18th AIVC Conference
Ventilation and Cooling

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Proceedings
Volume 1

PREFACE

International Energy Agency

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

Energy Conservation in Buildings and Community Systems

The IEA sponsors research and development in a number of areas related to energy. In one of these areas, energy conservation in buildings, the IEA is sponsoring various exercises to predict more accurately the energy use of buildings, including comparison of existing computer programs, building monitoring, comparison of calculation methods, as well as air quality and studies of occupancy.

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.

To date the following have been initiated by the Executive Committee (completed projects are identified by *):

Annex 1	Load Energy Determination of Buildings*
Annex 2	Ekistics and Advanced Community Energy Systems*
Annex 3	Energy Conservation in Residential Buildings*
Annex 4	Glasgow Commercial Building Monitoring*
Annex 5	Air Infiltration and Ventilation Centre
Annex 6	Energy Systems and Design of Communities*
Annex 7	Local Government Energy Planning*
Annex 8	Inhabitant Behaviour with Regard to Ventilation*
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Annex 12	Windows and Fenestration*
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Annex 21	Thermal Modelling*
Annex 22	Energy Efficient Communities*
Annex 23	Multizone Air Flow Modelling (COMIS)*
Annex 24	Heat Air and Moisture Transfer in Envelopes*
Annex 25	Real Time HEVAC Simulation*
Annex 26	Energy Efficient Ventilation of Large Enclosures*

Annex V Air Infiltration and Ventilation Centre

The Air Infiltration and Ventilation Centre was established by the Executive Committee following unanimous agreement that more needed to be understood about the impact of air change on energy use and indoor air quality. The aim of the Centre is to promote an understanding of the complex behaviour of air flow in buildings and to advance the effective application of associated energy saving measures in both the design of new buildings and the improvement of the existing building stock.

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

18TH AIVC CONFERENCE PROGRAMME
"Ventilation and Cooling"

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SESSION 7 - <u>INNOVATIVE COOLING</u>

Title: Ventilation and Cooling

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The main source of humidity in office buildings are the human occupant in the offices. Moisture is therefore a result of heat transmission from the person to the room air.

1. Heat transmission of the human body

The human heat transmission is done by convection, radiation and by evaporation of water to the environment. This physical transmissions cause the following six parameters of thermal comfort:

- activity level
- clothing
- air temperature
- air humidity
- air velocity
- wall temperature

The different heat transmission mechanism take over different parts of the total heat load. The ratio are depending on various parameters. With rising air-temperature the convection is decreasing meanwhile the latent heat by evaporation is increasing. In fig. 1 the influence of the activity level on the different ratios is shown. The total heat losses with an activity level related to 120 W (left part of Fig. 1) is fairly constant over a wide range of temperature. But the ratio between the latent heat and the sensible heat is very different at various air temperatures.
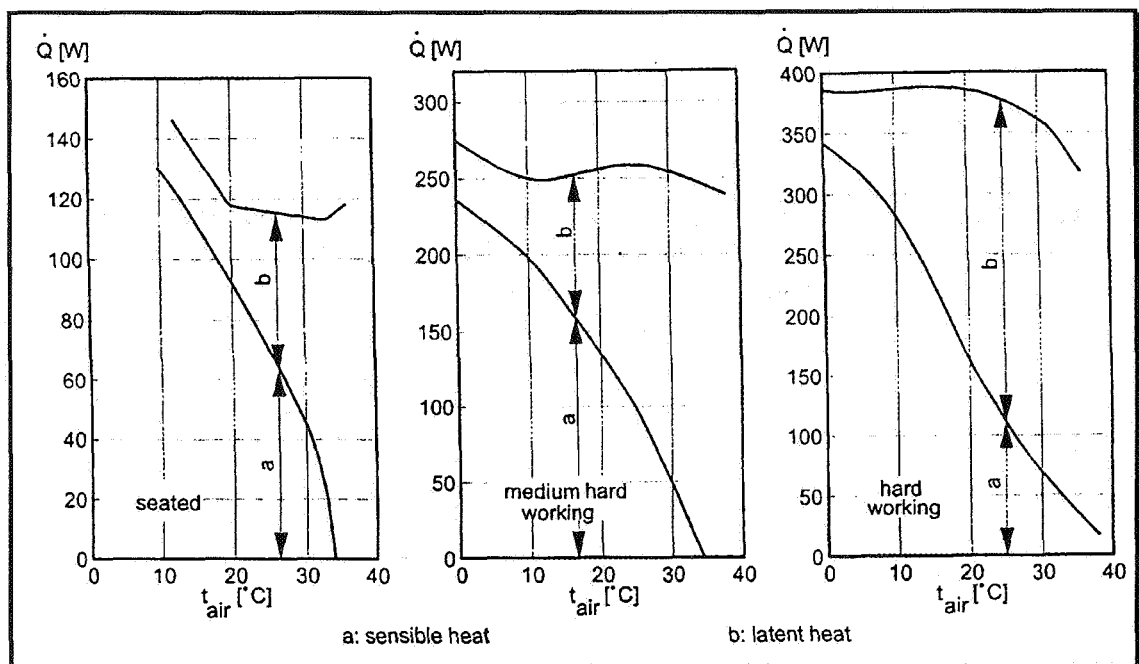
Fig. 1: Heat-transmission of persons in normal clothing /1/

Higher activity level related to 250 W (middle part of Fig. 1) or to 350 W (right part of Fig. 1) shows the same tendency. The sensible heat is always the sum of convection and radiation.

At higher temperatures the evaporation must take over sometimes incomming radiation. In cars e.g. the incoming solar radiation must be considered already at lower room-temperatures because this incoming energy must be bilanced by additional evaporaton. That is the reason why a large amount of water vapour ist transported to the air.

Main standards and regulations are done for offices with fairly the same activity level for all persons in the room. But also in these rooms a different heat transmission to the room can be seen in comparing the different individuals. /2/

Because the parameters of the heat convection are fairly stable, and because of the constant body temperature and a fixed clothing the control mechanism of the body temperature by changing the heat losses can only be done in the latent heat range. The rise of the temperature of the body surface can only be shifted in very small limits and the room air temperature and the air-velocity can not be ajusted individually. A change in the radiation is also not possible at a fixed wall temperature and given room configuration.

Fairly often different activity levels can be found in the same room. Some examples are shown with the following values (Tab. 1).

activity	range of heat transmission	mean value
1. seated	90 W - 150 W	120 W
2. typewriting	120 W - 170 W	150 W
3. speaker	160 W - 250 W	200 W
4. waiter	200 W - 300 W	250 W
5. dancing	200 W - 400 W	300 W

Tab. 1: Activity levels

These activity levels are very different but we can find them in the same room at the same time and with the same clothing. The distribution of the heat transmission to radiation, convection and evaporation therefore depends also on the room temperature. Calculated values on the bases on some measurements are shown in the next table (Tab. 2).

2. Humidity and comfort

If we regarde a ball-room we will have seated persons, speaker, waiter and dancing groups which have only a small difference in radiation and convection. To be able to meet the heat balance the human body sends water to the surface to reach evaporation cooling. The related amount of water will be at 20°C between 45 g/h and 220 g/h. The relative values at 26°C are 90 g/h and 300 g/h.

This amount of water must be transfered to the air. This can be reached by a high mass transfer coefficient or by a big difference between the absolut humidity at skin level and in the room air. An increase of the mass transfer coefficient will also give an increase of the heat transfer coefficient. This correlation is given in the Lewis law.

When we use the same values of the absolut humidity as in normal office buildings we will get no common comfort in rooms with very different activity levels.

		20°C				26°C			
	Heat	Radiation	Convection	Evaporation	Water	Radiation	Convection	Evaporation	Water
seated	120 W	45 W	45 W	30 W	45 g/h	30 W	30 W	60 W	90 g/h
typewriting	150 W	50 W	50 W	50 W	72 g/h	33 W	33 W	85 W	125 g/h
speaker	200 W	60 W	55 W	85 W	125 g/h	40 W	37 W	123 W	180 g/h
waiter	250 W	65 W	70 W	115 W	170 g/h	43 W	45 W	162 W	230 g/h
dancing	300 W	65 W	80 W	155 W	220 g/h	43 W	50 W	207 W	300 g/h

Tab. 2: Heat distribution with the same clothing and different activity levels

In table 2 we see that at 20°C a seated person will have a convective heat transfer of about 30 W and an evaporation heat of about 60 W. A dancing person, however, will have about 50 W of convective heat and more than 200 W by evaporation. If this will be done by a higher mass transfer coefficient the convective heat losses of a sitting person will increase too and will cause the feeling of draft.

To be able to avoid this the high water mass transfer must be reached with a big difference of absolut humidity instead of an increased transfer

coefficient. This means a temperature of 20°C and a low absolut humidity in the air with the standard air velocity will meet the comfort conditions for both groups of the population in the same room. The small temperature difference and a normal heat transfer coefficient will not give any draft for sitting persons and also give enough evaporation potential for people with higher activity level like speakers, waiters and dancers.

The relative humidity in the rooms should not be lower than 30 %. Below this level the nose and the throat can dry out and this must be avoided. In a lot of different materials which are used in buildings a low humidity can also give a high electrostatic load which also cause discomfort. /4/
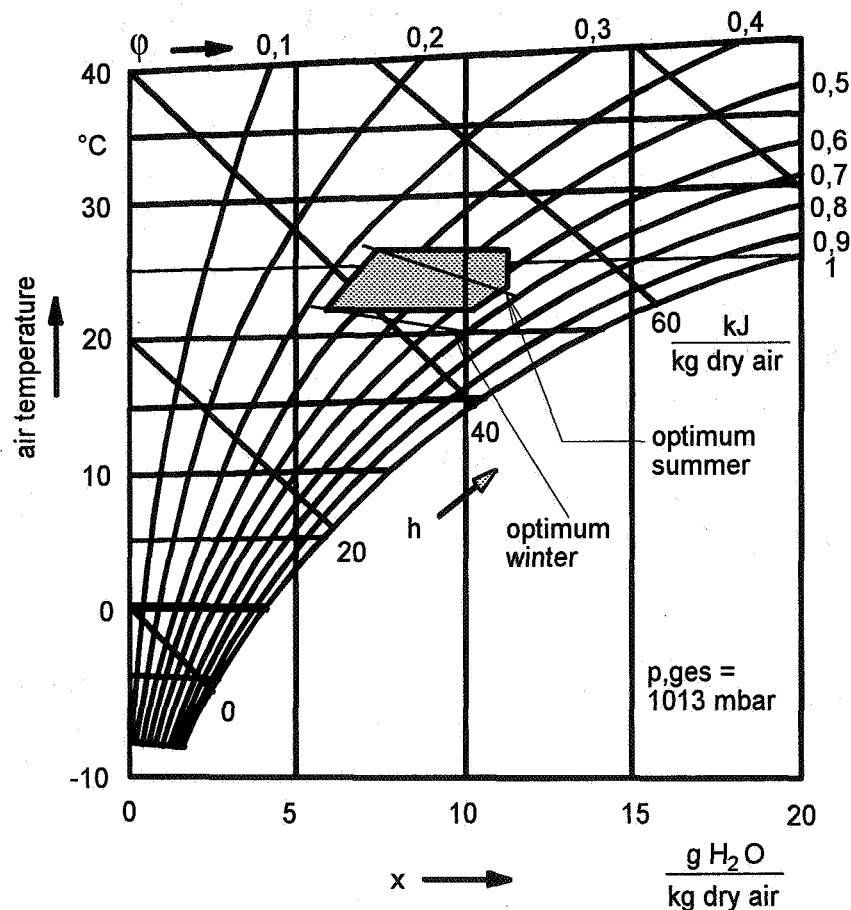
Fig. 2: Comfort zone DIN 1946 pt 2

The investigations of O. Fanger /5/ about the thermal comfort shows a much smaller influence of the humidity, but these values are only valuable for office buildings. The reason of these results is the very small change of activity level, a very simular clothing and a fairly stable air temperature. As shown the activity level is of great influence. It is not possible, therefore, to use the values for office buildings in a much broader scale. In figure 2 the optimal conditions are shown compared with the comfort zone of DIN 1946 part 2. These results can also be shown by experiments.

In air conditioned testrooms a group of about 30 people had to find out which rooms seems to be colder compared with the other one.

Unanimously they stated that a room with 26°C and 30% humidity is definetly cooler than a room with 24°C and 60 % humidity. These tests give the line of optimum conditions for summer (see figure 2).

3. Dehumidification load

The cooling load is not enough to discribe the refrigeration capacity because it is also necessary to consider the air changes. The air change rate cause especially in summertime a different dehumidification load which effects the refrigeration capacity. The minimum air changes are influenced strongly by the material which is used in the interial design. This can be shown in table 3 /6/. This example shows how great the influence of the material can be to the total energy consumption offer building. This influence is somewhat higher than the influence of the insulation.

	ventilation dehumidification		air changes		energy demand heating	
marble floor:	0,1 m³/m²h	⇨	0,04 ach	⇨	1 W/m²	2 W/m²
carpet floor:	2 m³/m²h	⇨	0,8 ach	⇨	25 W/m²	40 W/m²
	up to		up to		up to	up to
	8 m³/m²h	⇨	3,2 ach	⇨	100 W/m²	160 W/m²

Tab. 3: Ventilation rates

In summer everybody is speaking about cooling in air conditioning plants when he is going to degreese the air temperature. For the dehumidification we have to calculate in central Europe an enthalpy difference of about 25 kJ/kg dry air. It is very important to know that the highest enthalpy is not in the area of the highest temperature.

4. Cooling and Dehumidification

During the last years the reduction of energy consumption in buildings became one of the most important aims for the development of new technologies. A significant share of energy consumption in non-residential buildings bases on the requirement of cooling and air conditioning. Due to energy saving issues it is necessary to develop new air conditioning and cooling strategies, that lead to an continious reduction of energy consumption.

One possible starting point for the development of these systems is the separation of cooling and ventilation in air conditioning systems, because it

is more effective to transport energy by using water systems than to use only air to deliver the cooling energy to the rooms. This strategy was the basis for the development of hybrid systems. By using these systems, it is possible to reduce the ventilation rates to a minimum, which ensures dehumidification and guarantees a satisfying air exchange due to hygienic aspects. The main part of sensible cooling can be delivered to the room by induction-coils, heat exchangers with free convection or chilled structural elements in the room like chilled ceilings.

During the last few years many chilled ceiling systems and free convective cooling systems were developed and they are nowadays often installed in combination with ventilation systems, which guarantees the necessary ventilation rate. These systems can be installed as well in new as in retrofit commercial buildings. At the beginning of this period, there was no guide-line or technical standard available, which regulated or standardized the measurement of the cooling performance. So sometimes the given characteristic data for the design of these systems were related to different operating parameters like room temperature or mean temperature of the cold water, which is mainly used as transport medium to distribute the cooling energy within the building. Also it was possilble, that given data of cooling performance varied within a wide range for systems with nearly the same structure and design.

An accurate planning of these systems during the design period and an objective comparison of different systems is only possible, if the characteristic data for the description of heat transfer and cooling performance were investigated under comparable and clearly defined boundary conditions.

5. Refrigeration capacity

Mainly refrigeration systems for air conditioning are running with a temperature range of 6°C up to 12°C. Nearly all water chillers are designed for this temperature. The 6°C as supply temperature is necessary to reach the dehumidification of the outdoor air used for air renewal.

Normally the same system temperature is used to transport heat from inner heat gains to the central system. In chilled ceiling systems the temperature a the surface must be about 20°C in summertime to avoid condensing water at the surface.

If there is used the same water chiller either for the dehumiditation and for the cooled ceilings the 5°C or 6°C chiller water will be mixed with return water to read the higher supply temperature of about 15°C. If the water chiller will be devided in two parts working on a different temperature level a lot of energy can be saved. In table 4 the C.O.P. for different chiller temperatures are shown. They are measured in a water cooled condenser with 30°C/35°C. Compared with the supply temperature of 5°C the chiller

has 38 % increase in C.O.P. at 15°C water supply temperature. This is a very high energy saving which must be considered in new system design.

chilled water temperatures supply/ return	cooling capacity	electrical power input	C.O.P	relative C.O.P
5°C / 10°C	198 kW	42,5	4,6	1
6°C / 12°C	208 kW	43,1	4,82	1,048
10°C / 15°C	250 kW	45	5,55	1,21
15°C / 20°C	298 kW	47	6,34	1,38

Tab. 4: C.O.P.'s for different chiller temperatures

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VENTILATION AND COOLING
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Title: Design of Low Energy Office Buildings Combining Mechanical
             Ventilation for IAQ Control and Night-Time Ventilation for
             Thermal Comfort

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Design of low energy office buildings combining mechanical ventilation for IAQ control and night time ventilation for thermal comfort

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Summary

The design of low-energy office buildings requires specific attention to an energy efficient concept for providing good indoor air quality conditions. With this respect, mechanical ventilation shows undeniable advantages: it can be optimally controlled (infrared detection, CO_2 control,...), heat recovery is applicable, outdoor noise and pollution penetration can be minimised.

Another crucial challenge in low-energy office buildings is the avoidance or, if impossible, the minimisation of active cooling needs. Nighttime natural ventilation can play an essential part in an overheating prevention strategy.

This paper illustrates these concepts through several consulting projects:

- *new office building to be occupied by the owner;*

- *new office building by promoter to be rented by several companies;*

- *advanced double facade building in polluted area downtown;*

1. Introduction

The use of natural ventilation in non-domestic buildings is in principle an attractive option. In practice, one observes that there is often confusion with respect to the purpose of natural ventilation and the possible advantages and disadvantages. The Belgian Building Research Institute has been involved during the last few years in several consultancy studies for new buildings as well as for retrofitting projects and was also involved in several national and European projects in which the issue of natural ventilation was studied. This paper aims to give an overview of these experiences and it also gives some critical remarks with respect to natural ventilation in general and with the challenges for the design process.

2. Various functions of natural ventilation

When people speak about a 'naturally ventilated office building', it is often not evident to have a good understanding of what they really mean :

- in a number of countries and/or for a number of people, the meaning of natural ventilation is that the air supply and exhaust is assumed to happen through cracks and leakages in the facades as well as by opening the windows and doors. It is clear that such an approach may work for not too airtight buildings in very mild climates with good outdoor conditions (no noise problems and good outdoor air quality) but it is clear that in most circumstances such a strategy can not guarantee good indoor air quality conditions;

- others understand by 'natural ventilation design' that the required supply air for indoor air quality (IAQ) control is guaranteed by specifically designed supply and exhaust openings allowing to meet the IAQ needs and at the same time to keep the energy demand within reasonable limits. This strategy can be described as *'natural ventilation for IAQ control'*.

- others understand by 'natural ventilation design' that ventilation plays a crucial role for keeping thermal comfort conditions in summer. In general, night ventilation is used for cooling down the building structure at night in order to limit the indoor temperatures at daytime. This strategy can be described as *'natural ventilation for thermal comfort control'*.

It is essential that the designers fully understand the different purposes for 'natural ventilation for IAQ control' and 'natural ventilation for thermal comfort control'. As described in the paper presented at the AIVC conference in Gothenburg[1], the energy context for both strategies is completely different and therefore also the need for optimum control. In this paper, some practical aspects of both types of natural ventilation are discussed and illustrated with a number of practical examples.

3. Practical aspects for natural ventilation for IAQ control

Among the critical barriers for natural ventilation have to be mentioned :

- Optimum control of airflow rates
  Natural ventilation is the result of pressure differences due to temperature differences and wind effect. Outdoor temperature and wind varies substantially over time. If the supply openings are not controlled, the airflow rates will strongly vary over time. The users can achieve control but it is well known that such control is not optimal. Self-regulating devices, which adapt the cross section of the openings as function of needs and/or climatic conditions, are much better. At present, the number of such components is still very limited. It is expected that more and better components will come on the market in the future.

- Heat recovery
  Minimising the energy needs can be obtained by heat recovery. This is clearly not evident for natural ventilation designs.

- Acoustics
  A large majority of office buildings is located in areas with high acoustical outdoor levels. As a result, acoustical comfort can only be guaranteed if the most of the supply openings are acoustically insulated.

4. Practical aspects for natural ventilation for thermal comfort control

Natural ventilation for thermal comfort in summer is a very attractive option. The cooling potential is in principle very large, especially at night, but very large air quantities are needed since the thermal capacity of the air is very small (an outdoor air flow rate of 100 m³/h gives only 30 W/K cooling power).

Among the critical aspects have to be mentioned :

- Acoustics
  Intensive ventilation during working hours through open windows is not evident in urban areas for buildings in the vicinity of traffic roads, airports,... Night time ventilation is often not a problem since there will be no or only a very limited occupation and since the outdoor noise levels are in general lower than during the working hours.

- Outdoor pollution
  Traffic can create outdoor pollution problems due to which intensive ventilation through open windows at the streetside is not evident. At nighttime, pollution levels are in general lower and moreover no occupants.

- Burglary and insects
  At daytime and even more at nighttime, ventilation through open windows may not substantially increase the risk of burglary and the entrance of insects. Use of appropriate protection systems can guarantee this.

- Thermal mass
  Night ventilation can be an attractive option but a requirement is a minimum amount of thermal mass.

- Minimisation of thermal load (solar and internal)
  Even in buildings with a high thermal mass, thermal comfort without active cooling can only be guaranteed if the thermal load is reasonable. In practice, this often means good control of solar gains (by appropriate shading devices) and reasonable internal gains (efficient lighting, energy efficient computers and controls,...)

5. Design examples

5.1 General

A brief overview of some of the projects in which we were involved over the past few years is given in table 1.

	IVEG	Keppekouter	Pointcaré	Waregem	PROBE
Type of project	New building	New building	Renovation	Renovation	Renovation
Number of stories	3	3	6	2	2
Delivery	End 1998	Summer 1997	-	Summer 1997	Summer 1997
Architects	Lemaire & Musse	Declerck	Ph. Samyn & Partners		Y. Wauthy
Characteristics in relation to IAQ control					
• Ventilation strategy	Mechanical balanced	Mechanical balanced	Mechanical balanced	Natural	Mechanical supply
• Heat recovery	yes	Yes	Yes	No	No
• Noisy/polluted environment	yes	Yes	Yes	No	No
• Demand controlled ventilation	Yes	No	No	No	Yes
Characteristics in relation to temperature control in summer					
• Façade louvres for night ventilation	Yes, automatic	Yes	Rear façade	Yes	Yes
• Exhaust	Chimney	Extraction fan	Chimney	Chimney	-
• Solar protection	External screens	External screens	Double façade	External fixed	External screens
• Double façade ventilation	No	No	Yes	No	No

table 1 : overview of key features designs

In all these projects, the following features are found :

— Very good thermal insulation
A very good thermal insulation is important, not only for minimising the heating demand in wintertime but also for limiting the solar gains through opaque components in summer time.

— Good solar protection
Thermal comfort in summer with no or limited cooling power is only possible by minimising the solar gains. Therefore, solar protection systems are applied in all projects with g-values of the order of 0.10 or lower.

— Efficient intelligent lighting
In all projects, the use of high efficient fluorescent lamps with electronic ballast is foreseen resulting in installed lighting powers of the order of 10 W/m^2 (for 400 lux). Several projects make use of daylight compensated luminaries and in some cases also presence detection. These technologies have become mature technologies at reasonable additional costs.

In the next paragraphs, a more detailed description of the first 3 buildings in table 1 is given.

5.2 IVEG building

5.2.1 The context of the project and partners

This building will be the new head office of IVEG, which is public company for gas and distribution, which has some 70.000 clients in a number of villages in the region of Antwerp. A relatively long preparation period was possible (about 18 months) and the owner was very strongly motivated for realising a low energy building with good indoor climate conditions. Therefore, for a number of design decisions they were ready to take full responsibility and this of course simplifies certain decisions.

5.2.2 The overall design

It is a 3-story building with a total floor area of about 1.800 m². The plan of the top floor is shown in the figure.

5.2.3 Specific features relevant for this article

- Fire regulations
  The Belgian fire regulations specify that it is not allowed to have more than 2 stories in 1 fire compartment. The original scheme of 1 large central chimney for night ventilation had to be modified. As a result, 2 chimneys are now foreseen. An additional advantage of this scheme is that the ventilation of the top floor is substantially improved. In the original design, the neutral level was situated close to the middle height of the top floor whereas in the new design, it is situated above roof level.

- Design of chimney
  The net sections of both chimney openings are about 5 m² and the top of the chimneys is about 3 meter above roof level. Opening and closing of the chimneys is done automatically. These chimneys also allow the entrance of daylight.

- Top cooling as emergency
  Detailed dynamical simulations indicate that the applied strategies should result in acceptable thermal comfort in summer time. However, it is possible that the building use will change in time and/or that some of the assumptions are too optimistic. Therefore, the possibility for having in a later phase top-cooling is foreseen by having the supply ductwork insulated during the construction and by reserving the required space in the technical room for a chiller. Simulations indicate that top cooling will result in a 2...3°C temperature reduction and of course a lower relative humidity level.

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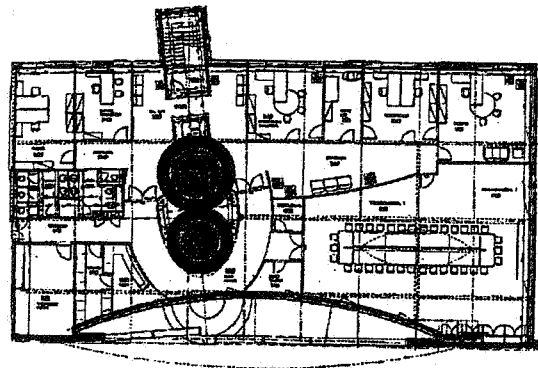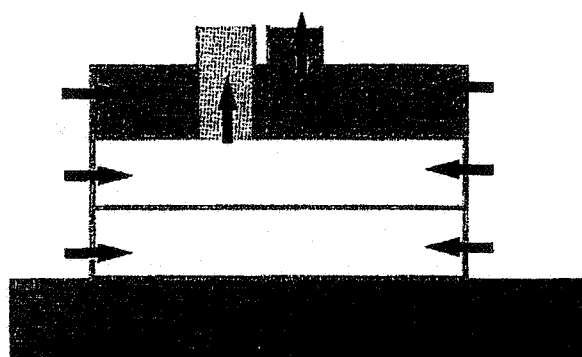
5.3 Project 'Keppekouter'

5.3.1 The context of the project and partners

Specific for this project is that it is project by a building promoter and that space will be rented. Therefore a very pragmatic approach with specific attention for the cost implications was essential. The building is situated at a distance of about 50 meters from the highway Brussels –Oostende and open windows during working hours is not evident.
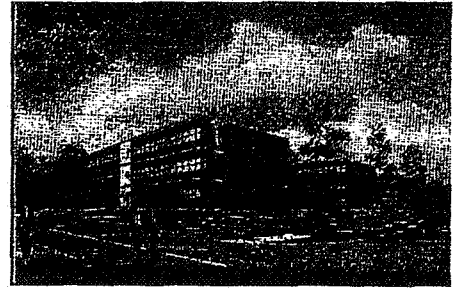
5.3.2 The overall design

It is a 3-story building. The spaces will be rented for office use, no details concerning the organisation of the space were available during the design phase. Landscape offices as well as cellular offices are possible.

5.3.3 Specific features relevant for this article

* As a result of the consultancy work, the thermal insulation of the building has been drastically improved and external solar shading is installed;

* Mechanical extraction for night ventilation, mainly because of budgetary reasons
Night ventilation is applied by having ventilation grills in slicing windows. BBRI proposed the use of natural extraction ducts, cross section 2 ducts of about 1 m² each per floor area. This results in loss of space, which can be rented or sold. The promoter came with the proposal to have mechanical extraction fans at roof level. Since they are only used at nighttime, the acoustical problem is marginal and the investment cost and space use rather low. The running costs of the fan seem to be acceptable.

* Thermal mass
The fact of having landscape offices strongly reduces the possibility of thermal mass in the separation walls. Moreover, the promoter wanted a false ceiling. Access to the thermal mass in the false ceiling is realised by having in the 4 corners of each room ceiling panels with each 9 openings 18 x 18 cm².

5.4 Project Pointcaré

5.4.1 The context of the project

This design for this renovation was prepared in the framework of an architectural competition. The design office Ph. Samyn and Partners (architects & engineers), wanted a low energy building with good indoor climate conditions. Specific attention is given to daylight features and summer comfort with minimum active cooling.

Proposed façade by Ph. Samyn and partners for Pointcaré building

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5.4.2 The overall design

The building is situated in Brussels along a road with very intensive traffic. The TGV station is at a distance of a few hundred meters. A complete renovation of the 6-story building is required.

5.4.3 Specific features relevant for this article

- **Double façade with various functions**
  The acoustical outdoor levels and the outdoor air quality do not allow the opening of the windows at street side. On the contrary, the noïse and air quality levels are quite satisfactory at the garden side. A double façade concept has been adopted in order to combine various functions :
  - efficient solar shading without specific maintenance problem
  - very good acoustical insulation of the street façade;
  - preheating of ventilation air during the heating season;
  - evacuation of room air during intensive night ventilation

  The operation mode in summer at night-time is shown in the figure above.
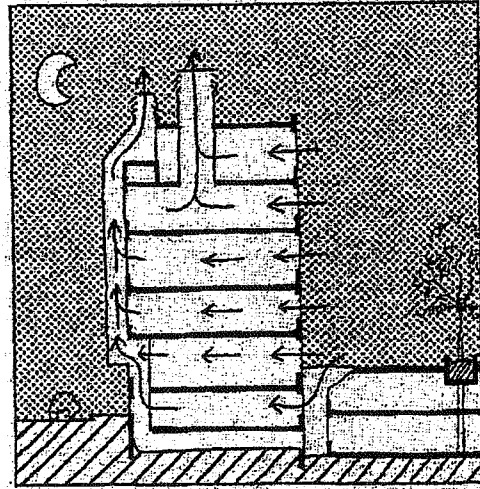
- **Avoidance of false ceilings in order to maximise thermal mass and use of daylight**
  The high floor to ceiling height and the concrete floors are attractive conditions for use of thermal mass and daylight.

- **Balanced ventilation with heat recovery for IAQ**
  In order to combine good IAQ with low energy demand, a balanced ventilation system with heat recovery was proposed. The supply can only be used for top cooling if needed.

6. Conclusions

1. Natural ventilation for IAQ control requires completely different provisions than natural ventilation for thermal comfort control in summer.

2. The use of natural ventilation for IAQ control in low-energy offices is of course possible but it is not necessarily the best choice in all circumstances.

3. Achieving an optimum balance between IAQ and energy use is for not evident with the natural ventilation devices on the market today. This situation may change in the future if self regulating devices become more widely available.

4. It is especially for summer ventilation very important that the issue of night ventilation is seriously considered from the beginning of the design process.

5. It is very important to have in the design process the appropriate knowledge, which allows quick decisions. Detailed simulations can be done afterwards for optimisation purposes and/of for checking if the proposed designs are feasible.

6. Optimising the airflow rate for thermal comfort in summer is in general from a numerical point of view not a real challenge. The higher the ventilation rate, the better the performances (unless draught problems occur). The real challenge for summer comfort is a combined design and analysis of solar and internal gains, thermal mass, thermal insulation, solar protection, ventilation,....

7. Further research is needed which focusses on the delivery of very practical design guides/rules to be used by designers in daily practice. The CIBSE guide on Natural ventilation in non-domestic buildings. Applications Manual AM10 : 1997.

7. Acknowledgements

8. References

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3. The CIBSE guide on Natural ventilation in non-domestic buildings. Applications Manual AM10 : 1997

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Title: Indoor Air Quality in Dwellings - A Comparison of the
Performance of Different Ventilation Systems.
IEA Annex 27 Evaluation of Ventilation Systems

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INTRODUCTION

The main goal of IEA Annex 27 "Evaluation of ventilation systems" is to develop tools to evaluate ventilation systems in an objective way in terms of indoor air quality, energy, comfort, noise, life cycle costs, reliability and other building related parameters.
To check the developed tools some measurements in real dwellings are necessary. The development of the tools is in its final stage. During the AIVC conference some of these tools will be presented. The indoor air quality tool is not yet ready. The results reported in this paper are investigations carried out in three groups of 10 dwellings with practically the same floor plan. Each group had a different ventilation system.
The ventilation systems are:

- Natural supply and passive stacks (natural)
- Natural supply with mechanical exhaust (mech. exhaust)
- Mechanical supply and exhaust (balanced)

This paper is the first analysis of the results. In a later stage the measurement results will be compared with the results of the tools developed within IEA Annex 27.

MEASUREMENTS

The measurements took place during the heating season 95/96 in Roermond, in the South East of the Netherlands.
The measurements carried out are:

- temperature and relative humidity
- PFT measurements to determine the total flow
- constant concentration tracer gas measurements to determine flow rates at room level
- IAQ measurements, CO_2, CO and TVOC's
- Air leakage tests
- system flow

During the measurements the occupants of the dwellings were asked to fill out a questionnaire form to determine their use of the ventilation provisions, such as the use of grids and vents and the switching of the mechanical system.
An overview of all measurements is given in figure 1.
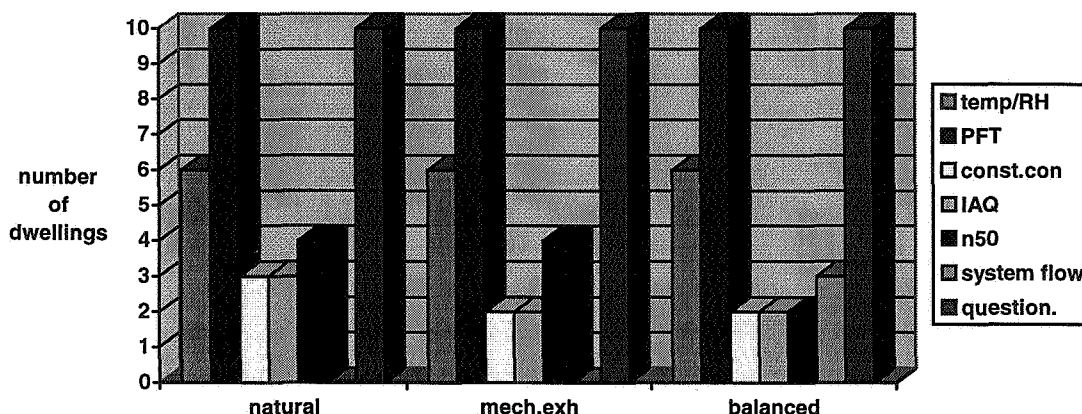
Figure 1   Overview of all measurements

DWELLING DATA

The dwellings are two storey single family houses with warm water heating systems.
The dwellings have a flat roof. All windows are double glazed.
The ground floor has an entrance hall, toilet and living with open kitchen.
The first floor has three bedrooms and a bathroom.

Ventilation systems

Due to building regulations a ventilation system in the Netherlands consists of:

Natural ventilation: grids and openable windows in all habitable rooms,
                        passive stacks in toilet, bathroom and kitchen.
Mechanical exhaust: vents and openable windows in all habitable rooms,
                        mechanical exhaust in toilet, bathroom and kitchen
Balanced system: mechanical supply to all habitable rooms
                        openable windows in all habitable rooms
                        mechanical exhaust in toilet, bathroom and kitchen

Not all these provisions were found in the dwelling which were investigated.
In case of natural ventilation the grids were not in the bedrooms and living, but only in the kitchen. In the dwellings with balanced systems the grids were in living room and kitchen. According to the building regulations they are not required there.

Air leakage data

The measured air leakage of the dwellings can be seen in figure 2.
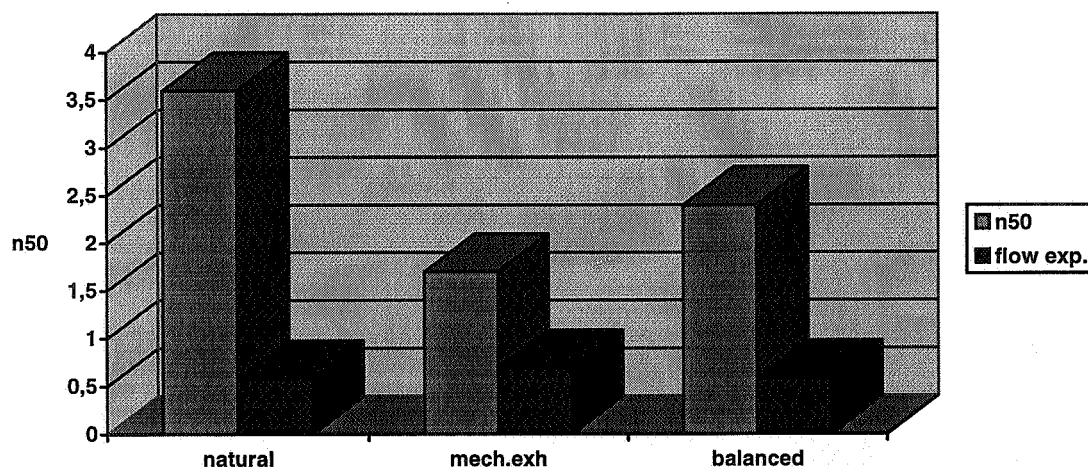
Figure 2   Results from pressurisation measurements

The air tightness of these dwellings can be considered as quite good. The differences between the three systems are however considerable.

Number of occupants per dwelling

The average number of occupants in the dwelling may be important for the intensity of the use of the dwelling. Data can be found in figure 3.
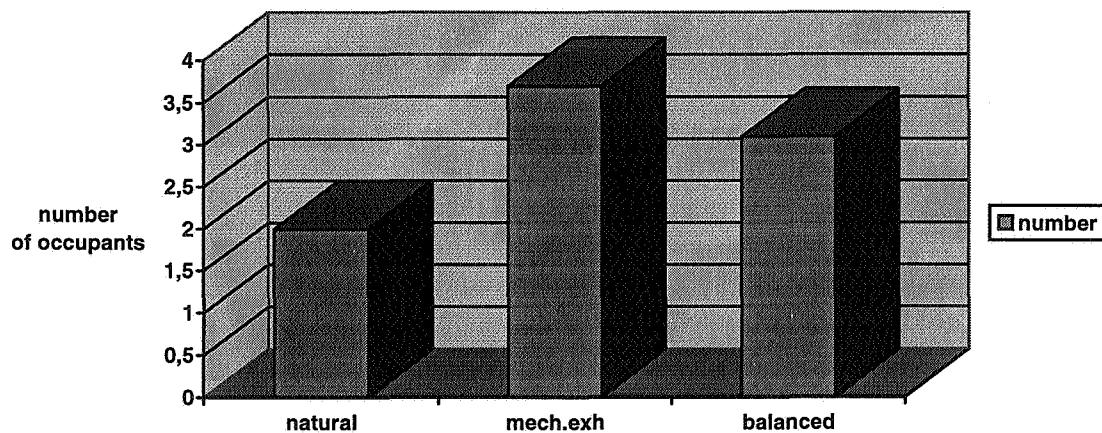
Figure 3   Average number of occupants grouped per ventilation system

The average number of occupants for all dwellings is about 2,9, which is a quite normal level. However the differences between the three groups are big. The natural ventilation system dwellings have about half of the occupancy than the two others.

MEASUREMENTS RESULTS

Temperature

Figure 4 shows the measured data.
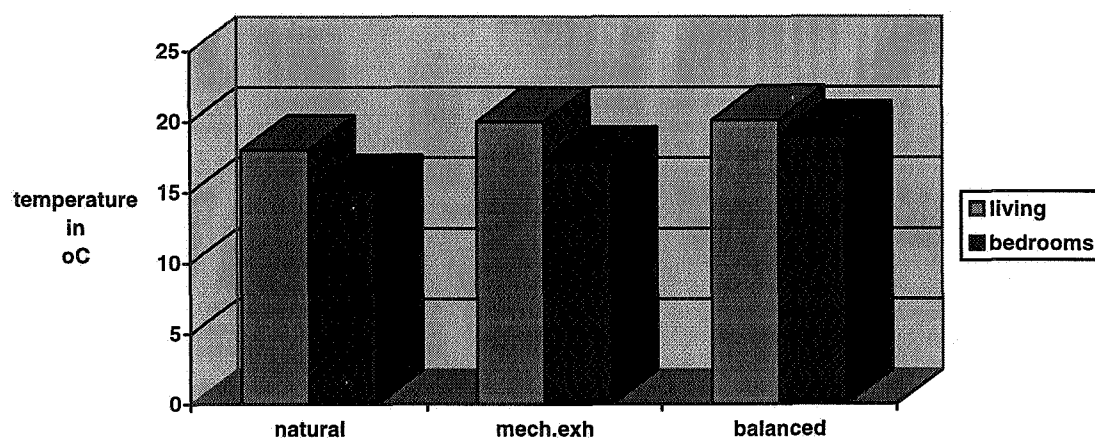
Figure 4   Temperatures in the dwellings

The following remarks can be made.

The temperature in the living is almost the same in the dwellings with mechanical exhaust and the balanced systems.

The naturally ventilated dwellings have the lowest temperature in the living as well as in the bedrooms. In the dwellings with balanced ventilation the difference between living and bedroom temperatures is as expected very low.

Moisture level

The results of the moisture level is presented in figure 5.
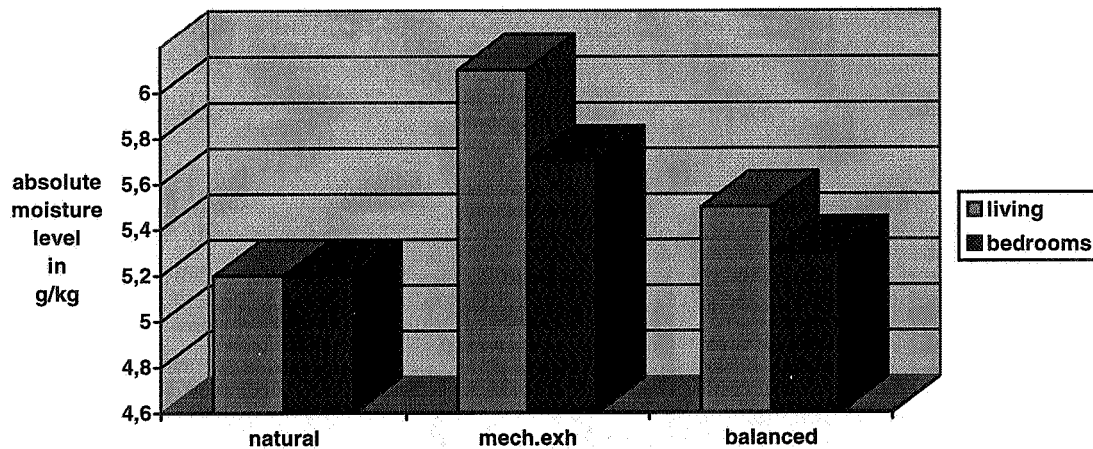
Figure 5   Absolute moisture level in the air

As can be seen the differences between the ventilation systems are considerable.

An important effect can be the occupancy. But also differences in ventilation rate may have their effect. The natural ventilated dwellings again are much lower than the dwellings of the two other systems.

Air Flow Rates
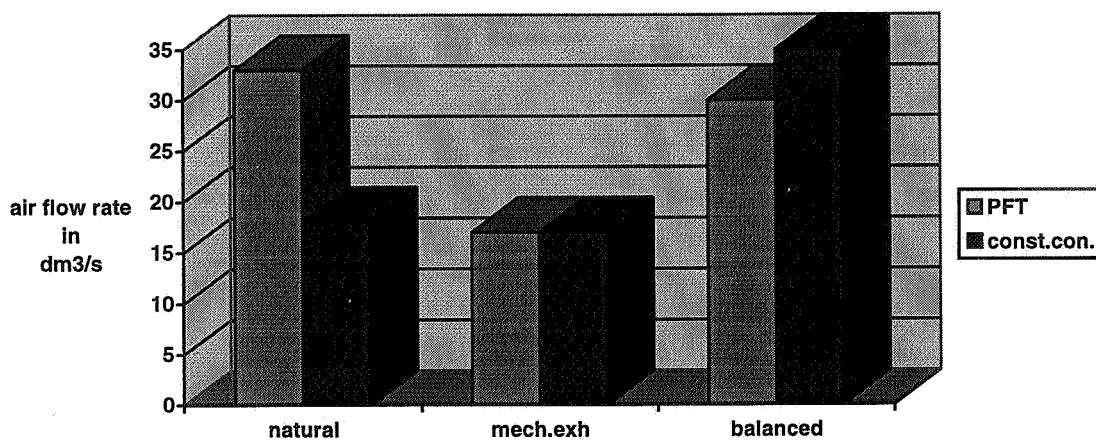
Figure 6   Measured air flow rates

There is a good correlation between the PFT measurements and the constant concentration measurements in case of mechanical exhaust and balanced systems.

The natural ventilated dwellings give a enormous difference between PFT and constant concentration results. Many checks have been carried out on the data analysis but no explanation can be given.

Indoor air quality

The indoor air quality is measured in terms of CO_2 , CO and TVOC concentrations.

The CO_2 data can be found in figure 7.

Again considerable differences between the groups of dwellings with different ventilation systems. The CO_2 concentration in the dwellings with the mechanical exhaust system is much higher than in the two other systems. This is partly due to the higher occupancy.

But this can not explain the whole effect. There must be an effect of the air flow rate.
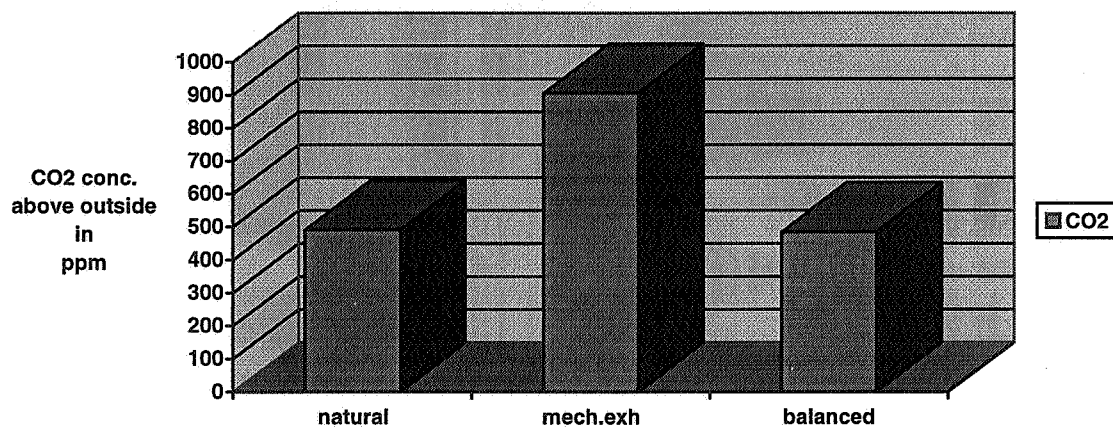
Figure 7   Measured CO_2 levels in dwellings

The measured data for CO and TVOC are shown in figure 8.
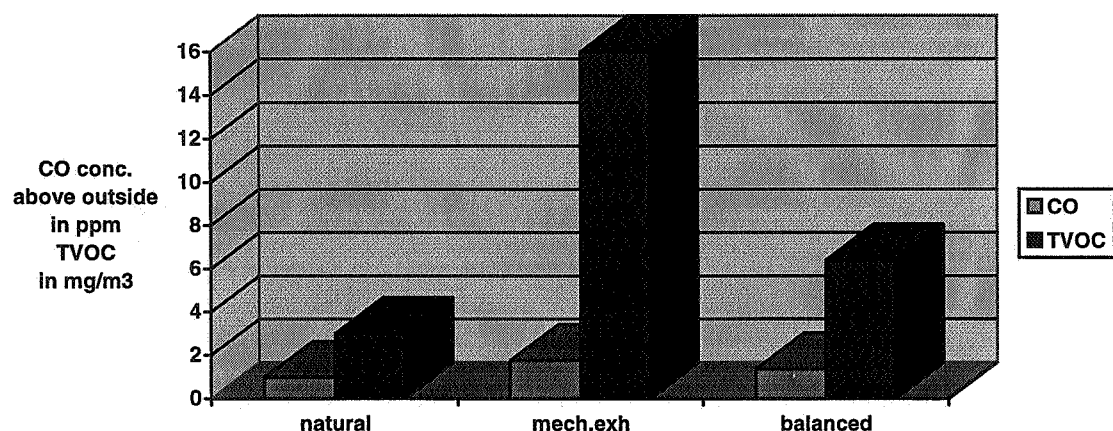
Figure 8   Other measured indoor air quality results.

The differences can't be explained on the bases of number of occupants. The habits of the occupants must be the risen for it.

Use of ventilation provisions

Grids
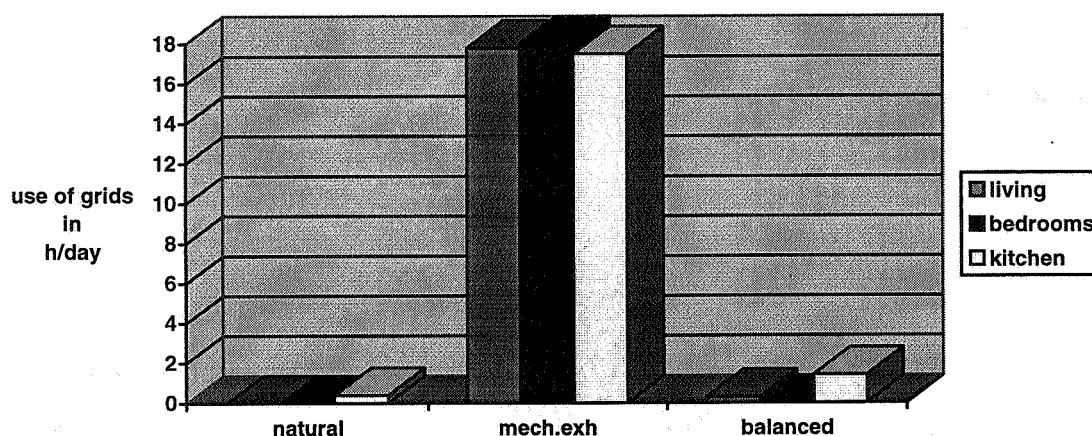
Figure 9   The use of grids by the occupants in hours per day.

Some remarks are necessary to understand these figures.
In the dwellings with natural ventilation system the grids are only present in the kitchen.
In the dwellings with balanced systems the grids, which are not a necessity at all, are only present in living room and kitchen.
The frequent use of these grids at average about 18 hours a day is remarkable. There effect on the flow rates are not very big. The sizing of the grids have to be checked.

Airing
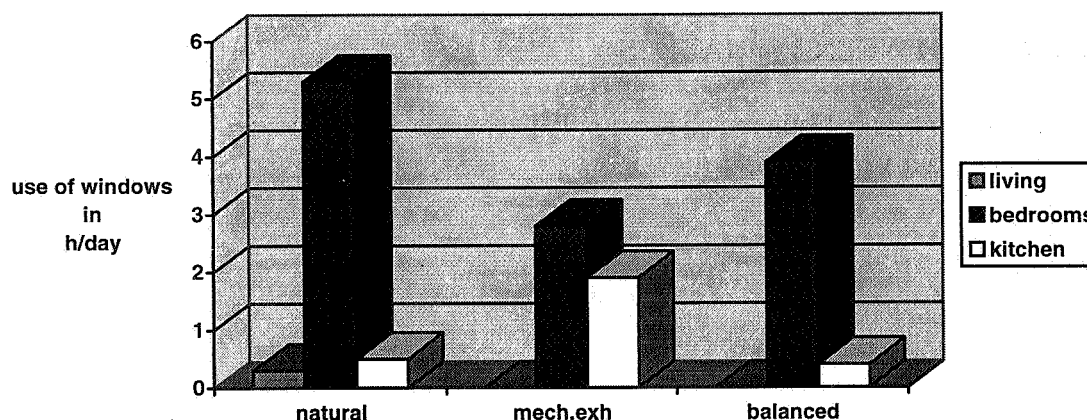
Figure 10   Use of windows for airing

The use of openable windows for airing give a more coherent picture. The result are shown in figure 10.

In the living the use is at minimal level. The use of openable windows in the kitchen is limited to less than two hours, which can be explained by the cooking periods.

In the bedrooms the lowest use is in the dwellings with mechanical exhaust which have the maximum use of the grids. (see figure 9)

The results are in agreement with earlier research of IEA annex 8 "Inhabitants behaviour with regard to ventilation".

Switch on time of balanced systems
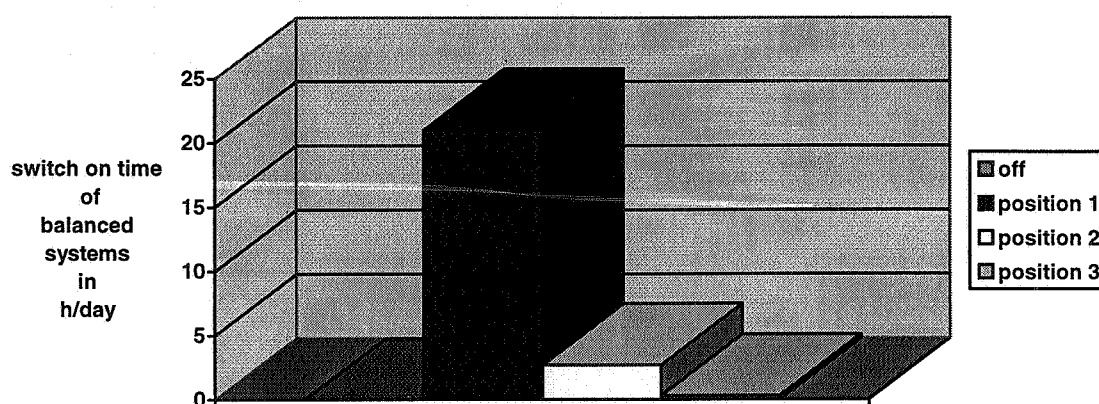
Figure 11   Switch on time of the balanced systems

As can be seen the balanced system are switched in position 1 for more than 20 hours. The are flow rate at that position 1 is aboutd16 dm^3/s. The air flow rate in position 2 is about 26 dm^3/s and in position 3 about 33 dm^3/s.

So a few hours per day, normally during cooking and washing, the air flow rate for the whole dwelling is at a level around 30 dm^3/s.

RELATION BETWEEN VARIABLES

As can be seen in figure 12 there is a relation between some of the variables of this study.
The most important variables are:
- CO_2 level
- air flow rate
- air leakage
- number of occupants
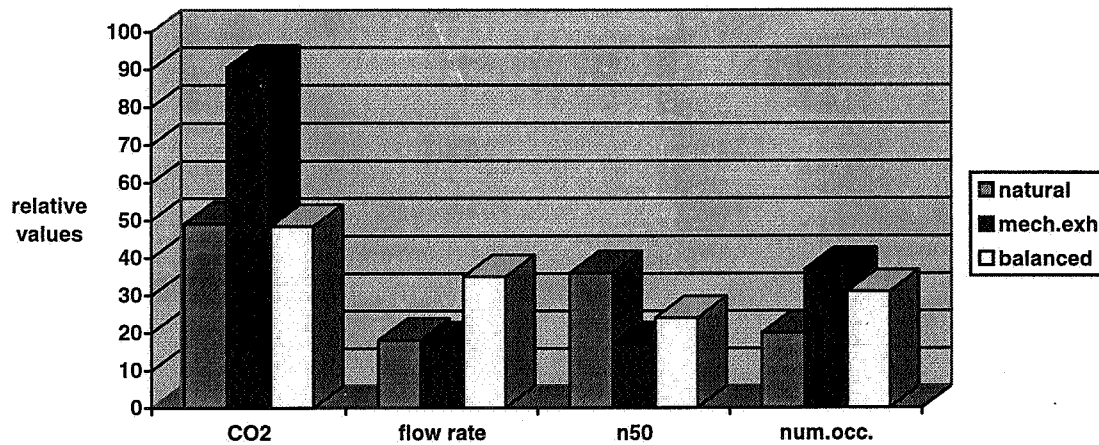
Figure 12   Relation between CO_2 , air flow rate, air leakage and the number of occupants

The mechanical exhaust systems have the highest CO_2 concentration in the most air tight dwellings with the highest occupancy. The air low rate is about the same as in case of the natural ventilated dwellings.

The CO_2 concentrations in the balanced ventilated dwellings is half of that of the mechanical exhaust systems. The flow rate in the balanced systems is about twice that of those in the mechanical exhausted dwellings, with about the same occupancy.

CONCLUSIONS

The measurement results show that the main parameter for the indoor air quality in these dwellings is not the ventilation system it self, but the use of the ventilation provisions in the dwellings by the occupants. Window airing can't be neglected in evaluating the indoor air quality in dwellings.

Other important parameters are:

• number of occupants
• air leakage level of the dwelling

For other pollutants such as CO and TVOC the main parameter is probably the activities by the occupant and not the ventilation system.

A tool to evaluate ventilation systems which don't take into account the inhabitants behaviour will not be very successful in predicting the right indoor air quality.

A conclusion that one of the ventilation systems is better that the other is difficult to draw. Natural and balanced systems had about the same CO_2 level . The air flow rate of the balanced system was twice as high as the natural one, but the number of occupants was also higher.

LBL

Title: Recommended Ventilatioan Strategies for New Energy-Efficient Production Homes

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VENTILATION AND COOLING

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RECOMMENDED VENTILATION STRATEGIES FOR NEW ENERGY-EFFICIENT PRODUCTION HOMES

[handwritten: signa mb beschouwd natural ventilation !]

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SYNOPSIS

The U.S. Environmental Protection Agency is seeking to improve the thermal quality of new homes, most of which are being built in the sunbelt by large building development companies. Low-infiltration production (tract) homes need ventilation systems that satisfy the low-cost priority of the builders as well as the safety, health and low operating cost expectations of homeowners. We evaluated ten ventilation strategies in order to recommend the most suitable systems for four climates: cold, mixed, hot-humid, and hot-arid.

We recommend that builders in mixed (cold and hot), hot-humid and hot-arid climates use supply ventilation, which provides the safety and health benefits of positive indoor pressure and the ability to filter and dehumidify ventilation air. When ventilation is integrated with forced-air conditioning, we recommend that ductwork be installed within conditioned space and buyers be offered the option of an efficient, variable-speed fan. In cold climates we recommend that builders offer buyers the option of balanced heat recovery units, which significantly reduce operating costs. In hot-humid climates, we recommend that builders offer buyers the option of dehumidifying supply ventilation to control indoor humidity and improve occupant comfort.

1. INTRODUCTION

The majority of new homes in the U.S. are built by large production building companies in fast-growing sunbelt cities from Florida to California. The Environmental Protection Agency (EPA), in its efforts to reduce greenhouse gas emissions, has introduced the Energy Star Homes program to encourage production homebuilders to voluntarily improve the energy-efficiency of their construction to beyond the levels required by the Model Energy Code. To achieve this, infiltration must be reduced to less than 0.20 (average annual) air changes per hour (ACH). This is below the level suggested to maintain indoor air quality (according to ASHRAE, the American Society of Heating, Refrigerating and Air Conditioning Engineers). These homes will need supplemental (active, mechanical) ventilation systems to provide fresh air and remove moisture and indoor pollutants.

Our task was to recommend the most affordable and effective ventilation systems in four climates. Suitable systems must meet or exceed ASHRAE ventilation and indoor air quality guidelines and be easy to implement by production residential builders and subcontractors. This is a challenging task because production builders' decisions are driven by cost, and though the Energy Star Homes program goal is to build efficient production homes at no additional cost, ventilation systems add to first-cost. Also, home ventilation systems currently available in the U.S. were developed for very cold climates, and even experienced HVAC contractors are unfamiliar with residential ventilation, and finally, unlike new custom homes, production home buyers have no input to the selection of their ventilation system.

2. SYSTEM DESCRIPTIONS

We determined that, at a minimum, ventilation systems must be able to deliver at least 0.35 ACH (daily average) ventilation and not cause or contribute to indoor depressurization. In order to account for the variation in effective ventilation rates, we normalized operating hours (operating costs) of all strategies to an effective rate of 0.50 ACH. Table 1 provides the name and description of the ventilation strategies evaluated.

Avoiding depressurization is a safety and health consideration. Negative indoor pressure can pull smoke from a fireplace, radon gas (if present) from the soil, auto exhaust from an attached garage, and pathogens from an attic, duct or building cavity into the home. It can also cause backdrafting (flue gas reversal) of combustion appliances that interact with indoor air. The Energy Star Homes program recommends that combustion appliances be direct-vent (sealed from indoor air), but it is reasonable to assume that most of these homes will have an attached garage, fireplace, and/or radon gas. Temporary depressurization can occur in any home (e.g., when a clothes dryer operates) but exhaust ventilation in a tight home without provision of supply air could intensify, prolong, or even sustain depressurization.

Table 1. Ventilation Strategies Evaluated	
Strategy Name	Description
1) *Forced-air Supply*	Outside air duct connected to the forced-air return
2) *Bath Exhaust w/ Vents*	An upgraded bath exhaust fan with passive vents
3) *Single-port Exhaust w/ Vents*	Ceiling- or remote-mount exhaust fan with passive vents
4) *Multi-port Supply*	Supply fan with ventilation ducts to living and bedrooms
5) *Forced-air Supply w/ Exhaust*	*Forced-air Supply* with a single-port exhaust fan
6) *Multi-port Supply w/Exhaust*	*Multi-port Supply* with a single-port exhaust fan
7) *Multi-port Exhaust w/ Vents*	Remote multi-port exhaust fan with passive vents
8) *Dehumidifying Forced-air Supply* (in Houston only)	Whole-house dehumidifying supply ventilation unit; ventilation air is distributed via forced-air ductwork
9) *ICM Forced-air Supply*	*Forced-air Supply* with an integrated-control motor[1]
10) *Balanced Heat Recovery*	Balanced heat-recovery ventilation unit with ductwork

3. MODELING

The following cities were selected to represent each of the four climates: Boston MA (cold), Washington DC (mixed), Houston TX (hot humid), and Phoenix AZ (hot arid). For modeling purposes, we assumed that the 2500 ft^2 Boston and Washington prototypical homes have 2-stories and basements. The 2000 ft^2 Houston and Phoenix homes are single story with a slab foundation.

Ventilation system performance was modeled using RESVENT software[2], hourly weather data, and the following assumptions:

- Energy Star homes have an annual average infiltration rate of 0.20 ACH.
- Mechanical ventilation systems are designed to provide, along with infiltration, a total annualized air change rate of 0.50 ACH.
- Windows remain closed, even in mild weather. [3]

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[1] Integrated-control motors have variable speed controls at the motor (rather than remotely-located, like ECMs).

[2] RESVENT was developed by the Energy Performance in Buildings Group of the Indoor Environment Program at Lawrence Berkeley National Laboratory (LBNL).

[3] Reasons that people might not open windows even in mild weather include noise, security, allergy and infirmity.

We used the ASHRAE 136 method to determine normalized leakage values corresponding to an annualized average infiltration rate of 0.20 ACH. We used DOE-2 to determine the hours of heating and cooling operation for forced-air integrated systems. Ventilation strategies were modeled with continual (24 hour) operation.

Modeling results show that, to provide an effective ventilation rate of 0.50 ACH, mechanical system design flow rates vary according to the climate, number of ventilation fans used, and whether the home is pressurized. The corresponding mechanical ventilation system design rates are given in Table 2. These results support the fact that ventilation contractors should take into account the climate, proposed ventilation system type and operating schedule when designing a residential ventilation system.

Table 2. Mechanical Ventilation System Design Rates (ACH) (For 0.50 ACH Annualized Average Effective Ventilation Rate)			
Ventilation Systems with: Strategies (from Table 1):	two fans #5, #6, #10	one fan and vents #2, #3, #7	one fan, no vents #1, #4, #8, #9
BOSTON	.23	.37	.41
WASHINGTON	.21	.36	.40
HOUSTON	.27	.40	.43
PHOENIX	.30	.43	.45

4. COSTS

Our installation (first) cost estimates include materials, labor and 25% overhead and profit. Costs were compiled from ventilation equipment manufacturers, distributors, contractors and consultants. Costs of all systems include a programmable timer with an on/off switch. For systems with passive vents, we assumed that one-story homes have five passive vents, and two-story homes have six passive vents. The heat recovery (HRV) system modeled has a 70% heat recovery efficiency. We assumed that installation costs are the same in each city (i.e., any variation is within the limits of our accuracy).

Annualized installation costs assume a 7% real discount rate. Installation costs were amortized assuming a 20-year lifetime for all ventilation systems, replacement after ten years of standard air-handler fans used intermittently for ventilation, and a 20-year lifetime for HRVs and other fans designed for continuous operation.

Table 3 lists the installation, operating and total annualized costs of ventilation systems in homes with two types of heating and cooling equipment – a furnace and air conditioner, and an electric heat pump.

Table 3. Ventilation System Costs

	Installation or First Cost ($)	Annualized Installation Cost ($)	Operating Costs by Heating & Cooling Equipment			
			Furnace/Air Conditioner OR Electric Heat Pump			
			Annual Operating Cost ($)	Total Annualized Cost ($)	Annual Operating Cost ($)	Total Annualized Cost ($)
BOSTON - cold climate						
1) Forced-air Supply	300	25	436	461	435	460
2) Upgraded Bath Exhaust with Vents	463	23	395	418	346	369
3) Single-port Exhaust with Vents	613	31	395	426	346	377
4) Multi-port Supply	650	33	388	421	342	375
5) Forced-air Supply with Exhaust	663	43	504	547	499	542
6) Multi-port Supply with Exhaust	763	38	419	457	370	408
7) Multi-port Exhaust with Vents	1063	53	395	448	346	399
8) ICM Forced-air Supply	1550	78	376	454	325	403
9) Balanced Heat Recovery	1838	92	324	416	289	381
WASHINGTON - mixed climate						
1) Forced-air Supply	300	25	323	348	290	315
2) Upgraded Bath Exhaust with Vents	463	23	290	313	290	313
3) Single-port Exhaust with Vents	613	31	290	321	220	251
4) Multi-port Supply	650	33	287	320	220	253
5) Forced-air Supply with Exhaust	663	43	364	407	184	227
6) Multi-port Supply with Exhaust	763	38	303	341	231	269
7) Multi-port Exhaust with Vents	1063	53	290	343	220	273
8) ICM Forced-air Supply	1550	78	282	360	212	290
9) Balanced Heat Recovery	1838	92	237	329	184	276
HOUSTON - hot humid climate						
1) Forced-air Supply	300	25	280	305	324	349
2) Upgraded Bath Exhaust with Vents	463	23	231	254	226	249
3) Single-port Exhaust with Vents	613	31	231	262	226	257
4) Multi-port Supply	650	33	227	260	223	256
5) Forced-air Supply with Exhaust	663	43	324	367	367	410
6) Multi-port Supply with Exhaust	763	38	252	290	247	285
7) Multi-port Exhaust with Vents	1063	53	231	284	226	279
8) Dehumidifying Forced-air Supply	1463	73	338	411	338	411
9) ICM Forced-air Supply	1550	78	228	306	220	298
10) Balanced Heat Recovery	1838	92	186	278	182	274
PHOENIX - hot arid climate						
1) Forced-air Supply	300	25	308	333	354	379
2) Upgraded Bath Exhaust with Vents	463	23	255	278	246	269
3) Single-port Exhaust with Vents	613	31	255	286	246	277
4) Multi-port Supply	650	33	249	282	240	273
5) Forced-air Supply with Exhaust	663	43	363	406	409	452
6) Multi-port Supply with Exhaust	763	38	283	321	274	312
7) Multi-port Exhaust with Vents	1063	53	255	308	246	299
8) ICM Forced-air Supply	1550	78	252	330	244	322
9) Balanced Heat Recovery	1838	92	206	298	201	293

Note: Total annualized cost is the sum of annualized installation cost and annual operating cost.

5. EVALUATION

Strategies were compared to each other according to five "priority" criteria: 1) installation cost, 2) operating cost, 3) indoor pressure, 4) effective distribution of ventilation air within the home, and 5) the potential for ventilation-related condensation in exterior walls. For each of our five evaluation criteria, we quantified the cost and effectiveness of ventilation strategies by assigning each a score (from –3 to 3) for each climate. Installation costs, indoor pressure and distribution scores are based on system types, and therefore, the same for each climate. Total score (overall cost and effectiveness) is the sum of the five scores. Finally, in each climate, we ranked strategies (with 1 as best), based on the total scores. Scores and ranking results are provided in Table 4. The scoring criteria is described below.

5.1. Installation Cost:

First Cost	$300-400	$401-500	$501-600	$601-700	$701-800	$801-1200	> $1200
Score	3	2	1	0	-1	-2	-3
Strategies	#1	#2	none	#3, 4, 5	#6	#7	#8, 9, 10

5.2. Annual Operating Cost

Average of the annual ventilation system operating costs for furnace/AC and electric heat pump:.

Score	Boston	Washington	Houston	Phoenix
3	$300-325	$200-225	$175-200	$200-225
2	$326-350	$226-250	$201-225	$226-250
1	$351-375	$251-275	$226-250	$251-275
0	$376-400	$276-300	$251-275	$276-300
-1	$401-425	$301-325	$276-300	$301-325
-2	$426-450		$301-325	$326-350
-3	>$450		$326-350	>$350

5.3. Indoor Pressure, from a safety and health perspective:

Score	Indoor Pressure	Strategies
3	positive	#1, 4, 8, 9
0	neutral, balanced	#2, 3, 5, 6, 7, 10
-3	negative	none evaluated

Table 4. Ventilation Systems Ranked by Cost and Effectiveness

Systems are Sorted by Rank Rank is based on Total Score	Rank	Total Score	EVALUATION CRITERIA				
			Installation Cost score	Operating Cost score	Indoor Pressure score	Distribution of Air score	Moisture Problems score
BOSTON - cold climate							
Multi-port Supply	1	3	1	1	3	1	-3
Balanced Heat Recovery	1	3	-3	3	0	3	0
Forced-air Supply	2	2	3	-2	3	1	-3
Multi-port Supply with Exhaust	2	2	0	0	0	2	0
ICM Forced-air Supply	3	0	-2	1	3	1	-3
Upgraded Bath Exhaust with Vents	3	0	2	1	0	-3	0
Single-port Exhaust with Vents	3	0	1	1	0	-2	0
Forced-air Supply with Exhaust	3	0	1	-3	0	2	0
Multi-port Exhaust with Vents	4	-1	-1	1	0	-1	0
WASHINGTON - mixed climate							
Multi-port Supply	1	6	1	1	3	1	0
Forced-air Supply	1	6	3	-1	3	1	0
ICM Forced-air Supply	2	4	-2	2	3	1	0
Forced-air Supply with Exhaust	2	4	1	1	0	2	0
Balanced Heat Recovery	3	3	-3	3	0	3	0
Multi-port Supply with Exhaust	3	3	0	1	0	2	0
Single-port Exhaust with Vents	4	0	1	1	0	-2	0
Multi-port Exhaust with Vents	5	-1	-1	1	0	-1	0
Upgraded Bath Exhaust with Vents	5	-1	2	0	0	-3	0
HOUSTON - hot humid climate							
Multi-port Supply	1	10	1	2	3	1	3
Forced-air Supply	2	8	3	-2	3	1	3
ICM Forced-air Supply	3	7	-2	2	3	1	3
Multi-port Supply with Exhaust	4	3	0	1	0	2	0
Balanced Heat Recovery	4	3	-3	3	0	3	0
Forced-air Supply w/ Dehumidifier	5	2	-2	-3	3	1	3
Forced-air Supply with Exhaust	6	0	1	-3	0	2	0
Upgraded Bath Exhaust with Vents	7	-1	2	1	0	-3	-1
Single-port Exhaust with Vents	7	-1	1	1	0	-2	-1
Multi-port Exhaust with Vents	8	-2	-1	1	0	-1	-1
PHOENIX - hot arid climate							
Multi-port Supply	1	7	1	2	3	1	0
Forced-air Supply	2	5	3	-2	3	1	0
ICM Forced-air Supply	3	4	-2	2	3	1	0
Balanced Heat Recovery	4	3	-3	3	0	3	0
Multi-port Supply with Exhaust	5	2	0	0	0	2	0
Upgraded Bath Exhaust with Vents	6	0	2	1	0	-3	0
Single-port Exhaust with Vents	6	0	1	1	0	-2	0
Forced-air Supply with Exhaust	6	0	1	-3	0	2	0
Multi-port Exhaust with Vents	7	-1	-1	1	0	-1	0

5.4. Distribution of Ventilation Air:

Score	Strategies	Distribution Effectiveness
3	#10	air is supplied to and exhausted from several rooms
2	#5, 6	air is supplied to several rooms, exhausted from a central location
1	#1, 4, 8, 9	air is supplied to several rooms
-1	#7	air is exhausted from each bath, closed doors can disrupt circulation
-2	#3,	air is exhausted from a central location, closed doors disrupt circulation
-3	#2	air is exhausted from one bath, closed doors definitely disrupt circulation

5.5 Moisture Problems:

Score	Potential for indoor pressure to cause condensation in exterior walls
3	ventilation-induced indoor pressure prevents moisture problems
0	indoor pressure is neutral, or there is no potential moisture problem
-1	indoor pressure may cause humid outdoor air to enter walls via infiltration
-3	ventilation-induced indoor pressure will push humid indoor air into walls

6. RECOMMENDATIONS

6.1. Cold Climate

In cold climates, we recommend production builders use exhaust ventilation with passive vents, or supply ventilation combined with measures to prevent condensation in exterior walls, or offer home buyers the option of paying for balanced heat recovery ventilation, which reduces operating costs.

For builders who use exhaust ventilation in cold climates, we recommend *Single-*or *Multi-port Exhaust with Vents*. *Single-port Exhaust with Vents* is less expensive to install; however *Multi-port Exhaust with Vents* provides better distribution of ventilation air. In multi-level homes, we recommend installing passive vents on the lower floor only, with the exhaust ventilation fan in the ceiling of the upper floor, and operating the system continuously, to help ensure that air enters the vents and exits via the fan.

In cold climates, we recommend supply ventilation only when combined with measures to prevent condensation in walls: 1) use a dehumidistat to control at least one bathroom exhaust fan and maintain indoor relative humidity $\leq 50\%$ and 2) install insulative vapor-permeable sheathing on exterior walls to keep wall temperature above the dew point of indoor air and facilitate drying. Builders using *Forced-air Supply* should also install ductwork in conditioned space and offer buyers the option of paying for a forced-air fan with a variable-speed integrated-control motor (ICM).

6.2. Hot Humid Climate

In hot humid climates, we recommend production builders install *Multi-port Supply* We recommend that builders using *Forced-air Supply* ventilation install ducts within conditioned space and offer buyers the option of paying for a dehumidifying supply ventilation unit to improve comfort.

6.3. Mixed and Hot Arid Climates

In these climates, we recommend production builders use *Multi-port Supply* ventilation. For builders using *Forced-air Supply* ventilation, we recommend installing ducts in conditioned space and offering buyers the option of paying for a forced-air fan with a variable-speed integrated-control motor (ICM).

7. CONCLUSION

Incorporating energy-efficient construction in U.S. production homebuilding is a task that requires the development of affordable and effective residential ventilation systems. Our investigation estimates the costs of residential ventilation systems in the U.S, offers a method to evaluate ventilation systems and their impact on the indoor environment, and provides usable information to the building community.

8. ACKNOWLEDGMENTS

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VENTILATION AND COOLING

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System Safety Analyses of the Performance of Mechanical Ventilation Systems - the quantitative approach

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1 Synopsis

A method for estimating the reliability of mechanical ventilation systems in dwellings has been developed. The analysis is based on component level reliability models interconnected by so called fault-tree schemes. A simplified model for maintenance is included. The analysis procedure is applied on an central exhaust ventilation system and on a central supply and exhaust ventilation system with heat recovery. For each system, three different quality standards have been defined and combined with three levels of maintenance. Work has also been done on collecting relevant input data, e.g. expected life-time values for ventilation components. The result of an analysis can be presented as a figure showing the reliability of the total system as a function of time. The result can also be presented in a compressed form giving the mean and minimum reliability values for a certain time period. Finally, a classification routine is proposed. This will transform the resulting mean and minimum values into a five graded classification scale.

2 Introduction

At the 1996 AIVC Conference, the basic principles of a method for estimating the reliability of mechanical ventilation systems was outlined (Kronvall, 1996). The concept of the analysis was based on general methods for system safety analysis, e.g. Rau, 1992 and Salem et al., 1976. However, no qualitative data were presented and the principles were shown only for a simple mechanical exhaust ventilation system. During the last year, extensive work has been done, not least in order to come up with quantitative data and an evaluation procedure for assessing the performance of different mechanical ventilation systems in terms of reliability.

The paper forms part of the Swedish contribution to the work of IEA-Annex 27 "Evaluation and Demonstration of Domestic Ventilation Systems".

3 Model for Reliability

3.1 Reliability on Component Level

The following model assumes that the life time of each component in a system has a normal distribution, i.e. it can be given a mean life time (m) and a standard deviation (s). Inserted into the general model of reliability this means that the reliability of the component (R) over time (t) can be expressed as follows, (Råde & Westergren, 1995):

if $t < m$ then; $$R(t,m,s) = \frac{1}{\left(1 + \exp^{-p(t,m,s)}\right)}$$

else $(t \geq s);$ $$R(t,m,s) = 1 - \frac{1}{\left(1 + \exp^{-p(t,m,s)}\right)}$$

Reliability = a probability (which is time depending)

where; $$p(t,m,s) = \frac{|t-m|}{s} \cdot \left[1,5976 + 0,070566 \cdot \left(\frac{t-m}{s}\right)^2 \right]$$

T = lifetime of a component

Depending on the type of component and its operational conditions, different values of *m* and *s* may be set. Figure 1 shows an example on how the reliability as a function of time depends on these values.

For reliability studies over time, the use of the so called Weibull distribution is more or less the standard procedure. In this work however, the normal distribution is used; the reason for that being that the input parameters for this model, mean life time and standard deviation, were assumed to be easier to understand and interpret for the practitioners than the more sophisticated parameters used for the Weibull model.

In this kind of component reliability analysis, *reliability* means *the probability that the component performs in such a way that the design flow rate within a certain interval is maintained.*

A simplified model for describing the influence of maintenance has also been incorporated into each component model. It assumes that after each maintenance interval the component is "as good as new". However, this is not shown in figure 1, but in figure 5.

greate ground dev, slope

mean large lifetime
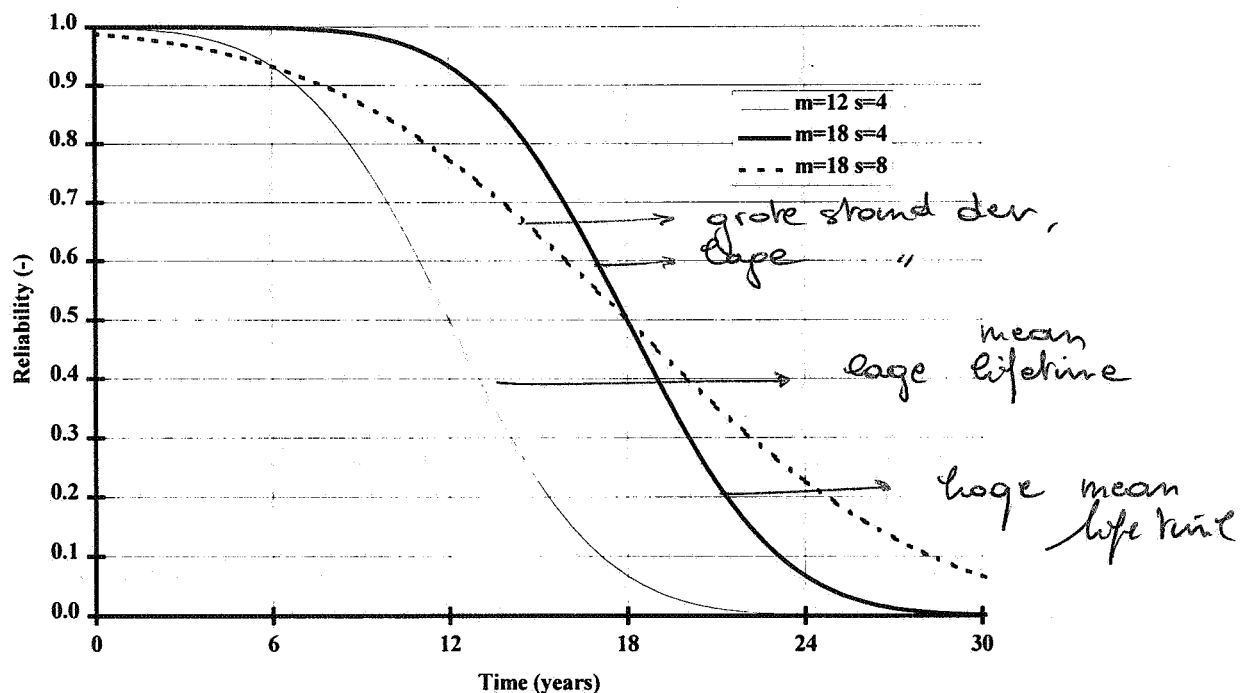
huge mean life time

Figure 1 Reliability as a function of mean life time (*m*) and standard deviation (*s*).

50 % reliability

41

3.2 Application on System Level

By connecting component models in a fault-tree scheme the analysis can be extended to a system level (Kronvall, 1996). This has in the present work been done on both an exhaust ventilation system and on a supply and exhaust ventilation system. The latter system including a heat exchanger is schematically shown in figure 2.
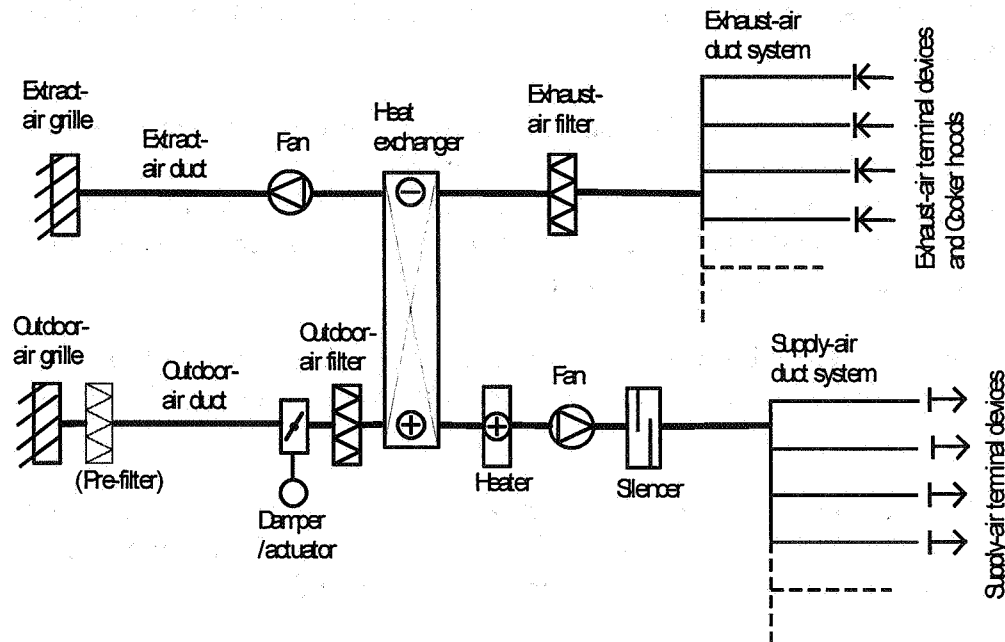
Figure 2 Central mechanical supply and exhaust ventilation system - Multi family house

The more simple exhaust ventilation system is schematically shown in figure 3.
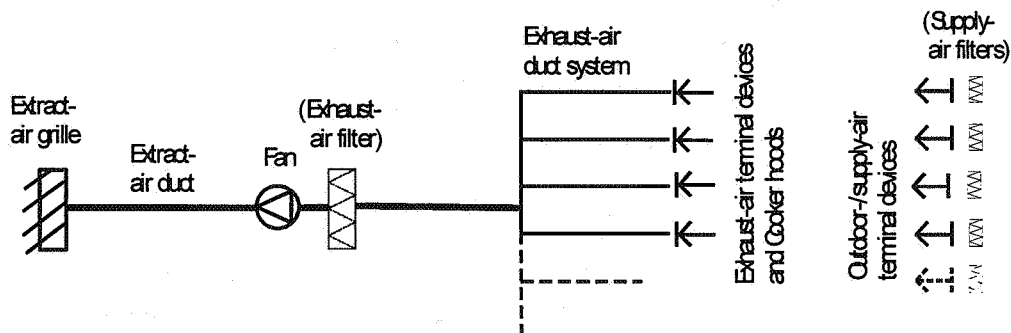
Figure 3 Central mechanical exhaust ventilation system - Multi family house

The fault-tree scheme for the exhaust system is shown in figure 4. The fault-tree for the supply and exhaust system, not shown, is more complex but built up in a similar way.

42
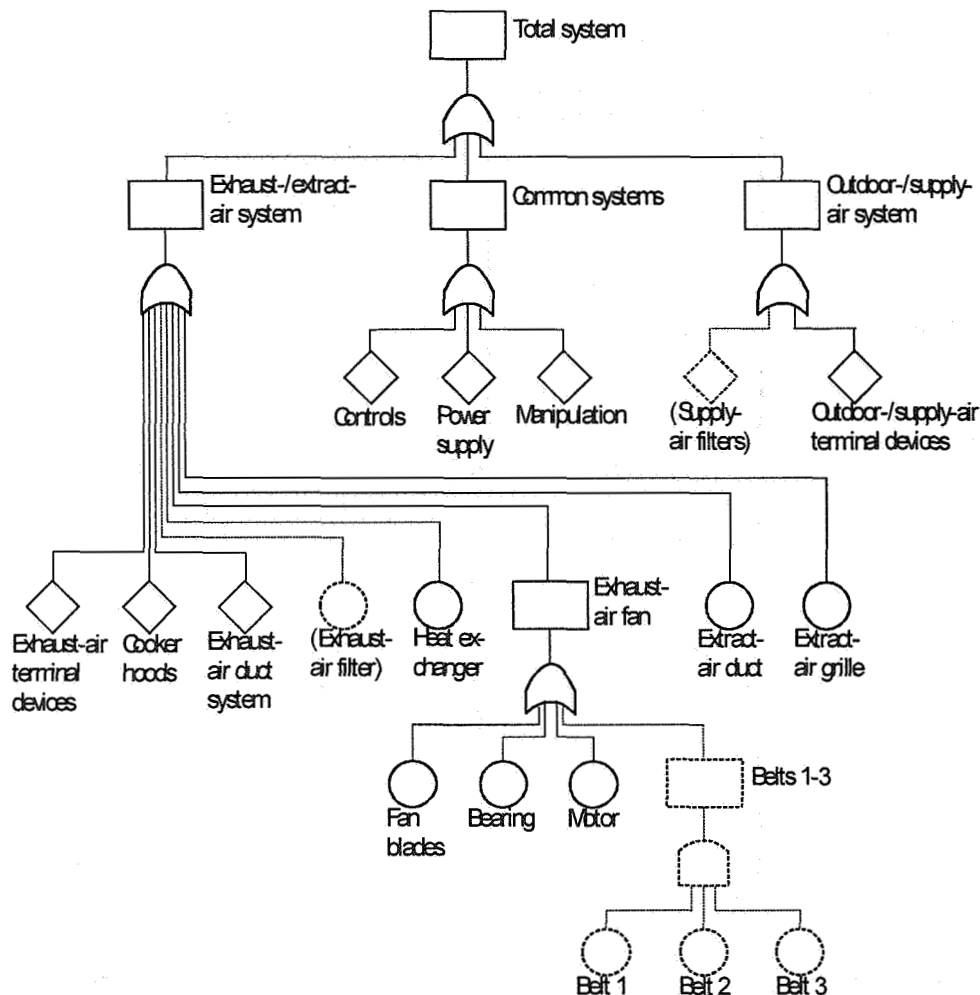
Figure 4 Fault three - Central mechanical exhaust ventilation system

For each type of system, three different quality standards have then been defined and combined with three different levels of maintenance intensity. This means a matrix with a total of nine different combinations for each ventilation system. An example of input data used for the exhaust system in this present work is shown in table 1. Relevant data have been collected from published and orally transferred empirical experiences from maintenance people and other researchers working with reliability or related matters, e.g. Myrefelt, 1996 and Gröninger & van Paassen, 1997. Certain basic data regarding maintenance origin from a Dutch investigation, (Op't Velt, 1997).

Table 1 Central mechanical exhaust ventilation system - Multi family house

Average system	* Mean life time (years)	Standard deviation (years)	** Maintenance intensity Interval (years)		
			High	Medium	Low
Outdoor-/supply-air components					
Outdoor-/supply-air devices	3	0.6	0.25	0.5	1
Outdoor-/supply-air filter	Not installed !				
Exhaust-/extract-air components					
Exhaust-air terminal devices	2	0.6	0.25	0.5	1
Cooker hoods	2	0.6	0.25	0.5	1
Exhaust-air duct system	10	2	4	6	8
Exhaust-air filter	Not installed !				
Fan blades	6	1.5	2	3	4
Bearing	18	3	8	12	16
Belt	8	2	4	6	8
Motor	20	4	8	12	16
Extract-air duct	20	4	8	12	16
Extract-air grille	20	4	8	12	16
Controls	12	2	4	6	8
Manipulation	50	20	4	8	16
Power supply	R=	0.999			

*   If no maintenance is performed

**   After each maintenance interval, the component is assumed to be "as good as new".

4   Results of application

The result of each combination in the matrixes can be presented in a figure showing the estimated reliability for the system as a function of time. An example of such a presentation is shown for the average system combined with medium maintenance in Figure 5.

The result is further evaluated by calculating the mean value and minimum value of the reliability for a time span of thirty years. The result for each system can be summarised in a matrix. The resulting matrixes for the two systems studied in this present work is shown in table 2 and table 3.

Reliability (-)

1.00
0.90
0.80
0.70
0.60
0.50
0.40
0.30
0.20
0.10
0.00

— · — Exhaust-air system

—— TOTAL SYSTEM

0   5   10   15   20   25   30

Time (years)
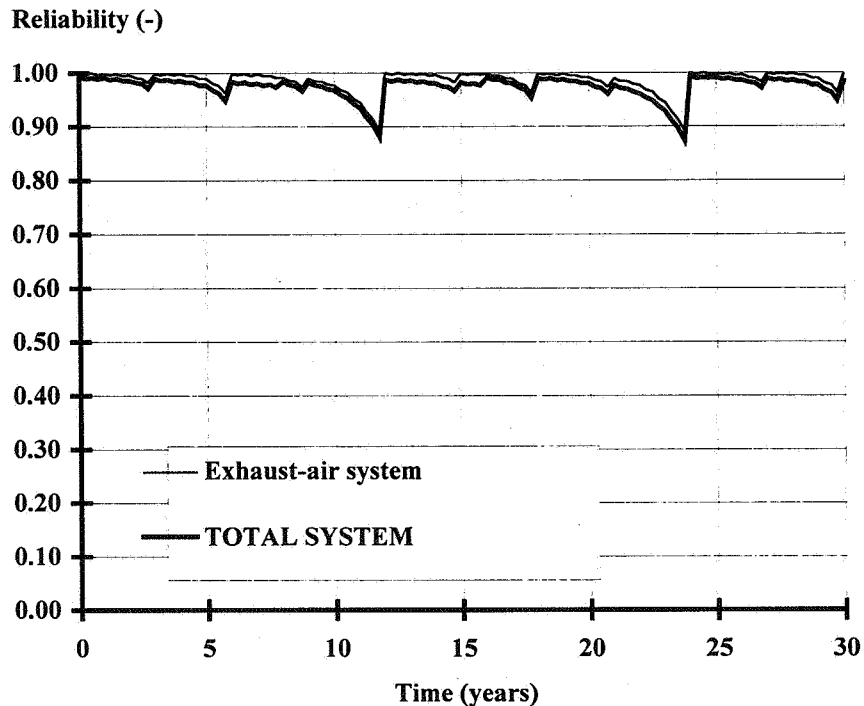
Figure 5 Exhaust ventilation - Average system - Medium maintenance

Lifetime of components vary (handwritten)

Table 2 Summary - System reliability. Exhaust ventilation system

Type of system	Reliability	Maintenance intensity		
		High	Medium	Low
Poor system	Mean	0.98	0.89	0.65
	Minimum	0.92	0.42	0.01
Average system	Mean	0.99	0.97	0.89
	Minimum	0.98	0.87	0.32
Best practice	Mean	0.99	0.99	0.96
	Minimum	0.99	0.97	0.76

Reliability mean & minimum (handwritten)

Table 3 Summary - System reliability. Supply and exhaust ventilation system

Type of system	Reliability	Maintenance intensity		
		High	Medium	Low
Poor system	Mean	0.93	0.70	0.46
	Minimum	0.61	0.13	0.00
Average system	Mean	0.98	0.92	0.73
	Minimum	0.93	0.70	0.07
Best practice	Mean	0.99	0.97	0.91
	Minimum	0.98	0.90	0.48

45

5 Discussion

The resulting mean and minimum values (of the reliability for a time span of thirty years) given in the matrixes may then finally be evaluated and transferred into a single classification system. A routine for such a classification is proposed in table 4. This means that each cell in the system matrix is given a reliability classification in a five graded scale, from very poor (- -) to very good (+ +). The result of such a classification is given in table 5a-b for the two systems studied in this present work.

Table 4 Proposed classification for reliability of ventilation systems

Mean reliability	Minimum reliability									
	0.90-1.00	0.80-0.89	0.70-0.79	0.60-0.69	0.50-0.59	0.40-0.49	0.30-0.39	0.20-0.29	0.10-0.19	0.00-0.09
0.90-1.00	++	++	+	+	+/-	+/-	-	-	- -	- -
0.80-0.89		+	+	+/-	+/-	-	-	- -	- -	- -
0.70-0.79			+/-	+/-	-	-	- -	- -	- -	- -
0.60-0.69				-	-	- -	- -	- -	- -	- -
0.50-0.59					- -	- -	- -	- -	- -	- -
0.40-0.49						- -	- -	- -	- -	- -
0.30-0.39							- -	- -	- -	- -
0.20-0.29								- -	- -	- -
0.10-0.19									- -	- -
0.00-0.09										- -

Table 5a-b Results of proposed classification for the studied systems

a: Central exhaust ventilation			
Type of system	Maintenance intensity		
	High	Medium	Low
Poor system	+ +	-	- -
Average system	+ +	+ +	-
Best practice	+ +	+ +	+

b: Central supply and exhaust ventilation			
Type of system	Maintenance intensity		
	High	Medium	Low
Poor system	+	- -	- -
Average system	+ +	+	- -
Best practice	+ +	+ +	+/-

6 Conclusions

From the work performed hitherto, the following conclusions can be drawn:

[handwritten annotation: data from DUTCH verificat. industry]

- The general concept of system safety analyses can be used in order to assess the performance over time for mechanical ventilation systems.
- There is a great lack of knowledge on basic reliability data for ventilation system *[handwritten annotation: wennyp data]* components, e.g. life-times; only more or less "best guesses" can be used with the present knowledge. *[handwritten annotation: veel]*
- The reliability of a mechanical ventilation system is determined both by the technical quality of the system and the maintenance intensity.

[handwritten annotations: wil analyze bexyket; deze zeer belangrijk; invloeden]

7 Acknowledgements

This work was sponsored by the Swedish Council for Building Research under research contract no. 940309-6. The support is gratefully acknowledged.

8 References

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Natural Cross Ventilation for Refrigerative Cooling Reduction
in a Well Insulated Apartment

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SYNOPSIS

In this paper the energy impact of natural cross ventilation is examined conducting a set of cross ventilation experiments in a well insulated apartment of a 5-storey building. The experimental results compared with simulation results derived from the combined use of the multizone air flow model COMIS and the thermal model Suncode.

A 24-hour lasting natural cross ventilation experiment was conducted, to monitor thermal comfort ventilation mainly during the day and night time cooling ventilation. A short lasting cross ventilation experiment was also conducted to monitor the impact of ventilation in midday hours. Both experiments compared with a 24-hour lasting single-sided ventilation experiment. Surface and air temperature, heat flux in building elements, ambient wind and solar conditions were monitored.

The ventilation and infiltration phenomena in the above experiments were modelled in detail using 30min / 1h time-step, taking into account landscape, wind environment, the building's shape as well as the leakage characteristics of the building's envelope.

1   INTRODUCTION

The knowledge of ventilation flow rates for all the ventilation phenomena in a building is necessary to estimate space heating and cooling loads, to determine IAQ and to make sizing calculations of air conditioning equipment (Liddament 1996). The study of cross ventilation and single-sided ventilation cooling potentials is aiming at the parallel use these techniques within the frames of ventilation strategies in order to reduce refrigerative cooling loads, and to simulate the energy impact of the user's behaviour. The ventilation phenomena and the thermal phenomena are very close related and strongly interacting and it is therefore essential to combine ventilation and thermal modelling to achieve adequate simulation of building energy performance.

The main scope of these experiments is to monitor and analyse natural cross ventilation potentials for refrigerative cooling reduction of common building types, which found to be a matter of great importance especially in the Mediterranean climates. The use of air conditioning in Greece has dramatically increased during the last decade.

2   EXPERIMENTS

2.1   Description

The natural cross ventilation experiments presented in this paper are:

- a 24-hour lasting, in order to combine the two aspects of passive cooling ventilation: thermal comfort ventilation mainly during the day and night time cooling ventilation.

- a short (4-hour lasting) in order to monitor the impact of midday hours ventilation on building' s thermal charge, in other words a «bad» example of summer ventilation.

A 24 hour lasting single sided ventilation experiment was also conducted to be compared with the above mentioned experiments and to examine the potentials of these two passive cooling methods for refrigerative cooling reduction.

All the experiments are focusing on the same room (bedroom) of a 60 m^2 well insulated apartment, although all the apartment rooms as well the adjacent apartments and the staircase were closely monitored.

The apartment where the experiments conducted is a reference one (not a passive solar one) and is located at the 4th (top) floor of a residential building of the Solar Village, a housing project of Greek Workers' Housing Organisation completed in the early 90's in Pefki-Attica, consisting of 435 apartments distributed in 30 buildings, an energy center, a solar information center and a commercial and community center, (figures 1, 2).

Figure 1: Aerial view of Solar Village
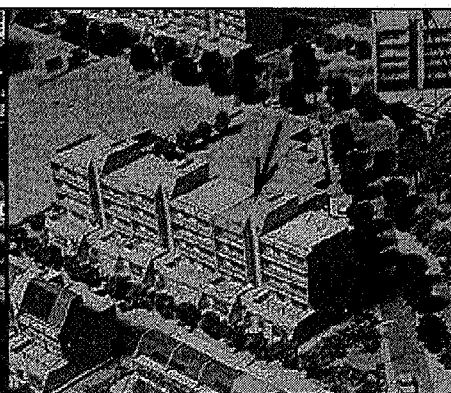
Figure 2: View of the building were the apartment is located

The measurements were performed implementing a the data-logger system that monitors the apart-ment. The sensors of the system monitored the indoor air temperature, the surface temperature of building elements (walls and roof) and the heat flux through the ceiling and the south walls (Interatom 1990, Koinakis 1992). Ambient conditions were monitored using wind velocity and wind direction sensors, temperature and related humidity sensors and a pressure sensor. Solar radiation (total hori-zontal and direct normal) was also monitored in order to provide the essential inputs for the thermal simulation. The ambient conditions as well the conditions inside the apartment were monitored every minute and half hour averages were produced, suitable for ventilation simulations. All the values were transformed to 1-hour averages for the simulation with the thermal model. This not very common technique was implemented to monitor closer the great variation of wind speed and direction and to validate ventilation simulation results with experimental data in a closer way.
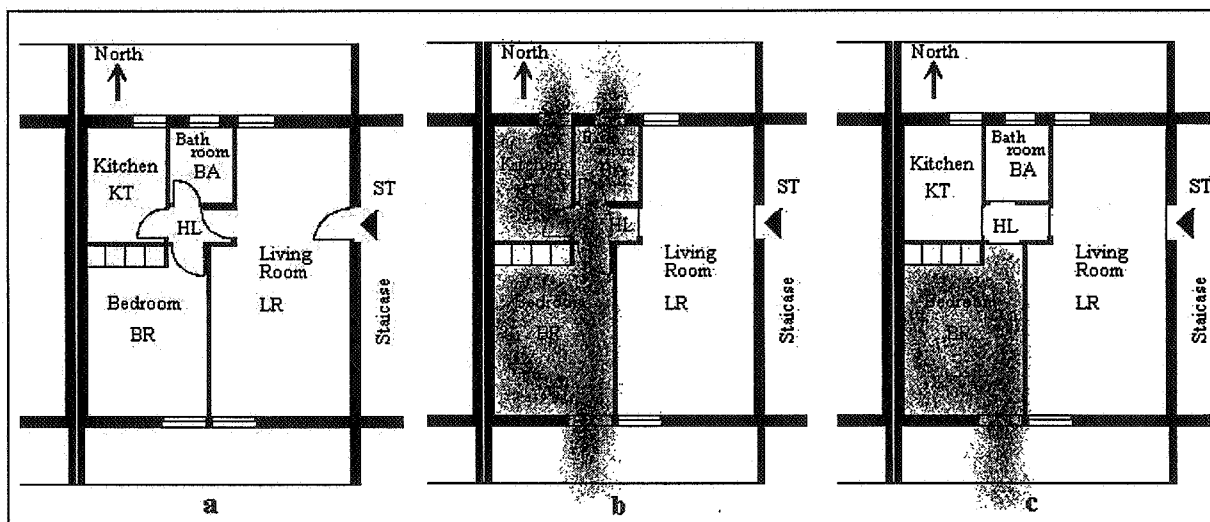
Figures 3 a., b, c: a. plan view of the apartment - zone names and symbols
b. configuration of the cross ventilation experiments and simulations (24h and 3h periods)
c. configuration of the single-sided ventilation experiments and simulations

The apartment where measurements took place could be considered as a typical modern apartment, been slightly more insulated than the rest modern apartments in Greece. The inner walls are of 10 cm thick brick covered with 2 cm plaster on each side and the outer envelope consists of 20 cm thick brick walls and 20 cm thick concrete frame and 15 cm thick slabs. It is heavily insulated with 10 cm of mineral wool in its external walls and with 10 cm polystyrene boards, on the roof. There is also 2

cm thick mineral wool insulation between the apartment and the adjacent spaces (the apartments at the same and the lower storey and the staircase). The apartment was uninhabited during the measurements and empty of furniture. The area of the room (bedroom) examined is 12.50 m^2 and its net volume approximately 31.88 m^3 .The balcony door on the south facade was a double glazed aluminium sliding door with a total area of 2.83 m^2 (WxH = 1.35x2.10 m) and a net opening area of 2.5 m^2 . The inner doors were common not weather-stripped 0.85x2.10 m timber doors.

2.2   24-hour lasting cross ventilation

The apartment was kept closed the last four days before the experiment, with short intervals of window and door opening in order to configure and test the data-lodging system. This was intended in order to increase the thermal charge of the building elements, to intensify the thermal transfer phenomena that occurred just after the beginning of cross ventilation and to monitor the heat rejection of the building elements of the room. to monitor .

The experiment started at 09:00 of Aug 25th and completed in the same time the next day. At the beginning air temperature in to the room was 26.7 ^0C while outdoor temperature was significantly lower not exceeding 22.1 ^0C . At the same time heat flux at BR south wall was lower than 0.5 W/m^2 indicating that thermal charge has been completed during the last days (figures 4, 5). Just after the formation of the cross ventilation flow due to the strong S-SW wind (first half hour average: 227 degrees, 10.44 m/s), a rapid decrease of air temperature started for the next $1^{1/2}$ hours at a mean rate of 1.5 $^0C/h$. Air temperatures at the two other rooms (KT & BA) were cross ventilation was formatted followed at a same time with zero time lag keeping almost steady temperature differences between them and BR: $\ddot{A}t_{BR}$ -$\ddot{A}t_{KT}$ ~ 2^0C and $\ddot{A}t_{BR}$ -$\ddot{A}t_{BA}$ ~ 1^0C. This is a result of orientation characteristics of the rooms. Heat flux at BR south wall followed immediately reaching 5.7 W/m^2 (thermal discharge) at 10:30. From that moment zone air temperatures started to increase following the ambient air temperature variations. Heat flux responded immediately to these variations forming maximums and minimums at the corresponding points.

The thermal charge/discharge was calculated based on experimental data, assuming uniform formation of thermal phenomena, implementing the equation:

$$Q_i = \int_t^{t+\Delta t} q_i dt$$

for 30 min time step. The results are presented in figure 5. From this analysis could be concluded that 24h excessive cross ventilation of the thermally charged room under the specific conditions lead to total thermal discharge equals to 2.08 KWh.
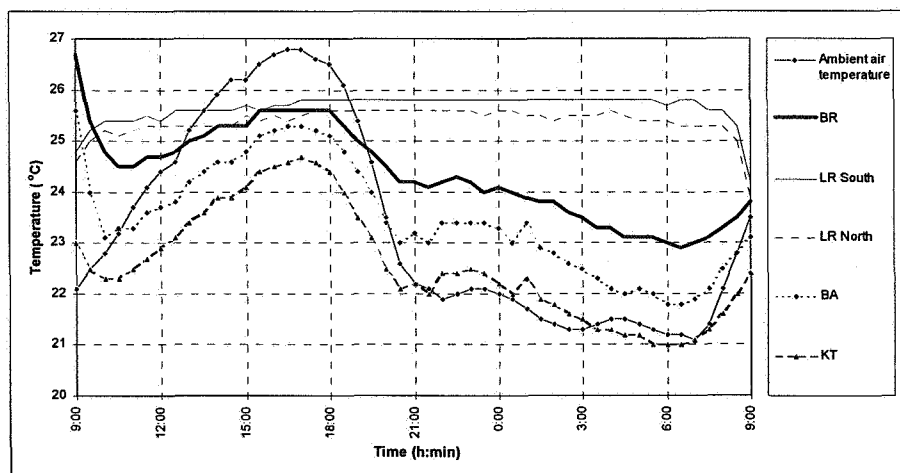
Figure 4: Air temperature variation during the 24h cross ventilation experiment
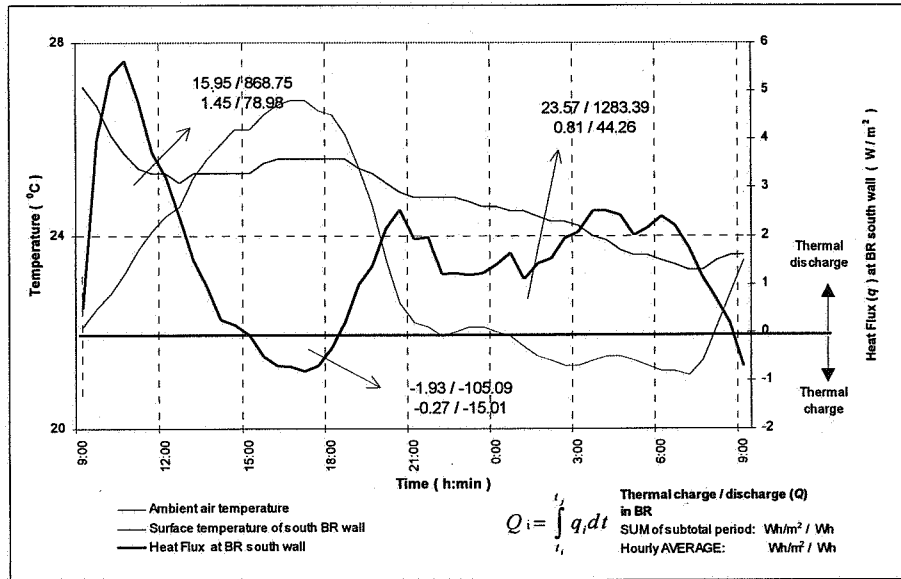
Figure 5: Heat flux and surface temperature variations during the 24h cross ventilation experiment

2.3   Short lasting cross ventilation

This 4-hour midday cross ventilation experiment was conducted between 10:00 and 14:00 in a hot summer day (July 31th). It is a «bad» example of ventilation not rare in Mediterranean climates, where airing could cause undesirable thermal charge. The aim of this experiment is to monitor the effect on inner temperatures and the role of thermal mass to prevent from excessive temperature variations.
As it is derived from figures 6 and 7, this 4-hout lasting noon ventilation leads to 0.5 ^{0}C temperature increase and to 214 Wh thermal charge. It should be mentioned that thermal charge was not too significant because of the heat capacity of the building elements as well as of the increased thermal charge of the room which remained closed the day before.
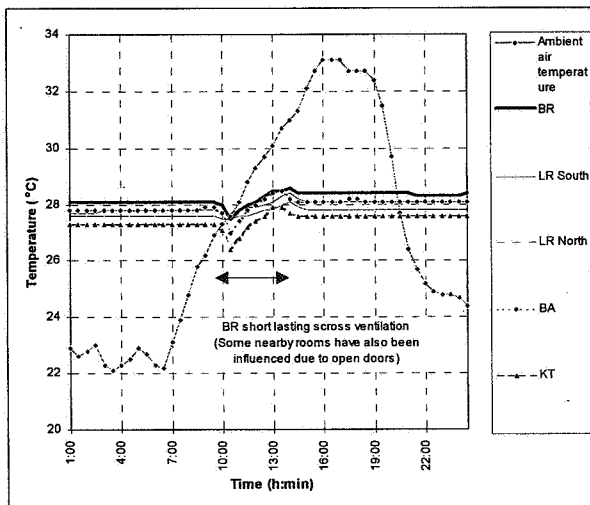
Figure 6:   Air temperature variations during the 4h cross ventilation experiment
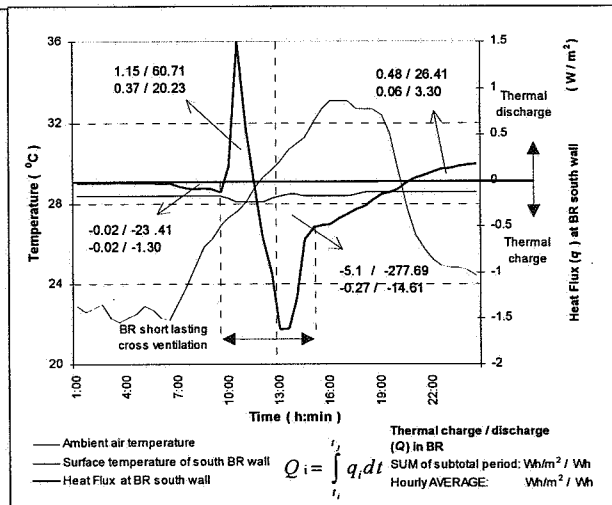
Figure 7:   Heat flux and surface temperature variations during the 4h cross ventilation experiment

2.4   24-hour lasting single sided ventilation

The 24 hour lasting single sided ventilation experiment was conducted to be compared with the above mentioned experiments and to examine the potentials of these two passive cooling methods for re-

frigerative cooling reduction. An extra reason is to evaluate the single-sided ventilation modelling (presented in paragraph 3), as the other two ventilation experiments.

In figures 8 and 9 the variations of air and surface temperatures and the heat flux at BR south wall are presented for a 3 days period, including the preceding day and the next day of the 24-hour single-sided ventilation, in order to monitor the disturbance caused to the thermal figures of the room. It can be seen that heat flux is significantly lower than cross ventilation heat flux, having maximum value 2 times smaller than the corresponding cross ventilation value. As a result room air temperature decreased about 1 ^{0}C after being almost steady for more than a day.
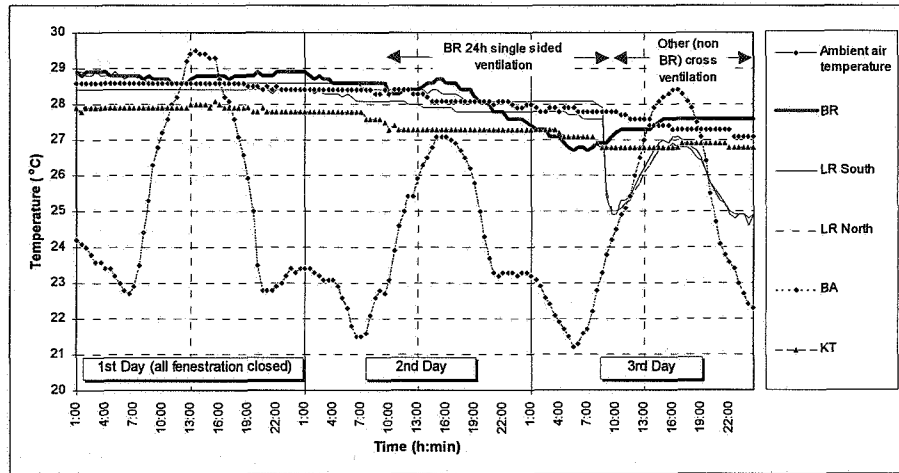
Figure 8: Air temperature variation during the 24h single-sided ventilation experiment

$$Q_i = \int_{t_i}^{t_j} q_i \, dt$$
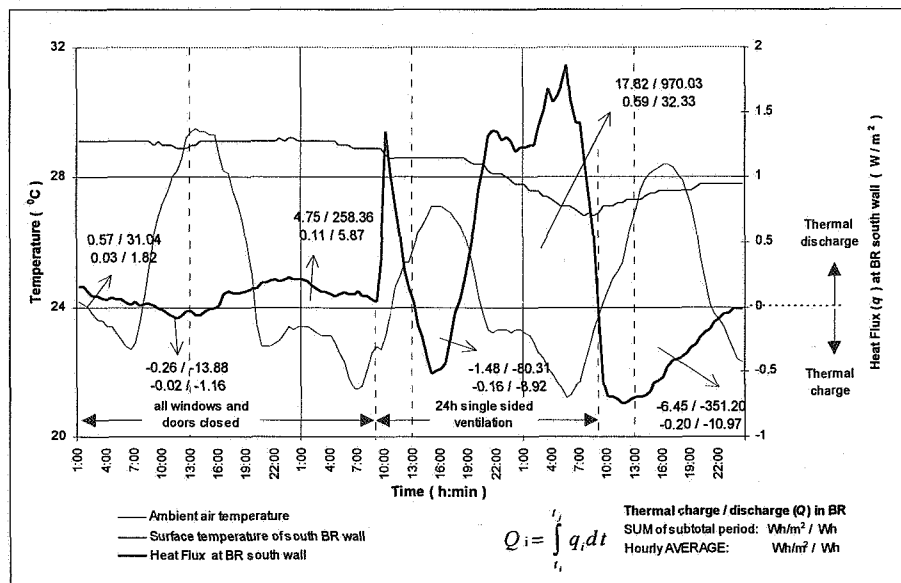
Figure 9: Heat flux and surface temperature variations during the 24-hour single-sided ventilation. experiment

3   SIMULATIONS

3.1   Integration of ventilation and thermal simulations

Energy flow due to infiltration and natural (cross and single-sided) ventilation is calculated separately for each zone of the apartment following the configuration of the experiments mentioned in paragraph 2 and only the values for the room tested above are presented. Energy flow is calculated in two ways:

- for incoming air flow from the ambient, due to infiltration and cross and single-sided ventilation
- for total incoming air flow (ambient and interzonal), due to infiltration and ventilation

Both air flows resulted from simulations implementing the air flow nodal model COMIS and the thermal nodal model Suncode (Feustel 1990 & 1995, Grosso 1992 and Weeling 1985).

The variation of meteorological parameters, especially wind speed and direction and ambient temperature cause systematic variation in infiltration and ventilation rates in building (ASHRAE 1985 & 1993, Blomsterberg 1990, Grosso 1992, Roulet 1991). Moreover the interaction of these parameters with user's behaviour expressed as the opening and closing of windows and doors using any possible combination, adds considerably on the complexity of the phenomena. From the one moment to the other completely different phenomena could take place in a common room starting for example with infiltration in a room closed for hours or days and overheated -as the one at the experiments- and then suddenly experiences massive air flow due to cross ventilation under strong 8-12 m/s wind. Soon after this would be followed by lower ventilation rates as a result of single-sided ventilation, possibly as a result of user's need for privacy or rest. It is therefore essential to examine in situ the ventilation phenomena and conditions in order to monitor every day's complexity and validate the capabilities of ventilation and thermal modelling.

For the purpose of ventilation and thermal simulations the apartment has been divided in five zones as presented in figure 3. An extra zone has been used for the staircase because it has found that significant interzonal air flow could occur between the living room (LR-zone) of the apartment and the staircase (Koinakis 1992). Although only the bedroom is examined in detail in this paper, it is preferable to produce a detailed ventilation model because the combination of the closed and opened doors and windows, could lead to various kinds of interzonal flows. On the contrary thermal modelling is a matter of less importance in this case study, because of the prevailing importance of the thermal phenomena. Air change rates in cross ventilation under strong wind often found to reach or exceed 100 ach and air flow velocity in selected points inside the apartment reached 1.8-2m/s making papers to blow. Although this seems excessive at first sight , it is a very common phenomenon in mild summer Mediterranean climate, where people find excessive air flow desirable and refreshing even at hot noon hours.

The adjacent spaces (the two adjacent apartment on the same and the lower storey were modelled as an extra zone using the temperature data of the data logger system. The building is assumed to be non conditioned and uninhabited as it was at the time of the measurements.

Assuming specific values for zone air temperature, the air change rate is calculated for each zone for the specific door and window configuration. This is used as an input for thermal simulation where new zone air temperature values are calculated. The air change rate is then recalculated using these new temperatures. This iterative procedure is repeated till convergence is less than the desired threshold (preferably from 0.01 to 0.05 ^{0}C).

The results derived from simulations were detailed including complete interzonal air flows, but only the results related to the examined room (bedroom) are presented in this paper.

The contribution of a special ventilation model as the COMIS model proved to be essential in modelling summer conditions where excessive ventilation is the main cause of heat transfer phenomena and adds considerably on the dynamic character of the phenomena. Therefore it is strongly recommended to implement an integrated calculation method for thermal simulation incorporated the estimation of ventilation and infiltration providing the essential wind and leakage data.

3.2   Simulation results compared with experimental data

The simulation results (figures 10, 11 and 12) were modified in order to show the energy flow per m^2 inner BR surface, following the guidelines mentioned in the previous paragraph. It derives that energy flow due to ventilation follows almost exactly the heat flux variation, proving that ventilation is by far the main cause of heat transfer phenomena.
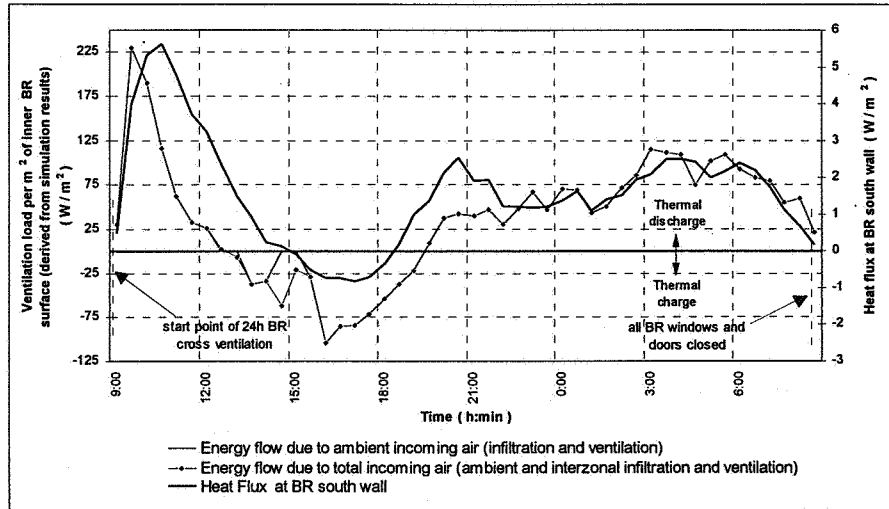
Figure 10: 24-hour lasting cross ventilation experiment: energy flow due to ambient air and total incoming air derived from simulation, versus heat flux experimental data in south BR wall.
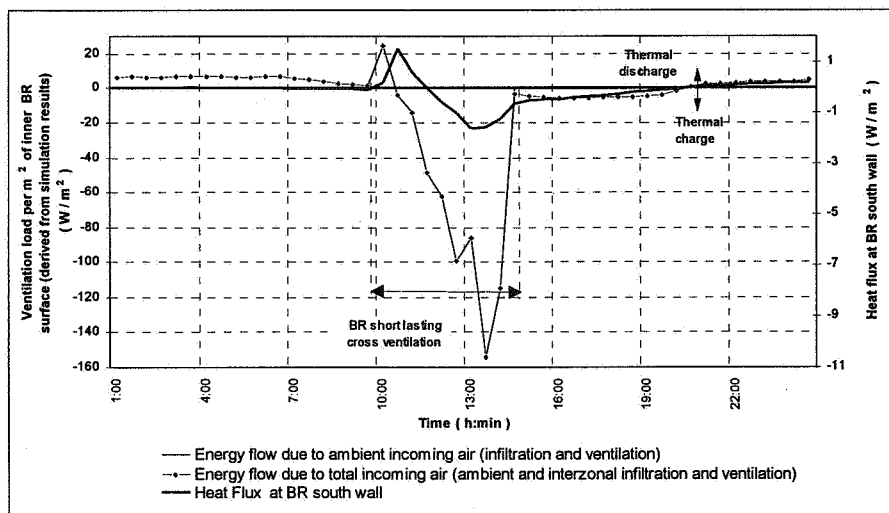
Figure 11: Short lasting cross ventilation: energy flow due to ambient air and total incoming air derived from simulation, versus heat flux experimental data in south BR wall.
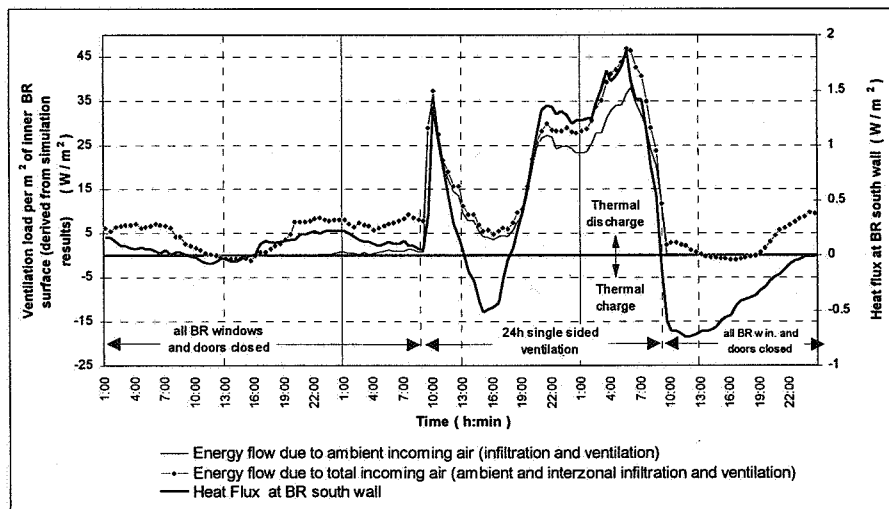
Figure 12: 24-hour lasting single-sided ventilation: energy flow due to ambient air and total incoming air derived from simulation, versus heat flux experimental data in south BR wall.

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4   CONCLUSIONS

As it derived from the three experiments and the corresponding simulations, air flow inside buildings is strongly influenced by (in order of importance):

- the configuration and the combination of opened and closed external and internal windows and doors and the time of the day they occurred,
- the outdoor wind conditions, mainly wind speed and direction,
- the thermal charge of the building elements and the ventilation period.

Air infiltration and leakage of the building's envelope tend to be negligible for buildings like the one examined in this paper in cases of summertime ventilation, because:

- increased ventilation combined with increased thermal mass placed in the inner building's surfaces proved to keep almost steady surface temperatures and heat flux. Only excessive air flow for more than a 4-hour period results significant temperature and heat flux change.
- the way that common Mediterranean buildings where built restricts infiltration flow paths only through the cracks of the external windows. The rest of the external building envelope is rather solid and airtight (Papamanolis 1996).

Heat flux in external building elements appear to form a sinus-like curve which was kept almost undisturbed by infiltration flow rates regardless the wind magnitude. As soon as ventilation begins, heat flux follows step by step and in zero time lag the ventilation heat flow which resulted from the simulations.

Taking into consideration the total incoming flows (ambient and interzonal) proved to be quite important in cases of excessive ventilation rates . Ventilation heat flow due to total incoming air (ambient and interzonal) follows heat flux in building's envelope more accurate than ventilation heat flow due to ambient incoming flow only.

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VENTILATION AND COOLING

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Hardware and controls for natural ventilation cooling

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Summary

This research is part of project NATVENT^{TM}, a concerted action of nine institutions of seven European countries under the Joule-3 program. It aims to open the barriers that blocks the use of natural ventilation systems in office buildings in cold and moderate climate zones.

Natural night-time ventilation cooling is a very effective means to remove the heat, accumulated in the building fabrics during office hours. Moreover, it requires no energy at all. Cooling with natural ventilation has it limits; more than 6 air changes per hour have no more cooling effect. So precautions have to be taken to reduce the heat gain during the day by limiting the glass area to approximately 40% and using effective sun shading devices. The internal heat load is also limited to 25 W.m^{-2}. Hardware for night cooling are the traditional types of windows and trickle ventilators. To obtain the most benefit of night cooling, automatic control is essential. For this the control strategies is of utmost importance. Specially for night-time ventilation cooling developed control strategies are incorporated in the predictive control, cooling day control, setpoint control, slab temperature control, degree hour control.

1. Introduction

For cooling, much higher air flow rates are required than for only refreshing the indoor atmosphere. These flow rates are not always available during the daily office hours, because, for reasons of comfort, the air velocity is limited. At night, without any occupation, these restrictions do not apply and much higher air velocities are allowed. To obtain the most benefits of natural night-time ventilation cooling, automatic control is essential. Automatically the windows or vent openings are closed, when the temperature has reached its setpoint, or when danger of ingress occurs or when wind speed exceed a given speed limit. In well designed office buildings with internal loads not exceeding 25 W.m^{-2}, automatic controlled natural ventilation with night cooling control will satisfy [1,2]

First a review of components applicable for natural night-time ventilation cooling, that are available in the market or under development will be given. This will be followed by a discussion of complete systems. Because the importance of automatic control with night-time ventilation, also an overview is given of strategies that can be applied for an optimal use of natural night-time ventilation cooling.

2. Cooling by natural ventilation

Natural ventilation is a free source of cooling at our disposal. The wind and the temperature difference between the outdoor and the indoor are the main driving forces for the transport of this free cooling medium. Unfortunately, during office hours, for comfort reasons,

the indoor air velocity is limited to 0.15 m.s^{-1}. At night, this constraint is not there and the windows can be more opened, but precautions have to be taken to prevent burglary. Cooling with natural ventilation has its limits. More than 6 air changes per hour (ach) have no more cooling effect [2]. So during the day the heat gain must be as small as possible. It can be stated that in buildings with maximum heat accumulation, 40% window area and with an effective automatic outside sun shao...g device, night-time ventilation cooling can keep indoor temperatures at an acceptable level with an internal load not exceeding 25 W.m^{-2} [1,2,6,7]. So only with automatic control the full benefits of natural cooling can be obtained.

3. The basic ventilation patterns

Three basic ventilation patterns cooling can be distinguished: a. single sided ventilation, b. cross ventilation and c. natural or fan assisted cross ventilation with an atrium or shaft.

3.1 Single sided ventilation

Single sided ventilation occurs, when the inlet and the outlet windows are placed in the same wall, see figure 1. The inlet and the outlet openings can be at different heights (left), or form one opening (right). The available air movement on the windward side is about 10% of the outdoor wind velocity at points up to a distance one sixth of the room depth from the window. Beyond this, the velocity decreases rapidly [3]. On the leeward side less air movement is produced. The ventilation is induced by the turbulence of the outside air that increases with higher wind speeds [2].

Figure 1   Single sided ventilation

3.2. Cross ventilation

Cross ventilation has a much higher ventilation rate than single sided ventilation. One configuration is shown in figure 2 (top); the air flow goes via the windows of the windward sided rooms and through vents near the ceiling via the corridor to the leeward sided rooms. Another configuration is shown in figure 2 (bottom): the air flow enters the windward sided rooms near the ceiling. There it sticks to the ceiling due to the Coanda effect and deep inside the room it drops to the floor where it goes to the corridor via the crack under the door. The first configuration is more comfortable for a summer situation because the fresh air passes the living zone, but for the winter situation the second configuration is better, because it prevents draught from entering the living zone. The disadvantage however is the transportation of the polluted air to the leeward sided rooms. However, this does not apply for night ventilation.

Figure 2   Cross ventilation

3.3 Natural or fan assisted cross ventilation with an atrium or shaft
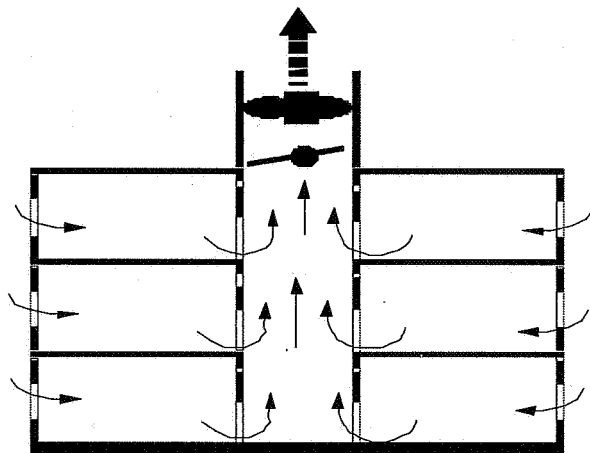
Figure 3 Fan or shaft assisted cross ventilation

With an atrium in the centre of the building the office rooms around this atrium are also cross ventilated, see figure 3. Due to the buoyancy forces in the atrium the air leaves the building from chimneys in the roof. The same pattern occurs also when there is a shaft in the building. But here we must observe stringent fire regulations. The air flow can be increased by adding an electric fan in the shaft or chimney. For a building with more floors the flow differs on the different floors due to the changing active height of the chimney. This depends on the wind speed, wind direction and temperature difference between indoor and outdoor.

3.4 Situation in some European countries on night-time ventilation cooling

In Switzerland research on night-time cooling is focused on optimising night-time cooling strategies for chilled ceilings. A pilot project of the Union Bank of Switzerland investigates passive night-time cooling by ventilation flaps and external blinds for solar protection, automatically set by a Building Management System [6]. In Norway many new office buildings are atrium buildings with glassed sun space between the buildings. These are normally naturally ventilated through motorised openings (shutters) in the roof. These openings are equipped with temperature and rain-sensors and therefore can be controlled for night-time cooling. Another system that include night-time cooling is the "ThermoDeck" system. It is a ventilation system integrated in the concrete structure of the building. Normally the floor have hollow core slabs where the supply air is circulated before entering the room [7]. In The Netherlands [1, 2, 8, 9] and The United Kingdom [10, 11, 12, 13, 14, 17] the research on natural ventilation with night cooling in office buildings is gaining momentum and many natural ventilated buildings with natural or fan assisted night-time cooling are being realised. Results of PASCOOL, a JOULE funded project, are collected in PASCOOL CD ROM [19].

4. Components for natural night-time ventilation cooling

For cooling, much higher air flow rates are required than necessary for refreshing the indoor climate. So in general the same components can be used: the traditional types of windows such as: inside/outside turning windows, trap windows, turning-trap windows, Louvre (glass) windows, vertical/horizontal sliding windows, etc. These components are all day life in building business, but as so not suitable for night cooling in office buildings because of security reasons and human behaviour. Besides keeping intruders out, these components must also prevent inrain and papers from flying all over the place. For optimal benefits of night cooling, automatic control of these components is necessary. Essential parts for night cooling are actuators and a automatic control system. The actuators can be connected to the window with a rod mechanism (figure 4), a special designed mechanism for Louvre windows (figure 5) or a chain (figure 6).
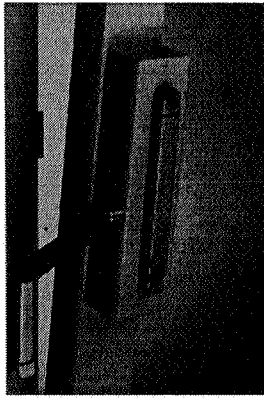
Figure 4 An actuator
with a rod mechanism
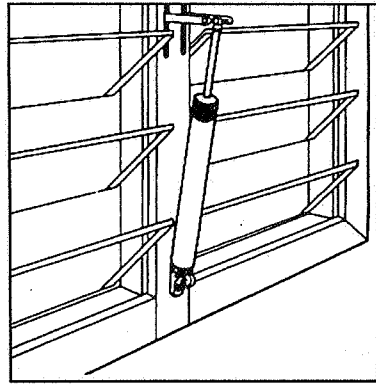
Figure 5 An actuator connected
to a Louvre window
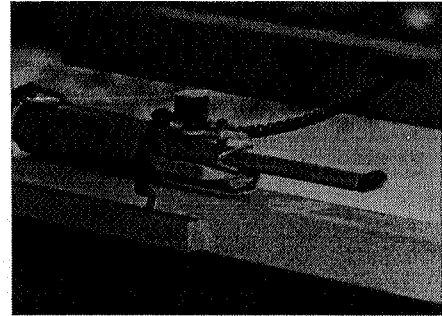
Figure 6 A chain connects the
the actuator with the window

At present, electronic controlled trickle ventilators, see figure 7, are already available on the market. Each ventilator is equipped with a print board, a duct, a sensor and a servo motor to control the moving sleeve. The air speed in the duct is measured based on the cooling time of a heated metal wire. The output of the sensor is converted into an electronic signal. This is passed to the controller which steers the servo motor to adjust the sleeves in the right position, such, that the air flow is constant, nominal 10-18 $[m^{-3}.s^{-1}$ per m length]. This ventilator is very suitable for night cooling because it is easy to connect it to a building management system. The capacity can be increased by adding a mechanical exhaust fan. No exact figures are available regarding the energy consumption and the life span of the servo motor. The price is comparable to a mechanical ventilation system.

Figure 7   The electronic trickle
            ventilator

A newly developed natural ventilation system, as applied in a new domestic housing project, called Urban Villa [15], is worth mentioning here, because this system is also applicable for office buildings. Cooling is provided by a combination of an opening in the parapet and an outside turning trap window above the vision window, see figure 8. The dimensions of the opening under the parapet is approximately 1 * 0.5 m. At the outside this is covered by an aluminium grill with fine small-mesh wire netting for insects. At the inside this can be closed by a well isolated hatch that can be manually opened with a spindle and positioned in any desired position. The air flow enters the room via the external louvres and leaves the room via the two upper trap windows, each also measuring approximately 1*0.5 m. A ventilation rate of 6 ach can be obtained in a summer night.
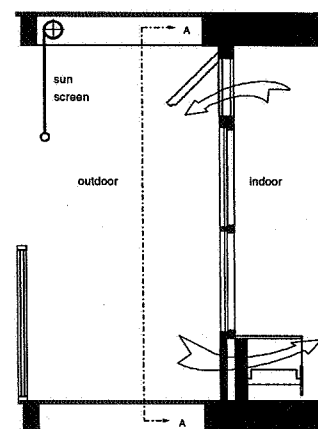
Figure 8 Vent openings in the
parapet and above the vision
windows
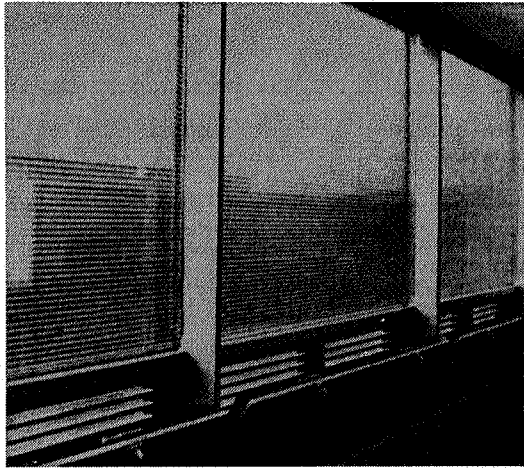
By the refurbishment of Regent House in Great Britain [15] is used the same principle. Around the whole perimeter of the building, below the fixed vision windows, mesh screens with external louvre (dimension 850*600 mm) provide 24-hour security and weather protection. Inside bottom hung hatches keep the cold out in winter. Above the vision windows are place cord operated trickle ventilators.

Figure 9   *A similar principle is used by the refurbishment of Regent House in London, England.*

A new development is an integrated self inducing trickle ventilator in the facade [16]. The advantage of this design is that the air flow is introduced into the room just below the ceiling; thus optimal use of the Coanda effect is guaranteed. The longer the air flow sticks to the ceiling, the deeper the air penetrates into the room and the more time is available to heat the cold air before it drops to the living zone. Slit 'B' can be adjusted manually or mechanically by valve 'O', so that the ventilation rate can be adjusted according seasonal needs. Noise reduction canals can be incorporated, see figure 10.
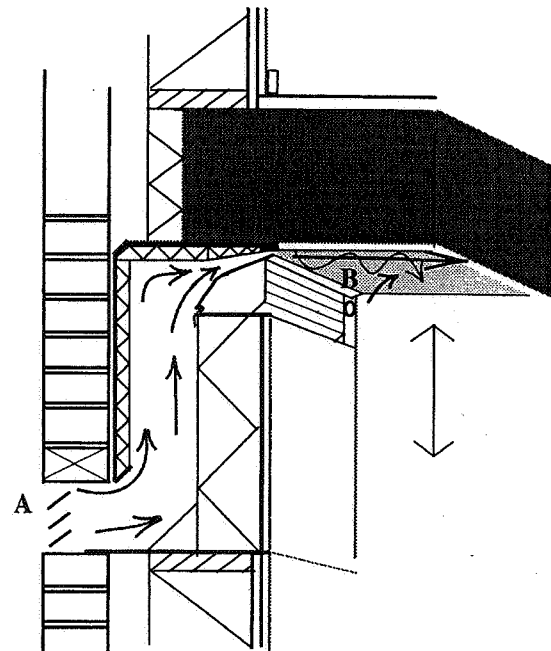
Figure 10 Integrated trickle vents in the facade with a noise reduction canal.

5.   Automatic controlled systems for natural night-time ventilation cooling

It has been mentioned in the preceding paragraphs, that optimal benefits from night-time cooling in office buildings are only possible with automatic control. Although developments in this field are very intensive, complete ventilation systems with night-time ventilation cooling designed for office buildings are not yet available on the market. Two systems having control features will be discussed here. These are the Passive Climate System and the Window Master.

5.1   The Passive Climate System

The Passive Climate System is an integrated system and consists of a central computer, local controllers and a weather station. The three most essential parts, the outdoor sun shading device controller, the room controller for the heating and the actuators for the windows are already available on the market.

Figure 11   The Passive Climate System
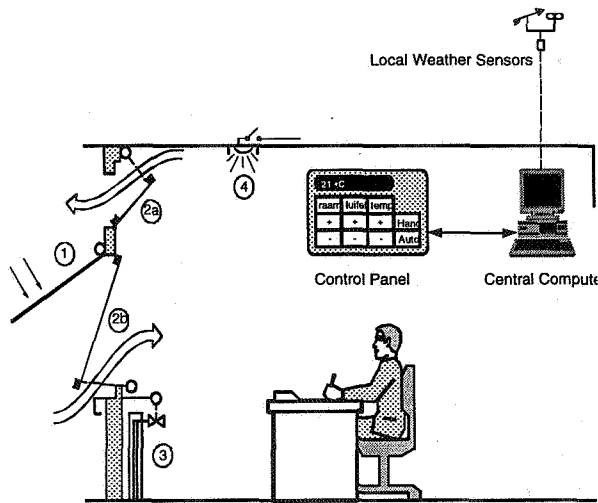
The central controller co-ordinates the information flow between the weather station and these local controllers. The system allows the occupant to overrule the control actions of the central controller during office hours with a remote controller. After working hours the central controller regains control and resets the manual set control actions of the day. A kind of predictive control action takes over to precool the office building during the night.

5.2   The Window Master

The Window Master is developed for ventilation purposes and security. This system consists of a central controller and a number of local controllers or under stations. They are connected by a local network. The central controller controls all windows of the whole building .Window Master with a restricted number of control functions is already on the market. This system is very suitable for controlling inside turning trap windows. In the trap position the window is automatically controlled by the system. During office hours the occupant can disconnect the spindle from the actuator (see figure 6) and operate the window manually. Before leaving the office it has to be reconnected again to the actuator before the central controller will regain control. With an infrared remote control unit the occupant can overrule the centrally realised action and open/close the motorised windows according its own wishes. This system is very suitable for night ventilation. Developments are going on to add more functions to this system (indoor temperature control with a timer and precipitation sensor for the outdoor sun shading device).
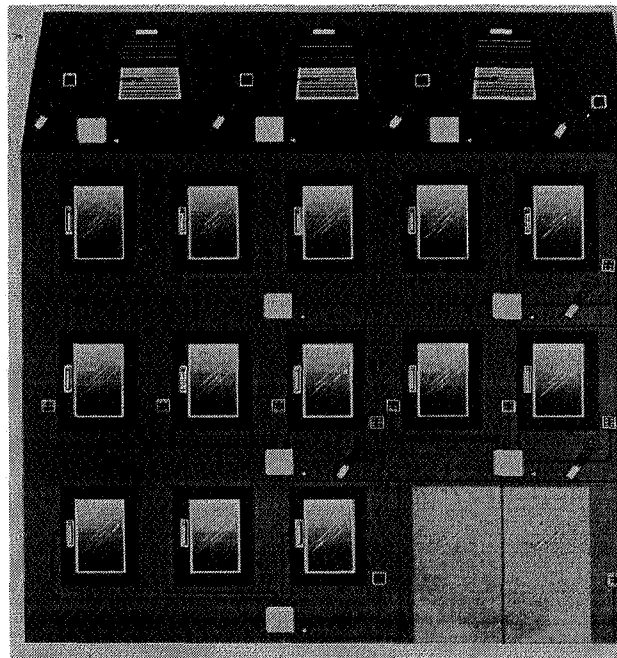
Figure 12   Window Master

6.   Control strategies for night cooling

A review of hardware components for night cooling is not complete without mentioning the control strategies. To mention some strategies that includes night cooling: predictive control, cooling day control, setpoint control, slab temperature control and degree hour control

6.1 Predictive control strategy [8]

Only in the summer the pre-cooling mode of the predictive controller is activated. This is done by a mathematical model that is identified from the measured data. With this model and the weather prediction, it calculates at what time the windows must be opened at night in order to cool the building in such a way that it will not be too hot the next day. Afterwards the prediction is compared with the real condition and the next prediction is adapted to these results. Wind and rain interlocks are utilised to prevent water ingress and damage due to high air velocities. A low external temperature interlock (e.g. 12 ^{0}C) is also provided.

6.2 Cooling day control strategy [2]

The predictive control can be simplified by using a rule based prediction: There are 2 setpoints, the upper temperature setpoint , θ_{SPH}, is for cooling, and the lower temperature setpoint, θ_{SPL}. is for heating, The strategy is based on the occurrence of a cooling day. This is defined as the day, that the indoor temperature during working hours exceeds θ_{SPH}. When a cooling day is detected, the windows are opened for night cooling. They are closed when the indoor temperature drops below the night setpoint temperature, $\theta_{SPH-night}$. If this is the second cooling day, $\theta_{SPH-night}$ is decreased one step, f.i. form 24 to 23 ^{0}C. If the same cooling situation occurs the next day, then $\theta_{SPH-night}$ is decreased again until finally the minimum value of $\theta_{SPH-night}$ = 18 ^{0}C is reached. When there is no cooling day, $\theta_{SPH-night}$ will be increased step by step until 24 ^{0}C is reached again. The windows stay closed.

Wind and rain interlocks are utilised to prevent water ingress and damage due to high air velocities. A low external temperature interlock (e.g. 12 ^{0}C) is also provided.

6.3 Setpoint control [13]

The mean outdoor temperature for a specific time interval during the afternoon is determined. In the event that this is above the "precool initial setpoint" (e.g. 20 ^{0}C) and the indoor temperature is greater than the outdoor temperature, precooling will starts after office hours. This will continue until the zone temperature drops to the minimum allowable space temperature, say 16 ^{0}C. All inlet and outlet vents will be closed. Due to the passive heating process of the building the indoor temperature rises again, say to 19 ^{0}C, at which point the inlet and outlet vents are again opened. This cooling and heating process continues until such time as the "preheat" period is reached. Wind and rain interlocks are utilised to prevent water ingress and damage due to high air velocities. A low external temperature interlock (e.g. 12 ^{0}C) is also provided to prevent any risk of condensation.

6.4 Slab temperature control strategy [13]

This precooling strategy aims to cool the slab to a predefined slab temperature setpoint during the night in order to offset the heating gains of the next day. This strategy is applicable for mixed-mode operation. In the event that the space temperature is more than, say 0.5 ^{0}C above the cooling setpoint (e.g 23 ^{0}C), passive cooling utilising the casement windows will be maintained in order to reduce the internal space temperature. This amount of cooling is controlled by the internal air temperature. At a predetermined time, and providing that the building is to be occupied the following day, the slab temperature is compared with the required slab temperature setpoint and if this is higher, then the slab cooling will commence. Passive cooling is initially used to facilitate cooling of the slab and this is controlled by the slab temperature setpoint. Wind and rain interlocks are utilised to prevent water ingress and damage

due to high air velocities as well as a low external temperature interlock (e.g. 12 ^{0}C) to prevent any risk of condensation.

In mixed-mode control, if at the start of the low electricity tariff period the slab temperature has not achieved the slab setpoint, the time is calculated that the fan assisted cooling is enabled in order to achieve the slab temperature setpoint by the end of the low tariff period. If the building is to be unoccupied for more than 24 hours then at the end of the occupancy period all the plant will shut sown. Natural ventilation will be employed to maintain the space temperature conditions. At say, 18:00 hours before the next occupancy period, the precooling strategy detailed above is initiated for slab cooling.

6.5   Degree hours control strategy [13]

This precooling strategy aims to maintain the equilibrium between the building fabric temperature and the space temperature. The daytime heat gains is estimated by measuring the degree hours of heating. This is defined as the number of hours that the temperature is above the chosen setpoint, totalled for all the hours in the period. The decision as to precool or not is based upon the number of hours that the internal temperature is above the room temperature setpoint. If at the end of the occupied period the degree hours are greater than say, three degree hours and the internal temperature is greater than the external temperature then the decision is made to precool the building during that night. Normal wind and rain interlocks still apply as well as the provision that the external temperature is above the low limit setpoint (12 ^{0}C) to prevent condensation.

7   Conclusions

For natural ventilation cooling much higher ventilation air flow rates are required then necessary for refreshing the indoor atmosphere. Because of comfort requirements, these air flow rates are always possible during working hours in an office. During office off hours, these restrictions do not apply anymore and higher air flow rates are allowed. However, other dangers, such as burglary, rain and dust ingress have to be accounted for.

For natural ventilation cooling, the same components can be applied as for traditional ventilation. But to obtain optimal benefits from night cooling, automatic control is essential. For this actuators and control systems are important hardware.

New developments are the electronic trickle ventilators, actuators with rod, chain or a mechanism to the windows and integrated ventilation systems

Control strategies for integrated systems such as predictive control, cooling day control, setpoint control, slab temperature control, degree hour control, are under development.

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VENTILATION AND COOLING

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(Title)

A STUDY OF WINDOW LOCATION AND FURNITURE LAYOUT TO MAXIMIZE THE COOLING EFFECT FOR AN URBAN TAIWANESE APARTMENT BY NIGHT VENTILATION

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A STUDY OF WINDOW LOCATION AND FURNITURE LAYOUT TO MAXIMIZE THE COOLING EFFECT FOR AN URBAN TAIWANESE APARTMENT BY NIGHT VENTILATION

SYNOPSIS

The year-round climate of Taiwan is warm and humid. Apart from the hottest months in summer, there are four months suitable for nocturnal ventilation to acquire indoor cooling. The urban Taiwanese apartments are small due the limited usable land. To maximize the spatial use, a relative large occupant-defined space is developed. This space can be divided into two to three sub-spaces with wall units or smaller pieces of furniture when needed. Based on a previous study in a typical occupant-defined space, some wintertime design principles of furniture layout to achieve high indoor air quality were obtained. To provide an overall picture of the natural ventilation design for such a space, this study investigates the impacts of window location and furniture layout on the summertime indoor thermal comfort and air quality by night ventilation. Different furniture layouts have neglected effects on indoor thermal comfort when the layout does not obstruct the primary supply air stream. More spatial divisions by wall units can help to removal CO_2 effectively by minimizing mixture among stratified thermal layers. Lower window location makes the penetration of supply air stream deeper into the room, which results in a cooler region away from the window. Lower window location achieves lower indoor CO_2 concentration level than higher window location.

INTRODUCTION

The subtropical climate of Taiwan is warm and humid. Apart from the hottest two months (July and August) of which the monthly average temperatures are over 28°C, there are four months the monthly average temperature are between 24°C and 28°C. During these months the daily temperature difference between the highest and lowest is around 6°C. For energy conservation, nocturnal ventilation could be a potential strategy for acquiring indoor cooling.

Due to limited usable land, the urban Taiwanese apartments are small. To maximize the utilization, a relative large space, called occupant-defined space, is developed. The occupant-defined space can be divided into two to three sub-spaces with wall units or smaller pieces of furniture when needed. It is important for designers to know how to arrange furniture in accordance with window locations to maximize the cooling effect and obtain high indoor air quality from night ventilation.

From a previous study [Chao et al. 1997] some design principles to sub-divide an occupant-defined space by wall units were obtained to achieve high indoor air quality in winter. High indoor air quality can be obtained if wall units can be arranged to be kept away from the primary air flow path. By studying the air flow pattern of a typical occupant-defined space two suggestions were made. To make the air circulate freely in a space, a gap should be left at the bottom of each wall unit. The height of a wall unit should be restricted when the wall unit is beside a heat/pollutant source, simulating a person. The heat/pollutant source can hardly be reached by the supply air stream if the wall unit is too high.

To provide an overall picture of the natural ventilation design for a typical occupant-defined space, this study examines the summer indoor thermal comfort and air quality when night ventilation is applied The goal of this study is to maximize the nocturnal cooling effect by

investigating the impacts from both of window location and furniture layout on indoor thermal comfort and air quality.

RESEARCH METHODS

CFD simulations

This study was carried out by computational fluid dynamics simulations. Since thermal buoyancy effect is prominent in this study, the applied turbulence model is the renormalization group k-ε model [Yakhot et al. 1992]. It has been found that the renormalization group k-ε model is more advanced in predicting the thermal buoyancy effect than the standard k-ε model and the modified k-ε model [Chen and Chao 1996]. The renormalization group k-ε model has the same form as the standard k-ε model, except for the model coefficients. The model coefficients in the renormalization group k-ε model are:

$$(\delta_k, \delta_\varepsilon, C_{1\varepsilon}, C_{2\varepsilon}, C_\mu) = (0.7194, 0.7194, 1.42, 1.68, 0.0845) \tag{1}$$

In addition, the dissipation-rate transport equation has an additional source term R:

$$R = \frac{C_\mu \eta^3 (1 - \eta/\eta_0)}{1 + \beta \eta^3} \frac{\varepsilon^2}{k} \tag{2}$$

where $\eta_0 = 4.8$, $\beta = 0.012$ and the dimensionless parameter, η, is defined by:

$$\eta = S \frac{k}{\varepsilon}, S = (2 S_{ij} S_{ij})^{1/2}, S_{ij} = \frac{1}{2}(U_{i,j} + U_{j,i}) \tag{3}$$

The computations are conducted by PHOENICS [Spalding 1994], a commercially available CFD code, which is popular among ventilation engineers. The governing equations are solved in the finite-volume method with a staggered grid system. A hybrid scheme is used for the numerical solution. The algorithm employed is SIMPLEST [Spalding 1994]. As a convergence criterion, the sum of the normalized absolute residuals in each control volume for all calculated variables should be maintained at less than 10^{-3}. To prevent the numerical solution process from oscillating or diverging, three methods are used. They are under-relaxation for the continuity equation, false time-steps for the other dependent variables, and source-term manipulation which treats positive source terms explicitly and negative source terms implicitly. A non-uniform mesh system is used with the finer mesh located in the near-wall region or the place with a large gradient of variables.

Apartment unit

A typical occupant-defined space was chosen from an apartment unit (Fig. 1) located in an apartment complex in Taipei. The left side of this apartment unit, denoted by dash line, is the occupant-defined space (Fig. 1). Three kinds of window locations and three types of spatial division were investigated (Fig. 2). Case B and C were chosen because of their good performance in obtaining high indoor air quality by the effective removal of CO_2 [Chao et al. 1997]. Three lying persons, each generating 75 w of heat and 4×10^{-6} m^3/s of carbon dioxide (CO_2), were simulated. The room and wall temperatures are 30°C which are considered to be 4°C higher than the outdoor temperature (26°C). The simulation cases and the corresponding dimensions are shown in Table 1 and Table 2. All simulations were conducted in three dimensions in 39 x 20 x 23 cells.

Evaluation models
To assess the performance of each design option, the average indoor thermal comfort and the average indoor concentration of CO_2 are evaluated. Indoor thermal comfort is evaluated on both Fanger's Predicted Mean Vote [Fanger 1982] and his draft risk model [Fanger et al. 1988]. The Predicted Mean Vote is determined by three personal parameters and four environmental parameters. The three personal parameters are metabolism, external work, and clothing insulation. The four environmental parameters are air temperature, mean radiant temperature, mean air velocity, and partial water vapor pressure. The draft risk model is a function of mean air velocity, turbulence intensity, and air temperature. To obtain a 90% level of satisfaction in thermal comfort, the value of the PMV should be kept between -0.5 and +0.5. A 15% or lower level of dissatisfaction in draft risk is desirable. Indoor air quality is evaluated on the average pollutant concentrations in three zones. A lower pollutant concentration in the evaluated region indicate an effective removal of CO_2. Six regions, two regions in each zone, are chosen for evaluating indoor thermal comfort and air quality. The location for each region is shown in Fig. 2.

RESULTS AND DISCUSSION

From Table 3, 4 and 5 one can find that thermal comfort level around each heat/pollutant source (zone 1) for all 9 cases is similar. The thermal comfort levels in region 1 of three zones for all cases are higher than 0.5, the comfort level, which means warm in the space. The percentages of dissatisfaction due to drafts in six regions for all cases are well below 15%, a desirable criterion.

Furniture layout
In this study different ways of spatial division by wall units have neglected effects on indoor thermal comfort level. The possible explanation is that the arrangement of wall units in case B and case C does not obstruct the air circulation in the whole space [Chao et al. 1997]. For each wall unit there is a gap left at the bottom to make the supply air stream go deeper into the space. In addition, the wall unit #1 and #3 are made shorter to make air circulation easier.

Different layouts of wall units have prominent influences on the ease in removing CO_2. Case C, the case with three wall units, obtains the lowest indoor CO_2 concentration level comparing with case A and B. The same situation has been found in the previous study in which less supply air volume was considered [Chao et al. 1997]. The possible explanation is that wall units are help to prevent mixture among different stratified thermal layers. The stratified thermal layers are generated by thermal buoyancy effect. More mixture among different thermal layers makes the removal of CO_2 difficult and consequently, results in a high indoor CO_2 concentration level.

Window location
Type 1 window location (case A-1, B-1 and C-1) can make the supply air stream penetrate deeper into the room comparing with higher window locations. This situation can be observed from all three cases in Table 3 in which the PMV value is relatively lower in zone 3 than in other zones. The higher the window the shorter the penetration depth is. Therefore, type 3 window location (case A-3, B-3 and C-3) has the lowest PMV value in zone 1 than in other zones.

The higher the window location, the higher the indoor CO_2 concentration level will be. Type 1 window location obtains the lowest CO_2 concentration level than the other types. The reason is that the position of type 1 window is lower the heat/pollutant sources. Type 2 window location is at the same level of the heat/pollutant sources while type 3 window location is higher than the heat/pollutant sources. Since the thermal buoyancy effect is

dominant around the heat/pollutant sources, a lower window location (lower than the heat/pollutant sources) helps to preserve the stratified thermal layers generated above the heat/pollutant sources. By preserving the stratified pattern, the indoor CO_2 can be removed effectively. The weakness of higher window location can somehow be adjusted by the gap beneath a wall unit. As one can observe from Table 3, 4, and 5, the difference in CO_2 concentration between case C and the other two cases increases with a raising window location. This situation states that the gap left beneath each wall unit regulates the supply air stream to reach the heat/pollutant source at a lower level, which helps to preserve the stratified thermal layers pattern.

CONCLUSION

This study investigate the impacts of furniture layout and window location on indoor thermal comfort and air quality by night ventilation. It was found that different ways of spatial division by wall units have neglected effects on indoor thermal comfort if the layout does not obstruct the primary supply air stream. Since thermal buoyancy effect is the dominant force around the heat/pollutant sources, more spatial divisions by wall units can minimize the opportunity of mixture among stratified thermal layers, which makes the removal of CO_2 effectively. Lower window location makes the penetration of supply air stream deeper into the room, which results in a cooler region away from the window. The higher the window location the less penetration depth can be made. Thus, higher window location brings more cooling effect in the area close to the window than the area far away. Lower window location, lower the heat/pollutant sources, achieves lower indoor CO_2 concentration level than higher location for the thermal buoyancy effect is prominent around the sources. It was also found that the cooling effect by a 5 ach supply air volume at 26°C is not enough for a room with a 30°C wall temperature and three heat sources, each with 75W. Further study is needed to identify the proper season and supply air volume to provide indoor thermal comfort by night ventilation.

ACKNOWLEDGMENTS

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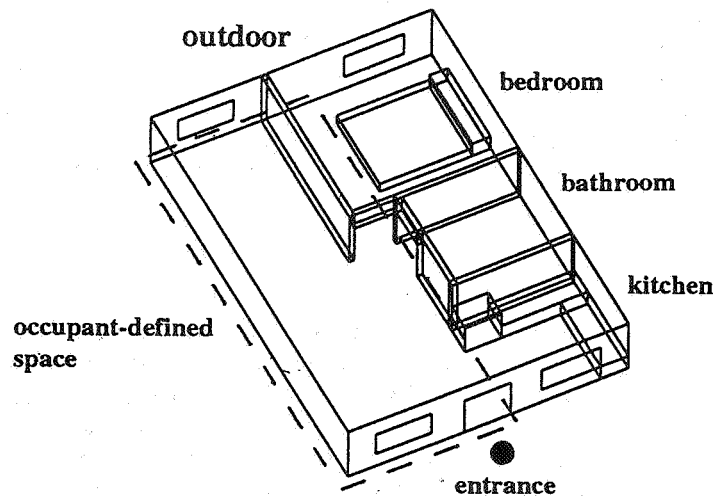
Fig. 1 Apartment unit.

Table 1 Simulation cases.

type	A			B			C		
case/window type	A-1	A-2	A-3	B-1	B-2	B-3	C-1	C-2	C-3
gap[1]	-	-	-	yes	yes	yes	yes	yes	yes
# of wall units	0			1			3		
ach[2]	5.								
inlet velocity (m/s)	0.2789								
inlet area (m^2)	0.36								
outlet area (m^2)	0.72								

[1]gap beneath wall units; [2]number of air changes per hour based on the volume of occupant-defined space.

Table 2 Dimensions and locations of windows and wall units.

DIMENSION (m)	X	Y	Z	LOCATION[1] (m)	X	Y	Z
space	8.74	3.42	3.00	type 1 inlet (A-1, B-1, C-1)	0.00	1.20	0.10
type 1 inlet (A-1, B-1, C-1)	0.00	1.20	0.30	type 2 inlet (A-2, B-2, C-2)	0.00	1.80	0.10
type 2 inlet (A-2, B-2, C-2)	0.00	0.60	0.60	type 3 inlet (A-3, B-3, C-3)	0.00	1.80	0.90
type 3 inlet (A-3, B-3, C-3)	0.00	0.60	0.60	outlet	8.74	0.00	2.10
outlet	0.00	2.40	0.30	wall unit #1	3.92	0.00	0.10
wall unit #1	0.40	1.20	1.40	wall unit #2	6.94	0.00	0.10
wall unit #2	0.40	1.20	2.00	wall unit #3	6.94	1.20	0.10
wall unit #3	0.40	1.20	1.10	block	3.92	2.40	0.00
block	3.92	1.02	3.00	heat / pollutant source #1	1.60	0.60	0.50
heat / pollutant source	0.20	0.20	0.30	heat / pollutant source #2	5.32	0.60	0.50
				heat / pollutant source #3	7.94	0.60	0.50

[1]measured from the origin to the lower left corner of each object, viewed from the direction of outlet.

1. inlet   3. wall unit #1   6. heat/pollutant source #1   9. zone 1, region 1   12. zone 1, region 2
2. outlet   4. wall unit #2   7. heat/pollutant source #2   10. zone 2, region 1   13. zone 2, region 2
             5. wall unit #3   8. heat/pollutant source #3   11. zone 3, region 1   14. zone 3, region 2
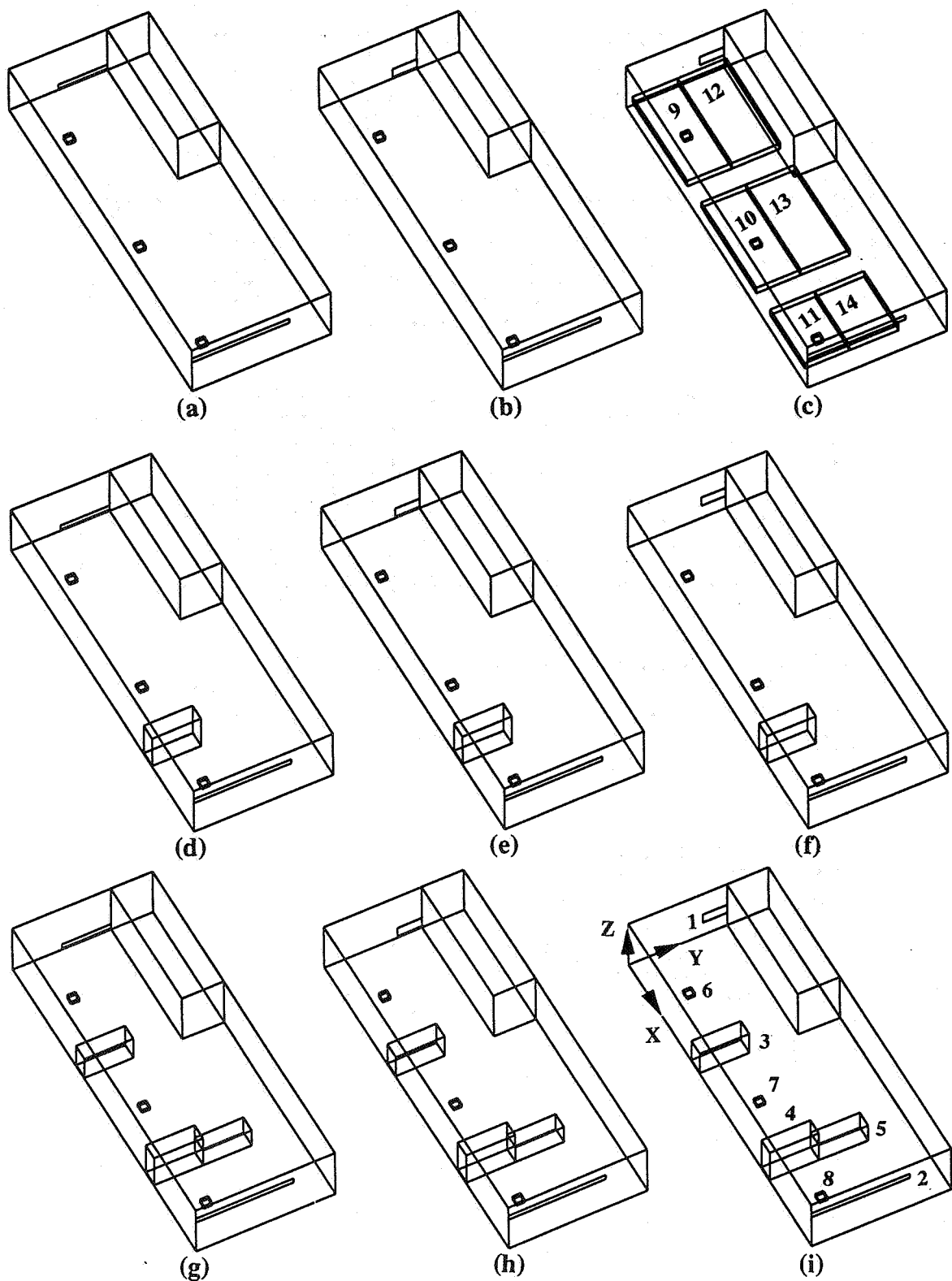
Fig. 2 Nine cases, (a) A-1, (b) A-2, (c) A-3, (d) B-1, (e) B-2, (f) B-3, (g) C-1, (h) C-2, and (i) C-3.

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Table 3 Performance of case A-1, B-1 and C-1.

type/case	A-1		B-1		C-1	
$T_{out}{}^{1}$ (°C)	30.08		30.07		30.03	
$C_{out}{}^{2}$ (ppm)	119.3		119.4		117.8	
thermal comfort in zone 1 (PMV)	1.36_1	1.33_2	1.38_1	1.34_2	1.35_1	1.33_2
thermal comfort in zone 2 (PMV)	1.32_1	1.28_2	1.32_1	1.29_2	1.31_1	1.28_2
thermal comfort in zone 3 (PMV)	1.29_1	1.24_2	1.31_1	1.27_2	1.32_1	1.27_2
draft risk in zone 1 (%)	0.73_1	0.60_2	0.24_1	0.51_2	0.07_1	0.25_2
draft risk in zone 2 (%)	0.48_1	1.65_2	0.06_1	0.69_2	0.17_1	0.06_2
draft risk in zone 3 (%)	0.11_1	0.41_2	0.17_1	0.00_2	0.13_1	0.01_2
CO_2 concentration in zone 1 (ppm)	14.79_1	6.42_2	14.96_1	8.12_2	13.92_1	7.94_2
CO_2 concentration in zone 2 (ppm)	10.36_1	4.40_2	16.24_1	7.51_2	13.94_1	7.94_2
CO_2 concentration in zone 3 (ppm)	10.28_1	3.14_2	12.70_1	4.85_2	12.32_1	4.59_2

$_1$lower left region in each zone; $_2$lower right region in each zone; $^{1}T_{out}$: average temperature at the outlet; $^{2}C_{out}$: average CO_2 concentration at the outlet.

Table 4 Performance of case A-2, B-2 and C-2.

type/case	A-2		B-2		C-2	
$T_{out}{}^{1}$ (°C)	30.05		30.04		30.03	
$C_{out}{}^{2}$ (ppm)	120.6		119.5		121.3	
thermal comfort in zone 1 (PMV)	1.35_1	1.29_2	1.36_1	1.30_2	1.31_1	1.25_2
thermal comfort in zone 2 (PMV)	1.33_1	1.28_2	1.30_1	1.28_2	1.27_1	1.24_2
thermal comfort in zone 3 (PMV)	1.30_1	1.25_2	1.33_1	1.28_2	1.32_1	1.27_2
draft risk in zone 1 (%)	0.91_1	1.63_2	0.63_1	1.66_2	0.09_1	1.19_2
draft risk in zone 2 (%)	0.81_1	1.84_2	0.40_1	1.11_2	0.41_1	0.21_2
draft risk in zone 3 (%)	0.42_1	0.70_2	0.16_1	0.00_2	0.11_1	0.00_2
CO_2 concentration in zone 1 (ppm)	24.53_1	17.89_2	22.64_1	17.25_2	16.15_1	8.04_2
CO_2 concentration in zone 2 (ppm)	18.73_1	12.25_2	21.76_1	11.94_2	11.31_1	6.28_2
CO_2 concentration in zone 3 (ppm)	18.26_1	9.84_2	16.62_1	9.68_2	11.69_1	4.60_2

$_1$lower left region in each zone; $_2$lower right region in each zone; $^{1}T_{out}$: average temperature at the outlet; $^{2}C_{out}$: average CO_2 concentration at the outlet.

Table 5 Performance of case A-3, B-3 and C-3.

type/case	A-3		B-3		C-3	
$T_{out}{}^{1}$ (°C)	29.88		29.86		29.89	
$C_{out}{}^{2}$ (ppm)	113.2		115.1		113.5	
thermal comfort in zone 1 (PMV)	1.31_1	1.16_2	1.32_1	1.16_2	1.20_1	1.09_2
thermal comfort in zone 2 (PMV)	1.33_1	1.27_2	1.27_1	1.27_2	1.25_1	1.22_2
thermal comfort in zone 3 (PMV)	1.31_1	1.26_2	1.36_1	1.30_2	1.35_1	1.28_2
draft risk in zone 1 (%)	0.34_1	2.62_2	0.27_1	2.71_2	0.52_1	2.78_2
draft risk in zone 2 (%)	0.64_1	1.65_2	0.93_1	1.08_2	0.52_1	0.55_2
draft risk in zone 3 (%)	0.69_1	0.12_2	0.14_1	0.00_2	0.17_1	0.03_2
CO_2 concentration in zone 1 (ppm)	33.99_1	27.68_2	38.29_1	32.39_2	21.56_1	19.17_2
CO_2 concentration in zone 2 (ppm)	39.19_1	29.12_2	40.40_1	33.39_2	25.02_1	21.19_2
CO_2 concentration in zone 3 (ppm)	39.91_1	29.74_2	36.31_1	31.39_2	26.19_1	20.46_2

$_1$lower left region in each zone; $_2$lower right region in each zone; $^{1}T_{out}$: average temperature at the outlet; $^{2}C_{out}$: average CO_2 concentration at the outlet.

VENTILATION AND COOLING

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USE OF SOLAR ENERGY FOR VENTILATION COOLING OF BUILDINGS

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SYNOPSIS

This paper discusses summer cooling of buildings by means of natural ventilation. Computational fluid dynamics is used to predict the ventilation rate in a room with a Trombe wall. The effect of Trombe wall insulation on the room thermal environment is investigated. It is shown that to maximise the effect of ventilation cooling, the interior surface of a Trombe wall should be insulated.

1 INTRODUCTION

A Trombe wall system consists of a massive storage wall and glazing. The massive storage wall serves to collect and store solar energy. The stored energy is transferred to the inside building for winter heating or enhances room air movement for summer cooling. Fig. 1 shows a Trombe wall for summer cooling of a room. During the operation, the buoyancy effect of air in the channel between the solar heated storage wall and glazing draws room air to the bottom vent. The air in the channel exits to the ambient through the top vent while cool outdoor air is drawn into the room through an open window in the opposite wall. Depending on the ambient temperature, this operation can be either for daytime ventilation or night cooling. If the summer temperature of outdoor air is not very high such that indoor air is warmer than outdoor air due to high heat loads through lighting and other sources, the Trombe wall can be used for ventilation cooling during daytime to reduce or eliminate the need for energy intensive refrigerative cooling. If the outdoor air is hot during the day, the Trombe wall can be used for night cooling by drawing the cool ambient air into the space and so removing heat from the interior of the building.

In this work computational fluid dynamics (CFD) is used to investigate the potential of a Trombe wall for summer ventilation of a room with solar heat gains and conduction heat transfer. The effect of insulation of Trombe wall on the indoor thermal comfort is assessed.

2 CFD TECHNIQUE

The CFD model consists of a set of governing equations representing the mean and turbulent velocities and enthalpy. The time-averaged steady-state air flow equations can be written in the following form

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

where ϕ is the flow variable, U_i is the mean velocity component in x_i direction, ρ is the air density, Γ_ϕ is the diffusion coefficient and S_ϕ is the source term.

The solution of the flow equations is based on the finite-volume TEAM code [1] and is validated against the experimental data for enclosures with Trombe wall geometries [2].

The CFD technique is applied to simulating ventilation cooling of an occupied room with a

Trombe wall system (see the schematic of the room in Fig. 1). The room is a test chamber at the University of Nottingham which has dimensions of 5 m long, 3 m wide and 2.4 high. The simulated Trombe wall is assumed to include a double glazing unit and a 0.3 m thick concrete wall with thermal conductivity of 1.4 W/m-K and insulated on the interior surface. It has the same width and height as the room and is situated at the south end of the room. The distance between the glazing and wall is 0.1 m. There are five 0.4 m X 0.1 m slots near the bottom of the storage wall and one single 3 m X 0.1 m slot at the top of glazing. Other room walls are insulated. In the north wall there is an openable window. In the summer ventilation mode, the outdoor air at a temperature of 20°C is assumed to enter the room through the window with an opening level of 1 m high and 0.5 m wide. It is further assumed that stack effect [3] is the sole driving force for air exchange between indoors and outdoors.

The storage wall solar heat gain is calculated from the daily mean total solar irradiance and mean solar gain factor on July 23. The daily mean solar irradiance on south wall is 165 W/m^2 [3] and mean solar gain factor for double glazing is 0.64. The room is occupied by two people with heat generation. The occupants' metabolic rate is taken to be 1.2 met and clothing level 0.6 clo. The heat gain due to lighting is 20 W/m^2 of floor area and uniformly distributed on the floor. The room is symmetrical and so only half of the room is used for simulation.

Solution of the flow field is considered to have converged when the sum of normalized residuals is less than 10^{-6} for enthalpy and less than 10^{-3} for other flow equations. Convergence is achieved after 10000 iterations. For a grid size of 68 x 40 x 27 for room length, height and half width, the CPU time for computation is 25 seconds per iteration on a Sun ULTRA server (Enterprise E3000 with three 250 MHz processors of 1 GB memory).

3   RESULTS AND DISCUSSION

Figures 2 and 3 show the predicted thermal environment on two vertical planes - symmetry plane and the plane through one of the occupant. It is seen from Fig. 2a that the outdoor air is drawn into the room through the window. The cool incoming air drops down to the floor due to the negative buoyancy effect and spreads over the floor. The buoyancy forces of air in the Trombe wall channel induce room air towards the bottom openings in the storage wall and eject the heated air in the channel through the outlet opening at the top of glazing. Because there is no strong air movement in the upper region of the room, the thermal plume created by the occupant reaches the ceiling (Fig. 3a and Fig. 3b). The predicted ventilation rate for the room is 103 l/s. This is well above the minimum fresh air requirement for two persons [3]. The predicted mean air velocity in the occupied zone (from floor to 1.8 high and 0.15 from walls) is 0.08 m/s. There exists a temperature gradient in the room but the mean gradient between 1.1 m and 0.1 m above the floor is only 1.1 K. The predicted mean air temperature in the occupied zone is 24.2°C. The mean radiant temperature varies from floor to ceiling due to heat gain from the floor but the variation is small (25.4°C on average) because of wall insulation (Fig. 2c and Fig. 3c). The predicted mean vote (PMV) [4] for the occupied zone is close to zero and the predicted percentage of dissatisfied (PPD) is 6.5%, within the comfort limit of 10% [5]. Thus, the average room environment is acceptable. Along the incoming air stream, air is slightly cool (PMV < -0.5) (Fig. 2d). This can be alleviated by adjusting the window opening level since the predicted ventilation rate is very high under the simulated conditions. However, since the air in the area where the occupant is situated is close

to a neutral temperature (i.e. PMV = 0) (Fig. 3d), such an operation may not be needed.

When the insulation on the interior surface of the storage wall is removed, part of the solar heat gain on the wall is transferred to the room through conduction and so less heat is utilised for generating the buoyancy forces in the channel of Trombe wall. The predicted mean air temperature in the occupied zone is increased to over 30°C due to the reduced ventilation rate and heat transfer from the storage wall by convection and radiation. The room is consequently far too hot for thermal comfort. However, the room thermal environment can be improved by opening extra vents in the north or other walls. Fig. 4 and Fig. 5 show the predicted room air conditions with the non-insulated Trombe wall and a slot vent near the top of the north wall whose size is the same as the outlet vent for the Trombe wall. The non-insulated warm storage wall induces an upward thermal plume inside the room (Fig. 4a) and when air flows towards the vent in the north wall it distorts the thermal plume created by the occupant (Fig. 5a). The predicted ventilation rate is increased to 167 l/s, of which 54% is induced by the Trombe wall. The mean air velocity in the occupied zone is also increased slightly to 0.09 m/s. The mean air temperature in the occupied zone is 23.6°C, which is slightly lower than that under the original room and Trombe wall conditions due to the increased ventilation heat loss over heat gain from the storage wall. Also, the vertical temperature gradient is increased, however it is still within the comfort limit of 3 K [5]. The average value of mean radiant temperature for the occupied zone remains the same as for the room with insulated storage wall but the distribution is not so uniform. The mean radiant temperature near the Trombe wall is higher than that near the north wall, resulting in the variation of mean radiant temperature principally along the horizontal rather than vertical direction. This is due to the increased temperature of the interior surface of storage wall. The mean temperature of the interior surface is 31.4°C compared with 24.6°C when the surface is insulated. The PPD value for the occupied zone is 7.1%. The room as a whole is also comfortable but the cool incoming air stream disperses to a larger area due to the higher air flow rate.

The above predictions are based on the occupants' clothing level of 0.6 clo. The clothing level could be reduced to 0.5 clo or lower for summer conditions. Hence the occupants could tolerate slightly higher outdoor air temperatures (e.g. 21°C) or internal heat gains (say 25 W/m^2) than used for the predictions without causing thermal discomfort provided that the interior surface of storage wall is insulated. In addition, opening more air vents can induce higher air exchange rates and thus allows the system to operate at even higher outdoor air temperatures. For example, with the storage wall insulated, when a vent is opened in the north wall in the same way as for the case with the non-insulated storage wall, the occupants with clothing level 0.4 clo will feel comfortable at outdoor air temperatures as high as 24°C. The predicted ventilation rate at the outdoor air temperature of 24°C is 160 l/s. The predicted mean air and radiant temperatures in the occupied zone become 26.4° and 27.5°C, respectively. The mean value of PPD for the occupied zone is 11.9% for the clothing level of 0.5 clo and decreases to 9.2% when the clothing level is taken to be 0.4 clo. If the clothing level has to be fixed at 0.5 clo, the maximum outdoor air temperature for acceptable indoor thermal comfort is 23°C (with the predicted mean air and radiant temperatures and PPD in the occupied zone of 25.5°, 26.7°C and 7.9%, respectively).

To demonstrate the effectiveness of the insulated Trombe wall for ventilation cooling, further predictions are made with the Trombe wall system removed but with the same amount of heat

gain imposed in the south wall as that from the non-insulated storage wall and with the slot vent opened at the top of both south and north walls. At an outdoor air temperature of 24°C, the predicted ventilation rate is reduced to 105 l/s (compared with 160 l/s for the room with insulated Trombe wall). The predicted mean air and radiant temperatures in the occupied zone reach 27.7°C and 28.7°C, respectively, and PPD is over 23% for clothing level 0.5 clo. However, when the outdoor air temperature is reduced by 4°C to 20°C, the predicted room thermal environment is acceptable with mean air and radiant temperatures reduced to 23.8°C and 24.9°C, respectively, and PPD for the occupied zone of 7.4%. Therefore, insulating the storage wall can provide ventilation cooling at higher outdoor air temperatures than without insulation (23°C compared with 20°C for instance).

4   CONCLUSIONS

In moderate sunny days, Trombe walls can be used for ventilation cooling. To maximise ventilation cooling, the interior surface of storage wall should be insulated. Additional vents should be provided to increase ventilation rates for high outdoor temperatures.

Trombe walls are however designed principally for winter heating. If ventilation cooling is the main purpose, it would be more effective to enhance and control the ventilation rate by using a solar chimney than a Trombe wall [2].

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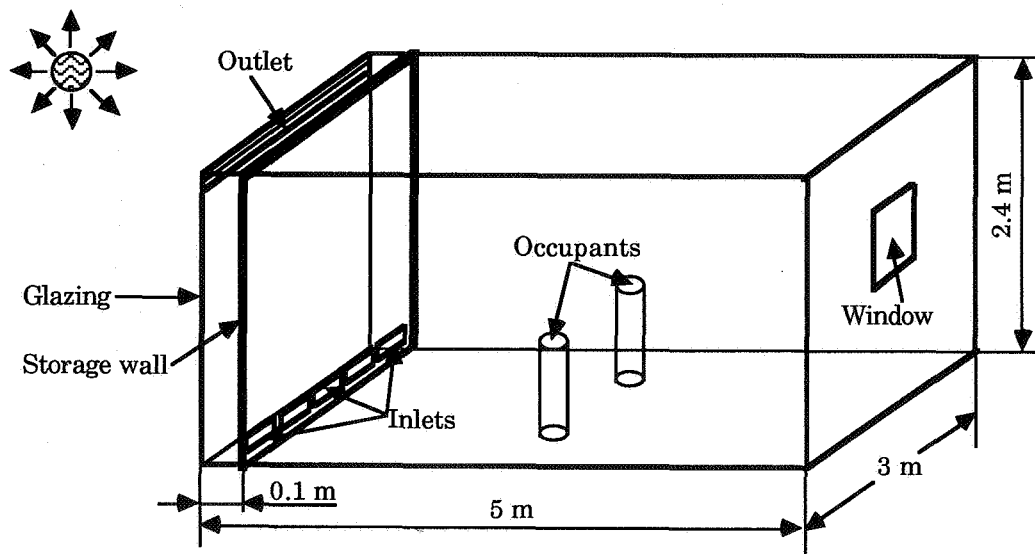
Fig. 1 Schematic diagram of a Trombe wall for ventilation cooling of a room
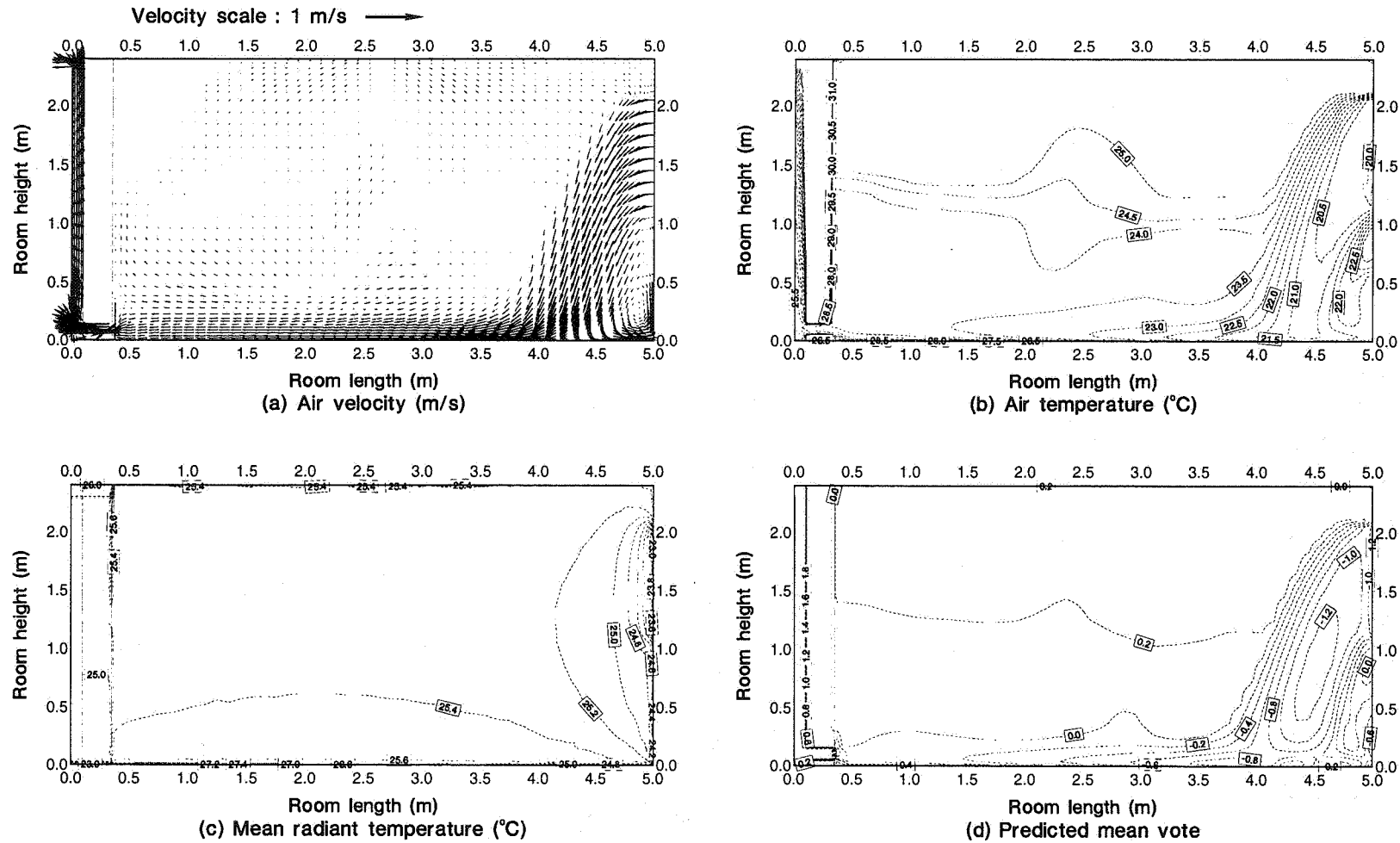
Velocity scale : 1 m/s ⟶

(a) Air velocity (m/s)

(b) Air temperature (°C)

(c) Mean radiant temperature (°C)

(d) Predicted mean vote

Fig. 2   Predicted air flow pattern and thermal environment on the symmetry plane
in the room with insulated Trombe wall

Velocity scale : 1 m/s ⟶

(a) Air velocity (m/s)

(b) Air temperature (°C)

(c) Mean radiant temperature (°C)

(d) Predicted mean vote
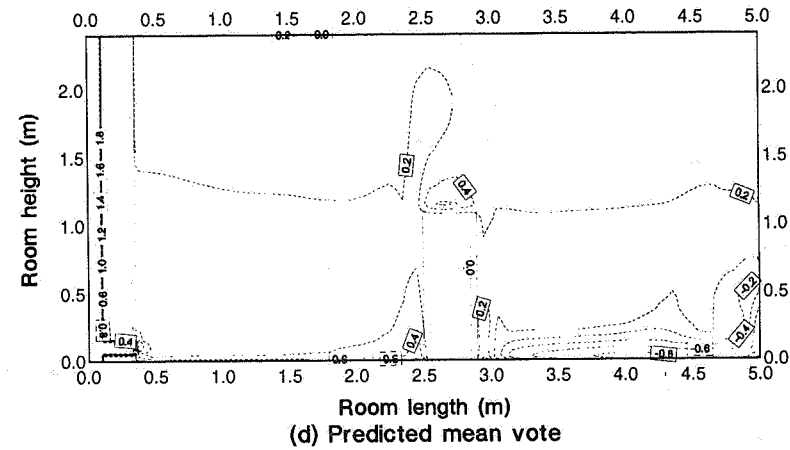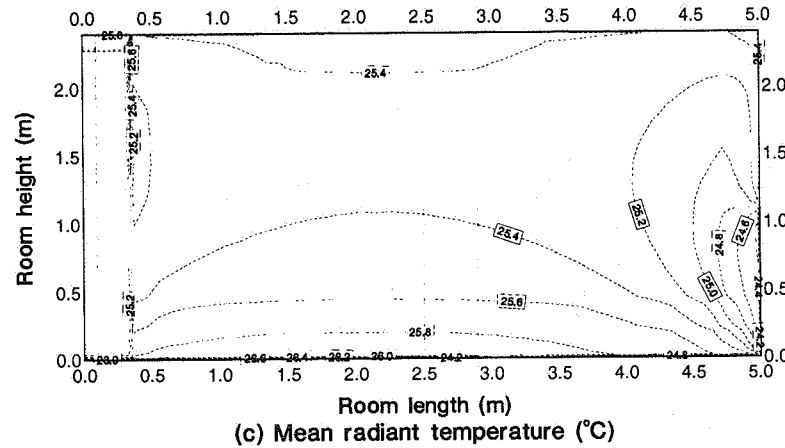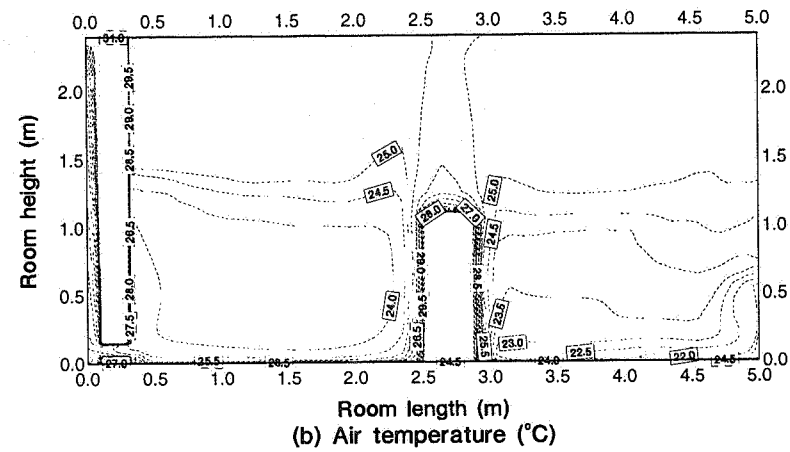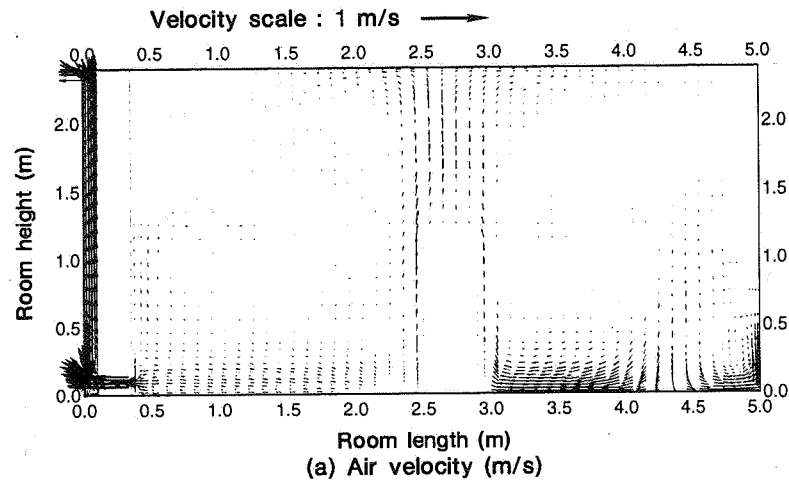
Fig. 3 Predicted air flow pattern and thermal environment on the plane through occupant
in the room with insulated Trombe wall

Velocity scale : 1 m/s �

(a) Air velocity (m/s)

(b) Air temperature (°C)

(c) Mean radiant temperature (°C)

(d) Predicted mean vote
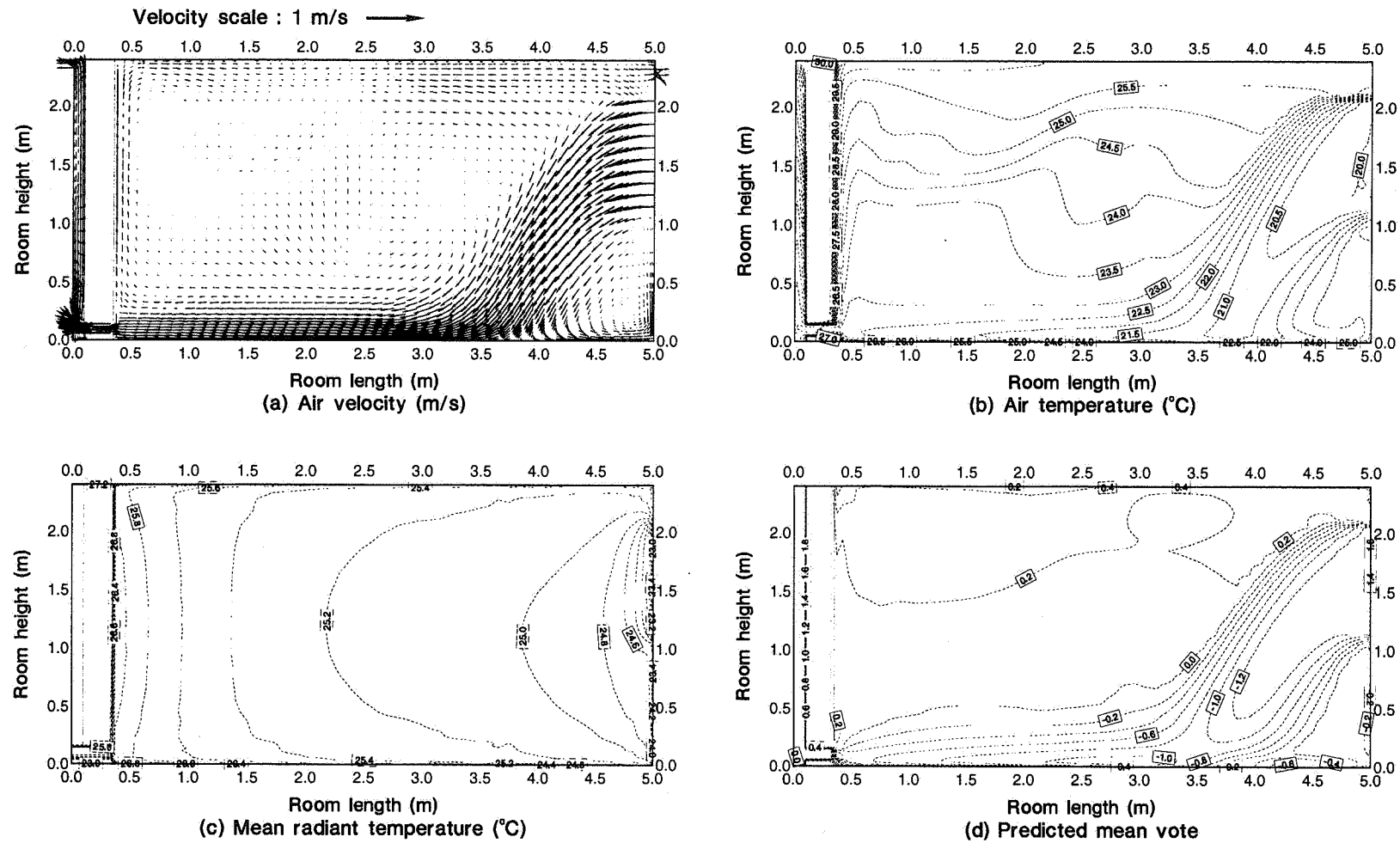
Fig. 4 Predicted air flow pattern and thermal environment on the symmetry plane
in the room with non-insulated Trombe wall

Velocity scale : 1 m/s ⟶

(a) Air velocity (m/s)

(b) Air temperature (°C)

(c) Mean radiant temperature (°C)

(d) Predicted mean vote
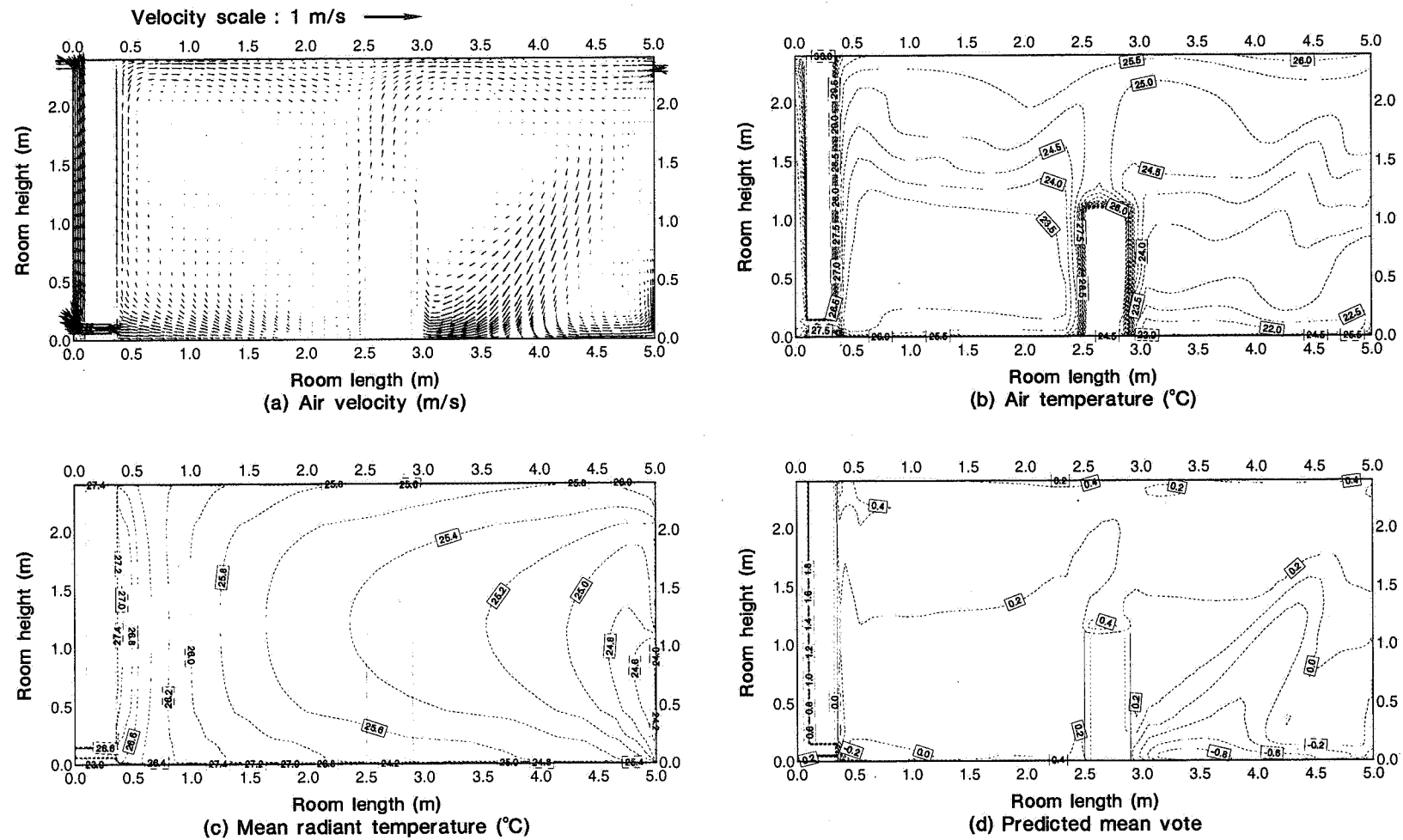
Fig. 5 Predicted air flow pattern and thermal environment on the plane through occupant in the room with non-insulated Trombe wall

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VENTILATION AND COOLING

18TH ANNUAL AIVC CONFERENCE
ATHENS, GREECE, 23-26 SEPTEMBER, 1997

STACK VENTILATION AND COOLING FOR URBAN SITES:
Natural ventilation with roof intake for improved air quality

Stephen A Gage, AADipl, RIBA

The Bartlett, University College London

STACK .VENTILATION AND COOLING FOR URBAN SITES:
Natural ventilation with roof intake for improved air quality

Stephen A Gage, AADipl, RIBA

Synopsis

The paper outlines the value of roof intake air ducts to serve largely passively ventilated and cooled buildings in urban areas. This approach improves air quality, reduces noise pollution and enhances security.

A diagrammatic representation of night cooling using this approach is given followed by a description of experimental work at the Bartlett. This work is directed at establishing methods of starting the ventilation process by overcoming buoyancy, and enhancing the cooling process by providing "mixed mode" cooling.

The author reports on "full size" experiments to establish wind driven ventilation techniques and experiments to establish whether roof planting could provide locally cooled air. Further experimental work on the latter is suggested.

The paper concludes by describing future work aimed at introducing refrigeration as a "mixed mode" ventilation and cooling strategy where passive night ventilation replaces the bulk of the cooling load, and discusses the architectural implications of the research. Further research to establish client attitudes to area loss in buildings as a result of installing large vertical passive stacks is recommended.

List of symbols

t(int)	internal temperature
t(ext)	external temperature
m/sec	metres per second
Pa	pressure in pascals

Introduction

Air conditioned commercial buildings have twice the fuel cost and CO_2 emissions when compared to non-air conditioned buildings. In addition, they are often perceived as uncomfortable and unhealthy. As a result there is a growing trend to avoid active cooling and mechanical ventilation, relying more on a passive control strategy. A new generation of naturally ventilated buildings has recently been constructed. As a result of their form and internal heat gains such buildings would have overheated if they had relied on conventional techniques. These naturally ventilated buildings are currently being monitored under the Sustainable Cities programme and research is currently being undertaken in this area as part of the NATVENT project. [1]

This new generation of naturally ventilated, deep-plan building often encourages natural ventilation by the use of stacks, and cools the building by night venting i.e. drawing cool night air into the warmer building. These techniques are however limited for the following reasons.

1.   Stack ventilation normally utilises air drawn in from the perimeter of the building at a low level. Therefore, in urban locations the air inlet is normally located at street level; this results in gaseous, particulate and noise pollution entering the building. This is currently a major drawback in utilising passive ventilation in urban sites as a result of the impact that such pollutants can have on the health and comfort of occupants. Filtration is

limited with natural ventilation due to the large pressure drops that filters introduce. In many cases it is impossible.

2.   Night cooling involves low level openings which often pose a security risk in urban areas. Also, night cooling has limited impact during the most uncomfortable spells when hot humid conditions prevail and night time temperatures do not drop substantially below day temperatures. However, during cold night conditions there is also a limited degree of available cooling as the structure of the building cannot be cooled below morning comfort temperatures.

Recent research has shown that pollution levels in street canyons fall to background levels at a height of approximately 13 metres. [2] Therefore, by bringing air from roof level or above, levels of pollution can be substantially reduced.

The paper outlines possible strategies for naturally ventilating buildings where the air can be drawn in from an unpolluted roof level site both in the winter and summer, day and night. Without such developments natural ventilation is likely to be limited to "Greenfield Sites".

The basis of any strategy for supplying air from roof level is to reduce intake air duct temperature, and thus buoyancy, to below building air temperature. This principle is illustrated in figure 1. As long as t(ext) is lower than t(int) and duct A is at t(int) and duct B is at t(ext) air will flow through the internal space and cool it down.

The approach has been successfully modelled by Dr Paul Linden at the University of Cambridge, Department of Applied Mathematics and Theoretical Physics. Details of this work will be the subject of a further paper.
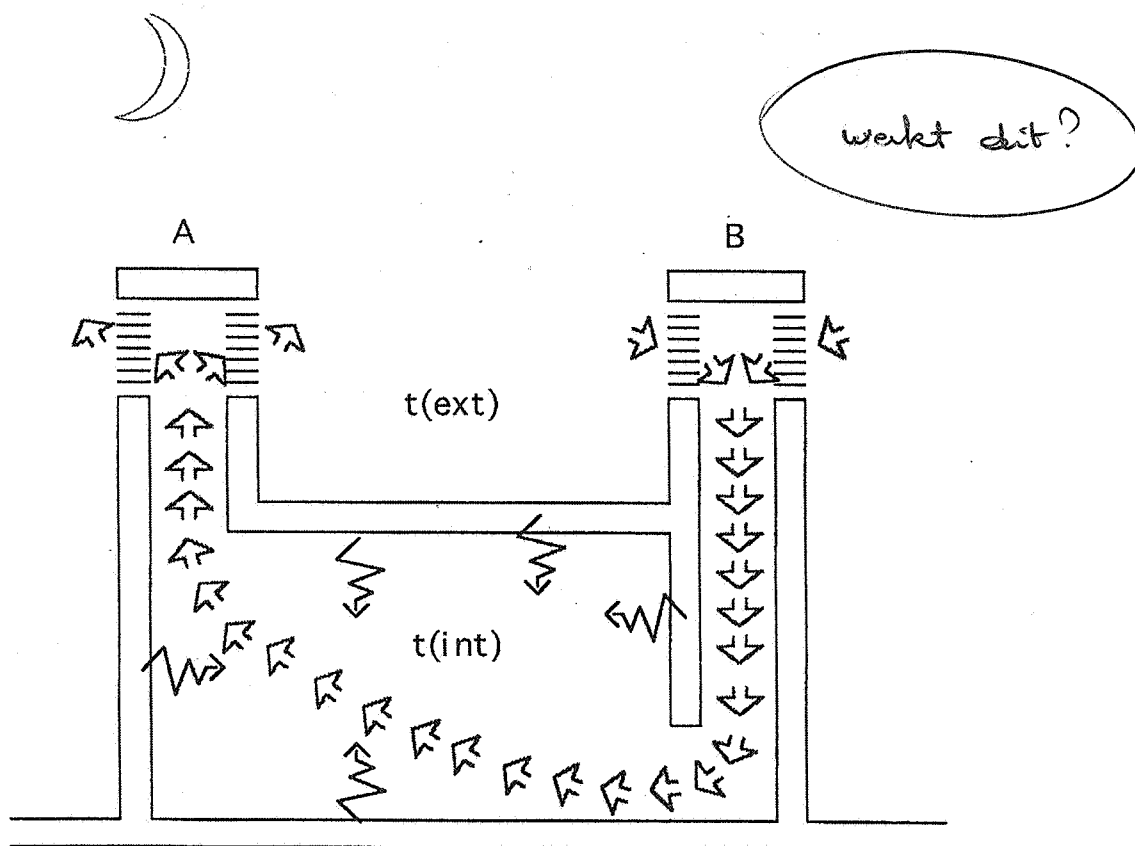
fig.1

PRINCIPLE OF VENTILATION (NIGHT)

Experimental work at the Bartlett

The experimental work at the Bartlett is focused on seeking ways to "kick start" and enhance this approach to passive cooling.

We are fortunate to have guidance from a number of industry partners. These include Monodraught Ltd., Trox UK Ltd and Max Fordham and Partners, Consulting Engineers. We are examining two different approaches:

* to use pressure differences created by the wind to drive air into the intake duct, so that it is cooled down to external air temperature
* to provide a source of cool air on top of or in the intake duct

The former approach will only provide cooling when the external shade air temperature is lower than the internal temperature. The latter offers the possibly of mixed mode cooling where daytime ventilation air can be cooled to comfort levels at times of excessive heat gain. We hope to examine the possibility of combining these approaches at a later stage in our work.

Wind driven ventilation

An obvious model for a wind driven intake stack is the traditional static Middle Eastern "wind catcher". [3] This is simple to build, but has the disadvantage that it will fail to work if the wind is in the wrong direction.

A substantial research project has been conducted jointly by Sir Michael Hopkins and Partners, Ove Arrup and Partners and CSTB in Nantes under the Joule 2 Programme. [4] This examines the feasibility of rotating wind catchers which combine intake and extract ventilation.

The basis of this work is to develop these devices to recover heat from outgoing air through a low resistance heat exchanger and to heat incoming air to a building. The same type of device has potential in passive cooling applications.

The Hopkins research was undertaken using CFD modelling and wind tunnel model tests. Conclusions are:

* hood type intakes are most efficient;
* the intake must be at least 2 metres above an adjoining flat roof to place it above boundary turbulence;
* inlets and outlets should be vertically separated to allow close spacing between ventilators;
* there is doubt that a wind vane will turn a large ventilator in light wind conditions.

We have taken the Hopkins research as a basis for further work. We are addressing the following issues:

* the design of an intake hood which can be shaded;
* strategies to drain rain water entry;
* the use of wind driven servos to turn this type of ventilator.

All our work is being undertaken using "full size" experimental devices. A full size device in this context is a device with airways not less than 200 mm diameter. This allows us to present experimental results giving absolute rather than relative air speeds and pressures.

The first experiment is the corrugated cardboard mock up shown in fig. 2. It has a long shaded intake hood and an intake drum which can be drained. There are seven right angle turns in the airflow through the experiment. The intake is 250% larger than the intake and extract ducts. The device was tested in a wind tunnel at the Building Research Establishment, Garston. Typical experimental results are shown in Table 1.

Table 1: First wind device.

First series of measurements

Tunnel airspeed (ms⁻¹) (u)	0.45	0.90	1.80	2.70	3.60

Tunnel airspeed (ms⁻¹) (u)	0.45	0.90	1.80	2.70	3.60
Reference dynamic pressure (Pa) ($m = 1.26$)	0.13	0.51	2.04	4.59	8.16
Tunnel air pressure (Pa)	1.40	3.50	6.30	*7.60	13.50
Pressure correction (Pa)	-1.27	-2.99	-4.26	-3.00	-5.34

Airspeed in device (ms⁻¹) (w) at position:

A	0.43	0.66	1.60	1.90	2.00
B	0.36	0.64	1.40	1.80	2.70
C	0.22	0.35	1.10	1.40	2.00
D	0.28	0.45	1.20	1.40	1.90

Ratio w/u at position:

A	0.96	0.73	0.89	0.70	0.56
B	0.80	0.71	0.78	0.67	0.75
C	0.49	0.39	0.61	0.52	0.56
D	0.62	0.50	0.67	0.52	0.53

Pressure measurements with plenum ducts closed (Pa) at position:

A (static)	1.50 (+0.23)	3.20 (+0.21)	2.60 (-1.66)	2.40 (-0.60)	0.50 (-4.84)
B (static)	1.60 (+0.33)	4.20 (+1.21)	7.80 (+3.54)	8.30 (+5.30)	13.70 (+8.36)
C (combined)	1.70 (+0.43)	4.10 (+1.11)	7.90 (+3.64)	7.50 (+4.50)	11.90 (+6.56)
D (combined)	1.40 (+0.13)	3.20 (+0.21)	2.10 (-2.16)	1.20 (-1.80)	0.02 (-5.54)

Pressure measurements with plenum ducts open (Pa) at position:

A (combined)	1.50 (+0.23)	3.20 (+0.21)	2.90 (-1.36)	2.70 (-0.30)	1.30 (-4.04)
B (combined)	1.70 (+0.43)	4.20 (+1.21)	6.50 (+2.24)	7.10 (+4.10)	10.70 (+5.36)
C (combined)	0.00 (-1.27)	3.00 (+0.01)	6.90 (+2.64)	8.20 (+5.20)	13.30 (+7.96)
D (combined)	1.50 (+0.23)	3.80 (+0.81)	2.70 (-1.56)	2.20 (-0.80)	1.70 (-3.64)

NB Pressure readings in brackets show pressures as corrected to atmosphere

* This is an odd figure which correlates with a non-linear arispeed in the device

Air speeds in the device are between 60% and 70% of wind tunnel speeds. We are currently constructing second experimental device to be tested in field conditions (see fig. 3). In this device intake and extract positions are vertically separated. The device will use a wind driven servo to turn it head to wind. The servo is derived from the tail fan wheels used to turn 19th century windmills in the UK.
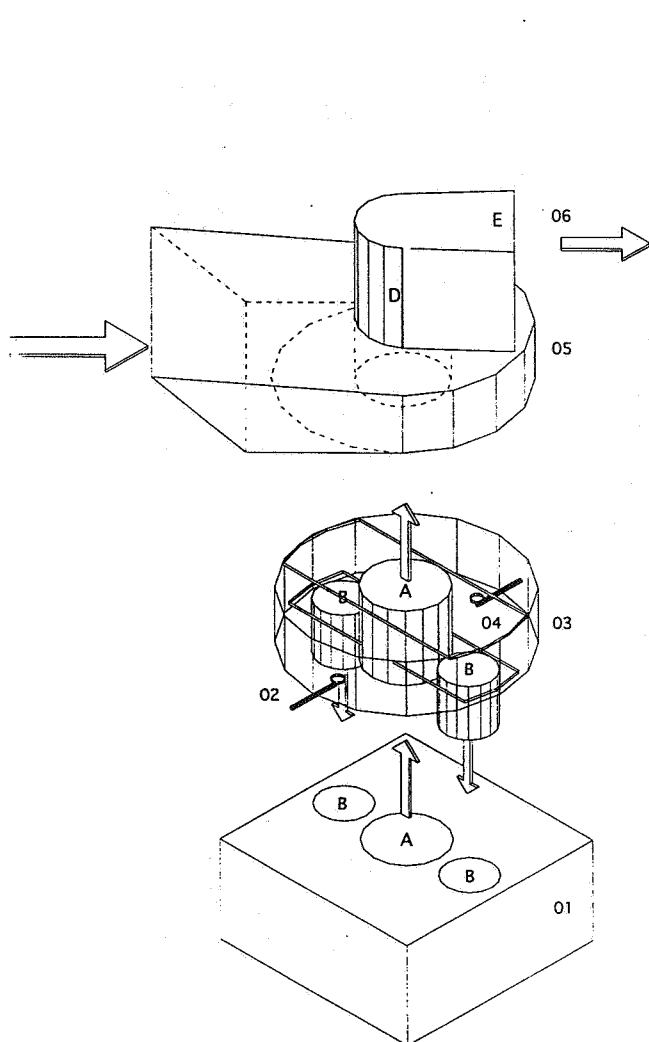
fig.2

FIRST WIND DRIVEN EXPERIMENT

01 plenum box   02 drain off positions   03 base drum
04 rainshield   05 intake hood   06 outlet vent

Refer to table.1
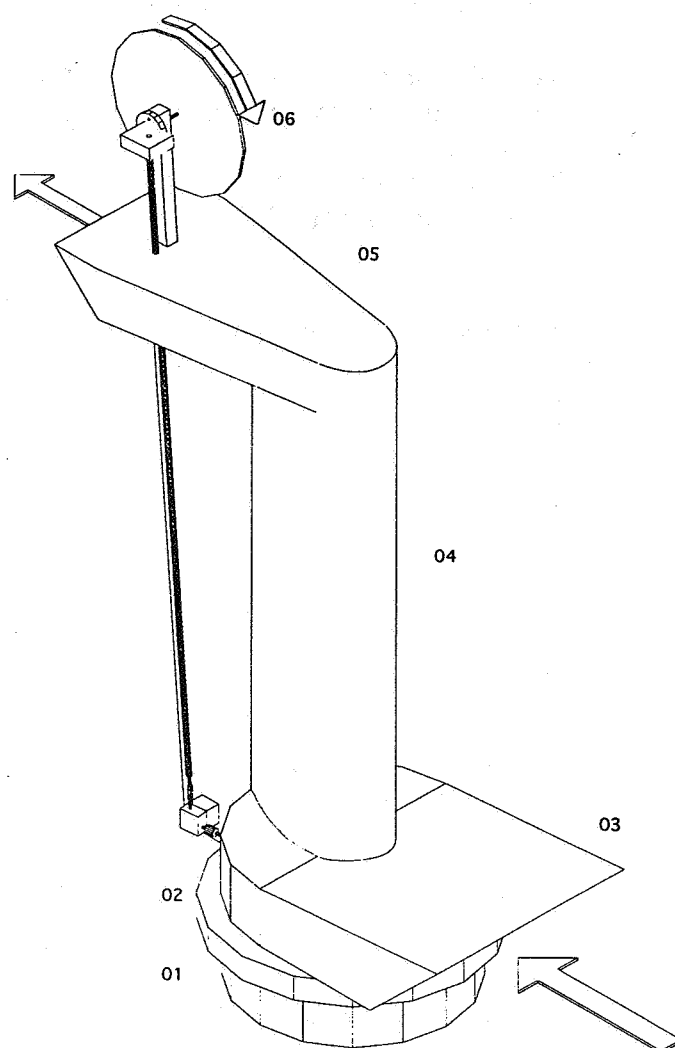
fig.3

SECOND WIND DRIVEN EXPERIMENT

01 base drum   02 revolve   03 intake hood   04 outlet vane
05 outlet vent   06 fanwheel

The initial results of our investigations into wind assisted ventilation and passive cooling systems are encouraging. We believe that intakes can be roof mounted in urban conditions with considerable improvement in indoor air quality and acoustic environment.

Two questions remain unanswered in this approach. What happens when the wind does not blow and what happens when the external shade air temperature is such that a fully passive cooling strategy will not succeed?

A response to the first question is the introduction of large low energy fans into the intake ducts. These are ideally suited to take them. The second question is much more difficult to resolve. We see two possible responses which could lead to mixed mode cooling systems.

Roof garden experiments

Evaporative cooling techniques were used in Middle Eastern wind catchers; water filled unglazed earthenware pots and damp materials were introduced into the intake shafts. This approach has been further explored by Cook *et al* where sprayed water is used both to cool and drive the air. [5] We are concerned that Legionella risks and the problems involved introducing a substantial amount of water into a building will limit the application of this concept.

The undoubted effectiveness of the phase change in water to assist in cooling has also been explored by Giabaklou and Ballinger. [6] We have looked at the possibility of using evapo-transpiration in plants to locally reduce the shade air temperature above intake shafts. The idea that roof gardens could be used to temper the air entering the buildings below them is very attractive because it combines indoor and outdoor amenity for building users.

It is difficult to find the empirical data that would enable a designer to establish the efficacy of this approach. Various researchers have investigated the thermal effects of planting in the built environment. Papers by Parker, [7] Kimura, [8] Euorfopoulou and Arauantinos, [9] Barrosso-Krause [10] and Onomina, Matsumoto and Hokoi [11] all indicate that evapo-transpiration will provide effective cooling. Parker's comments about the role of evapo-transpiration in cooling air are unquantified and all the other authors, with the exception of Kimura, concentrate on the role of vegetation to cool down the fabric of the building below usual (non vegetative covered) temperature levels. Kimura refers to experimental work undertaken at Yamashi where shade air temperatures are shown to be 1 - 2 °C lower in a densely planted garden that than the equivalent shade air temperatures outside the garden.

The subject of evapo-transpiration is also extensively studied by geographers and biologists. Oke, quoting experimental work by Long *et al*, [12] shows how; at 02.00 hours the temperature inside a field crop under an open sky is in excess of 2°C lower than the external air temperature; that at 06.00 hours the temperature is approximately 1°C lower; that at 12.00 hours the temperature is over 1°C higher and that at 18.00 hours the temperature was approximately 1°C lower. He suggests that this process points to an active surface just below the surface of the crop. It should be noted that these experiments took place in the UK, in summer and that the 06.00 measurement can be assumed to be 1.5 hours after dawn.

Forests show different characteristics. Oke refers to work by Jarvis to show that at midday the increase in temperature is lifted to the centre of the canopy layer, and that the forest floor temperature is that of the shade air temperature above the canopy.

We constructed an experimental "roof garden" 1.2 metres square and 600 mm deep at the Bartlett, planted with garden shade tolerant plants, placed in pots in a water tray. The high surround was provided to reduce wind effects. In order to mimic the effects of a forest canopy we shaded the garden with an artificial canopy consisting of two layer of white perforated profiled metal decking with an air cavity between them. This is shown in fig. 4.

Typical experimental results are shown in Table 2, which compare two days, 4th and 5th August 1996. Both were sunny but the 5th was a day of exceptionally high humidity. It can be seen that this experiment was only partially successful in that the "artificial canopy" did not stop radiant heat gain. Nevertheless in conditions of low external RH this approach appears to provide an extended morning period where the garden temperature is lower than the external temperature.

Relative humidity is very high when cooling effects are most apparent. This is probably a result both of stomatal evapo-transpiration and of evaporation from the water tray and the growing medium surface. The experiment also shows that, with this configuration heat once gained by the roof garden is trapped in it and that the early evening effects noted by Long do not occur.

Table 2: Experimental "garden".

Legend:
- ◯ garden temp
- ✳ garden rh
- △ met temp
- ◇ metrh
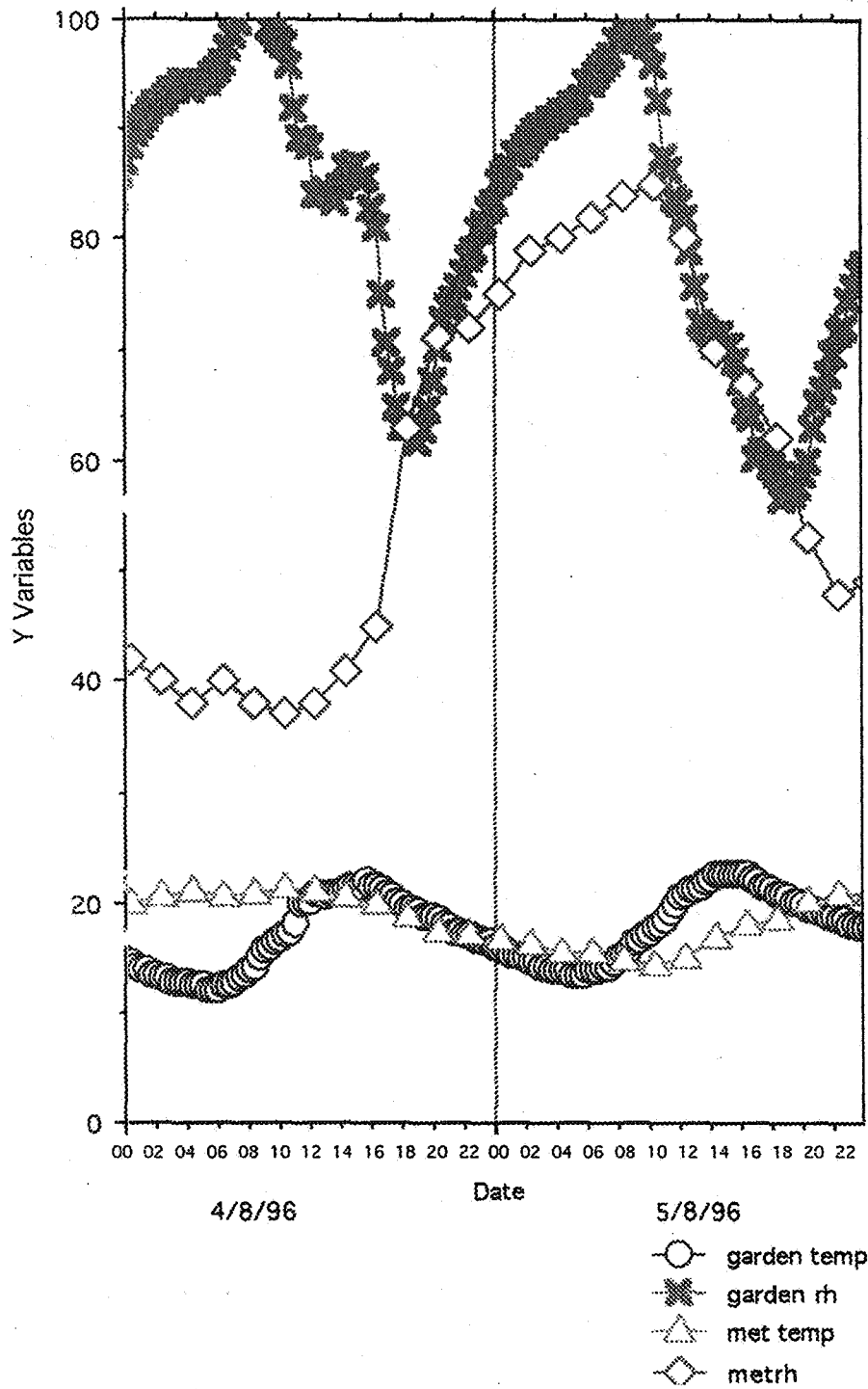
    In hot, dry climates it is probably advantageous to consider enclosed shaded roof
gardens under-planted with shade tolerant plants as sources of air for night cooling.
    Experimental work should be undertaken to establish temperature profiles in a walled
shaded and under-planted roof garden over Spring, Summer and Autumn in a hot, dry climate
to establish actual temperature differences in field conditions.
    We show a possible experiment in fig. 5. This shows a roof garden approximately 6
metres square surrounded by externally insulated walls 3 metres high. Alternative modes of
shading are indicated. A trellis and vine cover may be more advantageous than opening

insulated louvres. Our initial results suggest that the air temperature in the roof garden will be lower than the shade air temperature at night, in the morning and the late afternoon. We show a low energy intake fan fitted to the intake duct. This will be necessary if the building below the garden is successfully designed, because ventilation air in the early afternoon will be at a higher temperature than the internal air temperature. It must be driven down the duct. This will heat the inner duct walls which must be cooled down in the late afternoon to allow for subsequent night ventilation. The fan could be powered by photo-voltaic cells.
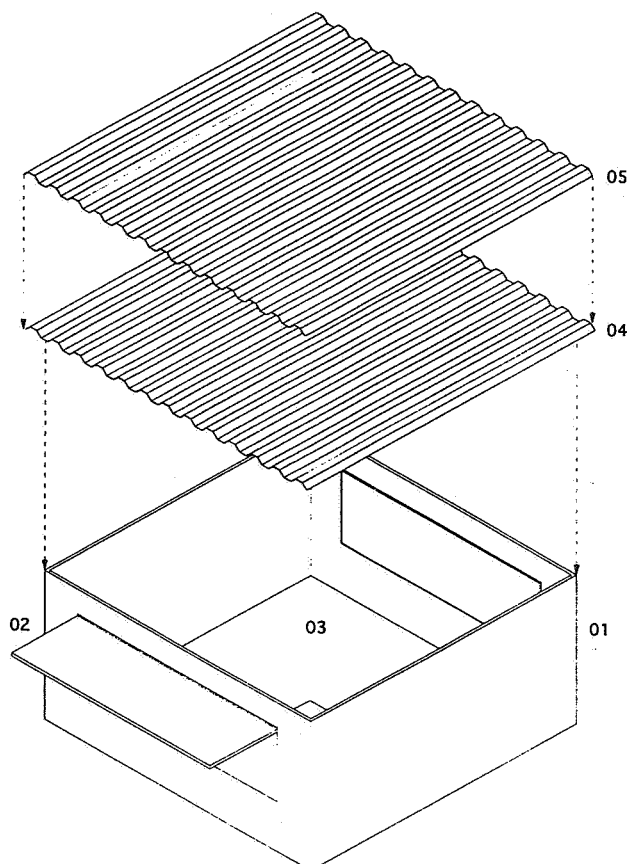
fig.4

SHADED PLANTER EXPERIMENT

01 main chamber 02 access hatch 03 planting
04 first canopy 05 second canopy

CORRUGATED ROOFS WHITE ALUMINIUM, PERFORATED 10% IN
CROWN & VALLEYS SEPARATED BY 15mm SPACERS.
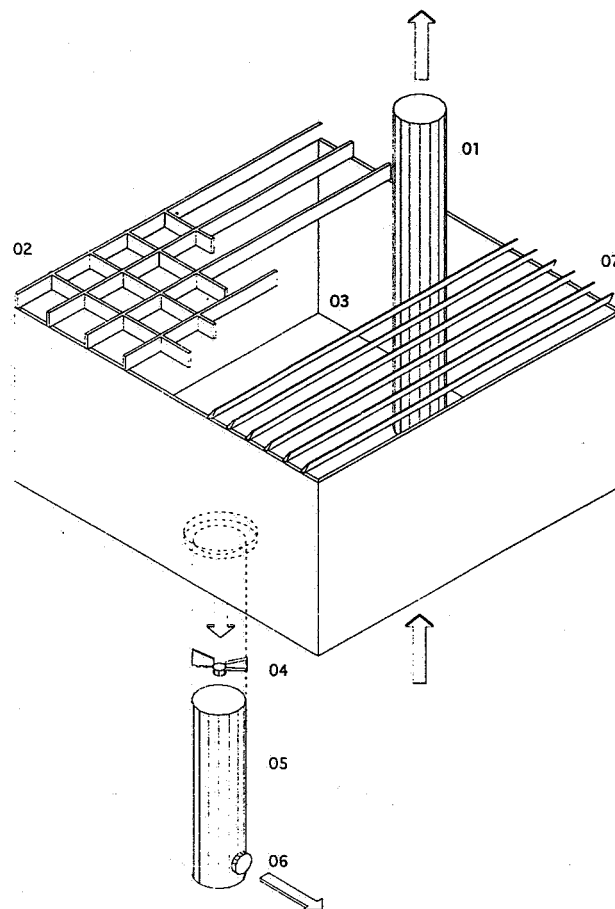
fig.5

POSSIBLE SHADE ROOFGARDEN

01 extract duct 02 trellis & vine 03 underplanting
04 low energy fan 05 insulated intake duct 06 damper
07 rotating insulated louvres

Future work at the Bartlett

Over the coming year we propose to investigate - again using 'full size' experimental devices - the approach shown in fig. 6. This type of "mixed mode" cooling is potentially attractive. The technique used to start night cooling is the same as that which could be used to "top up" cooling in extreme conditions. It should be noted that the vertical stacks designed for cooling must be large to allow for high volumes of air to move relatively slowly. If these are designed for air speeds of 2 m/sec then air speeds during the day will be approximately 0.2-0.4 m/sec in the stacks. The very slow speeds will aid heat transfer from chiller plates. These plates will run colder than the dew point of the incoming air and condensate drainage must be considered.
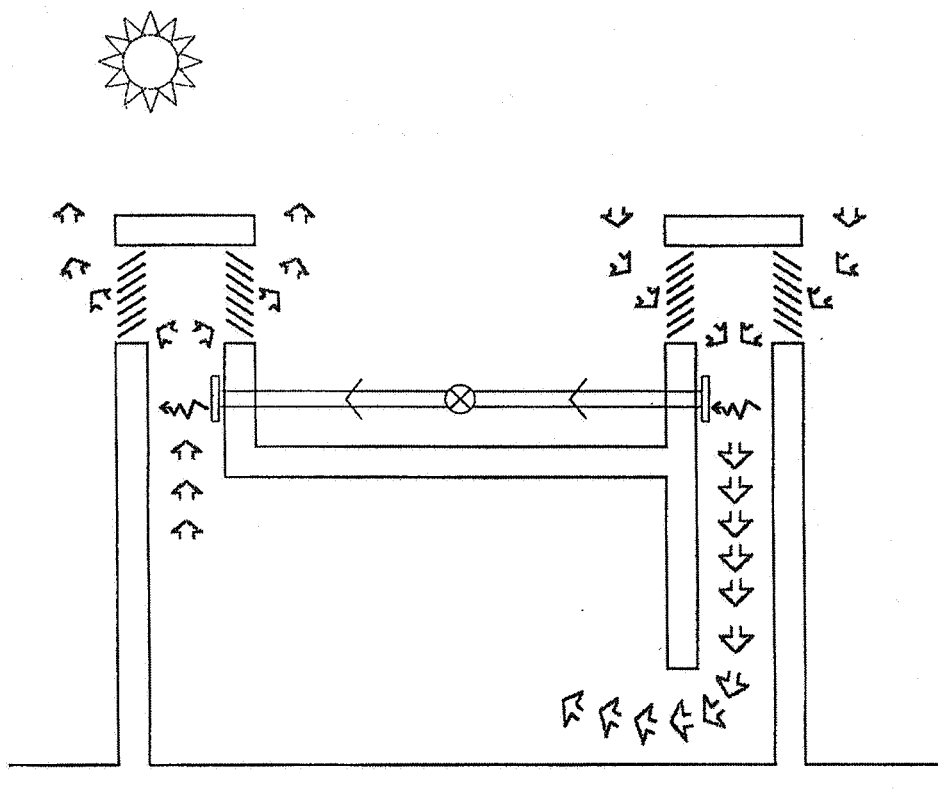
fig.6

PRINCIPLE OF ASSISTED COOLING (DAY)

Architectural consequences of vertical stack ventilation and cooling.

The architectural consequences of this approach to ventilation and cooling are such that it is unlikely to be applicable to buildings exceeding four stories in height. If we assume 10 air changes an hour, a duct speed of 2 m/sec and a floor to floor height of 3.6 metres the combined supply and extract stack area for any floor of a building is approximately 1%. A four storey building will have 4% of its top floor area occupied by stacks. This subject deserves further research. The judgement that no developer will contemplate more than 4% loss of building area is based on the author's experience as a practising architect. It may be that the current commercial climate is more enlightened and a survey of potential clients should be undertaken.

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VENTILATION AND COOLING

18TH ANNUAL AIVC CONFERENCE
ATHENS, GREECE, 23-26 SEPTEMBER, 1997

(Title)   **SORPTION-SUPPORTED AC-SYSTEM IN A PRINTING OFFICE**

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SORPTION-SUPPORTED AIR-CONDITIONING IN A PRINTING OFFICE

Synopsis

One of the first sorption-supported air-conditioning systems ("Desiccative Cooling Systems") in an industrial building in Germany was installed in a printing office in Waiblingen, a town in southern Germany. The circumstances for such a system showed to be optimal, as the printing office is equipped with its own co-generation system delivering a considerable amount of waste heat. The experiences made with the system in the hot and humid summer of 1995 were very positive. Even when the outdoor-temperature reached about 32 °C, the temperature and humidity set points of 20 °C and 80 % relative humidity could be realized.

1. Sorption-supported Air-conditioning – Innovative Ventilation Technology in the Industrial Environment

The main feature of sorption-supported air-conditioning is the separation of the cooling and dehumidification processes. When so far the water was removed from the air by cooling the air below the dew point, now the water vapour is linked to solid or liquid hygroscopical substances. The so-called "sorption regenerator" is widely known. It is different from the traditional regenerator only in so far as it uses another kind of storage medium for heat recovery purposes. The hygroscopical features are caused by impregnation with saline solutions or by deposition of solid adsorbents. This way of air dehumidification combined with other components leads to a completely new generation of air-conditioning systems. The integration of sorption-supported dehumidification with both heat recovery and air moisturisation by evaporative cooling is well advanced in Germany.

The most important progress in developing sorption-supported air-conditioning systems was to replace the driving forces of the refrigeration process. While classical chillers need electric current, sorption-supported air-conditioning uses heat as driving force.

2. An example from real life – the "Druckhaus Waiblingen", DHW (Waiblingen Printing Office)

In the meantime, the sorption-supported air-conditioning technology has finished the laboratory-stadium. The following report describes a printing-office in Waiblingen, Germany, which was equipped with sorption-supported air-conditioning. Two central preconditions had to be realized: on the one hand the surplus waste heat of the company co-generation system occuring during the summer season ought to be reintegrated into the energy-circuit. On the other hand the capacity of the co-generation system was not to be weakened by the chillers'

high connected load. Besides, the required indoor air quality level in printing-offices is very high. Especially 4-colour printing at high speeds needs constant levels of temperature and humidity. All these demands can be fulfilled by cold production based on dehumidification with subsequent evaporative cooling (Desiccative evaporative cooling, sorption-supported air-conditioning).

Fig. 1: The Waiblingen Printing Office ("Druckhaus Waiblingen", DHW)

3. No CFC-containing refrigerants

The following features of sorption-supported air-conditioning, also known as Desiccative Evaporative Cooling, DEC, characterize the Waiblingen Printery compared to conventional refrigeration:

- use of water instead of CFC-containing refrigerants
- possible use of free waste heat
- the connected load is only about 60 per cent
- the saving potential of electric energy is up to 40 per cent
- comparable investment

Both devices installed in the printing-office are laid out for a supply flow of 27,000 m³/h and a cooling capacity of 110 kW each. The temperature and humidity set points of 20 °C and 80 % relative humidity can be realized even at outdoor-temperatures above 30 °C in order to guarantee constantly good printing results.
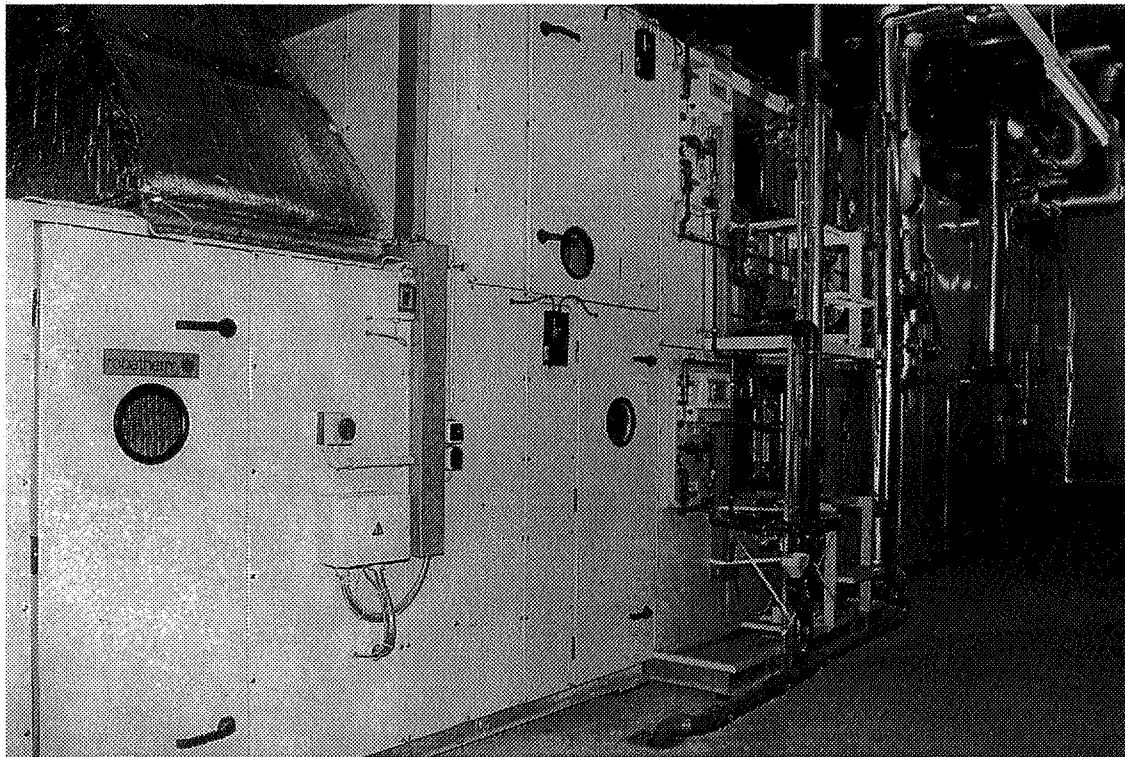
Fig. 2: View of the central unit of the sorption-supported air-conditioning system

The control parameters of sorption-supported air-conditioning are:

- the humidity degree of the infinitely variable air humidifier,
- the heat efficiency of the heat recovery unit,
- the thermal output of the regenerative air heater,
- the volume flow of the regenerative air ventilator.

Due to individually developped control strategies, it is possible to adjust the succession of the various control parameters in accordance with the system or the unit respectively. In this way, electric current and heat can be applied suitably to availability and need.

The experiences made during the hot and humid summer of 1995, when the printery was first air-conditioned by the sorption-supported system, confirmed all expectations.

4. Cooling by Evaporation

Sorption-supported air-conditioning is suitable for cooling and dehumidification of air both in human and process related air-conditioning, not, however, for the refrigeration of liquids or for the operation of cold-storage rooms. The operational principle is less complicated than could be assumed (see figure 1): The aspirated filtered outside air (1) is dried by means of a sorbate regenerator (2). During this process heat of condensation is released leading to a temperature increase. Afterwards the air is precooled in a heat recovery system (3) by heat transfer from the incoming part to the outgoing part. During winter time a reheater (4) has to assure good functioning. Further cooling to the temperature level of the supply air is realized by a controllable evaporative cooler (5).
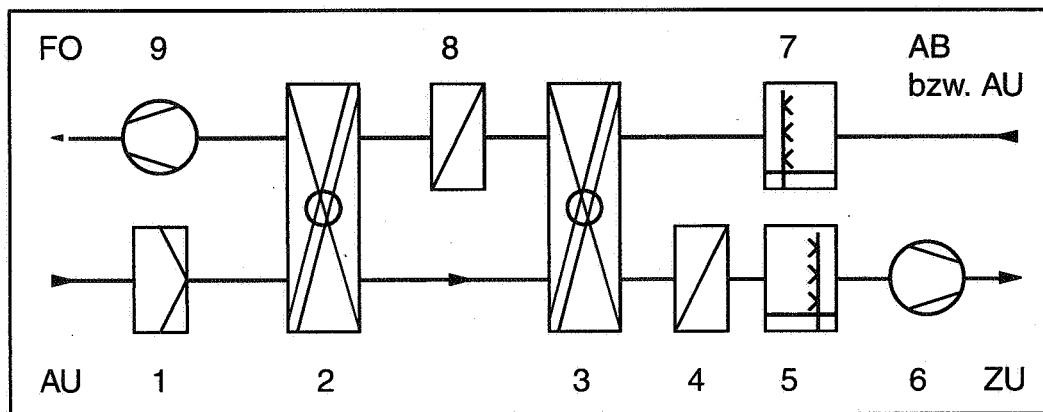
Fig. 3: Schematic description of a sorption-supported AC-system

Due to the effectiveness of evaporative cooling the prevailaing outdoor air condition results in a temperature decrease. The supply air is warmed up by the power input of the supply air fan; therefore this power input must be taken into account when the supply air temperature is to be determined.

The regeneration flow which is directed inversely to the supply air is usually realized by the outgoing air. First of all, the outgoing air is cooled down and humidified by means of a second evaporative cooler (7). Afterwards it flows through the heat recovery system (3) realizing hereby the precooling of the supply air as described above. After leaving the heat recovery system the outgoing air is heated and this process enables the desorption, i. e. the regeneration of the sorption regenerator (2).

5. Reduced Operating Expenses Allow Economical Operation

With regard to cost of production and investment the technologies of sorption-supported air-conditioning and of classical cold production have to be compared in relation to the project and in view of the specific basic conditions. In principle, these costs can be considered as being on the same level as conventional units for the fact that, in every respect, soprtion-supported air-conditioning is based on customary and reliable ventilation components. Whenever the use of sorption-supported air-conditioning brings in economic advantages, in most cases this is due to savings on the part of the operating expenses. Compared to air-conditioning units with electrically driven refrigerating machines, the consumption of the individual devices and unit components is differing, possibly leading to reduced costs of water, electric power and work. In addition, the heat recovery units guarantee that the year-round heat consumption corresponds roughly to the consumption of an air-conditioning system with refrigerating machine and standard heat recovery. The use of utility-supplied heating, solar heating and anyway available waste heat shows to be particularly advantageous.

6. Conclusion

The experiences made at the Waiblingen printing office (DHW) prove that ecologically desirable ways of cold production can definitely turn out economical. It ought to be mentioned yet that the exact comparison of costs as well as the determination of a possible saving potential must be carried out in relation to the project.

Further information

Further information on behalf of this subject is available from the *Fachinstitut Gebäude-Klima e. V.*, Danziger Str. 20, D-74321 Bietigheim-Bissingen. Tel. +49-7142-54498, Fax -61298, as well as from *Robatherm Wärme- und Klimatechnik GmbH*, Industriestr. 21-27, D-89331 Burgau, Tel. +49-8222-999-0, Fax -999-222, and from the *Institut für Luft- und Kältetechnik gGmbH*, Bertolt-Brecht-Allee 20, D-01309 Dresden, Tel. +49-351-4081-650, Fax -4081-655.

Title:　　　Air Conditioning of Internal Environment By Means of
　　　　　　BioClimatic Systems

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AIR CONDITIONING OF INTERNAL ENVIRONMENT BY MEANS OF BIOCLIMATIC SYSTEMS

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ABSTRACT

ICITE is carrying out a research program with the aim of studying and optimizing the energy performances of two prototypes of building components which are based on the utilization of hybrid solar system.

The first one consists of a sun-operated pre-fabricated modular system composed of a multi-layer element integrated with a module for energy management which causes a bidirectional air flow between the internal and external environment and through the air space of the element.

The second one consists of a ventilated double glazed window with varying transparency that under particular conditions causes different air flows in order to heat or cool the internal environment according to the season.

The two systems contain solar photovoltaic cells which are intended to provide energy supply and thus allow the components complete autonomy as regards energy. Furthermore their working systems are fully automatic so that the proposed technology does not interfere with the laying operations and helps manage the positive thermal contribution the external environment can provide.

Besides we thought it advisable to resort to a practical tool suitable for testing the performances of the two systems under different working and climatic conditions. This is the reason why a mathematical model was studied, optimized and validated on the basis of data obtained from experimentation carried out in real working conditions on the two systems. In the paper the outcome is presented and discussed.

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VENTILATION AND COOLING

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(Title)

Economical comparison of comfort ventilation and air-conditioning plants.

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Synopsis

Continuously rising energy costs, the demand for reduction of CO_2-emission and the prohibition of CFC-containing refrigerants create a base for new concepts of air-conditioning (A/C) systems. A primary action must be the prevention of heat consumption and cooling load by improvement of the building architecture. Additional the efficiency of the A/C process must be improved, in order to reduce the energy input.

In most cases the target is to replace the dehumidification process, which normally is realized by refrigerating cycles, by alternative systems. The air dehumidification by cooling the air below the dew point involves a high energy consumption.

The paper treats three different A/C systems for non industrial buildings. The task has been the economic evaluation of the air-conditioning plants and the demonstration of the different energy demand. Referring to the expected costs of the A/C systems the price of a new dehumidification unit, which is developed at the university of Essen (see figure 3), has been determinated. The results show that a higher price of the new system can be equalized by lower energy costs.

1. Introduction

The calculation bases on a model which describes the A/C of office buildings. In order to simulate the operation of the A/C-systems, four buildings with different air flow rates are designed. The design of the systems is carried out so that comfort room air condition will be reached for every outdoor air condition. The capital costs of the installed components are determinated with price lists of manufacturers. The operating costs for heat, electricity and water are calculated for a continuous plant operation over the year. The calculations are carried out referring to the German VDI-Standard 2067 by using a statistical procedure.

2. List of symbols

OA	Outdoor Air
SA	Supply Air
RA	Return Air
EA	Exhaust Air
P	Preheater
R	Reheater
C	Cooler

3. Presentation of the A/C-systems

(System 1)

The energy transport of the designed systems is guaranteed only by air. System 1 which presents the conventional A/C system is shown in figure 1. Cooling and dehumidification of the air is realized by a compression refrigerating cycle.
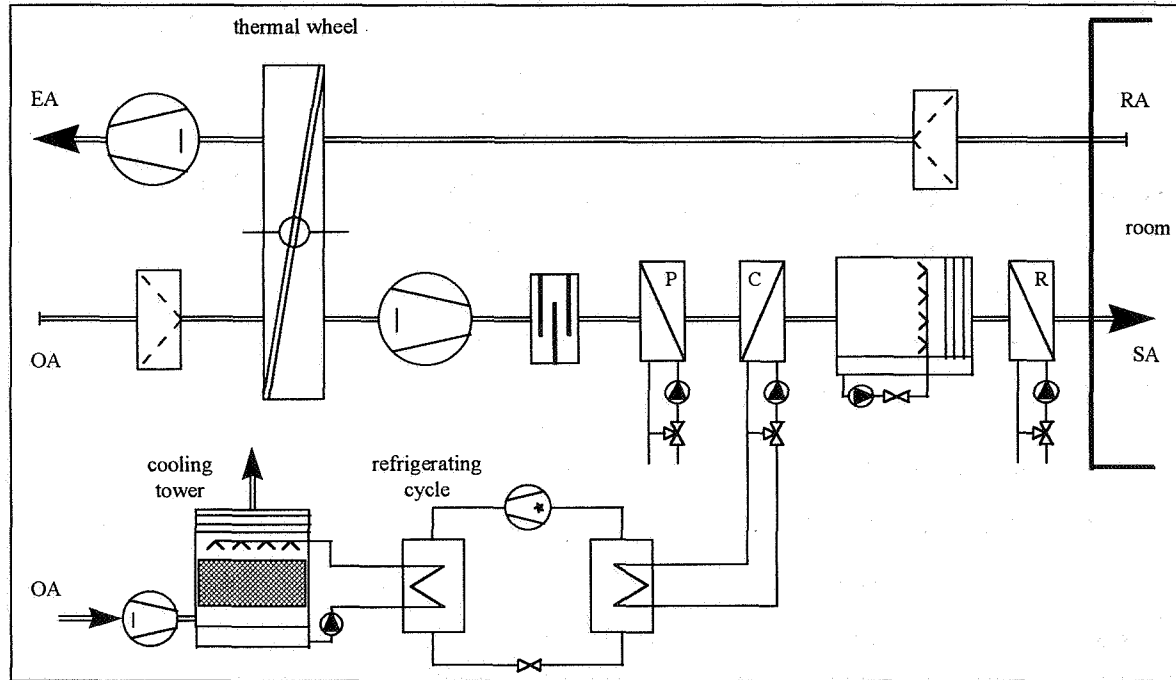
Fig. 1: System 1, with refrigerating cycle and dehumidification by cooling the air below the dew point.

The systems 2 and 3 are A/C plants, by which a comfortable room air is reached by sorptive dehumidification and cooling by evaporation of water (DEC = Desiccative and Evaporative Cooling), so that the thermodynamic functions cooling and dehumidification are seperated. The difference in the considered sorptiv systems is mainly the dehumidification process.

System 2 works with solid sorbent which becomes active on the surface layer of a dehumidification wheel. The regeneration of the sorbent is reached by heating a part of the return air which passes through the dehumidification wheel. The cooling of the supply air is realized by evaporation of water in humidifiers installed in the ducts of the return air (indirect cooling) and the supply air (direct cooling). The calculation of the air condition in system 2 shows that there could be some problems to reach low temperatures of the supply air when, the outdoor air is warm and humid. The lowest temperature is given by the boundary temperature of the evaporation process. Figure 2 shows the general assembly of system 2.
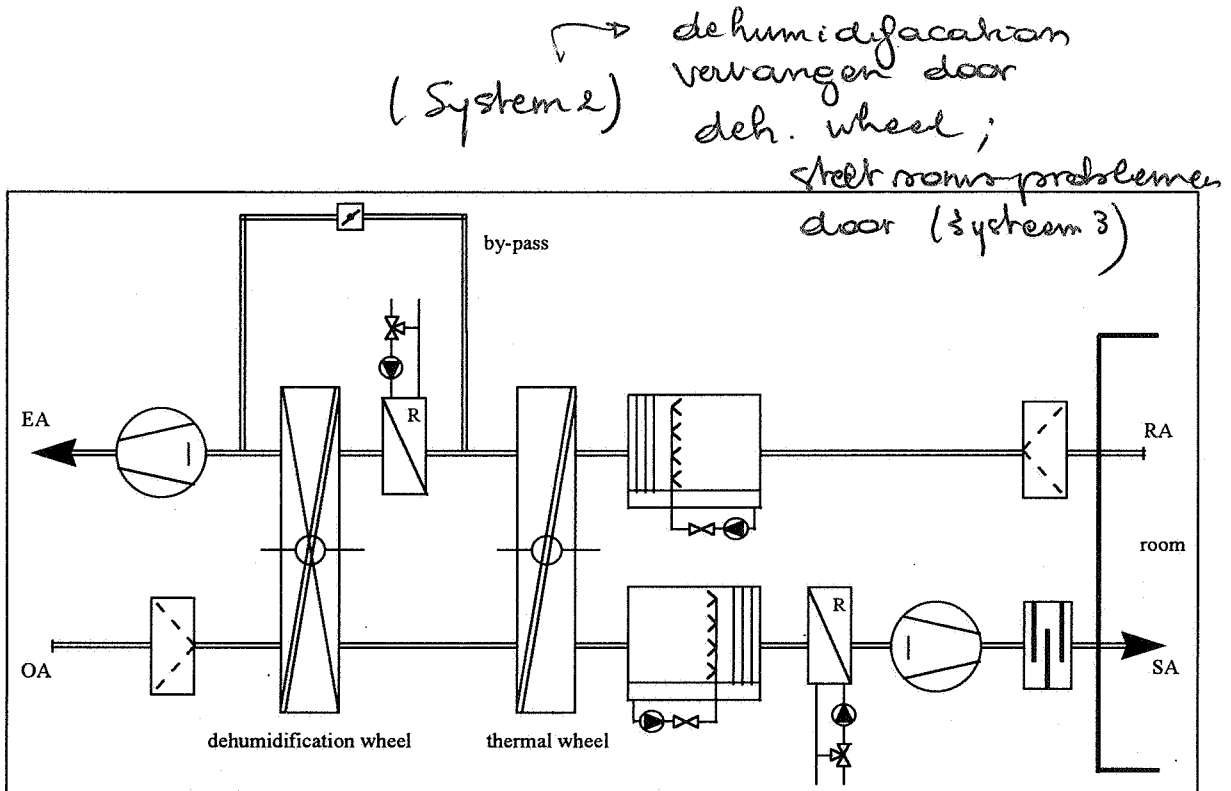
Fig. 2: System 2, Desiccative and evaporative cooling (DEC-system).

System 3 works with a new type of absorber which is integrated in a dehumidification unit. The outdoor air is dehumidified by a liquid desiccant and cooled indirectly by a cooling tower. The advantage of a cooled absorber is a lower process temperature, so that every supply air condition can be reached. Figure 3 shows the assembly of system 3.
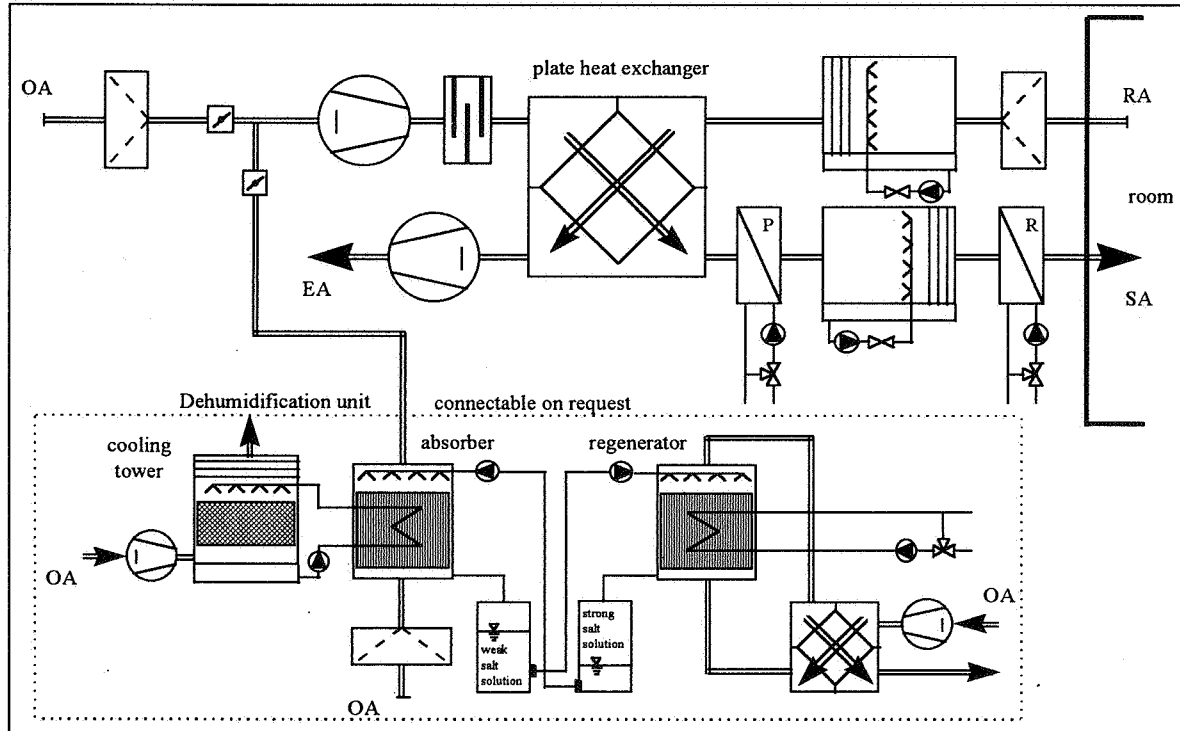
Fig. 3: System 3, evaporative cooling and dehumidification by absorption.

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The regeneration of the weak salt solution is carried out in a separate regenerator where the solution is heated and the water evaporates. In opposition to system 2 the heating of the regeneration process has no influence on the supply air. The use of a plate heat exchanger is advantageous. The supply air fan is installed in front of the heat exchanger, so that the heat gain of the fan can be released in the heat exchanger. As there is a complete separation between the return air duct and the supply air duct, it is possible to oversaturate the return air in order to increase the heat transfer.

4. Results of the economical comparison

The specific investment prices of the A/C-system in $[DM/(m^3/h)]$ correlate to the air flow rate. Figure 4 shows the expected prices of the regarded systems.
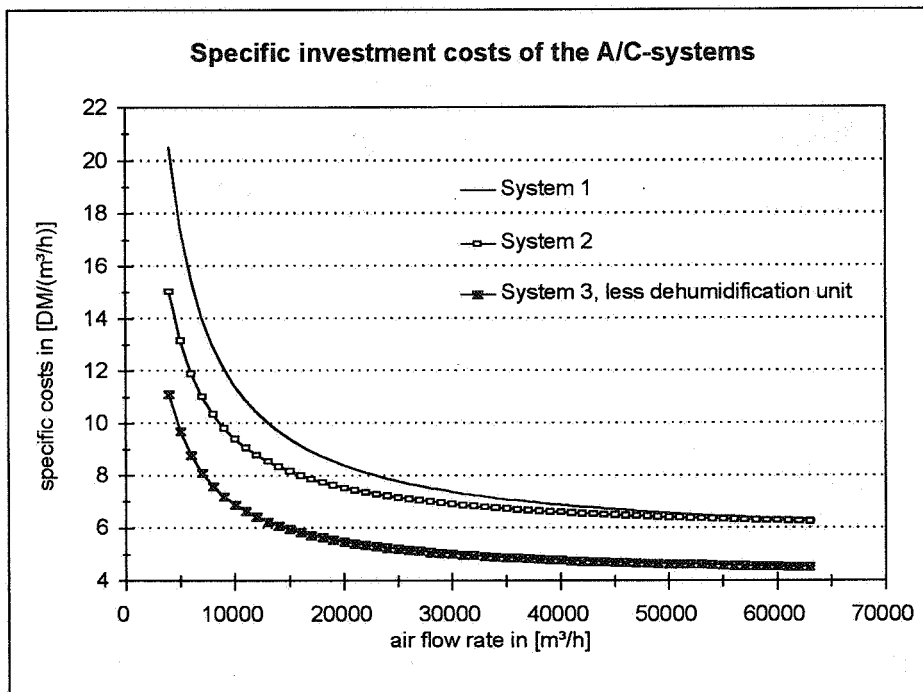
Fig. 4: Specific investment costs of the A/C-systems as a function of the air flow rate.

The specific costs of system 3 are charged without the costs for the dehumidification unit. That's why it is between 3,- and 2,- $DM/(m^3/h)$ cheaper than the other systems. The price of the dehumidification unit will be evaluated referring to the operating costs (see figure 8).

The following figures only pay attention to the energy costs, however it is possible to recalculate the energy consumption by using the energy prices. The prices for energy have been stated by data of energy supply companies.

heat: $k_H =$ 0,05 DM/kWh (natural gas)

electricity: $k_E =$ 0,19 DM/kWh

$k_{EP} =$ 160,00 DM/kW$_{el}$ (price for required power)

water: $k_W =$ 3,00 DM/m³

The analysis of the calculation shows that the distribution of the operating costs of one system doesn't change very much with the airflow rate. Figure 5 shows exemplary the energy costs for a building with a maximum air flow rate of 30 000 m³/h. The results are valid for a continuous plant operation over the year. For the comparison the costs are related to the costs of system 1.
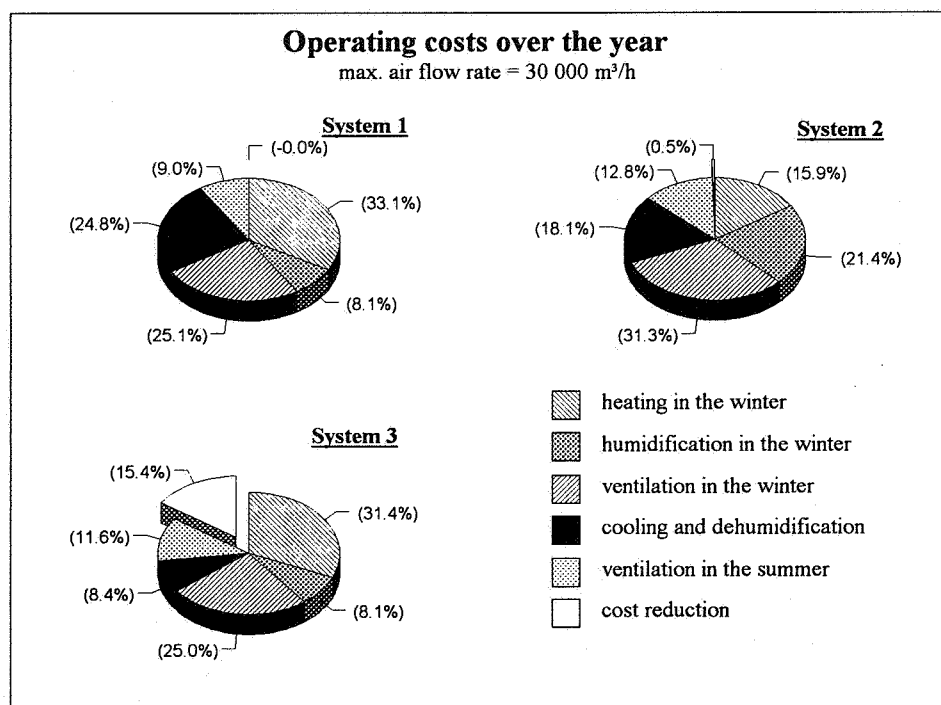
Fig. 5: Operating costs for a continuous plant operation over the year referring to system 1.

The absolute energy costs during the winter time are nearly equal for all systems. Approximately 65% of the energy costs of system 1 and 2 have to be payed for heating, humidification and ventilation in the winter. With system 3 the part of the energy costs in the winter is about 80% of the total energy costs. The distribution of the energy costs of system 2 in the winter is different to those of system 1 and 3. The reason for this difference is the use of the dehumidification wheel for recovery of moisture and the installation of a thermal wheel with a high efficiency for the indirect evaporative cooling in the summer. The calculation demonstrates that the annual energy demand of system 1 and 2 is nearly equal.

It becomes obvious that the 15% cost reduction of system 3 is a result of the replacement of the cooling and dehumidification process. For a better explanation the operating costs expected in the summer time are exactly described in figure 6.

Operating costs in the summer
max. air flow rate = 30 000 m³/h

System 1

System 2

System 3

ventilation in the summer

heating

refrigerating cycle

cooling tower

water

pumps and heat recovery

cost reduction
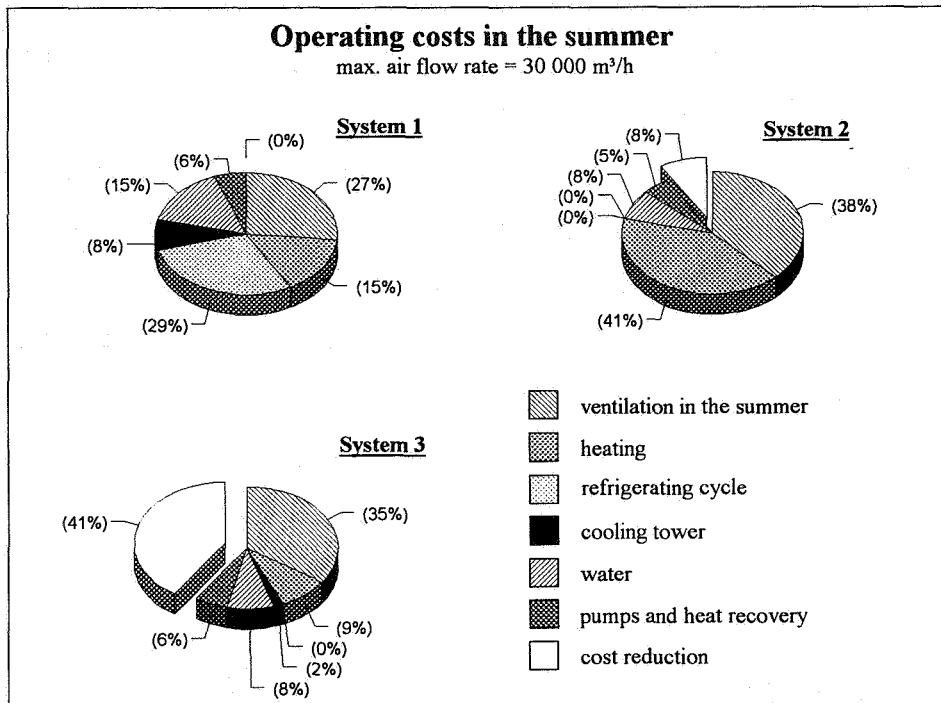
Fig. 6: Operating costs for a plant operation in the summer time referring to system 1.

The results of figure 6 show clearly that there are great differences between the regarded systems concerning the energy demand. The A/C-system with the refrigerating cycle causes the highest operating costs in the summer. By using system 2 it is possible to reach a cost reduction of about 8% and with system 3 it is possible to reduce the energy costs in the summer around 40%.

As there are more components in the A/C-plant which causes higher pressure differences, the ventilation costs of the sorptive systems are 30% higher than those of system 1. The input of heat in system 1 is necessary to reheat the air coming from the cooler up to a comfortable supply air condition. It is obvious that the highest heat demand is given in system 2. In the sorptive systems the heat is necessary for the regeneration of the sorbent. System 2 needs a three times higher energy input for the regeneration then system 3. The difference is given by the way of heating the sorbent. In system 2 the regeneration air flow must be heated which causes great heating loss over the exhaust air. Whereas in system 3 the heating of the sorbent is done directly by water and the process-heat is recovered in a heat exchanger. The need of

fresh water for cooling in system 1 is higher than that of the systems with evaporative cooling. The reason is that the amount of heat originated from the refrigerating cycle and released by the cooling tower is higher then the cooling load, which is released directly in system 2 and 3.

Up to now the considered costs exclusively refer to the energy costs. Supplementary costs must be expected for the disposition of electricity. This price has normally to be payed for a maximum power requested in a period. Paying attention to the costs for required power the distribution of the annual operating costs (shown in figure 5) changes considerably. Since the refrigeration cycle works with electricity, the operating costs of system 1 rise more then those of the sorptive systems. The influence of the costs for required power on the annual operating costs is shown in figure 7.
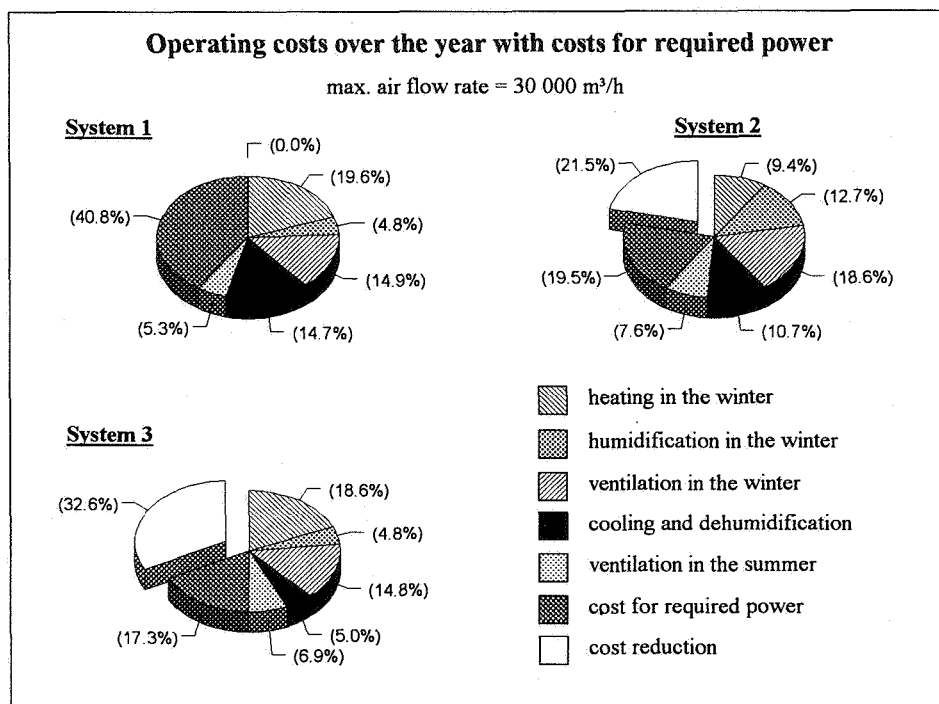
Fig. 7: Operating costs for a continuous plant operation over the year with costs for required power.

Figure 7 shows that the influence of the costs for required power can not be neglected. The use of the electrically driven refrigeration cycle causes annual cost reductions about 20% for system 2 and 30% for system 3.

The calculated cost reduction of system 3 has been taken to determine a price for the new absorptive dehumidification unit. The calculated price for the dehumidification unit so presents

the maximum costs in order to compete economically with the other systems. Figure 8 shows two curves in which the specific costs are presented as a function of the air flow rate.
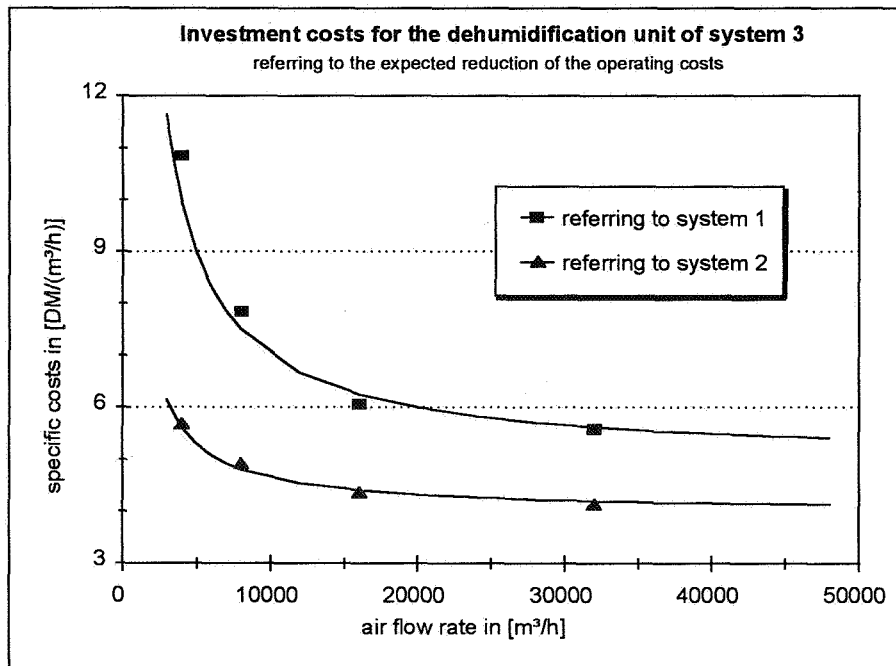
Fig. 8: Investment costs for the dehumidification unit of system 3, referring to the expected reduction of the operating costs of figure 7.

The price for the dehumidification unit resulting from the comparison of the investment costs (see figure 4) may come to an amount between 3,- and 2,- $DM/(m^3/h)$, so that the installation of system 3 will still be attractive. Paying attention to the reduction of the operating costs, the price of the dehumidification unit of system 3 could be between 9,- and 4,- $DM/(m^3/h)$, depending on the size of the A/C-plant and the reference A/C-system. The result shows that a higher price of the new system can be equalized by lower energy costs.

5. Acknowledgement

VENTILATION AND COOLING

(Title)

AIRTIGHTNESS OF APARTMENTS

BEFORE AND AFTER RENOVATION

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1. Synopsis

This study aimed to research the airtightness of the building envelope in apartments before and after renovation. Measurements were carried out in three apartment buildings. One to four apartments were examined in each building. Typical renovation measures included changing the windows and refurbishing the interior surfaces. In some cases the ventilation system was renovated as well. No special emphasis was placed on the sealing of the envelope. The airtightness of the apartments increased in most of the cases. There were, however, apartments that became leakier during the renovation. Poor workmanship was usually evident in these cases. The main leakage route in the measured apartments seemed to be the balcony door.

2. List of symbols

C	coefficient
n	coefficient
n_{50} is	air leakage rate at a 50 Pa pressure difference [1/h]
q is	air flow rate through the measured element [m^3/h]
q_m	air flow rate through the measuring device [m^3/h]
q_{50}	infiltration flow at a 50 Pa pressure difference [m^3/h]
T	temperature of air going through the measured element [K]
T_m	temperature of air going through the measuring device [K]
V	volume of the apartment [m^3]
Δp	pressure difference across the measured element [Pa]

3. Introduction

The airtightness of the building has a significant effect on the energy use and thermal comfort in apartments. The successful use of air-to-air heat recovery requires good airtightness to reduce the bypass of exhaust air due to exfiltration. On the other hand, sealing the envelope may lead to underventilation if the ventilation system is not renovated accordingly. This study aimed to research the airtightness of the building envelope in apartments before and after renovation. Measurements were carried out in three apartment buildings. The air change rate, pressure distribution and airtightness were measured in 8 apartments.

4. Methods

4.1 Buildings

Measurements were carried out in three apartment buildings that have a mechanical exhaust ventilation system. Building A is situated in the middle part of Finland. One apartment was examined in that building. Buildings B and C are situated in southern part of Finland. Four apartments were examined in building B and three apartments were examined in building C.

Building A, which is three storeys high, was built in the 1960s. The ventilation did not function properly before the renovation. The ductwork was leaky and the mechanical exhaust ventilation system was centrally controlled without external air intakes. After the renovation,

each apartment had its own mechanical exhaust ventilation system with new ductwork. Furthermore, new air intakes were installed below the new windows. Measurements were carried out before the renovation in January 1996 and after the renovation in February 1996. In both cases the wind velocity was below 5 m/s.

Buildings B and C, which are both eight storeys high, were built in the 1970s. The ventilation was inadequate in both buildings before the renovation. The mechanical exhaust ventilation system was centrally controlled without air intakes. Some of the top floor apartments were found to be overpressurized. This caused humid indoor air to penetrate the wall structures and led to problems with dampness. Some occupants even got symptoms from fungi. Furthermore, some ground floor apartments had a positive air pressure compared to the hallway. Odours from these apartments penetrated into the hallway and from there into the top floor apartments, which were underpressurized as compared with the hallway. The ventilation was adequate in each apartment after the renovation. New air intakes were installed above the new windows. Each apartment now also has a sauna and balcony. Measurements were carried out before the renovation during January - February 1995 and after the renovation in October '96. In both cases the wind velocity was below 5 m/s.

4.2 Airtightness of the whole apartment

A fan-pressurization test was used to get information on the airtightness of the whole apartment envelope /1/. A fan was mounted in a duct that pierced the window panel, creating excess underpressure inside the apartment (Figure 1).
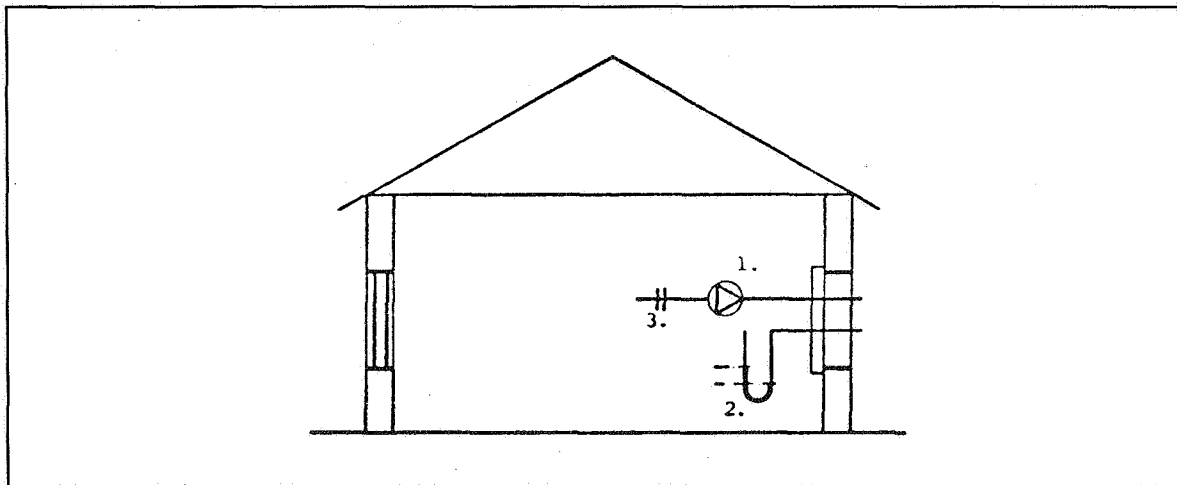
Figure 1. The principle of the fan-pressurization test. 1) adjustable fan 2) pressure difference meter 3) orifice plate for volume flow.

Both exhaust vents and external air intakes as well as the hallway door were sealed. The water seals of the floor drains and sinks were filled with water. All internal doors were left open. This caused air to enter the apartment via the cracks in the apartment envelope. The air flow rate of the incoming air was calculated from the air flow rate displaced by the fan and the density correction (equation 1).

$$q = q_m \frac{T}{T_m} \, . \tag{1}$$

The air leakage characteristics of the apartment envelope were determined by measuring the flow rate required to create a range of pressure differences (equation 2).

$$q = C\Delta p^n \, . \tag{2}$$

The coefficient C in equation 2 was corrected to standard conditions, that is 20 °C and 1 bar /1/. The final result of the airtightness is given by the air leakage rate at a 50 Pa pressure difference as

$$n_{50} = \frac{q_{50}}{V} \, . \tag{3}$$

The smaller the air leakage rate, the more airtight the apartment. In Finland there are currently no target values for the n_{50} in the whole apartment. However, a target value of $n_{50} = 1.0$ 1/h for apartment buildings is being planned. The target value for apartment buildings is 1.5 1/h in Canada, 2.2 1/h in the US and 4 1/h in Norway.

4.3 Airtightness of building components

Direct component testing was used in building A. The apartment was underpressurized as explained above. The individual building component was then sealed with a test chamber. The air that infiltrated through the component was conveyed to the orifice plates, whose pressure loss was compensated to zero by an auxiliary fan (Figure 2).
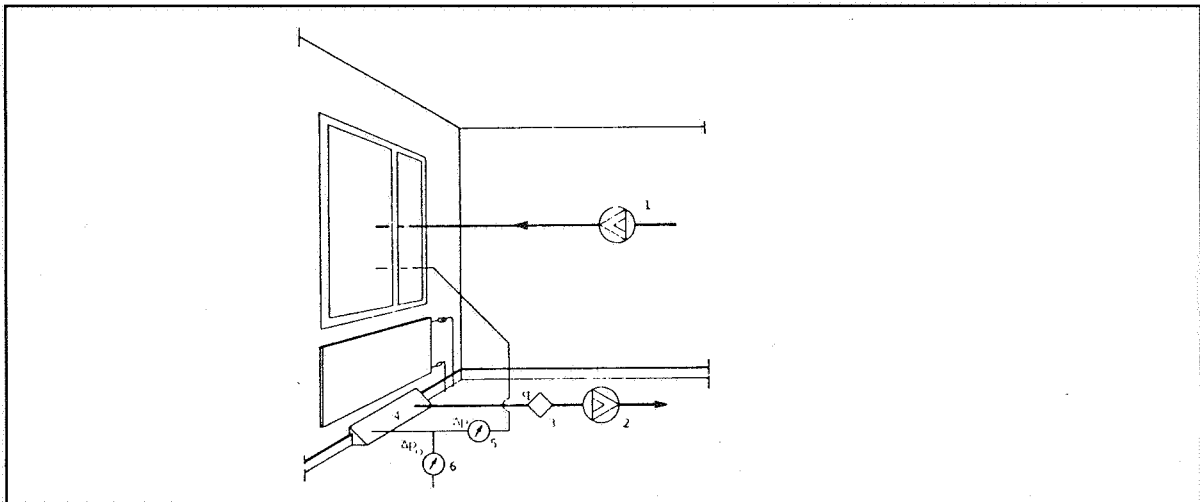
Figure 2. The principle of measuring airtightness of the building component (in this case the joint of the wall and floor). 1) adjustable fan 2) adjustable fan 3) orifice plate for volume flow 4) test chamber 5) pressure difference meter 6) pressure difference meter.

Indirect component testing through reductive sealing was used in buildings B and C. An airtightness test for the apartment was first carried out as explained in section 4.2. A component or group of components was then selected (e.g. windows) and sealed with adhesive tape and a polyethylene sheet. The airtightness test was then repeated. The difference between this and the first test was a measure of the airtightness attributable to the component or group of components which were sealed. Further components were then selected and the process continued.

4.4 Uncertainty of the measurements

The uncertainty factors in the airtightness measurements are the uncertainties of the air flow and pressure difference measurements and wind conditions. The measuring method (only an underpressurized situation has been examined) is also a source of uncertainty.

An orifice plate was used to measure air flows. The uncertainty of the air flow measurement with this method is estimated to be ±3 %. The wind velocity was below 5 m/s during the measurements, so its influence on the measured air flow can be estimated to be ±0.5 %. The influence of the measuring method on the measured air flow can be estimated to be ±1 %. Thus the expanded uncertainty of the measured air flow is ±3.2 %. An Alnor MP6KS micromanometer was used to measure the differences in pressure. The uncertainty of the pressure difference measurement is estimated to be ±0,6 Pa. The confidence level of the values given above is 95 %.

5. Results

5.1 Ventilation

The air change rate could be measured before and after the renovation only in buildings B and C. The results are presented in Figure 3: the ventilation unit operated at half capacity, which is the normal situation.
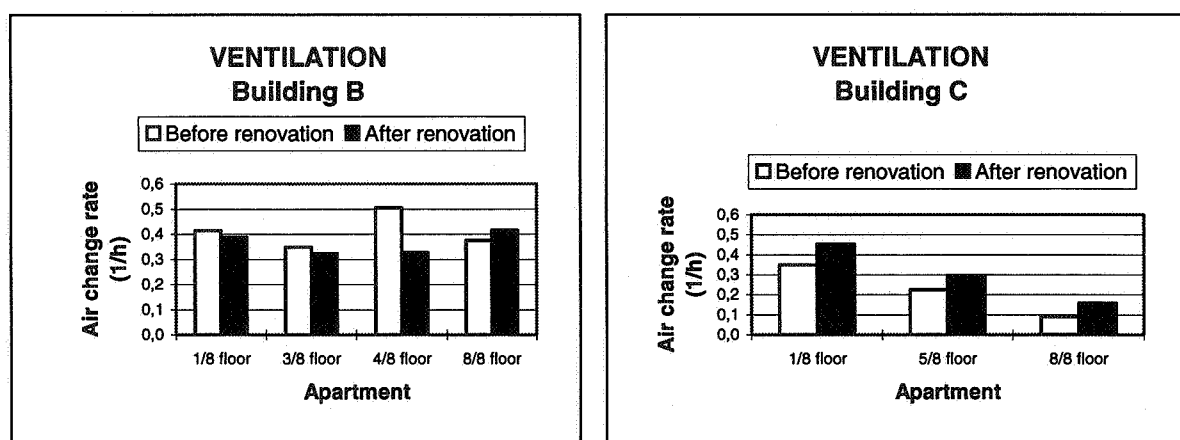
Figure 3. Air change rates in buildings B and C.

The air change rate in building B remained nearly the same after the renovation. In one apartment, the air change rate decreased remarkably. The ventilation improved in building C as a result of the renovation, but the air change rate in every apartment remained below the Finnish recommendation value of 0,5 1/h during normal ventilation conditions. The same situation existed in building B.

5.2 Pressure distribution

Pressure distribution could be measured before and after the renovation only in buildings B and C. The results are presented in Figures 4 and 5.

From Figure 4, one can see that the top apartments were underpressurized after the renovation.

The underpressure between the apartments and the hallway increased due to the renovation, but one apartment on the first floor of building C remained overpressurized (Figure 5).
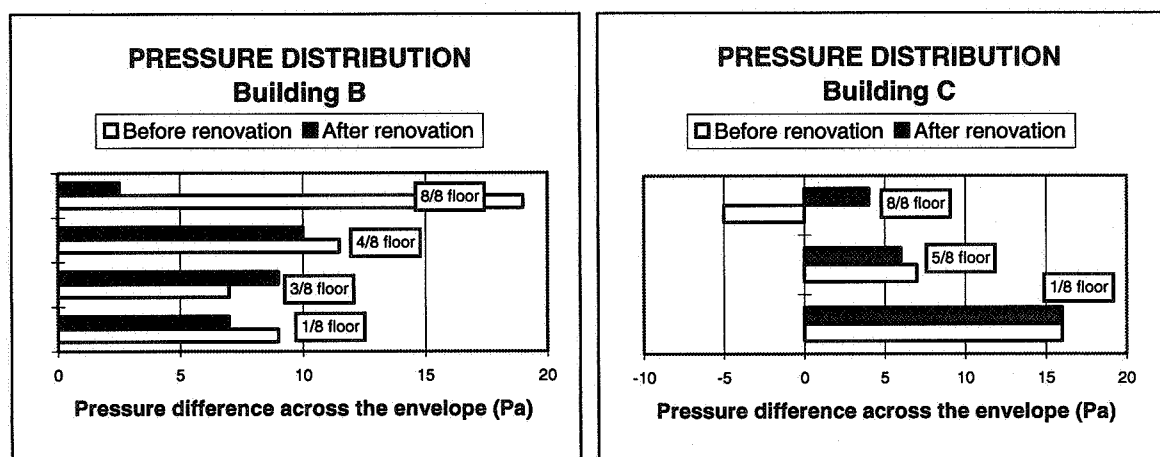
Figure 4. Pressure differences across the envelope in buildings B and C.
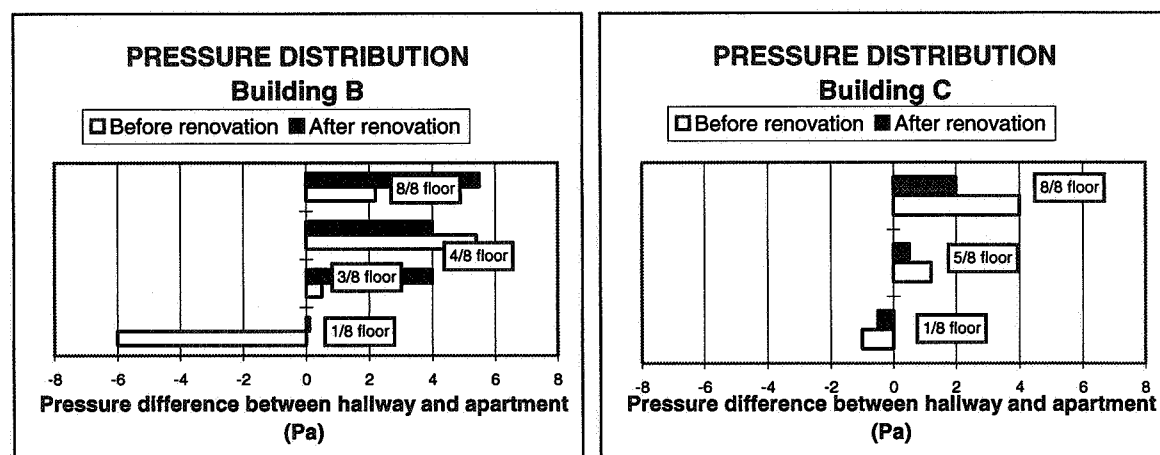
Figure 5. Pressure differences between hallway and apartment in buildings B and C.

5.3 Airtightness of the apartment

The airtightness of the whole apartment is presented in Figure 6. Airtightness is given by the air leakage rate at a 50 Pa pressure difference (n_{50}). The smaller the air leakage rate, the more airtight the apartment.

The airtightness of the apartment in building A improved as a result of the renovation (Figure 6). The air leakage rate value of 1.1 1/h after the renovation indicates a reasonably tight apartment.

In buildings B and C the airtightness of some apartments improved due to the renovation, but in a few other apartments it got worse (Figure 6). Poor workmanship was usually evident in these cases. A great difference in the airtightness of the apartments is apparent.
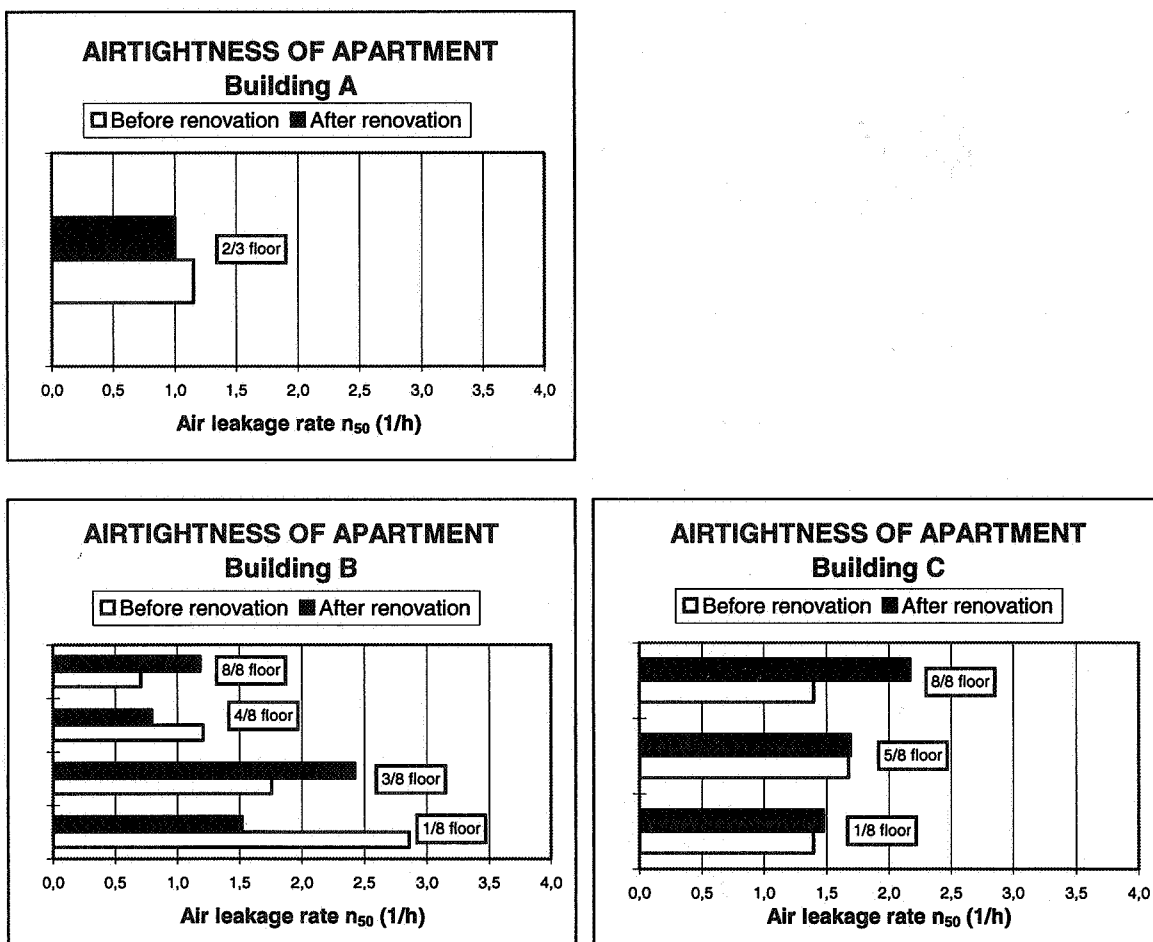
Figure 6. The air leakage rate n_{50} in three buildings.

5.4 Airtightness of building components

The airtightness of different walls was determined in building A (Figure 7). The direct component testing method was used, as explained in section 4.3. One can see from Figure 7 that the airtightness of every wall improved as a result of the renovation. One can also see that the balcony wall, including the door and window, was the leakiest one.

The airtightness of the windows, balcony wall and apartment door was determined in building B (Figure 8) by means of the indirect component testing method with reductive sealing, as explained in section 4.3. One can see from Figure 8 that the airtightness of the balcony wall worsened in many apartments as a result of the renovation. The airtightness of the windows and apartment doors improved in every apartment.
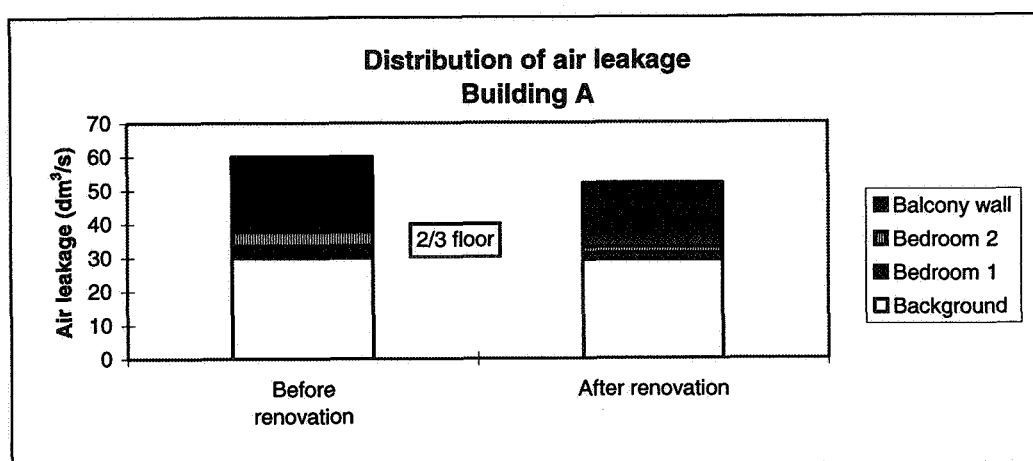
Figure 7. Airtightness of different walls in one apartment in building A.
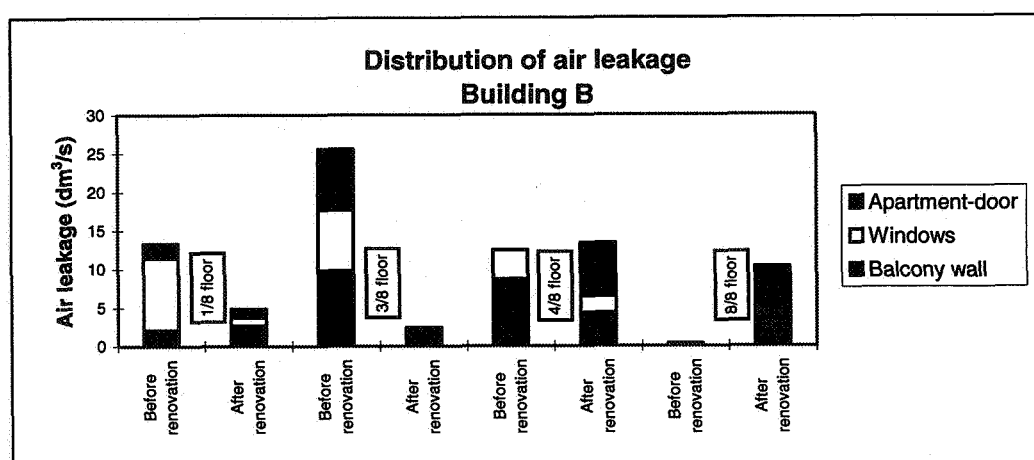
Figure 8. Airtightness of different building components in building B.
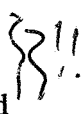
6. Conclusions

Eight apartments in three apartment buildings that have a mechanical exhaust ventilation system were examined before and after their renovation. The measurements that were made included examinations of the air change rate, pressure distribution and airtightness.

The apartments had no air intakes before the renovation and the ventilation was inadequate in every building. Some of the top floor apartments were overpressurized. In Finland's climatic conditions, this often leads to dampness and mould problems. Renovation measures included improving the ventilation system, changing the windows and refurbishing the interior surfaces. No special emphasis was placed on the sealing of the envelope.

After the renovation, every apartment had new air intakes below or above the new windows. The ventilation improved in many apartments as a result of the renovation. However, the air change rate in every single apartment was below the Finnish recommendation value of 0.5 1/h during normal ventilation conditions.

The underpressure between the apartments and the hallway increased due to the renovation, but one apartment on the first floor remained overpressurized. Every apartment was underpressurized after the renovation.

A great difference in the airtightness of the apartments was noticed. Airtightness ranged from 0.7 1/h to 3.1 1/h before the renovation and from 0.8 1/h to 2.9 1/h after the renovation. In most cases, the airtightness of the apartments increased due to the renovation. There were, however, apartments which became leakier. Poor workmanship was evident in these cases. The main leakage route seemed to be the balcony door. The airtightness of the windows and the apartment door improved in every single case.

Acknowledgements

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Title: Airtight Buildings - A Practical Guide

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Airtight Buildings - A Practical Guide

Although several investigations on how to design airtight buildings have been performed and the results furthermore have been published, many designers and contractors are still unaware of this knowledge. Therefore, the aim of this work is to collect existing knowledge and put it together to a practical guide. The target groups are architects, designers, contractors and building services engineers.

This paper is a summary of the report "Good Airtightness - guidelines to architects, building designers and contractors" published in Sweden during the autumn of 1997 [1]. The guide contains six different chapters: 1) motives for air-, diffusion- and windtightness; 2) materials that may be used for the purpose; 3) 70 drawings and specifications on how to make air-, diffusion- and windtight constructions; 4) theory; 5) measurements; and, 6) quality assurance system for gaining air-, diffusion and windtight buildings.

The content of chapter 1, 3 and 6 are emphasised in this conference paper.

1 Introduction

Almost all of us trust in the positive effect of having a wind barrier in a building. The wind barrier will prevent the wind from blowing into the building envelope, e g the external wall, and reduce the function of the thermal insulation. The wind barrier will also prevent rain, which has penetrated through the facade from finding its way into the building envelope.

In the same manner there are a number of motives why a building should be airtight (preventing air to leak through the envelope of the building): thermal comfort for the users, rational use of energy, control of ventilation and reducing the risk for moisture problems.

2 Objectives

Although several investigations on how to design airtight buildings have been performed and the results furthermore have been published, many designers and contractors are still unaware of this knowledge. Therefore, the aim of this work is to collect existing knowledge and put it together to a practical guide. The target groups are architects, designers, contractors and building services engineers.

3 Results

This paper is a summary of the report "Good Airtightness - guidelines to architects, building designers and contractors" published in Sweden during the autumn of 1997 [1]. The guide contains six different chapters: 1) *motives* for air-, diffusion- and windtightness; 2) materials that may be used for the purpose; 3) 70 drawings and specifications on how to make air-, diffusion- and windtight constructions; 4) theory; 5) measurements; and, 6) quality assurance system for gaining air-, diffusion and windtight buildings.

The content of chapter 1, 3 and 6 are emphasised in this conference paper.

3.1 Motives for making air-, diffusion- and windtightness

There are four major motives for making a building *airtight*. The first one is to save energy. In a building with poor airtightness the air will flow in and out through the envelope of the building. These air movements cause an extra energy flow. The size of the extra energy flow depends however on the ventilation system used. In natural ventilation systems and mechanical supply and exhaust ventilation air, the energy flow increases linearly with the air leakage through the envelope of the building. In mechanical exhaust ventilation the air leakage through the envelope is of minor importance for the extra energy flow, since the exhaust ventilation creates a negative pressure in the building, which prevents the exfiltration. In a building with a heat exchanger or heat pump in the ventilation system, the air tightness is of greater importance. In these systems the heat exchanger/heat pump will not perform to its fully extent, since all the air will not pass the exchanger.

The second motive is thermal comfort. When cold outdoor air is leaking through the envelope of the building, the indoor surfaces will be cooled. These surfaces will cause cold down draught. The consequence is unpleasancy for the user. Also direct draught is unpleasant. In order to solve this problem, the user might want to increase the indoor temperature, which means an increasing energy use.

The third motive is to prevent moisture problems. In a building with poor airtightness the air flows through the envelope. When air is flowing from indoor to outdoor, the air brings moisture since the indoor air contains more vapour than outdoor air. When indoor air flows through the envelope, humidity will increase when temperature decreases. If the envelope has a poor airtightness there might be moisture problems in the envelope.

The fourth motive is that if the airtightness of the envelope is poor, the wind and outdoor temperature will affect the ventilation flows in the building. A ventilation system that works properly in one outdoor climate may work poorly in another extreme situation. The consequence is that the air change rate sometimes is too high or too low, depending on the outdoor conditions. An increased air change rate will cause an extra energy use.

The motive for making a building diffusion tight is to prevent indoor air containing moisture to diffuse out in the envelope of the building. The moisture content in indoor air is normally larger than in outdoor air. The moisture flow caused by diffusion, is however smaller than the moisture flow due to poor airtightness (convection).

The motive of having a wind barrier in a building is to prevent the wind in getting into the envelope of the building and reduce the function of the thermal insulation. The second motive is to prevent rain, which has penetrated through the facade from getting into the building envelope.

3.2　　　Drawings and specifications

The hardest part in getting an air-, diffusion- and windtight building are the connections between different constructions and around services penetrations, Figure 1, and it's actually these parts that have the largest influence on the air- and diffusiontightness.

The original report "Good Airtightness" contains over 70 drawings and specifications on different connections. Figure 2 is an example of a connection between a crawl space, an intermediate floor and an external wall. Figure 3 is an example of a connection between an external wall, an intermediate floor and a roof.

Figure 4 is an example on how to make an airtight construction around a service penetration.
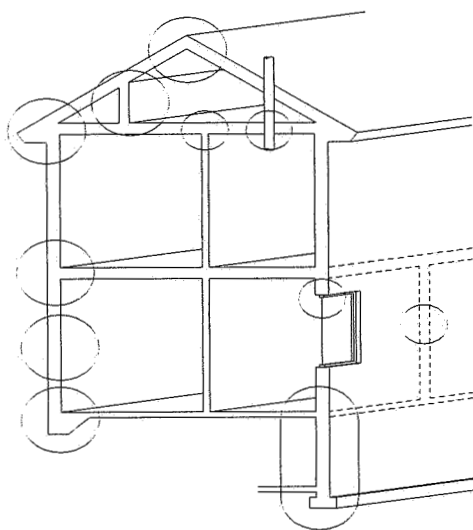
Figure 1: The hardest part in getting an air-, diffusion- and windtight building are the connections between different constructions and around services penetrations These areas are marked in the figure.
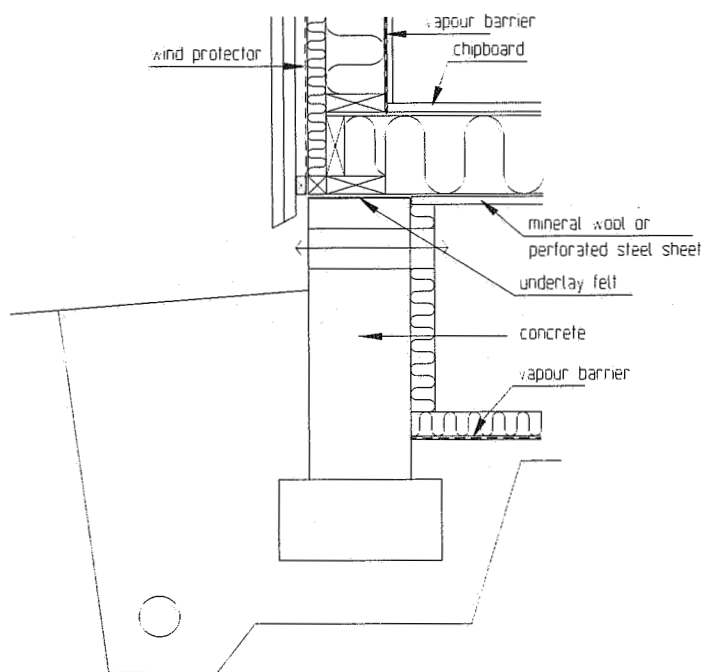
Figure 2: The figure shows how a connection between a crawl space, floor structure and external wall may be designed by means of air-, diffusion and windtightness [2]. The floor structure should have no diffusion barrier since the vapour content in the crawl space is sometimes higher and sometimes lower than indoor. Nevertheless the intermediate floor should have good airtightness. This may be achieved by having tongue-and-groove chipboards, glued and screwed to the joints. The counter floor is covered with a non organic material, e.g. mineral wool or perforated steel sheet.
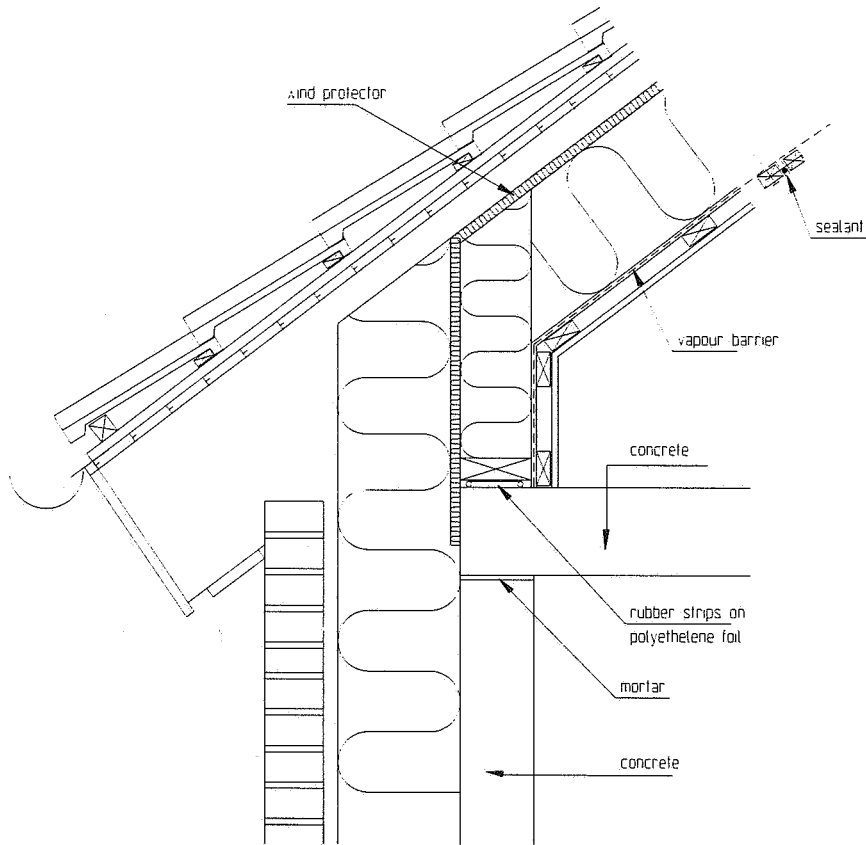
Figure 3: The figure shows a connection between an external wall, an intermediate floor and a roof. Mortar is placed in-between the external wall and the intermediate floor to achieve airtightness. Polyethelene foil with rubber strips are placed under the extra wooden sill between the roof trusses in order to achieve airtightness. The overlap joint of the polyethylene foil in the ceiling is clamped between two battens and to the sill.
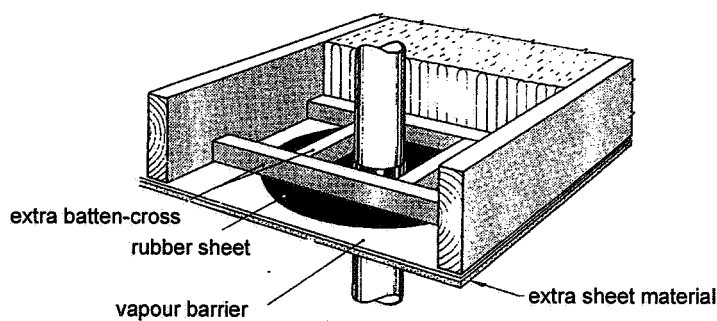
Figure 4: The figure shows how to achieve airtightness around a service penetration [3]. A rubber sheet is surrounded the pipe. The hole in the rubber sheet should be smaller than the dimension of the pipe. In this way the rubber sheet will surround the pipe airtight. The vapour barrier in the ceiling is clamped with the rubber sheet and pressed with a sheet material to the extra "batten-cross".

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3.3   Quality assurance system

If the building has a poor air-, diffusion- and windtightness, the risk of getting inconvenience increases. Table 1 shows what kind of inconvenience you may get if the constructions in the buildings have poor tightness.

airleakage through:	inconveniences:					
	moisture	energy	draught	radon	airflow	noise
foundation	x	x	x	X	X	
external wall	x[1]	x	x		x	x
sill and top plate	x[1]	x	X			x
around a window	x[1]	x	X			x
roof	X[1]	x				
walls between apartments					X	X

[1] only if there is a positive pressure indoor

Table 1: The table shows what kind of inconvenience you may get if the constructions in the buildings have poor tightness. The inconveniences are marked with a x. A capital X means a larger inconvenience than the smaller x.

In order to help designers and contractors to achieve good air-, diffusion- and windtightness, special check-lists have been made, see Table 2.

4   Conclusions

It is important to build a house with good air-, diffusion- and windtight envelope. Otherwise you will increase the energy use, reduce the thermal comfort, increase the risk of moisture problems and have less control over your ventilation.

Acknowledgements

Checklist for building contractors	OK	Date	Comment
Have the site manager and the tradesmen discussed the motives for air-, diffusion- and windtight?			
Have the site manager and the tradesmen overviewed the drawings of air-, diffusion- and windtightness produced by the designers?			
Have the building, electrical and vent contractors met an agreement of whom making airtightness at services penetrations?			
Have these connection been tightened:			
foundation/bottom floor/external wall			
external wall/windows			
external wall/doors			
external wall/intermediate floor			
external wall/roof			
external wall/internal wall			
roof/internal wall			
roof/attic door			
Are the services penetrations airtightned?			

Table 2: The table shows an example of a check-list for building contractors, which may be used for obtaining good air-, diffusion- and windtightness.

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2. Norwegian Building Research Institute, 1987: *Bindingsverk av tre - Varmeisolasjon og tettesjikt* (in Norwegian). Byggforskarserien. Byggdetaljer A 523.255. Oslo. Norway.

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VENTILATION AND COOLING

18TH ANNUAL AIVC CONFERENCE
ATHENS, GREECE, 23-26 SEPTEMBER, 1997

(Title) **PROBABILISTIC MODEL OF HEAT LOSS THROUGH THE BUILDING ENVELOPE**

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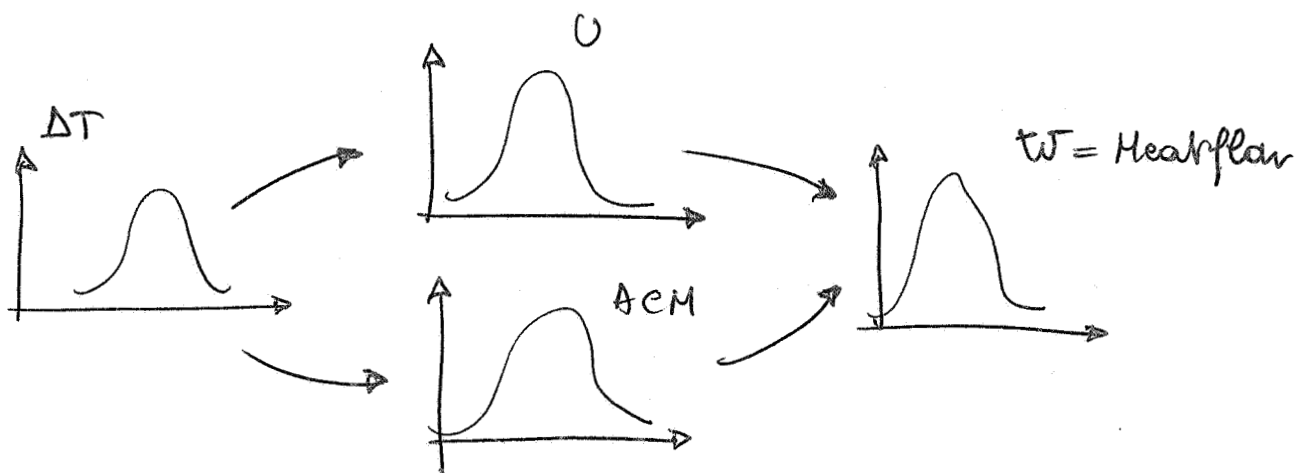
SYNOPSIS

A model for the application of probabilistic methods in the estimation of heat loss caused by convection and heat conduction through the material is developed. Temperature difference (ΔT) between inside and outside of a building, air change rate (ACH) and coefficient of thermal transmittance (U-value) of the building structure are treated as random variables. The mean value and the standard deviation of heat loss are estimated for different parameters of distributions for temperature difference, air change rate and thermal transmittance. The interaction between the above three quantities is described for different cases of tight and leaky envelopes by assuming certain degree of correlation between the random variables.

1.   INTRODUCTION

The total heat loss in a building is generally calculated by adding the contributions from the heat loss due to natural ventilation and transmission through the materials, where the material properties, thermal transmittance and convection are treated as deterministic quantities. Influence of variation of leakage properties of the building, climatic conditions and the material properties is not catered for in the assessment of heat loss.

Thermal performance of a building is associated with two main quantities selected from several parameters of interest, which can be defined as:

> *heat loss due to air flow through the construction*
> *heat loss due to transmission*

The heat loss due to natural ventilation and transmission through the building structure is determined in a complex interaction, by local climatic conditions and the factors related to the building and their surroundings. The heat loss due to transmission is governed by the temperature difference across the envelope, properties of material and the type and quality of construction. Air infiltration is governed by the pressure difference due to wind and temperature and the leakage characteristics of the house. Examination of the effects of air infiltration and heat transmission on the thermal performance of the building indicates that these are interdependent and climate is the common parameter, which is random in space and time.

2.   DETERMINISTIC MODEL FOR HEAT LOSSES IN A BUILDING

2.1   Heat loss due to ventilation

Heat loss caused by ventilation or infiltration (called later ventilation heat loss) forms a significant part of the energy consumption in buildings. To keep the internal temperature at a constant level, fresh air supplied to the house must be heated. The common measure of the amount of exchange of air is air change rate (ACH).

The energy needed for heating of the supplied air can be described by following equation:

$$W_v = \Delta T \rho c V \frac{ACH}{3600} \tag{2.1}$$

where

W_v - heat loss (kW)

ΔT - temperature difference between inside (T_{int}) and outside (T_{ext}) the building (K)

V   - volume of the house (m^3)

ρ - air density (kg/m^3)

c   - specific thermal capacity (kJ/kgK)

2.2 Heat loss due to transmission

Transmission loss through the building envelope is the sum of heat losses through the building components and can be described by the following equation [3]:

$$W_c = \Delta T \frac{1}{1000} \sum_{i=1}^{m} Nu_i U_i A_i \tag{2.2}$$

where:

W_c - transmission heat loss (kW)

ΔT - temperature difference between inside and outside the building (K)

m   - number of building components

U_i - overall average thermal transmittance of i:th component (W/m^2K)

A_i - area of the i:th component (m^2)

Nu_i - Nusselt's number for the i:th component describing the effect of convection flows (leakage and interstitial convection) on the thermal performance of a structure

In order to include the changes in the average thermal transmittance of the components caused by the influence of the convective flow on conduction heat losses, one can define the overall average thermal transmittance of a building envelope U_m as:

$$U_m = \frac{1}{A_m} \sum_{i=1}^{m} Nu_i U_i A_i \tag{2.3}$$

where: A_m- Area of the building envelope (m^2)

Equation 2.2 can be rewritten by including equation 2.3 as:

$$W_c = \Delta T \, U_m \, A_m / 1000 \tag{2.4}$$

3. PROBABILISTIC MODEL OF HEAT LOSSES THROUGH THE BUILDING ENVELOPE

A probabilistic model for the estimation of heat loss is developed by including the variations in the temperature difference ΔT, air change rate ACH and average heat transfer coefficient

U. Influence of variation in the climatic parameters and the material properties on the heat loss is studied for two types of buildings with tight and leaky envelope. It is assumed in the analysis that description of permeability of an envelope is related to the outer surface of the building. Total heat loss in a building at a time t can be described as a sum of ventilation and transmission heat losses:

$$W(t) = W_V(t) + W_C(t) \tag{3.1}$$

where:

$W(t)$ - total heat loss in a building (kW)

$W_V(t)$ - component due to ventilation heat loss (kW)

$W_C(t)$ - component due to transmission heat loss (kW)

Substituting expressions 2.1 and 2.4 into 3.1 gives:

$$W(t) = \frac{\rho c V}{3600} \Delta T(t) ACH(t) + \frac{Am}{1000} \Delta T(t) Um(t) \tag{3.2}$$

The main assumption is to include the randomness of the thermal transmittance and the climate in the model. The model should enable the estimation of the mean value and the standard deviation of the heat loss for the assumed statistical parameters of distributions of particular random variables. Four parameters from equation 3.2 are considered to vary with time.
Define

$\Delta T(t)$ $= X(t)$ inside-outside temperature difference

$ACH(t) = Y(t)$ air change rate

$U_m(t)$ $= Z(t)$ average overall thermal transmittance

$W(t)$ ventilation and transmission heat loss in a building

For a constant value of internal temperature of 20°C the air density becomes also constant. Equation 3.2 can now be written as:

$$W(t) = g(X,Y,Z) = a\, X(t)\, Y(t) + b\, X(t)\, Z(t) \tag{3.3}$$
where $a = V c \rho / 3600 \tag{3.4}$
   $b = A_m / 1000 \tag{3.5}$

It is important to emphasis that the temperature difference across the building envelope is a common parameter for both air change rate and the thermal transmittance. Air infiltration is governed by the pressure difference due to wind and temperature while thermal transmittance depends on the temperature of the material and the leakage paths through the insulation Approximate mean and variance of the function W may be obtained by expanding g(X,Y,Z) in a Taylor series about the mean values of the variables X, Y and Z. For practical reasons the first-order approximations for mean μ_w as well as for variance σ_w^2 of the function W have been used.

$$\mu_w = a\mu_x\mu_y + b\mu_x\mu_z \tag{3.6}$$

$$\sigma_w^2 = (a\mu_y + b\mu_z)^2\sigma_x^2 + (a\mu_x)^2\sigma_y^2 + (b\mu_x)^2\sigma_z^2$$
$$+2(a\mu_y + b\mu_z)a\mu_x\rho_{xy}\sigma_x\sigma_y$$
$$+2(a\mu_y + b\mu_z)b\mu_x\rho_{xz}\sigma_x\sigma_z + 2ab\mu_x^2\rho_{yz}\sigma_y\sigma_z \qquad (3.7)$$

where:

μ - mean value

σ - standard deviation

ρ - correlation coefficient

Assume that the mean values of X, Y, Z are the design values and the variations from the mean are caused by the influence of random parameters (climate etc.). It has been found [4] that for a naturally ventilated house with the uniformly distributed leakage over the building envelope, the distribution of air change rate can be defined by Normal probability density function. Temperature difference [4] and thermal transmittance are also assumed as normally distributed. A joint normal distribution function for the variables X,Y and Z has been applied in the model. The assumption of Normal probability density function for the heat losses has been verified for all cases of tight and leaky envelopes by comparing the statistical moments obtained by two different methods [2]. Normal distribution is found to be reasonable for a tight building with balanced or exhaust ventilation. However , for a leaky building, a slightly skewed distribution function gives a better fit.

Correlation coefficients among variables X, Y, Z show linear dependence between them and they are specific for individual case (tight envelope, infiltration, exfiltration).

The coefficient of partial correlation of air change rate and thermal transmittance, if the influence of temperature difference is eliminated, is of the form [1]:

$$\rho_{yz.x} = \frac{\rho_{yz} - \rho_{xy}\rho_{xz}}{\sqrt{(1-\rho_{xy}^2)(1-\rho_{xz}^2)}} \qquad (3.8)$$

For a tight building the coefficient of partial correlation $\rho_{yz.x}$ is equal to zero. It is negative for the case of infiltration of the cold air entering the building and positive for the case of exfiltration of the warm air out of the building.

The range of values of the correlation coefficient ρ_{yz} can be estimated from the knowledge of ρ_{xy}, ρ_{xz}, and $\rho_{yz.x}$. Limits for ρ_{yz} are presented in table 1.

1	Tight Envelop	$\rho_{yz.x} = 0$	$\rho_{xy}>0.\ \rho_{xz}<0.\ \rho_{yz}<0.$ $\rho_{yz}=\rho_{xy}\rho_{xz}$
2	Leaky Envelope Infiltration	$\rho_{yz.x} < 0$	$\rho_{xy}>0.\ \rho_{xz}<0.\ \rho_{yz}<0.$ $\rho_{xy}\rho_{xz} - \sqrt{(1-\rho_{xy}^2)(1-\rho_{xz}^2)} \leq \rho_{yz} < \rho_{xy}\rho_{xz}$
3	Leaky Envelope Exfiltration	$\rho_{yz.x} >0$	$\rho_{xy}>0.\ \rho_{xz}>0.\ \rho_{yz}>0$ $\rho_{xy}\rho_{xz} \leq \rho_{yz} \leq \rho_{xy}\rho_{xz} + \sqrt{(1-\rho_{xy}^2)(1-\rho_{xz}^2)}$

Table 1:   *The range of values assumed for the correlation coefficients among random variables X, Y and Z*

4. SENSITIVITY ANALYSIS OF HEAT LOSS IN A BUILDING

Sensitivity analysis of coefficient of variation of heat losses is carried out by considering certain ranges of values of the parameters ρ_{xy}, ρ_{xz}, ρ_{yz}, I_x, I_y, I_z and are estimated for the five cases shown in table 2

The coefficient of variation of heat loss is given by:

$$I_w = \sqrt{(A + B + C)} / (a + b\,s) \tag{4.1}$$

where
$$s = \mu_z / \mu_y.$$
$$A = (a + b\,s)^2 \, I_x^2$$
$$B = 2\,(a + b\,s)\,I_x\,[a\,I_y\,\rho_{xy} + b\,s\,I_z\,\rho_{xz}]$$
$$C = (a^2 I_y^2 + 2\,a\,b\,s\,I_y\,I_z\,\rho_{yz} + b^2\,s^2\,I_z^2)$$

Case No.	Building Type	Design Conditions
1	Tight building	ACH=0, *Only transmission heat losses are considered and is a purely theoretical case.*
2	Tight building	ACH=const, *Balanced ventilation is provided by mechanically operated inlets and outlets.*
3	Tight building	ACH≠const , *Exhaust ventilation is provided by mechanically operated outlets. The air inlets are specified openings through ducts.*
4	Leaky building	*(a) Infiltration is provided through the building envelope and the outlets are placed above the neutral pressure layer (Nu = 1.0)* *(b) Infiltration is provided through the building envelope and the mechanical outlets. (Nu = 0.8)*
5	Leaky building	*(a) Exfiltration is provided through the building envelope and the inlets are placed below the neutral pressure layer. (Nu = 1.0)* *(b) Exfiltration is provided through the building envelope and the inlets are placed below the neutral pressure layer (Nu = 1.2)*

Table 2:   Cases considered in the Analysis

Figure 1 shows the above fives cases considered in the analysis.

ACH = 0

Case 1

ACH = Constant

Case 2

ACH ≠ Constant

Case 3

NPL

Case 4

Infiltration

Neutral Pressure Layer (NPL)

Case 5

Exfiltration
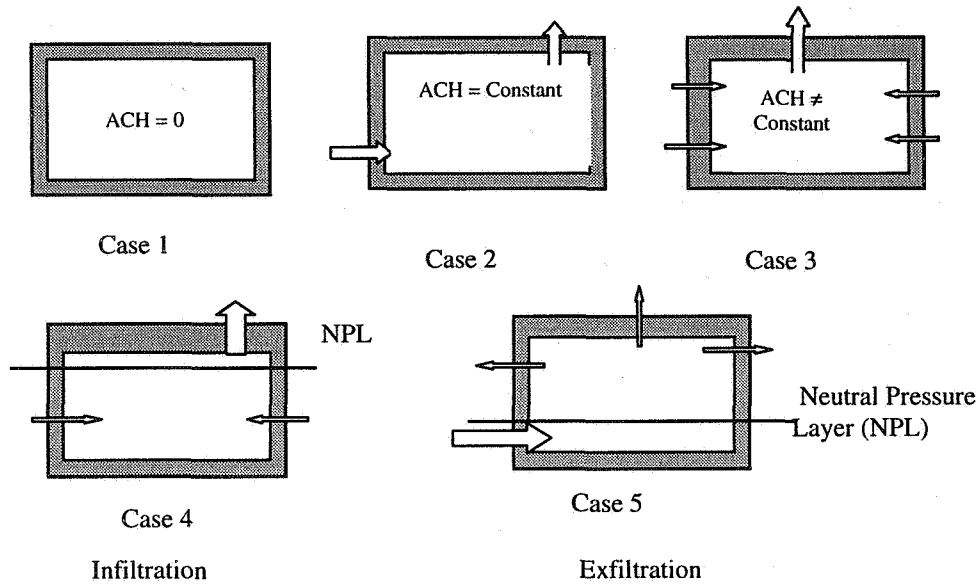
Figure 1   *Illustration for different cases.*

The influence of different parameters on the coefficient of variation of heat loss I_w is investigated for tight and leaky envelope, and table 3 presents the results of the investigation for a single family house.

	Case 1	*Case 2*	*Case 3*	*Case 4 (a)*	*Case 4 (b)*	*Case 5 (a)*	*Case 5 (b)*
Nu	-	-	-	1.0	0.8	1.0	1.2
I_x	0.25	0.25	0.25	0.25	0.25	0.25	0.25
I_y	0.00	0.00	0.01	0.25	0.01	0.25	0.25
I_z	0.025	0.025	0.025	0.05	0.10	0.05	0.10
ρ_{xy}	0.0	0.0	0.1	0.5	0.08	0.5	0.5
ρ_{xz}	-0.9	-0.9	-0.9	-0.9	-0.9	0.5	0.5
ρ_{yz}	0.0	0.0	-0.09	-0.5	-0.5	0.5	0.9
μ_w	1.32	3.60	3.60	3.60	3.33	3.60	3.86
σ_w	0.30	0.87	0.87	1.23	0.72	1.36	1.44
I_w	0.23	0.24	0.24	0.34	0.22	0.38	0.37

Table 3   *Estimation of transmission and ventilation heat flow in a single family house.*

Data for the numerical example used in the calculation is as follows.

Area of the house A = 413 m²,   Volume of the house V = 819 m³
Specific thermal capacity c = 1.0 kJ / kg K   Air density ρ = 1.25 kg / m³

Examination of the table shows that the influence of temperature is the governing factor in all cases. It should be noted that the influence of wind velocity is included in the evaluation of air change rate. Case 1 is a hypothetical case without any practical relevance. It is used as a reference case for transmission heat losses without the influence of ACH: Case 2 represents the influence of mechanically operated inlets and outlets. It can be noted that the coefficient of variation of the heat loss is approximately equal to the coefficient of variation of temperature difference and the influence of correlation is insignificant. The third case is similar to case 2 and again the variations in temperature are dominant.

Cases 4 and 5 are for the leaky buildings where the location of neutral pressure layer (NPL) is considered in the analysis. There can be infiltration or exfiltration through the envelope of the structure depending on the height of the NPL. Influence of Nusslet´s number (Nu) is also considered in cases 4 and 5.

Comparison of probability density function for different cases were studied. Figure 2 shows two cases corresponding to tight building with balanced ventilation (Case 2) and leaky building with exfiltration (Case 5 a). The dotted lines indicate the results obtained by using normal density function and the solid lines show the results based on first order reliability model (FORM) as given in reference 2. It can be noted that normal distribution gives a good fit for the tight building while the probability density function is skewed to the right for the leaky building. However, assumption of normal distribution is reasonable even for leaky structures and will be used in further investigations.

Heat loss W
(kW)
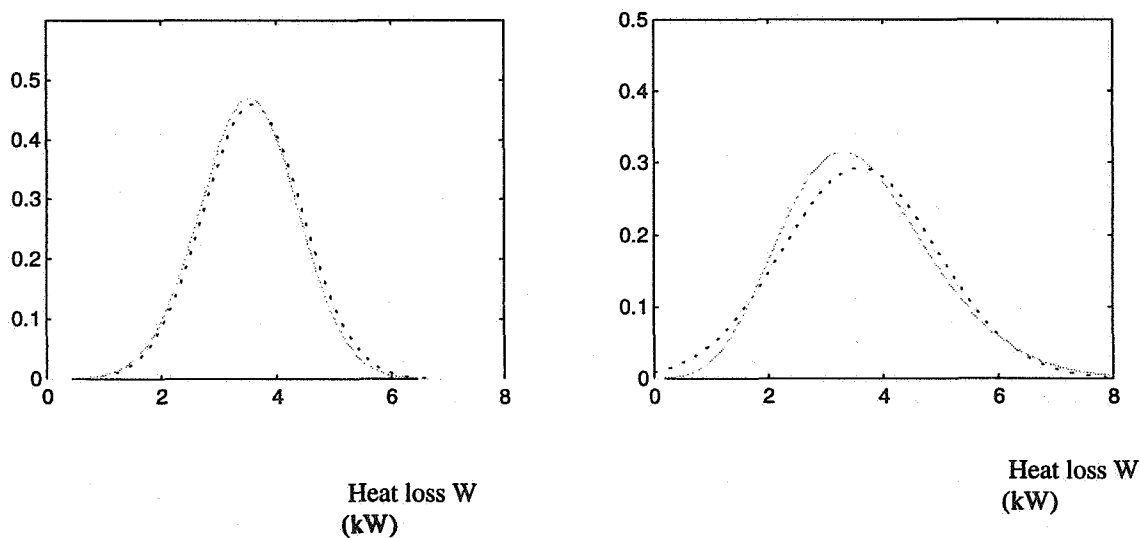
Heat loss W
(kW)

Figure 2   *Probability density functions for heat loss for tight and leaky building.*
(Cases 2 and 5)

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5. CONCLUSIONS

Application of probabilistic model for estimation the heat losses in a building gives us the possibility of taking into account the interactions between ventilation and transmission heat losses. The main conclusions from the work are:

1. The model calculates the first two moments (mean value, standard deviation and correlation) for the heat transmittance coefficient, air change rate and the temperature variations over a specified period of time. It is found that Normal density function is reasonably representative of the heat loss caused by conduction and ventilation.
2. Influence of variations in temperature and air change rate is important for calculating the coefficient of variation of heat loss.
3. Influence of variations in the transmittance properties of the material is not significant for this particular example. This may be due to the fact that small variations in U are assumed in the model.
4. Influence of Nusselt´s number on the total variation in the heat loss is also non-significant due to small mean temperature differences and quasi static heat transmittance coefficient.
5. The analysis have shown that correlation between temperature and thermal transmittance does not have any significant influence, while correlation between temperature difference and air change rate is important and should be considered in the calculations.
6. Further work is needed to develop a model which considers the non-linearities inherent in the analytical model and examine their influence on the total heat loss.

6. ACKNOWLEDGEMENTS

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Title: Airtightness of New Belgian Dwellings: An Overview Picture

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Airtightness of new Belgian dwellings : an overview picture

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Table of contents

1. Introduction

In the framework of the Flemish Impulse Programme on Energy Technology (VLIET), the project called SENVIVV is running from January 1995 till September 1997. The major objective is to obtain a detailed picture of various characteristics of dwellings constructed during the period 1990 – 1995. To achieve this, a representative sample of 200 dwellings is analysed in detail. The final report of this project is expected to be available at the end of 1997. This report gives an overview of detailed measurements carried out in 20 more or less representative recently constructed dwellings in Belgium.

2. Results of pressurisation measurements

All measurements were carried out on the so-called **protected volume**. This is the part of the house which is thermally insulated from the environment.

2.1 Global airtightness

An overview of the measured n50-values as a function of the year of construction is given in figure 1.
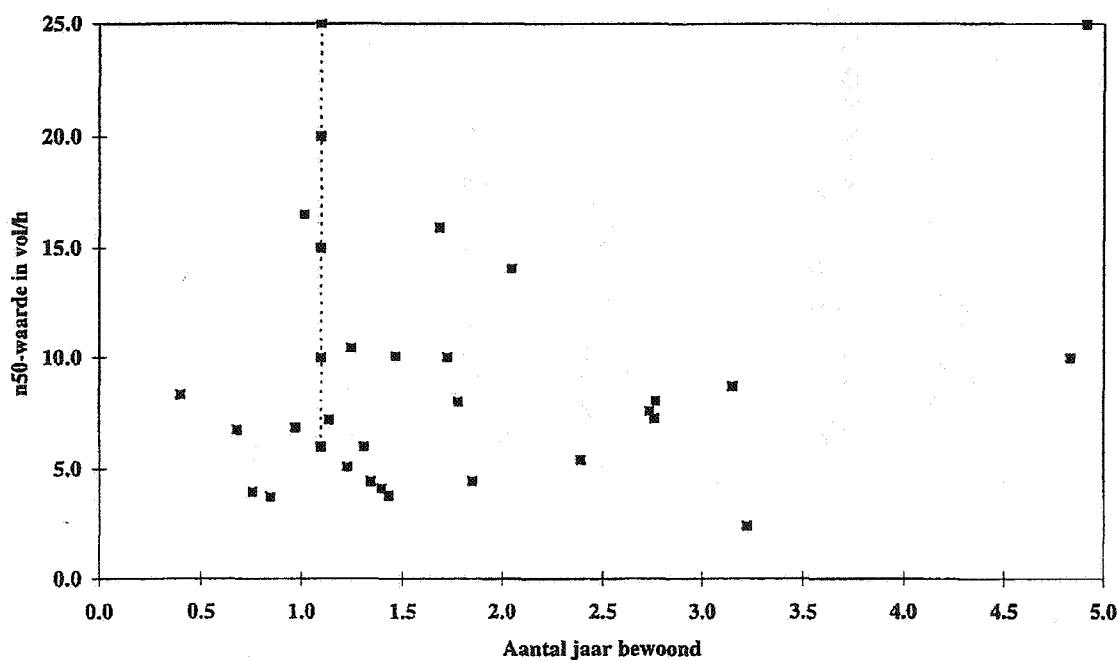
figure 1 : overview of measured n_{50}-values as function of duration of occupation

The following conclusions can be drawn :

- There is a very large spread in the results: the n_{50}-values range from 2.5 h^{-1} to 25 h^{-1} or a variation of a factor 10;

- The average value is about 8 h^{-1}. When applying the rule of thumb that the average seasonal air change rate is of the order of n50/20, one finds an average seasonal air change rate of 0.4 h^{-1}.

- There is no significant increase in the measured airtightness as a function of duration of occupation.

- The dotted line represents a dwelling of which the attic space was not yet finalised. The various points correspond with different airtightness situations.

figure 2 gives the histogram of the measured n50-values. A substantial difference is found when comparing individual dwellings and apartments. The average n50-values are the following :

- apartments : $n_{50} = 4.8 \text{ h}^{-1}$
- individual rowhouses : $n_{50} = 6.4 \text{ h}^{-1}$
- individual free-standing individual dwellings : $n_{50} = 9.6 \text{ h}^{-1}$
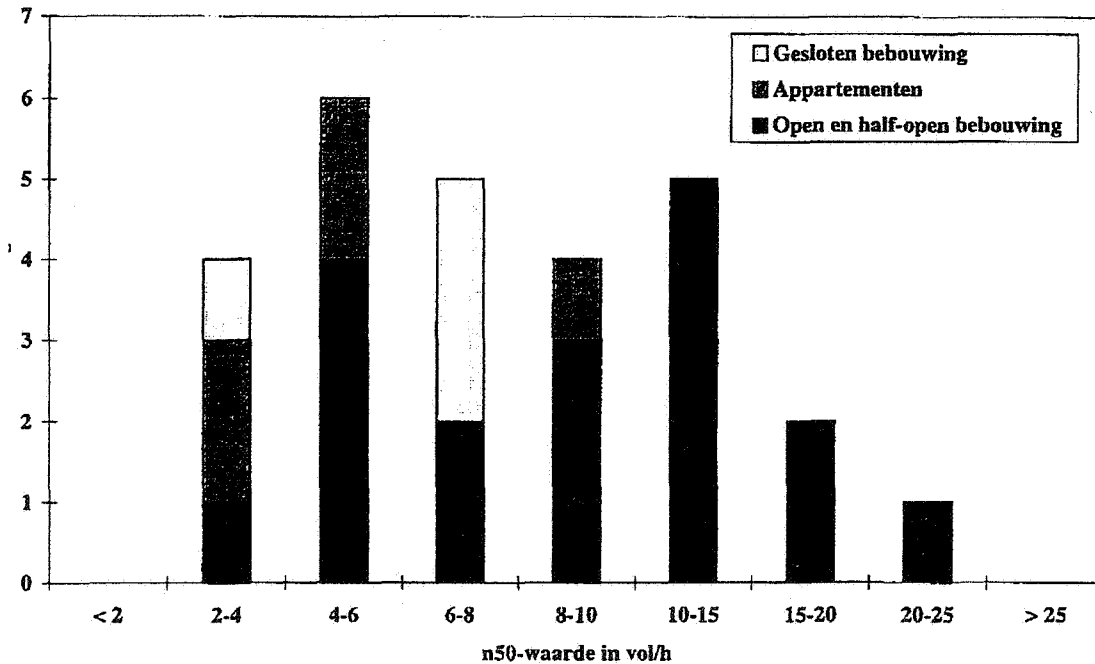
figure 2 : histogram of n50-values with splitting up as function of building type

An important information is the location of the leakages. This is discussed in the next paragraph.

2.2 Airtightness of the different leakages

- A first important observation is the fact that the most important leakages are situated in the non-occupied spaces : garage, insulated attics,...

- In case a garage exists, it represents about 1/3 of the total leakage.

- In case of an insulated attic, it represents on the average 50% of the total leakage.

- For a number of dwellings, the n50-value would drop down to less than 30% if the above mentioned spaces would be outside the protected volume.

Besides these very leaky spaces, there are also very airtight spaces. In most cases, the bedrooms are often very airtight. An overview of the measured Q50-values is presented in figure 3.
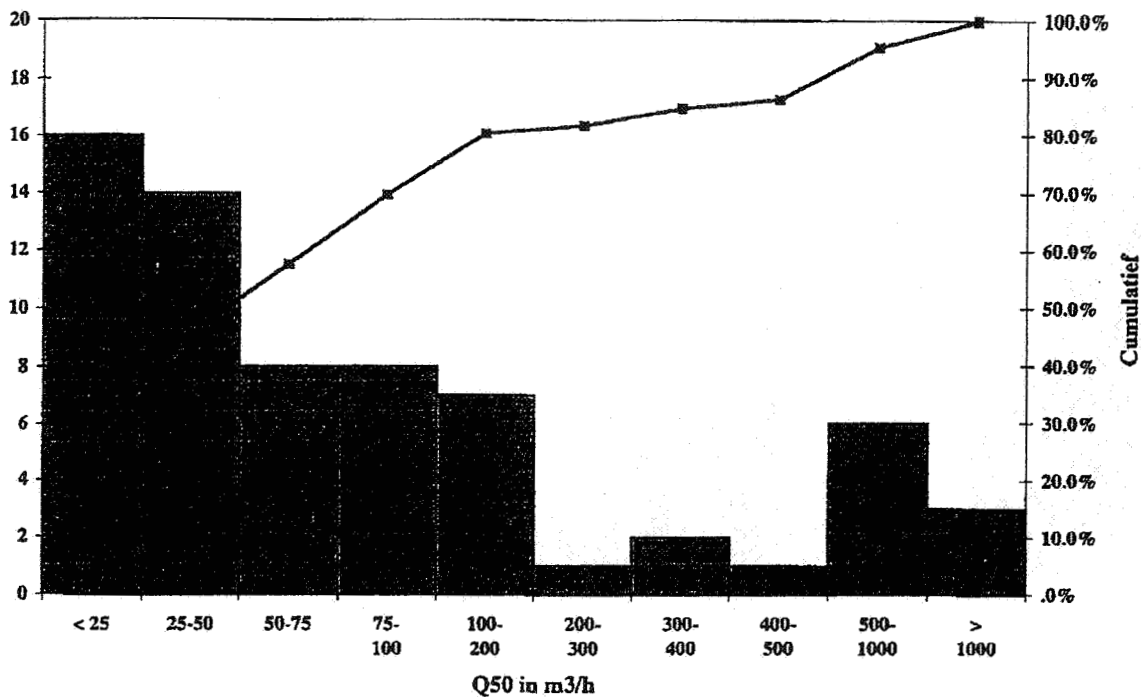
figure 3 : measured air flow rates (Q$_{50}$) in the bedrooms

One can observe that 70% of the bedrooms have a Q50-value which is lower than 100 m³/h. On the other hand, 15% of the bedrooms have a Q50-value which is higher than 500 m³/h. Most of the latter group are bedrooms situated below inclined roofs.

As far as the very airtight bedrooms is concerned, it is clear that the air infiltration through the leakages is insufficient for allowing a good IAQ.

Also the bathrooms are often very airtight. This is illustrated in figure 4.
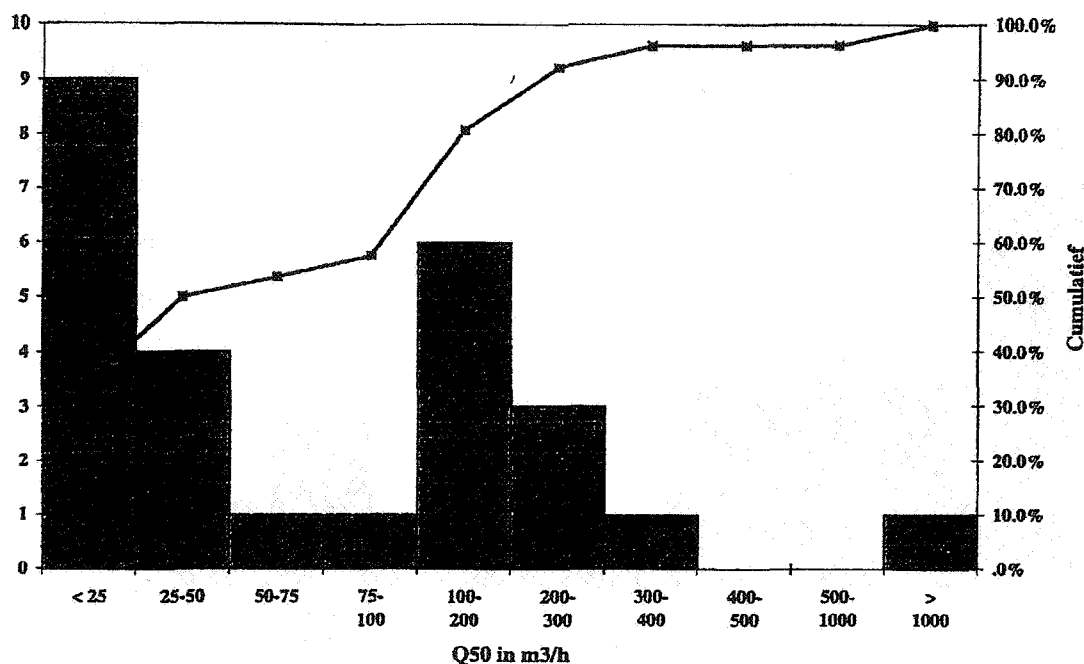
figure 4 : measured Q_{50}-values of bathrooms (vertical axis : number of bathrooms)

3. Visual estimation of the airtightness

One of the objectives of the SENVIVV study in relation to building airtightness is evaluating to what extent the airtightness can be estimated by visual means.

3.1 Approach

During the inspection visit, all visible leakages are registered. For each identified leakage, a leakage rate Q_{50} is associated. For this, a standard list is used. The following typical leakages are considered :

1. walls, floors and ceilings
2. connections between walls/floors/ceilings
3. joints in frames of windows and doors
4. connections between windows and walls
5. large openings and other leakages

The beginning of the list is given in table 1.

Walls in brickwork, concrete blocks,... : plastered	1
Brickwork not plastered nor painted	3
Concrete blocks not plastered nor painted	6
Gypsum board (joints plastered)	1
Gypsum board (joints not plastered)	3/10/(20)
Sandwich panels, joint finished	1.5
Sandwich panels, open joint	3/10/(20)
.....	...

table 1 : Part of list used for visual estimation of leakages

3.2 Results

The results of the comparison between the visual estimation and the measured values is given in figure 5 : comparison between measured and estimated Q50-values.
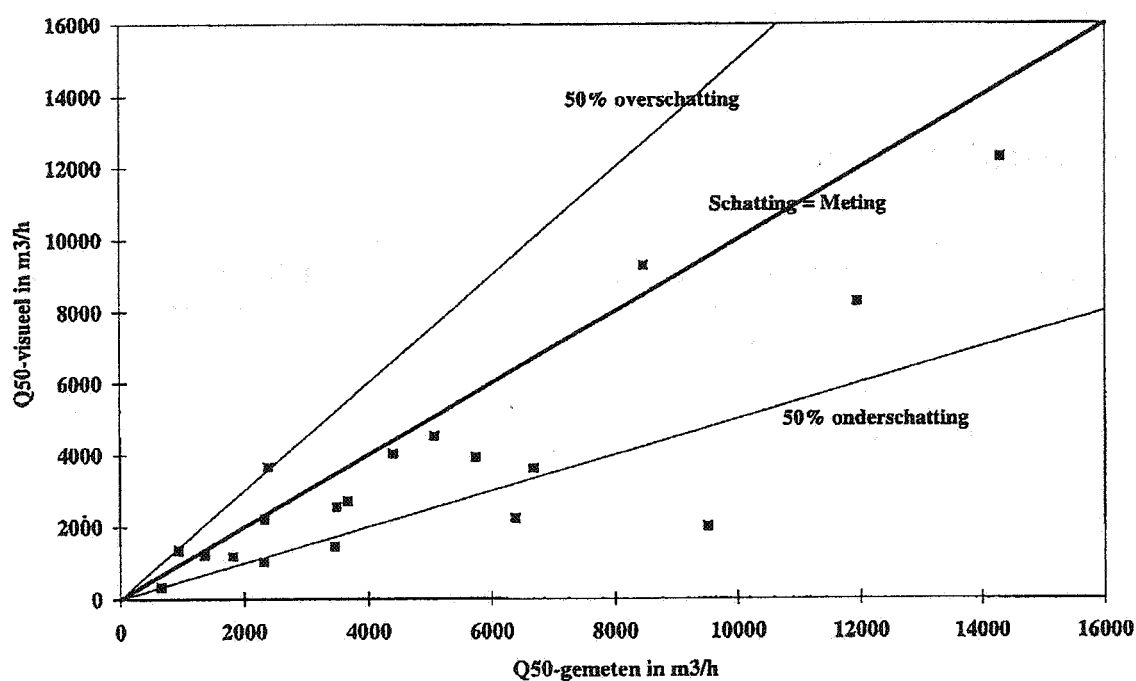
figure 5 : comparison between measured and estimated Q50-values

(horizontal axis: measured   vertical axis : estimated)

The following observations can be made :

- The estimated Q50-value by the visual method is rarely above the measured value (only 3 times), in most cases is there an underestimation.
- In only 1 case is the estimation more than 50% above the measured value;
- In 4 cases is the estimation less than 50% of the measured value.
- The results seem to indicate that the visual method allows a quite realistic estimation of the minimum leakage level.

4. Conclusions

1. The airtightness of recent Belgian dwellings is not at all very good. One explanation is the fact that the owners of many dwellings take care of the finishing of certain parts of the dwelling.
2. For the Belgian context, the visual estimation of the leakage rate seems to allow a first order estimation of the minimum leakage rate of the dwelling.
3. Even in new dwellings constructed after 1990, the presence of purpose provided ventilation supply and exhaust devices is seldom. This is mainly due to the fact that there is not yet a legislation in the Flemish Region imposing such provisions.

5. Acknowledgements

VENTILATION AND COOLING

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Controlling Ventilation and Space Depressurization in Restaurants in Hot and Humid Climates

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Synopsis

Testing was performed in 9 restaurants to identify uncontrolled air flows and pressure imbalances, building and duct system airtightness, building air barrier location, pressure differentials, building air flow balance, and ventilation rates. All restaurants are depressurized under normal operating conditions, ranging from -1.0 to -43 pascals. Space depressurization is a function of exhaust fan flow rates, missing or undersized make-up air, intermittent outdoor air caused by the cycling of air handlers, dirty outdoor air and make-up air filters, and building airtightness. Ventilation rates were found to be high, generally exceeding ASHRAE 62-1989 minimum recommended levels. Pressure imbalances and excessive ventilation rates impact energy use, heating/cooling system sizing, indoor comfort and humidity, building moisture damage, mold growth, combustion equipment problems, and indoor air quality.

The objectives of good restaurant air flow management (in hot and humid climates) are to: 1) achieve positive pressure in the building under a majority of operating conditions, 2) avoid excessive ventilation, and 3) maintain air flow from dining area to kitchen, all while minimizing heating/cooling energy use and achieving acceptable dehumidification (<60% RH most of the time). Recommendations are presented to achieve these objectives.

Symbols and definitions

ACH50 - a measure of building airtightness, expressed as air changes per hour when the building is depressurized to 50 pascals (with respect to outdoors) by a calibrated fan.

air handler (AH) - in a forced air heating or cooling system, the cabinet which contains the distribution blower and may contain heat exchangers and filters.

air distribution system (ADS) - includes all building elements (ducts, plenums, cavities of the building structure, and mechanical closets) through which air is transferred between the conditioned space and the space conditioning equipment.

backdrafting - a condition where flow in a combustion vent pipe is reversed so that air moves down the vent and into the building. Combustion gases therefore discharge into the space.

l/s @ 50 - a measure of building airtightness, expressed as air flow rate (l/s) through leaks in the *building envelope* when the building is depressurized to 50 pascals by a calibrated fan.

exhaust air (EA) - air drawn from a building, typically to remove unwanted elements such as heat, humidity, odors, and combustion fumes.

HAC - heating and air conditioning

HVAC - heating, ventilating, and air conditioning.

infiltration - air flow across the building envelope, from outdoors to indoor, that is largely unintended. It may be driven by wind, temperature difference, duct leakage, return air imbalance, exhaust fans, clothes dryers, etc. Infiltration may act as a source of ventilation.

make-up air (MA) - air pushed into a building to replace EA drawn from the building, is typically unconditioned, delivered in proximity to the EA intake, and is normally operated simultaneously with the EA (both EA and MA controlled by the same on/off switch).

make-up air capture rate - fraction of MA captured by EA before mixing into the building air volume. If 90% of MA is directly captured by the EA, then the MA capture rate is 0.90.

outdoor air (OA) - air moved from outdoors to the return side of the ADS, by the suction of the ADS blower or by a dedicated blower, to provide ventilation to the conditioned space.

uncontrolled air flow - air moving across the building envelope or between zones or compartments of a building, where the pathways of flow, the direction of flow, and the origin of the air are unknown, unspecified, or unintended.

ventilation- the intentional transport of air from outdoors to indoors or the transport of indoor air to outdoors, generally to remove or dilute pollutants and improve indoor air quality.

wrt - = "with respect to". Pressures are expressed as pressure in one location with respect to another location (e.g. "restaurant pressure was -8.4 pascals wrt outdoors").

1. Introduction

Restaurants generally have large exhaust fans, and therefore have the potential to experience space depressurization and high ventilation rates. In hot and humid climates, this combination of space depressurization and a high ventilation rate can produce undesirable consequences related to energy waste, moisture problems, combustion safety, and poor indoor air quality. To avert these problems, MA may be added to offset EA. When properly applied, **MA** can greatly reduce space depressurization and ventilation rates. When properly controlled, **OA** can produce positive pressure while achieving acceptable thermal and humidity conditions.

Space depressurization is a function of building airtightness and net exhaust air (EA_{net} = EA - MA - OA; calculated based on the absolute value of air flows). By use of a chart, building pressure may be determined if airtightness and EA_{net} are known (Figure 1; based on chart by John Tooley and Neil Moyer of Natural Florida Retrofit, Inc.). This figure assumes n=0.65 for the equation $Q = C(dP)^n$, where Q is EA_{net} (l/s), C is an air flow coefficient, dP is the indoor pressure wrt outdoors (pascals), and n is an air flow exponent. To the extent that n deviates from 0.65, the predicted pressures will be in error. **Example;** building airtightness is 2000 l/s @ 50 and EA_{net} is 1000 l/s. From Figure 1, find the intersection of 2000 l/s @ 50 and EA_{net} of 1000 l/s. The indicated pressure is -17.5 pascals. From the same figure, one can predict approximate building airtightness knowing EA_{net} and building pressure. **Example;** EA_{net} is 3500 l/s and building pressure is -10 pascals. From Figure 1, find the intersection of EA_{net} of 3500 and building pressure of 10 pascals. Indicated building airtightness is 10,000 l/s @ 50.

Building ventilation is a function of exhaust air, make-up air capture rate, and outdoor air, as defined by the following formulas.

Formula 1)   If OA > (EA-MA), then ventilation (l/s) = OA plus non-captured MA (MA that escapes into the room and finds its way into the ADS).

Formula 2)   If OA ≤ (EA-MA), then ventilation (l/s) = (EA-MA) plus non-captured MA.

Air Flow vs Airtightness Pressure Chart

(diagonal lines are pressure in pascals; chart assumes n = 0.65)
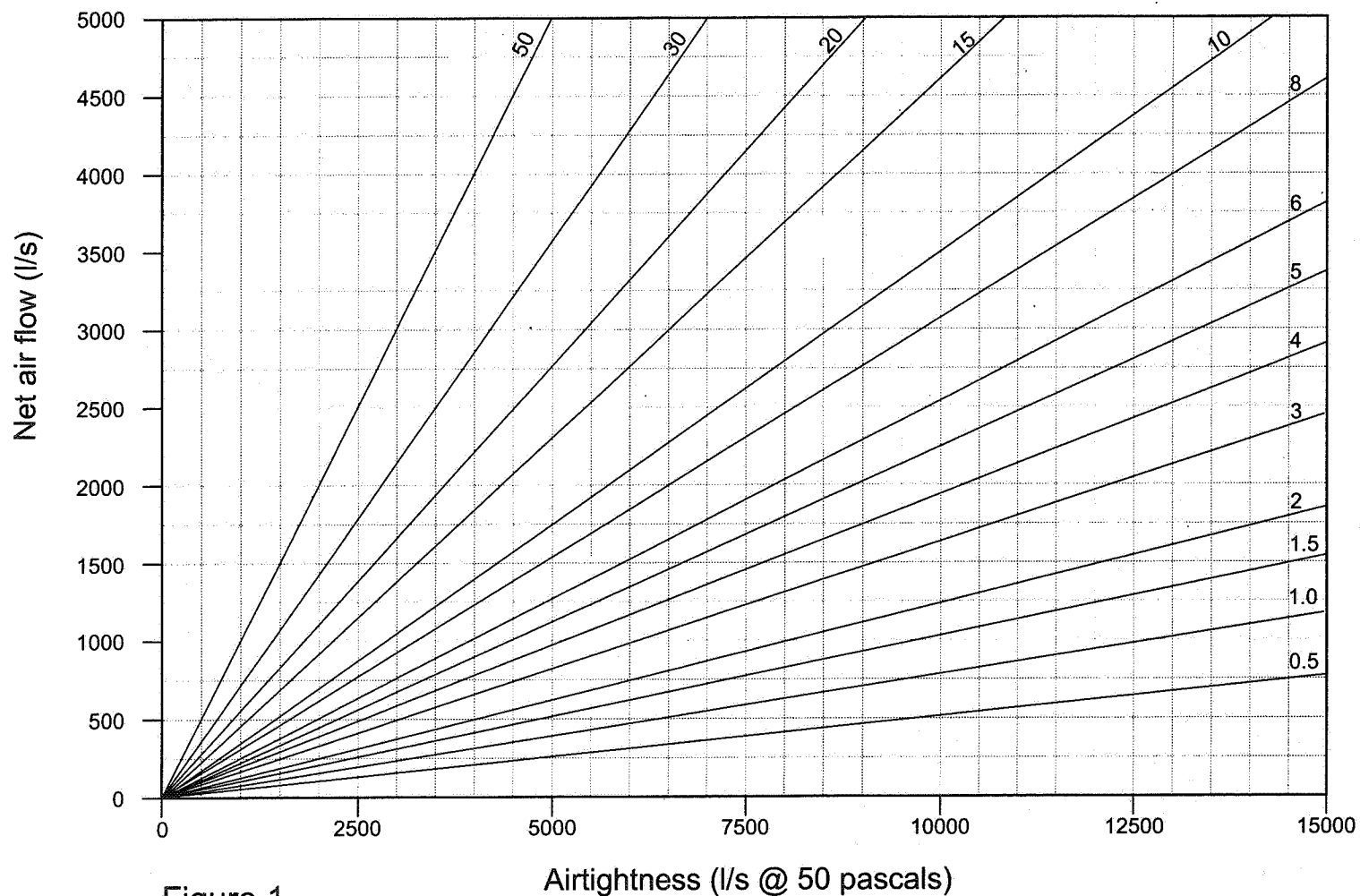
Figure 1

Two forms of uncontrolled air flow may also contribute to the building ventilation rate; duct leakage and unbalanced return air. Duct leakage, when it occurs outside the building air boundary, increases infiltration. *If return duct leakage*, then that leakage behaves like OA. To determine the effect of return leaks (drawing air from outside the building air boundary) on the building ventilation rate, add the return leak amount to OA and use formula 1 or 2. *If supply duct leakage*, then that leakage behaves like EA. To determine the effect of supply leaks (air spilling to outside the building air boundary) on the building ventilation rate, add the supply leak amount to EA and use formula 1 or 2. (Note that duct leaks that do not move air across the building air boundary do not influence the building ventilation rate.)

Unbalanced return air occurs when air flow from the conditioned space to the AH is restricted. This can be caused by closed interior doors when the return is located in a central zone, in which case the closed rooms go to a more positive pressure and the central zone goes to a more negative pressure. Positive pressure in the closed rooms pushes air out of the building and negative pressure in the central zone draws air into the building. Unbalanced return air can also occur when the ceiling space is used as a return plenum, fire walls subdivide this plenum, and the pathways through the fire walls are undersized or missing. Unbalanced return air can increase the building ventilation rate when the return air restriction causes one zone to be at positive pressure (wrt outdoors) and the other to be at negative pressure (wrt outdoors).

2. Research findings

Field testing was performed in 9 in central Florida restaurants (part of a sample of 70 small commercial buildings tested for uncontrolled air flow; Cummings et al., 1996), to characterize building air flow balance, airtightness, and pressures. Test data is presented in Table 1.

Table 1. Airtightness, ventilation, air flow, and pressure differential in 9 Florida restaurants.

restaurant type	floor area (m^2)	l/s @ 50	ACH50	vent rate[1] (ach)	exhaust (l/s)	make-up air (l/s)	outdoor air (l/s)	dP (pa)^2
pizza	180.4	3810	31.2	1.9	1496	0	0	-13
subs	325.4	1021	3.9	5.3	2645	1520	684	-25
bar	223.0	3139	17.5	2.3	466	0	0	-1.8
golf club house	404.2	3977	11.6	1.9	1434	523	0	-6.1
chicken 1	293.7	3302	14.8	11.2	5006	2906	1063	-8
chicken 2	308.5	1741	7.7	5.9	4353	2412	590	-43
Chinese	729.6	4771	8.2	2.2	3066	0	0	-4.6
hotel	1396.6	11002	7.6	1.7	6092	2428	385	-1.0
convenience store	401.3	5544	16.8	1.9	765	0	78	-1.8
average	473.6	4257	13.3	3.8	2814	1088	311	-11.6

[1] ventilation rate under normal building operation measured by tracer gas decay
[2] space pressure wrt outdoors

2.1 Ventilation rates

Ventilation rates averaged 3.8 ach under normal operation, ranging from 1.7 to 11.2 ach. Ventilation averaged 25.5 l/s per person in these 9 restaurants and ranged from 6.1 l/s per person to 82.1 l/s per person. ASHRAE Standard 62-1989 (ASHRAE 1989) calls for 10 l/s per person in restaurant dining areas, 15 l/s per person in bars and cocktail lounges, and 8 l/s per person in kitchen areas. Four of the 9 restaurants have bars or cocktail lounges. In 3 of these buildings, ventilation falls slightly below the recommended minimum during maximum occupancy. In most of the others, ventilation rates are excessive.

2.2 Depressurization caused by lack of make-up air

Four of the 9 restaurants have no MA, and 3 of the 4 have no OA. Consequently, these 4 restaurants experience negative pressure whenever the EA operates.

2.3 Depressurization caused by undersized make-up air

MA was provided in 5 of the 9 restaurants, but on average MA was only 51% of EA and OA was only 17% of EA (with all AHs operating and less if some were not operating). As a result, all 9 restaurants operated at negative pressures which ranged from -1.0 pascal to -43 pascals. The degree of depressurization is a function of the net exhaust air flow rate (EA_{net} = EA - MA - OA) and the airtightness of the building. The greater EA_{net} and the tighter the building envelope, the greater the building depressurization.

In all of these restaurants, the AHs do not run continuously. Rather they cycle on and off depending upon whether the thermostat calls for cooling or heating. Consequently OA is intermittent, turning on and off depending upon whether the AH is operating. Building pressure, therefore, varies as a function of which heating/cooling systems are operating.

In chicken restaurant 1, for example, each of the 4 AHs had OA and operated in on/off control depending upon whether the thermostat called for heating or cooling. When all AHs were operating, building pressure was -3 pascals. When AHs were turned off, incrementally one at time, pressure decreased to -6, -9, -13, and -18 pascals. Three consequences of this space depressurization were observed. First, mold growth was occurring behind vinyl wallpaper on exterior walls in this 9 month old building. The mold growth was occurring as humid outside air was drawn into wall cavities, accumulated on cool gypsum board surfaces inside the walls, and was prevented from drying to indoors by the vinyl wallpaper. Second, the gas water heater was backdrafting when building pressure was -9 pascals or greater causing combustion gases to spill into the occupied space. Third, on two occasions, the store owner reported water heater "flame rollout". These events occurred at closing time when the thermostats were changed to higher settings to reduce cooling energy use overnight. Flame roll-out occurs under the following conditions: room pressure is -18 pascals, the water heater is off, the vent pipe is backdrafting, air flow down the vent pipe pushes into the top of the flue (the flue is the vent stack inside the water heater), and then the water heater turns on. Since air is pushing down the flue, gas is pushed out of the burn chamber and combustion occurs partly outside the bottom portion of the water heater. On these two occasions, smoke poured from the water heater closet and staff had to extinguish the fire.

In chicken restaurant 2, two of the 4 AHs had OA. With both AHs operating, building pressure was -43 pascals. With both AHs off, building pressure was -63 pascals. In spite of the strong depressurization, combustion safety and mold problems were not identified. Backdrafting and flame roll-out from the water heater did not occur because it was located in an exterior closet, which was well connected to outdoors and isolated from indoors. When the building was at -63 pascals, the water heater closet was at neutral pressure wrt outdoors. The reason for no mold is uncertain, except that there is considerably less vinyl wallpaper compared to the other chicken restaurant, thereby allowing greater moisture migration through the building envelope.

In both of these restaurants, depressurization was exacerbated by the intermittency of OA. The reader may conclude, therefore, that the controls should be changed so that the AHs operate continuously when the EA operates. This would allow the OA to operate as a reliable form of MA. The problem with this strategy, however, is that continuous AH operation causes poor dehumidification performance, particularly with fixed-capacity cooling systems. This poor performance results from evaporation of moisture from the coil and drain pan during the periods when the compressor turns off (Khattar et al., 1987; Henderson et al., 1992). Depending upon the length of the compressor on-cycle, moisture collecting on the cooling coil may not have time to exit the pan before the compressor turns off and evaporation begins. Indoor relative humidity levels often increase by 10 percentage points or more as a result of continuous blower operation. In addition, moisture-laden outdoor air is supplied to the space when the compressors cycle off but the AHs continue to operate, further increasing indoor relative humidity during hot and humid weather conditions.

2.4 Depressurization due to dirty outside air and make-up air filters

In the sub sandwich restaurant, building air flow imbalance occurred as a result of dirty filters. These filters had not been cleaned or replaced in the 8 months since the building was constructed. At start-up, the HVAC contractor had demonstrated that the building was operating at positive pressure by showing smoke flowing out through a window. However, in the intervening months, building pressure had become progressively more negative (the authors surmise) as the filters became progressively more dirty. The building was operating at -25 pascals at the time it was tested. When the filters were cleaned or replaced, building pressure rose to -2 pascals. Two major consequences of depressurization were observed. First, the pilot light of the instantaneous gas water heater was repeatedly blown out by severe backdrafting in the water heater vent pipe resulting in unavailability of hot water. Restaurant staff had identified that opening an exterior door would allow lighting the pilot light and operation of the heater long enough to do washing. Second, sewer gas was entering the men's bathroom because of the -25 pascal pressure and a toilet that was improperly sealed to the sewer line.

In the convenience store, 765 l/s kitchen EA operated with no MA and limited OA. Intakes grills for the OA were located under an exterior eave unknown to store staff, and behind the grills were OA filters which had not been cleaned in years. The filters were clogged so total OA was only 77.9 l/s for combined cooling capacity of 61.5 kW (17.5 tons).

3. Solutions

There are several objectives of good restaurant air flow management; 1) achieve positive pressure in the building under a majority of operating conditions, 2) maintain flow of air from the dining area to the food preparation area, 3) avoid excessive ventilation, 4) minimize heating and cooling energy use, and 5) control the cooling system so that it achieves acceptable dehumidification (<60% RH most of the time). Achieving all of these objectives simultaneously is complicated. Following is one set of strategies to meet these objectives.

Positive pressure can be achieved by providing combined MA and OA greater than EA. As a first step, EA should be as small as possible while still meeting exhaust and ventilation requirements. As a second step, MA should be sized as large as possible, but normally not sized greater than 80% of EA. MA is sized at 80% or less of EA so that the food preparation area will be depressurized wrt to the dining area, air will flow from the dining area to the kitchen, and a large majority of the MA will be captured by the EA. To complete the building air flow balance, OA equal to 21% to 25% of EA should be provided to the dining area, thus enhancing air flow toward the kitchen.

Excessive ventilation is avoided by providing MA to a location proximate to both the cooking appliances and EA intake so that a large proportion of the MA is captured by the EA. Optimal design would indicate that the cooking appliances be located between the MA discharge and EA intake. MA discharge should be located as close as possible to the EA intake while still permitting capture of the vast majority of the heat, humidity, odors, grease, and combustion fumes associated with cooking, but not discharging onto restaurant staff or affecting pilot lights or burners. In hot and humid climates, care must be taken to avoid condensation of moisture contained in MA on interior surfaces.

HAC loads are minimized by minimizing ventilation (no more ventilation than is necessary), not conditioning MA, and optimizing design so that a maximum proportion of MA is captured by EA. [Note that modulation of EA flow in response to cooking intensity by means of fan speed controllers in conjunction with temperature and smoke density sensors is also an option.]

Finally, the cooling system control must meet two performance criteria: 1) provide OA to the space when the EA is operating and 2) control indoor temperature and relative humidity within acceptable limits. It is possible to have the AH blowers operate continuously whenever the EA is operating. In fact, it is not uncommon for commercial building AHs to operate continuously. However, this approach can seriously compromise the dehumidification performance of the cooling system as a result of evaporation of moisture from the coil and drain pan (see Section 2.3). One solution is to separate (to a large extent) space conditioning from ventilation, as follows.

Designate one heating/cooling system to treat OA. Its task is to provide conditioned OA to the dining area, and it will be controlled to operate simultaneously with EA and MA. While the AH blower will operate continuously in step with EA, the cooling compressor will cycle according to the requirements of the dining room thermostat. In order for the compressor to operate a large fraction of the time so that the OA will be cooled and dehumidified during hot and humid summer months, this designated OA unit can be controlled on the first stage of the thermostat. For example, if the thermostat setting for the dining area is 23.3°C, then the first

stage operation could control the compressor for the designated OA unit to operate at say 22.2°C. On most warm and hot days of the year, the compressor for this OA unit would then operate most of the time, thereby providing good dehumidification. The other cooling systems serving the dining area would not have OA and they would be controlled on the second stage at a 23.3°C setting. [Note that this control is best achieved by use of a two-stage thermostat. Use of individual thermostats with each set at different temperatures may be ineffective because the thermostat settings may differ or change over time, causing the compressor of the designated OA system to not operate a large proportion of the time and therefore not effectively dehumidify the OA.]

4. Wrap up

Field research has identified problems in restaurants associated with air flow balance, space depressurization, and HVAC system control. These problems can be characterized as primary or secondary consequences. Primary consequences are space depressurization and excessive ventilation rates. These primary consequences, in turn, cause secondary consequences including moisture accumulation in building cavities, combustion safety problems associated with backdrafting and flame roll-out, drawing sewer gases into the building, high relative humidity, moisture damage to building materials, and mold/mildew growth.

A strategy has been presented for avoiding space depressurization and excessive ventilation. This involves providing unconditioned MA in proximity to the EA at about 80% of the EA flow rate and providing OA equal to about 21% to 25% of EA into the dining space, thus creating positive pressure in the building and enhancing air flow from dining area to kitchen. OA is provided by means of a dedicated OA heating/cooling system. The AH blower for this unit would operate continuously during EA operation. The compressor or heating source for this unit would cycle in response to the first stage of the thermostat so that the system would effectively dehumidify the OA during most hours of the year.

5. Acknowledgments

6. References

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Example

→ heel wat opties die
nt direct zijn;
↓
* low airtightness =?
*

manual?
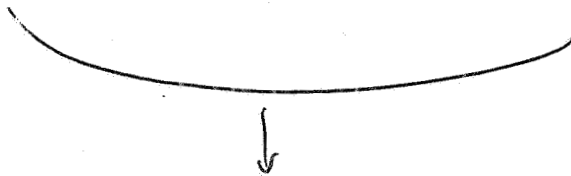
Duidelijk
— eenvoudije design &
education tool —

VRAGEN

→ spring / autumn; Results
↳ you can see in
year- around results

[Handwritten annotations: "Spring 1988" (circled), "thermal cap.?", "hi?" (circled), "Summer → 1 week"]

VENTILATION AND COOLING

18TH ANNUAL AIVC CONFERENCE
ATHENS, GREECE, 23-26 SEPTEMBER, 1997

[Handwritten: "ONE ZONE MODEL → eenvoudige input", "thermisch model + ventilatie model"]

A Design Tool for Natural Ventilation

[Handwritten: "output → Summer, Winter, TRY" ; "output → Max Temper. Ventilation rates, year-round perform. → indoor temp. → ventilat. rates"]

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A DESIGN TOOL FOR NATURAL VENTILATION

Synopsis

A difficulty when designing natural ventilation in office buildings is the lack of simple design tools.

In order to be able to predict natural ventilation air flow rates and indoor air temperatures at the design stage, a computer model has been developed within the EU-JOULE project $NatVent^{TM}$. The program is an integrated model with a thermal and an air flow model coupled together. It can be used early in the design process to determine possibilities and restrictions in the use of natural ventilation in an office building.

The most important objectives while developing the program have been to create a robust underlying theoretical model and an easy-to-use interface. Set in the Windows environment, the required input data are easily overviewed at all times. A key issue has been to use indata which are easy to quantify, even at an early stage in the design process.

The paper discusses briefly the theoretical model as well as the $NatVent^{TM}$ computer program.

The program will be subject to extensive user tests during the autumn of 1997 and will be released in the spring of 1998.

1 Introduction

In many countries there is a turn towards natural ventilation as an alternative to sometimes energy and cost demanding mechanical ventilation systems. The objective is to save money and energy while maintaining an acceptable indoor air quality and thermal climate, or even to improve the indoor environment by reducing noise levels and giving the user more control over the indoor climate etc.

The aim of the EC-JOULE project $NatVent^{TM}$ is to overcome technical barriers to low-energy natural ventilation in office-type buildings in moderate and cold climates. To identify the perceived barriers, a number of structured interviews among leading designers and decision makers, in each of the seven participating countries, have been made (Aggerholm, 1997). Many of the interviewees have given the development of a simple design tool, such as an easy-to-use computer program, as a key issue.

In order to meet this need and to be able to predict natural ventilation air flow rates and indoor air temperatures at the design stage of the process, this computer model was developed within the $NatVent^{TM}$ project. The program is an integrated model with a thermal and an air flow model coupled together.

The design tool was developed at J&W Consulting Engineers in Sweden in co-operation with the Danish Building Research Institute, SBI.

2 The *NatVent*^{TM} - program

The most important objectives while developing the program have been to create a robust underlying theoretical model and an easy-to-use interface.

2.1 The User Interface

The *NatVent*^{TM} program is set in the typical Windows environment. As a platform, a main window is created. Within this main window, input and output forms may be opened. The aim for the user interface is to facilitate the use of the program by any building designer, architect or engineer at an early stage. Therefore the interface uses input that are simple to quantify, even at an early stage in the design process.

The input is given by the user step-by-step in four forms describing: the Location, the Building, the Ventilation Strategy and the Windows. Under these four headings the relevant input is listed. The input forms are found in Appendix; an example is given in Figure 3.1.

Figure 3.1: One of the input forms.

3 The Single-zone model

The program uses a single zone model. Thus the entire building or a selected part, is represented by only one single zone. The selected part can be either one of the floors or a part of a single floor. The single zone has one temperature and one internal pressure. The zone is effected in many ways by the weather, the occupants and maybe by a mechanical ventilation system. To visually illustrate these factors Figure 4.1 shows a picture of the thermal paths and the air flow paths that create the temperature and ventilation system in the zone. When the system of equations is established, a tool for solving the system is needed.

Figure 4.1: A single zone model with air flow links and paths for heat flow

In order to reach the objective of creating a robust model, a stable iteration process is needed, coupling the air flow model and the thermal model. Below those two models are described.

3.1 Air Flow model

3.1.1 Pressure Distribution

Due to wind, thermal buoyancy and fans, if any, a pressure difference over the building envelope will be created. As a pressure difference occurs over the building envelope, the air is bound to flow from higher pressure to lower pressure and thus air flow to and from the building will arise.

3.1.1.1 Wind Pressure

The wind creates a pressure field around the building. The shape of this pressure field is determined by the surroundings, the shape of the building and of cause by the wind velocity and direction.

The wind velocity varies with height and roughness of the surroundings. Wind velocities given in the meteorological input should be the wind velocity at a height of ten meters in open surroundings. In order to use a value of the wind velocity that is appropriate for the specific conditions, the velocity is recalculated to the velocity at the top of the building with adjustments for the shielding conditions. The program uses the empirical relation:

$$u_{z,wind} = u_{m,wind} \times k_w \times z^{a_w} \qquad \text{[m/s]}$$

where: $u_{z,wind}$ wind velocity at the height z [m/s]

        $u_{m,wind}$ measured wind velocity at 10 meters height [m/s]

        k_w, a_w constants dependant on terrain [-]

The wind pressure on the building envelope is determined with the part of the Bernoulli equation describing the dynamic (velocity) pressure:

$$p_{wind} = C_p \times \frac{\rho_{air} \times u_0^2}{2} \qquad \text{[Pa]}$$

where: p_{wind} wind induced pressure [Pa]

        C_p pressure coefficient [-]

        ρ_{air} density of the air [kg/m^3]

        u_0 wind velocity in unrestricted air flow [m/s]

The pressure coefficient determines to which extent the wind pressure is present on the particular façade element. There are great difficulties in finding appropriate pressure coefficients as they need full scale measurements, wind tunnel experiments or extensive 3-d modelling. In a pre-design tool as this, that type of advanced input would be superfluous. Instead the C_p-values used are average values of the pressure coefficients for each façade of the building (Liddament 1996).

3.1.1.2 Thermal Buoyancy

Thermal buoyancy, or stack pressure, is caused by the difference in density between warm and cold air. The air pressure on a certain level is the pressure of the air pillar above this level. At a constant temperature the pressure declines linearly with the height. As warm air is lighter than cold, warm air gives a lighter pressure on the same height. This pressure difference between air of different temperatures at a certain height is described as:

$$\Delta p_{therm} = (\rho_{cold} - \rho_{warm}) \times g \times h \qquad \text{[Pa]}$$

where: Δp_{therm} pressure difference due to thermal buoyancy [Pa]

           ρ_{cold} air density of the colder air [kg/m^3]

           ρ_{warm} air density of the warmer air [kg/m^3]

           g   gravitation = 9.81 [m/s^2]

           h   height [m]

The density of air is affected by the temperature and the moisture content of the air. The density of air at the temperature zero degrees Celsius and a relative humidity of 50 % is 1.291 kg/m^3. The density at other temperatures is be derived from this.

The effect on air density of the moisture content in the air is quite small especially for the moderate temperature interval (about -20°C - +40°C) the program is dealing with. Therefore the effect of different relative humidities is neglected in the program.

3.1.1.3 Fans

As the program is dealing primarily with naturally ventilated buildings, fans are no big issue. Fans can only be simulated as a constant air flow rate. If the fans do not run continuously a schedule describing the air flow rate at different hours can be added.

3.1.2 Air Flows

3.1.2.1 General Air Flow Theory

Air flow through the building envelope can have many paths. Air flow through walls and ceiling, through small cracks and imperfections, through vents in the façade, through window airing, through ducts for supply air or passive stacks and forced flow through fans - if any. In order to get a realistic model of the building it is of great importance that the paths are described in a realistic way.

Air flow through cracks, small openings etc. is generally described with the equation (Kronvall, 1980):

$$\Delta p_{crack} = \left(\lambda_{fric} \times \frac{l_{crack}}{d_h} + \sum_{i=1}^{n} \xi_i \right) \times \frac{\rho_{air} \times u_m^2}{2} \qquad \text{[Pa]}$$

where: Δp_{crack} the pressure drop over the crack [Pa]

           λ_{fric} friction coefficient [-]

           l_{crack} length in flow direction [m]

           d_h hydraulic diameter [m]

           ξ_i loss factor, for contraction, expansion or bend losses [-]

           ρ_{air} density of the air [kg/m^3]

           u_m average air velocity [m/s]

It is possible to simplify the equation above and to summarise all pressure drops along the flow path. The air flow rate is presented as the equation:

$$q_v = a \times \Delta p^b \qquad \text{[m}^3\text{/s]}$$

where: q_v volumetric air flow rate [m^3/s]

           a   flow coefficient [m^3 / (s*Pa^b)]

           b   flow exponent [-]

3.1.2.2 Air Flow through the Building Envelope

The air leakage is specified, by the user, by choosing a low, medium or high air tightness. The program suggest an area related air leakage $(l/s/m^2)$, which is typical for the specific country. The flow coefficient is calculated from the air leakage data available with:

$$a = \rho_{crack} \times \frac{q_{leak} \times A_{build}}{1000 \times 50^{0,67}} \qquad [kg/ (s*Pa^b)]$$

where: ρ_{crack} mean of outdoor and indoor densities $[kg/m^3]$

q_{leak} air leakage expressed as leakage per square meter and second $[l/s/m^2]$

A_{build} envelope area $[m^2]$

The air leakage is distributed evenly over the walls and the ceiling. The floor is assumed to be an airtight construction. A flow exponent of 0.67 is empirically chosen, since many measurements on building envelopes show that this is a good estimation.

3.1.2.3 Air Flow through Vents

The pressure drop through a sharp edged hole in a thin wall can be described as (Dick's equation):

$$q_m = \rho_{crack} \times A_{hole} \times C_d \times \sqrt{\frac{2 \times \Delta p_{crack}}{\rho_{crack}}} \qquad [kg/s]$$

where: q_m mass air flow rate through the crack $[kg/s]$

A_{hole} equivalent area of the hole $[m^2]$

C_d coefficient of discharge ≈ 0.6 [-]

ρ_{crack} mean value air density of the internal and the external air

The equation is assumed to be applicable to the vents in the building. For the vents an equivalent area is given. These equivalent areas can normally be found in the manufacturers' specifications. The vents are by default placed at a height of two meters above the floor.

3.1.2.4 Air Flow through Windows and Skylight

The same principle as for the vents is applicable also for the windows. The difference is, as the window can be opened or closed, a schedule is needed. As a default the openable windows are open (ajar) during working hours and closed during the night. To describe a window ajar, it is assumed that 10 % of the open window area is actually open.

A large opening such as a window may have air flow that differs in direction top to bottom. To enable the option of a two way flow through the window, it is simulated as two links, one bottom half and one top half. The height of these links are on ¼ respectively ¾ of the height of the window.

3.1.2.5 Air Flow through Ducts

There are two kind of ducts: supply air ducts and passive stacks. They both include air flow through ducts and they are treated in a similar way. In the ducts the friction losses are not neglectable, but a considerable part of the total pressure drop.

$$\Delta p_{duct} = \Delta p_{sharpedge} + \Delta p_{friction} \qquad [Pa]$$

where: Δp_{duct} total pressure loss for the duct [Pa]

$\Delta p_{sharpedge}$ pressure loss for a sharp edged hole, calculated as before [Pa]

$\Delta p_{friction}$ pressure loss due to friction [Pa]

In order to use a realistic pressure drop over the ducts the friction part is studied. When calculating the friction losses the air velocity of the previous time step is used. The friction losses are calculated separately for each duct/stack.

3.1.3 System of Equations

The criteria for solving the infiltration part of the equation system is that at all times there should be mass balance in the zone. The criteria for mass balance in the calculation is set to a maximum difference for Σq_m of 0.0001 kg/s. This can be illustrated with:

$$\sum_{link=1}^{n} q_{m,link}(t) \leq 0.0001 \qquad [kg/s]$$

where: $q_{m,link}(t)$ mass flow for a certain link at time = t [kg/s]

The mass flow for each different link is calculated, as established earlier, with a non-linear equation written on the form:

$$q_{m,link} = a_{link} \times \Delta p_{link}^{b,link} \qquad [kg/s]$$

where: $\Delta p_{link} = p_0 - p_{wind} + \Delta p_{therm}$ [Pa]

where: p_0 unknown internal pressure at ground level [Pa]

The solution of the system must be found iteratively, where the internal pressure is the unknown parameter that are to be found. This is done by using the Newton-Raphson method. By setting a starting value of the p_0, new approximations are made with:

$$p_{0,n+1} = p_{0,n} - \frac{f(p_{0,n})}{f'(p_{0,n})} \qquad [Pa]$$

where: $f(p_0) = \sum_{link=1}^{n} \left[a \times \Delta p_{link}^{b} \right]$

For each time step the calculated internal pressure for the previous time step is used as a start value. As for the first time step the internal pressure = -1 Pa is used.

3.2 Weather data

Three types of weather data files are used in the program:

- Summer design weather data for estimation of maximum room temperatures in hot summer design periods.

- Winter design weather data for estimation of minimum ventilation rates in calm temperate winter design periods.

- Actual weather data measured or from test or design reference years. These weather data could be used either for design purpose (duration graph) or for estimating the average ventilation rates or room temperatures for a period.

The weather data files includes hourly values of: external air temperature, direct beam solar radiation, diffuse solar radiation on the horizontal, wind speed and wind direction.

The design weather data files can be generated by the program assuming period stationary weather with the same weather conditions each day in the period. The external air temperature is described by a sinusoidal curve where the average temperature, the daily maximum temperature and the peak hour are specified by the user of the program. The direct beam solar radiation and the diffuse solar radiation on the horizontal are calculated for clear sky conditions from the solar constant dependent on the latitude of the location, the time of the day and year and the turbidity of the atmosphere. The solar radiation data are only necessary for summer design. The wind speed and direction are assumed to be constant for the period and are specified by the user.

3.3 Solar radiation model

The solar radiation on external walls and the roof and the insolation through windows and skylight is calculated either from actual weather data e.g. in a test or design reference year or from summer design weather data generated for a stationary hot, calm, clear sky period. Solar radiation are not calculated in case of winter design of the ventilation system.

The solar radiation is calculated from the direct beam solar radiation and from the diffuse solar radiation on the horizontal, included in the weather data file. In calculating the diffuse radiation on the building clear sky conditions are assumed to simplify the calculation and avoiding the need for shadow conditions data. The solar radiation on the building also includes radiation reflected from the ground assuming a reflectance factor of 0.2. The direct beam solar radiation is neglected if the solar altitude is less than the horizon angle using $10°$ horizon angle in case of surrounding buildings and trees with half the height of the actual building and $25°$ in case of surrounding buildings and trees with the same height as the actual building.

The calculation of insolation through the windows and the skylight takes into account the type of glazing, the shading from overhangs and the effect of solar shading e.g. curtains, venetian blinds or protective glazing as specified by the user of the program.

3.4 Thermal model

The thermal model is a one zone, one time constant model. In the model it is assumed that all internal structures and surfaces have the same temperature and that the internal air temperature can be averaged to one air temperature representing the hole building or zone.

The heat exchange between the internal surfaces and the air is calculated assuming a heat transfer coefficient of 3 W/m^2 K and a total surface area of 4 times the floor area. The active heat capacity can be selected in a table by the user from the possibilities: very light, light, heavy and very heavy, or filled in directly by the user in Wh/m^2 K.

The internal heat gains and the insolation is assumed to be supplied half to the surfaces and half to the room air. The heat loss through the windows, through the skylight and by ventilation is assumed to be from the room air. The heat loss through external walls and roof and the transmission of solar radiation through the same constructions are assumed to be coupled to the internal surface temperature. The calculation takes into account the U-value of the constructions and the absorption of solar radiation on the external surfaces.

4 Future Development

During the autumn of 1997 extensive user tests will be performed by another member of the *NatVent*^{TM} consortium. These tests will aim to validate the results from the program and to give parametric studies of different office types. The program will also be available to the other *NatVent*^{TM} members during the autumn, to make evaluation of the user interface possible. The stability of the iteration process will also be evaluated. The final version of the program will be released in the spring of 1998.

5 Acknowledgements

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7 Appendix

NV the NatVent-program

File Edit View Run Options

| the Location | the Building | the Ventilation Strategy | the Windows |

About the ventilation strategy in the building

Vents

Equivalent size of one vent [100 ▼] cm2

Facade 1 [1] Vents per floor

Facade 2 [1] Vents per floor

Facade 3 [1] Vents per floor

Facade 4 [1] Vents per floor

Internal Heat Loads

Heat loads during working hours

[Medium ▼] [25] W/m2

Fans

[X] Exhaust fans [200 ▼] l/s. total

[ ] Supply fans [200 ▼] l/s. total

Ventilation strategy

[X] The building has a passive stack system

Height of the stack outlets [8.2] meters

Total area of the stack [0] m2

[X] The building has ducted air supply

Total length of the duct [25] meters

Total area of the duct [1.6] m2

(•) Fans running continuously

( ) Fans running working hours

[Advanced Alternatives] [Cancel] [Save] [Close] [Run Project]

NV the NatVent-program

File Edit View Run Options

| the Location | the Building | the Ventilation Strategy | the Windows |

About Windows

Windows - size

Determine the total percentage of windows and the percentage of windows open during work hours

	% windows of facade	% of windows openable
Facade 1	[15] %	[0] %
Facade 2	[15] %	[0] %
Facade 3	[15] %	[0] %
Facade 4	[15] %	[0] %
Skylight	[15] %	[0] %

Solar shading

Solar shading [No solar shading (1.0) ▼]

Overhang [No overhang (0 deg) ▼]

Position of openable windows

Give distance from floor to lower window frame and to upper window frame.

	Lower frame	Upper frame
Facade 1	[1] meters	[2] meters
Facade 2	[1] meters	[2] meters
Facade 3	[1] meters	[2] meters
Facade 4	[1] meters	[2] meters

Type of Windows

[double panes ▼]

U-Value [2.7] W/m2/K

Transmittance [0.75]

[Advanced Alternatives] [Cancel] [Save] [Close] [Run Project]

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* <u>Cooling capacity</u> !

{ * deplacement ventie
{ * mixing ventilation
} → totale ≠ effect op cooling

<u>3b</u>/ Stack effect + wind

int. gain source

nr afh vzd int. gains !!
wel van ↑→ openingen
↑→ ...

+
Wind

h ook afh. th~~e~~ ~FRVi~ number
verhouding tussen { stack
{ wind
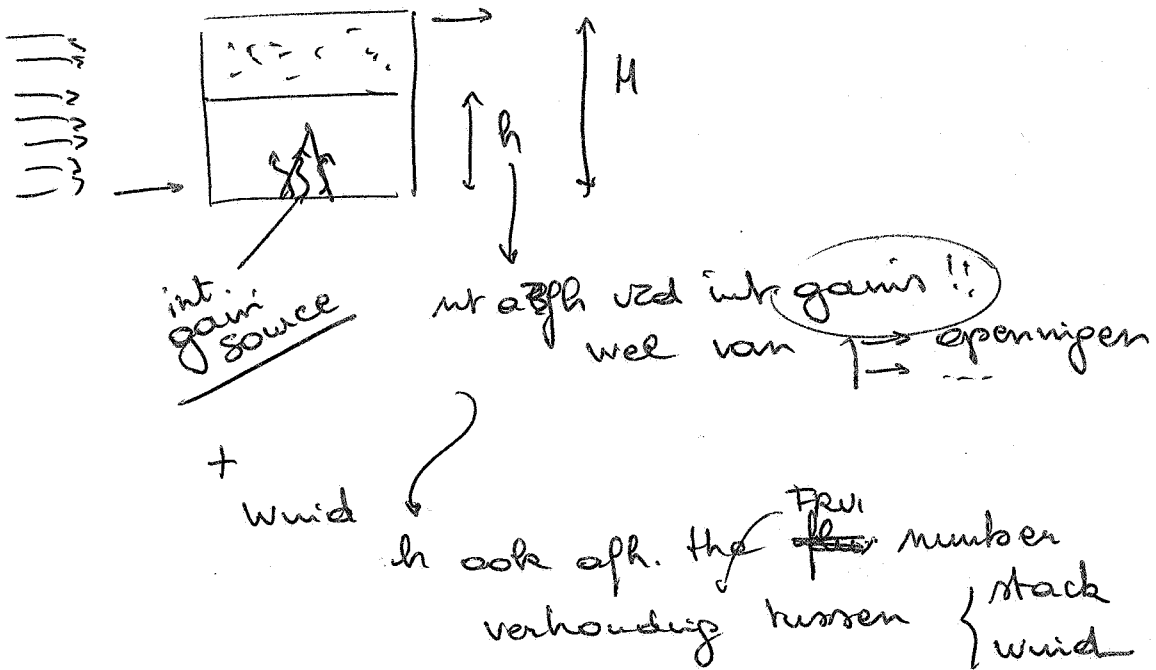
Wind toevoegen
h ↑ (VIDEO)
+ klein ↓ (= T° ↓)
want vert. rate ↑
MAAR stack is belangryk
behoudt stratificatie .

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zeer interessant voor displacement ventilation!

barnonderzoek
nieuwe modellen voor airflows...
↓ daarut
guidelines
rules of thumb

VENTILATION AND COOLING

18TH ANNUAL AIVC CONFERENCE
ATHENS, GREECE, 23-26 SEPTEMBER, 1997

(zeer interess!)

NEW SIMPLE STRATIFICATION MODELS

(Title)

Passive cooling by natural ventilation: Salt bath modelling of combined wind and buoyancy forces

opstelling in labo om model te chequen

Salt = warmte die komt uu warmtebron.

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Waarom water
— omdat in groot gebouw
α wrijving diffusie v. lucht
klein zy
→ voor water ook op die kleine schaal

Wat 's nachts → geen uit gains
(of rock (vloer) die afkoelt)

Passive cooling by natural ventilation:
Salt bath modelling of combined wind and buoyancy forces

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Synopsis

We examine conditions under which the natural forces of wind and buoyancy may be harnessed in order to provide ventilation for cooling. Steady-state, displacement flows driven by combined buoyancy and wind forces are simulated at small scale in the laboratory using a Perspex box to represent a generic room or single-spaced building. Density differences necessary to simulate the stack effect are produced using fresh and salt water solutions. Wind flow is simulated by placing the box in a flume tank; the flume produces a flow of water past the box and this flow is used to represent the wind. By measuring salinity and the position of the stratification within the box, equivalent temperature profiles and ventilation flow rates in naturally ventilated buildings are deduced.

Results of these experiments are compared with the predictions of a theoretical model. It is shown that if ventilation openings are located so the wind assists the stack-driven flow the ventilation may be significantly enhanced and passive cooling achieved. The cooling capacity of the ventilation system is shown to depend upon the relative magnitudes of the wind and buoyancy produced velocities, the area of the openings and the height of the space. It is shown that by harnessing the wind to assist the buoyancy-driven flow it is possible to i) reduce the temperature of the warm upper layer, ii) increase the depth of the lower layer at ambient temperature and iii) increase the ventilation flow rate.

List of symbols

A^*	effective area of openings (m^2)		g	acceleration due to gravity (ms^{-2})
ACH	number of air changes per hour		H	height of the enclosure (m)
a_t, a_b	areas of upper and lower vents (m^2)		h	height of interface above floor (m)
B	strength of source (m^4s^{-3})		Q	total ventilation flow rate (m^3s^{-1})
C_e	coefficient of expansion		T_a	temperature of ambient layer (°C)
C_d	coefficient of discharge		T_u	temperature of warm layer (°C)
C_{pi}	pressure coefficient at inlet		U_{wind}	mean wind speed (ms^{-1})
C_{po}	pressure coefficient at outlet		V	volume of enclosure (m^3)
C	plume entrainment constant		β	coefficient thermal expansion (°C^{-1})
c_p	specific heat capacity ($J kg^{-1}C^{-1}$)		Δ	wind pressure drop ($kg m^{-1}s^{-2}$)
d_c	distance between midpoint of upper opening and ceiling (m)		$\Delta\rho$	density step across interface ($kg m^{-3}$)
E	power of source (W)		ΔT	temperature step across interface $= T_u - T_a$ (°C)
Fr	Froude number		ρ	density of ambient fluid ($kg m^{-3}$)

1. Introduction

The cooling capacity of a ventilation system is determined by the rate of air exchange through a space and the air flow patterns within it. By supplying cool ambient air, either at low levels, in order to displace the warm air through high-level openings, or at high levels so as to mix with and temper the internal air, 'passive' cooling may be achieved. In displacement ventilation warm air is collected in an upper zone and cooling is achieved by flushing the lower zone with ambient air. The rate of flushing is determined by the area of the openings, the temperature and depth of the upper zone, and the driving produced by the wind.

If the system is designed to take advantage of the prevailing driving forces then energy may be saved as the periods in which mechanical cooling is needed may be reduced.

For stack-driven flows to be effective, the upper zone must be maintained at a depth and temperature sufficient to drive a ventilating flow at the required rate. The ventilation rate may be enhanced by increasing the depth and temperature of the upper layer, however, careful design is needed to ensure that occupants are not exposed to the high air temperatures and pollutant levels carried in this layer. If the area of the openings is too small or if their location is not carefully chosen the upper layer can descend to the occupied levels of the building which can have a dramatic effect on indoor air quality. The size and location of the ventilation openings and their effect on the depth and temperature of the upper zone must therefore be considered at the design stage of a naturally ventilated building to ensure the required ventilation will be provided. This task is further complicated as, over the majority of the year, the additional driving produced by the wind must also be taken into account.

In order to develop an understanding of temperature distribution and air movement within naturally ventilated enclosures a laboratory technique has been developed, see for example [1], which accurately reproduces stack-driven ventilation flows in small-scale models of buildings in water tanks. This technique, which is often referred to as the 'salt-bath' technique, has now been extended to include flows driven by the *combined* forces of wind and buoyancy, see for example [2]. In parallel with the development of these laboratory techniques, simple theoretical models have been developed which provide further insight into the parameters controlling these flows.

A theoretical model for natural ventilation flows in an enclosure containing a localised source of heat on the floor of the space is presented by [1] who show that a steady-state displacement flow is established when openings are made at high and low levels. The stratification developed consists of two homogeneous fluid layers, a warm upper layer and a lower layer at ambient temperature, separated by a horizontal interface. A key result of [1] is that the depth h of the layer at ambient temperature is determined by entrainment into the rising thermal plume and can be increased only by increasing the dimensionless area A^*/H^2 of the openings :

$$\frac{A^*}{H^2} = C^{3/2}\left(\frac{(h/H)^5}{1-(h/H)}\right)^{1/2} , \tag{1}$$

where the 'effective' area of the openings A^* is given by

$$A^* = a_t a_b \left(\frac{2C_e C_d}{C_d a_t^2 + C_e a_b^2}\right)^{1/2} , \tag{2}$$

H is the total height of the space, C_e and C_d denote the coefficients of expansion and discharge, respectively, C (≈ 0.14) is a constant dependent upon the entrainment into the plume, and a_t and a_b denote the respective areas of the upper and lower openings. Their results show that increasing the strength B of the source increases the temperature step across the interface but it does not alter the position of the interface. It has been shown by [3] that this latter result holds for any number of buoyant sources.

In this paper, we extend the work of [1] by considering the effect of a flow of wind past an enclosure containing a point source of heat. We focus our attention on steady-state displacement flows driven by buoyancy forces *assisted* by wind and present the results of laboratory experiments. These results are compared with the predictions of a theoretical model.

2. Laboratory Experiments

Laboratory experiments to simulate steady-state air flow patterns and temperature profiles in naturally ventilated buildings were conducted in a Perspex box (29.5 cm long, 25 cm high and 15 cm wide) which was suspended in a flume tank (length 2.65 m x width

0.30 m x depth 0.57 m) filled with fresh water. The box was used to represent a generic building or room and the large volume of water contained in the flume represented the external environment. The box had a number of circular holes in both windward and leeward faces and the total area of these ventilation openings could be varied by removing plastic plugs from the holes.

To simulate stack-driven flows, brine and fresh water were used to create density differences; brine is denser than fresh water and therefore the buoyancy forces act downwards. In order to model a localised source of heat in a building, brine was injected continuously, and at a constant rate, through a circular nozzle in the top of the box. This fluid descended as a turbulent plume and is the analogue of a thermal plume rising in air from a source of heat.

Wind flow past the box was simulated by a flow of water in the flume tank. The flow of water generated by the flume flowed around the box and resulted in a 'wind' pressure drop Δ between the windward and leeward openings which was measured using a manometer tube; details of this procedure are given in [4].

Flows were visualised by adding dye to the brine injected into the box and using a shadowgraph. The dye colours only the salty fluid so that regions of dense fluid (coloured) and regions of fluid at ambient density (uncoloured) can be clearly distinguished. The shadowgraph enhances the contrast between regions of different density and allows fine scale structures in the flow to be seen.

An experiment was started by removing a number of plugs from openings in the box, at high-level on the windward face and at low-level on the leeward face, and supplying dense salt solution to the plume. After some time a steady-state flow was established. The height of the interface and the density difference between the ambient fluid and the salty layer of fluid inside the box were then measured. Ventilation rates and equivalent temperature differences for air flows in buildings were then deduced from these measurements, see §4.

Using the techniques described to create wind and buoyancy forces it was possible to simulate natural ventilation flows for a wide range of conditions. The effect of the wind speed on the ventilation rate and stratification was examined by varying the mean flow speed in the flume; the effect of the strength of the heat source was examined by increasing the density and volume of salt solution injected into the box, and the effect of the area of the openings was investigated by varying the number of openings in the box.

3. Theoretical model

In order to determine the steady-state height h of the interface and the temperature of the warm zone when a displacement mode of ventilation is assisted by wind we make use of the theory of transient, wind-assisted displacement flows developed by [5] and apply the method described by [1]. By matching the flow in a turbulent plume with the flow driven through an enclosure by a warm upper layer, taking into account the additional driving produced by the wind, it is possible to show that the steady-state density step across the interface may be expressed as

$$\Delta\rho = \frac{\rho(B^2 h^{-5})^{1/3}}{gC},$$
(3)

and the height of the interface may be deduced from

$$\frac{A^*}{H^2} = C^{3/2}(h/H)^{5/3}\left(\frac{1-(h/H)-(d_c/H)}{(h/H)^{5/3}} + CFr^2\right)^{-1/2},$$
(4)

where Fr denotes the Froude number

$$Fr = \frac{(\Delta/\rho)^{1/2}}{(B/H)^{1/3}}.$$
(5)

The derivation of expressions (3)-(5) is given in [6]. The Froude number is a measure of the relative magnitudes of the wind-produced velocity $(\Delta/\rho)^{1/2}$ and the buoyancy-produced velocity $(B/H)^{1/3}$: for $Fr \ll 1$ buoyancy provides the dominant driving force and for $Fr \gg 1$ wind forces dominate the ventilation flow.

The theoretical model (3)-(5) predicts that when a displacement flow is assisted by wind, the height of the interface h/H is dependent upon entrainment into the rising plume, the wind pressure drop Δ, the dimensionless area of the openings A^*/H^2 and the strength B of the heat source. This is in contrast to flows driven by buoyancy forces alone where the position of the interface is independent of the strength of the source and depends only upon entrainment into the plume and the dimensionless area of the openings, see equation (1).

4. Application to building ventilation

The steady-state temperature distribution in a naturally ventilated enclosure may be determined from the model as follows. First, the power E (Watts) of the heat source in the space must be converted into an equivalent source strength B using

$$B = \frac{g\beta E}{\rho c_p}, \tag{6}$$

where the physical properties of the ambient air at $15°C$ are $\beta = 3.48x10^{-3}°C^{-1}$, $c_P = 1012 \text{ J kg}^{-1}°C^{-1}$ and $\rho = 1.225 \text{ kg m}^{-3}$. The wind pressure drop is related to the square of the wind speed, and hence, the Froude number (5) may be expressed as

$$Fr = U_{wind}\left(\frac{C_{pi} - C_{po}}{2}\right)^{1/2}\left(\frac{H\rho c_p}{g\beta E}\right)^{1/3}, \tag{7}$$

where U_{wind} denotes the mean wind speed, and C_{pi} and C_{po} denote the pressure coefficients at the inlet and outlet openings, respectively. By specifying the area of the openings and the height of the space, the height h of the interface separating the warm and cool layers of air can now be predicted from (4). Alternatively, a minimum acceptable interface height, e.g. $h = H/2$, may be specified by the designer and the area of openings required to achieve this height predicted from (4).

The density step $\Delta\rho$ across the interface (3) can be converted into an equivalent temperature difference using the equation of state, namely, $\Delta\rho/\rho = - \beta(T_u-T_a)$, where T_u and T_a denote the respective temperatures of the warm upper layer and the cool lower layer. Substituting for (6) into (3) yields the temperature of the warm layer of air, namely

$$T_u = T_a + \frac{1}{g\beta C h^{5/3}}\left(\frac{g\beta E}{\rho c_p}\right)^{2/3}. \tag{8}$$

Thus, by raising the height of the interface and/or reducing the power output of the heat source it is possible to decrease the temperature of the upper layer, and hence, passively cool the space.

Ventilation rates in buildings are normally expressed in terms of the number of air changes per hour ACH, and hence, if V is the total volume of the enclosure we have

$$ACH = 3600\frac{Q}{V} = 3600\frac{C}{V}\left(\frac{g\beta E h^5}{\rho c_p}\right)^{1/3}. \tag{9}$$

5. Results and Discussion

Results of the laboratory experiments are now described and compared with the theoretical predictions. In order to avoid confusion, these results will be described assuming the direction of motion in the plume is upwards as it is for the case of a thermal plume rising from a heat source in a building.

In the absence of wind, the plume ascends entraining ambient fluid as it rises. Due to entrainment, the volume of fluid carried in the plume increases with height and its temperature decreases. On reaching the ceiling of the space the plume spreads horizontally creating a layer of warm air that gradually increases in depth and temperature and drives a flow through the enclosure. Inflow of cool ambient air is through the low-level openings and outflow of warm air is through the upper openings and thus a displacement flow is set-up. A steady state is eventually established when the ventilation flow rate through the space is equal to the flow rate in the plume at the height of the interface which separates the warm and cool layers of air. The upper layer is then well-mixed and its temperature is identical to the temperature in the plume at the height of the interface.

a) Effect of wind speed. If the low-level openings are located in regions of positive wind pressure and the leeward openings in regions of negative wind pressure then the effect of a flow of wind around the building is to raise the height of the interface toward the ceiling. The interface continues to rise until a new steady-state flow is established. Both the depth and temperature of the warm upper layer are then less than for the no-wind case and the ventilation flow rate through the enclosure is increased. These effects are illustrated schematically in figure 1. Increasing the wind speed is equivalent to increasing the Froude
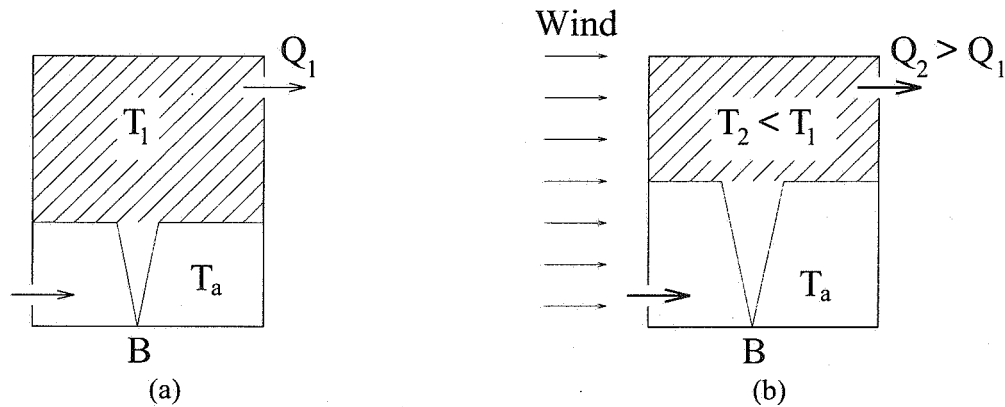
Figure 1. Schematic diagram showing the effect of wind on the interface height, temperature and flow rate established by displacement ventilation in an enclosure, a) stack-driven flow, and b) stack-driven flow assisted by wind.
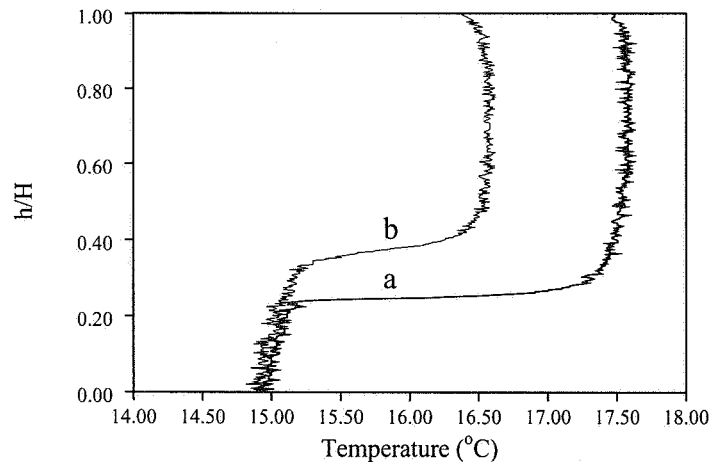
Figure 2. Typical temperature profiles for displacement flows driven by a) buoyancy forces alone ($Fr = 0$), note the interface at approximately $h/H = 0.25$, and b) buoyancy forces assisted by wind ($Fr = 9$), note the interface at approximately $h/H = 0.4$. The temperature step between the upper and lower layers is significantly reduced in b). In both a) and b) $A^*/H^2 = 4.3x10^{-2}$, $B = 2.3x10^{-6}$ m^{4}s^{-3}.

number and it has been demonstrated during the experiments that the displacement mode of ventilation is maintained for a wide range of Froude numbers. Typical temperature profiles, deduced from measurements of salinity, are illustrated in figure 2, where it can be seen that the wind has increased the depth of the ambient layer and decreased the temperature step across the interface.

b) Effect of opening area. Passive cooling may also be achieved by increasing the dimensionless area of the openings. The effect of an increase in the dimensionless area of the openings on the steady-state interface height and upper layer temperature is shown in figure 3. The experimental measurements are shown by symbols and the theoretical predictions by the continuous line. For a fixed wind speed and source strength, the interface height rises as the dimensionless area of the openings is increased (figure 3a). As a consequence of the rise in the interface height, the temperature (figure 3b) of the warm upper layer decreases and the ventilation rate increases. A factor of two increase in the dimensionless area of the openings A^*/H^2 represents the equivalent of a factor of two increase in the area A^* of the openings or a factor of $2^{-1/2}$ decrease in the height of the space.

(a)   (b)
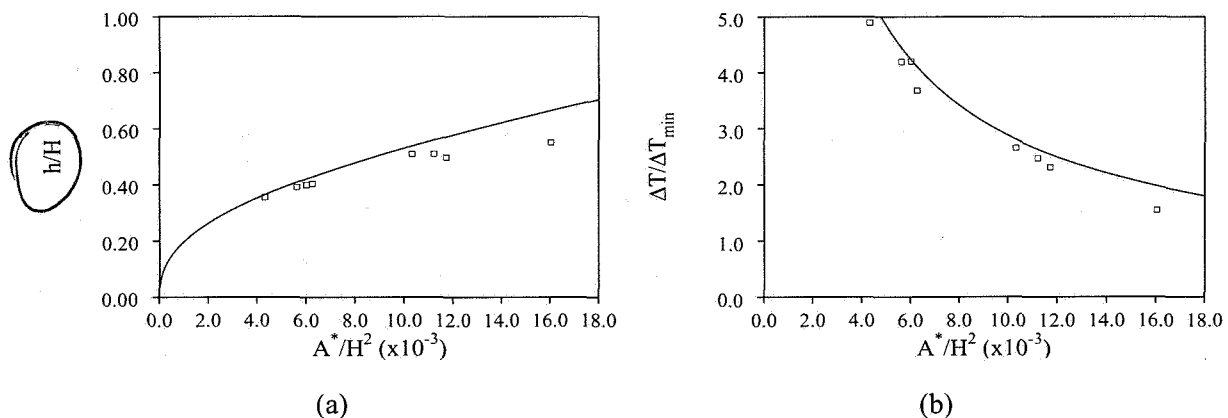
Figure 3. Effect of opening area: a) dimensionless interface height h/H vs. dimensionless area of the openings A^*/H^2; b) dimensionless temperature step across the interface $\Delta T/\Delta T_{min}$ vs. dimensionless area of openings A^*/H^2. ΔT_{min} is the difference in temperature between the ambient air and the air in the plume at ceiling height.

c) Effect of source strength. An increase in the source strength B results in a decrease in the Froude number, see (5), and hence, buoyancy forces become more significant and the depth of the ambient layer decreases (figure 4a). Furthermore, as B increases the temperature of the upper layer increases (figure 4b) thereby enhancing the buoyancy-driven flow and the overall ventilation flow rate. In this case cooling may be achieved only by decreasing the source strength. If, for example, we consider a wind speed of 2 ms^{-1} incident with a building 5m in height then, in figure 4b, a non-dimensional source strength of 0.5×10^{-2} corresponds to a temperature step across the interface of approximately 12°C in air (assuming $T_a = 15°C$ and $C_{pi} - C_{po} = 1$). Increasing the non-dimensional source strength to 1×10^{-2} results in a temperature step of approximately 20°C.

The interface height depends weakly upon the source strength as a large increase in B produces only a relatively small decrease in the interface height. However, the temperature step across the interface, and hence, the temperature of the upper layer, is strongly dependent upon the source strength.

Paradoxically, the effect of reducing the Froude number by increasing the strength of the source is to *increase* the ventilation flow rate through the enclosure. Recall that a reduction in the Froude number which results from a decrease in the wind speed *decreases* the ventilation flow rate (see §5b). These results are described in greater detail in [6].
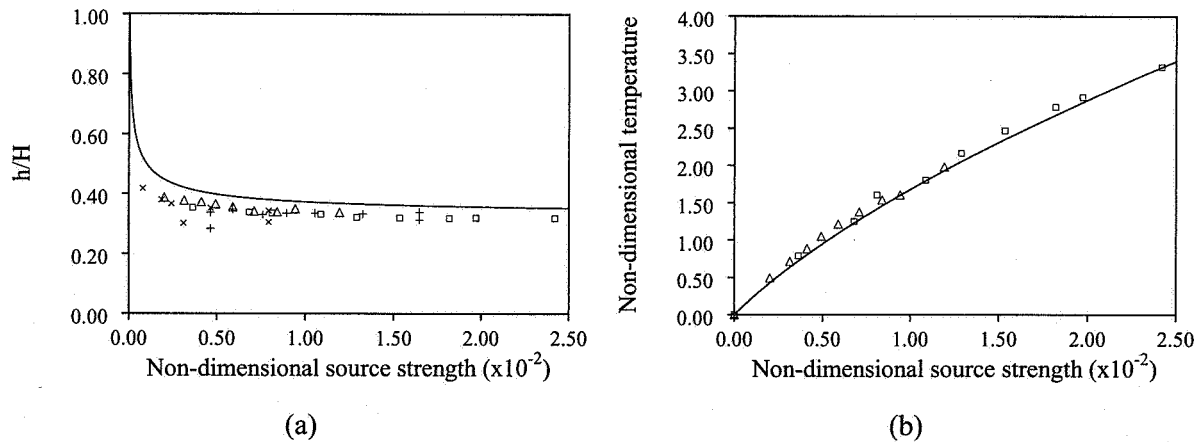
Figure 4. Effect of source strength: a) dimensionless interface height h/H vs. dimensionless source strength and b) dimensionless temperature step across the interface $\Delta T\, g\beta H/(\Delta/\rho)$ vs. dimensionless source strength $B/(H(\Delta/\rho)^{3/2})$.

6. Conclusions

Displacement flows driven by buoyancy forces *assisted* by wind have been simulated at small scale in the laboratory and the results of these experiments compared with the predictions of a theoretical model. The theoretical predictions are in good quantitative agreement with experimental measurements and the model may be used to estimate temperature profiles and ventilation flow rates within naturally ventilated buildings. Air flow and temperature conditions within an enclosure have been found to be dependent upon the relative magnitudes of the wind and buoyancy produced velocities (i.e. the Froude number), the area of the openings and the height of the space. Displacement flows established by a point source of buoyancy on the floor of the space are maintained for a wide range of Froude numbers when low-level openings are located in regions of positive wind pressure and high-level openings are located in regions of negative wind pressure. In this case, wind flow has a three-fold effect on the buoyancy-driven flow in the space: an increase in the wind speed i) raises the height of the interface above the floor of the space, thereby increasing the depth of the layer at ambient temperature; ii) results in a decrease in the temperature of the buoyant upper layer and iii) increases the ventilation flow rate through the enclosure. An increase in the strength of the heat source: i) decreases the depth of the layer at ambient temperature, ii) increases the temperature of the upper layer and iii) increases the ventilation flow rate. The height of the stratification is one of the main design parameters for displacement ventilation and by raising the height of the interface the reduced temperature of the upper layer may be used to provide additional cooling of the building fabric. In addition a greater area of the fabric is exposed to the ambient air thereby increasing the potential for cooling.

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VENTILATION AND COOLING

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HEAT-PIPE HEAT RECOVERY FOR PASSIVE STACK VENTILATION

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HEAT-PIPE HEAT RECOVERY FOR PASSIVE STACK VENTILATION

SYNOPSIS

Four types of heat-pipe heat recovery systems were tested for application in passive stack ventilation. The effects of fin shape, pipe arrangement and air velocity on the heat recovery effectiveness were investigated. The air velocity was found to have a significant effect on the effectiveness of heat recovery; the effectiveness decreasing with increasing air velocity.

The pressure loss coefficient for heat pipe units was also determined. It was found that at low velocities for natural ventilation the pressure loss coefficient decreased with increasing air velocity but the total pressure loss increased with the velocity. It is recommended that in naturally-ventilated low-rise buildings, without the wind effect or solar energy, the design duct mean velocity should be less than 1 m/s in order for a heat recovery system to function properly. The use of a solar chimney and/or wind turbine could increase the range of air velocity and so the amount of heat recovery.

INTRODUCTION

Ventilation accounts for 30% or more of space conditioning energy demand but as much as 70% of this energy can be recovered by the use of ventilation heat recovery systems [1]. Until now, effort has mainly been devoted to the design and development of heat recovery systems for mechanically-ventilated buildings. However, most domestic buildings are naturally ventilated and little consideration has been given to heat recovery from these buildings. A crucial parameter that limits the use of heat recovery with natural ventilation is pressure loss. The total pressure loss through a natural ventilation system should be much lower than in a mechanical ventilation system so that sufficient air flow can be achieved in the building. A system employing heat pipes would have the potential to provide substantial heat recovery without significant pressure loss.

A heat-pipe heat recovery unit is a heat exchanger consisting of externally-finned sealed pipes using a working fluid such as methanol or water. The unit is divided into two sections, i.e., the evaporator and the condenser, for heat exchange between exhaust and supply air (see Fig. 1). In addition to low flow resistance, a heat-pipe heat exchanger has a number of other advantages over conventional heat exchangers such as high reliability, no cross-contamination, compactness and suitability for both heating and cooling.

The objective of this study is to assess the performance of heat-pipe heat recovery units for naturally-ventilated buildings. The effectiveness of four heat-pipe units was measured in a two-zone chamber. The pressure loss characteristics of the units were determined by CFD modelling as well as measurement.

EFFECTIVENESS OF HEAT RECOVERY

The effectiveness of a heat-pipe heat recovery unit for sensible heat exchange between supply and exhaust air of the same flow rate, ε (%), is defined as:

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$$\epsilon = \frac{T_s - T_i}{T_r - T_i} \times 100\%$$ (1)

where T_i and T_s are the temperatures of inlet air before and supply air after the condenser section of heat exchanger in the supply duct (°C) respectively, and T_r is the temperature of return air before the evaporator section of heat exchanger in the exhaust duct (°C).

Measurements of the effectiveness were carried out in a vertical two-zone test chamber with a heat-pipe heat recovery unit. The two-zone chamber was designed to allow good mixing of supply air with room air in the lower zone and maintain a uniform temperature and concentration of return air in the upper zone. This ensured the reliability of temperature and air flow measurements.

Temperatures up and downstream of the heat recovery unit in both supply and exhaust ducts were measured using thermocouples (type T). The temperatures were recorded by a data logger. The air flow rate was measured using the constant-injection tracer-gas method [2].

Test chamber

Fig. 2 shows the schematic diagram of the test chamber. The chamber was made of plywood insulated with a layer of polyurethane. The chamber had a net interior base area of 1.169 X 1.133 m and a total height of 2.335 m. It was divided into two zones with a horizontal partition. There was an opening (0.215 X 0.215 m) in the middle of the partition to allow air to flow from one zone to another. Supply and exhaust ducts were connected to the chamber on one of the vertical walls. The air ducts were also made of plywood. When in operation, air entered the lower zone of the chamber via the supply duct and return air was extracted from the upper zone through the exhaust duct. A heat-pipe heat recovery unit was housed in the supply and exhaust ducts for heat exchange between return and supply air. An axial flow fan with adjustable speed was used to generate air flow through the chamber.

Heat pipes

Four types of heat-pipe heat recovery units were constructed and tested. Fig. 3 shows the cross-section of the heat pipes. The working fluid in the pipes was methanol with an operating temperature range from -40° to 100°C.

The first heat recovery unit (Type I) consisted of a bank of seven externally finned heat pipes. Each pipe was 0.0127 m in outside diameter and 0.45 m in length with 72 continuous plain fins on both the condenser and evaporator sections. The dimensions of each fin were 0.215 m long, 0.048 m high and 0.45 mm thick. There was a 0.02 m divider at the middle of the bank to prevent cross-contamination of return and supply air. The cross-sectional area of both the condenser and evaporator sections was 0.215 X 0.215 m. The total surface area of each finned pipe including fins and exposed pipe was 0.196 m². The whole unit was made of copper.

The second type heat pipe had cylindrical spine fins. The fins were made of copper wire. The

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unit with this type of fin consisted of three heat pipes of the same size and material as for Type I. There were eight continuous rows of fins on each of these pipes. Each row had about 300 spine fins and each spine fin was 0.7 mm in diameter and 30 mm long. The fins were soldered on the pipes and the tips of fins were fixed in such a way that they were uniformly distributed circumferentially. The estimated total surface area of the spine fins of each heat pipe was 0.158 m², which is about 19% less than that of the continuous plain fins.

The third type of heat recovery unit was made of two rows of staggered heat pipes. Each row consisted of three heat pipes. Each pipe was 18 mm in diameter and 365 mm in length with 70 continuous louvred aluminium fins on both the condenser and evaporator sections. The dimensions of each fin were 180 mm long and 60 mm high. Each fin had 96 louvres with 2 mm spacing and 0.65 mm gap. The length of the louvres varied from 5.5 mm at the centreline of each pipe row to 8.5 mm near the edge of pipes. The cross-sectional areas of the condenser and evaporator sections were 180 X 180 mm and 175 X 180 mm, respectively. The total surface area for heat transfer of the evaporator section was 1.5418 m²; this is about 8% more than that for Type I heat pipes.

The fourth type of heat recovery unit was made of five in-line heat pipes with wire fins. Each pipe was 19.05 mm in diameter and 450 mm long with 34.5 turns of copper wire fins on both the condenser and evaporator sections. Each turn of fins had 65 loops of wire 0.65 mm in diameter. The height of fins was about 12 mm. The cross-sectional area of the unit is the same as that of the first type of heat pipe. The total surface area for heat transfer of the evaporator or condenser section was 0.6035 m²; this is less than half of the first type.

Results and discussion

Fig. 4 shows a comparison of the effectiveness for the four types of heat pipes. It can be seen that the rate of heat recovery increases with decreasing air velocity. However, for a given heat recovery unit this does not necessarily increase the total amount of heat recovery (proportional to velocity). To achieve a required quantity of heat recovery at a lower velocity, the size of a heat exchanger needs to be increased but this will result in a higher initial cost. The main benefit of a lower velocity is the lower pressure loss through the ventilation system since the flow resistance is proportional to the square of velocity.

The effectiveness for the spine-fin heat pipes presented in Fig. 4 was for seven equivalent heat pipes. The effectiveness for this type heat recovery unit was much lower than that of plain-fin heat pipes. The main reason for the ineffectiveness of spine-fin heat pipes is the poor thermal contact between fins and pipes, resulting in a high contact resistance.

For the same cross-sectional area, the staggered heat pipes with louvred fins were more effective than plain fins, particularly at lower velocities. This may be attributed mainly to the increased external surface area available for heat transfer per unit cross section (55% more).

The effectiveness of the unit with wire fins was lower than that with plain fins. This is due to the lower external surface area for the wire-fin heat pipes. If the surface area was increased by say 50%, which would still be less than that of the plain-fin heat pipes, the effectiveness would be higher than that of the plain-fin heat pipe unit.

The effectiveness of heat pipes could be increased by employing more than one bank. For example, for Type I heat pipes, the measured heat recovery was between 16% and 17% more efficient using two banks than using one bank. It may be postulated that the effectiveness could be further increased by employing more banks of heat pipes but this would increase flow resistance and cost of installation. For natural ventilation the resulting pressure loss must be smaller than the driving force so that adequate air flow rates can still be achieved.

PRESSURE LOSS ACROSS HEAT PIPES

The pressure loss across a heat-pipe unit is represented by the pressure loss coefficient (k) as follows:

$$k = \frac{\Delta P_s}{\frac{1}{2}\rho V^2} \qquad (2)$$

where ΔP_s is the static pressure loss across the unit (Pa), V is the mean velocity of air flowing over the unit (m/s) and ρ is the air density (kg/m^3).

The pressure loss coefficient for Type I and III heat-pipe units was predicted by means of CFD modelling. The predictions were carried out using the CFD package FLUENT [3]. The CFD technique was validated for predicting the pressure loss coefficients for a number of duct fittings [2]. In the predictions, each row of heat pipes was modelled as one bank of rectangular cylinders such that it had the same free-area ratio and thickness as the real heat pipes. The fins were modelled as uniformly distributed rectangular studs on both sides of heat pipes such that the total cross-sectional area of the studs was the same as the sum of that of fins.

To determine the effect of fin shape on the pressure loss characteristics, flow resistance was also measured for two types of heat pipes - Type I and Type IV with pressure taps fitted on the up and downstream ducts of 0.15 m in diameter. Since the pressure loss at low values was difficult to determine, measurements were made at velocities higher than 2 m/s such that the resulting pressure loss was higher than the precision of instrumentation (1 Pa).

The pressure loss coefficient was found to decrease with increasing mean air velocity. The pressure loss coefficient for Type I heat pipes can be correlated to velocity between 0.25 and 10 m/s as follows:

$$k = (2.6 + 1.177n) \, V^{-0.03n^{3/4}} \qquad (3)$$

where n is the number of heat-pipe banks.

The pressure loss at a given velocity can be obtained from the pressure loss coefficient $(=\frac{1}{2}k\rho V^2)$. For example, at a velocity of 0.5 m/s, the pressure loss through a section of one bank of heat pipes is about 0.57 Pa and total pressure loss through the whole unit (both condenser and evaporator sections) is just over 1 Pa. Thus, if the driving pressure available for ventilation is, say, 1 Pa, the mean velocity through the heat-pipe unit should not be more

than 0.5 m/s. At a velocity of 1 m/s, the pressure loss through both sections of the unit is 4.5 Pa. Without the wind effect, this would require a stack height of about 10 m at a temperature difference between inlet and exhaust openings of 10 K, or 4 m height at 25 K temperature difference. In naturally-ventilated low-rise buildings, the average driving pressures are unlikely to exceed this value. Therefore, in designing ventilation ducts for housing this type heat recovery unit, the mean air velocity should be less than 1 m/s.

Fig. 5 shows a comparison of pressure loss coefficient for Type I and Type III heat pipes. The predicted loss coefficient for the six staggered 18 mm heat pipes was higher than that for one bank of seven heat pipes of 12.7 mm in diameter despite the porosity (free-area ratio) of the former being higher than that of the latter. When the six pipes were arranged as a two-row in-line bank, the predicted loss coefficient became lower than that of one bank of seven smaller heat pipes. For example, at a velocity 1 m/s, the predicted loss coefficient for the two-row in-line six pipes was 3.3, compared with 4.2 for the staggered pipes and 3.7 for the in-line seven smaller heat pipes.

In Fig. 6 the measured pressure loss coefficient for Type I and Type IV heat pipes is compared. At air velocities higher than 2 m/s, the pressure loss through the wire-fin heat pipe unit was higher than that for the plain-fin heat pipes particularly at high velocities. This may be explained by the opposite effect of the two types of fins on flow turbulence. The plain fins could act as a flow straightener whereas wire fins as a turbulence generator. The former decreased flow resistance whereas the latter increased flow resistance as velocity increased.

CONCLUSIONS

The experimental measurements show that air velocity has a significant effect on the effectiveness of heat-pipe heat recovery. The effectiveness decreases with increasing air velocity. For heat pipes with plain fins, at the same velocity the heat recovery is between 16% and 17% more efficient using two banks than using one bank. Poor thermal contact between fins and pipes can drastically reduce the effectiveness of heat pipes.

The numerical modelling indicates that at low velocities the pressure loss coefficient decreases with increasing air velocity but the total pressure loss still increases with the velocity. It is recommended that in naturally-ventilated low-rise buildings, without the wind effect or solar energy, the design mean air velocity should be less than 1 m/s in order for a heat recovery system to function properly. The use of a solar chimney and/or wind turbine could increase the range of air velocity and so the amount of heat recovery. For use in a natural ventilation system where low pressure losses are required, a staggered heat-pipe unit does not provide better overall performance than the in-line counterpart.

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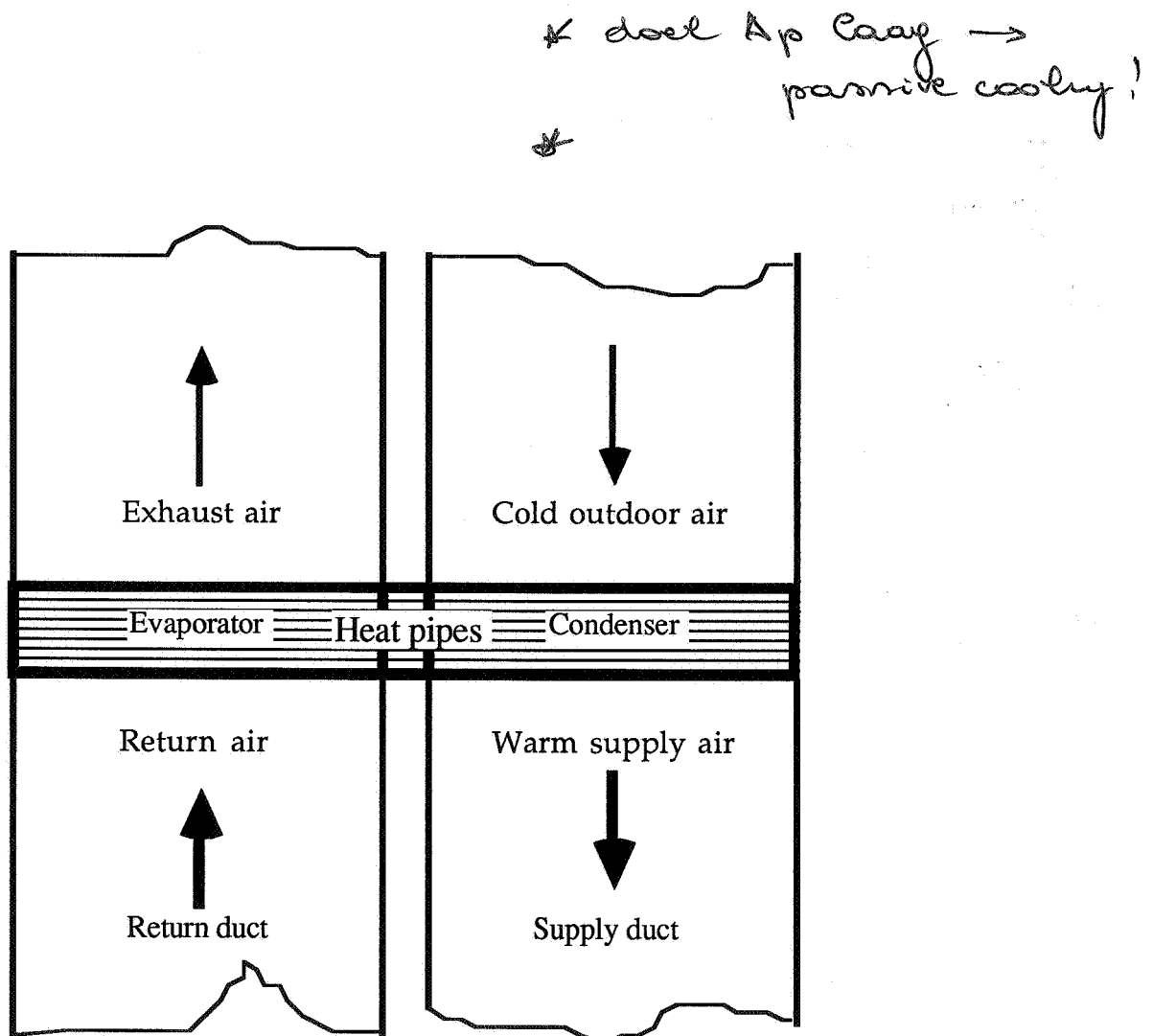
Fig. 1 Schematic representation of heat-pipe heat recovery
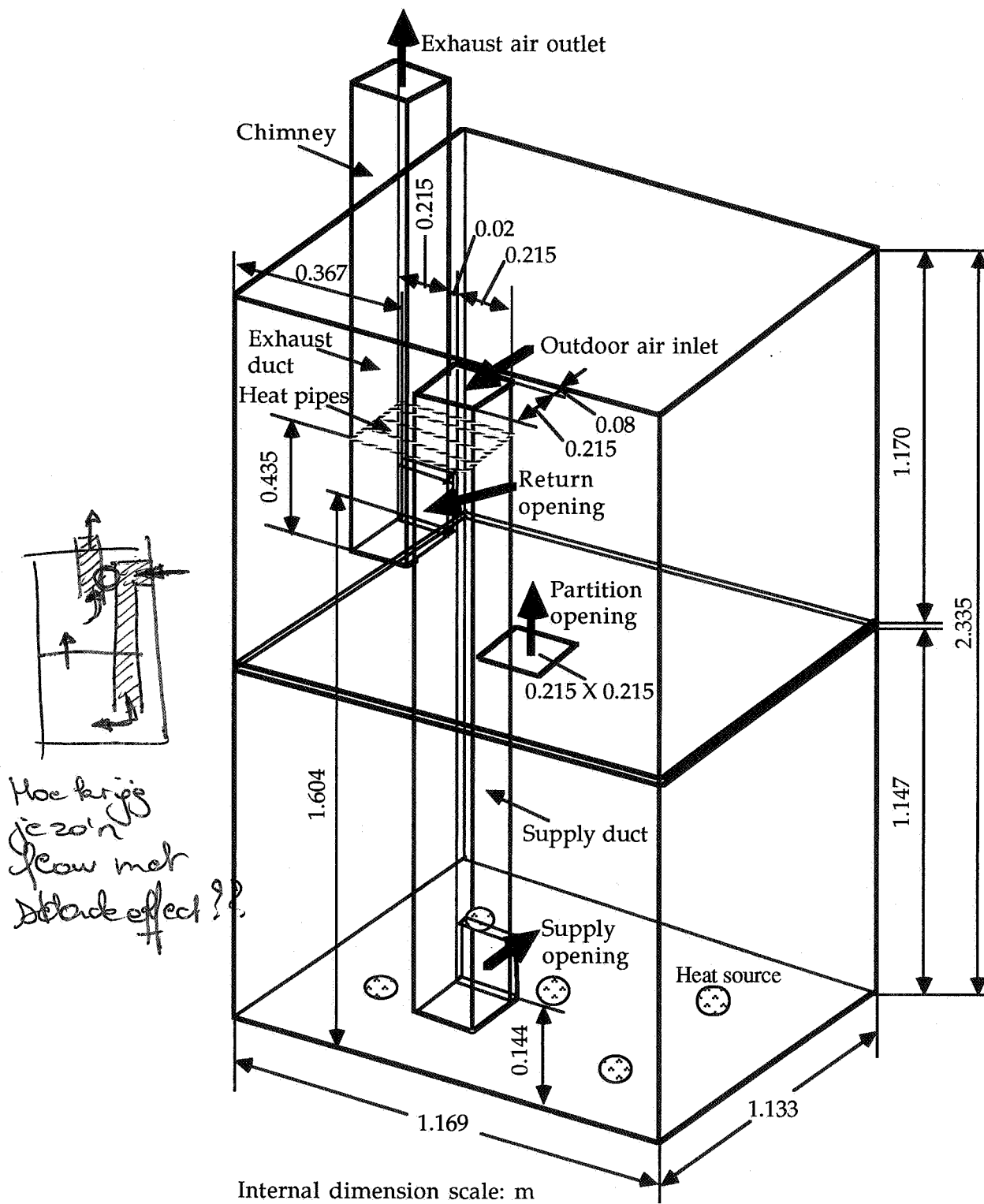
Fig. 2 Schematic of the two-zone test chamber with heat pipes
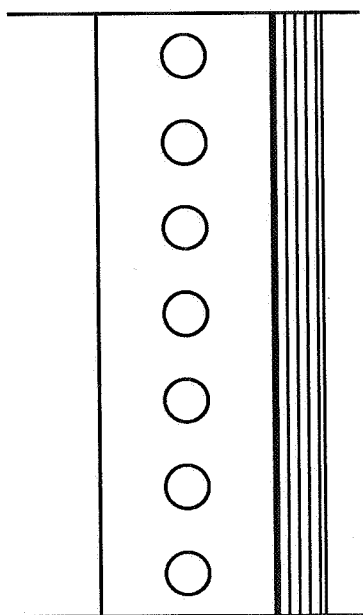
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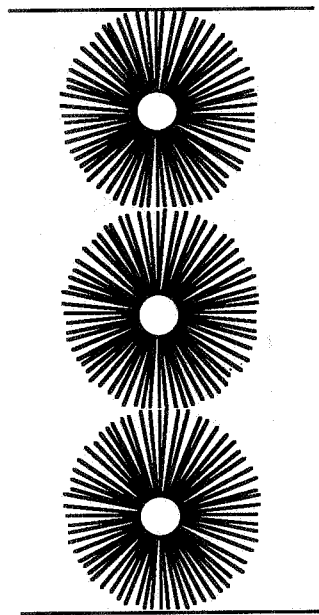
Type II (spine fins)
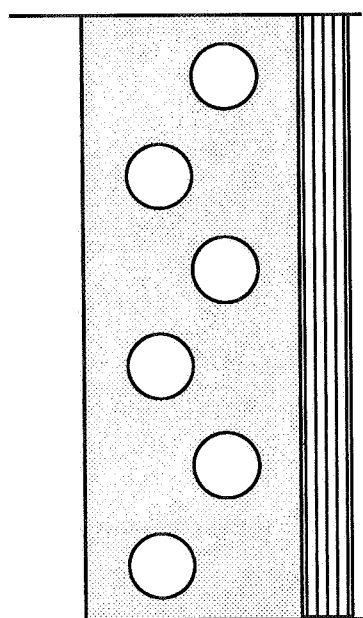
Type III (louvred fins)
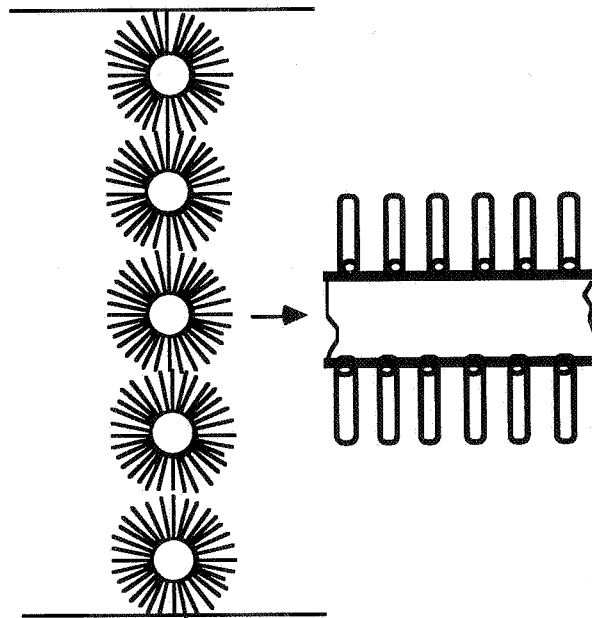
Type IV (wire fins)

Fig. 3 Cross sections of heat pipe units
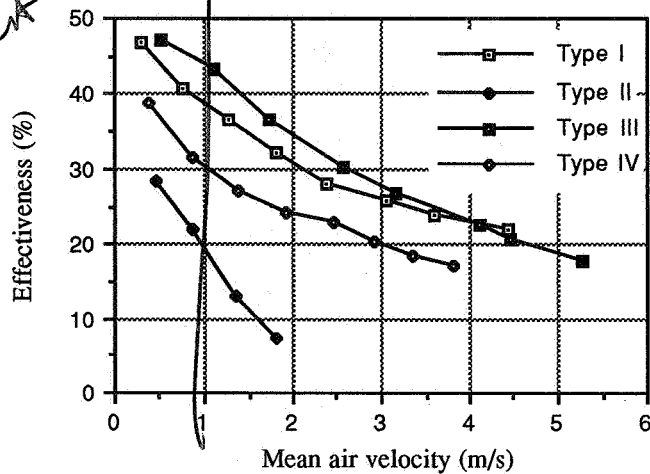
Fig. 4 Comparison of effectiveness for four types of heat pipes
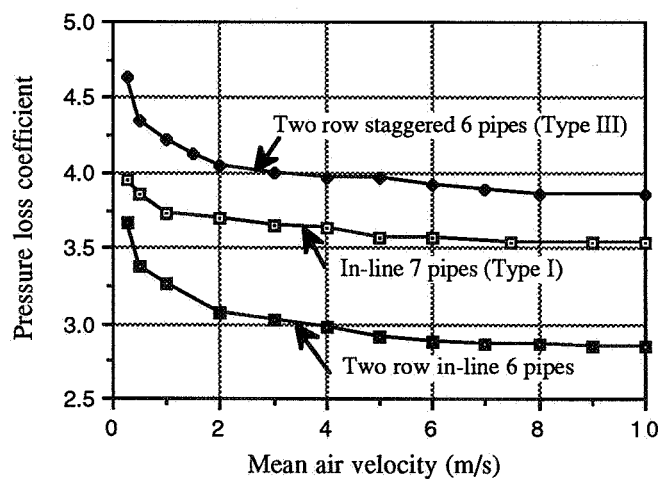
Fig. 5 Effect of pipe arrangement on the pressure loss coefficient
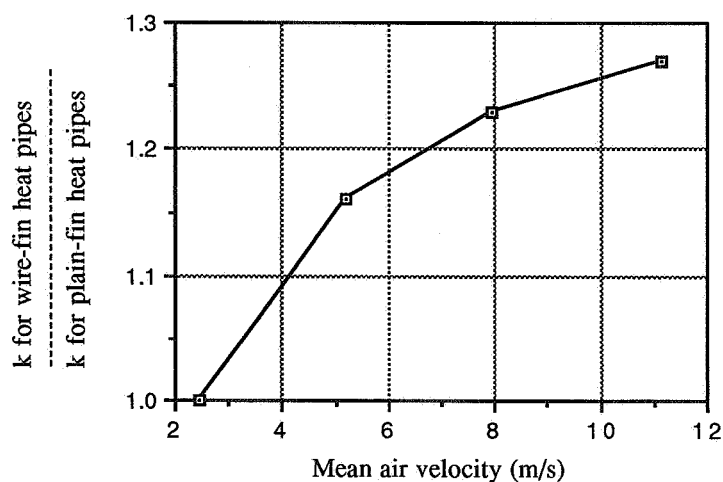
Fig. 6 Effect of fin shape on the pressure loss through heat pipes

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Title: Predicting Natural Ventilation Air Velocity Using Deterministic and Non-Deterministic Methodologies

Authors: E. Dascalaki, M. Santamouris

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SINGLE SIDE VENTILATION
(stokpaard Mat)

problem is
randomness
input param.

→ network
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⤷ fuzzy logic

grote Δ's
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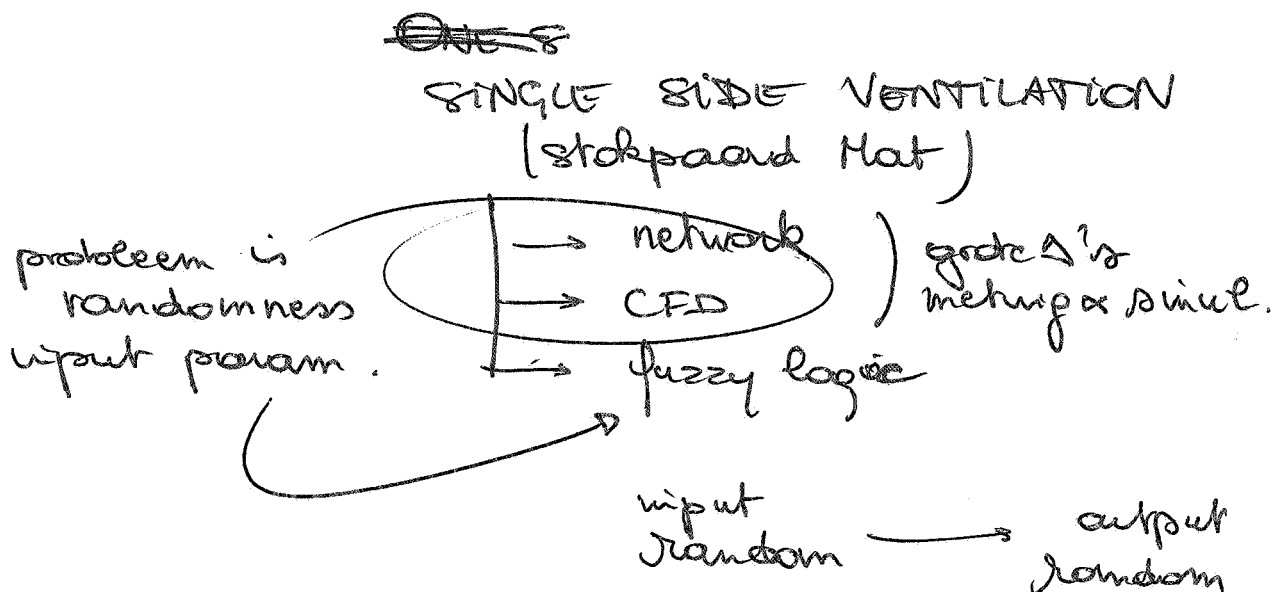
Fuzzy Logic ⟶ how does it work

PREDICTING NATURAL VENTILATION AIR VELOCITY USING DETERMINISTIC AND NON-DETERMINISTIC METHODOLOGIES

Elena Dascalaki, Mat Santamouris

GR-BES, University of Athens, Dept. of Applied Physics, Division of Meteorology, University Campus, Build. Phys.-V, Athens, GR157 84, Greece.

SYNOPSIS

An extensive experimental program on single sided natural ventilation was carried out within the frame of PASCOOL EC research project. Within the frame of these activities, four single sided natural ventilation experiments were carried out in a test cell, a full scale outdoor facility. Experimental data were used as input for numerical simulations that were carried out using air flow calculation tools based on network modeling as well as computational fluid dynamics (CFD). Finally, fuzzy logic techniques were used to predict the air velocity profile in the middle of the opening. This paper presents the simulation results using the above approaches as well as a comparison with measurements.

INTRODUCTION

Natural ventilation has proved to be an energy saving way to reduce indoor cooling load, achieve thermal comfort and also maintain a healthy indoor environment, in the case where the outdoor air conditions allow for its use. The physical processes involved in both cases are very complex and the interpretation of their role in ventilation effectiveness is a difficult task. When the building communicates with the outdoor environment through only one or more openings located at the same exterior wall, ventilation is single sided.

In order to study the physical phenomena related to single sided natural ventilation and their impact on the air velocity field at the opening level, four experiments were performed in a Test Cell, which is a full scale facility, during October 1993 in Athens, Greece. The tracer gas decay technique was used in order to measure the bulk air flow rate. Although tracer gas techniques provide an estimate of the bulk air flow rate in the investigated room(s), they do not give any information on the air flow/velocity field. This information is very important when treating problems of comfort or indoor air pollutant transport. Air velocity measurements were taken at various heights in the middle of the opening, to provide some insight of the air velocity field at the level of the opening. However, in the case of single sided natural ventilation, the uncontrollable nature of the wind, produces constantly changing air flow patterns through an exterior opening. Therefore, a lot of simultaneous multiple velocity measurements of high accuracy are required in order to predict the air flow rate successfully. This requirement implies that such an experiment is difficult and very expensive to perform.

Three modeling approaches were used in order to predict the air velocity profile in the middle of the opening: a) Bernoulli theory, b) CFD and c) fuzzy logic techniques. In the following sections, a brief description of the experimental procedure is given and the results from three modeling approaches are compared with measurements.

EXPERIMENTS

The PASSYS Test Cell is a fully equipped, two room, outdoor facility for thermal and solar monitoring [1]. Ventilation experiments were carried out in the "service room", while the door connecting it to the "test room" was kept closed and sealed. The service room has a floor area of 8.6 m^2 with a length of 2.4 m and a height of 3.29 m. It has an exterior door opening of 2.02 m^2 with a width equal to 1.01 m.

During the experiments indoor air temperature was measured by PT100 sensors (accuracy: $\pm$ 0.1 ^oC). A mast holding four PT100 sensors was used to monitor the vertical stratification. Their heights from the floor were: 0.42m, 1.76m, 2.37 m and 3.20m. Temperature stratification at the opening was monitored by an array of five T-fast sensors (accuracy ± 0.1 ^oC). The T-fast sensor is a 12.5E-6 m platinum wire, wired around an open Plexiglas base. The heights of these sensors from the floor were: 0.36m, 0.65m, 1.05m, 1.35m and 1.69m. Measurements of wind speed and direction were provided by a hot wire anemometer and a vane (accuracy: ± 5 deg) at a height of 1.5 m, 1m away from the Service room entrance door. At the same distance and at a height of 2 m, a T-fast sensor was placed to measure ambient temperature. Data on the wind at a 10 m height were also available. These measurements included 1min data, while the ones at 1.5 m were 1sec data. Data in front of the door were chosen for this analysis as more appropriate for studying the physical processes at the door level. The air exchange rates were derived using the single tracer gas decay technique. N_2O was used as tracer gas. Injection and sampling points were carefully chosen and distributed at various heights inside the studied rooms in order to supply the tracer gas homogeneously and also to monitor its spatial variation with time. The sampling period was set at 30 sec. Tracer gas concentration was measured by an infra-red gas analyser.

Comparison between indoor and outdoor average air temperature measurements shows that the average air temperature was higher indoors than outdoors. During the four experiments, this difference ranged between 0.5-3^oC. No significant temperature stratification was observed. Table 1 summarizes the mean characteristics of the Test Cell experiments.

Experiment	Mean Ambient Temperature	Mean Indoor Temperature	Mean Wind Speed, (ms^{-1}), at 10 m	Mean Measured Flow Rate (m^3h^{-1})
Exp. 1	24.1 $\pm$ 0.1	23.4 $\pm$ 0.1	3.3 $\pm$ 0.07	198 $\pm$ 27
Exp. 2	24.7 $\pm$ 0.1	24.3 $\pm$ 0.1	2.5 $\pm$ 0.05	202 $\pm$ 39
Exp. 3	25.7 $\pm$ 0.1	26.2 $\pm$ 0.1	3.8 $\pm$ 0.08	245 $\pm$ 65
Exp. 4	25.6 $\pm$ 0.1	26.6 $\pm$ 0.1	3.6 $\pm$ 0.072	322 $\pm$ 62

Table 1 : Characteristics of the Test Cell Experiments.

The air velocity at various heights in the middle of the opening was measured by an array of five triple hot wire anemometers (lower threshold: 0.2 m/s, accuracy: ± 0.02m/s) developed in the Laboratory of Meteorology of the University of Athens [2] and two commercial sensors manufactured by DANTEC (accuracy: ± 0.4%). The heights of the hot wires from the floor were 0.33 m, 0.63 m, 0.93 m, 1.43m and 1.73m. The heights of the commercial sensors from

the floor were 0.16 m and 1.87 m. Measurements were taken every second. Table 2 summarizes the mean air velocity values measured by each sensor.

Sensor Type	Height (m)	EXP1		EXP2		EXP3		EXP4	
		V(m/s)	SD	V(m/s)	SD	V(m/s)	SD	V(m/s)	SD
D1	1.87	0.47	0.09	0.54	0.12	0.62	0.14	0.57	0.14
HW1	1.73	0.35	0.05	0.39	0.09	0.44	0.10	0.39	0.09
HW2	1.43	0.44	0.12	0.42	0.12	0.48	0.10	0.43	0.10
HW3	0.93	0.65	0.27	0.43	0.08	0.54	0.13	0.50	0.12
HW4	0.63	0.61	0.27	0.39	0.07	0.47	0.12	0.45	0.11
HW5	0.33	0.64	0.27	0.42	0.11	0.51	0.14	0.49	0.10
D2	0.16	0.68	0.23	0.46	0.10	0.63	0.12	0.60	0.13

Table 2: Average air velocity measurements at the cell entrance

SIMULATIONS

a) NETWORK MODELS

Network models predict the air velocity at different heights at the opening level using the Bernoulli theory. According to the theory, the air velocity at a height z is:

$$V_B(z) = C_d \sqrt{2\Delta P/\rho} \ , \text{m/s} \tag{1}$$

where

$$\Delta P = P_{oi} - P_{dyn} + (\rho_a - \rho_i)gz \ , \text{Pa} \tag{2}$$

where P_{oi} and P_{dyn} are the reference and dynamic wind pressures (Pa), ρ_a, ρ_i are the outdoor and indoor air density (kg/m^3), while ρ is the air density in the direction of the flow.

Data (1 sec) from all experiments were included in a dataset resulting in a total of 2734 different measurements for each sensor. The wind speed in front of the cell, as well as the indoor-outdoor air temperatures were used as input values for simulations using PASSPORT-AIR [3] and the air velocity at the sensors' heights was calculated.
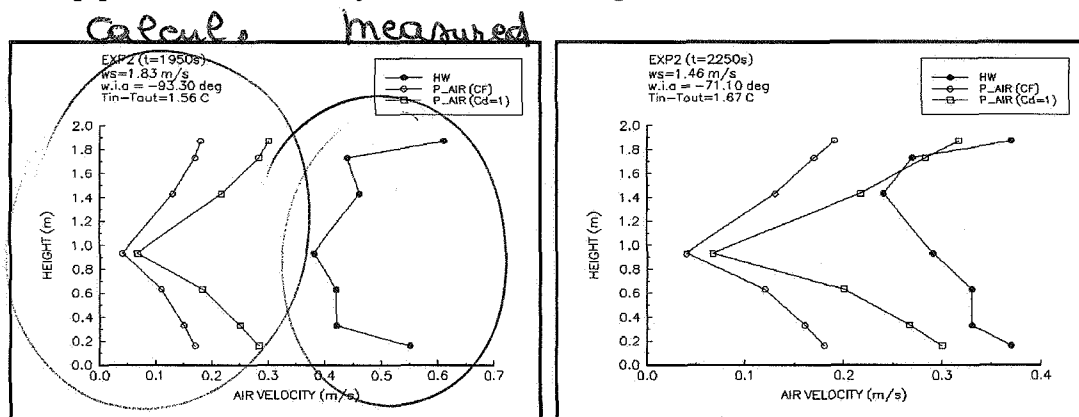
Fig.1: Predicted air velocity at the opening level using the Bernoulli theory. Comparison with measurements.

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Figure 1 gives a comparison between measured and predicted values. Predicted and measured values were not found to be in good agreement. The observed difference is attributed to the fact that network modeling considers homogenous pressure difference distribution along the opening surface. Simple observations of the flow characteristics near the openings in single sided naturally ventilated configurations reveal that the air velocity filed at the opening level is not homogenous. This absence of homogeneity introduces local turbulence, which is a very important driving force in the case of single sided ventilation. Network modeling practically neglects the effect of turbulence in the calculation of the air velocity. In reality, however, the air velocity field at the opening must be strongly affected by this factor, which is introduced by the constantly changing nature of the wind. Thus, despite its accuracy in predicting the air exchange rates, the Bernoulli theory does not provide much information on the flow field at the opening.

b) COMPUTATIONAL FLUID DYNAMICS (CFD)

CFD modeling is based on the solution of the Navier-Stokes system of equations for mass, momentum and energy conservation. This modeling type provides a detailed set of output on the air flow patterns and velocity fields. A very comprehensive set of input data is required, the accuracy of which determines the accuracy of the obtained results. The domain size and grid discretization also plays an important role towards achieving convergence. A computational fluid dynamics model, PHOENICS [4] was used in order to simulate single sided natural ventilation when the wind impinges normally on the opening (incidence angle $\cong 0$ degrees).

The domain size that was chosen for the simulation was 20m x 32.4m (height x width). The adopted grid discretization had 36 x 35 points and was finer in the viscinity of the opening and inside the cell. The upper and lower boundaries of the solution domain were defined as adiabatic walls while the right and left edges were defined as inlet and outlet respectively. The inlet was located at 10m in front of the cell entrance while the outlet was located at 20m behind the cell. These distances were found to ensure the formation of a logarithmic profile of the wind in the windward area as well as a total recovery of the flow after the formation of the wake in the leeward area. The temperature at the inlet was set equal to 24 °C. The wind speed at the inlet was taken uniform with height. As the wind speed components were not available at 10m away from the opening, various scenarios were tried by using different boundary conditions at the inlet. Figure 2 shows the results obtained using two scenarios. In the first scenario the horizontal and vertical component were taken equal to 2 m/s and 0.001 m/s respectively. In the second scenario the horizontal component was taken equal to 4 m/s. In both cases the solution domain and grid sizes are the same.

As shown, the derived air flow patterns are very different, though both scenarios give air velocity values that are close to the measured ones. Thus, the uncertainty of the boundary conditions at the inlet results in an uncertain flow pattern which implies that no unique solution can be derived. However, it is difficult to know in advance the exact values of the wind speed and its variation with height at any distance from the opening where this may be required according to the chosen simulation domain.
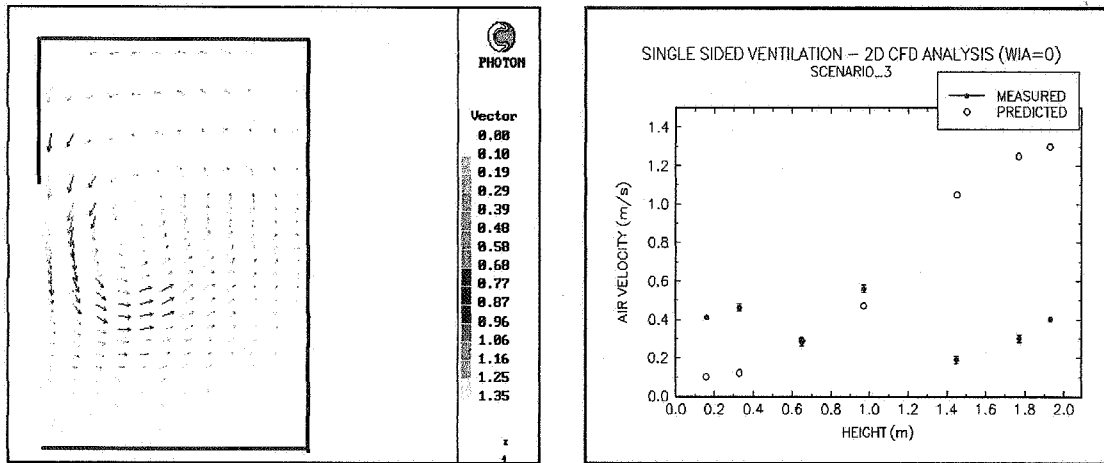
Fig.2a: Prediction of the air velocity at the opening level using CFD and comparison with measurements. Scenario 1 - grid size: 54 x 57 (wind speed at inlet : 2m/s (horizontal) and 0.001m/s (vertical)
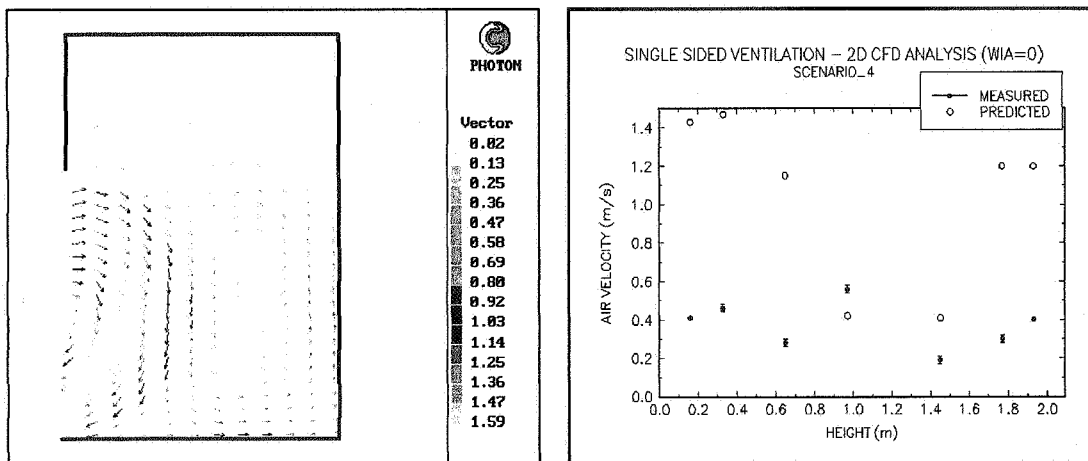
Fig.2b: Prediction of the air velocity at the opening level using CFD and comparison with measurements. Scenario 2 - grid size: 54 x 57 (wind speed at inlet : 4m/s (horizontal) and 0.001m/s (vertical)

[handwritten: problem is randomness DATA ...]

c) FUZZY LOGIC SYSTEMS

Fuzzy Logic Systems, have already been successfully applied to control and operate efficiently naturally ventilated buildings [5-6]. As fuzzy logic systems are considered as universal approximators, these systems are capable of uniformly approximate any non linear continuous unknown function, and thus can be used as non-linear dynamic system identifiers.

Air flow through large external openings, in single sided ventilation configurations, can be well approximated by non linear functions of specific meteorological and geometrical inputs. Therefore, it could be expected that such a function could be well approximated by a fuzzy system, when the necessary inputs and outputs are available.

[handwritten: training of the system with training data (input/output)]

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However, when the predictable parameter is not the air flow rate but the profile of the air velocity on the surface of the opening, the overall characteristics of the problem are completely different. The temporal variation of the profile of the air velocity on a large external opening is a random variable. The sample function of the air speed at a specific point (x,y) on the surface of the large opening represents a continuous non deterministic random process as future values of this function cannot be predicted from observed past values. The use of fuzzy techniques to describe such a process presents an important interest. What is interesting is to investigate in which cases appropriate fuzzy rules can be developed based on the existing inputs, which are also random variables, and also if such a fuzzy estimator has the ability for proper response to input patterns not presented during the training process (generalisation).

To reply to the above questions, a fuzzy estimator predicting the air speed at a specific point (x,y), on the surface of a large external opening has been developed. Three types of inputs have been considered : a) The wind velocity as measured in front of the Test cell, b) The wind direction measured also in front of the test cell, c) The mean temperature difference between indoor and outdoor of the test cell.
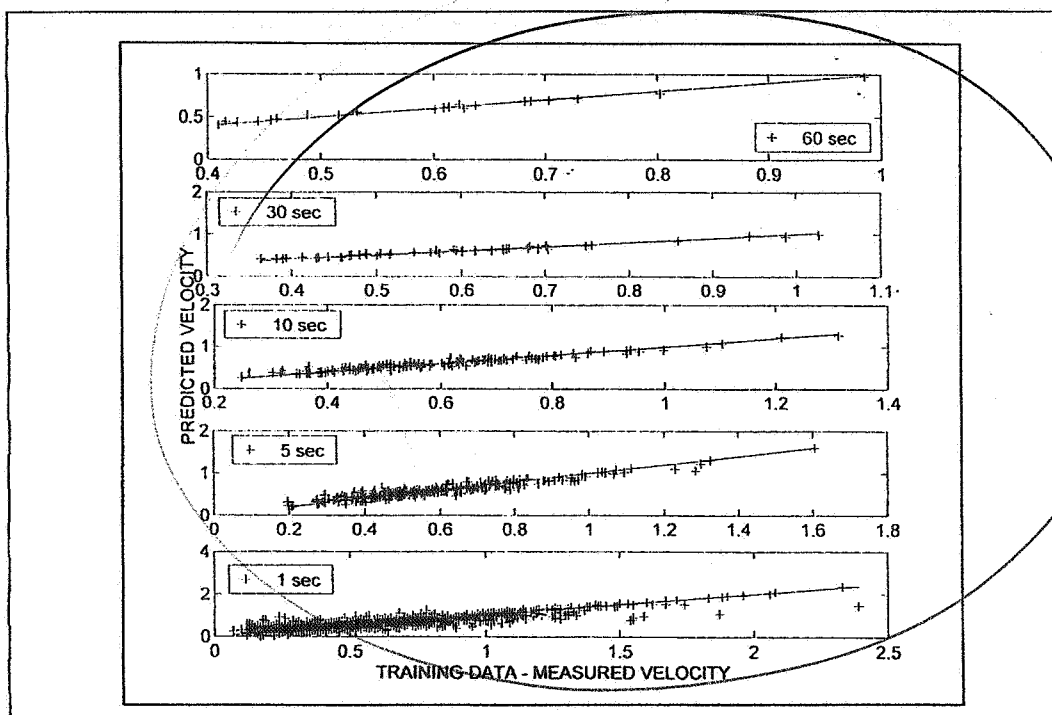
Fig.3: Training dataset: Predicted and Measured air velocity

Data measured at 1 sec time intervals have been used, and the corresponding air velocity on the surface of the opening for time intervals of one second, has been predicted. It should be noted that prediction of the air velocity at the opening surface for periods of one second has not any practical interest. However, one second data can be used to calculate mean values for larger time intervals, like 10 seconds, one minute or higher. The procedure to use one second input data instead of mean values for larger time intervals offers the advantage that a higher number of input data are used for training, while these inputs describe better the "physics" of

phenomenon than the mean values of larger the time intervals. In the following paragraphs, the discussion will focus on the data set composed by the values of the air velocity measured by one sensor at the highest measuring point of the door. A similar methodology can be applied to all other points at the door surface. The existing data set composed of 2734 data points has been divided in two almost equal parts. 1387 data points have been used to compose the training data set while the rest 1347 are used as a checking dataset.

Clustering techniques help to distill natural grouping of data from larger set producing a concise representation of the system's behavior. The objective in clustering is to partition a given data set into homogeneous clusters where all points in the same cluster share similar attributes and they do not share similar attributes with points in other clusters [7]. Crisp classification techniques oblige each point to belong to one of the clusters at least, and their membership in the cluster to which they are assigned is unity. In fuzzy clustering techniques each data point belongs to a cluster to a degree specified by a membership grade. One of the most popular fuzzy clustering techniques, is the so called "Fuzzy C-Means" [8]. This method uses concepts in n-dimensional Euclidean space to determine the geometric closeness of data points by assigning them to various clusters or classes and then determining the distance between the clusters. When the number of the clusters in a data set is not known, subtractive clustering techniques can be used to estimate the number of clusters and the cluster centers in the set of data.
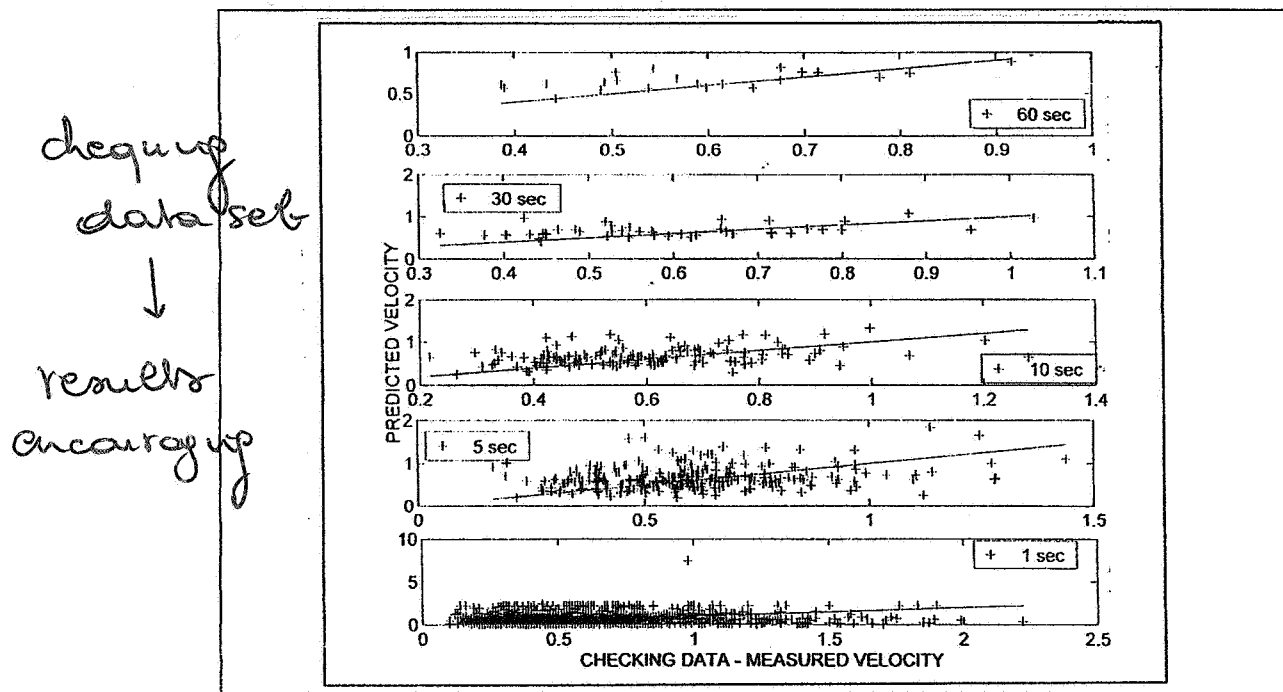
Fig.4: Checking dataset: Predicted and Measured air velocity

To improve classification of the input data, subtractive clustering techniques have been used. The range of influence for the wind speed, temperature difference, wind direction and air speed on the surface was set equal to 0.1,0.05, 10 and 0.05 respectively. An initial Fuzzy Inference System was generated and then back propagation techniques were used to train the

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system. The ANFIS - Adaptive Network based Fuzzy Inference System routine, operating in the frame of the MATLAB Fuzzy Logic Toolbox has also been used, for training.

The obtained results for both the training and checking data sets for one, five, ten, thirty and sixty seconds intervals are shown in figures 3 and 4 respectively. It is found that the present FIS predicts satisfactory the air velocity on the opening's surface for both the training and checking data while the trend of the predicted data is also satisfactory.

ACKNOWLEDGEMENTS

CONCLUSIONS

Deterministic methodologies were used in order to predict the air velocity in the case of single sided ventilation. Comparison with experimental values has proved their inability to give accurate information on the air velocity at the opening level. This is attributed to the fact that the air velocity at the opening is strongly influenced by ever changing parameters such as the wind speed and the temperature difference across the opening. The stochastic nature of the wind was found to be better approached by non-deterministic modeling procedures, based on the principles of fuzzy logic systems.

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Overall Strategy

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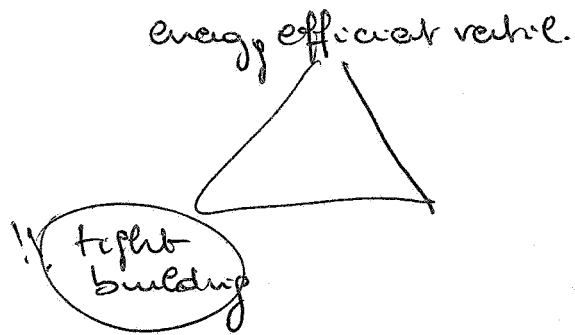
!! tight building

Airtightness Standard

→ Leakage index

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VENTILATION AND COOLING

18TH ANNUAL AIVC CONFERENCE
ATHENS, GREECE, 23-26 SEPTEMBER, 1997

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Predicting Envelope Air Leakage in Large Commercial Buildings Before Construction

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Predicting Envelope Air Leakage in Large, Commercial Building Before Construction

by

M D A E S Perera, J Henderson and B C Webb

SYNOPSIS

The concept of 'build tight - ventilate right' requires minimising air infiltration through the envelope of a building and then providing adequate ventilation in a controlled manner to satisfy the fresh air requirements of occupants. This paper describes a simple-to-use design tool (PC based and in spreadsheet format) for predicting the airtightness of office building envelopes either at the design stage or before a major refurbishment. The predicted value can be used as an indicator for post-completion air leakage testing to confirm the design assumptions and check on build quality.

Input of data is kept to a minimum. The simple design tool provides an overall leakage rate by considering the building size, glazed area, construction type and whether or not various tightness measures have been incorporated. The paper concludes with good comparisons between measured and predicted leakage rates in 10 office buildings. The paper also provides an example of how the tool can be used to assess the individual effectiveness of various key airtightness measures.

1. INTRODUCTION

This paper describes a prototype version of a 'simple to use' design tool (PC based and in spreadsheet format) to predict the airtightness of office building envelopes; either at the design stage or before a major refurbishment. This is a significant development since hitherto the only means available to quantify this leakage was through a fan pressurisation test [1] once the building was constructed.

1.1. Air infiltration in office buildings

Adequate ventilation is essential for the health, safety and comfort of building occupants, but excessive ventilation leads to energy waste and sometimes to discomfort. A building needs to be ventilated by design through controlled openings (e.g. openable windows) rather than by adventitious infiltration of air through the building envelope.

Air infiltration is the uncontrolled flow of air through cracks and gaps in the building envelope. It is driven by pressure and air temperature differences between the inside and outside of the building and is highly variable in response to changes in the weather. It cannot be designed for, and may therefore be considered as an overhead or penalty. Infiltration is not a reliable substitute for properly designed ventilation.

Good office design should therefore separate the mechanisms, which provide a good supply of fresh air to occupants, from the adverse and unpredictable effects of air infiltration. This demands good ventilation design and coupled to a clear and workable specification for an effective and maintainable airtightness layer. That is, the concept of 'build tight - ventilate

right'. It has to be emphasised that a building cannot be too tight - but it can be underventilated.

A building with an airtight envelope provides identifiable benefits to those who own, maintain and occupy it through the following:

- Energy savings since energy costs for space heating may be up to 20% less than for an equivalent but leaky building. Additionally, sophisticated energy-saving heating control systems and heat recovery systems can be economically viable options in tight buildings.
- Enhanced comfort since draughts and localised cold spots are minimised in a tight building. Providing controlled ventilation, e.g. at high levels, ensures adequate fresh air for occupants with a minimum of draughts around people.
- Reduced risk of deterioration in a properly ventilated but tightly constructed building; since otherwise air leaking out of the building will tend to pull warm and moist internal air through the fabric of the walls and roof.

1.2. Current requirements and guidance
Requirement L1 of the England and Wales Building Regulations requires that 'reasonable provision shall be made for the conservation of fuel and power in buildings by: limiting the heat loss through the fabric of the building ...'. Supporting this is Approved Document (AD) Part L [2] which provides guidance on some sealing measures for the more common building situations. For methods of reducing infiltration in larger and more complex buildings, the AD refers to the Building Research Establishment (BRE) Report on 'Minimising air infiltration in office buildings' [3].

This BRE Report is an outline guide setting out the principles of providing an effective airtightness layer through the design of a tight envelope and the sealing of air leakage paths. In addition to air leakage through the fabric of the walls and ceilings, infiltration can occur through the junctions between building elements such as:

- doors and windows and their frames,
- window and door frames and the walls,
- wall to ceiling joints, and
- wall to floor joints.

1.3. Ensuring compliance and performance specification
Using 'fan pressurisation' [1], the airtightness (or the leakiness) of the building envelope can be quantified. At present, however, this can only be done once the whole building has been constructed or, as an interim measure, when various stand-alone phases (of a multiphase project) are completed. This testing involves sealing a portable fan, such as the BREFAN system [1], into an outside doorway and measuring the air flow rates from the fans required to maintain a series of pressure differentials across the building envelope.

A suitable measure used to quantify the overall air leakiness of a building envelope is the 'leakage index', Q_{25}/S, where Q_{25} is the air flow rate at the imposed pressure differential of 25 Pa and S is the total permeable external envelope area. Airtightness target for a 'tight' UK building is set at [4] a value less than 5 m³/h per m², while an 'average' building should be less than 10 m³/h per m² and with a 'leaky' buildings in the region of 20 m³/h per m² or more [1].

2. 'AIRTIGHTNESS PREDICTOR' DESIGN TOOL

The objective was to produce a simple design tool which could be used at the design stage to predict the airtightness of office buildings. It was considered that this 'predictor' should have the following characteristics:

- Easy to use and requiring the minimum amount of data input.
- This input data should be readily available to the designer and should be at the gross building level rather than at any detailed component level.
- The output should be the predicted whole building leakage index.
- The tool should provide a clear indication of the contribution of each individual tightness measure to the overall building airtightness.
- In so doing, the tool should be flexible for the designer to consider 'what-if' scenarios of which tightness measures, when put together, will provide better return on investment while satisfying required tightness criteria.

2.1. Key parameters affecting airtightness

Extensive development work was carried out to determine the key parameters which affected the overall air leakage of office-type UK buildings. Key emphasis was placed on identifying the required gross characteristics of these parameters (rather than at any detailed component level) to ensure compliance with the design tool's requirements as set out above. The following is the final list of key parameters:

Building form

Users' input of actual or design values:

- Volume
- Surface permeable envelope area; i.e. the envelope area (usually walls and roof) separating the 'conditioned' space (i.e. heated and cooled) of the building from the unconditioned space or the outside
- Number of storeys
- Percentage of glazed wall area
- Number of single and double external doors

With these overall building characteristics and using an (unpublished) internal BRE database of UK office building characteristics, the total lengths of joints of the following possible leakage paths were derived:

- Between window glazing and frames
- Between window frames and walls
- Between walls and ceiling and between walls and floors

General component information

User input is through yes/no 'check' boxes for the following:

- Composition of walls, i.e. whether brick (bare, plastered or with wall board panelling), concrete block (bare or plastered), concrete panels (with or without gaskets), metal panels or curtain walling
- Whether windows and doors were weather-stripped or not
- Whether joints between walls and windows/doors, and between walls and floors/ceilings were caulked or not.
- Joints between walls and floor/ceiling (whether caulked or not)

2.2. Air leakage through components

Within the design tool, the air leakage Q induced through any porous surface areas or a leaky joint (at a pressure differential Δp across them) was determined using empirical equations of the form,

$$Q = k. \Delta p^n$$

where k is a flow coefficient expressed in terms of leakage per m^2 of porous surface area or for each metre length of crack. The flow exponent n characterises the type of flow and varies in value between 0.5 (fully turbulent) to 1.0 (completely laminar).

The values for k and n were obtained from compiled [5] measured data. In all instances, median values were used (to represent 'average' components) rather than the extremes (representing either high-performance or poor quality components). For example, for a caulked wall to wall timber joint, a k-value of 1.6×10^{-3} $dm^3.s^{-1}.m^{-1}.Pa^{-n}$ was used (rather than the quartiles 67×10^{-3} or 3.4×10^{-3}) with a common exponent 'n' of 0.6.

2.3. The prototype design tool

The prototype design tool is in spreadsheet form. The front worksheet receives input from the user and provides the leakage index and the percentage contribution made by each component and sealing measure. All calculations together with the database necessary to provide these results are contained in separate worksheets not obvious or accessible to the user.

The design tool works on the principle that leakage only occurs through the fabric or between building components. No specific calculations can be made if the building envelope is not designed from the outset for airtightness or this concept is not appreciated and significant large gaps are left in the building envelope. To ensure that the user is aware of the penalty for having large gaps, there is an additional check box which asks the users whether they are aware of these gaps in their building. If the answer is 'yes', then an empirical leakage (using previous in-house BRE measurements) of 10 m^3/hr per m^2 (at 25 Pa) is added to the overall calculated value.

The results are presented as both leakage indices (utilising permeable areas) and whole building leakage rates (using total building volume) at the standard pressure differential of 25 Pa for large buildings. However, corresponding values for a 50 Pa differential are also given even though it is difficult to establish this pressure in large UK office buildings with current pressurisation hardware [1].

3. RESULTS

3.1. Comparison of predictions with measurements

The design tool was evaluated using measured leakage rates in the following 10 UK office buildings held in an internal BRE database

Building		1	2	3	4	5	6	7	8	9	10
Year built		*1980*	*1963*	*1991*	*1965*	*1987*	*1990*	*1990*	*1971*	*1986*	*1985*
Volume (m³)		5,315	13,749	32,479	6,254	2,516	8,651	2,045	8,168	14,904	14,126
Surface area (m²)		1,750	3,769	8,189	2,195	1,105	2,508	829	3,056	4,726	4,394
% glazed		33	13	35	25	39	36	39	25	21	25
Large gaps		no	no	no	no	no	no	yes	yes	yes	yes
Wall[1]		br/p	bl/p	bl/p	br/p	bl/p	bl/p	bl/p	c_p	br/p	c_p
Window[2]		w	w	w	nw	w	w	w	nw	w	w
Doors[2]		w	w	w	w	w	w	w	w	w	w
Wall to window/door[3]		c	uc	c	uc	un	uc	c	uc	c	uc
Wall to floor/ceiling[3]		c	uc	c	uc	uc	uc	uc	uc	c	uc
Leak index (at 25 Pa)[4]	M	5.5	5.3	5.5	11.8	6.7	9.0	15.3	16.8	17.9	20.4
	P	3.5	4.3	3.2	9.1	6.3	5.3	15.8	20.6	13.2	16.0

Notes: *1 - bl(ock), br(ick), p(lastered), c_p (concrete panel); 2 - w(eatherproofed), nw (non- weatherproofed); 3 - c(aulked), uc (uncaulked); 4 - M(easured), P(redicted)*

The comparison between measured and predicted is encouraging given the generalised characteristics of UK buildings used in this prototype simplified tool. However, there are sizeable variations on some of the cases. Our view is that this could be improved by the following measures:

- Take account of the quality of the components used, i.e. whether good, average or poor, and use the appropriate component characteristic from the built-in database.
- Similarly, take account of infiltration through paths which are not obvious (and not component based) but is elsewhere on the building envelope and dependant on the care taken to minimise infiltration at both the design and construction stages.
- Extend the comparison study to encompass other buildings contained within the more extensive BRE database and determine whether other aspects (e.g. roof form) contribute to building tightness.

3.2. Identifying key tightness measures

The design tool can be used to identify cost-effective key tightness measures by determining the impact they have on the overall airtightness and their percentage contribution to this overall value. The following example in Building #1 (the BRE Low-energy office of 1980) shows the effectiveness of weather-stripped windows in contributing to the overall airtightness of the building.

	Weather-stripped windows	Windows without weather-stripping
Leakage index (m³/h per m² at 25 Pa)	*3.5*	*8.9*
Percentage contribution:		
through walls	15	6
through ceiling	36	15
through window glazing and frame	**34**	**73**
through door and frame	6	2
joints between wall and windows/doors	1	1
joints between walls and floor/ceiling	8	3

4. CONCLUSIONS

A prototype design tool to estimate the airtightness of large UK office buildings at the design stage is described here. This tool takes as input gross characteristics of the building such as volume and envelope area and converts these into possible leakage paths. A simple checklist then requests information about the methods used to seal these paths. The tool then uses this information and combines it with published leakage characteristics of individual components to provide an estimate of the leakiness of the building.

Comparison with full-scale results shows good agreement between measured and predicted. While the prototype design tool does require further refinement, it can be used to not only supply an airtightness estimate at the design stage but also a means by which the effectiveness of various airtightness measures can be considered.

However, it must be emphasised that the design tool is not a substitute for post-completion leakage measurement testing. Such testing is necessary to confirm design assumptions and provide a check on build quality. Data obtained through such testing could also be added to the growing database at BRE to refine this design tool as well as enhancing the derived practical guidance that could be provided to the UK construction industry.

ACKNOWLEDGEMENTS

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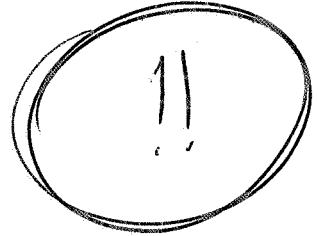
[handwritten: Last 6-7 years, a lot of low-energy buildings in UK → apply those techniques in more classical buildings]

NiteCool: Office Night Ventilation Pre-Design Tool

[handwritten: × Single zone × Pre defined ↓ you can't change mass ?]

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[handwritten: BRE]

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[handwritten: Results → o.a. cooling capacity worden dadelijk aangepast !! → hoe geen dyn berekn ≠ interess. grafieken]

[handwritten: Hypotheses ? × h_i ? × U-values → in code gaan magelijk]

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NiteCool: Office Night Ventilation Pre-Design Tool

M Kolokotroni, A Tindale and S J Irving

Abstract

NiteCool was developed under the Energy Related Environmental Issues in Buildings (EnREI) DOE Programme and is designed especially for the assessment of a range of night cooling ventilation strategies. The program is based on a single zone ventilation model and is configured to analyse a 10m x 6m x3m cell of an office building. It is intended to be used at the early stages in the design process to help the designer to make informed decisions on the construction, opening configuration and operation of the building. The user input is restricted to a few parameters from which a weekly internal temperature profile is predicted together with the energy consumption and the peak cooling capacity requirement relative to a reference system (with no night cooling). In this way, various building and system designs can be investigated by manually adjusting parameters until the comfort/energy consumption design criteria are met. The program can also be used to calculate the size of openings required to achieve a certain flow rate under given design conditions. This is a very quick and easy way to investigate the feasibility of using natural ventilation to improve comfort levels in buildings.

Introduction

It has been established over recent years through research work and built examples that night ventilation is an effective low energy cooling technique for appropriately designed modern buildings, especially in climates with relatively low peak summer temperatures during the day and medium to large diurnal temperature differences such as those of the UK. Such weather combination allows the thermal mass of the building to use the cool night air to discard the heat absorbed during the day. Therefore, cooling using night ventilation is particularly suited to office buildings which are usually unoccupied during the night so that relatively high air flows can be used to provide maximum cooling effect. Buildings using night ventilation for cooling in the UK have been evaluated and reported with encouraging results [1-3].

Low energy techniques to avoid overheating have also been central to research efforts across Europe, especially in southern European countries. Night ventilation is one of the main techniques being investigated. In order to help designers to explore its application, a number of easy-to-use pre-design computer tools have been developed. Examples include LESOCOOL [4] which deals with natural ventilation systems that can be used to passively cool a building and it is based on one design day and SUMMER -Building [5] which uses the admittance method for the thermal simulations with real weather data and includes natural ventilation as one of the four low energy cooling techniques. In both tools, the user can define the building details and predictions include hourly internal temperatures, air flow rates, and in the case of SUMMER, energy savings.

The design tool described in this paper has been developed especially for UK office buildings and climate with the aim to facilitate comparisons with energy consumption and comfort benchmarks and provide the opportunity to explore quickly variations in internal heat gains, ventilation rates, occupancy patterns and external temperatures. External temperatures are user defined so that the user can investigate various scenarios such as a few warm days followed by cool weather and vice versa. In developing the model, it was critical that user input parameters are kept to the minimum and simulation time is fast.

Building configuration and calculation algorithms

The building model is based on a typical cellular office with dimensions 10m width, 6m depth and 3m floor-to-ceiling height and therefore it is a single zone model. It is positioned in the middle of a row of offices on the middle floor of a 3-storey office. This module has been derived as a suitable office for night cooling through previous research work [6]. Three variations of thermal mass are included: LIGHT, MEDIUM and HEAVY. The definition of the types is based on the materials used (concrete) and area of exposed thermal mass which is provided by the ceiling. LIGHT construction includes a lightweight exposed concrete ceiling (specific heat 1kJ/kgK, density 1200 kg/m^3 and thickness 0.15m, heat capacity 180kJ/km^2) MEDIUM construction includes a heavy weight exposed concrete ceiling (specific heat 0.85kJ/kgK, density 2100 kg/m^3 and thickness 0.15m, heat capacity 270kJ/km^2). HEAVY construction includes a heavy weight waffle exposed concrete ceiling, thus offering an increased area of exposed thermal mass. The external wall is a granite clad wall in the case of MEDIUM and HEAVY types and a metal clad wall for LIGHT type, both insulated with 10cm mineral fibre. External windows are all assumed to be clear float double glazing, internal partitions are lightweight plasterboard and the floor is carpeted.

The 3TC (3 Time Constant) [7] lumped parameter simulation method is used for the thermal simulations. In this method, the thermal response of each room has three time constants and rooms are modelled as networks of thermal conductance and capacitance. The 3TC algorithms are implemented within the design tool and allow the modelling of long term storage of heat in the building fabric.

There are four natural ventilation modes incorporated in the model;
- single sided single opening
- single sided double opening
- cross ventilation
- stack ventilation (buoyancy and wind)

The ventilation algorithms for the first mode are based on the work by Warren and Parkins [8] and the algorithms for the last three modes are based on those described in the CIBSE Applications Manual on Natural Ventilation [9]. The tool also uses a methodology developed under an ETSU sponsored project [10] which automatically sizes openings in order to achieve a user specific flow. This facility is available for all the four modes of natural ventilation.

The structure of the tool

The data describing the current state of the buildings design is contained on five tabs on the main form of the program. These are;

- Building,
- Weather,
- Day Ventilation,
- Night Ventilation and
- Control Strategies.

Building input parameters

There are only eight main variables which describe the building and are controlled by the user:

- Internal Gains
- Infiltration airflow
- Orientation
- Glazing ratio
- Building weight
- Occupied period
- Solar protection and shading coefficient
- Site location for the calculation of solar position.

Weather

Calculations can be carried out either over a month or over the entire cooling season (assumed to be May to September). Each month is assessed by simulating the building for 7 days and energy data is multiplied up using appropriate factors. Data for each day can be set separately in terms of the risk that a certain weather combination will be exceeded. By defining a risk of (say) 5%, the weather data is selected such that the weather will only be hotter than the calculated weather data for 5% of days in the particular month. This defines the maximum and minimum daily temperatures. Hourly data is generated from this by assuming a daily sinusoidal temperature series. Each month has a fixed temperature lag from noon.

The weather data is calculated using the banded weather data in the CIBSE Guide [11]. This is used because it provides coincident data for external dry bulb temperature, solar radiation data and mean daily windspeed, all of which are important in assessing room heat gain and ventilation cooling potential. More specifically, the weather data banded on temperature rather than that banded on solar radiation has been used. This is because periods of high temperature rather than high radiation are likely to present a greater design risk in buildings which are using ventilation cooling, since by their nature, they are likely to be well shaded.

A different risk factor can be selected for each day in the week. This enables the user to vary the sequence of weather to which the building is subject. This could be a sustained period of very hot weather or a cooler period followed by a warming-up period (Figure 1).
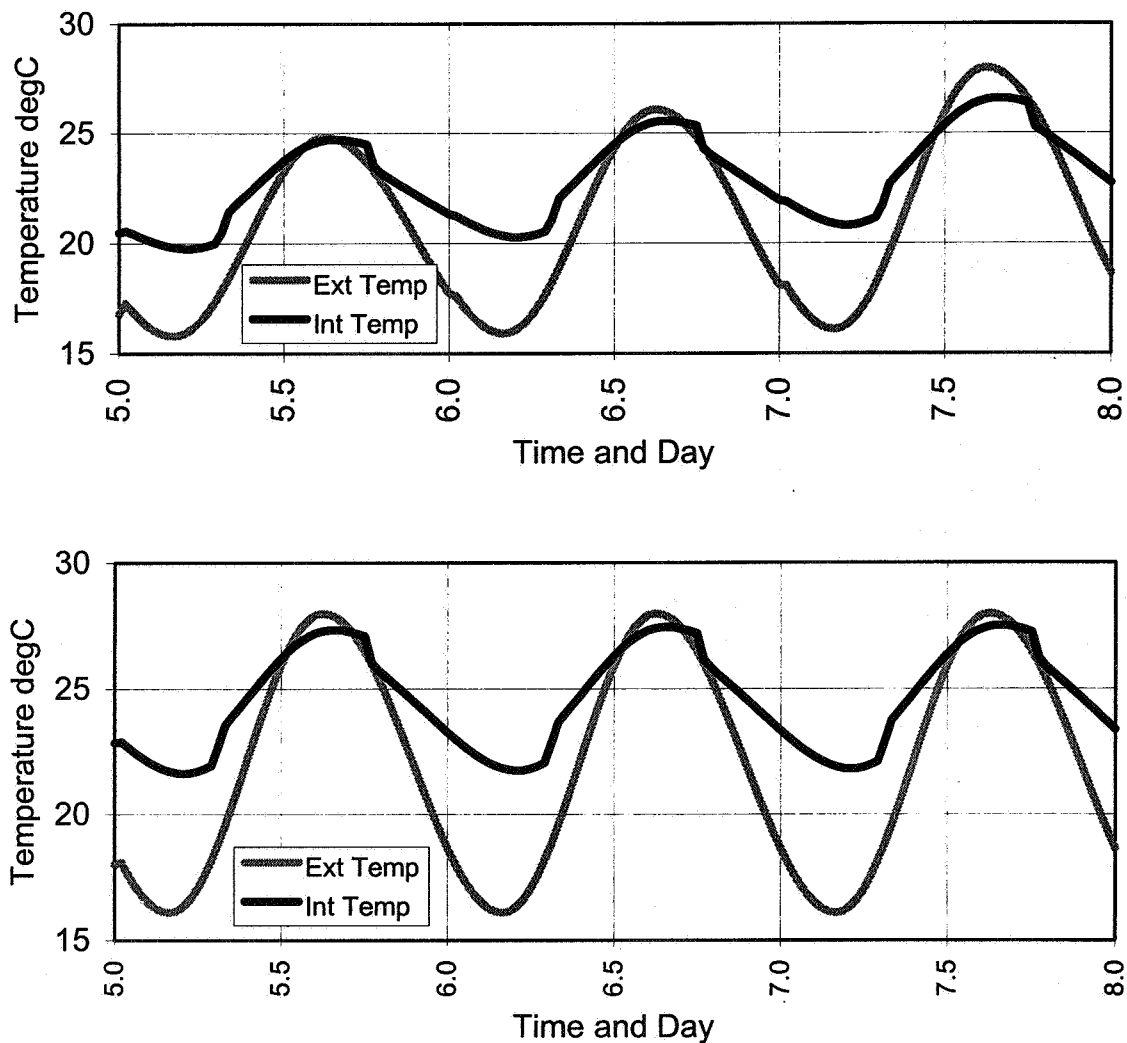
Figure 1: Shows the last three days of one week simulation to demonstrate the effect of external air temperature sequence on predicted internal dry resultant temperatures. A warming up period was chosen for the first graph. A risk factor of 20% was chosen for Saturday to Tuesday, 10% for Wednesday, 5% for Thursday and 2.5% for Friday. Internal temperature was predicted to be $26.6°C$ on Friday (last day). A risk factor of 2.5 % was selected for all week for the second graph. This resulted in a temperature of $27.5°C$ during the last day. Note that the peak temperature for the building exposed to gradually increasing external temperature (first graph) is almost 1K lower than that for a constant 2.5% risk.

Cooling systems and reference HVAC system

Two different systems may be used to cool the design building; one for day time and the other for night. Data for these is independent, and is contained on separate tabs but day occupied period cannot overlap with the night cooling period. There are nine different cooling systems incorporated in the program: three mechanical ventilation, four natural ventilation and two active cooling systems as follows:

<u>Mechanical Systems</u>
- Mechanical ventilation by supply fan
- Mechanical ventilation by extract fan
- Balanced mechanical ventilation by supply and extract fans

<u>Natural Ventilation Systems</u>
- Single Sided single opening
- Single Sided double opening
- Cross ventilation
- Stack ventilation

<u>Active Cooling Systems</u>
- Fan coil system
- Displacement ventilation system

It should be noted that mechanical systems are characterised by their Specific Fan Power and Coefficient of System Performance. The four natural ventilation systems have a design feature which allows the user to size and position openings to achieve a desired flow rate under design conditions.

Design Mode for Natural Ventilation Systems

Design mode can be used to size and position openings to achieve a design flow under design internal and external conditions. Outside temperature and wind speed are required along with the corresponding design inside air temperature. The program picks up as default, the inlet and outlet pressure coefficients from the Wind Pressure form. These defaults can be edited if required. The program then sizes the ventilation openings based on a methodology developed under an ETSU sponsored project [10] which is summarised in the CIBSE Applications Manual on Natural Ventilation [9].

A test mode is also available if the user knows the sizes of the openings; these can be specified and the flowrates calculated based on the design weather and internal air temperature. Other parameters (e.g. vertical separation of openings) can also be varied whilst keeping the opening sizes fixed in order to assess the impact on the flowrate through the room. Once the opening sizes have been finalised, these sizes can be copied to the system tab on the main form and the system is evaluated in the normal way.

Energy Savings Comparisons

One of the important features of the tool is the facility to compare the fuel used for the chosen design system to a reference HVAC system. The reference HVAC system is considered to be a standard solution to the design problem. No night ventilation is applied and the day cooling system defined on the main form is replaced with the system on the Reference HVAC form. Apart form the ventilation system, all other building and environmental parameters remain as for the design building. Two reference HVAC systems are available: (A) Displacement

ventilation system and (B) Fan coil system. There are two ways of defining the set point temperature of the reference HVAC system. The default option is to allow the program to automatically choose the set point such that the reference system provides a similar level of comfort to that maintained in the design building. The set point temperature is taken as the average temperature an hour either side of the peak temperature found in the design building. The second option is to enter a constant set point temperature.

As an example Tables 1-3 are provided tabulating the results of three different day systems using a range of night systems. The day ventilation rates are 4ACH for the fan coil system and natural ventilation systems and 3ACH for the displacement system. Night ventilation rates are 6ACH for all systems. Fan specific power is 0.75 for supply and 0.5 for extract. External temperatures are as specified for Figure 1A. Tables 1-3 show a range of potential energy savings or energy penalties when using mechanical ventilation at night. These values were predicted by specifying single values for delivered ventilation rates and fan specific power. In order to help the user to optimise the selection of the system parameters, the tool includes a parametric analysis facility. This is available for most of the user specified parameters in relation to the building, day and night ventilation systems. In the example demonstrated in Tables 1-3 the parametric analysis facility can be used to optimise the mechanical ventilation system by choosing appropriate ventilation rates and specific fan power values. Figure 2 shows four graphs as an example of such a parametric analysis.
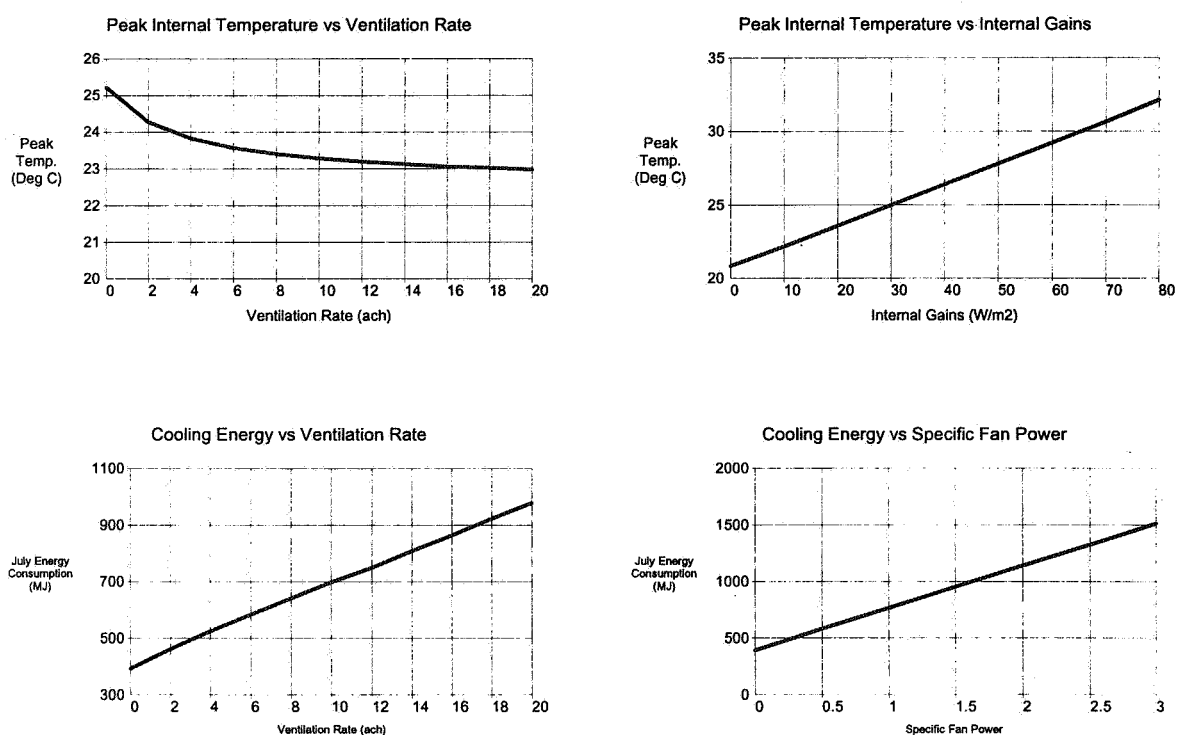
Figure 2: The effect of the delivered ventilation rate and the internal gains are shown in the first two graphs against the peak internal temperatures. The required cooling energy in relation to the delivered ventilation rate and specific fan power is shown in the next graphs.

Table 1: Day cooling is provided by a fan coil system using 100% fresh air. The same system is used as the reference HVAC system. It shows that night ventilation together with an active cooling system during the day provides the possibility of reduced cooling capacity and energy saving in some cases.

Night Cooling System	Maximum Internal Temperature °C	Energy Saving %	Cooling Capacity Saving %
No night system	24	0	0
Mechanical Supply	24	-8	22
Mechanical Extract	24	7	25
Mechanical Balanced	24	-33	22
NV cross	24	24	15
NV stack	24	28	20
Fan Coil	24	-46	19

Table 2: Day cooling is provided by a displacement ventilation system. A fan coil system with 100% fresh air is used as the reference HVAC system. It shows that night ventilation reduces the day peak temperature and achieve energy savings. It also shows that running the displacement system at night reduces day temperatures but the energy consumption is higher that the reference HVAC system.

Night Cooling System	Maximum Internal Temperature °C	Energy Saving %	Cooling Capacity Saving %
No night system	25.2	42	N/A
Mechanical Supply	23.8	5	N/A
Mechanical Extract	23.6	17	N/A
Mechanical Balanced	23.8	-20	N/A
NV cross	24.1	42	N/A
NV stack	23.9	42	N/A
Displacement Vent	24.2	-77	N/A

Table 3: Stack natural ventilation is utilised during the day. A fan coil system with 100% fresh air is used as the reference HVAC system. It shows that night ventilation reduces peak temperatures during the day by approximately 2°C.

Night Cooling System	Maximum Internal Temperature °C	Energy Saving %	Cooling Capacity Saving %
No night system	28.3	100	N/A
Mechanical Supply	26.6	62	N/A
Mechanical Extract	26.4	75	N/A
Mechanical Balanced	26.6	37	N/A
NV cross	27	100	N/A
NV stack	26.7	100	N/A

Control Strategies

Effective control of night cooling systems is important if the system is to operate in an energy efficient way and provide optimum comfort to the occupants. The Controls tab on the main form allows the user to customise the operation of the night cooling system, by controlling three aspects of the operation:

1) Operation times; ie the time to start and end the operation of night cooling system.
2) System initiation, i.e. the decision as to whether or not to initiate the night cooling each night. There are 4 control laws which can be selected for night cooling initiation:
   - Peak inside temperature during the previous occupied period must be greater than 23°C.
   - Average inside temperature during the previous day must be greater than 22°C.
   - Average outside air temperature during the previous afternoon (occupied period only) must be greater than 20°C.
   - Slab temperature at time of night cooling initiation must be greater than 23°C.
3) System continuation, i.e. if the system has been initiated, how long it should be on. Three control laws can be selected:
   - The current inside temperature is greater than the current outside temperature plus an offset of 2 degrees. This offset may be less for natural ventilation systems.
   - The current inside temperature is greater than the heating set point temperature (18°C). This is to avoid overcooling the space and having to heat in the morning to bring it up to acceptable comfort levels.
   - The current outside air temperature is greater than 12°C. Again this is to avoid overcooling the space.

All the above rules have been the result of an extensive study which included modelling and practical experience from monitored buildings [12].

Results and Parametric analysis

The following summary results are displayed in the main form of the tool:
1. Maximum and minimum dry resultant temperatures; which occur while the building is occupied over the analysis period.
2. Energy Saving; which is calculated as the percentage saving in cooling energy provided by the design system over the reference system.
3. Cooling Capacity Saving; which is calculated as the percentage saving in installed cooling capacity in the design building over that required for the building fitted with the reference system.
4. A graph button shows the variation of the dry resultant temperature with time over the last three days of the analysis.

In addition, the following graphical results can be obtained:
1. Parametric analysis graphs are available for all the important input variables (as demonstrated in Figure 2). These show the variation of internal comfort temperature, cooling energy or peak cooling capacity with the selected parameter.
2. Graph of dry resultant temperature and outside air temperature with time (Figure 1).

3. Solar with time shows the incident and transmitted radiation on the glazing plane in W/m^2.
4. Graph of cooling power and fuel use with time (Figure 3).
5. Graph of ventilation with time (Figure 4)
6. Temperature frequency distribution graphs.

As an example Figures 3 and 4 are presented where stack natural ventilation is used at night and mechanical extract at 2ACH during the day. The external temperatures are as specified for Figure 1A. The building characteristic are $20W/m^2$ internal heat gains, 0.5 ACH infiltration, south orientation, heavy construction, 0.4 glazing ratio, 8.00-18.00 occupancy hours and solar protection with 0.2 shading coefficient.
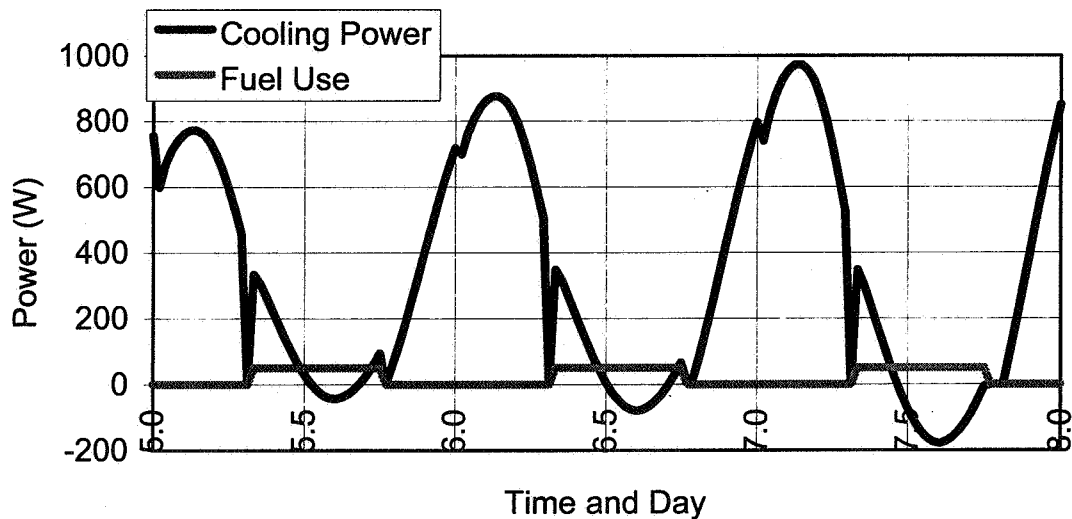
Figure 3: Cooling power and fuel use with time. The performance of the cooling systems can be assessed by observing the relationship between the two lines.
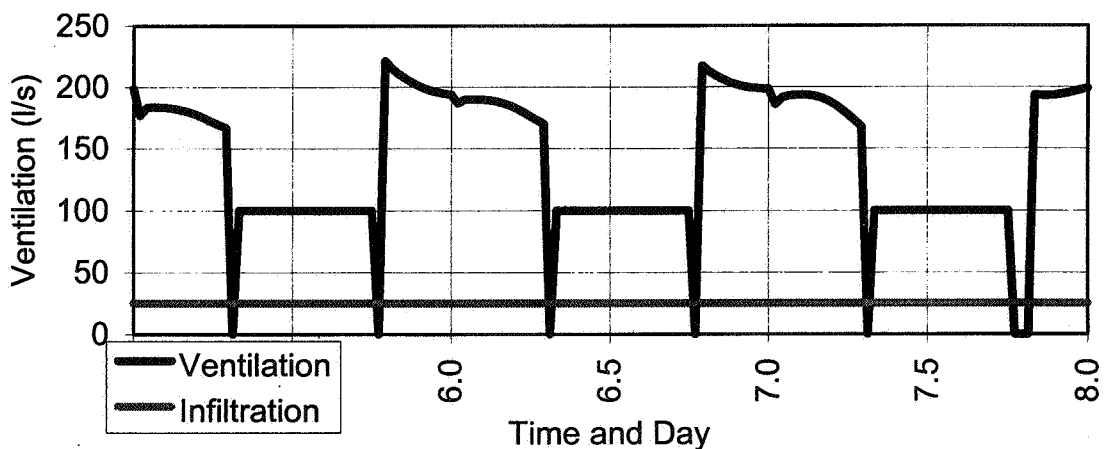
Figure 4: Graph of ventilation with time shows both the ventilation due to the cooling systems and that due to infiltration. This data may be useful in ensuring that sufficient fresh air is being provided.

Conclusions

This paper describes a pre-design tool developed especially for the UK office building and climate for assessing a range of night cooling ventilation strategies. The program is based on a single zone ventilation model, a given office configuration and the 3TC thermal simulation model. It is an easy to use pre-design tool with very fast simulation time which allows quick comparisons between three mechanical ventilation systems, four natural ventilation systems and two active cooling systems in terms of temperature, fuel consumption and cooling power. In the case of natural ventilation systems, the program has the facility of automatically sizing openings in order to achieve a user specified flow. The program also allows the user to investigate very quickly the effect of different external temperature sequences on the internal dry resultant temperature. It also includes a parametric analysis facility which can be used for design optimisation. Written in Visual Basic, the program has a user friendly interface, with a minimum number of input parameters and fast simulation time, all of which will allow its use by designers at the initial design stage.

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Title: On the ventilation and daylight efficiency of various solar shading devices

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ON THE VENTILATION AND DAYLIGHT EFFICIENCY OF VARIOUS SOLAR SHADING DEVICES

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Synopsis

Solar control devices placed in front of large building openings disturb air flow and the radiation transfer. Although solar radiation transfer through obstructed openings is a relatively well researched area, very little information is available regarding the air flow perturbations and daylighting alterations created by external solar control devices. The present paper reports a series of experiments aiming at investigating natural ventilation and daylight phenomena associated with the use of specific shading devices. Experiments have been carried out in outdoor test cells and twenty eight different configurations have been tested for several window characteristics under various climatic and radiation characteristics. Based on the experimental results, specific modeling activities have been undertaken and theoretical methods of calculating air flow and daylight through openings equipped with specific solar control devices have been developed and are now presented. Theoretical predictions are compared with the corresponding experimental data and a very satisfactory agreement has been found for both air flow and daylight processes.

List of symbols

$C(t)$ gas concentration at time t (ppm)
λ air changes per time unit
L_k is the luminance of the kth sky patch (cd/m^2)
a_k is the solid angle of the kth sky patch (sterad)
d_k is daylight coefficient of the kth sky patch ()
ρ density (kgr/m^3)
Q air flow rate (m^3/sec)
C_d discharge coefficient

A area of the opening (m^2)
Ar_D Archimedes number
Gr Grashof number
Re Reynolds number
g acceleration of gravity (m/sec^2)
H opening's height (m)
D depth of the test room (m)
T temperature (^{0}K)
U air speed (m/sec)

1. Introduction

Buildings are one of the most important energy consuming sectors. In Europe, buildings represent almost forty percent of its global energy consumption. A very high percentage of the consumed energy is used to cover both the cooling and the lighting needs of buildings as well. The shading effects of fixed and movable shading devices have been extensively studied, and very accurate models for their performance have been developed under dynamic or steady state conditions [1]. However, the impact of solar control devices on the airflow through obstructed openings is relatively poorly researched, while the existing knowledge on the air flow regime as well as on the

combined air flow and radiation transfer through openings equipped with solar control devices, is very limited [2]. In reality, phenomena of air flow through large openings are of a random nature and this because of the wind characteristics. Thus the proposed models should be designed to allow calculation of the air flow through the obstructed openings when the main climatic and geometrical characteristics are known.

Modelling of the performance of the louvers, as it concerns daylighting, can be performed by using empirical or flux transfer methods, [3], radiosity based methods, [4], or ray tracing techniques, [5]. Radiosity based models can treat only diffuse surfaces and present problems with highly reflected materials. Furthermore due to the need of accurate calculation of the form factors a high memory capacity is required in order to take into account detailed scenes such as an aperture equipped with blinds. Ray tracing techniques solve the rendering equation, [6], under most conditions including specular and diffuse reflection and transmittance in complicated curved geometries. However, as internal illumination has to be calculated in a dynamic way to take into account sky variability, these models require a very high computational effort and this because intereflection calculations have to be performed for each time step. Thus, there is a need for models that can calculate, in an accurate way, illuminance levels in complicated geometrical environments without to repeat time consuming intereflection calculations at every time step.

This paper presents the results of a series of 28 different experimental configurations where various shading devices have been tested in a real scale outdoor facility. It also presents appropriate theoretical models to evaluate air flow and daylight transfer through the studied solar control devices.

2. The Experimental Set Up

In order to expand existing knowledge on the physical phenomena related to the impact of movable solar control devices associated with large building openings a series of experiments have been carried out in a PASSYS test cell, [7]. Particularly, the experiments aimed to evaluate processes related to the air flow and daylight transfer through the openings under single side ventilation configurations. The PASSYS test cell is a fully equipped, two zones, outdoor facility for thermal and solar monitoring. Wall temperature, internal and external air temperature, internal and external (diffuse and global) illuminance, wind speed and direction were measured continuously in 2 min intervals. During the experiments and in order to estimate the luminance distribution of the sky, a luminance camera (Minolta LS-101) has been used to perform sky scanning. This procedure was based on the pattern recommended by the CIE [8], on 145 patches in 12 degree bands of altitude, centered in azimuth on the solar azimuth. The south facade of the cell is removable and allows installation and testing of specific building components. Experiments were carried out in the "test room" while the door connecting it to the "service room" was kept closed and sealed. The test room has a height of 2.72 m and a volume of 35 cubic meters.

To study the impact of the opening surface especially on the air flow through the obstructed window, a specific component has been constructed and attached to the removable south facade of the cell. The component covers the whole facade of the cell and has a transparent surface - opening of 4 m^2 located at the center of the facade. The maximum height and the width of the opening are equal to two meters. The opened area

of the component is adjustable in order to perform experiments under various opening surfaces. Two type of shading devices have been tested and in particular, movable vertical and horizontal louvers. The louvers are made from metallic sheets having a mat white finish of 0.1 m in width and 2 m in length. Experiments have been performed for various tilt angles of the louvers and various opening areas. Table 1 present the main characteristics of the experiments.

No	Area of the Opening, m^2	Type of Shading Device	Tilt of Louvers
1	0.5	Horizontal louvers	0^0
2	1	Horizontal louvers	0^0
3	2	Horizontal louvers	0^0
4	4	Horizontal louvers	0^0
5	0.5	Horizontal louvers	60^0
6	1	Horizontal louvers	60^0
7	2	Horizontal louvers	60^0
8	4	Horizontal louvers	60^0
9	0.5	Horizontal louvers	30^0
10	1	Horizontal louvers	30^0
11	2	Horizontal louvers	30^0
12	4	Horizontal louvers	30^0
13	0.5	Vertical louvers	45^0 east
14	1	Vertical louvers	45^0 east
15	2	Vertical louvers	45^0 east
16	4	Vertical louvers	45^0 east
17	0.5	Vertical louvers	0^0
18	1	Vertical louvers	0^0
19	2	Vertical louvers	0^0
20	4	Vertical louvers	0^0
21	0.5	Vertical louvers	45^0 west
22	1	Vertical louvers	45^0 west
23	2	Vertical louvers	45^0 west
24	4	Vertical louvers	45^0 west
25	0.5	No Shading	
26	1	No Shading	
27	2	No Shading	
28	4	No Shading	

Table 1. Main geometrical characteristics of the experiments.

Ventilation experiments were performed using a single tracer gas decay method. N_2O was used as tracer gas. Injection and sampling points were carefully chosen in order to achieve the necessary homogeneity. The sampling period was set to 30 sec. Tracer gas concentration was measured by an i.r. gas analyzer.

During the experiments N_2O was injected in the room, while the exterior opening was sealed. Small fans were used to enhance good mixing of the gas during the injection period. When the gas concentration reached the required levels and mixing was satisfactory, fans were turned off, the injection stopped and the window opened.

According to the decay method the decrease of the tracer gas concentration is given by the following equation:

$$C(t)=C(t0) \exp(-\lambda t) \qquad (1)$$

where $C(t)$ and $C(t0)$ are the tracer gas concentrations at time t and at $t=0$, respectively. The air changes per hour have been calculated for each sampling point and then the mean value for the whole room was calculated as the mean of all sampling points.

3. Daylight Results and Modeling

The developed method uses the Monte Carlo approach, a kind of backward ray-tracing in order to calculate the daylight coefficients. Daylight coefficient [9] is defined as the ratio between the luminance of a patch of sky and the illuminance in the building due to light from that patch.

The sky can be divided into zones of altitude and azimuth and the daylight coefficient can be found at each zone. Then total illuminance, at one point in a room, can be calculated using the following equation:

$$\text{Illuminance} = \sum_{k=1}^{\text{number of sky patches}} Lk \ ak \ dk \qquad (2)$$

where L_k is sky luminance, a_k is the subtended size of a sky patch and d_k is the daylight coefficient.

Following the daylight coefficients approach, the interreflection calculation is carried out once for each zone, and it doesn't have to be repeated if the sky luminance distribution changes. The advantage of this approach is that hourly calculations of interior lighting in a building, for a whole year, can be performed faster without repeating interreflection calculations. Additionally, because the sky is treated as a number of point sources, the contribution of direct and reflected sunlight in the interior lighting could easily be assessed by adding, to the sky zone where the sun is located, an additional luminance equal to the normal solar illuminance divided by the solid angle of the zone. Each emitted ray has an initial weight equal to 1. After each reflection or transmission the ray weight is multiplied by the corresponding reflectance or transmitance of the surface. If the resulting value is larger than a predefined threshold value the whole procedure is repeated; in the opposite case, the ray is considered to be absorbed. This developed method can deal with a large variety of reflection models. Particularly, the method considers specular, diffuse, specular and diffuse, reflective and diffusing glass. The model considers the ground as a separate surface with reflectance equal to 0.2. For all rays that do intersect with the ground a single ground reflected daylight coefficient is calculated.

The accuracy of the developed simulation method is affected by : a) the number of initial emitted rays, where the standard error regarding the estimation of the illuminance is inversely proportional to the number of emitted rays, and b) the limit value of ray-weight. This last variable affects the interreflection component of illuminance. Lower value contributes to higher interreflection values.

Twenty five thousand initial rays have been used by the present model, while the threshold ray weight has been considered equal to 0.1 The ratio of this threshold value to the average reflectance of the area is equal to the mean number of light bounces. For our experiments, the area weighted reflectance of the test cell was approximately equal to 0.4 thus, four light bounces have to be expected in average.

After the quality control tests twenty one data sets measured during June 1996 and September 1996 have been considered. Then for each specific experiment, simulations have been performed using the developed model. Finally, the calculated and measured data have been compared for both indoor measurement points.

Mean differences between predicted and measured values are close to 13 % for the first measuring point and to 18 % for the second one. Higher differences are observed for

point two because the local illuminance is reduced as louvers reduce the area of visible sky. Consequently, intereflections have a much higher contribution to the horizontal illuminance at points away from the window, and this is a source of computational error. Differences are smaller when no shading devices are used. In this case there is no redirection of light by the shading devices and thus intereflections are not important. Maximum errors are observed in configurations combining small surface openings and shading devices. The error of internal and external illuminance measurements is estimated close to 10%. The main source of error is the procedure of estimating the sky luminance distribution. As already mentioned, the time period for a complete sky scanning was 15 minutes. In order to reduce the error, two measurements of zenith luminance have been performed one at the beginning and the other one at the end of a scanning session. When the variance between both values was higher than 25% the data set was rejected. The scanner's acceptance angle was 1^0 so each point measurement is assigned practically to a sky patch, offering a small sky coverage. Errors associated with the model are of a random nature. The initial number of emitted rays is used to sample a continuous environment, resulting in an underestimation of the predicted illuminance. Furthermore, the number of the used light bounces is limited in comparison to infinite light bounces of real world. Increasing the number of rays and decreasing the threshold ray weight, predictions are improved, but the computational time increases.

4. Ventilation Results and Air Flow Modeling

Calculation of the air flow through large building openings can be achieved either by using empirical, network or computerized fluid dynamic models, [10]. Empirical models are in general of local validity, while do not take into account effects related to solar control devices. Computerized fluid dynamic, CFD, models are powerful tools, but are not so suitable for natural ventilation configurations, [11]. CFD tools are extremely sensitive to the initial and boundary conditions, which in natural ventilation studies are of a random nature and not well known.

Network prediction models are based on the mass flow balance of a space

$$\sum_{i_m=1}^{j_m} \rho_{im} \cdot Q_{im} = 0 \qquad (3)$$

and combine the effect of wind and buoyancy to calculate pressure differences. Network models have been proved to provide very reasonable predictions of the air flow levels in natural ventilation configurations, [12]. In equation (3), Q_{im} is the volumetric flow rate through the ith flow path of the mth node and ρ_{im} is the density of air flow through the ith flow path of the mth node (kg/m3).

The air flow through an opening is calculated by using the standard orifice equation :

$$Q = C_d \cdot A \cdot \sqrt{\frac{2 \cdot |\Delta P|}{\rho}} \qquad \text{(m}^3\text{/sec)} \qquad (4)$$

where Cd is the discharge coefficient of the opening, A is the flow area, ΔP is the pressure difference across the opening and ρ is the air density.

For large non obstructed openings, discharge coefficient varies between 0.6 and 1. An extensive review of the existing methodologies to calculate the discharge coefficient of large non obstructed internal openings is given by Santamouris et al. [13]. However,

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no methodologies have been developed to calculate discharge coefficients for large openings equipped with solar control devices.

A network air flow model, developed especially for natural ventilation configurations, [14], has been used to calculate the air flow for all the studied configurations. All calculations have been carried out using a discharge coefficient equal to one.

As it was expected the agreement between the experimental values and the results given by conventional network air flow model was not satisfactory. This conclusion agrees with results given by Dascalaki et al, [15], for single sided natural ventilation experiments carried out under quite high wind speeds. As found predictions can be highly improved if the relative importance of the inertia and gravitational forces is considered appropriately. It is found that the prediction error is proportional to the Archimedes number. Based on this, a new parameter CF has been proposed to be used to correct predictions of the network models, [15]:

CF = Measured Air Flow / Predicted Air Flow.

In the above mentioned paper it was found that CF can be expressed as an expotential function of the Archimedes number (Ar$_D$). Based on the above analysis an attempt has been made to study whether the observed differences between experimental and predicted values can be correlated with the Archimedes number:

$$Ar_D = \frac{Gr}{Re^2} = \frac{g \cdot H^3 \cdot \Delta T}{T \cdot U^2 \cdot D^2} \qquad (5)$$

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Figure 1 shows CF values versus Archimedes number in the case of horizontal louvers with 30 degrees tilt.

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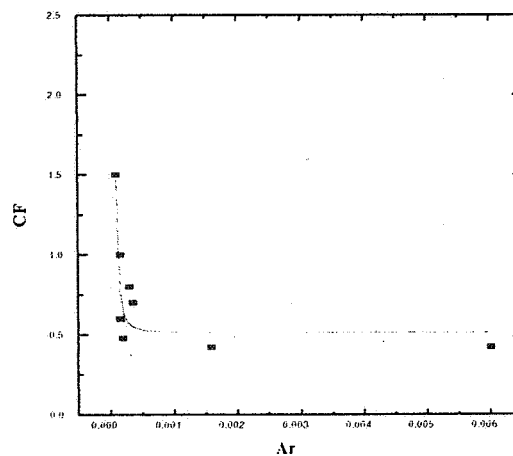
Fig.1 Measured/Predicted values of air flow rates versus Archmedes number

As shown, there is a very good agreement between the ratio of the measured to the predicted air flow (CF) and the Archimedes number. Based on the obtained results expressions to estimate CF, as a function of the Archimedes number have been obtained. For the case of horizontal louvers, it is found that Lorenzian type equations fit better the results :

$$CF = (2\ A1\ w1/3.14)/(w1^2 + 4(Ar - x_c)^2) + y_0 \qquad (6)$$

The obtained coefficients for equation 6 and for the various tilt angles are given in Table 2. The obtained correlation coefficients are given as well.

Shading	A1	w1	x_c	y_0	Correlation Coefficient
No	4.25E-4	4.48E-5	-1.24E-5	0.88	0.97
30 degrees	9.4E-4	1.75E-5	4.44E-5	0.53	0.72
60 degrees	9.39E-4	2.48E-4	-1.92E-4	0.39	0.64
0 degrees	6.33E-4	1.44E-4	-7.15E-5	0.56	0.94

Table 2 : Coefficients of the Lorenzian function and coefficient of determination for the cases where horizontal louvers were installed.

For vertical fins power functions fitted better the obtained results :

$$CF = A2\ Ar^{-B} \qquad (7)$$

The corresponding parameters A and B as well as the correlation coefficients for the various tilt angles are given in Table 3.

Shading: Vertical louvers	A2	B	Correlation Coefficient
45 degrees east	0.066	0.31	0.81
0 degrees	0.204	0.2	0.84
45 degrees west	0.06	0.3	0.74

Table 3. Coefficients for equation 7.

The obtained CF values have been used to recalculate the air flow for all the experiments. The corresponding correlation coefficients between the predicted and measured values when the CF coefficient is used, are given in Table 4.

Shading	Correlation coefficients
No shading	0.99
Horizontal louvers 30 degrees	0.98
Horizontal louvers 60 degrees	0.97
Horizontal louvers 0 degrees	0.9
Vertical louvers 45 degrees east	0.99
Vertical louvers 0 degrees	0.98
Vertical louvers 45 degrees west	0.96

Table 4. Correlation coefficients between measured and predicted air flow rates when the CF factor is used.

As shown the use of the appropriate CF coefficients considerably improves the accuracy of network models in predicting the air flow rate through large opening equipped with movable shading devices.

The developed algorithms are valid for the specific solar control devices and for the range of climatic parameters under which the experiments have been carried out.

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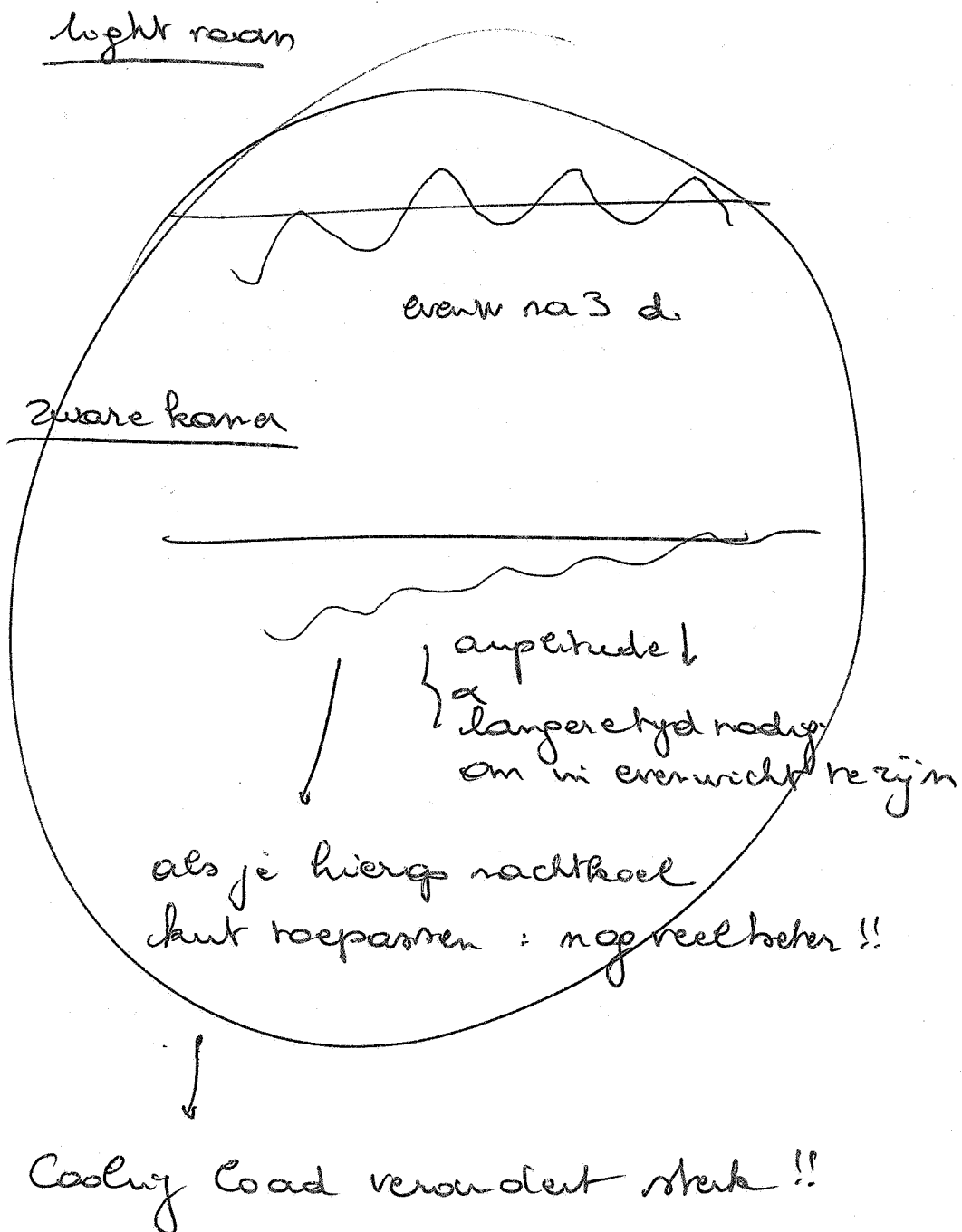
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VENTILATION AND COOLING
18TH ANNUAL AIVC CONFERENCE
ATHENS GREECE, 23 - 26 SEPTEMBER, 1997

Title

**Thermal analysis of rooms with diurnal periodic heat gain, ThermSim.
Part 2: Practical use and comparison**

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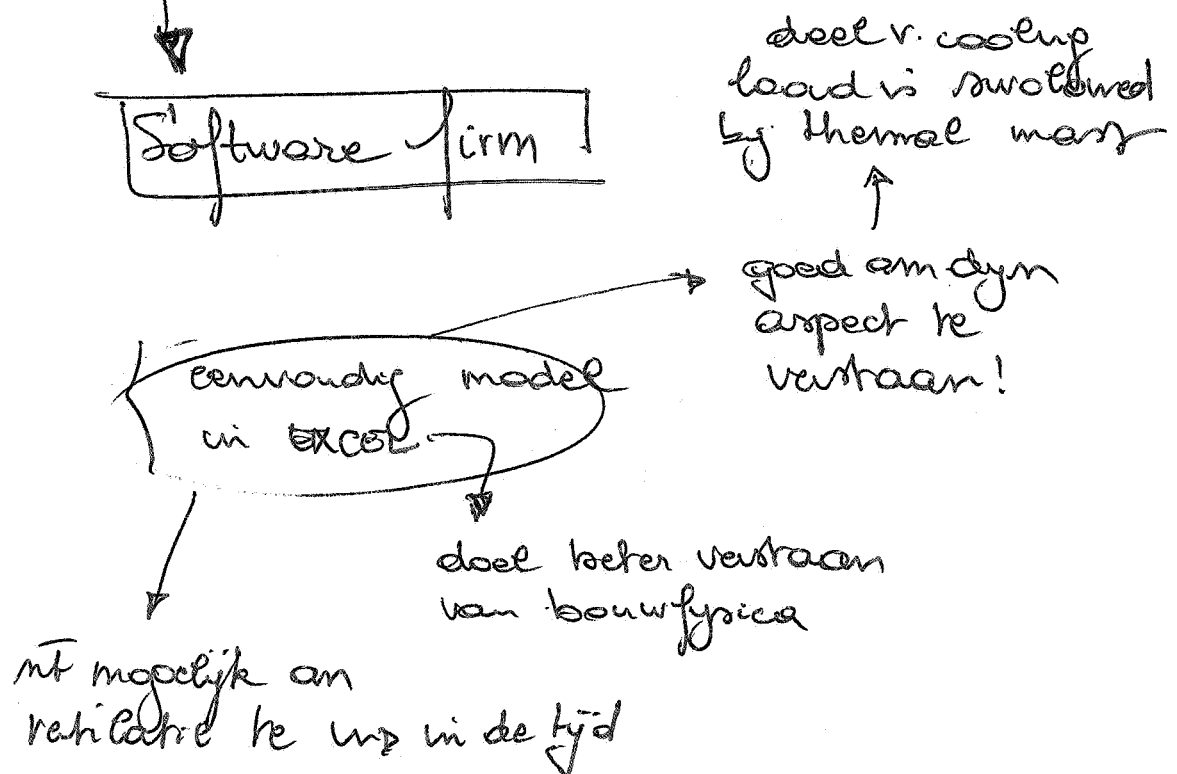
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Thermal analysis of rooms with diurnal periodic heat gain, ThermSim. Part 2 : Practical use and comparison

Synopsis

Temperature and cooling demand in a room summertime are influenced by numerous factors, like internal gains, ventilation, solar gain, behaviour of occupants, thermal inertia of the room and outdoor conditions (climate).

The thermal environment and cooling demand summertime are often analysed using detailed computer programs, which take into account the factors mentioned above (among others). Often the overview, transparency and some of the physical insight is lost using these advanced computer programs.

In a predesign phase of a project it is preferable to do simple calculations of the thermal behaviour of a room. These simple calculations often gives more physical insight and overview than using computer programs. Simple calculations also gives a quality assurance of later computer analysis of the room.

This is part 2 of two related papers concerning a simplified method for thermal design of rooms, called *ThermSim*. Part 1 (the accompanying paper) is concerned with derivation and interpretation of the model.

This paper is concerned with practical guidance in choosing appropriate input to the model. Comparison to the advanced simulation program BRIS is also presented.

The model shows good agreement with computer analysis when the model assumptions is fulfilled.

List of symbols

Symbol	Description	Unit
A_{fac}	Facade area	m^2
A_{floor}	Floor area	m^2
A_{win}	Area for whole window construction (including frame)	m^2
C_{air}	Heat capacity of air (can be set to 0.34 Wh/m^{3}K)	Wh/m^{3}K
F_{sh}	Effective total shading factor	-
L''	Normalized mechanical or natural air flow rate	m^3/hm^2
n	Air infiltration in ACH	1/h
n_{per}	Occupation time for persons	h
$n_{l\&a}$	Operation time for lighting and appliances	h
q_{per}	Heat gain from persons	W
$q_{l\&a}$	Heat gain from lighting and appliances	W
q''_{sol}	Solar intensity through a vertical pane	W/m^2
Q''_{sol}	Daily sum of solar gain through a vertical pane	Wh/m^2
$\overline{T}_e$	Mean daily external temperature	°C

$\hat{T}_e$	Daily amplitude external temperature	°C
ΔT_{fan}	Temperature rise over the supply fan	K
U_{fac}	U-value facade construction	W/m²K
U_{win}	U-value window construction	W/m²K
V	Air volume	m³

1 Introduction

This is part 2 of two related papers concerning a simplified method for thermal design of rooms. Part 1 (the accompanying paper) presents a method/model, called *ThermSim*, that can be used in the thermal design of rooms.

This paper is concerned with practical guidance in choosing appropriate input to the model. In addition this paper gives examples in the use of the model, and comparison to advanced computer simulation.

2 Guidance in choosing appropriate input data

This sections gives guidance in selecting appropriate input to the model. Most of the input can be "normalized" by dividing the value with the floor area. E.g. internal loads in Watt can be normalized into Watt per m² floor area (W/m²), which gives a much smaller range for the value. With this approach it is possible to make tables where normalized input values can be picked from, making the calculation process quick, and reducing the chance for calculation errors.

Values presented in the tables below are related to "Scandinavian" building standards, and may not be representative for other countries and other climates. It should however be easy to modify the tables to other standards and climates.

To exemplify how values in the tables have been determined, an office room is used as a case study throughout the section.

2.1 Normalized specific external loss
The specific loss to the external is comprised of window losses, facade losses and loss due to infiltration. In modern (Scandinavian) buildings the external walls are well insulated, and heat loss through windows are dominating. In less airtight buildings (older buildings) might infiltration have some impact on the external loss.

Given an office 3 x 4 m (12 m²) large, with a facade area of 9 m² (incl. window) and a window area of 2 m². U-value for the facade construction is 0.25 W/m²K (15 cm mineral wool) and the U-value for the window is 2 W/m²K. Infiltration is estimated to 0.3 ACH (Ceiling height is 3 m). Normalized specific external loss is then given by :

$$H''_{ext} = \frac{U_{fac}A_{fac} + U_{win}A_{win} + C_{air}nV}{A_{floor}} = \frac{(9-2) \cdot 0.25 + 2 \cdot 2 + 0.34 \cdot 0.3 \cdot 36}{12} = 0.785 \ W/m^2K$$

This figure is quite typical in office rooms, it normally lies between 0.4 W/m²K and 1.2 W/m²K. Table 1 presents typical values of normalized specific external loss, as a function of normalized windows- and facade loss, and infiltration (in ACH).

Table 1 : Specific external loss

Wind&Facad/ Infiltration	None	0.2W/m²K Low	0.5 W/m²K Medium	1.0 W/m²K High	3.0 W/m²K Very high
n = 0.1 ACH (Low)	0.1	0.3	0.6	1.1	3.1
n = 0.2 ACH	0.2	0.4	0.7	1.2	3.2
n = 0.3 ACH (Med)	0.3	0.5	0.8	1.3	3.3
n = 0.5 ACH	0.5	0.7	1.0	1.5	3.5
n = 0.7 ACH	0.7	0.9	1.2	1.7	3.7
n = 1.0 ACH (High)	1.0	1.2	1.5	3	4.0
n = 1.3 ACH	1.3	1.5	1.8	2.3	4.3

2.2 Normalized total specific loss

The total specific loss is the sum of the specific external loss and the ventilation loss. Air flow rate is often given in m³/h per m² floor area (normalized air flow), which is convenient here.

The room in subsection 2.1 is ventilated (balanced mechanical vent.) with 10 m³/hm² (120 m³/h). The total specific loss is the given by :

$$H''_{tot} = H''_{ext} + C_{air}L''_{vent} = 0.785 + 0.34 \cdot 10 = 4.185 \, W / m^2 K$$

Table 2 gives normalized total specific loss as a function of ventilation rate and normalized specific external loss.

Table 2 : Normalized total specific loss

External loss/ Ventilation rate	None	0.5W/m²K Low	1.0 W/m²K Medium	2.0 W/m²K High	4.0 W/m²K Very high
0 m³/hm²	0	0.5	1.0	2.0	4.0
3 m³/hm² (Low)	1.0	1.5	2.0	3.0	5.0
5 m³/hm²	1.7	2.2	2.7	3.7	5.7
8 m³/hm² (Medium)	2.7	3.2	3.7	4.7	6.7
11 m³/hm²	3.7	4.2	4.7	5.7	7.7
15 m³/hm² (High)	5.0	5.5	6.0	7.0	9.0

2.3 Normalized specific heat capacity, timeconstant and time-lag

The effective heat capacity of the room can be treated in the same manner as the specific losses. The effective heat capacity of a building construction exposed to a 24 hours cycle temperature variation, can be limited to the inside 10 cm of the construction, or inside the insulating layer. If heavy material as concrete or brick is covered with insulating materials (i.e. carpet or lowered ceiling), the accumulating layer is reduced considerably. These "rules" gives specific (per m²) heat capacity of : ~ 50 Wh/m²K for a massive concrete wall, ~ 35 Wh/m²K for a massive brick wall, ~ 4 Wh/m²K for a insulated composite wall with gypsum board or wood panelling, ~15- 25 Wh/m²K for a concrete slab covered with carpet or lowered ceiling.

Given the room in section 2.1 with concrete floors covered with carpet, mineral wool lowered ceiling (beneath concrete construction) and brick walls in facade and partition walls. The normalized heat capacity of the room can be calculated to :

$$C_a'' = \frac{C_{air}V + \sum C_a''A}{A_{floor}} = \frac{0.34 \cdot 36 + 12 \cdot 20 + 12 \cdot 20 + (9-2) \cdot 35 + (3+4+4) \cdot 3 \cdot 35}{12} = 158 \ W/m^2K$$

With the normalized heat capacity and normalized total specific loss, the timeconstant and time-lag can be readily calculated :

$$\tau = \frac{C_a''}{H_{tot}''} \qquad \tau_{lag} = \frac{\arctan[\tau\omega]}{\omega}$$

Table 3 gives timeconstant and time-lag values as a function of normalized heat capacity and total specific loss.

Table 3 : Timeconstant/time-lag

Total specific loss/ Normalized heat capacity	1.0 W/m²K Low	3.0 W/m²K	5.0 W/m²K Medium	7.0 W/m²K	9.0 W/m²K High
20 Wh/m²K (Very light)	20/5.3	7/4.1	4/3.1	3/2.5	2/1.8
40 Wh/m²K (Light)	40/5.6	13/4.9	8/4.3	6/3.8	4/3.1
80 Wh/m²K (medium)	80/5.8	27/5.5	16/5.1	11/4.7	9/4.5
140 Wh/m²K (Heavy)	140/5.9	47/5.7	28/5.5	20/5.3	16/5.1
260 Wh/m²K (Very heavy)	260/5.9	87	52/5.7	37/5.6	29/5.5

Example : Total specific loss : 3.0 W/m²K and specific heat capacity : 80, gives a timeconstant of 27 hours and a time-lag equal to 5 hours and 30 minutes.

2.4 Normalized internal load and solar gain

Heat gain from persons, light and appliances is often normalized with the floor area. In addition to the maximum instantaneous heat gain (to determine the amplitude heat gain), we have to estimate the diurnal mean heat gain. If balanced mechanical ventilation is used, we also have to estimate the heat gain from the supply fans.

The room in subsection 2.1 is occupied by one person (gain : 100 W) 8 hours a day. Lighting (120 W) and a computer (50 W) gives a mean heat gain of 170 W, and both are operated 8 hours a day. The supply fan rise the supply air flow 1 Kelvin (the fans are operated 24 hours a day). The normalized heat gain amplitude related to the internal load is then given by :

$$\hat{q}_{int} = \frac{q_{per} + q_{l\&a}}{2 \cdot A_{floor}} = \frac{100 + 170}{2 \cdot 12} = 11.25 \ W/m^2$$

The normalized mean heat gain related to the internal load becomes :

$$\bar{q}_{int} = \frac{q_{per}n_{per} + q_{l\&a}n_{l\&a}}{24 \cdot A_{floor}} + C_{air}L_{vent}''\Delta T_{fan} = \frac{100 \cdot 8 + 170 \cdot 8}{24 \cdot 12} + 0.34 \cdot 10 \cdot 1 = 10.9 \ W/m^2$$

Table 4 gives normalized amplitude heat gain and mean heat gain related to internal loads. It is given as a function of persons per 10 m² floor area (a normal office room) and the normalized gain from lighting and appliances. Heat gain from supply fan is included in the figures (air flow 10 m³/hm² and temperature rise 1 K). Operation of lighting and appliances is assumed to be 10 hours, and effective occupation time is set to 6 hours.

Table 4 : Normalized heat gain amplitude/mean heat gain

Lighting&Applianc./ Person density	5 W/m² Low	7 W/m²	10 W/m² Normal	15 W/m²	25 W/m² High
0.5 pers/10 m² (low)	5/7	6/8	8/9	10/11	15/15
1 pers/10 m² (office)	8/8	9/9	10/10	13/12	18/16
1.5 pers/10 m²	10/9	11/10	13/11	15/13	20/18
2 pers/10 m²	13/10	14/11	15/13	18/15	23/19
3 pers/10 m² (meet.room)	18/13	19/14	20/15	23/17	28/21
5 pers/10 m² (high)	28/18	29/19	30/20	33/22	38/26

Example : Normal lighting and appliances (10 W/m²) and 2 persons per 10 m², gives a amplitude heat gain of 15 W/m² and a mean heat gain of 13 W/m²

2.5 Specific solar gain

According to ,\1\, the maximum solar intensity through a vertical pane on a clear day can be approximated to 700 W/m². This figure can be used for facades facing East through South to West. Daily sum for the same facades can be approximated to 4700 Wh/m².

Solar shading in form of venetian blinds, curtains and building extensions can reduce the solar gain considerably. The shading effect for these shading devices is often taken as a constant shading factor. Typical values are 0.1 - 0.25 for external venetian blinds, 0.3 - 0.7 for inside venetian blinds and 0.5 - 0.8 for curtains. These values are for two pane windows with no coating, and has to be adjusted if low emessivity coating, reflective coating, absorbing glass or more panes are used. Shading from building extensions and nearby vegetation or buildings has to be estimated from case to case.

In the room from subsection 2.1, the facade is facing south, and there is inside venetian blinds with a shading factor of 0.5. The normalized amplitude gain can be estimated to :

$$\hat{q}_{sol} = \frac{q''_{sol} A_{win} F_{sh}}{2 \cdot A_{floor}} = \frac{700 \cdot 2 \cdot 0.5}{2 \cdot 12} = 29 \ W/m^2$$

The normalized mean solar gain can be estimated to :

$$\bar{q}_{sol} = \frac{Q''_{sol} A_{win} F_{sh}}{24 \cdot A_{floor}} = \frac{4700 \cdot 2 \cdot 0.5}{24 \cdot 12} = 16 \ W/m^2$$

Table 5 gives normalized solar gain (amplitude and mean) as a function of window area per m² floor area and total shading factor. Values are valid for facades facing east to west.

Table 5 : Normalized solar gain amplitude/mean heat gain (W/m²)

Window area pr m² / Total shading factor	0.05 m²/m² Low	0.10 m²/m²	0.2 m²/m² Normal	0.3 m²/m²	0.5 m²/m² High
0.85	15/8	30/16	60/33	89/48	149/80
0.75	13/7	26/14	52/29	79/42	131/70
0.55	10/5	19/10	39/22	58/30	96/50
0.40	7/4	14/8	28/16	42/24	70/40
0.25	4/2	9/5	18/10	26/15	44/25
0.10	2/1	4/2	7/4	11/6	18/10

· **Example** : With normal window area (0.2 m²/m²) and a total shading factor of 0.25, gives amplitude solar gain of 18 W/m² and a mean heat gain of 10 W/m²

Total heat gain (daily mean and amplitude) is the sum of the internal gain and solar gain.

2.6 Climatic data

In addition to the maximum solar intensity and daily solar gain treated in the previous subsection, we need to estimate the mean external temperature and its daily variation (amplitude). We also have to estimate the time for maximum heat gain (and external temperature).

The mean external temperature is normally found in meteorological journals. E.g. the highest five day mean temperature for the location in question could be used. This has to be evaluated against the use of the room from case to case.

The external temperature amplitude in Scandinavian climate varies between 5 - 7°C. If accurate information is not available a value of 6 °C can be used. If maximum external temperature isn't corresponding with the maximum heat gain, the temperature amplitude can be reduced a bit (0.5 - 2 °C).

If solar gain is dominating (compared to internal gains), which is one of the main assumptions in the model, the time for maximum heat gain is determined by the facade/window orientation. With daylight saving time (in Oslo) maximum solar gain is occurring : between 12.00 and 13.00 for south facades, between 9.00 and 10.00 for east facades, and between 17.00 and 18.00 for west facades. For other countries adjustment for time zone, longitude and daylight saving time has to be done.

3 Case study; comparison

A room which has been used in validation analysis of a computer program (TeknoSim), will be used as case study here, and compared to results from the widely used simulation program BRIS ,\2\. The room has width , depth and height equal to : 3.6 m x 4.2 m x 2.7 m (A_{floor} = 15.12 m², V = 40.82 m³) . The room has one facade, facing west, with one window (A_{win} = 1.92 m², U_{win} = 2.0 W/m²K). Infiltration is 0.2 ACH, and the room is ventilated continuously with 72 m³/h (4.8 m³/hm²). The room is occupied with one person (9 hours a day), and heat gain from lighting and computer is 270 W (9 hours a day). The supply fan rises the supply temperature 1 °C.

Two different building constructions has been simulated : one heavy room with concrete floor, ceiling and external wall; and one light room with insulated composite construction covered with gypsum boards or particle boards. Partition walls are insulated composite walls with gypsum board in both cases.

Calculation

Normalized total specific loss is calculated to : $H''_{tot} = 2.18$ W/m²K (both cases). Normalized heat capacity for the two cases are calculated to : $C''_{a,h} = 31.2$ Wh/m²K (light room) and $C''_{a,h} = 140.6$ Wh/m²K (heavy room). Timeconstant and time-lag for the light- and heavy room then become : $\tau_{light} = 14.3$ hours, $\tau_{lag,light} = 5$ hours (light) and $\tau_{heavy} = 64.6$ hours, $\tau_{lag,heavy} = 5.8$ hours (heavy). Normalized mean heat gain and heat gain amplitude is respectively : $\bar{q} = 20.4$ W/m² and $\hat{q} = 31.0$ W/m². Mean external temperature and temperature amplitude are respectively : $\overline{T}_e = 22$ °C and $\hat{T}_e = 6.5$ °C . This gives a stationary temperature of : $T_\infty = 31.7$ °C (both cases), and a temperature amplitude for the light and heavy room of respectively : $\hat{T}_e = 1.2$ °C (light) and $\hat{T}_e = 5.3$ °C (heavy). Transient temperature differences are calculated to : $\Delta T_{light} = -10.8$ °C (light) and $\Delta T_{heavy} = -9.9$ °C (heavy). Both cases are simulated for a period of five days.

Simulated operative temperature the fifth simulation day in BRIS is shown in figure 1 (light room) and figure 2 (heavy room), and is compared to calculation with ThermSim (fifth day). The operative temperature in BRIS is used for the comparison, since the calculated temperature in ThermSim is a "merged" room-, surface- and structure temperature.
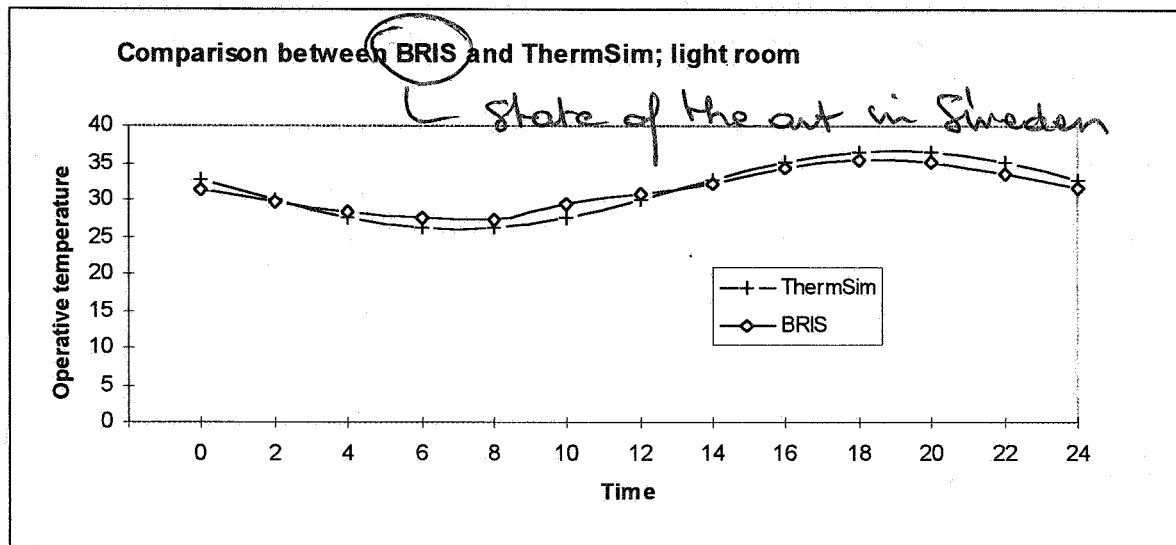
Figure 1 : Comparison between simulation in the advanced computer program BRIS and calculation with ThermSim, light room
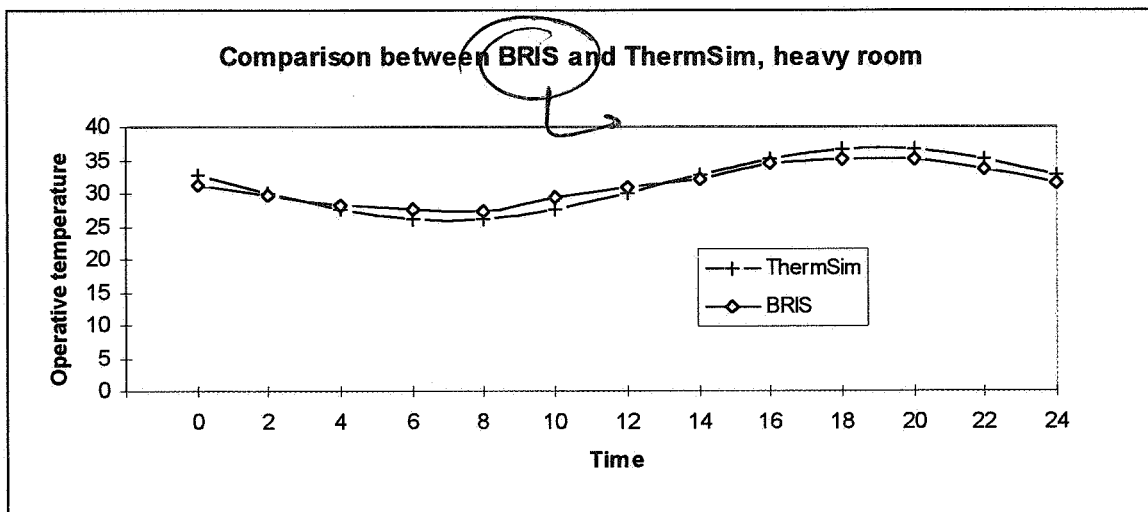
Figure 2 : Comparison between simulation in the advanced computer program BRIS and calculation with ThermSim, heavy room

5 Discussion and conclusions

- Temperature variation simulated with BRIS and calculated with ThermSim is similar, for both the light and heavy room
- Maximum temperature is somewhat higher calculated with ThermSim compared to BRIS (1.2 °C for light room and 1.7°C for heavy room).
- Maximum temperature occur a bit later in ThermSim than in BRIS ((1-2 hours in both rooms). This implicate that the calculated time-lag in ThermSim overestimate the "real" time-lag.
- Diurnal stationary conditions is reached after 5 days in the light room, but far from reached the heavy room (both with BRIS and ThermSim).
- Comparison between ThermSim and BRIS shows good agreement, and ThermSim should therefore be well suited for thermal analysis in a predesign phase of a project
- The simulations and calculations shows that large heat capacity reduce the daily temperature variation to a large extent, and prevent stationary condition being reached during a normal heat wave or a normal working week
- ThermSim is very well fitted for sensitivity analysis, because it only deals with the most important parameters affecting the thermal conditions in the room

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VENTILATION AND COOLING
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Title: **Controlled Air Flow Inlets**

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INTRODUCTION

Within the EU project NATVENT, which deals with the application of natural ventilation in office type buildings, one of the items to be studied was controlled air flow inlets. Natural air supply is a key part in the design of natural ventilation in offices. In case these air supplies are designed in the wrong way one may expect complaints in terms of draft and stuffiness. Size and controls on inlets are vital elements in design. Controlled air inlets may help to overcome the problems of draft and stuffiness, and may contribute to an energy efficient design of the building.

Several types of control can be considered such as: pressure control, humidity control, pollutant control and temperature control.

AVAILABILITY

Just a few of these innovative developments are available on the European market. Especially France have produced this type of inlets already for many years. One may say that France is really on the fore front in designing as well as in the application of these product. Because of the climate the Scandinavian countries have focused on control by outside temperature. Over the last few years in the Netherlands and France a number of new developments are going on.

The difference in philosophy of the countries involved on ventilation and the way the requirements are described in the building regulations in these countries have a large influence on the design and performance of the available controlled air inlets.

Some examples may explain the situation:

- in France air inlets may not be fully closed, while in The Netherlands they must be completely closable.
- in The Netherlands the controllability of these inlets are must be between 1 and 25 Pa, where in France the control may be at about 20 Pa.

These conflicting requirements don't miss their effect on the design of the controlled inlets.

PERFORMANCES

Pressure controlled inlets

The objective of these inlets can be expressed as a constant natural supply air flow independent of wind pressure and pressure differences due to buoyancy

Big differences are found in the capacity or sizes of the inlets available on the European market. As mentioned earlier the reason for this is mainly the difference in the requirements in the building regulations. As an example for the same size of room the difference in the capacity of air inlets expressed in cm^2 is in:

- France $20\,cm^2$
- UK $40\,cm^2$
- Belgium $70\,cm^2$
- the Netherlands $100\,cm^2$

As can be seen these differences are far from negligible, up to a factor of five! The difference in response pressure is also remarkable. The pressure at which air flow rate has reached an almost constant level differs from about 1 to 20 Pa. For the normal building environment the pressure differences across air inlets normally are in the range of 0 Pa to 50 Pa.
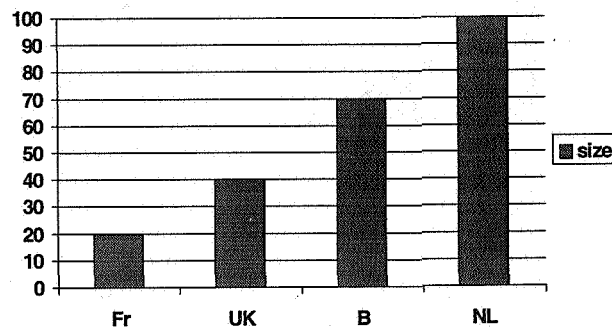
Figure 1 Relative size of inlets

The response time of the control of inlets is also quite different. Some inlets respond almost immediately while others react only after a few minutes.

Examples of passive inlets
An example of a pressure controlled inlet which acts in the way as described above can be found in figure 2.
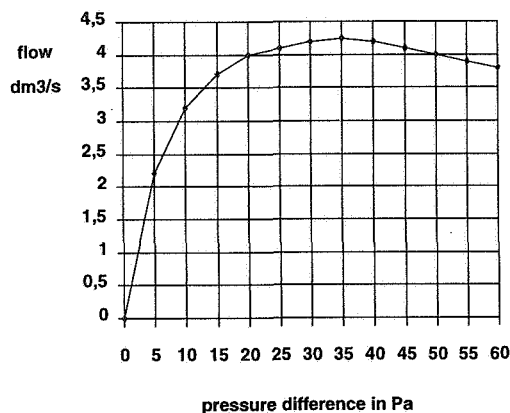
Figure 2 Characteristic of a pressure controlled inlet from France

A passive pressure controlled inlet which is developed by TNO and patented by a Dutch firm Compri is showed in figure 3.

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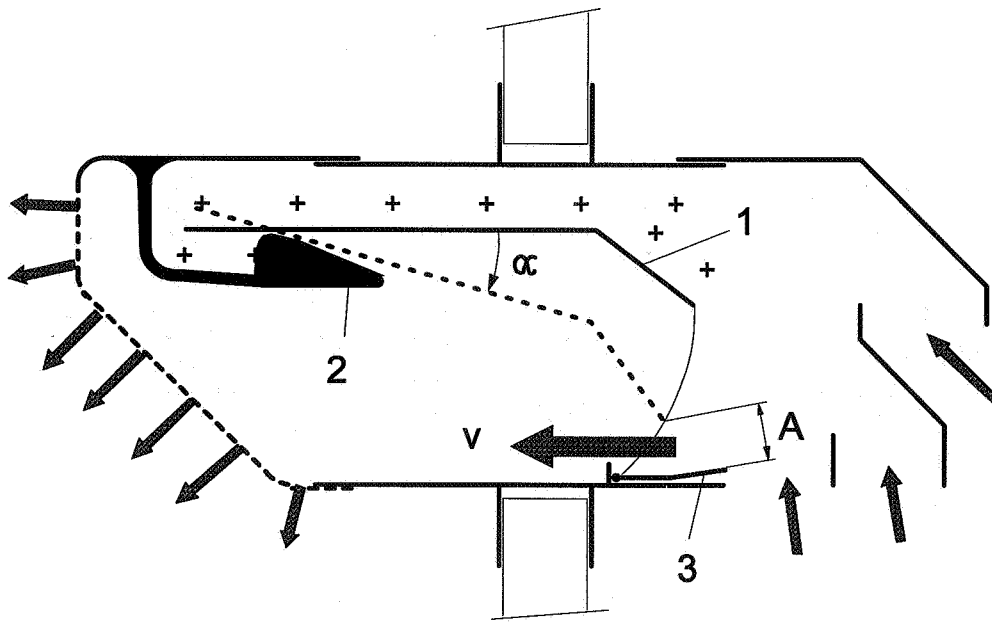
Figure 3 Cross section a passive pressure controlled inlet COMPRI IAQ

A performance curve of a pressure controlled inlet at 1 Pa can be seen in figure 7.
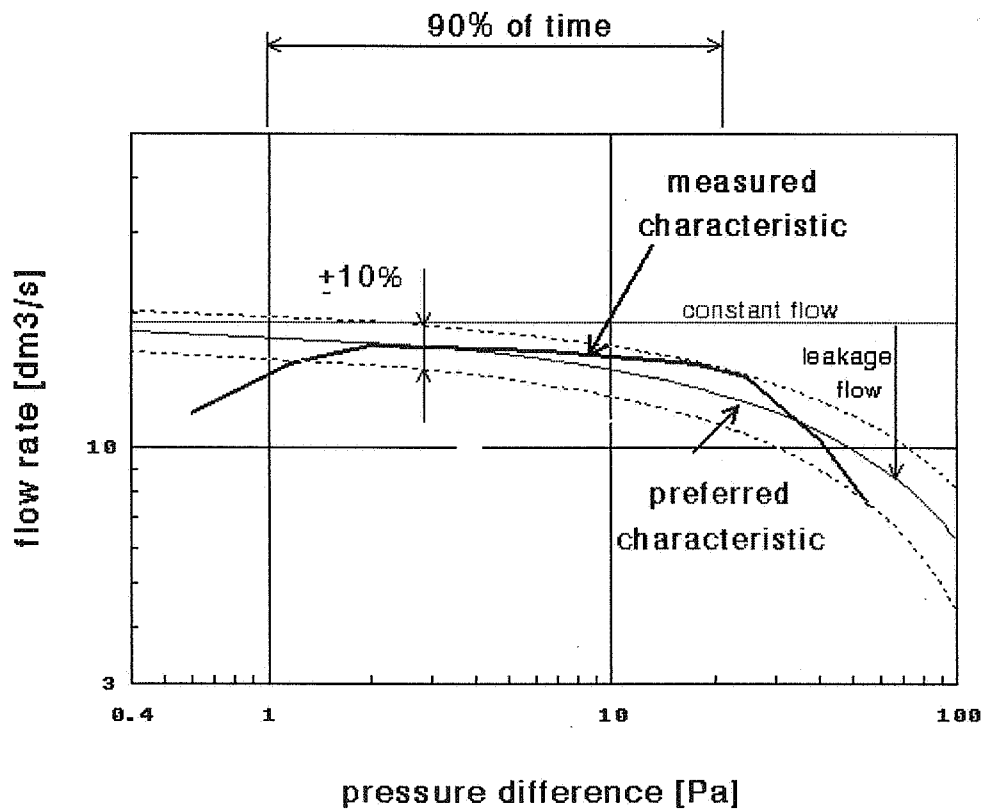
Figure 4 An example of the performance of a passive pressure controlled inlet from the Netherlands

Active inlets

There is also an active controlled air inlet on the market. In this inlet a device measures the pressure difference across it. With the help of a small motor the grid of the inlet is controlled. An example of this kind of inlet from the Netherlands is given in figure 5.
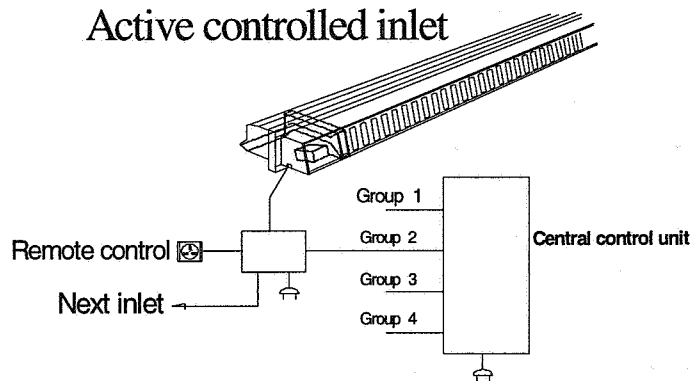
The advantage of this active type of control is of course the possible connecting with a building management system. This allows one to overrule the local control and open or close all the inlets in the building centrally.

Figure 5 An example of an active controlled air inlet

The disadvantage of this active inlets are price, about four times the passive ones and the fact that one needs electric power at each of the inlets. The performance of the active inlet can be seen in figure 6.
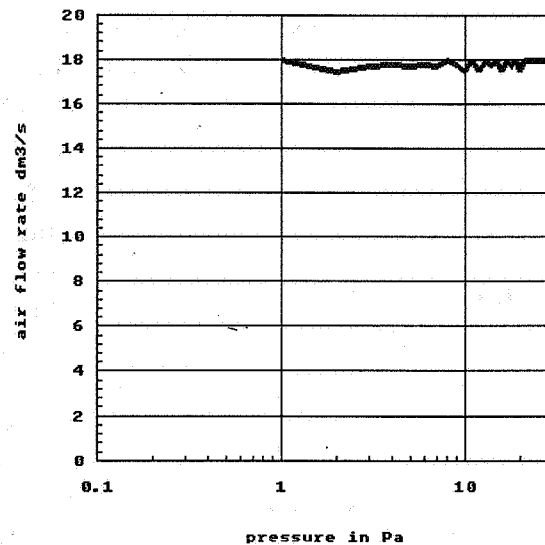
Figure 6 The performance of an active controlled inlet

Humidity controlled inlets

Most of the humidity inlets act on the relative humidity of the air. The calibration of the available inlets is quite reproducible and accurate. The principle of these type of inlets is mostly based on the change in length of a tape. An example of the characteristic of an inlets from France is given in figure 7.
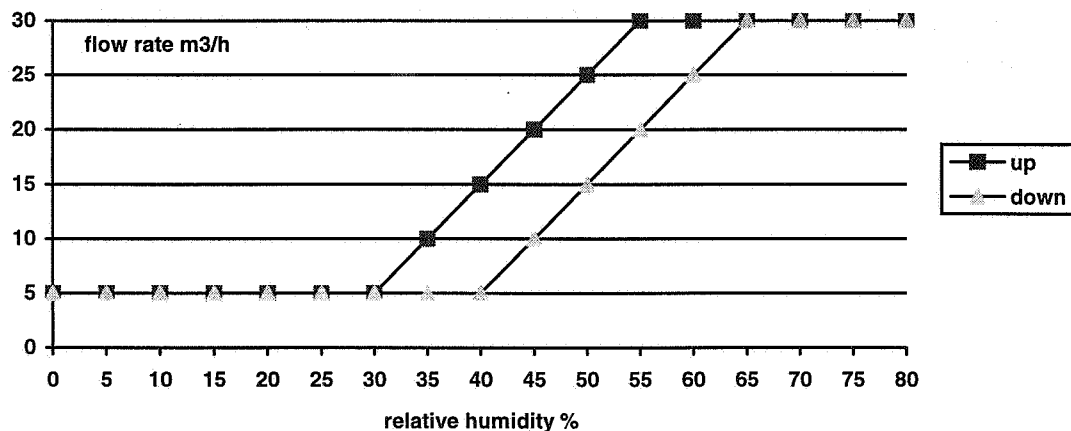
Figure 7 An example of the characteristic of an humidity controlled inlet

Pollutant controlled inlets

There are just a few pollutant controlled inlets available. In the Netherlands there is one manufacturer who claims to have a system where the pollutant level is measured with so called mixed gas sensors. The signal from this sensor can control the extract fan and the inlets with electronic controls. Calibration of the sensor seem to be a problem for which there is not yet a solution.

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Temperature controlled inlets

Temperature controlled inlets are on the market in countries where the outside temperature is more dependent for the driving force for ventilation. An example of the characteristic of a temperature controlled air inlet from Sweden is given in figure 8.

flowrate constant 4 dm3/s
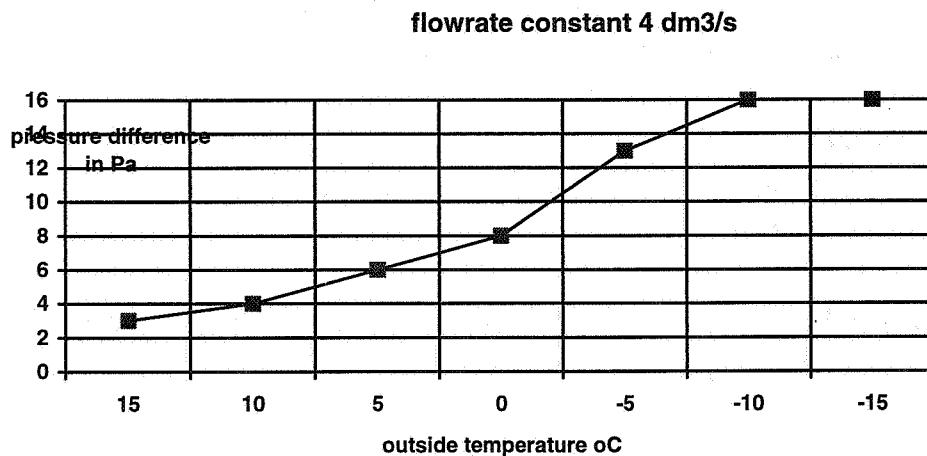
Figure 8 An example of the characteristic of a temperature controlled inlet
The lower the outside temperature the larger the pressure difference. The control of the inlet is a bi-metal sensor which by bending closes the inlet. Unclear is how these inlets are calibrated. Most of these inlets also have a considerable so called dead time and hysteresis.

APPLICATION OF CONTROLLED INLETS

General

The application of most of these controlled inlets are in dwellings. But even in dwellings there are only applied in a relative small part of the dwelling building stock. A rough estimate for the whole of Europe is less than a few percent of all dwellings have a kind of controlled inlet.

Due to the so called "green or sustainable design" of buildings a promising future market exists.

The building regulations and standards on ventilation throughout Europe don't push the application of controlled inlets.

Pressure

These inlets are applied mostly in dwellings, but in a minority of the dwelling building stock.

The application is sometimes in dwellings with indoor air quality problems. To overcome the problem people decide to try the application of pressure controlled inlets.

One of the barriers to apply them is the price of the inlets. Roughly spoken the passive inlets cost about three times a normal inlet. The application therefore will be in the more expensive dwelling types.

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Again price is a barrier because the price of these active inlets is ten times that of a normal one. Nevertheless the future looks very optimistic. In a number of plans at the moment architects are considering the application of pressure controlled inlets.

A potential market are offices where comfort and energy savings are the driving factors.

Some people claim that in comparison with a full mechanical system, the system with pressure controlled inlets and mechanical extraction is about one third of the cost of a full mechanical system.

Humidity and/or moisture

The application is mostly in the so called wet rooms in dwellings. In France an enormous amount of these inlets are applied in apartments. The moisture removal and control can be considered as good.

But in most rooms in buildings moisture or humidity is not the determining pollutant.

In the case of an office type building the application may not be obvious.

Pollutants

Pollutant controlled inlets are rarely available on the market. Nevertheless it is good to realise that pollutant controlled ventilation is applied in several types of buildings, such as schools, theatres, shopping malls, congress halls, parking lots etc.

CO_2 , CO and smoke control is mostly applied. This application is normally focusing on the control of the extract. In some cases the principle of a non selective sensor is applied. If these sensors will be more selective and sensitive, the possibilities of inlet control will have new chances.

Temperature

Although the application up till now is not very widely spread, the opportunities in the colder climates are quite good. The price normally is not a barrier. In relation to the prevention of draft problems and energy savings a wider application is possible in climates where thermal buoyancy is the dominant driving force. So in colder climates and high rise buildings the application of temperature controlled inlets are feasible. A good marketing and promotion plan is a necessity.

CONTROLLED INLETS IN VENTILATION SYSTEMS

General

Controlled inlets can be integrated with all types of ventilation systems. The application with passive extract is of course from the standpoint of promoting natural ventilation the most obvious one. The integration in a system with cross ventilation is however also possible. Relatively easily is the integration with mechanical exhaust systems. In cases the natural driving force is failing the mechanical extract system guaranties the minimum required flow through the building. The distribution may not be right but the flow at least is guarantied.

In studying the integration the following aspect can play an important role:

- indoor climate, mainly the prevention of draft problems,
- indoor air quality, mainly related to CO_2 levels,

- energy savings, mainly due to a better control

Indoor climate

Controlled air inlets have a big advantage above normal inlets. Because of the control on the flow rate the velocity of the jet entering the room normally will be under control even at high wind velocities. Inlets with fixed openings must be controlled manually by the occupant when the wind velocities increase to a level at which draft may occur. The most common reaction by people is to close the inlet instead of switching it to another control position. Fixed inlets therefore causes more draft problems then pressure controlled inlets.

A normal inlet with a fixed but controllable opening needs more occupant interaction which normally don't take place. Hence a part of the time the office is occupied the flow rate will be to high and unnecessary energy losses due to ventilation will occur.

Indoor air quality

A multi-zone model study carried out in the Netherlands shows the effect of pressure controlled inlets in combination with central mechanical exhaust at design level in an office under average weather conditions.
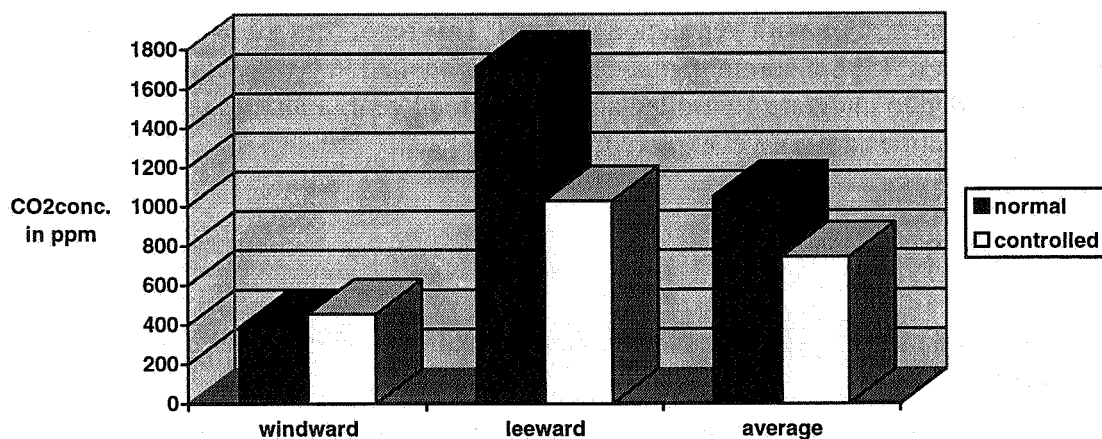
Figure 9 Increase of CO_2 concentrations above the outside level in office rooms with normal and controlled inlets in combination with mechanical exhaust.

The results given in the figure above are results under average weather conditions! The normal inlets are simulated to be controlled by the occupant in an almost perfect way. Both temperature differences and wind speed effects the position of the inlet.

It can be clearly seen that the pressure controlled inlets have a significant improvement on the CO_2 concentrations on the leeward sided office rooms.

IMPROVEMENTS IN CONTROLLED INLETS

Although the available controlled inlets can already be applied more widely than up till nowadays a number of improvements can be foreseen.

The most important ones are listed below:

- presence control
- interaction with building energy management systems
- sound attenuation
- air cleaning or filtering
- development of controlled inlets for ducted systems
- better integration in ventilation systems.

With regard to presence control simple technical solutions like infra red sensors can be easily applied to the already existing inlets. This option is important to decrease the ventilation heat. Interaction with building energy management systems is important because these systems can overrule local control if necessary. This can be of importance for energy as well as for safety for instance in case of fire or other hazard.

Natural inlets are is sometimes not applied because of high outdoor traffic noise levels. The barrier of noise can be taken in most situations with sound attenuation. Sound attenuated air inlets air widely available. It may be a good improvement on controlled inlets. Special attention is needed for ideas on controlled inlets for ducted systems.

In case of heat recovery a ducted system is necessary. A combination of the two leads to very energy efficient ventilation systems.

The last item is integration with ventilation systems. This report describes a number of solutions which are still not common practice but only applied in the special cases. More attention is needed to integrate controlled air inlets in total ventilation system design. A number of possibilities are not yet explored.

CONCLUSIONS

Controlled air inlets are available on the market but mainly applied up till now on dwelling type buildings.

The size and dimensions vary enormous because of local requirements throughout Europe.

Several types of control are available : pressure control, humidity control, pollutant control and temperature control.

As it stands now pressure controlled and temperature controlled inlets seems to have the most practical chances to be applied more in office type of buildings.

The performance of the controlled inlets are not always of that level that application in offices is obvious.

Some of the pressure controlled inlets are however very promising in terms of indoor air quality, comfort and energy.

Price is still a barrier to the wide application of these inlets.

Pressure controlled inlets improve the ventilation of offices. Controlled inlets can give significant better indoor air quality, better comfort and at the same time a lower energy use for ventilation.

A number of possible features as improvements on the existing ones can be easily foreseen.

The most important however is a better promotion of the lately developed controlled inlets. The products are there, the market is there, but the application still is far behind.

Demonstration project to show the real world that these inlets are vital parts of natural ventilation systems are necessary.

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VENTILATION AND COOLING

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The Significance of Urban Pollution and its dilution associated with height

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The significance of urban pollution and its dilution associated with height

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SYNOPSIS

This paper identifies the significance of pollution at five sites amongst the worst on the British mainland, hence indicative of other polluted areas within Europe. Three sites are located in London and one each in Birmingham and Cardiff. The pollutants examined are NO_2, SO_2, O_3 and PM_{10}. Newly proposed DOE figures defining poor air quality have been used to re-examine the frequency of excess pollution episodes between 1992 - 1995. The results identify the most appropriate periods for natural ventilation of offices in urban areas in terms of the hour in a day and time of year. Preliminary in-situ experiments also demonstrate that both PM_{10} and NO_2 concentrations decrease with increasing height from a busy road, and that this could be a useful strategy for reducing the impact of contaminants derived from vehicle emissions.

1. INTRODUCTION

The aim of this paper is to suggest ways in which barriers to natural ventilation may be overcome. The study forms part of the Pan-European project titled NatVent involving seven countries; the UK Building Research Establishment (BRE) are the co-ordinators. A widely perceived barrier to effective natural ventilation in urban areas is the ingress of external pollutants. Filtration of air supply is an option but may unacceptably restrict the flow of air into buildings. A possible solution is to vary ventilation periods to avoid 'polluted' occasions.

A general perception exists that pollution at roof level is likely to be less than by the road side, but there appears to be little quantitative evidence for this. Since the siting of offices varies substantially in height it is useful to examine how height influences the concentration of mainly traffic generated pollutants. If a dilution effect is evident it may be possible to recommend a suitable height for positioning air inlets in buildings to avoid or reduce the need for filtration. In section 3 of this paper the concentrations of PM_{10}, and NO_2 are examined at varying heights from a busy road.

2. THE EXTENT OF URBAN POLLUTION IN CITIES

2.1 Selection Of sample sites

A recent European report examined the urban pollution problem across Europe, using seven cities sited as representative "of their air quality, geographical location and characteristics of their vehicle fleet", (1). The locations chosen were, London, The Hague, Cologne, Lyon, Milan, Athens and Madrid. Of these cities London, The Hague and Cologne are located within an area of Europe covered by the NatVent

project, namely cold - temperate climatic regions. In each city a model was used to predict the reduction in pollution emissions necessary to achieve required standards. London, by inference, was the most polluted city and thus is a typical worst case North European area.

A study of urban pollution in the UK has been conducted for some time. Periods of poor air quality have been identified and published (2). An examination of poor air quality episodes reveals that PM_{10} and NO_2 have been major problems at a number of locations. SO_2 and O_3 concentrations have also presented problems, but to a lesser extent. The worst areas for PM_{10} excess pollution levels during 1992 - 1995 were, London Bloomsbury, Birmingham City Centre and Cardiff City Centre. During the same period NO_2 excess levels were a major problem in Cromwell Road and Bridge Place, both situated in London. Hence an assessment of air quality at these locations is a useful guide to the situation in similar regions of northern Europe. Additional information on each of the sites is provided below.

London, Bridge Place:
- Urban background site.
- Based on 2nd floor in a street near a busy area in Victoria.

London, Cromwell Road:
- Kerbside site.
- Based in central London at busy arterial road.
- High traffic density of approximately 60,000 vehicles per day.

London, Bloomsbury:
- Urban Centre site.
- Based at Russel Square 35m from Kerbside.

Birmingham City Centre:
- Kerbside site.
- Based on the busy Stratford road where gradient is 1:20.
- Traffic density is approximately 20,000 vehicles per day.

Cardiff City Centre:
- Kerbside site.
- Based on the busy Queen Street.
- Traffic density is approximately 30,000 vehicles per day.

2.2 Evaluation method

The pollutants reviewed at each location are highlighted in Table 1.

Table 1.   Pollutants reviewed between 1992 - 1995, at five UK sites

Site	NO_2	PM_{10}	SO_2	O_3
Bridge Place, London	√	-	√	√
Cromwell Rd, London	√	-	√	-
Bloomsbury, London	√	√	√	√
Birmingham City Centre	√	√	√	√
Cardiff City Centre	√	√	√	√

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The Department of the Environment (DoE) has published a 'consultation draft' outlining desirable maximum levels in ambient concentrations of pollutants to be achieved by 2005 (3). Table 2 identifies some of the pollutants of concern and the concentrations that must not be exceeded (unless figures are otherwise revised) by 2005. It also indicates some of the health problems attributed to each pollutant.

Table 2. **Proposed standards and specific objectives**

	Pollutant			
	NO_2	PM_{10}	SO_2	O_3
standard limits sample mean times	104.6 ppb 1 hour	50 μgm^{-3} 24 hours*	100 ppb 15 minutes	50 ppb 8 hours*
health problems	-respiratory	-respiratory &, cardiovascular	-respiratory & chest pains	-respiratory

The concentration values expressed in Table 2 are used to re-examine the magnitude of the problem between 1992 - 1995, at each site. The specific aim is to identify the time of day and period in the year when pollution levels would be termed excessive on the basis of the projected objectives for 2005. This highlights the degree of the problem as it currently stands.

A different criterion from that highlighted in Table 2 is used in evaluating SO_2 levels at the five UK sampling sites; 60 minute averaging times are used in place of 15 minute periods. This is largely due to the format of the data acquired. Although variation in concentrations are 'smoothed' out by this alternative approach the overall effect will not have a great impact on final conclusions.

From hourly means of NO_2 and SO_2 the percentage of time attributed to excess pollution levels is determined for the months between 1992 - 1995; each month consisting of a collection of four monthly sets of data. The same approach is adopted when examining PM_{10} and O_3. Although these pollutants are measured as running means (refer to Table 2) with overlapping periods between months the degree to which this occurs is constant for each pollutant and therefore has a minimal impact on overall results. When this approach is completed the month associated with poorest levels of each pollutant can be identified. During these periods diurnal variations are investigated.

2.3 Results

Figure 1 illustrates the percentage of time attributed to poor air quality due to individual pollutants. This applies to months between 1992 - 1995, so each month represents a total of four periods (eg Jan., 1992, 1993, 1994 and 1995).

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* These are running means.

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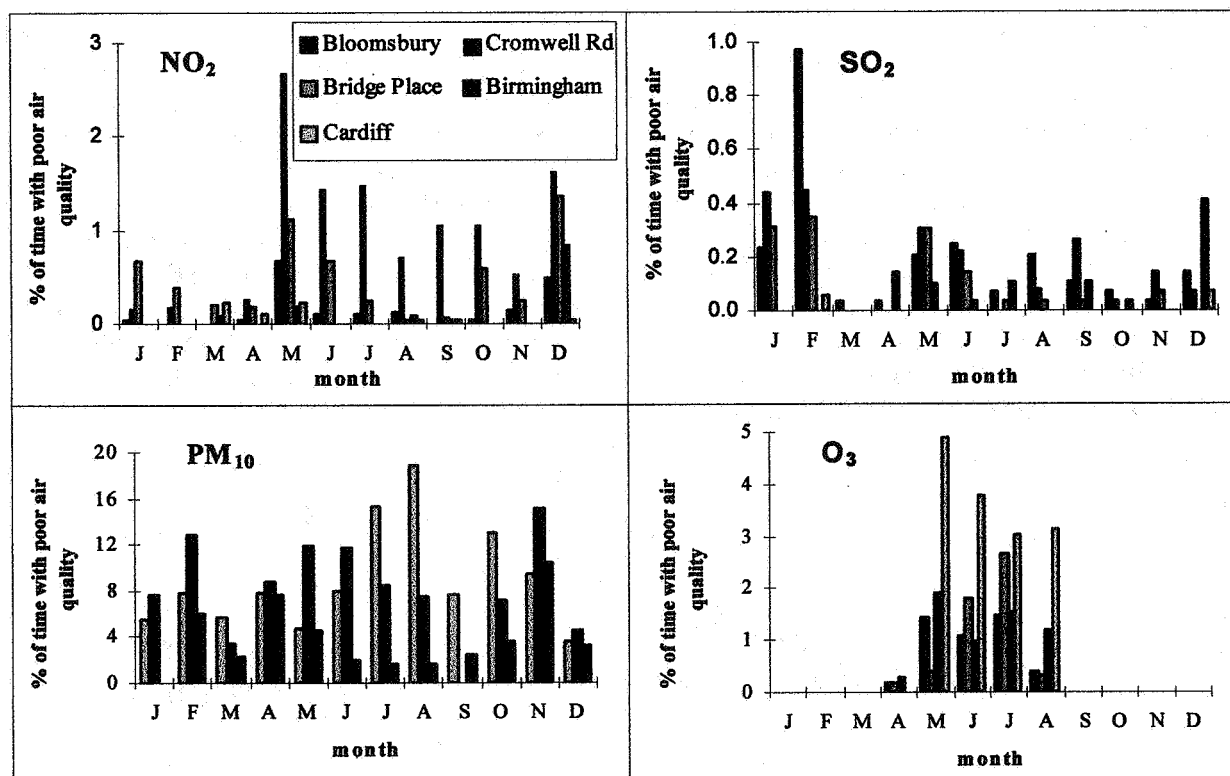
Figure 1 Amount of time associated with poor air quality during months between the years 1992 - 1995. The bench mark concentrations are those outlined in Table 2.

For each pollutant in Figure 1 the maximum period of time attributed to poor air quality can be identified during each month. The values are provided in table 3, and the figure typed in bold if it exceeds 5%.

Table 3. **Maximum percentage of time of poor air quality for each pollutant**

grouped months (during 1992-1995)	maximum percent of time attributed to excess pollution levels			
	NO_2	SO_2	PM_{10}*	O_3
January	0.7	0.4	**7.7**	0.0
February	0.4	1.0	**12.9**	0.0
March	0.2	0.0	**3.4**	0.0
April	0.3	0.1	**8.9**	0.3
May	2.7	0.3	**11.9**	4.9
June	1.4	0.3	**11.7**	3.8
July	1.5	0.1	**8.5**	3.0
August	0.7	0.2	**7.5**	3.1
September	1.1	0.3	2.4	0.0
October	1.0	0.1	**7.2**	0.0
November	0.5	0.1	**15.1**	0.0
December	1.6	0.4	4.5	0.0

Key

* These values do not include the Cardiff results because during 1994 the site was found to be 'contaminated' by building works. This had affected the normal distribution of particles at the site.

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PM$_{10}$ is the only pollutant that exceeds guideline values quoted in Table 2 for more than 5% of sampling periods. The worst period for PM$_{10}$ excesses is during November months between 1992 - 1995; 15.1% of this period is associated with excess concentrations of PM$_{10}$. February months are the worst for SO$_2$, and May months the poorest for NO$_2$ and O$_3$. These months are used to examine diurnal variations for the relevant pollutant. The data analysed is the set associated with the highest excess pollution episodes; for example it can be established that in May O$_3$ exceeded the bench mark most often. Reference to Figure 1 reveals that Cardiff is the location where excess O$_3$ concentrations occur for 4.9% of May months between 1992 - 1995. Hence diurnal variations during May months at Cardiff are studied. A similar approach is adopted for the other pollutants. Figure 2 illustrates diurnal variations for the pollutants in the way described above. NO$_2$ and SO$_2$ variations are expressed as percent excess episodes. This is not possible for PM$_{10}$ and O$_3$ as excess concentrations are analysed on a running mean basis, instead diurnal variation of concentration values are highlighted.
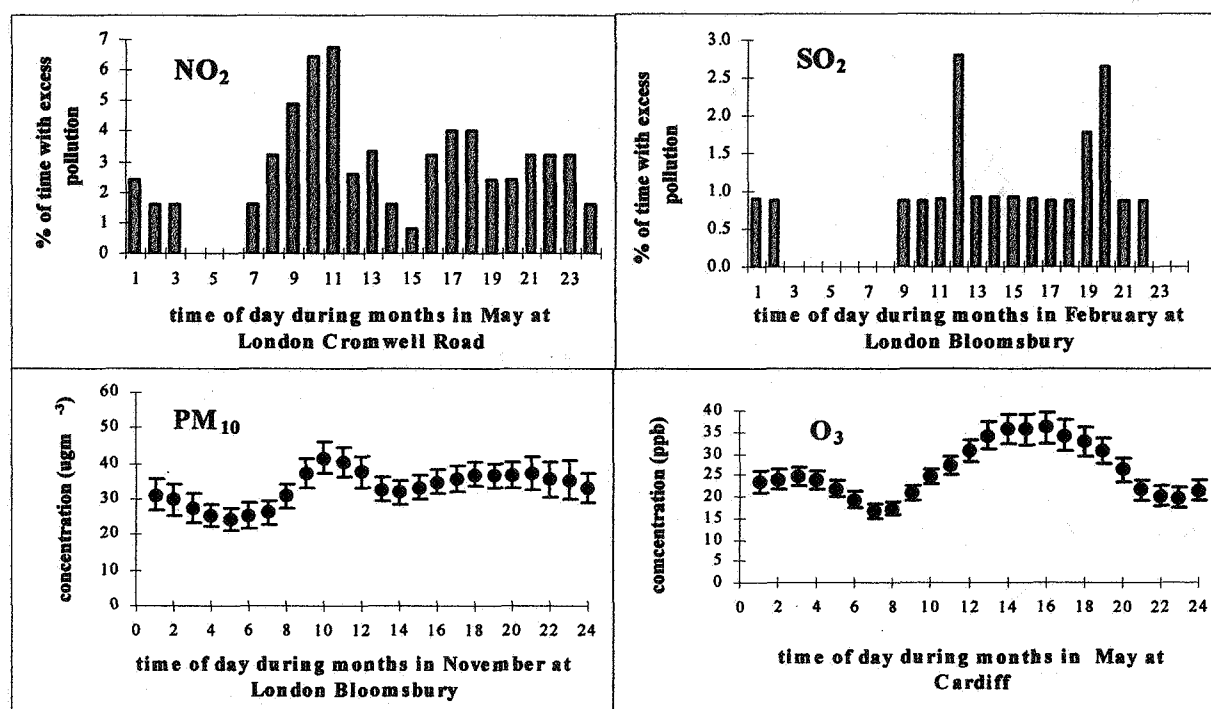
Figure 2   Diurnal variations in pollution levels between 1992 - 1995.

Figure 2 shows that excess episodes of SO$_2$ do not rise above 5% for any hour during February at London Bloomsbury, even though this month and site is the most onerous of all the areas studied. Excess episodes of NO$_2$ exceed 5% of sampling hours during May at London Cromwell Road on only two occasions (at 10.00 and 11.00 hours). Figure 2 provides means and associated standard error for diurnal variations of PM$_{10}$ and O$_3$. Between 09.00 - 11.00 hours PM$_{10}$ concentrations reach a significant peak. For O$_3$ this peak occurs between 14.00 - 16.00 hours. In the months when PM$_{10}$ and O$_3$ excess episodes are relatively high these peak periods are likely to be a real problem.

The data analysed above chiefly relates to pollutants monitored near road level. Natural ventilation systems with a central operating mechanism may draw air in at roof level. The advantage of this is that

ambient concentrations of pollutants may be reduced as a function of height. In section 3 a quantitative investigation of this issue is made.

3.   THE DILUTION OF TRAFFIC RELATED POLLUTANTS WITH HEIGHT

3.1 Evaluation method

Norfolk House, a ten storey building situated alongside a busy major road in Croydon London, was selected for the analysis. The windows on the road side of the office block were openable and allowed monitoring probes to be held outside.   The pollutants monitored were PM_{10} and NO_2, previously identified as the main contributors to poor air quality in British cities. Particle measurements were made using a light scattering particle counter with mean sampling times of 24 hours. NO_2 was analysed using a gas chemiluminescence techniques with sampling periods of an hour.

Air quality measurements were taken between 08.00 to 18.00 hours during five working days. On each day outdoor air quality was assessed at two heights; the lower position was fixed so that over the week dilution of each pollutant with increasing height could be examined. Table 4 summarises the differences in height between monitoring positions, on different days..

Table 4.   Analytical protocol of altitude monitoring exercise

Day	Control height (m)	Variable height (m)	difference in height (m)
1	5.00	11.10	6.10
2	5.00	17.20	12.20
3	5.00	23.30	18.30
4	5.00	29.40	24.40
5	5.00	32.45	27.45

3.2 Results

Differences in the concentration of pollutants between two heights were calculated as a percentage of the value at the lower fixed height of 5m. On each day an overall mean value was determined with standard error at the 95% confidence level. The daily means were contrasted to develop a profile of height against concentration. Although the distance between measuring points varied from one day to the next by calculating percent change in concentrations between points an overall picture of height effects is possible. Figure 3 illustrates this relationship for each pollutant.
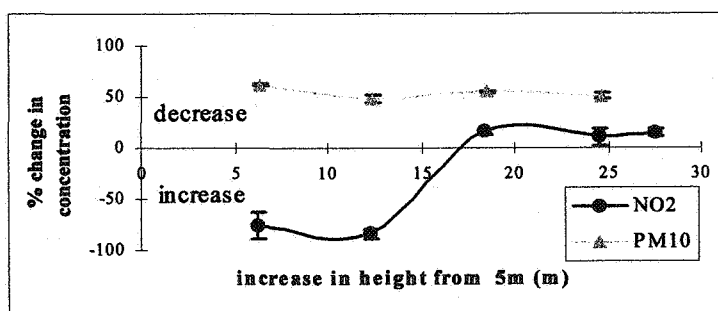
Figure 3   Variation in pollutant concentrations as a function of alterations in height from road level.

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PM$_{10}$ concentrations are diluted with height in all circumstances according to Figure 3; particle concentrations diminish by between 48 - 61%, between 6.1 - 24.4 m above 5 m. The relationship between PM$_{10}$ concentrations and height is not linear. NO$_2$ behaves differently in that between 6.1 - 12.2 m above 5 m its concentration increases. Beyond 18.4 m above 5 m NO$_2$ levels fall by between 10 - 16.

The significance of variations in meteorological conditions in a vertical plane has not been assessed in this exercise which imposes a limitation on the results obtained from this approach.. However this preliminary exercise is of use as it indicates potential improvements in the quality of air provided to buildings when supply is at a sufficient height from busy roads.

4. DISCUSSION

A potential strategy for natural ventilation of non domestic buildings in cities is to avoid periods when pollution loads from traffic may be high. The exercise in section 1 allows a decision to be made about the frequency of 'acceptable' excess pollution episodes in the outdoor environment. This is a possible design strategy for natural ventilation of non domestic buildings. An example of this approach is shown in Table 5 for the pollutants reviewed in section 1 of this paper. The assumption made in Table 5 is that excess outdoor pollution for up to 5% of sampling times is an acceptable frequency of episodes.

Table 5   **Suitable and unsuitable periods for natural ventilation due to pollutants in urban environments (letters in cells denote type of pollution problem)**

hour of day	month											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
1												
2												
3												
4												
5												
6												
7												
8												
9	p	p		p	p	p	p	p		p	p	
10	p	p		p	p & n	p	p	p		p	p	
11	p	p		p	p & n	p	p	p		p	p	
12												
13												
14					o							
15					o							
16					o							
17												
18												
19												
20												
21												
22												
23												
24												

KEY

p = PM$_{10}$ n = NO$_2$ o = O$_3$

The assumption behind Table 5 that 5% is a suitable upper bench mark for an acceptable frequency of excess pollution episodes is an important one as a different value may produce a dissimilar distribution of appropriate periods for natural ventilation. However 5% is a reasonable value given the additional dilution effect of pollutants with increased height from road traffic emissions. There is also evidence of much lower indoor concentrations of pollutants compared to outdoor levels, (4); up to 60% lower indoor concentrations in a naturally ventilated building compared to outdoor values at street level.

Table 5 demonstrates likely occasions when PM_{10} and O_3 ambient concentrations may be too high. Since this is based on calculations of percent excess episodes on monthly time scales it is possible that there are other periods in the day when the problem may also be consistent. Over the long term O_3 may become more significant in urban areas where other pollutants diminish; NO reacts with O_3 producing NO_2. If traffic emissions of NO decrease O_3 concentrations will rise. This is most likely to occur during peak hours of traffic intensity. However there is evidence that building materials such as masonry, are effective at breaking up O_3 molecules, hence overall the pollutant is not a severe problem (5).

Using the criteria set out in Table 5 SO_2 does not appear to be a significant urban pollutant. There were no periods when excess SO_2 concentrations prevailed for up to 5% of sampling periods. Although the sites examined were not the worst in Britain for this pollutant, they still represented areas where the problem was comparatively significant. Table 5 indicates that NO_2 concentrations can be persistently high, however these are very infrequent and the locations studied were the poorest for NO_2 levels in the UK. Further more tougher legislation to control vehicle emissions are likely to be enforced given ongoing concerns about air quality. This will assist in maintaining NO_2 concentrations below critical values. Section 3 also clearly demonstrated that height from roadsides is an important issue and can result in substantial reductions in pollutant concentrations, although further work is required to quantify the effects of meteorological considerations.

5. CONCLUSION

Table 2 indicates the adverse health effects associated with high concentrations of urban pollutants. It emphasises the need for unpolluted air when supplying non domestic buildings adequate ventilation. When natural ventilation is the favoured option the quality of the supply air is even more critical, given that low driving forces attributed to natural ventilation prohibits the inclusion of extensive air filtration mechanisms. Whilst air conditioning generates larger pressures that will cope with air cleaning processes it is not an ideal solution for strategies geared to reducing energy consumption. Thus an alternative approach is necessary.

A potential low energy solution is to supply air to buildings for ventilation purposes in a way that avoids the most onerous pollution periods. Either air inlets can be shut off during these occasions, or fan assisted ventilation utilised, with polluted air drawn in through a system of cleaning filters. Deciding when to switch to a fan assisted scheme is possible from the approach made in Table 5. Table 5 indicates the periods during a year when natural ventilation is a possible low energy option, and also indicates the occasions when a fan assisted scheme would need to be operated to allow for air cleaning via filters.

The analysis of pollution data from the British mainland examined areas where the problem appeared most evident. London is a good representative of high polluting areas of northern Europe, where the climate is cold to moderate. Thus measures identified in this paper geared to reducing the demand for energy due to ventilation can be adopted in similar regions of Europe. Whilst it is not possible to account

for all circumstances that may occur the approach described in this paper is a useful step towards the promotion of natural ventilation. It is also note worthy that section 3 of this paper suggests that intelligent location of air inlets may greatly reduce the need for costly air filtration systems.

6. ACKNOWLEDGEMENTS

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VENTILATION AND COOLING

18TH ANNUAL AIVC CONFERENCE
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NON-DIMENSIONAL GRAPHS FOR NATURAL VENTILATION DESIGN

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Non-dimensional Graphs for Natural Ventilation Design

Synopsis

There are certain conditions which are of interest when designing for natural ventilation of commercial buildings. These are:-

- summer cooling
- indoor air quality in winter
- night-time cooling.

For the first two conditions it is necessary to determine the distribution of open areas to give the desired distribution of flow rates. Since one is dealing with openings whose position and basic geometry are known, the problem is relatively simple compared to general ventilation problems. When buoyancy acts alone the position of the neutral layer can be specified and the size of the openings can be determined explicitly, as described by other authors. The paper takes this explicit approach further.

First it is shown how, for the summer and winter design conditions, one non-dimensional graph covers the buoyancy alone case with a uniform temperature and then how non-uniform temperatures can be covered by a few extra graphs. The approach is also extended to include the sizing of stacks as distinct from sharp-edged openings.

For the winter design condition where the openings are small it is important to estimate the effects of adventitious openings. It may also be desirable to determine the effects of wind. Suitable procedures and graphs for doing this are described.

Finally the possibility of using similar graphs in the evaluation of night-time cooling is briefly discussed.

List of symbols

a	leakage coefficient $[Pa.s/m^6]$	ΔC_p	pressure coefficient difference [-]
b	leakage coefficient $[Pa.s/m^3]$	Δp	pressure difference across envelope [Pa]
A	area of opening $[m^2]$		
A_o	area of outlet opening $[m^2]$	ΔT	temperature difference between interior and exterior [K]
Ar	Archimedes number, U_b/U [-]		
C_p	wind pressure coefficient [-]	$\Delta \rho$	density difference between interior and exterior $[kg/m^3]$
$C_,$	discharge coefficient of opening [-]		
d	diameter of stack [m]	ρ	density $[kg/m^3]$
h	height of outlet, see Fig, A1 [m]	ν	kinematic viscosity $[m^2/s]$
H	height of occupied space [m]	Subscripts	
L	length of stack [m]	i	opening number
Q	flow rate through opening $[m^3/s]$	E	exterior
Re	stack Reynolds number [-]	I	interior
T	temperature [K]	H	at z = H
U	wind speed [m/s]	0	at z = 0
U_b	$\equiv \sqrt{\Delta \rho . g H / \rho}$ [m/s]	L	leakage measurement
z	height [m]		

1. Introduction

Although sophisticated numerical methods such as CFD and envelope flow models are available to the designer (e.g. Refs. 1 and 2), in the early stages of a design it may be helpful to use simple manual calculations. Such calculations are facilitated by non-dimensional graphs, which present the results of numerical predictions in a general and easily used form (Ref. 3). Examples of this approach for natural ventilation design, using results from envelope flow models, form the subject of this paper.

2. Summer cooling

The usual aim of summer design is to provide sufficient fresh air entry to each space to prevent overheating of the occupants. The basic procedure is:- (i) decide an acceptable temperature rise above ambient e.g. $\Delta T = 3$ K, (ii) calculate the flow rates required for each space, Q_i, and (iii) calculate the area of openings required for each space when the wind speed is zero i.e. buoyancy alone. The third part of the procedure is of concern here.

The openings will be windows or air vents and they can be treated as sharp-edged with a square-law flow characteristic

$$Q = C_z A \sqrt{\frac{2\Delta p}{\rho}} \qquad (1)$$

Since the positions of the openings are known (for buoyancy alone it is only the height z_i which is relevant), the determination of the values of A_i for the required Q_i can be found by a relatively simple calculation for open-plan buildings with negligible internal flow resistance. This has been referred to as an explicit calculation (e.g. Ref. 2).

A typical problem is illustrated in Figure A1 in the Appendix. The height of the occupied part of the building is H and the height of the uppermost opening is (H + h). The values of Q_i and z_i are known and it is required to calculate the A_i such that fresh air enters each opening at the required rate, with the exception of the uppermost outlet opening where the air leaves the building (for convenience only a single outlet opening is considered here). This means that the neutral level, i.e. the height, z_n, at which the pressure difference across the envelope is zero, must lie somewhere close to H. By choosing a reasonable value for z_n, $z_n = $ H say, the pressure difference across each opening is known, because the relation between Δp and z is determined by the variation of ΔT with z and the height at which $\Delta p = 0$ is z_n. Using this relation, Δp_i for each opening can be found and substituted with Q_i in Eqn. (1) to determine A_i (C_z can be taken as 0.6). The area of the outlet opening is found in the same way, since the flow rate through it is given by the sum of the other flow rates.

The explicit procedure has been described in Ref. 2 for the case of uniform internal temperature. In the following the explicit procedure is expressed in non-dimensional form and is extended to include the case of non-uniform temperature and the use of a stack in place of a square-law outlet opening.

2.1 Buoyancy alone, uniform temperature

With a uniform internal temperature ($\Delta T_H / \Delta T_0 = 1$), the expression for the non-dimensional envelope pressure difference takes a very simple form

$$\frac{\Delta p}{\Delta \rho.gH} = 1 - \frac{z}{H}$$ (2)

Equation (2) is shown in Figure 1 for values of z/H up to 2.0. It is unlikely that the outlet opening will lie beyond z/H = 2.0.

2.2 Buoyancy alone, non-uniform temperature

With tall buildings such as atria, it is probably unrealistic to assume a uniform temperature. For design purposes a simple non-uniform temperature distribution can be assumed, whereby the temperature difference varies linearly between values of ΔT_0 and ΔT_H at z = 0 and z = H respectively (see Appendix).
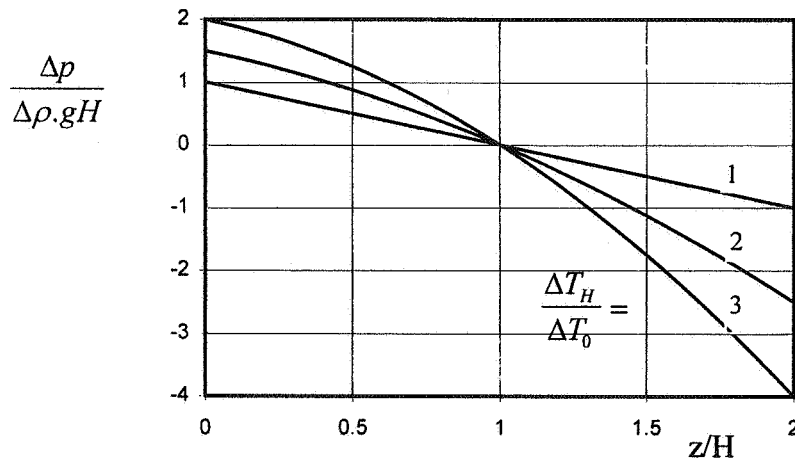
Figure 1   Non-dimensional pressure as a function of z/H for different temperature distributions.

The corresponding expression for the non-dimensional envelope pressure difference is

$$\frac{\Delta p}{\Delta \rho.gH} = 1 - \frac{z}{H} + 0.5 \left(\frac{\Delta T_H}{\Delta T_0} - 1 \right) \left(1 - \left(\frac{z}{H} \right)^2 \right)$$ (3)

Equation (3) is also shown in Figure 1, for values of $\Delta T_H/\Delta T_0$ of 2 and 3.

Figure 1 is particularly helpful for preliminary design of the outlet opening, because it can quickly show the trade-off between open area and height and the effect of non-uniform temperature.   For example, raising the outlet height from z/H = 1.25 to 1.50 leads to an increase in Δp and a corresponding decrease of 30 % in the required area. With z/H = 1.25, a non-uniform temperature corresponding to $\Delta T_H/ \Delta T_0$ = 2 will allow a similar reduction in area compared to that with a uniform temperature.

2.3 Buoyancy alone, with stack

Rather than use a simple opening for the outlet it may be desirable to use an internal or an external stack, as illustrated in Figure A2 in the Appendix.

It may often be reasonable to assume that the temperature distribution in the stack is the same as that in the building. This means that for the case of a uniform temperature the stack

can be treated as a pipe in a uniform density flow. The effect of the stack is then simply that it alters the area required to obtain the given outlet flow by virtue of the change in discharge coefficient. The discharge coefficient for a pipe with a circular cross-section and a bellmouth inlet is given by (see e.g. Section 3.2 of Ref. 4)

$$C_z = \sqrt{\frac{1}{\dfrac{0.316L}{R_e^{0.25}d} + 1}} \quad \text{and} \quad C_z = \sqrt{\frac{1}{\dfrac{96L}{R_e d} + 1.67}} \quad \text{for turbulent and laminar flow}$$

respectively, where C_z and Re are defined by $C_z = \dfrac{Q}{A}\sqrt{\dfrac{\rho}{2.\Delta p}}$, $Re = \dfrac{Qd}{Av}$.

For the design condition, the value of Δp is $\Delta \rho.gh$ i.e. the same as that for the square-law outlet at height (H + h), as shown in the Appendix.

The benefit of using a stack under these conditions can be seen in a plot of C_z against Re (Figure 2). The value of C_z for sharp-edged openings such as windows and vents is 0.6, which is significantly less than the values for a stack with turbulent flow, therefore requiring a larger opening area for a given Q. However this disadvantage of sharp-edged openings would be overcome if the opening were fitted with a short bellmouth inlet which would increase C_z to a value close to 1.0.
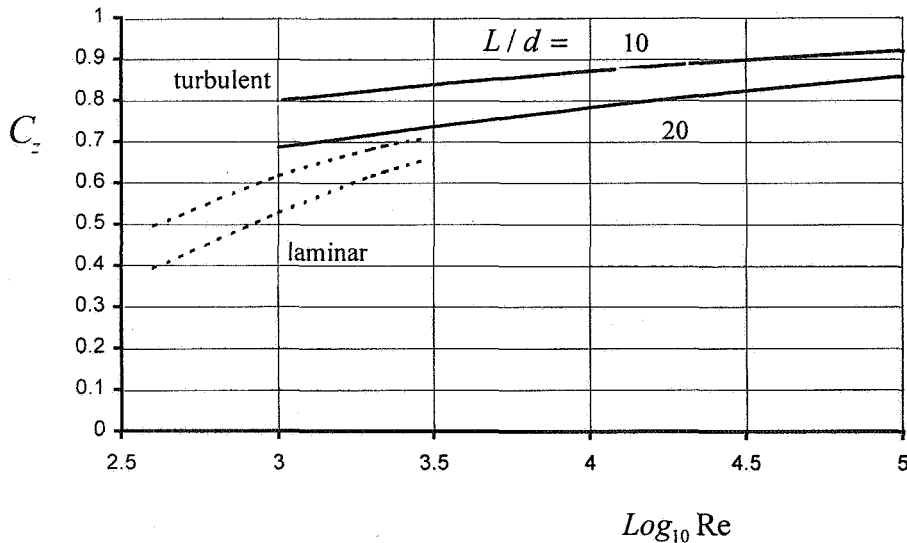
Figure 2 Discharge coefficient of stack as a function of Re and L/d.

3. Winter design

The usual aim of the winter design condition is to ensure adequate indoor air quality without excessive ventilation. This can be achieved by choosing air vents such that a fixed minimum area is always available and the area can be increased beyond this by the occupants as necessary. The design condition remains the same as for summer (i.e. z_n = H) and the procedure for determining the fixed minimum open area for each floor is the same i.e. Figure

1 can be used. The value of ΔT will be larger, but the values of Q_i will be much smaller than for the summer condition so that the vent areas, A_i, will be much smaller.

3.1 Buoyancy alone, effect of adventitious leakage

In view of the above it is quite possible that the adventitious leakage of the building will be significant i.e. it could exceed the leakage associated with the fixed minimum areas. Account should be taken of the adventitious leakage if excessive ventilation heat loss is to be avoided. One approach to the problem is to estimate the adventitious leakage which will give the design outlet flow rate with the same pressure distribution as the air vents. This equivalent leakage can be compared with either the measured value or the range of values likely to be encountered. If it is less than either of these values, there may be no need for a fixed minimum opening. If it is much less, it may be desirable to tighten the building envelope.

$$\frac{Q}{\sqrt{\Delta\rho.gH / 2a}}$$

$$1 / \sqrt{2\Delta\rho.gHa / b^2}$$
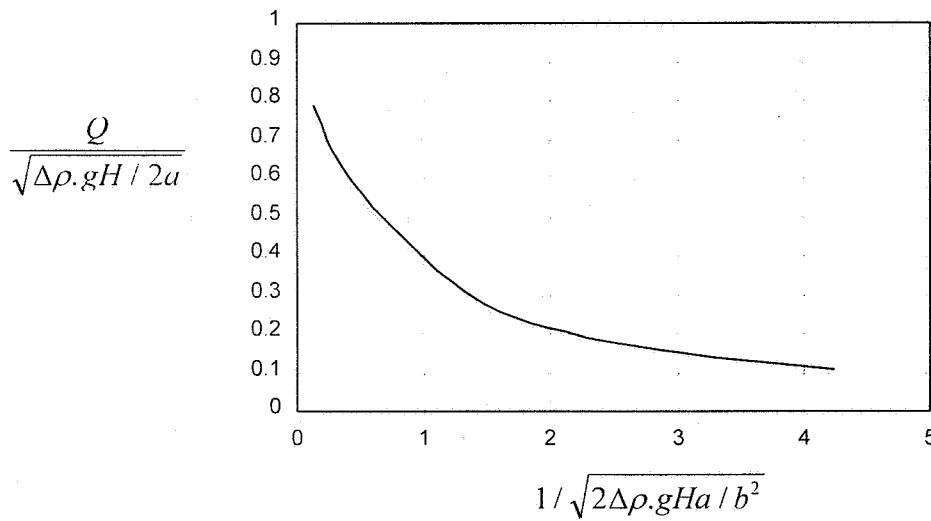
Figure 3 Non-dimensional ventilation rate for the winter design condition with a uniform distribution of adventitious openings.

The estimation of the equivalent adventitious leakage can easily be done with results from an envelope flow calculation method, such as *VENT* (Ref. 3), which assumes a quadratic leakage characteristic

$$\Delta p_L = aQ_L^2 + bQ_L \qquad (4)$$

where a and b are the leakage coefficients.

Figure 3 shows how the non-dimensional ventilation rate varies with the non-dimensional leakage parameter when the neutral level is at $z = H$. For this example it is assumed that the openings are uniformly distributed on the walls but curves for other distributions can easily be produced.

Knowing a/b^2, H and $\Delta\rho$, the parameter $2\Delta\rho.gH$ is known and Figure 3 then gives the corresponding value for $Q/\sqrt{\Delta\rho.gH / 2a}$. The value of Q is given by the design outlet flow rate and this enables the leakage coefficient a to be determined. Knowing a and a/b^2 the leakage at a given Δp_L can be found from Equn. (4). As an illustrative example, for H = 30 [m], $\Delta T = 20$ [K] and $a/b^2 = 0.1$, the parameter $1 / \sqrt{2\Delta\rho.gHa / b^2} = 0.42$ and hence

$Q/\sqrt{\Delta\rho.gH/2a}$ = 0.59. If the design ventilation rate is 10,000 [m3/h], the equivalent leakage is 25,530 [m3/h].

3.2 Effect of wind

If the building is in an exposed position, the designer may wish to investigate the effects of wind. This can be done quite easily using non-dimensional graphs of results for the design opening distribution. Figure 4 shows results from *VENT* for a building with h/H = 0.1 and with the openings on two walls. ΔC_{p1} is the difference between the wind pressure coefficients on the two walls and ΔC_{p2} is the difference between the coefficients on the windward wall and the roof. The effect of the Cp distribution can clearly be seen. It is a simple matter to generate curves for other values of h/H.
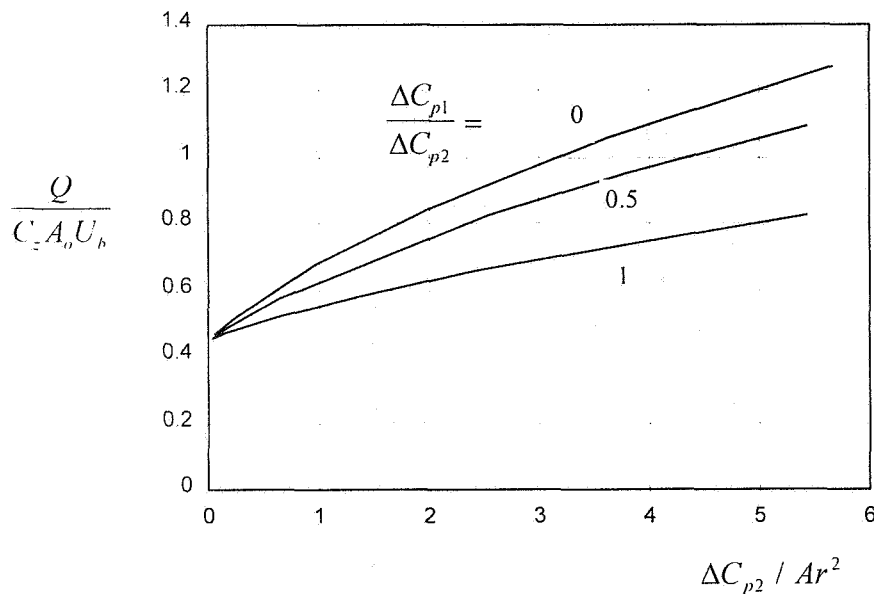
Figure 4 Effect of wind on non-dimensional ventilation rate for h/H =0.1.

4. Night-time cooling

Night-time cooling in summer will rely on large openings so that the calculations are again simplified by (a) the form of the flow characteristic, Equn. 1, and (b) the ability to neglect adventititious openings. For a given building layout and opening distribution it is relatively easy to generate a non-dimensional graph for the variation of Q with ΔT and wind speed, similar to Figure 4. This enables quick estimates to be made of ventilation rates under the conditions likely to be encountered during cooling. These could give a preliminary indication of whether or not the openings are of a sufficient size, prior to a full calculation with a combined thermal/ventilation model.

5. Conclusions

Results from envelope flow models can be expressed in a general non-dimensional form and thereby provide graphs which can be used for quick manual calculations in natural ventilation design. This is particularly true when the design involves large purpose-provided openings, because of the ability to neglect adventitious openings.

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APPENDIX.
Equations for buoyancy alone.
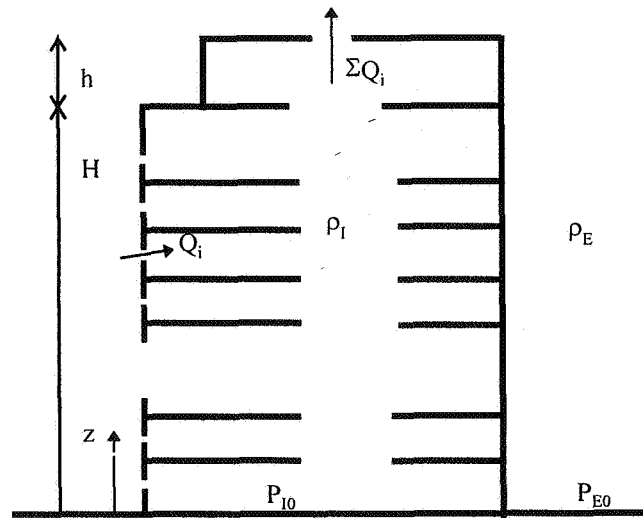
Figure A1

The absolute pressures at points outside and inside the building at height z are respectively

$$P_E\{z\} = P_{E0} - \rho_E g z \quad \text{and} \quad P_I\{z\} = P_{I0} - \int_0^z \rho_I\{z\}g.dz$$

and the pressure difference across the envelope is given by

$$\Delta p\{z\} = P_{E0} - P_{I0} - g[\rho_E z - \int_0^z \rho_I\{z\}.dz] \qquad \text{A.1}$$

The design condition is $\Delta p\{z\} = 0$ at $z = H$, so

$$P_{E0} - P_{I0} = g[\rho_E H - \int_0^H \rho_I\{z\}.dz] \qquad \text{A.2}$$

and substituting in Equn. A.1 gives

$$\Delta p\{z\}/g = \rho_E(H - z) - \int_z^H \rho_I\{z\}.dz$$

$$\Delta p\{z\}/\rho_E g H = (1 - z/H) - \int_{z/H}^1 \rho_I\{z\}.dz/ H\rho_E \qquad \text{A.3}$$

For a uniform internal temperature Equn. A.3 becomes

$$\Delta p\{z\}/ \Delta\rho.gH = (1 - z/H) \qquad\qquad A.4$$

For a linear variation of internal temperature given by

$$\Delta T\{z\} = \Delta T_0 + (\Delta T_H - \Delta T_0)z/H \qquad\qquad A.5$$

Equn. A.3 becomes

$$\Delta p\{z\}/ \Delta\rho.gH = (1 - z/H) - 0.5(\Delta T_H/\Delta T_0 - 1)(1 - (z/H)^2) \qquad\qquad A.6$$

Stack with uniform density

The pressure difference across the stack arises from the motion of the air. The implicit assumption is made that the flow establishes itself in an upward direction. There are two expressions for the absolute pressure at the top of the stack. Referring to Figure A2

$$P_2 = P_{EH} - \rho_E gh \qquad\qquad A.7$$

and

$$P_2 = P_{IH} - \rho_I gh + p_2 \qquad\qquad A.8$$

where p_2 denotes the pressure arising from the motion of the air. For the design condition $\Delta p = 0$ at $z = H$, so

$$P_{IH} = P_{EH}$$
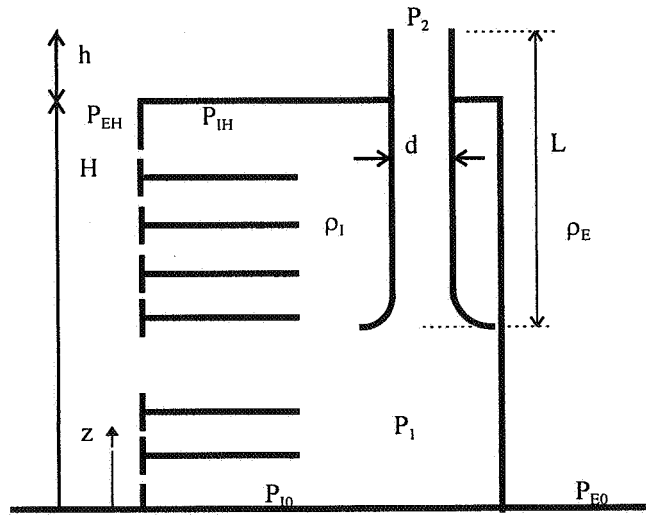
Figure A2

Thus from Equns. A.7 and A.8 it follows that

$$-\rho_E gh = -\rho_I gh + p_2$$

or

$$p_2 = \Delta\rho.gh \qquad\qquad A.9$$

The pressure due to motion in the room away from the inlet to the stack, p_1, is equal to zero, so the pressure difference across the stack is

$$\Delta p = p_1 - p_2 = -\Delta\rho.gh \qquad\qquad A.10$$

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Title: Prediction of the Potential of Self-Regulating Natural Ventilation
            Devices: Methodology and Practical Results

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PREDICTION OF THE POTENTIAL OF SELF-REGULATING NATURAL VENTILATION DEVICES : METHODOLOGY AND PRACTICAL RESULTS

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ABSTRACT

The performances of self regulating natural ventilation devices (devices of which the opening section varies as function of the pressure difference across the device) strongly depend on the type of building and its leakage characteristics. In like manner, the climatic conditions strongly impact on the achieved ventilation rates. As a result, it is not possible to express the potential benefit of self-regulating natural ventilation devices in an unambiguous way. This is not contributing to a good understanding of the potential of such devices in daily practice. In order to increase the transparency of the results, a method has been developed which allows comparing the performances of various natural ventilation devices (fixed devices, self-regulating devices, etc.) for a range of building types and climatic conditions. The paper presents the simulation concept and the possibilities of the VENTEX programme. Results obtained for a range of combinations of ventilation devices are presented and discussed.

1. THE SIMULATION CONCEPT

1.1 BASICS

At present, there is a range of programmes, which allows detailed simulation of airflows in buildings. The basic idea of this programme called 'Ventex' was the development of a quick simulation tool which is very transparent and which allows one to come to useful results for a range building types. The major limitation is the fact that the simulation tool can only treat buildings that can be modelled as a combination of:

- one *central zone* which can be in connection with the outside and with adjacent spaces;
- as many *adjacent zones* as needed through which the airflow passes from the outside to the central zone. The temperature of adjacent zones equals the temperature of the central zone.

These assumptions permit to treat the problem numerically as a single zone problem and results in a Visual Basic programme in a Microsoft Excel 97 environment and a short calculation time.

1.2 THE AVAILABLE COMPONENTS

The following figure shows the complexity of the models that can be calculated using Ventex.
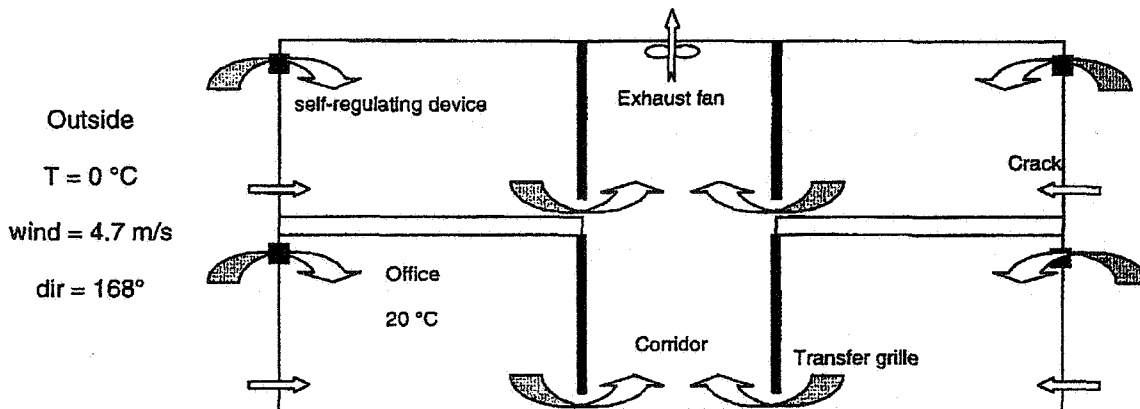
Figure 1 – Example of model that can be calculated with Ventex

Essentially, 4 types of components can be simulated:

1. A leakage opening with constant characteristics (trickle ventilators with constant section, cracks, etc.)

   Leakages are described by the equation $q = C.\Delta P^N$. The user must specify C and N.

2. Fans

   Fans can be described as a device with a fixed air flow rate but it is also possible to define the q-ΔP relation for various pressure differences. The programme interpolates for intermediate pressure differences. The figure shows an example . of fan characteristics as presented in the programme.
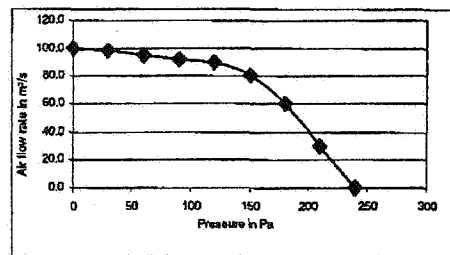
3. Leakage openings with variable characteristics

   The user can describe any kind of q-DP relation. The programme interpolates for intermediate pressure differences. The figure shows the q-ΔP relation of a self-regulating device.
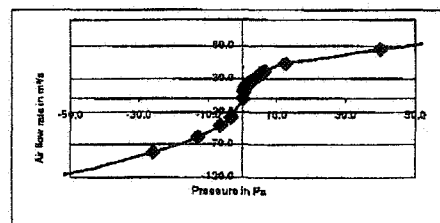
4. Equivalent leakage consisting of variable or fixed resistance in parallel and in series.

The airflow through adjacent rooms is unidirectional (from the central room to the outside or from the outside to the central room, no 2-way flows). Therefore, it is possible to use an equivalent resistance that characterises the airflow through the room.

Adjacent room

The example of Figure 1 shows an adjacent room with a ventilation device in the outside wall. There is a transfer grille in the inner door and a crack corresponding with the background leakages in the outside wall. Ventex determines the equivalent characteristics of the system consisting of these different components and simulates this system as one single component (see figure).
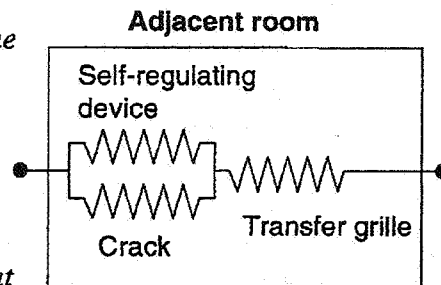
1.3 THE SIMULATION TOOL ©VENTEX

Ventex is running under Excel 97 and makes extensive use of Visual Basic programming. The user must define the number, type and characteristics of the components and the programme derives the equivalent characteristics and simulates the mass transport through the components.

The programme in its present version makes use of hourly data (e.g. the test reference year for Uccle in Belgium). The user can specify the time step (1 hour, 2 hours, etc.).

The output includes:

- Pressure differences across all components (for each time step);
- Air flow rates through all components (for each time step);
- Heating power for ventilation and monthly energy use for ventilation.

In principle, it is possible and not difficult to add prediction of CO_2 levels, which is not available at the moment.

2. EXAMPLE OF APPLICATION

2.1 THE BUILDING

The building taken as illustration has the following characteristics (see Figure 1):

- 2-storey building consisting of cellular offices at both sides of the corridor.
- Mechanical extraction with known q-ΔP characteristics.
- Natural supply openings in all offices (see Table 1).
- Background leakage corresponding with n_{50}-value of 2 h^{-1}.
- Each office has a floor area of 10 m², is occupied by one person.
- Air inlet of 30 m³/h are installed in each of the 4 offices.
- An exhaust fan is extracting air from the corridor. Its is nominal airflow rate is equal to the sum of the nominal airflow rates of all the ventilation supply device (number of offices X 30 m³/h).
- All windows and doors closed

Assuming that the resistance of the corridor and the open staircase is negligible, it is possible to model only a segment corresponding of 2 rooms at ground level, 2 rooms at top level and the corridor. This results in the model presented in the next figure.

The Cp-values for offices of the first floor are 0.2 (windward) and –0.2 (leeward) while for the ground floor, values of 0.15 and –0.15 were assumed. These values are related to the meteorological wind data.

Important remark :

It must be stressed that the results given in the paper are mainly aimed for illustration purposes. It is evident that the choice of the wind pressure coefficients and the local wind speed is extremely important. In the examples, we have assumed that the wind direction is always the same, this is also not a very realistic assumption.
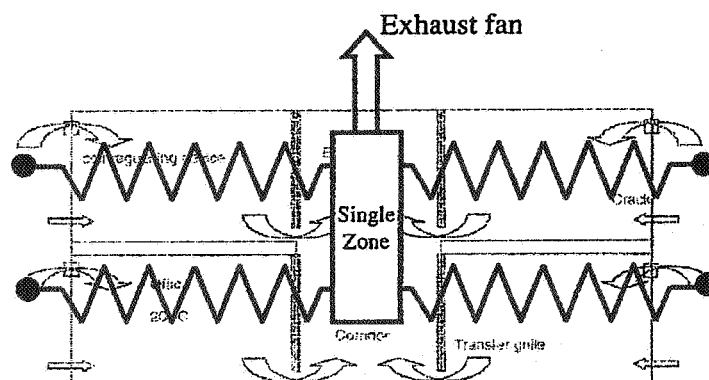
Figure 2 – Simulation model of the studied building

2.2 THE COMPONENTS UNDER CONSIDERATION

Table 1 described the different configurations that were simulated in the above-described building. Different ventilation supply devices were used as well as different kind of transfer openings between the corridor and the offices. Finally, simulations with open doors and windows were also tested.

The nominal pressure of the ventilation devices used is either 1 Pa (more or less representative for the approach used in e.g. the Netherlands) or 2 Pa (representative for the approach used in Belgium) or 20 Pa which corresponds with the approach of very small supply openings (to a certain extent the case in France).

The characteristics of the self-regulating devices used in the model are given in Table 2.

Case	Generic description	Transfer opening	Doors	Windows
1	Constant air flow rate device giving nominal air flow rate at pressure difference of 1 Pa	Fixed opening giving nominal air flow rate at difference of 1 Pa	All closed	All closed
2	Idem 1 at 2 Pa	Idem 1 at 2 Pa	All closed	All closed
3	Idem 1 at 20 Pa	Idem 2	All closed	All closed
4	Idem 1	Idem 1	All open	1 windward window open
5	Idem 3	Idem 1	All open	1 windward window open
6	Self-regulating device at 1 Pa	Idem 1	All closed	All closed
7	Self-regulating device at 2 Pa	Idem 2	All closed	All closed
8	Idem 1	Idem 1	All open	All closed
9	Idem 6	Idem 2	All open	All closed
10	Idem 2	Idem 1	All open	All closed
11	Idem 7	Idem 2	All open	All closed

Table 1 – Overview of the different simulation cases

Pressure difference	Fraction of nominal flow rate q_N (=air flow rate at the nominal pressure)
0-2 Pa	$\sqrt{P/P_{NOM}}$
5 Pa	1.2
10 Pa	1.2
25 Pa	1.2
50 Pa	1.5
100 Pa	2
200 Pa	3.0

Table 2 – Characteristics of self-regulating device (expressed as a fraction of the air flow rate at the nominal pressure)

2.3 SIMULATION RESULTS

2.3.1 IMPACT OF THE NOMINAL PRESSURE

The same building was simulated with the same climatic conditions using fixed ventilation devices giving the nominal airflow rate (30 m³/h for each office) at 1 Pa, 2 Pa and 20 Pa.

The simulations were performed for the whole test reference year with a time step of 1 hour.

The following figure shows the yearly average airflow rates in the 4 different offices as well as the standard deviations. The next one gives the yearly energy consumption due to ventilation per m² office floor area.
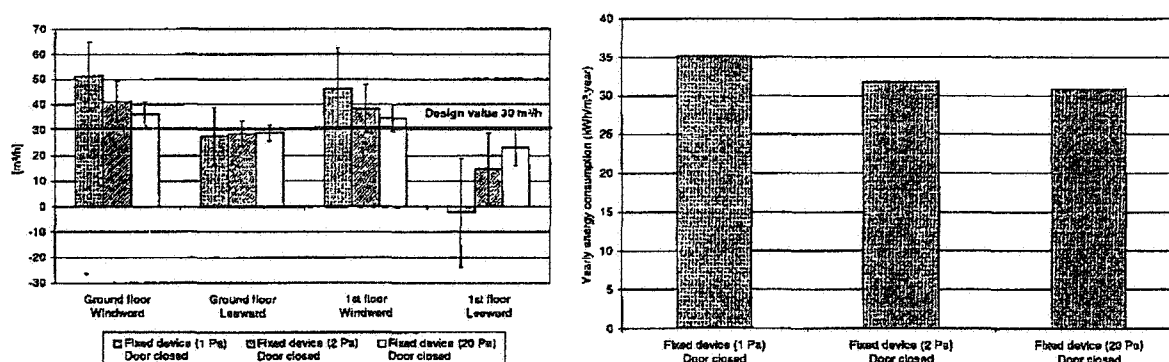
Figure 3 – Yearly average and standard deviation of the airflow rates in the 4 offices and total yearly energy consumption per m² floor area (case 1, 2 and 3).

The next figure represents histograms of the airflow rates in the different offices for a test reference year simulation (climatic conditions of Uccle, Belgium).
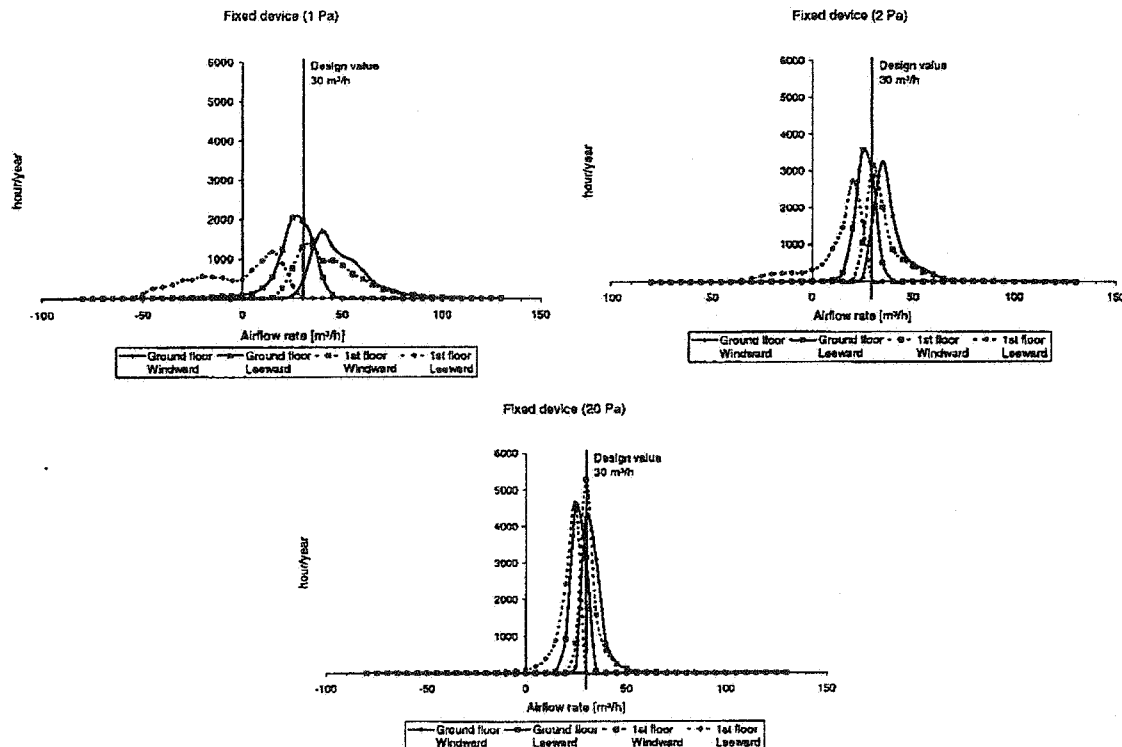
Figure 4 – Histograms of the airflow rates in the 4 offices (cases 1, 2 and 3).

As it can be seen on the previous figures, the higher the nominal pressure (the smaller the opening), the lower the variations of the air flow rates around the design values. The fan can always counteract the pressure drop through the ventilation devices (1, 2 or 20 Pa) but the wind induced and thermally induced pressure differences entail smaller airflow rates, as the openings are smaller.

The Figure 3 shows that the average airflow rates vary substantially from one room to another, the most underprivileged being the one situated leeward at the first floor because the stack effect and the wind effect play in the wrong direction. The energy consumption is about 12% less for a 20 Pa nominal pressure than for a 1 Pa nominal pressure.

2.3.2 WINDOW OPENING IN ONE OFFICE

The cases 1 and 3 were simulated again but with an open window (windward ground floor office) and all the internal doors open. Results are given in the next figures.
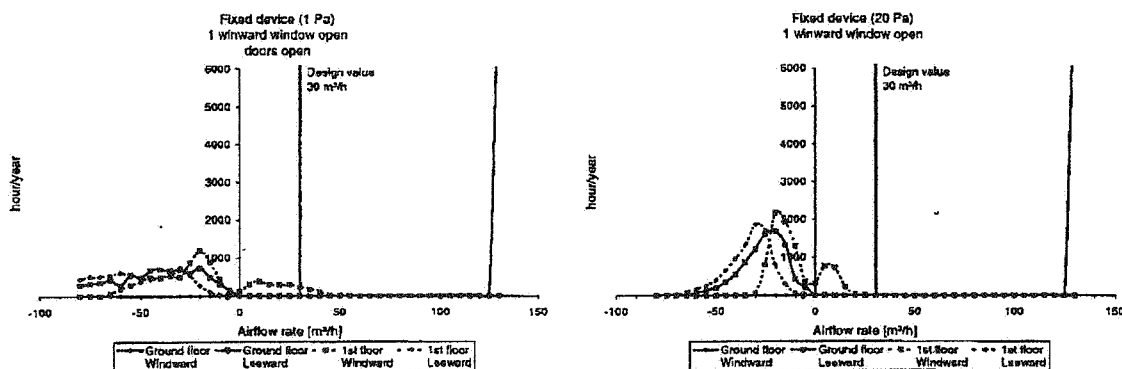
Figure 5 – Histograms of the airflow rates in the 4 offices (cases 4 and 5).

2.3.3 IMPACT OF SELF-REGULATING DEVICES

The next figures show the simulation results when using self-regulating devices instead of fixed ventilation devices. The doors are kept closed.
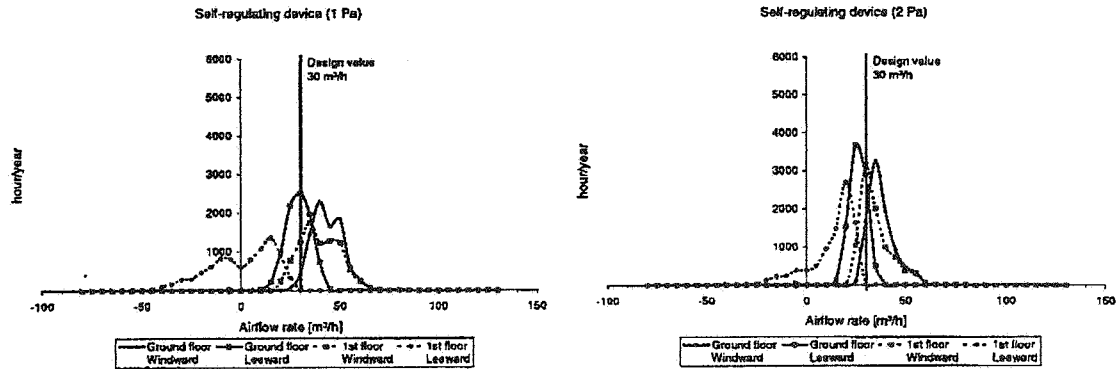
Figure 6 – Histograms of the airflow rates in the 4 offices (cases 6 and 7).

When comparing these results with the two first ones of Figure 4, one can see that the impact of the self-regulation is small. This is due to the relatively resistant transfer grilles that increase the pressure drop through the adjacent room and therefore limit the effect of the self-regulation.

The next 4 figures compare self-regulating and fixed devices for a situation where all the internal door between the corridor and the offices are open.
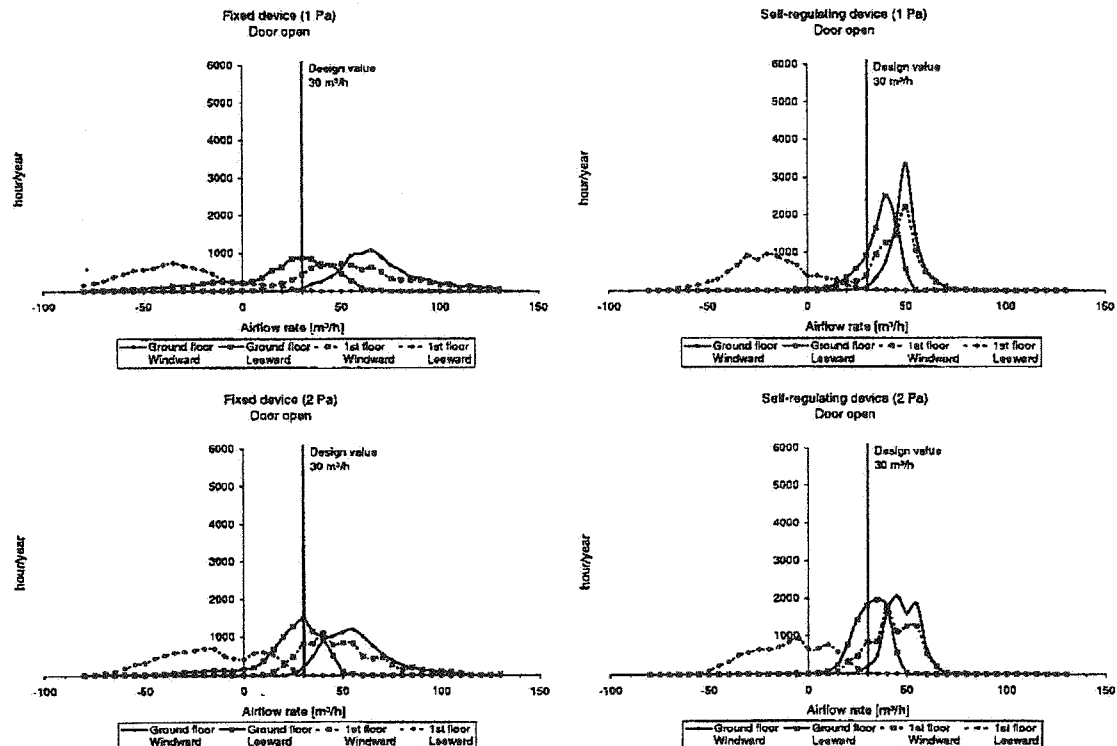
Figure 7 – Histograms of the airflow rates in the 4 offices (cases 8, 9, 10 and 11).

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As it can be seen, the impact of the self-regulation is much bigger.

3. CONCLUSIONS

1. This paper presents the concept of the Ventex programme, allowing to predict in a straightforward way air flows in a multizone building
2. Results of the impact of the nominal pressure for natural supply openings on the airflow rates are presented for an office building with mechanical extraction. The impact of use of internal doors is presented as well as the contribution of self-regulating devices.
3. Further work is needed before clear conclusions can be drawn on the impact of nominal pressure on airflow rates, indoor air quality and energy use.

4. REFERENCES

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2. A Guide to Energy Efficient Ventilation, March 1996.
3. BBRI Technical Note 203, Ventilatie van woningen – Deel II – Prestaties van veritilatiesystemen, BBRI 1997.

5. ACKNOWLEDGEMENTS

VENTILATION AND COOLING
18TH AIVC CONFERENCE, ATHENS, GREECE
23-26 SEPTEMBER, 1997

Title: **Natural Ventilation of the Contact Theatre**

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** Short Ford Associates, London, UK

Extended Abstract

This paper will describe the design and development of the natural ventilation system of the new Contact Theatre Complex designed by A.Goldrick of Short Ford Associates, now beginning construction in Manchester,UK.

The ventilation design is based on a stack dominant system using an 'H-Pot' chimney configuration. The paper describes the development of the design of both the studio theatre and main auditorium ventilation systems. These have been developed with feedback from wind tunnel and CFD testing so as to produce a strategy and design relatively insensitive to wind direction, yet providing sufficient ventilation to overcome the high heat gains expected from an audience and stage lighting. The potential for conflicts between wind and buoyancy forces have been reduced through the location and positioning of inlets and through the sizing and termination design of the stack.

Main	Rehearsal	Foyer	Foyer	Studio	Entrance
Stacks	Stacks	Stack	Inlet	Stack	
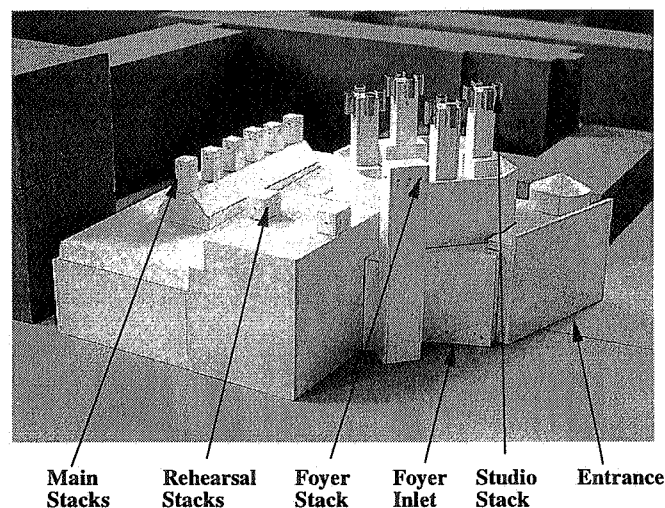
Figure 1 Wind Tunnel model of Contact Theatre initial design.
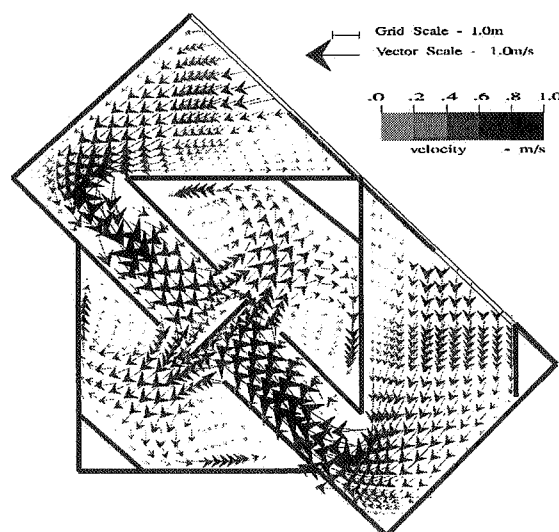
Figure 2 CFD analysis of airflows within mixing chamber beneath Studio theatre.

VENTILATION AND COOLING

18TH ANNUAL AIVC CONFERENCE
ATHENS, GREECE, 23-26 SEPTEMBER, 1997

NATURAL VENTILATION AND THE ROLE OF PASSIVE STACK CHIMNEYS IN TRADITIONAL EXCAVATED AND SURFACE DWELLINGS IN SANTORINI

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NATURAL VENTILATION AND THE ROLE OF PASSIVE STACK CHIMNEYS IN TRADITIONAL EXCAVATED AND SURFACE DWELLINGS IN SANTORINI, GREECE

K. TSIKOURIS and A. YOUNG

1. Synopsis

This paper considers the role of passive stack chimneys in controlling indoor thermal conditions in the vernacular houses on the volcanic island of Santorini . The quality of the environment within these dwellings is disputable, mainly because of the high humidity levels. A monitoring study was carried out in four actual dwellings in Santorini, two built on the surface and two excavated into the soft volcanic rock.. The temperature and relative humidity of their main space and their chimneys were monitored and compared to the simultaneous external conditions.

The results of this study were then used in a computer simulation package, modelling the performance of the dwellings and the chimneys in terms of air movement and air change rates. This showed that in most cases, chimneys proved to be efficient, establishing continuous air movement if located correctly, i.e. in the space where ventilation is mostly needed. The air flow characteristics of the chimneys seemed to be based on a diurnal cycle related to the external temperature fluctuation, but with a time lag.

By designing a chimney carefully and using the materials in an appropriate way, the ventilation problems of these dwellings can be solved at low cost, both in terms of running costs and energy consumption. In the last few years, natural ventilation has been adopted by many European designers, as the importance of energy conservation is increasingly realized. The study of the role of passive stack chimneys in natural ventilation can not only be useful for the restoration of such vernacular dwellings, but can also be used in the design of new, environmentally friendly, buildings.

2. The Indoor Environment of Dwellings in Santorini

2.1 Introduction

The volcanic island of Santorini belongs to the Cyclades, a cluster of over thirty islands in the Greek archipelago in the Aegean. Since the old geological periods, Santorini has been shaken by the explosions of the adjacent volcano, as well as by earthquakes, the most recent and destructive being the one that occured on the 9th of July 1956. The volcanic materials are visible everywhere and their presence has dominated the lives of the people, the agriculture and the architecture of the island. The volcanic rock which covers the entire island provides a firm and stable, yet easily worked, material for excavating dwellings that will not collapse, and also provides building blocks for surface dwellings. The uncommon landsacpe of Santorini was favourable to the development of a local architecture of extreme singularity, with excavated dwellings being the basic means of housing. A narrow facade and an elongated vault is the typical and most common layout of most of the excavated dwellings and indeed some of the surface ones.

2.2   The Potential of Natural Ventilation and the Role of Chimneys

An excavated dwelling is substantially a cave. It is surrounded by ground, which provides a huge thermal mass with an astonishing thermal capacity. Therefore this heavy construction has a thermal behaviour which is favourable to the dwelling with regard to temperature. Heat is absorbed by the ground during the hot days and given back at the cooler night-time, maintaining a stable temperature inside, pleasant in the summer and usually in winter, too. But the main disadvantage of an excavated dwelling is the poor air quality and lighting indoors. Since the building is surrounded by ground on all sides but the front, the later is the only way through to the outside air. Openings can only be situated at this front, normally narrow, facade. Daylight penetrates as deep as the first room of the house. Any space situated at the back of the house is hardly ever reached by any daylight at all.

Things get even worse when it comes to natural ventilation. Single-sided ventilation would only be adequate for a depth of up to 5 or 6 m. But excavated dwellings are usually much deeper than this and only the front room has proper ventilation. Since the planning is deep and cross-ventilation is impossible, the spaces at the rear of the house do not usually have any ventilation at all. The air in these spaces changes rarely. Excess moisture remains in the room and, consequently, extremely high relative humidities occur, as the temperature is most of the time kept at low levels. As a result, dampness, odours, mould grouth and condensation on the wall surfaces are common in excavated buildings.

The most common technique applied to assist natural ventilation was the chimney. The operation of a chimney is based on the stack effect. The air tends to move from a space with a high temperature to one with a lower temperature. In cold climates, where the chimneys are used exclusively for the removal of the (hot) smoke to the (cold) outside air, this means that there is a continuous air outflow from the chimney, via an upward air movement. In hot climates like the one of Santorini, where the chimneys also play the role of a ventilation opening in summertime, the flow of the air within the chimney may change direction depending on the various differences between the inside and outside temperatures.

The direction and speed of the wind also affect the performance of a chimney. The wind pressure at the top of the chimney is always negative and, therefore, if the temperature difference is very small, can produce an outflow at the top of the chimney irrespective of any stack effect which may want to drive the flow in the opposite direction. The construction, the shape and the orientation of the chimney determine its performance, as they relate closely to the effect of temperature and wind. Chimneys in excavated buildings in Santorini have a large thermal mass which is provided by the surrounding ground and respond very slowly to the changes of the external conditions. This time lag is expected to create large air temperature differences between the chimney, the external and the internal spaces of the house, which can be favourable to the ideal ventilation performance of the chimney.

A decisive factor for the efficiency of a chimney and its contribution to the natural ventilation of the house is the location within the building. Chimneys located in small isolated rooms are not likely to play an important role in the ventilation, since the air flow to the rest of the house is obstructed. Chimneys near large openings at the front of the house are also expected to be problematic, because the air moving in or out through the chimney will be expelled or drawn in through the front openings without ventilating the rest of the dwelling. Chimneys built at the back of the house or attached to the main room should prove more effective and contribute to the overall ventilation, because they will introduce an alternative to cross ventilation and create air draughts at the rear.

3. The Monitoring Study

3.1 Methodology

In order to demonstrate the effectiveness of chimneys as a means of natural ventilation an experimental study was carried out. Four buildings were selected for monitoring in terms of their environmental conditions. "Smart Reader"[1] data loggers were used, together with the "Trend Reader" [9] software package to record and download readings for temperature and relative humidity. Monitoring took place for one week. For each of the four buildings a data logger was positioned in the middle of the height of the chimney and another was hung in the internal space, at head height. Another logger for monitoring the external conditions, was positioned so that it was always in shade. The location of the loggers in each building is shown on the floor plans (fig. 1-4).

The temperatures obtained were used as input values to the natural ventilation computer simulation program, BREEZE, (Produced by the UK Building Research Establishment) [2] which was used to predict air flow rates in the chimneys and air change rates for the various spaces in the dwellings. The results were used as an interpretative tool to explain the operation of the chimneys and to determine the overall effectiveness of the natural ventilation in these dwellings.

3.2 The Four Case Study Buildings

Four buildings were selected for study from the many traditional buildings on the island, see figures 1 to 4, and were representative of the basic types to be found. They have all maintained their original layout with almost no damage or alteration to the original materials and architectural elements. An original chimney exists in every one of the selected buildings, without any significant change to its initial shape and materials. Buildings 1 and 2 are excavated into the ground whereas 3 and 4 are surface buildings.
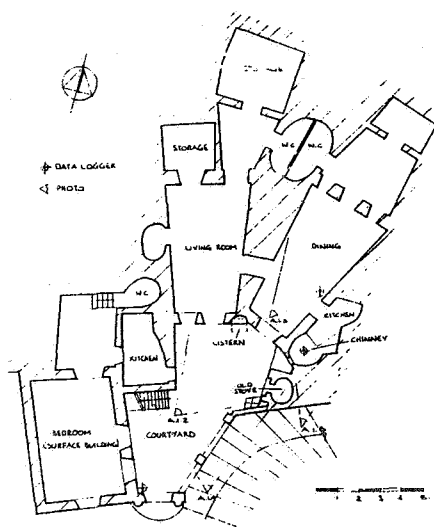
Fig. 1 Building 1: Floor Plan
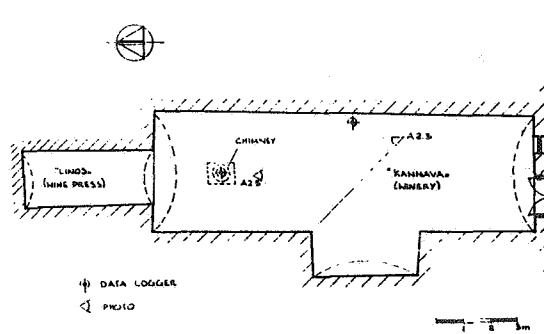
Fig. 2 Building 2: Floor Plan

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[1] Smart Reader and Trend Reader are registered trademarks of Status Instruments LTD

In building 1 the chimney is round and has been built in the kitchen above the stove (fig. 1). Half of it is excavated from the ground. When reaching the surface it forms a part of the adjacent public stair's parapet. The rear space of the house is not ventilated at all. In building 2 the chimney is actually a 60cm diameter hole on the roof, which brings light and fresh air into the deep-plan excavated space (fig. 2). Considering that the roof is in reality the ground above, the chimney is a 2.7m high excavated cylinder. The chimney in building 3 is partially excavated, with its main body built on the surface of the ground (fig. 3). In building 4 the chimney is located at the rear of the house, in the room which was probably initially a kitchen (fig. 4). It has an open top and the blowing winds (regardless of their direction) create a negative pressure and support an air movement in the chimney, even when the difference between the external and internal temperature is very low.
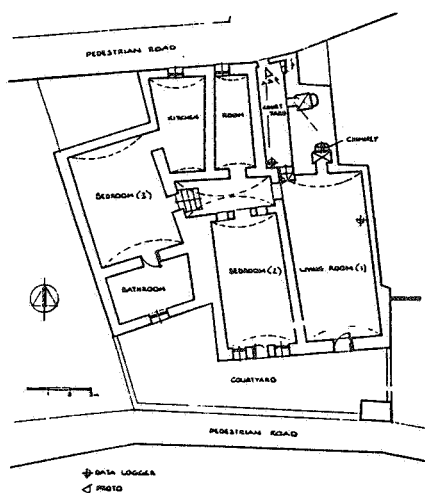
Fig. 3 Building 3: Floor Plan   Fig. 4 Building 4: Floor Plan
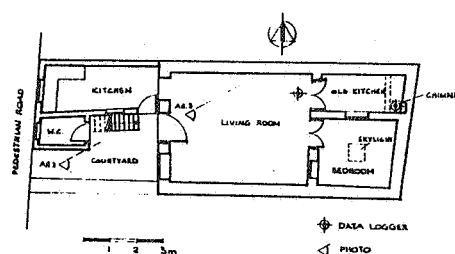
3.3   Results and Discussion

A summary of the results of the monitoring procedure is presented in Table 1 and sample graphs representing the monitored conditions in building 3 are given in Charts 1-2. In all cases the internal temperature is very stable, at relatively low levels, primarily due to the high thermal mass of the surrounding ground, or the thick walls (in the case of the surface dwellings) which are made of lava rock. The solar and internal heat gains are absorbed by the thermal mass at daytime and given back during the night, when the ambient temperature is lower.

The internal relative humidity is also relatively stable compared to the external one, although it follows the fluctuation of the external conditions by a time lag and it is very common for it to reach very high figures, see Chart 2. The monitored values for relative humidity and temperature for the four buildings were converted to absolute moisture content values, with the use of the equations suggested in [6]. Chart 3 shows the calculated moisture content for the external air and the internal ambient air for building 3, where it can be seen that the moisture content inside is at all times greater than the simultaneous moisture content outside but shows similar trend. This would seem to indicate that the moisture content inside is not determined purely by infiltration and natural ventilation from outside. In Building 3 it appears that other major sources are affecting the moisture content, such as moisture

absorption and release from the walls, which could be significant due to the high porosity of the volcanic material of which the building is constructed. However, the plot of moisture content values for building 4 (Chart 4) shows that the moisture content of the external and the internal air are not very different. In this case the moisture content inside is governed by the one outside. Natural ventilation is therefore probably more effective in Building 4, which could be attributed to the position of the chimney (at the rear space of the house).

		EXTERNAL				INTERNAL				CHIMNEY			
		Min	Mean	Max	Range	Min	Mean	Max	Range	Min	Mean	Max	Range
Building 1	Temp. (°C)	22.3	30.3	42.0	19.7	24.1	25.9	29.9	5.8	27.0	29.9	41.1	14.1
	R.H. (%)	16.3	38.0	78.0	61.7	37.1	54.2	64.6	27.5	33.2	46.8	70.9	37.7
Building 2	Temp. (°C)	22.3	30.3	42.0	19.7	22.0	23.8	26.3	4.3	22.0	25.0	29.2	7.2
	R.H. (%)	16.3	38.0	78.0	61.7	50.0	69.0	77.6	27.6	41.4	62.9	78.3	36.9
Building 3	Temp. (°C)	21.6	27.3	37.3	15.7	23.7	24.6	25.5	1.8	22.3	26.2	29.9	7.6
	R.H. (%)	20.7	48.5	86.9	66.2	58.2	73.6	80.4	22.2	30.6	60.4	87.3	56.7
Building 4	Temp. (°C)	21.6	27.3	37.3	15.7	26.2	27.0	28.0	1.8	23.4	27.6	34.9	11.5
	R.H. (%)	20.7	48.5	86.9	66.2	33.9	47.6	58.9	25.0	19.1	42.9	68.8	49.7

Table 1: Monitored Conditions for the four Buildings
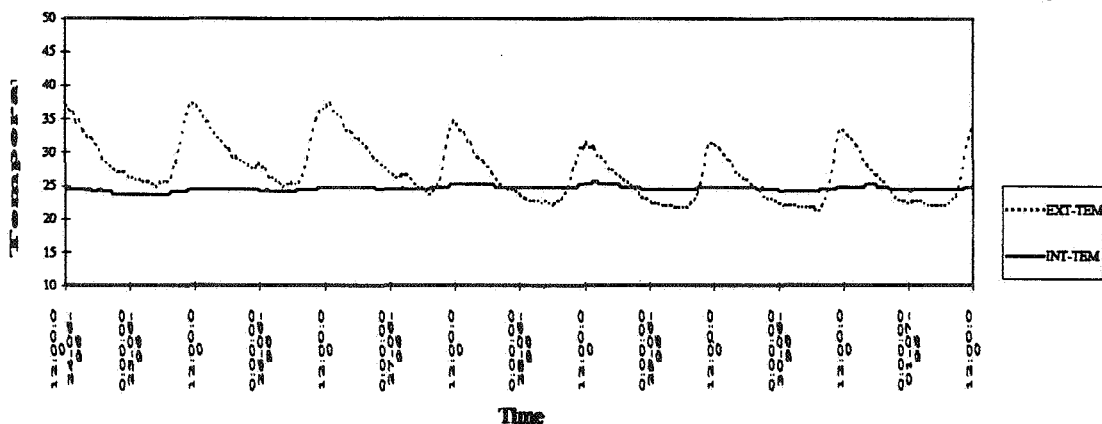
Chart 1: Building 3 - Temperature
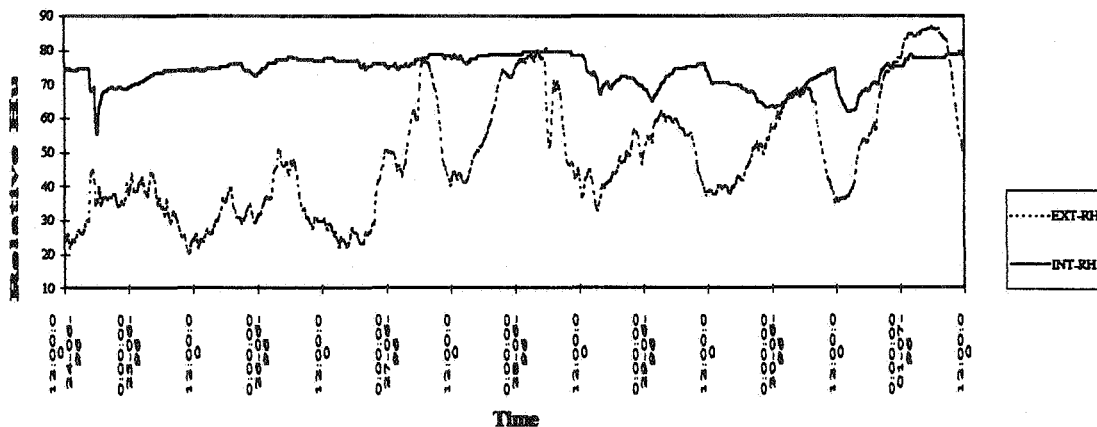
Chart 2: Building 3 - Relative Humidity
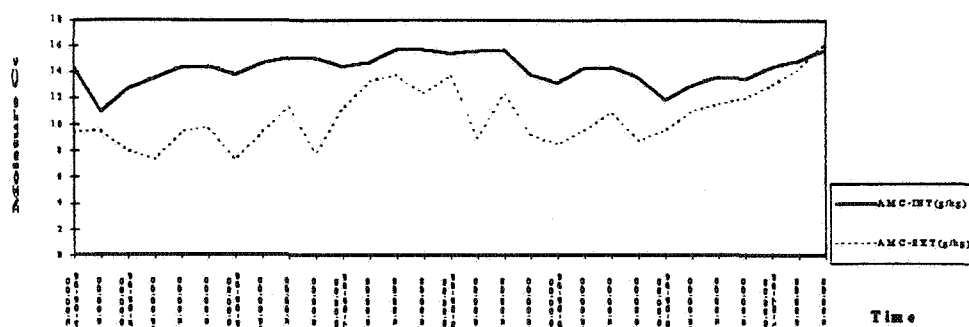
Chart 3: Absolute Moisture Content - Building 3
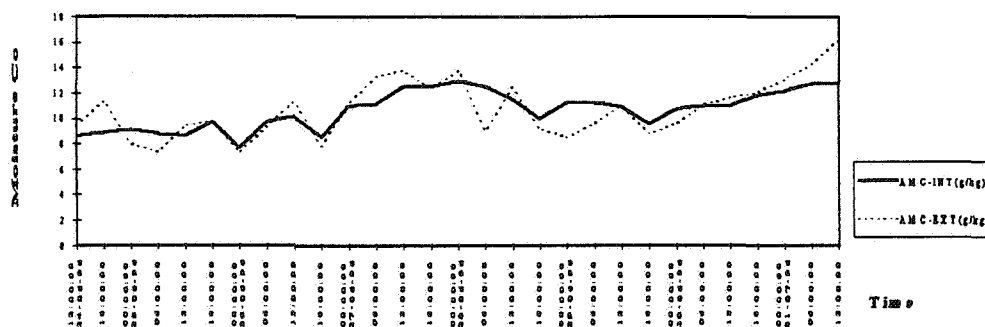
Chart 4: Absolute Moisture Content - Building 4

Initial consideration of the differences in the recorded temperature between the chimney and the external air would suggest that the chimneys operate on a diurnal cycle in the following way: the outside air temperature in the early afternoon is at its highest and is considerably higher than the inside, resulting in a downward air movement in the chimney. Direct gains from the sun and heat from the incoming air are absorbed by the walls and the roof of the house and from the chimney walls. The thermal mass gives back the heat at night, keeping the room temperature higher than the external which has decreased. Now the air flows in the chimney in the reverse direction, moving upwards. The temperature outside gradually increases as the day progresses and at some time the inside and outside temperatures become equal. At this instant, theoretically, there is no air flow in the chimney. In the early afternoon the outside temperature comes to its maximum again and the cycle repeats itself (See chart 5).

The monitored temperature data were used with the computer simulation software, BREEZE, [2] to model the natural ventilation of the buildings. These simulation results are given in Table 2, where column 3 gives the maximum daytime and the minimum night-time external temperatures, and column 7, the maximum and minimum temperatures inside the buildings and in the chimneys. The BREEZE-predicted total volumetric air-inflow and outflow rates for these temperatures are given in columns 8 and 10, respectively for the chimneys and the main spaces of the buildings. Also shown, where the flow through the chimney is non-zero, is the percentage contribution of the chimney to the overall air flow in the building. The final column expresses the volumetric air flow rates as air changes per hour for the whole building. It is reassuring to see that BREEZE predicts approximately equal air inflow and outflow for each building. Table 2 shows that the chimneys provide a significant contribution to the overall natural ventilation of the buildings. With the exception of Building 2, it can be seen that air flows down the chimney during the day, and up the chimney at night, as was expected. In Building 2, predicted air flow for the monitored temperature conditions is

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as was expected. In Building 2, predicted air flow for the monitored temperature conditions is always downwards. One possible explanation is that, because building 2 is deeply excavated, it stays cooler than the outside air at all times, thus creating a constant downward flow.

Chart 5 shows the performance of Building 4, in terms of air flow rate and direction through the chimney, and their change over time and with internal/external temperature difference. The external and room temperatures were those recorded over one day in the building during the monitored period, and the air flow rates are those predicted by BREEZE, using these temperatures. It can be seen how the direction of flow changes when the room temperature becomes greater or less than the external temperature. Table 2 shows that the buildings experience quite high ventilation rates. Although these fairly high ventilation rates maintain acceptable air quality in those rooms which experience the ventilation, humidity levels are generally unacceptably high in rooms isolated from the main air movement paths, and it is this aspect of these buildings which makes their rehabilitation most problematic.

The overall effectiveness of a chimney is determined by a number of factors, the most important of which are its location in the house and the height of the chimney. Chimneys located at the rear of an open-plan dwelling were shown to operate much more effectively than chimneys in small rooms or near large openings at the front, like the chimney in building 1 (fig. 1). Chimneys with a height greater than the surrounding buildings benefit from winds, when they are blowing, to create a wind-driven air flow, particularly useful when the temperature difference between the top and the bottom of the chimney is small.

It is difficult to improve a problematic chimney. Sometimes chimneys are in small, separate rooms, like the one in building 3. In this case, opening a large passage to the other rooms might improve the performacne of the chimney and its contribution to the overall ventilation. In the case of a chimney built lower than the surrounding buildings, the height of it should be increased. In vernacular buildings this is not always acceptable, because it alters the form of the house. However, it has been shown from this study that a well-designed chimney can be used as a means of improving natural ventilation, offering an alternative to cross ventilation, wherever the later is difficult to achieve.

		Ext.T. (°C)	Floor	Area m²	Volume m³	Temp. °C	INFLOW m³/h	%	OUTFLOW m³/h	%	A.C./h
Building 1	Day	42	Building	90.5	2 17.3	30.0	4268.7	83.3	5 104.6		
			Chimney	0.1	0.1	37.0	855.3	16.7%	0.0		3.94
			Total	90.6	217.4		5124.0		5104.6		23.57
	Night	22.5	Building			24.0	1939.7		1515.2	78	
			Chimney			27.0	0.0		428.8	22%	1.96
			Total	90.6	217.4		1939.7		1944.0		8.92
Building 2	Day	42	Building	81.8	245.4	26.5	2204.0	29.8	7264.4		
			Chimney	0.3	1.1	28.0	5187.1	70.2%	0.0		21.05
			Total	82.1	246.5		7391.1		7264.4		29.99
	Night	22.5	Building			22.0	337.0	22.5	1496.6		
			Chimney			22.0	1159.6	77.5%	0.0		4.70
			Total	82.1	246.5		1496.6		1496.6		6.07
Building 3	Day	37.5	Building	102.2	306.5	26.0	8882.7	93.7	9470.2		
			Chimney	0.4	0.9	30.0	595.4	6.3%	0.0		1.94
			Total	102.6	307.4		9478.1		9470.2		30.84
	Night	22	Building			23.5	3539.6		3334.2	94.2	
			Chimney			22.5	0.0		204.7	5.8%	0.67
			Total	102.6	307.4		3539.6		3538.9		11.52
Building 4	Day	37.5	Building	40.5	117.7	28.0	6114.0	85.5	7124.0		
			Chimney	0.1	0.1	35.0	1033.5	14.5%	0.0		8.8
			Total	40.6	117.8		7147.5		7124.0		60.7
	Night	22	Building			26.5	5093.1		4579.3	90	
			Chimney			23.0	0.0		507.7	10%	4.32
			Total	40.6	117.8		5093.1		5087.1		43.25

Table 2: Results of Modelling Natural Ventilation
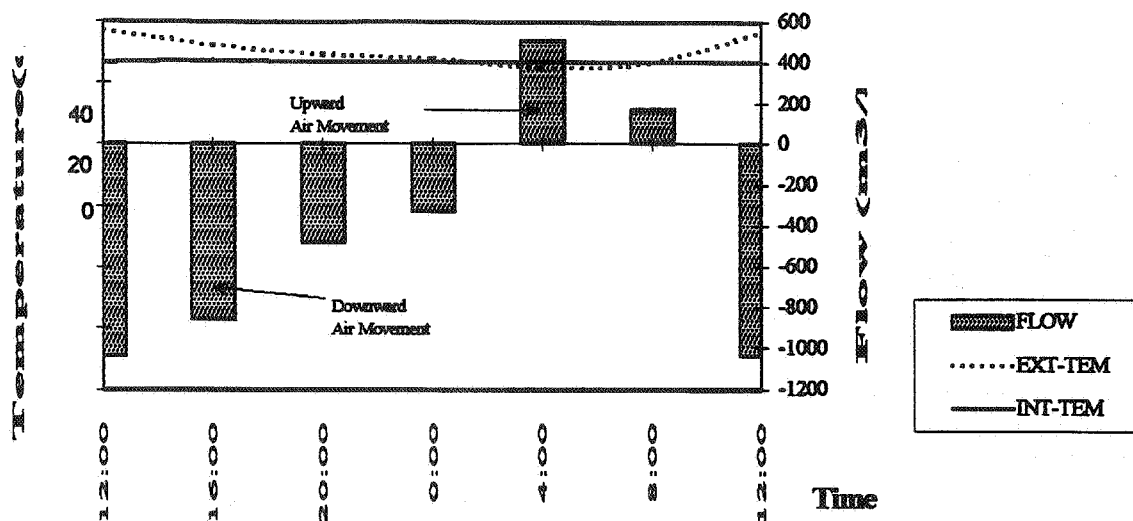
Chart 5: The Diurnal Ventilation Cycle for Building 4

4. Summary and Conclusions

The results of this study were based on the monitoring of four selected buildings. In all cases it is obvious that the high thermal mass has a very important role, keeping the temperature stable and at a low level. However, the excavated buildings are deep in plan and cross ventilation is impossible, resulting in high relative humidities. It is here that chimneys prove to be very important. The more effective chimneys, like the one in building 4 increased the ventilation, giving a high ventilation rate and maintaining the humidity to outdoor values. The inappropriate location of some of the existing chimneys for beneficial natural through-ventilation calls into question whether the original vernacular builders fully understood the role of the chimneys. The preferred location seems to have been determined by the position of the cooking stove (often positioned near the entrance to the dwelling) rather than to promote good overall air quality and humidity control.

The experience gained from this study can be used in new buildings in Santorini or elsewhere. A chimney carefully designed and located at the right place has been shown to be able to provide ventilation to deep planned buildings or ones which cannot have openings on all sides. Nowadays the construction methods and materials are different to the traditional ones. In Santorini, excavated dwellings are no longer constructed, although existing excavated dwellings are restored and deserted ones are rehabilitated. Concrete and brick are the materials used to build new buildings. Contemporary architects, designing new buildings for the island and re-furbishing the old, should learn from the experience gained from studying these houses. Their designs should reinforce the benefits of the old-style vernacular buildings, such as high thermal mass, and should tackle the draw-backs, such as poor lighting and air quality. By employing well-designed chimneys to promote high rates of natural ventilation to control humidity levels, for example, architects may be able to respond to the contradictory needs of providing comfortable modern housing, without spoiling the fragile architectural quality of the island.

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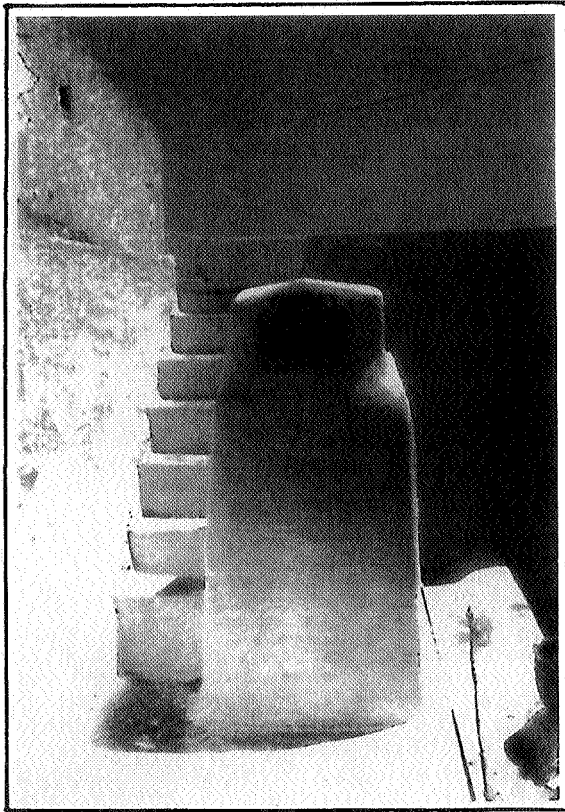
the chimney in Building 3
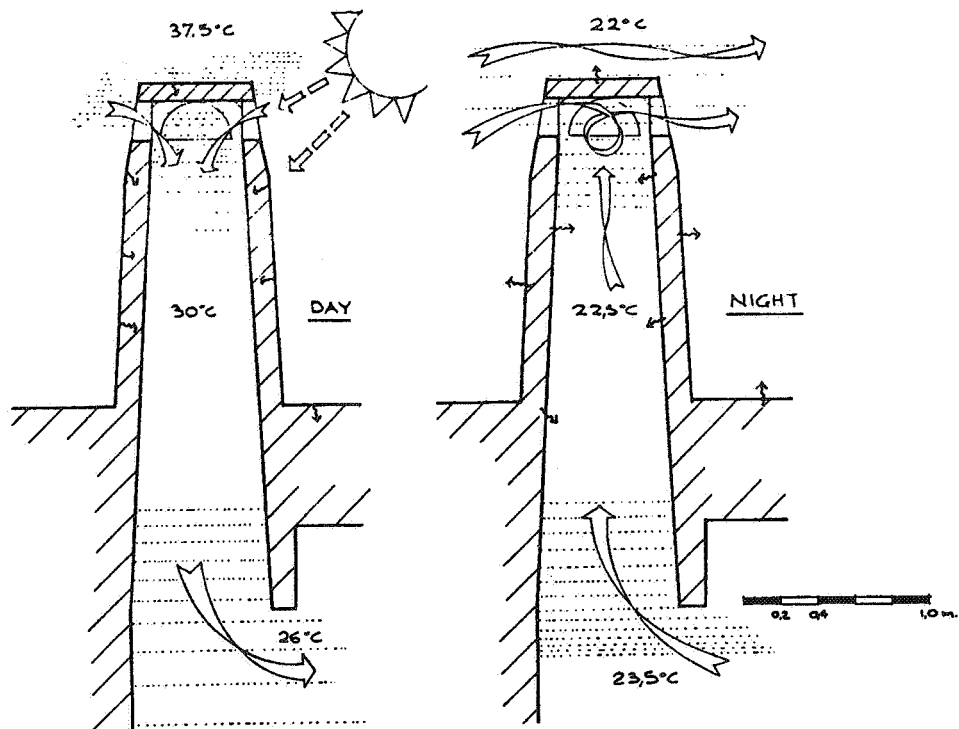
37.5°C

22°c

30°c DAY

22,5°C NIGHT

26°c

23,5°C

0,2 0,4 1,0 m.

VENTILATION AND COOLING

18TH ANNUAL AIVC CONFERENCE
ATHENS, GREECE, 23, 26 SEPTEMBER, 1997

INDOOR AIR QUALITY AND NATURAL CONTROLLED VENTILATION - EXIGENCE, PERFORMANCE AND STANDARD ASPECTS

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INDOOR AIR QUALITY AND NATURAL CONTROLLED VENTILATION - EXIGENCE, PERFORMANCE AND STANDARD ASPECTS

Synopsis

As everybody knows, today the air quality of an indoor environment may have several effects on our health; the beginning of serious breathing pathologies and of some forms of cancer, are with no doubt due to the presence of polluting and extremely noxious agents in the places we most frequently use.

That's the reason why it is very important that indoor rooms are correctly aired also in our homes where, due to several incidental factors, the healthiness of the environment is still guaranteed by the mere and discretionary operation of users of opening the windows.

In considering the growing attention drawn to these problems and in agreement with the provisions of the 3rd essential requirement laid down in Common Directive 89/106, 'Hygiene, Health and the Environment', ICITE has undertaken to develop a research and experimentation study aimed at establishing a device for the controlled natural ventilation of residential environments.

The main objectives have focused on the technical and performance-oriented characteristics of the devices that are already available on the Community markets and on the regulative aspects for what concerns air healthiness in domestic environments, while the final phase of the work, still in progress, will give new developping guidelines, both in regulative and productive terms.

1. Historical outline

The indoor ventilation problem is not related to modern history, but it concerns the human civilization from its origin. Cooling needs or simply smells removing, have beeen solved in different way over the centuries, according to the technological evolution and climatic conditions.

In any case, natural ventilation is based on the difference between indoor and outdoor pressure: when the indoor temperature exceeds the outdoor one, a continous air cycle is created throught the wall openings.

Early solutions were based on this very simple approach such as, for instance, wind towers, studied and realized during the Pre-Christian era from the Middle-East. These towers maintained their technical validity for many years.

The Roman age also considered ventilation as an health important factor. In fact they studied wind direction in order to improve building orientation. With regard to the Mitilene inhabitants, Vitruvio wrote: "When the Auster wind blows Mitilene's inhabitants fall ill, when the Mistral wind blows they have a cough, when the North wind blows they became healthy again".

The typical Roman house solved ventilation needs by creating open spaces inside: all rooms faced the impluvium and perystilium.

During the Middle Age new techniques were developed, based on the double air flow (drawing and ejection), and other passive techniques were developed to ventilate houses and castles. The ecclesiastical construction continued to apply Roman principles.

During the Renaissance the techniques based on wind studies were improved by introducing mechanical systems to stimulate air, and passive ventilation was left in part.

Also in Italy, mechanical ventilation was adopted, improving the performaces of early ventilators and using valves that, controlled by wires, sent air into those rooms only used from time to time.

Occasionally, wind towers were again adopted, which in Italy where known as "ventiere".
In the eighteenth century and during the Industrial Revolution, scientific progress led to the first forms of calculus for determining the unitary air flow to guarantee hygiene and health indoors.
England produced the most important resultus of this study at that time, where prisons were provided with heating and ventilation systems. The employment of prisoners to set their own ventilators was unique. Elsewhere, heating systems were steam-driven.
The systems set up at Westminster in London, in the first half of the nintheenth, re-adopted the concept of the wind towers (Victoria and Clock towers), by conducting air to the cellars, from where it reached the rooms and was then conveyed through the combustion systems which exploited the induction effect. Other examples of the survival of the wind towers were in Great Britain and America. Nevertheless the principle of ventilation linked to conditioning systems was maintained for a long time.
This principle is valid and operative even today, but the indoor ventilation problem for building without conditioning systems is still open.

2. Indoor air quality
The Common Directive on building products 89/106, relative to the proximaty of legislative, regulatory and administrative provisions of the Member States, adopted in Italy on 21 april 1993, with D.P.R. n. 246, specifies, among other things, the following essential requirements, to which every structure has to answer:
- Health, safety and environment
In order to satisfy this requirement, each structure has to be conceived and built so that it does not constitute a threat fto the health and safety of the occupants or neighbours. This threat is caused in particular by noxious gas, particles or dangerous gas present in the air, dangerous radiation emission, polluted or contaminated water or soil, evacuation defects of water, smoke and solid or liquid residuals and formation of humidity in some parts or within the interior surface of the structure.
In particular, the interpretative documents of the Product Directive, in underlining the principles to verify the respect of essential requirements, (re- point above-mentioned) indicate ventilation (natural or artificial) as one of the most important aspects in order to pursue healthy and safe environment in buildings.
In this way, the technical specifications require that the presence of polluting agents are brought back to safety levels, applying evalutation criteria of air quality, calculus methods to forecast the renovl rate, starting from climatic conditions and ventilation systems. Further measures referred to the calculus rate of the ventilation in building, determining its effectiveness, indentifying the kind of pollution and its concentration indoors.
In general, the complete characteristics of ventilation systems have to be verified in terms of their speed and air flow and differences of pressure.
The Directive 89/106 also highlights, through its *nouvelle approche* to building products, that past cares about indoor aeration have continued to increase during last few years and the absence of ventilation has been shown as a great threat to human health.
On the basis of a statistics study, carried out in the United States on a sample of 350 buildings, demonstrated that the effects due to the absence of ventilation (50%), the wrong air distribution, inadequate conditions of temperature and humidity, on top of tobacco smoke and endogenous pollution, are responsible for bodily discomfort known as 'sick building syndrome'. This syndrome causes various symptoms, such as headaches, fatigue, nose and throat irritation, etc.

The incidence of endogenous pollution is responsible for a smaller part of this temporary bodily discomfort, but they have a biggest responsibility to long term, in supporting great pathologies, not always curable with the up-to-date medicine. This is the case, for instance, of some adhesives and sealants delivering carcinogent and toxic substances, such as formaldhehyde, radon, etc., in the environment.

The absence of ventilation also allows the increase of relative humidity, which causes condensing phenomena and, as a consequence, mould formation accountable of respiratory pathologies.

3. The research proposal

These further elements of knowledge, point out very clearly that the termal and hygrometric performances of a building and its components, have to satisfy environmental safety and healty needs, with optimum energetic consumptions.

In fact, after superseding the economic restrictive worries, which imposed "hermetic sealing", the attention is today turned on indoor microclimate as an essential factor for the comfort of the occupants.

In this way, the frequence of air changes, especially in winter, is the crucial point of the problem, in considering that the energetic consumption has to be compared with air safety.

External windows have always met the primary need of air change allowing, with easy operations, the opening of one or more of its components, but living to frequency and the amount of air to be recycled to the discretionality of the occupants.

The increased attention drawn the objectives of safety and healthiness of domestic rooms has to be beyond the occupants' discretion, imposing on to external window additional performances able to guarantee permanent safety conditions.

The window, also known as 'the intelligence of walls', is still able to fulfil new expectations, as simply as in the past, but it needs receive new inputs in the technical and standard fields.

The modern construction sector in Italy, having to face several problems, doesn't seem to relying a satisfactory way on ventilation, and also the route of natural ventilation systems seems,to have still a long way to go.

In such a situation, indoor natural ventilation is affected by multidisciplinary factors, having to interact with technical, normative, sanitary and economic variables.

So, in order to give some acceptable answers to the up-to-date needs and to give useful orientation to the industry of the sector, and on the occasion of a specific demand coming from production world, the Systems and Components Department of Icite deemed it necessary to undertake a research study on the subject of natural controlled ventilation.

The work programme has prelimanarily, outlined the necessity to follow two directions:

- laboratory characterization and performance evaluation of a window filled with a ventilation unit;
- analysis and evaluation of national lows and standards, with reference to indoor ventilation criteria;
- analysis and evaluation of the effectiveness of operational instruments currently adopted.

In this field, the analytical phase of the study under way, aims at defining a reference frameworkin order to investigate and outline the existing gaps, concerning both need and performance aspects and legislative and normative aspects, all contributing to make a correct evaluation.

3.1 Laboratory characterization of the ventilation system and preparation of test methodology

The investigated ventilation system is of the self-adjusting type, it is conceived to be fitted in the upper part of the window, breadthwise, between the transom and the glazing unit, reducing this way the glazed surface.

Inside the system works a ventilation grid that with faint or no wind, allows for a satisfactory level of natural ventilation of the indoor environment, while it automatically stops working due to the action of the external pressure as soon as the atmospheric conditions change. The system is conceived in a way that it allows to keep acceptable conditions of indoor ventilation even when there is a strong wind outside.

The system works thanks to the internal grid with self-adjusting air intake consisting of punched thin plates oscillating perpendicularly to the ventilator and of two baffles, the former being placed on the outside for the natural collection of air, the latter inside, turned upwards, to convey the air flow indoors avoiding air draughts and rain water seepage.

According to the characteristics declared and checked in the laboratory, it is envisaged that with external air pressures greater that 20 Pa, the internal thin plates, by automatically placing themselves against the internal face of the system, manage to prevent a certain amount of air from seeping inside, by limiting the flow just to the air passing through the holes of the thin plates and throught the little gaps between them.

With pressures less than 20 Pa, the system must ensure a constant air flow lying between 15 m^3/h and 30 m^3/h, according to the size of the system.

In order to characterize the system, an "ad hoc" test chamber for the measurement of low pressures was used; the chamber was built according to the provisions contained in the Belgian standard NBN D 50-001.

The $3m^3$ test chamber was built with bolted and silicone-bonded sandwich panels and provided with vertical walls allowing to apply (see fig...) a constant air pressure in the inside.

The opposite closing walls were prepared to contain the test device and the air intake, with negative pressure, generated by a fan. A pressure gauge was used to measure the pressure difference between the test chamber and the laboratory, while inside the duct placed between the fan and the chamber, the air flow rate was measured by means of a Pitot tube. The device was installed with the internal face turned towards the laboratory and the external one towards the test chamber, in order to simulate, inside the chamber, the actual external atmospheric conditions.

3.2 Laboratory characterization of the window frame fitted with the ventilation system

The performance evaluation of the window system was carried out by means of official testing equipment for windows and structural glazings belonging to the "Components" service laboratory of ICITE, and thanks to the experience made over many years of activity in the field of compulsory and voluntary certification. In this connection, it must be remembered that ICITE is an active member of UEAtc and EOTA, the European organizations in charge of the harmonization of standards for granting technical agréments.

The whole system, consisting of an aluminium window frame with a horizontal bascule opening and provided with a ventilation system, was then fitted into the test wall and tested according to the methodologies envisaged by standards UNI EN 42-77.

Air permeability was measured by constantly increasing pressure at 50 Pa steps, from 50 to 500 Pa. The test was at first executed by sealing the system in order to assess the extent of the losses caused by the window frame, then, after removing the seal, the total losses were

recorded; the difference between the two values provided the amount of air passing through the system. Permeability values (table....) resulted to be quite high, thus contradicting the limits suggested by the official standards.

The watertightness test was not judged to be meaningful since it is not possible to make a comparison with the criteria suggested by the standards establishing that the nozzles used to spray water on the sample are to be fixed few centimeters below the upper transom of the window frame in the same place of the test set-up in which the ventilation system is to be installed.

The wind resistance tests previously carried out on the window frame not including the ventilation system, allowed to assign the highest resistance class. The same window frame, including the system and submitted again to the previous test, did not resist the 1800 Pa pressure due to the disjuntion of the lateral glazing beads.

4. National legislations and standards and assessment of the effectiveness of currently used working instruments

4.1 Indoor ventilation

The present national legislative situation on the subject of ventilation refers to Law No. 10 of 1991 "Standards for the accomplishment of the national energy Programme concerning the rational use of energy, energy saving and development of renewable energy sources" which somehow introduces the problem regarding the change of indoor air.

This law is very important since it goes beyond the restrictions imposed by Law 373/76, issued following the serious energy crisis which took place during the 70's, and it introduces important innovations about how to plan and realize living comfort and hygiene; moreover, it is structured on three levels of enforcement and this shows that the CPD has been satisfactorily adopted.

While the first level of the legislative apparatus provides the general directives, the intermediate level, consisting of the compulsory D.P.R. (Decree of the President of the Republic) No. 412, precisely defines and reaffirms the basic role of thermal insulation in view of energy saving and environmental welfare, delegating the application procedures to the technical standards drawn up by UNI.

In particular, D.P.R. No. 412 of 26 August 1993 "Regulations containing standards for the planning, installation, operation and maintenance of thermal plants of buildings in order to restrict energy consumption, in accordance with article 4, sub-section 4, of Law 10/'91 and following amendments" defines, among other things, the climatic areas, subdivides the buildings according to their intended use and provides the specifications of the plants. As regards dwelling buildings and buildings with similar intended uses, the D.P.R. assumes a room temperature, during the winter working period of the air-conditioning unit and a maximum value determined on the basis of the arithmetical mean of air temperatures of all the individual premises of the buildings, defined and measured according to the specifications contained in the technical standard UNI 5364.

Article 8 of the Decree also indicates the daily mean over 24 hours of the minimum number of air volumes that can be recycled in one hour, fixing it conventionally to 0,5 for dwelling buildings, if no controlled mechanical air changes are envisaged.

The whole regulative course outlines the calculation of energy requirements for an indoor environment as a physical magnitude depending on several interacting factors. In this sense, the ventilation requirement should act to correct and integrate the building's performances, to intermediate between the inside and the outside but , as specified by UNI 5364, it also depends on the intended use of the considered room, on its type, extent, orientation, on the resistance of frames, etc.

Nevertheless, as far as ventilation is concerned, the conventional value fixed by D.P.R. No. 412, is still 0,5 m^3/h.

What's more, in some cases, regional regulations provide for minimum ventilation limits; that's the case of Lombardia where the hygiene regulation in force establishes that for private premises the external filtered air change should not be less than 20m^3/h per person.

Uncertainties and misunderstandings could arise from such a complex situation; hence the need of having a national law taking charge of providing all necessary references, also related to the progress of other countries in this field.

For what concerns the dimensioning of ventilation grids, on the national level there are specific standards establishing the air flow rate values for ventilation according to the specific living premises. It's the case of the Belgian standard and of the Dutch standard:

NBN D 50-001

Destination des locaux	Débit (m^3/h)		
	Min.	Nominal	Max
Local de sejour	75		150
Chambre à coucher Chambre hobby ou étude	25		36 / personne
Cuisine Salle de bains Buanderie, local de sèchage et espaces analogues	50	surface (m^2) x 3,6	75
Couloirs, escaliers, hall de jour et de nuit et espaces de passages analogues	-	surface (m^2) x 3,6	-
Cuisines avec un passage ouvert vers d'auters espaces ou locaux	75		-
WC	-	25	-
Cage d'escalier	1/2 x volume par heure	-	-

NEN 1087

Destination des locaux	Débit nominal (dm^2/s)
Local de sejour	1 par m^2 min 21
Cuisine Salle de bains Buanderie Local de sèchage Espaces analogues	21 14 14 14 7
Cuisines avec un passage ouvert vers d'auters espaces ou locaux	1 par m^2 min 21
Chambre à coucher Chambre hobby ou étude	1 par m^2 min 7
Combles	1 par m^2 min 7
Garage	3 par m^2
Local poubelles > 3m^2	100
Ascenseur	1 par personne
Cage d'escalier	1 x volume par heure

4.2 Standards for windows

The standards sphere specifically related to windows and to the relationship existing between windows and the different natural ventilation systems, deserve a thorough investigation.

The national standard UNI 7979 establishes the criteria for classifying windows according to their performances related to air permeability, watertightness and wind resistance, tested in testing laboratories according to the methods specified by the relevant European normative provisions, that have been adopted in Italy as UNI EN 42, UNI EN 86 and UNI EN 77.

Air permeability can be defined as the air amount, expressed in m^3/h, managing to pass through the close window system, due to the pressure difference between the external and the internal surfaces. The value measured in the laboratory, by submitting the window to an air mass under static pressure, is the indication used to assign one of the three classes of performance. The results of this test refer to the ratio between the square metres of opening surface ($m^3/h.m^2$) and the linear metres of the opening joint ($m^3/h.m$). Class A3 represents the most favourable case and therefore indicates that only a very small amount of air manages to seep through the joints.

Such an appraisal can not obviously be extended to a window including the natural ventilation unit which, beside the assessment referring to the unit itself, is an integral part of a system with which it interacts, deserving this way a global appraisal.

A great care should also be taken over the tests for the classification of wind resistance as far as this system is concerned. Such a test is used to assess the ability of the window to withstand a certain wind pressure which should engender neither functional degradations of the window frame nor failures of the mechanical parts representing a possible danger for the users. Some experiments executed at ICITE's laboratories on bottom hinged windows with the ventilation unit inserted breadthwise in their top part, produced resistance values much below the minimum class provided for by the standard. This shows a weakening of the whole system that therefore needs the application of additional safety measures in case of wind.

5. International methodologies for the calculation of ventilation requirements in the indoor environment

The main objective of ventilation is to ensure the necessary air quality in an indoor environment. On the basis of this fundamental assumption, the first factor to be known is the air volume to be recycled, according to the pollution rate.

Polluting agents adding to endogenous noxious substances depend on how many people are in the room and on the type of activity that is carried out in it. In this connection, there are no official reference standards defining in full detail the overall pollution and relevant admitted levels, being the nature of the problem made more complex by the poor knowledge about the exact number of agents contained in the air and about the ensuing effects and the actions that can be synergically derived from them. The knowledge so far acquired is collected in the OSHA (Occupational Health and Safety Administration) Recommendations (see table...).

The presence of carbon dioxide, released by man in variable amounts, according to the activity he carries out in the indoor environment and to his metabolism, is a good indicator of air quality. ASHRAE standard 62-1989 "Ventilation for Acceptable Indoor Air Quality" assumes that the ventilation demand is acceptable whenever the concentration of carbon dioxide is not greater than 1000 ppm, as specified by the OSHA recommendations.

The revision of ASHRAE standard 62-89, which besides has not yet been adopted, has kept the assessment criterion based on the concentration of carbon dioxide; however, it has specified that the criterion is not based on sanitary principles but that it is just a parameter for the control of human smells, since it is mainly based on the individual input represented by crowding.

The American revised draft standard confirms the two previous methodological approaches; the compulsory approach and the performance-oriented approach, the former being used to calculate the minimum external air flow rate as the addition of two terms proportional to the index of indoor crowding and to the indoor area respectively, proposing proportional values that in the former case are of 3L/s per person and, in the latter case, are of $0,35 L/sm^2$.

The performance-oriented method needs a basic knowledge, namely:
- the nature of pollutants present indoors;
- the possible production of pollutants over a specific period of time;
- the admitted concentration in terms of health and comfort.
For the time being, it is extremely difficult to adopt this method since not all of the countless calculation inputs are known.

Another method used to dimension indoor ventilation was elaborated during the 80's in Northern Europe by a team of experts led by Prof. Fanger.
The theory in question was criticized by many but appears in the IAQ chapter of CEN/TC pre-standard 156 "Ventilation for Buildings: design criteria for the indoor environment" and suggests to dimension the air change requirements on the basis of the air quality that can be sensed by man himself. To this end, two units of measurement have been introduced: the *olf* after the Latin word"olfactus", that senses the polluting molecules dispersed in the air through the olfactory sensorial activity, taking place at the end of the nasal cavity; the *decipol*, after the Latin word "pollutio", turns the actions that irritate to a lesser or greater extent the mucous membranes of the upper tract of the respiratory system and of the eye, into different levels of pollution present in the area. In other words, decipol represents the air quality that can be sensed as a result of the interaction between pollutants and ventilation. This means that with a polluting source of one olf, detected in an indoor environment in which 10 l/s clean air are let in, the qualitative value expressed in decipols will amount to one and will prove that air and pollution have perfectly blended.
The equations proposed by Fanger are based on sensorial evaluations and, even if many people consider them rather superficial, still represent an effective and easy assessment instrument.

6. Future developments
The situation so far described clearly shows that future developments will mainly follow two ways: the technical/performance-oriented way and the legislative way.
The proposition of optimum ventilation values for indoor environments to be included in the Italian legislative and normative body becomes more and more meaningful also with reference to the present stage of the study which is trying to define, through a number of corrective actions to be applied to the studied prototype, performance-oriented criteria to be effectively applied to any atmospheric situation. To sum up, instead of completely closing the window when atmospheric conditions are bad, the application of a highly sensible mechanical regulator will ensure an air flow as an inverse funtion of the external pressure. Present difficulties concern the possibility of managing the air flow with pressures greater than 50 Pa, although even the most advanced standards ruling this subject do not envisage such a possibility.

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Title: Energy Recovery Possibilities in Natural Ventilation of Office Buildings

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Natural ventilation
in Sweden → voor verwarmen !
= heat - recovery

ENERGY RECOVERY POSSIBILITIES IN NATURAL VENTILATION OF OFFICE BUILDINGS

Vooral voor winter

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1. SUMMARY

The paper deals with energy consumption and heat recovery in office buildings with natural ventilation. Net energy consumption for ventilation is calculated for 7 European countries. The calculations are done with various air flow rates and occupancy. The calculations shows differences between the seven countries, but the net ventilation heat loss is substantial for all. Norway and Sweden will benefit most from heat recovery.
Several heat recovery concepts for natural ventilation are presented. Advantages and disadvantages with the various systems are discussed, also with respect to requirements as thermal comfort, air flow control, air cleaning and operation/maintenance. The paper also analyses the distribution of natural driving forces for ventilation both in various countries and in different parts of the heating season. The calculations indicate a need for assisting fans.

2. INTRODUCTION

A consortium of 7 countries conducts a project aiming at overcoming barriers to low-energy natural ventilation in office-type buildings in moderate and cold climates. The project, which is called NATVENT, is partly funded by the Commission of the European Union under the JOULE Program of the Fourth Framework.

Natural ventilation is the process of supplying outdoor air into a building and extracting the same amount of used or contaminated air from the premises utilising wind and thermal buoyancy as the driving forces. Natural ventilation systems can attain much more interest in the future in moderate and cold climate countries if heat recovery is included. Such systems use very little energy for air transport, they generate very little noise and a good air quality can be obtained. As a basis for further work on developing solutions for natural ventilation with heat recovery the paper focus on practical concepts:

- Controlling the air flow in accordance with what is demanded
- Preventing pollution in urban and industrial areas from entering a building
- Supplying the outdoor air without creating thermal discomfort
- Ensuring low energy consumption
- Ensuring user-friendliness in operation, service and maintenance
- Energy- and cost effectiveness as a total (installation and running costs)
- Preventing outdoor noise (traffic) from entering the building through ventilation openings

Sustainable development in the building sector requires energy efficient building design and operation. This paper shows that heat recovery cannot be neglected in natural ventilated office buildings. As a basis for further work on developing solutions for natural ventilation with heat recovery the paper focus on practical concepts. These concepts also pay attention to thermal comfort, air flow control, air cleaning and operation/maintenance.

3. ENERGY CONSUMPTION FOR VENTILATION

Outdoor air for ventilation must inevitably be heated to room temperature. The energy consumption for heating the ventilation air depends on air flow rates and the outdoor climate. It does not directly matter if the building is thermally well insulated or not, but in well insulated buildings more of the surplus heat can be used to cover ventilation heat losses than in buildings that are not thermally well insulated. Consequently, in order to calculate the net energy consumption for heating outdoor air for ventilation one must analyse the total energy balance of the building, taking into account all the internal heat loads. Table 2 shows calculated net energy consumption for ventilation in different countries. The calculation is done with tsbi-3, which is a Danish energy calculation program. Important input data in the program is listed in table 1. The building model used is taken as a multi-storey building with cellular offices. The office rooms are placed on both sides of a corridor. All the offices have windows. Solar shading devices and office equipment are the same for each country. Outdoor climate data files for a reference year are supplied by the partners.

Table 1. Building data and internal heat loads used in the energy calculations

Country	U-values Watt/m² K		Lighting	Office equipment	Infiltration
	Wall	Window	Watt/office	Watt/office	Ach
N, S	0,22	1,6	200	125	0,21
DK, GB, CH, B/NL	0,33	2,8	200	125	0,21

A lot of international research work is done in the indoor air quality field with respect to the minimum requirements for the supply of outdoor air in office buildings. Some of the reports conclude that for air quality reasons it is not recommended to go below an outdoor air supply of 10 l/s per person during occupancy hours [1]. For thermal reasons the air flow-rate requirements may become much higher than this. We have chosen to present the energy consumption for 10 l/s controlled ventilation and correction data per l/s deviation up and down from 10 l/s. The results are presented in table 2. The table also shows ventilation codes/guidelines in participating countries.

Table 2. Net energy consumption for ventilation

Country	Location	Ventilation codes/ guidelines	Energy consumption	Energy consumption	Energy consumption
				Correction if ventilation differs from 10 l/s per workplace (person)	
		l/s per m²	kWh per Year per 10 l/s	kWh per Year	kWh/m² per Year
			For one office or workplace	Per l/s	Acc. to10 l/s per person 15 m² per person
N	Oslo	1.4	363.1	39.7	24.2
DK	Copenhagen	0.4	316.8	32.6	21.1
S	Stockholm	0.7	380.8	41.3	25.4
B	-	0.8	238.8	25.9	15.9
NL	-	1.4	238.8	25.9	15.9
GB	London	0.8	193.9	22.6	12.9
CH	Zurich	0.4	279.4	30.1	18.6

The data reflects 12h controlled ventilation per day, 5 days a week.

Example - correction to 8 l/s ventilation per workplace for Denmark: 316.8- 2·32.6=251.6 kWh per year or 16.8 kWh/m² per year if the floor area is 15 m² per workplace.

The energy consumption data shown is only for controlled ventilation, 12 h per day, 5 days a week. It is the net energy consumption for ventilation which is shown, based on a complete simulation of two offices located on each side of a corridor. It is assumed that all the offices are equal. The simulations are done for single person cellular offices with a floor area of 12 m² + 3 m² corridor floor area. Changing the office size had only a minor influence on the ventilation energy consumption.

As we can see there are some differences between countries. The ratio between the highest and the lowest number is 1.87. As we can see from the table Norway and Sweden will benefit most from heat recovery, but the net ventilation energy consumption is substantial for all countries. In terms of percentage of the total energy consumption of the whole building, lighting and equipment included, the energy consumption for 10 l/s per person in this example is approximately. 30 % for N and 20 % for GB. The internal heat load used in the calculations are assumed to be average to high. A lower internal heat load will increase the net ventilation heat loss and thus make heat recovery more profitable and vice versa. Also, the ventilation requirements may be higher than 10 l/s, as is the case for Norway, or lower as may be the case for other countries. The energy consumption changes almost linear with ventilation flow rates, according to table 2 column 5. Running time for ventilation other than 12 h per working day matters also.

An ordinary mechanical ventilation system requires approx. a total fan power of 2 - 3 W per l/s for air transport. This represents an energy consumption of 10 - 20 kWh per year per l/s. Most of this energy will be saved in a natural ventilation system (energy consumption for assisting fans: less than 10 % of above).

4. AVAILABILITY OF NATURAL DRIVING FORCES

The driving force is the sum of buoyancy and wind effect. Based on the outdoor climate data files supplied by the participants, the distribution of available driving forces is calculated for the countries. In the calculations it is assumed that 80% of the dynamic pressure of the wind can be utilised in a natural ventilating system. The thermal stack height is taken as 10 m. Fig 1 shows the distribution for the autumn heating period and fig. 2 for the spring heating period. As we can see the frequency of very low driving force is rather high. For example the frequency of a driving force of 10 Pa and lower ranges between 1050 and 2460 hours, lowest for Copenhagen and highest for Oslo. Increasing the stack height to 20 m shifts the distribution some 5 Pa to the right. The ranging between countries becomes however different. The frequency of driving force, now equal to and less than 15 Pa, ranges between 630 and 1560 hours, the lowest value for Sweden and the highest for Belgium. For a 10 m stack height the frequency of driving force less than 15 Pa, fig. 1, would be very high.
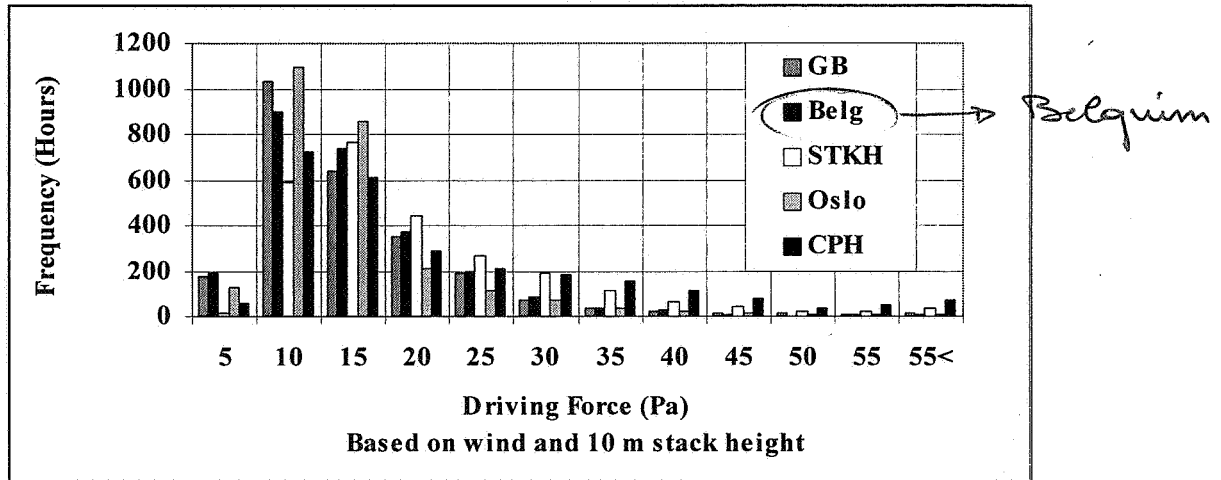
Fig. 1 The distribution of the natural driving forces in the heating season, autumn. The calculations are based on hourly weather data.
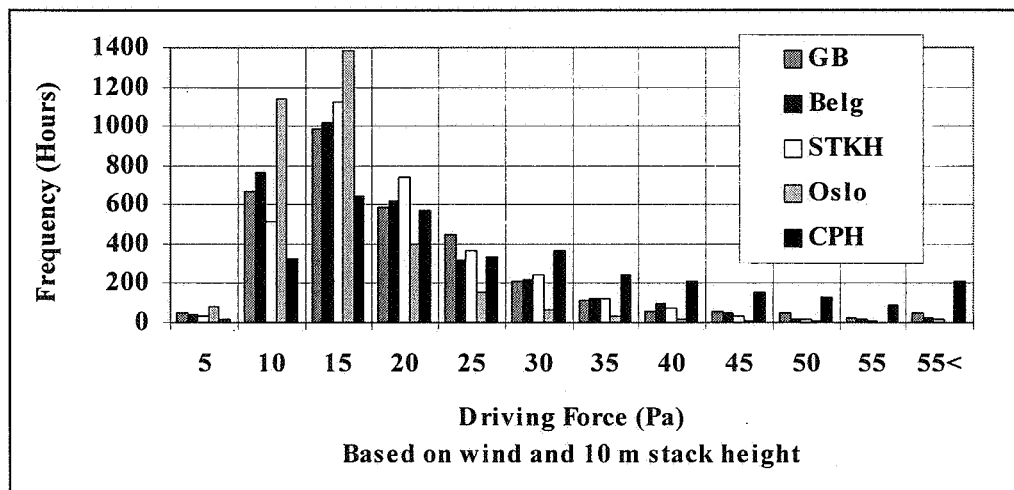
Fig. 2 The distribution of the natural driving forces in the heating season, spring. The calculations are based on hourly weather data.

Detailed calculations for a heat recovery system are not done, but a driving force of 15 Pa is certainly on the lower side.

5. HEAT RECOVERY CONCEPTS

5.1 Principles

There are a number of possibilities and concepts for heat recovery from exhaust air in natural ventilation. The concept to be chosen depends on the possibilities for utilising the recovered energy. In ordinary mechanical ventilation it is very common to use an air to air heat exchanger for direct transfer of heat between exhaust and supply air. It is important to preheat the outdoor air before entering the occupied zones in a building in cold climate countries. Heat recovery systems should consider this. Air to air heat exchangers may be used for this purpose. This solution requires, however, that supply- and exhaust air have to be joined, preferably in an air handling unit. This imposes some severe restrictions on the layout of the

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systems, especially the ducting system, which may lead to problems in natural ventilation. Introducing a liquid energy carrier instead, the energy can be distributed (run around) in small pipes and the systems become more flexible. Using liquid run around systems, the recovered energy can be used in the most suitable and practical way. The energy can either be used directly for preheating the supply air at appropriate locations in liquid to air heat exchangers or it can be used as the low temperature heat source in heat pump systems where the alternatives for using the energy is hot water heating, room heating or preheating of supply air. Liquid run around systems are widely used in mechanical ventilation systems.

5.2 A simple concept

In principle one way of achieving heat recovery in natural ventilation is to use an air to air recuperative heat exchanger as shown in fig. 3. This is the most simple concept. Using this concept the heat exchanger should be based on counter-flow. One should also bear in mind that the thermal driving force (buoyancy) is dependent both on the stack height and on the heat exchanger efficiency. Heat recovery will decrease the thermal stack effect while increased stack height increases the driving force. This solution is therefore not the most efficient one. The concept is studied by DTH in Denmark [2], but a practical solution is up to now not developed.
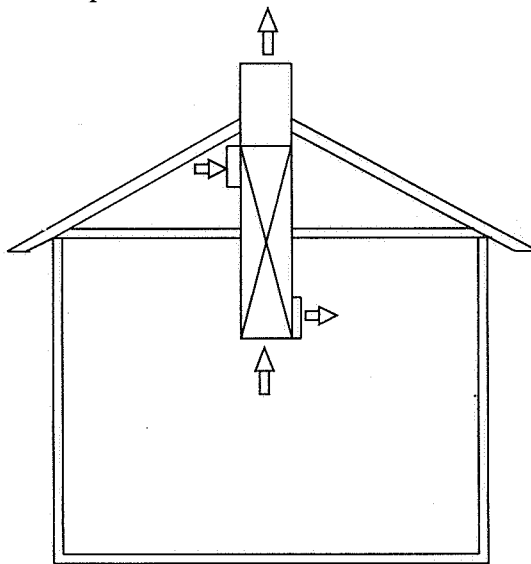
Fig. 3 A simple concept for heat recovery in natural ventilation

5.3 A practical air to air heat recovery concept

A practical and simple unit for a multi storey dwelling has been developed by NBI in co-operation with a Norwegian manufacturer [3], fig. 4. In this case the driving force will obviously be different for the different storeys. Because the heat recovery principle shown will decrease the thermal driving force, it is important to utilise the wind forces. Consequently, both the supply part and the exhaust part of the system should be optimised for that purpose. However, it turned out to be necessary to incorporate an assisting exhaust fan for use in periods with low natural driving forces.
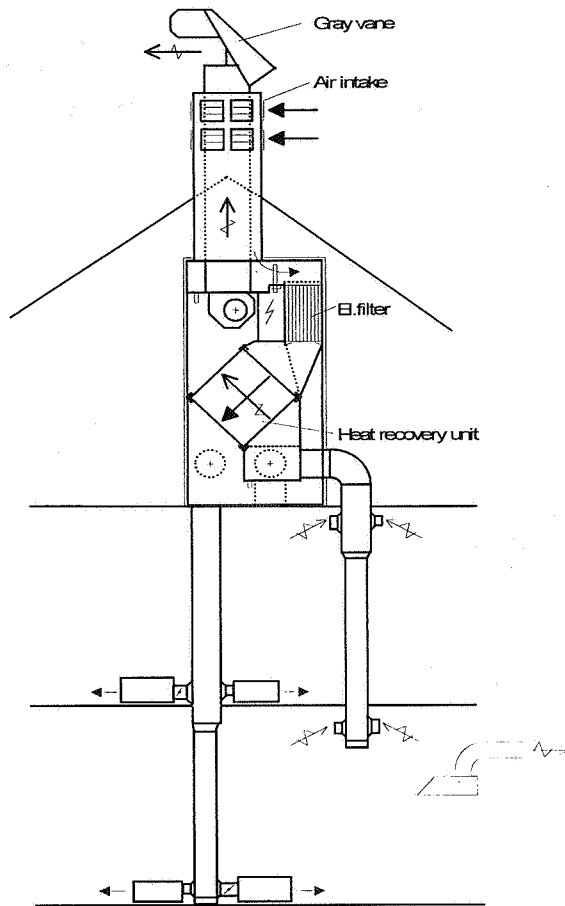
Fig.4. A practical simple unit for a dwelling for natural ventilation with heat recovery using an air to air heat exchanger.

5.4 Separated systems

A more rational approach in developing heat recovery concepts in natural ventilation is to separate exhaust air systems and supply air systems. However, in cold climate countries it is not acceptable, in the cold season, to supply outdoor air which is not preheated. Consequently, we are at the same time looking for air supply systems where the air can be preheated. The functional requirements for a good air supply system is the following:

- The air shall be distributed to where it is needed.
- The distribution should be independent on the wind conditions.
- It should be possible to clean the supplied outdoor air.

5.4.1 Separated heat recovery systems with outdoor air supply through the outer walls

A concept for separated systems is shown in fig. 6. The exhaust air heat is recovered in an air to liquid heat exchanger in the exhaust stack. A practical solution for multifamily houses is developed in Sweden [4]. Both wind- and thermal driving forces are utilised. The recovered heat can be utilised directly by pumping the liquid energy carrier through heat exchangers in the supply openings in the facades. Due to the small temperature difference the efficiency seldom becomes more than 50 %, which means that the air will be too little preheated. The

efficiency problem may be overcome combining heat recovery with a heat pump. The heat pump will raise the temperature leading to higher temperature differences and smaller heat exchangers. There are products in the market covering this function, also equipped with particulate air filters.
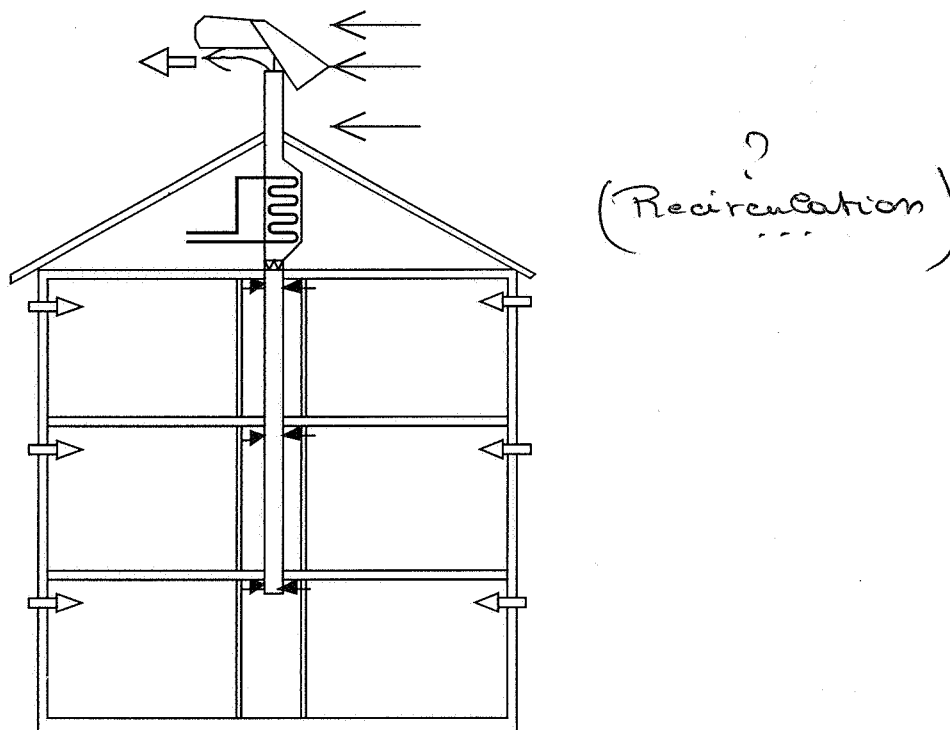
Fig. 6. Separated heat recovery system with air supply through the outer walls

In principle the described concept can work, but there are several objections to the solution. The air distribution is sensitive to the wind conditions and the driving force varies between each floor, highest at the bottom floor and lowest at the top floor. The air may go either way through the openings depending on the wind direction. It will be costly both with respect to installation and maintenance cost, and it will also introduce a substantial pressure drop to equip all inlets with heat exchangers to preheat the air. Also, in polluted areas like urban areas and other areas with busy traffic roads, it is a need for at least particulate air filtering. Good supply air filters would be costly and require an additional pressure drop. Sound attenuation may also be required to reduce traffic noise.

Assisting fans, to prevent a too low inside pressure would be difficult and costly to install in a proper way.

5.4.2 Separated heat recovery systems with internal air supply

The air distribution problem can be solved by using an internal air supply equivalent to the simple system in fig. 4. as shown in fig. 7. Because there is no coupling between the air supply system and the exhaust air system, there are several degrees of freedom in solving the air supply problem.

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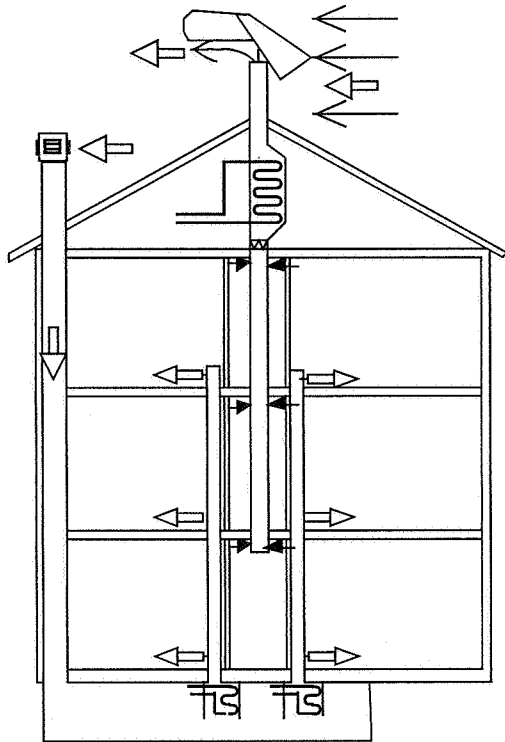
Fig. 7. Separated heat recovery system with internal air supply. Outdoor air intake at roof level.

The supply air system shown in fig 7 is taking the outdoor air in from above, but the most interesting feature in this concept is that the air is lead to underneath the building. Doing so the air can be preheated and fed to the different rooms through risers. In this way the driving forces will become equal for all and each of the storeys. The higher the building the higher the driving force. The outdoor air may be supplied from ground level as well as shown in fig.8. Note that wind forces are utilised for both air exhaust and air supply. More than one exhaust stack may be needed as well as more than one riser system.
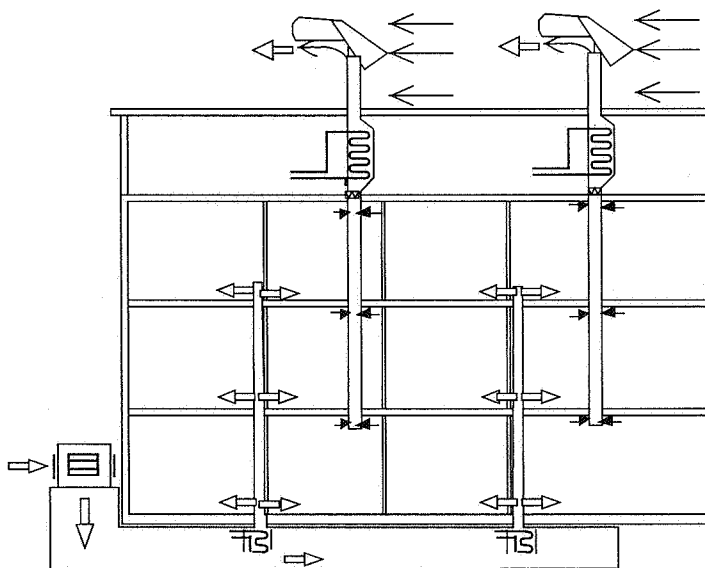
Fig. 8 Separated heat recovery system with internal air supply. Air is taken at ground level.

6. CHARACTERISTICS OF SYSTEMS AND COMPONENTS

6.1 The internal supply air system - characteristics

An internal air supply means to close the facades and supply the air from inside the building. This requires a ducted supply system. In principle there are two main solutions to this:

- Outdoor air is taken in through a vertical stack from the roof top of the building down to a tunnel under the building. From the tunnel the supply air is lead through risers and distributed to where it is needed. In open plan lay-outs, simple distribution systems, like supplying air only to corridors and atriums, may be used. The heat exchanger for the preheating should be located at the bottom of the risers. In this way the full thermal stack effect is utilised as driving force, fig 7. The driving force will further be equal for each floor.

- Outdoor air is taken from an air intake at ground level to a tunnel underneath the building. The solution for the risers and the preheating is the same.

The solutions offer a good possibility for central cleaning (filtering of the supplied outdoor air. Electrostatic filters offers especially high efficient particulate filtering with very low flow resistance (pressure drop). Also traffic noise can easily be reduced by this solution. Depending on the strategy the air intake may be designed to utilise or neutralise the wind forces. Utilising the wind forces is recommended because this will maximise the driving forces. Generally, utilising the wind forces results in periodically rapid changes in driving force which the control system must counteract in order to prevent rapid changing air flow rates. Changing air flow rates may cause both thermal discomfort and excessive energy use if not counteracted. Fortunately, constant air flow devices both exists and new and better ones may be developed and be used.

The concepts shown only show the principle. Practical solutions must involve a close integration with the building design, to reduce the need for ordinary ducting. The building design may be more open, simplifying the exhaust air stack. There are numerous architectural possibilities here. Other interesting features are the horizontal tunnels which opens possibilities for pre-heating and pre-cooling of the supplied outdoor air, by exchanging heat with the surrounding ground. This may specifically increase the efficiency of night-cooling.

6.2 The heat recovery system - characteristics

The exhaust air heat recovery system is the easiest part of the problem solving and the solutions are rather straight forward. In the market there are different types of both air to liquid and air to air heat exchangers to choose between. However, a basic consideration to take is that the pressure drop should be low. This means that the face velocity will be lower than in ordinary systems. This in turn means larger heat exchanging surfaces, and somewhat more expensive components. A benefit from this is possibilities for higher heat recovery efficiencies. An important thing is to keep the heat exchanger surfaces clean by placing a filter in front of the heat exchanger. Because of the larger face area and lower face velocity compared to common practice, cheap plane filters may be used for the exhaust air, which means that the combination heat exchanger/filter may not be more expensive than in ordinary systems. The ducting part of the system may also be rather simple since air can be extracted from a few locations.

Even if there already exist components suitable for heat recovery in natural ventilation, a challenge for the future is nevertheless to develop efficient and cost effective components, especially designed for natural ventilation.

6.3 Assisting fans and flow controllers

The variations in driving force will be large. This is illustrated in fig 1 and 2 in section 4. The figures shows the distribution of driving force for typical stations in the NATVENT participating countries. The necessary minimum driving force for a workable natural ventilating system with heat recovery is not accurately calculated, but a preliminary estimation is that it certainly will not be less than 15 Pa, indicating a need for assisting fans. We also see from the figures that the driving force may become higher than 60 Pa, indicating a need for efficient air flow controllers. Low pressure assisting fans are very efficient throttling devices and may be used for controlling and stabilising the air flow, and make other flow rate controllers superfluous. This means that assisting fans may replace separate flow controllers. The speed of the fans can be controlled based on a flow sensor, and a frequency converter for the power supply.

7. CONCLUSIONS

Heat recovery cannot be neglected in natural ventilated office buildings. Practical concepts exists and may be further developed. These must involve a close integration with the building design, to reduce the need for ordinary ducting. Calculations of available natural driving forces indicates that workable natural ventilating systems with heat recovery require assisting fans. Assisting fans may act as efficient flow controllers.

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VENTILATION AND COOLING

SOLAR ASSISTED NATURAL VENTILATION WITH HEAT PIPE HEAT RECOVERY
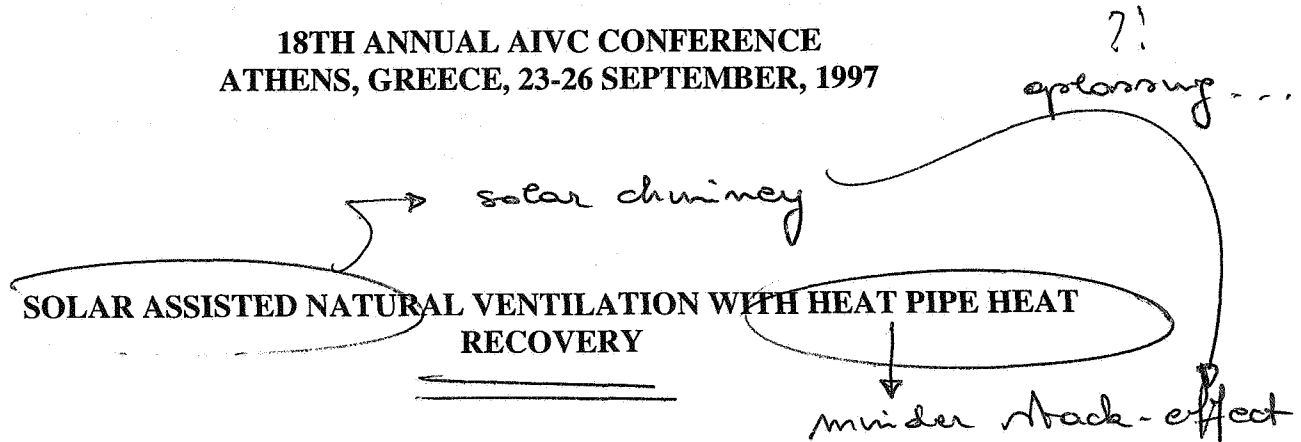
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SOLAR ASSISTED NATURAL VENTILATION WITH HEAT PIPE HEAT RECOVERY

SYNOPSIS

Natural passive stack ventilation (PSV) consumes no power and so produces no harmful emissions, has no running cost, no noise of operation, requires little maintenance and because it involves no moving parts, operation is reliable. However, virtually all PSV systems are designed and constructed without incorporating heat recovery, leading to wasteful heat loss. The goal of the research reported here, is to develop a passive stack ventilation system with heat recovery for use in naturally ventilated buildings.

The heat recovery unit is based on the heat-pipe principle. A recovery unit having a sufficiently high efficiency and a very low pressure drop is aimed at. The drawback of an efficient heat recovery is, that it reduces the stack pressure by reducing the temperature difference between the supply and exhaust air flows, which can cause the ventilation system to fail. To avoid this problem, a solar chimney and a wind generator driven fan are integrated into the system to assist the air flows and to maintain them on a sufficient level from the viewpoint of indoor air quality. On the other hand, for the air flows not to be too high, a control unit is added to the system. A pilot plant shall be constructed, where all above mentioned features are included and this plant shall be monitored for six months to find out its performance.

Until now several versions of the heat recovery unit have been tested. The highest effectiveness has been around 55% with two banks of heat pipes and a flow velocity of 1 m/s. With the same velocity the pressure loss through a two bank section is 4.5 Pa. The solar chimney has been studied both theoretically and experimentally. According to computations, the performance of the chimney is not very sensitive to small changes in direction on both sides of the south. Measurements indicate a 47 % increase in air flow compared to a conventional stack. The air flow control unit has been developed and tested. It works well according to a control strategy specified for the PSV system. The work continues by designing and optimisation of the pilot plant.

INTRODUCTION

Natural ventilation is being applied to an increasing number of new buildings across Europe to minimise reliance on mechanical ventilation and so reduce emission of greenhouse gases. The stack pressure created by the temperature difference between the indoor and outdoor air provides a driving force for natural ventilation and stack-driven ventilation has been applied to a wide range of modern buildings, including offices, schools and houses [1, 2]. Natural passive stack ventilation (PSV) consumes no power and so produces no harmful emissions, has no running cost, no noise of operation, requires little maintenance and because it involves no moving parts, operation is reliable. However, virtually all PSV systems are designed and constructed without incorporating heat recovery, leading to wasteful heat loss. It has been estimated that this heat loss amounts, depending of the location in Europe, to 3 - 15 GJ per

annum for a small family residence and much more for larger buildings, e.g. offices, which employ natural ventilation [2].

Table 1. An estimate of the annual ventilation energy consumption of a small European family residence equipped with different types of ventilation systems (η_{heat}=0.80, $\eta_{electri}$ =0.30)

Ventilation system	heat recovery efficiency [%]	consumption of heat [GJ]	electricity for fan operation [GJ]	total energy consumption [GJ]	primary energy consumption [GJ]
Mechanical exhaust, no heat recovery	0	9.79	0.78	10.6	14.8
Mechanical supply and exhaust with heat rec.	70	2.94	1.55	4.49	8.86
Passive stack ventilation, no heat recovery	0	9.79	0	9.79	12.2
Passive stack ventilation with heat recovery	50	4.90	0	4.90	6.12

According to the simple calculation presented in Table 1, the passive stack ventilation system with heat recovery (PSVH) would be a very attractive solution, especially from the primary energy point of view. There is obviously a need for appropriate and efficient heat recovery in natural ventilation systems to minimise waste of energy.

The SAVEHEAT research project, funded partly by the Commission of the European Union, aiming at a novel PSVH system, has started in January 1996. Following are described some features and ideas of the developing work as well as some results produced until now.

HEAT RECOVERY

The heat recovery unit is based on heat pipe technology. A nominal air velocity of 1.0 m/s has been chosen for the dimensioning. A very small pressure drop in the recovery unit is essential. At Nottingham University four types of extended surfaces on the air side of the recovery unit have been tested: I) plain fins, II) spine fins, III) louvred fins and IV) wire fins. The effectiveness of heat recovery of the types I and IV are shown in Fig. 1. A high thermal performance usually means a higher pressure loss and for that reason certain compromises, when choosing the best alternative, have to be done. A more detailed description and results of the recovery unit are given in reference [3].

SOLAR CHIMNEY

The heat recovery reduces the pressure difference and the air flows of the PSVH system in two ways: first, because of its own pressure loss and second, because of reducing the temperature difference between the supply and exhaust air flows. The higher is the efficiency of the heat recovery, the smaller is the stack effect. This could lead to insufficient small air

flows, moisture problems and a poor indoor air quality. The air flows can, however, be assisted using simple active solutions, such as directly heating the exhaust stack with solar energy or utilising wind energy as an additional driving force.
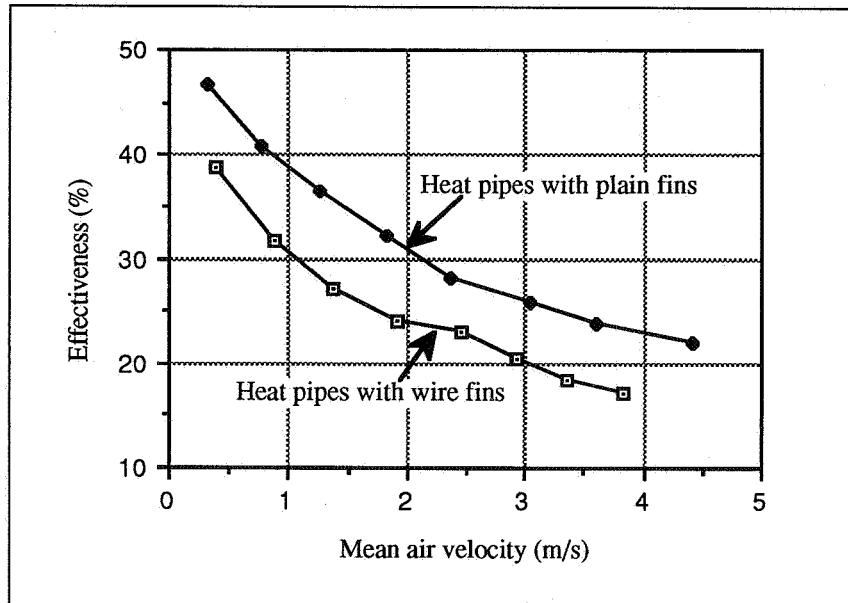
Fig. 1. Effectiveness of the recovery unit with plain fins and with wire fins.

Measurements of a solar chimney, to assist the natural ventilation, were carried out in a two zone test cell at University of Porto. The test rooms are identical, from a geometrical point of view, and each one was equipped with a heating facility, with a precise control of the inside air temperature. Both were provided an inlet and an exhaust air duct, through the roof. The exhaust chimneys have a similar geometry, but while one allows the collection of solar radiation (solar chimney), the other does not (conventional chimney). Both exhaust chimneys have an internal cross section of 0.2m x 1m and a height of 2m. The walls are made of brick (10 cm thick), with outside insulation (5 cm) for the solar chimney. In both rooms the air temperature is measured and controlled. The exhaust chimneys were fully instrumented with anemometers, thermocouples and fluxmeters.

Table 2.   Air exchange rate of the rooms in the test cell.

		ACH (h^{-1})	
	Sampling point		Average
Room 1	1	14.2	
(Convent chimney)	2	13.8	14.0
Room 2	3	20.6	
(Solar chimney)	4	20.5	20.6

Fig. 2 shows the variation of the flow rate in the solar chimney during a day-night period. A comparison between the air exchange rate created by the solar chimney and the conventional chimney is shown in Table 2. As can be seen, the contribution of solar energy to the air exchange rate of the room with solar chimney is about 47%, for the measuring period.
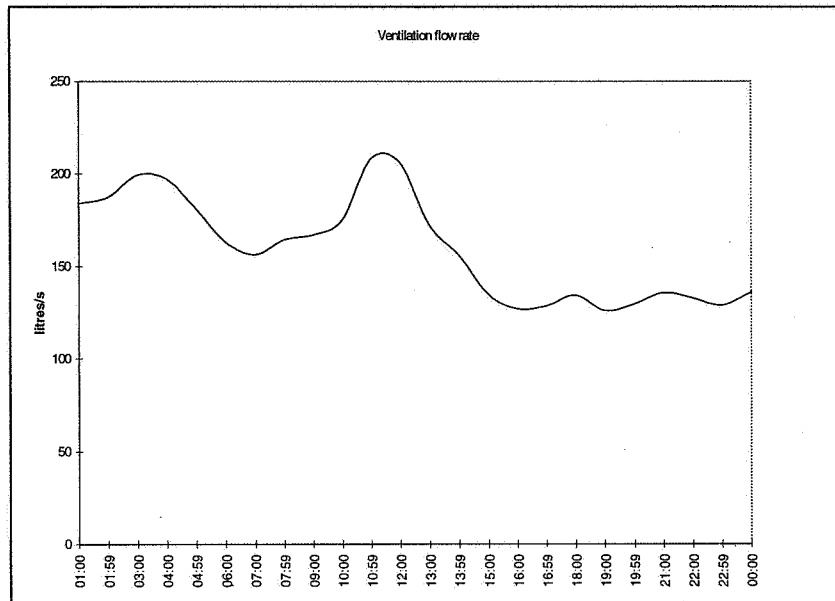
Fig. 2. Ventilation air flow rate with solar chimney, 15 Nov 96

WIND DRIVEN FAN

To further assist the ventilation flows in situations, when neither a temperature difference nor sun shine is available, a wind driven fan was constructed. A vertical axis wind generator was chosen. The purpose was to install it directly on the shaft of the fan. Preliminary modelling and computations were carried out to see what type of fan would match the generator and give the best performance. It was found out that a radial fan having forward curved blades would give roughly double air flow than an axial fan. This is a consequence of the better matching of the characteristics of the radial fan and the wind generator. Thus, a radial fan was chosen and combined with the generator to form a wind driven supply air unit.

CONTROLS

Because the PSVH system should work properly during different kind of conditions and in different climates, controls is needed. The main tasks of a control system would be to reduce the air flows in cases there is a too high driving force and to inactivate the heat pipe unit when there is no need for heat recovery. To accomplish this, a control strategy for the PSVH system was developed, Fig. 3.

Temperature:
Indoor - Outdoor

$T_{out} < T_{in}$

$T_{out} > T_{in}$

Heating
$T_{in} < T_{set}$

Cooling
$T_{in} > T_{set}$

Heating
$T_{in} < T_{set}$

Cooling
$T_{in} > T_{set}$

night / day

Alternatives

1.	2.	3.	4.	5.	6.
D_{exh} controlling D_{sup} open D_{fresh} closed	D_{exh} open D_{sup} closed D_{fresh} open	D_{exh} controlling D_{sup} closed D_{fresh} open	D_{exh} controlling D_{sup} closed D_{fresh} open	D_{exh} controlling D_{sup} open D_{fresh} closed	All dampers closed

Remarks:	Remarks:	Remarks:	Remarks:	Remarks:	Remarks:
-heat recovery	-no heat recovery -air-exchange rate may increase too much! -night cooling	-no heat recovery -building occupied	-no heat recovery -driving force?	-inversed heat recovery -driving force?	-no heat recovery -no air-exchange!

Fig. 3. Control strategy for the PSVH unit.

The control strategy is implemented using a modern programmable logic combined with conventional dampers, temperature sensors and a low velocity sensor measuring the air flow in the exhaust duct.

MODELLING

To prepare the selection of a test building for the installation of a pilot plant, the influence of some most essential parameters was investigated by a preliminary simulation. The computations were made with a simple aeraulic model describing the building and the passive stack system. The most important result of this exercise was, that the tightness of the building is crucial factor for the energy performance of the PSVH system. If the building is leaky, only a part of the air is guided through the heat recovery unit, thus greatly reducing the recovery potential.

Because the functioning of the PSVH system very much depends on the local weather parameters, different design for different climates has to be applied. Also the technical optimisation, where the minimum amount of primary energy is used maintaining the comfort conditions and the air quality at the same time, gives different results for different locations. To find out an optimal design for a specified location, a model describing the behaviour of the system is needed. A model consisting of the building structure, the ductwork, the recovery unit, the wind unit and the controls is at present under development. The developing environment, called IDA, is utilising the Neutral Model Format (NMF), has an efficient solver

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and is one of the most advanced simulation tools in the building/HVAC sector at present[4]. This model combination shall first be validated against measurement results from a pilot PSVH plant and after that used to look at the optimal design and performance in different European climates.

PILOT PLANT

A pilot plant of the PSVH system shall be built in an office building in Winterthur, Switzerland during the year 1997. The building, called SLM Building, is a five storey office with additional two basements and a technical level on the roof. It was completed in 1991 and is situated in a heavily urbanised part of Winterthur, with the main road to Zürich just in front of it. The area of the ground floor is 396 m2. The remaining four upper floors have the measures 37,8*19,8 m, giving 741 m2 to each of them. The building is planned for commerce and has approximately 180 workplaces.

An open office space in the building is prepared to serve as a test environment for the PSVH pilot plant. The space is isolated from the rest of the building to prevent the effect of the mechanical ventilation. The solar chimney will be built on the roof and the ductwork is guided in through one of the large windows. The plant will be equipped with a comprehensive monitoring equipment and monitored for a six months period to find out the performance of the pilot system.

CONCLUSIONS

A passive stack ventilation system including heat recovery, flow assistance and controls (PSVH) is under development and seems to have potential to be an alternative for the conventional ventilation systems.

The heat pipe based recovery unit has shown an efficiency of 55% and a pressure loss of 4.5 Pa at the same time.

The tests of a solar chimney indicate 47% increase in air flow compared to a conventional stack.

Preliminary modelling of the PSVH system show, that a good tightness of the building is essential for the energy performance of the system.

The work continues by the construction and monitoring of a pilot scale PSVH system.

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VENTILATION AND COOLING

18TH ANNUAL AIVC CONFERENCE
ATHENS, GREECE, 23-26 SEPTEMBER, 1997

IEA ANNEX 27: Evaluation and Demonstration of Domestic Ventilation Systems. A simplified tool for the assessment of LCC

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SYNOPSIS

Costs are one of the main decision factors for the selection of domestic ventilation systems. This often leads to a ventilation system that just meets the requirements of building regulations at the lowest initial costs. Decision makers are often not aware of the impact of the quality of the ventilation system on life cycle costs, not only for the ventilation system itself but also for the building, as a result of complaints or even damage due to a poor functioning ventilation system. In Annex 27 a simplified tool is developed to compare costs of ventilation systems, not only regarding the initial costs of the system, but also costs of maintenance and costs of complaints and failures related to ventilation and ventilation systems. Little is known about these costs, specially costs of maintenance and complaints. To collect some data a survey was carried out amongst 19 housing corporations, owning about 100,000 homes, and representatives of the Dutch ventilation industry (VLA).

The first step was to put up a framework for the tool. With data and information of the Dutch ventilation industry and four housing corporations, owning buildings and homes with different types of ventilation systems (ME and MVHR), types of maintenance activities as well as maintenance cycles were defined. The second step was to estimate a relation between the user frequency, basic quality and the expected maintenance level, depending on the type of ventilation system. As a prerequisite it was assumed that a certain maintenance is necessary to maintain the performances (IAQ) of the ventilation system. The third step was to collect data on costs for maintenance and complaints amongst the other 15 housing corporations. It appeared from this part of the research that very few corporations were aware of the necessity of planned maintenance and the costs for maintenance as well as the costs for complaints related to the quality of the ventilation system. Despite this it was possible to "fill" the tool with data on costs by comparing these data with more common data on *total* maintenance. The simplified tool can be used for estimating the expected maintenance and maintenance activity cycles as well as the total life cycle costs for the system, the maintenance of the system, the expected complaints and the ventilation related maintenance of the building.

The last step that is to make is linking this tool with the reliability tool of Annex 27. Svein Ruuds (1) and Johnny Kronvalls (2) work on reliability and system safety analyses is used as a basis for this.

INTRODUCTION

The objective of a tool for life cycle costs is to compare the total costs of ventilation systems and to make selections based on a cost comparison. The costs have to include:

- initial costs (investments)
- costs for maintenance of the system
- costs of maintenance of the building as a result of the ventilation system and its use.
- costs for energy.

One of the ideas was that investing more in the quality of the installation and selecting the right kind of ventilation system in relation to building properties and user characteristics could finally lead to lower maintenance costs, specially for complaint maintenance. This paper mainly focuses on the costs for maintenance of the ventilation system and ventilation related complaints in the building. Costs for energy is an output of the energy tool of Annex 27. The tool is mainly developed to be used by housing corporations and associations.

The development of the first idea for this tool was coached by chiefs of the technical department of four housing corporations in the Netherlands and representatives of the Dutch

ventilation industry (VLA). The first version of this tool was tested and applied by these housing corporations.

In order to get more data and to verify the first results a larger survey on maintenance costs was carried out amongst 15 other housing corporations. These corporations have stipulated the maintenance activities and the maintenance cycles for the ventilation system and the building itself. The first remark is that in practice there appears not to be any consensus about the required maintenance activities. Most of the corporations have their own way of planning maintenance and assessing the necessity of maintenance. There is a lack of knowledge about maintenance and its impact. Only 40 to 50 % of the required maintenance activities, as described by the Dutch ventilation industry is really carried out.

More over, only 50 % of the corporations could turn over detailed data about maintenance and complaint maintenance (= maintenance that is carried out if complaints occur and are reported to the corporation), concerning ventilation and ventilation systems. For the total maintenance activities and complaint maintenance much more data are available.

However, it was possible to draw some conclusions from this survey. One of the facts is that maintenance of ventilation systems and its costs show a relation with the total costs of maintenance.

DEFINITIONS IN RELATION TO MAINTENANCE

The following aspects in relation with maintenance are considered and definitions are used:

a. *Type of maintenance* in relation with the organisation.
The type of maintenance is determined by the way the maintenance is organised and managed. There is a difference in systematic maintenance (according to plan) and non-systematic maintenance. Systematic maintenance is based on the desired quality and need for maintenance of the building. Non-systematic maintenance is steered by signals like complaints and mutations.

b. *Maintenance activity* in relation to a technical property of a building or construction part.
Maintenance activities are the activities that are necessary during the life span of a building to maintain a certain level of quality. Each activity has its own cycle: maintaining, partial repairing, replacing.

c. *User frequency*
Building and construction parts are loaded by the environment and by daily use. This is an important factor to determine the need for maintenance. The design of the ventilation system in relation to the building must provide an optimal "load bearing" by a good selection of materials , construction and capacity. The *users frequency* is the load, in relation to the extent of use. This includes operating the ventilation provisions and cleaning.

d. *Basic quality*
There is a distinction between the basic quality of mechanical ventilation or balanced ventilation in quality of the building and quality of the ventilation system.
The basic quality of the building includes air tightness of the building envelope, location and type of supply devices in facade, ventilation windows etc. For the ventilation system the type of ventilation unit, fans, supply and exhaust air grilles and lay out and design of ducts.

e. *Maintenance classification*

The level of maintenance indicates the necessary intensity of the maintenance activity, as a result of the load by use and the design and construction. The sensitivity of a building or construction part can vary. The maintenance is classified in three categories:

* low level
* medium level (this should be according to common and accepted standards)
* high level

To determine the users frequency some indicators are used. These indicators give information about the way a user (occupant) operates and maintains ventilation provisions.

Table 1: User frequency

user frequency → maintenance activities--------------------------------V			low[1]	medium[1]	high[1]
use	installation	cleaning	regularly (monthly)	cleaning 4 times a year	not
		control and use	meet requirements	meet requirements	deranged
	building	ventilation	grilles always open	alternating grilles open	never
		cleaning grilles	cleaning regularly	cleaning not regularly	never

[1] This means: low, medium or high negative impact on the ventilation system

The maintenance activities must guarantee the original quality during the life span of the building. (Note: this definition will now be changed in "must guarantee a certain level of *reliability*").The parameters are:

* Cycle
* Costs (this means the quantity of the expected activities)
* Life cycle of construction parts (the moment of replacement)
* The chance of complaints/failures between planned maintenance activities. This is non-systematic maintenance.

Table 2: Maintenance activities

user frequency → maintenance activities---------------------------------V				low	mean	high
maintenance	installation	taking care of	measurements	standard costs/ standard cycle	standard costs/ standard cycle	standard costs/ standard cycle
			inspection	standard costs/ standard cycle	standard costs/ standard cycle	standard costs/ standard cycle
			cleaning	lower costs	reference costs	higher costs
		replace parts		longer life cycle	reference life cycle	shorter life cycle
	building	cleaning grilles		lower costs	reference costs	reference costs
		repairing grilles		longer life cycle	reference life cycle	shorter life cycle
complaints (design and construction meet requirements)				no extra maintenance	chance 1 to 30	chance 1 to 20

RESULTS OF THE SURVEY

The average complaint per dwelling per year, related to ventilation, is 0.05. The average costs for a single family dwelling are 50% higher than the costs for a multi-family dwelling. However, the average costs for repairing a complaint appeared to be much higher for multi-family dwellings with a central system. In this research the average costs for the *total* complaint/failure maintenance (i.e. all occurring complaints) are NLG 170 for single family dwellings and NLG 155 for multifamily dwellings. These costs seem to be rather low compared to known data for average costs of complaints from practice. These costs vary between NLG 150 and 400 per dwelling per year. The costs depend on the organisation, policy (relation planned and incidental or complaint maintenance) and age of installation and building.

Complaint maintenance

It appeared that the chance for a ventilation related complaint is 5 %. The chance for complaints differs with the type of ventilation system, especially between individual and central systems. This varies from approximately 7.5 % in single family dwellings to 2.5 % in multi-family dwellings with central systems. The costs for complaint maintenance per dwelling per year show big differences. According to results of this survey the average costs *per dwelling* are NLG 14. The costs for complaints in single family dwellings are a little higher than in multi-family dwellings:
- individual systems: NLG 17
- central systems: NLG 11

This leads to the following assumptions for costs per complaint:
- average: NLG 280 per complaint (= NLG 14/0.050)
- individual system: NLG 225 per complaint (= NLG 17/0.075)
- central system: NLG 440 per complaint (= NLG 11/0.025)

Relation use, basic quality and complaint maintenance

The housing corporations mentioned the following complaints and problems leading to complaint maintenance:

Table 3: Complaints and problems for different types of ventilation systems

ME in single family dwellings:	Central ME systems in multi-family dwellings	MVHR
• failing or displacement of grilles • failing of fan • smells of cooking • draught • mould grow • noise	• smells of cooking • draught • mould grow • noise	• occupants don't understand the system • wrong execution of the system • noise

Cycles for planned maintenance ME

From this research it occurs that planned maintenance of ventilation systems is quite unknown and uncommon. In order to get a definition, more close to practice, three levels are discriminated.

- Low: these are the activities, with corresponding data, carried out by the 15 housing corporations in at least more than 50 % of the cases.
- Medium: these are the activities that are carried out by at least 25% of the corporations.
- High: All required activities have to be carried out in practice, (but only less than 25% of the corporations actually do so).

In table 4 the results of the questionnaires are given. This table indicates in how many cases a maintenance activity is actually carried out (in %) and the average cycle, modified for the results in practice.

Table 4: Condensed results of research on maintenance activities

	Single family (15 corporations)		Multifamily (13 corporations)	
	cycle	% of activity that is applicable	cycle	% of activity that is applicable
- Measuring flow of grilles	7	27	5	38
- Cleaning grilles	5	38	5	38
- Measuring extract flow	8	38	7	46
- Inspecting ducts	7	23	5	31
- Cleaning ducts	14	15	13	15
- Cleaning extraction fan unit	3	54	4	85
- Replacing grilles	16	38	15	46
- Cleaning cookerhood	15	8	4	85
- Replacing cookerhood	18	23	18	23
- Replacing extraction fan unit	15	80	16	85
- Commissioning, control system	6	69	7	62

In table 5 cycles for planned maintenance are given. A comparison is made between the data as recommended by the Dutch ventilation industry and the data from this survey, representing maintenance cycles in practice.

Consequences for maintenance costs
Now the maintenance cycles are known maintenance costs can be calculated. Assumptions for these calculations are:
- considered period : 30 years
- nominal interest : 7 %
- inflation : 3.5 %
- maintenance class : as defined
- maintenance costs : Net Present Value (NPV) for complaints and planned maintenance.

Table 5: Cycles for planned maintenance; mechanical extract ventilation

Activities ↓	Number	Unit	Cycles, recommended by the Dutch ventilation industry			Cycles in practice (results of the survey)		
Maintenance level→			Low	Medium	High	Low	Medium	High
Installation								
- Measuring flow of grilles	4	piece	4	4	4	--	6	6
- Cleaning grilles	4	piece	4	4	4	--	6	6
- Measuring extract flow	1	post	4	4	4	--	6	6
- Inspecting ducts	1	post	4	4	4	--	--	6
- Cleaning ducts	1	post	8	8	8	--	--	6
- Cleaning extraction fan unit	1	post	4	4	4	3	3	3
- Replacing grilles	4	piece	21	16	13	--	18	18
- Cleaning cookerhood	1	piece	8	4	3	--	--	6
- Replacing cookerhood	1	piece	21	16	11	--	18	18
- Replacing extraction fan unit	1	piece	21	16	11	18	18	18
- Assess control systems	1	post	4	4	4	6	6	6
Building								
- Cleaning grilles (facade)	6	piece	8	5	2	--	--	--
- Repairing grilles	6	piece	15	10	5	--	12	12
- Repairing ventilation windows	2	piece	15	10	5	6	6	6

Table 6. Maintenance costs (NPV) for three levels; mechanical extract ventilation

Type of installation	Chance complaint maintenance	Maintenance costs in NLG per dwelling in practice	Maintenance costs in NLG per dwelling (Dutch ventilation industry)
Average	low	568	842
	medium	939	1366
	high	1410	2227
Individual	low	594	
	medium	991	
	high	1488	
Central	low	541	
	medium	885	
	high	1329	

Finally the maintenance costs in practice, as provided by the 15 housing corporations, have been compared with the maintenance costs for the maintenance classes as given in table 7. Assumptions for these calculations are:

- average cycles ("medium" as in table 5)
- the range of the maintenance of single activities is corrected with the percentages, given by the housing corporations.

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These calculations lead to maintenance costs for single family dwellings of NLG 1134. This value is a little higher than the costs for maintenance class "medium" from table 7 (NLG 991). This indicates that the medium level reflects realistic values from practice.

Next question is what the impact is of a "good" or a "poor" complex on the total maintenance costs. Because of the lack of information that was derived from the questionnaires (housing corporations do not know the differences in costs) it was impossible to calculate with average values. However one corporation was able to provide detailed information. If these data are used for the calculations the following results occur:

Table 7: Maintenance costs in practice for two levels of basic quality.

Building type Basic quality	Single family dwellings (individual)	Multifamily dwellings (central)
Good (optimal)	NLG 1532	NLG 1523
Poor (critical)	NLG 2535	NLG 2005
Difference	NLG 1003	NLG 482

For single family dwellings these results cope the calculations with the assumptions. However for multifamily dwellings there are large differences.

SIMPLIFIED TOOL FOR ESTIMATING MAINTENANCE COSTS

In four steps the maintenance costs, expressed as NPV, are estimated. Also maintenance activities and cycles are recommended to maintain these level and corresponding costs.

In step 1 the basic quality is determined by some qualitative descriptions of the installation and building qualities and properties. In step 2 the users influence or user frequency is determined, also by qualitative descriptions. If the majority of the descriptions apply to the site then the corresponding class can be used; otherwise class "average" is applicable. In step 3 the maintenance class is estimated in a graph as a function of the basic quality and user frequency. Also some cost ranges for maintenance (planned and complaints/failures) are given. In step 4 the expected costs, expressed as NPV for planned maintenance and for complaints and failures, are estimated. Also the recommended maintenance activities and cycles are given. Special attention is needed for level "low". Some of the maintenance activities are carried out by the users (as a result of a conscious behaviour, resulting in a "low user frequency") and some maintenance activities don't have to be carried out at all as a result of the basic quality of installation components. There are some differences in cycles between the tables 5 in the tool and table 5 in this paper. This is because the field research showed that some of the maintenance activities were not carried out or with a very low cycle. However we do some recommendations for (extra) maintenance activities or cycles.

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MPLIFIED TOOL FOR ESTIMATING COMPLAINTS AND MAINTENANCE
)STS

'pe of building: Single family
'pe of ventilation system: Mechanical extract, natural supply

ecific qualities that are of concern for assessment:
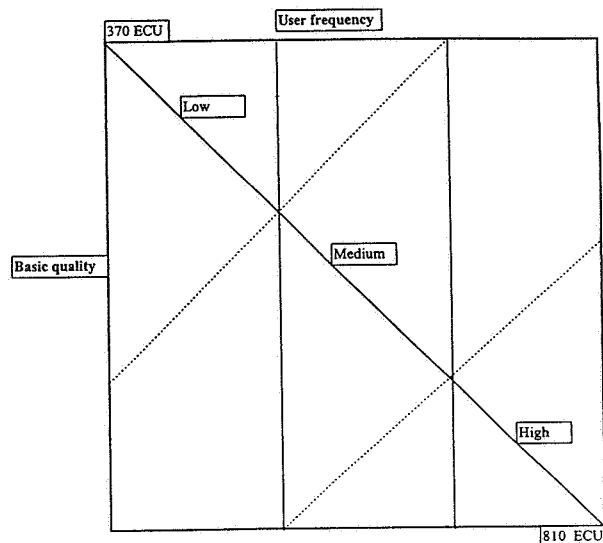
ntilation system, installation: *Building:*

nit: Purpose provided openings:
  location   - presence
  accessibility   - design, according to building regulations
  devises   - accessibility
uct and duct plan: - control possibilities
  lay out of the system   - cleaning possibilities
  devises   Building envelope:
  quality of design   - air tightness
  quality of execution
irilles:
  control possibilities
  cleaning possibilities

STEP 1: Estimate the basic quality by table 1

Table 1: BASIC QUALITY

	optimal	average	critical
Installation:			
unit	• easily accessible • mounted on construction with special sound proofing provisions	• accessible • flexible mounting on construction	• difficult accessible • no sound proofing provisions for mounting
ducts	• straight duct lay-out • short duct lengths • no sharp bends, flexible tubes • air tight sealing of ducts and connections • sound proofing provisions as silencers and flexible mounting on unit	• limited flexible bends • no special sealing of ducts; connections sealed with tape	• excessive duct lengths • many bends • flexible tubes • leaky ducts • no sound proofing provisions
grilles and adjustment	• adjustment and commissioning in every dwelling • completion reports of commissioning required • cleaning possibilities without disorder the adjustment (marking or securing)	• adjustment in limited number of test dwellings • completion report of test dwellings required • cleaning possibilities with limited chance of disorder the adjustment	• no measurements and/or completion reports required • no cleaning possibilities or cleaning possibilities with chance of disorder adjustment
Building:			
air tightness	• n50 = 3.. 5	• n50 = 5...6	• no requirements

Explanation:
Classification in category "optimal" or "critical" if most of the aspects are applicable for that specific category, otherwise category "average"

STEP 2: Estimate user frequency by table 2

Table 2: USER FREQUENCY

	low	average	intensive
Use of installation:			
instructions for use, control, maintenance etc.	personally addressed	written instructions are present	no instructions
cleaning grilles, unit	monthly cleaning	cleaning 4 x year	no cleaning
use	in accordance with manual und instructions	mostly in accordance with manual and instructions	disordering system
Building:			
ventilating, airing	conscious use of provisions	alternating use of provisions	incidental, very alternating or no use at all
cleaning vents	regularly	occasionally	no cleaning

STEP 3: After estimating the basic quality and expected user frequency the most suitable maintenance class can be established by table 3

Table 3: MAINTENANCE CLASSES

STEP 4: After estimating the most suitable maintenance class the expected maintenance costs can be estimated by table 4 and the planned maintenance activities by table 5

Table 4: EXPECTED COSTS OF MAINTENANCE
expressed as LCC over 30 years (ECU)

Maintenance costs	Planned maintenance	Complaints	Total
Low	300	72	372
Medium	522	145	667
High	590	218	808

Table 5: PLANNED MAINTENANCE ACTIVITIES

ACTIVITIES	CYCLE		
Maintenance class	Low	Medium	High
Installation:			
. measuring flow capacity	--	6	6
. cleaning grilles	1)	6	6
. inspecting ducts	18	9	6
. cleaning ducts	18	9	6
. cleaning extraction fan unit	3	3	3
. replacing grilles	--	18	18
. cleaning cookerhood	1)	--	6
. replacing cookerhood	--	18	15
. replacing extraction fan unit	18	18	15
. assessing control system	6	6	6
Building:			
. cleaning vents/grilles	1)	6	6
. repairing vents/grilles	--	12	12
. repairing ventilation windows	6	6	6

1) Activity carried out by occupant

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Ventilation and Cooling
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Introduction of Tools for Evaluating Domestic Ventilation Systems

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Synopsis

The IEA project Annex 27, Evaluation and Demonstration of Domestic Ventilation Systems, have come to the stage that simplified tools can be presented in a total scheme. At earlier AIVC conferences some of the tools have been presented in separate papers and still the tools are under development. In this paper a more general approach of the usage of the tools is to be presented.

The work is based on the joint work of participants from both AIVC countries (CAN, F, NL, S, UK, USA) and non-AIVC countries (I, J). The tools that are to be used are for indoor air quality, energy, noise, thermal comfort, life cycle cost, reliability, and building and user influences. A flow chart has been developed and is the procedure for the usage of the tools. The simplified evaluation tools are giving results both in a qualitatively and a quantitatively way. It has been applied on the four basic ventilation systems: adventitious, passive stack, mechanical exhaust, and mechanical supply and exhaust.

Background

The rate of outdoor air supply as well as comfort aspects associated with air distribution and the ability of the systems to remove pollutants are important factors to be considered at all stages in the building lifecycle. As distinct from a work place, residents can vary across a wide span from an allergic infant to a well trained sportsman, from active outgoing people to elderly confined to a life indoors. During the lifetime of a building the resident's pattern vary. This results in a varying need for supply air to obtain acceptable indoor climate and to avoid degradation of the fabric. Emis-sions from building materials are also time dependent. When the building is new or recently refurbished it may be necessary to dilute the emissions by extra outdoor air. In standards and codes the outdoor air needed in a dwelling is generally based on the maximum number of persons living in the dwelling, defined by the possible number of beds contained therein.

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

Objectives

The objectives of the IEA Annex 27 are: **to** develop tools to evaluate domestic ventilation systems; **to** validate the methods and tools with data obtained from measurements; **to** demonstrate and evaluate ventilation systems for different climates, building types, and use of the dwellings. The methods, tools, and systems are intended for existing and future residential buildings, that require heating. The target group is composed of standard and policy makers, developers in industry, and ventilation system designers. With this general objectives the Annex is divided in three Subtasks:
1.   State of the Art,
2.   Development and Validation of Evaluation Methods, and
3.   Evaluation, Demonstration, and Application Ventilation Systems.

Introduction

With the above objectives and scopes of the three Subtasks the Annex started in April 1993 and has today eight participants: Canada, France, Italy, Japan, Netherlands, Sweden, UK, and USA. Based on the subtask "State of the Art" assumptions have been set up to develop simplified tools for:

1. Indoor Air quality
2. Energy
3. Noise
4. Thermal Comfort
5. Life Cycle Cost
6. Reliability
7. Building and User aspects.

With the State of the Art Review, ref. 1, it is possible to give realistic assumptions of the most frequently used ventilation systems, the design of the dwellings, how many residents there usually are and when at home, the behaviour and which person in which room at what time. With these assumptions we can cover about 90 % of all possible cases, that are influencing the need of outdoor air supply. The usual levels of different pollutants in the dwellings are also given based on the review. The review report is based on and giving references to about 400 reports.

The OECD countries (14 of them) have 700 million inhabitants, 280 million dwellings with a floor space of 32 000 million m^2. The habitable space varies greatly and goes from 65 m^2/dwelling (Italy) to 152 m^2/dwelling (USA). There is also a great variation between the countries weather the dwelling is in a single family house or in a multi family building.

The number of persons/dwelling goes from 2.1 (Sweden) to 3.2 (Japan, Italy). Combined with the dwelling area it gives a floor space from 27 m^2/person (UK) to 61 m^2/person (USA). The crowdiness is defined by the number of persons/bedroom. From data can be seen that in 35 % - 50 % of all dwellings, there is less than 1 person/bedroom and in nearly all (90 - 95 %) less than 2 persons/bedroom.

A very important trend is that the number of one-person household is increasing. Today it goes from 20 % (Japan) to 40 % (Sweden). This trend has been observed during the last 45 years in all countries. A majority of the households have only two persons, except Japan (40 %). In the future it can be expected that we will have even more 1- and 2-person households as the number of persons older than 60 years during the next 40 years is growing from about 20 % today to 30 % of the population.

A survey amongst the AIVC countries gave that the most frequent ventilation system is either stack or simply window opening. However, in new constructions in most countries a fan is installed either for central exhaust or for local extraction in bath and/or kitchen.

Method

A complete computer simulation for various assumptions for the different parameters is never possible to be done for dwellings in the normal designing process. It is too expensive. With this in mind we are aiming at developing a set of tools that are easy enough to understand and use so that practitioner are able to use it even though not practised very frequent. The tools are

to be used in the design of new buildings or renovating existing. With the tools it may also be possible to give guidance to explain problems.

With this in mind the development of the tools aimed at having simplified tools based on the most sophisticated computer simulation codes that could solve our purposes. The tools had to be a mixture of qualitative and quantitative tools. All tools will have at least a qualitative way to make a judgement between the various systems, see table 1. By this we are covering most of the systems existing and possible as it can be combined.

Table 1. Combination of variables

Parameter	Window airing	Stack	Mech. exhaust	Mech. supply & exhaust
Local bath fan	Yes/no	Yes/no	Yes/no	Yes/no
Kitchen fan	Yes/no	Yes/no	Yes/no	Yes/no
Window airing pattern	3 cases	3 cases	3 cases	3 cases
Climate	Ottawa, London, Nice	Ottawa, London, Nice	Ottawa, London, Nice	Ottawa, London, Nice
Number of residents	2, 4, 5 persons	2, 4, 5 persons	2, 4, 5 persons	2, 4, 5 persons
Houses, flats	Detached. Ground & top floor. 4 storeys	Detached. Ground & top floor. 4 storeys	Detached. Ground & top floor. 4 storeys	Detached. Ground & top floor. 4 storeys
Exhaust flow rate [l/s]	-	-	15, 30, 45	15, 30, 45
Air tightness [N50]	House 2.5, 5, 10 Flat 1, 2.5, 5	House 2.5, 5, 10 Flat 1, 2.5, 5	House 2.5, 5, 10 Flat 1, 2.5, 5	House 1, 2.5, 5 Flat 1, 2.5, 5
Supply flow rate [l/s]	-	-	-	15, 30, 45
Supply area [cm^2]	0, 101, 410	0, 100, 400	0, 100, 400	-

For some of the tools we need to rely on laboratory tests and with more general assumption. By giving a tool that the user can feed his on data sets or experiences a workable and always up-to-date tool can be at hand.

The procedure how to use the different tools is shown by a flow chart, see figure 1.
All the tools are very briefly introduced. Some are given a more detailed introduction at this conference, IAQ was introduced at the 17th conference 1996. A more lengthy and detailed description is planned to be given at the AIVC conference 1998. Here after follows a brief introduction of the tools to be given.

Thermal Comfort
Different supply air devices, for location in external walls and windows, have been tested in laboratory. The thermal comfort in different points in the room have been measured at various external temperatures: Based on this a qualitatively based judgement table in 5 steps is set up for different outdoor temperatures device types as well as background leakage.

Noise
If the dwelling is situated in a noisy area like close to roads, rail - ways, airports also the choice of the ventilation system affects the noise reduction. A leaky house, windows with bad weather-stripping, or a device in the window or wall can ruin the good intentions. As the noise

Tool Flow Chart
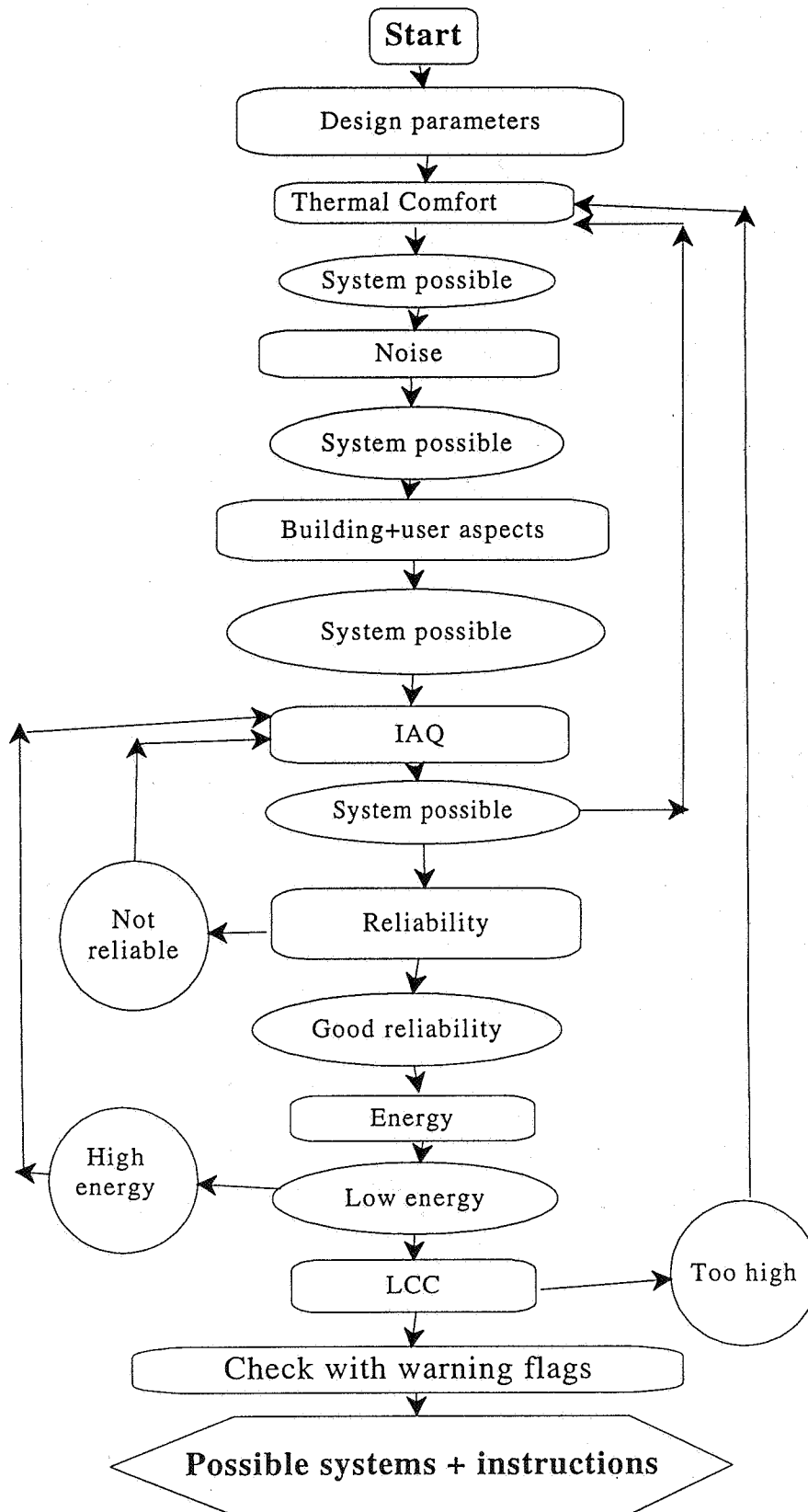
Figure 1. Tool flow chart for evaluating domestic ventilation systems

reduction is depending on many factors a set of cases have been calculated for the four main ventilation systems. Also the devices for outdoor air in stack and mechanical exhaust systems can be designed in many different ways giving more or less noise reduction. Tables are set up with the 5 step judgement to meet different required noise reductions taking into account the opening area for the devices with different noise reduction design. Ref 3.

Building and user aspects
The building aspects are listed and discussed how they might influence the various systems. The dwelling lay-out and height of the rooms are of more concern the more the ventilation is relaying on window openings. The system selection might also affect the architectural design for the room design, facades and roofs. Construction and services design are also depending on the ventilation system chosen.

Even though many assumptions are made also other aspects may influence the choice of the ventilation system. A check list or a qualitatively way of judgement is to be set up. The various aspects discussed are:
- User influence on how to use the system, obstruct the system, change the preadjustment
- Maintenance: Systems are more or less sensitive for cleaning the devices and filters or changing the filters.
- Furniture and decoration can make obstacles for the air flow. Most problems are caused by the curtains around the windows.
- User tendency to react on system failure either it is instant failure and slow degradation
- Cooking habit varies from country to country and over the long time. It is also affected by the family life stile. If a kitchen hood is used the sensitivity is less.
- Hygiene: There is a tendency to use more water giving a higher water vapour content in the dwelling. The use of chemical products are under the development to be less harmful.
- Redecoration interval. Usually it includes more or less extra water vapour to the dwelling in addition with extra VOCs. Usually it is linked to changing tenants or ownership.
- Other equipment installed or used: Here a lot of creativity is put in by quite another industries than the ventilation. Here can be foreseen many new machines that might affect the indoor environment more or less e.g. steam cleaners of floors, coffee machines, ovens not connected to the kitchen hood or gas fired grills not under a hood.
- Pets, the size and number varies a lot as well as the fur length and shedding
- Pot plants are giving some addition to the indoor water vapour content.
- Interest
- Knowledge how to handle the situation for improving the effectiveness of ventilation
- Manual capability
- Life style is affecting the family's attitudes towards all the behaviour. Some of the relationships are investigated. The life style can be divided into 10 styles. These are: Work geared, Nature-love, Family - attached, Well-informed, Consumption geared, Agrarian/Religious, Moralist, Collective bent, Climber, Economiser

IAQ
The intention is here to give both qualitative and quantitative tools. Calculations are made for the following parameters: Constant emission, CO_2, cooking products (incl. NO_x), tobacco smoke, pressure difference (positive and negative), relative humidity (mould growth & house dust mite growing risks), outdoor air change rate. The combination of all parameters and values gives 17 500 cases and are dealt with by multi-variat parametric study using COMIS and SIRÉN computer programs. Ref 2.

Reliability

Here assumptions have to be made on the lifetime of each component in the system selected. Also other the probability of a system to give the ventilation rate is here to be given. Sets of tables can be established for the probability and a computer code for the decrease of the flow rate at different replacement and maintenance intervals.

Energy

Fan and air heating energy is calculated taking into account flow rate, heat recovery efficiency, air leakage (n50), window airing habits, climate. A computer nomogram is to be the final tool for easy handling.

LCC

Mostly ventilation systems lack good maintenance. Some property owner have realised that each complaint cost money to deal with. In dwellings it is also a combination of maintenance made by the residents and by the professional organisations. A general trend is that it pays to do planned maintenance. The qualitatively approach is to combine the basic quality of the system for units, ducts, devices and possibilities to clean and adjust. After that the user's interest of the systems is estimated in three levels. By combining the two we will have three fields low, medium and high maintenance classes. The cost is estimated for the low and high for the various systems and building types. Planned maintenance costs is added with the cost for complaints.

Discussion

The tools are to be developed to a paper tool. Of course it is of great concern to go on to develop the tools to computer versions so that it can be easier to use and to include more climates and having a self - educating instruction for each tool. Our goal have been to give designers tools that they can bring with them at meetings giving answers on various questions that are arising in meetings concerning the choice of systems. There are still a few months of work before the final tools are fully developed. The tools are to be checked on the measurements made during the annex running.

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VENTILATION AND COOLING

18TH ANNUAL AIVC CONFERENCE
ATHENS, GREECE, 23-26 SEPTEMBER, 1997

EC-THERMIE project Heerlerbaan: Multi-functional appliances for retrofitting residential buildings

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SYNOPSIS

Many post-war residential buildings in the Netherlands have collective heating systems with poor energy efficiency. Also ventilation and DHW systems usually do not comply with current requirements. In Heerlen, the Netherlands, a demonstration was carried out in the framework of the EC-THERMIE programme in a residential building where the collective heating, DHW and ventilation systems are replaced by individual multi-functional appliances. These appliances are a recent development in the Netherlands, integrating different service functions. A measurement and evaluation programme was completed in May 1997, showing energy savings for space heating of 48 % over a period of 2.5 years and a significant improved ventilation and indoor air quality.

INTRODUCTION

In Heerlen, the Netherlands, an EC-THERMIE demonstration project is carried out, concerning the retrofitting of 120 apartments in a residential building. Individual multi-functional appliances replaced a collective central heating system, unvented hot water appliances and passive stack ventilation. The building is a typical example of a Dutch high rise flat for social housing, built in the sixties. In these types of buildings installations for space heating are usually energy inefficient. Poorly functioning ventilation systems and unvented hot water appliances ("geysers") often cause indoor air problems. Retrofitting installations for space heating DHW and ventilation is a complicated process in which the different components can have a major influence on one other.

In new and retrofitted airtight dwellings ventilation must be carefully designed. Natural supply by using vents in the facade can give problems due to a poor thermal comfort. Occupants close the vents in case of low outdoor temperatures or high wind velocities. This can lead to poor indoor air quality, moisture problems and mould grow. Also a high underpressure can occur if airtight dwellings have mechanical exhaust and vents are closed. This leads to serious problems with spillage of flue gasses from open combustion appliances. This means that also the selection of the type of ventilation system has a close relation with the selection of the types of combustion appliances. Balanced ventilation, especially in combination with closed combustion appliances, gives a good solution to deal with these problems. Even then indoor air quality problems can occur if occupants don't use the ventilation system, for example, by turning off the system. Because in a MFA ventilation is linked with hot drinking water the possibility of not using the ventilation system is minimised (no ventilation means no DHW). MFA's give an integral solution for the selection of ventilation system, combustion appliances for space heating and DHW and minimising malfunctioning of one or more system components because of occupants behaviour.

Another advantage of MFA's is that the number of ducts is limited. There are only two ducts necessary leading from the unit to the outside: one for fresh air and one for mixed exhaust air and flue gasses. For a comparable system (balanced ventilation and a closed combustion appliance) three or four ducts are necessary: fresh air, exhaust air, combustion air, flue gasses (flue gasses and combustion air can be combined in one duct). This can be of special interest for retrofitting when also ventilation and heating systems are involved.

Although originally designed for new built energy efficient dwellings, MFA's are also very suitable for energy efficient retrofitted dwellings. However there still remained risks on some technical and economical aspects for this application. These risks as well as the demonstration character for a large number of dwellings with similar problems all over the European Community, was a reason to submit this project in the EC-THERMIE programme.

Multi-functional appliances (MFA) are a new integral approach to space heating, hot drinking water and ventilation. These three functions are combined in one appliance. Heat recovery (8) takes place from the exhaust air for ventilation (1) and flue gasses (2). Moreover other waste heat such as convection and radiation losses and heat of fans is used. A MFA contains a combi-boiler for the heating systems (for feeding radiator panels 4) and hot drinking water (3). A MFA also contains a balanced ventilation system with mechanical supply (5,7) and exhaust of air (1,6). Extracted air (1) from bathroom, toilet and kitchen is used as combustion air for the boiler. Flue gasses (2) and extracted air (1) are mixed before they pass the heat recovery (8).
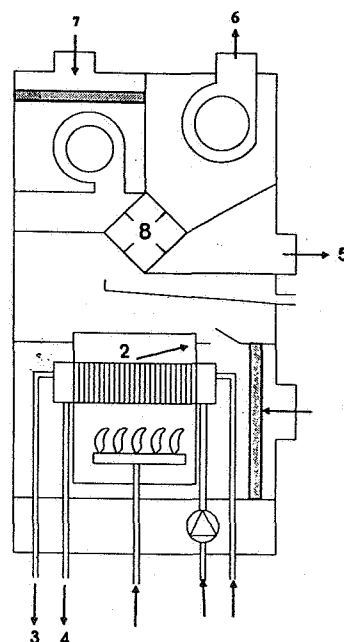

Figure 1: Principle of the appliances used in this EC-THERMIE demonstration project.

EC-THERMIE DEMONSTRATION PROJECT "HEERLERBAAN"

Multi-functional appliances are demonstrated in a retrofitted residential building in Heerlerbaan, municipality Heerlen, the Netherlands. This building is a part of a complex with five similar buildings. The building has 12 stocks and contains 120 apartments. The apartments have a floor area of 88 m^2 and have a living room, three bedrooms, a kitchen, bathroom and toilet. In the situation before retrofitting the building had a collective central heating system. The apartments had floor heating and one radiator in the living. There were no individual control devices. There was passive stack ventilation and natural supply through small ventilation windows. Hot drinking water was supplied by unvented hot water appliances in the kitchen. In the non-retrofitted situation there were many problems with the heating system. Specially the energy use was very high and the poor control systems cause overheating in many apartments. There were also indoor air quality problems caused by the hot water appliances (CO, NO_2). The natural ventilation system did not comply with Dutch ventilation standards.

A MFA is more or less designed for (new built) energy efficient single family dwellings and low rise residential buildings. However, there appeared to be no objectives using these appliances in this particular situation. As well, MFA's appear to be a very interesting solution for the complicated selection of heating, ventilation and DHW system for retrofitted buildings. Moreover, the occurring problems could be solved with one appliance. However, there were some technical risks for applying MFA's in high rise buildings. An important aspect was the functioning and the safety of the collective duct for combined exhaust of flue gas and stale air. In 12 apartments, connected to one collective duct, the performances and functioning of the different components were monitored and evaluated during one full year. After a successful test year the rest of the building was retrofitted between March and October 1996.

METHODS

The demonstration project is supported by an extended measurement and evaluation programme. This programme contains:
- Performances of the appliances and system components
- Ventilation rates measured with PFT and constant tracer gas technique.
- Indoor air quality by measuring TVOC, CO_2, CO, NO_2, relative humidity and temperature.
- Air tightness of the building envelope.

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- Monitoring monthly energy uses.
- Occupants survey on using ventilation provisions.

Ventilation rates were measured during one heating season by means of PFT technique. These measurements were carried out in five retrofitted dwellings and five non-retrofitted dwellings. In four retrofitted and four non-retrofitted dwellings ventilation rates were measured with constant tracer gas method during one week. Simultaneously In these dwellings indoor air quality parameters were measured. During the measurements occupants recorded the use of the ventilation provisions, window airing, inner doors and heating system. CO_2, CO, TVOC, temperature and relative humidity were measured during one week by using a Bruel & Kjaer 1302 gas monitor and 1303 sampler and doser unit. NO_2 was measured by using Palmes tubes.

RESULTS AND DISCUSSION

Energy saving in the complex

Energy use is monitored from October 1994 up to May 1997. From October 1994 till May 1996 monitoring took place in twelve test apartments and in the other non-retrofitted part of the same building (118 apartments with a collective central heating system). In October 1996 the renovation of the whole complex of 120 buildings was completed. Energy use was monitored in the total complex from October 1996 up to May 1997 as well as in two similar non-retrofitted complexes (240 apartments). In this way energy use before and after retrofitting could be compared. In figure 2 the gas use for space heating is given for the period from October 1994 till May 1996. In figure 3 the gas use for space heating is given for the period from October 1996 till May 1997. In the first period the average gas consumption for space heating decreased by 49 %, from 1688 m^3 nat.gas/dwelling per year to 862 m3 nat.gas/dwelling per year. In the second period the average gas consumption for space heating decreased by 48 %, from 1610 m^3 nat.gas/dwelling per year to 844 m3 nat.gas/dwelling per year. These reductions were achieved by only applying multi-functional appliances. No further measures for thermal insulation or air tightness were taken.
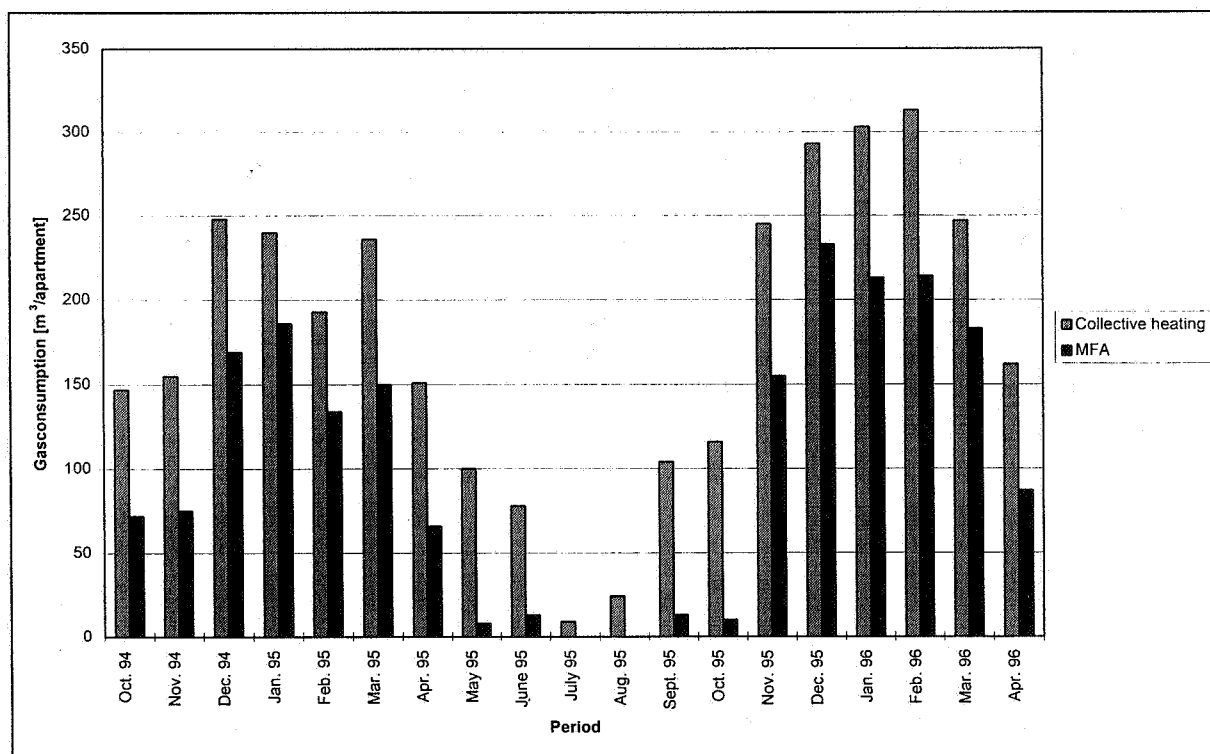
Figure 2 Gas consumption for space heating 12 retrofitted test apartments (MFA) and 118 non-retrofitted apartments in the same building.
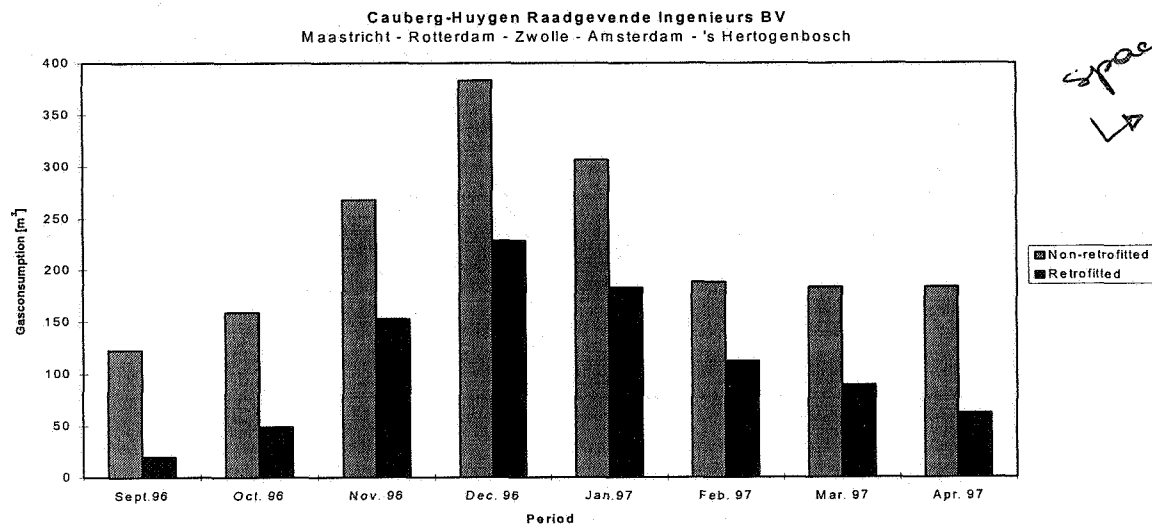

Figure 3 Gas consumption for space heating 120 retrofitted apartments (MFA) and 240 non-retrofitted apartments.

Also the electricity use of the MFA's (fans, pump, electronics) and the total electricity use (household) was monitored. Before renovation the electricity use for installations was 210 kWh/dwelling per year due to the central pumps of the heating system. The monitored electricity use of the MFA's during the first year was 850 kWh/year. This was 200 kWh more than the prognosis of 650 kWh. The total electricity use was 2783 kWh/dwelling which approaches the average electricity use for Dutch households (approximately 2900 kWh/year).

Energy performance of the appliances
The total energy performance of the installed multi-functional appliances is measured and evaluated under practice circumstances during 24 hours, an outdoor temperature of approximately 0° C and simulated habitants behaviour. This is done by real time measuring all input energy (gas and electricity, input air flow) and output energy (hot water and output air flow). Most energy flows are defined by their temperature, specific heat and mass flow. For the results of this measurements it cannot be claimed the same accuracy as measurements under laboratory circumstances. An energy balance for the complete period is based on the calculated energy transmissions.

The first conclusion of a total energy balance is that the total input energy is 13 % more than the total output energy. This is a common value for measurements under practice circumstances and is caused by electric losses (pump, valves and fans) and non-measurable radiation losses.

The net unit efficiency, defined as the useful output energy divided by the total input energy is 92,5 % while the gross unit efficiency, which is calculated without the electric input energy is 95,9 %. The efficiency for making domestic hot water is 81,7 %.

An indicator for the economic efficiency of the unit is the energy-index, defined as the total energy cost during one period divided by the total amount of output energy. For this unit the energy-index becomes DFL 20.50/ GJ (price level 1997).

Table 1: Input and output energy during measurement period

INPUT	Description	Value	Energy [MJ]	Cost [Dutch guilders]
Natural gas	total	4.41 m³	155.1	2.65
Electricity	total	3.9 kWh	14.0	
	peripherals	2.3 kWh	8.3	
	net	1.6 kWh	5.7	0.40
		Total:	**160.8**	**3.05**

	Description	Value	Energy [MJ]
Water	space heating	7112 kg / $\Delta T = 3.2°C$	95.1
	DHW	75 l / $\Delta T = 39°C$	12.5
			107.7
Air	removal dwelling	fluctuating	-46.0
	supply dwelling	fluctuating	41.1
	fresh air	fluctuating	-12.3
	exhaust flue gases	fluctuating	47.0
			29.8
		Total:	**137.5**

Ventilation rates and Indoor Air Quality

The air tightness of the dwellings in which the ventilation measurements took place was measured according to the Dutch standard NEN 2686 "Airpermeability of buildings. Method of measurement". There are no differences in the building envelopes of the retrofitted apartments with a MFA and the non-retrofitted apartments with collective heating. The air leakage flow by a pressure difference of 10 Pa (NEN 2686) is 43 dm³/s for the non-retrofitted apartments ($n_{50} = 1.71$) and 45 dm³/s for the retrofitted apartments ($n_{50} = 1.80$). There is no significant difference in air tightness for the retrofitted and non-retrofitted apartments.

In the non-retrofitted apartments (natural supply and PSV) the average ventilation rates are 37.8 dm³/s for the living room and 6.1 dm³/s for the bedrooms. The continuous constant tracer gas measurements show very strong variations of air flows from 0 up to about 300 dm³/s.

In the retrofitted apartments (balanced ventilation) the average ventilation rates are 15.3 dm³/s for the living room and 9.4 dm³/s for the bedrooms.

Table 2: Measured indoor air quality parameters and ventilation rates (PSV and MFA)

	non- retrofitted (natural supply + PSV)			retrofitted (balanced ventilation by MFA)			
	Living room	kitchen	bedrooms	living room	kitchen	bedrooms	guidelines
CO_2 (mg/m^3)	1863	2406	2621	1527	1610	1338	1800
CO (mg/m^3)	3.1	4.2	3.1	2.4	2.6	2.3	10 (8h)
TVOC ref. CH_4 (mg/m^3)	5.0	5.3	5.2	2.8	2.4	2.3	(3)
NO_2 $(\mu g/m^3)$		135			25		150 (24h)
Airflow const.tracergas method (dm^3/s)	37.8		6.1	15.3		9.4	-
Air flow PFT (dm^3/s)	31.3 (total dwelling, heating season)			55.1 (total dwelling, heating season)			-
RH (%)	34	36	36	40	42	41	30 - 70

Constant tracer gas measurements show that the air flows have much less variation and are rather constant in time. Although the average total air flow in the retrofitted apartments is smaller than in the non-retrofitted apartments the quality of ventilation is much better. Figure 4 shows that the ventilation pattern differs completely. In the non-retrofitted apartments the ventilation (and indoor air quality) is much more influenced by occupants behaviour and weather conditions. Occupants use more window airing and window ventilation causing great variations in air flows. Variations in air flows in the retrofitted apartments are mainly caused by occupants switching the fans (low/high) and the opening and closing of inner doors.
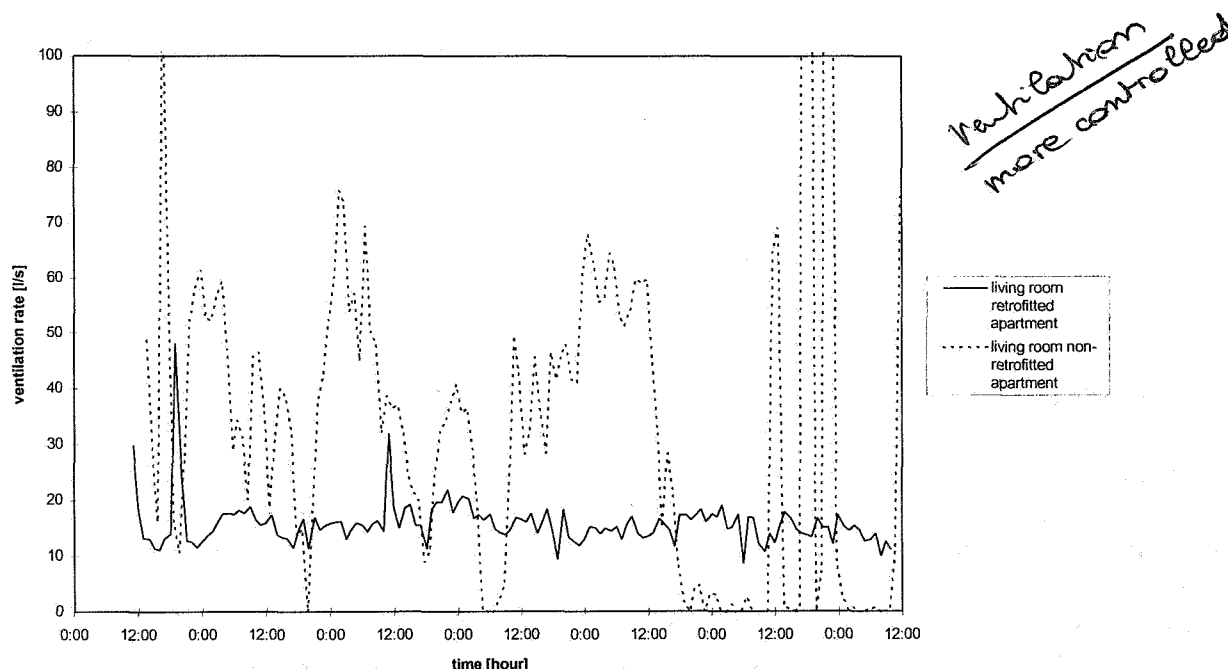

Figure 4. Ventilation rates (dm³/s) living room in a non-retrofitted and retrofitted apartment

In the non-retrofitted apartments the average CO_2 concentration exceeds the hygienic guideline level of 1800 mg/m³ . The average CO and NO_2 levels did not exceed the guideline levels. However, in one apartment a peak value occurred of 78 mg/m³ during 15 minutes in the kitchen. This level means a serious health risk. The main reason for these levels and the peak values are the unvented hot water appliances in combination with the lack of possibilities for controlling the ventilation exhaust. In the retrofitted situation all measured concentrations of indoor air quality parameters show a significant decrease for average levels as well as for peak levels. Source removal and improved ventilation show results in better indoor air quality.

Figure 5 shows CO levels in a non-retrofitted kitchen with PSV and an unvented hot water appliance and a retrofitted kitchen with mechanical extraction.

The ventilation rates measured by PFT and constant tracer gas method will be further analysed and linked with the air tightness measurements and the occupants survey on the use of ventilation and airing provisions in IEA-Annex 27: "Evaluation and demonstration of domestic ventilation systems" (2).
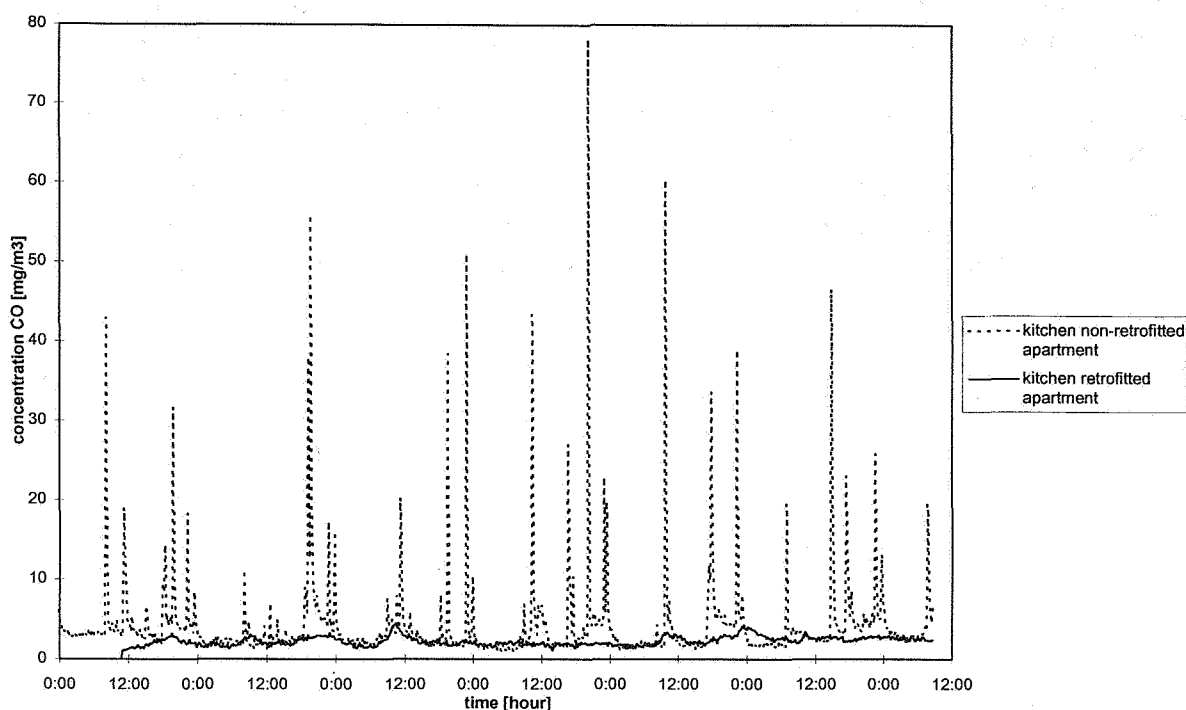
Figure 5. CO levels (mg/m^3) in a retrofitted and non-retrofitted kitchen

Occupants survey

The objective of the survey is to get general insight in the retrofitting as a process and the functioning and acceptance of the multi-functional appliances (MFA). For several reasons an oral survey was chosen. The response was 75 % out of 72 addresses.

Information and arguments provided by the housing corporation:

The housing corporation Heerlerbaan provided clear and understandable information. The arguments used such as individual temperature control, energy saving and improved ventilation were accepted by 75% of the occupants.

Appointments and realisation of the retrofitting:

Most appointments were carried out correctly. 20% Felt tied at home during the activities. Noise and dust caused inconvenience to 30 % of the occupants. A few damages due to the activities were reported; they were mostly repaired by the housing corporation. In some apartments earlier improvements made by the occupants, like an additional cooker hood in the kitchen, were removed with permission of the occupants.

Distribution of the costs:

There is no raise in rent or in service costs calculated for heating and ventilation investments done by the housing corporation. A small raise in rent, equal to the rent calculated by the local utilities (MEGA) for a similar DHW appliance, is made for the DHW system. This raise is accepted by 65%. After retrofitting occupants are now individual financial responsible for their energy consumption. The general opinion is that this is a more fair system than before.

Functioning and acceptance of the multi-functional appliances:

Based on the first the occupants appreciate the individual space heating system, the temperature control in different rooms and the increased fresh air rate in the bedroom. All occupants with more experience (a full winter season included) are very positive. A minority of the occupants has problems with the control of the MFA and low temperature level in the bathroom. The location and size of the radiators installed is judged positively. The MFA is installed in a cupboard in the hall. This solution means that this space is not available anymore for storage. This solution is reported as negative but necessary by most of the

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occupants. For this reason, additional storage capacity is installed by the housing corporation. The DHW system before retrofitting caused smell and very poor indoor air quality (NO_2, TVOC, CO). All occupants appreciate the comfort of the DHW system but the waiting time to get hot water is too long (a common problem with these kind of so called "combi-boilers"). Complaints of natural ventilation system before retrofitting concerned cooking smells and condensation. After retrofitting these complaints are far less and occupants report a significant increase of fresh air and improved indoor air quality. There is no univocal information on occupants with allergic sensitivities and their experiences. There are few negative reactions on the location of installed ducts. The size and finish of the ducts are reported positively. The fan is judged positively by most occupants. A few occupants report a great or, in contradiction, others a too small air capacity. Noise caused by the fan is reported by most occupants, especially at night. Half of the occupants wonder if the ventilator can be turned off for certain periods of time, for example at night and during the period of vacation or other absence. The general opinion is that the amount of draught has not changed; 15% of the occupants notice an increase in the amount of draught. This minority names several different rooms with draught, it is not clear where this draught is located exactly.

CONCLUSIONS

- During the demonstration between October 1996 and May 1997 no major occurred concerning the functioning and performance of the multi-functional appliances.
- Gas use for space heating decreased by 48 % compared to the adjacent similar blocks, only by retrofitting the heating system.
- Gas use for DHW increased. This problem is not inherent to MFA's but due to an increase in comfort of hot water supply in relation to small water heaters like unvented "geysers". As a result of this conclusion further experiments are going on in this project on measures to limit the hot water use without influencing the comfort.
- Although in retrofitted apartments average ventilation rates decreased, due to better control, the quality of the ventilation improved, resulting in better indoor air quality. Measurements of indoor air quality parameters show a significant decrease of concentrations CO, CO_2, NO_2 and TVOC.
- Electricity use of MFA's was about 130 kWh (20 %) higher than expected. In the next generation MFA's DC fans will be applied, decreasing electricity use with 50 %, and high efficiency heat recovery (laminar cross-flow heat exchangers) with an efficiency > 90 %.

ACKNOWLEDGEMENTS

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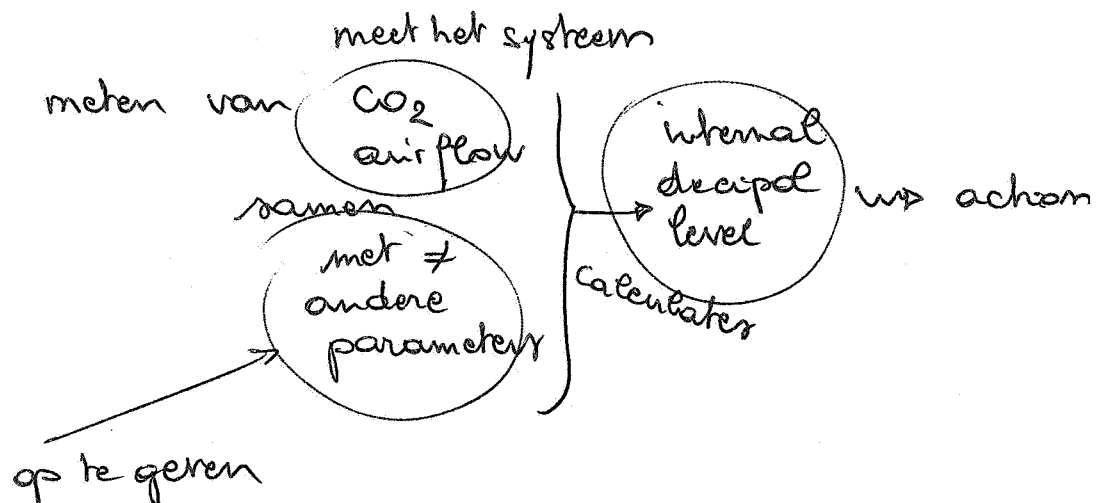
VENTILATION AND COOLING

A decipol Predictive Controller for VAV Systems

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Synopsis

Due to the lack of proper sensors for odours, the odour concept, involving the units olf and decipol, is of very little practical use with respect to automatic control of VAV systems. However, the decipol level in a room may be predicted from the concentration of CO_2 and the amount of fresh air supplied. By using the CO_2 level as a decisive variable of the occupant load within the room, the actual air quality (decipol level) can be predicted. Once the decipol level is known, it is compared to a given set point, thus enabling the controller to alter the air flow rate accordingly. Eventually the room air quality should become equal to the specified set point. In this study, such a demand controlled ventilation algorithm is presented. The performance of the controller is visualized by simulation, and the results show that the approach to control the decipol level based on measurements of the CO_2 level could be advantageous compared to commercial CO_2 control.

1 Introduction

The olf and decipol units were introduced by Fanger in 1988 [2]. The basic idea was to find a method that related a quantitative measure of odours (olf) to a quality measure of odours (decipol), exactly the same way as any contaminant mass flow is related to a concentration within a space. A consequence of the lack of odour sensors, was that the results (the olf and decipol units) were based on the subjective perception of trained odour panels. One olf is defined as the emission rate of odours (bioeffluents) from a standard person and one decipol is the perceived air quality in a room hosting a standard person, ventilated with 10 litres/sec of fresh air. Olf emissions from other sources, such as building materials, are expressed in terms of the emission from one standard person (a one-olf source). The work by Fanger (see [2] and [3]) and the general acceptance of the units have led to new regulations and standards for determining minimum air flow rates for ventilation in many countries, for instance the new building regulations in Norway. Odour emission sources can roughly be categorized as follows:

1. From human occupancy

2. From non human occupancy, such as building materials and indoor processes

3. From external sources, such as the outdoor environment and depositions in the ventilation system.

The odour emission due to human occupancy depends mainly on the number of persons present and their physical activity levels. The emission rates from non human occupancy and external sources can in many cases be considered to be constant over time. Theoretically, the decipol level in a room can be used as the decisive control variable in a demand controlled ventilation (DCV) system. By measuring the actual decipol level and comparing it to a given set-point level, the required amount of fresh air to the room can be computed by the flow controller. The obvious obstacle for enabling such a control system in practice is that no reliable odour sensors exist. In efforts to overcome this problem, mixed-gas sensor arrays and VOC sensors in combination with CO_2 sensors have been introduced (e.g. as discussed in [4] and [5]), showing quite promising results. The intension of such sensors is to provide an overall measure of the air quality in a room. Unfortunately, there are practical limitations with respect to the number of substances which can be detected.

Two important matters are (1) which of the many present substances should be given most weight, and (2) the magnitudes of the set-points which are to be given to the controller. Accuracy, stability (instrumental drift), costs for calibration and price of the sensor should also be taken into account.

The decipol level in a room can be achieved indirectly, by measuring the CO_2 concentration level. The CO_2 level provides a measure of the number of persons present in a room, hence the CO_2 level may be recognized as a surrogate for human related odours. In the following section, a new control algorithm based on the measured CO_2 level and ventilation air flow rate is discussed. Simulations of a single zone VAV system is used to visualize the capability and performance of the controller.

2 Control algorithm

The block diagram shown in figure 1 gives an overview of a possible configuration of a single zone VAV system. A ventilation system with a fan, ducts, a coil and a terminal device provides fresh air to a room. The measured CO_2 room concentration combined with some given control parameters are converted to a measure of the occupancy.
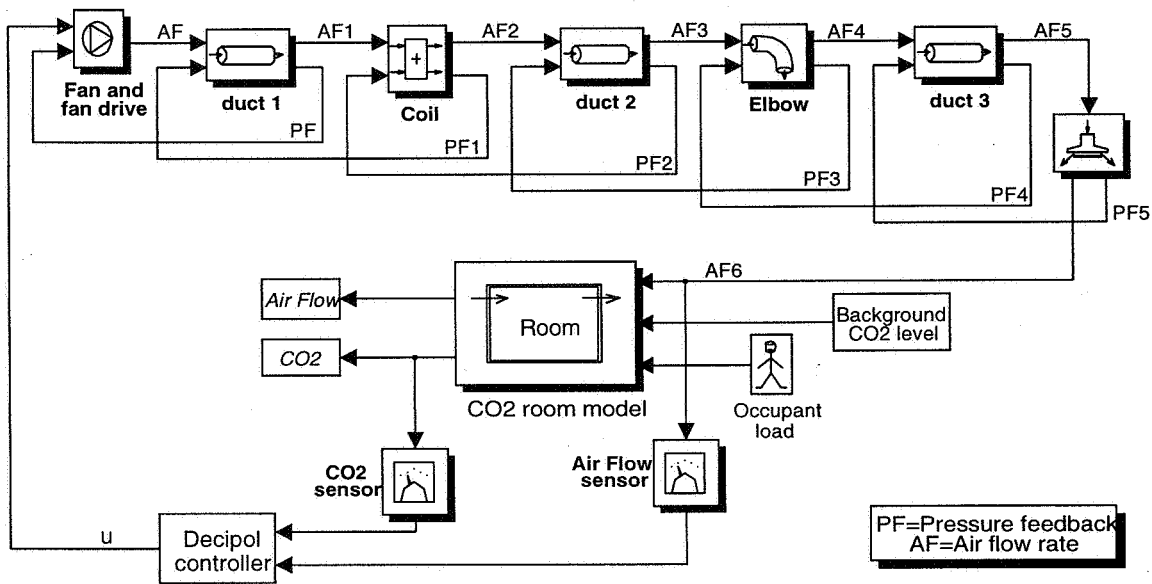

Figure 1: Block diagram of a odour controlled VAV system.

It has been assumed that the contaminants in the room (CO_2 and odours) is evenly distributed and that no infiltration air is present. These assumptions very much simplifies the room model, although it might be modified to take those effects into account.

The steady state CO_2 mass balance for a room with fully mixed conditions (and no infiltration present) can be written as:

$$C_s \cdot \rho_{CO_2} \cdot Q_a + S \cdot \rho_{CO_2} = C_r \cdot \rho_{CO_2} \cdot Q_a \qquad (1)$$

C_s : Supply (background) CO_2 concentration $[m^3/m^3]$
C_r : Room CO_2 concentration $[m^3/m^3]$
S : Total CO_2 emission $[m^3/h]$
ρ_{CO_2} : Density of CO_2 $[kg/m^3]$
Q_a : Ventilated (fresh) air flow rate $[m^3/h]$

The CO_2 emission from n persons can be expressed as a function of their mean activity level M as follows:

$$S = k \cdot M \cdot n \quad [m^3/h] \tag{2}$$

The magnitude of the proportional factor k is determined by the respiratory quotient (RQ) times the amount of oxygen consumed by respiration. Hence, RQ is defined as the volumetric ratio of metabolic produced CO_2 to consumed O_2 [1]. RQ is found to be constant, independent of the level of physical activity, thus k must be a constant too. For a person having a normal diet mix of fat, carbohydrate and protein, RQ is equal to 0.83 [1]. At an activity level of 1.2 met, the O_2 consumption is about 0.0216 m^3/hour [1], and hence the value of k becomes $\frac{0.83 \cdot 0.0216}{1.2} = 0.015$ m^3/hour per person and met.

A combination of the equations 1 and 2, reveals the following expression for the occupant load:

$$n = \frac{200 \cdot Q_a}{3 \cdot M} \cdot (C_r - C_s) \tag{3}$$

If the supply concentration varies significant over time, for instance due to heavily traffic outdoor or due to recirculation of air, this concentration should be measured as well. However, it has been assumed that no recirculation of air is present and that variations to the background level is negligible. Thus C_s is constant.

Now, establishing a similar balance for odours in the room:

$$G_s \cdot Q_a + 36 \cdot F = G_r \cdot Q_a \tag{4}$$

G_s : Supply (background) decipol level
G_r : Room decipol level
F : Total olf emission to the room

The background decipol level is in the range 0 (clean air) to 0.5 decipol (low AQ in cities). The scaling of the olf emission term $(36 \cdot F)$ of equation 4 comes from the definition of the decipol unit (1 olf ventilated with 10 litres/sec or 36 m^3/h of fresh air). The total olf production in the room (F) is the sum of occupant related odours (F_o) and building related odours (F_b). F_b emits from materials and processes within the room.

$$F = F_o + F_b \quad [olf] \tag{5}$$

F_o will of course vary by the occupant load, and the contribution to F_o from a single person is very much dependent on the level of activity. Also, a major influence is caused by tobacco smoking, if smoking is present. F_o can be written as:

$$F_o = f_o \cdot n \quad [olf] \tag{6}$$

where f_o is the olf emission from one person in the room. The magnitude of f_o as a function of the activity level can be found in various literature, for instance from Fanger [3]. A standard person conducting sedentary work , i.e. having an activity level of 1-1.2 met, emits 1.0 olf to the room air. f_o increases dramatically with the percentage number of occupants smoking.

The emission from non occupant sources (F_b) can be written as:

$$F_b = f_b \cdot A + F_p \quad [olf] \tag{7}$$

where f_b is the olf emission from building materials per. square meter floor area A, and where F_p represents the emissions from processes within the room. The first term of

equation 7 (f_b) will often decrease slowly, during a relatively long period of time. Fanger [3] found that the mean value of f_b for offices and schools was 0.3 olf/m^2. Although it is possible to take into account any variations of f_b, it is here, however, considered to be constant.

The combination of the equations 4 and 5 gives the following expression of the actual decipol level in the room:

$$G_r = \frac{36}{Q_a} \cdot (f_o \cdot n + f_b \cdot A + F_p) + G_s \quad [decipol] \tag{8}$$

The following algorithm should be implemented to the controller [6], [7]:

If	$G_r < G_{r,\min}$
then	Decrease the controller output signal (decrease the fan speed)
elseif	$G_r > G_{r,\max}$
then	Increase the controller output signal (increase the fan speed)
else	Do nothing

$G_{r,\min}$ and $G_{r,\max}$ refer to the lower and upper control limits of the room decipol level, respectively. The controller ensures that the room level is kept within these limits. For instance, one might want to allow an occupant dissatisfaction of maximum 20% in the room, meaning that the upper decipol level should be kept below 1.4 decipol, Fanger [3]. The lower limit, which also must be given as a control parameter, should be specified, for instance 0.1 to 0.2 decipol below the upper limit.

In figure 2, a block diagram of the controller model is shown. The blocks are named in accordance with the nomenclature used above. The controller has two inlets and a single outlet. The inlets are the measured CO_2 level (C_r) and the air flow rate (Q_a), and the outlet is the controller signal (u). Five different parameters have to be specified to the controller; the metabolism (M), the background CO_2 level (C_s), the olf emission from building materials per square meter floor area (f_b), the floor area (A) and the background decipol level (G_s). Two multiplexers are used to vectorize the different parameters and variables used by the equation blocks (Eq. 3 and Eq. 8). A relay modulates the controller output sign according to the computed decipol level (G_r), and a rate limiter ensures that the rate of changes to the output signal is not too high (or too low), causing possible problems of instability.

3 A brief description of the simulation system

The simulation system is built from different component models. The system contains models for ducts (straight and elbow), a fan, a terminal device and a coil (see [6] for further details). In addition, a simple room model is used to predict the CO_2 conditions. Based on turbulent flow conditions, implicit, nonlinear relations are employed to determine the pressure loss through the components. The pressure loss output from a model is fed backwards through the system, while the corresponding air flow output is fed forward (for an overview, see figure 1). The simulation system shown in figure 1, has the following properties:

- The fan model (fan motor included) has two inlets; (1) the controller output signal and (2) the total pressure loss. Ideal fan laws are employed to determine the corresponding air flow output from the fan.
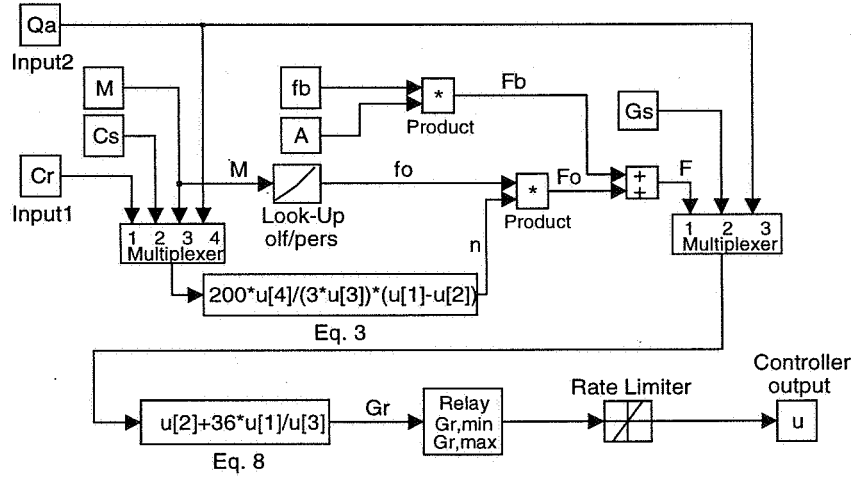
Figure 2: Block diagram of the controller.

- Both the duct and the elbow model account for pressure loss by friction. In addition, the elbow model has a table look up for single losses.

- The pressure losses in the coil and the terminal device are considered to be quadratically proportional to the air flow (i.e. fully turbulent flow).

4 Simulation case

In most cases a dynamic simulation system has a large number of parameters which must be specified, hence the number of possible simulation cases are almost infinite. In this study, only influences of the major load of the system have been investigated, i.e. the room occupancy.

The chosen room for the simulation case was typically a meeting room where the occupant density was expected to be high. The floor area was 40 m^2 and the ceiling height 2.5 m, giving a room volume of 100 m^3. Olf emission from building materials was expected to be 0.15 olf/m^2. The supply air quality was assumed to be around 0.05 decipol, constant over time, and no internal odour emitting processes were present. The ventilation system was able to keep the room decipol level below 1.4 with occupant loads up to 29 persons. Maximum air flow to the room was 925 m^3/h. Based on the olf emission from building materials, the flow controller determined a minimum air flow rate of 160 m^3/h.

5 Simulation results

The system was simulated for different occupant loads over a time period of six hours, using the RK45 numerical integration method with a simulation time step of 2 seconds. Figure 3 shows the occupant loads during the simulation. The actual load present in the room is represented by the dashed, stairs shaped line. The other line shown is the estimated number of occupants. As one might notice, the size of the inertia of the room model is important for the estimation to become accurate. If the occupant load was varying rapidly over time, the estimated load would probably not reach its steady state value, hence giving an inaccurate estimate of the actual load. In such cases, however, the actual decipol level

would not reach its steady state level either. For this reason, a poor load estimate due to large room inertia is not critical. Because of the low pass filter effect of the occupant load estimation, the controlled state amplitudes get more attenuated. One should have in mind though, that the system being simulated is dynamic and that the control algorithm is based on steady state formulas.
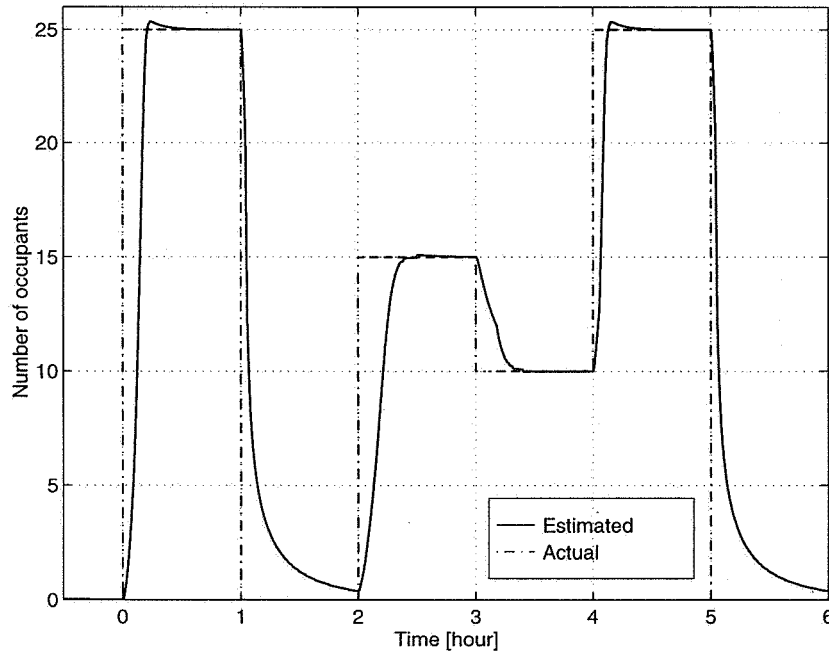
Figure 3: Actual and estimated occupant load in the room.

Now, focus on the controlled decipol level in the room (shown in figure 4). The control limits has been fixed to 1.4 and 1.2 decipol, upper and lower respectively. As 25 persons enter the room at time 0 hour, the room air quality deteriorates quickly. At a level of 1.4 decipol ($G_{r,\max}$), the controller begins to work, and the room decipol level is forced below the upper control limit. Most of the time between 0 and 1 hours, the room level stays between the control limits, and hence no control action is taken. At time 1 hour all the occupants leave the room. The room air quality improves very fast, down to a decipol value of 1.2 (the lower control limit, $G_{r,\min}$). When the lower limit is reached, the fan speed is reduced, forcing the air quality to stay around 1.2 decipol. The increasing, oscillating amplitude of the response just above the lower control limit stabilizes eventually between the control limits (a simulation period of one hour is to short to visualize this). As the simulation proceeds, the occupants enter and leave the room as shown in figure 3 and 4. The controlled state is kept within its limits, hence giving the desired room air quality.

In figure 5, the corresponding CO_2 responses from the simulation is shown. Starting with a background level of 350 ppm at time 0 hours, the concentration is rising rapidly as 25 persons enter the room. The maximum concentration during the simulation is below 900 ppm. If no occupants are present in the room, the CO_2 concentration drops towards the background level. The supply air flow rate, controlled by the decipol level in the room, is at this point 160 m^3/h.
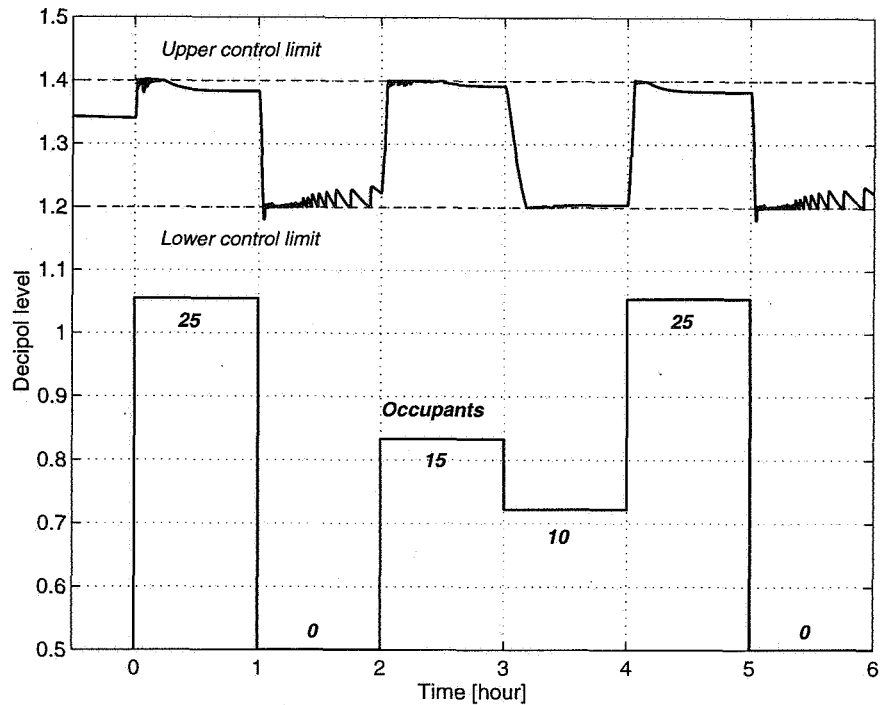
Figure 4: Simulated decipol level in the room. Upper and lower control limits are 1.4 and 1.2 decipol respectively.

6 Conclusions

A new control algorithm which enables predictive control of the air quality in a room has been presented. The control algorithm is based on measurements of the CO_2 level and ventilation air flow rate, giving an estimate of the occupant load in the room. The occupant load is then used to determine the air quality (represented by the decipol level) in the room. The algorithm is able to account for all odour sources in the room, for instance emissions from building materials. This ensures that the controller will provide a minimum ventilation air flow rate to the room at any time. However, the mean activity level of the occupants, the expected percentage of smokers (optional) and the emissions from non human sources have to be given as parameters to the algorithm. The predicted decipol level in the room is then compared to a given set point level, imposing the air flow to change properly. The following events should also be handled by the controller:

- Decreasing building odour emission rates over time

- Scheduling internal odour emitting processes

- Variations due to external sources

Simulations has been used to outline the controller performance, and the results show that the decipol level in the room can be kept within specified control limits, providing the desired room air quality. At this time, no efforts for a practical implementation of the algorithm have been conducted. Although the algorithm is functioning well for the ideal simulation case, this may not be the truth in reality. Some factors which may affect the performance are variable air quality background level, rapidly changes to any disturbance,
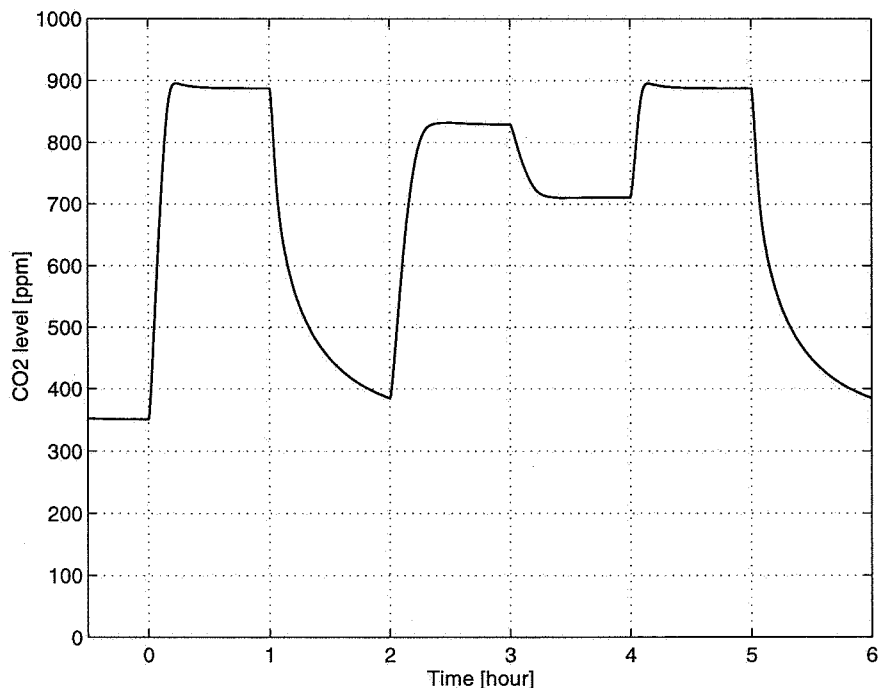
Figure 5: Simulated CO$_2$ response

placement of sensors, air flow patterns in the room, infiltration, recirculation and so on. The promising performance of the algorithm, shown by the simulations, has to be investigated further in practice.

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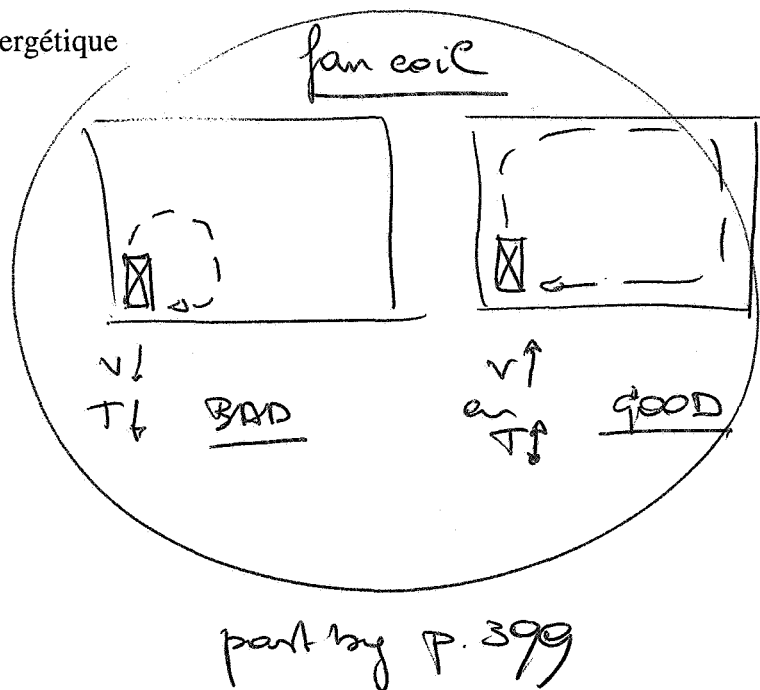
Introduction of Air Infiltration and Ventilation in a Simple Modelling for Energy Consumption Estimation in Air Conditionned Buildings

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Introduction of Air Infiltration and Ventilation in a Simple Modelling for Energy Consumption Estimation in Air Conditionned Buildings

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Synopsis- This study reports on the introduction of air infiltration and mechanical ventilation in a model for energy consumption estimation. The model applies to air conditionned non residential building and is developped to need few inputs. Existing air infiltration models are compared and three equivalent leakage area (ELA) databases are tested on the same case study. Calculations of air input throught opened-doors are made to compare flows due to air infiltration and due to natural ventilation. Simulations are made considering mean air infiltration value and hourly values. It appears that impact of air infiltration variation is no negligible in winter (+11% compared with mean air infiltration value).

Then concerning mechanical ventilation, this study reports on taking into consideration « fresh air flow » in the model. The first solution is to introduce mechanical outdoor air flow rate in building loads. The second solution is to calculate building loads without outdoor air flow rate and to introduce it in mixing box calculation. Both solutions are not equivalent concerning energy consumption (difference +16% for particular system used).

Keywords- air infiltration, mechanical ventilation, energy consumption, modelling

List of Symbols

A	area	m^2	n	flow exponent		
B	permeability coefficient	$\dfrac{m^3}{s.m^2.Pa^n}$	N_p	number of openings per hour	h^{-1}	
C	flow coefficient	$\dfrac{m^3}{s.Pa^n}$	P	pressure	Pa	
c_D	head loss coefficient		Q	air flow	m^3/s	
c_p	wind pressure coefficient		$\dot{Q}$	energy rate	W	
e	exposure coefficient to wind and stack effect	Pa^{-n}	T	temperature	K	
ELA	equivalent leakage area	m^2	W	wind speed	m/s	
K_i	empirical coefficient		z_o	opening door mean time	s	
n	air change rate due to infiltration	vol/h	ρ	density	kg/m^3	

subscripts: a : ambient   o : outside   area : unit per m^2

1. Introduction

Fresh air introduced in building is partly controlled (mechanical and natural ventilation) and partly uncontrolled (air infiltration). Natural ventilation comes through opened windows, doors and other intentional apertures. Mechanical ventilation uses fan and intake and/or exhaust vents specifically designed and installed for ventilation. Infiltration is uncontrolled air flow through cracks, interstices, etc [ASHRAE, 1993].

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Knowing and mastering air fresh flows rates are important for two main reasons:
- fresh air is used to improve indoor air quality and to dilute indoor air contaminants,
- energy used to handle fresh air is a significant load.

The magnitude of these air flow rates should be known at maximum load to properly size the plant. Minimum air exchange rates should be known to assure proper control of indoor contaminant levels. How to simply evaluate energy consumption due to air exchanges ? A simple model for estimating air inflitration in a building has been introduced into a model for estimating energy consumption of air conditionned buildings. Main aspect of this model is to need few inputs in such a way that it could be used very soon in the design process, when materials and building caracteristics are not fixed in details.

Existing models of air infiltration are compared and tested on the same simplified case study, a simple model is chosen. Then, two mechanical ventilation model are compared, first with mechanical air flow rate introduced in building load, second in system calculation level.

2. EXISTING MODELS IN AIR INFILTRATION

Infiltration is driven by wind, temperature difference creating stack effect and/or appliance-induced pressures across the building. Several methods detailed or simplified models are found in the literature.

2.1 AICVF's Guideline

AICVF[1] Guideline for heating [AICVF, 1989] considers contribution of wind and stack effect for winter. The pressure differences due to driving forces are considered in combination by adding them together and determining the airflow rate through each opening due to this total pressure difference. The flow equation is simplified choosing a sizing approach in: $Q = A.B.0,63.\Delta P^{2/3}$ considering $e = 0,63.\Delta P^{2/3}$, exposure coefficient to wind and stack effect. On the one hand, those formulas give data calculated with e_m (maximal value of exposure coefficient in winter) considering different types of buildings. On the other, the guideline gives permeability coefficient values established from CSTB[2] in situ experiments for different types of windows, doors, walls, etc.

2.2 AIVC Methods

AIVC guideline [AIVC, 1996] and technical note [AIVC, 1994] notice that it is very difficult to identify and quantify all airflows, driving forces , size and location of each opening. Simplifications lead to different models.

2.2.1 empirical model:
Few models establish correlation to determine air change rate due to infiltration. The simplest correlation is a linear formula [Etheridge and Sandberg, 1996]: $n = K_1 + K_2 |T_O - T_A| + K_3 W$

Type 19 of TRNSYS 14.2 gives values of coefficients depending on quality of building envelope construction. [TRNSYS, 1996]

2.2.2 estimation from building air-tightness
From sets of measurements, Kronvall suggests an approximate estimate of air infiltration which could be inferred from building air-tightness data. [AIVC, 1996]. This method concerns energy analysis and air infiltration impact estimation. This method is based on building air-

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[1] Association des Ingénieurs Chauffage Ventilation Froid: Refrigerating Ventilation Heating Engineers Ass.
[2] Centre Scientifique et Technique du Bâtiment: Technical and Scientific Center for Building

tightness mesure determined for a reference pressure of 50 Pa. This value is divided by an empirical coefficient representing forces influence. Simplified tables [AIVC, 1994] propose for six characteristical types of construction a basical value of ACH(50Pa).

2.2.3 simplified theorical model

This model incorporates the effects of air-tighness and natural and mechanical driving forces. The model has been developed at the Lawrence and Berkeley Laboratory [Sherman, 1980]. Equation are based on Equivalent Leakeage Area under 4 Pa (ELA_4). Flows due to the wind and stack effect are calculated independently. Equivalent leakeage area under 4 Pa can be extracted from tables building components as proposed in [ASHRAE, 1993].

2.2.4 zonal method

AIVC Guideline suggest a method in 5 steps [AIVC, 1996]:

1- calculation of flow throught the opening using detailed tables [AIVC, 1994] for several components;

2- development of a flow network identifying each source of air in building envelope (monozone model) or between each zone. Infiltration and natural ventilation are separated before combining the two networks.

3- pressure forces evaluation. [AIVC, 1994] suggests values for an evaluation of C_p (wind pressure coefficient depending on the building shape and wind direction)

4- mechanical ventilation determination (fixed at a constant rate)

5- air mass flow balance (volumic balance). Equation of the balance between inflow and outflow of air is solved.

2.3 Choice for the method of energy consumption

Main models evaluation criterias are summed up in the following chart:

	AICVF	empirical model	air-tightness estimation	theorical model	zonal method
splitting airflows into zones	yes				yes
no pressure calculations		yes	yes		
few parameters		yes	yes	yes	
parameterization		existing tables	easy		
introduction in the method	medium	easy	medium	medium	hard
precision	sizing	medium	low	good	very good

The field of energy consumption model development is non-residential buildings, in which air exchange depends more on mechanical ventilation than it does on envelope performance. So a medium precision to evaluate air infiltration is enough. Considering our aim, it is excluded to incorporate a detailed calculation of airflows. Consequently empirical model, easier to introduce in the existing method, is chosen.

The method must split airflows into different zones for a multizone building, and the chosen empirical model can not do that. Three configurations exist:

- Outlet-flow is fixed by mechanical ventilation: it is suggested to characterize each zone by a pseudo-ELA (sum of artificial and unvolontary openings). Inlet flows are shared proportionnaly to ELAs.

- Inlet flow is fixed by mechanical ventilation: outlet flows are shared proportionnaly to ELAs.

- Inlet flow and outlet flow are fixed zone by zone by mechanical ventilation. The sum of the flows for the building enable how to calculate air infiltration flow throught air-tightness default and specific openings. This global flow is divided proportionnaly according to the ELAs. If flows are balanced, there is no infiltration.

2.4 Test of ELA database on a same simplified case study

In order to determine repartition of air flow rates between different zones, each zone is characterized by an equivalent opening area (pseudo-ELA: adding of air openings and tightness fault). Three permeability or ELA databases (AICVF, ASHRAE and AIVC) are tested on the same case study. It is a monozone department store described on fig 1.
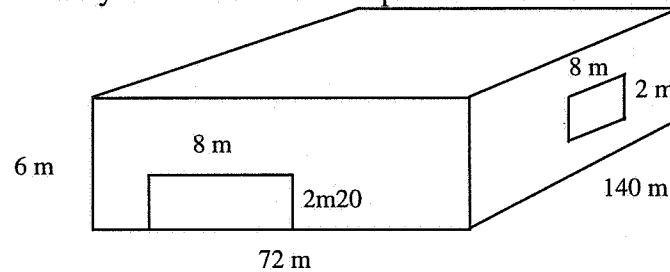

fig 1- Department store

The components chosen to describe building air-tightness are: ceiling, floor, wall, ceiling to wall joint, wall to floor joint, closed door, windows. In the 3 databases, the most important components concerning leakeage are ceiling and floor (85% of total area, source of 90% of total leakeage).

2.4.1 air infiltration

	AICVF	ASHRAE	AIVC
database reference	[AICVF, 1989]	[ASHRAE,1993]	[AIVC, 1994]
flow formula	$Q = B.A_e.\Delta P^n$	$Q = ELA.c_D.\sqrt{\dfrac{2\Delta P}{\rho}}$	$Q = C.\Delta P^n$
main hypothesis[3]	$\Delta P = 1 Pa$ applied uniformly	$\Delta P = 4 Pa$ $c_D = 1$	ΔP applied uniformly $n = 0,66$
air flow rate splitting key	$BA_e = 11,8 \dfrac{m^3}{s.Pa^n}$	$ELA = 4,5 \ m^2$	$C = 3 \dfrac{m^3}{s.Pa^n}$

2.4.2 natural ventilation

In building, natural ventilation is limited to opening doors. It is considered that windows are closed for an air conditionned building. AICVF's guideline suggest a formula that has been adapted to both other methods.

AICVF	AICVF adapted to ASHRAE	AICVF adapted to AIVC
$Q = 0,59 \ A \ \Delta P^{0,5} \ [m^3/s]$	identification with $\Delta P = 4$ Pa	identification with $n = 0,5$
$B_{area} = 0,59 \dfrac{m^3}{s.m^2 Pa^{0,5}}$ $BA_e = 0,59.z_0.A_e \dfrac{N_p}{3600}$	$ELA_{area} = 0,45 \dfrac{m^2}{m^2}$ $ELA = ELA_{area}.A_e.z_0.\dfrac{N_p}{3600}$	$C_{area} = 0,59 \dfrac{m^3}{s.m^2 Pa^{0,5}}$ $C = C_{area}.A_e.z_0.\dfrac{N_p}{3600}$

---

[3] The hypothesis of uniform pressure difference is correct when mechanical ventilation is preponderant.

For the department store, two scenarios are considered:
- 750 customers per hour by group of 6 persons, N_p = 125 openings per hour.
- 1500 customers per hour by group of 6 persons. N_p : 250 openings per hour.
Opening door mean time is fixed z_0 = 8 s.

	scenario low	scenario high
AIVCF BA_e [$\dfrac{m^3}{s.Pa^{0,5}}$]	2,83	5,70
ASHRAE ELA [m^2]	2,2	4,4
AIVC C [$\dfrac{m^3}{s.Pa^{0,5}}$]	2,88	5,77

2.4.3 Comparaison of results

Airflow due to a 4 Pa pressure difference have been calculated with the 3 methods in order to homogenize results. So the three methods could be compared.

m^3/h	AICVF	ASHRAE	AIVC
$Q_{infiltration}$	105730 m^3/h 1,7 vol/h	41 830 m^3/h 0,7 vol/h	26510 m^3/h 0,44 vol/h
$Q_{natural\ vent}^4$ min	20 400 m^3/h 0,34 vol/h	20 450 m^3/h 0,34 vol/h	20740 m^3/h 0,34 vol/h
max	41 060 m^3/h 0.67 vol/h	40 900 m^3/h 0,68 vol/h	41540 m^3/h 0,68 vol/h

For infiltrations, AICVF method gives value 4 times more important than AIVC method, but AICVF database is used to size installation. It is probably not adapted to a energy consumption model. Considering the simplicity of case of study, ASHRAE and AIVC results have the same range of values. The two databases are equivalent. We choose to extract from one of those two databases a simplified components database.

2.5 Estimation of air infiltration on HVAC energy consumption

2.5.1 Building, system and models

The store HVAC system is composed of 12 roof-top units described in the following table:

	cooling			heating			fans	
	nominal values			nominal values			cooling	heating
	$\dot{Q}_{cooling}$	$\dot{Q}_{compressor}$	COP	$\dot{Q}_{heating}$	$\dot{Q}_{heating\ coil}$	COP	$\dot{Q}$	$\dot{Q}$
12 units	1224 kW	452 kW	2,71	1056 kW	348 kW	3,03	109 kW	61 kW
1 units	102 kW	38 kW		88 kW	29 kW		9 kW	5 kW

set point temperatures		
occupation cooling	occup. heating	non occ. heating
23°C	17°C	12°C

Air flow rate in HVAC system are the same in cooling and heating modes: 216 000 m^3/h. Rooftop units operate with constant air flow. Schedule of occupancy is the following:

	Mon-Tue-Thu	Wednesday	Friday	Sathurday	Sunday
occupation 9-19h	1309 per/hour	1527 per/hour	1636 per/hour	1909 per/hour	0

---

[4] The natural ventilation flow rates are same for the 3 methods because the 3 methods use the same expression.

Mechanical ventilation is calculated from mean number of customers with 30 m³/h per person in the store and fixed to 0,75 vol/h. Meteorological datas are extracted from SRY file (2 caracteristic weeks per season) [Lund, 1985]. The simulations are made for first week of winter and first week of summer. The chosen location is Carpentras in the South of France.

Supply-air-conditions are a result of supply-air-flow-rate and thermal and hydric loads of the building, which are calculated using COMFIE as a building simulation model [Peuportier, 1992]. As roof-top air conditionners are running with a constant air flow, the cooling energy rate and the compressor electrical power are calculated, taking into consideration the fresh air flow rate, the return air flow rate, the supplied conditions at each calculation step. Air handling cycles in psychrometric chart, and finally energy consumption of roof-tops including auxiliaries (fans) are calculated, as described in [Morisot et al, 1997].
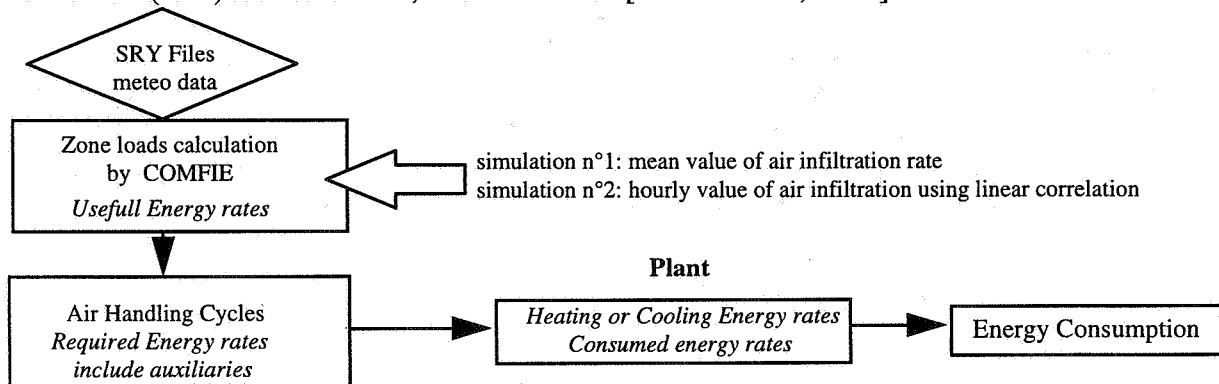

fig 2-flow chart of calculation

2.5.2 Simulations

Fresh air is introduced in the roof-top units. In simulation n°1, average value of air infiltration is used and in simulation n°2 hourly values are used (calculated using TRNSYS linear expression depending on temperature difference and wind). Air infiltrations are added to the loads of the building. Figure 3 shows evolution of air infiltration for the first week of summer and winter. Air infiltration rate is expressed in volume of the store per hour. Average value of air infiltration is calculated considering the 2 weeks of datas and its value is 0,44 vol/h.
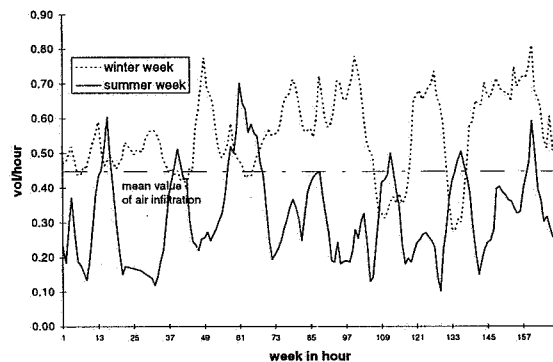

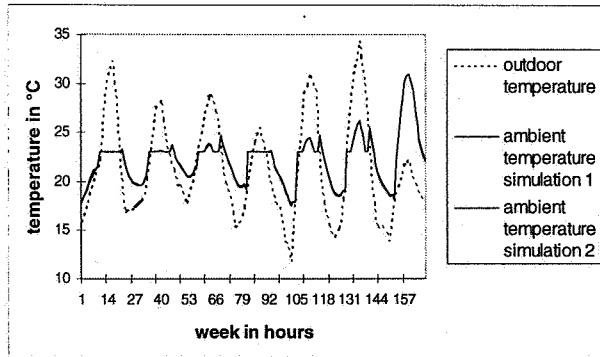fig 3- Air Infiltration Rate for two week of SRY files

fig 4- Outdoor and ambient temperature for
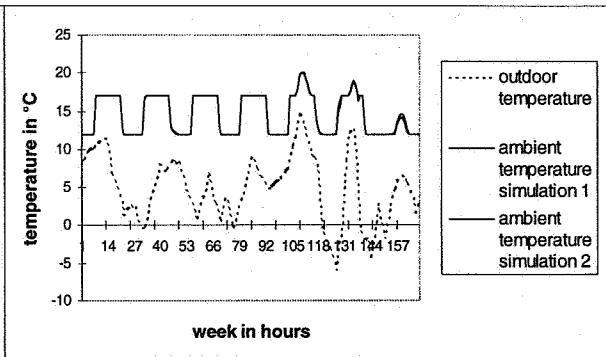the summer week in simulation 1 and 2

fig 5- Outdoor and ambient temperature for
the winter week in simulation 1 and 2

The ambient temperatures are nearly the same in winter and in summer for simulations 1 and 2. The building thermal response is not affected by using average or hourly values. Consumption is summed up in the following chart. 168 hours are simulated, 66 hours of occupancy and 102 hours of non occupancy. The chart gives the number of hours when the system is in the dead band. During cooling season, HVAC System is off in non occupancy periods.

	$\dot{Q}$ cooling	$\dot{Q}$ heating	
	hours in temp dead band in occup	hours in temp dead band in occup	hours in temp dead band in non occup
simulation 1 mean air infiltration	32 650 kWh 1 hour/66 hours	42 000 kWh 14 hours/66 hours	19 hours/102 hours
simulation 2 variable air infiltration	32 700 kWh 1 hour/66 hours	47 170 kWh 12 hours/66 hours	17 hours/102 hours

Difference between the two simulations is unsignificant concerning cooling season. On the contrary, the difference for the heating season is 11%. Two reasons explain that. Firstly, infiltration directly depends on absolute temperature difference between ambient and outdoor. This difference is 3,6°C in summer and 9,7°C in winter: so it is 3 time bigger in winter than in summer and it induces the same impact on air infiltration flow. In addition to that, consumption introduced by air infiltration is more important in winter considering absolute temperature difference between ambient and outdoor.

It appears to be necessary to take into account air infiltration variation for a consumption simulation during heating season. Conversely, using a constant air infiltration rate seems to be valid in cooling season considering its moderate impact on energy consumption but only on one condition: determine a correct mean value of air infiltration.

3. MECHANICAL VENTILATION

The aim is to answer this question: what is the most accurate way to take into consideration « fresh air flow » in the model? The first solution is to introduce mechanical outdoor air flow rate in building loads. This solution is adequate to take into account building behaviour inside temperature dead band but drives the control of fresh air quantity difficult. The second solution is to calculate building loads without outdoor air flow rate and to introduce it in mixing box calculation. This allows fresh air control easier but drives errors for evolution of the building in temperature dead band.

Simulations are made on the store already described for the two weeks of SRY files.

3.1 Simulation

Introducing mechanical ventilation in the mixing box is compared to introduce it in building loads.
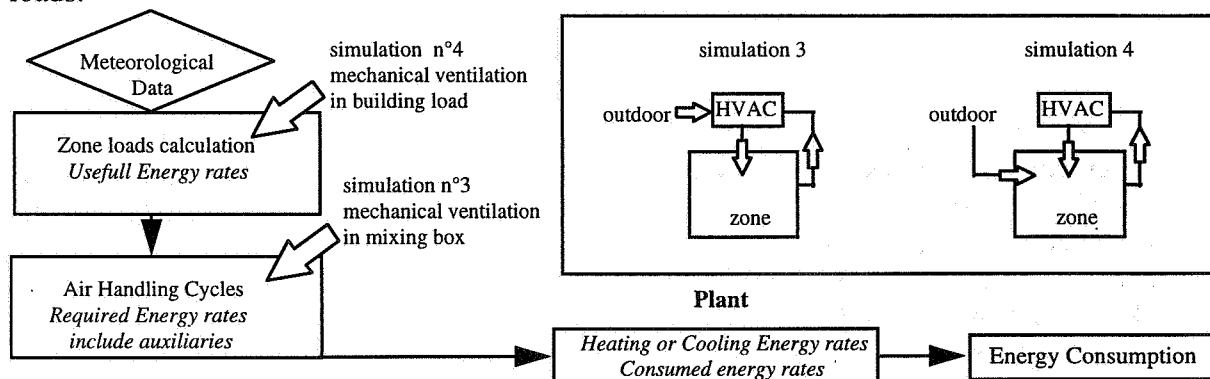

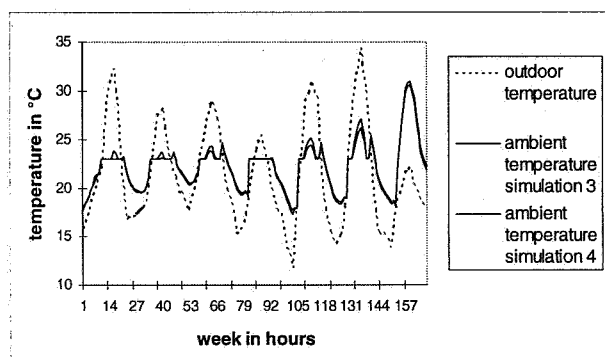fig 6- Two possible models for mechanical ventilation

3.2 Results

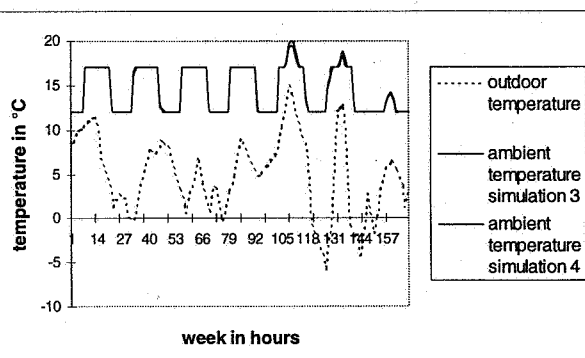| fig 7- Outdoor and ambiant temperature for the summer week in simulation 3 and 4 | fig 8- Outdoor and ambiant temperature for the winter week in simulation 3 and 4. |

	Q cooling	Q heating	
	hours in temp dead band in occup	hours in temp dead band in occup	hours in temp dead band in non occup
simulation 3 ventilation in system	32 700 kWh 1 hour/66 hours	47 170 kWh 12 hours/66 hours	17 hours/102 hours
simulation 4 ventilation in load	31 870 kWh 0 hour/66 hours	39 650 kWh 10 hours/66 hours	17 hours/102 hours

The building thermal response are nearly the same for the two models. On the other hand, consumption between simulation n°3 and 4 decreases significantly in winter (-16%) and decreases less in summer (-3%). The difference appears during the period when building temperature is in the temperature dead band. When ventilation is introduced in roof-top units, fresh air is supplied at ambient conditions as a « neutral air ». Indeed, constant air flow roof-top units chosen do not include a fresh air control. In our particular case, system never uses outdoor air as a cooling or heating source. That explains the artificial consumption increase when fresh air is introduced in the mixing box. Differences are clearly evident on figure 8. They correspond to three sequences of overheating (2 in occupancy and 1 in unoccupancy).

One can check that fresh air needs around 300 kW to become neutral (near 10000 kWh during the period).

However, considering fresh air as a building load prevents from simulating a fresh air control. In the particular case (no fresh air control and overheating in summer), model of simulation 4 must be used.

4. CONCLUSIONS

This study relates to the introduction of air infiltration and mechanical ventilation in a model for energy consumption estimation. A simplified tool of air infiltration evaluation is chosen because of its facilty to use and its medium precision. Two ELA or equivalent databases can be used to determine an air flow rate splitting key. Simulations have shown it is necessary to use hourly value of infiltration instead of mean value.

Models of mechanical ventilation have been compared. A simulation with particular system shows that both solutions are not equivalent for energy consumption (16% difference).

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VENTILATION AND COOLING

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Air distribution in an office building
as measured with a passive tracer gas technique

by

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only during occupation

Begrip : "average mean age of air"

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Synopsis

A passive tracer gas technique - the homogenous emission technique was utilised for measuring the air distribution in a part of an office building with displacement ventilation. Measurements were made during one winter period and one summer period. During the winter period the ventilation was run continuously, while on/off regulation was used during the summer period. The result from the winter measurement shows that the displacement effect was satisfactory but less pronounced close to the window-side of the office.

The paper discusses the effect of on/off ventilation strategy on the requirement of the measurement technique and the implications of such strategy on air quality. It is shown that the homogeneous emission tracer gas technique yields the true average mean age of air, also during transient conditions, but that special precautions must be taken when using intermittent sampling of the air.

1. Introduction

In most mechanically ventilated commercial buildings, the ventilation is varied depending on the occupancy. Usually ventilation is run in only two modes - on and off. During occupancy the ventilation is on, while it is off during nights and weekends. This type of operation has certain consequences on how to perform and interpret a ventilation measurement experiment and also affects the air quality during occupancy hours.

In the present paper, we present the result of ventilation measurements during two periods in the same office building. During the first measurement period, the ventilation was run continuously, while during the second period, ventilation was running only from 5 a.m. to 6 p.m. at weekdays.

In the first part of this paper, we discuss some general implication of varying flow rate on the measurement technique. In the second part we present the measurement and discuss the result in view of this general implications. Lastly, we discuss the effect of varying ventilation flow rate from an air quality and health aspect.

<u>1.1 Measurement technique and varying flow rate.</u>

The technique used in this work is a variant of a passive tracer gas technique called the *homogeneous emission technique*. This technique has been discussed in several recent papers (Stymne *et al.* 1992, Stymne 1994, Stymne and Boman 1994,1996). In essence, small passive tracer gas sources emitting a perfluorocarbon tracer (PFT) at a constant rate are distributed in a building in such a way that the emission is approximately homogeneous, which means that the emission rate per volume unit (S) is constant throughout the building space.

If the tracer emission is homogeneous, it has been shown (Sandberg and Sjöberg 1994, Stymne *et al.* 1994) that the local steady state concentration of tracer gas (C_p) will be directly proportional to the local mean age of air (τ_p), and that a measurement of local concentrations thus will yield estimates of the local mean ages of air and their distribution in space according to equation 1.

$$\overline{\tau}_p = \frac{C_p}{S} \qquad (1)$$

One advantage of using the homogeneous emission technique is that the local mean age of air is a good air quality indicator. Many continuously emitting contaminant sources can be considered more or less homogeneously distributed in the building space and the local contaminant concentrations will therefore be proportional to the local mean age of air.

The measurement of the local tracer concentration is performed by use of passive diffusive samplers, distributed in the building. The passive samplers sample the air during an extended time and thus yield time integrated values of the concentrations.

A problem may arise when the ventilation rate and thus the mean age of air changes during the measurement period. The steady state assumed in eq. 1 will in such a case not be valid all the time.

However, the validity of eq. 1 is not limited to the steady state case. If there is a homogeneous emission of tracer, the instantaneous local concentration will always be proportional to the local mean age of air, even during transient conditions, for example such as appear when the ventilation rate or flow patterns change from one value to another. No formal proof of this statement will be made here, only a rather intuitive indication of the correctness of this statement, when the tracer emission is truly homogeneous (a similar picture was presented by Sandberg and Sjöberg (1983)).

The air around a point in space can be considered as composed of air entities equal in volume having spent different times in the building i. e. having different ages. The ensemble of air entities around the point have a certain distribution of ages, with a mean value corresponding to the local mean age of air. As the air entities find their way from the inlet to the point in question, they travel through regions each emitting a constant amount of tracer per time unit and volume unit. As the air entities having equal volume, they will carry with them an amount of tracer, which is proportional to the time they have spent indoors. Thus the volume around a point can also be considered as an ensemble of air entities having different tracer concentrations. As the concentration of each entity is proportional to its age, the mean concentration of air around the point must be proportional to the mean age of air at that point. This reasoning is independent of whether a steady state is reached or not. The only suggestion made is that the tracer emitted from the sources is carried away with the air passing the tracer source. This is a reasonable suggestion, because there exists no other carrier for the tracer than the air.

Thus, as the tracer concentration is proportional to the instantaneous local mean age of air in a homogeneous emission experiment, integration of the local concentration over time yields a measure of the average of the mean age over that time. This is an important conclusion because it means that one can perform an integrating sampling during times when people are . present and get a fair estimate of the air quality, which the occupants are exposed for. If a measurement of the mean age of air instead is performed with tracer release only during the occupancy time, this could result in a misleading value, because very old air remaining from a low night ventilation rate remains a considerable time during daytime, increasing the average mean age of air well above the steady state value of daytime ventilation.

It is difficult to exactly calculate the transients of mean ages when a ventilation system changes between two flow rates in the general case of incomplete mixing. However, if the local mean ages do not differ very much from zone to zone, it is reasonable to make the approximation that after a step change in ventilation, the concentrations approach the eventual

new steady state values (C_{2e}) exponentially with a time constant close to the new steady state mean age of air (τ_2) according to eq. 2.

$$C - C_{2e} = (C_1 - C_{2e})e^{-\frac{t}{\tau_2}} \tag{2}$$

where C_1 is the concentration at the moment when the ventilation was changed (t=0) and C_{2e} is the eventual new steady state concentration.

Using this approximation it is possible to analytically calculate the time dependency of the tracer concentration. In figure 1 the computed time history under the assumption of a cyclic change between two different ventilation states are displayed.
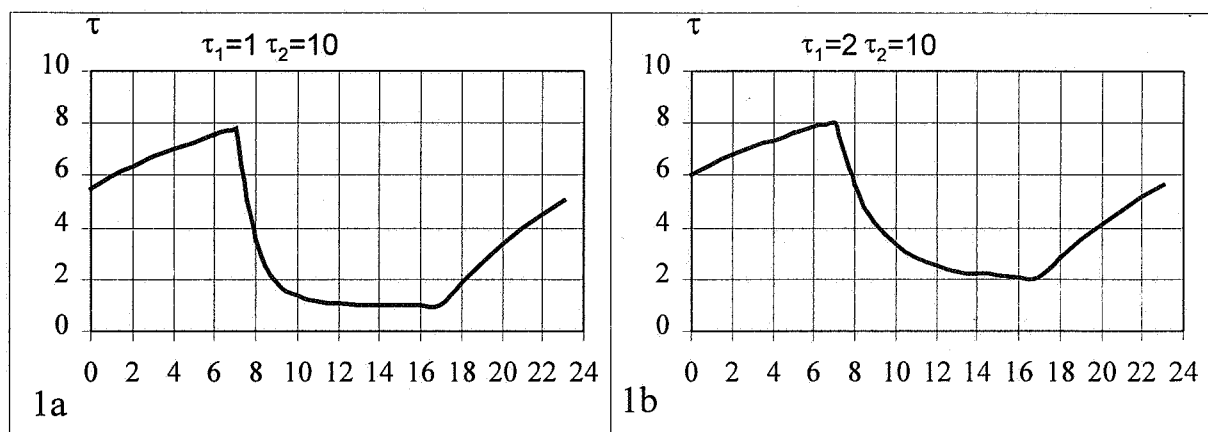
Figure 1. Display of the variation of the mean age of air, when the ventilation is cyclically changed between two modes, characterised of steady state mean ages of air τ_1 and τ_2 respectively. Mode 1 run between 7 a.m. and 17 p.m. and mode 2 during the rest of the day.

2. Description of the building and ventilation system

The investigated object is a part of an office building. The building was recently converted from an old naturally ventilated factory space into a modern office space with mechanical ventilation.

One part of the office building including 20 office modules (7 m^2 each) and 180 m^2 open area was investigated. The office modules consists of "boxes" along the walls of the space, leaving a corridor of approximately 3 m width between the two rows of office modules. The boxes, which are 2.10 m in height and open upwards have glazed walls with glass slide doors towards the corridor. The lower part of the glass walls are equipped with a steel mesh to allow air flow from the corridor into the boxes. The slide doors are normally left open by the occupants.

Ventilation air is admitted with under-temperature to the corridor with low velocity air supply devices. The return air is withdrawn at a few points close to the ceiling. It was suspected that the ventilation air may not easily penetrate to the office modules.

The ventilation is normally run only between 5 a.m. to 6 p.m. on weekdays. All other times the ventilation is off.

3. Experimental layout

3.1 Experiment 1

This experiment was performed during one week of cold winter conditions. During this experiment, the ventilation was run continuously at the same rate, day and night and weekend. This was made in order to be able to use integrating sampling during an extended period at approximately the same conditions which is valid during work hours.

Approximately 70 passive tracer gas sources were evenly distributed in the space to approximate a homogeneous emission. Approximately 50 samplers were distributed along the corridor and in some of the office modules at different heights in order to measure the local mean ages of air at different points. Samplers were also positioned close to air extract points.

3.2 Experiment 2

The second experiment was performed in the same object during summer conditions. The ventilation was run in its normal mode, which means that the ventilation is operating only on 13 hours each weekday.

The tracer gas sources were distributed in approximately the same patterns as in the first experiment. However, the number of sources were tripled to yield a higher emission rate in the space.

Approximately the same numbers and positions of passive samplers were also distributed in the space as in experiment 1. These samplers were left to sample the air continuously during 2 weeks i. e. also during hours with no mechanical ventilation.

However, a few additional diffusive samplers, which were electronically opened and closed were also positioned in the space. The latter samplers were meant to sample air only during 8 work hours during 5 days. As can be seen in the result and discussion sections of this paper, this part of the experiment was not entirely successful. However, the conclusions are valuable and worthwhile to report here.

4. Result

4.1 First experiment

In figure 2 the result from the first experiment is displayed in graphical form on a schematic plan of the office. The bars displayed in the plan, represent the local air change indices, i. e. the ratio between the mean age of the extract air and the local mean age of air. The lower lighter parts of the bars represent a unit air change index, that is the air change rate, which would have been expected if the ventilation had been of fully mixing type. The mean age of air at the extract point was determined to be 0.95 hours, which corresponds to an air change rate of 1.1 ACH.

As can be seen there is a pronounced displacement effect in the office. It is obvious that the displacement type of ventilation functions satisfactory. The computed total ventilation rate ($1203 \text{ m}^3/\text{h}$) is close to the design value ($1100 \text{ m}^3/\text{h}$).

▼ air supply   ▲ air extract at ceiling level
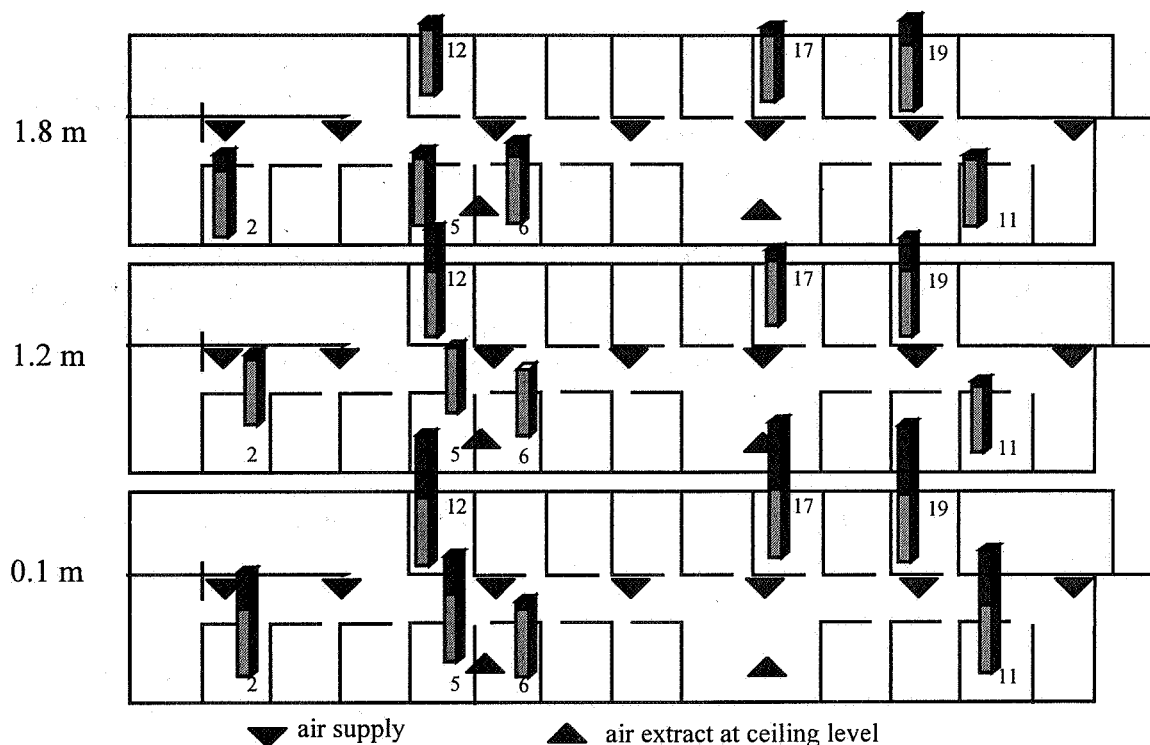
Figure 2. Plan of the office, showing the estimated local air change indices at different heights above the floor. The lower lighter part of the bars represents the air change rate corresponding to the air extract point, while the upper darker part of the bars represent the ventilation in excess of that which would be obtained with mixing ventilation.

4.2 Second experiment

4.2.1 Continuos sampling. During the second period, when the ventilation was run only between 5 a.m. to 6 p.m. on weekdays, integrating sampling during 14 days yielded local mean ages, which varied very little with the positions and height of sampling. Table 1 shows the computed mean ages of air. It is to be observed that the mean ages are computed from the total integrated sampling during the whole period and thus corresponds to an average of the mean age over that time. It is also to be noted that the computed mean age is significantly higher at the extract point than in the occupation zone. However, as there is no mechanical ventilation 61% of the total time the air at the extract point is not representative for air leaving the office.

Table 1. Estimated local mean ages [h] of air -averages over the whole period (337 hours).

	0.1 m	1.2 m	1.8 m	extract
rooms	4.5±0.4	4.8±0.3	4.8±0.4	5.7
corridor	4.7 ±0.3		5.0±0.3	

As can be seen from the small difference between different heights, there is no tendency of air stratification typical for displacement ventilation.

4.2.2 *Programmed sampling.* Programmed sampling was also performed at a few points in the office. The microprocessor controlled closing device was programmed to open the sampling tube at 8 a.m. and close it at 4 p.m. during 5 workdays. The result from the programmed sampler showed an average mean age of 1.5 hours.

5. Discussion.

5.1 First experiment.

The result (figure 2) shows that there is a marked tendency of stratified flow, typical for displacement type of ventilation. However, the different rooms are not equally well ventilated. Rooms no. 12, 17 and 19 on one side of the corridor are considerably better ventilated at breathing height (1.2 m) than rooms on the other side of the corridor (rooms no. 2, 5, 6 and 11). The latter rooms show mean ages close to that which can be expected with mixing ventilation. A plausible explanation is that these latter rooms are located along the window wall, while the former rooms are arranged along an internal wall. The cold window surfaces will create a mixing convection on that side of the office.

The total air change rate (total volume divided with the mean age of air at the extract point) is 1.1 ACH, which is close to the design value, but must be considered to be rather low for an office.

5.2 Second experiment

5.2.1 *Continuous sampling.* As discussed in the introductory section integrating sampling in a homogeneous emission tracer gas experiment will yield an average of the local mean age of air during the sampling time. In the present case the ventilation was operating at its design flow rate 13 hours each weekday (totally 130 hours) and was off rest of the time (207 hours). In figure 3 the computed variation of mean age of air during the exposure time is displayed, assuming 1 ACH during the operation hours and 0.1 ACH during the non-operation hours. A further assumption is the exponential approach towards steady state conditions as discussed in the introduction.
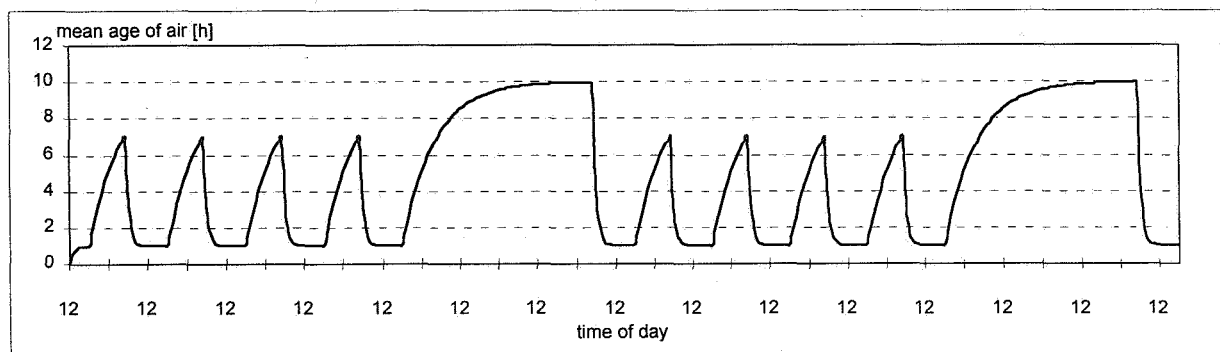

Figure 3. Computed time variation of the mean age of air assuming 1 ACH during operation (5 a.m. to 6 p.m. weekdays) and 0.1 ACH during off periods.

The average of the mean ages displayed in figure 3 is 4.7 hours, close to the value estimated from the integrating samplers in experiment 2. The value 1 ACH for the work hour ventilation rate was chosen from the result of experiment 1. However, the same result for the average mean age can be obtained for other combinations of on/off ACHs. Figure 4 displays the possible combinations of on/off ACHs yielding the same average
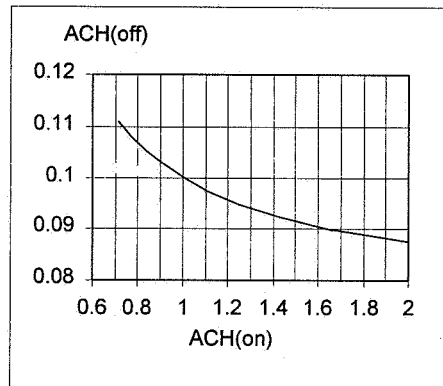

Figure 4. Relationship between ACHs with mechanically ventilation off and on respectively, in order to achieve an average mean age of air of 4.7 hours during a fortnight (13 hours on-mode each weekday).

It is noted from figure 4 that the result is very insensitive to variations in ACH(on), but very sensitive to variations in ACH(off). Whatever value for ACH(on) we choose in the interval 0.6 to 2 the ACH(on) must be around 0.1 in order to yield the measured integrated value. Thus we can conclude from this experiment that the air change rate during *operation-off* hours is approximately 0.1 [h^{-1}] corresponding to a steady state mean age of air of 10 [h]. However, we can not even guess the ventilation during *operation* hours from continuous integration data. It is necessary to have other supplementary information. Such supplementary data can for example be obtained from integrated sampling only during work-hours.

The aim of this experiment was also to test such sampling, the result of which is discussed under next heading.

5.2.2 *Programmed sampling*. The device for the electronic open/close mechanism for the passive sampler was programmed to open the sampler tube from 8 a.m. to 4 p.m. during 5 workdays (totally 40 hours). The result showed an average age of air during this time of 1.5 hours.

Assuming the air change rates 0.1 [h^{-1}] and 1 [h^{-1}] for non-operation and operation hours respectively, we can calculate the theoretical average of the mean age of air during the opening time of the device. This amounts to 1.02 hours. Thus the result is not very encouraging. The discrepancy could of course be explained if the ventilation rate is lower during daytime, than we assumed (0.67 ACH instead of 1 ACH). However, such a low ventilation rate is not very likely. Instead, we suspect a leakage through the sampler sealing during the closed periods. The concentration is high during these periods, and time in closed conditions is long compared to that in opened conditions.

A simple calculation (figure 5) shows the effect which a small leakage may have on the computed mean age. The leakage is defined as the air sampling rate of a closed tube in relation to that of an open tube.
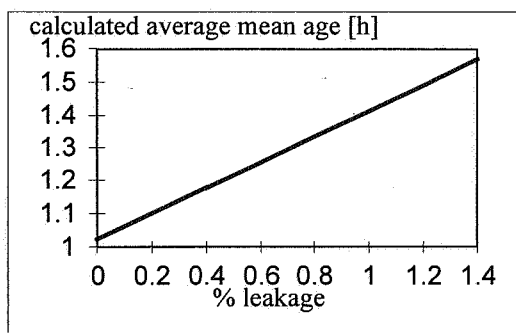
Figure 5. Effect of small leakage in the sealing of the sampler tube during closed conditions.

It is obvious that even small leaks during the closed condition of the programmed sampler tube have a deleterious effect on the result during the extreme conditions (high ACH(on)/ACH(off) ratio and short open compared to closed time) during this test. The erroneous result in the present test can be explained by a leakage of only 1.2%. Work is in progress to secure a 99.9 secure seal.

5.3 Air quality implications

As can be seen from figure 1, the local mean age of air, will only slowly attain the steady state value after a step change to a higher ventilation rate. The higher the ventilation rate, the quicker is the old air exchanged. Conditions according to figure 1a, will yield a 30 % extra contaminant dose for people present from 8 to 17, while the conditions according to figure 1b, yields a 90% extra dose during the same time, compared to the dose expected at constant ventilation rate.

6. Conclusions

The average local mean age during the occupation time in an office is suitable as an air quality indicator. It is concluded that the homogeneous emission tracer gas technique using passive tracer gas sources and integrating diffusive samplers is a convenient and satisfactory technique for measuring the average local mean ages and their distribution. However when the mechanical ventilation is intermittent or varied, special precautions must be taken when sampling only during occupancy hours. It is very important that the samplers are tightly sealed during non-sampling periods. This is especially true, when there is a large difference in ventilation rate, between sampling and non-sampling periods. Lower demand on tightness is required for example in naturally ventilated buildings, than in tight mechanically ventilated buildings with on/off regulation of ventilation.

On/off regulation of mechanical ventilation increases the mean age of air a considerable time after switching on ventilation in the morning. Examples are given, which may increase the integrated day-dose by 30-90% over that expected with constant ventilation rate.

It is shown that the displacement ventilation yields a satisfactory air stratification in part of the office, also during strong winter climate conditions. However, work stations close to the window wall, shows a mixed ventilation behaviour, probably due to air convection flows along the cold wall.

6. Acknowledgement

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VENTILATION AND COOLING

Part of RADON study

(Title)

Measured Air Exchange Rates at Workplaces Having Different Types of Ventilation

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1 SYNOPSIS

The aim of the study was to investigate the operation of different types of ventilation in places constructed underground and ground level; the effect of ventilation on indoor radon levels was also examined. Air exchange rates and radon concentrations were measured in underground (n=73) and ground level (n=64) workplaces. Air exchange rates, designed exhaust ventilation flows, ventilation rates per person and area were significantly higher in underground places than places constructed on the ground level. Mean of air exchange rate was significantly higher in places having only mechanical exhaust ventilation in the underground places than in the ground level places and indoor radon concentration was slightly higher in ground level, whereas in the places having mechanical exhaust and supply ventilation the mean radon concentration was almost twice higher in underground places. In general, all measures of ventilation were higher in underground places, except the ventilation rate per area against soil. This explains the radon concentration situation. The highest mean radon concentrations were found when air exchange rates were below 3 h^{-1}. Ventilation was mainly effective altough, only 30 % of originally designed ventilation flows were achieved.

2 INTRODUCTION

Finnish guidelines for ventilation are currently under reform. Old guide value for air exchange rate is 0.5 h^{-1} in dwellings /1/. The new proposal includes a range from 0.4 to 0.8 h^{-1} (table 1). These new values are mainly for dwellings, but they can also be used for offices, schools and day-care centers /2/. Good ventilation is a prerequisite for good indoor climate. When ventilation functions well and is in balance, it also prevents radon entry. In the study of Denman /3/, the improvement of the ventilation and sealing the constructions reduced the radon level up to 95 % in British hospital premises. In Belgian schools, effective ventilation together with pressurization and sealing the constructions decreased radon levels significantly (14-98 %) /4/. According to Nazaroff et al. /5/, the indoor radon concentration is expected to increase with ventilation rate if the entry mode is active and the ventilation is unbalanced. Similar findings were discovered by our group /6/ in Finnish homes, where the increase became significant when the pressure difference exceeded 5 Pa. Kim et al. /7/ have studied radon levels in 74 subway stations. They found the radon levels to vary within a wide range up to 677 Bq m^{-3} and they suggested that increasing ventilation may increase negative pressure in

subway stations with mechanical ventilation. In any case, the relationship between indoor radon level and ventilation is a complex that varies considerably with the particular circumstances of a house /5/.

Table 1. The guide values for dwellings, office buildings, schools, and factories.

	Air exchange rate, h^{-1}	Airflow $l.s^{-1}$, person	$l.s^{-1}$, m^2
Dwellings[2	>0.4->0.8	5-8	0.5-1
Office buildings[2			
Offices	>0.4->0.8	8-16	1-2
Training rooms		6-12	3-6
Schools, classrooms[21	>0.4->0.8	6-12	3-6
Factories[1			
Light or middle light work		10	1.5
Garages		4	7

3 MATERIALS AND METHODS

3.1 Workplaces

Totally 137 workrooms were measured in 32 different workplaces. Radon concentrations were measured continuously in 115 workrooms and air exchange rates in 97 workrooms. The places included offices, servicing rooms, schools, and telecommunication centers. The total number of employees in these rooms was almost 400. The room volumes varied from small office rooms of 20 m^3 to large research laboratories of 17 200 m^3. Slightly more than half of workplaces (n=73) were located underground. This also includes some places which were only partly underground. For comparison the study also included workrooms in the ground level, 47 % of the places. The most common type of ventilation was mechanical exhaust and supply (n=94).

3.2 Methods

Air exchange rates (h^{-1}) were measured during working hours by the tracer gas technique and the dilution method using freon-12, or later during the study, difluorodichloromethane as the tracer gas and an infrared spectrophotometer (Miran 1A) as the analyzer. Calculated air

exchange rates (h^{-1}) were calculated by dividing the designed exhaust air rate ($m^3\ h^{-1}$) by volume of the workroom. Ventilation flow rates ($m^3\ h^{-1}$) were calculated by multiplying the measured air exchange rate (h^{-1}) by volume of the workroom. Air flows were calculated by multiplying the measured air exchange rate by volume of the workroom and divided by the number of the persons ($l.s^{-1}$, p) or by the area of the workroom ($l.s^{-1}m^2$). Radon levels (Bq m^{-3}) were analyzed continuously using the Lucas cell method /8/ with a Pylon AB-5 assembly, which includes a detector, a photomultiplier and a system of data collection based on a microprocessor. The output data of the Pylon detector were processed with SP-55 software run on a PC. The flow rate of the pump was 0.4 l min^{-1}. The interval of continuous measurements was 30 minutes (averaged to one hour). Concentrations were measured during periods ranging from two hours to two weeks. Data concerning volumes, depth, working hours, number of employees, and types and operation times of ventilation were collected by questionnairies.

4 RESULTS

Measured air exchange rates distributed log-normally from 0.1 to 13.6 h^{-1} and calculated air exchange rates varied from 0.1 to 31.0 h^{-1} (figure 1). The total mean of air exchange rates in places having mechanical exhaust and combined mechanical exhaust and supply ventilation were 3.1 and 4.3 h^{-1}, respectively. The difference was not statistically significant. Air exchange rates in places having natural ventilation or mere mechanical supply ventilation were 0.1 and 0.4 h^{-1}, and 0.3 and 1.2 h^{-1}, respectively. The highest air exchange rate (13.6 h^{-1}) was measured in a workroom when the door was open to the adjacent corridor. Air exchange rate was 5.8 h^{-1} when door was closed. The calculated ventilation flow rates (calculated of designed ventilation flow) varied from 7 to 55 200 m^3h^{-1}. Calculated air flows per person were high ranging from 2 to 1179 $l.s^{-1}$,p. About 75 % of places exceeded the recommended air flows (4-16 $l.s^{-1}$). Air flows per area varied from 0.05 to 10.7 $l.s^{-1}$ which were mainly within the recommended levels (table 2). The means of measured air exchange rate, designed exhaust ventilation flow, and ventilation rates per person and area were signifigantly higher in underground places than in the places on the ground level. Mean indoor radon concentration was 42 % lower (325 Bq m^{-3}) in 90 workrooms where air exchange rate was more than the recommeded value of 0.4 h^{-1}. Workrooms having air exchange rate below 0.4 h^{-1} mean radon concentration was 559 Bq m^{-3} (n=6). Mean radon concentrations had no statistical difference between the underground and

ground level places. A decreasing air exchange rate and ventilation rates per area against soil seemed to be associated with higher and more variable radon concentrations (figure 2 and 3), especially in places constructed underground or in the hillside. Ventilation rates per area against soil had no statistical difference between the depth of the buildings.
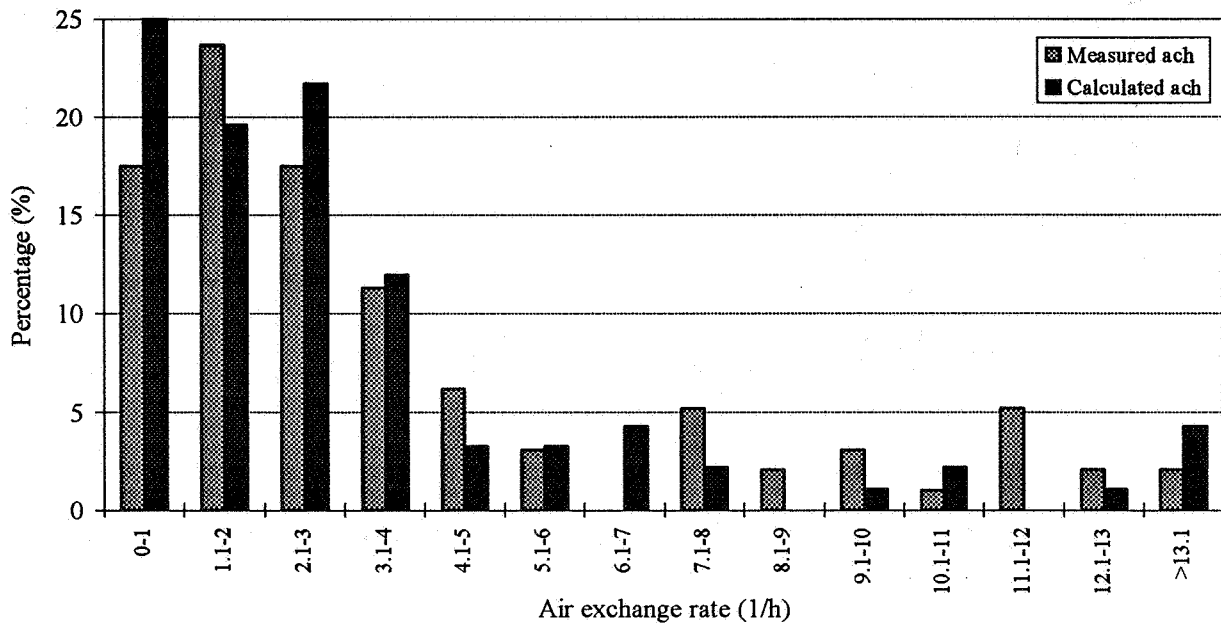

Figure 1. Distribution of the measured air exchange rates (n=97) and calculated air exchange rates (n=92).

Table 2. Measured and calculated air exchange rates (ach), ventilation flow rates, air flows per person, per area the underground (ug) and ground level (g) places and the p-values of varience analysis between underground and ground level places.

| | n | | Mean | | Std dev | | p- |
	g	ug	g	ug	g	ug	values
Measured ach,(1/h)	44	53	2.9	4.7	3.1	3.7	0.0119
Calculated ach,(1/h)	46	46	2.9	4.2	4.1	5.1	0.1909
Calculated ventilation flow, (m³/h)	41	44	279	2688	399	8692	0.0801
Designed exhaust ventilation flow, (m³/h)	47	47	283	2054	350	3419	0.0006
Ventilation rate per area against soil	38	44	7.5	6.2	8.3	3.8	0.4310
Ventilation rate, per person (l/s,p)	39	31	33	142	56	242	0.0084
Ventilation rate, per area (l/s,m²)	41	42	2.4	4.1	2.4	3.0	0.0038
Radon concentration (Bq/m³)	53	62	282	329	457	509	0.6086
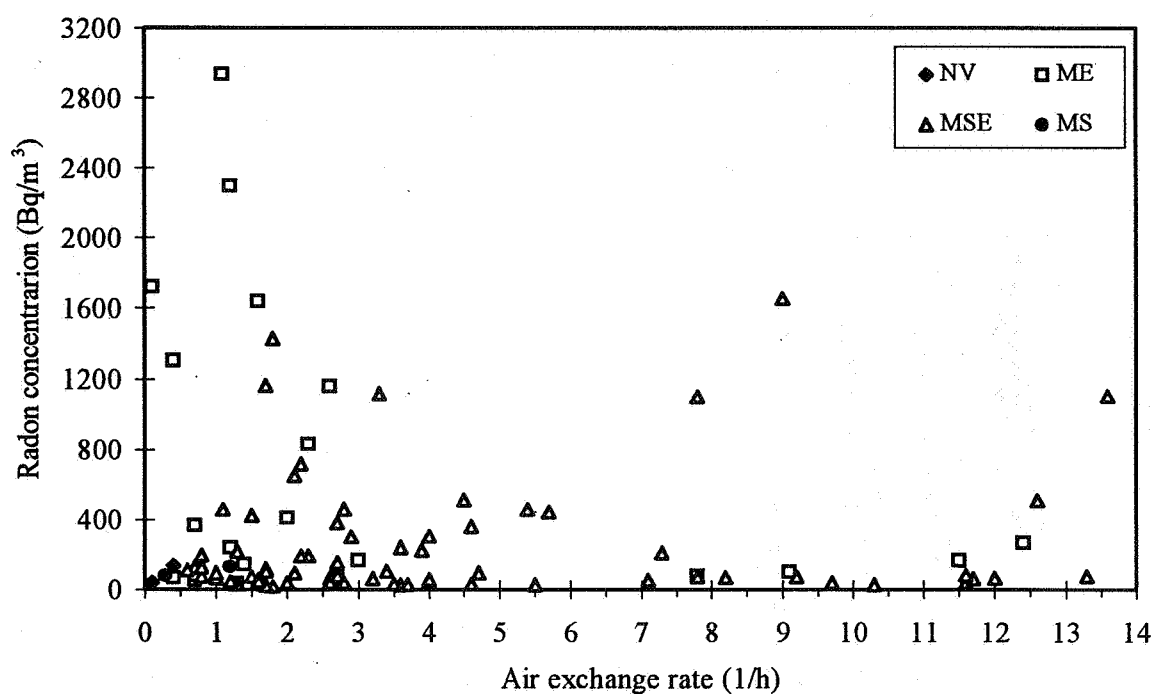
Figure 2. Radon concentrations (Bq m^{-3}) versus measured air exchange rates (h^{-1}) during working hours (n=91).
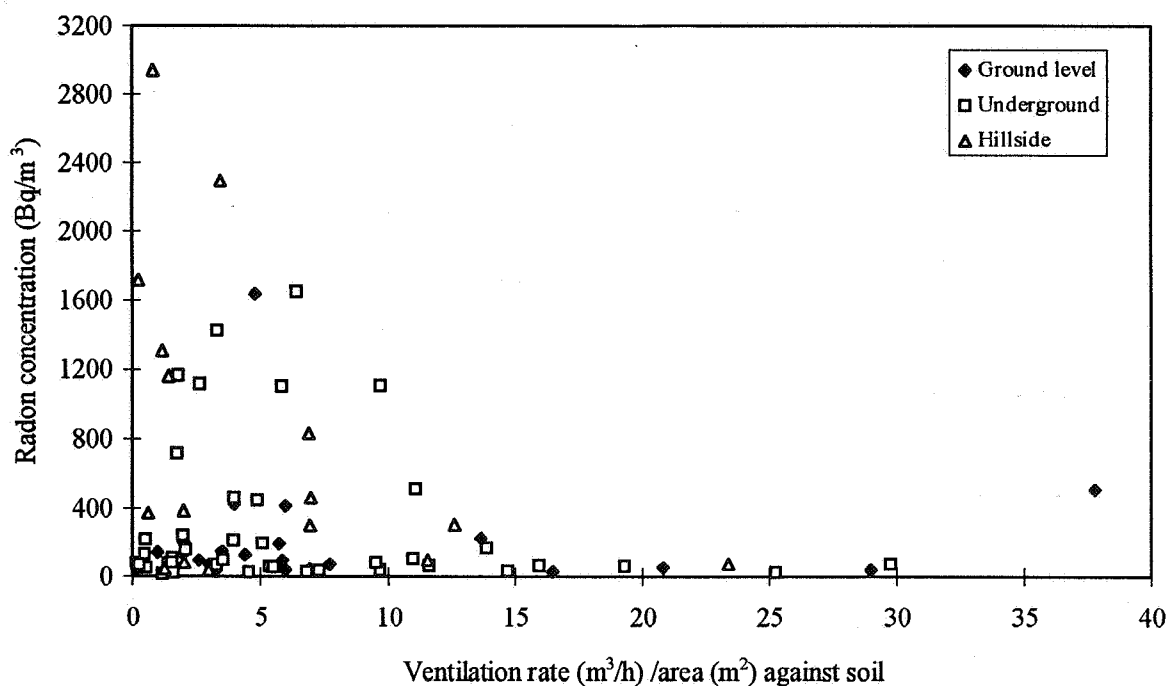

Figure 3. Radon concentretions (Bq m^{-3}) during working hours versus ventilation rate (m^3 h^{-1}) per area (m^2) against soil (n=81).

The mean air exchange rate in underground workrooms (4.7 h^{-1}) was almost twice compared to the rate in ground level (2.9 h^{-1}) workrooms. Air exchange rates were significantly different at different depths (p=0.0001), but not linearly (the linearity p=0.1523, R^2=0.015, variance analysis). The highest mean of radon concentration was measured in ground level places having mechanical exhaust ventilation. Mean air exchange rate was the lowest in these places. Air exchange rate was significantly higher in underground places having mechanical exhaust than in ground level. In places having mechanical exhaust and supply ventilation, radon concentration was significanlty higher in underground places than in ground level. The probable reason is the surrounding soil which is the main source of radon. The mean air exchange rate was at the same level in places having mechanical exhaust and supply ventilation in underground and ground level places (table 3). Ventilation was adjusted to operate during working hours with its full efficiency in 66 % of the places. During evening, nights or weekends the ventilation was not operating in 24 % of the places. Only in about 30 % of the workrooms, designed ventilation flows were achieved. Designed exhaust ventilation flows were not significantly different between the groups of underground and ground level.

Table 3. Mean of measured air exchange rates (Ach) and mean of radon concentrations (C$_{Rn}$) during working hours with mechanical exhaust (ME) and mechanical exhaust and supply (MSE) ventilation and the difference between them in ground level and underground workrooms.

Type of ventilation	n		Mean		Std		p-value	
	C$_{Rn}$	Ach	C$_{Rn}$	Ach	C$_{Rn}$	Ach	C$_{Rn}$	Ach
ME, ground level	15	13	644	1.4	734	0.9	0.8376	0.0070
ME, underground	9	8	572	5.7	952	5.0		
MSE, ground level	34	29	144	3.7	130	3.5	0.0345	0.2448
MSE, underground	49	43	299	4.7	408	3.5		

5 DISCUSSION

Mechanical ventilation is generally used in workplaces. Air exchange rates observed for mechanical exhaust and combined mechanical exhaust and supply ventilation were not significantly different. Measured air exchange rates varied largely during working hours. The highest air exchange rates were due to opened doors which was typical situation during workdays. Ventilation was generally in the recommended level in the underground places. More non-compliance was found in the ground level places. Ventilation rates per person were high.

Indoor radon concentrations were considerable high in some places. The highest concentrations were measured in places having only mechanical exhaust ventilation and constructed on the ground level. Some of these places were construced in the hillside, and part of their walls was against to ground. There were also places constructed in the hillside which were mainly underground and also grouped as an underground place. High radon levels were observed when the ventilation rate per area against soil was low. Effective ventilation seems to decreace indoor radon concentrations particularly in underground places and places constructed in the hillside.

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VENTILATION AND COOLING

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ADAPTATION OF A FAN COIL UNIT TO OPERATING CONDITIONS FOR OPTIMUM COMFORT

DANIEL MARCHAL, PIERRE BARLES

Commercial use

p 369

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ADAPTATION OF A FAN COIL UNIT TO OPERATING CONDITIONS FOR OPTIMUM COMFORT

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SYNOPSIS

The work discussed here concerns the conditions of comfort obtained in a room cooled by a fan coil in relation to the form of air flow obtained. It is based both on practical experiment and on numerical simulation using CFD code. Combining these methods allowed a large number of configurations to be studied, in association with different operating conditions for the appliance.

Using the results in combination enabled a relation to be established between the problem data, the device characteristics and the comfort conditions obtained. A simple rule was derived from this, which can be used in practice in air-conditioned premises, in order to make the right choice or scaling of the air-conditioning appliance depending upon its conditions of use.

LIST OF SYMBOLS

f : width of the fan-coil unit supply grille (m)
g : gravity acceleration (m/s^2)
h' : vertical distance between blowing plane and ceiling (m)
V : air blowing velocity (m/s)
β : air volume coefficient of thermal expansion (K^{-1})
Δt : temperature difference between intake and blowing (K)
v : cinematic viscosity of air (m^2/s)
Re : Reynolds number
Fr : Froude number

1. INTRODUCTION

To cool inhabited premises that are subject to heat loads, cold-air ventilation is generally used. Cold air is denser than the ambient air, and tends to fall, making it difficult to distribute to places where it is needed. To remedy this difficulty, the property of air jets to adhere to the walls, termed the Coanda effect, is resorted-to. Thus, for a fan coil unit like the one shown in the diagram in figure 1, the cold-air jet from the blower is directed towards the ceiling, to which it adheres right up to the end of the room, which is thus enveloped with cooled air, ensuring a satisfactory degree of comfort.
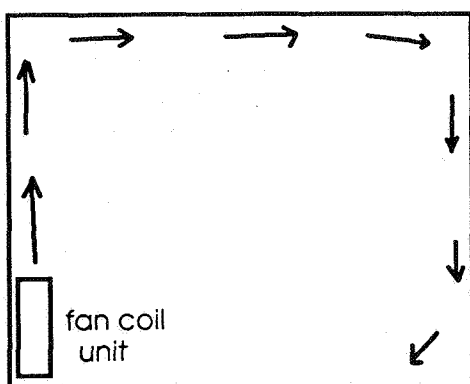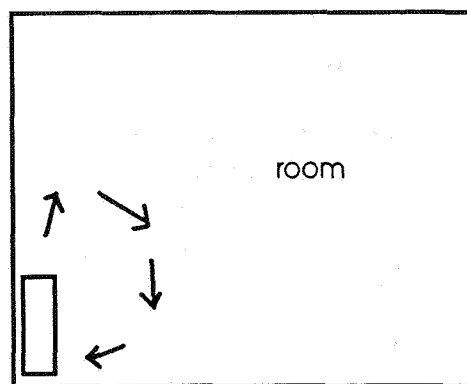
Figure 1

Figure 2

However, for this to operate correctly, certain conditions must be fulfilled, otherwise the flow shown in the diagram in figure 2 would be obtained, resulting in two zones with different conditions, e. g. cold in the vicinity of the appliance and hot elsewhere in the room.

The work discussed here is designed to define the conditions for correctly adapting the fan coil to its use. It is based on both experience and numerical simulation, the latter being used as a means of extending the scope of the experimental results. The study achieves a non-dimensional presentation of the results, which can be used in many situations for practical design of appliances.

2. EXPERIMENTAL STUDY AND COMPARISON WITH SIMULATION RESULTS

Reference [1] indicates the operating method used in a test room with scope for varying the heat loads and the operating conditions of the convector-ventilator installed. Two types of measurement were taken : measurements relating to the flow of the cold-air jet from the device, and the measurement of velocities and temperatures in the occupied area of the room. The latter measurements enabled the usual criteria to be used to describe the comfort of occupants, particularly the ADPI criterion [2], which takes account of both the homogeneousness of temperature and the lack of excessively ventilated areas.

The experimental work covered a number of situations, albeit limited by the dimensions of the test chamber and the air-conditioning appliance chosen. To extend the scope of the findings, we went on to use numerical simulation, performed using CFD code available on the market (*FLUENT*). The principle involved was of numerically integrating the Navier-Stokes equations in the field under study. The use for this purpose of the finite-volumes method requires prior meshing of the three-dimensional space represented. Other factors taken into account are turbulence, using a two-equation model (k-ε), heat transfer and the effects of gravity which entail gravity forces that vary with local air density.

The experimental results enabled the accuracy of the simulations to be verified for a number of cases, as regards both the velocity and temperature profiles in the air jet, and the temperature and velocity fields in the occupied zone, from which is derived the comfort criterion. Thus, the values of the measured and the computed ADPI criterion could be found to be identical, to within one point on an 80-point scale.

3. Using the numerical-simulation findings

Numerical simulation was then used to explore a large number of situations which may arise in practice. All the cases studied concerned empty, oblong rooms, with heat loads applied one the one hand by conduction through the upper part of the building-front against which the fan coil is installed (glazing), and on the other hand to the greatest extent, by internally-generated heat assumed to be uniformly distributed in the occupied zone. The study consisted of systematically varying the parameters listed in table 1, which shows the interval explored, representing some twenty configurations.

Variable (unit)	Minimum value	Maximum value
Room length (m)	3.0	6.0
Room width (m)	2.5	4.5
Room height (m)	2.5	3.5
Heat load (W)	650	1 600
Blower nozzle length (m)	0.5	1.2

In each of these cases, several simulations were performed in which the blowing velocity of the fan coil (assumed to be vertical and uniform) was varied simultaneously with the corresponding temperature, so as to compensate the heat load in order to achieve always the same air-intake temperature (25°C). This exploration of each configuration was so conducted as to highlight the transition between the two types of operation diagrammatically represented in figures 1 and 2.

4. Combining the results

Figure 3, relating to an average configuration (a room with dimensions of 4.5 x 3.5 x 2.8 m; blower nozzle dimensions 0.8 x 0.06 m; heat load 1600 W) shows the behaviour of the ADPI comfort criterion when the blower flow rate and temperature are simultaneously varied, in the manner indicated at the end of the previous paragraph. The value :

$$X = \frac{1}{\mathrm{Re}(f).\mathrm{Fr}(h')}$$

of the abscissa variable is the reciprocal of the product of two non-dimensional numbers :

— the Reynolds number relating to the width f of the supply grille of the device :

$$\mathrm{Re}(f) = \frac{f.V}{\nu}$$

in which V and ν refer to the blowing velocity and the kinematic viscosity of air, respectively ; and

– the Froude number relating to the height h' to which the air jet have to rise :

$$\mathrm{Fr}(h') = \frac{V^2}{h'.g.\beta.\Delta t}$$

where h' is the vertical distance separating the blowing plane of the device from the ceiling, g the acceleration due to gravity, β is the volume coefficient of thermal expansion of air, and Δt the temperature difference between the intake and blowing parts of the appliance.
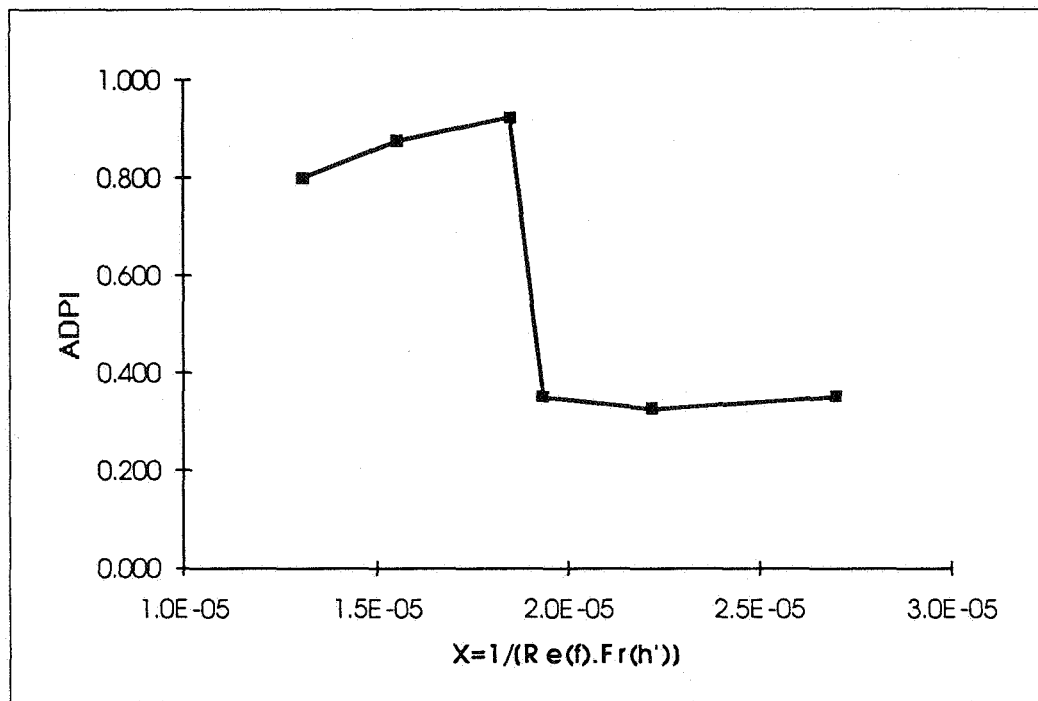
Figure 3

The choice of this expression of X, of which the value increases when the temperature difference is increased at the expense of velocity, is designed to account satisfactorily for the effects of inertia, diffusion and gravity, which involve forces the relationships of which taken in pairs are represented by the Reynolds and Froude numbers.

Figure 3 shows the rapid change in the ADPI criterion around a value for X of approximately $1.9 \ 10^{-5}$, which corresponds to the transition between the two flow situations described in figures 1 and 2. The right-hand part of the curve ($X > 1.9 \ 10^{-5}$) corresponds to the uncomfortable situation (ADPI < 0.4), in which only the immediate vicinity of the appliance is cooled. Conversely, if $X < 1.9 \ 10^{-5}$, a satisfactory degree of comfort is obtained (ADPI > 0.8), with a slight degradation if X is further reduced, on account of the increased air velocities generating local discomfort.

The interest in choosing parameter X, in preference over another non-dimensional expression (e.g. the so-called Archimedes number often used in the literature), can be seen when the geometrical parameters of the problem are varied.
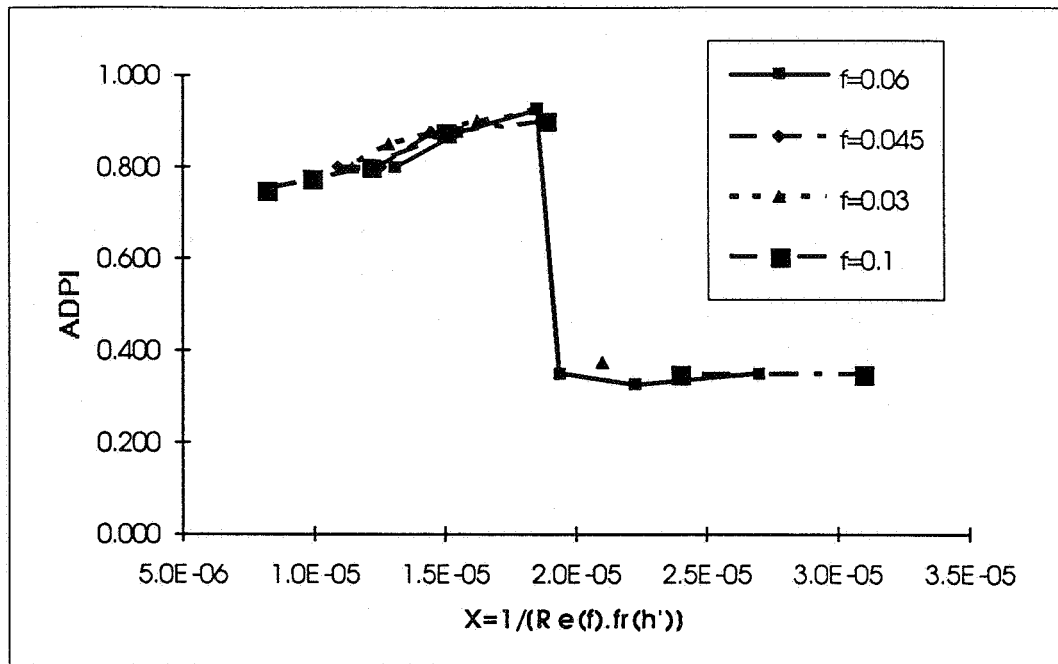
Figure 4

Thus, in figure 4 representing variations in the ADPI criterion, as in the previous figure, but here with different values for the width f of the supply grille, it can be seen that the transition between comfortable and uncomfortable situations takes place at about the same value of X as previously noted.
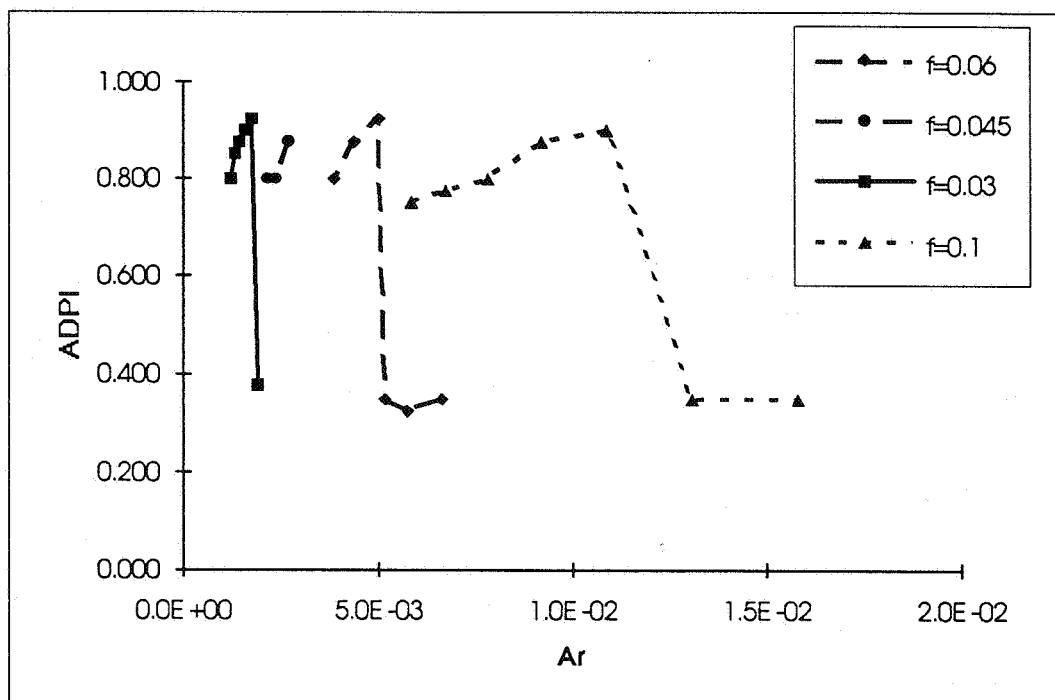

Figure 5

A similar representation using as the abscissa variable the Archimedes number :

$$\mathrm{Ar} = \frac{1}{\mathrm{Fr}(f)} = \frac{f.g.\beta.\Delta t}{V^2}$$

leads to the chart in figure 5, in which it will be noted that the dispersion of the characteristic curves does not allow generalisation. The inability of this number to represent the behaviour in all the situations is probably due to the fact that the Froude number Fr(f) on its own expresses only the relationship of the forces of inertia to the forces of gravity. It cannot therefore also allow for the effects of diffusion, and these are essential to the behaviour of the air jet.

The height h' through which the cold-air jet from the appliance must rise in order to reach the ceiling is also one of the important parameters of the problem. Its effect is correctly described by the number X, as shown in the chart in figure 6 in which it can be seen that, for all the values of h' considered, the transition between zones of discomfort and of comfort occurs for the same value of $X \approx 1.9 \ 10^{-5}$.
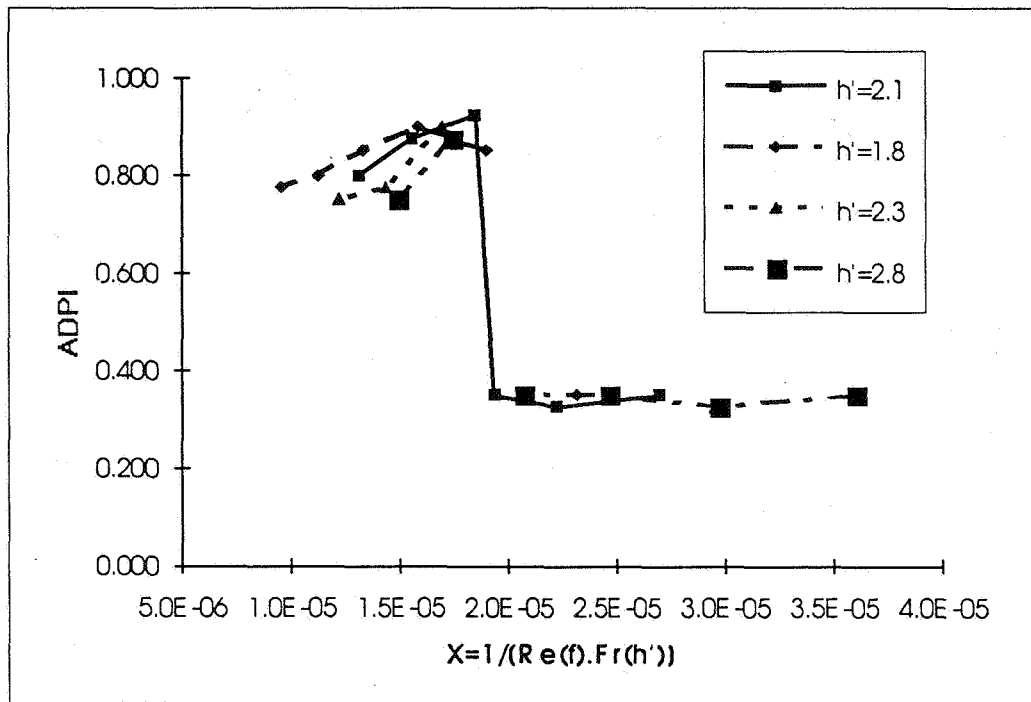

Figure 6

We also studied the effect of the room length, finding that it only slightly affects the value of X at which the transition occurs between discomfort and comfort, as shown in table 2.

Table 2

Room length (m)	Transition value of X
3	$2.4 \cdot 10^{-5}$
4,5	$1.9 \cdot 10^{-5}$
6	$1.8 \cdot 10^{-5}$

The results obtained show that the other parameters listed in table 1 also only exert secondary effects on the change in flow conditions. Accordingly, as the single condition for a satisfactory degree of comfort in terms of the ADPI, the following can be adopted :

$$\mathrm{Re}(f).\mathrm{Fr}(h') > 5.6.10^4$$

For shorter rooms (3 m or less), a lower value ($\approx 4.5 \cdot 10^4$) can be acceptable for this product.

5. CONCLUSION

The study presented leads to a non-dimensional relationship which allows foreknowledge of the flow behaviour caused by a fan coil unit used for air-conditioning an inhabited room, and the resulting comfort rating. This law derived from combining the results can be used in practice for the purposes of real-life projects for air-conditioned premises, in order to make the correct choice or scaling of the air-conditioning appliance, allowing for the conditions of its use.

The reliability of the formulation presented here obviously depends on the assumptions followed, basically regarding the homogeneousness of the air flow from the appliance blower nozzle, and that of the heat loads in the occupied area of the room. Although many ordinary situations do not meet these conditions, these assumptions are necessary for purposes of a general study, otherwise the diversity of cases to be studied would negate any attempt at generalisation. For these reasons, the results presented must nevertheless be applied with caution.

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VENTILATION AND COOLING
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Title: Study of the Ventilation in an Ancient Building Located in the
Centre of Rome and Now Used as a University Office

Author(s):   G. Fasano, G. Giorgiantoni, F. Raponi, V. Leonelli

Affiliation: Department of Energy, Div.Energy Rational Uses ENEA
(The Italian Committee for the New Technologies, Energy and the
Environment

Study of the Ventilation in an Ancient Building Located on the Centre of Rome and now Used as Universitary Office

G.Fasano[1], G. Giorgiantoni[2], F.Raponi[3], V. Leonelli[4]

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SYNOPSIS

An investigation of some Indoor Air Quality (IAQ) parameters in a significant office of the State in the area of Rome was undertaken.The aim, in future, will be to cover a wide range of situations in vaiious buildings and organizations, to achieve data to improve the working conditions, to have a more healthy working environment, to optimize energy consumption and energy management.Analisys of IAQ parameters was performed by field measurements; conditions of the ventilation system were checked by local inspections, code simulations by the multizone model CONTAM96 by NIST (National Insttute of Standard and Technologies USA) were undertaken and compared with the field measures, reproducing the local conditions with data extracted from the technical literature.

LIST OF SYMBOLS

Pa   = Pressure ($1 \ N/m^2$)
ppm   = Concentration (parts per million)

METHODS

The considered office is located in the City of Rome downtown. It is an hystorical building (16th century) and the activity of the organization is devoted to the scientific technical and administrative services of the Institute for Applied Physics of Nuclear Engineering.It is a multistorey building near a heavy all day traffic road and the office is a typical technical office with three employees located in the ground floor of the building.One of the threee smokes regularly and furthermore two personal computers are turned on.The duration of working activity is about eight hours (8-17) with one hour interval during lunchtime.Air handling system (AHU) is not provided in this room.The office door, to comply with office regulations is constantly kept opened.The investigation period is february 1997.
Therefore the main characteristics of the room can be summarized as follows:

Room dimensions:4.30*6.50 h 3.80
door: 1.24*2.10 kept opened
window:class A1,permeability 12.5 m^3/h/m of joint @ 100 Pa (from UNCSAAL Italian Windows Manufacturers Associations)

Outdoor conditions
Since the period is from Feb 3 to 8 the outside temperature varies from 0 to 14 °C.Usual metheorological conditions in this period for this climate lead to clear sky and weak wind velocity (max 5 knots).

These outdoor conditions which include the values for the pressure are available from the metheorological station of Urbe Airport located in the North of Rome approximately 10 km from the related building.

These data are important because they allow to assess the fluidodynamic behaviour of the building, in case of remarkable wind velocities leading to local variations of the pressure, this resulting in changes of the internal-external pressure differences and consequently in different trends of the contaminant migrations.All these data have been used to compile the metheorological *.wth* files used to make the simulations with the model.

Field measurements

The procedure includes two actions :
- information collection on the occupancy behaviour during working time
- measurement of the concentration of CO, CO_2,moisture, VOC.

The sampling points were chosen near the positions of the three employees at a height from the floor which is the most probable during work i.e. 1.10 m.

The adopted istrumentation is a BRUEL & KJEAR analyzer which includes :
- Multi Gas Monitor 1302
- Multipoint Sampler and Doser 1303
- Personal Computer

This instrumentation is based on the principle of photoacustic absorbtion using a photoacustic measurement chamber.The system sensibility is 0.16 ppm for carbon monoxide, 3.2 ppm for carbon dioxide and 50 ppm for water vapour ;the answering time,which includes the suction lines is about 10 s .Before the sampling, the following calibration tests were performed :
- zero calibration,through nitrogen N50
- humidity calibration, through humidified N50 nitrogen
- span calibration through gas mixtures having known concentration
- filters calibration to avoid the humidity interference

The ciclic sampling system operates round the clock alternatively in the selected points.The analyzed concentrations are stored in an ASCII file.The sampling time is from 11 :30 of Dec 4 1996, up to 10 :30 of Dec 6 1996.

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It was decided to undertake code runs to have much confidence with the related phenomena, to achieve a flexible and reproducible procedure to be used in other places and in other circumstances, because it may happen that the possibility of the measuring interval is shorter than the planned one. It is not easy to have a continuous operating analyzer during working time. The instrument produces a repetitive noise during the suction phase and the room is heavily crossed by the sampling pipes.These inconvenients in a productive structure have to be minimized and also occupancy may have obstacles to the normal activity.Then the field measurements might not be so abundant as required. Furthermore, multiple simulations can enlarge the investigation cases, different climatic conditions, different occupancy can be studied and evaluated.

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approach to airflow analysis.The building is treated as a collection of zones connected by airflow paths.These zones may represent groups of rooms,individual rooms,or even portion of rooms,as well as shafts and portion of the building air handling system.Within each zone the temperature and contaminant concentration is considered to be uniform.The airflow paths include doorways, small cracks in the building envelope, and eventually the air handling system (AHU) can be modeled. The main problems to be solved to perform the simulations were:

• Identification and emission rate of the sources

• airflow paths modeling

• Occupancy, windows, vent system timeschedule.

These values were on the literature(2) par.41.6-7 (3).Emissions from tobacco smoking were got from the same source.

Bioeffluents emission

CO_2 (p.m.44) $32*10^6$ µg/h/pp   C1

NH_3 (p.m.17) 15 600 µg/h/pp   C2

CO (p.m. 28) 10 000 µg/h/pp   C3

Tobacco smoke emission

CO_2 360 mg/sigarette   C1

NH_3 5.9 mg/sigarette   C2

CO 43 mg/sigarette   C3

lighting time 5 min/sigarette

max frequency: 2 sigarette/h/pp

Data files were processed through normal spreadsheets.

☐Ambt

* Via Larga

```
                              ⊚ C1
                              ⊚ C2      Win

                              h=3.80

                   doors    Office

          Stairs      GF      Secr              *

                                          Cso Vittorio
                                          Emanuele II
                *
          Hall

          h=4.80          Porter
```
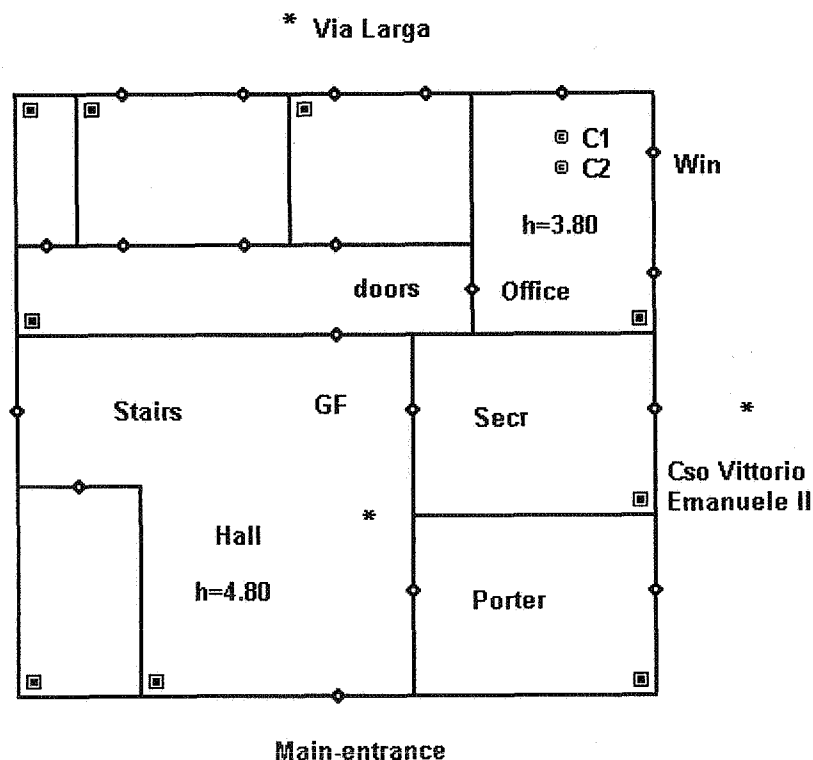
Main-entrance

Figure 1 Office scheme for simulations

RESULTS

The field measurements in the technical office gave the following average readings:

Table 1 Contaminants Concentrations in technical office, field readings

TWA Average Concentration ppm															
External (8-18)				External (18-24)				External (24-8)				Internal Office			
CO	CO_2	VOC	H_2O	CO	CO_2	VOC	H_2O	CO	CO_2	VOC	H_2O	CO	CO_2	VOC	H_2O
4.72	536	3.69	8347	10.57	604	4.57	10170	2.97	546	2.63	7471	5.58	607	4.19	8687

The average concentrations attributed to Dr.Fargione, the smoker are therefore (CO_2 607 ppm=1099 mg/m^3, CO 5.58 ppm= 6.42 mg/m^3 at 23.2 °C int.temp and 101325 Pa int.press).
External readings (CO_2 536 ppm=1045 mg/m^3, CO 4.72 ppm= 5.9 mg/m^3 at 0 °C ext.temp and 101325 Pa ext.press).

The readings in the technical office were taken from 9:00 Feb 3 up to 24:00 Feb 9 1997.
As far as the office is concerned, the simulations with the code gave the following results:

table 3 Simulations results for contaminants concentrations

Avg Concentrations (mg/m^3)			
CO_2 (mg/m^3)	Δ(%)	CO (mg/m^3)	Δ(%)
5356	512	58	983

Regarding the mass exchange, the simulations produced the following output :

Figure 2 Mass exchange

DISCUSSION

The first point to be recognized is that all the concentration readings are well below the limits recommended by **ACGIH** (Industrial Hygienist Association) (TWA CO_2 5000 ppm, CO 25 ppm). The smoker was only one and the sampling point was located near the employee.

As far as the simulations are concerned we can focus our attention to the following points :

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The evaluation of the air permeability would produce the test point which would refine the code input to achieve a better comprehension of the trend of the mass exchanges.

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□Ambt

* Via Larga

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C2
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VENTILATION AND COOLING

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Evaluation of thermal comfort impact of direct fresh air supply in Winter Part 2, Comparison of different ways of air supply to exhaust only ventilation

Takao Sawachi and Seto Hironao

Building Research Institute, Japan

1. SYNOPSIS

IEA Annex 27 "Evaluation and Demonstration of Domestic Ventilation Systems" has been engaged in developing the evaluation tools for various aspects of their performance. This paper describes the evaluation tool for thermal comfort impact by ventilation systems. The tool is based on the experiment by using a room inside artificial climate chamber, focusing upon the temperature difference and cold air supply rate into the room. As the evaluation index, the percentage of living space volume where a specified thermal comfort condition is satisfied is used. Grouping of supply air inlets is made and the score for each group can be obtained by referring tables. This tool is unique and helps practitioners very much knowing how serious or mild thermal comfort impact the ventilation system he/she is examining has.

2. INTRODUCTION

The improvement of the air tightness of building envelope in houses has made it indispensable to design rational ventilation system for houses, regardless of natural or mechanical. Although the technology for building ventilation has its long history and experiences in non residential buildings, it does not look easy to reach a solution which has satisfactory performance validated for residential building. Since man power and money available for maintaining domestic ventilation performance is limited, careful consideration for various aspects of ventilation system performance is necessary in development and design process. Thermal comfort impact of the ventilation systems is one of such aspects which sometimes determines the total performance of the ventilation system. Thermal comfort condition can be sensed sometimes much easier than quality of indoor air, and occupant's complaint about coldness due to the ventilation could make him/her to stop the system in the worst case. Designers of the domestic ventilation systems need to know what type of ventilation system has milder impact on winter indoor thermal condition, and how to minimize thermal discomfort risk for a certain type of system.

3. GROUPING OF AIR INLETS

Outdoor fresh air can be taken into rooms through various types of inlet from the view point of thermal comfort indoors. In heating season, indoor vertical temperature gradient is grown up by cold outdoor air coming into lower part of the room without being sufficiently mixed with room air, as well as by down air stream along indoor surface of outside walls caused by heat exchange. In addition, the air coming into lower part of the room contributes to make air velocity near the floor higher. For the ventilation systems of higher quality, preheating of supplied air is available in order to prevent outdoor air from disturbing indoor thermal condition in winter. In the previous paper (Sawachi T. et al. 1996), it was found that dry bulb temperature and air velocity in the lower part of the room are the critical factors when the thermal comfort impact of the ventilation systems are evaluated. Under the same air supply rate and temperature difference between indoors and outdoors, milder thermal comfort impact can be obtained by the air inlet whose shape and position contributes to better mixing of supplied fresh air and room air before its reaching the lower part of the room. There are four factors characterizing air inlets. The first one is whether air flow path is concentrated or distributed. Generally, the air inlet is a specially installed air flow path and has larger opening area than background leakage area. However, especially in exhaust only ventilation systems for houses more than two stories, additional air supply paths to the background leakage have

Table-1 Grouping of the air inlet

Generic type	flow direction	vertical position	form in Fig.-1
high induction	upward direction flow	high	(a)
	radiant flow	high	(b)
	horizontally straight	high	(c)
low induction	horizontally straight	high	(d)
	horizontally straight	medium	(e)
	horizontally straight	low	(f)
background leakage	horizontally straight	distributed	(g)

negative effect on evenly distributed fresh air supply pattern, due to stack effect. In such case, fresh air intake only through background leakage should be compared with the concentrated air inlet in addition to very small background leakage from the view point of thermal comfort impact. Secondly, the speed of incoming air is related to the extent how well the air is mixed with warmer room air. The difference can be expressed by high and low induction. Thirdly, when the air velocity is rather higher, the direction of air flow determines how much the air can be mixed before its reaching the lower part of the room. The last factor is the vertical position of the inlet. It is directly related to the distance between the air inlet and the lower part of the room.

Taking the above mentioned four factors into consideration, the grouping of the air inlet is made as shown in Table-1. An example of each group is shown in Figure-1. The inlet (a) is an example of the high induction / upward direction flow and has slit-shaped upward openings of which equivalent leakage area is 17.1 cm^2. It is usually installed above the window. The inlet (b) is an example of the high induction / radiant flow and has a round-shaped cap covering the end of cylindrical duct which is designed to make air flow going out radiantly along the wall surface. Its equivalent leakage area is 13.1 cm^2. The inlet (c) is a simple cylinder of which diameter is 50 mm. Even if the end of the cylinder is covered with a grille of which slats is horizontal, such inlet can be grouped into the same category as the inlet (c). The inlet (d), (e) and (f) are identical, but they are installed at different height of a window frame. The inlet is usually installed at the top of the window frame, but the same inlet was used to simulate rather large openings located at the center of the window and that located at the bottom of the window. Those two openings are assumed as two types of window which are opened ajar. The equivalent leakage area is 62.3 cm^2. The last category in Table-1 is the clacks that is background leakage. The inlet (g) is designed to simulate the clacks. It consists of two slits of which height, width and depth are 4 mm, 50 mm and 300 mm, respectively. In the experiment for other inlets, only one piece is used, while in the experiment for the inlet (g) ten pieces was used. A pair of slits has 1.5 cm^2 equivalent leakage area.

## 4. EVALUATION OF THERMAL COMFORT IMPACT BY AIR SUPPLY

4.1. EXPERIMENTAL METHOD

Both air flow rate and temperature of supplied air through different types of inlet are controlled. The pressure of the experimental room (2.6 m × 4.42 m × 2.55 m) is controlled by an exhaust fan to keep the pressure difference across the inlet to give air flow rate 35 m^3/h, 25 m^3/h, 15 m^3/h and 5 m^3/h (10 lit./s, 7 lit./s, 4 lit./s and 1.5 lit./s) as experimental condition. The
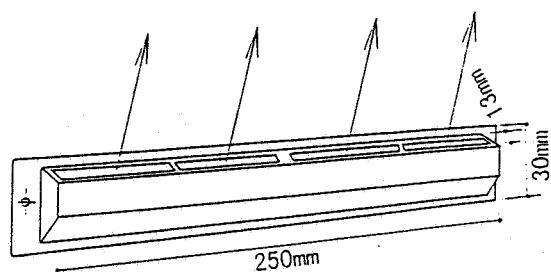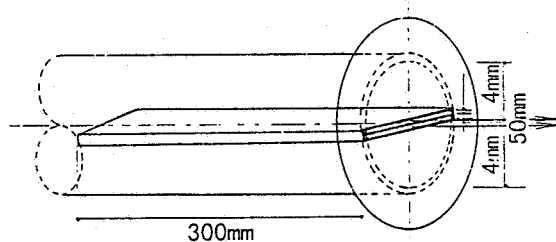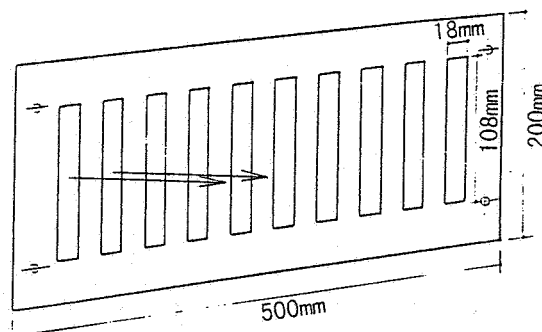
(a) high induction / upward direction flow
H=1,900 mm E.L.A.=17.1 cm^2
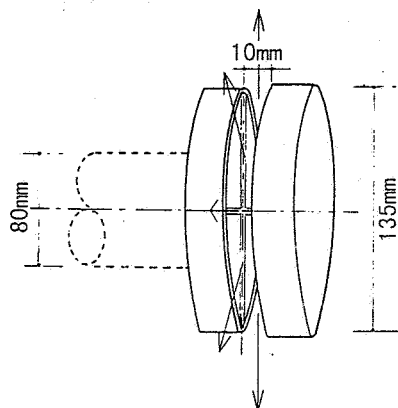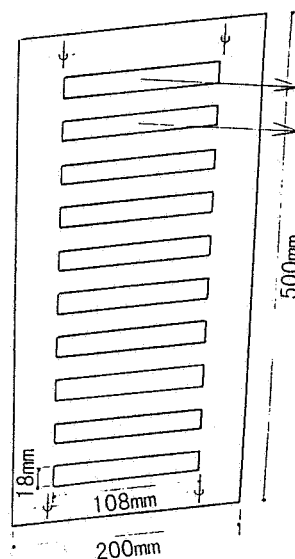
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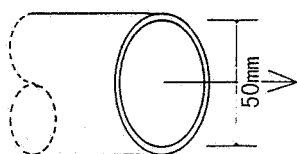
(c) high induction / horizontally straight flow
H=1,900 mm E.L.A.=15.5 cm^2

(g) background leakage simulated by deep slits
H=255 mm ~ 2,295 mm   E.L.A.=1.5 cm^2 × 10

(d) H=1,900 mm

(e) H=1,425 mm
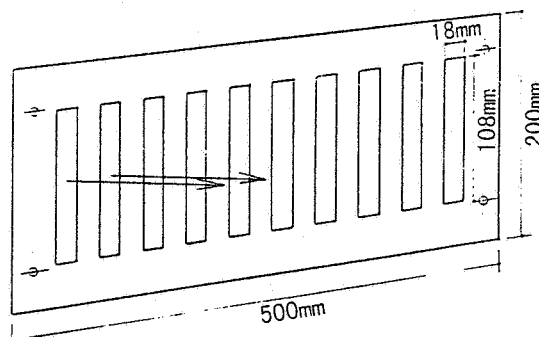
(f) H=950 mm

(d)(e)(f) common low induction inlet
installed at different height
E.L.A.=62.3 cm^2

H: height above the floor   E.L.A.: equivalent leakage area

Figure-1   Forms of the inlets used in the experiment

temperature outside the experimental room is controlled -10°C, 0°C, 10°C, and that inside the room is controlled to keep PMV (Fanger P.O. 1970) equal to zero at the center of the room by using two electric radiant heaters put below windows. Even though thermal comfort condition at the center of the room is kept nearly constant throughout whole series of experiment for different inlets, the distribution of thermal factors varies widely depending upon the route of cold air supplied through the inlets. Dry bulb temperature, globe temperature, air velocity and turbulence intensity are measured at one hundred and twelve points ($7 \times 4 \times 4 = 112$). The lowest position of the measurement is 10 cm above the floor.

4.2. EVALUATION INDEX

Thermal comfort impact by cold air supply can be evaluated by the number of points where the relatively large deviation from comfort condition is observed. As the criteria of the comfort condition, PMV and PD (Fanger P.O. et al. 1988) are used. The PD is a predicted percentage of occupants who feel uncomfortable due to draught, and is expressed as a function of air velocity, turbulence intensity and dry bulb temperature. Among one hundred and twelve points, the points where both conditions for PMV and PD expressed by inequalities (1) and (2) are satisfied are counted, and the percentage of the points is used as an index to evaluate the thermal comfort impact by the ventilation systems.

$$-0.2 < PMV < +0.2 \ (PPD < 7\,\%) \qquad \cdots\cdots \quad (1)$$
$$PD < 15\,\% \qquad\qquad\qquad\quad \cdots\cdots \quad (2)$$

4.3. BASIC DATA FOR THE EVALUATION AND ITS EXTENTION

The percentage of the points where both conditions (1) and (2) are satisfied is calculated for each combination of the air supply rate and the indoor-outdoor temperature difference. One hundred percent means the least impact of ventilation system and that at *all* of the one hundred and twelve points in the experimental room the deviations of PMV from the center and PD values are in the range of inequalities (1) and (2), respectively. On the other hand, smaller percentage means larger impact and extension of the space with difficulty to keep thermal comfort condition. In Figure-2, the contour lines of the percentage are drawn on air supply rate and temperature difference plane for each group of inlet.
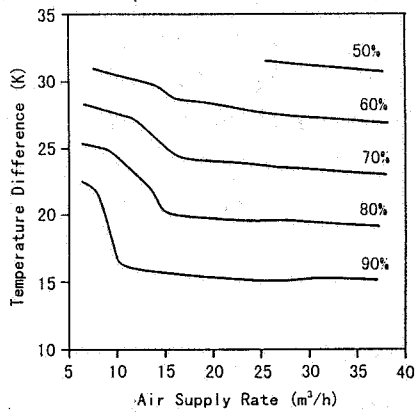
For the high induction / radiant flow inlet (b), the percentage becomes lower in higher air supply rate and larger temperature difference area. On the contrary, for the high induction / upper flow inlet, higher air supply rate does not necessary make the percentage lower, while larger temperature difference steadily makes the percentage lower. Higher air supply rate can contribute to mix supply air with room air and to weaken the cold air impact. It is general that the temperature difference is more influential on the percentage than the air supply rate. Among three directions of high induction inlets, thermal comfort impact by straight air flow is the largest one. For example, at 25 m³/h and 20 K, the percentage for high induction / straight flow inlet is approximately 55%, while the percentage for high induction / upper flow inlet is more than 90%. Comparing three positions of low induction inlet, the highest position (d) gives the least thermal comfort impact. It confirms the rule that the inlet of cold fresh air should be installed at higher position to avoid serious thermal comfort impact. Even when the same amount of air is supplied through distributed small clacks, the percentage is not higher than that for general types of inlet (a), (b) and (d). The rather high impact on thermal comfort by clacks seems to come from air supply through clack at lower position of the wall. In the
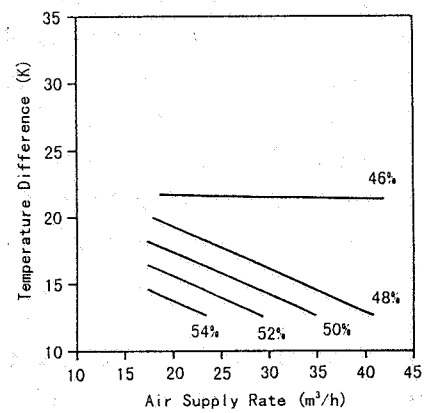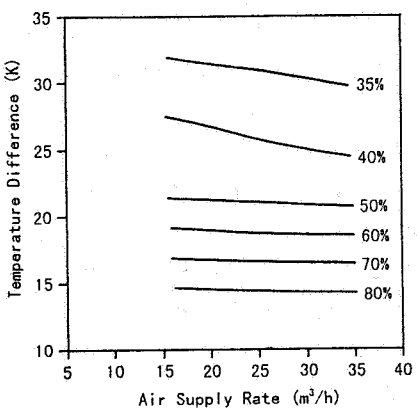
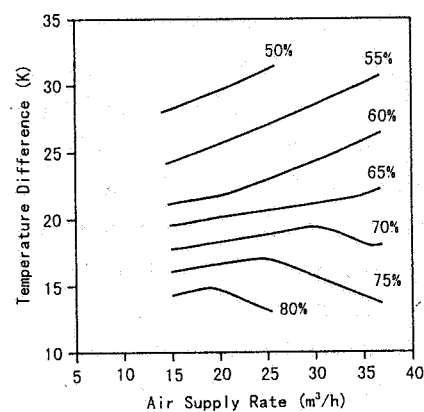(a) high induction / upward direction

(b) high induction / radiant flow

(c) high induction / horizontally straight
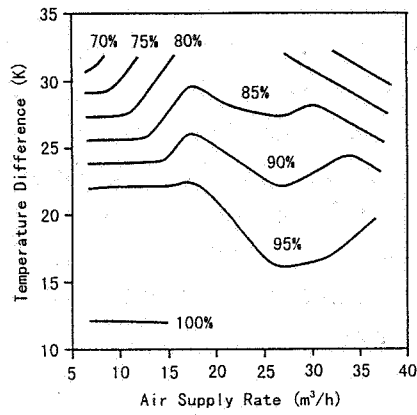
(d) low induction / high position
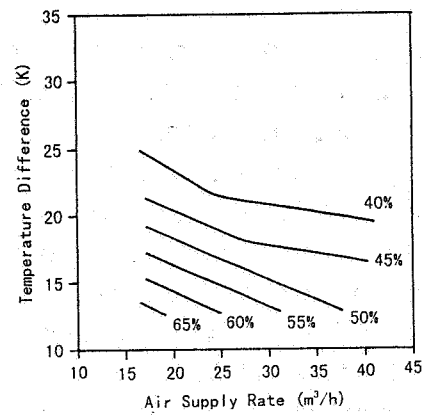
(e) low induction / medium position

(f) low induction / low position

(g) background leakage

Figure-2
Contour maps of the evaluation index drawn on the outdoor/indoor temperature difference and supply rate plane

experiment, ten clack-simulating slits were evenly distributed at different height. It means that a concentrated inlet in place of distributed clacks relieves the cold air impact on thermal comfort condition.

5. A SIMPLIFIED TOOL FOR THE THERMAL COMFORT IMPACT EVALUATION IN IEA ANNEX 27

In Annex 27 "Evaluation and Demonstration of Domestic Ventilation System", a set of simplified tools to evaluate domestic ventilation systems are being developed. Among the tools, a thermal comfort impact evaluation tool is included. As input information, type of ventilation system, outdoor temperature and air supply rate are requested. Indoor temperature is assumed to be 22 °C. The Score is expressed with five ranks from "- -" to "+ +", and is given by Table-2 and Table-3. In the tables, scores for different types of inlet and temperature have been obtained from contour maps of Figure-2.

Table-2 Thermal Comfort Impact of Fresh Air Supply   Q=35m³/h (10 l/s)

Type of Inlet (Vent)		Outdoor Temperature (°C)						
Generic type	Specific type	-15	-10	-5	0	5	10	15
High induction	upward direction flow	-	-	0	+	++	++	++
	radiant flow	--	--	-	-	0	++	++
	horizontally straight flow	--	--	--	--	-	+	++
Low induction	horizontal opening, high position	--	-	-	0	+	+	++
or	vertical opening, middle height	--	--	--	--	--	-	-
Window ajar	horizontal opening, low position	--	--	--	--	--	-	-
Background leakage	equally distributed leakage air flow path on exterior walls	--	-	-	-	-	0	0

Criteria for Scoring the percentage of lattice points satisfying thermal comfort conditions (1) and (2):
100-95% ++   95-85% +   85-75% 0   75-50% -   50-0% --

Table-3 Thermal Comfort Impact of Fresh Air Supply   Q=15m³/h (4 l/s)

Type of Inlet (Vent)		Outdoor Temperature (°C)						
Generic type	Specific type	-15	-10	-5	0	5	10	15
High induction	upward direction flow	-	0	+	+	++	++	++
	radiant flow	--	-	-	0	+	++	++
	horizontally straight flow	--	--	--	--	-	+	++
Low induction	horizontal opening, high position	--	-	-	-	+	++	++
or	vertical opening, middle height	--	--	--	--	-	-	-
Window ajar	horizontal opening, low position	--	--	--	--	-	-	0
Background leakage	equally distributed leakage air flow path on exterior walls	--	--	-	-	-	0	+

Criteria for Scoring the percentage of lattice points satisfying thermal comfort conditions (1) and (2):
100-95% ++   95-85% +   85-75% 0   75-50% -   50-0% --

(Example 1)
If a natural ventilation with a window open ajar as air supply at 1.9 m above floor is used, the air supply rate is 15 m³/h and outdoor temperature is 10 °C, the score is "++". The opening is considered as a low induction type inlet. If the change of supply rate due to wind is not negligible, the increase of turbulence intensity of indoor air flow can make draft risk higher than the evaluation. Therefore, in such case, the score should be carefully interpreted.

(Example 2)

When an exhaust only ventilation system with high induction / radiant flow natural supply vent is used, outdoor temperature is 0 °C and air supply rate is 35 m³/h, the score is "-", which means the percentage of lattice point without difficult to keep thermal comfort is 50-75%. Though thermal comfort situation depends not only upon ventilation system but also upon insulation, heating system etc., the percentage can be considered as an index for relative evaluation. The score, "-" means that there is still possibility for improvement.

6. SUMMARY

For domestic ventilation systems, appropriate air supply technique to avoid thermal comfort impact should be selected. The basic data from the experiment shows a wide variation of the impact by different types of inlet, even if the ventilation system is fixed. The selection of ventilation system also make the difference of thermal comfort impact. The impact can be evaluated to know the type of inlet, supply air rate and outdoor temperature by referring the tables 2 and 3.

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VENTILATION AND COOLING

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Measurements and control of the air motions within a building

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Measurements and Control of the Air Motions within a Building

C. Blomqvist, M. Sandberg

1. SYNOPSIS

There are a number of methods available concerning with distribution of air in buildings. Within control research, one can find new control algorithms which have not yet been used in practice. These new algorithms open the possibility of developing and implementing of new demand controlled ventilation systems.

In a building the internal air motions are due both to differences in temperature and pressure differences caused by the ventilation system. Therefore, one fundamental question is to what extent it's possible to control the air motions within a building using fan powered ventilation in combination with temperature control.

The aims of this paper is to report on measurements done to examine the influence of temperature differences between rooms on the air exchange through open doors in a building and to explore the use of modern control technique to minimise the temperature difference.

The result of the measurements shows that even very small (0.1-0.2°C) temperature differences between rooms cause bi-directional air flows in the doorways of a magnitude that exceed the flow rates caused by the mechanical ventilation system. Therefore it is necessary to control the temperatures in the rooms to make it possible for the ventilation system to distribute the air to those parts of the building where it is needed.

2. LIST OF SYMBOLS

$C(\Delta T)$	Coefficient
$g'\ [m/s^2] = g\Delta T/T$	Reduced gravitation constant
$g\ [m/s^2]$	Gravitation constant
$Gr\ [-] = g'h^3/v^2$	Densiometric Grashof number
$v\ [m^2/s]$	Kinematic viscosity
$Tc\ [K]$	Temperature in cold room
$Tw\ [K]$	Temperature in warm room
$\Delta T\ [K]$	Tw-Tc
$w\ [m]$	Width of door way
$h\ [m]$	Height of door way
$A=hw$	Area of door way

3. INTRODUCTION

3.1 Theory

The flow in a door way can be caused by several mechanisms of which the two most important are:

- Density difference caused by temperature differences, ΔT
- Pressure differences caused by mechanical ventilation

In practice there is a combination of both. A flow driven by density difference, characterised by its reduced gravity g'=gΔT/T, through an aperture with area A=hw and located in a partition wall with height H and width W can be written as:

$$q_e = f(\frac{w}{h}, \frac{w}{W}, \frac{h}{H}, Gr) \cdot A \cdot (g' \cdot h)^{1/2} \qquad (1)$$

Where the first three factors are aspect ratio of the door way, contraction in width and contraction in height. Gr is the densiometric Grashof number. For a given geometry equation(1) can be cast into :

$$q_e = f(Gr) \cdot A \cdot (g' \cdot h)^{1/2} \qquad (2)$$

For a fixed doorheight equation (2) implies the densiometric Grashof number is a function of the temperature (density) difference only. The presence of a temperature dependence has been observed by Kiel and Wilson (1989) and Fritsche and Lilienblum (1968).

For physically describing the flow through a door way there are two distinct models in use, both which make use of the Bernoulli equation. The orifice model makes use of the assumption that the pressure distribution is equal to that in the receiving room and the two-layer hydraulics model, Dalziel and Lane-Serff (1991), where the pressure distribution in the inflowing air is governed by its own density. In the two layer model flow separation is allowed for.

3.2 Ongoing research

To minimise energy consumption used for heating of ventilation air flow its is important that the air is distributed to the parts of a dwelling where the occupants are. To meet this demand one can make use of new control algorithms that have been developed but are not used in practice yet.

Björsell (1996) reports ongoing work on numerical simulation of air motions within a building in order to develop control algorithms for directing the ventilation air to the parts of the building where it is needed. Experimental work is carried out to develop methods to measure the air flow rates through open doors using velocity measurements in the door ways, Blomqvist, Sandberg (1996).

The aim of this paper is to continue development of experimental techniques to measure internal air movements within a multi room building using tracer gas and velocity measurements. The experimental work will develop various methods that can be used for validation of the simulation models.

4. TEST HOUSE

For the experimental work has been used a unique test house in the laboratory of the department. The house is built up to look like an ordinary Swedish apartment consisting of five rooms including hall, kitchen and bathroom (Figure 1). The height of the doors are 2.0m and the width 0.7m and 1.2m (living room door). The height of the apartment is 2.5m. The mechanical ventilation system of the house can easily be changed so it is possible to obtain any system desired.

The testroom is also equipped with computer controlled system for release of tracer gas. This makes it possible to use any type of tracer gas method. Furthermore the pressure and

temperature can be recorded in each room. To be able to measure the flows through the door ways there is a computer controlled traversing unit in each door way. Each of these traversing units is equipped with 10 thermistor anemometers mounted at different heights. The anemometer was developed in house by Lundström et al. (1990) and is of the omnidirectional type. To get information about the inflowing air in each room the pressure difference compared to the ambient space was monitored during the measurements. Constant concentration tracer gas measurements have been carried out to determine the size of the incoming air flows to the different rooms including the infiltration. The result of those measurements is presented in figure 2 where the air flows are plotted as functions of the pressure difference across the building envelope.

Figure 1 Test house
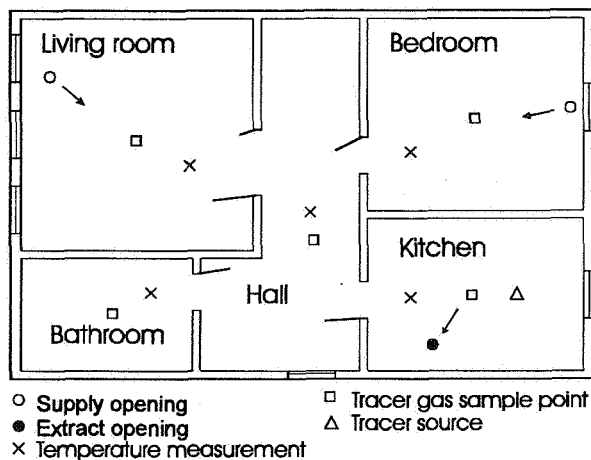
○ Supply opening
● Extract opening
× Temperature measurement
□ Tracer gas sample point
△ Tracer source
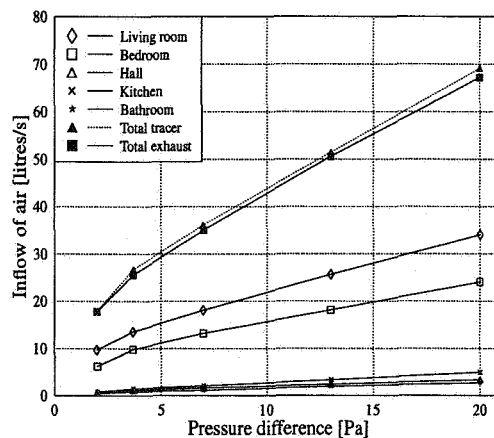
Figure 2 Measured inflow of air in each room versus pressure difference across the building envelope (doors closed).

5. EXPERIMENTAL DESIGN

5.1 Test conditions

In the test apartment was installed mechanical extract ventilation with one exhaust opening in the kitchen. The supply air was taken from the laboratory hall through two openings in the ceiling of the living room and bedroom. The exhaust air flow rate was measured by means of orifice plates, and the supply air was determined by constant concentration tracer gas measurements. To obtain a suitable temperature difference between the rooms electrically heated radiators were installed in all rooms. The electrical power of the heaters was 1000 W each. The apartment was originally equipped with ordinary commercial temperature regulators. The performance of those regulators has shown to cause temperature oscillations in the rooms. Because of the fact that even very small temperature differences between the rooms cause large bi-directional air flows in the door ways when the doors are open, it has been necessary to improve the temperature control in the test apartment in order to get steady state conditions. To solve the problem the regulators have been replaced by a computer based control system. Table 1 shows the temperature set values for the test cases and the actual measured values during the tests.

In order to measure the air flow rate through the door openings computer controlled traversing units equipped with thermistor anemometers in 10 different heights were installed in each door opening. The flow was visualised by smoke to determine the direction of the flow and the flow rate was calculated by integration over the each door aperture.

Temperature differences between hall and kitchen		
Flow rate [l/s]	Set value [°C]	Measured value [°C]
25.0	0.0	0.08
37.5	0.0	0.09
37.5	0.3	0.26
37.5	0.6	0.57
50.0	0.0	0.09
50.0	0.6	0.62

Table 1. Set values and measured values for the temperature differences for the six test cases.

To be able to trace the airflow on its way from the from the kitchen through the apartment, tracer gas was introduced in the kitchen at the triangle mark in figure 1. The concentrations were then monitored in each room at the square marks in figure 1.

At the cross marks in figure 1 the temperature was measured at four different heights by means of thermocouples connected to a data acquisition system.

5.2. Measurements

Measurements have been carried out for three different flow rates, (25, 37.5 and 50 litres/s) and for three different temperatures in the kitchen (23.0, 23.3, and 23.6°C). The temperature in the rest of the apartment has been 23.0°C.

In the door openings the air speed has been recorded in 12 (living room) resp 7 different horizontal positions in the cross section of the doors. The larger number of horizontal positions in the living room door is because of its larger width (1.2m). The values recorded are averaged over a time period of ten minutes. To determine the direction of the air movements the flow patterns have been visualised using smoke technique.

The exhaust air flow rates are measured by means of orifice plates and the supply air entering each room is measured by constant concentration tracer gas technique.

To follow the air flow patterns in the door ways tracer gas has been injected at the triangle mark in the kitchen (figure 1). The gas concentration has been measured in each room at the square marks in the same figure. The sampling points consisted of vertical tubes with holes at different heights to get an average of the concentration.

6. RESULT

6.1 Temperature and velocity

In figure 3 the result of the measurements is shown for the isothermal case. The curves show the temperature gradients in the kitchen compared to the gradient in the hall for three different extract air flow rates. In the same diagram is also shown the average air speed at the different horizontal levels. Air speed is defined as positive in the direction towards the hall. In the diagrams the upper x-axis refer to velocity and the lower x-axis refer to temperature.

In figure 4 the result is shown in the same way for the case where the temperature in the kitchen was 0.6°C higher than in the hall and the ventilation air flow rate was 50 litres/s.

6.2. Tracer gas measurements

To determine the air exchange through the kitchen door a constant tracer gas flow of 7.0 litres/h was introduced in the kitchen at the triangle mark in figure 1. To accomplish good

mixing of the tracer gas a small fan is used. The tracer gas concentration was monitored at the square mark in each room until steady state conditions were reached.

Figure 5 shows the result of those measurements in the isothermal case. The curves represent the concentrations in the kitchen the hall and in the extract opening. The calculated concentration in the extract air at steady state conditions is added as a separate curve for comparison. The concentration in the extract air is calculated as the tracer gas flow rate divided by the extract air flow rate. The measured values in the extract corresponds well to the estimate. The measured average concentration in the kitchen does also correspond fairly well to the expected value which indicates that the tracer gas is mixed well into the room air.

Figure 6 shows the measurements for the case when the temperature in the kitchen is 0.6°C higher than in the hall and the extract air flow rate is 50 litres/s.

In figure 7 the result of the measurements is summarised in a bar graph which shows the steady state concentration in the hall divided by the calculated concentration in the kitchen for the six test cases.
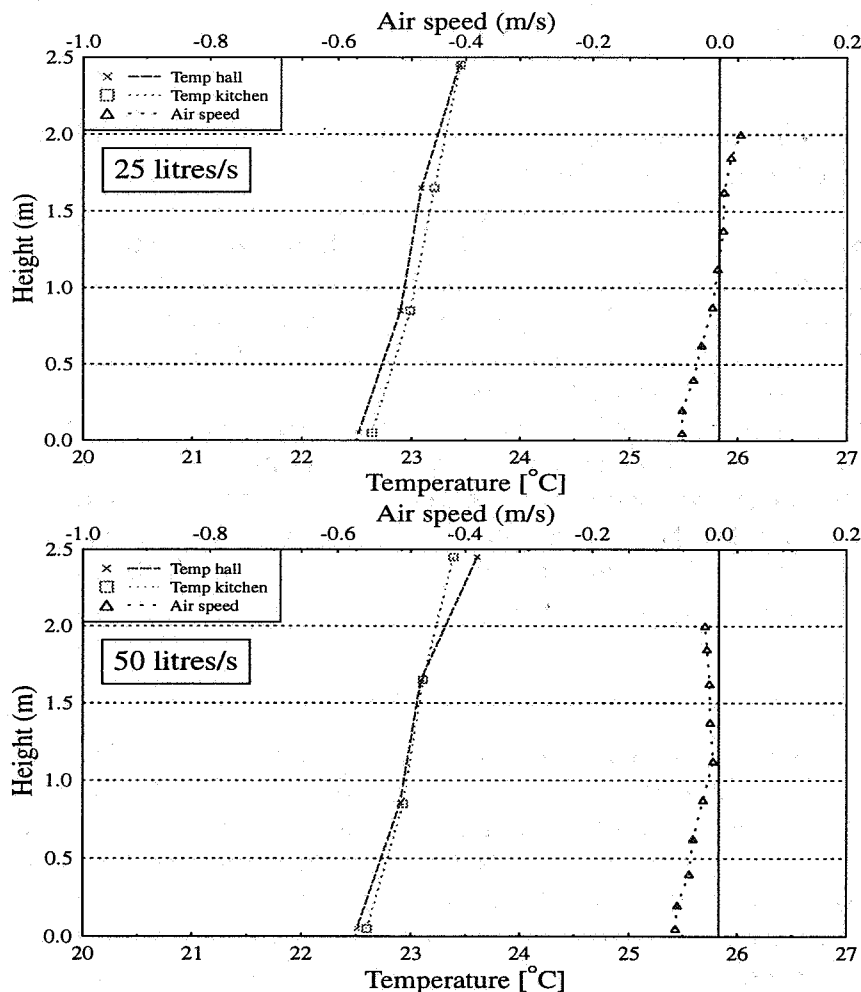
Figure 3. Air velocities and temperature differences in the kitchen door. Temperature set values are equal in kitchen and hall. Air flow rates : 25 and 50 l/s
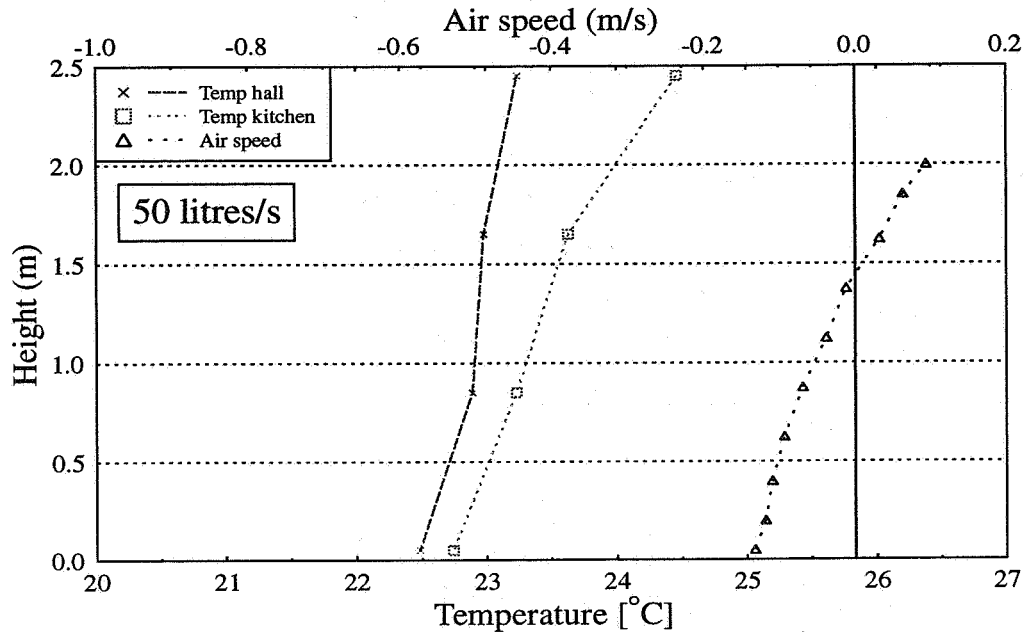
Figure 4. Air velocities and temperature differences in kitchen door. Temperature set value is 0.6°C higher in kitchen. Air flow rate : 50 l/s.

7. DISCUSSION

Using the orifice model and assuming parallel temperature distribution in the two adjacent rooms the total flow through an opening caused by temperature difference can be written as:

$$q_e = C(\Delta T) \cdot A \cdot (g' \cdot h)^{1/2} \qquad \text{(Etheridge,Sandberg)} \qquad (3)$$

Where C is an experimentally determined coefficient. When a forced flow is imposed in one direction the minimum forced flow needed to in the mean get unidirectional flow in the opening can be written as:

$$q_n = \sqrt{8} \cdot q_e \qquad \text{(Etheridge,Sandberg)} \qquad (4)$$

Figure 8 shows the air flow caused by temperature difference in the kitchen door calculated using equation(4) and the coefficient C set to 0.15. The estimated flow at a temperature difference of 0.1°C corresponds rather well with the result of the measurements shown in figure 3 where unidirectional flow has been obtained at a flow rate of 50 l/s and the actual temperature difference has been ≈0.1°C. However, the tracer gas measurements on the same case (figure 7) show that the tracer gas concentration differs from zero in the hall which indicates that the flow is not completely unidirectional. This can be explained by turbulence effects causing time dependent air flow in both directions.

It is obvious that even very small differences in temperature between rooms generates air flows which are large compared to the flows caused by the mechanical ventilation. As a comparison it can be mentioned that the Swedish building code prescribes a total forced ventilation flow
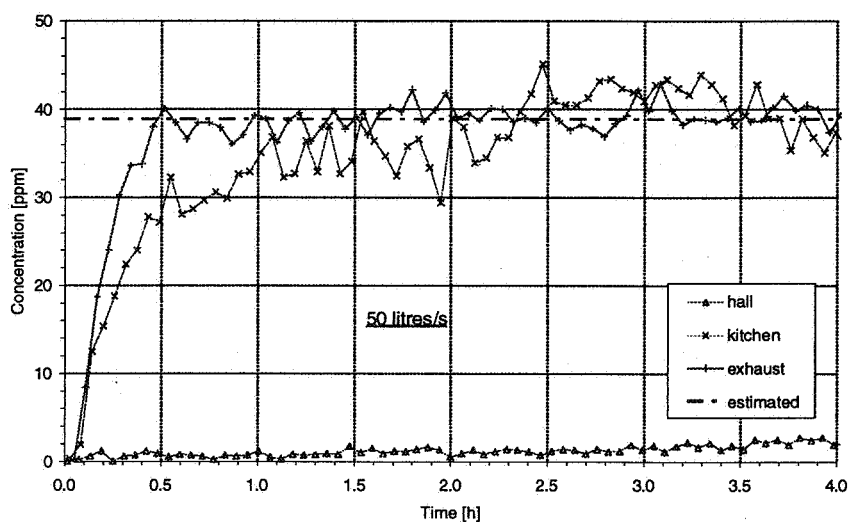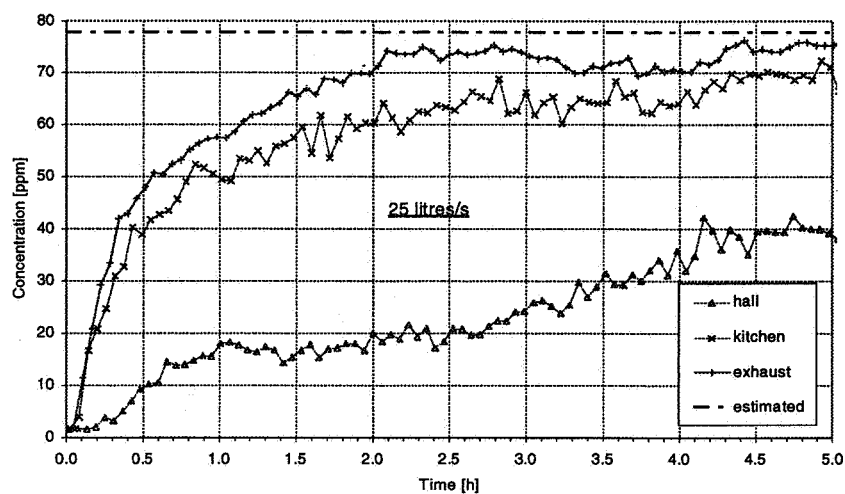
Figure 5. Spread of tracer gas from kitchen for different air flow rates. Isothermal.
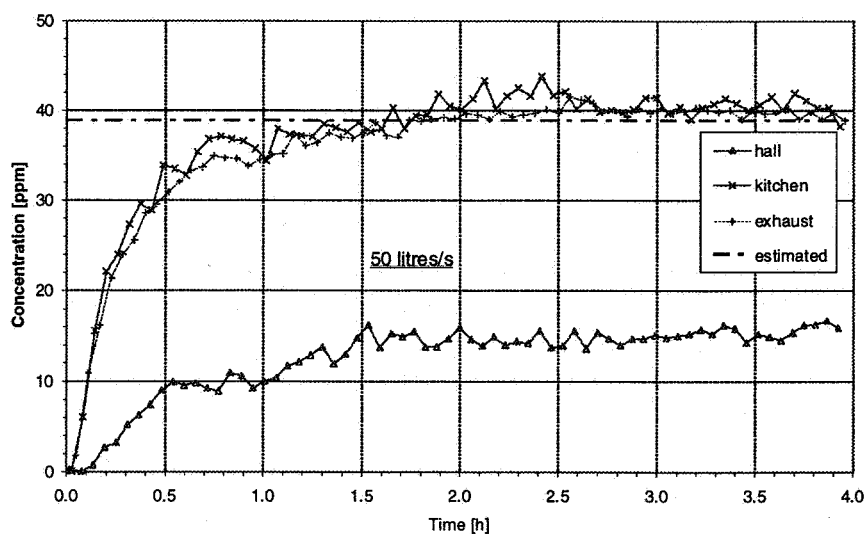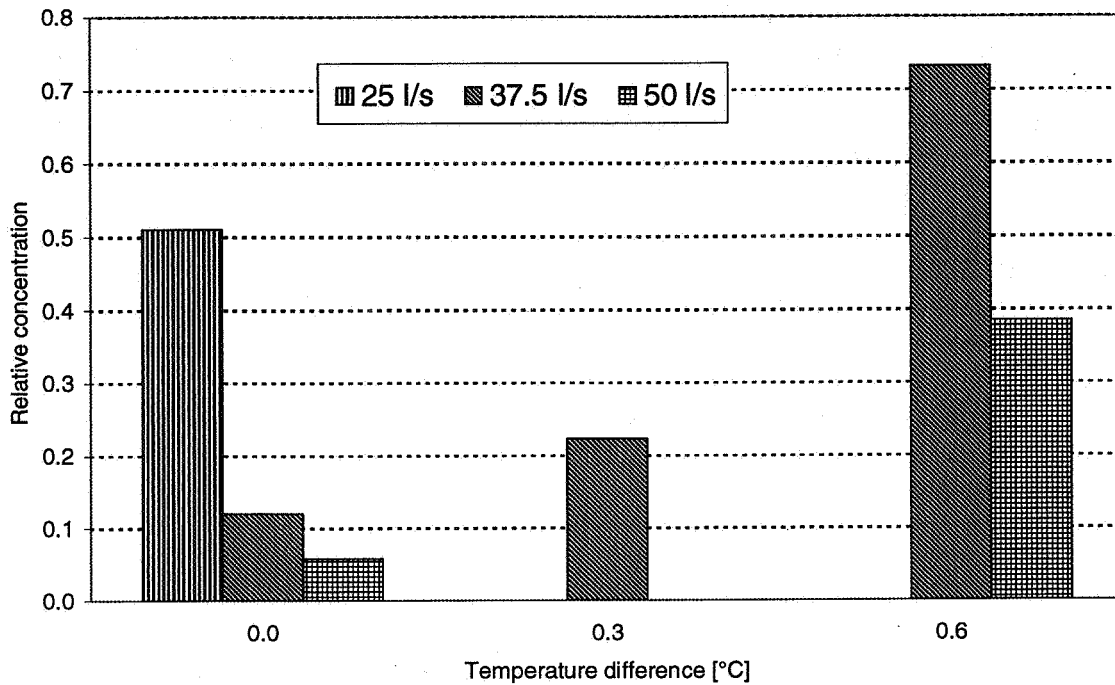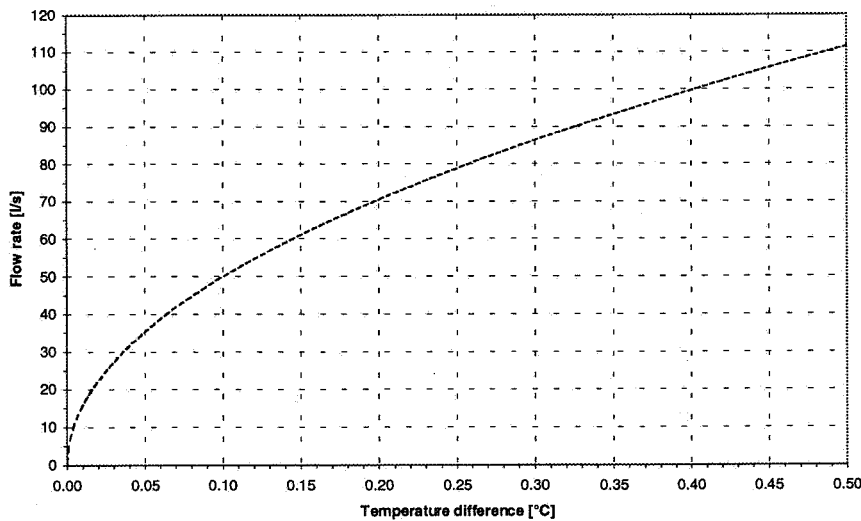

Figure 6. Spread of tracer gas from kitchen. Temperature difference = 0.6°C

rate of 25 l/s in an apartment of the same size as the test building. Usually the flow is equally distributed between kitchen and bathroom.

Figure 7. Summarised result of tracer gas measurements. Concentration in hall divided by concentration in kitchen at steady state conditions.

Figure 8. Estimated air flow rate through a door aperture using equation (4) (C=0.15).

8. CONCLUSIONS

The bi-directional flow caused even by very small temperature differences between adjoining rooms in a building are large compared to the ventilation flow rates. Therefore it is important to keep those differences small. Using modern computer based control technique it is possible to achieve temperature differences between rooms as small as 0.1°C.

The result of the measurements presented in this paper shows that in a door way of the size 0.7x2.0 m (WxH) it is required a ventilation flow rate of 50 l/s to get unidirectional flow in the mean when the temperature difference between the rooms is 0.1°C. This is in good accordance with the theoretical value.

The tracer gas measurements show however that even if the mean value of the flow is unidirectional turbulence can cause leakage through the door way. In the test case with 50 l/s and an actual temperature difference of 0.1°C the tracer concentration in the room nearest to the source room was 5% of the concentration in the source room.

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