16th AIVC Conference
Implementing the Results of Ventilation Research

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Volume 2

Preface

International Energy Agency

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

Energy Conservation in Buildings and Community Systems

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

The Executive Committee

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

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*

ANNEX 9 Minimum Ventilation Rates*
ANNEX 10 Building HVAC Systems Simulation*
ANNEX 11 Energy Auditing*
ANNEX 12 Windows and Fenestration*
ANNEX 13 Energy Management in Hospitals*
ANNEX 14 Condensation*
ANNEX 15 Energy Efficiency in Schools*
ANNEX 16 BEMS - 1: Energy Management Procedures*
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ANNEX 20 Air Flow Patterns within Buildings*
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ANNEX 27 Evaluation and Demonstration of Domestic Ventilation Systems
ANNEX 28 Low Energy Cooling Systems
ANNEX 29 Daylighting in Buildings
ANNEX 30 Bringing Simulation to Application

Annex V Air Infiltration and Ventilation Centre

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

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

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

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Implementing the Results of Ventilation Research
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The Design and Development of Two Energy and Environmentally Sustainable Prototype Office Buildings

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SYNOPSIS

The C-2000 program for advanced commercial buildings is an awards program to assist in the development of energy efficient and sustainable building technologies and design in Canada. The objectives of the C-2000 program are to develop energy efficient buildings using sustainable materials and technologies. The buildings must provide a high level of occupant comfort. The technology must be transferable to the current building industry and must meet market constraints. This paper presents a case study of the design and development of two C-2000 office buildings, in which innovative energy efficient ventilation strategies were implemented. The buildings are located in the Pacific Northwest in British Columbia and are projected to use between 30% - 50% of the energy used by a base building, performing to the ASHRAE/IES 90.1 Standards.

1.0 INTRODUCTION

The C-2000 program for advanced commercial buildings is an awards program to assist in the development of energy efficient and sustainable building technologies and design in Canada. The Bentall Crestwood Corporate Centre has two buildings as a single project under the C-2000 Awards Program. The Buildings are located in a campus style business park in Richmond, British Columbia and are referred to in this paper as Building 2 and Building 8.

The objectives of the C-2000 program are to develop energy efficient buildings using sustainable materials and technologies. The buildings must provide a high level of occupant comfort. The technology must be transferable to the current building industry and must meet market constraints.

In order to achieve the objectives of the C-2000 award program, the designers of the Bentall Crestwood Corporate Centre developed an innovative team approach, integrating all disciplines through all stages of the design. The design process was coupled with a computer modeling process to provide feedback on the effects of design decisions on energy demand and to provide a quantitative basis for measurement of performance measures. The design process progressed through a series of eight key steps; Orientation and Configuration Modeling, Envelope Design, Lighting and Power Design, Heating and Cooling Design, Ventilation Design, Building Materials Selection, Site Design and Commissioning/Quality Assurance. The design team; including architects, energy, mechanical, electrical and environmental consultants worked together to create two buildings that perform functionally, aesthetically and environmentally.

The Bentall Crestwood Corporate Centre buildings use between 30% - 50% of the energy used by a base building, performing to the ASHRAE/IES 90.1 Standards (ASHRAE/IES, 1990). The design team applied advanced window design, a self-balancing interior atrium space (volarium), advanced envelope technology, advanced and innovative systems design and window shading to achieve the energy objectives.

The Bentall Crestwood Corporate Centre buildings are thermally balanced, have good acoustic ratings, optimize daylighting, have operable windows and a once through fresh air system for ventilation. Interior ambient lighting is maintained at approximately 60 foot candles. In addition, a Commissioning/Quality Assurance process was designed specifically for the project to ensure that the buildings are properly designed, constructed, operated and maintained.

This paper presents a case study of the design and development of the two Bentall Crestwood Corporate Centre C-2000 office buildings focusing on the innovative energy efficient ventilation strategies.

2.0 VENTILATION STRATEGY FOR BENTALL CRESTWOOD CORPORATE CENTER C-2000 BUILDINGS

2.1 DESIGN PROCESS

The entire design team was involved from the beginning of the concept design phase. At this time, critical information regarding energy budgeting, client parameters, site restraints and basic technical information were exchanged. Each member of the group was made aware of every other members' concerns and ideas and contributed equally during the concept design process. The exchange of ideas between team members was facilitated by adding a series of eight meetings to the design process. The meetings were structured as follows:

Meeting 1:
- Entire team with client and funding partners.
- Discussion to resolve parameters for energy and environmental goals.
- Action to develop preliminary design goals.

Meeting 2:
- Entire design team.
- Discussion to refine preliminary design approach.
- Action to sketch basic configuration options responding to energy and environmental goals.

Meeting 3:
- Entire team including client, funding partners and experts.
- Discuss research information available as presented by guest experts in atrium design and daylighting design.
- Action to model basic configuration options.

Meeting 4:
- Architectural and energy team members.
- Review results of preliminary orientation, configuration and atrium options on DOE-2.1E.
- Action to refine modeling.

Meeting 5:
- Architectural and energy team members.
- Review results of refined modeling.
- Action to finish modeling all options.

Meeting 6:
- Architectural and client team members.
- Develop options for envelope treatment.
- Action to sketch detailed options.

Meeting 7:
- Architectural, energy and client team members.
- Select best orientation and configuration option.
- Action to input envelope design option values and model for selected configuration/orientation option.

Meeting 8:
- Entire design team.
- Review modeling results.
- Action to tabulate cost benefit analysis of various options and develop concept design approach for presentation.

The energy and environmental goals were outlined in these eight meetings. The team developed a slightly different approach to each of the two buildings. For Building 8, because the form was fixed due to owner site constraints, the design team chose to achieve C-2000 performance levels using high performance, but relatively non-exotic technologies applied in innovative and effective ways. The result is a building which can be easily reproduced by the design community at large without highly specialized resources or the assumption of a high level of risk. The design team took a much more adventurous and leading-edge approach to Building 2, relying on a super-performance envelope and a high degree of thermodynamic synergy to produce very low energy loads which can be met by extremely efficient, yet extremely simple mechanical systems.

2.2 CONCEPT DESIGN

For both buildings the objectives of the overall ventilation strategy were to:

1. Reduce the source level of volatile organic compounds in the interior spaces
2. Reduce the source level of indoor particulates.
3. Reduce the potential for indoor microbial contamination.
4. Provide alternatives to back up ventilation systems.
5. Reduce overall energy consumption.
6. Reduce the use of ozone depleting refrigerants.
7. Minimize the entry indoors of outdoor ambient pollutants such as carbon monoxide, oxides of nitrogen and particulates.

These seven objectives were to be met within the constraints of the owners' functional program. Constraints imposed upon the design of the ventilation systems included:

- Provision of outside air ventilation rates exceeding current standards (ASHRAE/ANSI, 62-1989).
- Maintenance of constant volume air flow to maximizing occupant comfort.
- Multiple zoning allowing a maximum of 1,000 ft^2/zone in the interior space and 500 ft^2/zone in the perimeter space.
- Flexibility to add further capacity and zoning to the HVAC system

- Cost efficient system design based on life-cycle costing balanced with a low capital cost.

In meeting these constraints the HVAC strategy for both Buildings 8 and 2 is predicated on the fundamental concept of maximum compartmentalization of HVAC functions. Meeting thermal and ventilation requirements on a highly local basis eliminates the intrinsic zone control (i.e. reheat), ventilation, and energy transport inefficiencies of conventional central HVAC systems while offering an extremely high level of individual zone control and flexibility.

2.3 DESIGN DEVELOPMENT

Options for ventilation were not modeled separately but formed an intrinsic part of the considerations for both buildings. As options were investigated throughout the design development, the following opportunities for improving the ventilation and energy efficiency of both buildings became apparent.

2.3.1 Operable Windows

The HVAC system for Building 2 is a low volume system based on the low heating and cooling loads the building is expected to generate. Operable windows were considered as a back up air supply system. The operable windows are tied to a simple interlock system that shuts the air supply off at one diffuser if a window is opened. Generally this is a cost effective and viable solution. The operable windows comprise approximately 10% of the general window area. However, operable windows were determined to not be an effective solution as back up ventilation for Building 8.

2.3.2 Volarium

For the internal space in Building 2 a "Volarium" was evaluated. The "Volarium" is a hybrid or modified atrium which is completely internal to the building envelope. For the purpose of ventilation, the volarium is intended to provide mixed air that has been marginally cooled or heated. The air supplied through the volarium may be freshened with specific plants that have been found to cleanse the air of specific toxins. If the volarium is not used for this purpose, the system may be reversed to exhaust the stale air from the building.

2.3.3 100% Direct-Ducted Fresh Air Systems

The design team explored direct-ducted 100% fresh air systems for both Buildings 8 and 2. The design supply of outside air is 30 cubic feet per minute (cfm) per person. Studies undertaken to determine the local relative humidity within buildings in the Pacific Northwest show that the relative humidity design goal of 50% is achievable in a low volume ventilation system without the use of a supplemental humidification.

2.3.4 Indoor Air Quality

The design team investigated the option of restricting the selection of materials for construction and interior finishing to those with minimal contaminant emission problems. Emissions of concern included off-gassing of harmful vapors (commonly found in carpet glues, wall vinyl and some paints) and particulates (from the fine breakdown of unstable materials such as insulation, cloth fabrics and carpets).

Locations of outside air intakes were evaluated to minimize the intake of carbon monoxide and other contaminants commonly found in outdoor air. Commissioning Specifications were written with the intent of eliminating the potential for microbial contamination.

2.4 HEATING, VENTILATION AND AIR CONDITIONING SYSTEMS

The ventilation system in Building 8 provides unmixed, outside air directly to a four-pipe heat/cool fan coil unit and ceiling diffuser system in each individual HVAC zone using a roof-mounted ventilation air handler with hot water pre-heat. The outside air is mixed with local return air in the fan coil and subsequently provided directly to the space. This positive ventilation delivery directly to the zone avoids the inherent loss of overall ventilation effectiveness associated with centralized variable air volume (VAV) systems. Combined with the constant volume air flow characteristics of the fan coil system, estimated net ventilation effectiveness is 0.90. Passive relief air dampers in the ceiling space to the outside wall at several locations on each floor provide system trim balancing in conjunction with mechanical washroom and other specialized exhaust (e.g. photocopy rooms, kitchens, etc.). The system is fully flexible to accommodate tenant improvements, and includes accessible exhaust risers for tenant connection.

Installed system capacity accommodates a net effective ventilation rate of up to 30 cfm per person for initial and periodic building flushout purposes, although the normal final operational rate is anticipated to be 20 cfm per person. Ongoing air quality monitoring may allow the rate to be reduced further. Gross fan coil air supply will provide a minimum zone air circulation rate of four air changes per hour.

Given the relatively simple ventilation configuration and the associated ease of maintenance (the most catastrophic failure would consist of a motor replacement which could be accomplished in a few hours), provision for backup ventilation was not considered economically justifiable.

Building 2 extends the ventilation concepts introduced for Building 8 even further by providing unmixed, outside air directly to overhead high induction room diffusers in a "once through" ventilation configuration. The ventilation air is tempered by zone heat/cool coils and is the sole primary diffuser supply. Secondary induction occurs at the room level, and there is subsequently no inter-room air mixing or no inter-room air recirculation. Estimated net ventilation effectiveness is 0.90. Relief and exhaust provisions are similar to Building 8, as are the installed and planned ventilation capacities of 30 and 20 cfm per person respectively. Effective induced room air movement will be validated through laboratory verification of individual products, but is not expected to be less than four air changes per hour. This system is also fully flexible with respect to accommodating tenant improvements. Backup ventilation for Building 2 will be provided by openable windows. Outside air filtration is minimum 50% dust spot efficiency for both buildings.

2.5 ENERGY EFFICIENCY

A preliminary energy efficiency plan was developed at the Concept Design phase of the project. Although a number of modifications to specific strategies had to be made as the analyses proceeded through Design Development, the initial premises and performance targets remained intact.

Simplicity, elegance, robustness, and cost-effectiveness were the key criteria for all strategies. Complex, highly exotic or specialized, or "fragile" technologies were avoided. The introspective question continuously asked by the design team was whether or not a proposed approach or technology would be likely to be embraced and reproduced by the mainstream building industry. If not, it was abandoned as unsuitable.

Reflecting this premise, Building 8 was developed using entirely mainstream and readily reproducible technologies applied in an effective and integrated manner. The resulting building, a visual twin to an existing adjacent building achieves the 50%-of-ASHRAE/IES 90.1 energy performance target with negligible projected net incremental capital cost compared to the baseline market building (less than 2%).

Building 2 was developed using slightly more advanced, (but not exotic) technologies with the objective of significantly exceeding the 50% of ASHRAE/IES target. At the present level of development, the building energy use approaches 30% of ASHRAE/IES 90.1.

DOE 2.1e (integrated with LBL Window 4.1) was used for all energy analyses. In addition to the Building 8 and Building 2 ASHRAE/IES Reference design models, two variations of a Building 8 "market" or baseline building were modeled for life-cycle costing purposes.

3.0 CONCLUDING COMMENTS

Early in the project, modelling of the energy demands for both buildings identified an important practical design consideration that carried through to the final design of the HVAC systems. Put simply, the design team determined that while the magnitude of envelope and electrical energy loads is clearly a cornerstone of building energy performance, the single most significant factor in the performance equation is often the response of the HVAC system to these loads. In this regard popular central mixed-air VAV systems do not perform well. They generally address multiple-zone load variations by supplying cooling air to meet the worst-zone cooling load, and then relying on VAV box shutdown combined with reheating to control overcooling in less critical areas. In practical reality, the minimum VAV box position is dictated by ventilation and/or air circulation requirements, and since this is often still well above what is required to meet the cooling load, the zone operates in reheat mode for extended periods of time. Most central VAV-reheat systems do in fact operate as constant volume reheat systems most of the year. The wider the zone thermal variances, the more severe the effect.

One solution to this problem is to compartmentalize HVAC systems as much as possible, minimizing the number of zones served by any one system. In this respect the HVAC systems for both Buildings 8 and 2 extend this practical concept to its logical conclusion by meeting heating and cooling loads at the zone, or "terminal" level. The resulting buildings use between 30% - 50% of the energy used by a base building, performing to the ASHRAE/IES 90.1 standards. To achieve this goal, the design team made use of advanced window design, an interior self-balancing atrium space (volarium), advanced envelope technology, advanced and innovative systems design and window shading.

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**Implementing the Results of Ventilation Research
16th AIVC Conference, Palm Springs, USA
19-22 September, 1995**

Field Survey of Residential Heat Recovery Ventilation Systems: Occupant Interactions

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TITLE:

FIELD SURVEY OF RESIDENTIAL HEAT RECOVERY VENTILATION SYSTEMS: OCCUPANT INTERACTIONS

SYNOPSIS:

The installation of packaged heat recovery ventilation (HRV) systems has recently become common practise in new homes in Canada. Despite improvements in product quality and reliability, HRV systems are only capable of providing safe, continuous, efficient and effective ventilation if homeholders have a understanding of the basic operation and maintenance procedures and the system's interaction with other house systems. Furthermore, homeholders must be able to perceive the value of HRV systems if they are expected to operate them.

Canada Mortgage and Housing Corporation initiated a research project to determine the degree to which homeholders are capable, or willing, to interact with HRV systems. HRV systems within fifty-eight, regionally representative houses of various age were inspected to characterise the condition and performance of the systems as found. Interviews were conducted with the occupants to determine their understanding of the operation and maintenance requirements of their HRV systems and their perceptions of system value, effectiveness and efficiency.

While most occupants reported an understanding of the operation and maintenance needs of their HRV systems certain disparities exist. For instance, more than half of the systems surveyed had immediate service requirements such as filter, heat recovery core and intake grille cleaning. More than half of the systems had unbalanced supply and exhaust air flows. Ventilation rates were found to be substandard in 60% of the homes surveyed. The occupants of tract built homes demonstrated the least appreciation of the operating and maintenance requirements of HRV systems. The configuration of the HRV systems ductwork and the availability of controls also was found to have an influence on occupant interactions. This investigation demonstrated that most homeholders appear willing and able to interact with HRV systems. More consumer education and refined (user-friendly) control and maintenance strategies are required to ensure the successful adoption of HRV systems within Canadian homes.

1.0 INTRODUCTION

In Canada, heat recovery ventilators (HRV) are commonly installed in new homes to meet building code ventilation requirements or in existing homes to improve indoor air quality. Although HRV systems have been routinely installed in energy efficient and custom homes for the past 15 years, their application in conventional homes is a relatively new phenomenon. While the evolution of the design, fabrication and installation processes has significantly improved the reliability and performance of HRVs, questions remain unanswered concerning the impact of occupant-system interactions and HRV installation configurations on the ability of the systems to safely, efficiently and effectively ventilate homes.

Recent surveys have indicated that homeholders that dwell in new homes with HRV systems are very knowledgeable about the operation of their ventilation systems and are generally very satisfied with the performance (1). However, HRV systems are relatively complex for homeholders to understand, operate and maintain, particularly when compared with more familiar approaches to ventilation such as operable windows and localised exhaust-only systems. Many components such as motors, filters, heat recovery cores, air intake grilles require routine service. Field investigations of HRV systems have found that homeholder interactions, or a lack thereof, can be problematic as poorly maintained HRVs can fail to operate properly (2).

Additionally, homeholders must be able to control the ventilation system in order to meet ever changing needs. Control strategies often include combinations of manual and automatic switches that are either remotely located (e.g.; in kitchens and bathrooms) or are mounted on the HRV itself (which are usually found in relatively inconvenient locations such as basements or service rooms). Occupants must also be aware of how HRV systems interact with central forced air heating/cooling systems, combustion venting systems and the building envelope if the systems are to be operated in a safe, efficient and effective manner. The failure of occupants to properly operate and maintain HRV systems could cause indoor air quality problems due to inadequate ventilation and combustion venting failures due to unbalanced air flows. Building envelope durability could also suffer due to high indoor humidity levels and forced exfiltration due to insufficient and unbalanced ventilation, respectively.

The objective of this study is to assess the ability of homeholders to interact with residential HRV systems and the subsequent impacts on system performance, system condition and occupant perceptions of indoor air quality, and the overall value of HRV systems.

2.0 METHOD:

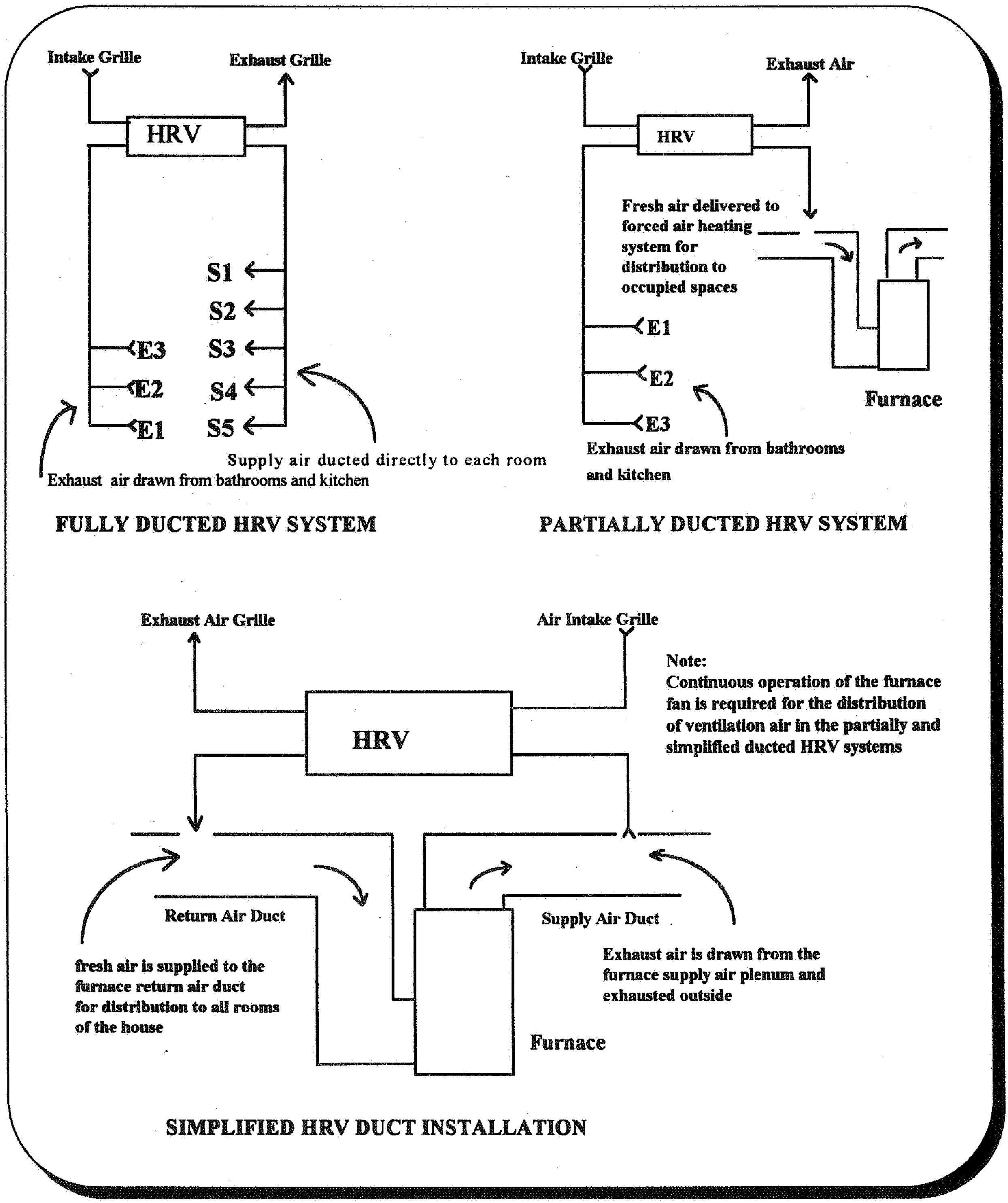
Fifty-eight houses located in the central, Western and maritime regions of Canada were selected for the field investigation to characterise regional differences in installation practises, occupant awareness and market penetration of HRV systems (Table 1).

Type	Region	2-4 years old	5 -7 years old	8 years and older	Subtotal
Fully ducted	Western	3	4	2	
	Central	4	0	1	
	Eastern	3	2	4	
Subtotal					23
Partially Ducted	Western	3	3	0	
	Central	7	1	4	
	Eastern	4	1	1	
Subtotal					24
Simplified Ducting	Central	11	0	0	
Subtotal					11
Total		35	11	12	58

Table 1.0: HRV System Type, Region and Age Distribution

The HRV systems selected were representative of the three most prevalent types of installation configurations: fully ducted, partially ducted and simplified systems (Figure 1). The sample set of houses contained HRV systems that were between 2 and 15 years old.

FIGURE 1: HRV INSTALLATION CONFIGURATIONS



The field investigation consisted of an inspection of each HRV system and an interview with the associated homeholder. The HRV inspection included an general assessment of the condition of the HRV and duct system, measurement of air flows to and from the house, and the compliance of the system with appropriate codes and standards. The interview was performed to assess the occupant's understanding of the HRV operation and maintenance needs, control strategies, interaction with other house systems, and the implications of improper system operation and maintenance on health and safety and building durability. The occupant's perceptions of indoor air quality were also solicited.

The investigation was conducted during January, February and March to assess the performance of HRV systems during extreme conditions. Given cold outdoor air temperatures, closed windows and high occupancy levels, homeholders would be more likely to be conscious of poor indoor air quality conditions. Additionally, under such conditions, any failure on the part of homeholders to properly operate and maintain their HRV systems would be more noticeable to the investigators.

3.0 RESULTS:

The results of the study can be presented in terms of the occupant survey results, HRV system inspection results and the apparent relationships between occupant interactions with the HRV systems and subsequent system performance and condition.

3.1 Occupant Survey Results:

Of the 58 occupants interviewed:

- 69% reported having had their system explained to them,
- 84% had reported having manuals for the HRV (although only 67% indicated that they had read them),
- HRV service agreements were not at all common,
- 77% reported to understand the purpose and operation of their HRV systems,
- 26% reported indoor air quality problems (95% of the problems reported concerned overly dry indoor air)
- 81% indicated that they performed regular maintenance on the HRV systems,
- while 70% reported that they understood the HRV system controls, only 42% understood the automatic humidity controls,
- all respondents used the HRV system high speed activation switches located in bathroom and kitchen areas when they were provided,
- controls mounted on the HRV cabinet were the least likely to be used or understood,
- 81% reported operating the HRV system continuously during the winter (the apparent lack of need, energy concerns and cold drafts were primary reasons for deactivation in the winter),
- 50% reported operating the HRV continuously during the summer (a preference for opening windows for ventilation during the summer was the most common reason for the deactivation of the HRVs at this time),

- 66% of the occupants with partially ducted systems understood the importance of operating the fan of the forced air heating system to distribute the ventilation air,
- 33% of the occupants with simplified systems understood the importance of operating the fan of the forced air heating system to distribute the ventilation air.

3.2 HRV Inspection Results:

Of the 58 HRV systems inspected:

- 71% were installed in accordance with good practise guidelines,
- simplified HRV systems were mostly likely to be improperly installed and commissioned,
- 75% had remote controls in the living area while the remaining 25% had controls mounted on the HRV only,
- 50% of the systems had airflow balancing equipment installed,
- 60% of the system airflow rates are below current Canadian ventilation standards, 7% were found to be too high.
- 55% of the systems had unbalanced supply and exhaust air flows (29% of the systems were unbalanced by more than 40%)
- 50% of the systems had filters, heat recovery cores, HRV cabinet interiors that required cleaning,
- 14% of the systems inspected had exterior air intake grilles that were loaded with debris,
- Of the systems integrated with forced air heating systems, only 45% of the occupants reported operating the fan of the forced air system continuously to distribute the ventilation air,
- Of the 15% of the systems that had major repair or service performed, all were greater than 4 years old and 50% were greater than 7 years old.

3.3 Occupant Interactions with the HRV Systems:

Of the occupants who reported that they understood their HRV systems (77%):

- 50% had HRV systems that were unbalanced
- 60% had substandard ventilation
- 52% of occupants with partially ducted or simplified HRV systems were not aware of the need to operate the fan of the forced air heating system.

Of the occupants who reported indoor air quality problems (26%):
- 60% had substandard ventilation
- 62% had unbalanced HRV system supply and exhaust air flows
- 56% had HRV systems with dirty filters, heat recovery cores, HRV cabinets.

Of the occupants who reported that they performed regular maintenance (81%):
- 42% had systems with dirty filters, cores, cabinets
- 15% had blocked air intakes
- 46% had unbalanced HRV system supply and exhaust air flows

Of occupants who reported having read the HRV operation and maintenance manuals (32%):
- 43% had dirty filters, cores, cabinets
- 49% had unbalanced HRV supply and exhaust air flows
- 40% operated the HRV systems at recommended ventilation rates

Of the occupants who reported that they did **not** read the HRV manuals (68%):
- 77% had dirty filters, cores, cabinets
- 63% had unbalanced HRV supply and exhaust air flows
- 17% operated the HRV systems at recommended ventilation rates

Of the HRV systems that had been provided with remote controls in the living areas:
- 80% of the occupants understood how to use the controls

Of the HRV systems that had been provided with no remote controls:
- 36% of the occupants understood how to use the controls

4.0 DISCUSSION:

In Canada, the increased airtightness of houses, growing occupant awareness of indoor air quality and more demanding regulatory requirements have created a need for effective and efficient residential ventilation systems. While packaged residential heat recovery ventilation systems represent one of the most promising means of meeting this need, little was known about the ability, or desire, of homeholders to properly operate and maintain these relatively new and sophisticated systems. However, based on the results of this investigation, it has been demonstrated that, for the most part, homeholders are willing to operate and maintain HRV systems even though they often lack the technical knowledge to properly do so. Nevertheless, the majority of the homeholders surveyed are satisfied with their HRV systems and the high level of indoor air quality the systems provide.

The results of the occupant interview and HRV system inspection are representative a wide cross section of homeholder demographics, regional differences, system configurations and age. Accordingly, the results presented do not identify specific concerns or observations that pertain to certain aspects of HRV installations. For instance, of the occupants surveyed, it was discovered that knowledge of HRV system operation and maintenance requirements was greater for those who live in custom homes as opposed to those who live in tract built housing. Custom homeholders usually were responsible for the decision to install their HRV systems and were more likely to appreciate the value of the systems. Homeholders living in tract housing often moved into homes where HRV systems had been already provided with little or no discussion regarding the value of the systems.

The type of HRV installation was also found to have bearing on homeholder use and maintenance patterns. Simplified systems were far less likely to be understood, operated or maintained by homeholders, particularly when no remote controls in the living areas were provided. The absence of a day to day presence of simplified systems tended to promote an "out of sight, out of mind" attitude on the part of homeholders. These observations are of significant concern as the application of HRV systems in tract built housing represents the fastest growing market for HRV systems in Canada.

While homeholders tend to believe that they understand the purpose and operating principles of their HRV systems, the control strategies were generally less well understood. Dehumidistat controls were found to be the most poorly understood control mechanism. Improper settings may have been responsible for homeholder complaints of uncomfortably dry indoor air. It was noted that in many of the houses where the homeholders complained of dry air, humidification equipment had been subsequently installed. This corrective action may not have been necessary if the homeholders had a better understanding of the relationships between the dehumidistat setting, HRV continuous airflow speed setting and the relative humidity of the indoor air. Furthermore, few homeholders understood the interaction of their HRVs with other house systems. The need to operate the fan of the forced air heating systems when simplified and extended ducting configurations are used was not readily apparent to most homeholders. The potential impact of unbalanced airflow rates on the venting of other household combustion appliances and the building envelope was also unknown. This lack of understanding could be detrimental to the health of unsuspecting homeholders and the durability of the building envelope although no evidence of either concern was noted during the inspections.

In general, the successful maintenance of HRV systems is not beyond the capabilities of the average homeholder. However, there is a concern that maintenance activities are not as rigorous or frequent as required. Dirty filters, cores and intake grilles were found in many installations where the homeholder was under the impression that his maintenance activities were sufficient. There was also a general lack of awareness concerning HRV defrost systems and motor oiling requirements. In the majority of cases, homeholders are not provided with the means to balance and/or verify the HRV system's air flows. This was evident in the number of systems that were found to be unbalanced and /or operating at substandard ventilation rates. While formal repair and service agreements could help alleviate many of these concerns, most homeholders do not recognise that such arrangements may be necessary.

Given the successful evolution of the packaged residential heat recovery ventilator, the delivery of comprehensive HRV system design and installation courses for contractors and the pending adoption of specific performance and installation related requirements for ventilation systems in the 1995 National Building Code of Canada, problems relating to product quality and installation practise can be expected to occur less frequently. Homeholder related operational and maintenance related concerns will continue to occur and possibly increase as HRVs proliferate in the conventional housing stock. HRV manufacturers and installing contractors must continue to develop and refine the information transfer processes used to educate homeholders about HRV systems. Additionally, operation and maintenance strategies must be refined to make them more obvious and accessible for the average homeholder. As this technology improves and becomes more common in housing, the occurrence of many of the operation and maintenance related problems noted in this investigation should decrease dramatically.

ACKNOWLEDGEMENTS:

The author of this paper gratefully acknowledges the efforts of Buchan, Lawton, Parent Ltd in the preparation and execution of the HRV inspections and occupant interviews and the subsequent data analysis. This project was made possible through funding from Canada Mortgage and Housing Corporation and the Program of Energy Research and Development.

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Implementing the Results of Ventilation Research
16th AIVC Conference, Palm Springs, USA
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Improvement of Mechanical Ventilation Systems Regarding the Utilization of Outdoor Air

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Synopsis

Nowadays it is rather common with demand controlled ventilation in public buildings and offices. The purpose of demand controlled ventilation is to adapt the ventilation to the varying needs of the occupations. In dwellings it is rather unusual with demand controlled system. The main reason for that is the high investment cost for the system. The outdoor air used for ventilation in dwellings is therefore not effectively used. For example in a mechanical exhaust ventilation system 50 % of outdoor air is leaving the house without being used of the people.

In a multi zone computer program simulations baced on tracer gas measurements were done for two modern typical houses. We did calculations for outdoor air flow for the whole and for a single room, during a heating season. Two different ventilation systems were simulated. The air flows between rooms and between inside and outside were calculated.

We found that we could utilize the outdoor air better if the systems were reconstructed. The reconstructed system contains the same amount of air inlets and outlets, but all outdoor air is supplied to the bedrooms. Further calculations show us the ratio between the interzonal air flow created by the fan(s) and interzonal air flow created by differences in temperature was 1 to 10.

The paper presents the results from calculations of air flows for two diffrent ventilation systems. Further on the paper discusses how a system should be constructed to utilize the outdoor air in an optimal way.

1. INTRODUCTION

An investigation into various ventilation principles employed in modern single-family houses has been carried out by, and on the initiative of, the Swedish National Testing and Research Institute (SP), financed by the Swedish Council for Building Research (BFR), the National Board of Housing, Building and Planning and the Swedish building industry's Development Fund. The investigation involved 40 single-family houses, built between 1988 and 1992.

One of the objectives of the project was to 'determine, by means of measurements and calculations, how well different types of ventilation systems ventilate individual rooms.' In addition, another objective has been to investigate the air source, i.e. outdoor air or transferred air (air flowing into one room from an adjacent room).

In order to obtain this data, outdoor air flows and transferred air flows to an individual room have been measured in 40 houses. These measurements were made over a period of a month in a large bedroom, using the passive tracer gas method. The MOVECOMP-PC program (Bring 1988, Herrlin 1987) has been used to generalise the results. The program allows air flows between rooms, outdoor air flows, infiltration and exfiltration to be simulated.

This paper describes the results from calculations and measurements of ventilation in two of the 40 houses having different types of ventilation systems. The measured results from the other houses have been described in a report (Blomsterberg, Carlsson, 1995).

2. DWELLING WITH MECHANICAL EXHAUST VENTILATION SYSTEM

2.1 General

In a house having a mechanical exhaust ventilation system (see Figure 2.1), the proportion of outdoor air entering an individual room depends on the number of outdoor air inlets, air leaks in the building envelope and the pressure difference across the building envelope. This pressure difference depends on the general airtightness of the building, the exhaust air flow rate, wind velocity, wind direction and the indoor/outdoor temperature difference.

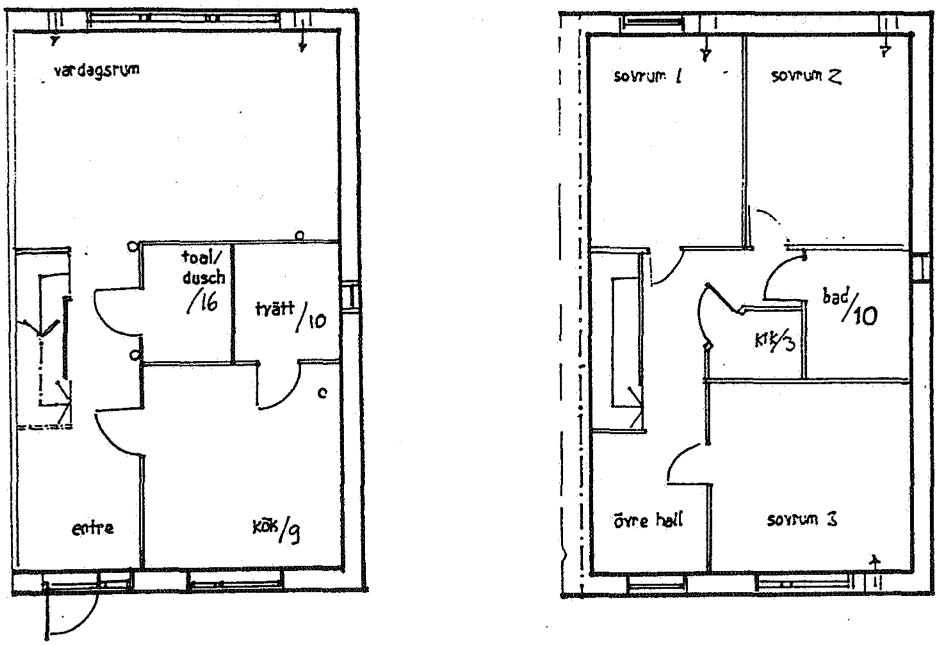

Figure 2.1 Positions of ventilation air inlets in a 1½-storey house with a mechanical exhaust system.

Single-family houses generally have outdoor air inlets in bedrooms, the sitting room and in any general-purpose room. If these inlets are closed in the sitting room and the general-purpose room, the proportion of outdoor air entering via the bedrooms will increase.

By how much will the outdoor air flow rate to the bedroom increase if the outdoor air inlets in the other rooms are closed? What will be the effect on air change rate in the sitting room and general-purpose room when the outdoor air inlets in those rooms are closed?

2.2 Input data for simulation

One of the houses in the survey, having mechanical exhaust ventilation, has been simulated by the program. This is a 1½-storey house, with a conventional floor plan (see Figure 2.1), with bedrooms and bathroom on the upper floor and the remaining rooms on the ground floor. The ventilation system is typical of that in modern Swedish single-family houses with mechanical exhaust ventilation. The house fulfils Swedish Building Regulations requirements in respect of airtightness of the building envelope (3.0 air changes/h, 50 Pa).

The measured pressure/flow characteristics of the outdoor air inlets have been used in the simulations (Mellin, 1980). The house's wind pressure coefficients have been taken from wind tunnel investigations (Wiren, 1985). The actual airtightness of the house was measured during the diagnostic tests after construction (2.8 $m^3/h, m^2$). On the basis of our experience of leakage measurements through building envelopes in conventional single-family houses, we feel that the actual air leakage in a 1½-storey house can be regarded as being distributed as follows: 60 % through the floor/ceiling structure between the ground floor and upper floor, 20 % via penetrations, 10 % at the edges of floors, 5 % around the edges of the ceiling and 5 % through windows. The exhaust air flow rate used in the program (176 m^3/h) is that as measured for the house. Room temperatures and temperature gradients are based on measurements made when checking the performance of the system (see Table 2.2). Climate data for the 1971 reference year for Stockholm has been used when calculating heating and ventilation requirements for a heating season.

Table 2.2 Room temperatures and temperature gradients as used in the simulation.

Room	Room temperature, °C (at floor level)	Temperature gradient, °C/m (vertical)
Sitting room	18.4	0.6
Utility room	18.5	0.6
Kitchen	19.1	1.0
Hall, ground floor	19.1	0.6
Bedroom 1	18.8	0.5
Bedroom 2	19.1	0.5
Bedroom 3	19.8	0.3
Landing, upper floor	19.3	0.3

2.3 Comparison between measured and calculated values

The first computer run was performed in order to compare the actual measured results obtained from tracer gas measurements with the calculated outdoor air flow rate for bedroom 3 (see Figure 2.1). Simulation was performed for two different outdoor temperatures, with a wind velocity of 1 m/s. The model results were in good agreement with the measured values. The measured outdoor air flow rate to the bedroom was 3.9 l/s, while the calculated outdoor air flow rates to the bedroom at ambient temperatures of 0 °C and 10 °C were 3.4 l/s and 4.0 l/s respectively.

The computer model calculations were made on the basis of open bedroom doors and at two different ambient temperature levels in order to illustrate the effects of temperature. During the actual measurement period, the ambient temperature varied between 0 °C and 10 °C, while the indoor temperature was 20 °C. As indicated by the model simulation, the outdoor air flow rate to the bedroom increased when the ambient temperature increased, as a result of thermal driving forces.

2.4 Calculations for a whole heating season

The outdoor air flow rate is less than 8 l/s (the Swedish Building Regulations requirement for bedrooms with two persons) for most of the heating season (see Diagram 2.4.1). The reason for the variation in flow rate is partly thermal driving forces and partly the effects of wind.

The house used for the model contained two outdoor air inlets in the sitting room (see Figure 2.1). If these inlets are relocated to bedrooms 2 and 3, the outdoor air flow rate to these bedrooms will increase (see Diagram 2.4.2). The average outdoor air flow rate over the period increases from 2.6 l/s to 5.8 l/s in bedroom 3 if the sitting room air inlets are 'transferred' to the
bedrooms.

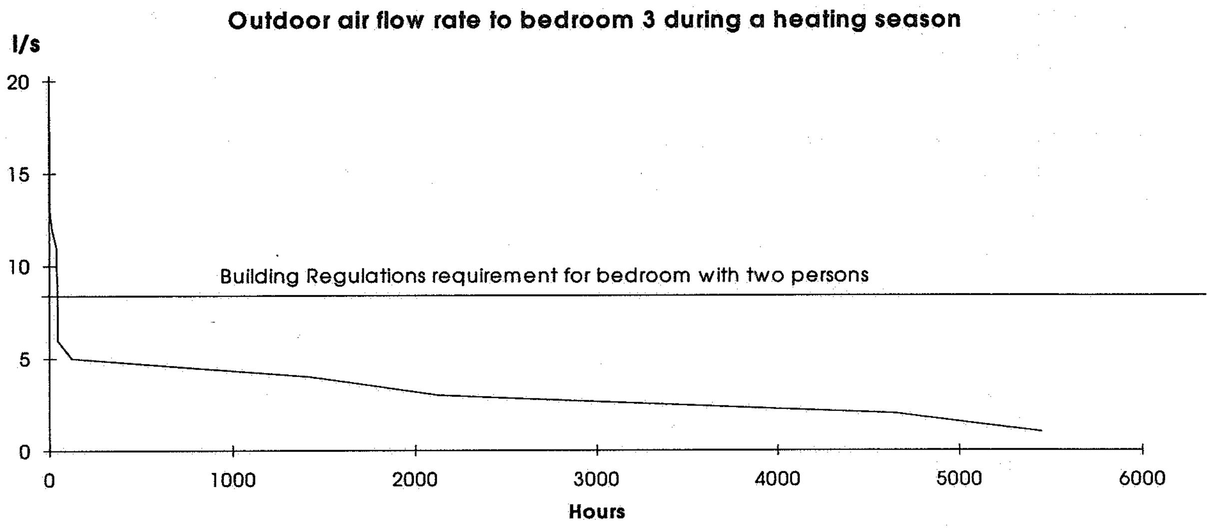

Diagram 2.4.1 Duration diagram (showing the Building Regulations requirement for 8 l/s outdoor air flow rate) for the outdoor air flow rate to bedroom 3 (standard type ventilation system, see Figure 2.1). Over the heating season, the average value of outdoor air flow rate is 2.6 l/s.

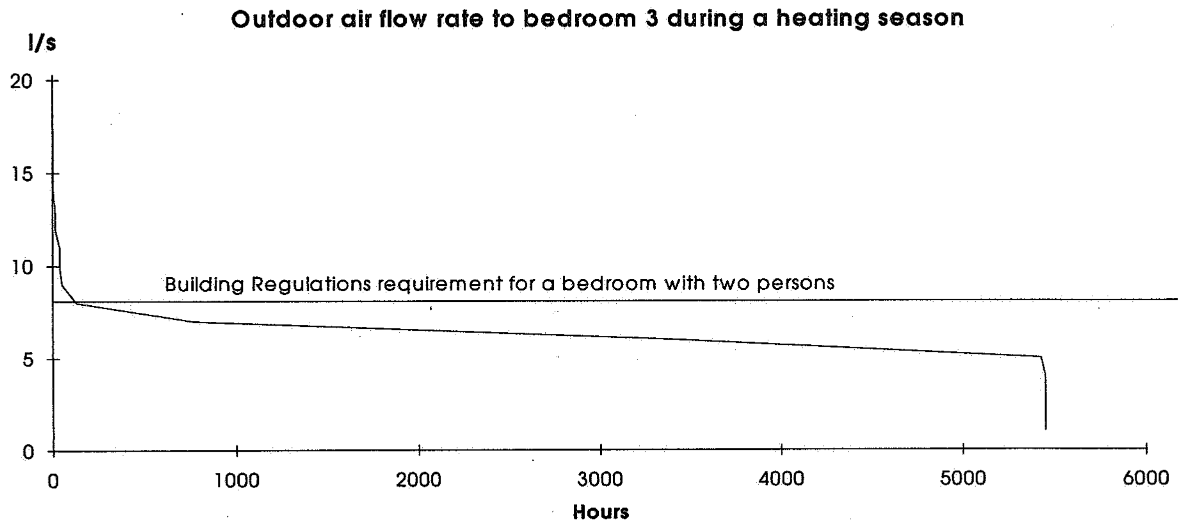

Diagram 2.4.2 Duration diagram (showing the Building Regulations requirement for 8 l/s outdoor air flow rate) for the outdoor air flow rate to bedroom 3 (with closed outdoor air inlets in the sitting room and an extra outdoor air inlet in the bedroom). Over the heating season, the average value of outdoor air flow rate is 5.8 l/s. With open bedroom door.

Both the simulations were performed assuming an open bedroom door. If the door is closed, the outdoor air flow rate is reduced as a result of the pressure drop across the door. A further simulation was performed without air inlets in the sitting room, but with a closed bedroom door. This indicated that the average outdoor air flow rate during the heating season fell from 5.8 l/s to 4.9 l/s. Moving the outdoor air inlets from the sitting room to the bedrooms thus greatly increases the proportion of outdoor air to the bedrooms.

3. THE EFFECT OF INDOOR TEMPERATURE ON INTERNAL AIR CHANGE RATES

Three simulations were performed in order to investigate the effects on air change rates in individual rooms with different types of ventilation arrangements. Airtightness parameters were assumed to be in accordance with Swedish standard requirements for the building envelope. The temperature difference between indoor and outdoor temperatures was taken to be about 20 °C, with a wind velocity of 1.0 m/s.

1. Air change rate in the room, as powered by mechanical ventilation only, i.e. no temperature difference between the rooms (standard ventilation system).
2. Air change rate in the room, as powered by mechanical ventilation and temperature differences (see Table 8.1) between the rooms (standard ventilation system).

3. As simulation No. 2, but with outdoor air inlets only in the bedrooms.

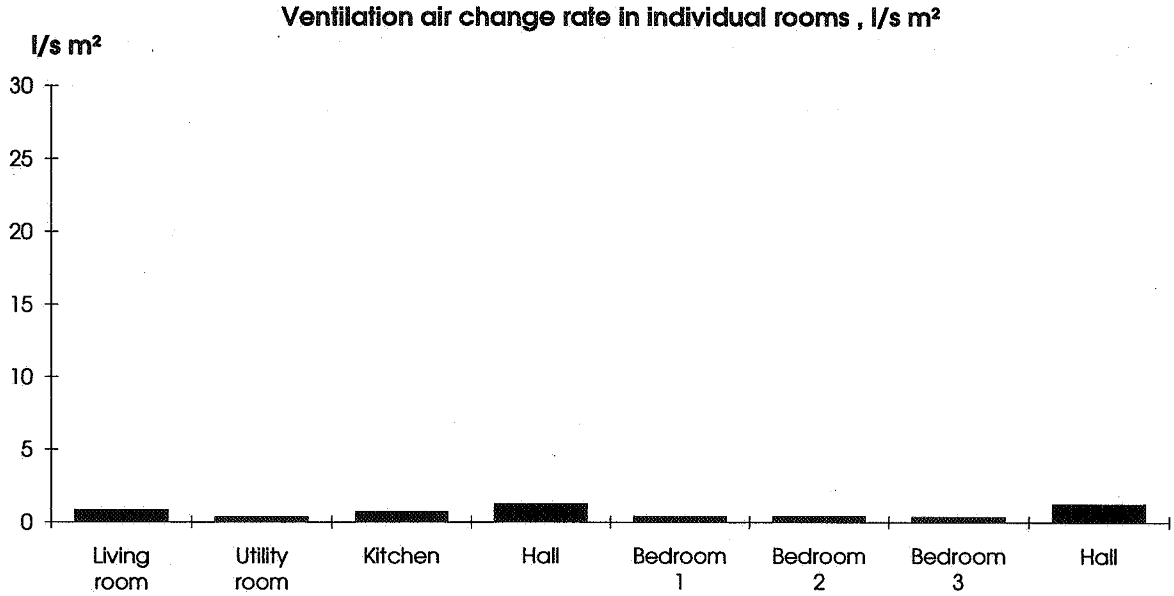

Diagram 3.1.1 **Simulation no. 1.** Ventilation air change rates (l/s, m² of floor area) as generated by mechanical ventilation only. Simulation results based on the standard design of ventilation system. Rooms as shown in Figure 2.1.

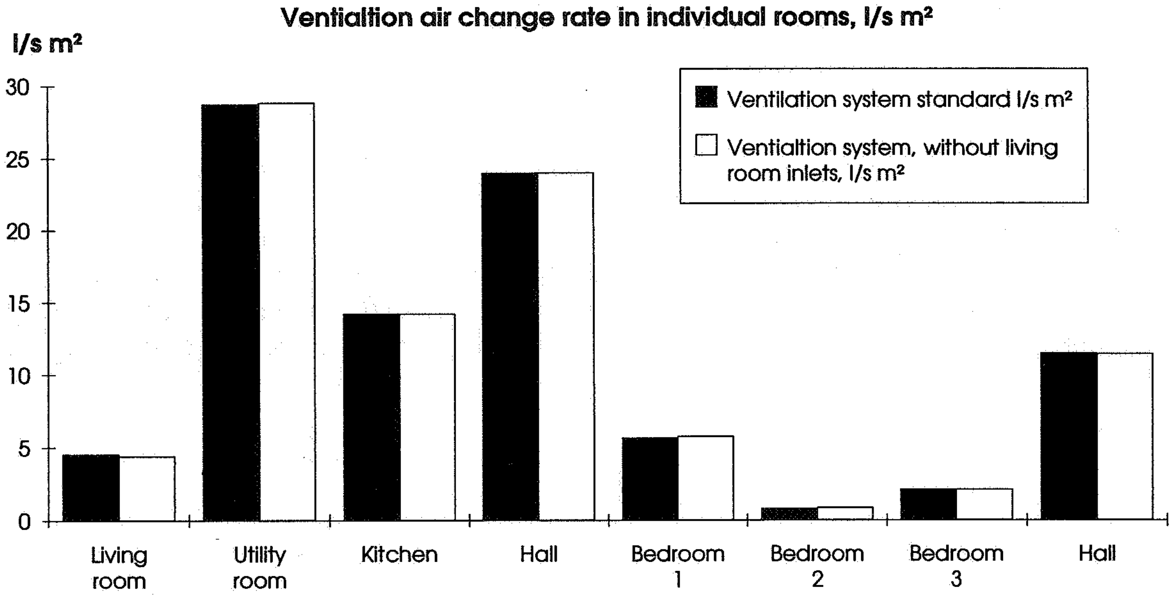

Diagram 3.1.2 **Simulations 2 and 3.** Ventilation air change rates (l/s, m² of floor area) as generated by mechanical ventilation and thermal driving forces. Simulation results based on the standard design of ventilation system and as modified with inlets in the bedrooms only. Rooms as shown in Figure 2.1.

The model simulation calculations assumed that all doors were open between all rooms. If the door to a room is closed, air change will not be affected by thermal drive forces between the rooms. Modern houses generally have no internal doors between the hall, sitting room, general-purpose room and kitchen.

The earlier simulations showed that the outdoor air flow rate to the bedrooms increased substantially, from 2.6 l/s to 5.8 l/s (see Diagrams 2.4.1 and 2.4.2) if the outdoor air inlets in the sitting room were closed. Air change in the rooms is strongly affected if temperature differences between the rooms arise when the internal doors are open (see Table 2.2, Diagram 8.4 and Diagram 8.5). If the internal doors are open, air change in the rooms is affected only insignificantly by whether or not there are outdoor air inlets in the rooms. The proportion of air change between rooms that can be affected by the fans when the doors are open is of the order of 10 %.

Further simulations that we performed showed that if the outdoor air inlets in the sitting room are closed and the total exhaust air flow rate is reduced from 178 m³/h to 88 m³/h, the outdoor air flow rate to the bedroom investigated increases to 2.8 l/s instead of 2.6 l/s (the mean value for the heating season). In other words, the bedrooms can be ventilated just as effectively with only half the exhaust air flow rate if there are outdoor air inlets in the bedrooms only.

4. DWELLING WITH BALANCED MECHANICAL VENTILATION SYSTEM

4.1 General

One of the houses in the investigation, a 1½-storey house having a balanced mechanical ventilation system combined with air heating, has been simulated by the computer program. The house had a relatively conventional layout, with bedrooms and bathroom on the upper floor and with the remaining rooms on the ground floor (see Figure 4.1). The ventilation system, shown in Figure 4.2, is of a type commonly employed in Swedish houses with air heating systems. The house complied with Swedish standards in respect of airtightness.

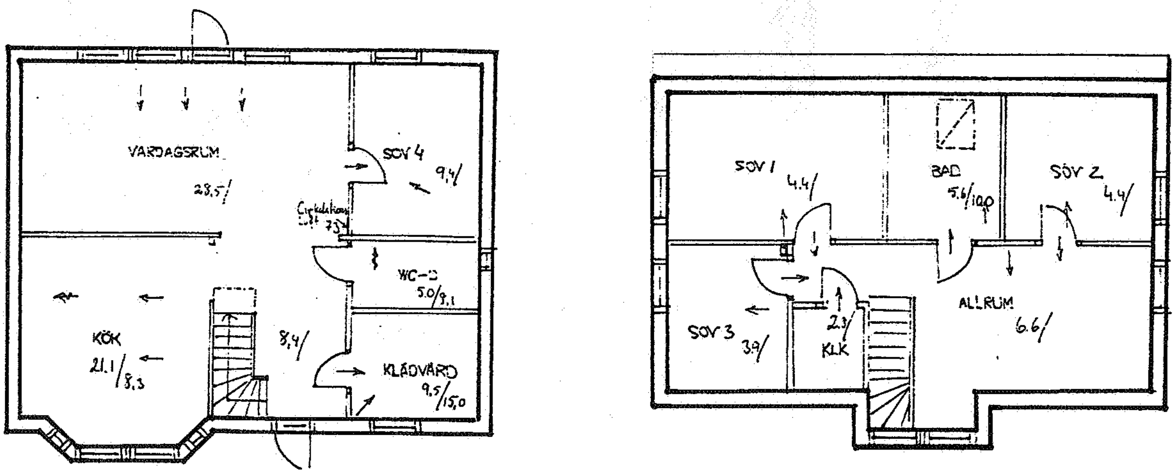

Figure 4.1 A plan of the house, showing positions of inlets and outlets for a balanced ventilation system. Supply and exhaust air flow rates are shown in l/s.

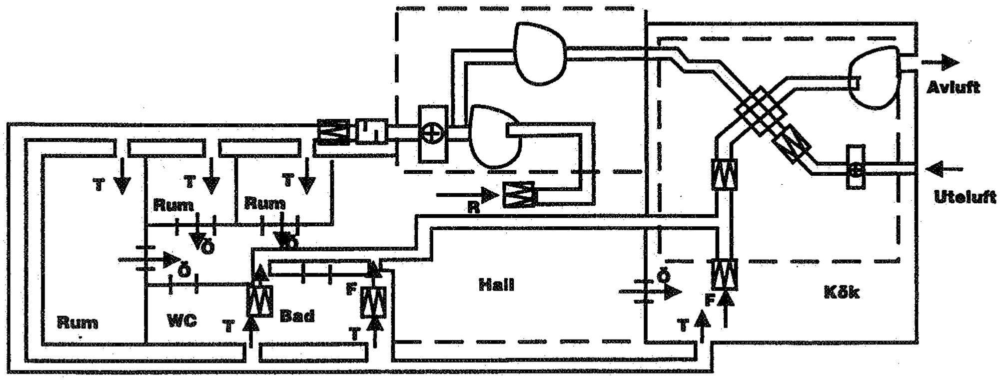

Figure 4.2 Schematic diagram of ventilation system for a single-family house with air heating. R = circulated air, T = supply air, F = exhaust air.

4.2 Input data for simulation

The house's form factors have been taken from wind tunnel investigations (Wiren, 1987). The actual airtightness of the house was measured during the diagnostic tests after construction (2.6 $m^3/h, m^2$). Leakage has been assumed to be distributed as follows: 60 % through the floor/ceiling structure between the ground floor and upper floor, 20 % via penetrations, 10 % around the edges of floors, 5 % around the edges of the ceiling and 5 % through windows. The measured air flow rates were 149 m^3/h of exhaust air flow, 95 m^3/h of outdoor air flow and 300 m^3/h of circulated air flow. Room temperatures and temperature gradients are based on measurements made when checking the performance of the system (see Table 4.2). Climate data for the 1971 reference year for Stockholm has been used when calculating heating and ventilation requirements for a heating season.

Table 4.2 Room temperatures and temperature gradients as used in the simulation.

Room	**Room temperature, °C (at floor level)**	**Temperature gradient, °C/m (vertical)**
Sitting room	20.0	0.5
Utility room	20.0	0.5
Kitchen	20.0	0.5
Hall, ground floor	20.0	0.5
Bedroom 1	19.9	0.5
Bedroom 2	19.7	0.2
Bedroom 3	20.0	0.3
Bedroom 4	19.7	0.4
General-purpose room	20.0	0.3

4.3 Comparison between measured and calculated values

The first computer run was performed in order to compare the actual measured results obtained from tracer gas measurements with the calculated outdoor air flow rate. Simulation was performed for two different outdoor temperatures, with a wind velocity of 1 m/s. The model results were in good agreement with the measured values. The measured outdoor air flow rate to the bedroom was 1.9 l/s, while the theoretical outdoor air flow rates to the bedroom at ambient temperatures of 0 °C and 10 °C were 1.2 l/s and 1.4 l/s respectively.

The computer model calculations were made on the basis of two different ambient temperature levels in order to illustrate the effects of temperature. During the actual measurement period, the ambient temperature varied between 0 °C and 10 °C, while the indoor temperature was 20 °C. As indicated by the model simulation, the outdoor air flow rate to the bedroom increased when the ambient temperature increased, as a result of thermal driving forces.

The low outdoor air flow rate to the room is due to the use of a combined heating and ventilation system (see Figure 4.2). Bedroom 1, which was studied is on the upper floor (see Figure 4.1). Most of its heating demand is met by heat from the ground floor. This means that there must be a lower outdoor air flow rate to the upper floor than to the lower floor in order to prevent excessive temperatures.

4.4 Calculations for a whole heating season

Both the computer simulation and the measured values indicate a low value of outdoor air flow rate. The requirements in the Building Regulations for a ventilation air flow rate of 8 l/s are fulfilled in the master bedroom for only a few hours during the period (see Diagram 4.4.1). One way of improving the outdoor air supply to the bedrooms with this type of ventilation system would be to divide the system up into two zones. The supply air to the bedrooms would then consist solely of outdoor air, while the supply air to other rooms would consist of recirculated air (see Figure 4.4).

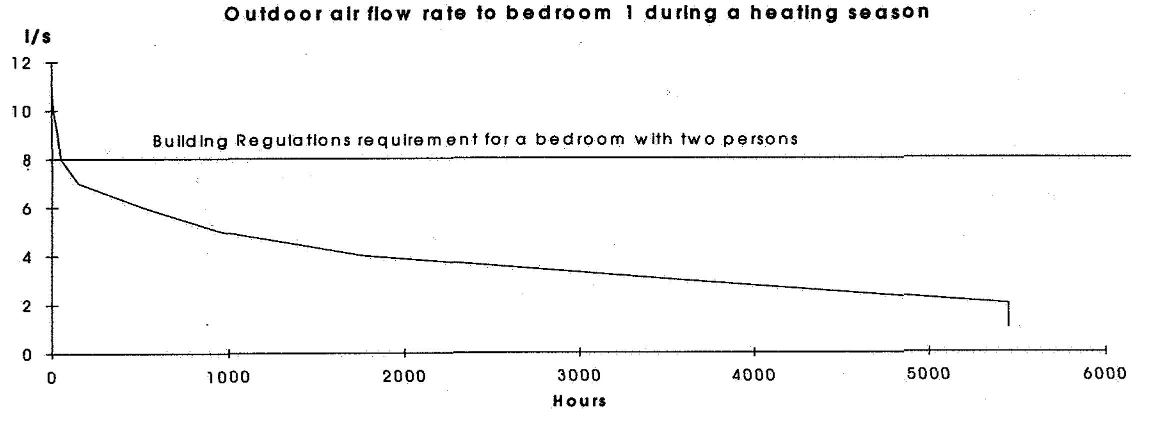

Diagram 4.4.1 Duration diagram for outdoor air flow rate to a two-person bedroom during a heating season. The diagram above is based on an open bedroom door. The average air flow rate during the period is 3.3 l/s (of which 2.3 l/s are infiltration).

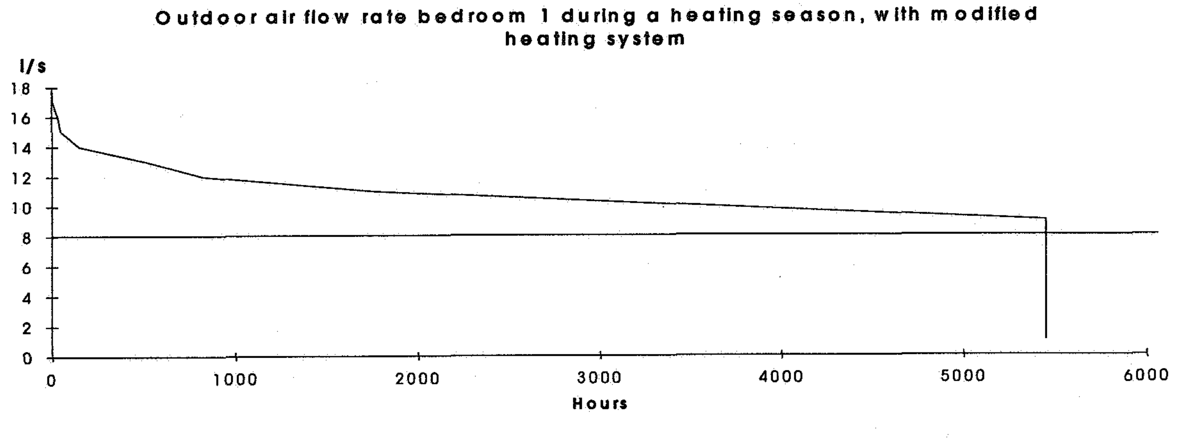

Diagram 4.4.2 Duration diagram of outdoor air flow rate to bedroom 1, with the heating and ventilation system modified as shown in Figure 4.2. The average air flow rate during the period is 10.3 l/s (of which 1.7 l/s are infiltration).

The measured results for the individual bedroom indicate low outdoor air flow rates (1.9 l/s). By modifying the system so that outdoor air is supplied via the bedrooms, the outdoor air flow rate increases from 3.3 l/s to 10.3 l/s, although the total air flow rate to the house remains unaltered. In the existing system, most of the fresh air is supplied to the ground floor.

When the bedrooms are occupied, all outdoor air supplied to the house will benefit those in the rooms. When the bedrooms are unoccupied, the outdoor air will pass through them and benefit the rest of the house.

The result is that better use would be made of the outdoor air than with the present system design. With the same outdoor air flow rate to the house, but with the modified system arrangement, the outdoor air flow rate to bedroom 1 would exceed 8 l/s throughout the entire heating season, as shown in Diagram 4.4.2.

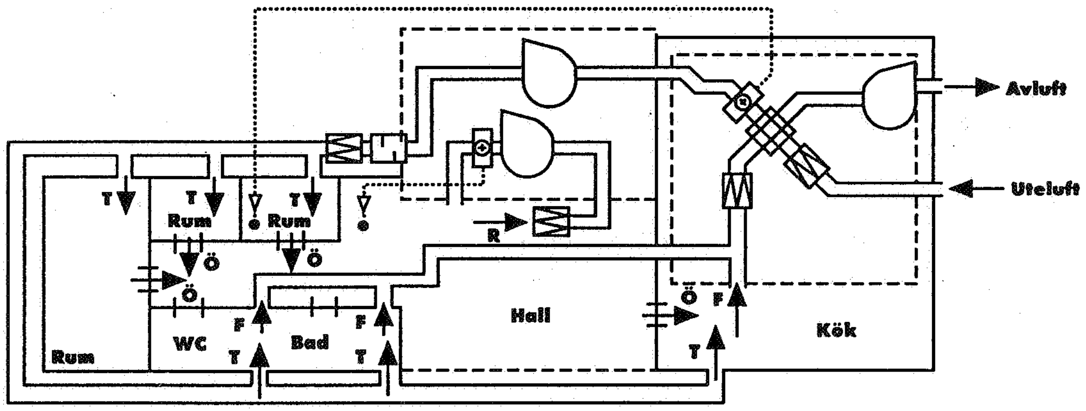

Figure 4.4 Modified air heating system

5. Conclusions

Ventilation systems in Sweden have generally been designed so that outdoor air is supplied to the bedrooms, sitting room and general-purpose room. This means that when the bedrooms are occupied, the fresh air supplied to the sitting room and general-purpose room eventually leaves the house without having ventilated the bedrooms: it is effectively wasted.

If, instead, all the incoming fresh air is supplied via the bedrooms, it will be utilised more effectively. When the bedrooms are unoccupied, the air will flow through them without being contaminated by recirculated air and will then benefit the other rooms.

The general conclusion to be drawn from these measurements and calculations is that all outdoor air should be supplied via the bedrooms, regardless of which of the types of ventilation systems investigated is used. This would mean that the outdoor air flow rate to the bedrooms will increase (c.f. Diagrams 2.4.1 and 2.4.2 and Diagrams 4.4.1 and 4.4.2). Diagram 3.1.2 shows that the incoming fresh air will benefit the rest of the house when the bedrooms are unoccupied.

If it is felt that the quantity of outdoor air supplied by the present system design is sufficient, the proportion of outdoor air can be reduced using the suggested system design. As described in Section 3.1, the outdoor air flow rate can be reduced by 50 % in a system in which outdoor air is supplied only to the bedrooms, and yet still provide the same quantity of outdoor air to the bedrooms. A reduction in outdoor air flow rate will result in reduced energy use and a quieter system. When the building is new increased ventilation might be necessary during 0,5-1 years to remove emissions from new serface materials.One important condition is that there are no harmful emissions from building materials and furniture.

Maintenance of a ventilation system in a house with lowered ventialtion rate is very important, as the marginal to too low ventilation rate is reduced.

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Efficient Work Environment Ventilation

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***National Institute of Occupational Health, Solna, Sweden**
****Finnish Institute of Occupational Health, Vantaa, Finland** **Air Dehumidification by Absorptive and Evaporative Cooling**

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Abstract

A breakthrough in ventilation research was made once it was realized that ventilation principles based on mixed flow patterns are not optimal and that further energy savings can be achieved if an alternative technique could be developed. Several researchers, particularly in the Nordic countries, have shown by theoretical studies that replacing mixed ventilation flow by displacement flow increases ventilation efficiency. This also results in decreased air supply volumes and thus decreased energy requirements. In addition, lower air velocities may reduce problems of comfort and noise.
Soon, however, practical experience showed that a displacement ventilation system must be very carefully designed in order to work as theoretically expected. Not all systems were successful initially and difficulties were encountered in implementing the new technology.

This paper discusses the design basics for practical displacement ventilation systems. An example is taken from a plastics industry, where horizontal displacement ventilation is applied to a real work environment situation. Field measurements are made. Results of the field measurements, carried out in the plastics industry, are surprisingly good. For a channel flow ventilation situation, an air change efficiency of around 80 % has been achieved. It is also shown that worker exposure to styrene vapor can be kept within acceptable limits. The results are affected by the room geometry and the nature of the air supply. The energy gains from high air change efficiencies are discussed, particularly with regard to cold climates where there are large heating requirements.

Introduction

Displacement ventilation systems have become popular in a growing number of applications, because of their efficient contaminant removal and high air change efficiency. Conventional mixing ventilation can never be more efficient than 50%, in terms of air change efficiency. This is one of the main reasons for the development of displacement ventilation systems [1]. Another is the increasing demands for better thermal comfort conditions. In processes where contaminant emission and heat generation are not strongly coupled, momentum driven supply air can flow horizontally through a room [2]. Laboratory work for a better understanding of this type of displaced flow has been reported [3].The distribution of the supply air has been shown here to have a great influence on the flow pattern in a ventilated space.

Successful industrial applications of horizontal displacement ventilation have been recently reported [4,5]. These measurements are discussed further here and compared to numerical simulations of the same industrial work place situation. The process described is the manufacture of polyester products reinforced with glass-fibre. With mixing ventilation, this process requires extremely high air change rates to keep the styrene concentrations in the lamination hall within acceptable limits. This paper reports the results and discusses the research efforts for better ventilation principles in this type of industrial application.

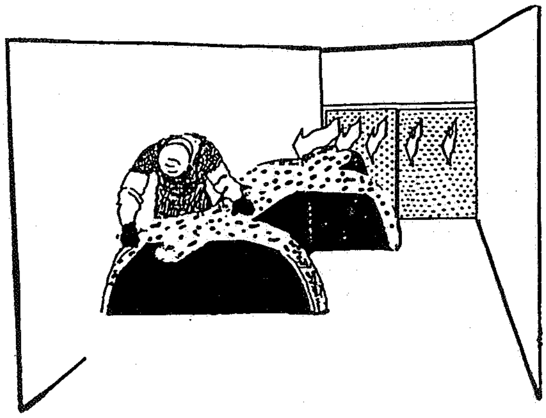


Figure 1. Worker exposure to styrene during lamination with polyester shows great variation with worker position and the principle of ventilation air transport.

Numerical simulations

Numerical simulations were used to analyze the air flow and concentration patterns. The governing equations to describe the three-dimensional transport mechanisms are the equations for continuity, momentum, concentration and the local mean age of air. The k-ε turbulence model, Jones and Launder [6], uses two additional equations, one for the kinetic energy of turbulence k and one for the dissipation of this energy ε. Both these equations are of the same type as those mentioned earlier. In general form, the partial differential equations are:

$$\frac{\partial(\rho u_i \phi)}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\Gamma_\phi \frac{\partial \phi}{\partial x_i}\right) + S_\phi \qquad (1)$$

where S_ϕ is the source term [7]. In these calculations, the dependent variable ϕ takes the forms: u, v, w, c, $\bar{\tau}_p$, k, ε. For the continuity equation, $\phi = 1$. The SIMPLE algorithm [8] and staggered grid arrangements were used to give discrete solutions of the equations. The discreet equations for each node point were derived by hybrid upwind central differencing [9]. A stable iterative solving of the equations was thus achieved. All calculations were performed in three dimensions on a 40*40*36 grid with constant spacing. The solution time on a Risc-6000 work station was around three hours per dependent variable ϕ.

Boundary conditions

All surfaces were assumed to be adiabatic (zero heat flux). Zero-gradient boundary conditions were assumed for incoming and outgoing air. The boundary conditions for the k-ε turbulence model used were given for isotropic turbulence. Wall functions of the conventional type were used. The ventilation flow rate was 1,9 m^3/s. In the numerical simulations, the styrene emission rate was set to 60 mg/s, and the location of the source was fixed. The laminar and turbulent Prandtl numbers (Schmidt numbers for concentration) σ and σ_ϕ were set to 0,72 and 0,90 respectively in all calculations. The diffusion coefficient, Γ_Φ, took the form:

$$\Gamma_\Phi = \frac{\mu_t}{\sigma_\phi} + \frac{\mu}{\sigma} \tag{2}$$

where μ and μ_t are the laminar and the turbulent dynamic viscosities [6]. A steady-state method was used for the numerical prediction of the mean age of air [7]. The final differential equation for the local mean age of air $\bar{\tau}_p$ includes the density ρ as the source term:

$$\frac{\partial(\rho u_i \bar{\tau}_p)}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\Gamma_{\bar{\tau}_p} \frac{\partial \bar{\tau}_p}{\partial x_i}\right) + \rho \tag{3}$$

The boundary condition is $\bar{\tau}_p = 0$ at the inlet.

Measurements in the work environment

Some results from the plastics industry [4,5] are compared with the numerical simulations. The most useful result from the measurements was the extremely high air change efficiency found in a channel-shaped lamination room with horizontal displacement ventilation. The results also indicated how much air is required for ventilation if a mixing ventilation principle is used. These figures are compared to corresponding figures when horizontal displacement ventilation is used. A pattern of the channel flow is described by measured vertical velocity profiles.

Simulated air flows

The lamination chamber, including the air terminal for the one-way orientation of the ventilation flow, has been described [4]. All the dimensions are straightforward and easy to model in a computer program. The mold , however, is more complicated and its shape had to be simplified for the computer program. Figure 2 shows a plan view (x-y) of the simulated 6-metre long, simplified, double-cross mold model. The whole chamber width was open for supply and exhaust air, see Figures 1 and 2.

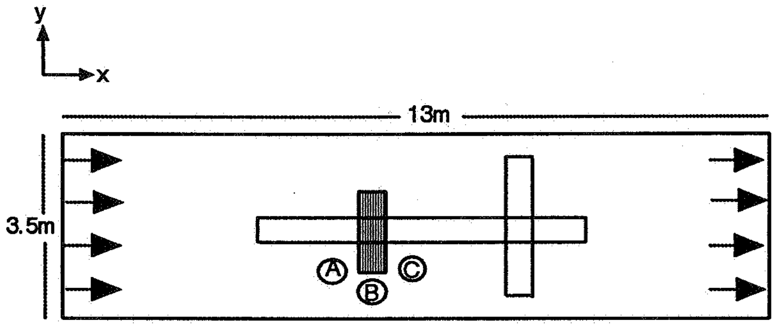


Figure 2. Plan view of the lamination hall, where both industrial measurements and numerical calculations were performed. Points A, B and C are worker positions. The styrene concentration was estimated at these points in the breathing zone 1,5 m above the floor. The simulated styrene emission (60 mg/s) was equally distributed over the striped source area.

The air supply velocity u_0 for both measurements and simulations was 0,3 m/s. The Reynolds number for the flow, Re, is based on the active (open) air supply area A and takes the form:

$$Re = \frac{u_0\sqrt{A}}{\nu} \tag{4}$$

where ν is the kinematic viscosity of the flow. For the displacement flow shown here, a Reynolds number of 55200 was used.

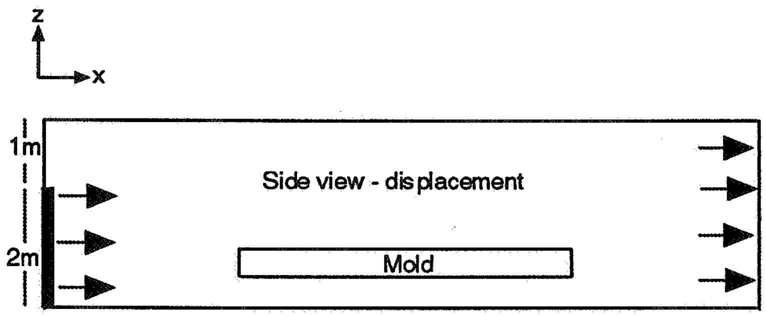


Figure 3. Elevation of lamination hall and mold. In the calculations, the mold geometry was simplified. In reality, the upper corners were a bit curved [4].

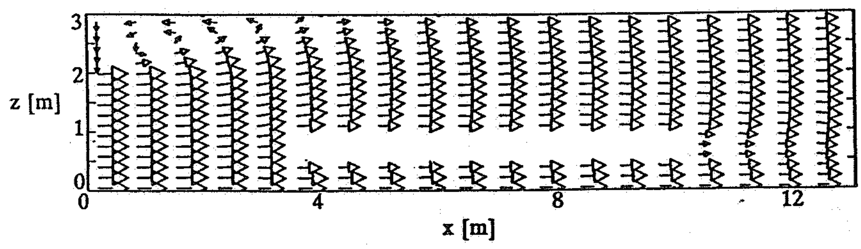
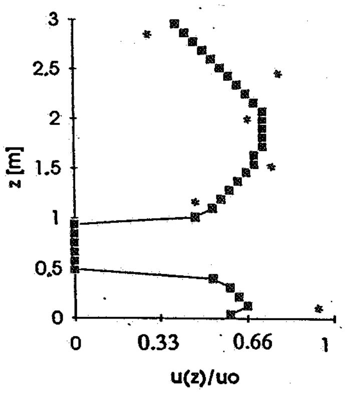


Figure 4. Simulated air flow structure in the lamination hall with horizontal displacement ventilation. Recirculation occurred only in a small area above the air supply. Center-plane crossing the mold; y =1,75 m.



* = measurements [4]

Figure 5. Numerically calculated vertical velocity profile over the mold compared to the measured profile (x = 7 m, y = 1,75 m). Supply velocity u_0 = 0,3 m/s. The velocity is blocked by the mold from z = 0,5 m to z = 1,0 m.

Simulated mixing ventilation as reference

Mixing ventilation was used as a reference and compared to the displacement flow. The comparison was done both for the work place and in the numerical calculations. The mixing model (plant) used was identical to the displacement model, except for the size and shape of the air supply inlet and exhaust outlet. Both were reduced here to a 0,35-m-high slit extending the full width of the room. A constant ventilation flow rate of 1,9 m^3/s was used. This gave supply and exhaust velocities almost six times greater than those in the displacement flow.

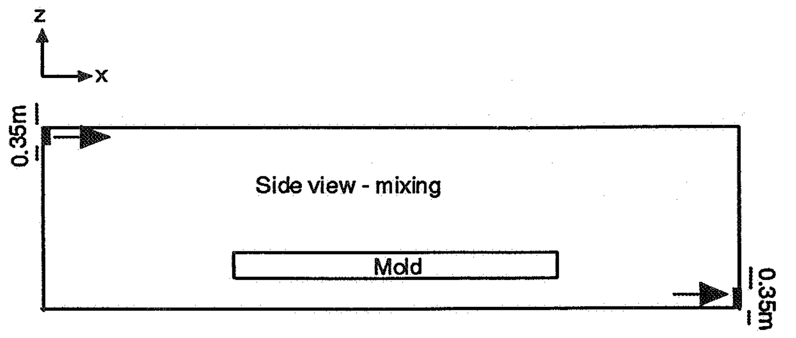


Figure 6. The mixing ventilation reference simulations were also performed in the 13-m-long chamber. The air supply and exhaust velocities were higher than for the displaced flow.

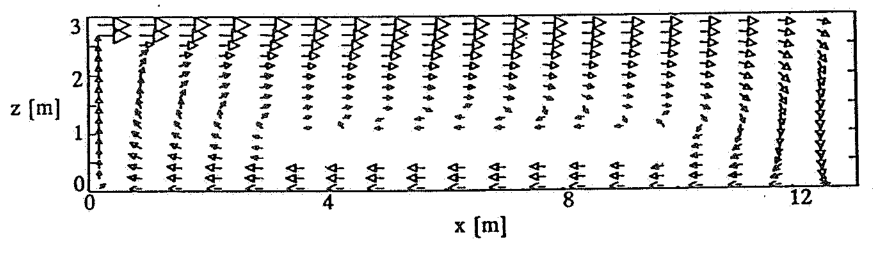


Figure 7. Simulated air flow structure in the lamination hall with mixing ventilation. Recirculation occurs both on the upper and lower side of the mold. Center-plane crossing the mold; y =1,75 m.

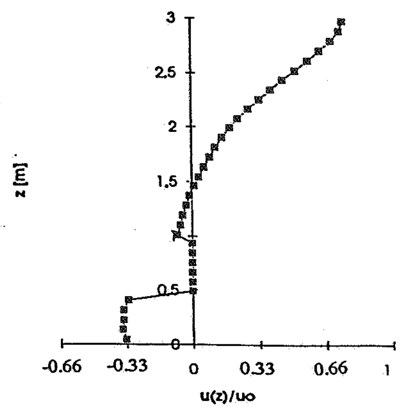


Figure 8. Simulated vertical velocity profile through the center line of the styrene-emitting area of the mold (striped area in Figure 2). A Reynolds number of 132000 was used.

Ventilation efficiency

A tracer gas pulse method was used to measure the mean age of air in the lamination hall [5]. In the simulations, equation (3) was used to calculate the local mean age of air at every grid point in the room. The room mean age, $\langle\bar{\tau}\rangle$, was then calculated as the average value of the local mean ages. For the air change efficiency, ε_a, the following definition was used:

$$\epsilon_a = \frac{\tau_n}{2\langle\bar{\tau}\rangle} 100\% \tag{5}$$

where $\tau_n = V/q$ (=room volume/ventilation flow rate) is the nominal time constant of the room [10].

The contaminant removal effectiveness $\langle\varepsilon\rangle$ is defined as the ratio between the steady-state contaminant concentration in the exhaust outlet and the steady-state average contaminant concentration in the room [11,12]:

$$\langle\epsilon\rangle = \frac{c_e(\infty)}{\langle c(\infty)\rangle} \left(= \frac{\tau_n}{\tau_e}\right) \tag{6}$$

Equation (6) shows that $\langle\varepsilon\rangle$ can also be expressed as the ratio between the nominal time constant of the room, τ_n, and the mean residence time of the contaminants, τ_e [13]. It is a measure of how quickly the contaminants are removed from the room. A comparison of measured and simulated ventilation efficiencies is given in Table 1.

Table 1. Measured and simulated ventilation efficiency measures.

	Ventilation efficiency			
	Air change efficiency, ε_a %		Contaminant removal effectiveness, $\langle\varepsilon\rangle$	
	Displacement	Mixing	Displacement	Mixing
Simulated	85	41	1,36	0,54
Measured	81	51*	1,90	-

* The measured industrial plant for mixing ventilation [14] was different from the simulated mixing reference plant. An air change efficiency of over 50 % indicates some degree of displacement flow.

Table 1 shows that the agreement between simulated and measured air change efficiencies in the displacement chamber (Figures 1-3) is good. The measured contaminant removal effectiveness value is higher than the simulated. A probable reason for the deviation is the location of the contaminant source, which was 0,75 m closer to the exhaust outlet during measurement than in the simulations.

For mixing ventilation, the measured industrial plant was not identical to the simulated plant. The ventilation systems, including air terminals, were also different. The simulated system, shown in Figure 6, was inefficient. The supply air passed through the room without contacting the contaminants. The recirculated air came in close contact with the contaminants and thus blew contaminants the wrong way. This lead to a contaminant removal effectiveness of only 0,54.

By far most efficient ventilation was achieved with the channel-flow arrangement with horizontal displacement ventilation. A buoyancy influence from the exothermal lamination process, not considered in the simulations, could be one reason for the small difference in measured (81 %) and simulated (85 %) air change efficiencies.

Simulated exposure rates with different ventilation principles

Measured exposure rates are not directly dealt with in this report. The reason is that the appropriate measured data were not available. It was difficult to generate steady-state conditions in the lamination hall, and difficult and expensive to measure at many points simultaneously. A constant, time-independent styrene-emission from the mold is difficult to arrange. The comparison of exposure rates here is based on numerical simulations only. Constant boundary conditions are easy to arrange. So it is possible to have a fair comparison of the effects of different air flow principles on the exposure rates.

Table 2. Numerically calculated exposure rates with different ventilation principles at different worker place locations A, B and C. All three locations are 0,3 m from the mold and 1,5 m above floor (see Figure 2). The upper surface of the mold i.e. the styrene source, is 1 m above floor level.

Worker position	Simulated exposure rates [mg/m^3]	
	Displacement ventilation	Mixing ventilation
A	< 0,1	212
B	0,2	150
C	18	143

Displacement ventilation gave lower exposure rates than mixing ventilation, even at worker position C. This is because of a low vertical concentration diffusion, up to the breathing zone, with this type of ventilation flow. At the source level, 1m above floor level, dangerous concentrations occur. Table 2 shows that the two types of ventilation can be expected to give very different exposure rates.

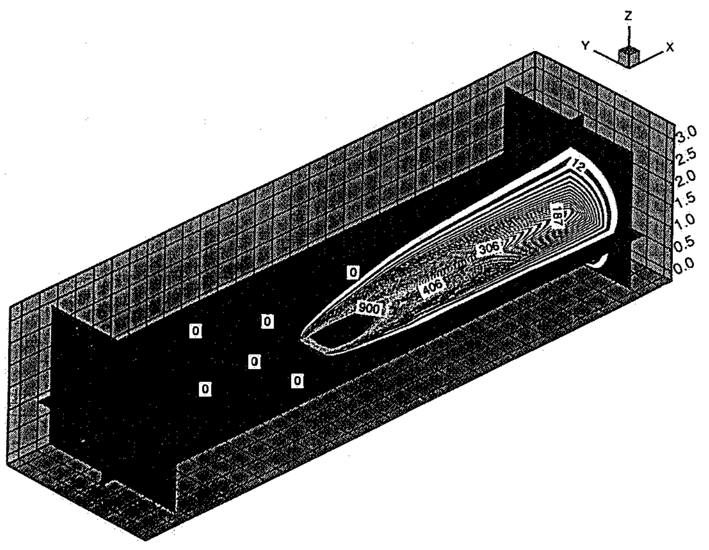


Figure 9. Styrene exposure rates [mg/m^3] with horizontal displacement ventilation. In the zero-concentration zones the exposure is guaranteed to be less than 6 mg/m^3. Fence plot showing center-planes for x-y, x-z and exhaust outlet plane for y-z.

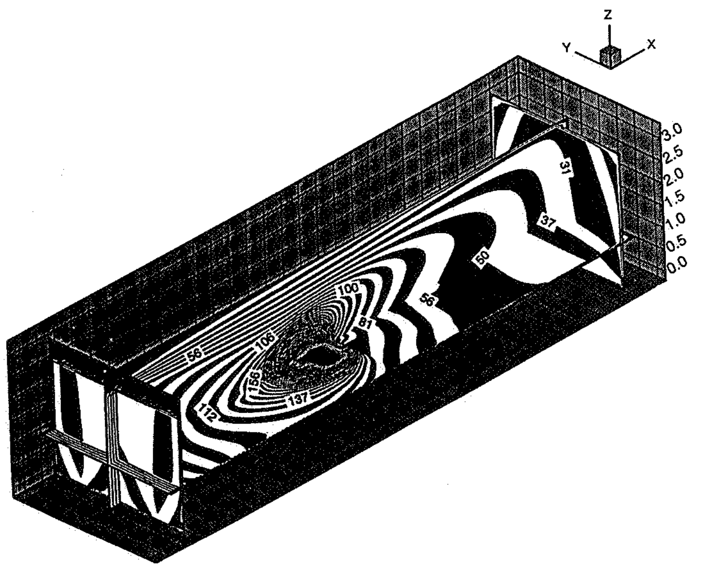


Figure 10. Styrene exposure rates [mg/m^3] with mixing ventilation. The value (value label) positions are the same as for the previous figure. This makes it possible to compare exposure rates with the two ventilation principles.

Energy and health aspects

For mixing ventilation, there is a well-known relation between the contaminant generation $\dot{m}_c$ and the ventilation flow rate q, which gives the contaminant concentration c in the room.

$$c = \frac{\dot{m}_c}{q} \tag{7}$$

There is a great influence of the ventilation efficiency ε and the spatial positions (x,y,z) on the styrene exposure rates. This leads to a new principal equation for the local styrene concentration c_p in the room:

$$c_p = \frac{\dot{m}_c \cdot f(\epsilon, p(x,y,z))}{q} \tag{8}$$

where the ventilation efficiency ε is a measure of the stored styrene quantity in the room, and p(x,y,z) indicates the spread of this quantity. The latter is a position factor that depends on the contaminant source position and the positions of the ventilation air supply and exhaust. Equation (8) states that in all parts of the room the concentrations are directly influenced by the total ventilation flow rate q. The term $\dot{m}_c \cdot f(\varepsilon,p(x,y,z))$ determines the level and the distribution of the local room concentrations.

For worker position C in Table 2, the mixing ventilation flow rate has to be increased by a factor of 8 to bring the exposure rate down to a level achieved by horizontal displacement ventilation (18 mg/m^3). Air consumption data from measurements [4] and concentration extrapolation show that 4-5 times more air has to be used with mixing ventilation. The differences in air change efficiency between displacement and mixing were: for the simulations 85% - 41% = 44%; and for measurements 81% - 51% = 30%, Table 1. The energy E, required to heat ventilation air is given by:

$$E = q \cdot \rho \cdot \Delta t \cdot c_p \cdot \Delta T \tag{9}$$

where
q = ventilation flow rate
ρ = density of air
Δt = heating duration
c_p = specific thermal capacity
ΔT = temperature rise of the heated air.

It has been shown that efficient ventilation means low ventilation flow rates, q. Equation (9) shows that this also means low energy requirements, E. In the equation no account is taken of the heat losses during heating and transport, the efficiency of the heater, or the energy used by the fans. Countries with high heating requirements for ventilation air normally also have a long heating season, which affects Δt, and a large indoor-outdoor temperature difference, which affects ΔT. To compensate for this, ventilation flow rates are often lowered. In this situation, efficient ventilation should be introduced for health reasons. Also, knowledge about correct work practice and favorable worker positions are important. Recently the exposure limits for styrene have been lowered; the eight-hour exposure limit in Sweden and Finland is today 20 ppm (90 mg/m^3). This once again puts higher demands on future ventilation systems.

Conclusions

Because of large variations in the size of manufactured products and the movable nature of work activities, general ventilation is frequently used for controlling airborne styrene. High exposure rates can be avoided by using horizontal displacement ventilation that is properly positioned in the room. Factors that need more study, because of their influence on the ventilation air flow, include room geometries, air supply and exhaust arrangements, and thermal loads. Concentrated air supply diffusers [15] can probably reduce costs further.

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Air Dehumidification by Absorptive and Evaporative Cooling

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Air dehumidification by absorptive and evaporative cooling

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Synopsis

Especially in modern buildings with small capacity of humidity storage it is necessary to reduce the humidity in the supply air. Normally a refrigeration system containing CFC's is used. There are some alternative fluids available, but mostly they show a high global warming potential. These systems all need electrical energy to be driven and therefore it is necessary to consider other possibilities with alternative systems.

The most promissing systems are sorptive systems which are used now in open cycles. In these systems the air is dehumidified by a liquid sorbens and cooled indirectly by evaporating water in an open circuit. In this paper a design of an open cycle liquid desiccant system is shown as well as two possible system configurations. Very interesting possibilities for the regeneration are given in using low temperature energy and also solar radiation in sunny areas.

1. Introduction

Heating, ventilating and air conditioning (HAVC) systems are build to process outdoor air to a special indoor air condition. The demand of the air quality depends on the kind of building. In an industrial building the quality of the products is very important, but in an office building the thermal comfort of the employees must be guaranteed. Basically the parameters temperature and humidity can be changed by special components of a HVAC-system.

Especially in modern office buildings with small capacity of humidity storage it is necessary to reduce the humidity in the supply air. The classical way to dehumidify the outdoor air is using a refrigeration system. Everybody knows the problems and the discussions about the refrigerants. For that it is very important to find alternative components to dehumidify the air.

An interesting technique to dehumidify air is using hygroscopical solid or liquid substances. The ad- respectively absorption technology used in HVAC-systems is nowadays a good enlargement to the existing refrigeration systems but particulary by using liquid desiccants a high demand of research work is necessary. In this paper an open cycle liquid desiccant dehumidification system combined with an evaporative cooling system is described.

2. Methodes of dehumidification

In the HVAC technology the following dehumidification systems are usual:

❶ Condensation on cold surfaces of chillers or water droplets.

❷ Desiccating by the contact with hygroscopic materials.

To ❶:

This kind of dehumidification is the frequent applied technology. To get condensate temperatures below the dew point of the dehumidifying air are necessary. These low temperatures can be realised by an evaporating refrigerant (direct evaporater) or by cold water (water cooled chiller).

The cold water is made by a refrigeration plant and its usual preliminary temperatur is approximately 6°C. The refrigeration systems are formed basically in the three following groups:

- Compression-refrigeration system
- Absorption-refrigeration system
- Steam jet-refrigeration system

The employment of steam jet-refrigerators in HVAC-systems is secondary. The both common refrigeration systems are schematically shown in figure 1 and 2. Further the process is presented in a lg p, h respectively a lg p, 1/T-diagram.

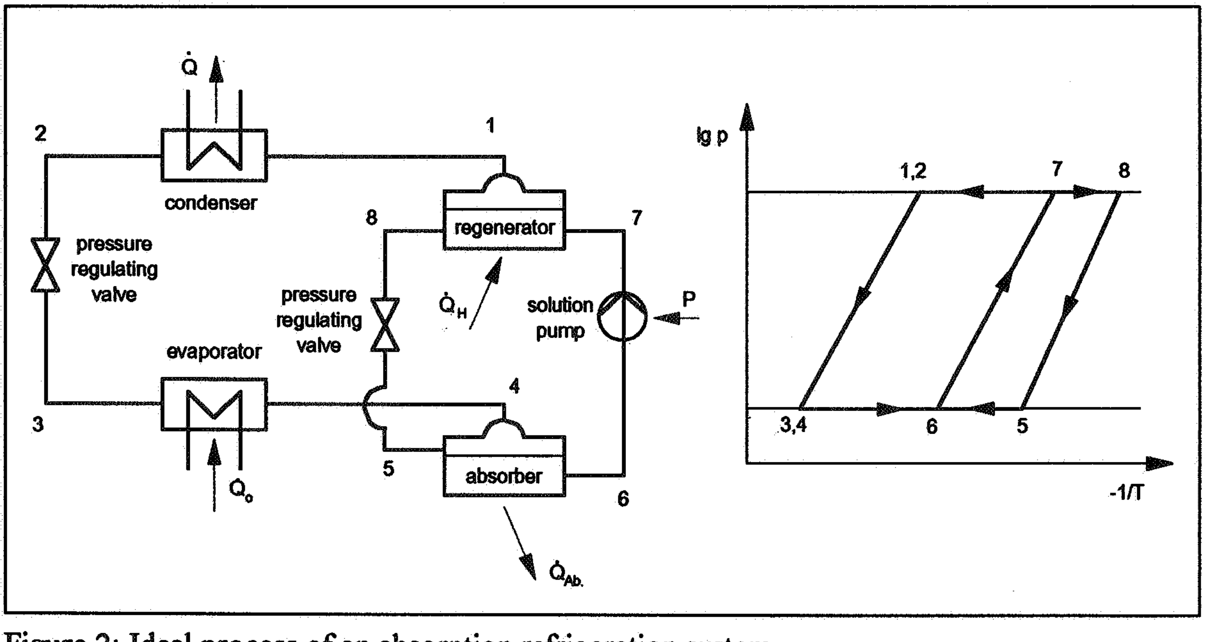

Figure 2: Ideal process of an absorption refrigeration system.

Disadvantageous by using this kind of systems is:

- Poor controllability due to constant water temperature in the cold water chiller.
- Disadvantageous high energy demand for the refrigeration.
  - ⇨ For the air conditioning it is not always necessary to have preliminary water temperatures of 6°C.
  - ⇨ Energy demand for the compressor.
- Reheating of the dehumidified air is often necessary.
- Only limited usability of low temperature heat by the refrigeration systems.

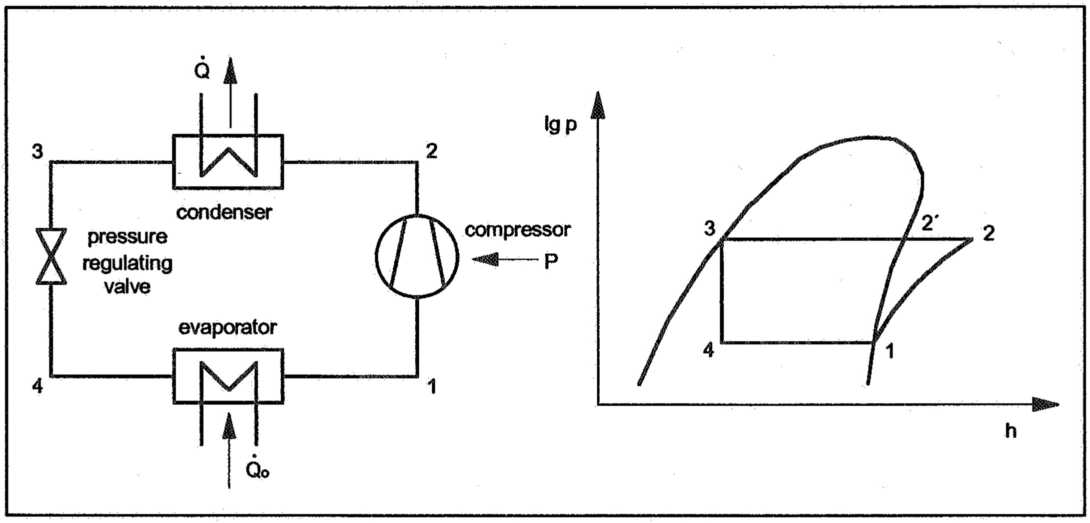

Figure 1: Ideal process of a compression refrigeration system.

By increasing the preliminary water temperature it is possible to reduce the energy demand for preparing the cooling water with refrigeration systems. By increasing the water temperature of about 1 K the improvement of the coefficient of performance (COP) is approximately 3 %.

To ❷:

Desiccating by contact with hygroscopic materials are distingushed by the kind of the used materials:

- solid hygroscopic materials and
- liquid hygroscopic materials.

There is a great variety of solid materials which can be used for dehumidifying air. Active carbon, active alumina, silicagel, zeolithes as well as hygroscopic salts are mostly used for technical drying. Silicagel and hygroscopic salts are privileged for the air dehumidifier. Continuous working rotary wheels or discontinuous working packed beds are equipped with such solid desiccants. In this paper the solid desiccants and the design of belonging plants is not described because the main emphasis is the use of liquid desiccants.

One of the first liquid desiccants which was used for the dehumidification of air was triethylenglycol. Due to the high vapor pressure the application of this desiccant in open cycle systems is unfit because there are high losses of desiccants which have a negative influence to the environment. However open desiccant cycles working with solutions of lithium chloride- respectively calcium chloride are partly successful in practice or investigated in promising research projects.

3. Design of open absorption systems

The described system is an absorptive system for air conditioning. The outdoor air is dehumidified by a liquid desiccant and cooled indirect by evaporation of water in an open cycle. The following figures show schematically possibilities of different processes. Figure 3 presents the arrangement of an open liquid desiccative and evaporative system with the main belonging components. It describes a system working with 100% outdoor air.

Warm humid outdoor air passing the salt solution is dehumidified in a crossflow absorber. To

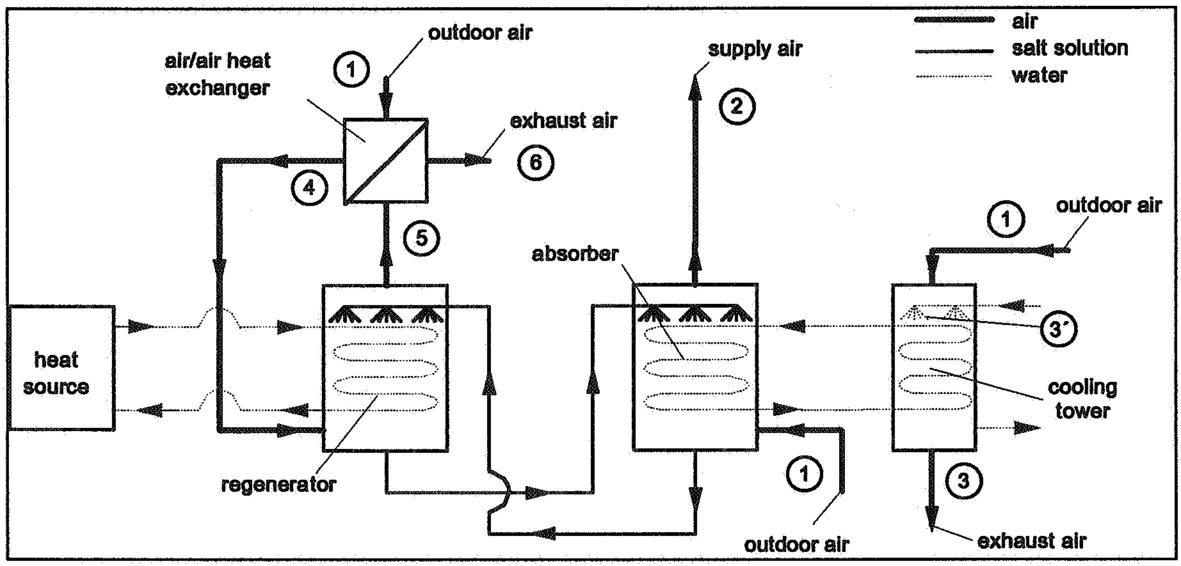

Figure 3: Open cycle liquid desiccant cooling system (100% outdoor air).

get a constant dehumification performance it is necessary to take away the absorption heat by a cooling tower.

By the absorption of steam the salt concentration of the solution is decreasing so the ability of the absorption is reduced. To achieve a constant concentration of the salt solution a regeneration after the absorption is necessary. This is happend in the regenerator where the solution is heated and the water evaporates. The steam and the regeneration air (outdoor air) pass an air to air heat exchanger and leave to the environment.

The change of air conditions for the case of 100% outdoor air is sketched in a Mollier-h,x-diagram in figure 4. Point ① describes the outdoor air situation on a warm humid summerday. The change of condition from ① to ② shows the dehumidification in the absorber with simultaneously cooling by an indirect cooling tower. So the supply air ② has a lower temperatur as well as a lower water content than the outdoor air.

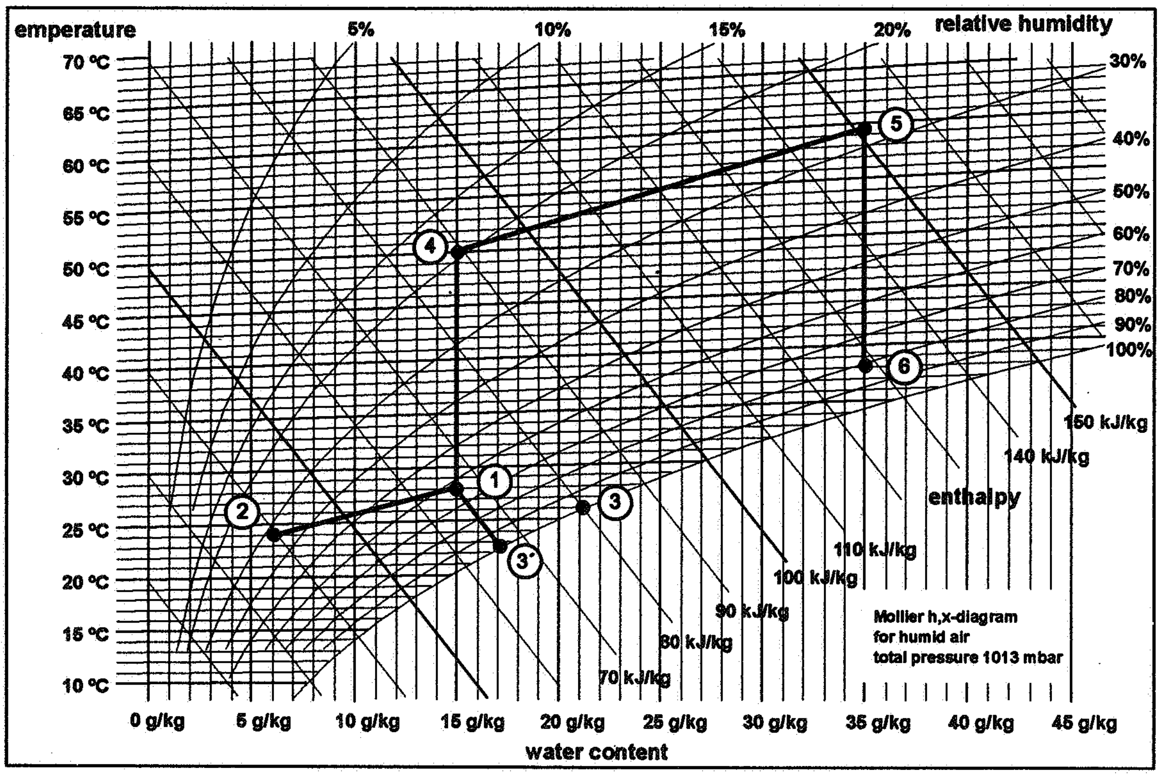

Figure 4: Change of air condition in a h,x-diagram (100% outdoor air).

Moreover in figure 4 also the change of air conditions in the cooling tower as well as the regeneration process are skeched. Such as already mentioned the outdoor air is humidified in the cooling tower by the principle of evaporation (① → ③′). With assumption of a complete wet mode of operation in the cooling tower it is possible to reach point ③. The outdoor air which is used for the regeneration is warmed up from ① → ④. By heating the salt solution the water evaporates in the regenerator (④ → ⑤) and leaves together the system with the air after the heat exchanger (⑤ → ⑥).

The air conditioning of buildings is normally done with a part of return air. It means, that a part of the exhaust air is mixed with the conditioned outdoor air in a mixing chamber. Such air conditioning process is shown in figure 5. The main difference in the absorption process to the system which is explained before is after point ②. In this process it is necessary to dehumidify the outdoor air to a low water content because there is a mixing of humid return ⑦ air with already conditioned outdoor air ②.

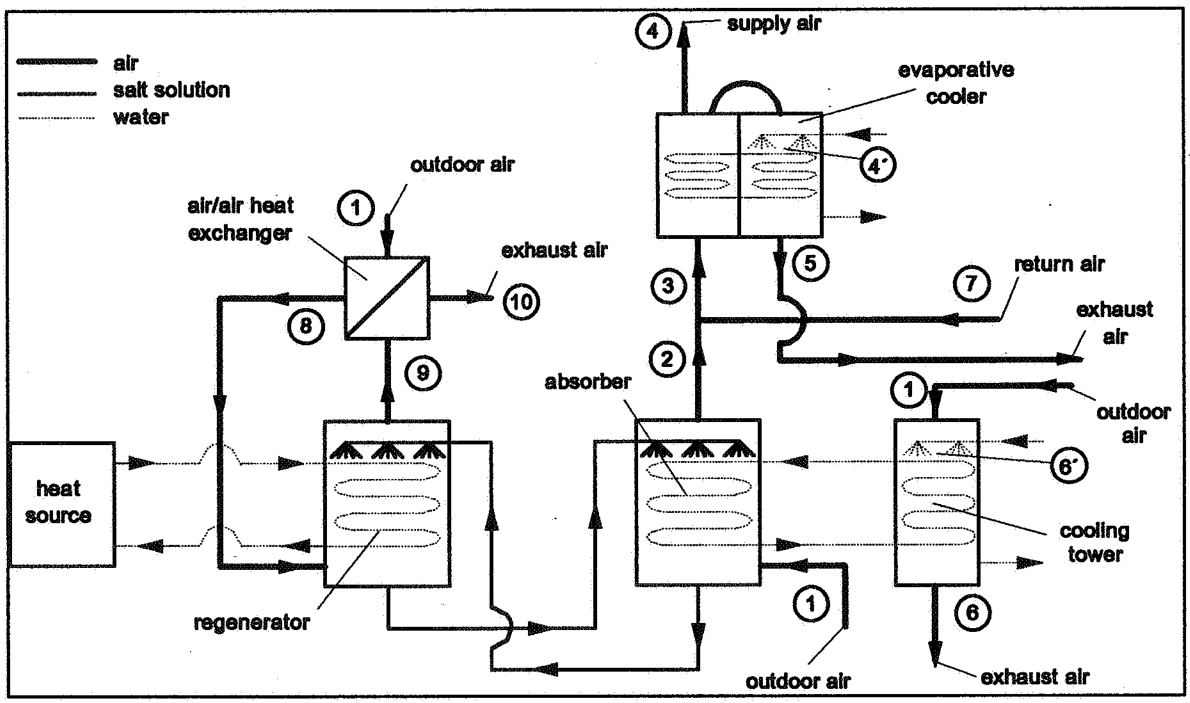

Figure 5: Open cycle liquid desiccant cooling system (With a part of return air).

By mixing the return air ⑦ and outdoor air ② the temperature in point ③ is increasing so that it is necessary to cool the mixed air before entering the room. The cooling of this air is done by an indirect evaporative cooler which operates with a part of the supply air. The

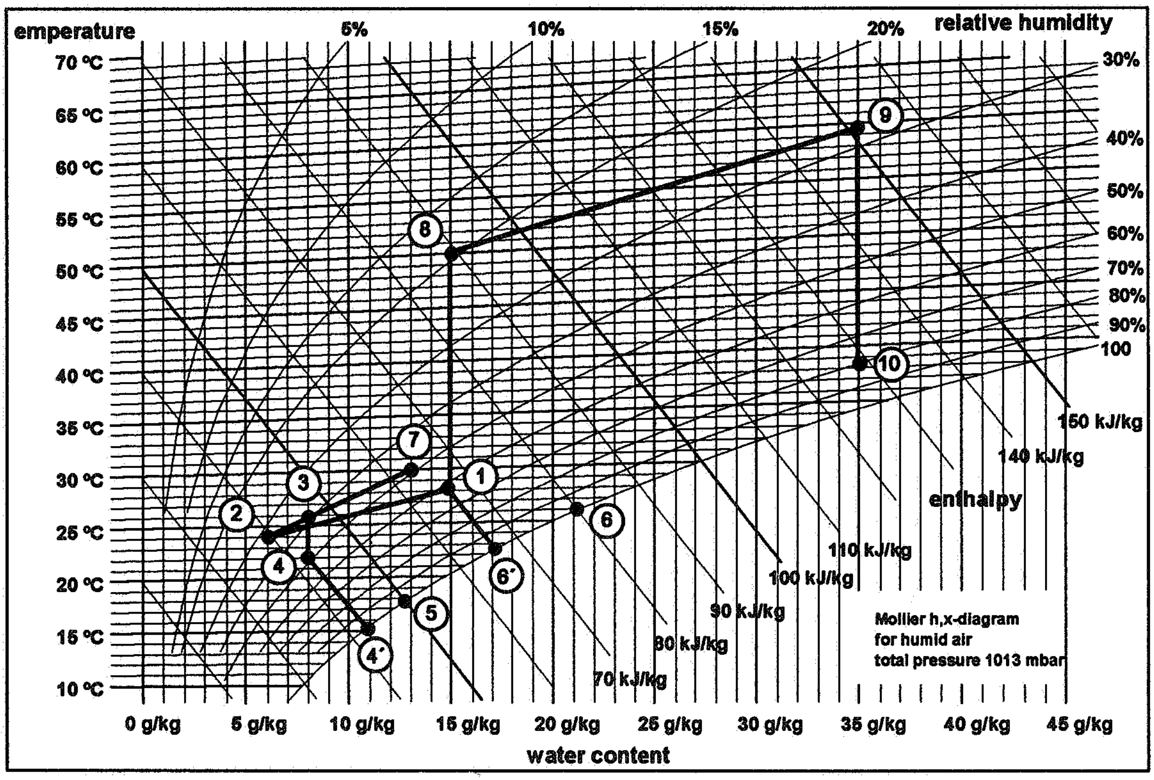

Figure 6: Change of air condition in a h,x-diagram (With a part of return air).

change of the mixed air condition from ③ → ⑤ is sketched in figure 6 on the condition of the complete wet operation of the cooler to reach point ⑤.

3.1 Dehumidification by the "Kathabar"-system

The "Kathabar"-sytem is used in different fields of air conditioning. The desiccant (the so-called Kathene solution) of this system is a 40 % Lithium chloride solution. Figure 7 shows a schematic diagram of the system with the main components: a heating- and a cooling coil, a conditioner, a regenerator, a sump and a pumping unit.

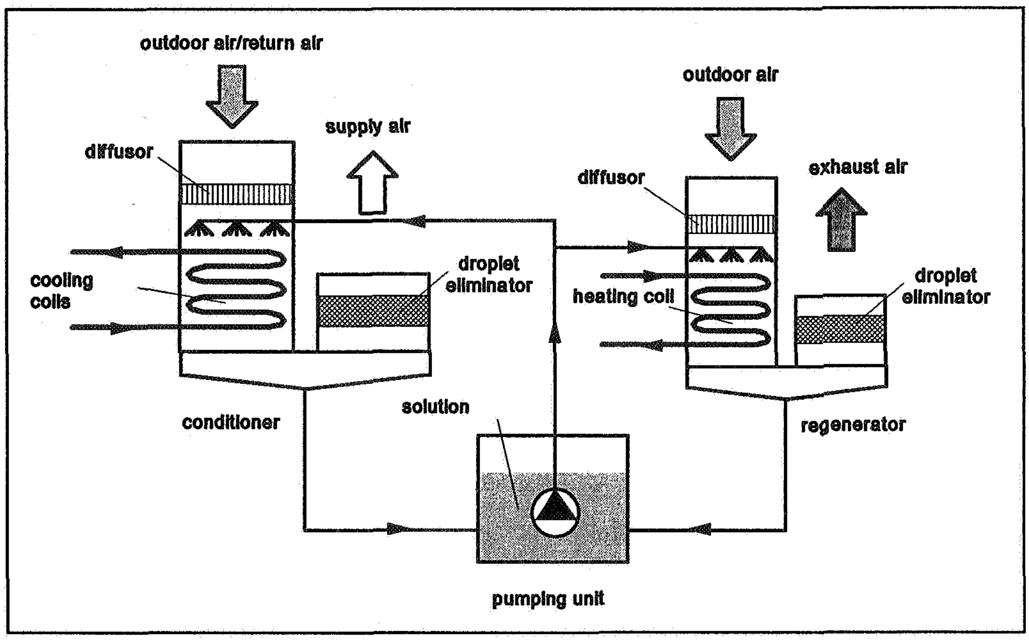

Figure 7: Schematic diagram of the "Kathabar"-system.

The solution is sprayed over the cooling coil in the conditioner and the outdoor air respectively the return air comes in direct contact with the solution. The absorption heat is taken away by the cooling coil and the dehumidified air passes a droplet eliminator to guarrantee that there are no particles of solution in the supply air. The weak solution flows back into the pumping unit. Steam is pumped to the regenerator heating coil to warm up the solution. For that reason the water is evaporated and is exhausted with the air flow.

3.2 Regenerators

To guarantee a stationary operation of the dehumidifying process it is necessary to regenerate the weak solution. This can be made by different ways. Basically it is possible to use the same construction for the regenator as for the absorber. Instead of a cooling unit a heating unit is installed and it is only low temperature energy necessary to drive out the water of the solution. Temperatures between 60°C to 80°C are sufficient so that it is possible to use district heating

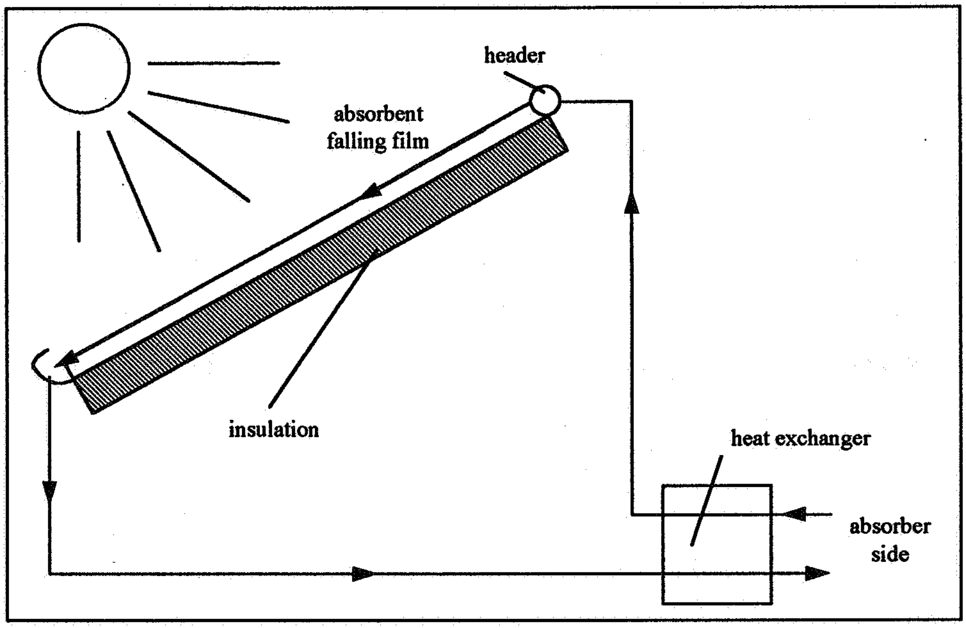

Figure 8: Example of a solar-collector-regenerator.

for example. Further the use of solar-collector-regenerators in sunny areas is an efficient way to regenerate the weak solution. Figure 8 shows an example for a solar-collector-regenerator. Usually the surface of a solar-collector is blackened to facilitate the absorption of solar energy and the back is well insulated. The surface can be corrugated, V-grooved or a textured surface to aid the even distribution of liquid film.

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The New Energy Conservation Code in the Federal Republic of Germany

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Synopsis

With effect from January 1st, 1995 the amended Heat Transfer Barrier Act („Wärmeschutzverordnung“) was introduced in the Federal Republic of Germany, replacing the 1982 version. This decree is binding on all houses to be built so that they reach the low energy standard. Former decrees envisaged mainly the reduction of the transmission heat loss while the amended version takes into account all other relevant aspects such as internal and solar heat gains as well as ventilation heat losses, and includes them into an energy balance procedure. In this way the new Heat Transfer Barrier Act does not lay down certain construction types or heating techniques, but focuses on the annual heat energy demand. With this energy conservation code the German Government aims at a 30 per cent reduction of the heat energy demand of new houses

1. The Heat Transfer Barrier Act

1.1 The Legal Background

The legal background of the Heat Transfer Barrier Act, HTBA, ("Wärmeschutzverordnung") was fixed in 1976 when the „Energieeinsparungsgesetz“ (EnEG), an energy conservation code, was passed. With the first Heat Transfer Barrier Act in 1977 the energy conservation code was put into practice. The necessity of the reinforced conservation of energy together with the development of modern construction techniques then lead to the amendment of the Heat Transfer Barrier Act in 1982 as well as to its introduction into planning and building laws and regulations in 1984.

The latest HTBA came into force on January 1st, 1995 after four years of difficult negotiations: Within the framework of its strategy to reduce the energy related emissions of the greenhouse gas CO_2, the government of the Federal Republic of Germany decided in November 1990 to amend both the Heat Transfer Barrier Act and the Heating Systems Act. Further impetus gave the report of the commission of enquiry „Precautions for the Protection of the Global Atmosphere“ („Vorsorge zum Schutz der Erdatmosphäre“) which was adopted by the German Parliament in September 1991. When the reflections on the part of the European partners regarding any hindrance of the common market were dispelled the cabinet decided the final version in July 1994.

On the occasion of the hearing in the German Ministry of Trade and Commerce the Fachinstitut Gebäude-Klima e. V. commented on the important topics of ventilation and heat recovery.

1.2 Area of Application of the ‘95 Heat Transfer Barrier Act

The main area of application of the ‘95 HTBA are buildings to be constructed. The latters are divided into three sections or types, depending on their use and indoor temperature level. Accordingly, they have to fulfill graded requirement profiles. Furthermore, the HTBA contains rules and requirements as for constructional changes like extension or heightening of existing buildings. Figure 1 gives an overview of the area of application of the ‘95 HTBA.

In the following areas, HTBA-requirements have to be met:

- Buildings to be constructed with normal indoor temperatures such as dwellings, office and administration buildings, schools etc.
- Buildings to be constructed with indoor temperatures between 12 and 19 °C such as industrial buildings
- Extension or heightening of existing buildings

- Change of individual heat transferring components of existing buildings (replacement, renewing, installation)
- Particularly relevant individual components such as slats for shutters, surface heating, ventilation systems etc.

2. The main changes of the HTBA

2.1 Increased requirements: With the help of the new HTBA the heat energy requirements and the pollutant emission of new buildings will be reduced by 30 per cent compared to the previous standard. In order to achieve these values, the heat requirement of new buildings follows the standard of low energy buildings, which is 54 to 100 kWh/m^2a. Comparing calculations show that the annual oil consumption can be decreased to 6-12 l/m^2 and the gas consumption to 6-12 m^3. Before, the standard consumption of oil driven heating systems amounted to more than 20 l/m^2a.

2.2 New proof procedure: While in the previous calculations only the heat loss through the building envelope (transmission heat loss) was considered, the new HTBA lays down an energy balance procedure. The u-value (k-value) calculation is only valid for (normally) heated small dwellings, i. e. buildings with up to two complete floors and not more than three accomodation units. Here, however, the requirements regarding the heat transfer are tightened compared to the 1982 HTBA. In all cases one of the two possible proofs of the heat transfer barrier have to be furnished. Figure 2 shows the area of application of the proof procedures for buildings to be constructed as well as for the extension of existing buildings.

Fig. 2: Area of application of the proof procedures according to the '95 HTBA

proof of the annual heat energy demand (heat balance procedure)		simplified proof for small dwellings (so called building component procedure)
binding	dwellings to be built with more than two storeys or more than three accomodation units	
optional	dwellings to be built with not more than two storeys and not more than three accomodation units	optional
binding	buildings to be constructed with normal indoor temperatures such as office and administration buildings, schools, hospitals, old people's homes, restaurants etc.	
optional	constructional extension or heightening of existing buildings by at least one heated room or more than 10 m^2 of joined heatable and useable surface	optional

The new calculation procedure is based on a heat balance of the entire heated building, i. e. not only the transmission heat loss, but the ventilation heat loss as well as solar and internal heat gains are considered also. The transmission heat loss informs about how much heat is transferred through the building envelope (external building components). The ventilation heat loss represents the amount of heat lost due to the exchange of warm indoor air against cold outdoor air. Solar heat gains indicate the amount of heat resulting from solar radiation into the building while internal heat gains result from several factors occurring inside such as occupancy, quantity of computers and other electronic devices used, lighting, etc. The calculated value now represents an energy parameter of the considered building on which the calculation of the annual heat energy demand is based.

2.3 Bonus arrangement for energy saving ventilation systems: The installation of ventilation systems can help reduce ventilation heat losses. Reduction factors according to the ventilation system that is built in may be used now for calculating the ventilation heat loss. The reduction factor of supply/exhaust systems is 0,95, the factor for systems with heat recovery and systems with heat recovery and heat pump is 0,80.
In order to prevent the operation of ventilation systems from being more energy consuming than energy saving, limiting values for the energy use of each system type have been fixed. As the annual heat energy demand of each building has been fixed, too, the reduction of ventilation heat losses thanks to a ventilation system allows the constructional heat insulation to be reduced. In order to limit constructional ventilation heat losses, the requirements in view of building airtightness were tightened, too.

2.4 The heat demand certificate for new buildings represents another important innovation of the '95 HTBA. It informs authorities, builders, tenants and buyers clearly about the energy relevant features of a building. This heat demand certificate contains a written synopsis of the calculated results of the heat transfer barrier proof. The builder or the owner of a building has to show the certificate to potential buyers oder tenants of his building. Normally the architect or the construction engineer issues the certificate and delivers it to the builder. The heat demand certificate is always issued for the entire building and not only for one accomodation unit. It informs about the determined annual heat energy demand, but not about the real energy demand of a building per year.

The proof procedure represents the core of the amended Heat Transfer Barrier Act. In the following, it is explained more precisely therefore:

3. Course of the proof procedure

At first, the ratio that the heat transferring parts of the building envelope A bears to the enclosed building volume V is calculated for a certain building plan (the same procedure as before).
Secondly, the upper limiting value of the annual heat energy demand is evaluated. This is done by the calculation of the A/V-ratio and ranges between 17,3 and 32 kWh per m^3 of the heated building volume and per year.
Only with buildings with a clear height equal to or smaller than 2,6 m, the maximum values can refer to the effective area as well. In this way they cover a range from 54 to 100 kWh/m^2 useable floor space and year. According to the '95 HTBA „useable floor space" means 0,32 times the heated building volume V. The staircase, basement and loft do not belong to the heated building volume V.

Fig. 3: Maximum values of the annual heat energy demand which is based on the heated building volume or on the useable floor space A_N in dependance on the ratio A/V.

Ratio A/V	Maximum annual heat energy demand referring to V Q'_H [1] according to clause 1.6.6	Maximum annual heat energy demand referring to A_N Q''_H [2] according to clause 1.6.7
in m^{-1}	in kWh/m^3a	in kWh/m^2a
1	2	3
< 0,2	17,3	54,0
0,3	19,0	59,4
0,4	20,7	64,8
0,5	22,5	70,2
0,6	24,2	75,6
0,7	25,9	81,1
0,8	27,7	86,5
0,9	29,4	91,9
1,0	31,1	97,3
< 1,05	32,0	100,0

Intermediate values are to be determined according to the following equation:

1) $Q'_H = 13{,}82 + 17{,}32\ (A/V)$ in kWh/m^3a
2) $Q''_H = Q'_H/0{,}32$ in kWh/m^2a

Dwellings with an A/V ratio of 0,72 m^2/m^3 and a headway of equal to or smaller than 2,60 m show a maximum annual heat energy demand of 82 kWh per m^2 useable floor space and year.

Following now is the calculation of the annual heat energy demand of the entire building and after this it has to be checked whether the calculated result exceeds the upper limiting value of the heat energy demand or not. If yes, constructional changes have to be integrated into the plans until the recalculated results meet the required values.

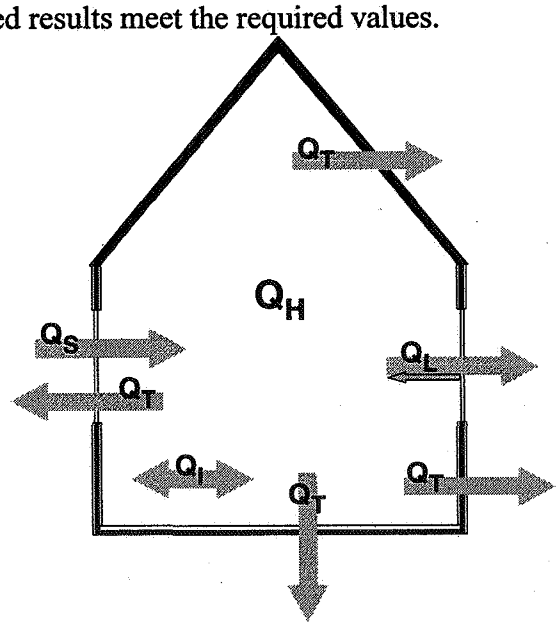


Fig. 4: Annual heat energy demand

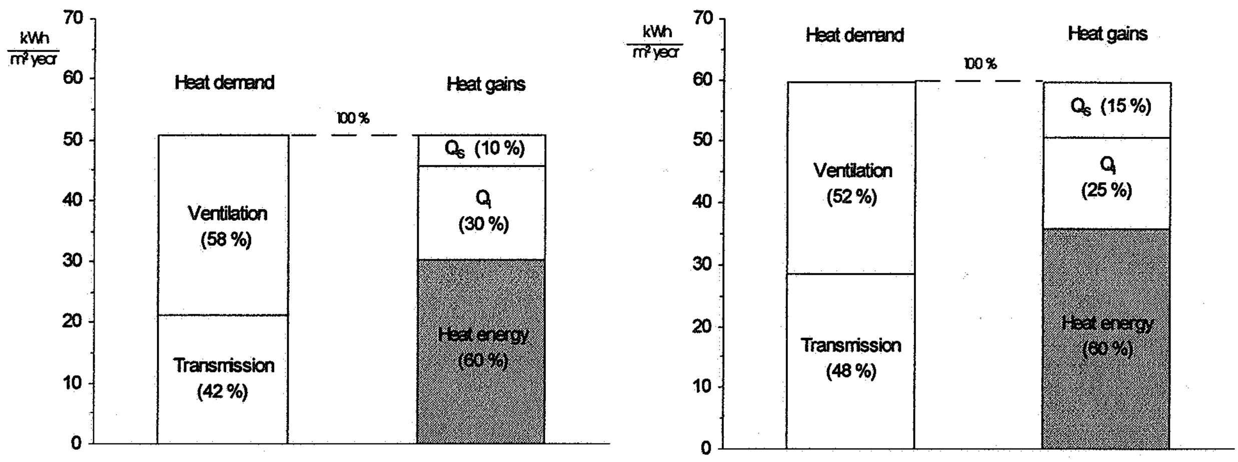
The annual heat energy demand Q_H of a building is composed of

- the transmission heat demand Q_T
- the ventilation heat demand Q_L
- the useable internal heat gain Q_I caused by the heat emission of persons and technical devices
- useable solar heat gains Q_S.

It is to be calculated for the entire building according to the following equation:

$$Q_H = 0.9\,(Q_T + Q_L) - (Q_I + Q_S)$$

The time spans of lowered indoor temperatures are taken into account with factor 0.9. Each single demand and all heat gains are to be determined according to the HTBA procedures. With the ventilation heat demand it has to be distinguished whether the building is ventilated through window opening or through a mechanical supply/exhaust system with or without heat recovery. As for solar heat gains, the solar radiation is evaluated according to the orientation and the angle of inclination of the window or the french window considered. Before starting the calculations therefore the orientation and the angles of inclination of all windows and french windows have to be determined.



Apartment building with six accomodation units

Useable floor space:	544 m^2
„A/V" ratio	0,49 m^2/m^3
Upper limiting value of the annual heat energy demand	69,7 kWh/m^2 year (100 %)
Annual heat energy demand:	50,6 kWh/m^2 year (73 %)

Semi-detached corner house

Useable floor space:	123 m^2
„A/V" ratio	0,72 m^2/m^3
Upper limiting value of the annual heat energy demand	82,2 kWh/m^2 year (100 %)
Annual heat energy demand:	59,7 kWh/m^2 year (73 %)

Fig. 5: Graphic representation of two calculation examples

Figure 5 shows the calculated results of the annual heat energy demand of an apartment building with six accomodation units and of a semi-detached corner house. The annual heat energy demand of each house remains under the upper limiting value by 27 per cent thanks to the building components and building constructions used. It is characteristic of both buildings that the useable internal and solar heat gains almost meet the respective transmission heat demand and that the ventilation heat demand corresponds to the scale of the annual heat energy demand, respectively. These results are applicable to other dwellings without heat recovery from exhaust air. It is clearly shown that the considerable influence of the ventilation heat demand on the annual heat energy demand is getting ever more important the better the building envelope is thermally insulated.

'95 Heat Transfer Barrier Act/Calculated results of the annual heat energy demand of two kinds of dwellings

4. Conclusion

The examples clearly show that the '95 Heat Transfer Barrier Act can be an effective instrument to reduce the pollutant emissions of new buildings. However, each regulation is only as effective as the possibility of controlling it. The annual heat energy demand simply represents an approximate value of the resulting heat energy demand. This is due on the one hand to the simplicity of the algorithm. Besides, the execution of the construction work as well as the heating and ventilating habits of the occupants can have a considerable influence on the heat energy demand. The problem of controlling the real annual energy consumption can not be solved therefore.
The current Heat Transfer Barrier Act is to be reamended in 1999. It is desirable that the amendment will not only bring along a further intensification of the constructional heat insulation and of the annual heat energy demand, but all energy relevant factors should become more severely integrated than before. In this way the Heat Transfer Barrier Act could be transferred definitely into an energy conservation code.

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Automatic Control of Natural Ventilation and Passive Cooling

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AUTOMATIC CONTROL OF NATURAL VENTILATION AND PASSIVE COOLING

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Natural ventilation has the potential to replace or supplement air conditioning, comfort cooling and mechanical ventilation. Whilst there are obvious environmental advantages, there are problems of achieving adequate control. The flow of air must be controlled to ensure a comfortable environment in the building whilst limiting energy consumption and maintaining indoor air quality.

Several UK buildings now use natural ventilation in conjunction with automatic control systems. This technique has a number of important benefits including:

- reduced building energy consumption
- potentially lower equipment costs
- lower operation and maintenance costs
- improved temperature control (primarily in passive buildings), especially during hot periods
- possible integration with mechanical ventilation systems.

This work was carried out as part of the Department of the Environment's EnREI (Energy Related Environmental Issues in Buildings) programme and was managed by the Building Research Establishment. The material presented in this paper highlights some aspects of two research projects, 'The Control of Natural Ventilation'[1] and 'Night Cooling Control Strategies'.

The research undertaken has led to the development of generic control strategies. These have evolved from consideration of the control strategies used in naturally ventilated buildings utilising Building Management System (BMS) control together with experience obtained from monitoring three naturally ventilated buildings. The site monitoring has also led to recommendations being provided for commissioning and fine tuning procedures.

1 INTRODUCTION

1.1 The decision to use automatic controls

The decision to use automatic controls is primarily based on the expected heat gains in the occupied space. If the heat gains are expected to be below approximately 25 W/m^2[2] throughout the year then manual opening of windows will generally allow summer comfort conditions to be maintained. The use of automatic controls fitted to the inlet and outlet ventilation path within the building can extend this, through the use of night-cooling control strategies and an appropriate building design, to heat gains of approximately 40 W/m^2 or more. These cooling strategies allow the inlet and outlet vents to be opened during the night thus allowing the cool night time air to flow through the building, cooling the fabric, furniture and fittings. In this way the heat that has built up during the previous day is removed and storage of the cool air ('coolth') within these components is achieved, consequently providing a cooling effect the following day. The building fabric may be designed to have exposed concrete ceilings or other exposed mass to facilitate the storage of additional 'coolth', thus enhancing the effect.

The use of a lighting control system may extend this even further since it will reduce the heat gains during the peak cooling season (perhaps by 10 W/m^2) when natural light levels will be at their highest. Above heat gains of approximately 45 W/m^2 mechanical ventilation in conjunction with comfort cooling or air conditioning is more likely to be used.

Automatic controls are not only used to ameliorate heat gains. Some buildings may not be suited to manual operation of windows especially public buildings where damage may occur to the vents when opened by occupants unfamiliar with their operation. Other buildings may not have accessible windows such as atria, sports halls, etc.

Automatic control of the windows and vents may take place in conjunction with manual control. Research has been carried out[2] showing that where occupants of a space have control over their local area they are more willing to accept a wider comfort band. This may be in the form of two separate ventilation components, one under manual control and the other under BMS control (for night-cooling). Alternatively, an occupant controlled pushbutton may be used to electrically drive a window or vent open during the day with the BMS overriding this at night.

1.2 Open/closed or modulating control

Modulation of the vents is advantageous since it allows a reasonable level of ventilation to take place when there is rain or high winds present. This is achieved by restricting the opening of the vents to a position to prevent the ingress of water and high air velocities. Other building designs allow ventilation in poor weather conditions by protecting the air inlet path by the use of shades, grilles, cowls and baffles, thus allowing some or all of the vents to remain open.

Modulation of the vents may not be necessary as opening and closing the device in steps can be acceptable, or alternatively, the successive opening of ventilation devices in sequence may be possible until the ventilation demand is satisfied. The air flow rate through the device will, in any case, be generally coarsely controlled due to the fluctuations in the wind speed. This renders close control of air flow rates impracticable. However, if ventilation openings with a large 'free area' are provided relative to the volume of the space, it is likely that they will be required to have modulating control in order that some control of the air flow rate can be achieved.

The research carried out has led to the development of generic control strategies for buildings with automatic control of natural ventilation only, buildings with mechanical and natural ventilation control (mixed mode) as well as two generic control strategies for night cooling. These evolved from consideration of the control strategies used on ten sites together with experience from monitoring three of these sites.

2 MIXED MODE CONTROL STRATEGY

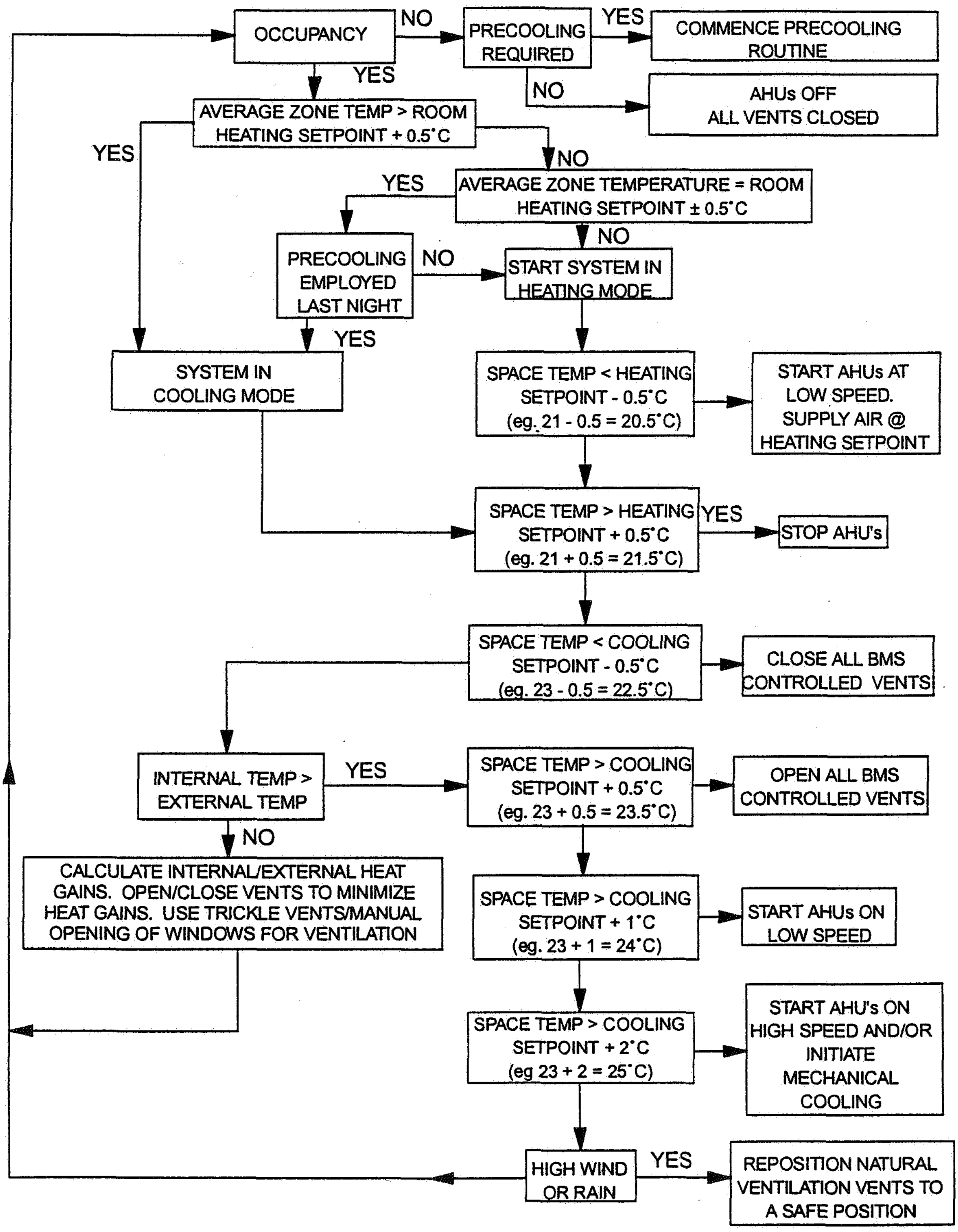
The control system determines whether heating or cooling is required depending upon the average zone temperature and selects the appropriate ventilation system following the logic presented in figure 1. In the event that mechanical cooling is not available and both the internal and external temperatures rise above say, 23°C, then the optimum control strategy (ie whether to open or close the vents not required to satisfy the minimum fresh air requirement) is dependent upon a number of issues:

i) Selecting the minimum source of heat gain - ie the external air will supply a heat gain to the space depending upon the external air temperature and the number of air changes per hour. This is approximately 0.35 W/m^3 for each 1°C difference between internal and external temperature.

ii) Maintaining the inlet and outlet vents open will maximise the air movement in the building, but at the cost of increasing the rate of internal temperature rise. However, the increased air movement will be beneficial to comfort in preference to slightly lower internal temperatures and less air movement. The extent of the perceived benefit of air movement on people has been discussed by Oseland[2] who suggests that " the graphs and algorithms used in current standards (CIBSE, ASHRAE Standard 55 and ISO 7730 [1993]) which show the increase in temperature required to compensate for an increase in air velocity can equally well be used to show the increase in summer temperature that will be tolerated with an increase in air velocity." This would suggest that an air velocity of 1 m/s will correspond to a perceived temperature decrease of about 2°C.

iii) A further consideration is that the requirement to manually close vents, especially windows, is counter-intuitive to human nature when it is hot in countries where air conditioning is not prevalent. The benefit of providing mechanical ventilation plant simply to increase air flow under hot conditions should also be carefully considered. The use of a 'punkah' fan is likely to be more beneficial.

Figure 1 - Mixed Mode Ventilation Control Strategy



3 NIGHT-COOLING CONTROL STRATEGY

The choice of a night-cooling control strategy is dependent upon achieving the optimum transfer of cool night air into the building fabric, furniture and fittings. One control strategy for allowing this is presented in figure 2. This strategy aims to measure the day-time heat gains in the space and then remove the equivalent amount of heat at night, thus maintaining the equilibrium between the building fabric temperature and the space temperature. The method of quantifying the daytime heat gains is based upon measuring the number of hours that the internal space temperature is above room temperature setpoint, totalled for all the hours in this period. The cooling gain in degree hours is defined as the number of hours that the internal temperature is below the room temperature setpoint, totalled for all the hours in the period.

The decision as to night-cool or not is based upon the number of degree hours that the internal temperature is above the room temperature setpoint. If, at the end of the occupied period the internal temperature has been above the room temperature setpoint for more than say, three degree hours and the internal temperature is higher than the external temperature, then the decision is made to cool the building that night. The normal wind, rain and low external temperature interlocks still apply. In the event of these interlocks occurring the ventilation openings will either close or they will be limited, if possible, to a practical operating position. A multiplying factor may be provided so that the amount of cooling degree hours can be varied above or below the monitored daytime heating degree hours.

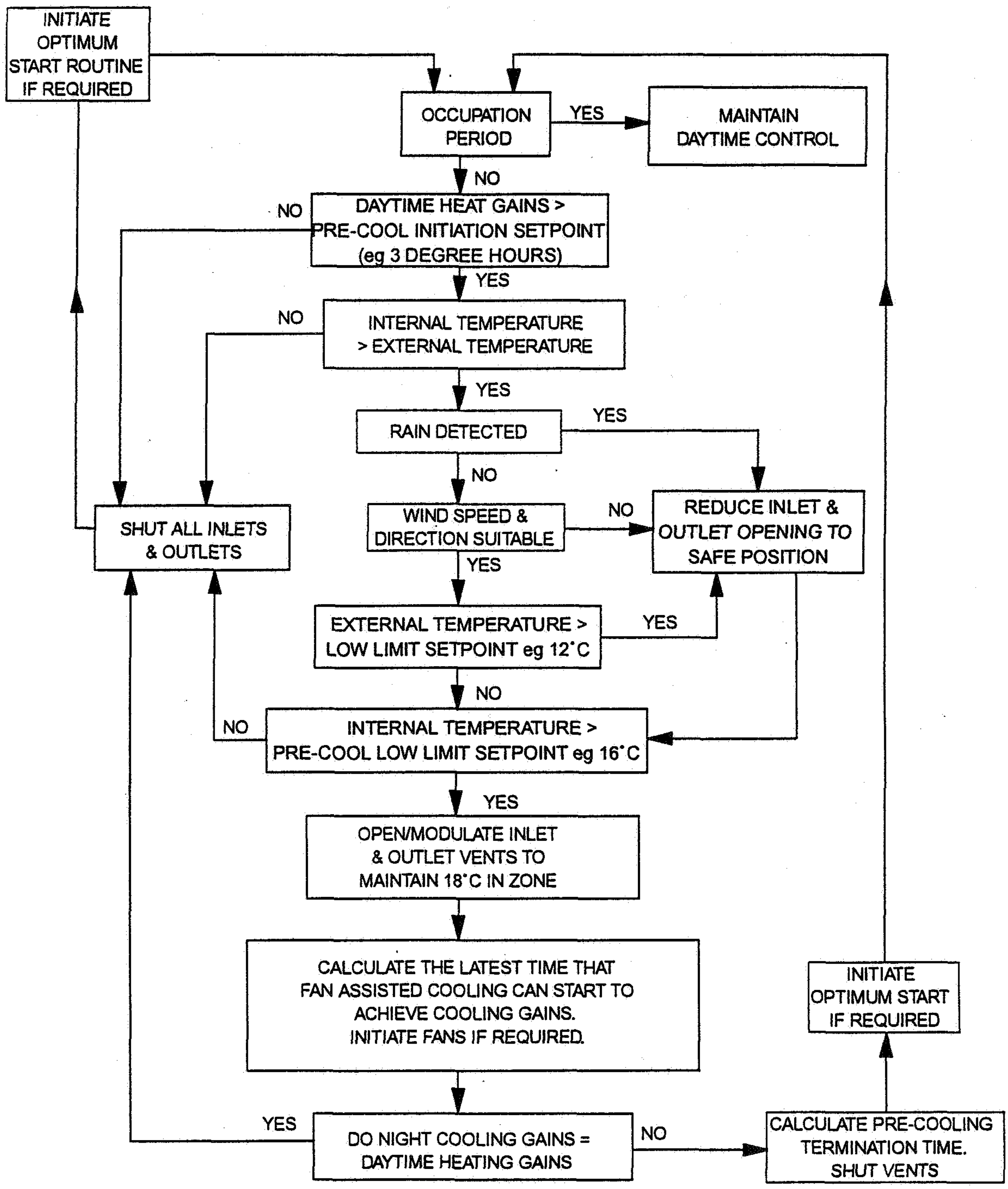
Other night cooling control strategies allow the internal temperature to fall to a lower value (eg 14°C) thus increasing the amount of passive cooling provided. This may add an additional control problem - when to terminate the night cooling. Typically, an exposed ceiling slab will still be at a temperature of approximately 20 - 23°C even after a full night of cooling (it is unlikely that a concrete slab will change temperature by more than 1°C overnight and 0.5°C is more likely). This exposed slab thus provides a heating source capable of raising the internal air temperature towards the room heating temperature setpoint. If the night-cooling is not completed then the optimum time to terminate the night-cooling strategy is not clear. The two possibilities are:

a) disabling the night-cooling earlier in the morning thus allowing the space to be heated by the heat re-emitted from the building fabric or,

b) extending the night-cooling and initiating the heating system for a short period. This is believed to provide additional stored cooling for use later in the day.

In option a) the control system calculates the time that the ventilation system should shut down in order that the heat gains re-emitted from the building fabric will provide sufficient heating to raise the space temperature to the setpoint by the start of the occupation period. The intensity of this heating effect is typically expected to be in the region of 1 to 1.5 K per hour and will depend in particular, upon the effectiveness of the night-cooling strategy in conjunction with the thermal characteristics of the fabric, furniture and fittings. If there is insufficient passive heat available then it may be

Figure 2 - Night Cooling Control Strategy



necessary to provide heating. It is, in any case, recommended that the conventional optimum start algorithm is provided to initiate the heating if the building is unlikely to achieve the space temperature setpoint by the start of the occupation period. This is analogous to option b) and the efficacy of this is based on the requirement that the internal temperature is at the space temperature setpoint by the start of occupancy. The comparatively small amount of heat input necessary to reach this state is expected to be justified by the additional cooling stored in the slab, which will benefit the space with a reduced air temperature during the ensuing day. It is, of course, necessary that the space temperature (heating) setpoint is below that of the slab temperature to prevent the slab absorbing the heat given out by the terminal heat emitters.

Where mechanical ventilation is available a separate algorithm calculates the latest time that the supply and extract fans should start in order that night-cooling can be completed. There may be an interlock that only allows the fans to run during the low tariff period thus minimising electricity charges.

Three buildings were monitored as part of the project, one of which is presented below.

4 MONITORING OF A MIXED MODE BUILDING

The building is a three storey office building built around a central atrium. The building utilises BMS control of the casement vents with manually controlled centre pivot windows beneath. A mechanical ventilation system can provide either 2 or 4 air changes per hour to the open plan office space. The building utilises the generic ventilation control strategy presented in figure 1. The night-cooling strategy is based upon ventilating all night using natural ventilation to obtain a calculated slab temperature setpoint. This setpoint calculation is self learning according to the deviation from the setpoint at the start of the following day and is based upon equalising the slab temperature, the room temperature and the slab temperature setpoint. An adjustment factor is provided in order that towards the end of the occupancy period a cooling effect is still available from the slab. Mechanical ventilation is initiated during the low tariff period if necessary. The mechanical ventilation system is also initiated in the event of the high wind and rain interlocks operating.

Monitoring of this building highlighted the following:

- Night-cooling of the exposed concrete slab enabled the average space temperature to be held at 26°C with an external temperature of 31.5°C (windows open),

- Bad weather (high wind or rain) prevented the natural ventilation system from operating for approximately 30% of the time. Consequently the mechanical ventilation system ran thus increasing the energy use of the building. Fine tuning of the control setpoints will help to alleviate this,

- However, mechanical ventilation to assist cooling during the daytime was initiated on only 9 of the days during the 4 month monitoring period. On 7 of these days the outside air temperature was above the internal temperature and therefore the mechanical ventilation was not suitable for cooling. This would suggest that if passive winter ventilation was provided together with either an improvement in the building design to facilitate natural ventilation during poor weather conditions and/or fine tuning of the weather control setpoints, then there is little requirement for mechanical ventilation,

- The use of night-cooling using natural ventilation appeared to provide more effective cooling of the exposed slab (and the space temperature) than the use of night-cooling using mechanical ventilation. This is despite the natural ventilation working with higher temperature external air. This effect may be due to fan pick up. A better technique may be to use the extract fans to 'drag' air through the casement windows in order to cool the space,

The initiation of the heating system following night-cooling occurred on 25% of the monitored days. The night-cooling and heating controls were not linked other than to inhibit the optimum start routine if night-cooling was applied. It is suggested that for this building consideration should be given to linking the strategies in order to reduce the hours of operation of the heating system.

The analysis carried out regarding the eleven buildings utilising automatic control as part of this project suggests that the buildings would benefit from further fine tuning of the control systems in order to enhance their performance, both in terms of the internal temperature as achieved and in a reduction in energy consumption. This fine tuning should be an additional contract following completion of the building and should be undertaken throughout the year following handover, perhaps involving three site visits. The additional cost of this can often be justified by the energy savings that can be achieved as well as the improved comfort conditions that will result. However, it is important that the contractor has a pre-conceived plan for the fine tuning method.

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Air Flow Distribution in a Mechanically Ventilated High-Rise Residential Building

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Air Flow Distribution in a Mechanically-Ventilated High-Rise Residential Building

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Synopsis: Air flow measurements and simulations were made on a 13-story apartment building to characterize the ventilation rates for the individual apartments. Parametric runs were performed for specific conditions, e.g., height, orientation, outside temperature and wind speed. Our analysis of the air flow simulations suggest that the ventilation to the individual units varies considerably. With the mechanical ventilation system disabled, units at the lower level of the building have adequate ventilation only on days with high inside-outside temperature differences, while units on higher floors have no ventilation at all. Units facing the windward side will be over-ventilated when the building experiences wind directions between west and north. At the same time, leeward-side apartments will not experience any fresh air--the air flows enter the apartments from the corridor and exit through the exhaust shafts and the cracks in the facade. Even with the mechanical ventilation system operating, we found wide variation in the air flows to the individual apartments. In addition to the specific case presented here, these findings have more general implications for energy retrofits and health and comfort of occupants in high-rise apartment buildings.

1.0. Background

Airflow in high-rise apartment buildings has been the subject of sporadic research over the past three decades. From the early work by Shaw and colleagues in the early 1970s, researchers have tried to understand the driving forces for air flow in order to recommend energy efficiency measures that do not jeopardize the health and comfort of building occupants.

Our recent activity in this area came about through the DOE-HUD Initiative, a response to the U.S. National Energy Strategy's directive to improve the energy efficiency in public housing. Under the Initiative's guidance a collaborative project was established to demonstrate energy efficiency in public housing as part of a utility's Demand Side Management (DSM) Program.

The demonstration site is the Margolis Apartments, a modern 150-unit high-rise apartment building for the elderly and handicapped, located in Chelsea, Massachusetts, in the greater Boston, Massachusetts, metropolitan area.

2.0 Building Description

The Margolis Apartment building was designed in 1973-1974 and is typical of high-rise construction from that period. The building has thirteen stories and is of steel-frame construction. The individual apartments have electric-resistance heaters in each room, and double-pane windows and sliding balcony doors. A typical floor plan is shown in Figure 1.

The building has a mechanical ventilation system, with kitchen and bathroom exhaust fans for each apartment leading into separate vertical shafts that have additional exhaust fans located on the roof. The supply air system for the building is provided by a fan and heating unit on the roof which connects to a vertical shaft with supply registers to the main hallway on each of the floors. Supply air then enters the apartments by a slot under the front door of each unit.

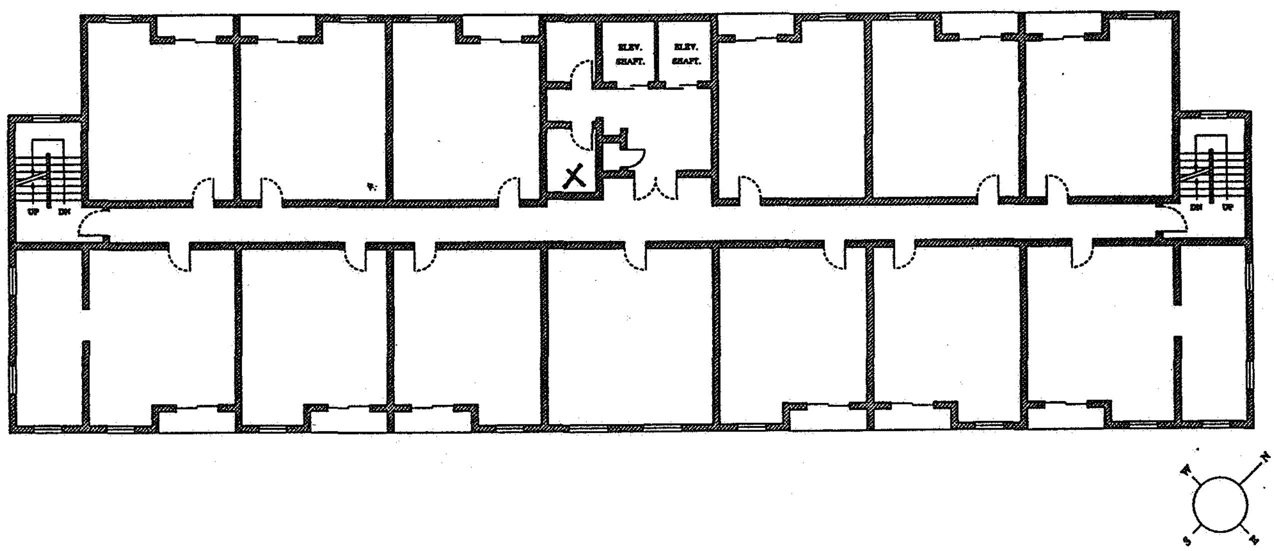

Figure 1. Typical floor plan (floors 2-5), Margolis Apartments, Chelsea, MA. The "x" shows the location of the supply ventilation register for each corridor.

The building is exposed on all sides to the wind, and is located less than 5 km from the airport weather station. Airport weather data records a mean annual wind speed of 6 m/s with up to 26 m/s wind speeds in winter. The winter wind is primarily from the northwest; the wind in spring through fall is from the southwest.

In December, 1993, the building underwent extensive retrofits. New double-pane, low-e windows replaced the old windows throughout the building. A computerized energy management system was installed that allowed for tracking and controlling of the thermostats in the individual apartments. Efficient light bulbs were installed in the individual apartments. A new sprinkler system was installed throughout the building. The balconies were screened in to prevent the pigeons from roosting. A second phase of retrofit activity a year later involved improvements to the abandoned ventilation system.

Prior to the window retrofit, drafts were a major complaints expressed by the tenants, but since the retrofit, there have been--according to building management--fewer complaints about window drafts. There was mention of the windows being hard to open for some of the residents, both from the latching mechanism and the effort needed to lift the double-hung sash. No problems with condensation on the windows were reported since the retrofit.

The northwest-facing units (weather side) continue to be the hardest units to maintain thermal comfort. Also the second floor units (above the open parking areas) continue to be a problem in cold weather.

3.0 Measurements & Preliminary Findings

The measurements and analysis that we are reporting here consist of two parts: 1) Air leakage measurements of the apartments measured pre- and post-retrofit, and 2) Computer simulations of the air flows in the building under different weather conditions.

3.1 Air Leakage Measurements

We measured using blower doors the air leakage in nine apartments, before and after the new windows were installed. The average pre-retrofit total effective leakage area for the one-bedroom apartments was 241 cm^2 and 256 cm^2 for the two-bedroom apartments. The post-retrofit total effective leakage area for the one-bedroom apartments was 230 cm^2 and 248 cm^2 for the two-bedroom apartments.

These measurements suggest little or no reduction in air leakage due to the new windows, which is surprising given that tenants who had previously complained of drafts were now satisfied. One explanation is that tenants were previously experiencing down drafts at the window due to cold surface temperatures, which no longer occur because of the new double-pane, low-e windows.

We also note that these measurements were made in very windy conditions--beyond the limits allowed for standard blower-door tests. While this problem is not uncommon in low-rise buildings, it is an even bigger problem in high-rise buildings, where wind speeds are often much higher than for buildings at ground level. Furthermore, the measurement technique used is based on a reference pressure describing the pressure field around the building. In large buildings, it is very difficult to find a pressure point which acts as the reference pressure for the apartment being investigated. There is also the possibility that the measurement technique itself, i.e., depressurization with a blower door, temporarily seals the windows and distorts the findings.

By way of comparison, Kelley et al. (Kelly 1992) measured the air leakage pre-and post-retrofit in a high-rise apartment in Revere, Massachusetts, a few kilometers north of the Margolis apartment. They found an average pre-retrofit leakage for 17 of the apartments of 904 m^3/h at 50 Pascals, and a post-retrofit leakage of 763 m^3/h at 50 Pascals, a reduction of 15%. The comparable flows at Margolis were higher, and showed no reduction after the retrofit, with an average of 1183 m^3/h pre-retrofit and 1214 m^3/h post-retrofit.

We also measured the leakage from one apartment to another, using tracer gases, and found little communication between units--less than 4% of the total leakage was to adjacent apartments. This was not altogether surprising given the concrete construction of the building.

3.2 Ventilation Simulations

Based on the measured air leakage data from the building we conducted extensive air flow modeling of the apartments using the multizone air flow model COMIS, a simulation tool, developed at Lawrence Berkeley Laboratory, which calculates air flows based on mass balance calculations for individual zones (Feustel, 1990).

In order to limit the amount of input needed for the simulation model, each apartment was modeled as one zone, assuming the internal doors to be open. In order to account for the stack effect and the inter-zonal flows between the floors, all 13 floors were modeled.

The results show, that with wind blowing perpendicular to the windward side and no stack effect present, air moves from the windward side facade through the corridors into the leeward side apartments. Under the previous conditions with no ventilation system present, only a small portion of the infiltration air is exhausted through the vertical shafts of the exhaust system. Dampers at the apartment level and on top of each of the shafts restrict the exhaust flow.
When the building is operating without the mechanical ventilation system, the air mass flow distribution for windward side apartments on different floors follows a predictable pattern (Figure 2). With increasing wind speed, the distribution of infiltration becomes more pronounced, showing a minimum at the level of the third floor and a maximum at the 11th floor. The leeward side apartments do not experience any infiltration.

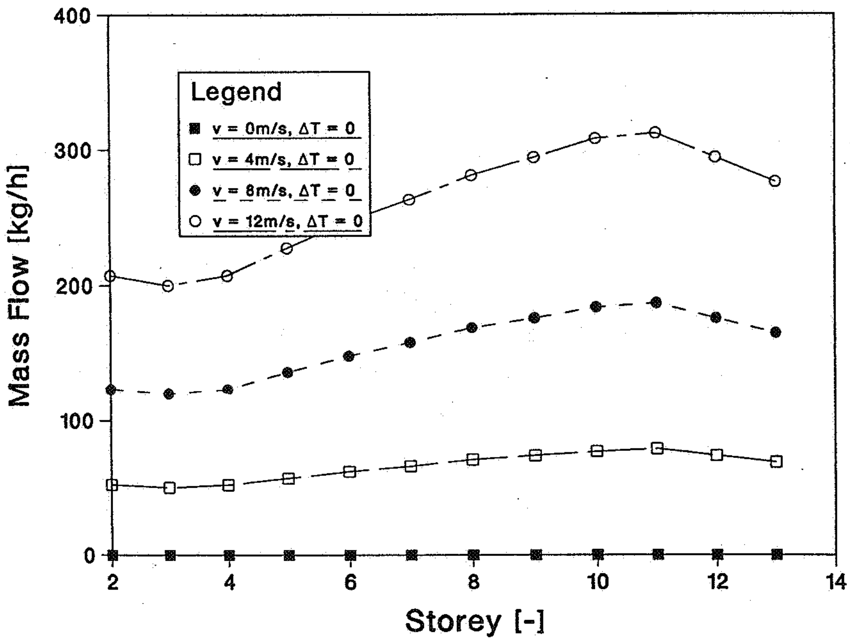

Figure 2. Mass air flow at different wind speeds and no inside/outside temperature difference for the windward apartments with the mechanical ventilation system off. The ASHRAE Standard 62 recommends a ventilation rate of 0.35 ACH, which corresponds to a mass air flow of 50 kg/h.

With a larger inside/outside temperature difference of 20 ^{o}C and zero wind speed, the air flow for the windward apartments decreases with height above ground from 100 kg/h (50 cfm) on the second floor to zero at the level of the 11th floor. With increasing wind speed the air flow curves show a more balanced air flow distribution until the velocity driven air flows override the stack effect (Figure 3). As the pressures forcing the air flow can be added, the air flows for any given wind speed is higher if stack pressure is present.

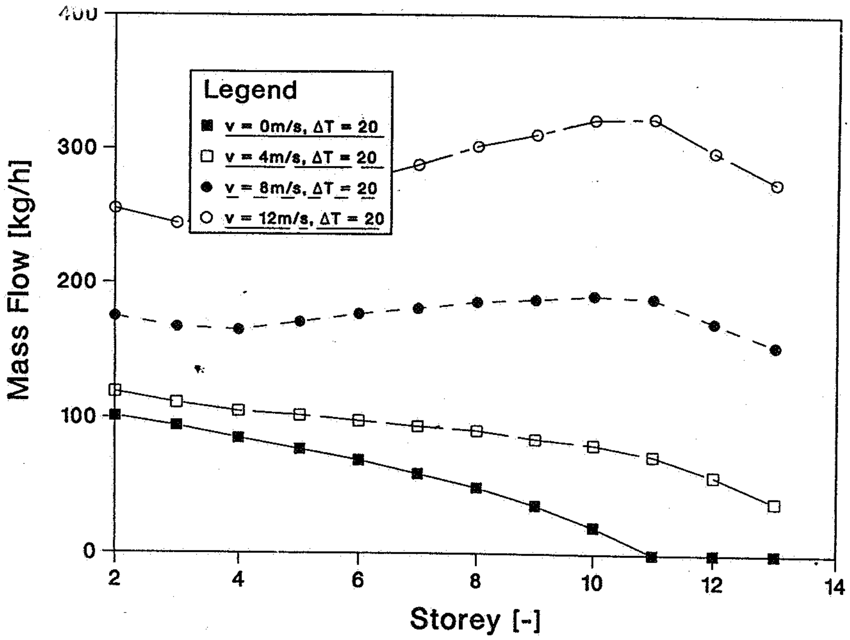

Figure 3. Mass air flow at different wind speeds and an inside/outside temperature difference of 20 K, for the windward apartments with the mechanical ventilation system off.

The air flows for the leeward side is shown in Figure 4. With increasing wind speed the air flow entering the apartments through the outside wall is getting smaller. The zero wind speed curve is the same for the windward side and the leeward side. The top floors do not experience any infiltration. Higher wind speeds cause higher negative pressures on the facade, which lower the level for the neutral pressure. At wind speeds of 12 m/s no infiltration occurs at the apartments facing the leeward side.

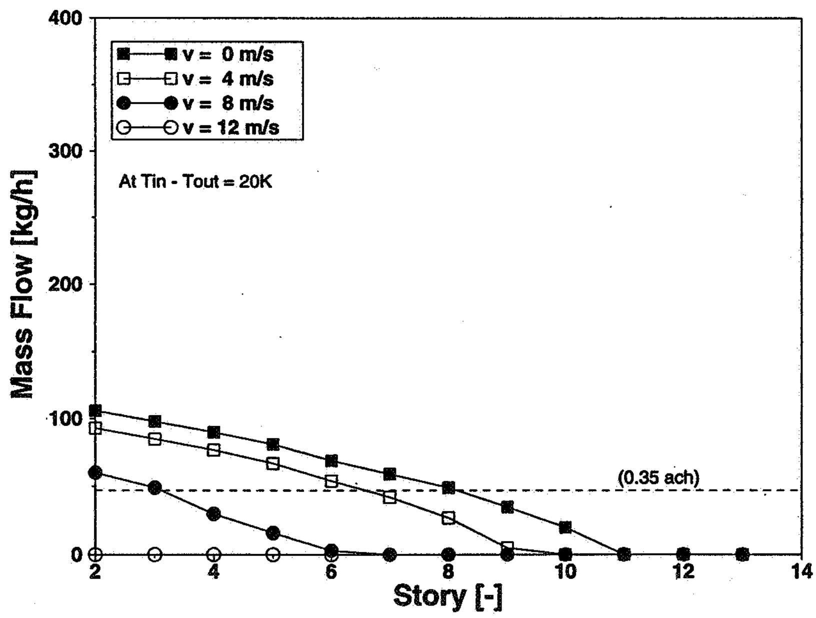

Figure 4. Mass air flow at different wind speeds and an inside/outside temperature difference of 20 K, for the leeward apartments with the mechanical ventilation system off.

Air flows into the apartments are slightly higher when the ventilation system is in operation. Figure 5 shows the air flows entering the apartments located on the windward side through the facade for different wind speeds when no stack effect is present. At low wind conditions, infiltration is almost independent of the height above ground. With higher wind speeds, we see that the infiltration flows follow the wind pressure profile.

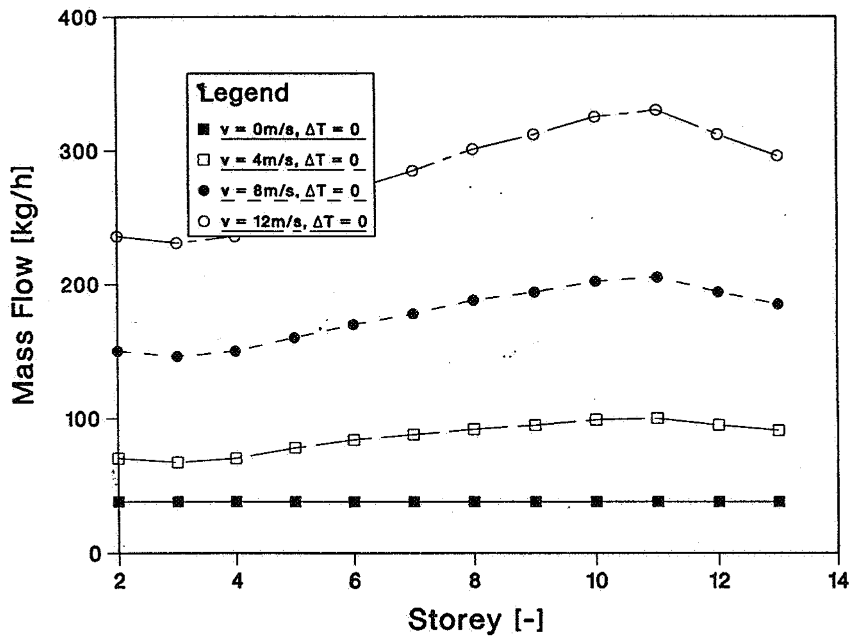

Figure 5. Mass air flow at different wind speeds and no inside/outside temperature difference, for the windward apartments with the mechanical ventilation system **on**.

The ventilation system is designed to provide the necessary "fresh" air by means of supplying the air to the corridor. The direction of the air flow through the doorway of the apartment door determines whether the supplied air is entering the apartments. For the two higher wind speeds, the air flow passing through the doorways are shown for the apartments on both sides of the corridor. We see, that at higher wind speeds the windward side apartments do not receive any of the air supplied to the corridor (see Figure 6). At lower wind speeds, the windward side apartments located on the lower floors participate slightly in the air exchange provided by the supply system. This means, that at lower wind speeds nearly all the air entering through the facade is being exhausted directly into the vertical exhaust shafts. At higher wind speeds, air from these apartments is being pressed into the corridor. All leeward side apartments receive between 50 and 75 kg/h air from the corridor. For higher wind speeds, this amount of air is smaller than the air which enters the corridor from the windward side apartments. The excess air is leaving the corridor through the elevator shaft (which has a large opening to the leeward side at the penthouse level).

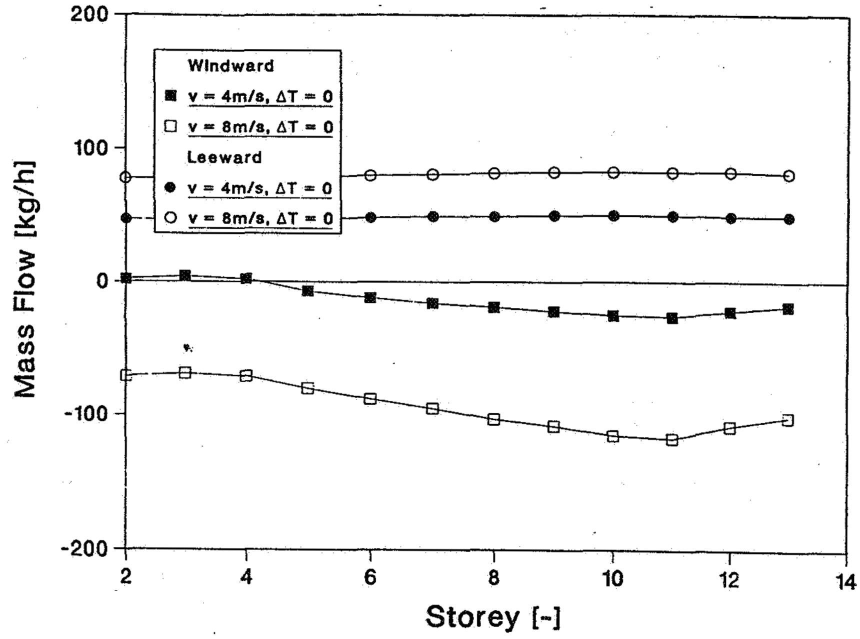

Figure 6. Mass air flow between the apartments and the corridor at different wind speeds and no inside/outside temperature difference, for all apartments and the mechanical ventilation system on.

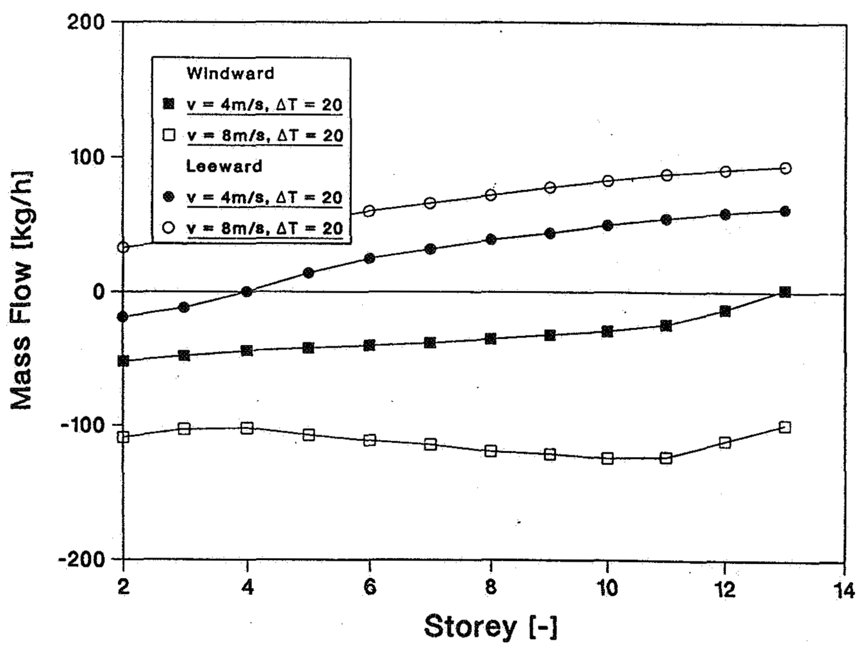

Figure 7. Mass air flow between the apartments and the corridor at different wind speeds and an inside/outside temperature difference of 20 K, and with the mechanical ventilation system on.

With larger temperature differences between inside and outside present (Figure 7), the infiltration flows for the lower windward side apartments increase significantly. The flows for the apartments on the higher floors keep constant. The stack effect also causes the distribution of air flow through the doors to change. At wind speeds of 4 m/s and temperature differences of 20 °C all apartments on lower floors provide air flow to the corridor, rather than receiving ventilation air. Higher up in the building, leeward side apartments receive air from the corridor while windward side apartments exhaust air into the corridor. With increasing temperature difference, the stack effect is amplified.

4.0 Conclusions

Our analysis of the air flow simulations shows that the ventilation to the individual units varies considerably. With the mechanical ventilation system disabled (pre-retrofit case), units at the lower level of the building had adequate ventilation only on days with high temperature differences, while units on higher floors had no ventilation at all. Units facing the windward side were over-ventilated when the building experienced wind directions between west and north. At the same time, leeward side apartments would not experience any fresh air--air flows would enter the apartments from the corridor and exit through the exhaust shafts and the cracks in the facade. Even with the mechanical ventilation system operating, we found wide variation in the air flows to the individual apartments.

A fundamental issue here is the design question of how to best supply ventilation to individual apartments in a highrise building. Using the corridor as the supply route has several challenges, including the control of the temperature of the supply air, the temperature of the corridor, the access from the corridor to the apartment, and the balance between supply and apartment exhaust.

On the exhaust side, studies have shown that when apartment occupants have local control over bathroom and kitchen exhaust, they use them less than one hour per day, if at all (Shapiro-Baruch, 1993), which makes it difficult to size the supply ventilation system. Continuous exhaust ventilation, however, presents the possibility of over ventilation and unnecessary use of energy.

Efforts to improve the energy efficiency of high-rise apartment buildings have been frustrated because of the lack of knowledge on air flows for individual apartments. Ventilation rates for individual apartments vary greatly due to height, orientation, and wind speed and outdoor temperature. Any recommendations for reducing air leakage will have to take these variables into account, so that efforts to tighten the shell for energy efficiency do not create health and comfort problems for the residents.

5.0 Acknowledgments

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Energy Impacts of Air Leakage in US Office Buildings

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SYNOPSIS

Airtightness and infiltration rate measurements in office and other commercial buildings have shown that these buildings can experience significant levels of air leakage [1,2]. The energy impact of air leakage in U.S. office buildings was estimated based on the analysis of a set of 25 buildings used in previous studies of energy consumption [3,4]. Each of these buildings represents a portion of the U.S. office building stock as of 1995. The energy impact of air leakage in each building was estimated by performing an hourly analysis over one year, with the infiltration rates varying linearly with the wind speed. The energy associated with each of the 25 buildings was then summed to estimate the national energy cost of air leakage. The results show that infiltration accounts for roughly 15% of the heating load in all office buildings nationwide, and a higher percentage in recently constructed buildings. A sensitivity analysis showed that the heating loads due to infiltration were particularly sensitive to uncertainty in the balance point temperature and nighttime thermostat setback. The results also show that infiltration has very little impact on cooling loads in office buildings. The results for office buildings are presented and discussed, along with the implications for the energy impacts of air leakage for the total commercial building stock in the U.S.

1. INTRODUCTION

Despite common assumptions that envelope air leakage is not significant in office and other commercial buildings, airtightness and infiltration rate measurements have shown that these buildings are subject to significant levels of air leakage [1,2]. Air leakage in commercial buildings can have several negative consequences, including reduced thermal comfort, interference with the proper operation of mechanical ventilation systems, degraded indoor air quality due to the infiltration of unfiltered outdoor air, moisture damage of building envelope components, and increased energy consumption. For these reasons, attention is being given to methods of improving airtightness both in existing buildings and new construction [5]. However, in order to evaluate the cost effectiveness of such measures, an estimate of the impact of air leakage on energy is needed. While there have been many studies of energy consumption in office and other commercial buildings using building energy simulation programs [3,6], these programs typically employ a simple approach to infiltration. For example, the DOE-2 program requires the user to specify an air change rate, which is then adjusted hourly depending on the wind speed [3]. However, outdoor infiltration in multizone, mechanically ventilated buildings is a complex phenomenon, with the infiltration rates depending on the indoor-outdoor temperature difference, wind speed and direction, the airtightness of exterior walls and interior partitions, and mechanical ventilation system airflow rates. In order to determine the impact of air leakage on energy consumption and to evaluate the benefits of various leakage mitigation strategies, a detailed multizone network airflow analysis, which calculates infiltration based on pressure distributions and effective leakage areas must be included in the energy simulation. While such an approach is currently being pursued at NIST, the objective of the current study is to make a preliminary estimate of the annual energy cost of infiltration in commercial buildings.

2. BUILDING SET

The calculations were performed for a set of 25 office buildings, each of which represents a portion of the office building stock in the U.S. Twenty of the buildings were developed by Briggs, Crawley, and Belzer [7] to represent the existing office building stock as of 1979; the other five buildings represent construction between 1980 and 1995 [4]. In both cases, cluster analysis was used to separate the total building population into several groups, within each of which certain physical characteristics and estimated annual loads of the buildings were relatively uniform. The characteristics on which the clusters were based were floor area, year of construction, number of floors, climate, and census region. For each group, a prototypical building was defined, using the mean values of the relevant properties of the member buildings. The source for the building characteristics was the Nonresidential Building Energy Consumption Survey database developed by the U.S. Energy Information Administration [8]. A summary of the salient features of the buildings appears in Table 1.

Bldg. No.	Floor Area (m^2)	No. of Floors	Year Built	Location	Floor Area Represented (10^6 m^2)	Air Change Rate w/ Fans Off (h^{-1})
1	576	1	1939	Indianapolis, IN	15.6	0.53
2	604	3	1920	Cleveland, OH	24.8	1.00
3	743	1	1954	El Paso, TX	21.5	0.43
4	929	2	1970	Washington, DC	26.5	0.33
5	1486	2	1969	Madison, WI	51.7	0.28
6	2044	2	1953	Lake Charles, LA	31.0	0.42
7	2601	4	1925	Des Moines, IA	68.2	0.65
8	3716	5	1908	St. Louis, MO	28.3	0.70
9	3902	2	1967	Las Vegas, NV	43.2	0.18
10	4273	3	1967	Salt Lake City, UT	35.5	0.21
11	13935	6	1968	Cheyenne, WY	28.6	0.19
12	16722	6	1918	Portland, OR	27.9	0.54
13	26941	11	1929	Pittsburgh, PA	58.5	0.62
14	26941	6	1948	Amarillo, TX	37.3	0.31
15	27870	12	1966	Raleigh, NC	32.7	0.22
16	28799	10	1964	Dallas, TX	22.9	0.18
17	53882	19	1965	Minneapolis, MN	27.6	0.26
18	67817	10	1957	Boston, MA	16.3	0.16
19	68746	28	1967	New York, NY	43.4	0.32
20	230392	45	1971	Los Angeles, CA	40.8	0.26
21	1022	2	1986[1]	Raleigh, NC	117.0	0.62
22	1208	2	1986[1]	Phoenix, AZ	92.2	0.58
23	1579	2	1986[1]	Pittsburgh, PA	101.0	0.53
24	38089	9	1986[1]	Pittsburgh, PA	64.5	0.21
25	46450	14	1986[1]	Charleston, SC	54.0	0.23

1. Each of buildings 21 - 25 represents a mix of construction in 1986 and 1995.

Table 1. Summary of Representative Building Set

This set of buildings has been the subject of previous studies of building energy consumption. The total heating and cooling coil loads experienced annually in each of the 25 buildings has been estimated using the DOE-2 building energy simulation program [3,4]. It was therefore possible to estimate the percentage of the total annual load that is attributable to infiltration.

3. DESCRIPTION OF APPROACH

The energy associated with infiltration in each of the buildings was estimated by summing the hourly infiltration load over one year. This analysis was performed with a program called AILOAD written in Microsoft® Visual Basic™. The algorithm for calculating infiltration loads for a given building consists of the following steps:

1. Obtain weather conditions for the current hour: outdoor temperature, humidity, and wind speed.

2. Determine the infiltration rate for the current hour, based on wind speed and HVAC system status.

3. Determine the appropriate thermostat setpoints of the HVAC system, based on the building occupancy schedule.

4. Compare the temperature of the outdoor air with the thermostat setpoints and building balance points to determine whether the infiltrating air needs to be heated or cooled.

5. If cooling is necessary, compare the humidity of the outdoor air to the desired humidity to determine whether latent cooling loads exist.

6. Calculate the hourly sensible and latent loads using equations (a) and (b).

   a) $Q_s = \rho * C_p * \Delta T * ACH * V$

   b) $Q_l = \rho * h_{fg} * \Delta W * ACH * V$

7. Add the hourly infiltration load to the cumulative total heating or cooling load.

In equations (a) and (b), Q_s is the sensible load due to infiltration, Q_l is the latent load, ρ is the density of the infiltrating air, C_p is the sensible heat capacity of air, h_{fg} is the latent heat capacity of air, ΔT is the indoor-outdoor temperature difference, ΔW is the indoor-outdoor humidity ratio difference, ACH is the infiltration rate in air changes per hour, and V is the total volume of the building. ACH * V is, therefore, the volume of outdoor air that enters the building in one hour. The specific data and other input parameters that are required at each of the steps are discussed in section 4.

The loads calculated in equations (a) and (b) are the space conditioning loads, indicating the amount of heat that must be added to or removed from the space to offset the heat loss or gain due to infiltration. In general, the total load on equipment is the sum of the conditioning loads for all the spaces it serves plus any losses in the air distribution system and any heat that must be added to or removed from ventilation air. Because the aim of this study is to assess the impact of infiltration only, the coil load is considered equivalent to the space heating load due to infiltration. Infiltration coil load intensities in MJ per m^2 of building floor area were calculated for each building. These values were compared to the total coil load intensities as predicted by the DOE-2 energy simulations in previous studies of these buildings. In order to convert the coil loads into energy use, some knowledge of the fuel types and efficiency of each building's HVAC system was needed. This information was drawn from the results of the previous studies [3,4], which calculated the energy use associated with the annual cooling and heating loads for each building. It was assumed that the energy required to meet the infiltration coil loads would be proportional to the energy required to meet the overall coil loads of the same building. Different ratios were used for heating and cooling energy estimates.

4. INPUT PARAMETERS

Implementation of this algorithm required specific information regarding the weather conditions, leakage characteristics of the buildings and HVAC system parameters. Much of the necessary information was provided by Briggs et al. [3] in the descriptions of the prototypical buildings and the input files for the DOE-2 energy simulations. Whenever possible, the parameter values for the infiltration load calculations were taken directly from the DOE-2 input files. However, in the cases of indoor humidity levels and building balance temperatures, no specific information was available, so additional assumptions were necessary. This section describes the important input parameters and the methods used to define their values.

4.1 Weather Data

Hourly weather data was provided by a WYEC (Weather Year for Energy Calculations) file for each of the 22 cities in which the prototypical buildings were located. Each file consists of a full year (8760 hours) of weather measurements, taken from U.S. Weather Service records for the month during which temperatures were closest to the long-term mean [9]. The specific data garnered from this source were the temperature, humidity, and wind speed for each hour of the typical year.

4.2 Infiltration Rates

Infiltration rates for each of the representative buildings were generated by Briggs et al. [3,4] for a wind speed of 4.5 m/s (10 m.p.h.) based on the age and height of the building and the average annual temperature difference. For the infiltration load calculations, as in the DOE-2 analysis, the baseline air change rates were adjusted hourly according to the current wind speed, assuming a linear relationship with zero infiltration in perfectly still conditions. No adjustment was made for the temperature difference across the building envelope; the baseline air change rates take into account the average influence of stack effects by including the building height and the average yearly temperature difference. However, the infiltration load estimation program used in this study allows specification of air change rates that vary with ΔT, and future analyses are planned to include this dependence.

The infiltration rates in Table 1 are valid when the HVAC system fans are off and at a wind speed of 4.5 m/s. During hours of system operation, the resulting pressurization of the building is assumed to limit the leakage of air through the building envelope. The previous DOE-2 analysis reflected this through reduced air change rates during the operating hours of the building. The amount of this reduction was based on the height of the building. For buildings of five stories or less, infiltration was reduced to 25% of the fans-off rate; in taller buildings, it was reduced to 50% of the fans-off rate.

4.3 Building Volume

Infiltration rates were multiplied by the building volume to calculate the amount of air entering the building during an hour. Building volumes contained a 90% correction factor that adjusted for the presence of unconditioned spaces, walls, and furniture within the building:

$$V = 0.90HA$$

A is the floor area represented by the building, and H is the floor-to-floor height.

4.4 HVAC System Parameters

Due to the effect of building pressurization on the infiltration rate, it was necessary to know whether or not the HVAC system fans were running during any given hour of the day. The operating hours for each building were derived from occupancy schedules developed by Briggs et al. [3], which were in turn based on hourly lighting and receptacle load data compiled during the End-Use Load and Consumer Assessment Program, a survey of electrical loads in commercial buildings in the Pacific Northwest [10]. The occupancy schedules used in the DOE-2 analysis contain hourly fractions of maximum occupant density, and the fans were assumed to be operating during all hours in which the scheduled occupancy was greater than 5% of the maximum. Each prototypical building was assigned one of five different schedules, which were scaled to reflect the average number of operating hours per weekday among the buildings represented, as reported in the NBECS [8]. On weekend days, the operating schedules were typically one hour shorter than during the work week.

The temperature setpoints reflected the common practice of changing thermostat settings in order to conserve energy at times when the building is unoccupied. Using the values as they appear in the DOE-2 input files, heating setbacks were 2.8 °C below the corresponding occupied-hours heating setpoint, which ranged from 21.1 °C to 22.2 °C. Setpoints for cooling fell between 23.3 °C and 25.0 °C. Cooling setups were fixed at 37 °C for every building, essentially ensuring that no cooling would occur during unoccupied hours. In general, setbacks and setups were in effect from the time the HVAC system fans cut off in the evening until one hour before they restarted in the morning. The existing building descriptions do not include a setpoint, per se, for the humidity of the indoor air. However, the input files for the system subprogram of DOE-2 include a listing for the maximum humidity of the system air. When calculating latent cooling loads, it was assumed that all infiltrating air that needed to be cooled was also dehumidified to the maximum level indicated for that building. The maximums were 70% relative humidity for the 20 original buildings, and 60% for the 5 buildings representing recent construction.

4.5 Balance Points

Another building parameter was introduced to account for the presence of internal heat sources, such as occupants, lighting, and electrical equipment. When the outdoor temperature is below the thermostat setpoint, infiltrating air may not need to be mechanically heated due to the heat generated by internal sources. The temperature above which this is true is called the balance temperature, or balance point, of the building. In order to account for the 'free' heating effect of a building's internal heat sources, a balance temperature was calculated for each of the representative buildings. If during any hour the temperature of infiltrating air fell between the balance temperature and the heating setpoint, no heating load was assessed. A balance temperature was estimated for each building, based on properties provided in the DOE-2 input files, using the following equation [11]:

$$t_{bal} = t_i - \frac{q_{gain}}{K_{tot}}$$

The total rate of heat gain, q_{gain}, includes internal sources such as occupants, lighting, equipment, solar gains through fenestration, and radiative gains through the walls and roof. K_{tot} is the total heat loss coefficient of the building (in W/K) due to infiltration, ventilation, and conduction. For the DOE-2 simulation, Briggs et al. [3] separated the representative buildings into distinct thermal zones. In general, each building comprised 5 zones: one interior zone and four perimeter zones, one facing each cardinal direction. If one assumes that heat transfer between the zones of a building is negligible, then each zone will exhibit its own characteristic balance temperature. Since most heat loss occurs across the building envelope, the limiting balance temperature (i.e., the highest) will be that of the zones having exterior walls. For this reason only the heat sources in the perimeter zones were included in the heat gain term when calculating the balance point for multizone buildings. For each building, separate balance points were calculated for occupied and unoccupied hours, based on the internal load intensities and schedules in Appendix C of reference [3]. Balance point temperatures for the 25 prototypical buildings ranged from -5.5 °C to 15 °C during the day, and from 10 °C to 17 °C at night, with averages of 4.5 °C and 14 °C, respectively. These ranges are comparable with the values of 1.1 °C and 11.1 °C calculated by Norford [12] for a modern, 3-story office building.

5. RESULTS

The results of the infiltration load calculations appear in Table 2. For each of the 25 buildings, the infiltration load estimates are shown, along with the total annual heating or cooling load predicted with DOE-2 [3,4] and the percentage of this total that is due to infiltration. Note that these values are the loads on the heating and cooling coils, and not the actual energy consumption, which depends on the source of energy.

The results indicate that, nationwide, infiltration is responsible for about 15% of the total annual heating load of the office building stock, but only 1% of the cooling load. The heating and cooling percentages are different because of the different extent to which these loads depend on ΔT. Heating loads arise from heat loss due to ventilation, conduction, and infiltration, all of which depend on ΔT. On the other hand, cooling loads have a substantial contribution from internal gains and solar gains, which do not depend on ΔT. Thus the portion of the total load that arises from ΔT-driven mechanisms, including infiltration, is smaller for cooling than for heating.

BLDG	LOCATION	HEATING LOADS (MJ/m^2)			COOLING LOADS (MJ/m^2)		
		Total	Inf.	% of Total Due to Inf.	Total	Inf.	% of Total Due to Inf.
1	Indianapolis, IN	656.0	104.6	16%	233.8	4.6	2%
2	Cleveland, OH	2127.0	345.9	16%	355.3	14.4	4%
3	El Paso, TX	162.3	25.5	16%	429.0	4.0	1%
4	Washington, DC	340.5	34.1	10%	355.3	3.7	1%
5	Madison, WI	313.3	44.8	14%	254.2	1.5	1%
6	Lake Charles, LA	120.3	19.5	16%	620.8	13.2	2%
7	Des Moines, IA	1087.3	151.2	14%	400.7	6.6	2%
8	St. Louis, MO	744.6	104.2	14%	763.9	24.3	3%
9	Las Vegas, NV	132.8	15.5	12%	420.0	4.0	1%
10	Salt Lake City, UT	225.9	24.7	11%	547.1	1.5	0%
11	Cheyenne, WY	382.5	64.7	17%	534.6	1.0	0%
12	Portland, OR	724.1	69.5	10%	198.6	1.3	1%
13	Pittsburgh, PA	1357.5	78.2	6%	615.2	5.2	1%
14	Amarillo, TX	190.7	72.9	38%	516.4	6.4	1%
15	Raleigh, NC	639.0	14.7	2%	1208.8	5.7	0%
16	Dallas, TX	185.0	20.3	11%	1087.3	11.0	1%
17	Minneapolis, MN	651.5	70.1	11%	479.0	3.0	1%
18	Boston, MA	990.9	45.7	5%	989.7	1.2	0%
19	New York City, NY	232.7	84.4	36%	291.7	4.3	1%
20	Los Angeles, CA	65.8	6.3	10%	999.9	0.2	0%
21	Raleigh, NC	97.6	42.7	44%	565.2	8.6	2%
22	Pheonix, AZ	48.8	12.9	26%	363.2	12.4	3%
23	Pittsburgh, PA	155.5	71.6	46%	184.4	2.7	1%
24	Pittsburgh, PA	48.8	41.6	85%	245.7	2.2	1%
25	Charlotte, SC	63.6	17.1	27%	443.8	14.3	3%
	All Buildings	380.2	58.5	15%	494.3	6.4	1%

Table 2. Annual Heating and Cooling Loads

A closer look at the results for individual building categories reveals that the percentage of the heating load due to infiltration varies from building to building. In particular, the estimated percentage for all five of the recent building classes (21 through 25) are significantly above the mean of 15%. In the DOE-2 analysis, these buildings were assumed to meet the building energy efficiency guidelines of ASHRAE Standard 90.1-1989. The more stringent envelope insulation values prescribed therein decrease conductive losses, making infiltration loads a higher percentage of the total. In buildings 13 and 15, infiltration causes a far smaller percentage of the heating load than average, partly because the HVAC systems of these buildings operate 24 hours per day. This has the dual effect of eliminating thermostat setbacks, thus increasing the total heating coil load, and reducing the infiltration loads because the building is pressurized day and night.

The results in Table 2 were calculated assuming that air exchange rates were reduced by one half or three quarters during hours of fan operation, depending on the height of the building. In actuality the relationship between HVAC system operation and infiltration is not nearly so simple. These reductions were intended to reflect the fact that in some buildings, the systems are designed to maintain positive pressure inside the building, eliminating infiltration entirely. Since, in reality, the ability of an HVAC system to maintain positive pressure varies from building to building, it is informative to look at two extreme cases. The first assumes that the buildings are completely pressurized while the system fans are running, eliminating any infiltration during occupied hours. In this case, the mean heating load due to infiltration drops to 9% of the total annual heating load, and the cooling load due to infiltration is effectively eliminated. On the other hand, if it is assumed that infiltration is unabated during hours of fan operation, the portion of the total heating load attributable to infiltration climbs to 20%; the portion of the cooling load increases to 4%.

6. Sensitivity Analysis

Given the approximate nature of many of the inputs to the infiltration load calculations, a sensitivity analysis was performed to determine how the uncertainty in the inputs affects the results.

6.1 Concept

Sensitivity analysis is a statistical technique which measures the relative importance of each input parameter in terms of its effect on the output. The importance of each variable x_i is represented by its 'main effect' - the percentage change in the output, y, as x_i changes from its lowest value to its highest value [13]. It is also possible to determine the effect of a nonlinear interaction between two or more variables. The effects are determined by running the simulation numerous times while systematically varying the values of the input parameters. In a factorial design, each input is assigned one low and one high value, or level, and with every run one variable is toggled between its low and high level. For n variables, this method requires 2^n runs to exhaust all combinations.

A fractional factorial design is a way to reduce the number of runs by varying more than one parameter with each run [14]. Reducing the number of runs introduces a certain amount of ambiguity to the results of the analysis: certain effects, as calculated, will actually represent the sum of the effects of more than one variable or interaction. The confounding pattern is known, however, so with some knowledge of the physical processes involved in the algorithm, reasonable conclusions can be drawn about which variable's effect is represented by each coefficient.

6.2 Experiment Design

The experiment measured the sensitivity of both the nationwide annual heating load and nationwide annual cooling load to eight different input parameters. The variables and their levels are listed in Table 3. The levels of the variables are given as a range above and below their nominal values, which vary from building to building. In some cases, the nominal values are different from the input values described in section 4, e.g., the cooling setup value. Therefore, the results of the sensitivity analysis provide insight only into the relative impact of the inputs and not into the uncertainty of the estimate of the energy use due to infiltration presented earlier.

The ranges are intended to be large enough to include all but the most extreme cases for each variable. By using a half factorial design when assigning variable levels for each simulation (a 2_{IV}^{8-4} design [14]), the number of runs was reduced to 16 (from 256 for a full factorial design). Interactions between three or more parameters were assumed to be negligible. By recognizing that certain variables could influence only the heating load and others only the cooling load, all two-factor interactions were isolated from their confounding effects

Number	Parameter	Nominal Value	Range
1	Heating Setpoint	21.1 °C - 22.2 °C	± .55 °C
2	Cooling Setpoint	23.3 °C - 25.0 °C	± .55 °C
3	Heating Setback	2.8 °C below setpoint	± 2.8 °C
4	Cooling Setup	2.8 °C above setpoint	± 2.8 °C
5	Balance Point	As calculated	± 2.8 °C
6	Maximum Humidity	60%	± 10%
7	Operating Hours/day	14 - 16 hours	± 2
8	Volume Correction	0.9	± 0.1

Table 3. Variables and Levels for Sensitivity Analysis

6.3 Results

Figure 1 summarizes the main effects of the variables listed in Table 3 with respect to the total annual infiltration loads for all 25 buildings. The values in Figure 1 are the percentage change in the output when each input is varied from its lowest to its highest level, as given in Table 3. The effect of each variable on the individual building loads varied widely from building to building, but in general the values of the heating setback and the balance point temperature had the greatest influence on heating loads. The largest changes in cooling loads were a result of varying the humidity setpoint and the thermostat setpoint and setback. The effect of the volume correction factor was nearly the same for all buildings; the 22% change in the output reflects a linear effect due to varying the effective volume of the building by ±11%. An overall level of uncertainty was estimated for the nationwide annual infiltration load estimates based on the nominal inputs in Table 3, by taking the square root of the sum of the squares of each of the main effects. This yielded an uncertainty of 44% for heating loads and 70% for cooling loads. As stated earlier, these uncertainty estimates do not apply to the infiltration loads presented in Section 5 due to some slight differences in the input values. Again, the overall uncertainty for individual building loads varied widely among buildings; between 36% and 83% for heating, and between 45% and 95% for cooling.

One important parameter was excluded from the sensitivity analysis - the infiltration rates. Based on equation (a) it is clear that if the air change rates were all adjusted by the same amount regardless of weather conditions, it would have a linear effect on the output, in the same way that the variation of the building volume does. Therefore the uncertainty in the output due to this parameter is the same as the uncertainty in the input parameter itself, which in the case of infiltration rates is relatively large.

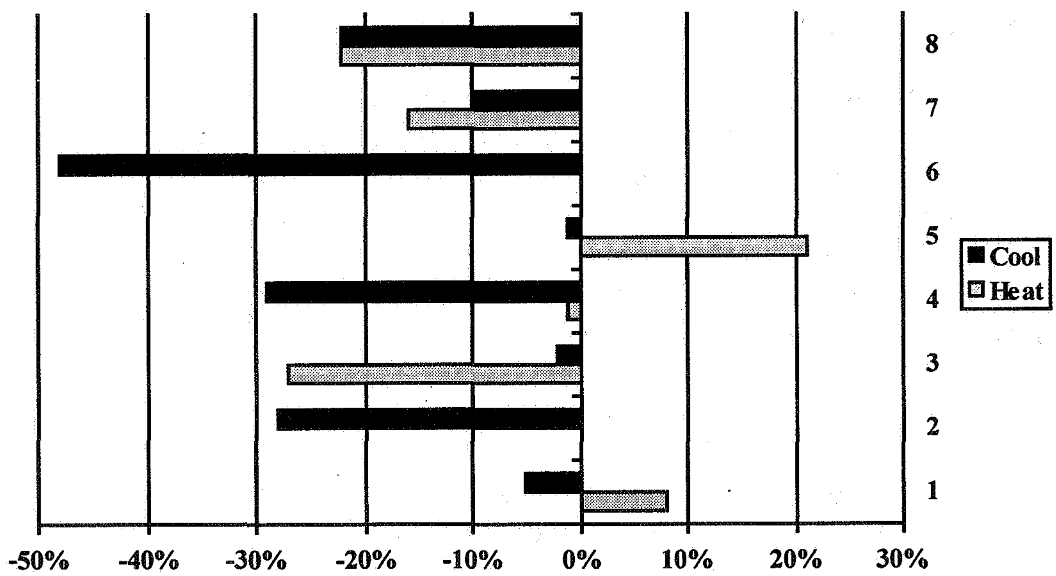

Figure 1. Main Effects of 8 Parameters on Nationwide Annual Infiltration Loads

7. Discussion

The earlier DOE-2 analysis of these buildings [3,4] includes an estimate of the annual energy use accounting for conversion efficiencies of HVAC system components and the source of energy. The energy used to cool and heat each building for a year was multiplied by the ratio of infiltration loads to total conditioning loads in order to estimate the annual energy cost of infiltration. For cooling, the total infiltration energy for all 25 buildings was 2.5 PJ (1 PJ = 10^{15} J), as compared to the total cooling usage of 145 PJ, i.e., infiltration was responsible for 2% of the cooling energy consumption. For heating, infiltration consumed 70 PJ or about 18% of the total of 410 PJ. Considering only the buildings constructed over the last 10 years (buildings 21 - 25), the portion due to infiltration is 45% of the heating energy, showing the increased impact of infiltration in newer, better insulated buildings. According to the Energy Information Administration [6], office buildings consumed a total of 1.3 EJ (1 EJ = 10^{18} J) of energy in 1989. Altogether, commercial buildings of all types consumed 6.1 EJ of site energy in 1989, 2.1 EJ of which went toward space heating. Assuming the portion of heating energy use due to infiltration is 18% for all commercial buildings, the nationwide cost of air leakage in commercial buildings is 0.38 EJ.

The accuracy of this estimate is limited by input uncertainty and the crude approach used to estimate infiltration rates. A sensitivity analysis of eight system parameters and building properties, detailed in section 6, revealed an overall uncertainty of 44% in the total heating load estimate. The assumptions made regarding infiltration rates are another source of uncertainty. The weather dependence of the air change rates was represented crudely, not taking into account the temperature difference across the building envelope. The interaction between air leakage and system operation was simplified to a constant reduction of infiltration during system operating hours.

Despite the large overall uncertainty of the infiltration energy estimates, they indicate that air leakage may be responsible for a significant portion of the energy used in U.S. office buildings. In order to estimate to what extent this energy usage could be reduced, more sophisticated methods of analyzing infiltration energy costs are necessary. The next phase of this project will involve using a building energy simulation program combined with network airflow analysis to account for the dependence of air change rates on weather conditions and on the interactions between system operation and infiltration.

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Implementing the Results of Ventilation Research
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TWINFACE - First Results About a New Developed Double Facade System

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Abstract for

16th AIVC Conference

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Title: TWINFACE - first results about a new developed double facade system

Abstract:
A modern two-storey office building in Berlin was designed and built with a special double facade. The thermosyphoning facade, type TWINFACE, consists of box-windows with fresh air inlet vents and adjoining vertical shafts to exhaust the room air. The facade should insure energy efficient and comfortable natural ventilation throughout the year. It has its main potential in high rise office buildings. Openable windows, low noise entry, weather-protected solar blinds and passive night cooling are some features of the design.

The building is equipped with a data aquisition system to monitor air temperatures inbetween the facade and in the rooms, window opening angle, air change rates and other parameters. One aim of the project is to validate a new developed simulation code, which consists of the dynamic thermal building simulation programme 'APACHE' coupled with the multi zone infiltration programme 'TWIN'. The paper outlines results of the first measurement phase and compares predicted and measured data.

The second aim is to improve the facade design and functioning. New ideas are introduced.

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Design and Installation Guide of Passive Stack Ventilation Systems in Retrofitted Apartment Buildings

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SYNOPSIS

Almost 60% of French residential buildings were built before the seventies, and an important part of those is to be retrofitted for complying with new needs with regard to acoustic insulation and energy saving.

Retrofitting modifies the airtightness of the building envelope and can lead to an insufficient air change rate in passive stack ventilated buildings ; the existing ventilation system has therefore to be redesigned in order to insure adequate indoor air quality.

Dimensioning of the stack effect ventilation system for multi-storey dwellings is a critical issue, as the air flow rate depends on many parameters as outdoor temperature, wind, distribution of air inlets and envelope air leakages, ducts area and length, characteristics of outlets and cowls.

Research work has been initiated by the CSTB, GDF and SOCOTEC (commissioning authority) to produce a design and installation guide of ventilation system and flue system for gas appliances by passive stack effect in apartment buildings. The guide is aims at helping engineers, building consultants and professional building owners to properly choose and dimension the ventilation system based on general and continuous air renewal.

Results of this research work and main features of the guide are presented in the paper, in a first part the suitable system design is given according to the typology of the existing building and the different measures to be taken are listed. The choice and dimensioning of ventilation components are operated in order to comply with indoor air quality requirements. Dimensioning was derived from simulation results obtained by a numerical model developed at CSTB, predicting the ventilation rates, which has been described in the 15th AIVC Conference paper. The dimensioning of the ventilation system depends on the stack height and section area of ducts.

The guide also deals with requirements and specifications concerning gas appliances, schedule of operation conditions of renovation work taking into account personal protection during this work, installation, commissioning and maintenance, use instructions for the occupants and building managers.

LIST OF SYMBOL

St	sum of exhaust ducts cross sectional areas (cm^2)
n1	height between the highest floor of the group of dwellings and the cowl (number of floors)
n2	number of floors served by the same group of ducts (if individual ducts n2=1)
n	number of habitable rooms in the dwelling (the living room is taken as two rooms)
np	number of habitable rooms in the group of dwellings (np = n2 x n)
C	suction coefficient of the cowl
ζ_c	pressure drop coefficient of the cowl

1. - INTRODUCTION

In the French apartment building stock two thirds of buildings were constructed before 1971. A third of the existing apartment building stock is composed of dwellings without ventilation system, another third with passive stack ventilation system. The apartment building stock is in need of renovation for various reasons : refurbishment of interiors, creation of new service rooms, retrofitting of heating installation, replacement of windows, insulation of façades,...

Today, retrofitting measures represents a significant proportion of the total construction market and concern a lot of passive stack ventilated existing buildings. Energy saving measures make the building envelope so airtight that hygro-thermal behaviour of the building can be completely modified and ventilation operation can be disrupt. Attention must be paid to attaining an acceptable air quality in dwellings by ensuring a minimum ventilation rate.

The renovation measures (in particular airtightness measures) often modify the state of ventilation so much that the requirements with regard to indoor air quality, comfort, security and risk of moisture damage do not meet anymore. Thus the ventilation shall be carefully considered when selecting renovation measures. A new ventilation system has to be designed, reusing the existent ducts in order to comply with indoor air quality, comfort and security requirements.

In order to assist engineers and building professionals owners and building consultants in obtaining a satisfactory ventilation system, a guide has been produced [1].

A CSTB's numerical model [2] was adapted and used to perform numerous simulations for dimensioning the ventilation system (more than one thousand runs). Recommendations based on a simplified description of the existing building were derived from this work.

The guide gives practical solutions which can be achieved from a typology of the existing dwellings and installations.

2. - TYPOLOGY

Right from the beginning of the century, French regulations have been the mover of the technological progress in ventilation and the present state of ventilation in France is the result of a long sequence of habits and requirements [3]. The following typology of existing apartments can be done according to the date of completion of buildings.

before 1906 : no regulation on flue or ventilation systems ; dwellings can be without any chimney.

from 1907 to 1937 : Individual flue system is required in the kitchen and in each habitable room of the dwelling. No regulation on ventilation : airing by window opening, by air leakage of the building envelope and by flues.

from 1937 to 1955 : Individual flue system is required in the kitchen and in each habitable room of the dwelling. Continuous ventilation of the toilet is required.

from 1955 to 1958 : Shunt duct is allowed for exhaust combustion products and polluted air in service rooms.

from 1958 to 1969 : Individual or collective flue system is required in the kitchen and only one flue system is required for three habitable rooms. Ventilation operates room by room : the habitable rooms are ventilated by infiltration and exfiltration through the envelope, the kitchen by a chimney and/or ventilation duct, the bathroom and toilets by two ventilation ducts in lower and higher locations.

from 1969 to 1982 : No more requirement about flue system. General and continuous ventilation is required. Fresh air enters habitable rooms via air inlets and polluted air exhausts from service rooms via exhaust vents. Two systems can be used : passive stack ventilation or mechanical exhaust.

since 1982 : General and continuous ventilation principle remains but the ventilation rate is modified.

3. - GENERAL APPROACH

Ventilation strategy applied to retrofitted dwellings is the general and permanent ventilation with air flow route control. Fresh air is distributed in habitable rooms and polluted air is expelled from service rooms through the exhaust vents : air circulates from the least polluted rooms to the most ones. The guide is intended to provide solutions as easy as possible with dimensioning calculations reduced to the essential. It is applied to the ventilation systems and flue systems of gas appliances operating by passive stack effect in apartment buildings. Passive stack ventilation is achieved by using the former flues as ventilation ducts as far as possible.

Each stage from design ventilation system up to maintenance via achievement is described in the guide (cf. figure 1). At the design stage, the technical solutions to be implemented from the existing building typology, the choice and the dimensioning of ventilation components are given in the guide. The design of ventilation system requires a prerequisite diagnosis of the installation and a consistent choice of components. At the realization stage, the guide explains how to implement the different ventilation components and operate the renovation work taking into account personal protection. The last part of the guide deals with commissioning, maintenance and use instructions for the occupants and building managers.

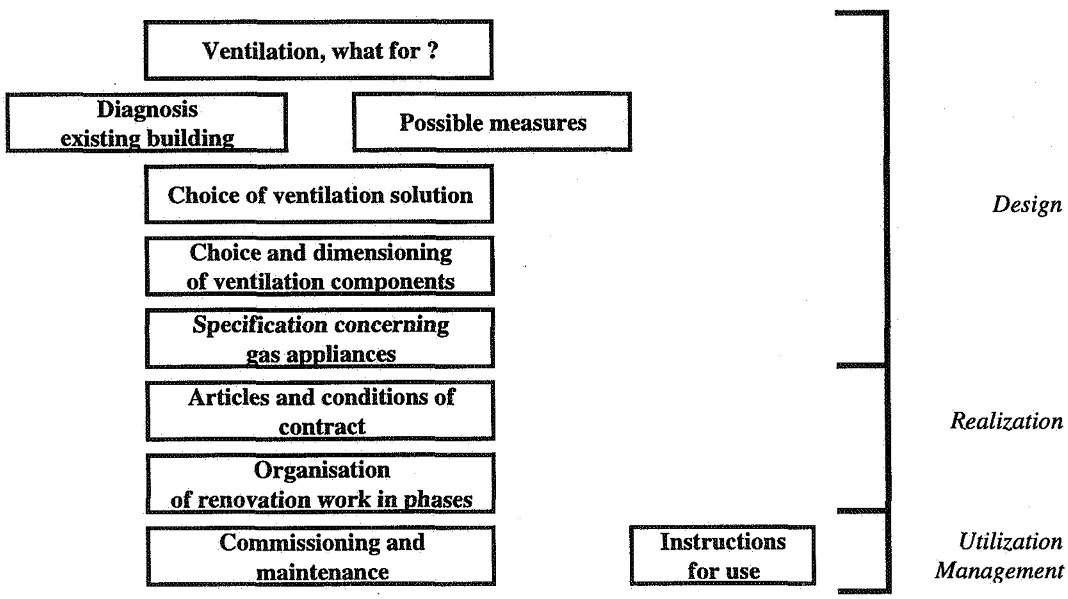

Figure 1: different stages in renovation work

4. - DIMENSIONING CALCULATION

Each component of the ventilation system has to be properly dimensioned in order to obtain satisfactory ventilation.

To make the work easier for the guide users the area values of the air inlets, the inter-rooms openings and the air outlets to be used are directly given in the guide and are the same for all the dwellings ; to limit the energy losses due to ventilation, the designer can choose self-regulated or humidity-controlled outlets instead of fixed outlets.

inlets : one self regulated inlet in each habitable room ; equivalent area 30 cm^2 (the living room is taken as two rooms).

inter rooms openings : kitchen door 160 cm^2, other doors 80 cm^2.

outlets : in the kitchen, a gas boiler is connected to a flue ; equivalent area 120 cm^2
in toilets and bathroom one outlet grille; equivalent area 75 to 150 cm^2
with possibility of self regulated or humidity controlled outlet for saving energy (all the outlets grilles mounted in the building are the very same).

In France, in buildings built after 1906 ducts exist : consequently there is no, strictly speaking, need to design and dimension the ducts ; however section area and height of ducts have to be considered in this design stage.

Extract flow rate depends on the sum of exhaust ducts cross sectional areas (**St** in cm^2) divided by the number of habitable rooms in the group of dwellings (**np.n2**) and the height of air column between the highest floor of the group of dwellings and the cowl (**n1** given in number of floors).

The solutions given in the guide comply with the ventilation requirements in terms of minimal exhaust flow rates during the most part of the heating period. Six different ventilation systems (depending on characteristics of outlets and cowls) are taken into account :

system 1 : fixed exhaust grilles and cowl class B
(French standard P 50-413 : $\zeta_c \leq 2$ and $C \leq -0.65$).

system 2 : self regulated outlet or humidity controlled outlet and cowl class B

system 3 : fixed exhaust grilles and motorised cowl class B
(these cowls are motorised and can give an additional pressure of 15 pascals when necessary).

system 4 : self regulated outlet or humidity controlled outlet and motorised cowl class B

system 5 : fixed exhaust grilles and two speeds motorised cowl class B
(these cowls are motorised and can give an additional pressure of 15 pascals when outdoor conditions are insufficient and an additional pressure of 35 pascals during cooking time).

system 6 : self regulated outlets or humidity controlled outlets
and two speeds motorised cowl class B.

An example of solutions for collective ducts is given in figure 2.

n1 from 0 to 3

	System					
St / np n2	1	2	3	4	5	6
< 20			-5°C	-5°C	-5°C	-5°C
[20 to 25 [			0°C	0°C	0°C	0°C
[25 to 30 [			4°C	4°C	4°C	4°C
[30 to 35 [			8°C	8°C	8°C	8°C
[35 to 40 [			12°C	12°C	12°C	12°C
[40 to 45 [			16°C	16°C	16°C	16°C
[45 to 50 [			16°C	16°C	16°C	16°C
[50 to 70 [			16°C	16°C	16°C	16°C
≥ 70			16°C	16°C	16°C	16°C

n1 7 and more

	System					
St / np n2	1	2	3	4	5	6
< 20			8°C	8°C	8°C	8°C
[20 to 25 [			12°C	12°C	12°C	12°C
[25 to 30 [			16°C	16°C	16°C	16°C
[30 to 35 [			16°C	16°C	16°C	16°C
[35 to 40 [			16°C	16°C	16°C	16°C
[40 to 45 [			16°C	16°C	16°C	16°C
[45 to 50 [			16°C	16°C	16°C	16°C
[50 to 70 [			16°C	16°C	16°C	16°C
≥ 70			16°C	16°C	16°C	16°C

In each box you find the outdoor temperature at which the motorised cowl interlocks.

not recommended inadequate air quality

acceptable acceptable air quality or
adequate air quality but immoderate energy losses

recommended adequate air quality
without immoderate energy losses

Figure 2: Example of exhaust ventilation system dimensioning

5. - ACHIEVEMENT AND GUIDELINES

5.1 - Diagnosis

This chapter of the guide gives the check list of the parameters to be taken into account :

- year of construction of the building,
- existing ducts (number, state - air tightness, emptiness -, location, ...)
- existing cowls (characteristics, location on the roof)
- gas equipment
- reusable ventilation equipments (inlets, transfer grilles, outlets, ...)

5.2 - Conditions of contract

This chapter describes who does what and indicates the documents to be produced by the parties involved.

5.3 - Organisation of renovation work in phases

The objective is to insure the security of the occupant during the work ; the chapter gives the sequential organisation of the work :

- put the ducts to rights,
- take up the old outlets,
- put in the new outlets,
- connect the gas appliances,
- change windows (possibly),
- put in the inlets in main rooms,
- put in the inter room openings,
- take up the inlets in the service rooms and stop up the openings,
- carry out the self inspection check list.

5.4 - Commissioning and maintenance

This chapter gives directions for use addressed to for users and managers.

6. - EXAMPLE OF APPLICATION

6.1 - Description of the building

Five storey building built in 1958.

Kitchen with inlet grille (VG) and smoke flue (SF) ; bathroom and toilets with windows.

One smoke flue in the living room (not used)

All the smoke flues are shunt type flue systems (collector 20 x 20 cm)

6.2 - Propounded solution

The two smoke flues are in good state : they can be reused

The bathroom and toilet will be linked with an horizontal duct to the living room smoke flue which becomes a ventilation duct (VD).

Data of the problem :

- $n2 = 5$ $n1 = 0$.
- $St = 2 \times (20 \times 20) = 800\ cm^2$
- 3 principal rooms : $np = 3 + 1 = 4$
- $St / np\ n2 = 40$

It is not necessary to use motorised cowls (see table figure 2) if an acceptable solution is provided for but a motorised cowl can be used, interlocked at 12°C outdoor temperature for a better air quality.

Inlets, inter-room openings and outlets are dimensioned according to paragraph 4

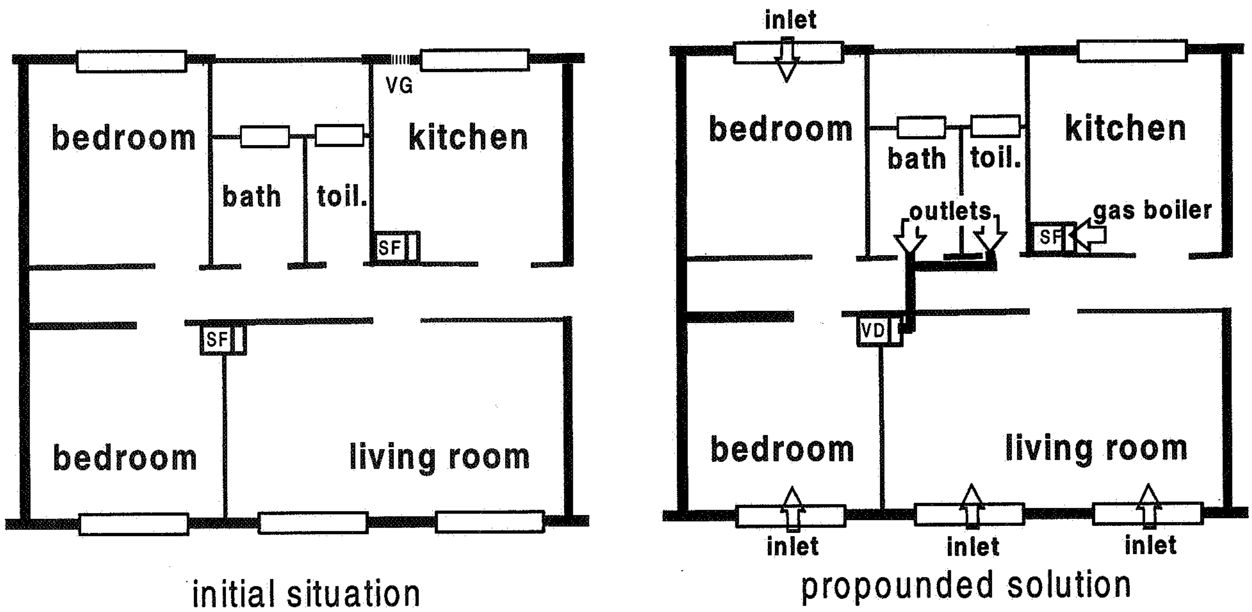

figure 3 : dwelling before and after retrofitting

7. - CONCLUSION

A methodology to design and achieve passive stack ventilation systems for retrofitted apartment buildings was developed and led up to the production of a guide. The guide which was issued in April 95 meets success because it fulfils requirements of engineers and consultants. Indeed, before its issuing there was no practical document to assist the building professionals in achieving proper passive stack ventilation. The technical solutions given in the guide involve ventilation components available on the French market. Nevertheless, the main features of this guide could be applied in other countries considering national building typologies, requirements and habits.

ACKNOWLEDGEMENTS

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The Impact of Various Ventilation Remedies on Radon Levels and Local Building Environment in a UK Test House - Some Preliminary Results

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THE IMPACT OF VARIOUS VENTILATION REMEDIES ON RADON LEVELS AND LOCAL BUILDING ENVIRONMENT IN A UK TEST HOUSE - SOME PRELIMINARY RESULTS

Paul A Welsh

1.0 Summary

The Building Research Establishment is currently investigating the impact of various radon remedies at a radon affected test house. Tests aim to assess how different ventilation strategies affect indoor radon levels and the building environment. Those examined include natural underfloor ventilation, mechanical underfloor ventilation (supply and extract), and whole house pressurisation.

The test house has a suspended timber floor with an inaccessible underfloor space and is typical of much of the UK housing stock except for indoor radon levels regularly in excess of $1000Bqm^{-3}$. It is fitted with a comprehensive range of monitoring equipment which closely records important environmental parameters.

This paper presents some preliminary results including radon levels and whole house ventilation rates. Mechanical underfloor extract ventilation is shown to produce a 94% reduction in indoor radon levels, the greatest recorded. Mechanical underfloor supply ventilation causes the largest change in house ventilation rate with an approximate energy cost of £50 per year. These results, and future data, will allow for the assessment of remedies to be based on their total impact to the building rather than radon reduction alone.

2.0 Definitions

Airbrick: a purpose built vent in the external wall of an underfloor space to promote natural underfloor ventilation.

Underfloor space: the space beneath a suspended floor. In the US this is often referred to as a crawl-space. Underfloor spaces in the UK are usually inaccessible and poorly ventilated.

Underfloor extract ventilation: mechanical extract ventilation of the underfloor space. Air is discharged to outside.

Underfloor supply ventilation: mechanical supply ventilation of the underfloor space using outside air.

3.0 Introduction

Radon remedies for houses with inaccessible underfloor spaces often involve some form of underfloor or whole house ventilation [1]. Very large radon reductions have been achieved using these approaches [2] but in general they are not well demonstrated or well understood. In addition there are many concerns about possible side-effects such as the cost of changes in house ventilation rates, the risk of freezing to underfloor water pipes and the possibility of combustion products from combustion appliances spilling into living areas.

To help increase our understanding of radon remedies involving ventilation BRE (the Building Research Establishment) have purchased a radon affected house with a suspended timber floor. Controlled tests will help answer a number of important points allowing for a more informed choice of solution, based not only on radon reduction as is often the case.

4.0 The test house

The house is typical of much of the UK housing stock being semi-detached (internal volume of about 200 m^3) with a timber floor suspended about 0.35 m above the soil. It was built in the 1930's and has a slightly higher than typical UK leakage [3] of about 17.6 air changes at 50 Pa. Figure 1 shows a plan of the ground floor.

The tongue and grooved timber boards sit on timber joists which in turn, sit on a timber wall plate. Generally the floor is in good condition expect for a cupboard area where there are signs of advanced wood rot and large gaps between the floor boards. The living room, dining room and hall area are covered with carpet tiles which are taped in position. The kitchen has a poor linoleum finish. The area beneath the stairs is bare.

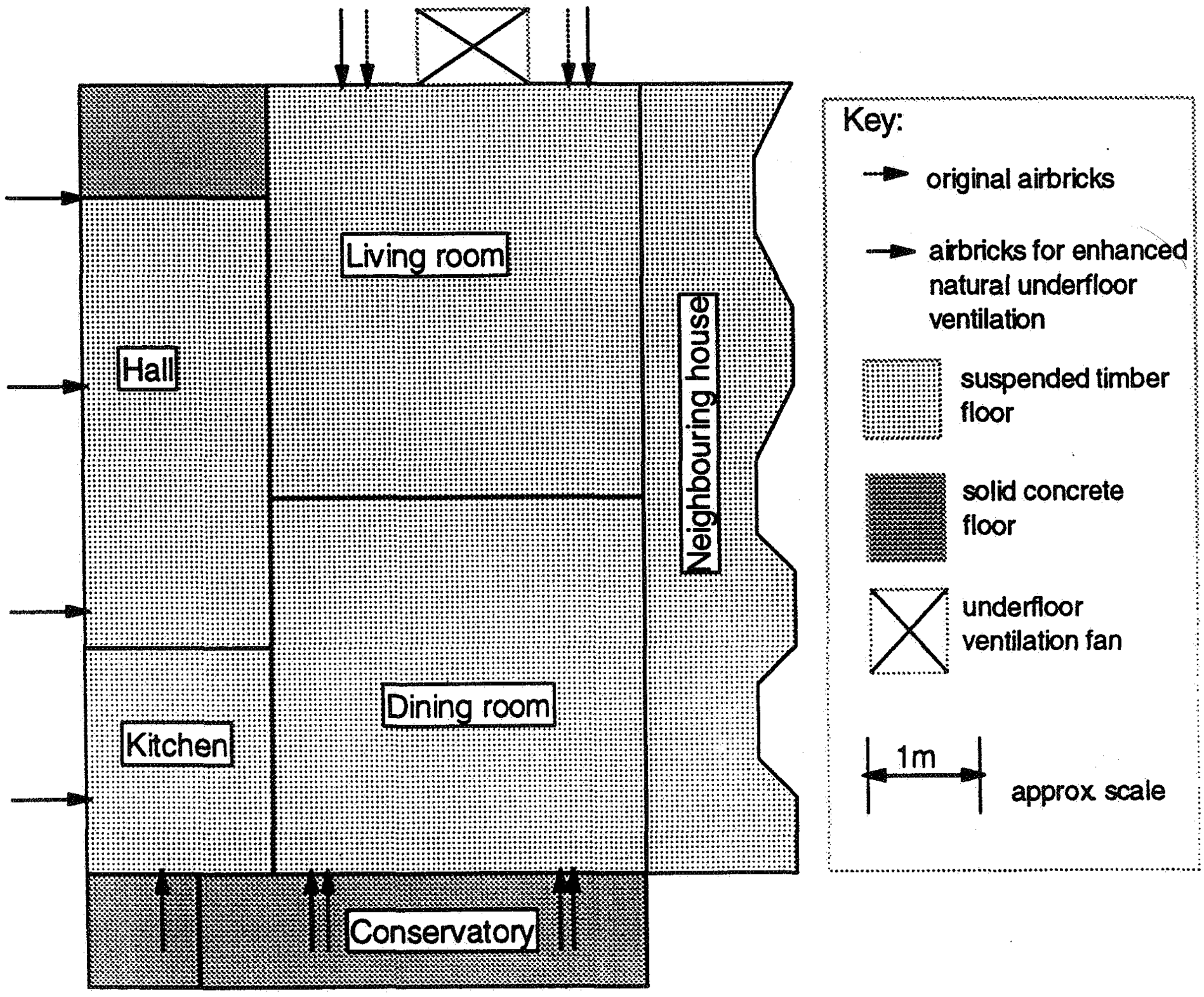

Figure 1: Ground floor plan of test house

The underfloor space is split into four sections by underfloor walls but cross ventilation is promoted through purpose built gaps. The underfloor volume is about 14 m^3. When purchased there were four airbricks, two at the front which were largely blocked, and two at the rear venting into the conservatory. The free area of these [4] is estimated to be about 10,000 mm^2.

During the heating season the temperature within the house is controlled using a gas central heating system on a timer and remote thermostatic radiator valves. Downstairs is at about 21 ^{0}C with upstairs being a couple of degrees cooler. A small humidifier maintains the living room relative humidity above 45%.

5.0 Monitoring equipment

Living room and living room underfloor radon levels are continuously monitored using 'Alphaguard' units. These units use a pulse ionisation chamber to monitor radon levels and have an accuracy of about ±7%.

A separate data-logger continuously monitors a number of environmental parameters. These include:

- 11 temperatures (4 underfloor, 4 ground floor, 1 bedroom, 1 loft and 1 external)
- 3 relative humidities (living room, living room underfloor, external)
- atmospheric pressure
- 4 floor pressure differences (each ground floor room)
- 4 room to outside pressure differences (each ground floor room)
- wind speed and wind direction

All but the wind direction are monitored every two minutes and half hourly averages are stored. The wind direction is monitored as a spot reading every half hour.

In addition to the above, timber moisture levels at eight locations are recorded manually before and after tests. Three locations are at the front of the living room, three at the rear of the dining room, and two beneath the stairs. For each location the moisture level is recorded for the joist top, joist bottom, and wall plate.

6.0 Radon remedies

The radon remedies installed in the house are discussed below.

6.1 Whole house pressurisation

A fan mounted in the loft space forces air from the loft, into the area at the top of the stairs. The fan has a maximum flow rate of about 180 m^3h^{-1}. It is claimed that these systems reduce radon levels by positively pressurising the house with respect to the soil. However field experience carried out by BRE has shown that the pressure changes are often very small, commonly being less than 1 Pa. This small amount of pressurisation indicates that the increase in house ventilation rate may be responsible for a significant proportion of any radon reduction.

6.2 Enhanced natural underfloor ventilation
Initially the underfloor space was poorly ventilated by four airbricks with a total free area of about 10,000 mm^2. To enhance the natural ventilation the number of airbricks was increased from four to nine, increasing the free area to about 41,000 mm^2. They are positioned on three walls (Fig. 1) and can be open or closed according to requirements.

Enhancement of the natural underfloor ventilation can reduce indoor radon levels by lowering the level of natural depressurisation and by dilution.

6.3 Mechanical underfloor ventilation
A fan at the front of the house (Fig. 1) is used to force the ventilation of the underfloor space. It is connected to a pipe which runs to beneath the centre of the living room. The inclusion of the pipe aims to increase the area of influence and helps prevent air short-circuiting the system by passing through any airbricks. The airbricks can be open or closed according to requirements. Opening them increases the underfloor ventilation and decreases any pressure effects. With the airbricks closed the converse is true.

The fan can be set to obtain the following flow rates: 430, 390, 195 and 75 m^3h^{-1}. Thus the underfloor air change rate can be varied from 5 to 30 per hour. This air change rate will not be uniformly distributed throughout the space because of the irregularity of the air flow paths and fan pressure distribution.

Extract ventilation draws air from beneath the floor and discharges it outside. It reduces indoor radon levels by two mechanisms namely dilution and changes in pressure differences. Dilution takes place as the extracted underfloor air is replaced by 'radon-free' air. The pressure effect reduces, or ideally reverses the floor pressure difference, which reduces the radon flow through the floor. These two beneficial responses are partially offset by an increase in the level of underfloor depressurisation which increases the pressure driven radon entry rate from the soil.

Supply ventilation blows air from outside to beneath the floor. The mechanisms responsible for the radon reduction are dilution together with pressurisation (which reduces the pressure driven radon entry rate from the soil). This reduction will be partially offset by an increase in the air flowing from the underfloor space to the rooms above.

7.0 Test details

7.1 Radon reduction study
Remedies are left running for a period of a least 25 days to obtain sufficient data for reliable assessment. During the test period all internal doors are left open and all external windows and doors are closed. The house is left undisturbed.

To date five different scenarios have been closely monitored. The first is with the house 'as purchased' without any remedy and with poor natural underfloor ventilation. Second is the whole house pressurisation system operating on the house in the 'as purchased' condition. This was followed by enhanced natural underfloor ventilation. The final two tests were on underfloor extract and supply ventilation. For both of these the fan was set to full speed and all of the airbricks from the enhanced natural ventilation were left open.

Radon reductions caused by the different remedies are directly compared to each other. However since the tests are held at different times of the year any seasonal variations in radon levels will be superimposed on the results. To assess the level of seasonal variation, remedies will be tested twice at different times of the year. To date this has only been performed for enhanced natural underfloor ventilation.

7.2 Impact of remedies on ventilation rate

Ventilation tests were held separately to the testing mentioned above. A gas analyser and injection system continuously monitored the fresh air change rate of the whole house by injecting a tracer gas, Sulphur Hexafloride, to two downstairs areas and two areas upstairs at such a rate as to maintain a constant concentration of 10ppm. The gas concentration was monitored at six different locations and a number of fans were used to ensure adequate mixing throughout the house.

To investigate the impact of the different remedies on the whole house ventilation rate each remedy is cycled on for two hours then off for two. Provided that the weather between the on and off periods is similar, a direct comparison of the ventilation rates can be made. The remedies investigated are listed in Table 2 of the results section 8.2.

8.0 Results

The following sections detail some of the results as recorded for each remedy. Details are given on radon levels and whole house ventilation changes. The results given are discussed very briefly.

8.1 Variation of radon levels

Table 1 summarises the radon results. The percentage radon reductions relate either to the 'as purchased' results or the 'enhanced natural underfloor ventilation' results depending on the condition the house was in before the remedy was used.

Initial tests in the 'as purchased' condition show that the radon level above the floor is roughly half that below. Thus, assuming the radon levels are representative of the average level of the whole space in which the measurements are taken, and that the underfloor space is the only radon source, half of the house ventilation air comes via the underfloor space. The level above the floor varies in a similar fashion to the levels below the floor, highlighting the underfloor space as a major radon entry route. During this test the average ground floor pressure difference is about -0.3 Pa, indicating an upwards flow (the pressure difference transducer takes the underfloor pressure from the room pressure). It ranges from 0.0 to -1.0 Pa and is similar for each room.

On average the whole house pressurisation system reduced the indoor radon level by 52%, however it was below the UK Action level (200 Bqm^{-3}) for 20% of the time, most of which was over one period. This emphasises the importance of long term monitoring. The floor pressure difference was reduced to an average of between -0.1 and -0.2 Pa. Thus air stills flows upwards through the floor but in reduced quantity. Assuming that this flow is proportional to the root of the pressure difference [4], this remedy reduces the upwards flow by about 30%.

Remedy type	**Period of test**	**Location**	**Rn Average (Bqm^{-3})**	**Rn Range (Bqm^{-3})**	**Indoor Rn reduction**
House as purchased (no remedy and poor underfloor ventilation)	24 May - 25 Aug 1994	living room	1,600	110 - 6,020	-
		underfloor	3,210	650 - 10,110	-
Whole house pressurisation (with poor underfloor ventilation)	22 Oct - 21 Nov 1994	living room	770	50 - 2,450	52%
		underfloor	unavailable	unavailable	-
Enhanced natural underfloor ventilation (nine airbricks open)	23 Dec 1994 - 30 Jan 1995	living room	1,130	97 - 3,540	29%
		underfloor	2,250	250 - 11,000	-
	8 Apr - 16 May 1995	living room	1,096	74 - 3,632	32%
		underfloor	2,030	426 - 11,392	-
Underfloor extract ventilation (fan on full and nine airbricks open)	1 Feb - 26 Feb 1995	living room	66	12 - 380	94%
		underfloor	2,570	1,300 - 5,890	-
Underfloor supply ventilation (fan on full and nine airbricks open)	1 Mar - 26 Mar 1995	living room	170	25 - 1,300	85%
		underfloor	140	45 - 680	-

Table 1: Radon statistics and reductions

The two tests with enhanced natural underfloor ventilation indicate little seasonal variation between the monitoring periods. This remedy reduced indoor radon levels by about 30%. Again the level above the floor is roughly half that below which, by the same assumptions as before, indicates that half of the house ventilation air comes via the underfloor space. The floor pressure differences average to about -0.9 Pa indicating an increase in air flow through the floor. This is to be expected as air can move freely through the airbricks to feed the stack effect, reducing the natural level of underfloor depressurisation. With the same assumption as before this increase in pressure difference represents an increase in flow of about 70%, representing an increase in house ventilation rate of 70%.

Underfloor extract ventilation reduces indoor radon levels to an average of 66 Bqm^{-3}, a reduction of 94%. The floor pressure difference in the living room averages to about +0.4 Pa, indicating that on average, air flows downwards through the floor eliminating pressure driven radon entry. For the areas further from the fan the floor pressures range between -0.2 and -0.3 Pa, indicating an upwards air flow. Thus the fan does not depressurise the whole underfloor area with respect to the rooms above. Rather it causes a pressure gradient which diminishes with the distance from the fan because of the underfloor walls and gaps in the floor.

Extract ventilation gives a high radon level beneath the floor due to the depressurising effect of the fan which increases the pressure driven soil gas flow into the underfloor space. As the underfloor level is similar to that with enhanced natural ventilation any increased dilution of the underfloor radon caused by the fan, appears to be offset by the increase in soil gas entry. This suggests that the floor pressure difference is the main radon reduction mechanism. Assuming the exhaust concentration to be similar to that measured in the void (2570 Bqm^{-3}) the possibility of exhaust air entering into living areas is a worry . However, provided the exhaust is sited away from windows and doors this should not prove a problem.

Underfloor supply ventilation reduces indoor levels by 85%, to 170 Bqm^{-3}, and underfloor levels to 140 Bqm^{-3}. The low underfloor level reflects the fact that outside air is supplied directly beneath the living room where the radon is measured. The floor pressures vary between -1.4 and -2.3 Pa with the largest difference being across the living room floor which is closest to the fan. These pressures show an increase in upwards air flow through the floor.

Assuming the underfloor space is the only radon source for the test house, the lower level in the living room underfloor space indicates the living room is being contaminated via routes other than through the living room floor. It is suggested that the radon levels in the underfloor areas far from the fan are higher than those close to the fan, causing a radon gradient within the ground floor rooms. Of these the living room is the lowest and thus susceptible to room to room cross-contamination. Alternatively radon may be travelling through the cavity walls or via some other convoluted route.

8.2 Changes in whole house ventilation rate

The various remedies investigated for their impact on whole house ventilation rates are listed in Table 2. Any remedy using mechanical ventilation has the fan set on maximum flow rate.

The tests are split into two categories according to the condition of the nine airbricks used for 'enhanced natural underfloor ventilation'. The table shows the impact of the remedy as a percentage change in air change rate per hour. In addition the average air changes are given for the conditions with remedy on and remedy off. All differences are highly significant at the 99% level.

Condition of airbricks (from 'enhanced natural underfloor ventilation')	**Remedy cycled on and off**	**Average air change rate (h^{-1}) with remedy..**		**change in air change rate (h^{-1}) caused by remedy**
		OFF	**ON**	
Airbricks closed	Airbricks open	0.84	0.96	+14%
	Whole house pressurisation	0.75	1.09	+45%
	Underfloor extract ventilation	1.02	1.30	+27%
	Underfloor supply ventilation	0.76	1.24	+62%
Airbricks open	Whole house pressurisation	1.14	1.46	+29%
	Underfloor extract ventilation (test 1)	1.43	1.31	-8%
	Underfloor extract ventilation (test 2)	1.04	0.97	-7%
	Underfloor supply ventilation	1.04	1.29	+24%

Table 2: Remedies tested for impact on house ventilation rate

Enhancing natural underfloor ventilation is seen to increase the house ventilation rate by about 14%; a much smaller figure than that predicted in section 8.1. This may indicate that the radon measurements in the house do not facilitate ventilation predictions. For example the radon measurement taken under the living room may not truly reflect the whole underfloor average, which is possible if the radon entry across the soil area is irregular and there is poor underfloor mixing. Alternatively it may indicate the difficulty in measuring very small pressures such as those across the floor. According to the manufacturers literature the error in this measurement should be less than ± 0.2 Pa, but this is still considerable when the pressures differences are often less than 1 Pa.

The changes in air change rates for remedies using fans are, as expected, significantly higher with the airbricks closed. Extract ventilation with airbricks open actually decreases the ventilation rate, probably as it competes against the natural upwards flow of house air. All other remedies and airbrick combinations cause an increase of between 24% and 62%.

The measured ventilation changes can be used to estimate a ventilation energy cost for each strategy. To do this a number of assumptions are made to keep the analysis simple. These include:

- whole house air change rate of 1.0 per hour (with remedy off)
- average outside temperature of 7 °C during the heating season
- average indoor temperature of 20 °C
- heating on for 6 hours a day, October to April inclusive
- thermal storage effects ignored
- 100% efficient heating system
- cost of fuel £0.07 per kWh

For the worst case scenario experienced with the test house these assumptions lead to a ventilation energy cost of about £50 per year, which is comparable to the cost of running a 75 W fan all year round.

9.0 Conclusions

This short report demonstrates the abilities of different ventilation strategies with regards to reductions in indoor radon levels.

The underfloor extract system proves to be the most effective, producing a 94% reduction getting indoor levels down to 66 Bqm^{-3}. The levels beneath the floor averaged to 2570 Bqm^{-3} presenting a small risk should any of the exhaust air find its way back into the house. The high level beneath the floor and low level above indicates that the floor pressure difference is the crucial radon reduction mechanism. Any increase in underfloor ventilation, causing dilution of the underfloor radon, appears to be offset by an increase in radon entry rate due to the level of underfloor depressurisation. Floor pressure data show that the fan does not manage to depressurise the whole underfloor space with respect to the rooms above. It does manage to reverse the floor air flow in the living room (directly above the fan) but not for any other areas.

Underfloor supply ventilation is seen to produce a reduction of 85%, indoor levels falling to an average of 170 Bqm^{-3}. As with extract ventilation, pressure readings show that the fan produces a floor pressure gradient which diminishes with the distance from the fan. Floor pressures indicate an increase in upwards air flow through the floor.

Although only the above two remedies 'solved' the radon problem (ie. reduced the indoor level below the UK Action level of 200 Bqm^{-3}) the others tested did reduce indoor radon figures. Enhanced natural underfloor ventilation gave a 30% reduction and the whole house pressurisation system achieved 52%. These reductions are considerable and the reason for them not solving the house only reflects the very high indoor radon concentrations initially associated with the house.

A large level of fluctuation in indoor radon levels is demonstrated reinforcing the philosophy of long term measurements if an accurate yearly average is to be obtained. For example a one week monitor during the whole house pressurisation test could have obtained the result 221 Bqm^{-3} when in fact the monthly average gave 770 Bqm^{-3}.

Using the radon levels as a tracer gas technique for assessing the amount of house ventilation coming through the underfloor space gave some interesting results. For both poor and enhanced natural underfloor ventilation, the radon level above the floor was half that below it, indicating that about 50% of the house ventilation air comes through the floor (with certain assumptions as given in section 8.1). Floor pressure differences suggested that by enhancing the natural underfloor ventilation, the flow through the floor increased by about 70%, representing a 70% whole house ventilation increase. However results from whole house tracer gas experiments disagree with this figure, indicating an increase of only 14%. This discrepancy is probably due to the difficulty in monitoring very small pressures. In addition any differences in the weather conditions during the different tests will be superimposed over the results.

The brief analysis of the changes in house ventilation rate caused by the various remedies show that the likely ventilation energy cost should be no more than about £50 per year. Mechanical supply ventilation increased the house air change rate by 62% when the airbricks were closed, which represents the largest individual change. All of the mechanical remedies (those with fans) cause larger changes in ventilation rates when the airbricks are sealed compared to when they are open. The measured increase in house ventilation rate caused by enhanced natural ventilation was much smaller than that predicted by the floor pressure measurements.

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Particulate Deposition on Indoor Surfaces - Its Role, with Ventilation, in Indoor Air Quality Prediction

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Particulate Deposition on Indoor Surfaces - its Role, with Ventilation, in Indoor Air Quality Prediction.

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Synopsis.

There is an increasing concern at the possible health effects of fine suspended particulate (aerosol) upon human health, particularly in the urban environment. Aerosol infiltrating indoors may arise from transport, power generation and natural sources. Aerosol also arises from indoor sources, through cooking processes for example, and from animal dander.

In zones within a building, within which the air is reasonably well-mixed, the levels of aerosol will depend upon the ventilation rate and the rate of deposition on indoor surfaces. For conditions of low air exchange rate, surface deposition will be the dominant removal process. It will depend upon the nature and orientation of surfaces and on the airflow conditions and is usually expressed in terms of a characteristic deposition velocity.

This paper describes a sensitive aerosol labelling technique which allows experiments to be conducted, in both occupied test houses and aerosol test chambers, aimed at measuring aerosol deposition velocities to a range of surface types. Progress is also reported in developing a CFD code capable of making use of the boundary layer parameters which result from the experimental work.

1. Introduction.

Since the human population spends significant time periods in enclosed environments, a subject of primary importance is the effect of indoor air quality on the health of building occupants. Indoor air may contain a variety of components in particulate form which arise from both indoor sources and outdoor sources. Significant aerosol hazards of outdoor origin are those arising from vehicular emissions; there is strong recent evidence that fine particles, particularly those less than 10 µm in diameter (PM_{10}) produced by diesel combustion, are linked to respiratory problems and an increase in mortality (1). Environmental tobacco smoke is a major source of genotoxicity indoors; tobacco smoke particles span a significant size range and are highly respirable. Pollens are allergenic aerosols of outdoor origin, while aerosols associated with animal dander and house-mite excreta arise when indoor surfaces are disturbed. In respect of airborne pathogenic particles, notably *legionellis,* it is ironic that one of the major transfer mechanisms into the indoor environment is through air-conditioning systems.

Airborne contaminant transport modelling has an important role in health risk assessment and is also a useful tool in the design of ventilation strategies for improving indoor air quality. However, in formulating a model which includes aerosol contaminants, there is a need to recognise that the behaviour of airborne particles is complex relative to the gaseous constituents of air. Although aerosol particles approaching molecular dimensions are subject only to diffusive forces and behave approximately as gases, the movement of larger airborne particles, notably those in the 0.1-1 µm range, has been shown to be strongly influenced by electrostatic and thermal factors while particles of super-micrometre dimensions settle readily from an airstream under gravitational influences (2).

Figure 1 shows the predicted equilibrium indoor/outdoor aerosol concentration ratio for an outdoor aerosol pollutant source, computed using a single compartment model (3). Experimental aerosol deposition rate data generated by the authors (4) has been used to make predictions at a range of air exchange rates. It can be seen that, particularly at low air exchange rates, aerosol deposition is an important modifier of indoor air quality. If aerosol deposition is neglected in indoor air quality prediction (refer to the bold line on Figure 1) the inhalation exposure of building occupants may be overestimated. An additional important consequence of ignoring the aerosol deposition process is that human exposure to particulate pollutants by routes other than inhalation will be neglected. Ingressed radioactive aerosol, the product of accidental industrial emissions, presents a risk of carcinogenisis not only through inhalation but also through ingestion and dermal penetration. In addition, it has been shown (5) that the primary route of child exposure to metal dusts from ingressed vehicular emissions is through hand-to-mouth transfer from contaminated indoor surfaces after dust deposition. Surface soiling through aerosol deposition is also of significance in a context unrelated to health; in many non-domestic environments, such as art galleries and semi-conductor fabrication plants, the economic consequences of surface degradation following particulate deposition can be severe.

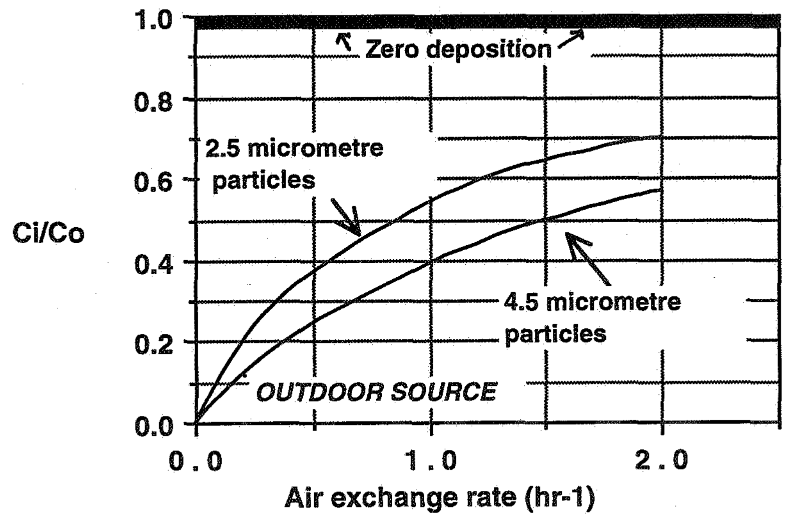

Figure 1. Calculated steady-state indoor/outdoor aerosol concentration ratio (Ci/Co), using experimentally-determined deposition rate data for 2.5 μm and 4.5 μm particles. The case for zero deposition is also shown. A building fabric filtration factor of unity is assumed.

In summary, the consideration of the aerosol deposition process when developing computational models leads to a more accurate indoor air quality predictor and increases a model's applicability to a greater range of pollutant types. The development of novel measurement techniques has allowed the authors to determine the most significant factors governing the behaviour of aerosol in both test chamber and real house environments. These data can be used to aid the development of computational codes for predicting indoor air quality. The aerosol measurement techniques, experimental results and computational developments are described in the remaining sections of this paper.

2. Experimental.

2.1 Aerosol generation and labelling.

In designing aerosol deposition experiments under conditions of building occupancy, careful consideration should be given to the type of tracer aerosol and detection technique used. A tracer aerosol deposition study should span the particle size range of real pollutant aerosol. This range is extensive; tobacco smoke particles can be as small as 0.01 μm while pollens of several tens of micrometres in diameter exist (6). Tracers which have low natural concentrations and high analytical detection sensitivities are ideal as aerosol labels since experiments can then be conducted using aerosol concentrations close to ambient levels, thus avoiding the occurrence of non-representative effects, such as thermal coagulation. Since the low aerosol levels constitute a negligible risk to a building occupant (or experimentalist), realistic simulation of a wide range of building occupancy conditions is possible. In addition, the use of a tracer with a high detection sensitivity presents the potential for surface analysis to complement aerosol concentration decay measurements.

A technique for generating, dispersing and detecting tracer labelled particles has been developed in a collaborative effort between the Energy Systems Section at Imperial College, the Danish National Laboratory at Risø and the Imperial College Centre for Analytical Research in the Environment. Porous silica particles, available in a variety of super-micrometre uniform size distributions, are agitated in a tracer salt solution so that tracer ions become bound to the particles' surfaces. Further details of the labelling procedure are presented elsewhere (7). The labelled particles are dispersed using a rotating brush aerosol generator. Sub-micrometre tracer particles are generated by atomisation and subsequent evaporation of a tracer salt.

Salts containing the rare earth elements dysprosium and indium are used as tracers in this work. Both exist naturally in a stable state but become unstable (i.e. radioactive) when bombarded with neutrons. Subsequent to a period of aerosol deposition in a test room/chamber, neutron irradiation of tracer aerosol-bearing materials (such as air filter papers or samples of domestic furnishing materials) in a nuclear reactor, followed by gamma-spectrometry, allows a quantitative determination of the aerosol mass present. Since dysprosium and indium occur naturally in low concentrations, the possibility of analytical interferences from particle-bearing media is minimised.

2.2 Aerosol deposition measurements in occupied rooms.

The tracer particles described in the previous section have been used in aerosol deposition measurements in three Danish and one British house, under furnished and unfurnished conditions. The details of the experimental procedures are described elsewhere (8); basically, a full-scale aerosol deposition experiment involves establishing a well-mixed tracer aerosol concentration in a suitable room and monitoring the aerosol concentration decay by sequential air filter sampling. If the air exchange in the room is simultaneously measured, by tracer gas monitoring, the rate constant for aerosol deposition can be calculated by subtracting the air exchange rate constant from the aerosol decay rate observed. Figure 2 shows the aerosol concentration decay and tracer gas decay curves generated in a typical experiment, plotted with the same arbitrary units to illustrate that air pollutants in particulate form behave quite differently from gaseous contaminants.

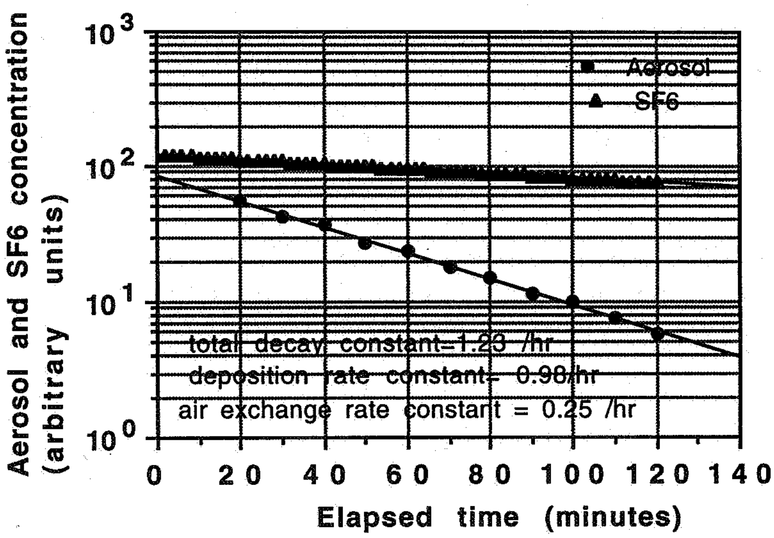

Figure 2. Aerosol and tracer gas concentration decay data, from a measurement in a single room. The decay rate constants, calculated from the data, are shown.

The average *aerosol deposition velocity*, a quantity which describes the aerosol flux to a surface for a given air concentration, to all the surfaces of a room can be determined by multiplying the aerosol decay rate constant by the volume to surface area ratio for the room. Figure 3 shows a representative selection of data; although the degree of furnishing and human occupancy was different in each test series, some general trends can be identified i.e. the average aerosol deposition velocity was seen to increase with particle size. In addition, the whole dataset indicates that, particularly for the larger aerosol particle sizes, average aerosol deposition velocities values are enhanced by the presence of furnishings.

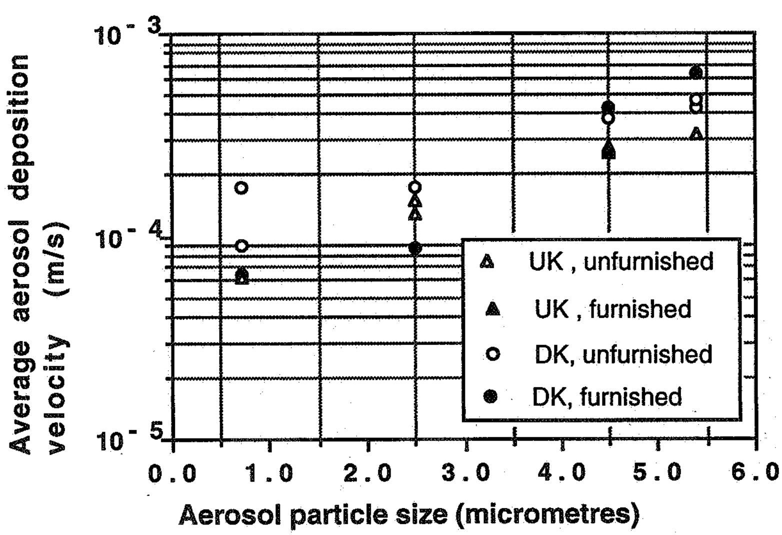

Figure 3. A representative sample of aerosol deposition data generated under furnished and unfurnished conditions in occupied houses in Denmark (DK) and the UK.

Considering the non-uniform test conditions, the results shown above are reasonably consistent, illustrating the value of sensitive detection techniques in identifying the likely influencing factors on air pollutant behaviour in single rooms. However, for multi-zone enclosures, computational models have an important role since the accurate tracking of aerosol particles approaches the limits of even the most sensitive measurement techniques.

An understanding of the detailed physics of aerosol transport required in the formulation of a computational model cannot be inferred exclusively from data generated in real houses since too many influences on aerosol behaviour simultaneously exist. Experiments in test chambers provide data for code development and are described in the next section.

2.3 Aerosol deposition measurements in test chambers.

While contaminant mass transport in laminar flows is relatively well understood, further knowledge is required concerning the mechanisms for mass dispersion and deposition under turbulent flow conditions such as might prevail in the occupied indoor environment. The sensitive aerosol detection techniques described above have been employed to measure aerosol deposition velocities in an $8m^3$ aluminium test chamber (cubic) under turbulent conditions; the turbulence intensity generated by a small fan, suspended from the test chamber ceiling was in the range 22-43%.

Average aerosol deposition velocities were measured for particles in the size range 0.7-5.4μm; the results were found to be in good agreement with theory and are discussed elsewhere (9). Since the use of neutron activatable tracers facilitates surface sampling, additional information can be obtained by the analysis of filter papers, attached in regular arrays to each interior chamber surface for the duration of the aerosol concentration decay period. The relative contributions to the deposition process of gravitational settling (i.e. deposition to the floor) and eddy and Brownian diffusion (deposition to the floor, walls and ceiling) can thus be identified.

Figure 4 shows the relative vertical (one wall) and horizontal (floor) particle fluxes measured for four particle size distributions. It can be seen that, as a proportion of the average aerosol deposition velocity to all the surfaces, wall deposition decreases as particle size increases, in opposition to floor deposition. The figure also indicates that, for the smallest particle size tested, total deposition to the four walls becoming comparable to floor deposition; this has implications for the design of decontamination strategies for internal building surfaces.

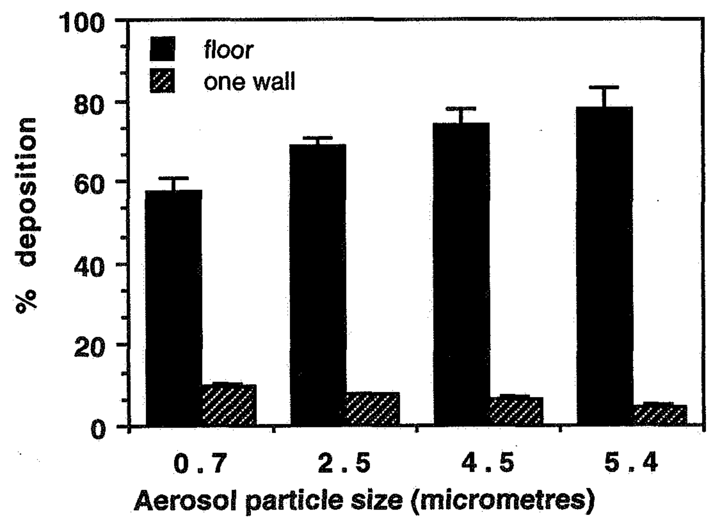

Figure 4. Measured relative aerosol particle mass fluxes to the floor and one wall of a test chamber.

The furnishings and wall and floor coverings which exist in the real indoor environment span a wide range of roughness characteristics. Having determined the average aerosol deposition velocity in the test chamber and the relative flux to each surface, the effect of surface roughness on aerosol deposition can be observed by covering one of the internal chamber surfaces with a rough material and combining the deposition velocity data for this case with the original chamber data. The aerosol deposition velocities to three rough vertical surfaces: wallpaper, short-pile carpet, and astroturf were determined by this method. Figure 5 shows the effect on vertical aerosol deposition velocity of attaching these rough surfaces to the chamber walls, for 4.5 μm particles; the roughness of the surfaces were quantified by friction velocity measurements using the method described in (10). As might be expected, greater deposition velocities were measured for the rough wallpaper, carpet and astroturf surfaces than for the visibly smoother aluminium surface. Although more data are clearly necessary to obtain a more reliable relationship between the variables, particularly in the intermediate friction velocity range, Figure 5 indicates that the particle deposition velocity to a vertical surface varies in a linear fashion with the friction velocity of that surface; it is likely that an increase in surface roughness will increase the rate of turbulent diffusion, and subsequent inertial impaction, to that surface but a limiting roughness will exist above which no further increase will increase the vertical particle deposition velocity, due to gravitational attraction of the particles to the floor.

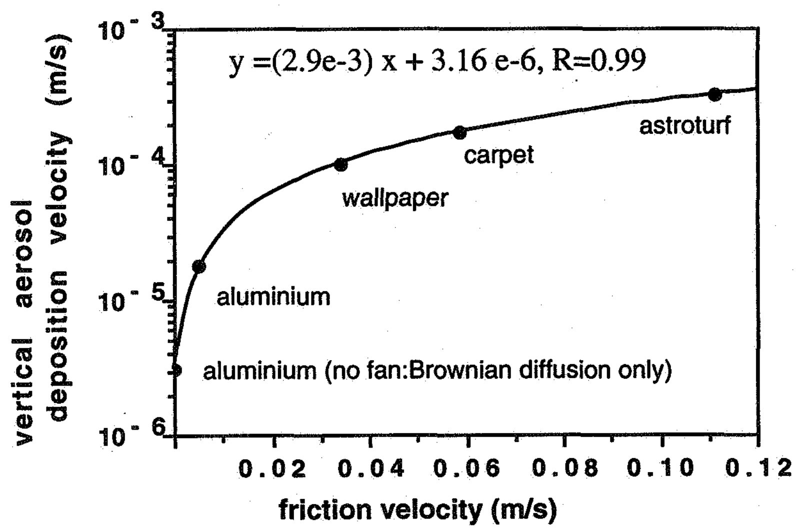

Figure 5. Measured aerosol deposition velocities (for 4.5 μm particles) to a vertical wall of a test chamber covered with materials of varying roughness.

3. Computational.

A model is under development at Imperial College which simulates the movement of aerosol particles in a turbulent airstream. The model has three key components: the gas-turbulence interaction in the main-stream, the particle-turbulence interaction in the main-stream and the particle-surface interaction in the boundary layer close to a wall.

A two equation K-E model (11) is used to model the gas-phase turbulence. A finite difference / finite volume technique is employed to solve the gas-phase governing equations on a staggered grid arrangement. Further details of the gas-phase modelling are available elsewhere (12).

A model has been developed to account for the frequently-overlooked mechanism of turbulent transport of particles to the boundary layer close to a surface by the turbulent diffusion of particles in the main stream; a large number of researchers have studied, both experimentally as well as analytically, the physical mechanism underlying particle deposition, but in the viscous sub-layer only, and as a process isolated from the main turbulent flow field. In the present approach, the computationally-economical "Prediction of Evolving Probabilities" (PEP) model (13) is used to simulate the particle-turbulence interaction; it is assumed that the joint gas-particle velocity distribution is Gaussian. An equation which correctly embodies particle momentum conservation is derived for the evolution of the probability of particle velocity and by implication, its position. The probability density function is discretised within the range of physically probable velocities. A Lagrangian formulation is used to calculate the particulate phase by tracking the particles from a finite number of starting locations.

The motion of a particle within the turbulent external region is dominated by turbulent dispersion until the particle reaches the edge of the boundary layer, whereas within the boundary layer, the deposition becomes the controlling parameter until the particle is either deposited or rebounds. In the boundary-layer, a 'free-flight 'model (described in (14)) is used to account for particle inertia ; the possibility of particle rebounce from a surface is treated by considering that all particles approaching the surface with a velocity less than a critical value (15) should stick to that surface.

Figure 6 shows the two-dimensional simulation of the gas-turbulence interaction in the test chamber which was described in section 2.3 of this paper. The direction of the airflow, as dictated by the presence of the fan in the test chamber, indicates that the preferential surface for particle deposition is likely to be the floor of the enclosure; as was seen in Figure 4, this was found, by experiment, to be the case.

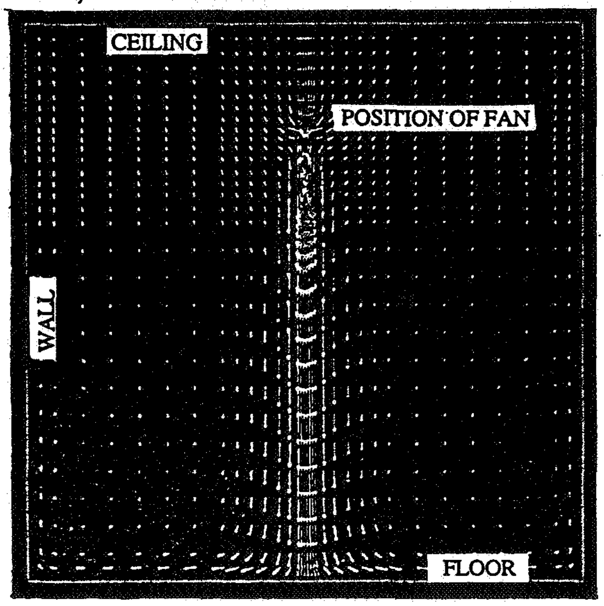

Figure 6. CFD simulation (axisymmetric) of the airflow generated by use of a fan in the aerosol test chamber described in section 2.3

4. Conclusion

It has been shown that the application of sensitive tracer detection techniques to aerosol deposition measurement facilitates studies of the effect of internal building surface character on indoor aerosol deposition, which complements aerosol concentration decay rate measurement. The data presented indicate that the orientation and roughness of indoor surfaces may significantly influence aerosol deposition velocity; these effects are relevant for environmental control considerations. These data can be used to aid in the development of computational models for indoor air quality prediction.

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Occupant Response to Passive Stack Ventilation: A UK Postal Survey

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OCCUPANT RESPONSE TO PASSIVE STACK VENTILATION: A UK POSTAL SURVEY

SYNOPSIS

A study was set up to compare the effectiveness of passive stack ventilators (PSV) with mechanical extract fans (MEF) in providing adequate ventilation in UK homes. New build and refurbished homes with PSV and MEF were identified and questionnaires posted to 3000 households of which 1223 were returned.

The survey showed that in homes installed with a PSV system, only 7% of those in the kitchen and only 8% of those in the bathroom were reported as blocked up. There were also few cases in which the MEF was blocked up or disconnected: 1.5% in kitchens and about 5% in bathrooms. However, less than a half (40%) of the respondents "usually or always" used their MEFs whereas most (93%) of the PSV systems were in constant use.

The respondents were asked to rate how problematic condensation, mould and noise were in their home. Approximately one-third of the respondents had at least a moderate problem with condensation in winter (34%) and noise from extract fans (36%) and one-fifth (19%) had moderate problems with mould. Fewer occupants had problems in homes installed with a PSV system than homes fitted with a MEF, whether manually operated or humidistat-controlled.

1.0 INTRODUCTION

The new UK *Building Regulations Approved Documents, Part F* [1] explicitly allows the use of passive stack ventilation (PSV) systems as a means of providing adequate ventilation in homes whereas mechanical extract fans (MEF) were the only solution explicitly allowed in the previous version. Previous work has compared the performance of PSV with other ventilation devices, including MEF, in reducing moisture content [2]. However, occupant appreciation and the way in which occupants use such devices are also important if these devices are to be used effectively. A study was therefore set up to examine theefficacy of PSV and MEF devices in providing adequate ventilation in homes, as perceived by the occupants.

2.0 METHOD

2.2 Questionnaire

A postal survey was used as an initial means of studying views on PSV and MEF devices as it permitted the occupants of a large number of homes fitted with PSV to be approached for comment. The survey allowed BRE to discover basic information about the respondent's opinion of the adequacy of the ventilation in their home.

As the study was designed as a postal survey, the questionnaire was kept short and simple. The respondents were asked to rate how problematic a series of issues (eg condensation, mould and noise from extract fans) were in their kitchen, their bathroom, and their home as a whole. The issues were rated on a five point scale labelled "no problem" (1), "moderate problem" (3) and "major problem" (5). The questionnaire also included a separate section enquiring about the main details of the ventilation devices (PSV, MEF or extract cooker hood) installed in the kitchen and the bathroom.

2.2 Sample selection

The sample needed to be balanced between the two methods of extract ventilation (MEF versus PSV), and also home size, type of housing, number of occupants, and geographical location. Furthermore, so that homes of similar building standards could be compared, the homes sampled all needed to be newly built or refurbished since 1985 when the previous version of the *Approved Documents* [3] was published.

A market research company (MRC) was contracted to locate the addresses of approximately 3000 homes installed with PSV and MEF systems. Addresses were obtained from local authorities (LAs), district councils, housing associations, the National House Building Council, builders and PSV manufacturers. The MRC was therefore asked to obtain a sample in which at least ⅓ of the sampled homes were fitted with PSV systems and at least ¼ of the sampled homes had humidistat-controlled MEF and ¼ had manually-operated MEF. A balance of the type of ventilation system in new-build and refurbished homes was also requested.

The LAs who responded supplied a total of 5199 addresses. Unexpectedly, in many cases (53.2%) it was not known whether the home was new build or refurbished. There were also fewer homes with PSV systems (8.2%) or humidistat-controlled MEF (7.8%) than with manually operated MEF (63.0%). Thus, for the final sample size it was decided to exclude the (10.1%) homes in which the type of home and type of ventilation system was not known and to also reduce the unknown home types with manual MEF (from 36.9% to 4.7%). This results in a final sample size of N=3000 homes of which 14% had PSV, 14% had humidistat-controlled MEF, and the remaining 53% had manually controlled MEF. The ventilation device in 19% of the homes was unknown. Furthermore, 58% of the homes sampled were definitely new build and 18% were definitely refurbishment.

3.0 RESULTS AND DISCUSSION

3.1 Final sample

Questionnaires were posted to 3000 occupants and 1223 completed questionnaires were returned resulting in a response rate of 41%. A recent BRE postal survey achieved a response rate of 38% and 20-30% is generally considered typical of postal surveys [4]. During posting it was found that the information supplied by the LAs was not particularly accurate and it was not clear how many of the addresses were unoccupied or unbuilt, which may have reduced the recorded response rate.

Table 1 shows the proportion of new-build and refurbished homes with each type of extract ventilation device. The table shows that the number of PSVs and humidistat-controlled MEF is less than anticipated in the original sample frame. Furthermore, the proportions of refurbished and new-build homes with the various ventilation devices are not as similar intended as there are more MEFs in new-build homes and more PSVs in refurbished homes.

Ventilation device →	PSV (%)		Manual MEF (%)		Humidistat MEF (%)		Don't know (%)		Total	
Data source →	LA	OR	LA	OR	LA	OR	LA	OR	LA	OR
New-build	5.1	2.2	35.2	36.5	2.9	2.0	18.5	21.0	61.7	61.7
Refurbished	9.2	5.9	16.8	9.3	0.0	2.4	0.1	8.4	26.1	26.0
Don't know	0.4	0.4	8.3	5.1	3.7	2.4	0.0	4.4	12.4	12.3
Column Total	14.7	8.5	60.3	50.9	4.6	6.8	18.6	33.8	100.2	100.0

Data source: LA = Local Authority, OR = occupant response,
Percentages are of total sample (N=1223) for each data source.

Table 1. Ventilation devices installed in new-build and refurbished home

Table 1 includes figures based on the information collected from the LAs and the occupants. The information from both sources is similar and there is close agreement on which homes are refurbished or new. However, according to the occupant response there are fewer PSVs and fewer manually operated MEFs than predicted by the LA. If the "don't know" categories are excluded, and manual and humidistat MEFs are grouped together, then concordance is quite high (80.7% matching). As mentioned earlier, it was found that the LA records were not as reliable as expected when it came to identifying the addresses of existing homes, thus reducing confidence in the supplied data. The majority of the following analysis is therefore based on occupant response and not the LA data.

3.2 Description of homes

Most the sample were houses (73.3%) or flats (19.9%) with a few bungalows (6.7%). Approximately one-third (37.1%) of the houses were semidetached and another third (35.9%, 331) were detached. These figures are of similar proportions to those in the whole UK housing stock ie one-fifth (19%) flats but mostly (80%) houses and bungalows of which one-third (33%) are semidetached and one-fifth (20%) detached [5]. The sample therefore covered a good range of dwelling types built or refurbished since 1985 and is representative of the UK housing stock.

One-quarter (25.2%) of the homes were built between 1985 and 1992 with just over one-third (37.7%) built before 1985 and the remainder (37.1%) built after 1992. Most (61.6%) of the homes were new-build and a quarter (26.1%) were refurbished; the building status of 12.3% was unknown. As expected, most (91.0%) of the refurbished homes were built before 1985 whereas most (81.3%) of the new-build, ie non-refurbished, homes were built after 1985; of the homes built post 1992, most (93.2%) were not refurbished.

3.3 Ventilation devices

Overall, 27.1% of the occupants said that they had no main ventilation device (PSV or MEF) in the kitchen; this is possible but, of course, they would be expected to have a window. Similarly, 15.7% said there was no PSV or MEF in the bathroom, but 81.6% of the bathrooms had a window and only 0.4% of the respondents reported having no window or ventilation device in the bathroom; these few cases may be due to the device going unnoticed (which is more likely in the case of PSV systems).

More detailed analysis showed that in homes with a MEF in the kitchen, 60.8% are manually-operated, 13.3% are light-pull activated and 15.8% are humidistat-controlled; the operation of the remaining 10.1% is unknown. In contrast, in the bathroom 47.3% of the MEFs are manually-operated, 35.5% are light-pull activated and 10.7% are humidistat-controlled. So, as expected, light-pull activated MEFs are more common in bathrooms than kitchens. However, one-tenth (9.9%) of the homes with a manually-operated MEF did not have a bathroom window, but unexpectedly most (70.8%) of the homes with a light-pull activated or humidistat-controlled MEF in the bathroom also had a window.

Homes with PSV systems were more likely to be refurbished (73.7%) than new-build (26.2%) ones whereas homes with a manual MEF were more likely to be new-build (79.6%) than refurbished (20.4%) homes. Homes with a humidistat-controlled MEF have an equal chance of being newly built or refurbished. However, overall there were more refurbished homes in the sample with a MEF (35.7%) than with a PSV (22.9%) system. 12.6% of the homes sampled did not have any ventilation device and a higher percentage of the homes with no device were new-build, ie 14.5% of new homes compared with 6.0% of refurbished homes.

Anecdotal evidence suggests that occupants block up or switch off ventilation devices because of draughts or noise. Therefore, the occupants were asked about whether or not they used the devices installed in their home. Table 2 shows that of the homes installed with a PSV system, only 6.6% of those in the kitchen and 7.7% of those in the bathroom were reported as blocked up. Table 2 also shows that there were also few cases in which the MEF was blocked up or disconnected: 1.2% in kitchens and 4.6% in bathroom. However, less than a half (39.4 to 49.0%) of the respondents "usually or always" used their MEFs whereas most (92.3 to 93.4%) of the PSV systems were in constant use. Thus, from a usage point of view, there appears to be some advantage to installing PSV systems.

Ventilation device	**Usage**	**Kitchen** % (*N*)	**Bathroom** % (*N*)
MEF or cooker hood: (N = 606)	Usually or always	39.4 (239)	49.0 (412)
	Sometimes	48.8 (296)	32.1 (270)
	Rarely or never	10.6 (64)	14.2 (119)
	Disconnected or blocked	1.2 (7)	4.6 (39)
PSV: (N = 152)	In use	93.4 (142)	92.3 (229)
	Blocked up	6.6 (10)	7.7 (19)

Percentages are of homes with each ventilation device.

Table 2. Use of ventilation devices

The occupants were also asked what ventilation devices existed in their home in addition to PSV and MEF systems. In general the number of alternative ventilation devices was similar in new-build and refurbished homes. One observation was that trickle ventilators were slightly more common in new-build homes and other background ventilation devices were slightly more common in refurbished homes. However, this observation is probably due to the age of the building rather than building status *per se*.

3.4 Subjective rating scales

Table 3 shows that in general the mean rating for each potential problem issue, rated on a 5-point scale (see table key), is around 2 or less, indicating that in general they were not problematic issues. However, approximately one-third of the respondents had at least a moderate problem (ie 3 on the rating scale) with condensation in winter (34.3%) and almost one-fifth (18.6%) had moderate problems with mould.

Room → **Problem issue ↓**	**Kitchen**		**Bathroom**		**Whole home**	
	$\bar{x}$ (σ)	% ≥ 3	$\bar{x}$ (σ)	% ≥ 3	$\bar{x}$ (σ)	% ≥ 3
Condensation in winter	2.2 (1.2)	37.3	2.1 (1.2)	33.9	2.1 (1.2)	34.3
Mould in winter	1.5 (1.0)	14.7	1.6 (1.1)	16.1	1.6 (1.1)	18.6
Odours	1.9 (1.1)	28.8	1.5 (1.0)	15.1	1.6 (1.0)	17.2
Noise from extract fans†	2.4 (1.2)	45.5	2.3 (1.3)	37.2	2.2 (1.2)	36.2

1 = "no problem", 3 = "moderate problem", 5 = "major problem",
$\bar{x}$ = mean, σ = standard deviation, † for homes with extract fans and/or extract cooker hoods.

Table 3. Ratings of ventilation-related problems

Table 3 also shows that a third of occupants had problems with noise from extract fans, ie 36.2% of those who had at least one MEF or cooker hood. Furthermore, almost one half (45.5%) of the respondents in homes with an extract fan or extract cooker hood had at least a moderate problem with noise from extract fans in the kitchen. The table also shows that approximately one-quarter of the occupants reported having some problem with odours in kitchens (28.8%). Overall, 22.7% of the respondents said that they had a major problem (ie rating 5 on the subjective scales) with one of the issues listed in Table 3 in either the kitchen, bathroom or home as a whole; 15.6% said they had major problems either just in the kitchen or just in the bathroom.

Further analysis (paired t-tests) showed that the respondents consider the issues listed in Table 3 to be significantly more problematic in the kitchen than in the bathroom ($2.7>t>14.4$, $1165<df<1175$, $p<0.01$), except for mould which was more problematic in the bathroom (t=3.0, df=1146, $p<0.01$).

3.5 Relationship between rating scales and physical properties

Table 4 lists the percentage of respondents reporting at least a moderate problem (ie rating 3 on subjective scales) with condensation or mould in the kitchen, bathroom and whole home broken down by the type of ventilation device installed in each room.

Ventilation device	Kitchen		Bathroom		Whole home	
	Cond.	Mould	Cond.	Mould	Cond.	Mould
No main device	55.4	22.4	46.8	22.2	46.5	17.7
Manual MEF	37.1	15.5	31.9	15.3		
Cooker hood	16.4	2.0	n/a		35.4	21.2
MEF + cooker hood	16.3	5.5	n/a			
Humidistat MEF	39.5	15.9	39.3	18.1	40.7	22.5
PSV	29.9	12.6	30.1	14.0	17.0	11.1
Other combinations	27.9	8.9	24.4	12.1	29.8	14.5

Cells show % of respondents reporting at least a moderate problem (≥3 on rating scales) in homes with corresponding devices.

Table 4. Percentage of respondents with a moderate mould or condensation problem

Table 4 shows that in homes reported not to have any main ventilation device, the occupant is more likely to report a condensation or mould problem. The table also shows that fewer occupants have problems in homes installed with a PSV system than in homes fitted with a MEF, whether manually-operated or humidistat-controlled. The difference is minimal for condensation and mould in the bathroom and for mould in the kitchen, but larger for condensation in the kitchen and mould and condensation in the home as a whole. Kitchens fitted with an extract cooker hood, or PSV and extract cooker hood, are considerably less problematic than those without cooker hoods. Overall, one-third (34.3%) of all the homes had at least a moderate problem with condensation and one-fifth (18.6%) had problems with mould.

Table 5 is similar to Table 4 but shows the mean rating on the subjective scales, rather than percentages, and the results of an analysis of variance (ANOVA). The ANOVA tests whether the subjective ratings are significantly affected by the ventilation device present in the home. Table 4 shows that, on average, condensation and mould are most problematic if the home does not have any ventilation device (the ANOVA statistics are included in the table). Overall, homes with PSV systems were rated to have significantly fewer mould and condensation problems than those with manual or humidistat-controlled MEF. Cooker hoods were also shown to produce the lowest level of problems in the kitchen and are significantly better than standard MEF devices. PSV systems and cooker hoods are therefore considered better for reducing mould and condensation problems, however as mentioned above there were actually relatively few homes with problems.

Ventilation device	Kitchen		Bathroom		Home	
	Cond.	Mould	Cond.	Mould	Cond.	Mould
No device	2.7^{abc}_{def}	1.8^{jkl}	2.4^{nop}	1.8^{qrs}	2.4^{tu}	1.7
Manual MEF	2.1^{agh}	1.5^{jm}	2.1^{n}	1.5^{q}		
Cooker hood	1.7^{bgi}	1.1^{km}	n/a		2.1^{v}	1.7^{x}
MEF + cooker hood	1.7^{ch}	1.2^{l}	n/a			
Humidistat MEF	2.2^{di}	1.5	2.2	1.7	2.3^{w}	1.7
PSV	2.0^{e}	1.5	1.9^{o}	1.4^{r}	1.7^{tvw}	1.4^{x}
Other combinations	1.9^{f}	1.3	1.8^{p}	1.4^{s}	2.0^{u}	1.5
Statistics: F	17.5	11.6	6.5	3.8	7.3	3.2
df	6,1148	6,1122	4,1155	4,1140	4,1151	4,1136
p	<0.001	<0.001	<0.001	<0.01	<0.001	<0.05

Means sharing common postscript are significantly different ($p<0.05$).

Table 5. Mean rating of problems with condensation and mould

T-tests showed that the occupants rate new-build homes ($\bar{x}=2.2$) as having significantly more condensation problems than refurbished (1.9) homes ($t=3.3$, $df=1,1016$, $p=0.001$). However, a 2-way analysis of variance (ANOVA) of building status (new-build versus refurbished homes) and the type ventilation device installed show that both the ventilation type ($F=5.2$, $df=4,1008$, $p<0.001$) and building status ($F=5.3$, $df=1,1008$, $p<0.05$) significantly affect the level of problems with condensation. A further 2-way ANOVA showed that ventilation ($F=3.8$, $df=4,998$, $p<0.01$) significantly affected the level of problems with mould whereas building status did not ($F=0.7$, $df=1,998$, $p=0.42$). Whether the property is new-build or refurbished *per se* therefore does not affect mould but the effect on condensation is additional to the effect of the type of ventilation device installed.

Odours were more problematic if there was no device in the kitchen ($F=8.0$, $df=6,1142$, $p<0.001$). There were no significant effects of type of ventilation device on odour problems in the bathroom or home as a whole.

Table 2 lists the mean noise ratings of occupants in homes installed with an extract fan or cooker hood. However, ratings of noise were made by most respondents (80.1%), regardless of whether they had an extract fan or cooker hood, thus allowing a comparison of the noise due to the different ventilation devices. As expected, the occupants considered the noise from extract fans to be significantly more problematic if their homes had (in order of increasing problems): a cooker hood (2.2), manual MEF (2.4), MEF plus cooker hood (2.5) and humidistat MEF (2.7), and least problematic if there was no ventilation device (1.2) or a PSV (1.3) in the kitchen ($F=32.6$, $df=6,951$, $p<0.001$). Similar ratings were made of the bathroom; there were significantly more problems with noise if there was manual (2.3) or humidistat (2.2) MEF installed rather than a PSV (1.1) or no device (1.1) at all ($F=26.8$, $df=4,1005$, $p<0.001$). So, PSV seem to be a more favourable ventilation system from a noise point of view, but in general the problem was at a low level (ie subjective scales were rated <3). Similar results were also found for the home as a whole (F=31.8, df=4,1103, p<0.001).

T-tests also showed that new-build homes (2.1) have significantly more problems with noise from extract fans than refurbished (1.9) ones (t=4.7, df=1,979, p<0.001). This result could be because homes with extract fans are more likely to be new-build ones (see Table 7) and homes with PSV are more likely to be refurbished.

4.0 Conclusions

The respondents of a national postal survey did not, on the whole, consider themselves to have major problems with their home due to condensation or mould. However, their responses showed that on average homes with passive stack ventilation (PSV) systems are rated less problematic than those fitted with mechanical extract fans (MEF). Further analysis showed that homes installed with cooker hoods were most effective in reducing mould, condensation and odour related problems in kitchens. As expected, there were fewer problems with noise from PSV systems than MEF (particularly humidistat-controlled MEF). Therefore, standards and guidelines can include the use of PSV systems and extract cooker hoods in new build homes and as part of refurbishment schemes.

5.0 Acknowledgement

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Implementing the Results of Ventilation Research
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Evaluation and Demonstration of Domestic Ventilation. State of the Art

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Synopsis

The IEA Annex 27, "Evaluation and Demonstration of Domestic Ventilation Systems" is aiming at developing tools by using the most developed computer models and equations available including modul development. Before staring up all the simulations an in depth review of the variables influencing the evaluation of a ventilation system have been done and a report is to be published. All parameters are needed to be mapped so that realistic assumptions can be made for the simulation phase. In the review is also included the models that can be possible to use for simulations of indoor air quality, thermal comfort, water vapour content, giving the possibilities to calculate the life cycle cost.

In the review report is given facts about housing statistics, population densities, moving pattern. The residents' behaviour is given for time spent in the dwellings, airing pattern etc. The loads, that have to be dealt with, are given by reviewing the present results on particles, NO_x, VOCs. The AIVC TNs have been used for the chapters concerning standards and leakages in dwellings. Also emissions from radon, landfill spillage, garage, and combustion can be influenced by the ventilation system and that have also been dealt with.

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 its dwelling occupational pattern vary. This results in a varying need for supply air to obtain acceptable indoor air climate and to avoid degradation of the fabric. Emissions 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 A 27 are: **to** develop tools, for evaluating 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 of Current and Innovative Ventilation Systems.

With the above objectives and scopes of the three subtasks the Annex started in April 1993 and has today nine participants: Canada, France, Italy, Japan, Netherlands, Sweden, UK, and USA. The specific objectives of the first subtask "State of the Art" are:

1. Give an overview of typical and frequently used systems,
2. Background and reasons for existing systems and standards,
3. Review existing evaluation methods.

Introduction

With the State of the Art Review, Månsson (1995), 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, the behaviour, and the time spent in dwellings. This means that one aim has been to 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 content of the report is also the headings of the following text in this paper. However, it should be noted that the chapter "Statistical Data on Housing" and the subchapter "User Behaviour and Perception" under chapter "Evaluation Approach" was presented at the AIVC conference in Buxton, UK, 1994, see Månsson (1994).

Statistical Data on Housing

The 14 OECD countries studied have 700 million inhabitants, 280 million dwellings and a useful floor space of 32 000 million m^2. The variation is great 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, see figure 1.

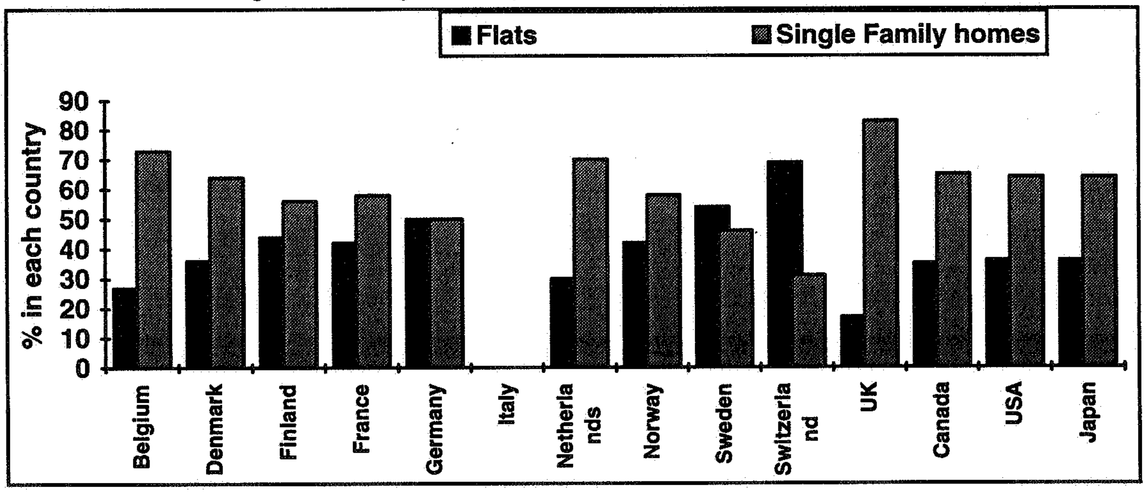

Figure 1. Single family homes and flats, percentage (%).

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/dwelling. Moving frequency studies show, that after 35 years of age the family has settled and will remain living in that dwelling.

A very important trend is that the number of one-person household is increasing. Today it goes from 20 % to 40 %. The 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 % to 30 % of the population.

Ventilation Performance

In order to get an overview of the most frequently used ventilation systems and how they perform, a survey has been done giving the ventilation rate and tightness reported from different measurement campaigns. As expected the most used system in existing dwellings is natural ventilation, see table 1. However, in new constructions a fan is always installed either for local exhaust or for general exhaust with a fan running all the time, see table 2.

Table 1. Distribution of ventilation systems in the existing dwelling stock

Country	Single family houses					Multi family houses				
	Natural			*Mechanical*		*Natural*			*Mechanical*	
	Adventi tious	Stack (S)	S+Kitch en hood	Exhaust	Supply+ Exhaust	Adventi tious	Stack (S)	S+Kitch en hood	Exhaust	Supply+ Exhaust
Belgium	100					95	5			
Canada		15	85							
Denmark		50		48	2		50		50	
France	40	15	20	22	3	40	20	10	30	
Italy	80		10	10		75			25	
Netherlands		62		38			37		63	
Norway			80	15	5		60		30	10
Sweden		12	63	14	11		40		44	16
Switzerland	70		30			40		60		
UK		95	5				100			
USA	60			40						

Table 2. Distribution of ventilation systems in the new constructed dwellings

Country	Single family houses				Multi family houses			
	Natural		*Mechanical*		*Natural*		*Mechanical*	
	Local exhaust	S+local exhaust	Exhaust	Supply+ Exhaust	Local exhaust	S+local exh	Exhaust	Supply+ Exhaust
Canada				100				
France		20	75	5	1		99	
Italy	80		20		90		10	
Netherlands		20	80			20	80	
Sweden			80	20			20	80
UK	100				100			
USA	90	10			90	10		

From various reports are given results of the ventilation rate in dwellings measured with active or passive methods. In table 3 is given the results from countries or regions with a need for heating during a part of the year. The air change rate is usually lower in single family houses than in multi family buildings. From Orme et al (1994) can be seen that the air tightness goes from 1 h^{-1} ($_n50$) to 30 h^{-1} ($_n50$) and with average values from 3 to 14 h^{-1} ($_n50$).

Table 3. Ventilation in dwellings

Country	No of dwell.s	Year of constr	Method p or a	Single family houses h^{-1}	l/s,p	l/s,m^2	system	Multi family buildings h^{-1}	l/s,p	l/s,m^2	system
Belgium	17 many	1980	a	0.5 0.75			N				
Canada	40			0.2-0.6							
Denmark	200 ? 150	1930-60 >1982	p p	 0.35				0.4	8	0.27 0.6	N E
Finland	242	-1982 all	p	0.40 0.42 0.45 0.45				0.62 0.64 0.60 0.64			N E SE
Germany		all		0.8				0.8			
Japan	10	1984	p	0 -0.7							
Netherlands						40 l/dw					
Norway		<1951 51-65 >65		0.5 0.4 0.3							
Sweden	≈2000	all	p	0.34 0.36 0.43	12 12 14	0.23 0.24 0.29	N E SE	0.49 0.58 0.60	12 14 16	0.33 0.39 0.40	N E SE
Switzerland	5	mv1980s mv1980 all	a a	 0.7				0.5[1] 0.1[2] 0.7	11		N N
USA, NY Cal, L.A. Georgia All states North states	30 640 22 500 ?	 all all	p p[3] p a a	0.2 0.6 0.1-6 0.83 <0.2							

p=passive & a=active tracer gas method, 1) closed & 2)opened bedroom doors 3) measured in January
N=natural, E=mechanical exhaust, SE=mechanical supply and exhaust

New systems that are on its way into the market or under testing are also briefly described. A trend is to install demand related systems and components, which make it possible to use the supplied air more efficient. Humidity is the most common controlling parameter, but also others are under development. A system under development is to use a timer to direct the constant flow of outdoor air to different rooms at various time, when the rooms are occupied. The location of the supply devices have always been a matter of experiments. Today also displacement ventilation is tested in dwellings. Various improvement measures of natural ventilation systems are under development in particular for existing dwellings. Such systems might be the improvement of the stack effect by small exhaust fans controlled by humidity, timer, pressure difference etc. Also improvemet of the trickle ventilator, usually located in the window frame or casement, so the opening is varied depending on the wind velocity, humidity, or temperature.

Standards

Indoor climate have been discussed almost as long as man built the first dwelling. Documented comments and recommendations were given in the ancient societies Egypt, Greece, and Rome. But a more detailed study started at first in the 19th century by distinguishing between undesireble unvoluntary and acceptance of voluntary exposure of pollutants. Today there are different ideas of how to give a standard for ventilation. It is or has been based on:

1. Outdoor environment
2. Hazardous air pollutants
3. Work place emissions
4. Causing chronic effects
5. Associated with threshold values
6. Specific pollutants/indicators (CO_2, particles, total hydro carbons)
7. Irritant properties of chemicals
8. Odour criterion

Standards are mostly based on perceived body odour and some specific pollutants. Moisture seems to be forgotten in the discussion of giving a ventilation standard in dwellings. All flow rates are based on the maximum number of residents defined by the number of beds possible to furnish all the bedrooms with.

Pollutant Loads

The main purpose with a review of pollutants, that may occur indoors, is to give information on normal levels and the range. The loads can be emissions of three categories : ① Constant (more than a few days), ② Variable, or ③ Outside sources. Some pollutants are given when the house is constructed or refurbished and others are unavoidable or linked to the living in a dwelling. Other pollutants are possible to avoid e.g. tobacco smoke.

Moisture

The water vapour content has very seldom been monitored in large investigations. A large study in Sweden reports average values of 38 % Relative Humidity (RH) in single family houses and 32 %RH in flats. This levels are out of the risk for house dust mites and mould growth. The comfort interval is reported to be 30 - 70 % RH (acceptable interval 20 - 90 % RH). It is recommended to have at least one month with RH < 45 - 50 % RH to avoid the growth of house dust mites and always try to keep it below 55 % RH (all figures as monthly average). Mould growth is avoided if the RH is kept below 75 % RH, weekly average.

VOCs

Volatile Organic Compounds (VOCs) are always present indoors and the emission is depending both on resident's behaviour and on building materials. Numerous studies have been conducted both large surveys in many ordinary dwellings and in problem buildings. Sometimes it is not possible to distinguish between dwellings with smokers and non-smokers. Other influencing factors are sometimes also at hand. Different sampling and analysing methods are giving different results. There is also a great variation during the day and can be in the ratio 1:5. E.g. in Japan is 27 000 t of para-chlorobenzene produced every year and used for deodorants (and moth-balls) and released usually in small rooms. The hygenic limits for work places is 300 mg/m^3 15 min and might be excedeed every day in some bathrooms or bedrooms. In table 4 is given some values indicating a range of 50 - 800 µg/m^3 and can easily reach over 10 000 µg/m^3 in new houses. A level usually mentioned is 300 µg/m^3 that should not be exceeded.

Table 4. Average concentration of VOC, 95 % confidence intervals [µ g/m^3]						
	Sweden		**Canada**	**UK**	**Germany**	**Switzerland**
	Single family	**Flats**	**Single family**	**All bldg types**	**Single family**	**New or refurbished flats**
No of dwellings	101	92	754	120	180	22
Analyse method	Tenax, MS	Tenax, MS	OVM 3500, MS	Tenax, FID	Home-made FID	
VOC conc	470 ± 180[1)]	310 ± 40	4 - 11	110	90 45 - 886	13 000 700 - 35 600

Particles

High particle concentration is usually a matter of smoking or not indoors. Particles originating from outdoors is less than 10 % as long as the air change rate is kept below 1 h^{-1}. In table 5 is given values for average and peak situations. It must be noted that peak values are not very well studied.

Table 5. Particle levels in homes, average and peak values.			
	Average mg/m^3		Peak mg/m^3
	Normal areas	Polluted areas	
Nonsmokers	0.020	0.050	0.1 - 0.2
Smokers	0.10 - 0.20	-	<1.0
Standards Netherlands	0.070		
USA federal	0.150		
USA California	0.050		

Bioeffluents and CO_2

The tracergas for bioeffluents is CO_2 even if it is known that the compounds giving the bioeffluents are chemical unstable and is rapidly decomposed to less odorous compounds. The result is that the odour is vanishing faster than a result of the dilution. The perception of odour has two cases: 1. Visitors entering a room with people, 2. Occupants. Flow rate is usually not discussed in the perspective of the occupants' case. Studies with occupants' percep-tion have given very small differences of the perception of annoyance of 800 or 1500 ppm CO_2.

NO_x

In dwellings the pollutant can originate from unvented gasheaters, a stove (range), an oven, and domestic hot water heater(s). A pilot flame for easy access results in a constant NO_x emission even if it is at a low level. In the future it is foreseen that more vented appliances are installed. Together with an increased number of installed kitchen hoods, better outdoor conditions and the use of outdoor vented appliances, the NO_x levels can be expected to decrease during the next decade. Large monitoring and epidemiological studies have been finalised the last few years.

Sensitive individuals have been observed to feel annoyance at 100 µg/m^3 NO_2. Measured values have been reported as high as 800 µg/m^3 NO_2 as weekly average in kitchens. Peak values during cooking can be up at 2200 µg/m^3 NO_2. However, it should be noted that the level originating from traffic can be in the range of 5 - 75 µg/m^3 NO_2 measured in homes without any natural gas appliance, but mean values is often at 20 µg/m^3 NO_2. As the energy used for domestic hot water can be up to 10 times the energy used for cooking the NO_2 level is consequently much higher. As a guidance the standards in dwellings are 60 µg/m^3 in Japan, 100 µg/m^3 in USA and Canada, and 300 µg/m^3 in The Netherlands for 24 h level.

Interaction with combustion. Radon and landfill spillage
Open-flued natural draught combustion appliances are very sensitive to the pressure difference. In calm weather conditions spillage can be caused at a pressure difference of 4 - 5 Pa when the vent is cold. With a hot flue spillage can be the case at 10 - 20 Pa. When installing mechanical ventilation this must be observed. Also warning flags must be rised if there is a risk for radon in the ground under the house and in particular if the indoor is given an underpressure.

Evaluation Approach

When evaluating ventilation systems the residents' behaviour must be taken into account. This must be made by assuming behaviour based on studies that might not be found in the ordinary technical reports. Usually it is market researches or sociological studies of smaller or larger groups. This assumptions will be used in simulation programs for evaluating the various systems.

User behaviour
This was presented last year at the AIVC conference, Månsson (1994), and here briefly summarised. In itself the discussion of how many persons/household is also a sort of behaviour. Assumptions must be made as close as possible to general behaviour.

Presence in the dwelling and in individual rooms:
Emploied men are away from the dwelling between 7 h to 17 h. In some countries it is more frequent, up to 60 %, to have lunch at home but not more 1 hour, whilst in other countries it si 40 %. Lowest frequence during a day is 20 %. The pattern for emploied women si similar but at a higher level at lunch and coming home 2 hours earlier. The housewives are away most frequently during a couple of hours in the morning and in the afternoon. Studies have also been made on the pattern of people at different ages. In the study of the time spent in the kitchen it was proportional to the area of the kitchen. A cultural habit is that the time in the kitchen is doubled in France compared to USA. This was valid both for housewives and women working also outside the home. Time spent on household work (cooking, cleaning, washing etc) is 3 - 4.5 h.

Body washing
One very important source for water vapour production is body washing. Some ideas of how much water vapour that is needed to be transported away, might be given by the use of water that goes from 150 to 260 l/person,day. Variation of hot water use is very high and can be in the ratio of 1:20. Most common is to take a shower. Studies in Europe indicates a frequency of once a day to 4 times a week and the duration seems to be about 10 min/shower.

Window airing
Todays knowledge of the behaviour can be summarised to be:
* The same daytime and at night
* Proportional to the outdoor temperature (higher when warmer)
* Proportional to the wind speed (lower with higher wind speed)
* Windows are not left opened when no person is present in the dwelling.
* Doubled when tobacco smoking is allowed in the dwellings. If smoking only takes place in the living room it is only in this room the opening is doubled.
* Regulating occationally high temperature, eg at parties.
* Regulating the temperature in general.

* Depending of the housewives' habits when making up the beds and cleaning the dwelling.
* Less when higher indoor temperature was prefered.
* Less amongst elderly people.
* No socio-economic correlation
* Increased when the room has direct solar radiation
* More when sunny than cloudy

Energy models
Simple heat loss calculations may be adequate for determining thermal liabilities if an appropriate balance point is chosen (usually 4 - 6 ^{o}C below room temperature). Only those hours where the external ambient temperature is less than the balance point are included in the calculation. Of course fan energy must be included if mechanical systems are evaluated. Simplified calculations can be performed on an hourly basis, but using daily or weekly averages for temperatures and average net flow rates results in only minor loss of accuracy. It is about 1 - 3 % relative to thermal simulation results.

Indoor Air Quality Models
As the main aim with Annex 27 is to evaluate ventilation systems the influence of many variables must be able to be dynamically handled by the model. Examples of variables are:

1. External: temperature, wind velocity, sunshine, treeshading.
2. Building envelope: air tightness, unintentional cracks, intentional leakage paths
3. Internal: pollutants (average, resident depending, local), ventilation system, room location, doors opened or closed, room volume.

Only very few models take into account all variables. In all the multi zone models it is assumed to be complete mixing in each zone. If an individual room is to be investigated on the consequences of not having complete mixing it can be handled by the discussion of ventilation efficiency and the use of computational fluid dynamic models (CFD). In such case most often only one room is dealt with. Models can include adsorbtion and emission of different gaseous pollutants as well as the treatment of water vapour. The greatest problem is to find material data.

Orme (1995) shows that there is no great differences between four tested models. However, some models can only make simulations for a day. The comparison gave identical results or less than 1 % deviation between the average of the four models and an indivual one.
The conclusion is that it is not the model that is most important, but the careful selection of the input data.

Thermal Comfort Model (Draught Equation)
As the equation with its constants is an empiric equation a sensitivity analysis will show how small errors will influence the prediction of dissatisfied.

Draught Equation $$PD = (34-t_i)\times(v-0.5)^{0.62}\times(3.14+0.37\times v\times T_u)$$

where	PD	Predicted dissatisfied in percentage (%)
	t_i	Indoor room temperature (^{o}C)
	v	Air velocity (m/s)
	T_u	Turbulence intencity in percentage (%)

With a fault of 3 % of the constants the PD can deviate up to 20 %. For greater faults of about 10 - 15 % the result will deviating up to 100 %. Various anemometers for the measured turbulence intencity are reported to give a 10 % difference at 10 % turbulence intencity and a 30 % deviation at 30 % turbulence intencity. With such a large range of uncertainty it may cause problems to interprete the measured values as well as the formula above as uncertainty may arise in what type of anemometer, that was used when the data were collected.

If the aim is to have the PD-value of 15 % it might also be up to 30 % with the given measured values for air velocity, turbulence intencity and room air temperature. If the target is a PD-value of 10 % with the uncertainty it might also be PD=20%. The practical way to use the equation is to compare different solutions in the seletion phase. Another means can be to recommend how close to a device, external wall or a window a person can sit or stand. Thus the equation can be used as a quality index.

Ventilation Efficiency

All the various methods to express the ventilation efficiency criteria for service and habitable rooms are given, see table 6

Table 6. Ventilation efficieny criteria	
Service rooms	**Habitable rooms**
Ventilation efficiency Pollutant removal efficiency Capture efficiency Removal efficiency Collection efficiency Pollutant index Room pollutant index	Nominal time constant Air change time Air change efficiency

Life Cycle Cost (LCC)

In order to compare different systems economically investment and LCC calculation methods are described. The total LCC must include factors like initial investment, maintenance cost, replacement cost, operating cost (heating the supply air, electricity for fans), damages caused by bad ventilation (eg refurbishment caused by mould growth). The best way to calculate is to use the Net Present Value (NPV). As the housing sector is dealing with a very long time perspective it is usually not possible to use the simple payback method. Sometimes there are other factors to concider such as noise and noise reduction, when dwellings are close to heavy traffic. A sensitivity analysis will show the consequenses of minor changings in the assumptions and is recommended to be made, as predictions of the future always are uncertain. With this type of calculations taking into account both the direct investment and the cost of the consequences of inadequate choice of ventilation system it might be possible to change the view of a ventilation system **from a cost to an investment.**

Noise

Noise related to domestic ventilation systems can be divided into three main areas. Depending on which system, that the dwelling has, the consequences varies, see table 7. Noise reduction goes hand in hand with air tightness. A good single weatherstripped window in a facade of brickwork will reduce the noise even if the ventilation provision through the wall has no soundproofing (25 dB(A) compared to 21 dB(A) without). If there is no ventilation opening (eg.

Table 7 Noise consequence

	Natural	Mechanical exhaust	Mechanical supply and exhaust
Outdoor noise	x	o	o
System Noise	-	x	x
Sound transmission	o	o	x

mechanical supply and exhaust) the reduction is even more (29 dB(A) compared to 22 dB(A)). It should be noted that a stack can transmit noise from aircraft and elevated roads and this must be observed when the facades are well soundproofed.

In many countries the maximum allowed indoor noise level is 30 dB(A). Usually a fan in a single family house gives a noise level of 30 - 45 dB(A) in rooms without sound reduction in the ductwork. With a silencer it is cheap and easy to take action. Another matter to consider is the vibration from a fan unit and in particular if a heat recovery unit is installed. Cross-talk especially between flats is a problem, that is of particular concern in mechanical supply and exhaust systems. It can be brought about by the following ways:

- Through the ductwork
- Transfer openings (within the same dwelling)
- Duct transition

Reliability
The reliability of a ventilation system is how well it can provide a minimum air flow rate and keep it. How well a certain selected system can give a required flow rate in individual rooms during a year under certain weather conditions is exemplified in the paper presented at this conference by Kronvall, Blomsterberg (1995). Other very important factors are dust accumulation in ducts and other components and malfunction of system equipment.

The malfunction of equipment is a matter of collecting information of how long the life time is expected to be. With a technique of safety analysis either with "Event Tree Analysis" or "Fault Tree Analysis" this is treated.

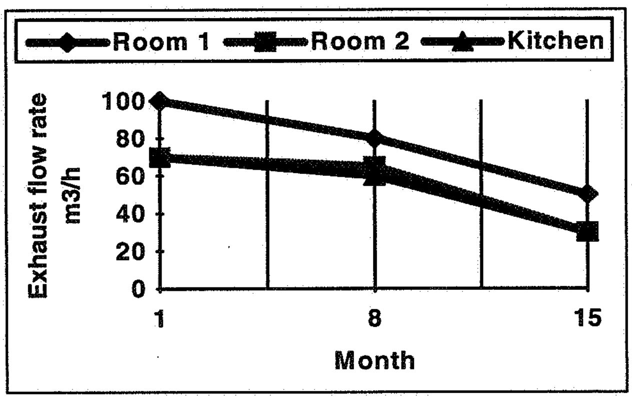

Figure 2. Measured reduction of the air flow rate due to dust accumulation

Dust accumulation gives a reduction of the flow rate, thus resulting in a risk for not keeping the required flow rate. If the dust can be calculated and the reduction of the flow estimated it is possible to give the duct cleaning intervals. In figure 2 is shown results of measured exhaust flow. A method to calculate has been developed by Wallin (1994) gave a flow rate reduction of about 20 % within 2 to 3 years. Another year without cleaning might result in a dramatic decrease of the flow rate to less than 50 %.

Conclusions

Dealing with the housing sector and with residents means to deal with a great variation in technical status and behaviour. Cases are not always well defined as it usually are in work places eg. offices. In some areas there is a lack of data or only studied in one or two countries. However, there is enough data on technical status and on loads to formulate assumptions, see table 8, that is expected to cover most of the residential behaviour and systems. Reliable simulation programs for energy, noise, IAQ are developed using the most recent knowledge. Modeling of water vapour and sorption might need to be developed further. The draught equation has to be used with great care. Data are still the most crucial point.

Table 8. Assumptions for the simulations
Design assumptions
1. Example dwellings
2. Ventilation systems
3. Leakage values
Residents' behaviour
4. Standard families
5. Combination of families and type plans
6. Time spent at home and in individual rooms
7. Window airing pattern
8. Internal door positions, indoor temperature
9. Metabolism, water vapour production
10. Criteria for house dust mites and mould growth
Simulations
Indoor air quality
Energy
Thermal comfort
Life Cycle cost
Noise
Reliability

Acknowledgement

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Indoor Climate and User Interaction in Modern Swedish One-Family Houses - Results Using a Questionnaire

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Synopsis

Disadvantages and advantages with different heating and ventilation systems in modern housing have been discussed during many years in Sweden. The discussion has intensified for modern low energy houses, where the use of forced air heating has increased during the last fifteen years, mostly in one-family houses. In many articles and the general debate diverging opinions have been presented concerning the thermal comfort, the air quality, the ventilation and the energy use in modern one-family houses with forced air heating. All modern Swedish one-family houses are very well insulated, fairly airtight and are equipped with mechanical ventilation.

The indoor climate (thermal comfort and ventilation), the energy use and the user interaction were examined using questionnaires in 236 houses built after 1988, 144 houses with exhaust fan ventilation and radiator heating and 92 houses with balanced ventilation and forced air heating. In 50 of the 236 houses measurements have been carried out, one-time tests of mechanical air flows, air tightness, status of heating system, indoor temperature, particles and long-term tests of outdoor air ventilation, humidity, carbon dioxide and indoor temperature during a winter month.

This report presents and discusses the quality of the indoor climate and the user interaction in modern Swedish one-family houses. Improvements to future houses are proposed e.g. reduce sound from the outside in "exhaust" houses, reduce draught from outdoor air vents in "exhaust" houses, reduce sound from the ventilation system in "balanced" houses, make more readable operating and maintenance instructions for the heating and ventilating systems, provide better air temperature control of individual rooms, reduce the ventilation rate?, supply all outdoor air to the bedrooms?

The investigated houses show appr. the same level of complaints concerning the indoor climate as in a study of the Swedish building stock i e there are no serious problems. There are some differences between houses with the two types of heating and ventilating systems.

1. INTRODUCTION

Disadvantages and advantages with different heating and ventilation systems in modern housing have been discussed during many years in Sweden. The discussion has intensified for modern low energy houses, where the use of forced air heating has increased during the eighties and nineties, mostly in one-family houses. In many articles and the general debate diverging opinions have been presented concerning the thermal comfort, the air quality, the ventilation and the energy use in modern one-family houses with forced air heating.

A nordic seminar concerning "Functional requirements on ventilation systems and their use for space heating and improvement in indoor air quality" took place in September 1992. The 35 participants agreed upon a number of disadvantages and advantages with forced air heating (Andersson 1993).

This project was initiated by the Swedish National Testing and Research Institute and was funded by the Swedish Council for Building Research, The Development Fund of the

Swedish Building Branch and Swedish National Board of Housing. During the project fruitful discussions took place with a reference group. The overall aim has been to evaluate modern housing and to evaluate disadvantages and advantages of forced air heating compared with radiator heating. This with respect to thermal comfort, ventilation, air quality and energy use (Blomsterberg 1995).

Indoor climate questionnaires were mailed during 1993 to 449 one-family houses built after 1988. Answers were received from 53 % of the houses. Of the houses 172 were equipped with forced air heating, 255 with exhaust ventilation and radiator heating, and 22 with exhaust-supply ventilation and radiator heating. Measurements of indoor climate were carried out as a case study during the winter of 1993-4 in 50 of the houses, where the occupants experienced problems with the indoor climate.

2. THE HOUSES TESTED

The houses which were examined in this project are representative of modern Swedish one-family houses. The houses with exhaust fan ventilation have exhaust air terminal devices in rooms such as bathrooms, kitchens and laundryrooms and outdoor air supply to the other rooms through outdoor air vents near windows. Space heating in most of the houses is provided for by radiators located below windows. More than 50 % of the houses have an exhaust air heat pump and thereby also a prefilter in the exhaust air duct.

All of the houses with forced air heating have exhaust and supply ventilation, and circulated air. Air is exhausted from rooms such as bathrooms, kitchens and laundryrooms and air is mainly blown into bedrooms and living-rooms. Most of the houses have a prefilter in the outdoor air and the circulated air ducts. There is usually no filters in the exhaust air duct. All of the houses are equipped with some kind of heat recovery.

The houses were chosen randomly. Important criteria were however; the houses should have been built after 1988, occupied for at least one year, located in the southern or middle part of Sweden, the number of houses with different brands of heating and ventilation systems should agree with their share of the market, different manufacturers of houses should be represented.

3. METHODS

3.1 Questionnaires

A survey of standardized questionnaires which are being used to investigate the perception of the indoor climate by the occupants, shows that there are mainly two more worked through questionnaires. One was developed by the hospital in Örebro. The Örebro-questionnaire deals mainly with the individual persons and their health. The other questionnaire has mainly been used to examine buildings which have had or are suspected to have problems with the indoor climate. In the ELIB-study the Örebro-questionnaire has been used in order to determine the indoor climate in the Swedish housing stock (Norlén 1993).

The second questionnaire "SABO" has been developed by the office of investigation and statistics in Stockholm. The questionnaire was recently used in a survey of the perceived indoor climate in the housing stock of Stockholm (Engvall 1992). The questionnaire uses as a starting point the dwelling, its use and indoor environment with regard to thermal comfort, air quality, sound and light level and the estimation by the occupants of problems. The questions concerning the indoor environment are supposed to be detailed enough and enough directed towards measures so that the answers can lead to measures.

As one purpose of this project was to be able to use the ELIB-study as a reference the Örebro-questionnaire was employed. As the purpose also was to be able to make a more detailed determination and thereby facilitate a comparison between air-heated houses and radiator-heated houses, with regard to thermal comfort, air quality, sound and light level and an estimation by the occupants of possible problems, the SABO-questionnaire was used.

3.2 Measurements

The measurements were started with an inspection and diagnostic testing of the 50 houses during the winter of 1994. The purpose was to document the status of the heating and ventilation system. If the performance deviated from the design values then the intention was to carry out an adjustment. An evaluation would otherwise not make sense. In reality it turned out that no system was in such a bad condition that an adjustment was really necessary.

The following inspections and diagnostic tests were performed:
- The total air flows, the air flows at the air terminal devices and the filters in the ventilation systems were measured.
- The insulation of the ductwork in the air-heated houses was inspected.
- The airtightness of the building envelope was measured and the main leakage paths were located.
- The fouling at the air terminal devices were inspected
- The function of the heating system was checked.
- The surface temperature of the floor and the indoor air temperature (absolute level and gradients) were measured in all rooms.
- The air velocity was measured in all rooms.
- The sound level was measured in all rooms.
- The number of particles outside, in one bedroom, in the supply air and in the circulated air were measured.

The following long-term measurements were carried out in the houses:
- The relative humidity was measured as monthly average in the middle of the house and in one bedroom. A diffusion tube with an absorption material was used.
- The indoor air temperature was measured as monthly average in the middle of the house and in one bedroom. The measurements were performed using electronic temperature meters, where the temperature is converted into pulses.
- The ventilation was measured as a monthly average in the whole house and in one bedroom. A passive tracer gas technique with two different tracer gases was employed.
- The carbondioxide concentration was measured during one night. A Drägertube was used.

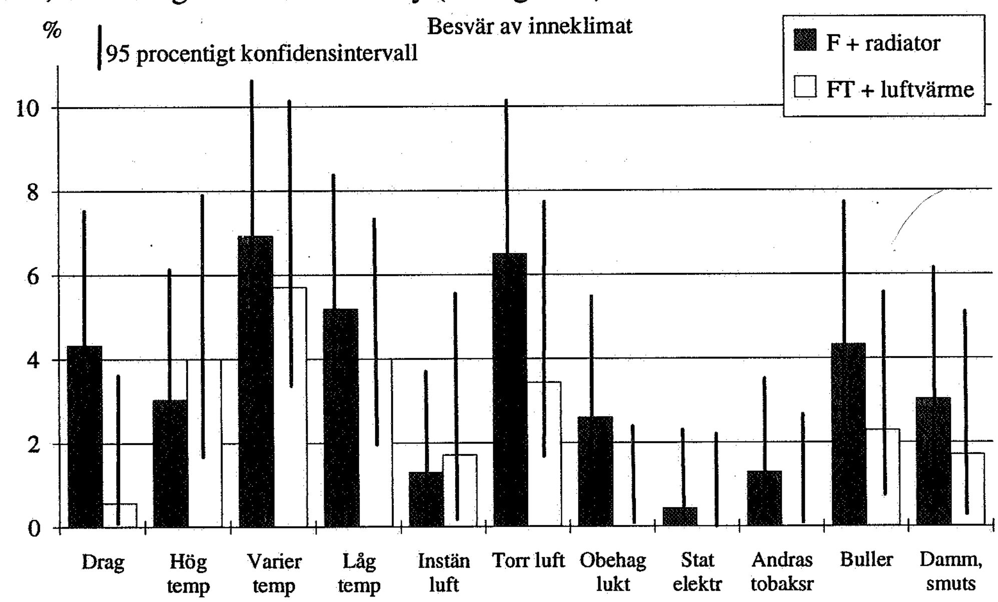
The relative humidity, the indoor air temperature and the ventilation were measured using the same equipment as in the ELIB-study.

4. RESULTS

4.1 Indoor climate

The Örebro-questionnaire presents the perception of the indoor climate by the occupants as profiles of complaints and profiles of symptoms. The questionnaire has been used in order to be able to make a comparison with the Swedish one-family housing stock, as presented in the ELIB-study. A complaint or symptom must occur often to be regarded as a complaint or a symptom.

The frequencies of complaints in general are low for the examined one-family houses (see figure 1). The highest values were obtained for dry air and varying indoor temperature. Appr. 6 % regard the indoor temperature as often (every week) varying and appr 5 % that the air is often (every week) too dry. For two questions, draught and unpleasant smell, there is a statistically guarenteed difference between radiator-heated houses and air-heated houses and for both cases the complaints are higher for the radiator-heated houses. There is no statistically guarenteed difference between this study and the Swedish one-family housing stock, according to the ELIB-study (see figure 2).



Drau-ght	High temp	Varia-tion in temp	Low temp	Stuffy air	Dry air	Unple-asant smell	Stat electr	Other's tobac smoke	Noise	Dust, dirt

Figure 1. Frequencies of complaint ("often") in the examined houses. F + radiator = exhaust ventilation and radiator heating, FT + luftvärme = balanced ventilation and air heating.

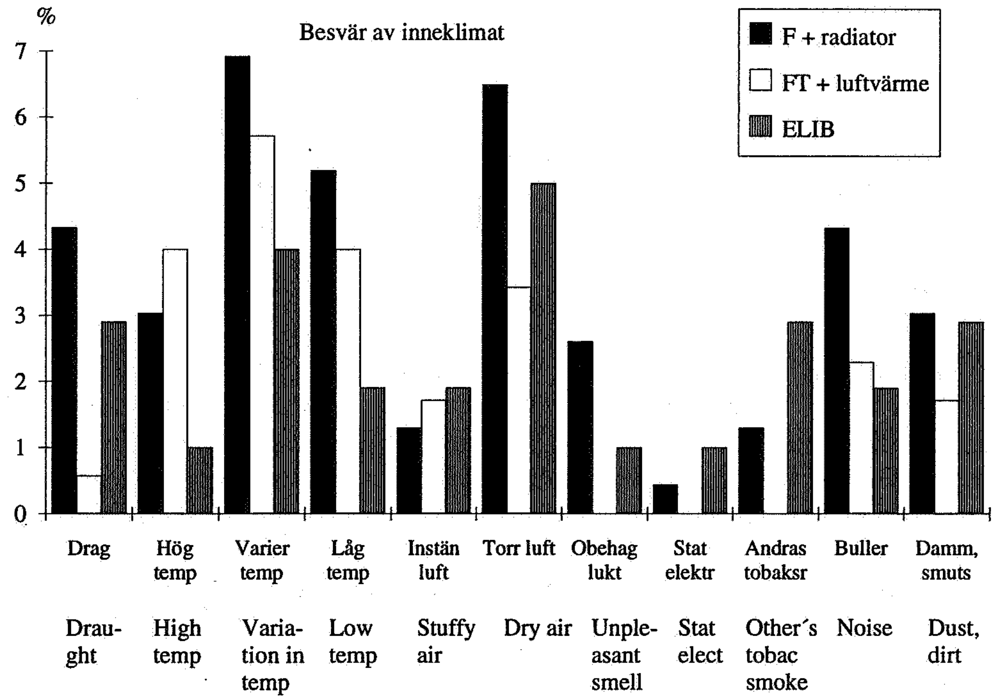


Figure 2. Frequencies of complaint ("often") in the examined houses. Comparison with the Swedish one-family housing stock according to the ELIB-study. F + radiator = exhaust ventilation and radiator heating, FT + luftvärme = balanced ventilation and air heating.

The frequencies of symptoms are generally low for the examined houses (see figure 3). The highest values are obtained for tiredness. Appr. 8 % feel tiredness often (every week). For peeling/itching in scalp/ears there is a statistically guarenteed higher frequency of symptoms for radiator-heated houses than for air-heated houses, the levels are however low. There is no statistically guarenteed difference between this study and the Swedish one-family housing stock, according to the ELIB-study (see figure 4).

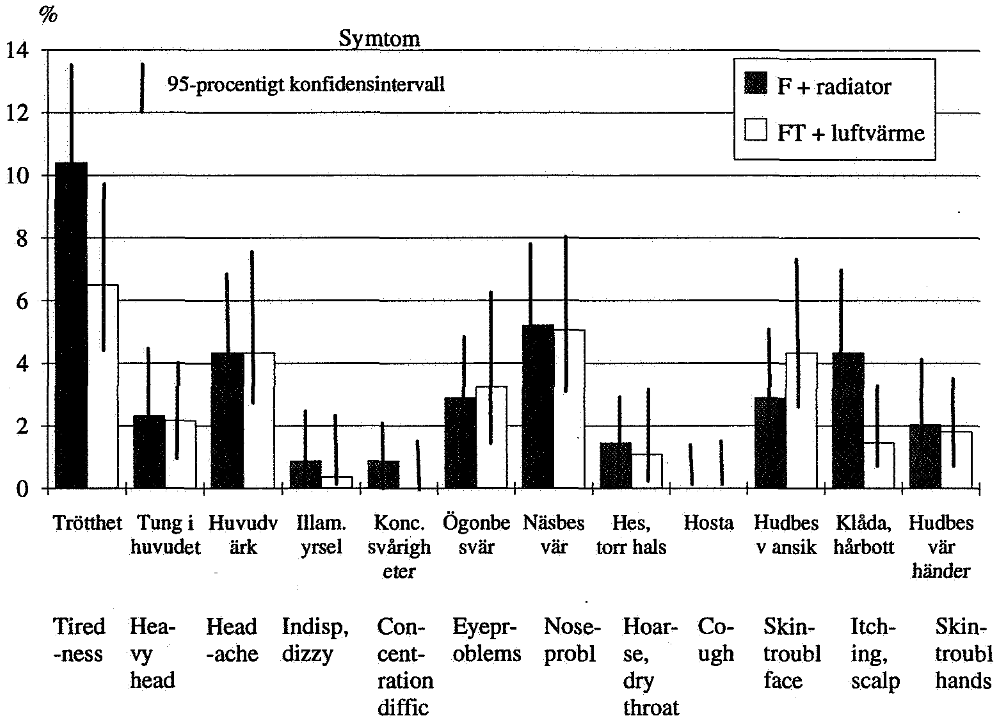


Figure 3. Frequencies of symptoms in the examined houses. F + radiator = exhaust ventilation and radiator heating, FT + luftvärme = balanced ventilation and air heating.

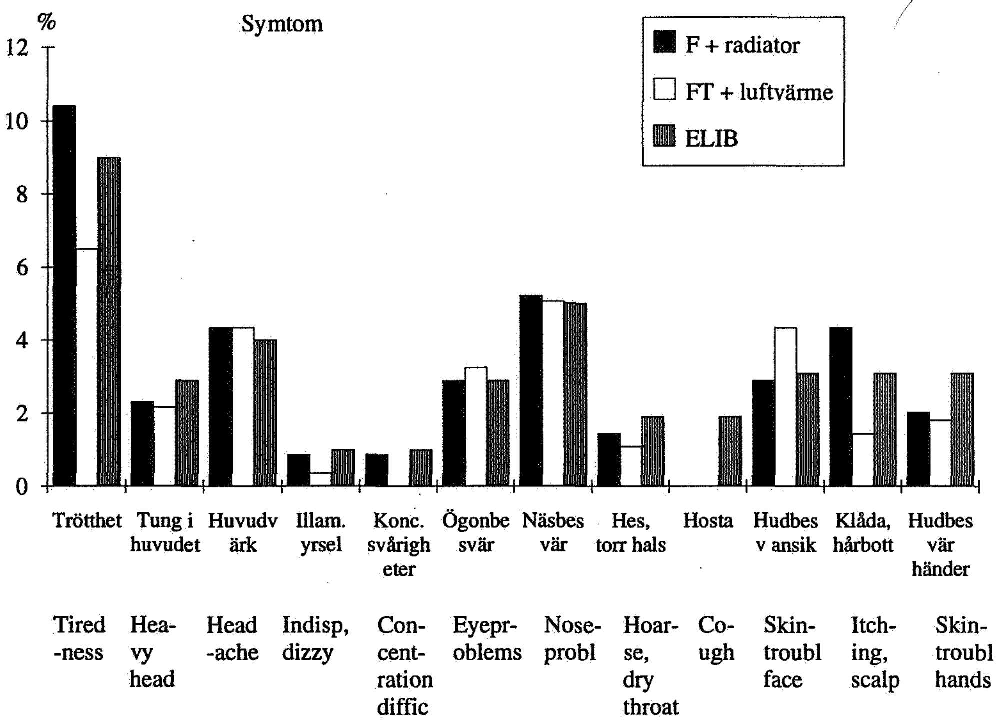


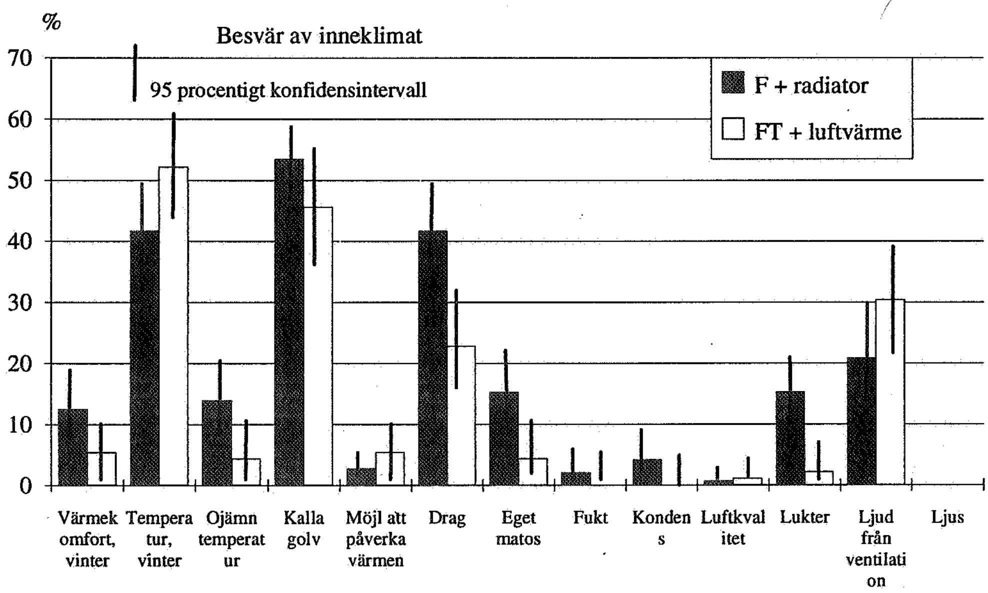
Figure 4. Frequencies of symptoms in the examined houses. Comparison with the Swedish one-family housing stock according to the ELIB-study. F + radiator = exhaust ventilation and radiator heating, FT + luftvärme = balanced ventilation and air heating..

The SABO-questionnaire has been used in order to be able to answer detailed questions and thereby also enable a comparison between radiator-heated houses and air-heated houses regarding the use and the indoor environment of the dwelling with respect to thermal comfort, air quality, sound and light level and an estimation of problems.

What concerns the field of heat appr 90 % of the occupants consider the thermal comfort during the winter season as good or acceptable (see figure 5). The occupants perceived most complaints concerning the indoor temperature during the winter season and cold floors. Appr 45 % claim that one or a couple of rooms are very cold or too cold. The frequency of complaints for cold floors is 50 %. A statistically guarenteed difference between radiator-heated houses (14 %) and air-heated houses (4 %) exist only for often varying indoor temperature due to changes in outdoor temperature.

What concerns the field of ventilation 99 % of the occupants answer that the indoor air quality is good or acceptable. The highest frequency of complaint concerns draught, appr 30 % are troubled by draught somewhere in their house. A significant difference exists between radiator-heated houses (42 %) and air-heated houses (23 %).

As to sound 25 % of the occupants in both types of houses consider the sound from the ventilation system as sometimes or often disturbing, a few more in air-heated houses than in radiator-heated houses. In radiator-heated houses however sound from the outside is perceived by more occupants as often or sometimes disturbing than in air-heated houses, 24 % resp. 6 %.



Ther-	Temp,	Un-	Cold	Pos-	Drau-	Coo-	Moi-	Con-	Air	Sm	Sound	Light
mal	winter	even	floors	sible	ght	king	sture	dens-	qua	ells	from	
comf,		temp		influe		fum-		ation	lity		ventil	
winter				heat		es						

Figure 5. Frequencies of complaint (many alternatives include "sometimes" and "often") in the examined houses. F + radiator = exhaust ventilation and radiator heating, FT + luftvärme = balanced ventilation and air heating.

A comparison with the one-family housing stock in Stockholm (Engvall 1992) show a couple of higher frequencies of complaints than for modern one-family houses (see figure 6). The Stockholm one- family housing stock consists of houses of different years of construction, 3/4 of them have passive stack ventilation. The frequencies are higher for temperature during the winter season (one or a couple of rooms are very cold or too cold), cold floors and draught.

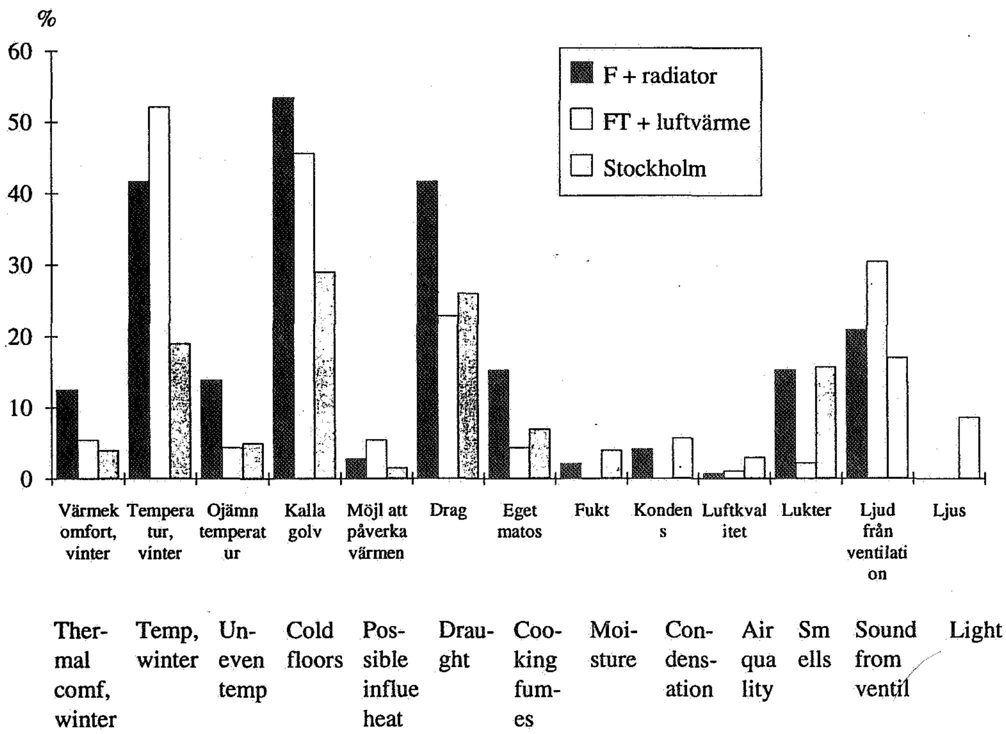


Figure 6. Frequencies of complaint (many alternatives include "sometimes" and "often") in the examined houses. Comparison with the one-family housing stock in Stockholm. F + radiator = exhaust ventilation and radiator heating, FT + luftvärme = balanced ventilation and air heating.

If thermal comfort, temperature, sound from ventilation, dry air and sound from the outside only include "often answers" then the frequencies of complaints are reduced by at least 2/3.

4.2 User interaction

An interesting question for modern houses is whether the occupants think that the heating and ventilation system gives any possibility to influence the indoor climate. Only 5 % of the households answer that there is no possibility to influence the heating of their dwelling i e the heating system gives according to the occupant of 5 % of the houses no means of influencing the indoor air temperature. The same percentage applies to the answer that there is no possibility to influence the ventilation of the dwelling i e the ventilation system gives according to the occupant of 5 % of the houses no means of influencing the air quality.

In every second to every fifth one-family house the operating and maintenance instructions for the heating and ventilaiton system are considered to be "not so easy to understand". The problem is more common in radiator-heated houses than in air-heated houses. 2/3 of the radiator-heated houses have an exhaust air heat pump. In every tenth house the occupants do not know whom to contact if problems occur with the heating and ventilation system. In appr 90 % of the houses the occupants claim that they vacuum clean their house at least once a week. In 60 % of the air-heated houses and 10 % of the radiator-heated houses this is done using a central vacuum cleaner.

The filters are cleaned more often in air-heated houses than in exhaust-ventilated houses. In 85 % of the air-heated houses they are cleaned every month and in 35 % of the exhaust-ventilated houses according to the occupants. Most of the houses have a range hood with a grease filter. This filter is cleaned in almost 50 % of the houses every month. In a couple of % the houses it is done very seldom or never. Most of the houses are equiped with an air-to-air heat exhanger or an exhaust air heat pump. In 1/3 of the houses this component is cleaned every month. In 15 % of the exhaust-ventilated houses the exhaust air heat pump is cleaned very seldom or never.

Airing is more frequent in exhaust-ventilated houses than in air-heated houses during the heating season (September - April) (see figure 7). In exhaust-ventilated houses 60 % the occupants claim that they air daily or almost every day compared with 25 % in air-heated houses. It is quite common in air-heated houses that no airing takes place, 40 % of the houses. In exhaust-ventilated houses the number is 10 %. When airing the most common technique is to cross ventilate for a couple of minutes. This is in done in 50 % in both types of houses. Some 10 % of the houses air continuously for a day or a night using windows or airing panels.

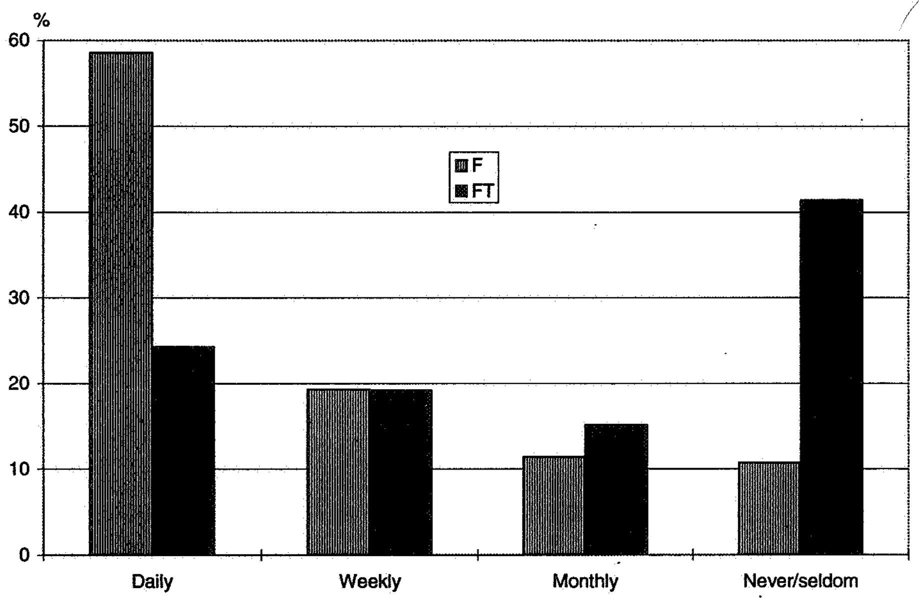


Figure 7. Airing habits in the examined houses. F = exhaust ventilation and radiator heating, FT = balanced ventilation and air heating.

4.3 Measurements

The inspection and diagnostic testing of 50 houses show that

- the air flows in the ventilation systems fullfill the requirement of an overall ventilation of 0.5 ach of outdoor air (Nybyggnadsregler 1989) apart from the exhaust-ventilated houses, where 50 % had a ventilation rate which was somewhat too low
- the sound from the ventilation system is considerably lower in the exhaust-ventilated houses than the houses with balanced ventilation, which in some cases do not fullfill the requirements of the Swedish building code of 1988 (Nybyggnadsregler 1989)
- the measured temperature difference between the warmest and coldest room is appr the same in all houses, appr 1 °C, during an ordinary Swedish winter day
- the requirement of a maximum vertical temperature gradient (between 0.1 and 1.1 m) of 3 °C is not fullfilled in ¼ of the houses, when the outdoor temperature is - 10 °C
- for several of the houses the requirement of a lowest surface temperature on the floor of 19 °C is not fullfilled during an ordinary Swedish winter day
- the air velocity does not exceed 0,15 m/s during an ordinary Swedish winter day
- most of the houses fullfill the requirement on airtightness of the Swedish building code
- it is not possible to see any difference in the content of particles in the air of exhaust-ventilated houses and air-heated houses.

The long-term measurements of 50 houses show that
- the outdoor air flow to the rooms upstairs is lower in air-heated houses than in houses with balanced ventilation
- the outdoor air flow to bedrooms upstairs is often too low in exhaust-ventilated houses (Blomsterberg 1991)
- the purging air flow (Merkell 1993) to bedrooms is larger than the design outdoor air flow in all houses, which means that if the air quality of the transferred air is acceptable then the ventilation requirements (4 l/(s and bed)) of the Swedish building code is fullfilled
- the purging air flow to bedrooms is smaller in exhaust-ventilated houses than the other houses, which indicates that the ventilating air is being used more efficiently in the other houses as all houses have basically the same overall air change rate of outdoor air
- the exhaust-ventilated houses exceed 1000 ppm carbondioxid and have higher levels than the air-heated houses, which can be explained by the difference in the purging air flow
- the relative humidity indoors is somewhat lower than it should be during the winter season and there is no difference between the ventilation systems
- the indoor air temperature is 22 °C and no difference between the ventilation systems.

In the questionnaire the occupants were asked to write down their use of energy during 1992. Most of the houses seem to have a low and reasonable level of energy use.

5. CONCLUSIONS

According to the questionnaire survey there is no difference compared with the Swedish housing stock as to frequencies of complaints and symptoms concerning the indoor climate. The Swedish housing stock was examined in the ELIB-study and the conclusion from that study is that between 7 % and 11 % of the Swedish population is subject to an indoor climate in their homes which can influence the health and the comfort. The questionnaire survey and measurements in this project does not show any serious differences between houses with radiator and air heating as to indoor climate.

This project and a previous project (Johansson 1993) have shown that air heating and radiator heating can result in a thermal comfort of equal quality. In the previous project it was shown that both systems meet criteria for thermal comfort even at low outdoor temperatures.

This project has on a few points confirmed and other points not confirmed the disadvantages and advantages with air heating, which were presented at a nordic seminar.

Nordic seminar on warm air heating	**According to SABO-questionnaire**	**According to diagnostic tests/long-term measurements**
Advantages with warm air heating:		
- Possibilities of limiting noise from outside.	Confirmed i. e. in air-heated houses fewer occupants are disturbed by noise from the outside than in exhaust-ventilated houses.	No conclusions possible.
- Less risk of draught (compared with exhaust and naturally ventilated houses)	Confirmed for exhaust-ventilated houses: In air-heated houses fewer occupants are disturbed by draught in one or some rooms than in exhaust houses.	Not confirmed.
- Fast control of temperature in the whole house-/zone (compared with hydronic heating i e naturally, exhaust or balanced ventilated houses)	Partly confirmed: Fewer occupants of air-heated houses are often disturbed by varying temperatures due to changes in outdoor climate, than in radiator-heated houses.	Not tested.
- Guaranteed air flow to each room (compared with exhaust and naturally ventilated houses).	No conclusions possible.	Confirmed in the bedrooms: The purging and the outdoor air flow were lowest in the exhaust-ventilated houses.
Disadvantages with warm air heating:		
- Less possibility of control of temperature of indivdual rooms (compared with exhaust and naturally ventilated radiator-heated houses)	Not confirmed: No difference between air and radiator-heated houses as to the perception of one or some rooms being too cold.	Not confirmed.
- Risk of annoying noise from ventilation system.	Confirmed: According to 30 % of the occupants in air-heated houses sound from the ventilation system is sometimes or often disturbing. The corresponding value for exhaust-ventilated houses is 23 %.	Confirmed: The highest sound level from ventilation was measured in balanced-ventilated houses, some were above the code requirement.

According to the questionnaire the occupants air more in exhaust-ventilated houses than in balanced-ventilated houses. The reason could be that according to the measurements the ventilation rates, in particular in bedrooms, are lower and the carbondioxide concentration higher in bedrooms of exhaust-ventilated houses.

The following improvements should be made in the one-family houses of the future:
- improve upon the abatement of sound from the outside in exhaust houses
- reduce the draught from outdoor air vents
- improve upon the control of the indoor temperature i e the coupling to the outdoor climate
- develop quieter ventilation systems for air-heated and balanced-ventilated houses
- make the operating and maintenance instructions easier to understand for the occupants
- reduce the spread of cooking fumes within the dwelling
- investigate the possibility of reducing the outdoor air flow below 0,5 ach
- investigate the possibility of supplying all outdoor through the bedrooms

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Implementing the Results of Ventilation Research
16th AIVC Conference, Palm Springs, USA
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Feasibility of Ventilation Heat Recovery in Retrofitting Multi-Family Buildings

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SYNOPSIS

The work concentrated on estimating the effects of building leakages and terrain parameters on the air infiltration. The analysis was performed mainly using a multi zone airflow model Movecomp with which the building and its ventilation system could be described in detail. The computations were performed for a flat in a 4/8-storey building. The highest infiltration occurred in an untight two-facade flat in open terrain. The calculations gave valuable information on the effect of the location of the leakage. The vertical distribution of the leakage had the most significant effect on infiltration. On the other hand, the tightness of the floor/ceiling and the apartment door did not have significant effect on the whole building infiltration, nor did the number of storeys. The knowledge gained from these simulations will be used in designing sealing techniques for existing multi-family buildings.

The results revealed a significant reduction in the economic feasibility of heat recovery ventilation when the air-tightness of the envelope decreased. It was estimated that, in Finnish climate and energy prices, the air-tightness of the envelope should be in the range of 2-3 air changes at 50 Pa, or better, in order to air to air heat recovery to be economically feasible in existing buildings. This means that the renovation of the ventilation system to include air to air heat recovery should almost always be connected with sealing of the building envelope.

1. INTRODUCTION

The renovation of existing high-rise residential buildings is becoming a major part of the construction work in Finland. At the same time, reduction of energy consumption is required to conserve the global environment. In 1992, the Finnish government decided to reduce the consumption of residential heating energy by 15 % by the year 2005. It has been estimated that 40% of residential heating energy is used for ventilation, which means that the renovation of ventilation systems play a major role in decreasing the overall energy consumption in buildings. The possible improvements to existing systems were described in an earlier paper /1/. Since then the research has focused on the feasibility of air to air heat recovery in residential buildings. Air to air heat recovery has become popular in new single family buildings during the 80's. Many manufacturers provide integrated ventilation units and product development has cut the size and price of the systems. Although many single family buildings are equipped with balanced ventilation including heat recovery, its share in the whole residential building stock remains low. The aim of this research was to study the technical and economical barriers which hinder the use of air to air heat recovery in existing buildings. The research included designing system concepts based on discussions with manufacturers and decision makers in the building process, computer simulations of air flows in buildings and life cycle cost calculations of the feasibility of air to air heat recovery.

2. SYSTEM CONCEPTS FOR AIR TO AIR HEAT RECOVERY

2.1 Typical present system

The majority (ca. 70%) of Finnish apartment buildings is equipped with mechanical exhaust ventilation. Each apartment has typically 2-3 exhaust points, at kitchen and at bathroom. The exhaust valves have a high pressure drop and usually an integrated sound attenuator. Common ductwork connects the apartments in the same vertical shaft. Circular sheet metal duct has been used since 1970's, but older buildings have masonry ducts. Typical pressure loss in the ductwork is 50-100 Pa. The system is usually equipped with a clock-controlled two-speed exhaust fan in the roof. The outdoor air usually enters through cracks in the windows or other components of the envelope. Purpose-built openings for incoming air, or air inlets, were taken into building practice since 1988 when the latest building code came into force. At the same time exhaust hoods in the kitchen became practically mandatory.

2.1 Central ventilation unit with ducted supply

One solution to air to air heat recovery is to install a new supply system and connect the existing, possibly refurbished, exhaust system to it via a heat exchanger. A plan of such solution is presented in Fig. 1. The system needs only one supply fan, which brings benefits in maintenance costs compared to apartment based systems. The installation of supply ductwork in an existing building can, however, cause significant costs. Using the staircase as a supply chamber would reduce the installation costs, but fire safety reasons and fear of contamination have restricted the use of such systems. According to the Finnish Building Code, fire dampers would be required in each apartment and the supply fan should be equipped with a fire emergency circuit breaker. A promising solution for the ducting problem would be placing the ducts outside the building envelope. Architectural reasons can be a major restriction if outside ducting is planned without redesigning the whole facade.

2.2 Apartment based ventilation units

Installing apartment based ventilation units is another possibility for air to air heat recovery. A system with both supply and exhaust fans, heat exchanger, filter and preheater can be installed in a top part of a standard cupboard. Only short ductwork is needed, see Fig. 2. The exhaust air can be discharged on the wall, or the existing ductwork can be used to lead the exhaust air to the roof. Unfortunately, neither alternative is accepted in the present Finnish Building Code as overpressurized ducts from bathrooms are not allowed. Experimental buildings and research projects have, however, removed some resistance against wall exhaust discharges /6/. The ease of installation is the major advantage of apartment based systems. The maintenance costs of apartment based systems are expected to be higher than in central systems due to a greater number of components and the difficulties of having to visit homes in order to carry out the maintenance. Another drawback of installing the system in the apartments is the fact that the cost of the preheating electricity is transferred from the building owner to the apartment owner/lodger. This can be prevented by installing a separate electricity network for the ventilation units or by using a hot water preheting coil connected to the central heating network.

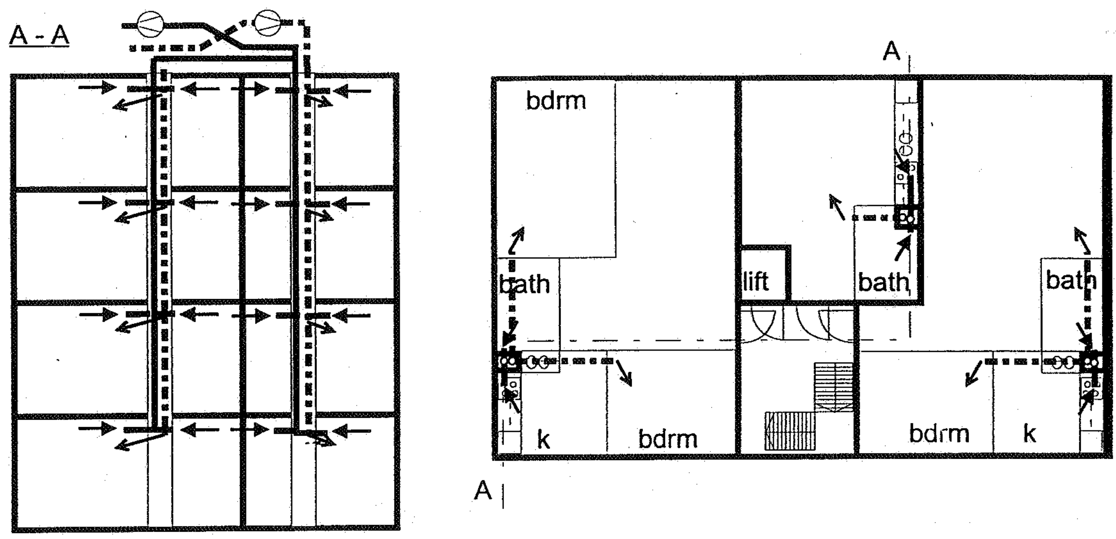

Fig. 1. Plan and section of a typical Finnish apartment block with a central ventilation unit with ducted supply.

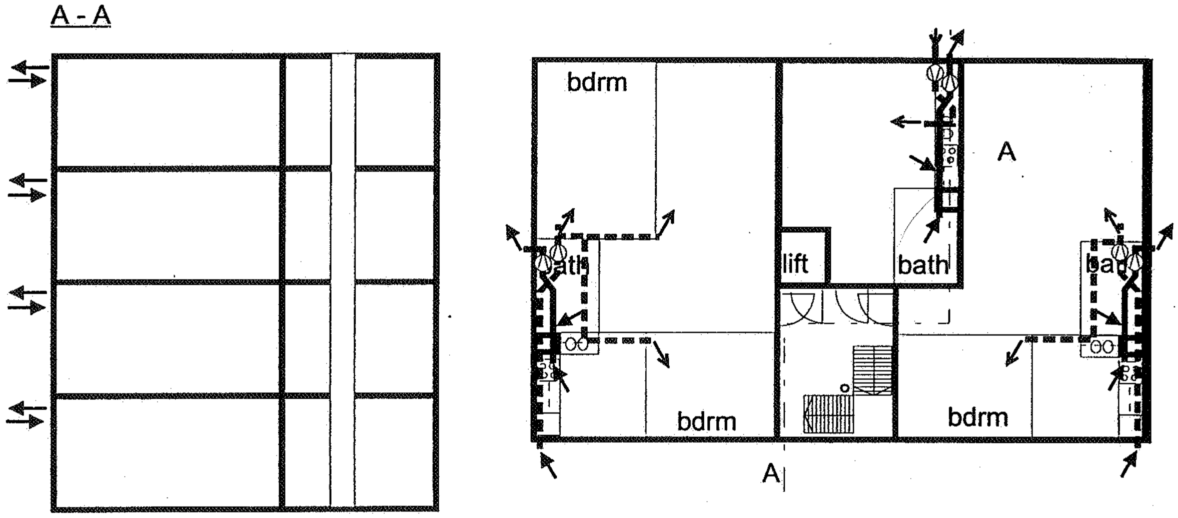

Fig. 2. Plan and section of a typical Finnish apartment block with apartment based ventilation units.

3. COMPUTER SIMULATIONS

3.1 The simulation model

The air flow analysis was performed using a multi zone airflow model Movecomp /2/, with which the building and the ventilation system could be described in detail. The building has a basement and 3 or 7 inhabited floors. The length, width and height of the building are 75 m, 12 m and 14 m respectively. Most of the flats have only two walls facing the outside. Therefore, it was considered reasonable to compute only one 63 m^2 flat on each floor. The air leakages are set to measured values in the actual building, see /1/. The pressure coefficients for the 12 wind directions were taken from reference /5/.

The simulations were performed for a total of 182 weather conditions. The annual results were obtained using the probability of each weather condition at Helsinki-Vantaa and at Rovaniemi airport, Finland, over 30 years.

3.2 Studied parameters

In order to be able to estimate the economic feasibility of heat recovery it is essential to know how much air is exfiltrated from the building. The effects of the following parameters were studied:

- terrain shielding
- tightness of the envelope
- tightness of the intermediate floor
- tightness of the apartment door
- weather data (Helsinki vs. Rovaniemi)
- supply/exhaust air flow difference
- the height of the building
- location of the outdoor air leakage

The terrain shielding was estimated based on the method described in AIVC's Numerical Database /4/. The following equation was used to estimate the wind speed (v_h) at height (h): $v_h = v_m K h^a$, coefficients K and a are given in table 1.

Table 1. Air leakages of the investigated building /4/.

Terrain type	K	a
Open flat country	0.68	0.17
Urban	0.35	0.25
City	0.21	0.33

3.3 Results

The main results of the air flow calculations are shown in figures 3 and 4. It can be seen that the air tightness of the envelope and the terrain shielding are the dominating factors in air infiltration. The effects of supply/exhaust air flow difference and the height of the building are negligible compared to these. The vertical distribution of the leakage had a significant effect on infiltration. In tightening the building it is essential to concentrate on the leakages on the outside wall near the floor and the ceiling. On the other hand, the tightness of the floor/ceiling and the tightness of the apartment door did not have any notable effect on the whole building infiltration. However, their role may be important in the transport of contaminants between apartments.

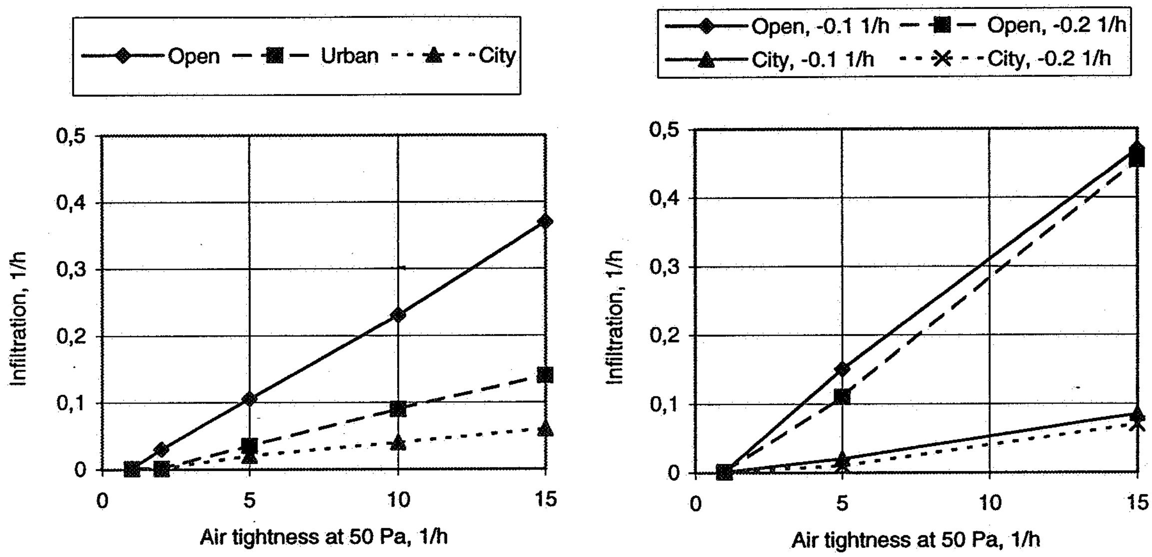

Fig. 3. a. Infiltration at different terrain conditions (Helsinki); b. Infiltration at different supply/exhaust air flow difference (exhaust 0,5 1/h, supply 0,3 1/h or 0,4 1/h) and terrain conditions (Rovaniemi).

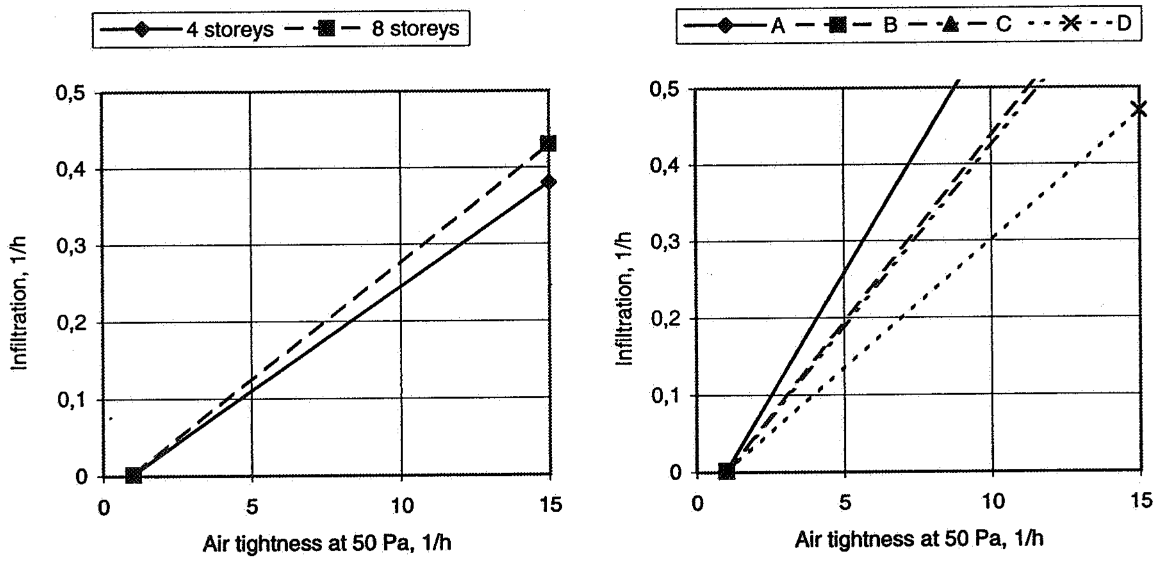

Fig. 4. a. Infiltration at buildings of different height (Helsinki); b. Infiltration depending on the location of the outdoor air leakage (Rovaniemi, A: cracks at 0.0 m and 2.5 m; B: 50% of leakage at 1.25 m, 25 % at 0.0 m and 2.5 m; C: cracks at 1.0 m and 2.0 m; D: one crack at 1.25 m).

4. LIFE CYCLE COST CALCULATIONS

4.1 Calculation method

A simple steady-state energy balance was used to calculate the energy recovered by air to air heat exchanger. The balance consisted of losses due to transmission, ventilation and infiltration, and internal gains including solar radiation. The heating energy was calculated for the whole year at 2°C intervals, the recovered ventilation energy was calculated only when external heating was required and when the outdoor temperature was above the anti-freezing set point of heat recovery air exchange.

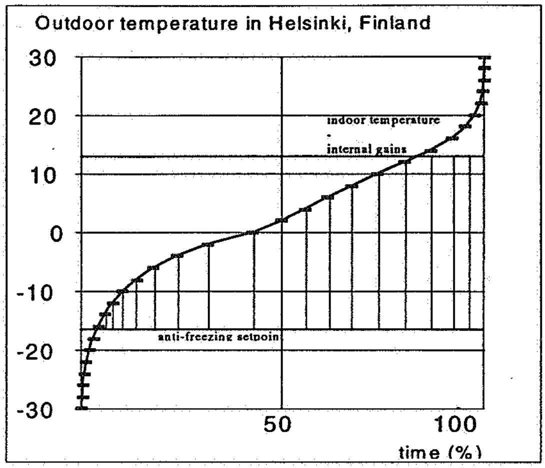

Fig. 5. The calculation of the energy saved by the air to air heat recovery (vertical hatch).

The life-cycle cost calculation is an analysis of the present values of future costs. The economically feasible investment to a given system was calculated based on the estimated maintenance and repair costs. The maintenance costs included: the cost of heating the ventilation air, the reduction in the initiation fee, the regular maintenance costs, and the cost of the electricity used by the fans.

4.2 Studied parameters

The various system solutions to air to air heat recovery were compared to a standard mechanical exhaust system. A similar base level of investment costs was assumed for all systems (even the existing exhaust system needs repairing to meet the present requirements). The total air exchange rate was set at 0.5 1/h. Infiltration was included in the total rate in order to achieve a comparable indoor air quality in all systems. The infiltration was subtracted from the exhaust air flow in the balanced systems to give the appropriate energy penalty. For example, with an infiltration of 0.15 1/h, only 0.35 1/h of exhaust air goes through the heat exchanger.

4.3 Results

The results of the air flow calculations were combined with the life cycle cost calculations and the results were presented in nomograms, see Fig. 6 for an example. At an air tightness of 8 1/h at 50 Pa, the economically feasible investment to air to air heat recovery is 80 FIM/m^2 in urban terrain, but only 30 FIM/m^2 in open terrain. The results depend strongly on the weather data, the costs of heat and electricity, and on the rate of the interest. Therefore, is it difficult to apply the nomograms outside Finland. They will, however, give an indicative picture of the effect of air infiltration on the economic feasibility of air to air heat recovery.

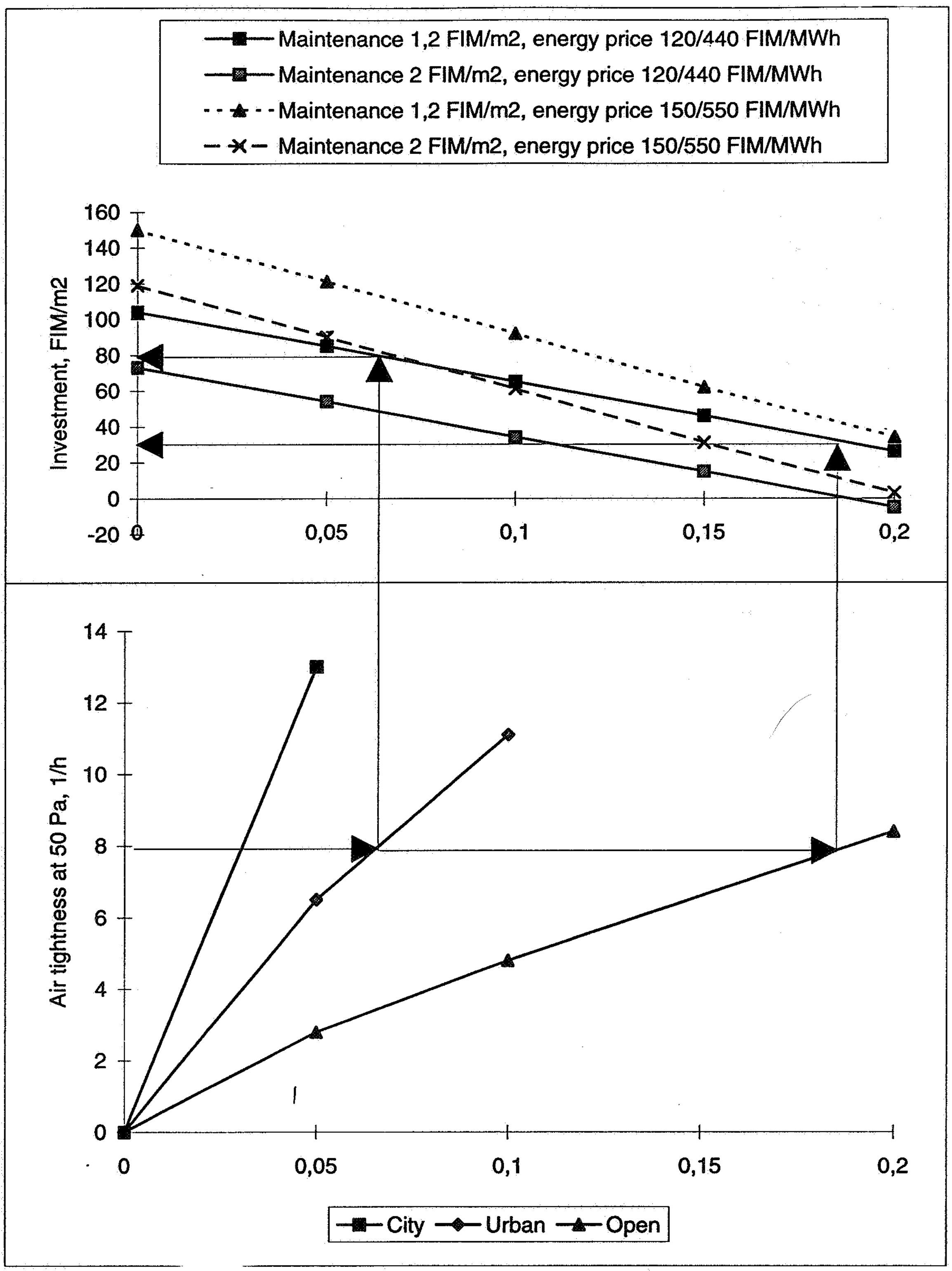

Fig. 6. The economically feasible investment to the air to air heat recovery in Finnish climate. The two levels of energy prices are: heat 120 FIM/MWh, electricity 440 FIM/MWh; and heat 150 FIM/MWh, electricity 550 FIM/MWh. The efficiency of heat recovery is 65 %.

5. CONCLUSIONS

Main part of the work concentrated on estimating the effect of building leakages and terrain parameters on the infiltration. The aim was to be able to give guidance on the feasibility of ventilation heat recovery in the design phase. The computations were performed for a flat in a 4/8-storey building. The highest infiltration occurred in an untight two-facade flat in open terrain. The calculations gave valuable information on the effect of the location of the leakage. The vertical distribution of the leakage had the most significant effect on infiltration. On the other hand, the tightness of the floor/ceiling and the apartment door did not have significant effect on the whole building infiltration, nor did the number of storeys. The knowledge gained from these simulations will be used in designing sealing techniques for existing multi-family buildings.

The results of the simulations were used as a basis for estimating the (uncontrolled) infiltration and its effects on the economic feasibility of air to air heat recovery ventilation. Life-cycle cost calculations were performed for various heat recovery ventilation systems which were compared with a mechanical exhaust system. The result revealed a significant reduction in the economic feasibility of heat recovery ventilation when the air-tightness of the envelope decreased. It was estimated that, in Finnish climate and energy prices, the air-tightness of the envelope should be in the range of 2-3 air changes at 50 Pa, or better, in order to air to air heat recovery to be economically feasible in existing buildings. This means that the renovation of the ventilation system to include air to air heat recovery should almost always be connected with sealing of the building envelope.

ACKNOWLEDGEMENTS

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Implementing the Results of Ventilation Research
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The Indoor Air Quality and the Ventilation Performance of Four Residential Buildings with Dynamic Insulation

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THE INDOOR AIR QUALITY AND THE VENTILATION PERFORMANCE IN FOUR RESIDENTIAL BUILDINGS WITH DYNAMIC INSULATION

SYNOPSIS

Dynamic insulation has been used in non domestic buildings for 20- 30 years in order to reduce the heat loss and to bring preheated air into the buildings. Dynamic insulation means a construction where the air is being forced through the insulation, usually from the colder outside air into the heated building.
The Norwegian Building Research Institute has been engaged to evaluate 12 row houses, with dynamic insulation used in the roof, which has been built in the Oslo area. 4 of the houses were monitored over a period of time. In two of the houses the thermal performance were monitored when the houses were new. In the other two the indoor air quality, the ventilation performance etc. were measured after two years of occupation. The paper presents the results from these measurements. The main conclusion from the work is that dynamic insulation gives a good indoor climate with less conduction heat loss than an ordinary construction with the same thickness of insulation.

THEORY

The air-flow is normally forced the opposite direction of the conduction, thus the term *contraflux insulation*. When the airflow has the same direction as the conduction, *proflux insulation*, there is a big risk of moisture condensation inside the construction. All the tests described in this paper is carried out with contraflux insulation.

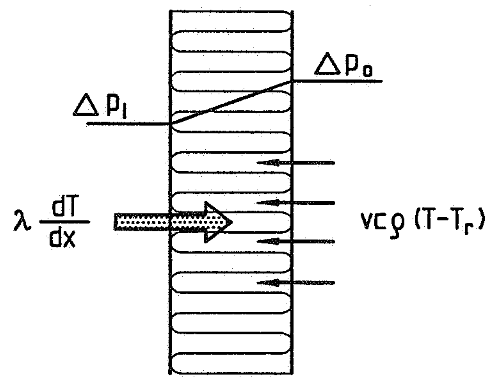

Figure 1.
Direction of conduction and convection in a contraflux insulation.

The conduction and the convection heat flow can be described mathematically as follows:

$$q_d = \lambda \frac{dT}{dx}$$

$$q_v = v\rho c_p\,(T-T_r)$$

In a steady-state situation Anderlind (1) first showed that the temperature distribution in a homogeneous insulation layer can be described as:

$$T = T_u + (T_i - T_u)\frac{e^{-\frac{ax}{d}} - e^{-a}}{1 - e^{-a} + \frac{a}{b-a}}$$

where $a = \frac{v\rho c_p d}{\lambda}$ and $b = \frac{\alpha_i d}{\lambda}$

The concept "dynamic U-value" can be defined as the U-value of a construction with "static" insulation with the same conduction heat loss as the construction with the dynamic insulation. For a homogeneous insulation layer the dynamic U-value in the steady state situation can be shown to be:

$$U = \frac{\lambda}{d} \cdot \frac{a\, e^{-a_1}}{1 - e^{-a} + \frac{a}{b-a}}$$

The U-value as a function of the air velocity and the thickness of the insulation is presented in figure 2.

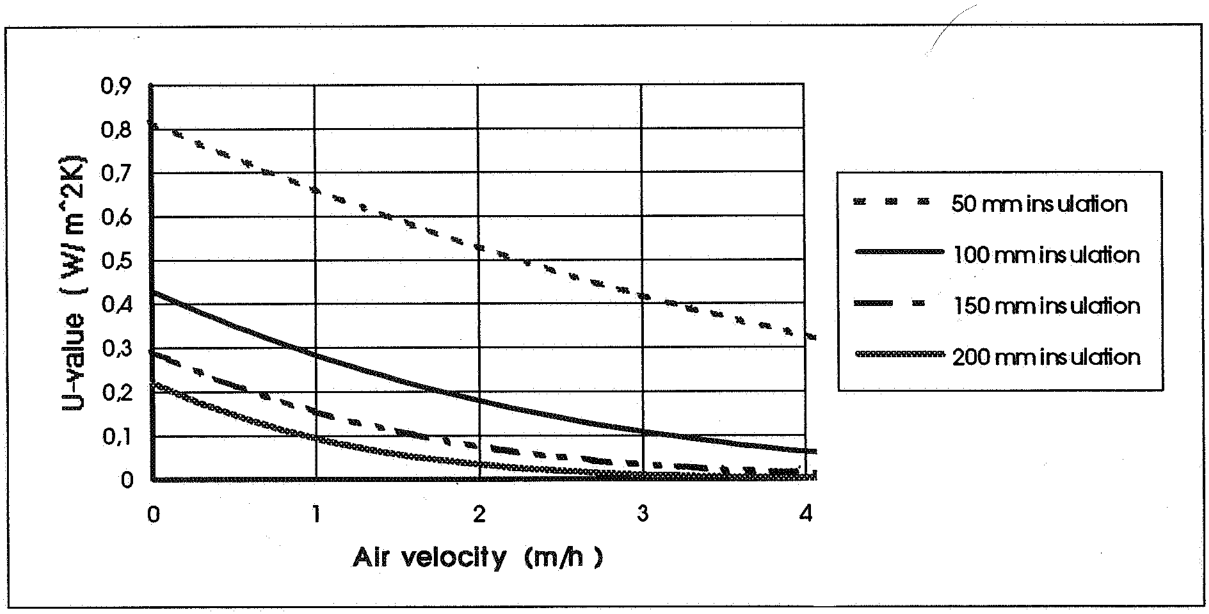

Figure 2
"Dynamic U-value" for a construction with 50 -200 mm dynamic insulation as a function of the air velocity.

FIELD EXPERIMENTS

The construction of the 12 houses with dynamic insulation in the roof is shown in figure 3. Several strategies can be used to force the air through the insulation:

1. Overpressure in the attic
2. Underpressure inside the house
3. Combination of 1. and 2..

The necessary and acceptable pressure difference will also vary with the different strategies and type of houses.

For the 12 row houses the strategy 2. was chosen.
The underpressure was planned to be 10 Pa. We considered 10 Pa to be the highest pressure difference that could be acceptable for the users (opening of windows and doors etc.). 10 Pa was also considered enough to avoid the wind causing a change of the flow direction through the roof or the walls more than 5 % of the time. The velocity through the insulation with 10 Pa pressure difference should be 2 m/h which should give a ventilation rate alone of 0.8 ach. If the n_{50} value, based on leakages from the rest of the house, was less than 1.0 ach, an airflow which corresponds to about 0.3 ach would come through the leaks in the walls and the floor at a pressure difference of 10 Pa. Together this will give a total ventilation rate of 1.1 ach.

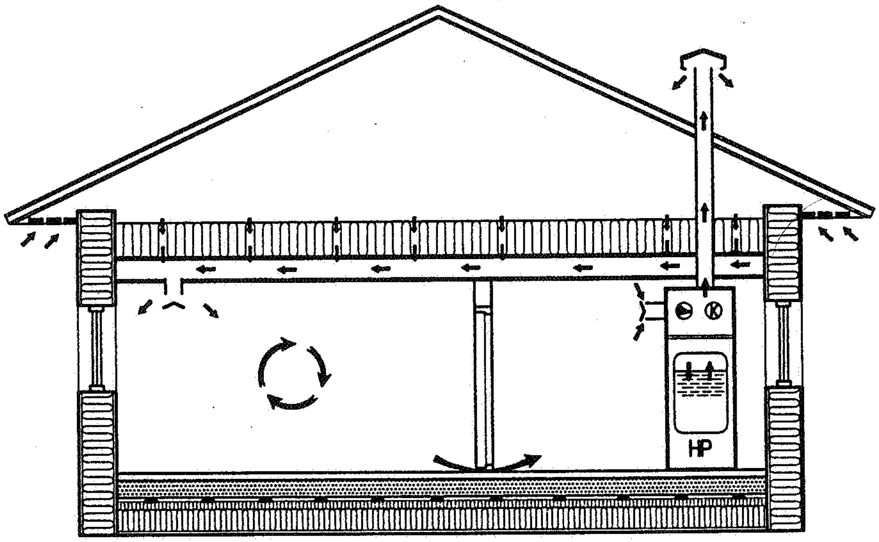

Figure 3.
The 12 row-houses have dynamic insulation in the roof. The outside air is sucked into the attic and then down through the insulation and into the house through inlets. The driving force is an underpressure inside the house created by fans in the heatpump which also transfer the heat from the exhaust air to the hot water supply. The houses were built without any influence from the evaluation group.

The following parameters were measured in the houses (not all the parameters were measured in all of the four houses):

- Ventilation rate measured with tracer gas technique inside and in the attic
- Volume flow through the outlets
- Temperature inside, outside and the profile through the insulation
- Pressure differences between the outside, the attic, through the insulation and the inside
- Wind speed and wind direction
- TVOC, CO and CO_2 concentrations outdoors, in the incoming air and in the indoor air
- Number of particles outdoors and in the in incoming air
- Concentration of fungal propagules

Ventilation rate measurements were carried out with the constant concentration method using Brüel and Kjær equipment model 1302 and 1303. We had 6 channels for dosing and 6 for sampling. Values for the tracer gas measurements were logged every minute, then averaged values for 10 minutes were stored.
CO-, CO_2- and TVOC- concentration is continuously measured with the same instrument.

Values for temperature, pressure differences, wind speed and wind velocity were logged every 10th second and then the averaged values for 10 minutes were stored.

Concentration of fungal propagules is measured with a BIAP Slitsampler with DG18.

RESULTS

House 1 were occupied by the owners, an elderly couple, during the measurements. After the termination of the measurements and all the equipment were removed, they informed us that they had tried to air out the tracer gas by opening windows and doors. The pressure difference across the thermal envelope disappeared of course during these periods. In addition we had some problems with the tracer gas equipment and therefore there are no results from these measurements in house 1.

House 2 was unoccupied during the measurement period which can be divided into two parts. During the first period we did only tracer gas measurements inside the house. During the second period we had tracer gas measurements in the attic and sampler tubes in the insulation, in the air inlet and inside the livingroom.

House 3 was occupied by an elderly woman. When the measurements started we realized that the flow through the outlets were only 40 % of the planned value. This was due to the filter in front of the heat pump which had not been changed for two years and because the fan had been adjusted because of noise. A new filter was installed and the fan was adjusted the last day of the measurement in house 3 and the ventilation rate went up from about 0.4 ach to about 1.0 ach.

House 4 was also occupied by an elderly woman.

House 1 and 2 were measured 4 months after the houses were finished while house 3 and 4 were monitored after 2 years.

The main results are given as averaged values in table 1.

Table 1.
Averaged values from all the four houses. Each measurement lasted from 5 to 8 days.
1): Measured with Brüel and Kjær model 1302
2): Measured with adsorption for two hours on tenax TA and analysed with GC - MS.
3) 3 other of the 12 row houses were measured and the n_{50} values were: 4.8, 4.4, and 4.6 ach at 50 Pa pressure difference
- means either that there has not been measured any values or that the measurements were unsuccessful

Parameter	House 1	Outs.	House 2 -1	House 2 - 2	House 2 Attic	House 3	Outs.	House 4	Outs.
Temperatures(C)	21.1	5.3	-	25.6	13.9	24.1	1.2	22.3	-7.9
Temp air inlets (C)	18.0	-	-	-	-	21.9	-	12.0	-
Pressure diff (Pa)	8.2	-	-	6.2	-	3.9	-	10.4	-
CO livingr. mg/m^3 1)	2.6	-	3.3	3.4	2.7	-	-	-	-
CO_2 livingr. mg/m^3 1)	1170	-	954	1163	910	1127	784	1125	1048
TVOC livingr. mg/m^3 1	7.9	-	11.8	11.2	6.3	7.2	5.0	5.0	5.0
TVOC inlets µg/m^3 2)	-	-	-	-	-	225	380	289	380
Air tightness, n_{50}, 3)	-	-	4.2	-	-	4.2	-	-	-
Air change rate	-	-	0.58	-	2.0	0.54	-	1.09	-

Concentration of gas contaminants

Together with the tracer gas measurements some gas contaminants were monitored as well. The results do not indicate that outside air, going through this kind of insulation, brings contaminants from the insulation into the house as some have feared. This depends on the kind of insulation and other materials which are used in the dynamic construction.
These measurements are not fully analysed and will therefore not be discussed further in this paper.

Particles

When the houses were new the total concentration of particles in the incoming air was measured. In addition the amount of particles was measured in house 3 and after two years of operation.

Table 2.
Concentration of particles in the air inlets in house 5 and 6.

	House 5	House 6
Cons: - 2.5 µm (µg/m^3)	16.6	4.4
Cons. 2.5 - 15 µm (µg/m^3)	4.4	4.8
Total: - 15 µm (µg/m^3)	21	9.2

After 2 years the number of particles in house 3 and 4 was counted and results are shown in table 3.

Table 3.
Number of particles in house 3 and 4 and outside measured in January 1995, two years after the houses were new.

	House 3, inside	House 3, air inlet	Outside	House 4, inside	House 4, air inlet	Outside
Number of part.-2.5µm	41200	5850	86500		19800	249500
Number of part 2.5-15µm	1100	9	325		50	400

Fungal propagules

The concentration of fungal propagules was measured in house 3 and 4. The concentration, which is given in Colony Forming Unit (CFU), was measured in the air inlet, in the attic and outside. The results are given in table 4.

Table 4
Concentration of fungal propagules

	House 3 air inlet	House 3 attic	House 3 outside	House 4 air inlet	House 4 attic	House 4 outside
Colony forming units (CFU)	0	36	39	21	39	42

Tracer gas measurements

The ventilation rate for the Houses 2, 3 and 4 are given in figure 4.

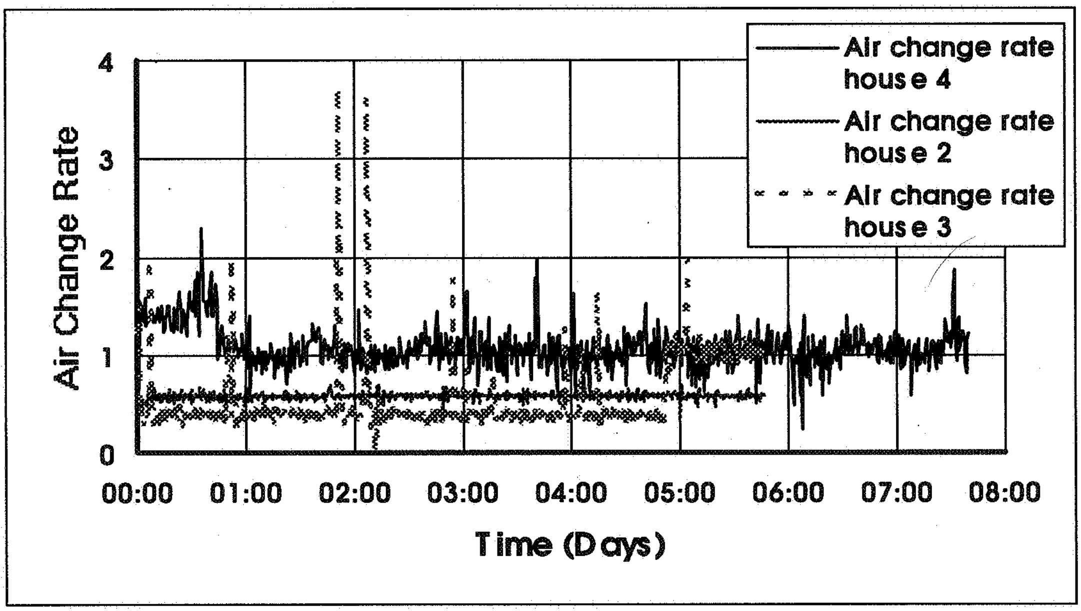

Figure 4
Ventilation rate for house 2, 3 and 4. The first half day in house 4 the pressure difference was not adjusted to the right level. For house 3 the pressure difference and therefore the air change rate was right only the 5th day.

The ventilation rate for the whole house 2 is rather constant as expected. With an underpressure of 6-7 Pa there must be a wind speed above 3-4 m/s towards a wall to change the flow direction through the leaks in the leeward wall and change the air change rate significantly.

In house 4 the average pressure difference was higher, 10.4 Pa, and hence the higher ventilation rate. Figure 4 also show the difference in the variation of the ventilation rate between an unoccupied house and an occupied house.

Pressure differences, temperatures and wind speed

These measurements were carried out in order to better evaluate the performance of the "dynamic construction".

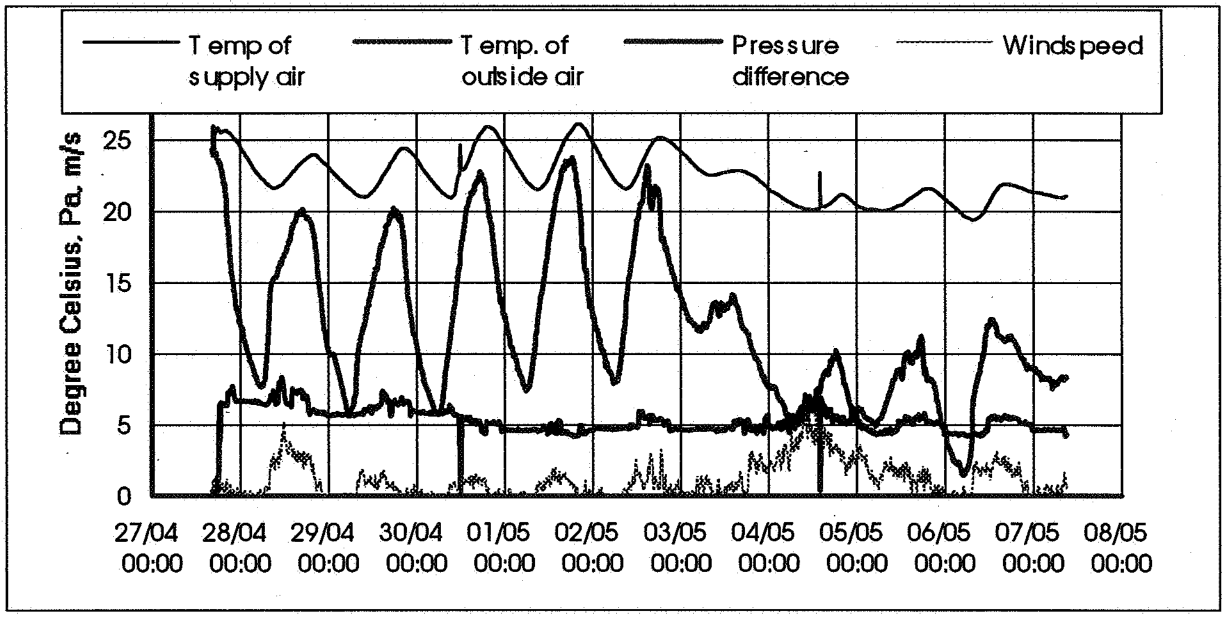

Figure 6
Temperature of supply air, pressure difference between the inside and the attic, temperature of the outside air and wind speed measured for house 2. The pressure difference is smaller than for house 1 but it is stable around 5-6 Pa.

For both house 1 and 2 we had 2 columns with temperature sensors through the insulation. For all the four columns we found the curved temperature profile which correspond to the theoretical curve.

DISCUSSIONS

From the measurements in house 1 and 2 it is clear that a smaller proportion of the total ventilation rate is coming through the roof than planned. There are two reasons for this:

- The n_{50} value is higher than planned
- There are leaks from the air layer and out through the walls

The measurements in house 1 and 2 also show that the total flow rate was smaller than planned even if the houses were leakier than planned. The reason for this is bad or none adjustment of the fan and all the outlets and inlets after the houses were completed.

The strategy for bringing in the air to the house has been to establish an underpressure inside. Since the houses are leakier than expected, more air will be sucked through the leaks. To avoid this, one action can be to establish an overpressure in the attic. This will give the possibility also to clean the air before it enters the insulation.

From the temperature measurements on both sides of the insulation and inside the insulation it is possible to calculate a "dynamic U - value" for house 1 and 2. The average U-values for the roofs for the measured periods for these two houses are:

House 1 $U = 0.10\ W/m^2 {}^\circ K$
House 2 $U = 0.06\ W/m^2 {}^\circ K$

The periods are too short however to find a representative "dynamic U-value" for the roofs. The values indicate though that the heat loss through the roof have been reduced with 55 % and 73 % respectively for the two houses compared to the conduction heat loss through a similar construction with static insulation. The U-value without any convection is $U = 0.22\ W/m^2 {}^\circ K$.

Air Quality

The results from these measurements show that the incoming air in these houses with dynamic insulation have a very good quality after two years of operation. The measurements show that there are no mould growth in the insulation. The content of particles in the air which has passed the insulation is significantly lower than in the outside air. Since the particles will be filtered in the insulation there is a possibility that this could cause problems after some years of operation . One could fear that there would be a growth of bacteria and mould which could result in spread of contaminants, e.g. endotoxin and spores, to the indoor environment. This has, however, not been found in this project (endotoxin is not measured in this project) nor been reported by others . The reason might be one or several of the following facts:

1. Buildings with dynamic insulation has been used for only 25 years.
2. The volume flow, through the insulation is very low.
3. There have not been many evaluations of indoor air quality of the oldest buildings

Re. 1:
There are older buildings with dynamic insulation than 25 years but only very few

Re. 2:
The air velocity through a construction with dynamic insulation has a typical value of about 2 m/h. In an ordinary filter the typical velocity through the filter material is about 400 m/h which is a factor of 200 higher. If it takes half a year to fill an ordinary filter it will take significantly more time before the insulation have to be changed. There are lots of uncertainties in such a simple calculation but it gives an idea of how fast the insulation will be contaminated.

Re 3:
The author is aware of just a few 10 years old buildings with dynamic insulation in which the indoor air quality have been evaluated. There are not reported any mould growth or high concentration of endotoxin in the insulation.

Since there are uncertainties, and "better to be safe than sorry", we recommend to have a filter on the incoming air before it enters the insulation. This is a necessity where the insulation, not in a simple way, can be removed and replaced by a clean insulation if the insulation will cause problems before the building is ready to be condemned

CONCLUSIONS

The following conclusions can be drawn from the field experiments on four residential buildings:

- Dynamic insulation can give good indoor air quality and ventilation without draught in residential buildings.
- Dynamic insulation can reduce conduction heat loss from the construction to 0.
- It is important to choose a strategy on how to get the right air flow rate into the house without influence of the weather condition, the air tightness of the house and the users.
- The incoming air in house 3 and 4, after it has passed the insulation, contain only 7% and 8% of the number of particles in the outside air.
- The measurements do not indicate any mould growth in the insulation

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LIST OF SYMBOLS

A	Roof area	m^2
Q_a	Volume of air flow rate from attic to apartment	m^3/h
Q_t	Volume of total air flow rate through the apartment	m^3/h
T_i	Temperature inside	°C
T_o	Temperature outside	°C
T_r	Reference temperature	°C
U	Thermal permeance	$W/m^2{}^\circ C$
V	Volume of the apartment	m^3
c_a	Concentration of tracer gas in the attic	ppm
c_i	Concentration of tracer gas in the apartment	ppm
c_p	Specific heat capacity	J/kg°C
d	Thickness	m
n	ventilation rate	h^{-1}
n_{50}	ventilation rate at 50 Pa pressure difference	h^{-1}
q_d	Conduction heat flow	W/m^2
q_v	Convection heat flow	W/m^2
v	Air velocity	m/h
α_i	Surface film coefficient for heat transfer	$W/m^2{}^\circ C$
λ	Thermal conductivity	W/m°C
ρ	Density	kg/m^3

Implementing the Results of Ventilation Research
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Criteria for Heat Recovery and Dehumidification

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CRITERIA FOR HEAT RECOVERY AND DEHUMIDIFICATION

1. Synopsis

Two factors - CO_2 emissions from heating and cooling systems and restrictions on the use of CFC refrigerants - have accelerated the development and introduction of new and more environmentally friendly cooling systems. These new cooling systems also include the so-called "Desiccant Cooling Systems (DCS)" [1]. The desiccant cooling systems consist of a rotating dehumidifier, a rotating heat exchanger and evaporative coolers. For design, control and operation of desiccant cooling systems new criteria have to be considered because of the specific properties of these new technologies. Therefore consulting engineers as well as installers of air conditioning systems are hesitating to trust the efficiency and performance of these new components. Dehumdification of moist air is one of the least known and understood thermodynamic processes. On the other hand the same rotating desiccant wheels and the same rotating heat exchangers are used for many years in a similar joint combination in thousands of installations in the field of air dehumidification for industrial processes for heat recovery and cooling.

In general the dehumidification rotor is devided into two sectors (Fig. 1). One is the process zone where humidity is accumulated. The other zone is the regeneration zone where humidity is removed by the counterflow of heated air. The typical operation is unbalanced with a ratio of 1:3 for regeneration to process zone area or air flow. For a balanced ratio the regeneration temperatures can be below 80° C. To reach high efficiency for the dehumidification process the rotor is divided in three sectors to have in a multistep operation an intermediate cooling in the so-called purge sector.

In the following a standard design of an industrial air dehumidification system is described in order to show that design criteria as well as experience can be derived and transferred to the desiccant cooling systems in the comfort field application.

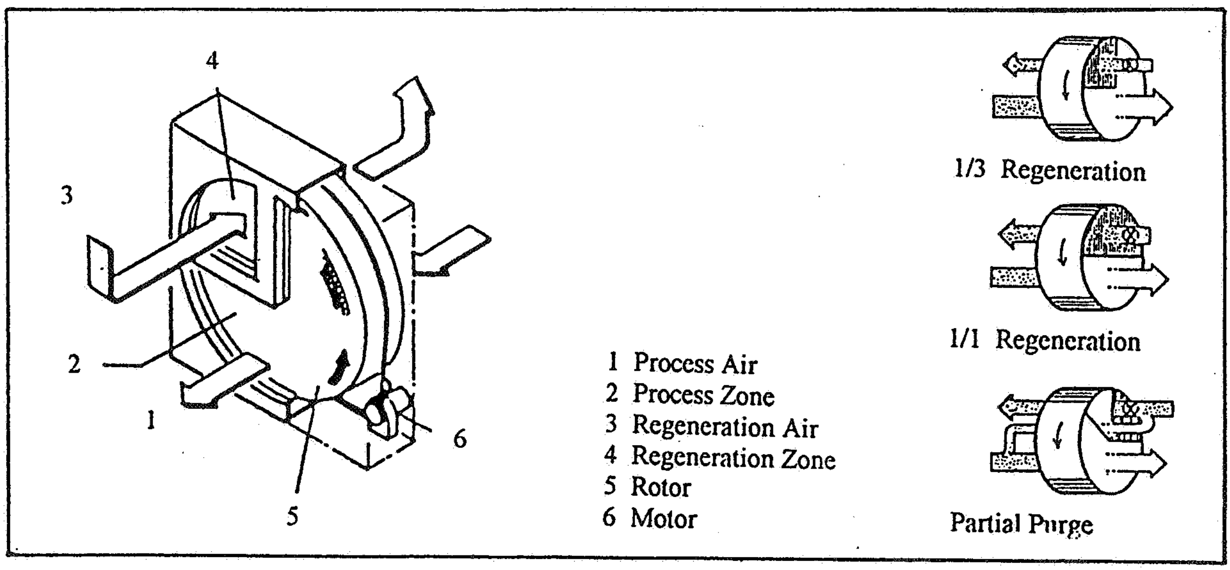

Fig. 1: Desiccant Wheel for Dehumidification of Process Air

2. Desiccant Cooling Systems

According to Fig. 2 (diagram and psychrometric chart) the desiccant wheel "A" rotates within the outdoor air stream and removes the moisture from it (1→2). The most suitable rotor is fabricated of silica gel reinforced with inorganic fibres and formed into a honeycomb shape. It has an excellent water adsorbing ability. The adsorption of moisture on the silica gel causes the temperature of the air to rise. The heat generated during the drying step is removed from the air by the rotating heat recovery wheel "B" with high efficiency (2→3) . This heat recovery wheel is non-hygroscopic and made of corrugated aluminium. The evaporative cooler "C" humidifies the dried air to further reduce the dry bulb temperature of the supply air stream (3→4). For the reactivation cycle the return air is used by first reducing the dry bulb temperature in the evaporative cooler "D". (5→6). The heat originally generated during dehumification of the supply air is then removed and transferred back into the reactivation cycle by the heat recovery wheel "B" (6→7). In the heat exchanger "E" external heat energy brings the reactivation air to the required temperature for desorbing the desiccant wheel "A" (7→8). Due to the synthesized silica gel this temperature can be set to a minimum which allows to use low level waste heat available from many heat processes, cogeneration processes and also solar energy. When desorbing the desiccant wheel "A" the exhaust air temperature is reduced by increasing the absolute humidity (8→9).

The new desiccant cooling systems can operate during winter (Fig. 3) and summer seasons for heating, humidifying, cooling and dehumidifying the supply air in the same way as a traditional air conditioning device according to a special control equipment. A lot of proposals were added to the basic configuration and are compared to each other and the conventional mechanical cooling systems in terms of performance coefficients. This values depend strongly on the efficiency of each component of the desiccant cooling system as well as the total configuration. The highest efficiency is required in any case for the heat recovery exchangers since the enthalpy reduction of the supply air and the heat recovery for regeneration process occur just there. It is important to understand that the evaporative coolers cannot reduce the enthalpy of the air leaving the dehumidifier in spite of reducing the air temperatures. As the dehumidification is also an adiabatic process, the dehumidification requires the reduction of air temperature with the mentioned high heat recovery efficiencies and the subsequent step of evaporative cooling. Only under this conditions the desiccant cooling systems are able to operate in terms of competing with mechanical systems.

Heat recovery devices and evaporative coolers are very well known in many engineering applications and their efficiency has been improved close to the maximum especially for rotating heat exchangers. The continuous desiccant dehumidification is a relatively recent technology and its operation characteristics and design procedures remain fixed in many respects to the product manufacturers and specialized companies working in the field of industrial air dehumidification and the endusers applying the main technology - the regenerative desiccant dehumidifier.

Controlling both supply air temperature and supply air humidity in an overall yearly desiccant cooling operation mode needs very new design strategies. Supply air temperature and humidity depend on the rotation speed of the adsorption wheel and the heat recovery wheel as well as on the reactivation temperature and the inlet moisture content and temperature of both air streams. However the main characteristics for the design of a desiccant cooling system depend of the properties of the various desiccant materials used such as silica gel, molcular sieves, lithium chloride and activated carbon. The structure and the material of the honeycomb core are also very important for fixing the desiccant on the surface of the rotor matrix.

3. Industrial Dehumidification Systems

Dehumidification of air is essential in nearly every segment of industry, in research and in preservation and storage of products, raw materials and even foofstuffs. Dehumidification is often the key to higher productivity by significantly reducing product rejection and production time.

As industry looks for new ways to improve productivity, moisture control is experiencing broader application as a means of producing higher quality products or greater product volume using existing manufacturing equipment.

Many areas of high technology cannot function today without controlled climate. Development of efficient, dependable and versatile dehumidification systems has frequently been the key to the success of a new technology.

The pharmaceutical industry is a classic example of both productivity improvement and new technology availability attributable directly to the utilization of dehumidification. Extremly dry air is required for processing hygroscopic pharmaceutical products such as capsule forming, tablet compressing, sterile filling, powder drying, ultra clean handling, gelatine based coatings, fabrication of laminated materials and so on. The factories are therefore classified according to the room air humidity. The requirements for controlled low humidity room air conditions are during the winter operation 20 to 22° C at 15 to 30 % r. H. and during the summer season 24 to 26° C at 15 to 25 % r. H. . To guarantee these room air conditions it is necessary to have highly qualified air condition systems with dehumidification of the supply air to dewpoints down to -15° C and lower in a 24 hour operation per day.

Choosing a dehumidification system involves a series of decisions, beginning with a review of the comparative merits of refrigerant and desiccant dehumidifiers. Basic factors are capital investment, operating costs and maintenance costs. In terms of capital investment, mechanical refrigeration systems are usually less costly for delivered air dew points above 5° C. Maintenance costs may be equal to or higher than desiccant systems. In general, operating costs will be a very important selection factor. However, as a guideline for dewpoints less than 5° C , desiccant dehumidifiers should be used. Above the 5° C dew point both mechanical refrigeration and desiccant systems can be evaluated on the basis of comparing operating costs. A comprehensive assessment requires consideration of refrigeration efficiency, dehumidifier efficiency, heat recovery efficiency, moisture removal rates, delivered dry bulb and dew point temperatures.

A typical application for dehumidification of room air and process air in the pharmaceutical industry is presented according to Fig. 4 and Fig. 5. The air treatment for heating, cooling and dehumidifying outdoor air is done by the combination of three heat recovery wheels, a desiccant dehumidifier wheel and a cooling coil. Normally process air is outdoor air only without mixing return air to prevent process air contamination.

During summer operation (Fig. 4) the outdoor air is cooled and dehumidified by the enthalpy of the room return air with a rotating total heat exchanger "A" (1→2). The return air is raising humidity and temperature (6→7). The dehumidification of the outdoor air down to the required dew point of -15° C (1 g/kg) follows with the desiccant wheel "B" (2→3). The first cooling step of the process air is in the rotating sensible heat exchanger "C" (3→4) using the return air after the total heat exchanger (7→8). The second cooling step is the cooling coil (4→5) down to the required supply air condition. The main advantages of this design is an impressive reduction of the cooling capacity and cooling energy by the combination of two heat recovery wheels with the desiccant wheel.

The winter operation (Fig. 5) is very simple. After the total heat exchanger (1→2) which is operating as a preheater the outdoor air is dehumidified to the supply air condition by controlling the rotation speed of both rotors (2→3).

The reactivation of the desiccant wheel needs thermal energy for summer and winter operation. The required capacity and the energy demand for this reactivation cycle is also reduced by using a rotating sensible heat exchanger "D" with an efficiency of up to 80 %.

4. Summary

For both desiccant cooling systems and dehumidification systems humidity and temperature control are important to meet the room air or process air conditions. Such cases require that dehumidifiers and heat recovery systems together with other components as cooling and heating coils, evaporative coolers, filters, air-moving devices and sensors are integrated into an overall efficient and effective air handling system. The basic design parameters of these air handling systems are different from conventional air conditioning systems. It is obvious that longlasting experience and research with desiccant and heat recovery wheels in industrial dehumidification systems can be transferred to the new desiccant cooling systems and design and operation criteria can be applied without any restriction [2].

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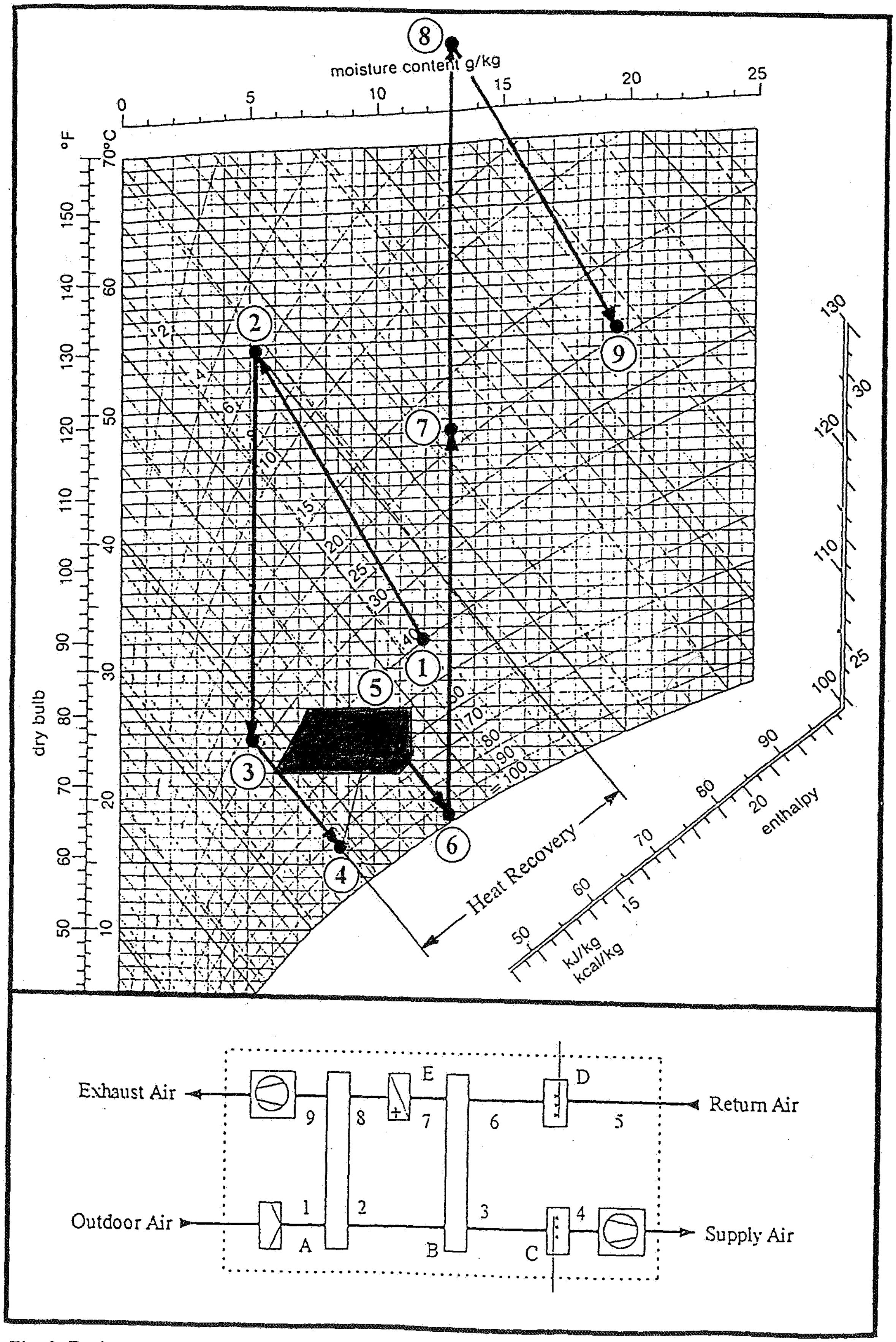

Fig. 2: Desiccant Cooling Summer Operation

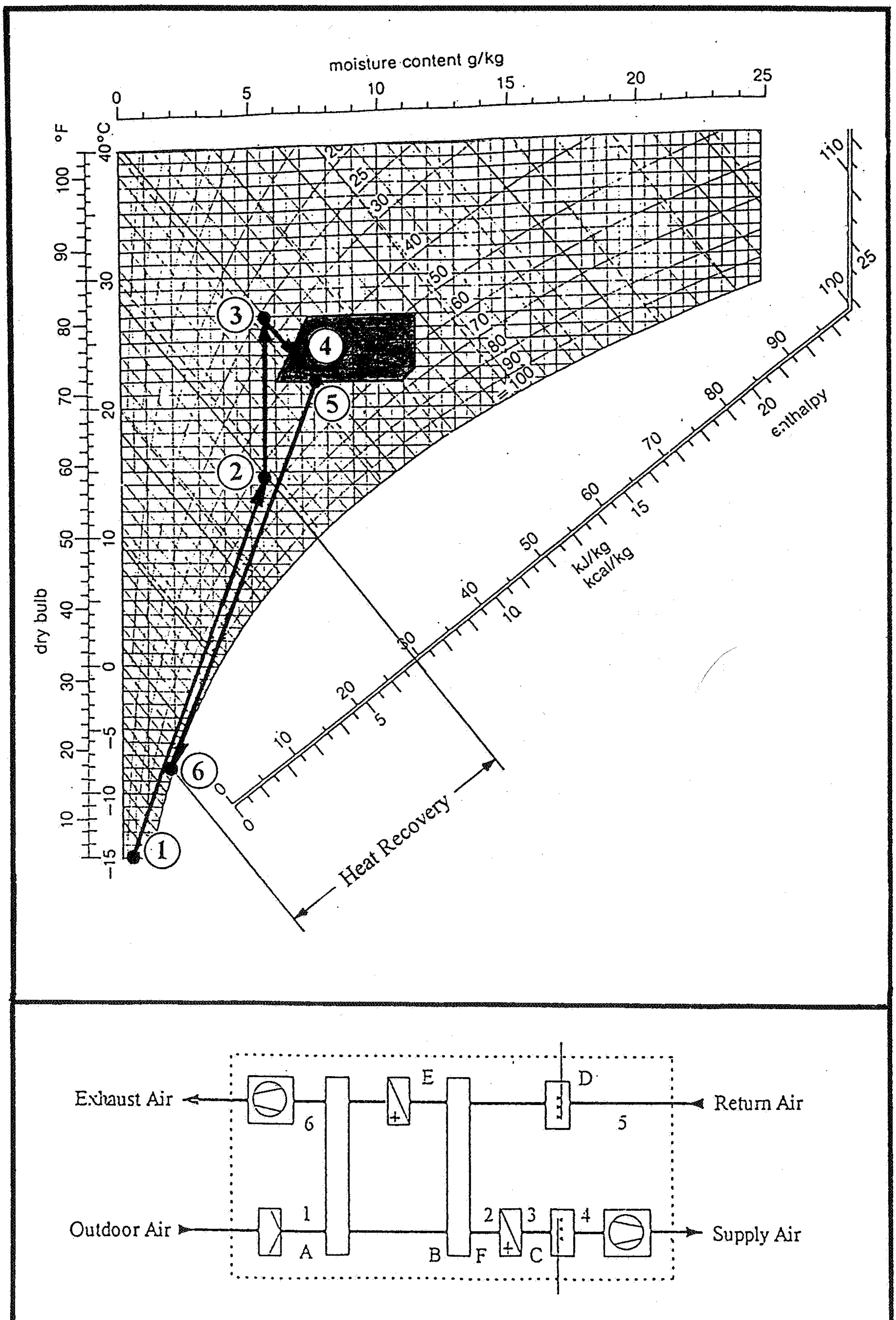

Fig. 3: Desiccant Cooling Winter Operation

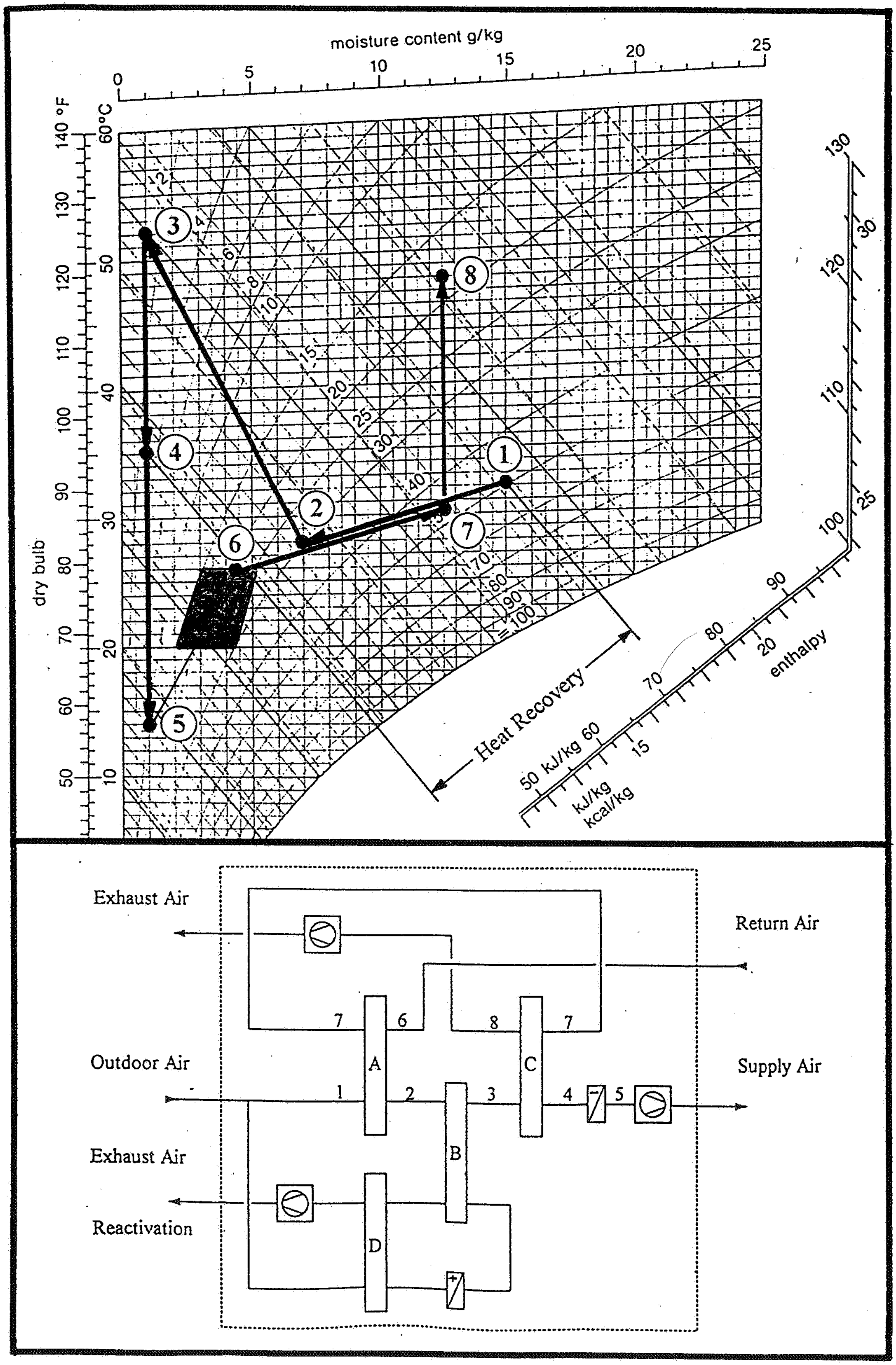

Fig. 4: Industrial Dehumidification Summer Operation

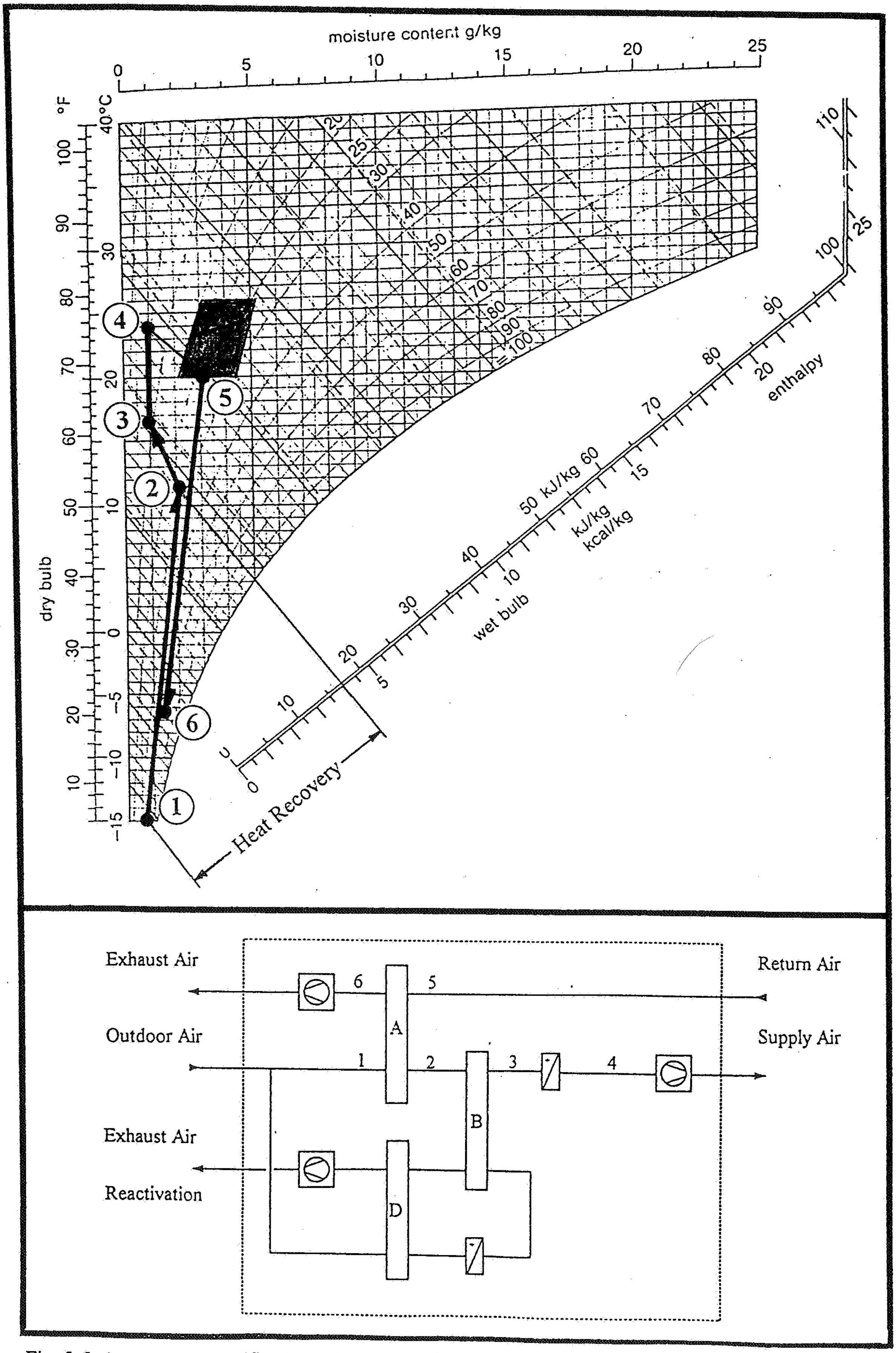

Fig. 5: Industrial Dehumidification Winter Operation

Implementing the Results of Ventilation Research
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A New Ventilation Strategy for Humidity Control in Dwellings - A Demonstration Project

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A new Ventilation Strategy for Humidity Control in Dwellings - A Demonstration Project

Synopsis

A demand controlled ventilation system with humidity as the control parameter was tested in an experimental demonstration project in 16 apartments. In the same housing complex 16 identical apartments with a constant exhaust airflow rate were included in the test as a reference group.

The purpose of the study was to investigate whether satisfactory physical health conditions could be reached in the humidity-controlled in apartments, while at the same time reducing the use of energy for heating.

In the apartments with humidity-controlled ventilation the air supply was regulated so that the humidity in the indoor air in all rooms was kept at a level which was below the limit for the growth of house dust mites and just sufficient to prevent condensation on all indoor surfaces of the building. The airflow could increase up to a maximum to the requirements as stated in the Nordic Committee on Building Regulations and in the Danish Building Code. The airflow in the reference apartments was at a constant level, according to the requirements.

In the humidity-controlled apartments the total outdoor air change, the mechanical exhaust airflow rate, and the energy consumption for heating were significantly reduced compared to the reference apartments, at mean temperatures per day below 9°C and at the same time the humidity criteria were met.

In addition to the technical study, a resident indoor climate satisfaction study was carried out in the housing complex. The main purpose of this study was to gain knowledge of the residents' appraisals of the indoor climate.

1. Introduction

A high concentration of humidity in the indoor air of apartments provides suitable growing conditions for house dust mites and mould. Both house dust mites and mould cause allergies in human beings. In addition, rot and mould seriously damage the building construction. The criteria in building codes and ventilation standards for mechanical ventilation of apartments have the primary aim of ensuring the exhaust of humidity from the indoor air, and thus avoid the disadvantages of excessive humidity concentrations. The use of mechanical ventilation guarantees a minimum level of ventilation in dwellings. However, if the ventilation has a constant air volume, regardless of the ventilation needed, the outdoor air exchange and the use of energy for heating may in certain periods be unnecessarily high. Therefore, it is obvious to attempt to regulate the ventilation rate by humidity and only ventilate *where* and *when* humidity occurs. This condition can be established through a ventilation system that has individual control of air volumes for each room of a dwelling. For instance, humidity can be kept at a level which is just sufficient to prevent condensation on indoor surfaces of the building and below 45 % of relative humidity which is the limit necessary to reduce the growth of house dust mites [1,2].

2. Purpose

The purpose of the present study was to investigate if satisfactory physical health conditions could be obtained in apartments with humidity-controlled ventilation while at the same time reducing the use of energy for heating.

In this context, satisfactory health conditions means humidity conditions that can reduce the occurrence of house dust mites and fungus spores and prevent condensation on indoor surfaces of the building.

3. Method

3.1. Material

3.1.1 The apartments and the ventilation system

Two groups of 16 identical apartments were provided with mechanical ventilation systems. In group one, a humidity-controlled ventilation system was installed where the ventilation in each room of the apartments was regulated by the humidity level of the indoor air. In group two (the 16 reference apartments), the ventilation system comprised of a constant mechanical exhaust with an airflow rate according to the building code [2,3]. Each group of apartments was ventilated by two exhaust fans. In the investigation period some occupants moved from their apartment and others would not participate in the investigation. Therefore only 13 apartments having each type of ventilation system were used in the investigation. While originally identical in pairs of two, the size of the remaining 26 test apartments now varied slightly. The average net floor area in the humidity-controlled apartments and in the reference apartments was respectively 49 m^2 and 55 m^2, and the average number of inhabitants per apartment was respectively 1.7 inhabitants and 1.4 inhabitants.

Connected to the living room of the apartments there was an enclosed veranda with an external wall constructed with two layers of glass. Glass doors separated the veranda from the living room.

In both types of apartments, the indoor air was exhausted from the kitchen and the bathroom and the outdoor air was supplied through a fresh air valve in each room. In the kitchen there was a manually controlled cooker hood which was connected to the ventilation system. In the humidity-controlled apartments, each fresh outdoor air valve was controlled by a humidity sensor, except for some of the apartments where the fresh air valve in the living room was manually controlled as in the rooms in the reference group. Here, the manual fresh air valve was designed as a slide valve in the window frame. A ground-plan of an apartment with the positions of the fresh air valves and corresponding humidity sensors and the mechanical exhausts is shown in figure 1.

The humidity sensors were constructed as capillary hygrostats.

3.1.2 Principle of regulation

At outdoor air temperatures greater than 1°C, the relative humidity in all bedrooms was regulated up to a maximum of 40 % and in all other rooms up to a maximum of 45 %. When a humidity sensor registered higher relative humidity than the set point, the on-off regulated fresh air valve in the same room was opened, and again closed when the relative humidity had decreased to a value of approximately 5 % relative humidity below the set point. In the kitchen and the bathroom the exhaust airflow rate was regulated

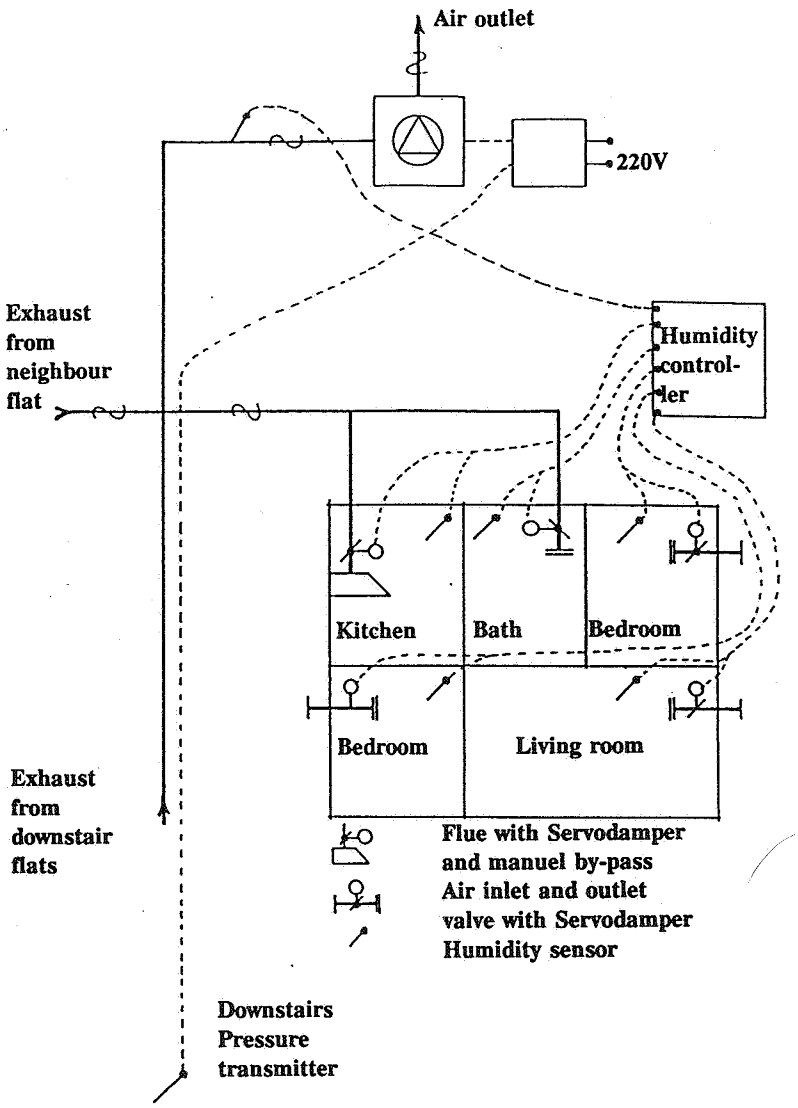

Fig. 1. The position of the fresh air valves and the humidity sensors in an apartment.

automatically by a motor driven exhaust air valve. The airflow rate of the fan was automatically controlled relative to the ventilation need by the difference between the pressure in the apartments and the pressure in the fan inlet. The minimum exhausted airflow rates for apartments with respectively small (< 7 m^2) and big (> 7 m^2) kitchens were respectively 14 l/s and 20 l/s. When dehumidification was needed both in the kitchen and the bathroom, the airflow rates could rise to the requirements in the building code [2,3], respectively 30 l/s and 35 l/s. The opening of fresh air valves in the other rooms did not change the total exhausted air volume but secured the supply of outdoor air to the these rooms.

At outdoor air temperatures less than 1°C, a criterion to prevent condensation on window panes was added to the control criteria of humidity limits of 40% and 45% relative humidity. The limit of occurrence of condensation was determined by the relation between the relative humidity in the indoor air (RH_i) and the outdoor temperature (t_u) by the model:

$$RH_i < EXP(0.0360 * t_u + 3.85) - 3 \quad (\%)$$

The exponential term of the model expresses the limit for the indoor relative humidity when condensation occurs, based on a calculation of the heat transmission of a 1 meter high thermo window with two layers of glass. The heat radiation and the convection have been calculated individually and apply to the average glass surface. To simplify the regulation system, the relative humidity was registered in a main exhaust duct that covered eight apartments. The value, -3, in the model should partly compensate for this simplification. When the relative humidity in the main duct was higher than the one calculated by the model, all fresh air valves opened and the exhausted flow rate from all eight apartments was increased to the requirement in the building code. If the indoor relative humidity then decreased to 5 % below the value calculated by the model, the hygrostats regulated the fan's performance and the air valve positions to the level adjusted at the hygrostats.

The limits for the relative humidity that regulate the valve and the fan are shown in figure 2.

When forced ventilation was needed in the kitchen, it was possible to increase the airflow rate manually up to approximately 50 l/s by means of opening a valve in the cooker hood. In the bathroom, the exhaust air flow rate could be increased (by turning on the light switch) to 15 l/s, which is the requirement in the building code [2,3].

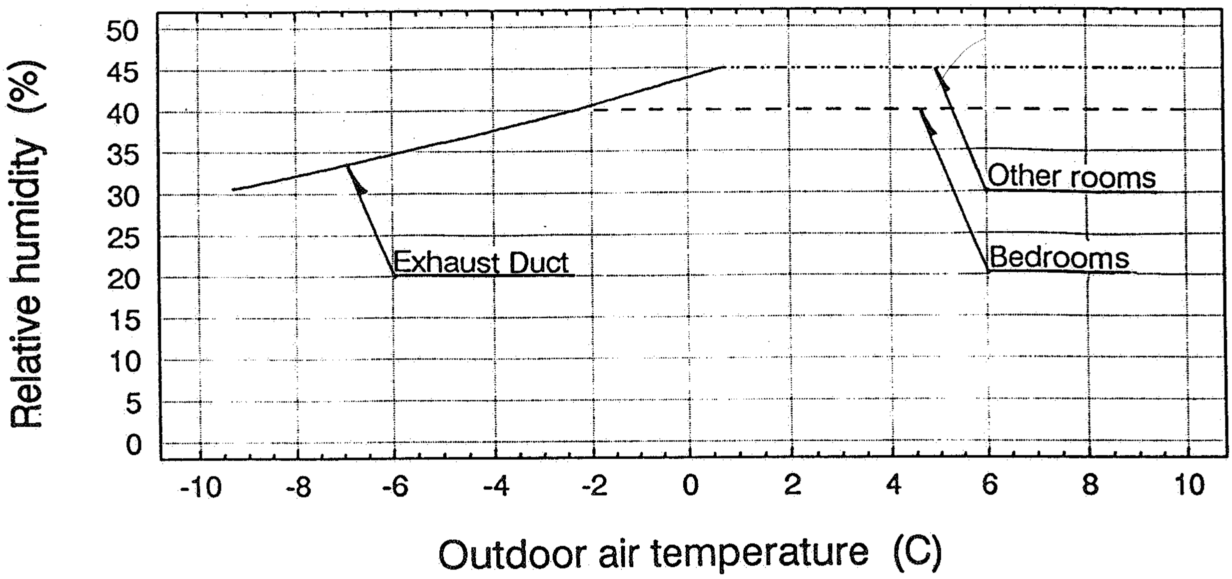

Fig. 2. The limits for the relative humidity that control the regulation of the valves and the fan. The upper value at the dead band of the hygrostats at 5 % relative humidity was adjusted at the drawn limits at 40 % and 45 %. The registered relative humidity in the exhaust duct is the mean value from eight apartments.

3.2 Investigation

3.2.1 Control and registrations

Before starting the registrations, all hygrostats and humidity sensors were calibrated and all airflow rates and regulation functions were checked. The registrations were then performed during three periods.

During two weeks at a mean outdoor air temperature per day varying from -5°C to 3°C and at an average of -0.1°C and a mean relative humidity at 86 % the registrations consisted of:

- The total mean air change rate for the apartments and the air change rate for the bedroom in all apartments. Use of passive tracer gas technique.
- The mean relative humidity and the mean air temperature in the living room and the bedroom in all apartments. Use of thermohygrograph.
- A questionnaire investigation concerning the indoor activities which could influence the amount of indoor air humidity. The answers were focused on the two week registration period.

During three months at mean outdoor temperatures per day varying from -3°C to 15°C and an average of 7°C the mean total exhaust flow rate per day was registered for each apartment with humidity-controlled ventilation. Use of hot wire anemometer in each main exhaust duct.

One year after the investigation was started the hygrostat readings were checked. In the first half year the consumption of energy for heating was registered.

3.2.2 User satisfaction study

A study of users' satisfaction with the indoor climate of the apartments was included in the research design for the evaluation of the humidity-regulated ventilation system. The purpose of the user study was to investigate interaction effects between the perceptions and behaviour of users (i.e the residents in each apartment), the characteristics of the ventilation systems and the physical indoor climate characteristics.

The hypothesis was that residents in apartments with humidity-regulated ventilation would experience no significant defecit in satisfaction with the indoor climate compared with residents in apartments with constant ventilation at the normal rate.

The user study included a series of pilot interviews with 5 households and the caretaker, and a postal questionnaire survey of all residents.

The "satisfaction model" for the interviews and survey can be described as a series of inter-related but conceptually discrete "facets", some comprised of objective conditions, others subjective; each facet being made up of different "elements"[4, 5]. Using a so-called "mapping sentence" (- being a shorthand, textual presentation of a research problem), the following definition expreses how the level of user satisfaction can be conceptualized as a function of relations between 6 main facets:

Mapping sentence: *Residents' LEVEL OF SATISFACTION with indoor climate (expressed as high/low) is a function of:*
Facet (A) - her own situation ("biographical" characteristics),
(B) - her experience of physical (climate) impacts
(C) - her home-use and behaviour
(D) - her health and welfare, (and possible irritation/ill-health symptoms)
and (E) - actual physical conditions in the apartment.

Pilot Interviews were tape-recorded and analysed in relation to each of the 5 facets (above). The *questionnaire survey* was used to record 1. user behaviour which could affect the humidity of the apartment, and 2. users' observation of condensation/humidity in the apartment.

4. Results

4.1 Physical condition

The mean exhaust airflow rate per day from one apartment is plotted as a function of the mean outdoor temperature in figure 4. The mean airflow rate is estimated by three linear regression equations.

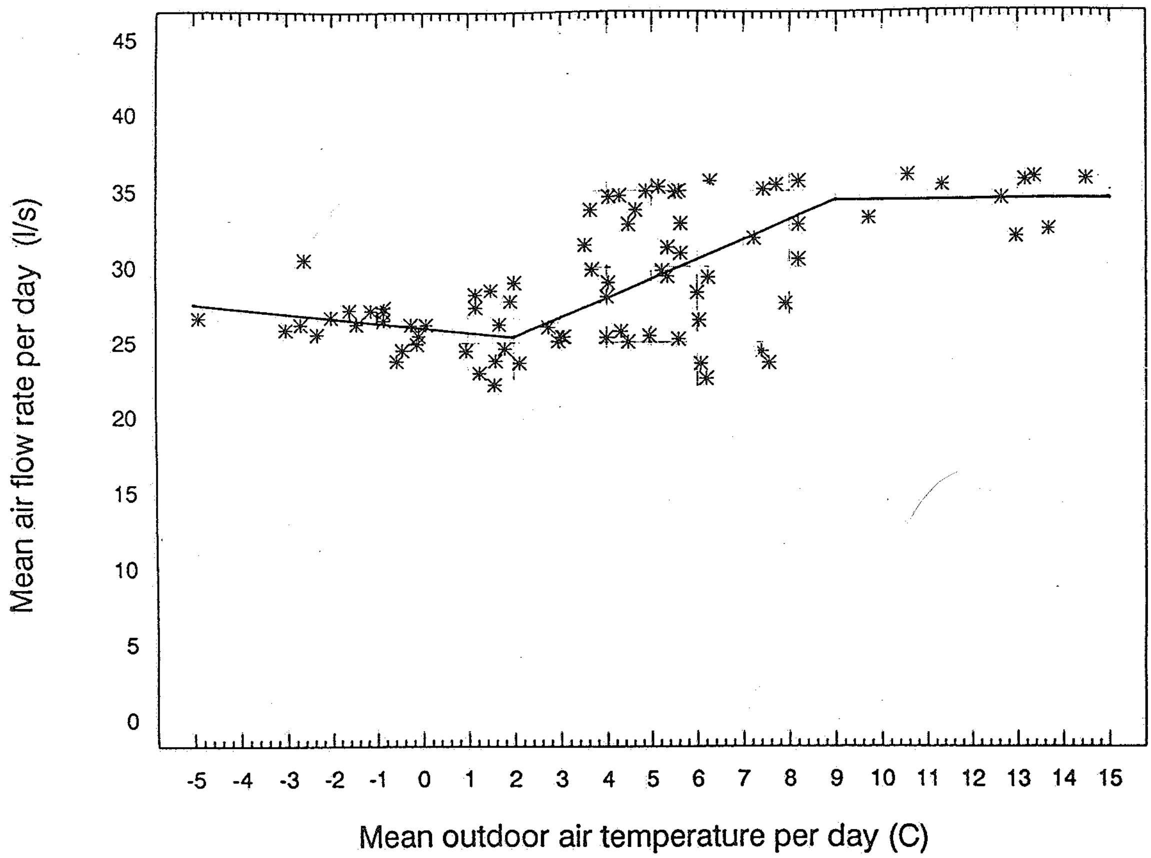

Fig. 4. Plot of the mean airflow rate per day versus the mean outdoor temperature per day.

The percentage reduction of the mean airflow rate - compared with the flow rate measured in the reference apartments - is calculated as a function of the mean air temperature per day for each apartment with humidity-controlled ventilation. The average percentage reduction for all apartments is shown in figure 5.

In the living room and the bath room for both type of apartments all registrations of the relative humidity was below 45 %. In the bedroom in the humidity-controlled ventilated apartment and in the reference apartment the variation of the relative humidity were respectively 32-43 % and 27-48 %.

On the basis of the measured room air temperatures and relative humidity the vapour pressure and the vapour content of the indoor air can be calculated. The difference between the vapour content of the indoor air in the living room and in the bedroom and the vapour content of the outdoor air is shown in figure 6.

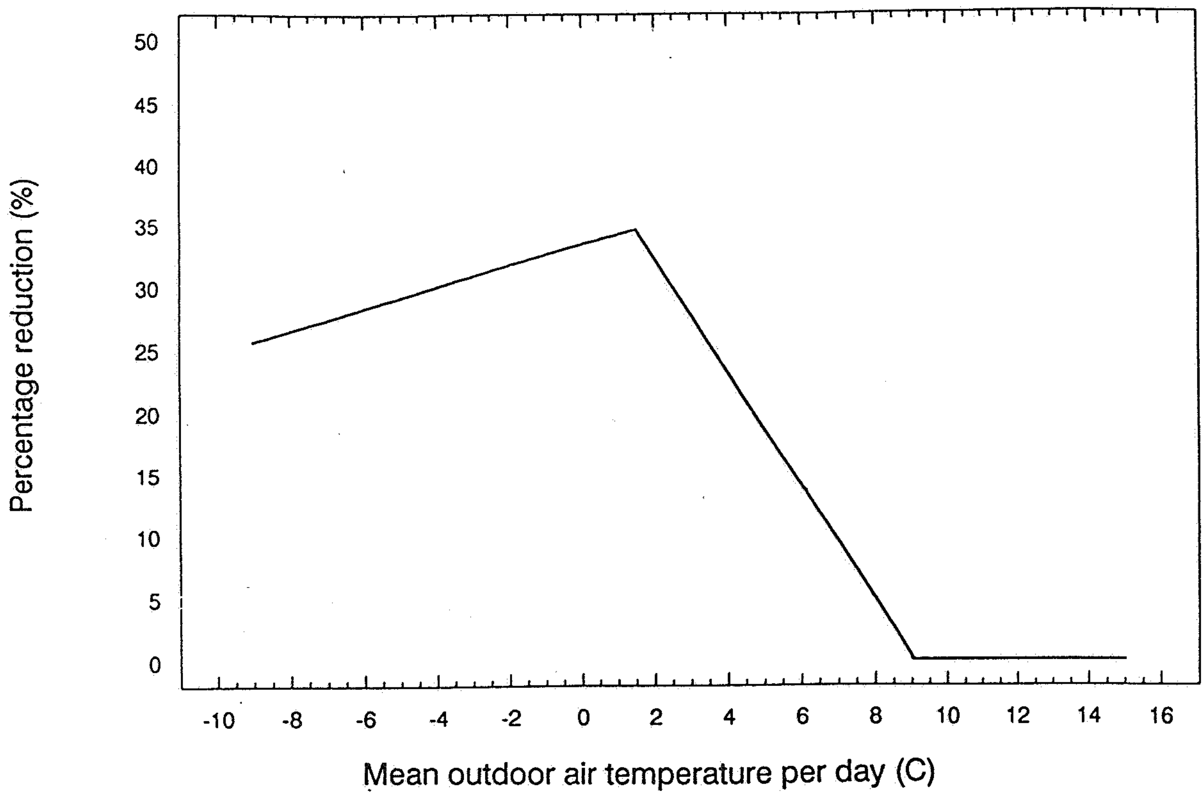

Fig. 5. The percentage average reduction in mean airflow rate.

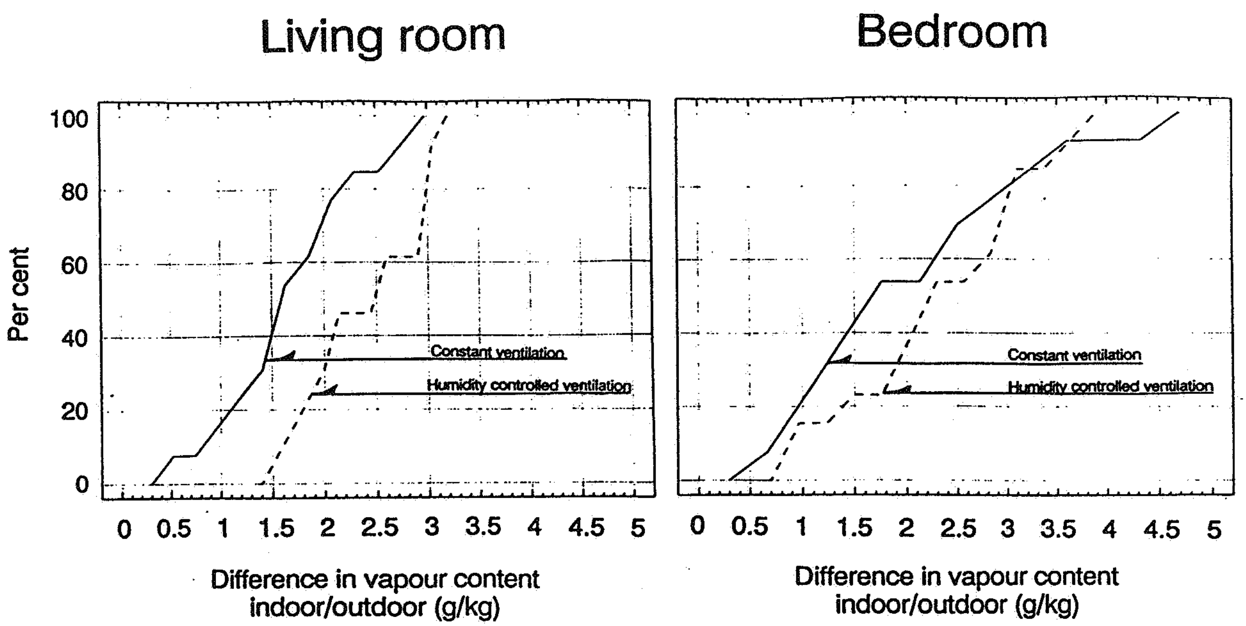

Fig. 6. Cummulative, relative fequency of the difference in vapour content in the indoor air of the living room and the bed room and in the outdoor air illustrated.

A statistical presentation of data for the first two weeks' registration is shown in table 1.

Table 1. Results of the registrations.

Parameter	Part of apartment	Humidity-controlled ventilation Mean ± [2]Std.	Constant ventilation Mean ± [2]Std.	[1)]t-test Level
Temperature	Living room	20.2 ± 2.2°C	21.0 ± 2.2 °C	
	Bedroom	20.6 ± 1.5°C	21.2 ± 1.4 °C	
Difference in vapour content indoor/outdoor	Living room	2.37 ± 0.58 g H_2O/kg air	1.66 ± 0.69 g H_2O/kg air	p=0.004
	Bedroom	2.30 ± 0.97 g H_2O/kg air	2.07 ± 1.16 g H_2O/kg air	p=0.29
	Bathroom	2.09 ± 1.54 g H_2O/kg air	1.54 ± 0.61 g H_2O/kg air	p=0.020
Relative Humidity	Living room	37.5 ± 3.1 %	32.2 ± 1.7 %	p=0.0001
	Bedroom	36.6 ± 3.7 %	35.2 ± 6.2 %	p=0.25
	Bathroom	33.0 ± 4.6 %	30.4 ± 4.1 %	p=0.06
Moisture generation rate	Whole apartment	6.25 ± 2.0 kg H_2O/day	6.69 ± 3.5 kg H_2O/day	p=0.35
	Bedroom	1.38 ± 0.46 kg H_2O/day	2.07 ± 1.4 kg H_2O/day	p=0.1
Total outdoor air change	Whole apartment	0.77 ± 0.22 h^{-1}	1.08 ± 0.34 h^{1-}	p=0.002
	Bedroom	0.63 ± 0.22 h^{-1}	1.24 ± 0.86 h^{1-}	p=0.002
Total outdoor air supply	Whole apartment	25.5 ± 6.1 l/s	39.0 ± 10 l/s	p=0.002
	Bedroom	5.64 ± 1.9 l/s	10.6 ± 7.5 l/s	p=0.003

[1] *Testing of the difference between the mean values. Where the test level p is higher than 0.05 the two mean values can be considered as equal. In all tests the standard deviations are pooled.*
[2] *Standard deviation.*

4.2 User satisfaction study

Resident characteristics

Residents who were interviewed included the following persons :

- a retired married couple, wife asthmatic (apartment with constant ventilation)
- a young, female single student, smoker (humidity-controlled ventilation)
- an elderly wheelchair user, heavy smoker and his family (humidity-controlled ventilation)
- a young mother and 2 small children (humidity-controlled ventilation)
- a young couple, no children (humidity-controlled ventilation).

Knowledge of system: Neither the residents nor the caretaker were informed in detail about the measurement principles of the humidity-controlled ventilation system. This made it impossible for the caretaker to explain to the residents precisely when and why the humidity-regulated ventilation system came into effect. Some residents felt that during periods of peak-ventilation need (food preparation in evenings), the flow was too low, as smells could come from other flats. Several residents would like to have had the possiblity to shut outside air vents completely in windy periods.

General satisfaction with apartments: Was not recorded in detail in the pilot interviews but was indicated by residents as largely positive. The apartments were regarded as very attractive and functional. West-facing apartments receive plenty of daylight. It was observed that *outdoor climate* effects are particularly strong due to the exposed sight of the housing, which faces the prevailing south-westerly wind. Strong winds, driving rain, snow, strong sunshine and heat have led to rain-water seapage around window-frames and concrete elements in some flats.

General perception of indoor climate: An interview with the housing caretaker, who received queries and complaints about the ventilation systems revealed that particular problems were related to the noise nuisance and draughts from air intake valves (vents) in the walls of apartments with humidity-regulated ventilation.

Noise and draughts: Positioning of air intake vents (in bedrooms and kitchens) was regarded as particularly irritating on cold or windy days. Due to the limited possibilities for arranging furniture, residents found that cold air fell directly onto the bed and gave draughts when sitting at the kitchen table. Noise from the ventilation system (experienced as "whistling" at the air outlet points in bathrooms and kitchens) was irritating for some residents. After the experimental period and measurements were completed, some residents have had covers fitted to the outlet valves to reduce noise, and one couple has since blocked the valve completely.

Health: One resident, who previously had an asthma condition, has used much less medicine than she did in the former dwelling (- a traditional, brick-built house with a basement). She regards the indoor climate as being conducive to her improved health. She lives in an apartment with constant mechanical ventilation.

5. Discussion

In figure 4, the three linear regression equations are divided by the mean air temperature at 2 °C and at 9°C. At a mean air temperature lower than 2°C, the criterion of avoiding condensation significantly affects the airflow rate which is increased by lower outdoor air temperatures. Since the need for dehumidification to prevent condensation grows significantly at low outdoor air temperatures, the airflow rate will also increase as shown in the figure. At mean air temperatures between 2°C and 9°C, the airflow rate increases up to around 35 l/s. At a mean air temperature per day higher than 9°C, the airflow is constant, i.e. the outdoor air does not have any dehumidification potential with respect to the indoor air.

The three regression equations are estimated for all apartments. The abscissa value for the points of discontinuity of the estimations and the high scatter of the points of the airflow rate do not differ essentially for the other apartments from what is shown in figure 4. However, the differences between the levels of the airflow rate of some apartments are more than 7 l/s at 2°C.

The high scatter of points of the mean airflow rate in figure 4 shows that the ventilation needs differ greatly in the individual apartments. The differences are due to a high variation in the moisture generation rate and in the dehumidification potential of the outdoor air in combination with irregularity of airing.

Figure 5 shows that the achieved maximum percentage reduction of the mean airflow rate as average for all apartments was around 35 % (12 l/s) at 1.5°C and the outdoor air had no dehumidification potential at a mean air temperature per day higher than 9°C. As a consequence of the lower airflow rate, a saving in energy for heating is obtained. From a mean outdoor air temperature per day between -2°C, which was the lowest temperature during the energy registration, and 6°C, the registrations of energy for heating showed a linear increase in the difference of energy consumption for the two types of apartments. At a decrease of one degree in the outdoor air, the difference in the needed heating effect was increased by 0.75 W/(m^2 net floor area), i.e. the save in energy for heating for an apartment at a net floor area at e.g. 70 m^2 and a mean outdoor temperature at 0°C is approximately 7.6 kWh per day.

The criterion for condensation for the humidity-controlled ventilation system is in operation at mean air temperatures less than 1°C. In the two weeks' registration, during which it was tested if the regulation system could prevent condensation, 70 percent of the outdoor air temperatures registered each hour was below 1°C. Also, knowing from figure 5 that the minimum mean ventilation need appeared at 1.5°C, the outdoor temperature in the test period was estimated to be low enough to perform an assessment of the criterion of condensation. At a difference in the vapour content of the indoor and outdoor air of 2.5-3.0 g H_2O/ kg air, condensation on window panes normally will not occur. At a difference of 4.0-5.0 g H_2O/ kg air, condensation problems in houses with windows with 2 layers of glass may occur when the indoor air temperature is decreased and curtains cover the windows. Figure 6 shows that in 15-20 % of the registrations, the difference in vapour content of the indoor and outdoor air in the apartments with humidity-controlled ventilation system exceeds 3.0 g H_2O/ kg air, but no values exceed 3.9 g H_2O/ kg. The scatter of the registrations for both rooms in the reference apartments in the figure is much higher, which may be due to the fact that the valve in the reference apartment was not activated as often as the automatical humidity-controlled valve. From the interviews of the occupants it appeared that the valves were almost always open or closed. Therefore, the difference in the vapour content in the indoor and the outdoor air also in some cases can be high. In one bedroom the difference was more than 4.5 g H_2O/ kg air.

The test of the difference between the mean values in table 1 shows that there was no significant difference in the moisture generation rate in the two types of apartments, both when calculated for the whole apartment ($p=0.35$) and for the bedroom alone ($p=0.10$).

In the bedroom, the relative humidity and the differences in vapour content in the indoor and outdoor air did not differ. However, the difference in the outdoor air supply rate to the room was large ($p=0.002$). These relationships strongly indicate that the removal of vapour content from the bedroom happened more effectively by the humidity-controlled ventilation system, i.e. when the vapour content increases, the ventilation airflow rate increases. Calculating the removal factor as gram removed vapour per liter supplied outdoor air per second for each apartment, the mean removal factor for the apartments with humidity-controlled ventilation was 1.4 times higher than that for the reference apartments. For the living room this factor was the same, but here the relative humidity and the vapour content in the indoor air differs significantly. The regulation of humidity in the living room is less effective, presumably due to two main reasons:

1) In the half of the apartments the living rooms had only a manually controlled fresh air valve by which a part of the outdoor air needed for dehumidification was not led through the room if the valve was closed.
2) The hygrostat in other living rooms were placed at the door leading to the corridor and the fresh air valve in the opposite wall close to the kitchen where the air was exhausted. In that way, the hygrostat often could be surrounded by air coming from the adjacent room into the corridor.

Selecting the few apartments left for which the above reasons did not apply, the average removal factor for the living room was more than three times higher for the apartments with humidity-controlled ventilation than for the reference apartments. The analysis indicates that simple fundamental physical changes would be able to improve the regulation system and probably further reduce the energy consumption for heating.

In the questionnaire, the occupants were asked if they had observed condensation on the window panes during the two week period. A few more occupants in the apartments with humidity-controlled ventilation than in the reference apartments had observed condensation. It sometimes occurred in the living room, when the occupants dried clothes in the room or took a shower in the bathroom. In the apartments with humidity-controlled ventilation, all the condensation was observed in the type of apartments with less effective humidity regulation due to a manually controlled fresh air valve, as described in reason 1) above.

When the criterion to prevent condensation controlled the ventilation, one humidity sensor in the main exhaust duct functioned as reference sensor for eight apartments. If one sensor was used in the main exhaust duct from each apartment, the regulation probably would be substantially improved.

The check of the hygrostat readings one year after they had been calibrated showed that the hygrostats should be recalibrated at least once a year, e.g. just at the time when the mean outdoor temperature per day is decreasing below 9°C.

The mean total outdoor air supply in the two weeks' registration period was 28 % less than the requirement in the building codes [2,3]. However, the vapour content in the indoor air was on a level which normally meets the requirements for humidity conditions in the indoor air of a dwelling. The humidity-controlled ventilation system did not in the controlling take into account the removing of other pollutants, such as, for example, radon or gases emanating from furniture and building materials. However, only one of the registered total outdoor air change was below the limit at 0.5 h^{-1} as required in the Danish Building Code [3]. It was 0.46 h^{-1}.

Results of the user satisfaction pilot interviews indicate that there are only few perceived problems concerning humidity in the apartments with humidity-controlled ventilation. However, there may be greater dissatisfaction due to noise and draughts in these apartments due to the construction and positioning of the air intake vents. The questionnaire investigation in the first two weeks' registration period showed that the occupants in 72 % of the humidity-controlled apartments and 45 % in the reference apartment complained about draught. Further investigations based on a questionnaire survey will reveal the extent of user satisfaction with the two ventilation systems.

Conclusion

The functioning of a demand controlled ventilation system with humidity as control parameter has been demonstrated in 16 apartments, of which the humidity condition in the indoor air was compared with 16 identical reference apartments having a constant mechanical exhaust airflow rate according to the Nordic Building Code.

At lower mean outdoor temperatures per day than 9°C, the airflow rate was reduced in the apartments with humidity-controlled ventilation. The minimum airflow rate was obtained at a mean temperature at 1.5°C, where the reduction was 35 %. This reduced the effect for heating 0.75 W/(m^2 net floor area) for each degree the mean outdoor temperature is lower than 6°C.

The mean relative humidity did not exceed 43% which was below the controlling criterion at 45%. The registration of the difference of the vapour content in the indoor and outdoor air did not indicate the occurrence of condensation on the window glass. However, a few more occupants in apartments with humidity-controlled ventilation had observed condensation in the living room when they dried clothes in the room or took a shower. The humidity-control in these apartments was not at its optimum, but simple fundamental physical changes here would be able to improve the regulation system and probably further reduce the energy consumption for heating.

In rooms and apartments where the regulation of the humidity-controlled ventilation was at its optimum, the indoor level of humidity condition in the apartments with humidity-controlled ventilation did not differ from the level in the reference apartments, but the total outdoor air change was less. Hence, the removal of vapour content from the indoor air was more effective with the humidity-controlled ventilation system.

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Trickle Ventilators: Effective Natural Background Ventilation for Offices

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SYNOPSIS

Approved Document (AD Part F1) of the Building Regulations [1] for England and Wales identifies trickle ventilators as an option for providing natural background ventilation in commercial buildings. This paper reports the results of a field measurement study carried out at BRE during the winter of 1994/95 to assess the effectiveness of trickle ventilators. Two occupied office rooms were equipped with trickle ventilators and measurements were carried out for a fortnight period in each office, with the ventilators closed during the first week followed by a week with them open. Internal and external parameters of significance were monitored including level of metabolic CO_2 within the rooms, draught speeds near the ventilators and occupancy.

The study has confirmed results of a previous modelling study [2] which showed that trickle ventilators of sizing given in the AD Part F1 provide adequate background ventilation of about 5l/s per person in typical UK offices. Furthermore, it has also confirmed the results of an earlier laboratory study [3] where it was shown that for average and high occupancy (8-10m^2 per person) trickle ventilators with similar sizing were capable of providing the fresh air required to maintain CO_2 levels at or below 1000ppm (which equated to about 5l/s per person) for average UK external weather conditions.

In conclusion, the present field study *in occupied offices* has shown that trickle ventilators of suitable design and openable area of 400mm^2 per m^2 (of floor area), as advocated in AD Part F1, are effective in providing adequate background fresh air in UK offices during the heating season without creating occupant discomfort.

INTRODUCTION

The new Approved Document Part F1 (Ventilation) of the Building Regulations [1] which have come into force in July 1995 requires that all offices in new build commercial buildings are provided with background ventilation. Guidance is that this is provided via openings having a total area of at least 400mm^2 per m^2 of floor area. Trickle ventilators are a common means of providing this degree of ventilation and the revised Regulations are likely to make the use of these devices more widespread.

Trickle ventilators consist of slot ventilators located either in the window frame or incorporated into the window pane. In the latter, the ventilator has a flange of the same thickness as the glass and lies in the same plane forming a structural part of the glazing. Trickle ventilators are also frequently provided with a damper mechanism which allows user control of the ventilation opening and usually comprises of an adjustable plate mounted in front of the slot. This configuration deflects incoming air vertically upwards or downwards from the ventilator.

This paper describes the third stage of a research project to assess the effectiveness of trickle ventilators to provide adequate and controlled background ventilation in office buildings during the heating season.

Previous Work

In 1992, a study [2] using a multi-zone air flow prediction computer program, BREEZE [4], was carried out to assess whether the inclusion of permanent and controllable background trickle ventilators could provide adequate background ventilation in commercial buildings. The study recommended that 4000 mm^2 open-area ventilators be used in rooms with floor areas less than 10 m^2, or 400 mm^2 per m^2 (of floor area) for those which were larger.

Following on from this, a full scale laboratory study [3] was carried out in the heating season of 1993/94. This was done in two deep-plan office rooms with a depth of 7.5m and an area of approximately 26 m^2. One room was used as a control while the other was fitted with a trickle ventilator, (with opening area of 10 500 mm^2, corresponding to 400 mm^2 per m^2 of floor area) fixed to its main south facing window. Varying levels of occupancy were simulated in both rooms by constant CO_2 injection (to simulate metabolic rate) and heat sources (lamps). Internal measurements included the monitoring of CO_2 levels, air velocity and temperature as well as the ventilation rate using the SF_6 tracer gas decay technique. External measurements included wind velocity, wind direction and temperature.

The study concluded the following:

(a) Trickle ventilators with a minimum openable area of 400 mm^2 per m^2 (floor area) can provide adequate fresh air during winter in a typical office room (located in a suburban site) with maximum occupancy density of 8 m^2 floor area per person.

(b) Cold draughts did not appear to be a problem either at desk or head levels at distances up to 2 m from the ventilators. Measured velocities were well below the accepted threshold of 0.3 m/s for discomfort. However, velocities measured at ankle height were found to be somewhat higher averaging at about 0.5 m/s. While this could cause discomfort, good ventilator design could correct it.

(c) Long term CO_2 monitoring could be used as a marker of indoor air quality in naturally ventilated buildings in the same way it is used for mechanical ventilation systems. No significant variation in CO_2 concentration levels was found with varying heights thus indicating good internal mixing of fresh air.

The above conclusions have indicated that trickle ventilators can be effective in offices. To obtain confirmation of this in the field, measurements were made in two occupied office rooms equipped with trickle ventilators. Monitoring was made during the winter of 1994/95.

EXPERIMENTAL ROOMS AND EQUIPMENT USED

Office A is cellular, with single-sided ventilation and situated on the first floor of a four storey building with a floor area of 12.5 m^2 (3.5 m x 3.6 m). Using the criteria of 400mm^2 per m^2 of floor area, a trickle ventilator with a 5000 mm^2 openable area was fitted to the west facing single glazed window at a height of 2.4 m above floor level. The spinal corridor running outside of this office is 25 m long and 1.8 m wide, is fitted with fire doors at either end and none of the office doors are opposite one another . Ventilation is therefore single-sided and through the windows in each of the offices.

Office B is situated on the ground floor of an one storey building with a floor area of 16 m^2

(4.2 m x 3.8 m). A trickle ventilator with a 7000 mm^2 openable area was fitted using the same criteria as in office A. The ventilator was also placed 2.4 m above floor level and fitted to an east facing single glazed window. The corridor running outside of this room is 12 m long and 1.8 m wide, and, as before, fitted with fire doors at either end. However, unlike that for office A, all of the office doors within this corridor are opposite to one another. During the experiment, occupant in office B kept the corridor door to the corridor open most of the time, while the occupant opposite did the same. Therefore the ventilation flow in office B was generally through cross ventilation.

The following measuring equipment were used in each room:

1) Two ultrasonic anemometers to assess possibility of cold draughts leading to occupant discomfort. One of these was set at desk height (~ 1.5 m above floor level), and the other at ankle height (approximately ~0.25 m above floor level). The anemometer loggers were programmed to measure air velocity in the three x-y-z directions, at intervals of 15 min.
2) Two electrochemical CO_2 sensors to measure the effectiveness of the trickle ventilator to provide the required fresh air. One was placed within the room at approximately 1.75 m above floor level. The second was placed in the corridor and attached to the door frame by a bracket at 2 m above floor level. Data was collected at 15 min intervals.
3) A humidity/temperature sensor: The temperature was measured to assess any discomfort associated with cold air coming through the ventilator. The humidity reading was used as an indicator for calibrating the CO_2 sensors since low humidity were found to affect the range parameters of these instruments.

All equipment used within anyone room were mounted on a stand positioned in each office approximately 1 metre from the window. In addition, a micro-switch was fitted to the door frame and measured when and for how long the door was opened or shut. A questionnaire was provided to the office occupant for completion on a daily basis to note occupancy levels, whether the window was open or shut and whether internal window blinds were either up or down. Outside weather information of wind speed, direction and air temperature were also monitored.

Monitoring was carried out for two weeks in each office; with the ventilator closed during the first week and opened during the second week. A week prior to testing, the equipment stand was placed 2 m from the window (windows shut, ventilators open) in the offices to compare air velocity values. During monitoring the stands were moved to 1 metre from the windows.

RESULTS AND DISCUSSION

Internal air velocity

Air velocity was measured at desk and ankle height at 1 m and 2 m away from the window with the trickle ventilators. Previous laboratory measurements have shown that cold draughts at ankle height could cause discomfort. For this reason, air coming through the ventilators was directed upwards by positioning deflector vanes. All the measurements then made showed

(Figure 1) that the air velocities within the rooms were significantly lower than 0.3m/s, the value usually considered as the upper limit above which discomfort is likely to be felt by draughts during the heating season [5]. A further confirmation was that the occupants did not report any complains concerning cold draughts.

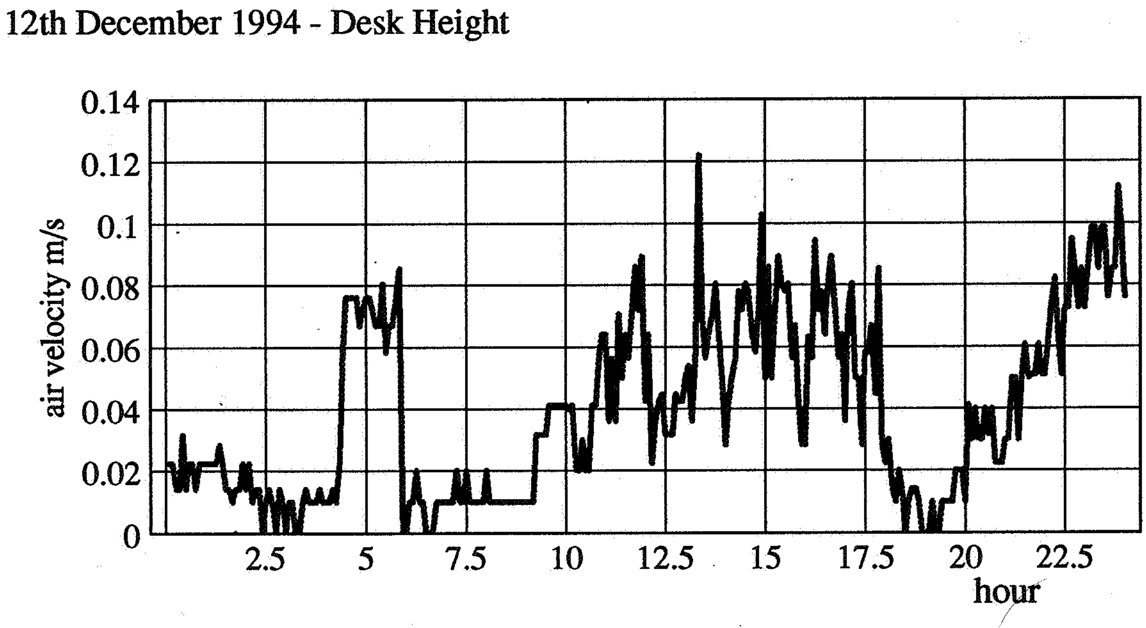

Figure 1A: Internal air velocity at desk height with the trickle ventilator open

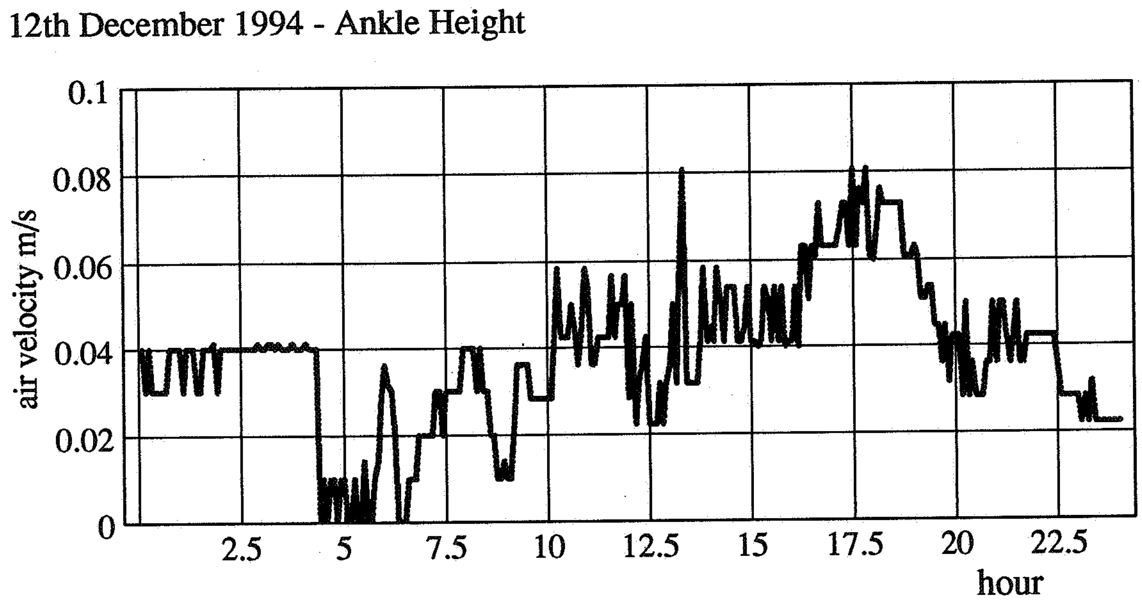

Figure 1B: Internal air velocity at ankle height with the trickle ventilator open

Carbon dioxide

The CO_2 levels recorded in both occupied offices have reconfirmed the findings of the laboratory study which had shown that CO_2 did not exceed 1000ppm when the correct sized ventilator was used for the occupancy of the office. It should be noted that a level below 1000 ppm of CO_2 in an occupied conventional office room indicates a fresh air inflow of about 5 l/s per person [6]. When the ventilators were closed, the CO_2 level was higher and reached that found in the corridors. Figure 2 shows the CO_2 levels recorded for the two monitored weeks in office A and the levels in the corridor outside the office.

It can be seen that during the first week (Fig 2A) the levels in the room and corridor follow each other fairly closely, rising to a peak of approximately 1400 ppm on Thursday. The office was unoccupied on Friday so the room peak level is much lower at about 800 ppm. Apart from the Friday when the room was empty in the afternoon, the CO_2 levels in both corridor and room exceed the 1000 ppm limit. During the weekend, both levels drop to the outside level of about 350-400 ppm. They begin rising again on Monday reaching a peak of ~1100 ppm.

On Tuesday of the second week, the vent was opened about 12 noon, and the room level begins to fall off from a peak of 900 ppm. The corridor level reaches a peak of 1300 ppm. During this second week (Fig 2B), the CO_2 levels within the room were well below the corridor levels. The overall peak recorded for the corridor was in the region of 1600 ppm. During the week when the office was occupied by one person the peak levels did not in any instance exceed 1000 ppm, indicating a fresh air ventilation rate of about 5 l/s per person. However, on the Friday during this phase of the experiment, the peak within the office reached ~1400 ppm arising from increased occupancy (three people) because of a meeting held in that room during the afternoon.

Temperature

Internal air temperatures were very similar in all cases. Figure 3 presents the results for two weeks in office A. It can be seen that although the outside temperatures was much colder during the second week (ventilator open), the internal temperatures were maintained at the same level during both weeks. The following table shows the average temperature inside and outside during both weeks:

	First Week	Second Week
T_{int} (average)	16.8 ^{0}C	15.7 ^{0}C
T_{ext} (average)	10.0 ^{0}C	3.8 ^{0}C

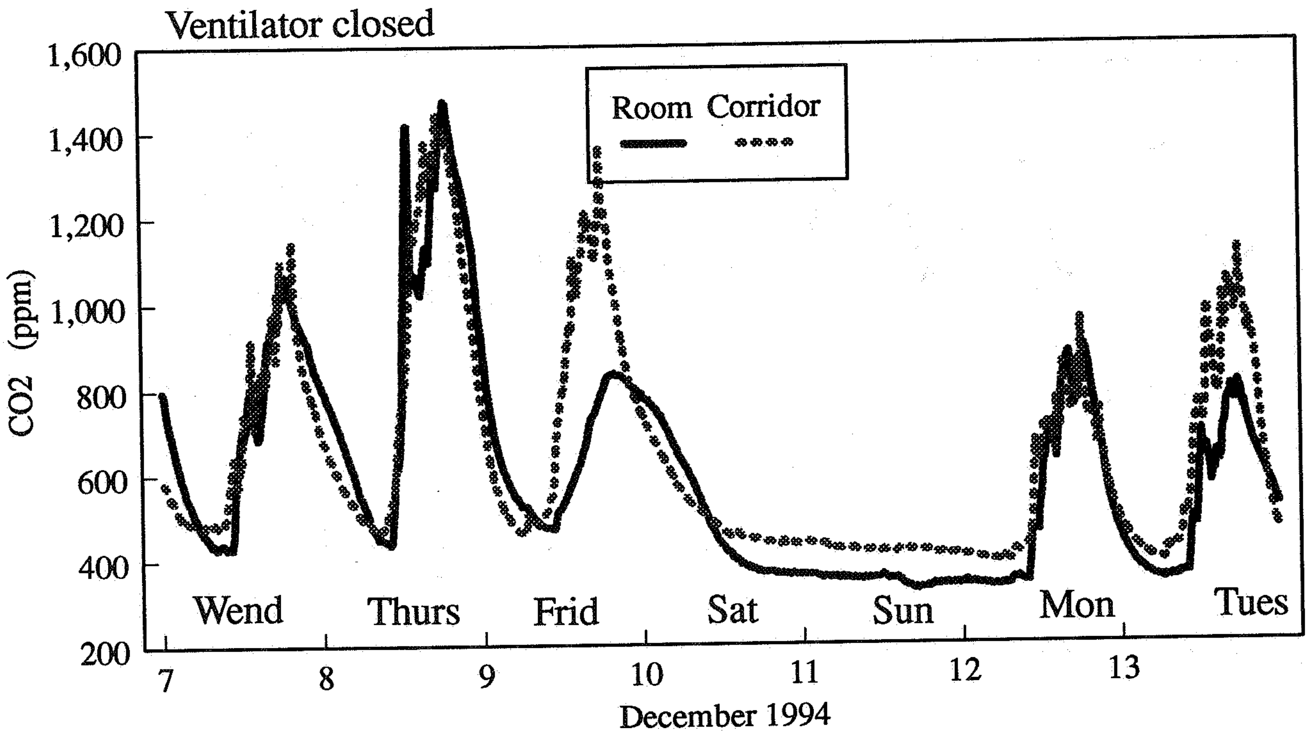

Figure 2A: CO_2 levels in office A and corridor with the trickle ventilator closed

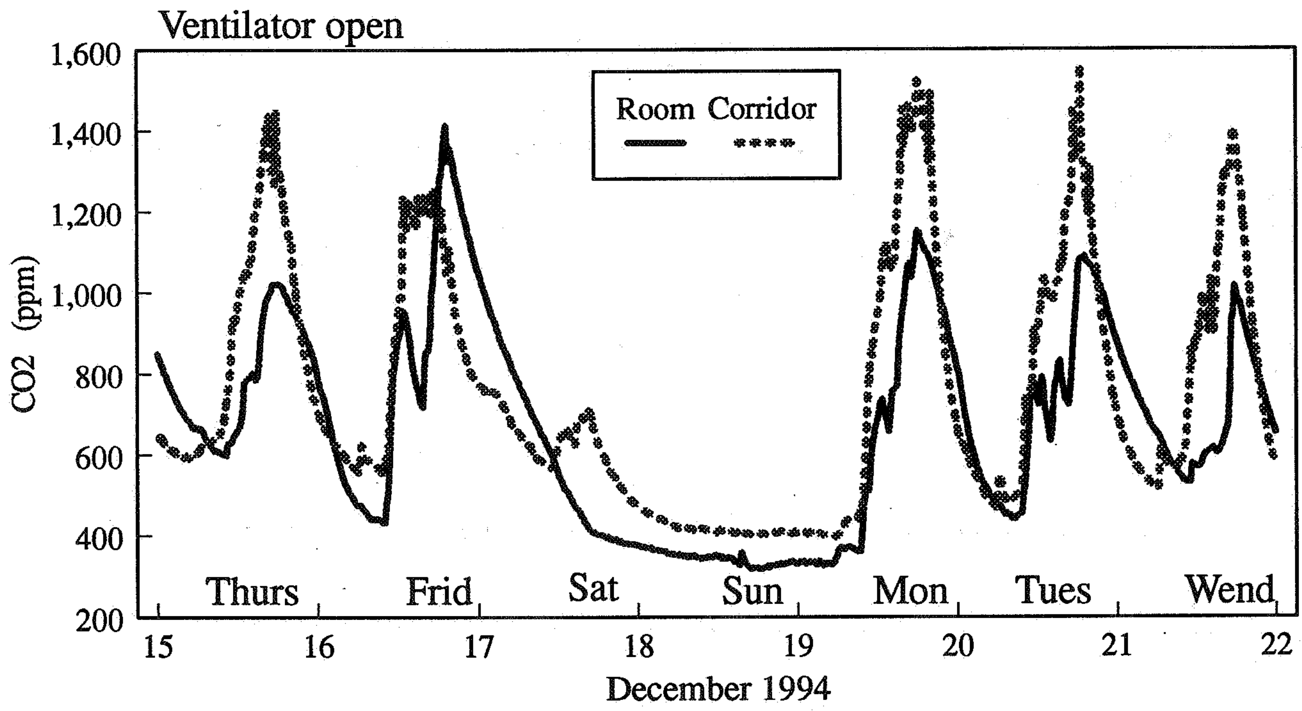

Figure 2B: CO_2 levels in office B and corridor with the trickle ventilator open

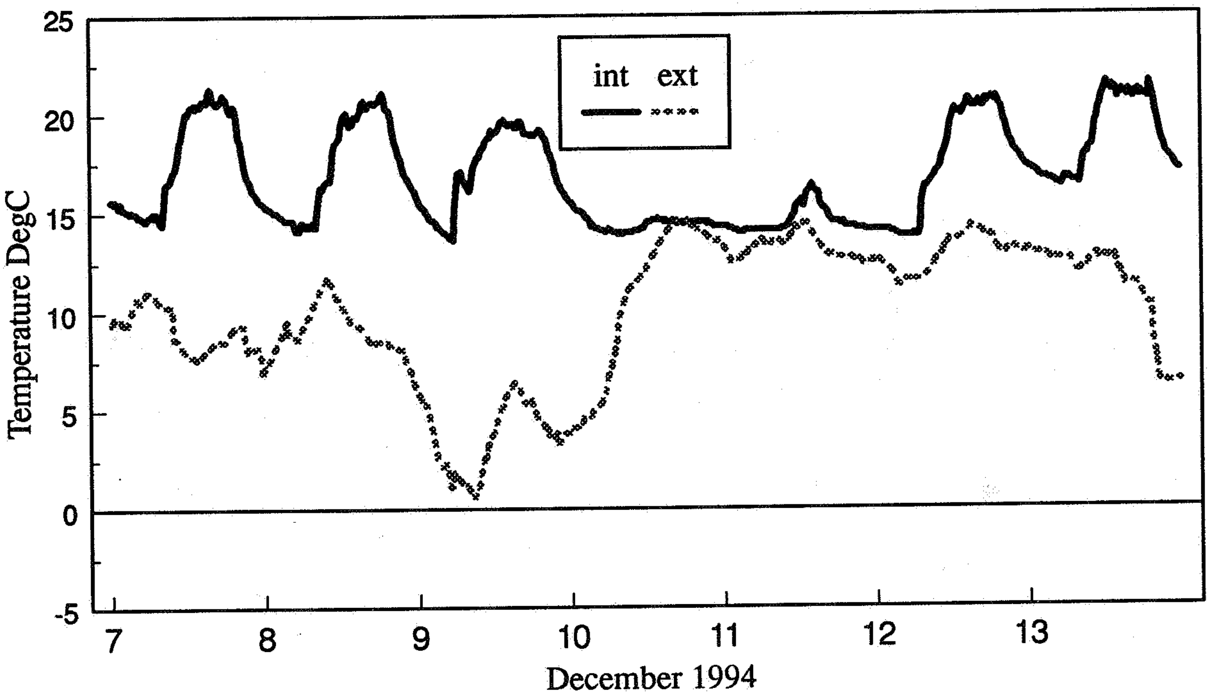

Figure 3A: Internal and external temperature with the trickle ventilator closed

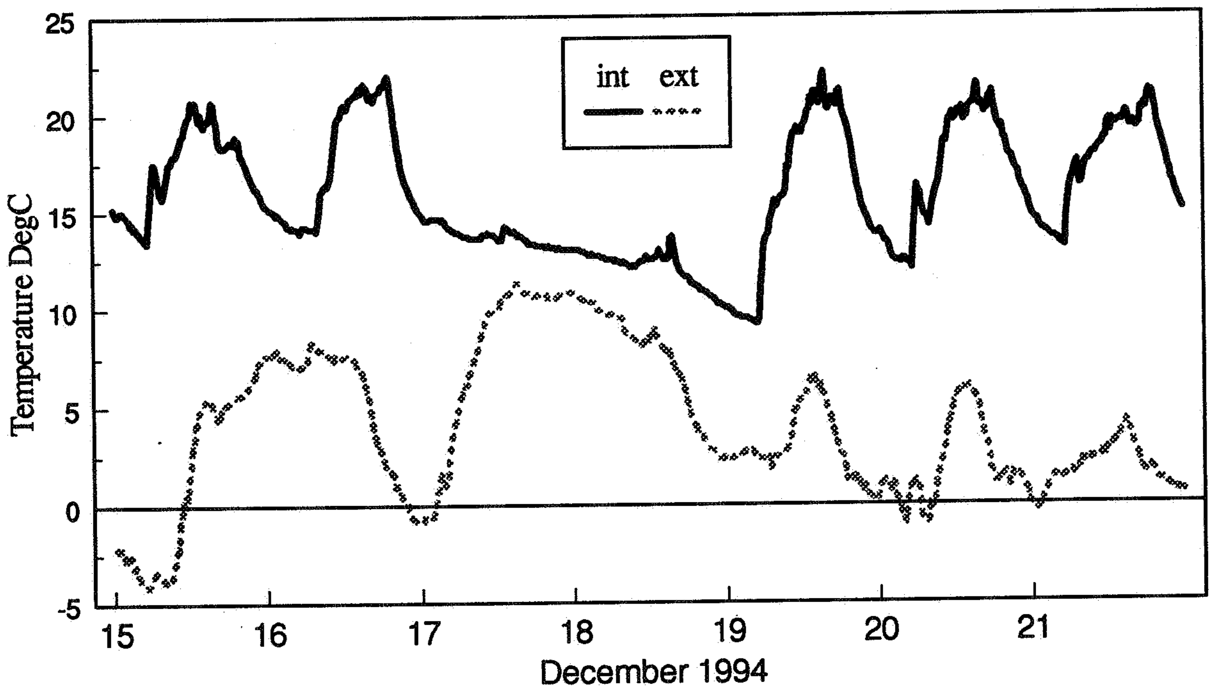

Figure 3B: Internal and external temperature with the trickle ventilator open

CONCLUSIONS

There are four main conclusions:

(a) Trickle ventilators with a minimum openable area of $400mm^2$ per m^2 (of office floor area) can provide adequate fresh air during winter and maintain good IAQ (using CO_2 as the surrogate indicator) within a typical office room located in a suburban site.

(b) Maximum metabolic CO_2 levels in Office A were maintained below 1000 ppm when the occupancy of the office was one person with the ventilator open, indicating a fresh air ventilation rate of about 5l/s per person. Transient increased occupancy increased this level but was quickly restored to normal levels when normal occupancy (of one) resumed. It is interesting to note that this level fell rapidly rather than the slower natural decay as is observable when the offices are left for the night. This is most likely due to the increased ventilation (by opening windows and doors) usually undertaken after a space has been occupied at levels higher than normal occupancy.

(c) Ventilators of poor design, positioned incorrectly and with unfavourable external weather conditions may produce cold draughts. However, cold draughts do not appear to present problem either at desk or ankle height during these field tests. This is probably due to the choice of ventilator which projected incoming air upwards. Measured internal velocities were significantly below the accepted draught threshold of 0.3m/s.

(d) Internal temperature was maintained during the week that the ventilator was open even though there was a significant fall in the external temperature.

Therefore, to conclude, ventilators with adequate sizing and correctly positioned, can provide comfortable and sufficient ventilation for occupants.

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Evaluation of an IR-Controlled Ventilation System in an Occupied Office Building

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EVALUATION OF AN IR-CONTROLLED VENTILATION SYSTEM IN AN OCCUPIED OFFICE BUILDING

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SYNOPSIS

The use of IR detectors to steer the ventilation is in principle an attractive approach for optimising the ventilation according to the occupants needs. In order to evaluate the performances under real conditions, one of the BBRI office buildings in Limelette (some 31 offices with in total 51 persons and a variable occupation load) was equipped with a mechanical supply ventilation system in which each terminal is controlled by an IR detector. During a two week period, the performances of the ventilation system were measured in detail (total air flow rate, functioning of each individual terminal, pressure control function, energy consumption, acoustical measurements,...).

The paper presents first the concept of the ventilation system and some of the measured performances at component level. The overall performance of the system is then discussed as well as the impact of the building and ductwork airtightness. Finally conclusions and recommendations for further improvements are given.

1. INTRODUCTION

Ventilation is a major energy consumer in office buildings and it is therefore interesting to control it so as to find a compromise between indoor air quality and energy consumption.

Although research in the recent years has showed that occupants are not the only source of pollution in office building (but also building materials), it is clear that, up to the present, it is the only source upon which a realistic ventilation control can be based.

The use of infrared detectors to steer the ventilation is a possible approach for optimising the ventilation as a function of the occupancy in the ventilated spaces. Other methods like people counting or carbon dioxide measurements can be appropriate as well.

In order to evaluate its performances under real conditions, one of the BBRI office buildings was equipped with a mechanical supply system in which each terminal is controlled by an infrared sensor.

After a description of the investigated system, this paper presents the results of the performed monitoring: evaluations at component level (pressure control, ductwork,...) are first presented, secondly the overall system performances. Then comes a chapter that shows how the ductwork airtightness and the building airtightness impact on the performances of the system. Finally conclusions and recommendations are given.

2. SYSTEM DESCRIPTION

2.1 GENERAL

The studied IR controlled ventilation system is installed in BBRI office building B which is situated in Limelette, 20 km South of Brussels, Belgium. It is a 2-storey office building with a volume of 3000 m^3, counting 31 rooms with 1 to 4 working places each, or 51 working places in total.

The mechanical air supply installation is made of two separated systems, one for each storey. They consists of a constant speed fan, a filter providing the pressure control function (F.A.R. type manufactured by the French firm ALDES), a heating exchanger, ducts and ventilation terminals (OPTO type manufactured by the French firm AERECO).

The system is designed to provide 25 m^3/h of outside air per working place in the rooms that are effectively occupied. When all the rooms are occupied, this provides a total air flow rate of 51 x 25 = 1275 m^3/h for the whole building.

The ventilation system is operating from 6:00 AM to 8:00 PM.

2.2 Supply terminals

The OPTO ventilation terminal is controlled by a motion sensor based upon infrared detection. If presence of occupants is detected in a room, the ventilation is activated thanks to a electro/pneumatic device. When the occupants leave the room, the ventilation remains activated during 15 minutes before switching off.

The terminal can be manually set in four different opening positions corresponding with four different air flow rates: 25, 50, 75 and 100 m^3/h. This setting is made once for all in function of the number of working places in the ventilated space, one working place corresponding with a rate of 25 m^3/h. The pressure in the ductwork should stay between 70 Pa and 130 Pa to ensure a proper operation of the terminals.

2.3 Autocontrolling filter

2.3.1 Filtering and pressure control functions

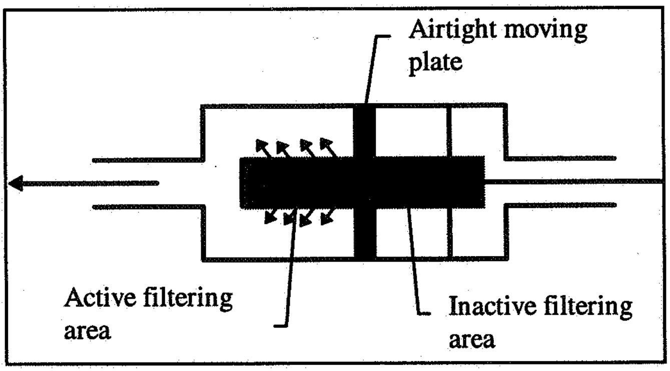

Figure 1 - Pressure control function of the filter

Besides the usual filtering function, this autocontrolling filter is designed to maintain the pressure in the ductwork between certain limits. This control function is of course essential for a ventilation system allowing the total air flow rate to vary with the demand.

The pressure control function is obtained by moving an airtight plate which modifies the active filtering area through which the air is passing (see figure) hence the pressure drop through the filter. The pressure in the ductwork is measured and the position of the moving plate is adjusted in order to keep this pressure between the chosen limits.

Furthermore, this equipment is designed to change automatically the filter when the accumulated particulate matters provoke too high pressure losses in the filter to ensure a normal operation of the ventilation system.

3. Measurement set-up

Various parameters were continuously recorded during the monitoring of the whole system operation: the total supply air flow rates using tracer gas technique, the outside temperature, the temperatures after the pre-heating units, the pressure at the end of the ductwork and just after the fan, the state (open or closed) of each OPTO ventilation terminal of the ground floor system and the position of the airtight moving plate in the autocontrolling filters.

Figure 2 shows the ventilation system with its different components and the positions of the different measurement points.

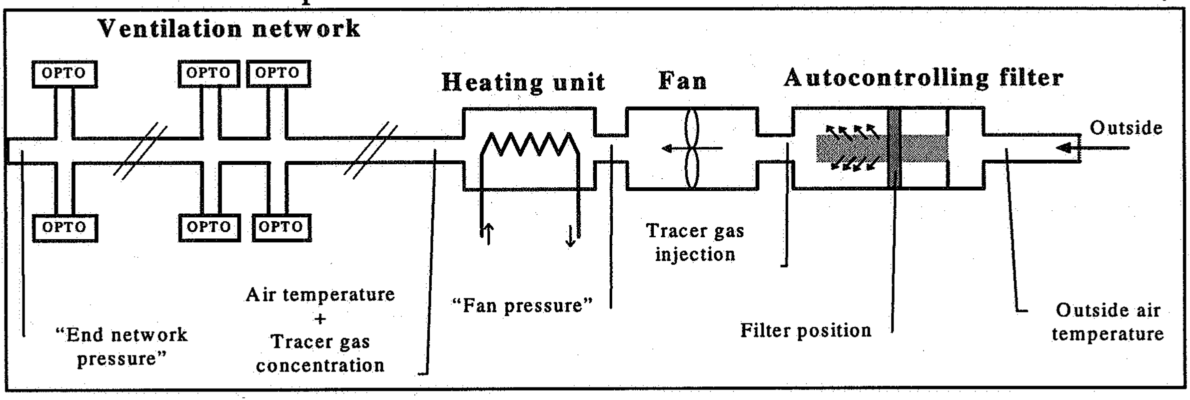

Figure 2 - Monitoring of the whole system operation - Measurement points

Spot measurements of the overall building airtightness, the ductwork airtightness, the acoustical level, the air flow characteristics of terminals and air flow rates at ventilation terminals were also performed.

4. EVALUATION AT COMPONENT LEVEL

4.1 DUCTWORK

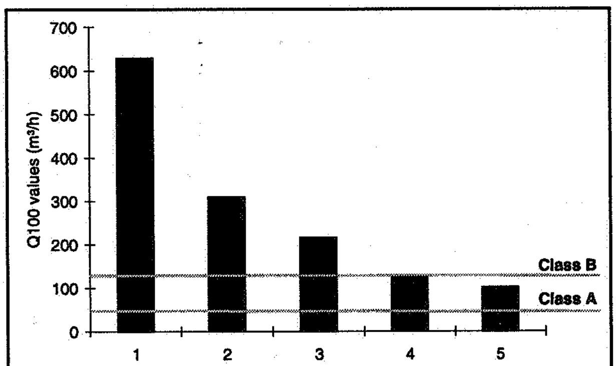

Figure 3 - Ductwork airtightness measurements and EUROVENT 2/2 standard classes

Figure 3 shows the results of the airtightness measurements performed on the ductwork of the first floor ventilation system. They are presented in terms of Q_{100} values, i.e. the air flow rate flowing through the ductwork leakages at a pressure difference of 100 Pa, which is the average pressure of the normal operation range of the ventilation terminals (70-130 Pa).

The EUROVENT 2/2 standard, "Air leakage rate in sheet metal air distribution systems", proposed two classes for ductwork airtightness. The Q_{100} values according to this standard are also shown.

As it can be seen, the initial Q_{100} value was about 630 m³/h which is almost equal to the nominal ventilation air flow rate (625 m³/h). Under these conditions, the system could not operate normally. Several improvements were necessary to reach an acceptable value. The last test (n°5) gave a value slightly better than the class B of the EUROVENT 2/2 standard.

4.2 AUTOCONTROLLING FILTER

The pressure control function should allow in theory to maintain the pressure in the ductwork in the operating range of the ventilation terminal (70-130 Pa)

The following figure gives, for the ground floor, the measured pressures just after the fan and at the end of the ductwork as well as the filter position during a typical day. The maximum filtering area corresponds with the top of the scale (38 cm) and the minimum filter area with the bottom of the scale (0 cm).

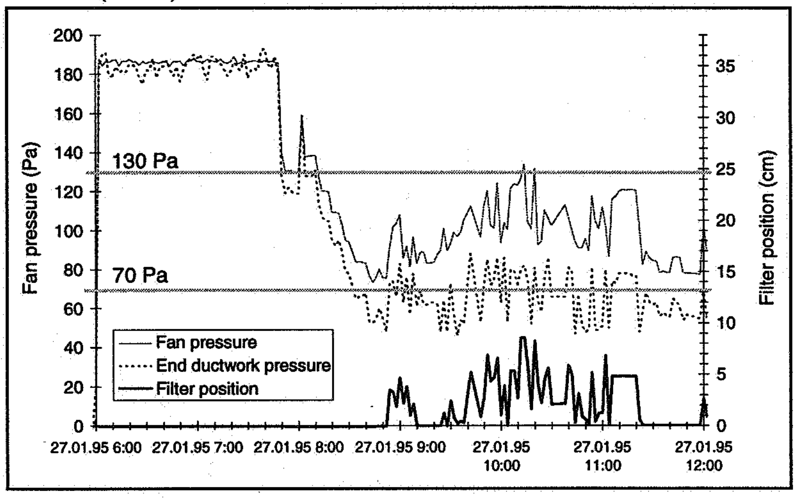

Figure 4 - Filter position, fan pressure and end ductwork pressure (ground floor)

The following remarks can be made:

- In the morning, when the ventilation demand is zero or very low, the filtering area is minimal which should provoke a sufficient pressure drop to decrease the air flow rate through the fan so as to obtain a ductwork pressure between 70 and 130 Pa. However, the measured ductwork pressures are near 180 Pa. This suggests that either the fan is too powerful or the maximal pressure drop the filter can provoke is too small.
- Since the pressure at all ventilation terminals is somewhere between the two measured pressures, it is likely that most of them function within the operating range (70-130 Pa). However, the pressure at the end of the ductwork falls under 70 Pa several times which should be prevented thanks to a better pressure control. Moreover, too low pressures prevent the terminal to close completely and some air can flow into unoccupied rooms.

The following figures show the fan pressure as a function of the filter position and the filter position as a function of the ventilation demand.

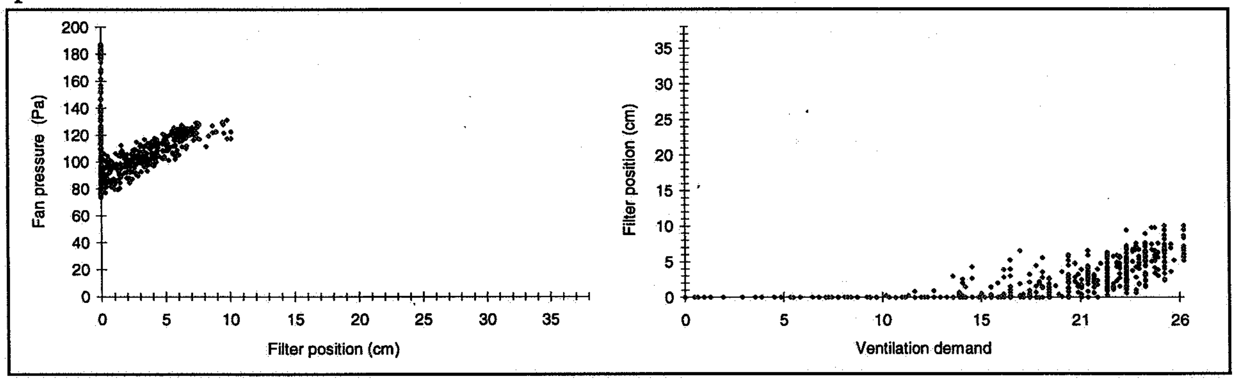

Figure 5 - Operation of the pressure control function of the filter

As it can be seen, the pressure after the fan can vary from 180 Pa to about 75 Pa when the active filtering area is the smallest possible (filter position = 0 cm). One can also observe that the active filtering area of the filter remains quite small (not more than 11 cm over 38 cm).

The pressure control function of the filter operates only when the ventilation demand goes over 10 people. Below this value, the active filtering area is minimal and the pressure varies in function of the air flow rate, it is no more controlled by the filter and, as a consequence, the supplied air flow rate per occupant can strongly vary in the time.

5. OVERALL PERFORMANCES

The overall performance analysis is based on the ground floor measurements (26 places).

The next figure shows the air flow rate effectively supplied to the ground floor in function of the ventilation demand as well as the air flow rate that would be supplied by a perfect system (25 m^3/h per person). The air flow rate supplied by a constant air flow rate system installed in the same building, that is 26 x 25 m^3/h, is also shown.

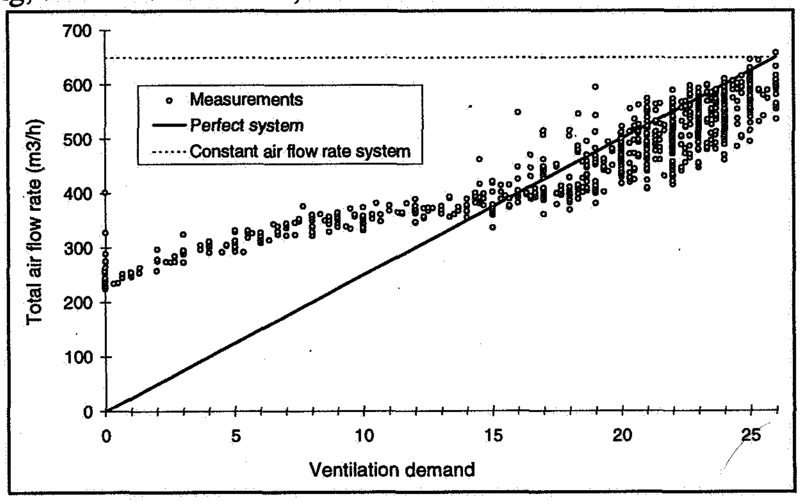

Figure 6 -Air flow rate supplied in function of the ventilation demand - Measurements, perfect controlled system and constant air flow strategy

One can observe that the total air flow rate is quite higher than expected for the low ventilation demand (< 15 people). This is on the one hand due to the bad operation of the pressure control for low demand and, on the other hand, to the ductwork leakages.

It must be stressed that the total air flow rate shown on the previous figure represents the amount of air supplied by the fan and not the amount of air supplied to the occupied rooms. Indeed, a part of the air is supplied through the ductwork leakages to the corridor and another part is supplied through the closed ventilation terminals to the unoccupied rooms.

Briefly put, it can be said that the previous figure gives a correct image of the system performance from the **energy point of view** but does not allow to evaluate the system performance from the **IAQ point of view**.

5.1 SPLIT-UP OF THE TOTAL AIR FLOW RATE

The total air flow rate supplied by the ventilation system (ground floor) is made of three components: the air flow rate supplied to occupied rooms through open terminals; the air flow rate supplied to unoccupied rooms through closed terminals; the air flow rate supplied to the corridor or the basement through ductwork leakages.

On the basis of the air flow characteristics of the ventilation terminals and the ductwork leakages, it has been possible to model the behaviour of the ventilation system and to split the total air flow rate into its different components. This is shown on the next figure as well as the average air flow rate per person in occupied and unoccupied rooms.

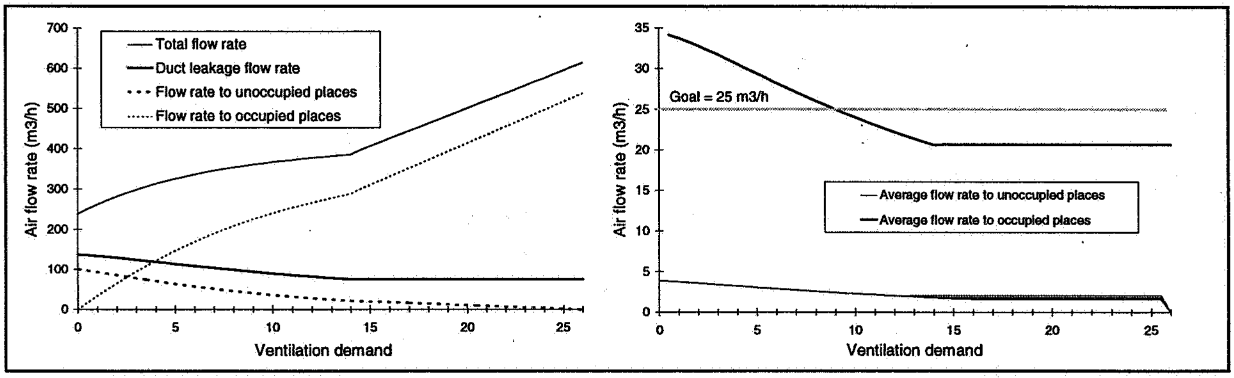

Figure 7 - Split-up of the total air flow rate supplied to the ground floor and average air flow rate per person.

As one can see, the air flow rate through the ductwork leakages varies from 100 m^3/h for the high demand up to 130 m^3/h for the low demand. Moreover, when the ventilation demand is very low, the air flow rate to the unoccupied rooms is about $100^3/h$.

The average air flow rate supplied to the people (in occupied rooms) can vary from 34 m^3/h to about 20 m^3/h which is due to the pressure variation in the ductwork owing to the incorrect operation of the pressure control function of the filter.

6. Impact of Ductwork and Building Airtightness

6.1 Building Airtightness

It is clear that the building airtightness plays a very important role in the efficiency of the IR controlled ventilation system. Indeed, the air infiltration results in an additional air flow rate which is totally uncontrolled and thus reduces the saving that can be obtained.

The BBRI office building B has an average air infiltration of about 400 m^3/h (simply evaluated from the Q_{50} value) and the ventilation system does not induce any over-pressure in the building which could somehow counteract the air infiltration through the envelope.

The following figure shows the energy savings that can be obtained thanks to an IR controlled ventilation system in comparison with a constant air flow strategy and for 3 degrees of airtightness. It was assumed that the pressure control was perfect (100 Pa).

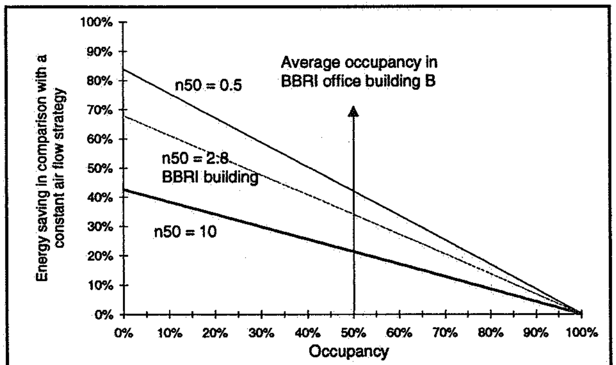

Figure 8 - Energy saving in comparison with a constant air flow strategy for different building airtightness values

As it can be seen, the energy saving is about 34% for the present characteristic of building B (n_{50}=2.8 vol./h) for the average measured occupancy (50%). A very good airtightness in such a building would be about 0.5 vol./h which would results in a gain of 42%. A very bad airtightness (n_{50}=10 vol./h) would give a gain of only 21%.

6.2 Ductwork Airtightness

The ductwork airtightness is a critical aspect in controlled ventilation systems. We will consider two different cases to illustrate that matter.

First case

It will be assumed that the installed fan has been chosen to provide a maximal air flow rate which takes into account normal ductwork leakages (class B in EUROVENT 2/2 standard) and 25 m^3/h for each working place. The fan is strictly limited to that maximal value.

In this case, if the airtightness of the ductwork is worse than expected, it would result in a too low operating pressure hence air flow rate when the demand is high.

This case is illustrated on the next figures. The first one represents the energy saving that can be achieved in comparison with a constant air flow strategy for two degrees of airtightness.

Normal ductwork leakages (class B in EUROVENT 2/2 standard) are assumed for the constant air flow strategy. The ductwork airtightness of the "leaky ductwork" is taken equal to the first measured value in the BBRI building B, before any improvement and the ductwork airtightness of the "normal ductwork" is taken equal to the last measured value in the BBRI building B, after all the performed improvements.

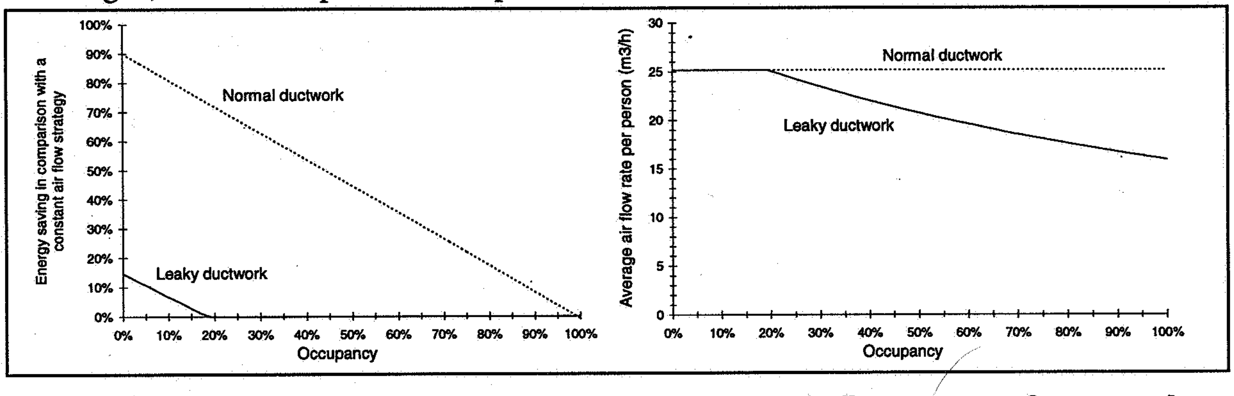

Figure 9 - Energy saving in comparison with a constant air flow strategy for two values of ductwork airtightness - Average air flow rate per person

The impact of the ductwork airtightness is very clear. For the "leaky ductwork", when the occupancy is higher than 20%, the fan gives its maximal air flow rate and every new increase of the occupancy results in decreasing the ductwork pressure and accordingly the air flow rate per person. A lot of air is supplied to the corridor and the basement. Moreover, since the maximal air flow is supplied, the IR control has no impact on the energy consumption. The performance is disastrous from the energy point of view and from the IAQ point of view.

Second case

The chosen fan can supply more than the air flow rate needed. This is a very interesting solution in a building where the number of working places is very likely to change in the time. In this case, if the pressure control operates correctly, the air flow rate supplied in the occupied rooms will be as expected but, due the ductwork leakages, an additional air flow will be supplied to the corridor or the basement which will result in additional thermal losses.

Figure 10 compares the energy saving achieved in comparison with a constant air flow strategy for both values of airtightness.

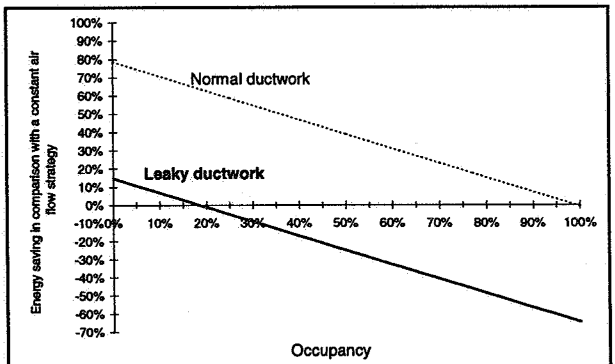

Figure 10 - Energy saving in comparison with a constant air flow strategy for two ductwork airtightness

In this case, the air flow rate per person is as required, i.e. 25 m^3/h, because it is assumed that the pressure control is perfect. However, a large part of the air leaves the ductwork through its leakages which explains why, when compared to the reference case, the energy savings are negative for high occupancy.

7. CONCLUSIONS

The following conclusions can be drawn from the measurements and observations made.

Regarding the performances of the ventilation terminals:

- The detection function of the ventilation terminal works apparently in a satisfactory way.
- Observations showed that under normal conditions, the air velocities in occupied spaces are low enough to avoid draught complaints. Furthermore, the noise levels are acceptable.

Regarding the overall system performances

- The measurements have highlighted the tremendous importance of the duct airtightness. In the investigated building and system, the original airtightness was so poor that the IR control had little impact on the total air flow rate and as a consequence on the energy bill.
- During periods of non-occupation of the building or periods of very low ventilation demand, the noise coming from the ventilation system is not negligible because of the high pressure in the ductwork (up to 180 Pa). A good pressure control should in principle solve this problem.
- A lot of efforts were needed to improve the ducts airtightness so as to come near an acceptable value. It is a very labour intensive activity hence a high cost activity and, therefore, it is clear that the only realistic solution is to achieve a better airtightness from the beginning and clear performance requirements in the technical prescription.
- The pressure variation in the ductwork were considerable (from 60 to 180 Pa). This means that the compromise between indoor air quality and energy consumption is not so optimal. We believe that an improvement is required.

Recommendations

- A special attention should be paid to the ductwork airtightness. We would suggest the following procedure:
    1. clearly specifying a minimum level for duct airtightness in the technical prescriptions.
    2. a systematic testing of the duct airtightness after the installation.
    3. perform improvements if needed.
- The pressure in the duct should be better controlled. We see the following possibilities:
    - a variable fan speed, which is probably a rather expensive solution but it has the advantage that the fan energy bill will drop substantially.
    - a wider operating range for the filters permitting to keep the pressure more or less constant over the complete range of air flow rates.
    - a variable air volume outlet in the non-heated area with a control of the terminal opening position which leads to a constant pressure in the ductwork. This terminal should be placed after the fans but before the pre-heating unit.

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Cooling Performance of Silent Cooling Systems Built by Free Convective Coolers

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Synopsis

For the planning of "silent cooling" systems built by free convective coolers, it is necessary to support characteristic data for the cooling performance and the effect of different installation and operating parameters on the cooling performance. At the "Institut für Angewandte Thermodynamik und Klimatechnik" at the University of Essen measurements of the cooling performance of free convective coolers were carried out by using a testing chamber as well an enlarged and modified testing room with dimensions near to practise.

The investigations have shown that the cooling performance of convective coolers varies with different parameters like dimensions of the room, distibution of heat sources and positioning of the coolers in the room. A realistic investigation of these coolers and an objective comparison with other systems like chilled ceilings is only possible, if the investigation takes place under realistic operating conditions and arrangements of coolers and heat sources. Otherwise, the results from different systems are not comparable. The effects of different installation parameters have shown, that an accurate planning of the installation is necessary to guarantee sufficient cooling performance.

List of symbols

C		regression coefficient
ε	[-]	emission coefficient
n		regression coefficient
$\dot{q}$	[W/m^2]	heat-flux density
t_i	[°C]	initial temperature of the heat distribution medium
t_r	[°C]	return temperature of the heat distribution medium
t_R	[°C]	room temperature inside the testing chamber
Δt_R	[K]	logarithmic temperature gradient between room and heat carrier

1. Introduction

Nowadays chilled ceilings in combination with ventilation systems are often installed, so that the total energy consumption can be reduced by the separation of cooling and ventilation. This reduction can be reached because it is more effective to transport energy using water systems instead of air to deliver cooling energy to the consumers. Instead of using chilled ceilings to deliver the sensible cooling to the room it is possible to install free convective coolers inside the room or above an intermediate ceiling to compensate the cooling loads.

For the planning of these systems it is necessary to support characteristic data, which describe the cooling performance of these systems. In Germany there are two different standards for measurements of cooling performance at these systems available. The DIN 4715 /1/ of the German standard organisation regulates the measurements at chilled ceilings as well as the investigation of the cooling performance of free convective systems. The guide-line /2/ of the the German FGK e. V. (Fachverband Gebäude-Klima e. V.), in which the leading manufacturing, planning and installation companies are represented, defines the boundary conditions for measurements of cooling performance of chilled ceiling modules in a testing chamber.

During the last few years these two standards were often used to investigate the cooling performance of chilled ceilings, where the heat transfer to the room bases mainly on radiation. But up to now little has been published about the use of these standards in the determination of cooling performance of free convective coolers. In this paper the experiences with measurements of the cooling performance of free convective coolers in the testing chamber, which is described in the guide-line of the FGK e. V., at the "Institut für Angewandte Thermodynamik und Klimatechnik" at the University of Essen will be presented as well as results from measurements in a modified enlarged testing room. A proposal for measurements at free convective coolers and the main aspects of the installation of these systems will be discussed.

2. Testing chamber and conditions for measurements

The testing facility, that is explained in the guide-line of the FGK e. V./2/, bases on thermal measurements at chilled ceiling elements with closed or open surfaces in a testing chamber as shown with plan and side view in Fig. 1. The conditions for measurements and the testing chamber are described in detail in the guide-line of the FGK e. V. as well as in /3/, so that only the main parameters will be repeated.

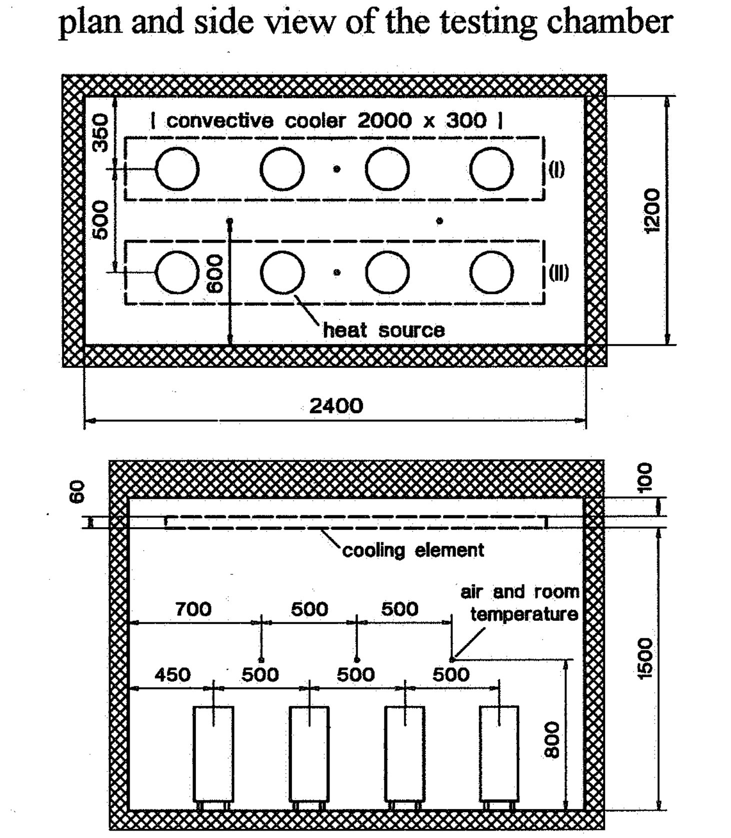

Figure 1

All elements of this testing chamber have to be well insulated. The thickness of insulation with a thermal conductivity less than 0.04 W/mK must be more than 0.1 m for walls and floor. For the ceiling the thickness has to be more than 0.2 m. The emission coefficient ε of the inside surfaces of the chamber must be higher than 0.9. Also it must be possible, that air circulates around the chamber while the temperature difference between the ambient air and the air inside the testing chamber has to be less than 1 K. The reference temperature inside the chamber is the so called room temperature, which will be measured by a temperature sensor inside a black ball (diameter of 35 mm). The shown installation in Fig. 1 with two convective coolers (2000 x 300 mm) is not provided in this guide-line. The cooling load will be simulated by 8 clearly defined cylindrical heat sources with a black surface ($\varepsilon > 0.9$).

The room and air temperatures inside the testing chamber have to be measured at at least 4 positions as marked in Fig. 1. Additional temperature sensors have to be installed at the center of the wall and bottom surfaces. Each water temperature has to be measured by two separate temperature sensors, while a difference of less than 0.05 K between these sensors is allowed. Otherwise they have to be exchanged.

The investigation of the characteristic of a chilled ceiling element includes at least 3 series of measurements with different initial temperatures of heat carrier medium. The rated temperatures are 12, 14 and 16°C with a tolerance of ± 0.5 K, while the room temperature inside the chamber has to be 26°C ± 0.2 K.

During a period of at least one hour with stationary operating conditions 10 measurements have to be taken to describe the cooling performance of the chilled ceiling elements. The chosen measuring device allows up to 60 measurements during a period of one hour.

3. Analysis of measurements

If the heat transfer bases mainly on convection, the cooling performance should be described by a "room characteristic". The results of the measurements of cooling performance in the testing chamber can be described by

$$\dot{q} = C \cdot \Delta t_R^n \qquad\qquad \Delta t_R = \frac{t_i - t_r}{\ln \frac{t_R - t_r}{t_R - t_i}}$$

The parameters C and n have to be determined by a regression basing on the measured parameters and Δt_R, the logarithmic mean temperature gradient between heat carrier and room temperature, as measured with the "black balls".

4. Results from measurements

Investigations at free convective coolers, which are installed in the testing chamber as shown in Fig. 1, were carried out to characterise the cooling performance and the effects of operating conditions and distribution of heat sources on the performance. The cooling performance of the convective coolers has been described in a modified room characteristic, in which the cooling performance per length of cooler instead of the cooling performance related to the surface of the cooler (2000 mm x 300 mm) was chosen.

Figure 2 shows three room characteristics of the measurements that were carried out to describe the effect of asymmetrical and symmetrical arrangements of coolers and heat sources on the cooling performance. The room characteristic of the cooler I with both coolers and groups of heat sources active was chosen as reference for the two asymmetrical cases, where only the cooler I and the heat sources beside or below the convective cooler were active. The maximum cooling performance can be reached with an asymmetrical arrangement of coolers and heat sources while the positioning of the coolers just above the heat sources leads to a minimum of cooling performance, while the differences are near neglectible.

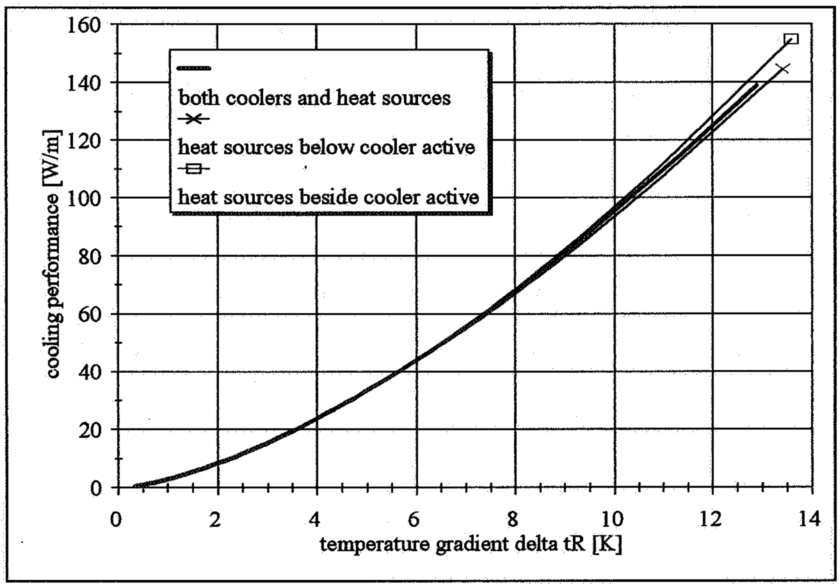

Figure 2: "room characteristic of cooler I with symmetrical and asymmetrical arrangement of coolers and heat sources

Parallel investigations of the cooling performance of convective coolers in a test room with the dimensions of a "real" office room had shown significant higher cooling performance. The small distance between the floor and the cooling element in the testing chamber may have been one reason for this effect, because natural convection at the cooling elements can be limited by this kind of installation. More realistic values of the cooling performance can be determined by investigations in a larger "full scale" testing room with dimensions similar to the dimensions of offices in reality. Such a testing room is shown in fig. 3. It was built according to most of the requirements described in the FGK guide-line. But the dimensions of the testing room deviate from some paramters fixed in the guide-line and the German DIN-standard 4715 as well as the chosen asymmetrical arrangement of the heat sources and coolers.

The width of the testing room is characteristic for single office rooms with two axis of the buil-ding while the length was limited because of the available space in the laboratory. The testing room represents an office room with a length of three axis with a board at the inside wall. The test room is as well insulated as the smaller testing chamber. The heat sources were located at the "outside wall", because the desks are normaly positioned near the windows. Additional to the internal heat sources the direct solar gain will be absorbed near the windows, so that this part of the cooling load will be set free near the outside wall too. So it is possible to simulate main parts of internal as well as external cooling loads by the installed heat sources. The convective coolers should be installed as described by the manufacturing and installation companies.

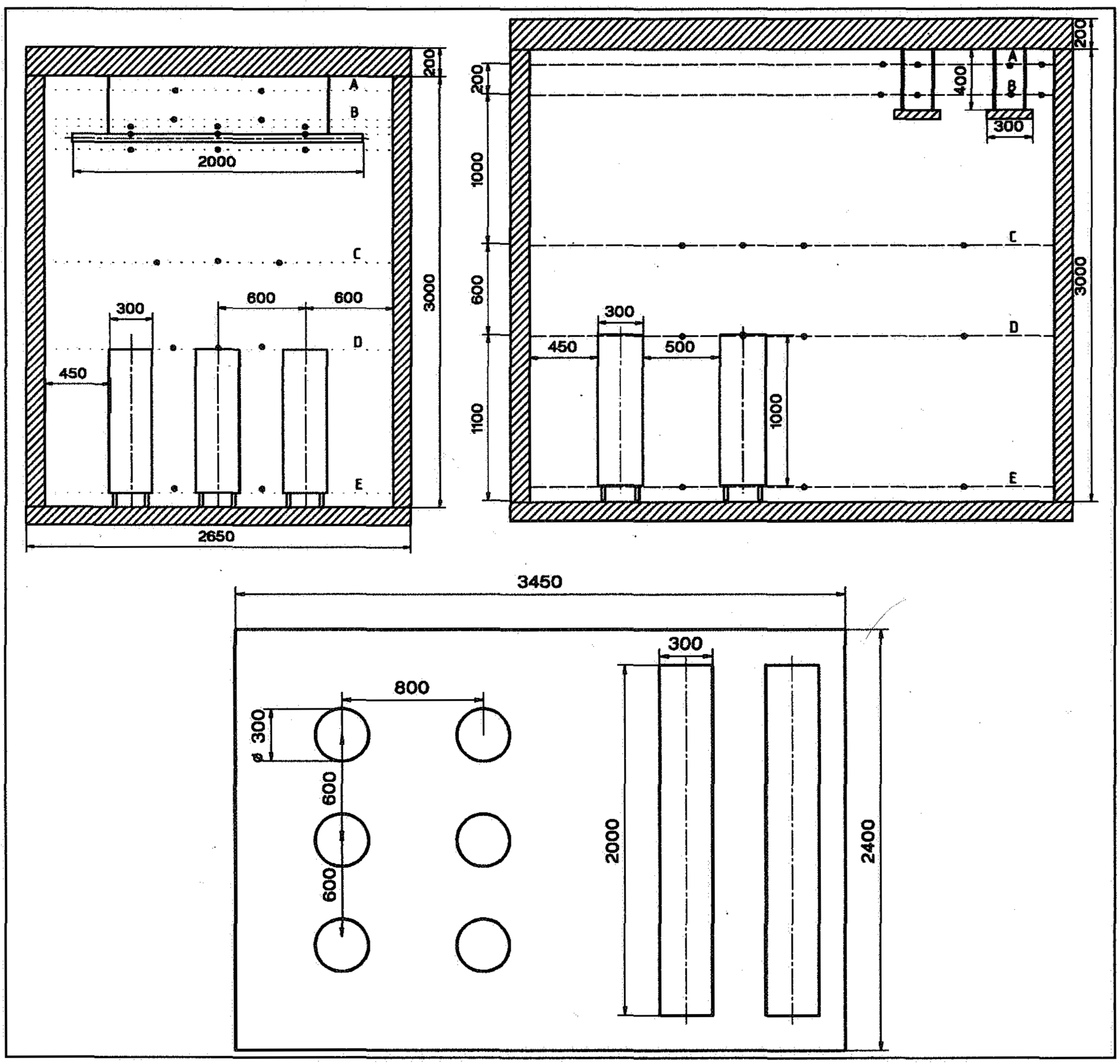

Figure 3: plan and side views of the enlarged testing room

A realistic cooling load of 55 W/m^2 leads to a total demand of 455 W of cooling inside the testing room. The estimated cooling performance of the investigated convective coolers is about 165 Watt per length of cooler at a difference of 10 K between mean water temperature and air temperature inside the testing room, so that two coolers with a length of 2000 mm and a width of 300 mm had been installed near the inside wall as shown in figure 3. This kind of installation was prefered by the installation company, which uses this kind of cooler to build the convective cooling systems.

During the measurements of cooling performance in the enlarged testing room the differences between simulated cooling load and total performance of the coolers was always less than 30 watt, so that the requirements fixed in the German DIN 4715 were met. During some measurements this difference was less than 12 watt, so that the requirements of the FGK guide-line could be fulfilled although the test room is much larger than the FGK testing chamber. Also the parameters to guarantee stationarity during a measuring period of at least one hour could be met as it is fixed in the FGK guide-line, so that the air flow pattern inside the test room appears to be stable and stationary.

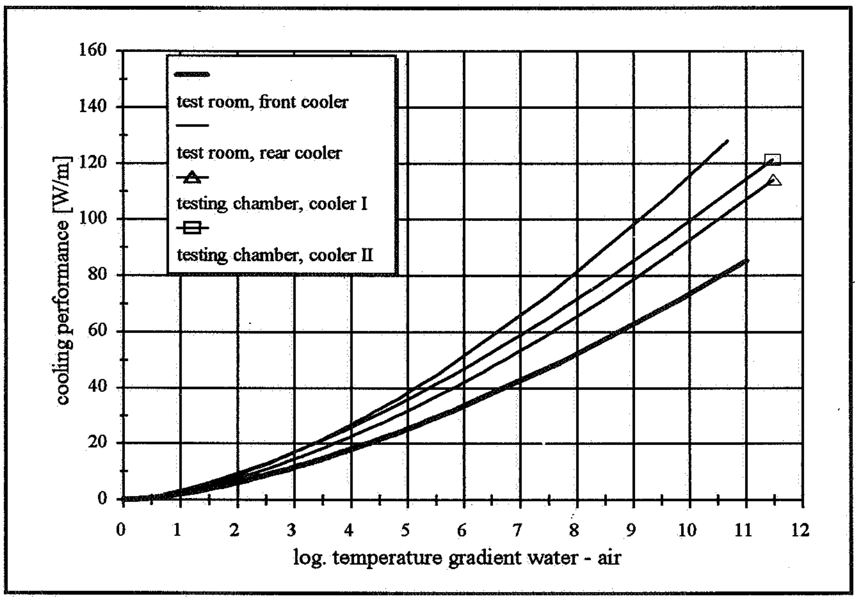

Figure 4: comparison of measured cooling performance of convective coolers -results from testing chamber and large test room

Figure 4 shows the room characteristic of the investigated convective coolers basing on measurements in the testing chamber and the test room. It has to be mentioned that different water flow rates per cooler had been adjusted (test room: 100 l/h, testing chamber: 140 l/h). But various measurements at these coolers with different water flow rates had shown, that these differences will have only a small effect on the cooling performance. The distance between cooler and ceiling inside the larger test room is twice the distance, that was adjusted in the testing chamber. This corresponds with the enlargement of the test room and the higher cooling load. The figure shows, that the arrangement of the coolers and the dimensions of the room have a significant effect on the performance of the convective coolers. There is only a small difference of performance between the two coolers during the investigations in the small testing chamber in comparison to the results for the large test room. The cooling performance of the rear cooler near the "inside wall" is much higher than the values, which could be found out for the front cooler. An analysis of the temperature distribution has shown, that there is a stable circulation of the air inside the test room. The buoyancy at the heat sources and the natural convection at the coolers leads to a stable air flow pattern. Because of the temperature distribution at the front cooler it could be estimated, that the air flows over the front cooler and the main part of the cooling will be supported only by the rear cooler. The natural convection at the front cooler is not able to sustain a sufficient air flow rate through the front cooler. This shows, that only the larger testing chamber allows an investigation of the convective coolers under operating conditions near to practice. During additional series of measurements the effect of different distances ot the coolers from the ceiling in a range from 50 mm to 400 mm on the cooling performance were examined as well as the effect of shafts of different length directly below the coolers to increase the natural convection (figure 5 - 6).

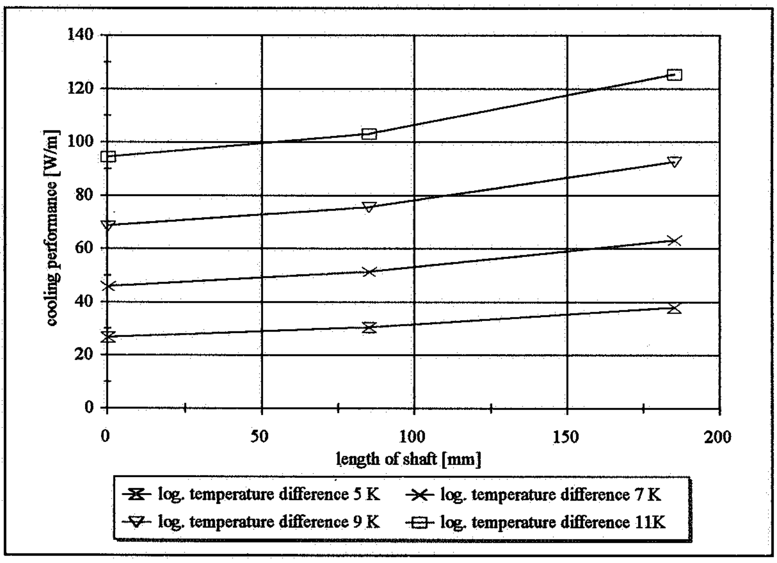

Figure 5: room characteristic of front cooler, 200 mm distance to ceiling

The analysis of the measurements has shown, that the cooling performance of both convective coolers increases nearly linear by with the distance from the ceiling in a range from 100 mm to 400 mm, while the gradient increases with the temperature difference between cold water and air temperature. The installation of shafts directly below the coolers leads to significantly higher cooling performances of both coolers (figure 5 and 6). So the investigations have shown, that there has to be sufficient distance between the coolers and the ceiling to guarantee the cooling performance. To increase cooling performance it is at first necessary to provide a sufficient distance between cooler and ceiling. If there is additional space available, it is possible to install shafts below the coolers, while the length of these shafts has to be at least 1.5 times the depth of the fins of the convective coolers. Otherwise the distance of the coolers to the ceiling should be further increased.

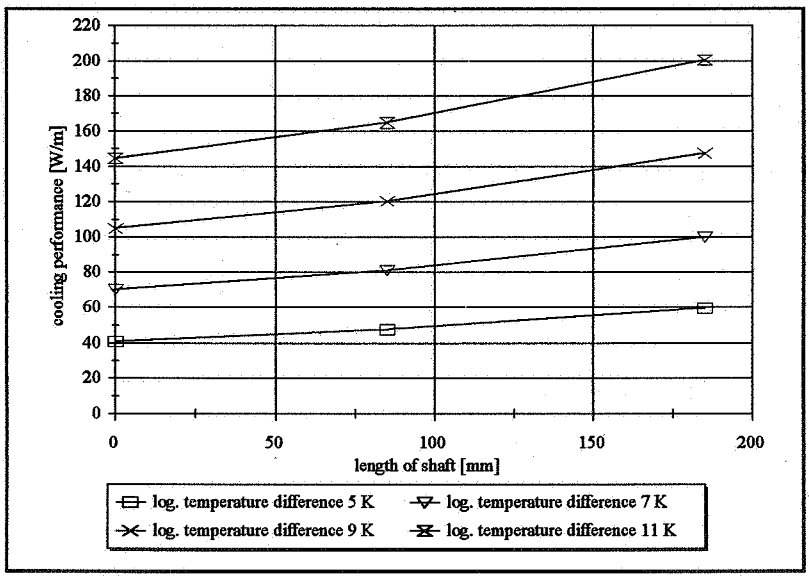

Figure 6: room characteristic of rear cooler, 200 mm distance to ceiling

Acknowledgement

The measurements were part of a research project, which was supported by the Bundesministerium für Bildung, Technologie und Forschung under the contract number 0329399A.

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High Resolution Particle-Imaging Velocimetry for Full-Scale Indoor Air Flows

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ABSTRACT

A high resolution particle-imaging velocimetry has been developed and applied to study full-scale room air flows. The system is designed to study local field quantities in occupied zones (microclimate), ventilation effectiveness, and airborne pollutant transport in the indoor environment. The system can be applied to evaluate indoor environment in typical commercial and residential settings. The technique and instrumentation have been applied successfully to study localized air flow patterns and particle concentration distribution in the indoor environment. The results of this research can be used to assess the ventilation effectiveness and energy effeciecy in rooms and buildins.

INTRODUCTION

To maintain comfort and suitable indoor air quality within occupied zones of a building, conditioned outdoor air usually is introduced by a heating, ventilating, and air-conditioning (HVAC) system. Due to different arrangements of rooms and of diffusers and returns, different air flow patterns can be obtained in rooms (Figures 1, 2, and 3). Improper air distribution patterns not only reduce ventilation effectiveness but also cause occupant discomfort even when the overall ventilation rate is sufficient. In fact, most complaints by the building occupants about discomfort are associated with non-uniformity of air temperature, large velocity gradients (air drafts), and localized microclimates in occupied spaces[1]. The room air distribution greatly affects the distribution and transport of airborne pollutants in buildings and rooms and occupant exposure to these pollutants. In many cases, emission rates of particulate and gaseous air pollutants depend on the air flow patterns near the pollutant sources (e.g., product usage, building materials, furniture, and occupants). To best address these issues, the microstructure of indoor air distribution is being studied.

In the present study, a high resolution particle-imaging velocimetry has been developed. This system consists of three sub-systems which illuminate flow structures, acquire images, and interrogate these images to obtain the velocity and particle concentration distribution.

ILLUMINATION

The illumination system is the starting point of this research. The basic function of this system is to visualize the flow patterns and/or particles. These visualized flow and particle movements can be recorded and processed. Since the light energy per unit area is inversely proportional to the total area illuminated, illumination of a full-scale room is one of the most important parts of this research. To obtain uniformity of illumination over a large space, four halogen lamps are used. The light is controlled by four cylindrical lenses installed in front of the light bulbs. Lens focal length is 90 mm. The depth of the field, δz, is calculated as[2]:

$$\delta z = 4(1+M)^2 f^{\#2} \lambda, \tag{1}$$

where M is the camera magnification, $f^{\#}$ is the camera f-number, and λ is the wave length of the illumination light. The cylindrical lens used actually changes the point light source to a line light source. To maximize light intensity, the minimum value of the f-number (maximum aperture) is used in the experiments. The thickness of the light sheet is chosen as the depth of field to minimize the background noise caused by light scattered by the particles out of the depth of field. To reduce the light scattering from the space around the lens, the lamps are placed in an air-cooled cabinet which is sealed so that light comes out only through a thin slot. Each of the four lamps consists of an aluminum chassis with a 1,500 W bulb. Bulbs are cylindrical and behave like a line source. The light sheet generated is 50 mm thick across the test room. The thickness of the light sheet is uniform.

In the present study two types of particles are used. The first type are neutrally buoyant, helium-filled bubbles 1 mm in diameter. Since these bubbles behave as a low pass filter, the flow structure is obtained only when this structure is larger than 1 mm. Flow structures smaller than 1 mm are obtained as averaged effects. The helium in the bubbles is kept at the same temperature as the room air. Bubbles are seeded uniformly in the test room and they follow flows well. Plastic microspheres are the second type of particles. These particles are heavier than air. The size distribution is from 10 to 100 μm. Tests on aerosol and dust particles with different sizes are in progress.

Since the large room size (2.53 by 3.60 by 2.43 m) requires a strong illuminating light source to make small particles visible, the parameters need to be optimized to ensure image quality. Mean exposure ($\bar{\varepsilon}$) averaged over the area of a particle image is given by[2]:

$$\bar{\varepsilon} = \frac{\lambda^2 W \int |\sigma| \, d\Omega}{\pi^3[M^2 d_p^2 + 2.44^2(1+M)^2 f^{\#2}\lambda]\Delta y_0 \Delta z_0}, \tag{2}$$

$$\sim \frac{\lambda^2 W d_p^n D_a^2}{\lambda^n d_o^2(M^2 d_p^2 + 2.44^2 d_i^2 \lambda^2 / D_a^2)\Delta y_0 \Delta z_0}, \tag{3}$$

where W is the energy of the light pulse, σ is the Mie scattering coefficient of the particle, Ω is the solid angle, D_a is the lens aperture diameter, and n is the power-law exponent describing the scattered light energy. In our case, W is the light energy transmitted to the film each time the camera shutter is opened. As particle size increases into the geometric scattering regime, the equation becomes:

$$\bar{\varepsilon} \propto \frac{W D_a^2}{d_0^2 M^2 \Delta y_0 \Delta z_0}, \tag{4}$$

which implies that light intensity, shutter speed, and lens aperture diameter control film mean exposure.

IMAGE ACQUISITION

Although the designed resolution of the system is 1 mm, the system needs to acquire high quality images of the helium-filled bubbles with 1 mm diameter. Therefore, much higher resolution is needed for particle images. If the film used has a resolution of 320 line pairs per mm (Kodak Technical Pan film) and one line pair is needed to resolve 1 mm, the film size needed is about 6 by 4 mm for a view area of 2,000 by 1,500 mm. Actually, more than one line pair are needed to obtain a high quality image of 1 mm size. The film size (36 by 24 mm) used in this research is much larger than the calculated minimum size. With the frame size of 36 by 24 mm, the camera can provide six line pairs per mm for the size of the given objective field to obtain circular bubble images.

Shutter speed, characterized by the opening time, δt, and time interval between the exposures, Δt, are the critical parameters for the image acquisition in the tests. The former is a measure of the distance particles travel when the shutter is open causing the elongated image. The latter is a measure of the distance particles travel between two sequential openings of the camera shutter, which determines the distance between particle images on the film. To obtain the best results, Δt is bounded by the optimized displacement of particle images (3 to 5 particle image diameters). The ratio $\Delta x/\delta x$ is chosen as 10 in current experiments. Therefore, the displacement of particles is $0.3d_p$ for maximum velocity in the room, where d_p is particle image diameter.

Since room flows have large inverse velocities, image-shift techniques are used to determine the direction of velocity. Shift velocity is calibrated, using both stationary images and measurement of shift velocity. Velocity of the shift (U_s) is determined by camera magnification (M) and the maximum flow velocity in the opposite direction of the shift (u_{max}). Based on the measurement of air jet velocity (0.9 m/s) near the diffuser and analysis of some test images, the minimum speed of the shift has been determined. The camera is shifted by a stepmotor which moves at a constant speed of 0.2 m/s.

When the camera shutter opens, intensity $I_{oi}(\mathbf{x}); i = 1,2, ..., n$, produces a multiple-exposed single photographic film with particle images. Each of these intensities is separated by a time interval, Δt, from a constantly illuminated light sheet of thickness, Δz. Simultaneous in-plane velocity measurements can be obtained from these images.

A sample image is shown in Figure 4. The image has been shifted to the left to solve direction ambiguity. The distribution of the bubbles is uniform and the density is high. To increase the strength of the signal, quadruple exposures are used.

Concentration has been measured using a Charge-Coupled Device (CCD) camera. Since the light intensity scattered by the particles is proportional to their concentration, the camera is not required to resolve individual particles, although it is the ultimate goal of the measurement.

INTERROGATION

The images acquired are interrogated to obtain the information needed. For velocity measurements, the images were processed by a computer system for Particle-Imaging Velocimetry (PIV)[3]. An analog image signal is sent from a CCD camera to the frame grabber, where it is digitized and stored as a 1024 by 1024 pixel image. Then the digitized image is divided into eight sub-images and processed in two array processor boards (MC860VS) each with four i860 microprocessors.

When a multiple-exposed photograph is interrogated by a light beam of intensity $I_I(\mathbf{X} - \mathbf{X}_I)$, centered at $\mathbf{X}_I$, the transmitted light intensity after the photograph is:

$$I(\mathbf{X}) = I_I(\mathbf{X} - \mathbf{X}_I)\tau(\mathbf{X}), \tag{5}$$

where $\tau(\mathbf{X})$, the intensity transmissivity of the photograph for multiple exposures equally spaced in time, is:

$$\tau(\mathbf{X}) = \sum_i \sum_{j=1}^{n} I_{oj}\tau_o[\mathbf{X} - M\mathbf{x}_i(t + (j-1)\Delta t)]. \tag{6}$$

The spatial autocorrelation of $I(\mathbf{X})$ with separation $\mathbf{s}$ is approximated by the spatial average over the interrogation spot:

$$R(\mathbf{s}) = \int I(\mathbf{X})I(\mathbf{X} + \mathbf{s})d\mathbf{X}. \tag{7}$$

It consists of five components[4]:

$$R(\mathbf{s}) = R_C(\mathbf{s}) + R_P(\mathbf{s}) + R_{D^+}(\mathbf{s}) + R_{D^-} + R_F(\mathbf{s}), \tag{8}$$

where R_{D^+} is the correlation of all earlier images shifted by $\mathbf{s}$ with subsequent unshifted images, while R_{D^-} is the correlation of all later images shifted by $\mathbf{s}$ with prior unshifted images. Both R_{D^+} and R_{D^-} can be decomposed further when the pulse separation is constant:

$$R_{D^+} = R_{D^+}^{(1)} + R_{D^+}^{(2)} + \cdots + R_{D^+}^{(n-1)}, \tag{9}$$

where $R_{D^+}^{(k)}(\mathbf{s})$ is the correlation of all earlier images shifted by $\mathbf{s}$ with subsequent unshifted images at a later time separation, $k\Delta t^5$. A similar decomposition exists for $R_{D^-}^{(1)}(\mathbf{s})$.

To determine the mean image displacement across the interrogation spot between successive pulses, the centroid of $R_{D^+}^{(1)}(\mathbf{s})$ is located by:

$$\mu_D^{(1)} = \frac{\int \mathbf{s}R_D^{(1)}(\mathbf{s})d\mathbf{s}}{\int R_D^{(1)}(\mathbf{s})d\mathbf{s}}. \tag{10}$$

If the successive pulse intervals are equal to Δt, the measured in-plane velocity is calculated as:

$$\mathbf{u}(\mathbf{x}_l) = \frac{\mu_D^{(1)}}{M\Delta t}. \tag{11}$$

For concentration measurements, the particle concentration distribution is calculated from the number of the particles per unit volume:

$$C = \frac{n}{V}. \tag{12}$$

TEST ROOM AND PROCEDURE

The Room Ventilation Simulator (RVS) at UIUC is used to study air and air contaminant distributions within ventilated rooms[6]. The RVS consists of an adjustable inner room and an outer room for controlling ambient environmental conditions of the inner test room. The outer room of the RVS (Figure 5) is an insulated, 12 by 9 by 3.6 m building used to simulate climatic conditions ranging from cold winter to hot summer around the inner test room. The outer room HVAC system, which includes an air cooling condenser, compressors, an evaporator, electric heaters, a supply fan, and a control system, is designed to provide temperatures raging from -27 to 40°C. The inner room of the RVS is modular so that different room configurations and sizes (up to 10 by 7 by 3 m) can be modeled conveniently. Cold or warm air can enter the inner test room directly from the outer room. Alternatively, the independent HVAC system for the inner room provides constantly conditioned supply air (ranging from -27 to 40°C) for the inner test room. At the same time, conditions around the inner test room can be maintained at a different temperature.

A test room, 2.53 by 3.60 by 2.43 m, has been constructed inside the RVS. The front and the left sides of the test room are clear, tempered glass to permit optical access to the interior of the room. The other two walls are wood and painted black to obtain a perfect optical environment (Figures 6 and 7). The test room has been designed and constructed to make it easy to change configurations of the room, such as the locations of the diffusers and air returns, furniture, carpet, and occupants (models or real people). The test room is supplied with air through a closed-loop fan system. A configuration with both a linear diffuser and a linear exhaust has been designed and constructed. Four supply tubes connect the high pressure side of the fan to a diffuser box mounted on top of the test room, and the exhaust is connected to the low pressure side via a sealed room next to the test room. Figure 8 outlines the diffuser box and the positions of supply tubes. Each of the supply tubes has a damper for individual control of air flow. A porous screen is mounted inside the diffuser box to make the outgoing velocity profile uniform (Figure 9). The damper positions are adjusted to ensure a uniform velocity. Velocity profiles for different damper positions are shown in Figure 10.

The HVAC system is turned on for 30 minutes for each air change rate to ensure that the room air flow reaches a steady state. The light is switched on only during image acquisition, which only takes 0.5 min. If a large number of frames are needed for statistical analysis, the process is split into a few short sessions. Room temperature is maintained at 21±0.5°C (70±0.9°F) throughout the tests.

RESULTS AND DISCUSSIONS

Experiments have been conducted with air change rates of 5 and 10 air changes per hour (ACH). Figure 11 shows the size and position of the objective plan in the test room at 5 ACH. Figure 12 shows the correspondent velocity distribution for 5 ACH, and Figures 13 and 14 show the objective plan and velocity distribution for 10 ACH. For 10 ACH, the camera was placed closer to the objective plan which then gave a smaller view field and a lower relative bubble density.

Depending on the structures of room air flows and seeding particle density, 2,000 to 8,000 vectors can be obtained for the size of the given two-dimensional space. The small scale structures of air flows can be seen clearly in Figures 12 and 14. This capability is critical for the study of diffusion effect dominated by small structures in the indoor environment, which in turn is important for particle transport.

The results also show that the air flows in most parts of the room are slower than 0.1 m/s at a normal range of air change rates (~ 3 to 10 ACH). There is no instrumentation commercially available to measure slow velocity in this region. The capability to measure such low velocities gives the described system another unique feature which will be useful in helping aerosol manufactures and other industries to improve their products.

A contour plot of particle concentration with 5 ACH is shown in Figure 15. Although the concentration distribution of the particles appears to be complicated, the particles are denser in the lower part of the room and more dilute in the upper part of the room.

CONCLUSIONS

A non-intrusive, whole-field measurement technique for studying indoor air quality has been developed. Two-dimensional structures of full-scale room air flows and particle concentration have been determined. The technique shows great potential for helping us to understand ventilation effectiveness and indoor air quality since mixing efficiency, fresh air delivery rate to occupied zone, and transport of particulate air pollutants can be calculated from instantaneous velocity and concentration distribution. It also can serve as a bench mark test for global measurement techniques, such as tracer gas techniques, and computational fluid mechanics models.

ACKNOWLEDGEMENTS

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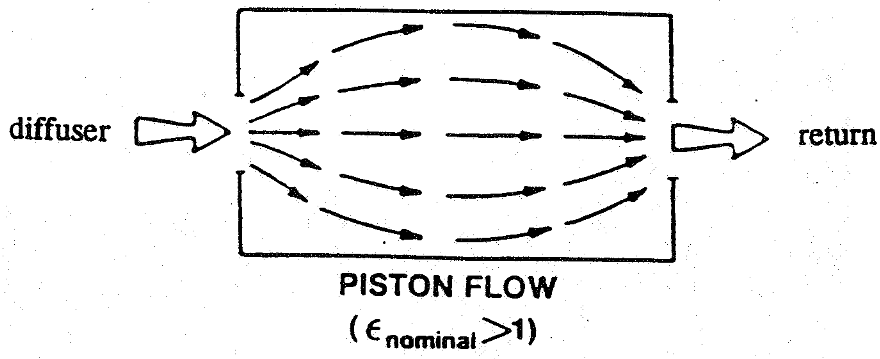

Figure 1. Piston flow. (Source: U.S. Department of Energy.)

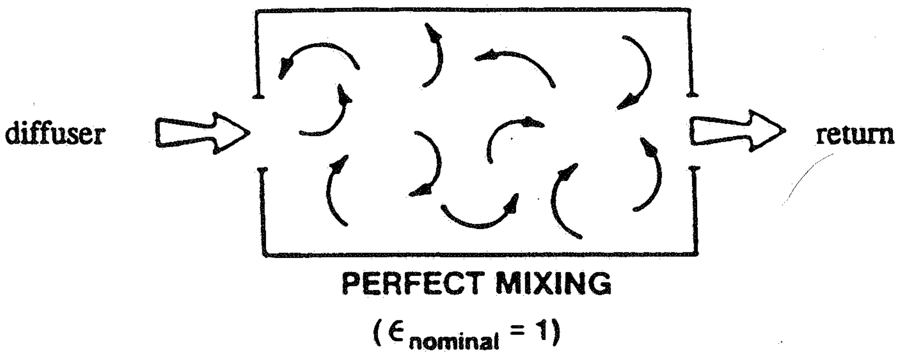

Figure 2. Perfect mixing. (Source: U.S. Department of Energy.)

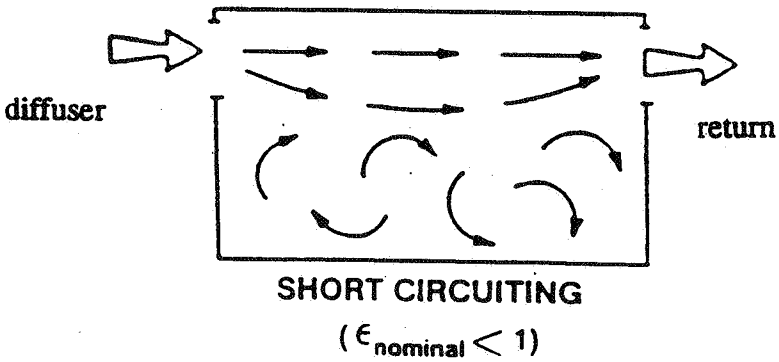

Figure 3. Short circuiting. (Source: U.S. Department of Energy.)

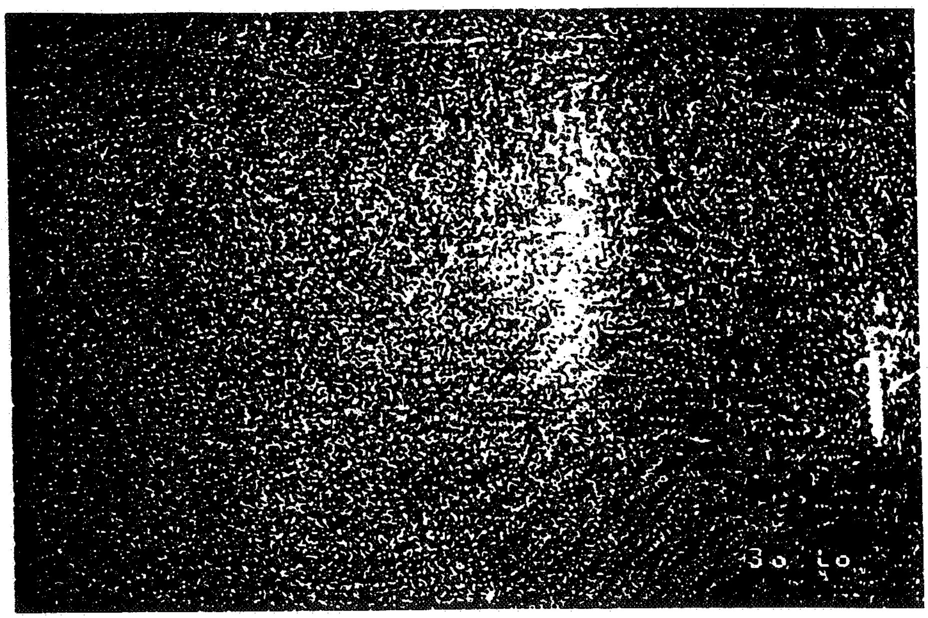

Figure 4. Room air flow with bubbles. (Air change rate = 5 ACH.)

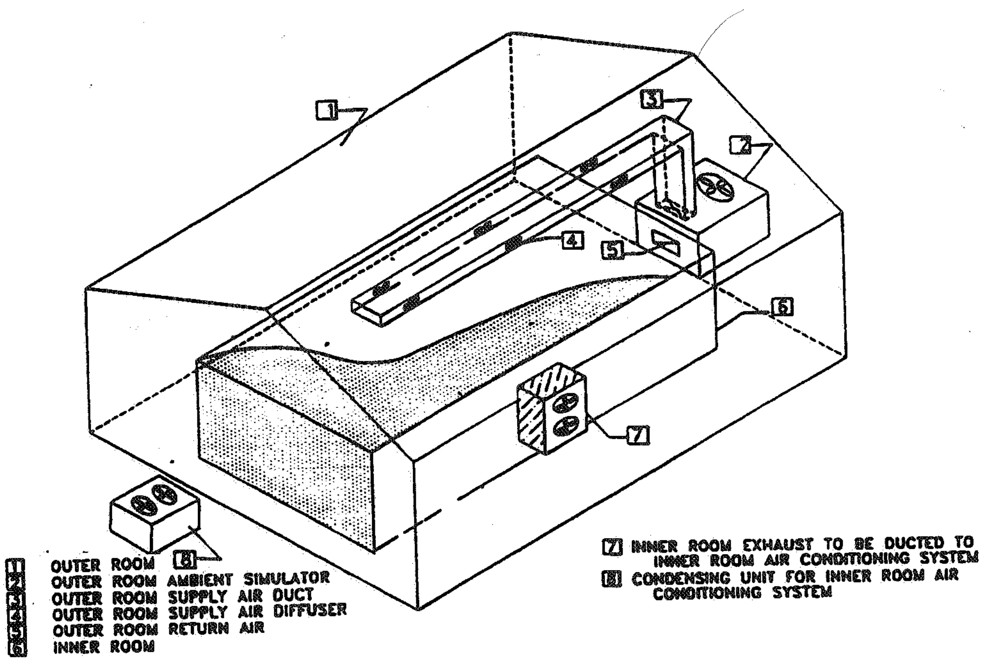

Figure 5. Room Ventilation Simulator.

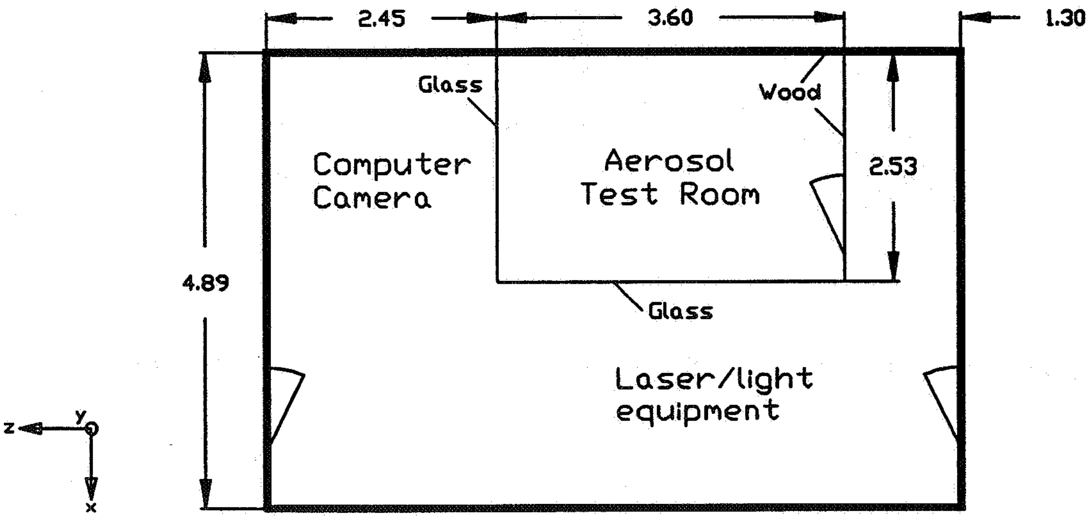

Figure 6. Plan drawing of the inner room and the test room. (All dimensions in meters.)

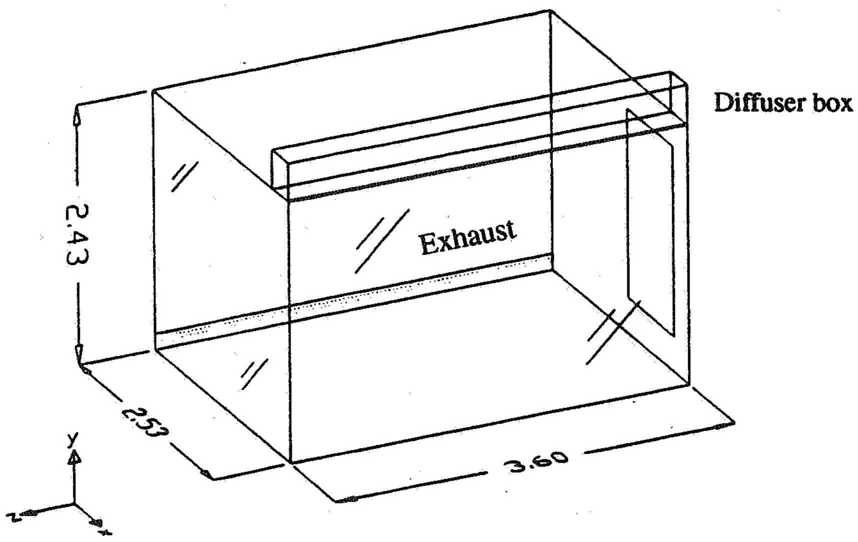

Figure 7. Dimensions and configuration of the test room. (All dimensions in meters.)

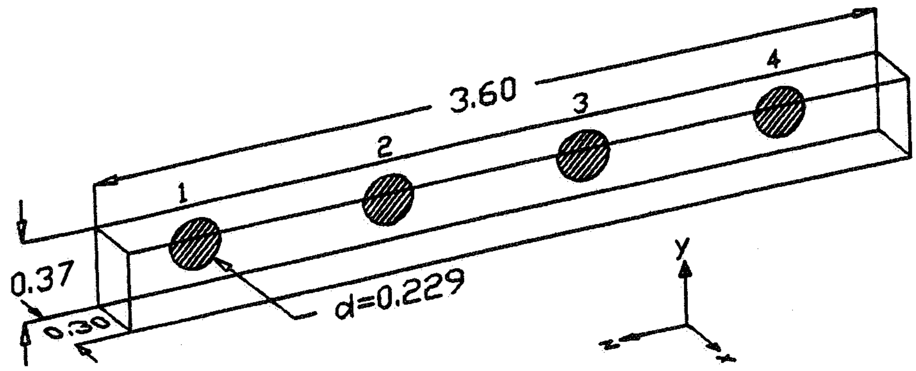

Figure 8. Diffuser box for providing the room with a two-dimensional air flow. (All dimensions in meters.)

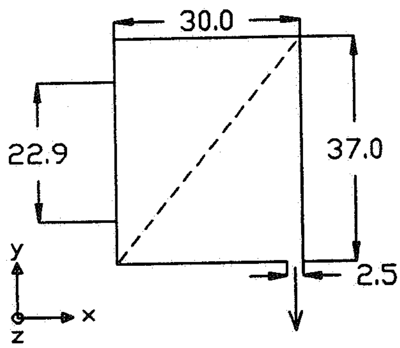

Figure 9. Cross-section of diffuser box. (All dimensions in centimeters.)

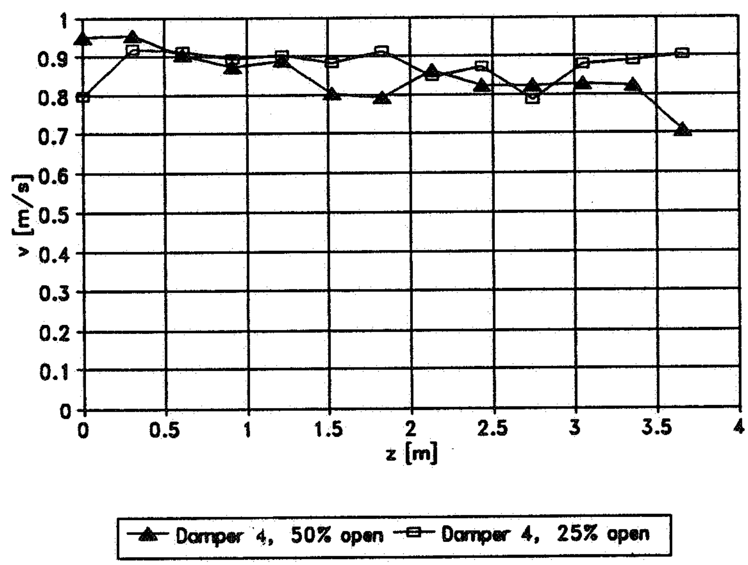

Figure 10. Inlet velocity profiles along the z-axis for two positions of the left damper.

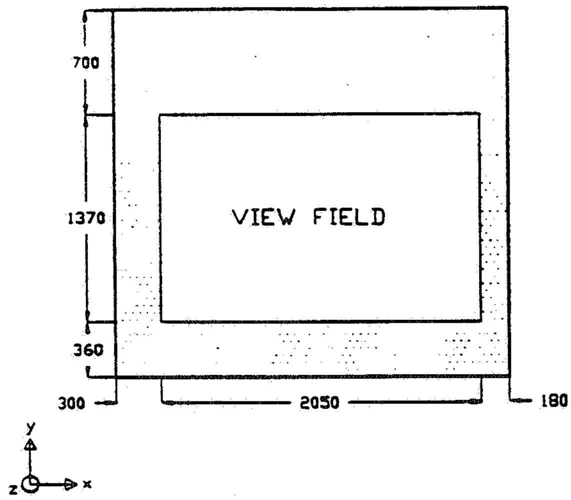

Figure 11. Size and position of the objective plan in the test room for air change rate = 5 ACH. (All dimensions in millimeters.)

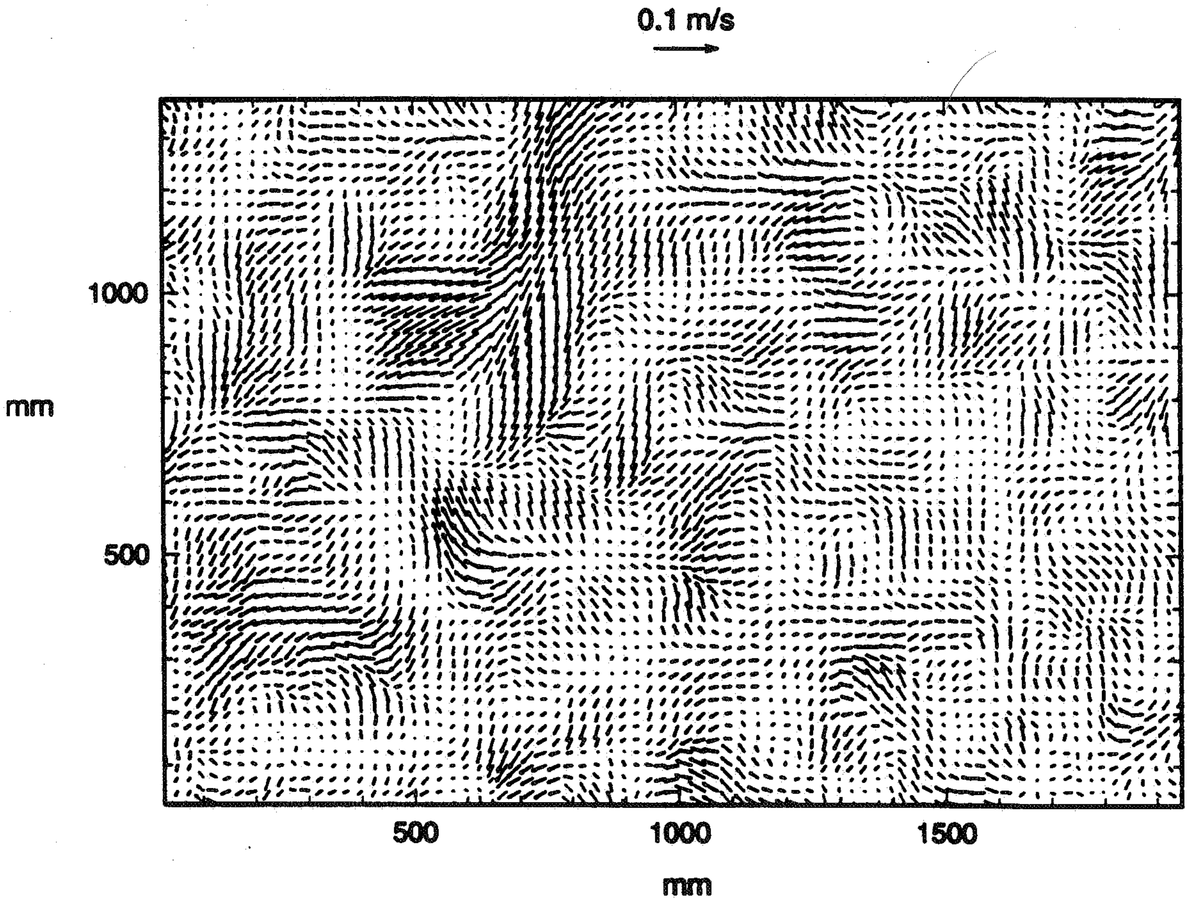

Figure 12. Vector map of room air flow for air change rate = 5 ACH.

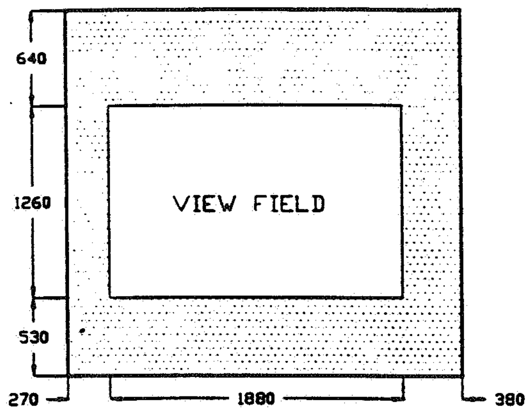

Figure 13. Size and position of the objective plan in the test room for air change rate = 10 ACH. (All dimensions in millimeters.)

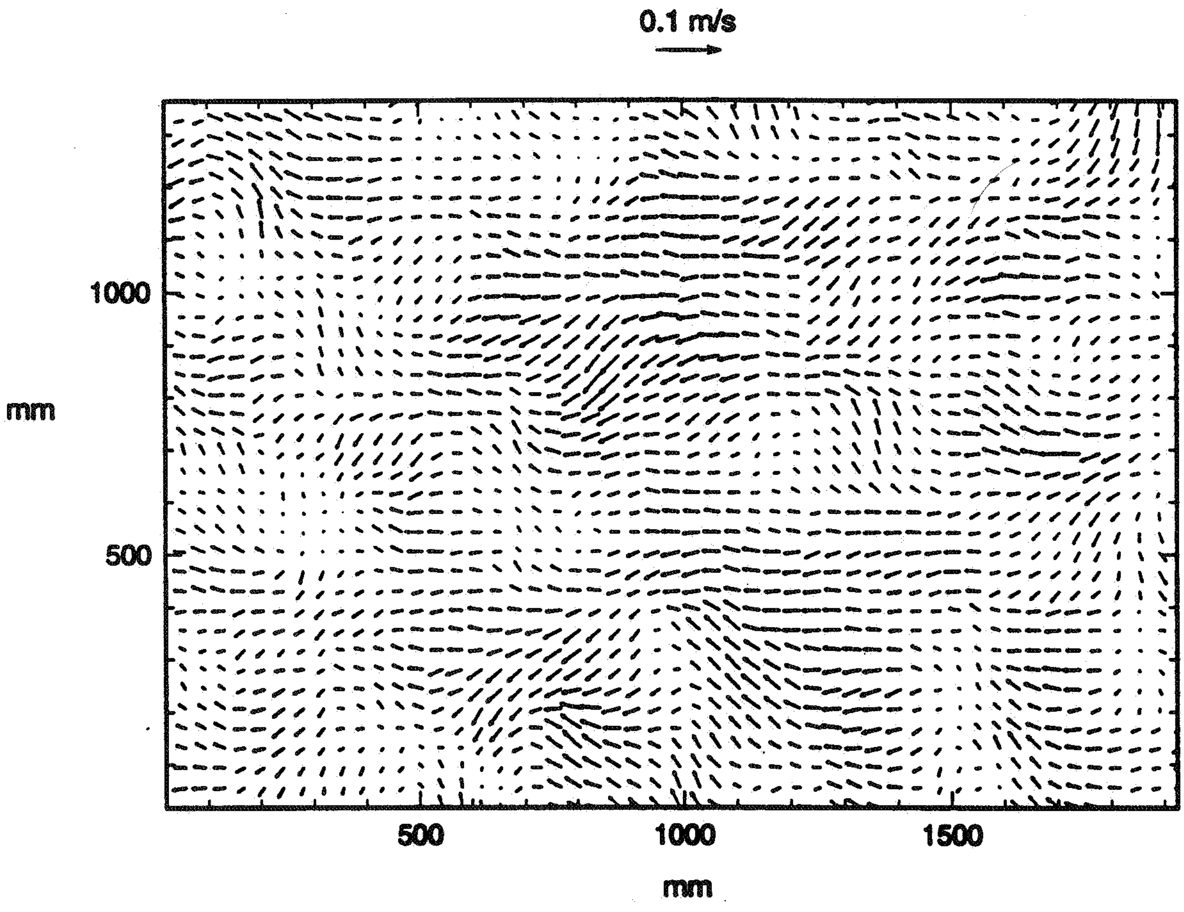

Figure 14. Vector map of room air flow for air change rate = 10 ACH.

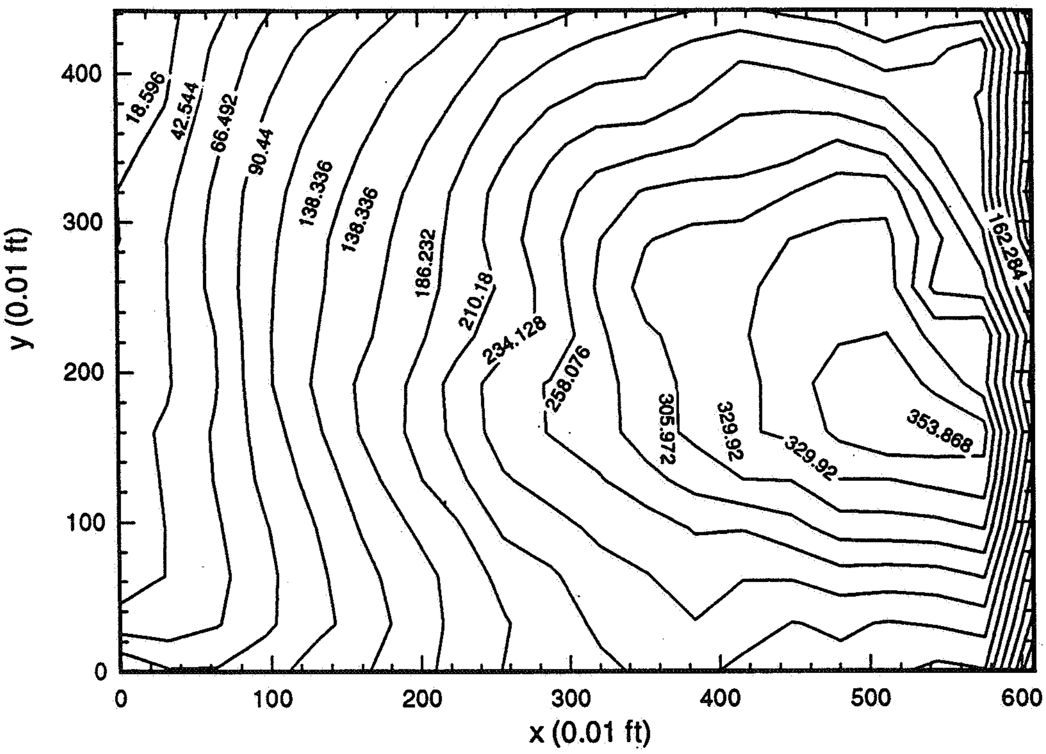

Figure 15. Contour plot of particle concentration distribution in cubic feet (×1000) for air change rate = 5 ACH.

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Reducing the Permeability of Residential Duct Systems

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REDUCING THE PERMEABILITY OF RESIDENTIAL DUCT SYSTEMS

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SYNOPSIS

In this paper, we investigate the potential of an aerosol-based technique to significantly reduce the leakiness of residential air distribution systems (ADS). The first part is dedicated to a short review of theoretical analyses of particle transport and deposition in an ADS as well as particle removal in the leaks. The purpose of this review is to pre-determine the ranges of the flow rates, pressure differentials and miscellaneous characteristics of the particles that would allow plugging of the leaks in a relatively short time. The remainder of the paper deals with our experimental investigation and includes a description of experiments performed to assess the feasibility of the technique. We found that an aerosol alone made out of a liquid suspension of a vinyl polymer can plug 16 cm^2 of Effective Leakage Area in a branch in less than 30 minutes. Based on our theoretical and experimental results, we designed a field device and performed laboratory experiments on a small-scale duct system. We showed that with this portable unit, the Effective Leakage Area of a typical duct network in California can be reduced by about 80%.

LIST OF SYMBOLS

C_m	aerosol mass concentration [kg/m^3]
D	duct diameter [m]
$d()$	differential [-]
d_p	particle diameter [m]
e	duct wall thickness [m]
h	leak-width [m]
i	integer [-]
L	duct length [m]
P	penetration [-]
Q	flow rate [m^3/s]
Re	Reynolds number [-]
SE	sealing efficiency [-]
Stk	Stokes number [-]
t	time [s]
t_i	i-th characteristic sealing time [s]
t_{res}	residence time in the separation zone [s]
U_s	velocity upstream of the slot at $y=y_s$ [m/s]
u	velocity along x [m/s]
v	velocity along y [m/s]
v_s	bulk velocity through the slot [m/s]
w	thickness of the seal [m]
x	horizontal coordinate [m]
y	vertical coordinate [m]
y_s	height of the dividing suction streamline [m]

Greek symbols:

ΔP	pressure differential [Pa]
η	deposition efficiency [-]
ν	kinematic viscosity of the fluid of interest [m^2/s]
ρ	density [kg/m^3]
τ	particle relaxation time [s]

Subscripts and superscripts:

o	at t = 0
D	pertaining to duct
f	pertaining to the fluid of interest
p	pertaining to particle
ref	at the reference pressure differential
s	pertaining to slot
$seal$	pertaining to particle build up
$\overline{\ }$	average value

Abbreviations:

ADS	Air Distribution System
ELA	Effective Leakage Area [m^2]

1. INTRODUCTION

During the past five years, research has quantified the impacts of residential duct system leakage on HVAC energy consumption and peak electricity demand. A typical California house with ducts located in the attic or crawlspace wastes approximately 20% of heating and cooling energy through leaks and draws approximately 0.5 kW more electricity during peak cooling periods (Modera, 1993). Besides, given that 25% to 75% of the leaks are not accessible (Robison and Lambert, 1989), conventional technologies such as using duct tape or mastic are often not satisfactory. Existing remote sealing technologies have been examined (e.g. introducing a rolling mechanical cart, or unfolding a cylinder in the duct network). Although the application of these techniques might be of interest for some pipe networks (the cart was patented and used for gas pipes (Smith, 1983) and the unfolding cylinder is used to seal large underground pipes), the complexity of a residential duct system does not allow straightforward application of these technologies. Our attention is thus focused on the use of aerosol sealants, the versatility of this technique allowing us to deal with bends and bifurcations without significantly affecting its performance.

The proposed concept involves blowing an aerosol through a duct system to seal the leaks from the inside, the principle being that the aerosol particles would deposit in the cracks of the ductwork as they try to escape because of the pressure differential. The reader may find similarities between this internal-access sealing and the after-market automotive sealants for tires although proper use of those sealants is rather restrictive (e.g. it requires the spinning of the tire after foam injection). One company was marketing a vinyl polymer suspension to immobilize dust in residential air-distribution systems. The company used a nozzle-produced jet to entrain air and generate an aerosol, and to blow the aerosol through the ducts. Filter paper was used in all the registers to block the fog from directly entering the house. The sealing effectiveness of this aerosol was tested at Lawrence Berkeley Laboratory for naturally occurring and for artificially created leaks in a duct system. Although the pressure-driven flow of aerosol was a good vehicle for transporting the sealant through the system, the particles' size and type as well as the degree of control of pressure and flow rate were apparently inappropriate for sealing the typical leaks encountered. Thus, it appeared key to better understand aerosol transport in an ADS as well as particle deposition in the leaks.

This papers begins with a theoretical approach to the problem which includes: a) aerosol transport in an ADS and b) particle deposition in leaks. The second part of this paper is dedicated to our experimental investigation and describes the different series of experiments we did in order to obtain proof-of-concept and assess the applicability of this technique *in situ*. The third part deals with the design of a prototype field device and its testing on a small-scale ADS and in a house.

2. OVERVIEW OF THEORETICAL INVESTIGATIONS

To develop a successful technology for remotely sealing leaks with an aerosol, we need to solve two fundamental problems: how to transport aerosol particles from the injection to the vicinity of the leaks, and how to have them deposit in the cracks to be sealed. As for the first problem, literature was reviewed and we subsequently developed a simplified model that can predict particle deposition in a duct system as a function of the total air flow rate and the miscellaneous characteristics of the aerosol (Carrié, 1994).This model is based on Agarwal's (Agarwal, 1975) work on turbulent deposition of particles in a straight tube. A similar approach to that of Balásházy (Balásházy *et al.*, 1990) is used to quantify aerosol penetration in bends, tees and wyes. To solve the second problem, which is key to a) assessing the feasibility of the technique and b) optimizing the sealing effectiveness, we started with an identification of the different types of leaks. We were able to divide the leaks in two major categories, which are: a) simple holes in the duct wall (Type-I, Figure 1), and b) annular channels that are characteristic of leaks at duct joints (Type-II, Figure 1).

Because of the complexity of the assessment of particle behaviour in such configurations, we limited our detailed investigation to two-dimensional Type-I leaks (Figure 2). Even though restricted, this research presented the advantage of being "experimentally verifiable" while providing us with some understanding of aerosol collection in leaks. In our approximate analysis, we found that the deposition efficiency, defined as the ratio of the mass flux of particles that deposit in the crack due to impaction versus that of particles that are originally on streamlines that exit through the orifice ($y_p \le y_s$), may be expressed as follows (Carrié and Modera, 1994) :

$$\eta = \frac{\tau v_s^2}{y_s} \frac{e}{v_s h} = Stk \frac{e}{h} \tag{1}$$

The theoretical limits of the model are the following:

1. $\mathrm{Re}_p \sim Stk \dfrac{v_s d_p}{\nu} << 1$
2. $Stk \sim \dfrac{\tau}{t_{res}} << 1$
3. $\dfrac{\tau v_s}{e} >> 1$

We further assumed that the only geometrical parameter that varies as impaction occurs is the slot-width. Thus, if the aerosol is monodisperse (or in general, if the mean deposition efficiency ($\overline{\eta}$) can be evaluated over the particle size range) and if the sticking probability of the aerosol is 1 (ideal case), the rate at which the leak-width decreases is:

$$-\frac{dh}{dt} = \overline{\eta} \times \frac{h}{\rho_{seal}\, w} \times v_s \times C_m \tag{2}$$

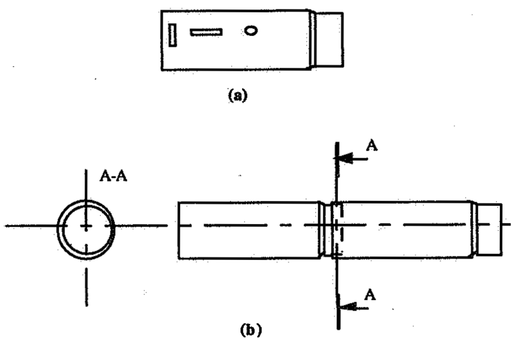

Figure 1: Identification of the different types of leaks; (a) Type-I leaks; (b) Type-II leak.

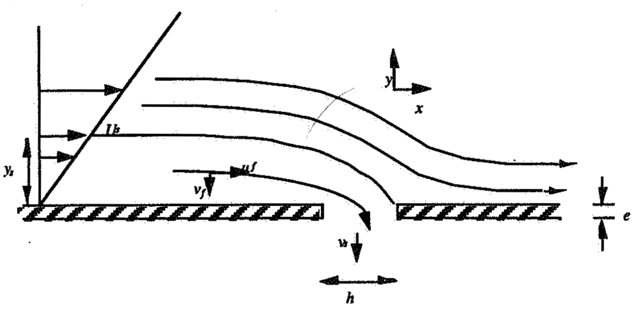

Figure 2: Uniform shear flow approaching a two-dimensional slot (see (Thomas and Cornelius, 1982)).

Videotaping the sealing process to monitor leak-width as a function of time under varying conditions, we found good agreement between theoretical and observed leak behaviours for low pressure differentials ($\Delta P \le 60$ Pascals). Assuming that the deposition efficiency is well assessed by this model in a typical duct system, we developed simplified calculations to approximate key parameters that will affect the sealing time. Given a two-dimensional slot at a distance L from where the concentration C_m is measured, the leak-width decrease rate is given by Equation (2), where the left hand side is multiplied by the aerosol penetration P over the distance L. On this basis, we define the sealing efficiency[1]:

$$SE = \overline{\eta} \times P \tag{3}$$

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[1] Calculations of P are detailed in (Anand and McFarland, 1989)

Also, the time required to obtain a slot width decrease of $\frac{1}{i}h_o$ (i-th characteristic time, t_i) appeared to be an interesting indicator of the sealing effectiveness. Assuming that the sealing efficiency computed by Equation (3) remains constant within that time frame we obtain:

$$t_i = \frac{\rho_p e}{SE \times v_s \times C_m} \times \ln\left(\frac{i}{i-1}\right) \tag{4}$$

where v_s and y_s (that are needed to compute SE) can be approximated with the following equations:

$$v_s = 0.6\sqrt{\frac{2\Delta P}{\rho_f}} \quad (5) \qquad\qquad y_s = D\sqrt{50.63\frac{\mathrm{Re}_s}{\mathrm{Re}_D^{7/4}}} \quad (6)$$

and the variation of SE over t_i is:

$$\frac{\Delta SE}{SE} = \left(\frac{i}{i-1}\right)^{3/2} - 1 \tag{7}$$

Obviously this i-th characteristic time is all the more realistic as i is sufficiently large so as to ensure that SE is constant over t_i (for i = 2, $\frac{\Delta SE}{SE} = 0.41$; for i = 20, $\frac{\Delta SE}{SE} = 0.08$).

Calculations to assess the sealing efficiency and the i-th characteristic time can be performed with the help of a spread-sheet program (see Table 1).

Table 1: Sample calculations of the penetration, the sealing efficiency, and the i-th characteristic sealing time (D=0.15 m; L=10 m; h=3 mm ; e=0.6 mm; ΔP=40 Pa; C_m=5.0 mg/l; d_p=10 μm).

Flow Rate [m³/h]	Penetration [-] (in %)	Sealing Efficiency [-] (in %)	t_2 [s]	t_{20} [s]
14.6	32.7	0.3	6558	485
29.3	57.2	0.8	2046	151
43.9	68.9	1.4	1191	88
58.5	75.6	2.0	844	62
73.2	80.0	2.6	656	49
87.8	83.0	3.1	539	40
102.4	85.2	3.7	459	34
117.1	86.9	4.2	401	30
131.7	88.2	4.8	356	26
146.3	89.2	5.3	321	24

3. LABORATORY EXPERIMENTS

Because we previously assumed that all of the deposited particles contribute to a decrease of the opening size, an important aspect of the issue of particle deposition in the leaks that remains is to understand how the particles are going to stick to the duct wall and eventually "build a bridge" between the boundaries. Vinyl plastics seem appropriate to achieve this goal since they are widely used to smooth and waterproof surfaces. However, it is key to understand that particles need to hold their shape in order to "build a bridge". If the particles are too deformable, they will tend to spread over the duct wall preventing any particle build up.

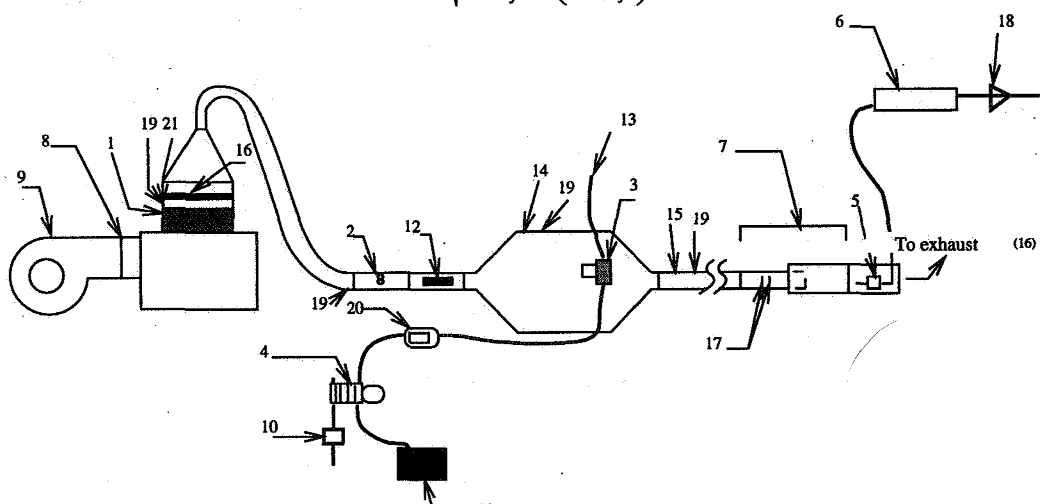
A preliminary approach to the problem (see (Carrié *et al.*, 1994)) enabled us to show that by drying our aerosol we obtain solid particles that can stick to the duct wall and to one another and thus allow the decrease of the leak size. Enlightened by this preliminary research, the apparatus described in Figure 3 was designed. It includes: 1) a desiccant drying section, 2) a vane-anemometer flow measurement device, 3) an assortment of commercially available sprayers, 4) a peristaltic pump to control and measure the liquid flow rate to the spray nozzle,

5) a sampling unit to measure aerosol concentration, 6) a mass-flow controller network to ensure isokinetic sampling and 7) precisely constructed Type-I and Type-II leaks. The data from all of the measurement devices described in Figure 3 is sent to an IBM-PC and stored electronically.

With this apparatus, we found that round Type-I leaks as large as 3 mm in diameter could be sealed in 10-20 minutes. 3 mm Type-II leaks were found to be partly sealed during the same experiments. These results were all the more promising as they were repeatable and could be obtained with a two-branch duct network as well (see (Carrié *et al.*, 1994)).

In order to quantify the sealing effectiveness, it was further decided to measure the permeability of the network before and after injection of the aerosol. To this end, the Effective Leakage Area (ELA) (see (Modera, 1993)) commonly employed to measure the leakiness of a building envelope or an ADS was used. The equation linking the leakage flow rate to the pressure differential is given by:

$$Q = ELA\sqrt{\frac{2\Delta P_{ref}}{\rho_f}}\left(\frac{\Delta P}{\Delta P_{ref}}\right)^n \tag{8}$$



1: Desiccant drying section
2: Vane anemometer
3: Sprayer
4: Peristaltic pump
5: Sampling unit
6: Mass flow controller network
7: Section of leaks
8: Damper
9: Fan
10: Mass flow meter
11: Pressure tank
12: In-line Heater
13: To compressed-air
14: Injection chamber
15: 15 cm diameter duct
16: Filter
17: Slots
18: Pump
19: Thermocouple
20: Temperature controller
21: Relative humidity sensor

Figure 3: Schematic diagram of the experimental apparatus in use to quantify aerosol sealing effectiveness.

Thus, by artificially creating a pressure differential in the test section and by measuring the leakage flow rate, one can calculate the ELA. The physical meaning of the Effective Leakage Area is that, at the reference pressure differential, the flow rate passing through the leaks would be the same as that leaking through a sharp-edged orifice of this same area under the same pressure differential. The reference pressure differential is usually set to 4 Pascals in building science applications in the USA, which is typical of wind pressure and stack effect in that country (Sherman *et al.*, 1984). However, because the pressure differential across duct leaks is significantly higher than 4 Pa when the system is in operation, characterizing duct leaks at a reference pressure of 25 Pa is more appropriate (Modera, 1993). Therefore, the 25-Pa characterization is utilized in this paper. Precision errors are calculated with the following equation:

$$\frac{\delta ELA}{ELA} = \delta\left(\ln\left(Q_{ref}\right)\right) \tag{9}$$

where $\delta(\ln(Q_{ref}))$ is estimated with the standard deviation of the extrapolation of $\ln(Q)$ at the reference pressure differential.

Sealing tests showed that with our unit (Figure 3) we could seal up to 88% of the Effective Leakage Area of our system in 45 minutes (see (Carrié *et al.*, 1994)). A preliminary approach to the design of a field device showed that an effective way of preventing aerosol entry into the dwelling without adversely affecting the sealing process is to block the ends of the ducts (rather than filtering particles from the flow exiting the registers). This way, filtration is complete and it avoids the additional requirement of having to place filters (that would eventually clog) at the registers. Tests were performed with the apparatus described in Figure 3 under the conditions quoted above (i.e. the ends of the ductwork were sealed). We found that the ELA of the ADS could be reduced by more than 90% in 20 to 45 minutes . In addition, the initial air flow rate could be lowered down to 40 m^3/h per branch and still provide us with sufficient aerosol penetration and significant ELA reduction. Plugging an equivalent of 16 cm^2 is possible in less than 30 minutes according to our experiments (see (Carrié *et al.*, 1994)). However, larger initial ELAs require more time. Furthermore, we noticed that an in-line heater was sufficient to lower the water content of the aerosol particles, eliminating the need of a desiccant drying section.

4. IN SITU SEALING APPARATUS

The portable unit that we have developed includes: a) a blower (Centrimax CXH 33A2B), b) an in-line resistance heater and c) a commercially available spray nozzle (SUE15B manufactured by Spraying Systems Co.). For both safety and control purposes, a thermostat and a pressure switch are also included in the apparatus. Injection is realized by pressurizing a liquid container to a specified value, and connecting the container to the spray nozzle[2]. Compressed-air is supplied with a separate portable unit.

Because the ends of the ducts are blocked when aerosol injection is performed, the fan curve is of paramount importance for designing the field device. Indeed, as the cracks are being plugged, the pressure in the system increases. When the pressure exceeds a pre-set value (mainly determined by the maximum pressure ratings of the ducts), the pressure switch disables the unit. Thus, once the stop-pressure is set, the final ELA[3] may be determined as a function of the flow exponent for a given fan curve. We found that our blower should give satisfactory results[4] when operated successively at full-speed (speed 1) and at reduced-speed (speed 2) with a stop-pressure set at 300 Pascals. At full-speed, the fan ensures proper aerosol transport throughout the system and will significantly reduce the ELA. Further reduction of the ELA has to be obtained by reducing the air flow rate (and thus the pressure) through the ADS. This way, particle transport may not be as effective; however, the sealing process however can be continued without exceeding the stop-pressure.

We built the small-scale ADS to perform preliminary laboratory experiments. Although our unit was tested several times, only one complete test has been performed to date. The results of this experiment (in terms of ELA) are listed in Table 2. As expected, the pressure increased as soon as the aerosol injection began, and reached the stop-pressure (300 Pascals) in about 5 minutes. Then, we switched the unit to reduced flow and pressurized the liquid container to its new specified value. As the leaks were being sealed at reduced flow, the pressure differential in the duct system increased from 40 Pascals to 300 Pascals in about 15 minutes.

As for the types of leaks we had in the system, most of the joints between the different ducts were not taped at all; we also voluntarily made cracks at the registers boots; finally, some simple holes or slots were included in the ADS. The plugging of relatively large-size cracks

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[2] Note that this pressurization sets the liquid flow rate through the nozzle.

[3] Assuming that the stop-pressure is reached.

[4] "Satisfactory results" assume a final ELA on the order of 20-25 cm^2. Given that the ELA (at 25 Pa) of a residential ADS is typically on the order of 180 cm^2, that would constitute an 85% decay.

(~ 1-3 mm) was observed. Although significant aerosol collection was observed in the largest holes (~ 5-10 mm), these were found to be difficult to seal.
Our results may be summarized as follows:

1. The dimensions of the injection chamber of the original apparatus (see Figure 3) can be reduced without significantly affecting the sealing process.
2. The use of a variable-flow fan seems necessary.
3. Any type of leak can be sealed provided that they are sufficiently small (~ 1-3 mm).
4. This technique appears efficient for straight sheet metal ducts, sheet metal ducts with bends, and the plastic flexible ducts often encountered *in situ*.

Table 2: Measured and predicted ELA† before and after aerosol injection with field device in the laboratory‡.

	ELA [cm^2]	Precision Errors (in %) [-]	Exponent [-]
Before aerosol injection (measured)	124	6.7	0.55
After aerosol injection (measured)	26	5.6	0.51
After aerosol injection (predicted)	25	-	-

† ELA at 25 Pascals. ‡ Initial pressure: 170 Pa; Stop-pressure: 300 Pa.

The first field test was performed in a single-family house in Berkeley, CA in 1994. The device was found to seal approximately 60% of the leakage in the duct system within 15 minutes using about $6 worth of sealing material. However, the set-up time far exceeded the sealing time. In addition to attaching the device to the HVAC system and simple leakage measurements, these field tests included measurements of particle and volatile organic compound (VOC) concentrations before and after sealing. After the sealing process, total suspended particles were found to decrease and there was no detectable change in VOC concentrations. These results appear extremely promising. We may however propose a few improvements to the present apparatus. First, we noticed that the aerosol jet tended to directly impinge on the walls of the injection chamber. Slightly modifying the nozzle angle may help avoid this problem. Second, with minor changes, our device could measure leakage of the duct system before and after injection, eliminating the use of additional hardware.

5. CONCLUSIONS AND FUTURE EFFORTS

The practical objective of this research was to develop an aerosol-based technique to remotely seal leaks in residential air-distribution systems. Our theoretical investigations lead to the design of a laboratory apparatus capable of sealing 1 to 3 mm cracks in 10 to 20 minutes. Based upon a number of laboratory tests with this relatively large injection unit under controlled conditions we can conclude that this technique is capable of remotely sealing leaks.
Based on our theoretical investigations and our expertise, we constructed and tested a prototype field device. The results of laboratory experiments on a small-scale duct system to assess the performance of our unit under realistic conditions combined with a field test in a single-family house in Berkeley, CA allow us to conclude that *in situ* sealing of residential duct systems is achievable with this unit.
In the future, we will explore the longevity of the seals by tracking the air tightness of the Berkeley house duct system. We also plan to focus in the future on accelerated laboratory testing of the seals produced by the aerosol, designing reusable, quick-installation seals for the registers and HVAC heat exchangers, followed by larger-scale field testing, construction of a second prototype sealing apparatus, and related activities required for the technique's commercialization.

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A Report on the Radon Measurement in a Single Family House

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A Report on the Radon Measurement in a Single Family House

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Summery: This paper presents the results of a series of measurements made in an occupied family house. Long and short-term measurements of the concentration of radon gas in the cellar and other living areas of the house were carried out. Subsequently a mechanical ventilation system was installed in the cellar and operated in both supply and extract mode with different air change rates. Further measurements of radon concentrations were carried out along with other physical parameters. The results of these tests are reported in the paper and indicate that the rate of ventilation is important in reducing the concentration of radon gas in the dwelling.

1 Definitions/Terms

Fan System / Mechanical Ventilation System: In this paper it refers to the system installed in the cellar of the house. It consists of an in-line centrifugal fan, a speed controller, connecting ducts and a sound attenuator.

Air Change Rate (ACH): It is defined as the total air volume entering the cellar per hour divided by the effective volûme of the cellar. Negative ACH means that the mechanical ventilation system was extracting air from the cellar, while positive ACH means supplying.

Hence positive ACH does not always mean the cellar is being pressurised. For example in natural ventilation state, the cellar is de-pressurised by stack effect, but ACH is positive because the air is introduced into cellar from outdoors.

Average Concentration in Living Area (ACILA): In the paper this parameter is used to represent the radon level in the house. It is the average of all levels measured in the seven rooms which where regularly occupied.

Radon Reduction Rate / Reduction Rate: It is defined as the difference in radon levels measured before and after the application of a mitigation procedure divided by the level before the application.

2 Introduction

Spaces/voids in buildings' substructures, such as basements, crawl-spaces or cellars have been identified as areas where radon gas can enter a building . The mechanisms for radon entry and mitigation measures have been studied and solutions developed. Two remedial measures, pressurisation and de-pressurisation have been shown to reduce the concentration level in existing single family houses with a basement or crawl-space. (1)

• The mechanisms of these two measures are simple. In neutral condition houses are de-pressurised because of the stack effect created by indoor - outdoor temperature difference in winter. The radon in the soil is sucked into the substructures through the floor or walls and then enters the living areas via cracks or staircase. Using the fans in the pressurisation mode increases the pressure in basement hence reduces the radon entry rate, while de-pressurisation dilutes and removes the radon gas and reverses the differential pressures of routes of air flow linking substructure with the rest of building although it may introduce more radon into substructure.

Cellars, unlike the other two sub-structure spaces, have their own characteristics. In the case being considered the cellar is beneath only part of the house. In this case the cellar may not be the only source of indoor radon. The possible cracks in the floor above the soil may also be ways for radon to enter.

A preliminary study using computer modelling indicated that both approaches, either pressurisation and de-pressurisation in the cellar were effective in reducing the radon problem (2). The model, however has not been verified by any physical validation, which requires a series of real site measurements.

It is therefore the intention of this paper to report on the physical modelling carried out in an occupied house and to indicate the benefits or otherwise of varying the air flow rates into/ out of the cellar.

3 The House

3.1 General Description

The building being investigated is a single family end terrace house about 90 years old. It is built of stone and built along an east-west axis. It has three stories and a cellar under part of the ground floor. On ground floor there is a lounge, hallway, dinning room and a kitchen. The timber floor in the hallway is not covered by carpet, and some cracks are visible. The main bedroom is to the north on the first floor, there is a further bedroom, a toilet and a bathroom. There are two bedrooms on the second floor.

There is a conservatory to east facet of the house and access to the front door is through this structure.

All windows are double glazed. Two external doors face east. The one of main entrance links the conservatory and hallway, the other links outdoor and kitchen. Cracks between doors and their frames can be seen. The door to cellar is not well sealed.

3.2 The Cellar

The cellar is principally used as a store and workshop. Usage is very irregular. It contains both gas and electricity meters, water service pipes and electrical distribution.

The cellar is about same size as the lounge directly beneath it under ground level. Access to it is via a stone staircase which is under a wooden staircase (the latter from ground floor to the first floor).

One window is single glazed on the north wall below ground level. Half of the window has been replaced by the mechanical ventilation system ductwork. The walls are stone and painted white. The floor is made of stone slabs

3.3 The Floor on Ground-floor

The floor on ground-floor plays an important role in radon problems in this type of building In this house the floor of dinning room and kitchen is concrete. The age and depth of the concrete is unknown. The house owner recently replaced the floor by a concrete of 4 inches minimum thickness incorporating a plastic membrane. The lounge floor over the cellar is a normal timber on joists construction. The floor of the hallway is very interesting. It is timber, and partly over soil and rock and partly over a door way to the cellar. Cracks on that floor can be easily seen, which are believed to link with both the cellar and the soil.

3.4 The Mechanical Ventilation System

A mechanical ventilation system was installed in October 1984, which consists of an in-line central fan, a controller, an attenuator and two pieces of connecting duct. The system was switched on at a low speed extracting air out of the cellar until the monitor was available in February 1995. Two reasons were as follows.

Firstly, according to the computer modelling, de-pressurisation seemed more efficient than pressurisation when the fan system was running at low speed.

Secondly, at this low speed, the differential pressure across the cellar door (pressure in ground floor minus pressure in cellar)was just slightly positive. The air moved from the hallway down to the cellar, so the radon gas did not travel to the upper floors through this path.

4 Instrumentation and Measurement

Three methods of monitoring were carried out. Long term, short term (or dynamic monitoring) and instantaneous.

4.1 Long-term Radon Concentration Measuring

The long-term radon exposure was measured by using three etch-track detectors placed in the cellar, lounge and main bedroom from 4th December 1993 to 7th September 1994.

4.2 Short-term Radon Concentration Monitoring

The short term monitoring was aimed at measuring the radon concentrations in response to the controlled ventilation. It was carried out in the cellar as only one continuous radon monitor (CRM), Alpha Guard PQ2000 was available.

The cycle of the monitor was 60 minutes in dynamic radon concentration monitoring while the periods were about 2 to 4 days. These intervals were appropriate for the pressure and radon concentration to settle down after a power change of the ventilation system. This was confirmed by the measurement (refer to 5.5) and reference (3)

4.3 Radon Concentrations Measuring

The instantaneous monitoring concentrated on measuring levels in the other rooms in the house using the Alpha Guard. Each reading was the arithmetic mean of a ten minute measuring period.

4.4 Air Change Rate

averages over a period of 9 months which included a summer season. The opining of windows and doors during this time diluted the concentration in the upper floors. On the other hand, the values of instantaneous readings in winter should be higher than the long-term average ones. The increased ventilation in the upper floors had less effect on the radon level in the cellar. Hence the difference between values of long-term and short-term in the cellar are smaller than these in the lounge and the main bedroom.

5.2 Weather Data

The weather data were collected from Buxton. As the wind speed data was only recorded at 9:00 am every day, hence each value could not represent the wind conditions on that day. However they gave some clues to some extraordinary readings of radon levels (section to 6.1). The values of barometric pressure are greater than that measured by the monitor. These differences cannot yet be explained.

Date	Fan Setting	Wind Direction	Wind Speed(m/s)	Temp. C	Barometric Pre. (mBar)	Relative Humidity
06/02	de-pre 2	W	5.0	7.0	1016.2	93
08/02	de-pre 10	N	5.0	6.7	1010.4	88
14/02	de-pre 8	SW	7.5	4.6	999.4	89
21/02	de-pre 5	SW	11.0	2.0	1009.6	89
22/02	de-pre 3	SW	7.5	5.9	1006.1	89
24/02	de-pre 1	WNW	3.0	2.0	995.0	94
27/02	Natural	SW	4.0	2.5	1014.5	100
28/02	Natural	SW	11.0	9.1	1009.5	92
02/03	Pre 3	SW	3.0	-.1	1002.5	92
06/03	Pre 5	W	7.5	2.4	999.6	84
09/03	Pre 8	SW	3.0	2.4	1011.5	84
13/03	pre 3	SW	1.0	6.5	1028.8	78

Table 2 Weather data in the days when the instantaneous monitoring was done.

5.3 Radon Concentration / Air Change Rate in the Cellar

From figure 5.1 it can be seen that the concentration in the cellar increased as the power of the ventilation system was reduced in extracting mode. The radon concentration reached its highest level at de pressurisation fan setting 2. It then dropped as the extraction rate decreased. When the system was switched off, the cellar was under natural ventilation mode and being resulted in it being de-pressurised.

When the ventilation system was supplying air to the basement the increased pressure reduced the rate of radon entering the cellar, while ventilation removed radon out of the cellar. Hence the level dropped dramatically. The pressurisation ventilation seemed better than de-pressurisation if the level in the cellar were the main interest.

5.4 Average Concentration in Living Area (ACILA) / Air Change Rate in the Cellar

The solution to the problem of radon must address the concentrations found in living areas rather than in a intermittently used cellar. A new parameter, average concentration in the living area was introduced to compare the efficiency of the two ventilation approaches which were investigated.

From Figure 5.1 it can be seen that for the same air change rate in the cellar, the average level of radon concentration in all seven rooms on the upper-floors was higher in pressurisation than de-pressurisation mode. This finding suggests that de-pressurisation is a better ventilation strategy than pressurisation.

The concentration decay method was used to measure air change rate in the cellar. SF_6 was released into the cellar and mixed with the air. The decay in the concentration level was measured and the decay curve evaluated.

The accuracy of the method depends largely on how well the gas is mixed with indoor air. In order to ensure good mixing three approaches were used.

Firstly, a small table fan was used to mix SF_6 with indoor air. An evenly distributed tracer concentration was relatively easy to achieve.

Secondly, the tracer was sampled at two locations as shown in Figure 4.1.

Lastly, several tests were carried out when the fan was set at a high level as the air change rates were higher. The average values gave the final ACH for the particular fan power setting.

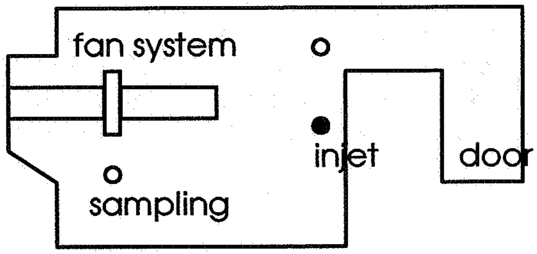

Fig 4.1, Injection and sampling points in the cellar

4.5 Differential Pressure Measurement

Two differential pressures were measured during the tests, one across the cellar door and the other across the external wall. The former represents the pressure difference between cellar and the ground floor while the later represents that between the cellar and outdoors.

Only one point outside the dwelling was taken for differential pressure measurement. The pressure was subject to wind effects, thus fluctuation can be seen (Fig. 5.4.1). Consequently, the measurement yielded a poor correlation between air change rate and wind speed.

The differential pressure across the cellar door was steady at each ventilation mode. The maxium deviation was 41%. This parameter was measured as -16.7 pas. at full power supplying ventilation and 15.5 pas full power extraction. It was 0.7 pas. at speed 2 extracting model, which was believed the best mode.

5 Results

5.1 Long-term Monitoring

The laboratory analysis on the three detectors of long-term monitoring was carried out by BRE. The results were 660 Bq/m^3 in the cellar, 210 Bq/m^3 in the lounge and 190 Bq/m^3 in the main bedroom. The results may have been underestimated by up to 30% because the detectors were issued more than one years ago and age may result in underestimating the exposure.(4)

The values of long-term monitoring are smaller than the other two . This is because the values of the long-term monitoring were the

Room	Cellar	Lounge	Main Bedroom
Long-term	660 +198	210 +63	190 +57
Short-term *	737 ±115		
Instantaneous**		369 ±75	378 ±74

Table 1. Comparison of three types of monitoring of radon concentration level (Bq/m^3)

* Average over 4 days. ** average of 8 readings in two days.

In Figure 5.1 it can also be seen that the ACILAs were higher than that in the cellar in pressurisation mode, such as in Pre 03, Pre 05, Pre 08 and Pre 10. This phenomena can be explained by the fact that other routes for radon entry to upper-floor beside the cracks linking with the cellar exist. This same phenomenon was also reported in work from Princeton University (5).

Figure 5.2 shows the reduction in radon concentration rates for each power setting. This also implies that de-pressurisation is a more efficient remedial approach than pressurisation.

5.5 Concentrations Distribution in Living Area / Air Change Rate of Cellar

From figure 5.3, it is difficult to infer a general pattern of radon distribution. But the levels in the rooms on 1st floor were always the lowest.

Although the radon level in the upper floors depends on the level in cellar, they are subject to the buildings infiltration characteristics and wind conditions. This can be clearly seen in Figure 5.3

5.5 Radon Concentration Short-term Monitoring in the Cellar

From figure A in Figure 5.4, it can be seen that the radon level decreased from about 2500 Bq/m3 within 12 hours when the power setting changed (dep 2 to dep 10), then fluctuated near 1200 Bq/m3 until the next power setting which took place two days later. It seemed that it took a time for the radon level to settle down after a large power change in ventilation system. From all other profiles, no general tendency of increase could be found while the fan power was decreased.

5.6 Radon Level / Barometric Pressure in Cellar

Interestingly, there is a large peak in the radon level when a valley in barometric pressure occurred as seen in figure a, Figure 5.5. Statistical analysis did not indicate that there was any correlation between concentration and air pressure (totally, about 2100 pairs of data of the two parameters in four figures in Fig 5.6).

5.7 Radon Concentration in the Cellar / Differential Pressure Across External Wall

A test was carried out to look at the relationship between radon concentration in the cellar and the differential pressure across the external . This did not provide any significant results. During the measurement period fluctuations in differential pressure varied over a wide range. It can be suggested that to obtain significant results then it would be necessary to measure the air pressure at more than one point outside the house.

6 Conclusions & Suggestions:

The measurements carried out highlighted several aspects of radon control in such situations.

6.1 Mechanical Ventilation in the Cellar

Although the mechanical ventilation in the cellar, both pressurisation and de-pressurisation could be the solution, de-pressurisation was more effective than pressurisation at low power settings (lower air change rates) in reducing the radon level in the living areas in the house.

Higher ventilation rates in the cellar was more effective at reducing the radon concentration in the living spaces in the house although this may have an adverse effect on the heating bills.

Care is needed if there are open-flued combustion appliances in house. The de pressurisation in the cellar may reverse the pressure in chimney hence resulting in spillage.(6). An additional air-brick in the cellar may prevent the spillage as it provides an easy inlet for a cross ventilation in the cellar.

6.2 Simplified Model Study

It is difficult to draw a simple relationship between radon levels in the cellar and the average concentration in the living areas, because of the structural complexity of the building and the unpredictability of the wind conditions.

Further studies on a simplified model under fully controlled conditions is needed to investigate the radon mitigation measures in a wider range before applying them to all other buildings of this type.

6.3 Computer Model Modification

Numerical simulation can be used to study the problems for other house types, provided a adequate model of the building can be developed. The model of the house in BREEZE(2) dose not model the house well, which results in a poor correlation between the measurements and the simulations. Modifications to the model are therefore necessary and should include:

1. Changeable contaminant feeding rate;
2. Multi-route for radon entering into living area;
3. Staircase modelling.

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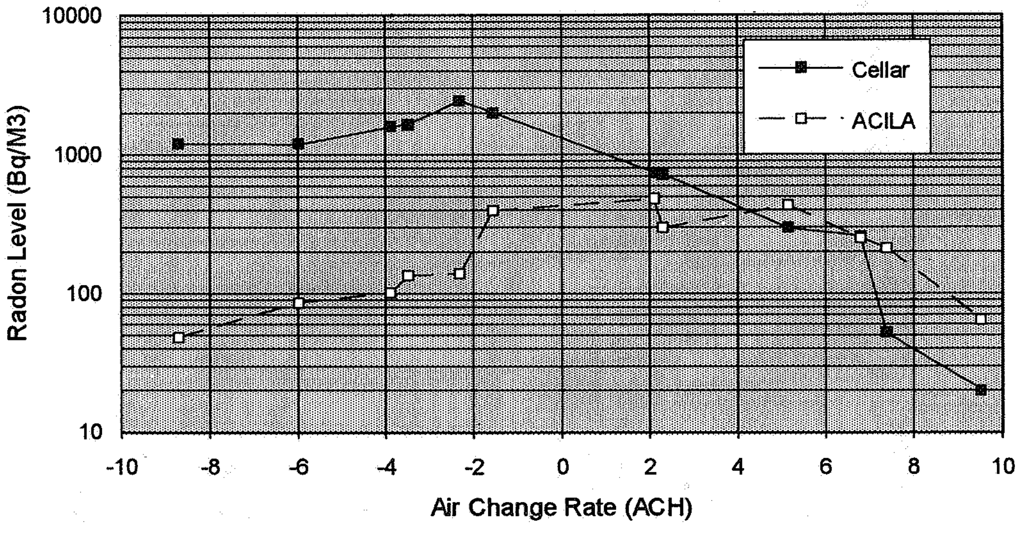


Fig. 5.1 Radon Concentration and ACOLA vs. Air Change Rate in cellar

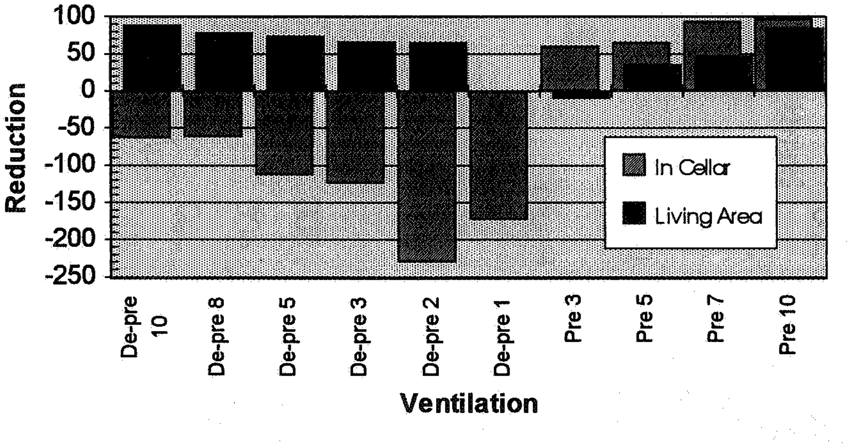
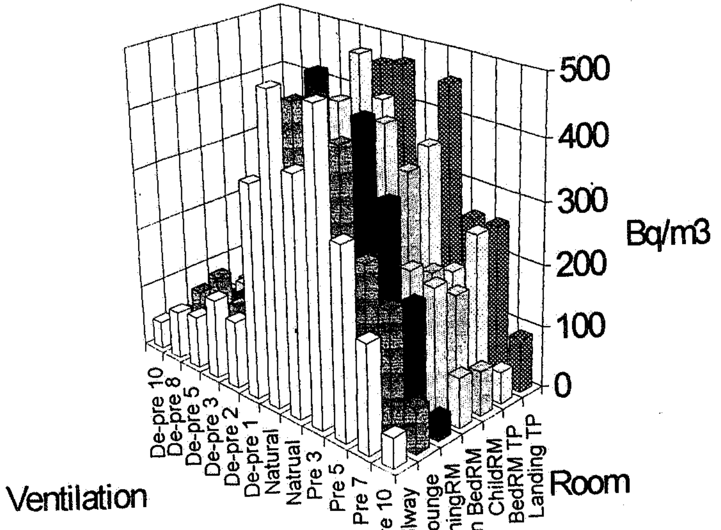
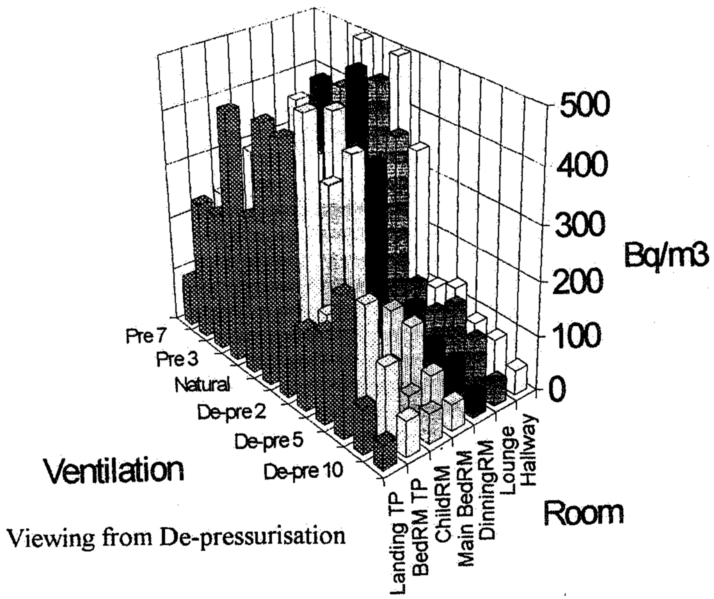


Fig 5.2 Reducation of Radon concentration in Cellar and ACOLA

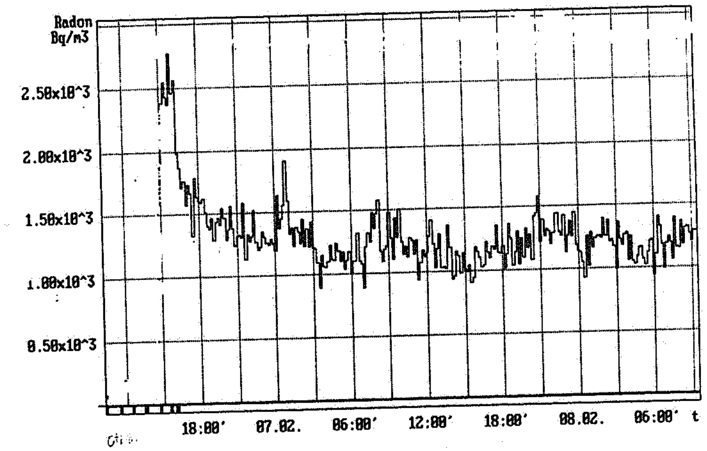


Viewing from Pressurisation

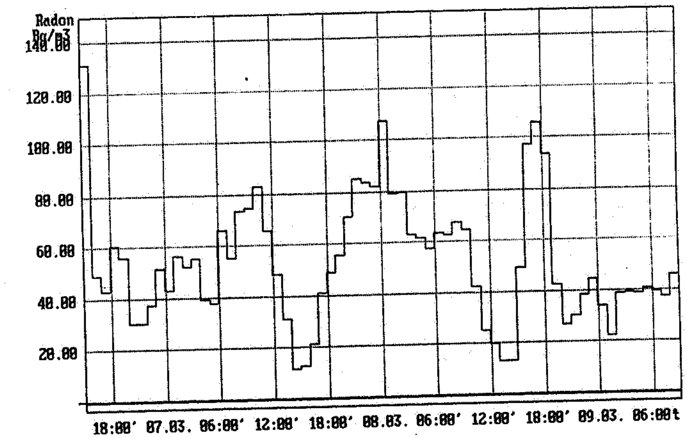


Viewing from De-pressurisation

Fig 5.3 Radon distribution, Levels in seven rooms of the house

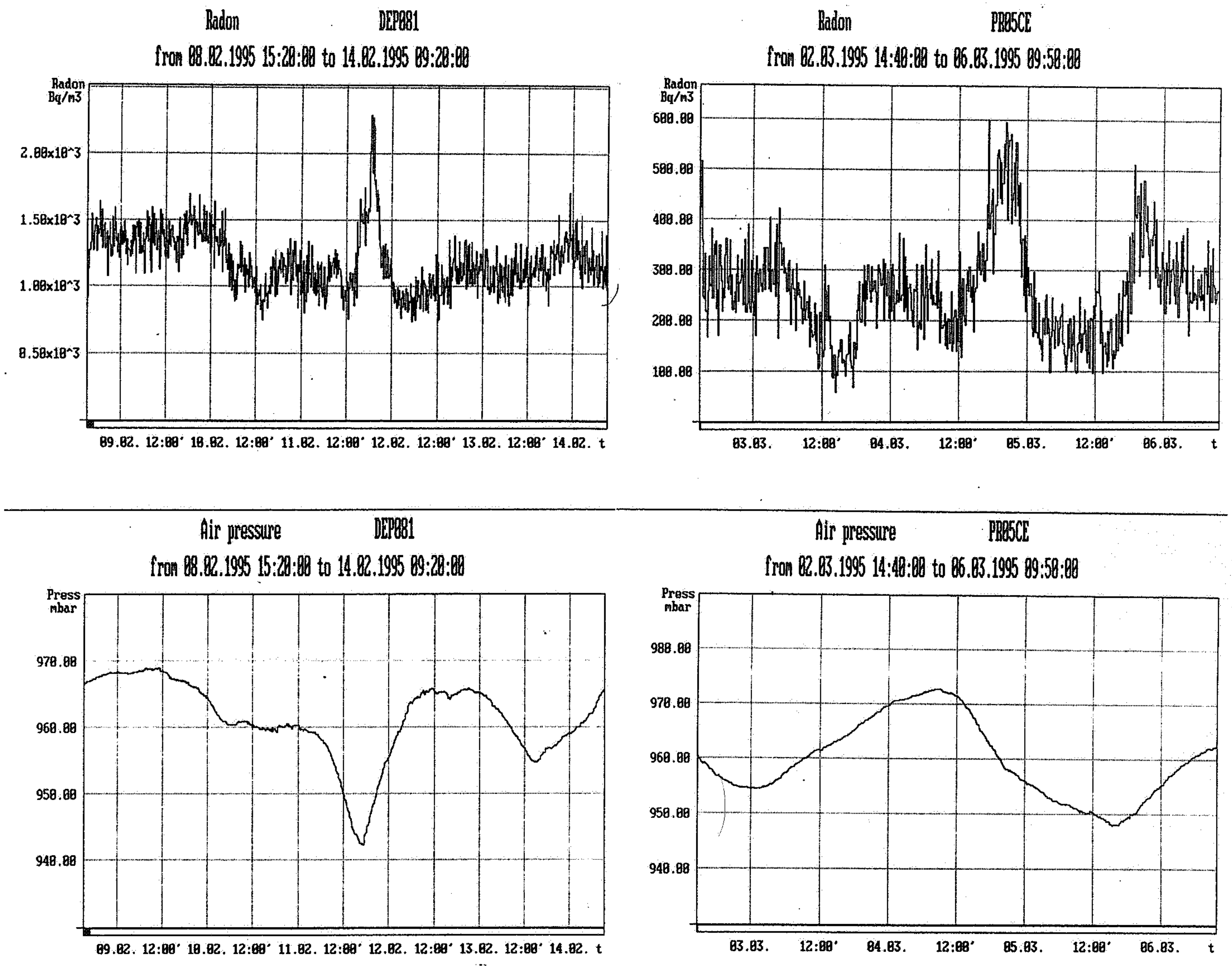


a) after power setting of the system changed from De-pre 2 to De-pre 10



b) after power setting changed from Pre 5 to Pre 8

Fig 5.4 Continuous radon monitoring in cellar after fan power changed



a) fan extracting at power 8

b) fan supplying at power 5

Fig 5.5 Continuous monitoring on radon and pressure in cellar

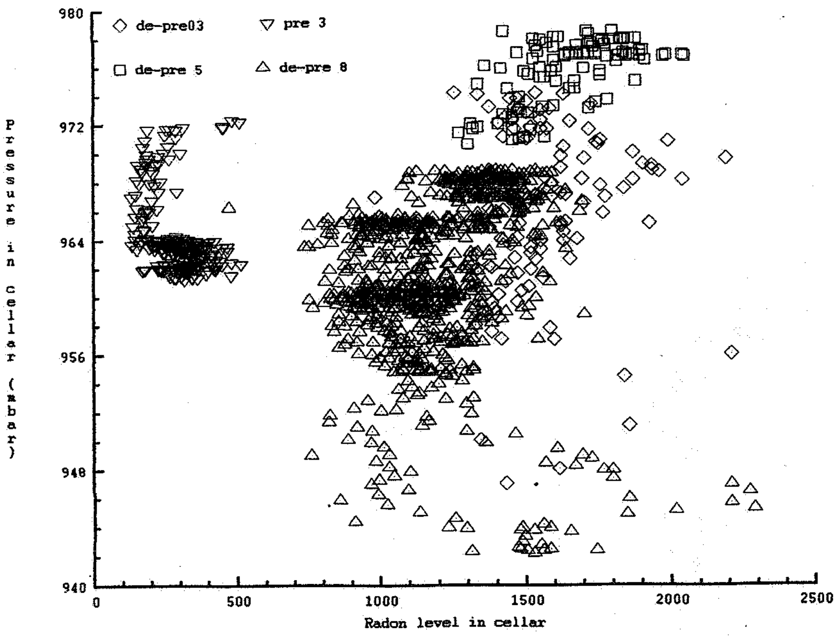


Fig 5.6 Correlation between radon level and barometric pressure in cellar