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INFLUENCE OF KINETIC ENERGY SOURCES AND INTERNAL OBSTRUCTIONS ON ROOM AIR CONDITIONING STRATEGY, EFFICIENCY OF VENTILATION AND ROOM VELOCITY CONDITIONS

REPORT A4

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ABSTRACT

There is a variety of different methods consulting engineers use to design room system, room air diffusion, such as assumption of perfect mixing, design methods employing the empirical relations determined through research, air jet theory and computational fluid dynamics (CFD) codes. The most common design methods based on air jet theory allows only for the prediction of extreme values of air velocities and air temperatures in the occupied zone. However, the results of most analytical and experimental studies has been received from tests in empty rooms and do not reflect the influence of the obstructions or other kinetic energy sources on the room conditions, air distribution and ventilation efficiency.

The objectives of the study have been to investigate the influence of different factors on the room air conditions, airflow pattern and efficiency of ventilation and to utilize the collected information to improve current design practices.

Scale models and full scale experiments and computational fluid dynamics simulations were conducted in order to study the influence of an occupied zone obstruction level, air distribution method, air change rate, heat and contaminant source plus non-uniformity on the room system performance and the efficiency of ventilation.

A new room air conditioning strategy classification was developed. In classification the zonal strategy is introduced to separate room flow situations that cannot be explained by mixing strategy. It is suggested the room air conditioning strategy should be used as a target for design of the room air conditioning system.

A simple method for the calculation of the room average velocity conditions was developed. The method is based on the kinetic energy balance of the room space, thus taking into account both air jets and heat sources. Following the presented design algorithm, a designer can estimate the average velocity level within a ventilated room and furthermore utilize it for evaluation of comfort conditions. The calculation method developed is reasonably accurate in mixed conditions, but additional development is needed to take into account zone effects.

The room obstructions do not influence on room contaminant distribution within the studied ranges and air distribution methods. The room heat sources are important factors for contaminant removal effectiveness and contaminant uniformity inside the occupied zone with zonal air distribution methods. There exists non-uniformity of the contaminant concentrations within the occupied zone that should not be neglected when designing room air distribution. The non-uniform distribution of heat and contaminant sources within the ventilated space can have a remarkable influence on the contaminant removal effectiveness and especially on the contaminant distribution within the occupied zone.

A straightforward comparisons between the measurement data and CFD simulation results is difficult because the comfort oriented, omni-directional air speed measurements are not directly comparable with the air velocity computed with the turbulence models. The use of an artificial, modified velocity method gives especially in low speed areas better correspondence between the measured and calculated speeds.

PREWORD

This work has been carried out at the Laboratory of Heating, Ventilating and Air-Conditioning of Helsinki University of Technology in co-operation with University of Illinois at Urbana-Champaign, USA and is based on the results from "Development of a method for air distribution design in industrial environments" –project. The project was funded by the Finnish Technology Development Center (TEKES) and the companies ABB, HALTON, JP-Building Engineering and VALMET. University of Illinois provided the laboratory facilities and support for extensive scale model experiments. All the sponsors are gratefully acknowledged for sponsoring this work.

I wish to express my gratitude to my supervisor Prof. Kai Sirén for his guidance and valuable support to my work. Special thanks to Prof. Leslie Christianson and Dr. Alexander Zhivov for guiding me through their scientific background and expertise. Without the experiences you shared with me the result of this work would definitely look different. I also want to thank Laboratory manager Steve Ford for his invaluable support in my research work.

I am most grateful to my co-authors Esa Sandberg, Hannu Koskela, Timo Hautalampi, Juha-Veikko Alajuusela and Jorma Heikkinen for the joint enjoyable learning experience.

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Most of all this work wouldn't be realized without the flexibility of my dear wife Lotta and our children Emil, Fanny and Hugo. Thank you for carrying me to this milestone – now its time for us all to go for new challenges.

TABLE OF CONTENTS

LIST OF PUBLICATIONS

I. Hagström K, Zhivov A M, Sirén K, Christianson L L, Influence of the floor-based obstructions on contaminant removal efficiency and effectiveness. Building & Environment 37 (2002), 55-66.

II. Hagström K,Sirén K, The Influence of heat and contaminant source non uniformity on the performance of three different room air distribution methods, ASHRAE Transactions, Vol. 105, Part 2, 1999, pp. 750-758.

III. Hagström K, Sandberg E, Koskela H, Hautalampi T, Classification for the room air conditioning strategies, Building & Environment 35 (2000), 699-707.

IV. Hagström K, Sandberg E, Koskela H, Hautalampi T., A strategic approach for the room air conditioning design, Proceedings of the RoomVent 2000 conference, Reading, 9-12 July 2000.

V. Hagström K, Sirén K, Calculation of the Room Velocity Using Kinetic Energy Balance, ASHRAE Transactions, Vol. 106, Part 2, 2000, pp. 3-12.

VI. Alajuusela J, Heikkinen J, Hagström K, Numerical and Experimental Studies on room airflow, HUT, Laboratory of Applied Thermodynamics, Report 132, August 08, 2001, 66pp.

The Author is the principal author of five publications (I-V). In VI the author was responsible for carrying out the physical experiments and the CFD simulations were carried by co-authors Jorma Heikkinen and Juhaveikko Alajuusela. All authors made the analysis and the conclusions jointly.

1. INTRODUCTION

Until now there has not been presented any unambiguous classification for the room air conditioning strategies or terminology. Traditionally the room air conditioning classification has been based on the room air distribution methods. The most used division has been the division into mixing and displacement, while the other methods have been varied (ASHRAE 1979, Tapola 1987). In German guidelines the division has been made based on the resulting air flow pattern within the room rather than distribution methods (VDI 1994). Etheridge and Sandberg suggested the air distribution methods to be classified as jet controlled or thermally controlled, which raises the important question how well the room air flow patterns are controlled by the air distribution method (Etheridge 1996).

The direct application of air distribution methods to describe the strategies has led to the wild usage of different terms with unclear definition. Additionally, in some cases the same term has been used to describe both the air distribution method and air supply devices or in some cases even the whole air conditioning system. Using a wrong term can also lead to a complete misunderstanding of the physical phenomenon in the room. For example, the term "displacement" is currently used for the room air distribution method in which room air flows are primarily driven by the buoyancy of the thermal sources inside the room and not by the supply air that is introduced to replace (substitute) the air removed by the sources in order to prevent the return flow back to the occupied zone. Thus the term "replace" (substitute) gives a correct picture of the phenomenon, while "displace" may mislead the user to believe that the flow field is created primarily by the air distribution method. The results of this inconsistency can also be seen in the presentations of experts in scientific conferences, but much more confusion is caused in everyday construction business, where the customer often doesn't have any expertise in the technology field.

The objectives of air distribution systems for warm air heating, ventilating, and air-conditioning are to create the proper thermal environment conditions in the occupied zone (combination of temperature, humidity, and air movement), and to control vapor and air born particle concentration within the target levels set by the process requirements and/or threshold limit values based on health effects, fire and explosion prevention, or other considerations. Another objective challenge for the designer is to choose and design the ventilation system energy efficient.

Dimensioning of dilution ventilation is often made using the perfect mixing approximation, assuming uniform contaminant concentration throughout the room space. However, also the efficiency of air conditioning system should be accounted for during design. The efficiency of a room air system consists of two factors: the first factor is the ability of the system to remove heat and contaminants out from the ventilated room. This is most often characterized using temperature and contaminant removal effectiveness (ε_c). Another, seldom considered, efficiency factor is the uniformity of the temperature and contaminant distribution within the ventilated space. In this context the uniformity of the contaminant distribution within the occupied zone is defined using relative occupied zone concentration standard deviation (*Sc*). This factor shows how the maximum contaminant concentration differs from the average concentration. If this factor is neglected, half of the room space has a higher concentration than the actual target level.

There is a variety of different methods consulting engineers use to design room system, room air diffusion and to select and size air diffusers, such as assumption of perfect mixing, design methods employing the empirical relations determined through research, such as the air diffusion performance index (ADPI), air jet theory and computational fluid dynamics (CFD) codes.

Air supplied into the room through the various types of outlets (grills, ceiling mounted air diffusers, perforated panels etc.), is distributed by turbulent air jets. In mixing type air distribution methods, these air jets are often the primary factor affecting room air motion. However, the results of this work show that other kinetic energy sources, such as heat sources affects the room system performance, too. Numerous theoretical and experimental studies that developed a solid base for turbulent air jets theory were conducted concurrently in different countries (Germany, Sweden, Russia, U.K., USA) from the 1930's through the 1980's (Hagström 1999).

Design methods based on air jet theory allows only for the prediction of extreme values of air velocities and air temperatures in the occupied zone of empty spaces. Current air jet theory techniques account for the effects of buoyancy, confinement and jets interaction.

However, the results of most analytical and experimental studies has been received from tests in empty rooms and do not reflect the influence of the obstructions or other kinetic energy sources on the room conditions, air distribution and ventilation efficiency. Attempts have been made to utilize statistical data from the occupied zone conditions in order to extend predictions from extreme parameters to the rest of the occupied zone. The efficiency of the ventilation has been analyzed using zonal models.

Researchers have for a long time tried to establish the relationships between jet momentum and room velocities (Li 1995, Chow1996 & 1997). However, room air movement is to a greater degree three dimensional without a specific direction, which makes it difficult to describe in a general case using momentum that is a vector quantity. (Priest, 1996) Additionally, the room air velocities are significantly influenced by the internal heat sources.

The importance of the room kinetic energy was initially introduced by Elterman as he stated that volumetric kinetic energy influences the turbulent exchange between the supply air jet and the bulk flow and the convective heat and mass transfer between different zones in the room (Elterman, 1980). According to Elterman, the total kinetic energy can be generated from three sources: the energy of incoming air, convective jets attenuating in the room, and the energy introduced by moving objects.

Priest and Zhivov presented a very straightforward approach (Priest, 1996, Zhivov 1996): They proposed that it is possible to calculate the average room "dissipation" velocity assuming that the kinetic energy flux introduced into the space is dissipated into heat by the room volume.

The objectives of this study have been to investigate influence of different factors on the room air conditions, air flow pattern and efficiency of ventilation and to utilize the collected information to improve current design practices.

The specific objectives for the research were:

- To study the influence of the floor mounted obstructions with and without heat generation to the air flow patterns and room conditions. (I)
- To study influence of the source non uniformity. (II)
- To develop a framework to explain experimental findings and to allow their utilization in improving design of room air conditioning. (III&IV)
- To improve the design of occudied zone conditions in realistic rooms by taking into account room obstructions and kinetic energy sources. (V)
- To study applicability of computational fluid dynamics simulations for quantitative design of room air conditioning. (VI)

The focus segment for the research was industrial manufacturing hall. The air distribution methods, cooling load and obstruction levels, and room layout were based on this selection. No specific installation was modeled, but based on the information of existing factories the parameters were selected in order to cover large range of different types of industries.

2. METHODS

Scale models and full scale experiments, tracer gas technique and computational fluid dynamics simulations were conducted in order to study the influence of an occupied zone obstruction level, air distribution method, air change rate, heat and contaminant source plus non-uniformity on the room system performance and the efficiency of ventilation. The methods and experimental set-ups used are described in this chapter.

2.1. Physical experiments

2.1.1 Reduced Scale Experiments

Experimental data was collected in reduced scale for three different air supply methods:

1. Horizontal, **concentrated supply,** in which the occupied zone was ventilated by reverse flow. The air was supplied from nozzles horizontally by the west wall to the upper room level at a height of 2.1m. The outlet diameter of the nozzles was 78mm.

2. **Horizontal direct supply**, in which the occupied zone was ventilated directly by the jet. The air was supplied by grilles horizontally from the south wall at a height of 1.2m from the floor. The outlet size of the grille was 102mm x 114mm. The vertical vanes of the grille were adjusted at the 90° angles for greater expansion of the jet in a horizontal direction.

3. Vertical supply by jets projected downwards. The air was supplied from grilles vertically from a height of 2.1m. The grilles and air velocities were the same as in method 2.

The scheme of the air supply methods is shown in Figure 1. The jet momentum through a single inlet was kept constant. Thus, the number of air supply depended on the air change rate: at an air change rate (1/h) of 2, two nozzles or three grilles were used; at 4 1/h, four nozzles or six grilles, and at 6 1/h, six nozzles or nine grilles were used, respectively. The air supply devices were located symmetrically in each measurement.

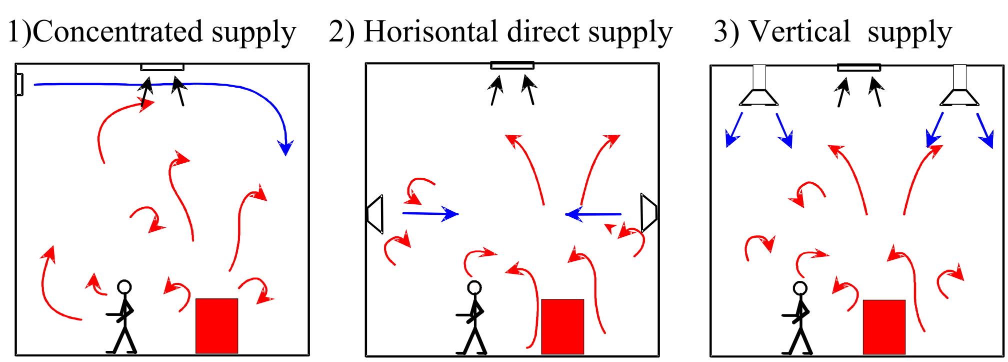

Figure 1. Studied air supply methods

Room Ventilation Simulator

The experiments were conducted in the Room Ventilation Simulator (RVS) of the University of Illinois at Urbana-Champaign (Wu 1990). The RVS consists of an adjustable inner test room and an outer room for controlling the ambient environmental conditions of the inner test room. During the experiments, the inner test room was set at 7.2m x 3.6m x 2.4m to model a full-scale ventilated room, and the structures were set at 3:10 scale, see Figure 2. The independent HVAC system provides a constantly conditioned supply of air for the inner test room.

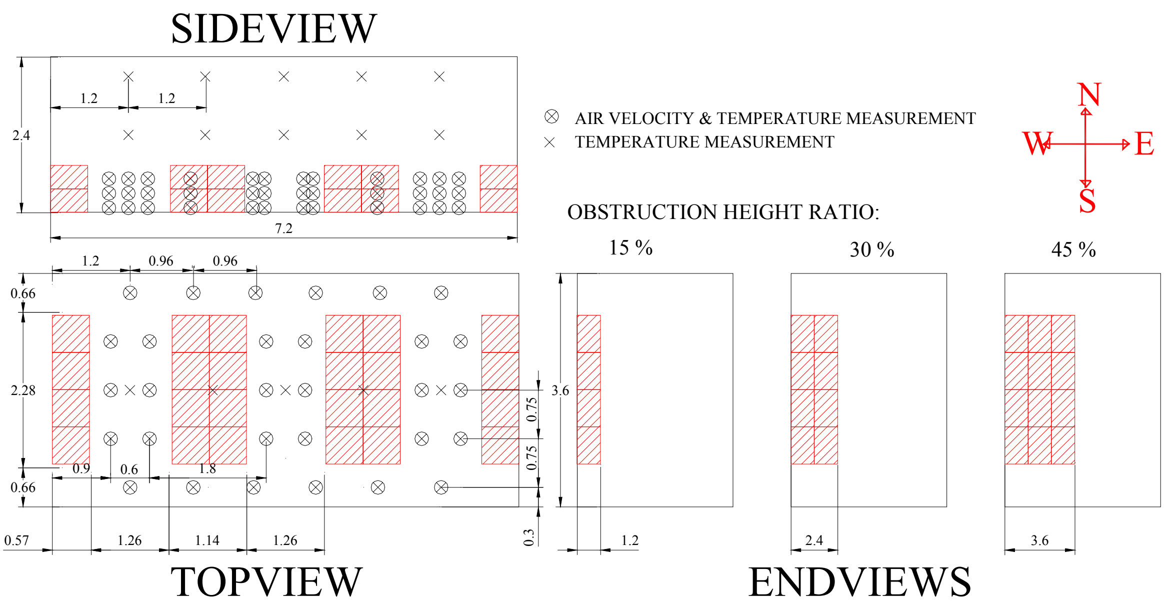

Figure 2. Experimental room layout and air temperature and velocity measurement points, Obstruction area ratio 30%. Dimensions in meters.

Obstructions and Heat Sources

Sheet metal boxes measuring 0.57m x 0.57m x 0.36m were used as obstructions in the experiments. The boxes, 72 in all, were positioned in the room in different ways in order to create the desired room layout, obstruction area and height ratios, see Figure 2. The following obstruction levels were used in this study:

- Height ratio, (h_{obs}/h_{rm}): 0%, 15%, 30%, 45% of room height
- Floor area ratio (A_{obs}/A_{rm}): 0%, 15%, 30% of room area

The heat source inside the obstructions was created with light bulbs. The actual power consumption and obstruction surface temperature were monitored during each experiment.

Measurement and Control Systems and Equipment

Supply and exhaust volume flow rates were controlled with frequency transformers. The volume flow rates of individual supply openings were adjusted using a factory-made measurement and adjustment unit (accuracy ±5%) connected to the ductwork. The pressure difference between the inner and outer room was kept at zero by adjusting the exhaust air flow rate.

A temperature controller kept the supply temperature at the set point during the test. A data acquisition system monitored the air temperatures in the supply, exhaust, test room (10 points) and outer room. Data acquisition system with T-type thermocouples was used to monitor air temperatures in supply, exhaust, test room (10 points) and outer room.

A constant temperature anemometer with 30 omni-directional thermistor probes was used to measure air temperature and velocity in the occupied zone. The measurement system is described in detail in (Kovanen 1986). The inaccuracy of it is estimated to be ± (0.03 + 0.03v) m/s, when the air velocity is between 0.05 m/s and 1.0 m/s. The measurement set up was created in such a way that it was possible to change probe height from outside of the test room without interference to the measurements. During each test altogether 90 measurements were made in thirty locations and at three different heights, 45 mm, 330 mm and 520mm. Location of measurement points is shown in Figure 2.

Sampling interval 0.2 s and integration time 60 s were used in the measurements. The integration time was chosen to ensure high quality of data with a reasonable duration of the test period. By this way the duration of one measuring cycle was two hours. The influence of the integration time was studied prior to actual measurements and no influence was found to the measurement results, when the integration time was reduced from 180 s to 60 s.

Tracer Gas Supply

Tracer gas SF_6 was used to model contaminant sources inside the room. Pure tracer gas was mixed with outdoor air and pumped inside the room. The tracer gas concentration in the supply was less than 0.2%. Supply air was taken outdoors to ensure absolute purity from tracer gas. The implementation was also controlled during the measurements. Exhaust air from the test room was exhausted directly outdoors to the opposite end of the simulator building to prevent short-circuiting of the tracer gas into supply air.

Tracer gas measurement system.

Tracer gas concentrations in the occupied zone were measured in fifteen points at 1'10" (0.55m) height (corresponding 6 ft (1.8m) in full scale room) and exhaust air concentration was measured from the exhaust duct. See Figure 3. for the location of the measurement points. Air was pumped constantly (total rate 5.5 ml/min) through measurement hoses from the measurement points inside the room and exhaust duct. Air samples were taken from each hose and injected into the Gas Chromatograph column. It was used to separate SF_6 from the air, and an electron capture detector was used to measure the concentration. The whole measurement system was calibrated using premixed 500 ppb and 1000 ppb solutions of SF_6 and nitrogen.

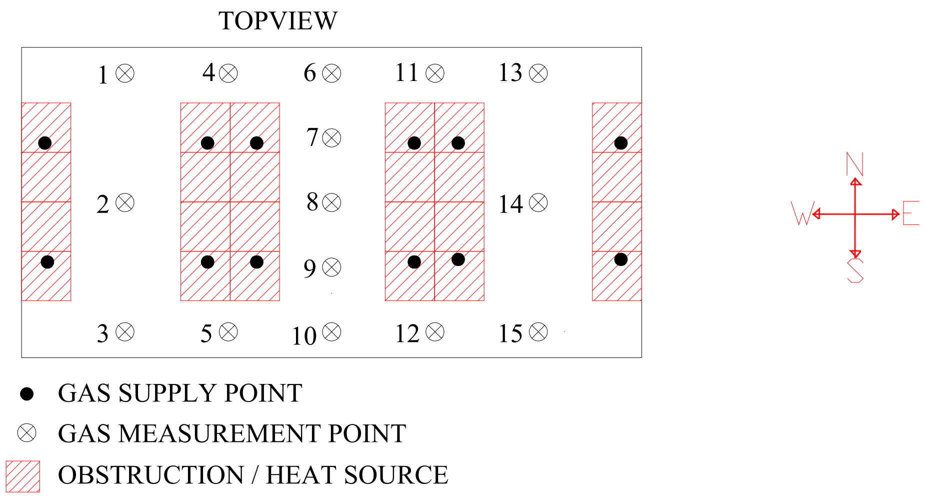

Figure 3. Tracer gas (contaminant source) supply and measurement points in experimental room.

Method for Scaling

Air flow similarity. Experiments were conducted using an approximate modeling method on a reduced scale of 3:10 to simulate air-conditioning in a large industrial hall. For approximate modeling of turbulent flow, it is sufficient to ensure that the dimensionless Archimedes and Prandtl numbers are identical in the full scale reference and in the scale model. Moreover, the dimensionless Reynolds number in the supply outlet has to exceed a threshold limit value in both the full scale reference and in the scale model. To maintain the similarity of a turbulence spectrum in ventilation models, the Reynolds number to be exceeded is 10^4 (Mierzwinski 1987, Popiolek 1998). During the experiments, the Reynolds numbers for the inlets were: nozzles $4*10^4$ and grilles $1.6*10^4$.

Similarity of thermal boundary conditions.

Natural convection. When natural convection of a thermal plume is modeled under ventilation conditions, turbulent convection flow occurs (Shilcrot 1994), if

$$(GrPr) / 2*10^7, \tag{1}$$

where Gr is a dimensionless Grasshoff number and Pr is a dimensionless Prandtl number. This criteria was met during the simulations.

Partitions. The thermal exchange between the environment and the test room was eliminated during simulations. From (Baturin 1972) the necessary condition for this assumption is that the temperature in the test room is the same as the average temperature in the model room and the walls and ceiling are insulated. In such a situation the heat flow through partitions is very small compared with the heat flow due to ventilation and can thus be neglected

2.1.2 Full scale laboratory experiments

One of the air distribution methods was tested in a full scale experiments in the laboratory of Halton Oy in Finland during the summer 1998. The dimensions of the laboratory hall were 25 m by 12 m by 8m. The measurement and control systems and test setup were identical to reduced scale experiments except the geometrical scale.

2.2 Computational fluid dynamics simulations

Air velocity and temperature measurements in a reduced scale industrial hall were compared with the simulations, which were performed using two different computing codes in paper (VI). The simulations using the commercial Fluent code with a high-Reynolds number standard k-ε model and wall functions represent the usual practice in CFD simulations, whereas the simulations with the university code FINFLO represent a more scientific approach without using the wall functions. The latter practice necessitates a low-Reynolds number turbulence model as well as a much finer computing mesh near the walls to capture the actual wall boundary layers. In the high-Reynolds number model simulations the first cell height was selected to be out of the actual operating range of the turbulence model in order to get better resolution near the heating elements.

The supply air terminal boundary conditions remain one of the main difficulties in the room airflow simulations. The flow boundary conditions at the supply air grille used in horizontal direct supply were especially difficult to set accurately because it was not possible to describe the small details of the grille in the computational geometry. Therefore, the flow in front of the grille was separately measured using Laser-Doppler anemometry. Still it was necessary to simplify the measured flow field to be able to use it as a boundary condition in the simulations. The turbulence quantities were manipulated in order to adjust the jet profile similar to the measurements.

The radiation heat transfer between heat sources and large room surfaces was evaluated before the simulations by using a simple radiation model utilizing view factors and blackbody assumption for the room surfaces and equivalent convection heat flux was set as a boundary condition.

Symmetry was utilized in the simulations but the measurements showed that flow was not fully symmetric in reality (VI).

3. RESULTS

3.1 Importance of floor based obstructions on contaminant distribution

Influence of an occupied zone obstruction level, air distribution method, air change rate, heat and contaminant source plus non-uniformity on the contaminant removal effectiveness and occupied zone contaminant concentration uniformity were studied in scale model in paper (I). The results suggest that the room obstructions do not influence on room contaminant distribution at least within studied ranges and air distribution methods.

Heat sources can be an important factor for contaminant removal effectiveness and efficiency with some air distribution methods. With the horizontal direct supply increasing heat power improved both efficiency parameters clearly. Also in vertical supply case the contaminant removal effectiveness was improved, but only a small change in the efficiency was found due to heat power. Whereas the heat sources did not have any influence either on the contaminant removal effectiveness or efficiency in concentrated supply.

However, the results of the experiments show that there exists non-uniformity of the contaminant concentrations inside the occupied zone with all supply methods. This deviation was not significantly affected by any of the studied parameters. The average standard deviation of the occupied zone contaminant concentrations for all the concentrated distribution cases was 0.12. The average of the measured contaminant removal effectiveness was 1.0. The corresponding values for vertical supply were 0.10 and 1.26. In horizontal direct supply, the average standard deviation of the occupied zone concentrations for all the cases was 0.08. Heat power of the sources increased the contaminant removal effectiveness. The average contaminant removal effectiveness for all isothermal cases was 0.92 and for the non-isothermal cases 1.14.

Design of dilution ventilation is based on the target level for the contaminant concentration inside the occupied zone. The contaminant concentration non-uniformity in the occupied zone should not be neglected when designing room air distribution. Thus, the Threshold Limit Value (TLV) of specific contaminant should not be used as target level, because TLV would be exceeded in part of the space due to non-uniformity. Also, the dilution air flow should be increased from the value calculated based on total mixing to keep the maximum concentration inside the occupied zone below the target level. This can be taken into account by using a safety factor,

$$SF = 1 + 2S_c \,, \tag{2}$$

where S_c is the standard deviation of the occupied zone concentrations.

The safety factor could be developed for different methods of air supply from statistical measurement data. The necessary amount of supply airflow, which guarantees desired conditions throughout the occupied zone with the 95% confidence5 interval, could then be calculated from

$$q_s = \frac{mSF}{C_{oz}\varepsilon_c}, \qquad (3)$$

where m is the contaminant generation rate [kg/s] and C_{oz} is target level of the contaminant concentration within the occupied zone [kg/m^3].
Applying the results of the experiments in paper (I), the following safety factors, SF, can be developed for the studied air distribution methods:

- Concentrated supply	1.24
- Horizontal direct supply	1.16
- Vertical supply	1.20

However, it must be noted that the use of these safety factors is limited to the room conditions close to the experiments and cases with uniform contaminant sources only. Especially for horizontal grilles and vertical air supply methods the source non-uniformity was found to have major influence on the concentration uniformity conditions. More research would be needed to develop safety factors for different types of air distribution in uniform and non-uniform situations.

3.2 Sensitivity of air flow pattern on source non uniformity

3.2.1 Introduction

In chapter 3.1 it was shown that heat and contaminant sources influence on room airflow pattern and on efficiency of the ventilation. However, all the experiments were made in such a way that the sources for heat and contaminant release were located uniformly inside the room. Additionally, the non-uniform source location might have a great influence on ventilation efficiency. The importance of the contaminant source location to the concentration distribution has also been addressed by Nielsen based on his work with computational fluid dynamics models and by Heiselberg based on laboratory measurements with single point source (Nielsen 1994, Heiselberg, 1996).

The objective of the study reported in paper (II) was to find out how non-uniformity of heat and contaminant sources influences on the efficiency of ventilation with different air distribution methods. Scale model experiments were conducted to find out the influence on the occupied zone temperature and contaminant distribution and the temperature effectiveness and contaminant removal effectiveness.

3.2.2. Method of non uniformity

Heat Source non uniformity
The influence of the heat source non uniformity was studied by supplying heat on one half of the room only, both lengthwise and width wise division of the room was tested with each air distribution system. The principle of division is illustrated for horizontal direct supply in Figure 4. Following measurements were made with different air supply methods:

1.) Concentrated supply from west wall

a) The heat source was in the west half of the room close to the wall, where supply jets were located (3.6m x 3.6m area).
b) The heat source was in the east half of the room opposite to the wall, where supply jets were located (3.6m x 3.6m area).
c) The heat source was located on one (north) side of the room in the direction of the supply jets (7.2m x 1.8m area).

2.) Horizontal direct supply from the south wall
a) The heat source was in the south half of the room close to the wall, where supply jets were located (7.2m x 1.8m area).
b) The heat source was in the north half of the room opposite to the wall, where supply jets were located (7.2m x 1.8m area).
c) The heat source was located on one (west) side of the room in the direction of the supply jets (3.6m x 3.6m area, lengthwise division).

3.) Air supply with vertical downwards projected jets.
a) The room was divided lengthwise and the heat source was located on the one (west) "side", half of the room (3.6m x 3.6m area).
b) The room was divided widthwise and the heat source was located on the further (north) side of the room (7.2m x 1.8m area).

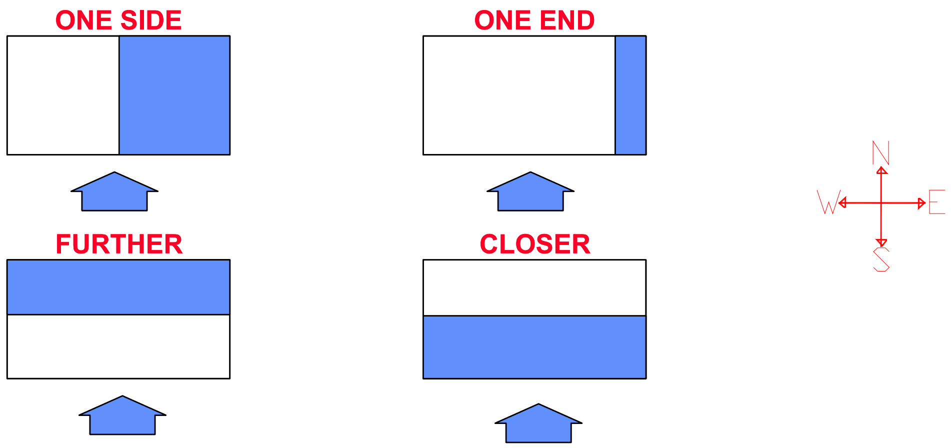

Figure 4. The principle of the room division in source non uniformity studies, horizontal direct supply. The arrows indicate location of the supply air diffusers

Contaminant source non uniformity

The contaminant source non-uniformity studies were done with the same approach as heat source non uniformity studies. The only difference was, that during these tests the heat source was kept uniform and location of contaminant source was changed. Additional tests were made with the gas source on one end of the room, when the tracer gas was supplied only through the points on the obstruction by the east-end wall.

Combined heat and contaminant source non uniformity

The influence of the combined heat and contaminant source non uniformity was studied by supplying heat only on one half of the room, both lengthwise (7.2m x 1.8m area) and widthwise (3.6m x 3.6m area) division of the room was tested with each air distribution system. In each situation two tests were run; firstly, the contaminant source was located on the same side as the heat source (a buoyant contaminant), secondly it was located on the opposite side than the heat source (a non buoyant contaminant).

3.2.3 Results

Both heat and contaminant source non-uniformity and location was found to be major factors on the efficiency of the studied room air systems.

Heat Source non uniformity
The heat source non-uniformity increased the temperature effectiveness with all air distribution methods. On the other hand, it had only minor influence on the temperature non-uniformity inside the occupied zone with vertical supply and no influence with the other methods. In the case of concentrated air supply the temperature effectiveness improved by 10% in each case compared to the uniform situation. Heat source location was not found to have influence on the effectiveness. In the case of the horizontal direct air supply the temperature removal efficiency was improved in all cases but especially, when the heat source was located on one side of the room the improvement was 28%. In the case of vertical air supply the temperature effectiveness improved by 11%, when the room was divided widthwise and by 24%, when the room was divided lengthwise.

Contaminant source non uniformity
The contaminant source non-uniformity has a major influence on the conditions with the horizontal direct supply and the vertical supply methods, while the concentrated supply was less sensitive method towards this factor. The biggest difference was found in the occupied zone conditions, when the room was divided lengthwise. The contaminant source located on one end of the room increased standard deviation of the occupied zone concentration fivefold in the case of horizontal direct supply and vertical supply methods. At the same time also contaminant removal efficiency was decreased by 10-15%. Applying equation (3), this would result in about two times the supply airflow rate needed in the case with uniform sources. The result of the contaminant source non uniformity studies in case of a uniform heat source are presented in Figure 5.

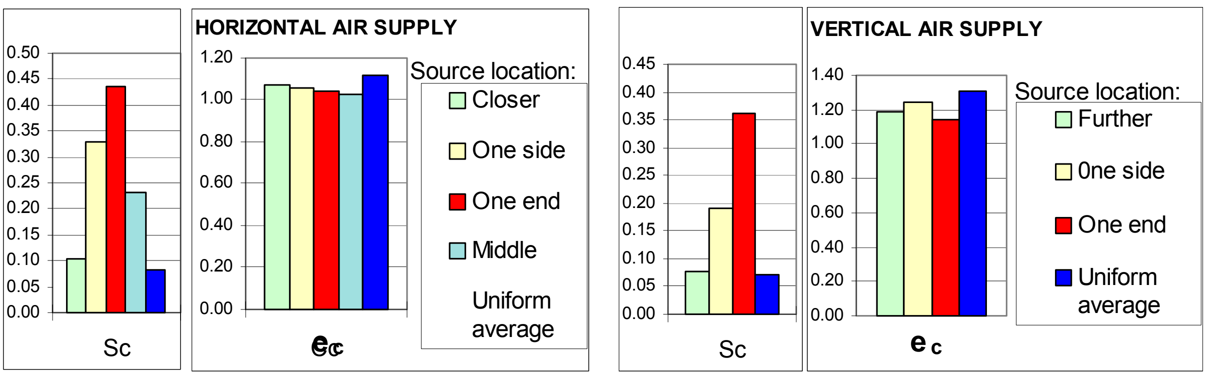

Figure 5. Influence of the contaminant source non uniformity on the occupied zone contaminant concentration standard deviation (S_c) and on the contaminant removal effectiveness (e_c).

Although the contaminant source non-uniformity influenced in a similar way with the horizontal direct supply and the vertical supply methods, the concentration distribution profiles were different. With the vertical supply the borderline between the high and the low concentration area was very sharp, while the change was smoother along the room with the horizontal direct supply, see Figure 6.

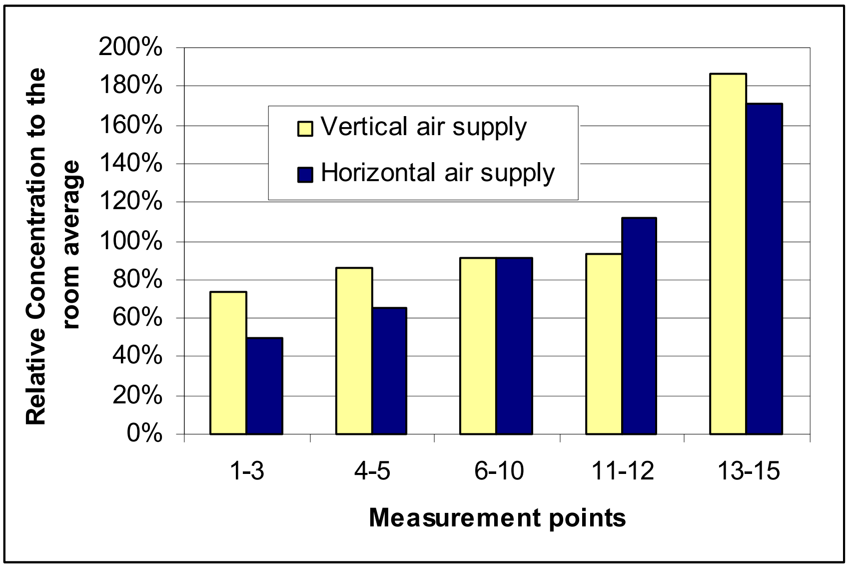

Figure 6. Occupied zone concentration distribution profiles with vertical and horizontal air supply, contaminant source at the east end of the room. Average concentration of measurement points; see location of the points Figure 3.

Combined heat and contaminant source non-uniformity

The influence of the contaminant source buoyancy and location of the heat source in relation to supply air outlets were found to have a greater influence on both occupied zone concentration uniformity and contaminant removal effectiveness with both horizontal direct supply and vertical supply methods. When the contaminant source was non-uniform, the heat source non-uniformity reduced standard deviation of the occupied zone concentration compared to the situation with the uniform heat source. Still, the difference to the cases with uniform heat and contaminant sources were remarkable.

With the horizontal direct supply the contaminant removal effectiveness was decreased below 1.0 in all the cases with the non-uniform heat source, while it was 1.12 with the uniform heat source. In the case of vertical supply the contaminant removal effectiveness was increased with buoyant source and decreased with non-buoyant source. The probable reason for the results is that the non-uniform heat source tends to create additional circulation of the room air that in some cases is perpendicular to the flow pattern created by supply jets. Depending on the situation this either improves or hinders the transportation of the contaminants from the occupied zone.

3.3 Room air conditioning strategies

A new strategy approach has been introduced in paper (III) for the room air conditioning including classification and terminology. The basis of the classification is different aims or ideas of the temperature, gas, particle or humidity distributions and air flow patterns that can be created within the room. The distributions are often described by using contaminant removal and temperature effectiveness-coefficients (Etheridge 1996). The classification of ideal room air conditioning strategies is summarized in Figure 7. Though the main emphasis of this presentation is on the general room air conditioning, the same ideas of different strategies can be used also for local ventilation. Additionally, as ideal, the classification doesn't make a difference whether the flow direction is horizontal or vertical (upwards or downwards).

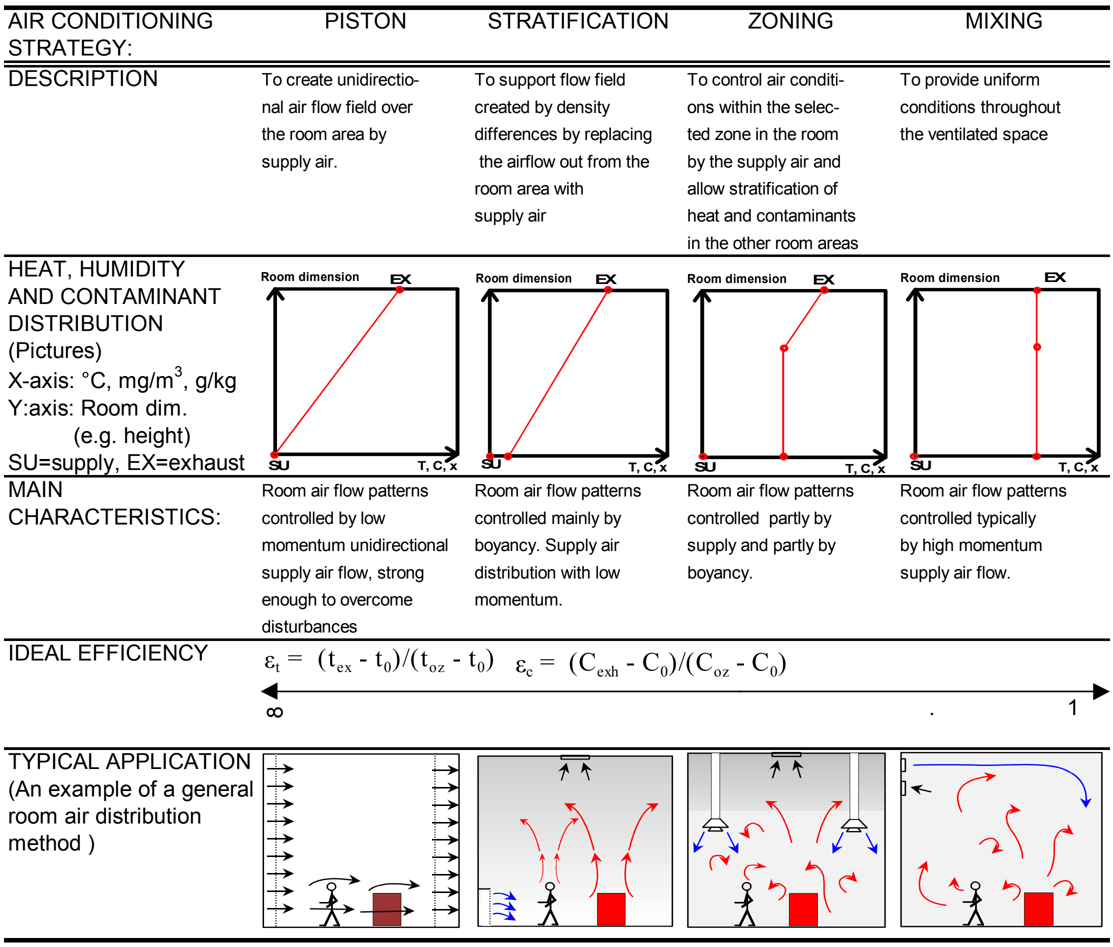

FIGURE 7. The summary of the ideal room air conditioning strategies

As the focus of the proposed classification differs from the present practice it is necessary to explain the terminology used. The aim of the room air conditioning is to maintain desired conditions, **target levels**, in the room during different operating conditions in the most economical way (energy, cost efficiency). Depending on the design criteria the designer has different strategies to choose in order to achieve specified targets. The room air conditioning **strategy** is a fundamental scheme that describes the targeted temperature, humidity and contaminant distributions as well as air flow patterns within the air-conditioned room. The room air conditioning **system** consists of different methods and their controls that all together create the system performance. The system performance is evaluated by comparing the reached conditions to the chosen strategy. Both the **methods** (room air distribution, exhaust, room heating and cooling, etc.) and processes and disturbances inside the room influence the resulting conditions

The aim of this classification is not to value one strategy over another. They all have their advantages and disadvantages and it is up to the designer to select the most desirable strategy for each case. In practice a certain type of room air conditioning strategy can be applied by using different kinds of air distribution installations and air supply devices. How well the real situation will fulfill the aim of the ideal strategy is dependent not only on the physical installation itself, but also on the operating parameters as well as the characteristics of other internal sources that influence the supply air flow patterns and the room air flows, such as heat and contaminant sources, cold drafts, room heating and cooling methods. It is therefore important to separate the ideal strategies from the practical room air conditioning solutions.

3.4 Design of room air conditioning

The new strategy approach presented in paper (III) provides grounds for new strategic room air conditioning design process as well as for explanation of the findings presented in papers (I, II) , which have been discussed in paper (IV).

3.4.1 Strategic design process

Suggested room air conditioning design and evaluation process is illustrated in Figure 8. The main difference to the current design practice is that the consideration of the room air conditioning strategy has been added as a separate design stage between indoor air targets and selection of the room air conditioning system. Also, it is suggested that during commissioning the realization of the strategy as well as loads should be checked – not just the target values. Only checking of the strategy ensures that the room system operates optimally and as it has been designed.

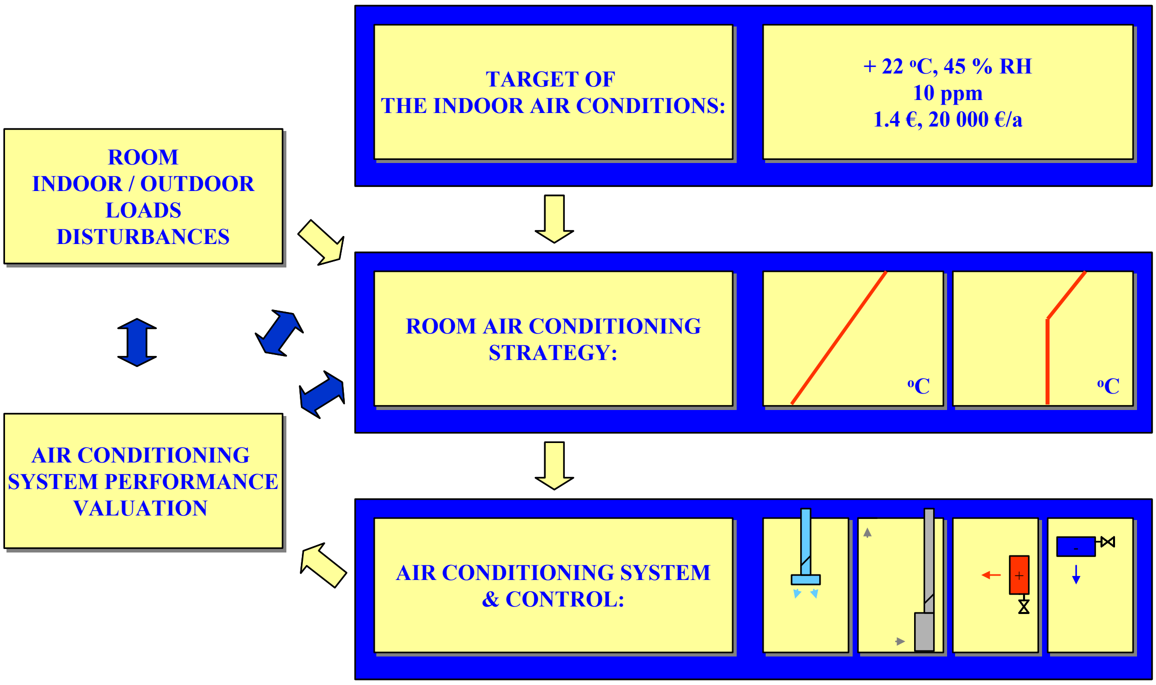

Figure 8. The Room Air conditioning Design and Evaluation Process. Light arrows show the process flow and the dark arrows illustrate the factors to be evaluated during commissioning.

3.4.2 Practical examples on the implementation of the strategy approach

There are three very common misunderstandings among practitioners that can be explained with the strategy approach:

(1) *The displacement ventilation is much the same as piston flow.* Displacement ventilation is stratification approach. The strategy classification shows the difference between stratification and piston strategies.

(2) *Designing displacement ventilation air supply leads automatically to stratification strategy.* Stratification strategy is based on buoyancy forces, thus, it is very sensitive to total system approach. In the following it is given an example, where actually mixing strategy is applied despite the displacement type air supply.

(3) *The use of air jets leads to the complete mixing.* The results of this study show clearly zonal effects within room airflow pattern. The zonal strategy can be utilized to explain these findings. The difference between mixing and zoning is demonstrated below.

Stratification versus Mixing

In Roomvent '98 conference there were presented three papers (Alamdari 1998, Brohus 1998, Tan 1998), which showed the low velocity air supply and cooled ceiling system to behave almost as a complete mixing strategy instead of stratification strategy. The transition from stratification strategy to mixing strategy was demonstrated by (Tan 1998) et al, Figure 9. Using traditional terminology one would consider this system as displacement ventilation and automatically assume temperature stratification.

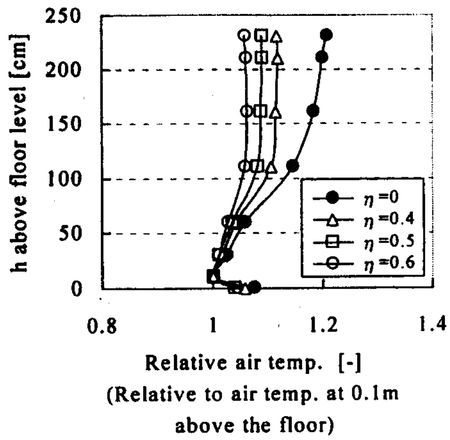

Figure 9. Relative vertical air temperature distribution by η (η = ratio of the cooled ceiling cooling output to the total cooling output) (Tan 1998).

Zoning versus Mixing

The studies of three air diffusion methods in both isothermal and non-isothermal situations were reported in papers (I & II). The average temperature effectiveness and contaminant removal effectiveness values of the measured cases are presented in Table 1.

The results show that different air diffusion methods together with the room situation result in different strategies in the room. The concentrated air diffusion represents close to the ideal mixing strategy. Horizontal air supply, where the air supply units were located mid-height of the wall, results in the mixing strategy in isothermal conditions, but in non-isothermal conditions the zoning strategy is applied. The vertical air supply results in zoning strategy in both isothermal and non- isothermal situations. This is due to the special air supply arrangement, where the supply air was directed into the corridors between the heat and contaminant sources, thus creating zoning in horizontal direction and encouraging upward air movement on top of the sources.

Table 1. The average temperature effectiveness and contaminant removal effectiveness values of the measured cases.

Temperature effectiveness

	Average	STDEV
CONCENTRATED	0.98	0.02
HORIZONTAL	1.21	0.04
VERTICAL	1.27	0.1

Contaminant removal effectiveness

	Average	STDEV
CONCENTRATED		
-Isothermal	0.99	0.06
-Non-isothermal	1.01	0.03
HORIZONTAL		
-Isothermal	0.92	0.03
-Non-isothermal	1.14	0.09
VERTICAL		
-Isothermal	1.23	0.07
-Non-isothermal	1.30	0.05

Another finding from the study emphasizes the sensitiveness of the zoning strategy to non-uniform source conditions. In situation when the contaminant source was located on one end of the room, the variation of the contaminant concentration within the occupied zone increased significantly compared to the situation with uniform source distribution. Corresponding influence was not found in concentrated air supply. Results can be seen from Figure 10.

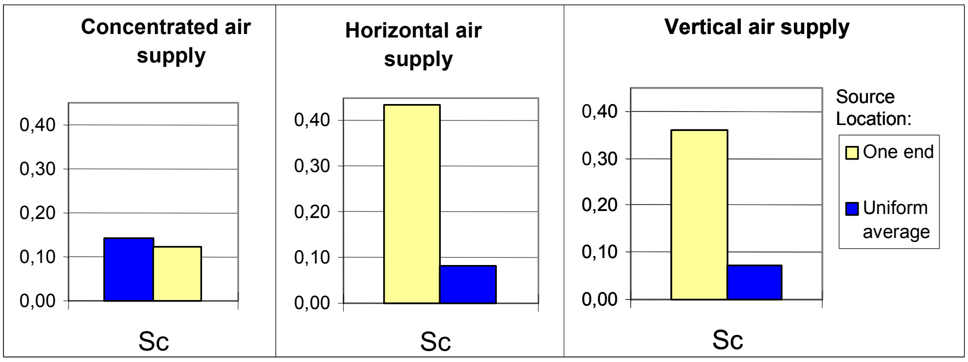

Figure 10. The standard deviation of the contaminant concentrations within the occupied zone with different air diffusion methods. Influence of the contaminant source location non-uniformity.

3.5 Room average velocity as a function of kinetic energy balance

3.5.1 Introduction

In paper (V) a thorough analysis of the kinetic energy balance is conducted in order to find the exact correlation between kinetic energy flux and the room average velocity. As a result a method for the room average velocity calculation is developed and validated, and an algorithm for its application is presented.

3.5.2 Room Kinetic Energy Analysis

Defining a room as a control volume, the conservation of kinetic energy can be expressed as

$$\frac{dE_r}{dt} = e_{input} - e_{output} + e_{sources} - e_{sinks} \, , \tag{4}$$

where E_r (J) is the room kinetic energy and e (J/s) is the kinetic energy flux influencing the room space. As the kinetic energy balance is applied, the following assumption can be made about the room air motion: Any addition into the room kinetic energy will increase the room velocity level without delay. Thus, there is no “kinetic energy capacity” and the room air momentum can be considered as a chain of steady state conditions. In such conditions

$$\frac{dE_r}{dt} = 0 \,. \tag{5}$$

The introduced kinetic energy flux is

$$e = \frac{1}{2}\rho u^3 A = \frac{1}{2}\rho u^2 q = \frac{1}{2}\rho \frac{q^3}{A^2} \,, \tag{6}$$

where A [m^2] is the outlet area, u [m/s] is the initial velocity and q [m^3/s] is the volume flow rate from the source.

The potential kinetic energy sources and sinks in the room space are shown in Figure 11. It is also important to differentiate the dissipation zone from the primary zones of the supply and thermal jets. In the dissipation zone, the velocity level will be a function of the room kinetic energy. In the primary zones of the jets the velocity depends mainly on the source.

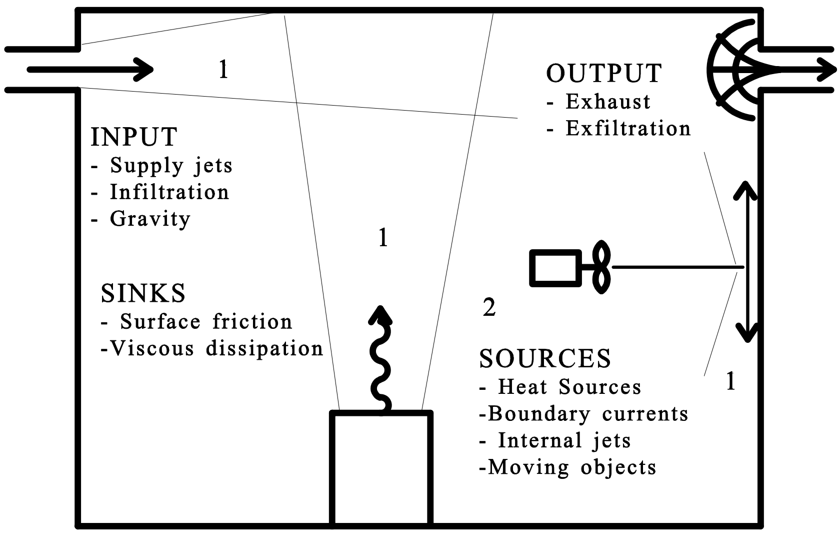

Figure 11. Kinetic energy components and zones in a room space; 1) the jet (supply or thermal) primary zone, 2) the dissipation zone.

Following simplifying assumptions were made:

- The bulk room airflow close to the wall boundaries is turbulent with no defined direction.
- The surface friction in the room occurs in a laminar sub layer, because the conditions for the development of turbulent boundary layer do not exist at the wall surfaces outside the core areas of the wall jet and boundary layer flow.
- Thus, the equations for the laminar boundary layer are valid for the most of the room surfaces.
- Turbulent stress layers in the room space outside the laminar wall boundary can be

characterized by turbulent intensity.

3.5.2 Room average velocity equation

The equation for the room average velocity that was developed through kinetic energy analysis is

$$u_r = \left(\frac{C_x^{1/2}}{\rho} \frac{e_{input} + e_{sources}}{0.664\, A_s} \right)^{1/2} \left(\frac{V_r}{A_s} \right)^{\frac{1}{6}} , \tag{7}$$

where C_x= 1.40 [$m^{11/3}/s^{5/3}$] is an empirical coefficient, ρ is air density, [kg/m^3], e_{input} and $e_{sources}$, [W], are kinetic energy fluxes from external and internal sources, A_s,[m^2], is an area of the room surfaces and V_r, [m^3], is a room volume. The ratio $C_x^{1/2}/\rho$ can be neglected in the normal range of room temperatures because the influence on the velocity is only ±2. If we reorganize equation (7), the equation for the room average velocity gets the form:

$$u_r = \left(\frac{1}{0.664} \frac{e_{input} + e_{sources}}{V_r} \right)^{1/2} \left(\frac{V_r}{A_s} \right)^{\frac{2}{3}} . \tag{8}$$

An interesting feature is that the turbulence intensity was subtracted out from the average velocity equation. This could be explained by the fact that globally, all the kinetic energy dissipation into thermal energy occurs at the wall boundaries, whereas the net dissipation inside the fluid is equal or close to zero. This makes sense if we consider a fluid cell that is heated through viscous dissipation. The heated fluid expands and forces other fluid cells to move around, thus the instantaneous heating is transformed back to kinetic energy inside the fluid.

Additional speculation could be made about the turbulent kinetic energy. Since the dissipation is completely defined by mean air motion, the turbulent intensity does not have any physical importance from the energy point of view. As a matter of fact, the whole room flow outside the jet core area is to some degree a turbulent, fluctuating motion, and the mean velocity describes its magnitude. The turbulent intensity simply describes the scale of the fluctuation. Using a similar conclusion to the supply air jet boundary, one can ignore turbulence intensity and just use the average velocity at the outlet to calculate imported kinetic energy.

3.5.3 Design algorithm - a practical application of the method

The application of the proposed calculation method is a straightforward process for evaluation of comfort conditions, in which a little input information is needed. The design algorithm consists of four steps:

(1) A collection of the input data from external and internal kinetic energy sources and room dimensions;

(2) Calculation of the kinetic energy supplied into the room, the room volume and the area of the room surfaces;

(3) Calculation of the resulting room average velocity.

(4) Together with information on room temperature conditions the average velocity can be used to estimate thermal comfort conditions of the room using PMV –index (ISO7730,1993).

3.5.4 Validation of the calculation method with experimental data

The calculated room velocity from equation (26) covers the whole room space excluding the primary zones of supply and convective jets. The occupied zone average velocity can be used for validation of the presented method given that it is not in the primary jet zone. And more importantly, the method can be utilized in practise for the evaluation of the occupied zone average velocity as well. The results of the validation are presented in Figures 12 through 15, where the predicted room average velocities are drawn as a function of the measured occupied zone average velocities.

Case 1, Concentrated supply, reduced scale. The results show an excellent correlation with high reliability between the calculated and measured velocities of fifty-six experiments, Figure 12. The slope of the correlation line is 1.0 and the correlation coefficient R^2 =0.91. The average error of the prediction is 3mm/s.

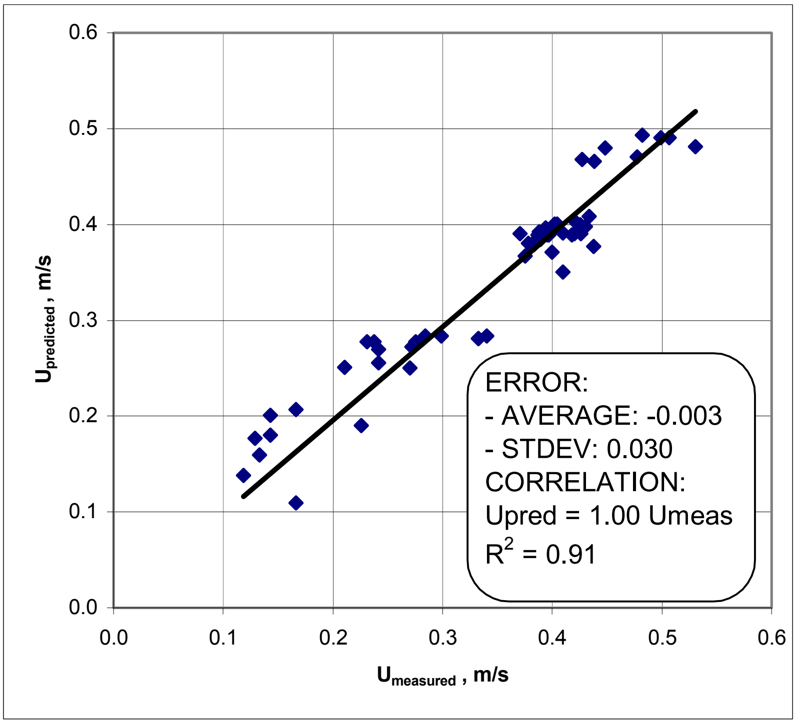

Figure 12. Concentrated supply, Reduced scale, Correlation of measured and predicted occupied zone average velocities, 56 experiments.

Case 2, Horizontal direct supply, reduced scale. The validation with thirty experiments results in a correlation curve with the slope 1.0 as well. But the R^2 -value is much lower, 0.41, than in case 1, because of the disintegration at the lowest velocities, Figure 13. This, however, is more likely a result of the measurement accuracy at such a low velocity, below 0.07 m/s, than of the calculation method. The average error of the prediction was still very small, 3mm/s.

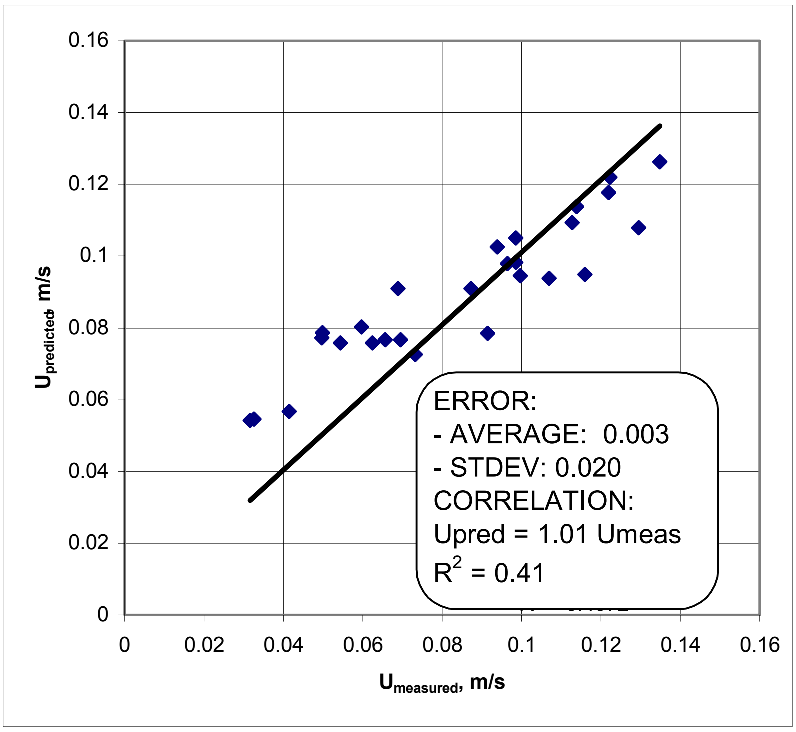

Figure 13. Horizontal direct supply, Correlation of measured and predicted occupied zone average velocities, 30 experiments.

Case 3, Vertical supply, reduced scale. The validation of twenty-seven experiments shows a clear correlation as well, but the calculated values are only 54 percent of the measured ones as can be seen from Figure 14. The reason for this result is that the occupied zone is partly within the main zone of the vertical supply jets. Thus, the surplus kinetic energy introduced directly by the supply air jet into the occupied zone increases the zonal velocity above the room average. As a result, the measured occupied zone average velocities are twice the calculated room average velocities, although a linear correlation is found. The preliminary results indicate, however, that using additional correction also the influence of the surplus kinetic energy could be taken into account when calculating the zonal average velocity.

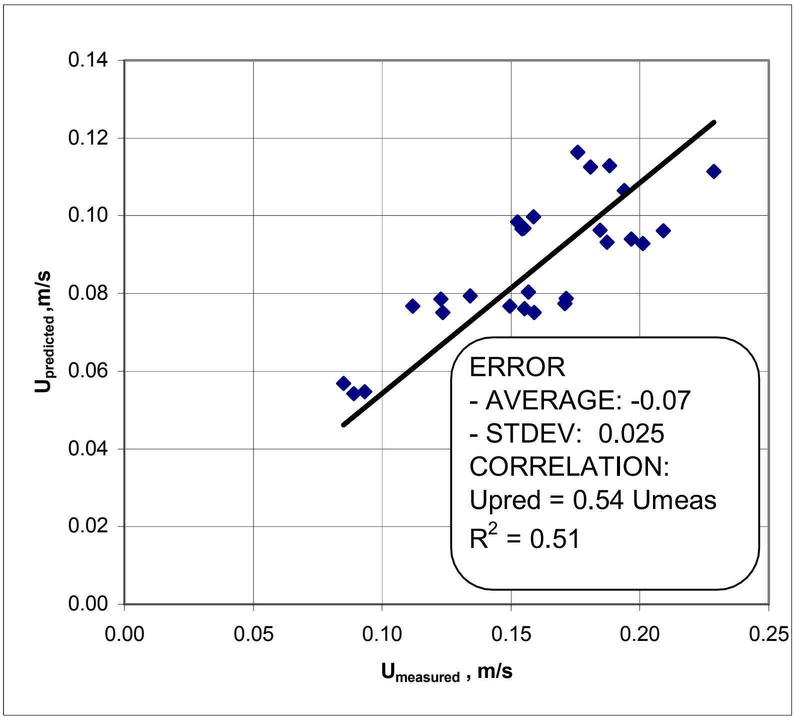

Figure 14. Vertical supply, Correlation of measured and predicted occupied zone average velocities, 27 experiments.

Case 4, Concentrated supply, full scale. The validation of the fifteen full-scale experiments with the concentrated supply method proves that the calculation method is applicable to various room sizes as well, Figure 15. The slope of the correlation line is 1.0 and R^2 is 0.87. The average error of the prediction is 3mm/s.

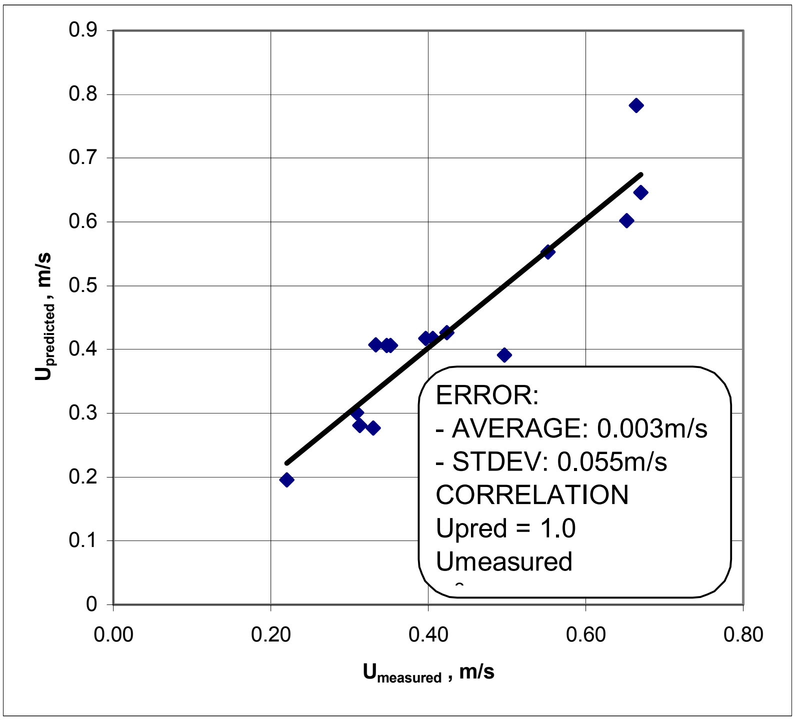

Figure 15. Concentrated jets, full scale, Correlation of measured and predicted occupied zone average velocities, 15 experiments.

3.6 Comparative analysis of measurements and CFD simulation results

3.6.1 Introduction

In the study reported in paper (VI) the airflow in a reduced scale industrial hall was simulated numerically using two different computing codes, the commercial flow solver, Fluent, with a standard k-ε turbulence model (called below as high Reynolds number code) and a university code, Finflo, with a low-Reynolds number turbulence model. The results were compared with measurements. Two different air supply arrangements, horizontal direct supply and concentrated air supply were studied, both with isothermal and non-isothermal boundary conditions.

3.6.2 Results

The measured temperature distribution was generally well predicted but there were bigger differences in the air velocity. In some parts of the room even the qualitative airflow pattern was different in the two simulations, most notably in the regions where the buoyancy forces and the inertia forces of the supply air jet are of the same magnitude and the supply air jet is deflecting near the solid boundaries. In those parts of the flow the low-Reynolds number model seems to perform better, probably because the natural convection boundary layer flow is better resolved.

Some interesting findings can be highlighted from the comparison between measurements and different computational methods using global average velocity, Table 2, and turbulence intensity, Table 3. An additional comparison is made using modified average velocities, Table 4.

Average velocity
The two simulation methods gave different quantitative results. Better results were achieved either with the low-Reynolds number code or with the high Reynolds number code, depending on the case. The air velocities with the high Reynolds number code were consistently higher than the measured ones. The low-Reynolds number simulations predicted too low velocities for concentrated supply. For comparison the kinetic energy approach was applied to these cases as well, Table 2.

Table 2. Comparison of global results, occupied zone average velocity, [m/s]

Method	Horizontal Direct Supply		Concentrated Supply	
	Isothermal	Nonisothermal	Isothermal	Nonisothermal
Measured	0.032	0.070	0.242	0.282
FINFLO	0.053	0.076	0.101	0.141
FLUENT	0.046	0.137	0.276	0.274
Kinetic Energy	0.054	0.077	0.277	0.285

Turbulence intensity.
The comparison of turbulence intensities from different methods is shown in table 3. However, one must keep in mind, that the comparison of the turbulence intensities from measurements and computations is artificial, because their physical meaning is different (Koskela 1996). Thus, the numerical turbulence values cannot be used for example for the calculation of comfort conditions. The low-Reynolds number model showed extremely high turbulence and low velocity results for the concentrated supply. This shows that major part of the kinetic energy in the computation was generated through the turbulence and not the mean flow.

Table 3. Comparison of global results, occupied zone average turbulence intensity, [%]

Method	Horizontal Direct Supply		Concentrated Supply	
	Isothermal	Nonisothermal	Isothermal	Nonisothermal
Measured	82	74	35	38
FINFLO	61	136	914	539
FLUENT	106	73	81	68

Modified average velocity.
The usage of a modified velocity is an attempt to make measurement and numerical results comparable by compensating the different treatment of the turbulence in the methods. This problem is issued in (Koskela 1996). In the approach developed both measured and computed values are corrected. However, based on the conclusions on turbulent intensity, which were drawn in chapter 3.5, it is questionable whether the measured velocity should be at all corrected. Another approach suggested for the correction of computational and directional velocity measurements, when they are used for comfort conditions evaluation, is corrected velocity presented by (Koskela et al, 2000).

The modification increased the computational values more than measured values, Table 4. An important feature was that the correction increased low-Reynolds number model velocities for concentrated supply, which were out of the class, close to the level of the other methods. The reason for this is that the turbulent kinetic energy that was seen as extremely high turbulent intensity is taken into account through modification. The results support the conclusion that it is necessary to modify CFD velocity results before comparison with comfort oriented measurements.

Table 4. Comparison of global results, modified occupied zone average velocity, [m/s]

Method	Horizontal Direct Supply		Concentrated Supply	
	Isothermal	Nonisothermal	Isothermal	Nonisothermal
Measured	0.046	0.079	0.255	0.301
FINFLO	0.071	0.130	0.200	0.262
FLUENT	0.078	0.190	0.342	0.355

4. DISCUSSION

The starting point for the research was to study room air conditioning in large spaces, especially industrial manufacturing hall environments in which the efficiency of ventilation, zonal effects and influence of the internal loads on room conditions are especially important. The research set-up and the parameters were designed to cover large range of different types of industries. The applicability of the specific results to other types of environments lies on how well the selected parameters and set-up fit with the specific environment. However, the broad results that are discussed here can be considered as generally applicable to all kind of spaces.

A classification of room air conditioning strategies was developed. The room air conditioning strategy defines the operational target for design of the room air conditioning system.
The selection of the system and the set of methods should be made in such a way that the different strategies could be applied most efficiently. Often it would be desirable to apply a different air conditioning strategy within different seasons or operating conditions. It is demonstrated in paper (IV) how the strategy can be changed by using different combinations of available methods instead of having duplicated systems.
Traditionally the room air distribution methods have been used as a basis for classification. Naming a practical room air distribution method according to a certain strategy may lead to misunderstanding of its performance in varying operating conditions. Though such a naming probably can't be avoided in business, the understanding of the basic strategies helps customers, while evaluating offered practical air distribution methods and system solutions.
Clearly, there is a need for change in current design practices. Currently the design is too much focused on evaluation of single flow elements without considering their interaction to the whole. The design of room air conditioning should be more application oriented with specific targets for the total system performance. The presented classification offers a platform for a strategic design and commissioning procedure that will result in good indoor environment that is produced with optimized and energy efficient room system.

In chapter 3.1 it was given a method to develop a safety factor to characterize non-uniformity of the contaminant concentration. It must be noted that the applicability of the safety factors developed are limited to the room conditions close to the experiments and cases with uniform contaminant sources only. Especially for zonal methods, horizontal air supply and vertical air supply, the source non-uniformity was found to have major influence on the concentration uniformity conditions.
The results of this study demonstrate the importance and the magnitude of the non-uniformity, but more research would be needed to develop safety factors for different types of air distribution in uniform and non-uniform situations.

Using the proposed kinetic energy method, a designer can estimate the average room velocity level and the acceptability of the conditions in the ventilated room. The accuracy of the calculation depends on how well the kinetic energy sources are characterized. Thus, the boundary conditions of sources, such as supply air diffusers and heat sources, should be defined carefully. In practice moving objects do also create additional movement into the room space. A method to calculate the influence of moving objects has been presented in paper (V). Some preliminary results, however, suggest that more work should be done with the moving objects,

because the draft coefficients available in handbooks don’t seem to be applicable for moving objects in a room. The influence of infiltration was not measured during the studies, but it is not likely to be a major concern in most cases as it was discussed in paper (V).
The validity of the average velocity method is limited to the dissipation zone, which is the room volume outside the primary zones of supply and convective. In those areas, the velocity distribution is still governed by the kinetic energy sources. Further development is needed to assess the influence of this remaining “excess” kinetic energy directed into the specific room zone. The preliminary results indicate, however, that using additional correction also the influence of the surplus kinetic energy could be taken into account when calculating the zonal average velocity.

CFD is a promising design tool for complex situations. However, more validation work would be needed to adopt CFD as a reliable, consistent tool for evaluation of comfort conditions. Following problem areas that were found during comparison with the measurements are:

- Diffuser boundary conditions are one of the main difficulties in the room air simulations.
- A straightforward comparisons between the measurement data and CFD simulation results is difficult because the comfort oriented, omni-directional air speed measurements are not directly comparable with the air velocity computed with the turbulence models.
- The quantitative velocity results differed from measured and they were not consistent.
- The use of turbulent kinetic energy from CFD simulations for the calculation of the turbulent intensity may result in unrealistic values.

5. CONCLUSIONS

The influence of different factors on the room air conditions, airflow pattern and efficiency of ventilation was studied in this work.

The room air conditioning strategy developed serves as a benchmark to which room system performance is evaluated. The use of the strategy approach clarifies design process as it separates the system performance from the air distribution method. The selection of the room system and the set of methods (room air distribution, exhaust, heating, cooling) should be made in such a way that the different strategies could be applied most efficiently.

A simple method for the calculation of the room average velocity was developed utilizing kinetic energy analysis and collected experimental data. The calculation method is suggested to be generally applicable. It can also be used for estimation of occupied zone average velocity given that the occupied zone is not within primary areas of supply and convective jets.
The application of the calculation method is a straightforward design process for evaluation of thermal comfort conditions as it enables the estimation of the velocity conditions that has not earlier been available for the designer.

The room obstructions do not influence on room contaminant distribution within studied obstruction levels. Whereas the room heat sources are an important factor for contaminant removal efficiency and contaminant uniformity inside the occupied zone with zonal air distribution methods. The results of the experiments show that there exists non-uniformity of the contaminant concentrations inside the occupied zone that should not be neglected when designing room air distribution.

Non-uniform distribution of heat and contaminant sources influence on the contaminant removal effectiveness and especially on the contaminant distribution within the occupied zone. However, the influence is unique for each air supply method and their ability to eliminate the influence of the source non-uniformity varies greatly. However, it can be stated that influence of non-uniformity should be carefully considered especially, when designing for stratification or zonal room air conditioning strategies.

Air velocity and temperature measurements were compared with the CFD simulations that was performed using two different computing codes. The measured temperature distribution was generally well predicted but there were bigger differences in the air velocity.
Omni directional velocities are used to determine thermal comfort conditions in standards. The results support the conclusion that it is necessary to modify CFD velocity results before comparing them with comfort oriented standards or measurements. Also, more validation work would be needed to adopt CFD as a consistent tool for quantitative evaluation of comfort conditions.

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III. Classification for the room air conditioning strategies.

IV. A strategic approach for the room air conditioning design.

V. Calculation of the Room Velocity Using Kinetic Energy Balance.

VI. Numerical and Experimental Studies on room airflow

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