Development of mechanical ventilation system with low energy consumption for renovation of buildings

Søren Terkildsen

PhD Thesis

Department of Civil Engineering
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Preface

This thesis is submitted as a partial fulfilment of the requirements for the Danish PhD degree. The research was carried out in the Section of Building Physics and Services at the Department of Civil Engineering at the Technical University of Denmark. The project was financed by a Scholarship from DTU and by the project Plan C under Gate 21. The main supervisor has been Professor Svend Svendsen, and Associate Professor Toke Rammer Nielsen has been co-supervisor, both from the Section of Building Physics and Services at the Technical University of Denmark.

Gate 21 is a non-profit organisation that creates a platform for partnerships between local authorities, private businesses and knowledge institutions to develop, test and demonstrate the energy solutions of tomorrow. The aim and objective of the project Plan C is to promote energy renovation solutions for the good of the environment and the economy. The project was funded by the European Regional Development Fund and the Trade and Industry Agency for the Copenhagen Region (Vækstforum hovedstaden).

The thesis is divided into three parts. In the first, a comprehensive study of the literature introduces the research field and the relevance of the research. The second part summarises and adds to the four scientific papers that represents the main body of work and which are appended in the third part of the thesis.

Kgs. Lyngby, June 2013

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Søren Terkildsen

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Finally, special thanks are due to the Plan C partners and project leaders involved for being an inspiring contrast to academia and keeping me in touch with the world outside, especially Per Boesgaard and Lau Markussen Raffnsøe for their input and help in carrying out the demonstration project at Vallensbæk School.

Abstract

A general reduction in total energy consumption is needed, due to the increasing concerns about climate change caused by CO_2-emmissions from fossil fuels. In 2004, the building sector accounted for 40% of the total energy consumption in the EU and the US and therefore must play a crucial role in reducing CO_2-emmissions. Over the last decade, initiatives have been taken to reduce its energy consumption e.g. by the European Union, national governments or NGOs. The initiatives have mostly focused on improving the thermal properties of the building envelope to reduce heat losses. Building services, including ventilation, therefore now represent a larger part of the total energy consumption. Mechanical ventilation has been the most widely used principle of ventilation over the last 50 years, but the conventional system design needs revising to meet future energy requirements. The increase in the use of natural and hybrid ventilation systems is intended to reduce the energy consumption for ventilation, specifically the power consumption of fans in mechanical systems, but these alternative systems have other flaws, e.g. higher ventilation heat loss. Meanwhile, little has been done to improve the performance of mechanical ventilation systems. The power consumption of mechanical ventilation depends on the flow rate, fan efficiency and pressure loss in the system. This thesis examines the options and develops a concept and components for the design of low-pressure mechanical ventilation. The hypothesis is that

A new type of low-pressure mechanical ventilation with improved indoor environment and energy performance can be developed, by optimizing and redesigning each constituent element of conventional mechanical ventilation systems with respect to pressure and the development of new low-pressure components.

The goal was to develop a mechanical system with an SFP-value of 0.5 kJ/m^3 and a heat recovery efficiency of 85% that can meet current indoor environment requirements without discomfort in terms of thermal, acoustic and draught issues. The concept was developed for a temperate climate, such as Denmark's, and the objective was to provide comfort ventilation all year round and avoid overheating through increased ventilation and night cooling. This would mean that only one system needs to be installed and mechanical cooling is unnecessary. The potential to reduce pressure losses was examined for the main constituting parts of a mechanical ventilation system and the parts that are critical for the hypothesis were identified. The system proposed consists of electrostatic precipitators for filtration, an "oversized" heat exchanger to reduce pressure loss and improve heat recovery efficiency, diffuse ceiling ventilation for air distribution, and a static pressure reset control system to control the airflow to the individual rooms. The investigation of the hypothesis is reported in four papers appended to the thesis, and the thesis summarises the results and adds further discussion and an extensive study of the literature.

Paper I introduces the concept and its performance is evaluated through simulations of a system designed for a test-case building. All the components were designed to minimize pressure losses and therefore the fan power needed to operate the system. The total pressure loss was 30-75 Pa depending on the operating conditions. The annual average specific fan power was 0.33 kJ/m^3 of airflow rate. This corresponds to 10-15% of the power consumption for conventional mechanical ventilation systems, enabling the system to help meet future energy requirements in buildings. Paper II describes the development of a static pressure reset control system using a new type of flow control damper. The performance of the control system was examined using a test set-up duct system. Measurements showed that the developed control algorithm and the flow dampers were able to regulate the airflow accurately down to 5 Pa. Paper III reports on an investigation into the performance of diffuse ceiling ventilation in a school classroom. The investigation included tracer gas, air velocity and temperature measurements and showed perfect mixing of the air in the room without any discomfort issues. The diffuse ceiling ventilation was part of a low-pressure mechanical system that included an “oversized” air handling unit and duct system and a new type of flow damper to regulate the demand-controlled airflow. The performance of the system in terms of indoor environment, pressure loss, energy consumption and life cycle cost are reported in Paper IV. The system was able to provide an acceptable indoor environment and the annual average SFP-value of the system was 0.61 J/m^3. The life-cycle cost investigation showed that some components (measures) were cost-effective but the total cost of the system as a whole was higher than the reference system.

In theory, it is possible to fulfil the claims of the hypothesis and the goals stated, but it was not possible to reach that level in practice mainly due to limitations in the conventional solutions used in the pilot systems. However, the concept and the solutions developed are believed to be a contribution to making the design of low-pressure mechanical ventilation systems realistic in the future.

Resume

Der er behov for en general reduktion af det samlede energiforbrug på grund af den stigende bekymring for klima forandringer forårsaget af CO_2-udledningen fra fossile brændsler. I 2004 udgjorde energiforbruget i bygninger cirka 40 % af det samlede energiforbrug i EU og USA og spiller derfor en afgørende rolle i at reducere af CO_2-udledningen. I løbet af det sidste årti EU, nationale regeringer og interesseorganisationer har igangsat forskellige initiativer for at reducere energiforbruget i bygninger. Initiativerne har mest fokuseret på at forbedre de termiske egenskaber af klimaskærmen for at reducere varmetabet. Installationer herunder ventilation udgør derfor nu en større andel af det samlede energiforbrug. Gennem de sidste 50 år har mekanisk ventilation været det mest anvendte ventilationsprincip, men det konventionelle design behøver en revidering for at kunne overholde fremtidens krav til energiforbruget. Naturlig- og hybridventilationssystemer bliver i stigende grad installeret for at reducere energiforbruget, specielt elforbruget til ventilatorer i mekaniske anlæg, men disse systemer have andre mangler som højere ventilationsvarmetab. Imens er det begrænset hvad der blevet gjort for at forbedre mekanisk ventilation. Energiforbruget til mekanisk ventilation afhænger af volumenstrømmen, ventilator effektiviteten og tryktabet i systemet. Denne afhandling undersøger mulighederne og udvikler et koncept og komponenter til design af mekanisk lavtryksventilation. Hypotesen er at.

En ny type mekanisk lavtryksventilation med forbedret indeklima og lavere energiforbrug kan udvikles ved at optimere og redesigne delkomponenterne i konventionel mekanisk ventilationsanlæg med hensyn til tryktab og udvikling af nye lavtrykskomponenter.

Målet er at udvikle et mekanisk anlæg med en SEL-værdi på 0.5 kJ/m^3 og varmegenindvindingsgrad på 85 % som kan opfylde gældende indeklima krav uden ubehag i form af termisk, akustisk eller træk gener. Konceptet var udviklet for et tempereret klima som i Danmark og formålet var at levere komfort ventilation året rundt og undgå over temperaturer ved øget luftskifte og natkøling. Det betyder at kun et anlæg skal installeres og brug af mekanisk køling kan undgås. Potentialet for at reducere tryktabet var undersøgt for hovedkomponenterne i et mekanisk ventilationsanlæg og de afgørende for komponenter for realisering af hypotesen blev identificeret. Det foreslåede anlæg består af elektrostatiske filtre, en "over dimensioneret" varmeveksler for at reducere tryktab og øge virkningsgraden, diffust ventilationsloft til indblæsning og dynamisk trykstyring til regulering af luftmængderne til de enkelte rum. Undersøgelsen af hypotesen er rapporteret i fire videnskabelige artikler som er vedlagt afhandlingen, og afhandlingen sammenfatter resultaterne med supplerende diskussion samt et omfattende litteraturstudie.

Konceptet er introduceret i artikel I og ydeevnen er evalueret gennem simuleringer af et forsøgsanlæg designet til en typisk kontorbygning. Alle komponenter var designet til at minimere tryktabet og dermed elforbruget til ventilatorer til at drive anlægget. Det samlede tryktab var 30-75 Pa afhængig af driftssituationen og den gennemsnitlige årlige var SEL-værdi 0.33 kJ/m^3. Dette

svarer til 10-15 % af elforbruget i konventionelle mekaniske ventilationsanlæg og kan dermed være med til at opfylde fremtidens energikrav for bygninger. Artikel II beskriver udviklingen af et dynamisk tryk (static pressure reset) styresystem med en ny type reguleringsspjæld. Ydeevnen af styresystemet var undersøgt i en forsøgsopstilling og målinger viste af den udviklede styrealgoritme og reguleringsspjældene var i stand til at regulere luftstrømmene præcist ned til 5 Pa tryktab. I artikel III er rapporteret resultaterne af en undersøgelserne af ydeevnen af diffust ventilationsloft installeret i et klasselokale. Undersøgelsen indeholdt bland andet sporgas målinger, lufthastighed og -temperatur og viste perfekt opblanding af indblæsningsluften uden nogen trækgener. Det diffuse ventilationsloft var en del af et mekanisk lavtryksventilationsanlæg som derudover bestod af et "overdimensioneret" aggregat and kanalsystem og en ny type reguleringsspjæld til at regulere de behovsstyrede luftstrømme. Ydeevnen af ventilationsanlægget med hensyn til indeklima, tryktab, energiforbrug og total økonomi er præsenteret i artikel IV. Anlægget var i stand til at opretholde et acceptabelt indeklima og det gennemsnitlige årlige SEL-værdi for anlægget var 0.61 kJ/m^3. Total økonomi beregningerne viste at nogle af delkomponenterne var omkostningseffektive, men at total økonomien for hele anlægget var højere end for reference anlægget.

I teoretiske beregninger og simuleringer var det muligt at opfylde påstandene i hypotesen og de opstillede mål. Det var dog ikke muligt at opnå samme niveau i praksis primært på af begrænsningerne ved de konventionelle løsninger brugt i forsøgsanlæggene. Konceptet og løsningerne udviklet kan dog bidrage til at gøre det muligt at designe mekaniske lavtryksventilationsanlæg i fremtiden.

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Nomenclature

Latin letters

A	Area	m^2
A	Ampere	mA
C	Concentration	ppm
E	Yearly energy consumption	kWh/m^2 per year
n	Air change rate	h^{-1}
N	Fan speed	rpm
p	Pressure	Pa
P	Fan power	W
q_v	Ventilation air flow rate	l/s
R	Pressure gradient	Pa/m
SFP	Specific Fan Power	kJ/m^3
t	Time	s
T	Temperature	°C
Tu	Turbulence intensity	-
V	Volt	V

Greek letters

ϵ	Air change efficiency	%
η	Efficiency	%
Δ	Pressure difference	Pa
σ	Standard deviation	-
$\langle\bar{\tau}\rangle$	Mean age of air (whole room)	s
$\bar{\tau}_p$	Mean age of air (point)	s
τ_n	Nominal time	s
$\bar{\tau}_r$	Actual air change time	s
$\bar{v}$	Mean air velocity	m/s

Subscript

a	Air
e	Exhaust
l	Local
n	nominal
p	Point
r	Residence
tot	Total
v	Ventilation
0	Initial
1	First operating condition
2	Second operating condition
3	Third operating condition

Abbreviations

AHU	Air Handling Unit
CAV	Constant Air Volume

COP	Coefficient of Performance
CO_2	Carbon Dioxide
CFD	Computational Fluid Dynamics
DCV	Demand Control Ventilation
DDC	Direct Digital Control
DKK	Danish Kroner
DR	Draught Rating
EC	Electronic Commutation
EPBD	Energy Performance of Buildings Directive
ESP	Electrostatic precipitator
FEG	Fan Efficiency Grade
HEPA	High Efficiency Particle Air
HVAC	Heating, Ventilation and Air Conditioning
IE	International Efficiency
IEA	International Energy Agency
LCC	Life Cycle Cost
SBS	Sick Building Syndrome
SEK	Swedish kroner
SFP	Specific Fan Power
SPR	Static Pressure Reset
ULPA	Ultra-Low Penetration Air
VAT	Value Added Tax
VAV	Variable Air Volume
VOC	Volatile Organic Compounds

Part I:

Introduction and literary review

1. Introduction

The focus on reducing energy consumption in buildings has revolved around improving the thermal properties of the building envelope. This focus has successfully reduced the energy consumption for heating in new low-energy buildings and many renovated buildings. As a result, building services, including ventilation, now constitute a larger part of the total energy consumption in buildings. The operation of building services requires power that is currently expensive to produce both in terms of investment and CO_2-emmisions. The focus is therefore wisely shifting towards the development of energy-efficient building services.

As the name implies, building services are installed to service the occupants and enable them to thrive and perform to the best of their ability. Technology today is constantly improving and the field of ventilation is no exception. Nevertheless, it will not be possible to meet future energy requirements with conventional ventilation systems (mechanical or natural) without compromising the indoor environment. This thesis presents the research performed during a PhD study conducted over the past 3 years. The research focused on developing a concept and components for low-pressure mechanical ventilation systems that can fulfil future energy requirements and help achieve the goal of a fossil-fuel-free society. An important aspect of the research was to develop and document solutions and components for the concept, so it would be accepted and used by the industry in practice. The building industry is very conservative. So, for the concept and components to be accepted and used, it is important not only to validate the performance through calculation and simulations, but also by measurements and experience from full-scale tests on demonstration projects. The thesis contains four papers that represent the main body of work, an extended summary with a literary review to link the papers together, and a general conclusion with suggestions for future work.

1.1 Aim and objective

The aim of the research presented in this thesis was to examine and document the possibilities for improving the energy efficiency and resultant indoor environment of mechanical ventilation systems. The hypothesis in this thesis is that:

A new type of low-pressure mechanical ventilation with improved indoor environment and energy performance can be developed, by optimizing and redesigning each constituent element of conventional mechanical ventilation systems in respect to pressure and the development of new low-pressure components.

The objective was to develop solutions, components and an overall concept for low-pressure mechanical ventilation. The goal was to develop a system that could fulfil the ventilation demand all year round and;

- Achieve an annual average Specific Fan Power (SFP) value of 0.5 kJ/m^3.
- Heat recovery efficiency of 85%.
- Maintain the CO_2-concentration below 1000 ppm.
- Without any draught, thermal or acoustic discomfort by complying with respective standards.

The project focused on solutions for the renovation of buildings, but the solutions are expected to be applicable for new buildings as well and be suitable for various building types, e.g. offices, apartments, schools and day-care facilities.

1.1.1 Scope

The Danish climate, building tradition and style of architecture was used as the point of reference and the focus was on balanced mechanical ventilation for comfort ventilation alone. This was the context in which the work was conducted and should be evaluated.

1.2 Methodology

Mechanical ventilation systems can be divided into the constituent elements shown in Figure 1. The methodology was to start with a study of the literature to:

1. Establish the state of the art of ventilation at system level. The focus was on balanced mechanical ventilation, but natural and hybrid ventilation was explored as well for possible solutions and inspiration.

2. Establish the state of the art of all the constituent elements, so as to identify obstacles, options, and the potential to reduce pressure losses and fulfil the aim and objective.

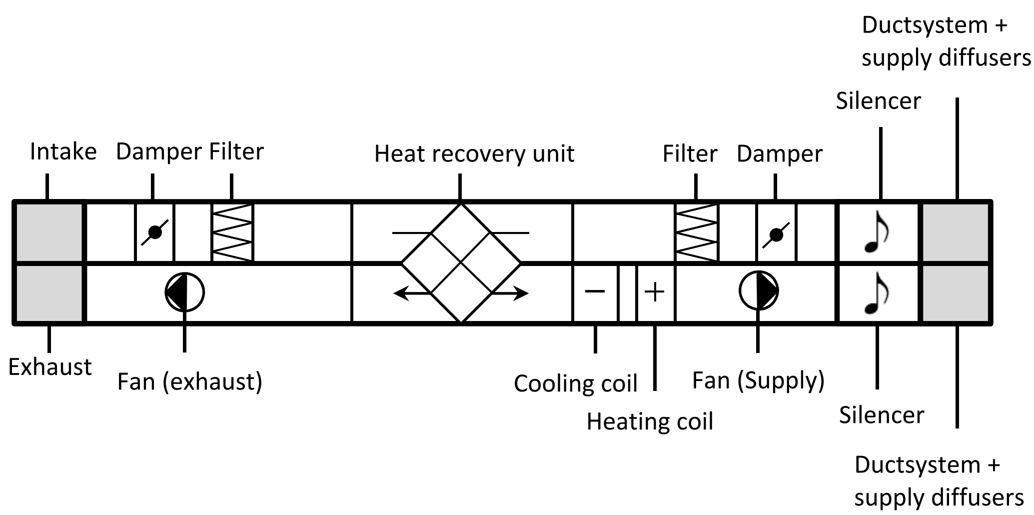

Figure 1: Constituent elements of a mechanical ventilation system (Hvenegaard, 2007).

Based on the literature study, the following focus areas were set up covering the key aspects required to satisfy the hypothesis. In parentheses is where each element or aspect is dealt with.

- Development of low-pressure ventilation concept (Paper I)
- Design and dimensioning methods for duct systems (thesis)
- Low-pressure supply concepts and diffusers (Papers I and III)
- Demand control ventilation systems for low-pressure systems (Paper II)
- Efficient heat recovery with low pressure loss (thesis)
- Low-pressure air filtration (thesis)
- Design of low-pressure air handling units (Papers I and IV)
- Effect of indoor environment on the performance of occupants (Paper III)
- Motor and fan efficiency (thesis – cases)
- Life cycle cost of low-pressure ventilation concept (Paper IV)

The research work could not and does not provide an in-depth analysis of all the focus areas listed. For the focus areas covered in the papers, however, specific solutions are developed. The performance is comprehensively documented through calculations, simulations, literature studies and measurements on test set-ups or full-scale demonstration systems. For the focus areas covered in the thesis, possible solutions are suggested and the performance is documented through literature studies and calculations alone. The solutions might be state-of-the-art components currently available, new dimensioning standards for conventional components to reduce pressure loss, or newly developed components where necessary. The focus is to document improved performance in low-pressure mechanical ventilation using the solutions or components suggested. Performance is a broad term, in this context, the focus is on reductions in pressure loss and energy consumption, but other aspects, such as indoor environment (air quality, thermal comfort and noise), comfort, cost, applicability, safety, durability and maintenance must not be compromised.

2. Background

Due to the increasing concern about climate changes caused by CO_2-emissions from fossil fuels, a general reduction in total energy consumption is needed. The building sector can play an essential role in achieving that. In 2004, the building sector accounted for approximately 40% of the total primary energy consumption in the US and the EU (Perez Lombard *et al*., 2008, EPBD, 2010). Of this, heating, ventilation and air conditioning (HVAC) systems accounted for 48% in the EU and 57% in the US, of which fans accounted for 15-50% depending on the type, design and performance of the system (Blomsterberg *et al*., 2001, Perez-Lombard *et al*., 2011). This indicates that there is a huge energy savings potential if we can develop and employ more energy-efficient ventilation systems.

To initiate and promote improvements in the overall energy performance of buildings, in 2002 the European Union launched the Energy Performance of Buildings Directive (EPBD) and in 2010 a recast was published (EPBD, 2010). Amongst other things, the directive set up a framework for the energy performance of buildings that covers space-heating, domestic hot water, cooling and lighting. The directive requires a revision of the requirements every 5 years. In the Danish context, this has already been implemented by defining 3 new energy classes for buildings in the building code, which reduce the energy framework by 25%, 50% and 75% of the 2006 level. The classes are respectively denoted energy class 2010, 2015 and 2020 after the year they become the current requirement (DBC, 2010).

The energy framework only applies to new buildings, which constitute less than 5% of the annual construction work in Denmark, and the overall goal in the Danish Building Code is to have a fossil-fuel-free building sector by the year 2035. So a critical aspect in achieving this is how to bring the large existing building mass up to an acceptable energy standard. This means that it is important to develop technical solutions that can comply with the requirements for 2020 and are suitable not only for new building but also for renovation projects.

Improving the indoor environment is another key aspect of renovating our buildings. The indoor environment is increasingly considered unsatisfactory, because of poor air quality and thermal comfort (Delsante *et al*., 2002). These aspects have led to the general term “sick building syndrome” (SBS). Several studies have documented that poor indoor environment adversely affects the performance, well-being and health of office workers (Seppänen *et al*., 2003, Wargocki *et al*., 1999, Wargocki *et al*., 2002), and similar evidence has been found for school pupils (Wargocki *et al*., 2007). Other studies have shown that the cost of this decreased productivity is far greater than the total cost of improving the ventilation systems (Djukanovic *et al*., 2002, Wargocki *et al*., 2005).

2.1 Project framework

Large parts of suburban Copenhagen were developed and built through the 1960s and 70s. The buildings from that period do not comply with current energy requirements because of their high heating and power consumption. Many of them suffer from a poor indoor environment and they now represent a substantial renovation backlog for the community. What is needed is knowledge on how to update the building mass to meet future energy requirements. In this context, the local authority of Albertslund initiated an environmental knowledge forum (Miljøvidenparken), which was a platform for companies interested in the development of energy-efficient renovation solutions. In 2009, the name was changed to Gate 21, and the partnership has since expanded to 16 local authorities, 23 companies, 2 housing associations and 5 research institutions, including the Technical University of Denmark. The overall goal is to develop and promote energy-efficient solutions to achieve a sustainable society. The purpose is to create innovative private-public partnerships and projects in the sectors of building and transport, city planning, energy and resources. One project, called Plan C is financed by the European fund for regional development and the trade and industry agency for the Copenhagen region (Vækstforum hovedstaden), and it is working on various aspects of energy renovation of buildings. This PhD research is a part of that project, focusing on the development of concepts, solutions and components for mechanical ventilation that can be used in the renovation of these buildings, reduce their energy consumption, and improve their indoor environment.

2.2 Ventilation strategies

The main purpose of a ventilation system is to provide an acceptable air quality and temperature in our buildings, so that the indoor environment does not compromise the occupant's health, well-being or performance. There are three main ventilation principles: natural ventilation, hybrid ventilation and mechanical ventilation. Each principle has a variety of different designs and system configurations, see Figure 2. It is easy to distinguish between natural and mechanical ventilation, while hybrid ventilation is a more blurry concept with no clear distinction between the designs as shown in Figure 3.

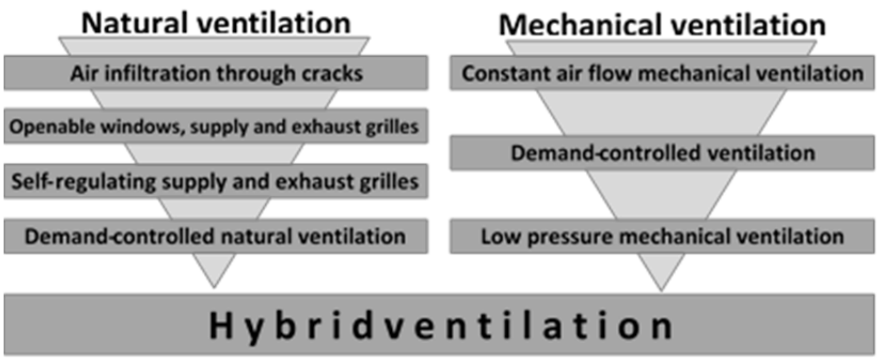

Figure 2: Development of natural and mechanical ventilation systems (Heiselberg *et al*., 2002).

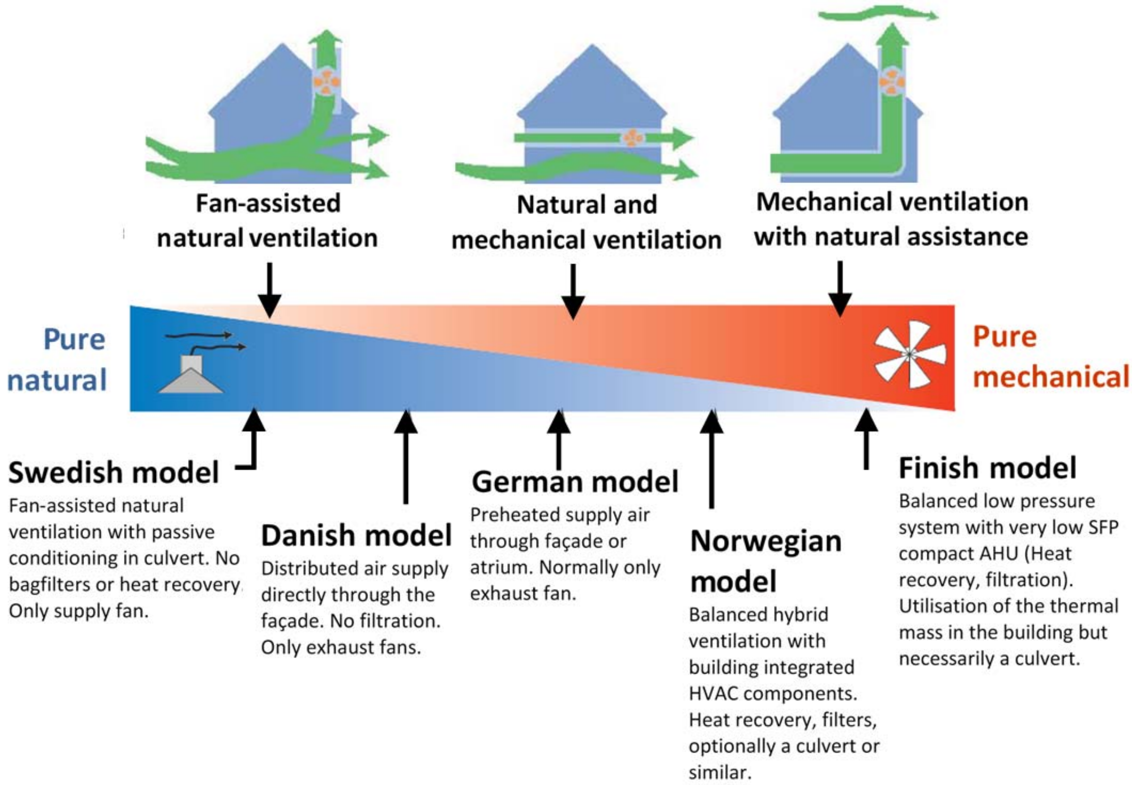

Figure 3: Illustration of the spectrum of different types of hybrid ventilation (Dokka *et al*., 2003)

Historically, natural ventilation has been used in various forms to ventilate buildings, but over the last 50 years, mechanical ventilation has been the most commonly used principle due to increased air tightness of buildings to reduce heat losses (Dokka *et al*., 2003).

Mechanical ventilation systems were introduced in the first half of the 20th century to meet indoor environment needs in our buildings. Before the introduction of mechanical systems, the climate was the determining factor in building form, not style and appearance, and comfort was achieved by passive means and architectural features built into the design (Heiselberg, 2012). Natural ventilation driven by wind and buoyancy forces makes use of building spaces to supply and extract air, typically through openings in the façade and roof which enable free cooling. In recent years, natural ventilation has experienced a paradoxical rebirth as an energy-saving measure to avoid the fan power connected with mechanical systems. Each ventilation principle has its strengths and weaknesses and Table 1 lists the typical properties of current systems (for more comprehensive lists see Liddament (1996) and Dokka *et al*. (2003). For natural ventilation, the lack of heat recovery, filtration and ability to condition and control the supply limits the performance, especially in winter (Larsen *et al*., 2006). The systems are often unable to provide the air quality and thermal comfort required and have high heat losses (Heiselberg *et al*., 2002, Delsante *et al*., 2002).

Table 1: Strength and weaknesses of natural, hybrid and mechanical ventilation, (÷)=weakness, (-)=neutral, (+)=strength.

	Natural	Mechanical	Hybrid
Heat recovery	÷	+	-
Fan power	+	÷	+
Particle filtration	÷	+	-
Air flow control	÷	+	+
Air conditioning	÷	+	-
Free cooling	+	÷	+
Thermal comfort	÷	+	+
Overall applicability	-	-	-

As a result, the focus has switched to hybrid ventilation systems with numerous projects with various approaches, while little attention is given to improving energy efficiency and solving the flaws of mechanical systems.

Hybrid ventilation systems try to combine the strengths of natural and mechanical ventilation without the flaws, but there are concerns about current hybrid designs. Many hybrid systems are two-mode, which means installing two separate autonomous natural and mechanical systems for the summer and winter period respectively. This results in excessive investment cost, and operation in the spring and autumn is challenging. Other hybrid systems are mixed-mode or single-mode and operate using thermal stacks and/or wind with an assisting fan, but they have inferior performance of e.g. draught, airflow control, filtration and inefficient heat recovery compared to mechanical systems. Several projects are described in Heiselberg *et al*. (2002) and Delsante *et al*. (2002), and these systems can fulfil the indoor environment requirements and reduce the energy consumption compared to conventional systems. However, these designs cannot be used for renovation projects without excessive cost. Solutions for the renovation of buildings are therefore needed and the potential of mechanical ventilation systems is almost uncharted. But to reduce energy consumption and improve the indoor environment will require a revision of the conventional design methods for mechanical ventilation systems.

Mechanical and natural ventilation have opposite strengths. The drawback of conventional mechanical systems is the relatively high fan power consumption required to operate the system. This is due to the high pressure losses in current systems, which are the result of the industry's focus on minimizing space use and poor integration in the building. The high pressure losses also lead to disturbance and discomfort because of high air velocities that increase noise generation and complicate the distribution of the supply air if no draught is to be caused. However, most of the flaws of current mechanical systems are in some way or another due to poor engineering and not so much the principle involved (Delsante *et al*., 2002). Natural and hybrid ventilation interact with the building elements, utilizing them to heat, cool and distribute the air, and this requires good integration in the building to function satisfactorily. Mechanical ventilation is not usually integrated in the building in the same way, because historically it has been customary to separate activity in the design process into an installations side and a construction side. In this way, integration of the installations and their interaction with the building gets neglected. This results in poor design and unnecessarily high pressure losses and power consumption.

To sum up, current ventilation solutions are not capable of fulfilling future energy requirements and the increased focus on indoor environment. New ventilation solutions are therefore needed, especially for renovation projects. There is no universal solution that will cover all building types and solve all issues, because solutions vary according to the design and use of the building, but one of the solutions could be low-pressure mechanical ventilation. By reducing the pressure loss, power consumption for fans is reduced, while the advantages of mechanical ventilation are maintained. This is challenging in renovation projects, where the building and structural design is fixed, because for some constituent elements extra space is required to reduce pressure losses. So we need to develop new components and design solutions with lower pressure losses that can improve integration and interaction with the building – to minimize space use and enable the utilization of free cooling.

3. State of the art

An immense amount of research and development has been carried out on a global scale on all aspects of ventilation e.g. indoor environment, component development, system design, control systems and energy optimization. All these aspects intertwine and they can rarely be dealt with separately, and they must all be considered in the development of new ventilation concepts and solutions. Much of the research has been carried out by the ventilation industry and is not publicly accessible. The other part has been carried out by research institutions and universities alone or in collaboration with the industry. This chapter presents the state of the art in the field of ventilation divided in as follows:

- Books and guidelines (standards)
- Papers – research and demonstration projects
- Conventional design
- Energy requirements
- Indoor environment
- Economics.

This review focuses on balanced mechanical systems for comfort ventilation. Natural and hybrid ventilation are touched upon where solutions and components could be relevant and transferable to mechanical systems. The review focuses on systems, while the state of the art for components is presented separately in the respective sub-chapters of Chapter 5.

3.1 Books and guidelines

Books in the field of ventilation are mostly textbooks describing the mathematical and physical theory necessary to design all aspects of ventilation, e.g. indoor environment, heat and mass transfer, pressure, energy, etc., along with the current best practice e.g. Danvak (2007), Ludvigsen *et al*. (2001). A textbook from Awbi (2003) focuses on the design of air distribution in rooms, and documentation of the airflow distribution through Computational Fluid Dynamics calculations (CFD) and measurements. There is also a chapter on low-energy ventilation, but that only deals with natural and hybrid ventilation; the book does not describe innovative solutions to improve energy efficiency – only the current best practice.

Several guidelines have been published by ventilation industry associations or as part of larger research projects. Liddament (1996) lists the advantages and disadvantages of all the different ventilation strategies of natural and mechanical ventilation. Thoroughly naming appropriate applications for each strategy, and gives an in-depth description of the components needed as well as examples and experience useful for the design of ventilation systems using the particular ventilation strategy. However, the dimensioning of components and recommended pressure losses are not described, and with regard to energy consumption, it is only stated that good systems have an SFP value of 1.0 kJ/m^3 while poor systems have an SFP value of 3.0 kJ/m^3. Blomsterberg *et al*. (2001) lists recommended pressure losses for all constituent components, see

Table 2, and the SFP value for a well-designed system is again 1.0 kJ/m^3 while normal-designed systems have SFP values of 5.5-13 kJ/m^3. Furthermore it is noted that very good systems have SFP values of 0.5 kJ/m^3. Almost exactly the same pressure losses are recommended in Nilsson (1995), who also gives an SFP value of 1.0 kJ/m^3 for well-designed systems and 10.0 kJ/m^3 for poorly designed systems. The average pressure losses for 100 Danish systems are presented in Hvenegaard (2007), along with recommendations for optimal pressure losses, but the power consumption of the systems was not presented. The recommended pressure losses correlate well with the previously mentioned recommendations. The data from the 100 systems showed that the duct system pressure loss was high, matching the "poor" and "current" design values from respectively Schild *et al*. (2009) and Blomsterberg *et al*. (2001). The other values correlate well (except for system effects) and this indicates that the greatest potential for reducing SFP of current systems is in the duct system.

The recent guideline by Schild *et al*. (2009) focuses on fans and motors, providing up-to-date knowledge in these areas and giving recommendations for good design of each constituent component. Also included are examples of pressure losses for good and poor design along with component pressure losses in a hybrid ventilation system, see Table 2. Again an SFP value of 1.0 kJ/m^3 is recommended for good design, while the hybrid ventilation system achieves an SFP value of 0.2 kJ/m^3. This is mainly due to reduced pressure losses in hybrid ventilation components, with little contribution from the utilization of natural driving forces, as it is stated in (Schild *et al*. 2009).

In hybrid ventilation systems, the contribution from natural driving forces (wind and stack effect) accounts for less than 1% of the energy savings in comparison to conventional ventilation systems with high pressure loss. The remaining 99% of the savings is actually a result of reduced flow resistance.

The four guidelines are useful references for design of good mechanical ventilation systems, and although three of them are 15 years old, all four recommend an SFP value of 1.0 kJ/m^3, which could fulfil Denmark's 2020 requirements, see Chapter 3.4. But none of them give examples of components, solutions or systems that live up to the recommendations.

Table 2: Recommendations and examples of pressure losses in conventional and custom-designed systems (denoted by first author).

	Blomsterberg		**Schild**			**Hvenegaard**		**Berry**	**Tjelflaat**	**Hestad**
Component	Current practice	Efficient design	Poor design	Good design	Hybrid vent.	Normal design	Optimal design	Comfort vent. mode	Grong school	NBI
Air flow [m^3/h]	-	-	-	-	-	8,000		15,840	8,000	1,440
Supply side										
Duct system [Pa]	150	100	150	100	1	280	150	11	2	19
Sound attenuator Pa]	60	0	200	0	0	Incl. duct system	Incl. duct system	9	0	0
Heating coil [Pa]	100	40	100	40	0	45	40	0	4	0
Heat exchanger [Pa]	250	100	250	100	13	140	100	60	14	7
Filter [Pa]	250	50	250	50	27	60	50	16	13	1
Air Terminal device [Pa]	50	30	50	30	12	Incl. duct system	Incl. duct system	Incl. duct system	1	Incl. duct system
Air intake [Pa]	70	25	70	25	0	-	-	20	0	11
System effect [Pa]	100	0	330	0	0	140	50	37	0	0
Exhaust side										
Duct system incl. exhaust [Pa]	150	100	370	110	1	290	120	5	5	7
Sound attenuator [Pa]	100	0	100	0	0	Incl. duct system	Incl. duct system	10	0	0
Heat exchanger [Pa]	200	100	250	100	13	145	100	60	29	3
Filter [Pa]	250	50	250	50	0	60	50	0	0	-
Air terminal devices [Pa]	70	20	30	20	0	Incl. duct system	Incl. duct system	-	0	1
System effects [Pa]	100	30	330	30	0	130	50	51	0	0
Sum [Pa]	1950	645	1800	645	84	1160	710	279	68	51
Fan efficiency [%]	15-35	62	28	63	40	-	-	-	-	-
Specific fan power [J/m^3]	5.5-13.0	1.0	10.0	1.0	0.2	-	-	0.4	-	0.14

3.2 Research and demonstration projects

Several research project annexes in the framework of the International Energy Agency (IEA) have dealt with ventilation challenges. These mostly deal with simulation (Annexes 10+25+34), control and commissioning (18+40), and ventilation systems in general (26+27) (Lebrun *et al*., 1988, Mansson *et al*., 1993, Hyvarinen *et al*., 1996, Moser *et al*., 1998, Concannon *et al*., 2002, Jagpal, 2006, Visier, 2004). Little work has been done on the development of new, strictly mechanical ventilation concepts in this framework. At the concept level the research in Annex 35, (Dokka *et al*., 2003, Heiselberg *et al*., 2002), was focused on the development of various hybrid combinations of natural and mechanical ventilation systems with a view to achieving the best of both concepts. Some of the development in fan-assisted natural ventilation and stack and wind-assisted mechanical ventilation is certainly relevant, because some of the solutions could be applied in conventional mechanical ventilation.

The few projects involving purely mechanical ventilation show that significant fan power reductions can be achieved by reducing pressure losses. Berry (2000) reports on the design and performance of an innovative mechanical system at the University of Nottingham in the UK. The system has four operation modes: summer (max), winter (max), night, and comfort ventilation, which bypasses components not in use to reduce pressure losses. The Air Handling Unit (AHU) was custom-made with specially designed intake and exhaust solutions, electrostatic precipitators for filtration, integrated ducts in the floor, and floor inlets to distribute the supply air. In this way, good contact was achieved between supply air and the thermal mass of the building, and this, combined with night ventilation and a cooling pump, was estimated to provide 5 °C of passive cooling. The total pressure loss varies between 279-343 Pa depending on the operating mode, resulting in an annual average SFP value of 0.4 kJ/m^3.

Another project is at the Media School in Grong, Norway, where a balanced mechanical ventilation system has been installed that makes use of stack and wind effects (Tjelflaat, 2000). The system has a central intake outside the building through an underground culvert. The almost constant ground temperature preheats or precools the supply air depending on the seasonal outside temperature. The culvert also acts as a part of the filtration system, in combination with a EU7 filter wall, because larger particles in the air settle on the culvert floor. The air is distributed to the rooms through a plenum under the building, and liquid coupled heat-exchangers are used for heat recovery. Table 2 lists the pressure losses for the components in the system. The total energy consumption for the system is 50 kWh/m^2 per year, but neither the system's heat recovery efficiency nor its SFP value have been disclosed.

At the Norwegian building research institute was installed a test system designed to minimize pressure losses and utilize stack and wind effect (Hestad *et al*., 1998). The duct system was dimensioned with a pressure gradient of 0.15 Pa/m, and the AHU consists of a special intake/exhaust, electrostatic precipitator and a liquid coupled heat-exchanger with an efficiency of

50%. The pressure losses in the components are listed in Table 2 and the SFP value for the system was measured to 0.14 kJ/m^3. To achieve such low energy consumptions and provide a good indoor environment, the systems mentioned require a great deal of integration. This strongly affects the building design and makes them less than ideal for use in renovation cases, where plug-and-play solutions are required.

3.3 Conventional design of mechanical ventilation systems

The conventional design approach of ventilation systems focuses on fulfilling specific indoor environment parameters and current energy requirements. Providing a good indoor environment and reducing energy consumption are currently two conflicting aspects. National and international standards list quantifiable design criteria for thermal comfort, air quality, flow rates, noise and energy consumption that are used as specifications in the tender process and the dimensioning of ventilation systems (DBC, 2010, EN 15251, 2007, CR1752, 1998).

Consulting engineers and manufacturers strive to fulfil the requirements to the best of their ability, but in the building industry the focus revolves around reducing investment costs and space use. In most cases, this leads to the installation of standard prefabricated AHUs with relatively high pressure losses. Industry tends to design and produce components that fulfil market needs and put little effort into the design of AHUs that are more efficient in terms of both energy consumption and indoor environment, because there is no demand for them. It is therefore difficult to design and validate the performance of a more energy-efficient AHU, because there is little knowledge or experience on how to go about it. This means that the AHU has more or less to be custom-made, which makes it exorbitantly expensive. For the duct systems, a pressure gradient of 1.0 Pa/m is the standard rule of thumb and is recommended in (Nilsson, 1995, ASHRAE, 2006), while it is reduced to 0.8 Pa/m in Schild *et al*. (2009). Similar recommendations are given in Malmstrom (2002) with pressure gradients between 0.5-1.5 Pa/m depending on the size of the duct, and Hvenegaard (2007) recommends 1.0 Pa/m for systems operating 16-24 hours/day, but sees 1.5-2.0 Pa/m as acceptable for lower operating hours. The origin of rule of thumb was to avoid excessive noise generation in the duct system, and it therefore rarely results in the optimal pressure loss from the perspective of energy and life cycle cost (LCC). The relatively high pressure losses in the duct system are also maintained and even desired because they ease control and distribution of the supply air. In conventional design, dampers (actuators) are inserted in the duct system to control the air flow, and it is beneficial to have the majority of the pressure loss in the duct system across the dampers. This makes the control easier and more robust against outside influences. The high pressure losses are also required in conventional diffusers to ensure efficient mixing of the supply air in the rooms. All this is what current control systems and diffusers struggle find difficult to do precisely and efficiently at low pressure losses.

Surveys made to examine the energy consumption of mechanical systems in existing buildings show that they cannot live up to the guideline recommendations. However, even the most recent data is quite old. An audit of 500 balanced mechanical systems in Sweden indicated an average SFP value of 3.0 kJ/m^3, and studies in other countries have shown similar or higher values (Nilsson, 1995 and Schild *et al*., 2009). The most recent data presented for Danish systems showed that the energy consumption follows the energy requirements, see Table 3 (Jagemar, 2001). This study showed that systems from the 1990s had SFP values of around 3.0 kJ/m^3 and newer systems from around the year 2000 had SFP values around 2.5 kJ/m^3. This shows that the energy efficiency of conventional mechanical ventilation systems is intertwined with the development of the standards and regulations, and not the best practice guidelines. Current ventilation systems cannot meet future energy requirements, whereas other areas, such as windows or lighting systems already have solutions that can meet the requirements. In other words, the ventilation industry is lagging behind. There is a sensible trend towards including energy consumption and taking the LCC (and not just the initial investment) into account when choosing the appropriate ventilation system. This will help the implementation of more energy-efficient systems, but it does not take into account the benefits of improving the indoor environment beyond the requirements.

3.4 Current and future energy requirements

As part of the EPBD framework, new low-energy classes have been defined in the Danish Building Code (DBC, 2010). Denmark is currently the only member with requirements defined until the year 2020, also referred to as the “Energy framework”. The energy framework is the maximum allowed energy requirement (*E*) for a building and includes the energy consumption for heating, ventilation, domestic hot water, cooling and lighting (only non-residential buildings). Renewable energy production from e.g. photo-voltaic or solar collectors can be subtracted to help fulfil the energy framework. For mechanical ventilation systems, the requirements for heat recovery efficiency and the SFP value are tightening as shown in Table 3.

Table 3: Evolution of maximum SFP value in the Danish Building Code and energy framework for 2020 (DBC, 2010).

Commercial buildings	**1995**	**2006**	**2010**	**2015**	**2020**
Maximum SFP value VAV [kJ/m^3]	3.2	2.5	2.1	-	1.5
Heat recovery efficiency [%]	65	65	70	-	75
Reduction in SFP value [%]	-	0	16	-	40
Energy framework					
$E_{\text{Residential}}$ [kWh/m^2]	-	$30+\frac{10}{A}$	$52.5+\frac{1650}{A}$	$30+\frac{1000}{A}$	20
$E_{\text{Non-residential}}$ [kWh/m^2]	-	$30+\frac{10}{A}$	$71.3+\frac{1650}{A}$	$41+\frac{1000}{A}$	25
Energy framework reduction [%]	-	0	25	50	75
Primary energy factors					
Electricity [-]	-	2.5	0.8	2.5	1.8
Heating (gas or oil) [-]	-	1	1	1	1
District heating [-]	-	1	2.5	0.8	0.6

The requirements for 2015 have not been specified, but in 2020 systems must have a maximum SFP value of no more than 1.5 kJ/m^3. This is a 40% reduction compared to 2006 level, but the energy framework is reduced by 75% in that period. In the calculation of the energy framework, heating and electricity are multiplied by primary energy factors to reflect the CO_2 footprint of the production and distribution. These also change due to the development of district heating, wind power and renewable energy technologies. The annual power consumption for mechanical ventilation in a typical office building is around 13-15 kWh/m^2 with an SFP value of 1.5 kJ/m^3 and including the primary energy factor for 2020. This means it will constitute more than 50% of the energy framework. This leaves little room for lighting, heating and hot water installations, and it will be almost impossible to meet the requirements without the use of renewable energy sources. The building code does not give separate specifications for building services for renovation projects and by default they should comply with the requirements for new buildings, though exemptions can be made if building design or exorbitant cost hinders installation.

3.5 Motor and fan efficiency

New European regulations on motor efficiency were introduced in 2011 (IEC 60034-30, 2011) and removed a large part of the current motors on the market. The standard classifies motors over 0.75 kW into four classes, denoted IE1-4 (International Efficiency), see Table 4.

Table 4: Classification of motors in IEC 60034-30 (2011).

Class	Name	Year
IE1	Standard efficiency	
IE2	High efficiency	2011
IE3	Premium efficiency	2015 (7.5-375 kW) 2017 (0.75-375 kW)
IE4	Super premium efficiency	-

IE2 is the current standard and IE3 will be introduced in 2015 or 2017 depending on the motor size. Motors that can fulfil IE4 are already on the market, and by 2020 it is possible an IE5 will have been developed, which will help reduce energy consumption. Most current ventilation systems have IE2 motors as standard, but IE3 or 4 can be chosen at additional cost. The efficiency of a motor depends on the motor size and the load, but there is considerable variation between different types and manufacturers. Figure 3 shows (left) a comparison of the efficiency of IE2 and IE3 motor depending on the motor size, and (right) the efficiency of an 11 kW IE2 and IE3 motor operating at different loads (Hvenegaard, 2007).

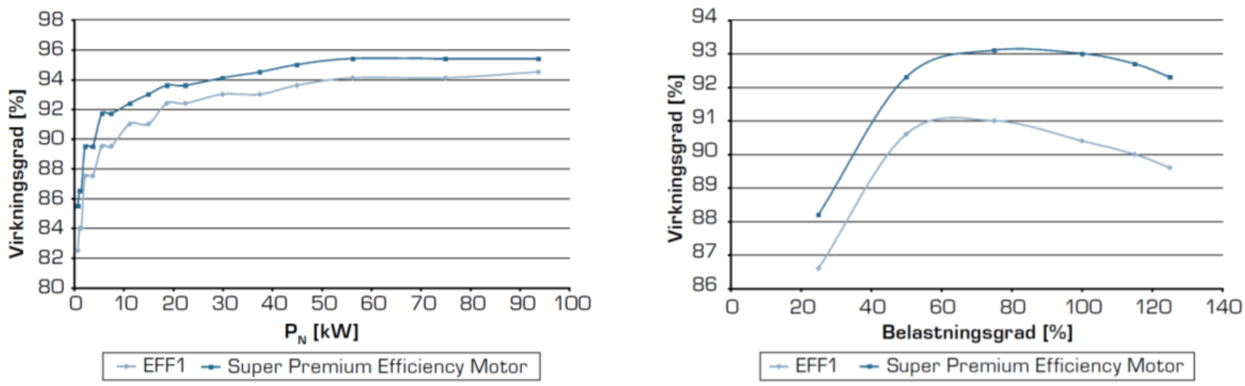

Figure 3: (left) comparison of IE2 and IE3 motor efficiency depending on motor size, (right) comparison of IE2 and IE3 motor efficiency operating at partial load.

The Figure shows that the IE3 motor has 2% higher efficiency and that it maintains this higher efficiency at partial load. This is useful in VAV systems with variable loads. New EC motors (Electronic Commutation) can fulfil class IE4 and are already on the market. Their advantage is that the speed can be regulated without the use of frequency control, which normally has efficiencies of 94-98% depending on the motor size. The standard ISO 12759 (2010) defines a grading system for fans denoted Fan Efficiency Grade (FEG). The FEG grades are arranged in steps of 6% starting at 95% and the grading applies to all fans irrespective of type, e.g. axial or centrifugal. The challenge is to choose the fan type and size that will operate at, or close to, peak efficiency under the operating conditions. This is quite simple for constant air volume (CAV) systems but challenging for variable air volume (VAV) systems because you need to know the operation range and duration at partial loads to select the optimal fan size. Centrifugal fans with backward-bending blades (B-wheel) are the most commonly used in conventional systems, while axial fans are the most appropriate for low-pressure systems. Most fan manufacturers have selection and sizing software to estimate the efficiency of a fan for a given combination of flow rate and pressure loss. Unfortunately, fans are commonly sized so that their peak efficiency occurs at peak load (Schild *et al*., 2009). This results in fans operating with reduced efficiency most of the time under partial load. Many fans are also sold in standard AHUs sized for a given range of pressure loss and flow rate and therefore not optimized to the actual partial load combinations.

3.6 Indoor environment

People in the developed part of the world spend on average 90-95% of their time indoors and their health, comfort and performance are strongly associated with the indoor environment in our buildings. So it is important to provide adequate ventilation to ensure a good indoor environment.

Ventilation is strongly associated with comfort and health, and is a key factor in maintaining a good indoor air quality for occupants (Wargocki *et al*., 2002). Several studies have showed that poor air quality adversely affects the health and performance of human beings. Most of these studies were carried out in laboratories where the subjects knew they were not performing real office work, only staged examples of it, to avoid outside influences. The studies mostly focused on adults performing office work, and it has been established that adult performance can be increased by up to 5% by maintaining the CO_2-concentration below 1000 ppm (Mendell *et al*., 1993, Seppänen *et al*., 1999, Wargocki *et al*., 1999 and 2002). The effect on school pupils is less documented, but there have been several studies showing that the air quality has greater effect on children compared to adults and their performance can be improved by up to 15% (Myrhvold *et al*., (1996), de Gids 2006, Wargocki *et al*., 2007 and Shaughnessy *et al*., 2007, Bakó-Biró *et al*., 2011). The most comprehensive study by Wargocki *et al*. documented a 2% performance increase per degree below 26 °C and an increase of 14.5% when the supply flow rate was doubled. Despite this knowledge, occupants in new buildings increasingly perceive the indoor environment as unacceptable due to poor air quality and thermal comfort (Delsante, 2000). This situation is the same in existing buildings. A recent study of Danish schools showed that 56% had CO2-concentrations above the maximum level recommended in the Danish Building Code and the Working Environment Authority of 1000 ppm (DBC, 2010, DWEA, 2012). Similar findings are reported for Holland and Sweden (Jacobs *et al*., 2008 and Smedje *et al*., 2002).

The air quality in buildings depends on the number of occupants, their activities, the building, its materials, the interior, and the ventilation system. Health effects are typically connected to specific substances and particles, like CO_2 or dust. Attempts to reduce the concentrations of individual substances or particles will automatically also lead to a reduction of others and improve the indoor environment. Comfort is more indefinable and depends on the individual person, as it is connected to the perceived air quality and thermal sensation. The air quality is immediately perceived as good or bad when entering a room, but the human body quickly adapts and does not notice increases in CO_2-concentrations, for instance, until affected by odours, headache or irritation of mucous membranes or respiratory system. Long-term exposure to poor air quality, however, can lead to more serious effects like asthma or cancer. If the air flow rate and air quality comply with current requirements (EN 15251, 2007), most health and comfort problems can be avoided. Thermal comfort depends on the temperature, air velocity and the person's activity level and clothing level. The thermal comfort is immediately perceived as either too hot or too cold which more directly affects the comfort of occupants, so the temperature should comply with the requirements to achieve the least amount of dissatisfaction (CR1752, 1998).

3.7 Economics

The cost of a ventilation system or concept is probably the most important parameter to achieve realization in practice. Several literature sources discuss the economics of systems that in some way differ from conventional mechanical ventilation. Most of the studies elaborate on hybrid or natural ventilation, while there is little data on improvements to conventional mechanical ventilation, such as low-pressure mechanical ventilation. Nevertheless, these studies are relevant because they give an indication of the cost of new ventilation concepts and can be used as references. In IEA Annex 35 (van der Aa, 2002) the cost of several hybrid systems installed around the world are presented. Many of the systems are located in Scandinavia, and were therefore designed to meet indoor environment requirements under the same climate conditions as the work of this thesis. The systems were found to be neither as cheap as the cheapest nor as expensive as the most expensive mechanical system in a reference building. Various types of hybrid system are presented and data for the reference system and building are not provided, so it is difficult to examine the cost of the different types of systems. In Berry (2000) and Tjelflaat *et al*. (2002), the cost of the system is not elaborated. Hestad *et al*. (1998) estimated the pilot system presented to have 50% higher initial cost than a standard system. However, the LCC over 20 years was only 10% higher with the current electricity price. If the expected increase in electricity prices were taken into account, the pilot system would probably have the lowest total cost.

A popular article by Førland-Larsen *et al*. (2009), calculated the LCC for five different systems: natural ventilation, hybrid ventilation, and mechanical ventilation with CAV, VAV or VAV+cooling. The hybrid system was two-mode with natural ventilation in the summer and decentralized mechanical ventilation with an SFP value of 2.1 kJ/m^3 and heat recovery efficiency of 80% in the winter. The mechanical ventilation systems were centralized with similar SFP values and heat recovery efficiency and the cooling coefficient of performance (COP) was 3. The LCC was determined based on construction cost, energy consumption and energy price for heating and power (including primary energy factors) and an interest rate of 4%. The results were very favourable for natural ventilation with 40% lower LCC over 20 years than hybrid and mechanical CAV and 45% and 50% lower compared to VAV and VAV+cooling, respectively. The high SFP values for the mechanical systems were the main reason for the higher LCC, but higher installation cost also contributed. However, the study was somewhat flawed. It is well established that natural ventilation causes draught problems and high heat loss under Danish climate conditions. So pure natural ventilation is not a valid solution. Moreover, the maintenance cost for the mechanical systems was assumed to be the same, although it requires more time to service the decentralized unit in the hybrid solution.

There is no definitive way to determine whether one concept is cheaper than another because it depends on the design and use of the building and therefore on a lot of assumptions. In addition, the discussion is influenced by personal agendas from manufacturers and invalid conclusions. This chapter was not intended to resolve the discussion, but only to give the general conclusion that the LCC of new ventilation concepts is neither lower nor higher than that of conventional systems.

3.8 Concluding remarks

There is quite a big discrepancy between the energy consumption data in Hvenegaard (2007), Jagemar (2001) and Nilsson (1995) on current standard designed systems and the best practice of the guidelines. All the systems reported had SFP values of 2.5-3.0 kJ/m^3, with newer systems around 2.0-2.5 kJ/m^3, and no examples of systems with SFP values of 1.0 kJ/m^3 were found elsewhere in the literature. This is despite the fact that guidelines for the last 15 years have unanimously recommended 1.0 kJ/m^3 as best practice. Only a few custom-designed mechanical ventilation systems were found that could live up to the recommendations. For one thing this is due to the industry focusing on minimizing space use and reducing initial cost while just fulfilling the energy requirements, but it is also due to the lack of low-pressure components and solutions on the market. The market for energy-efficient ventilation focuses on natural and hybrid solutions, leaving mechanical ventilation side-lined with a reputation of being energy-consuming and noisy. Instead, we should be trying to solve and improve the design of conventional mechanical ventilation systems in terms of not only energy efficiency but also indoor environment aspects such as air quality, draught and noise.

In general, building services are treated as a segregated activity from the construction design in the early design phases. But this means that the interaction with the building, e.g. its thermal mass, is ignored, and this is often the case with mechanical systems. Hybrid systems strongly affect the building design both in order to function and to achieve low energy consumption, so they require a different collaborative approach to building design and this is looked upon more positively because it is an intuitive way to save energy. Huge efforts are therefore made in the design process to integrate natural or hybrid ventilation systems, while the integration of mechanical does not get the same attention. If the same attention and liberties were given to mechanical ventilation, larger AHU and duct systems with reduced pressure loss could be installed. The improvement in motor and fan efficiency would help reduce the power consumption of mechanical systems, but new and better components and solutions for the other constituent elements would also be required – especially in the area of control and distribution to improve the energy efficiency to the level of the custom-designed examples. Of course, this is more challenging in renovation projects, but Berry (2000), Hestad *et al*. (1998) and Tjelflaat *et al*. (2001) have shown that it is possible to make mechanical systems with very low energy consumption. This would help meet future energy requirements and it could be a better economic solution to invest in efficient ventilation systems instead of investing in renewable energy sources. It is well established that the use of new ventilation concepts increases initial costs, but there is no such certainty about the LCC. The literature examined shows that the savings potential in mechanical ventilation is huge and that reductions are needed to meet future energy requirements and the goal of a fossil-fuel-free building sector. But knowledge and solutions are lacking on how to design and develop the mechanical ventilation systems of the future.

Part II:

Summary of work

4. Concept proposal

The literature study shows that there is a huge potential in reducing pressure losses to improve the efficiency of mechanical ventilation systems. In this chapter, the low-pressure concept presented in Papers I and IV is further described with the development process and the theory behind it.

4.1 Development of concept

Heat recovery is essential if we are to minimize energy consumption and it is a well-established part of current mechanical ventilation systems, but only minor improvements can be obtained here, see Chapter 5.1. The largest energy-saving potential is in the fan power used to transport the air in the system. The fan power P required to transport the air depends on the volume flow rate q_v, the total pressure loss Δp_{tot} in the system, and fan and motor efficiency η_{tot}.

$$SFP = \frac{\sum P}{q_v} \qquad (1)$$

$$P = \frac{q_v \cdot \Delta p_{tot}}{\eta_{tot}} \qquad (2)$$

$$SFP = \frac{\Delta p}{\eta_{tot}} \qquad (3)$$

The relatively high recommended pressure losses listed in Table 2 of conventional mechanical systems is therefore the direct cause of the fan power consumption needed to operate the system. It is therefore essential to improve the integration of ventilation systems in buildings and to decrease pressure losses in components and AHUs if we are to reduce energy consumption (Dokka, 2003, Blomsterberg *et al.*, 2001). Reducing the high pressure losses will also avoid the consequent high air velocities which can create discomfort problems with noise and draught (Delsante, 2000 and Malmstrom, 2002). The power consumption of a fan is governed by three fan laws or affinity laws.

$$\left(\frac{q_{v1}}{q_{v2}}\right) = \left(\frac{N_1}{N_2}\right) \qquad (4)$$

$$\left(\frac{\Delta p_1}{\Delta p_2}\right) = \left(\frac{N_1}{N_2}\right)^2 \qquad (5)$$

$$\left(\frac{P_1}{P_2}\right) = \left(\frac{N_1}{N_2}\right)^3 \qquad (6)$$

The laws describe the relationship between volume flow rate, pressure loss p, fan speed $N$and power consumption with the fan size held constant. Simply stated, the laws say that twice the air flow requires twice the fan speed and that twice the pressure loss increases the fan speed by a factor of 4 and the fan power by a factor of 8. This shows that the greatest potential is in reducing the pressure losses throughout the system, but that another key factor is to reduce the flow rate. This can be done by only ventilating when necessary, and in accordance with the actual demand; studies show that the control system has a significant effect on energy consumption in terms of both operating hours and required fan power (Seppänen, 2007 and Wei, 2004).

These changes will lead to higher initial costs, but since ventilation systems remain in service for 30 years, the lower operational costs for energy will outweigh the higher installation and purchase costs (Dokka, 2003). When developing, optimizing and designing a new ventilation system, the challenge is to find the optimal balance between economics, air quality, thermal comfort, energy consumption, and environmental impact in periods with cooling or heating over the year (Heiselberg, 2002). The primary purpose of a ventilation system is to maintain an acceptable indoor air quality and thermal comfort in our buildings. The ventilation system must therefore be able to meet this requirement because otherwise it loses its function (Blomsterberg *et al*., 2001).

4.2 Proposal

The low-pressure mechanical ventilation concept presented here is believed to eliminate the flaws of conventional mechanical ventilation systems without introducing the flaws of the presently known concepts for hybrid and natural ventilation.

The ventilation system must provide an acceptable atmospheric and thermal indoor environment that meets the ventilation requirement all year round with minimal energy use. This means that only one system has to be installed and this reduces costs compared to a 2-mode hybrid system and it makes control easier because only one system is needed. The concept was developed for a temperate climate like Denmark's; this implies the use of heat recovery in the winter period, the option of increased air flow rate with a bypass of the heat exchanger in the summer, and night cooling in warm periods to reduce the cooling demand during the day. Night cooling is defined as any ventilation required outside working hours to lower the temperature in the building. In this way, all ventilation needs can be met, and it is only necessary to install one system without active cooling to obtain an acceptable indoor environment in the building. To achieve this without excessive energy use, the fan power required must be reduced, by optimizing and dimensioning every component and the design as a whole to minimize pressure loss. The ventilation system was designed as a conventional mechanical system with an air handling unit and a duct system to distribute the air, and using components available on the market to reduce development and purchase costs.

5. Components

The realization and implementation of low-pressure mechanical ventilation will require the development of new solutions and components. This chapter discusses the status of the main constituent parts listed in Chapter 3.1 and their potential for reducing pressure loss. For some parts, new components and solutions have been developed and tested and these are described in detail, while others are only discussed theoretically in a study of the literature. Solutions are evaluated primarily on their reduction in pressure loss, and secondly on their cost, integration and efficiency.

5.1 Heat recovery

Heat recovery is an essential and well-established part of a modern AHU and, in terms energy efficiency, it is the key advantage of mechanical systems over natural and hybrid systems. The heat recovery efficiency in new systems is about 80-85%, so they can already meet the efficiency requirements for 2020 listed in Table 3. The potential for additional energy savings for heating are therefore limited, especially considering the need to remove internal gains to avoid overheating. Focus should instead be on reducing the pressure loss that is between 100-250 Pa in current systems, see Table 2. Several heat recovery solutions have been developed for hybrid systems, such as counter-flow stack-driven air-to-air heat exchangers (Schultz *et al*., 1994), heat pipes interjected in exhaust chimneys (Shao *et al*., 1998), and liquid-coupled heat exchangers (Overgaard *et al*., 2002). These alternative solutions reduce the pressure loss to about 1-5 Pa, but the efficiency also decreases to about 40%. Moreover, the liquid-coupled heat exchangers use power to drive the liquid circuit. In other research, liquid-coupled heat exchangers have achieved efficiencies of 58% (Hestad *et al*., 1998) and 55-60% (Tjelflaat *et al*., 2000), the latter though at relatively high pressure losses of 14 and 29 Pa on the supply and exhaust sides, respectively. The development of heat exchangers is on-going. In a recent study, the efficiency of a liquid-coupled heat exchanger developed for hybrid ventilation systems was improved to about 70% (Hviid *et al*., 2010). This heat exchanger, however, is quite large (3.0x1.92x0.31) and not suitable for most renovation cases. Moreover, it has issues with high power consumption for the liquid circuit.

Another approach is to reduce the pressure loss by increasing the size of a conventional solution, such as a rotary or counter-flow heat exchanger, thereby increasing the heat exchanger area and reducing the pressure loss. A commercial rotary heat exchanger with an efficiency of 80% and pressure loss of 49 Pa is used as reference in Hviid *et al*. (2010). Its size is considerably smaller (1.08x1.08x0.32) than the liquid-coupled heat exchanger they developed. A rotary heat exchanger was also used in Berry (2000), and its pressure loss was 60 Pa with an efficiency of 84%. The heat exchanger was quite large (Ø2.9m), but it also handled comparably more air and the unit had a bypass option to reduce pressure loss outside the heating season. A bypass option, either partial or full, is a good and simple way of reducing pressure loss outside the heating season. It can also be used to adjust the heat recovery efficiency in periods with high thermal loads where maximum efficiency is not needed.

A study of heat exchangers from different manufacturers (Enventus, 2011, Hoval, 2011, Klingenburg, 2011) shows that rotary heat exchangers have the greatest potential to reduce pressure losses (~30 Pa) with increasing the size. At the same time, their heat recovery efficiency increases to above 90%. The pressure loss cannot be further reduced because temperature stratification can occur and cause freezing, leading to impaired performance and a risk of breakdown. Furthermore, a rotary heat exchanger requires a motor to operate and, depending on the exchanger size, this will use about 100 W. This aspect was not discussed in Berry (2000) or Hviid *et al*. (2010).

The research for Paper I used a counter-flow heat exchanger with an efficiency of 89% and a pressure loss of 45 Pa. The exchanger was larger (2.23x2.23x1.24) than an equivalent rotary exchanger, but the total annual energy consumption was almost identical and about 15% lower than the annual consumption of a liquid-coupled heat exchanger, see Table 5.

Table 5: Calculation of annual energy consumption for different heat exchangers.

	Pressure loss [Pa]	Heat recovery efficiency [%]	Power use [W]	Operating hours winter [h]	Fan power [kWh]	Heat exchanger power [kWh]	Heat loss [kWh]	Total energy consumption [kWh]
Counter-flow heat exchanger	45	89	-	1449	585	-	1869	2454
Liquid-coupled heat exchanger	2	70	45	1449	410	65	2376	2851
Rotary heat exchanger	27	94	100	1449	506	145	1769	2420

All three solutions could applicable in specific cases without significant increases in energy consumption, e.g. a rotary heat exchanger to limit space requirements or a liquid-coupled heat exchanger if the intake and exhaust are not connected. With the liquid-coupled heat exchanger, the reduction in pressure loss and fan power does not make up for the increased heating loss due to lower efficiency. This, combined with the additional power use for pumps, gives this solution the highest total energy consumption. The motor power required for the rotary heat exchanger combined with fan power results in the highest total power consumption, but this is outweighed by the increase in heat recovery efficiency, if primary energy factors are not taken into account. The primary energy factors for the year 2020 are listed in Table 3. When these are taken into account, the total energy consumption is 2174 kWh for the counter-flow, 2281 kWh for the liquid-coupled, and 2233 kWh for the rotary heat exchanger. The counter-flow heat exchanger was therefore chosen for its lower energy consumption under the energy framework, but also because it has fewer moving parts that need maintenance and does not suffer from the risk of leakage between air streams in rotary heat exchangers, especially at low pressure (Sørensen, 2007 and Jensen, 2008).

5.2 Filtration

In conventional mechanical ventilation systems, the exhaust and supply are filtered by bag filters made of synthetic materials (glass fibre, polyester). The filters have two objectives: 1) to remove pollutants from the supply air that could be harmful to the occupants, primarily pollen and traffic soot, and 2) to protect the AHU components and duct system from soiling that otherwise will result in impaired performance. During operation, the bag filters collect pollutants that can have negative effects on the indoor environment, so they need to be changed regularly (Bekö *et al*., 2008). Furthermore, the pressure loss in bag filters is relatively high and increases as the filters get soiled, see Table 2. This increases the fan power required to operate the system and is not compatible with the low-pressure concept. In the hybrid systems listed in Table 2, the filtration pressure loss is reduced by either increasing the filter area (Tjelflaat *et al*., 1997) or by using electrostatic precipitators (ESPs) (Berry, 2000, Hestad *et al*., 1998). Increasing the filter area requires space that is usually a scare resource in renovation projects, but there is constant development in filter materials using nanotechnology (Barhate *et al*., 2007, Hassan *et al*., 2013). The current focus, however, is on improving filter efficiency for the filtration of oil, water, chemicals, gases and combustion processes, and not on reducing pressure loss. In the field of ventilation the technology is only beginning to be used for HEPA (High Efficiency Particle Air) and ULPA (Ultra-Low Penetration Air) filters that are not commonly used in comfort ventilation systems. If this technology is applied to conventional air filters, it is believed that pressure losses could be reduced. The use of ESPs in low-pressure systems is promising because the pressure loss is about 2-10 Pa and they have high filtration efficiency. Moreover, the particles accumulated in the filter are removed regularly, avoiding the negative aspects of bag filters described in Bekö *et al*. (2008). The negative side of ESPs is their high purchase cost, high operation cost due to power consumption and maintenance because the collector plates accumulating the particles need to be cleaned. Furthermore, old ESPs produce ozone rich in particles and create Volatile Organic Compounds (VOCs) which are harmful to human beings, but this problem has been solved in new ESPs. The power consumption depends on the number and size of the particles ionized in the air, so it is difficult quantify the power required precisely because it depends on the surrounding environment. Manufacturers of ESPs and studies using ESPs also provide varying information on the power consumption, as listed in Table 6.

Table 6: Pressure loss and power consumption of various electrostatic precipitators.

	Peak pure air (2011)	AP electrostatic filters (2011)	United air specialists (2011)	Berry (2000)	Hestad *et al*. (1998)	Tjelflaat *et al*. (1997)
Flow rate [m^3/h]	4100	1300	1700	14400-25920	1440	-
Pressure loss [Pa]	2	10	2	16	1	12
Power consumption [W]	60	16	104	28	60	0

The United Air Specialist ESP was used in Hviid *et al*. (2010b), and it accounted for 89% of the total power consumption of the one-mode passive ventilation system. In Paper I, two Peak Pure Air ESPs were used, and their power consumption accounted for 57% of the total power consumption of the system. These studies were both theoretical, and the hybrid systems in Berry (2000) and Tjelflaat *et al*. (1997) do not elaborate on or discuss the annual power consumption of the ESPs or include it in the SFP-value of the system. Whether ESPs are an energy-saving solution compared to bag filters very much depends on the power consumption. Table 7 shows the annual power use of various ESPs and their effect on fan power consumption. The calculations were made during the optimization process of the system described in Paper I by varying the filter properties and otherwise using the same components, flow rate and calculation method. The power consumption for the ESPs was multiplied to match the required flow rate of 1.132 m^3/s (4075 m^3/h) based on the design flow rates in Table 6. The power consumption of the ESPs from Berry, however, was not reduced to match required flow rate, and the energy consumptions listed are excluding primary energy factors.

Table 7: Calculation of annual energy consumption for electrostatic precipitators.

	Operating hours [h]	Fan power [kWh]	Filter power [kWh]	Total power consumption [kWh]
Peak Pure Air	3128	585	751	1335
AP electrostatic filter	3128	614	601	1215
United Air Specialist	3128	585	3902	4487
Berry (2000)	3128	637	1050	1687
Hestad *et al*. 1998	3128	585	2252	2837
Tjelflaat *et al*. 1997	3128	623	0	623

Increasing the filter area of bag filters as in Tjelflaat *et al*. (1997) resulted in the lowest total power consumption. This solution requires more space, which is usually a scarce resource in renovation projects, and it incurs high maintenance costs changing filters to maintain the low pressure loss. The AP electrostatic filter had the lowest total power consumption of the ESPs, but the Peak Pure Air ESP was used in Paper I. This was a conservative choice because the documentation was found to be most valid and because the product had been used before in Hestad *et al*. (1998). The documentation for the United Air Specialist ESP specifies a 5 mA electrical charge for the ioniser and 0.5 mA for the collector plates at 230 V, – information that is not given for the other products. This would give a power consumption of 1.25 W, well below the specified value of 104 W, so the actual power consumption is difficult to quantify. Documentation of ESP power consumption in practice is needed to determine whether this technology might be a beneficial solution for comfort ventilation. An overall investigation of the relationship between filter power, filter efficiency, pressure loss, fan power, cost and maintenance is needed to find the optimal filter solution, but this was beyond the scope of this project.

5.3 Duct systems

Mechanical ventilation systems use ducts to distribute the supply air and extract the exhaust air. The use of duct systems has several advantages, e.g. it controls air flow rates within the building, separates supply air from polluted air, and makes highly efficient heat recovery possible. On the other hand, there are some disadvantages, e.g. increased pressure loss, cleaning and maintenance of duct systems to avoid contamination of supply air, and fire protection so that fire and smoke cannot spread through the ducts. As discussed in Chapter 3.3, a pressure gradient of about 1.0 Pa/m is commonly recommended for the design of duct systems. This rule of thumb was mainly developed to avoid excessive noise generation, and aspects such as maintenance, fire safety and energy use (pressure loss) were not taken into account. Circular ducts are normally the most cost effective (Evans *et al*., 1996), and their use is recommended by ASHRAE (2007) when feasible. The advantages of circular ducts are that they are easier to manufacture, make tight, insulate, handle and install (Ekelund, 2001). Circular ducts are standard in Scandinavian ventilation systems, but rectangular ducts are still used in many countries. Their advantages are well-illustrated in Malmstrom *et al*. (2002a) where leakage tests were performed on French, Belgian and Swedish duct systems. The study showed that French and Belgian systems were up to 27 times less air-tight due to the use of rectangular ducts. This is something to consider in the design of duct systems, but the use of low-pressure mechanical ventilation will result in reduced leakage in general.

There are several sizing methods to design ventilation systems: constant velocity, equal friction, "most economic", static regain, constant diameter, 30% rule, and T-method (Malmstrom *et al*., 2002b). Each method has its pros and cons, but Bouwman (1982) showed that the total cost of the different methods did not vary much. The T-method is the only method that takes the economic aspect into account (Tsal *et al*., 1990, Malmstrom *et al*., 2002b), but it is best suited for CAV-systems (Johansson, 2005). Besant *et al*. (2000) present another approach, called the Initial Duct Sizing, Pressure Augmentation and Size Augmentation method, and claim that this handles duct system constraints better than the T-method. Bouwman (1982) found that the "economical pressure gradient" was 2 Pa/m based on the first year's costs, and a similar result was found in Kuehn *et al*. (1998). If an increase in energy prices is assumed, a lower value should be used, and Bouwman recommends 1 Pa/m. More recently, Johansson (2005) came up with the same result; this investigation used an electricity price of 0.8 SEK/kWh (0.71 DKK/kWh), an interest rate of 3%, and a lifetime of 50 years. The current electricity price in Denmark is about 1.6 DKK/kWh excl. VAT (Johansson does not state whether his electricity price includes VAT). Based on this rather recent study it is assumed that a pressure gradient lower than 1.0 Pa/m would be the most cost effective, due to the increase in electricity price but to make a detailed analysis was beyond the scope of this project. For the systems presented in Papers I and IV, a pressure gradient of 0.1 Pa/m was used, in line with other systems using low-pressure duct systems (Hestad *et al*., 1998, Hviid *et al*., 2013).

5.4 Diffuse ceiling ventilation

How fresh air is supplied to and distributed in a room has a huge influence on the air quality and thermal comfort. In conventional mechanical ventilation systems, mixing or displacement diffusers are used to distribute the fresh air. In mixing ventilation, the fresh air is diluted with the "polluted" room air by inducing high impulse air streams through one or more diffusers placed outside the occupant zone, usually in the ceiling. In displacement ventilation, the fresh air is supplied at floor level and utilizes the thermal plumes from people and heat loads to create stratification that replaces the polluted room air. Displacement ventilation is usually used in rooms with high occupancy and/or thermal load, e.g. conference rooms and classrooms, while the mixing principle is preferred in offices. There are numerous manufacturers of diffusers and various models that all rely on the same principles and theory well described in the textbooks (Ståbi, 2002, Danvak, 2008, Awbi, 2007) and the performance of the diffusers has been tested and reported in the literature (Lee *et al*., 2007, Nielsen *et al*., 2009). Because the diffusers rely on the same principles, they all require a pressure drop of 30 Pa or more to ensure a proper distribution and avoid discomfort due to draught or noise generation. This relatively high pressure loss is not compatible with low-pressure concepts, and alternative solutions are therefore needed. There has been development in the traditional diffuser design, especially with regard to diffusers with self-adjusting vanes or opening area (Acticon, 2013, Drivsholm, 2013). This prevents the pressure loss from increasing under most operating conditions, but the pressure loss is still about 30 Pa. Under-floor air diffuser (UFAD) concepts with low pressure losses were used in Berry (2000) and Tjelflaat *et al*. (1997); no discomfort issues were reported and the indoor environment met the design criteria. These solutions, however, are not applicable for renovation cases without excessive cost.

A promising alternative ventilation concept is diffuse ceiling ventilation or diffuse ceiling inlet, where the fresh air supplied through perforations in a suspended ceiling. The principle of diffuse ceiling ventilation is to inject the supply air into the plenum above a standard suspended ceiling, which functions as a distribution chamber. A small overpressure is created in the plenum and the air is forced down into the room through cracks and perforations in the ceiling surface. This gives a very large inlet area with low air velocities, which reduces the risk of draught and noise generation, facilitates efficient mixing of the supply air, and increases comfort. The air flow through the ceiling is called diffuse because it has random directions when it enters the room. The air jets through the ceiling are too small to mix with or displace the room air. The mixing with room air is generated by movement and buoyancy forces from people and heat loads. The concept is commonly used in livestock buildings, and here a study showed that the location of the heat loads controls the air distribution in the buildings (Jacobsen *et al*., 2004). The thermal plumes create the strongest air currents in the buildings and deflect the cold downward air current from the ceiling. In this way, large vortices are created that remove polluted air through a combination of displacement and mixing.

The concept is being increasingly used for comfort ventilation, but research in this area has been limited and mostly relies on laboratory experiments; results, however, have been promising. Diffuse ceiling ventilation came out on top in comparison with 5 conventional air distribution systems in terms of ability to supply high flow rates and air at lower temperatures without causing draught (Nielsen *et al*., 2009). Tracer gas measurements on two ceiling types in Hviid *et al*. (2013) in a test facility office room showed that diffuse ventilation inlet provided perfect mixing. Moreover, air temperature and velocity measurements disclosed no local discomfort in the occupied zone over a broad range of flow rates and inlet temperatures. A third ceiling type was examined at the same test facility, and the measurements showed comparable results (Fan *et al*., 2013). Similar finding are reported in Jacobs *et al*. (2008), who carried out measurements in a test facility resembling a small classroom. Experience and measurements from a pilot study where a diffuse ceiling inlet was installed in a school classroom were also reported, but the measurements included no quantifiable measurements of air temperature and velocity or ventilation efficiency. The phenomenon of thermal plumes obstructing supply air is well-illustrated in numerical analysis of diffuse ceiling ventilation by CFD (Computational Fluid Dynamics) (Hviid *et al*., 2013, Jakubowska, 2007 and Fan *et al*., 2013). The supply air is pushed to areas with no heat loads, where it drops down to and flows along the floor. This could lead to stagnant air in the occupant zone and/or short circuits depending on the position of the exhaust diffuser, as well as draught problems at ankle height.

Overall, the test results reported are promising, but the concept's performance in practice has not yet been documented. To investigate the performance of the concept under real conditions, diffuse ceiling ventilation was installed in two classrooms at Vallensbæk School and the results are reported in Paper III. The diffuse ventilation ceiling was made with cement-bonded wood wool panels consisting of active panels that are air permeable and passive panels that are non-permeable. The investigation encompassed several elements to document the thermal comfort and map the air distribution in the classrooms, including: air temperature and air velocity measurements, tracer gas, pressure drop across the ceiling panels, thermal camera pictures and smoke visualization of air movements in the room.

5.4.1 Draught rate

Air temperature and velocity were measured at different locations in the occupant zone and were used to determine local discomfort due to draught. Based on the measurements, the draught rating (DR) can be calculated (CR 1752, 1998).

$$DR = (34 - T_{a,l})(\bar{v}_{a,l} - 0.05)^{0.62}(0.37\bar{v}_{a,l}Tu_{a,l} + 3.14) \quad (7)$$

$$Tu = \frac{\sigma}{\bar{v}} \quad (8)$$

The draught rate in an empirically determined equation that expresses the relationship between the air temperature (T), mean air velocity $\bar{v}$ and turbulence intensity (Tu) and predicts the percentage of dissatisfied people at the specific conditions. The definition of the Tu is the ratio between the standard deviation σ of the mean air velocity and the mean air velocity measured, and the indices *a,l* denote the local air in question. Figure 4 shows the results of the DR calculations for a supply temperature of 17 °C.

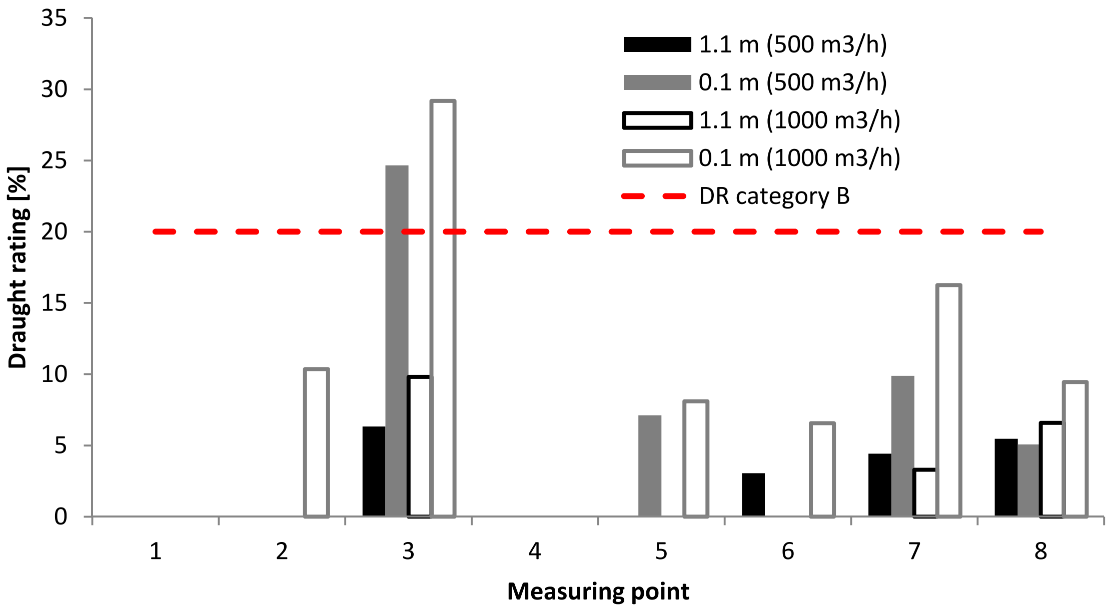

Figure 4: Draught rating at 0.1 and 1.1 m above the floor and flow rates of 500 and 1000 m^3/h with a supply temperature of 17 °C.

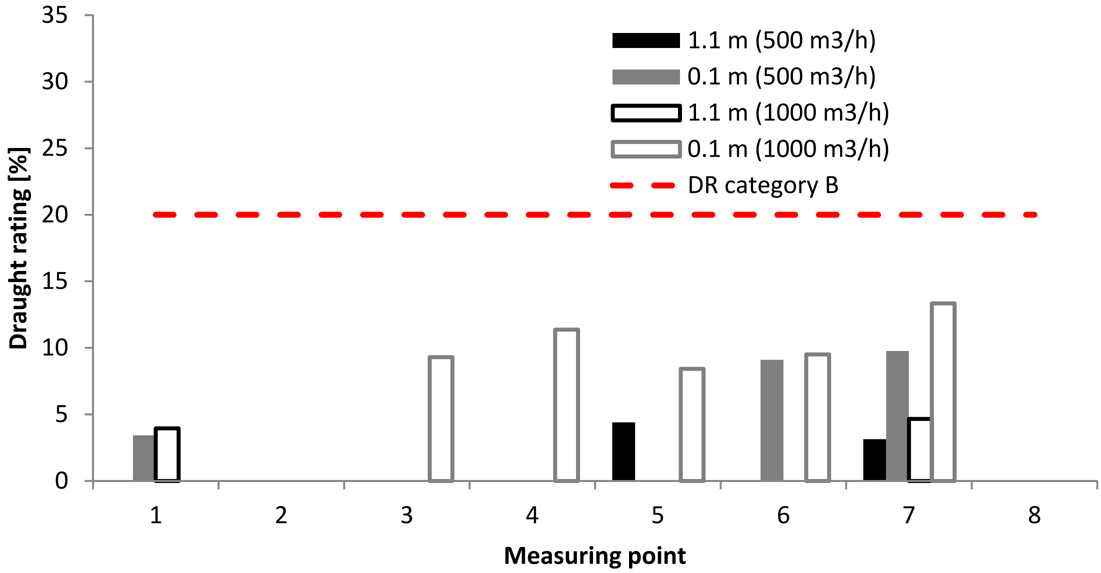

Figure 5: Draught rating at 0.1 and 1.1 m above the floor and flow rates of 500 and 1000 m^3/h with a supply temperature of 13 °C.

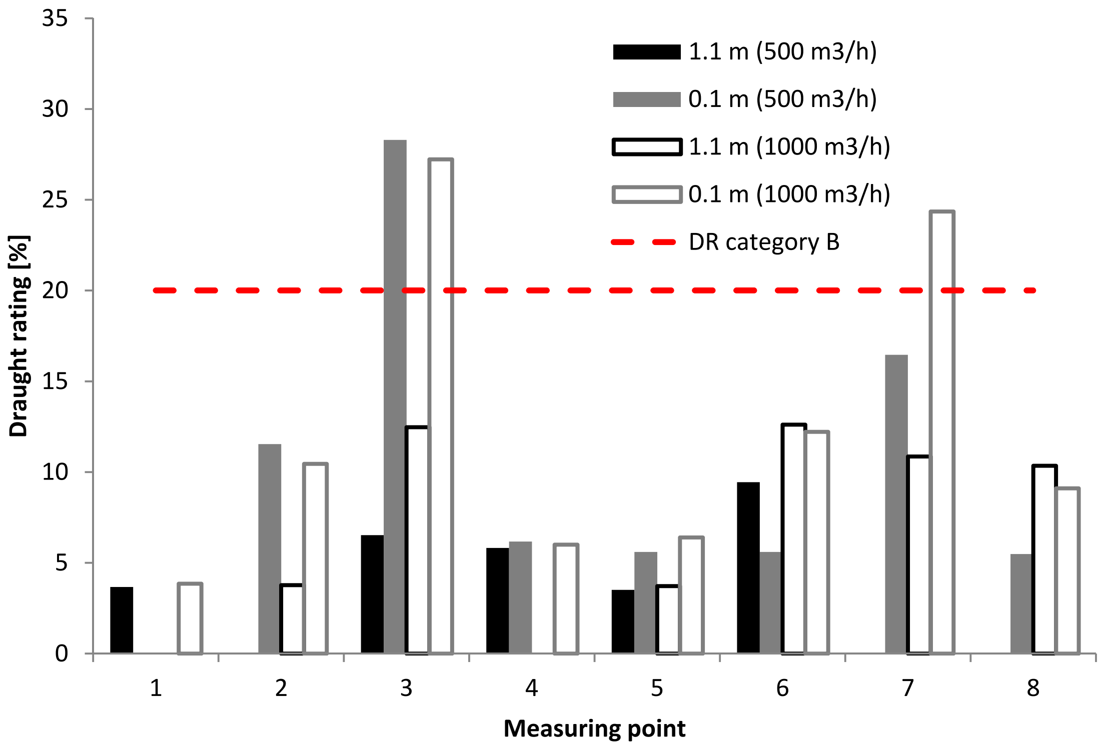

Figure 6: Draught rating at 0.1 and 1.1 m above the floor and flow rates of 500 and 1000 m^3/h with a supply temperature of 10 °C.

Figures 4, 5 and 6 show that there is a risk of discomfort due to draught at points 3 and 7, which is elaborated on in Paper III. The Figures also show that the DR is not an appropriate way to present the results, because the equation only is valid for air velocities above 0.05 m/s, so data for several points are not presented. The air temperatures measured were in the lower end of the comfort range, increasing the risk of draught at high air velocities. When the results are presented as DR, it is unclear whether the risk of draught is caused by low air temperatures or high air velocities. The air velocities in the occupant zone were the key parameter for determining draught issues caused by diffuse ceiling ventilation. So Paper III used the raw temperature and air velocity readings to analyse the risk of discomfort. The analysis showed a low risk of draught in the room except for two points, and they were most likely caused by outside influences and not the diffuse ceiling ventilation inlet, see Paper III.

5.4.2 Age of air and air change efficiency

In mixing ventilation, it is often assumed that all the supply air mixes perfectly with the polluted room air and dilutes any indoor contaminant. This is, however, rarely the case. If the supply does not mix perfectly, it can lead to the fresh air being exhausted before it has diluted its share of the indoor contaminants in the occupant zone (Awbi, 2007). This leads to either decreased air quality or higher energy use to supply more fresh air. The effectiveness of an air distribution system in supplying fresh air to a room is called its air change efficiency. Another concept is the age of air, defined as the length of time the fresh air supplied remains in the room before it is exhausted, also denoted the residence time $\bar{\tau}_r$.

The concept of the age of air was introduced by Sandberg (1982) and is determined by tracking the movement of particles in the air. In experiments, this is usually done by inducing tracer gas into the room. In practice, a large number of particles are induced and the age of air and residence time will vary from one particle to another. The local mean age of air $\bar{\tau}_p$can be determined by integrating the local tracer gas concentration *Cp(t)* at point p with time and dividing by the initial concentration *C(0)* at time zero (Awbi, 2007).

$$\bar{\tau}_p = \frac{\int_0^\infty C_p(t)dt}{C(0)} \qquad (9)$$

The mean age of air for the whole room $\langle\bar{\tau}\rangle$ can be quantified by measuring the tracer gas concentration at the exhaust *Ce(t)* and integrating with respect to time.

$$\langle\bar{\tau}\rangle = \frac{\int_0^\infty tC_e(t)dt}{\int_0^\infty C_e(t)dt} \qquad (10)$$

In the exhaust, the local mean age of air is equal to the inverse of the air exchange, also denoted nominal time constant, τ_n. The air change efficiency ϵ_a is the average time it takes to replace the room air compared to the shortest air change time possible. The definition is the ratio of the shortest possible air change time in the room (nominal time τ_n) and the average time it actually takes to replace the air at a point (actual air change time $\bar{\tau}_r$).

$$\epsilon^a = \frac{\tau_n}{\bar{\tau}_r} \text{ x } 100\% \qquad (11)$$

The actual air change time $\bar{\tau}_r$ can be derived from the mean age of air in the room.

$$\langle\bar{\tau}_r\rangle = 2\langle\bar{\tau}\rangle \qquad (12)$$

Figure 7 shows the mean age of air at five sampling points in the room. Paper III gives the local air change index results, along with a more detailed description of the experiments performed.

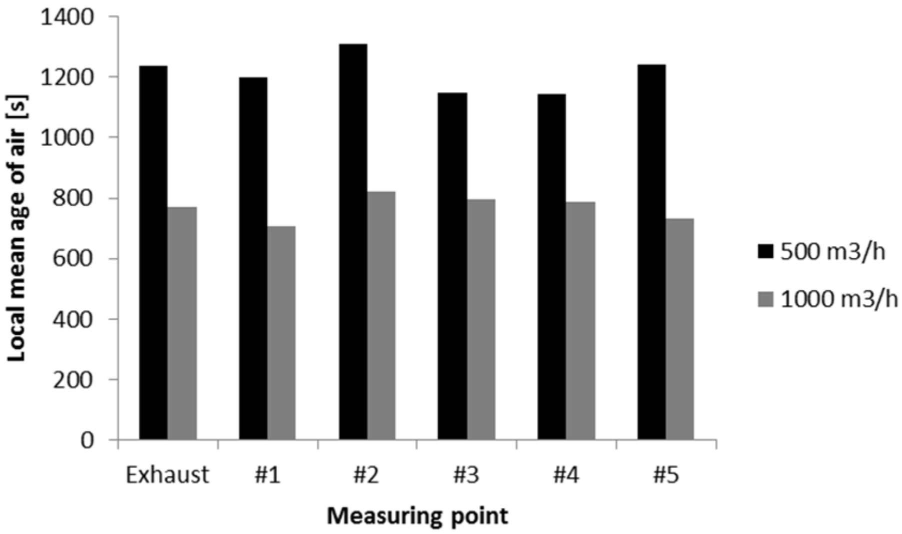

Figure 7: Local mean age of air at the exhaust and 5 sampling points in the occupant zone.

The local mean age of air at all five sampling points in the occupant zone was within ±10% of the exhaust local mean age of air at both flow rates. This shows that the supply air is distributed equally to all sampling points in the occupants indicating no short circuit or stagnant zones. The mean age of air, however, gives no intuitive indication of the effectiveness of the air distribution concept. For that, the air change efficiency is more appropriate and was therefore used in Paper III to present the results and elaborate on the effectiveness of diffuse ceiling ventilation. Table 8 lists the expected air change efficiency for different flow patterns. Mixing ventilation usually has efficiencies slightly below 50% because the supply air is rarely fully mixed and displacement ventilation has efficiencies between 60-70% (Rehva, 2001).

Table 8: Air change efficiencies for different ventilation principles.

Flow pattern	Air change efficiency [%]
Ideal piston flow	100
Displacement	50-100
Perfect mixing	50
Short circuit flow	<50

The air change efficiency results presented in Paper III showed perfect mixing in the classroom. The measurements, however, did show a slight transition towards displacement ventilation at the high flow rate with a room air change efficiency of 53%.

5.4.3 Pressure loss

One key advantage of diffuse ceiling ventilation is the low pressure loss compared to conventional diffusers. Pressure losses of 0.5-2.5 Pa depending on the flow rate were found for two typical types of suspended acoustic ceilings (aluminium and gypsum) (Hviid *et al.*, 2013). This correlates well with the pressure loss measured across the cement-bonded wood wool panels used at Vallensbæk School, see Paper III. The panels themselves were not developed specifically for diffuse ventilation, but the mineral wool on the back permitted this use; otherwise the panels probably would have been too permeable to ensure uniform distribution. Only the products presented in Jacobs *et al*. (2008) were found to be specifically designed for diffuse ceiling ventilation. At Hvidovre Community Centre, diffuse ceiling ventilation was used as part of a general renovation of the ventilation system for the offices, see Chapter 6.3. Gypsum tiles were used as in Hviid *et al*. (2013), and Figure 8 shows the measured pressure loss characteristic across the ceiling.

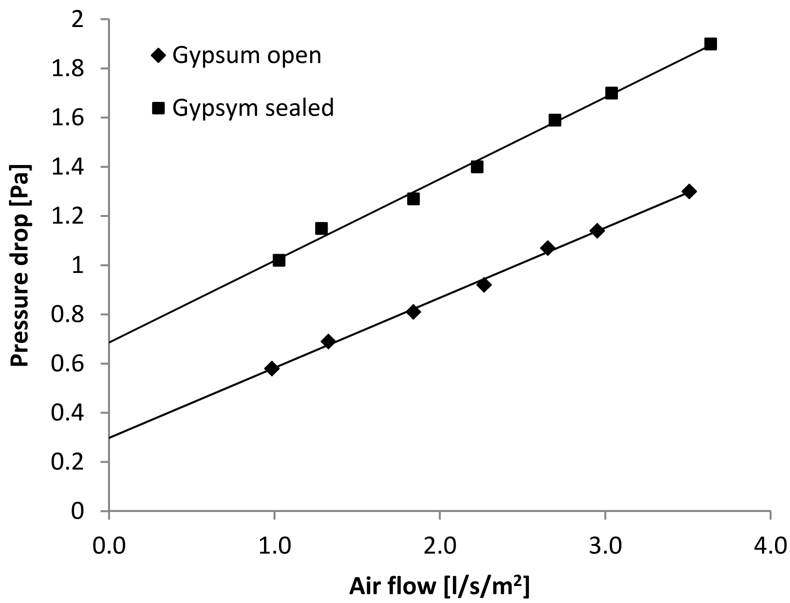

Figure 8: Pressure loss characteristic across suspended acoustic ceiling at Hvidovre Community Centre.

The lighting fixtures were integrated in the ceiling and they were not airtight, so the supply air could penetrate through. The pressure loss characteristic was therefore also measured with the lighting fixtures sealed to determine the percentage of supply air that penetrated through the lighting fixtures. The two characteristics are almost parallel and show that at a pressure loss of 1 Pa two thirds of the air penetrates through the lighting fixtures. As with the diffuse ceiling ventilation at Vallensbæk School, this shows that it is important to keep track of leaks in the ceiling to control the air flow. There are various types of suspended acoustic ceilings on the market, but hardly any are designed for diffuse ceiling ventilation. Nevertheless, the concept seems to work for a broad range of types and configurations, just as long as the ceiling surface is permeable or has small cracks and/or openings. If products were developed specifically for air distribution, the performance of the concept could be improved in terms of better design, the predictability of the air flow pattern in the room, and the capacity to control the air flow.

5.4.4 Learning ability and perceived air quality

The new ventilation system in the two classrooms was able to improve the indoor air quality and maintain a CO_2-concentration below 1000 ppm, see Paper IV. To examine how the new supply concept and improved ventilation affected the pupils' learning ability and well-being, and how they perceived the indoor environment, learning performance tests and questionnaires on the perceived indoor environment and Sick Building Syndrome (SBS) were given to the pupils.

In the performance tests, the pupils occupying the two rooms were exposed to different air supply rates. The experiment had what is called a crossover design in which the two rooms were exposed to different air supply rates in one week, and the conditions were then switched the following week. One condition was with an air supply rate of 500 m^3/h corresponding to Category 3 in EN15251 (2007) of 5 l/s per person and 0.35 l/s per m^2. The other condition was with an air supply rate of 0 m^3/h corresponding to conditions before the new mechanical ventilation system was installed. Estimation of the infiltration air flow showed an air change rate of approximately 0.3 h^{-1} (80 m^3/h), based on the CO_2-concentration decay after the pupils left. The experiments were carried out in the first two weeks of September and the outdoor temperature during the day was about 20 °C. Under both conditions, the teachers and pupils were allowed to open the windows and doors as usual. The tests were given in the classes as part of their normal schedule to avoid changes in their routines and teaching environment, and they were given on Thursday and Friday so the pupils had time to get acclimatized to the specific condition that week. The pupils were not aware that the tests were part of an experiment, and neither the teachers nor pupils were aware of the changes in air supply flow rate. The CO_2-concentration, air temperature and air humidity in the rooms was continuously monitored over the two weeks. The CO_2-concentration was measured with a VAISALA GM20 CO_2-transmitter connected an Onset HOBO U12-012 data logger that measured the air temperature and relative humidity.

5.4.5 Learning performance tests

The tests were a mathematics test, in which the pupils had to subtract two four digit numbers, and a reading and comprehension text, in which the pupils had to choose the correct word out of three. All three words fitted in the context of the sentence but only one was correct in the context of the text as a whole. Sections of the two tests are shown in Figure 9.

3746
\- 2345

234,92
-123,50

Det var i onsdags, og Beate kom hjem fra en tur i skoven. Men mor så ikke på hende, mens hun talte. Hun sad ved maskinen og syede som gjaldt det livet - hvad det for resten også gjorde. Mor [syede, strikkede, hæklede] kjoler til damer og børn. Og flittig var hun, og dygtig, det sagde alle mennesker. Men hvor flittig hun end var, hvor sparsommeligt de end levede, ville hun aldrig kunne skaffe de nødvendige penge. Og det var fordi far var død og ikke havde efterladt dem andet end en [kæmpeformue, kæmpegæld, kæmpekapital].

Figure 9: (left) Section of mathematics test, (right) reading and comprehension test.

For a thorough description of the test and method, see Wargocki *et al*. (2007) where the tests used in this paper were also used. The pupils had 10 minutes for each test and if someone finished before the allocated time, the teacher stopped the test and noted the time spent.

5.4.6 Perceived air quality

Every Friday after the pupils had taken the last test, they filled out a visual analogue scale questionnaire to indicate the perceived air quality and the intensity of various SBS symptoms. The questionnaire included 6 parameters on the indoor environment in the classroom (temperature, air movement, air dryness, air freshness, illumination and noise), and 10 questions on SBS symptoms and their ability and motivation to perform school work (nose congestion, throat, lip, skin dryness, hunger, sleep at night, fatigue, enough sleep, motivation and headache), see Figure 10.

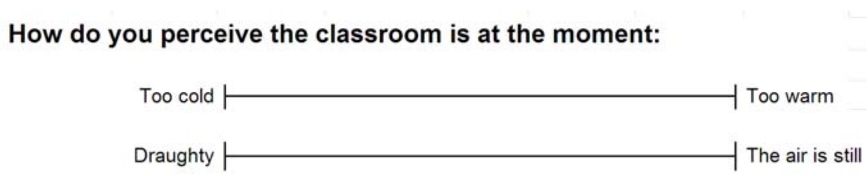

Figure 10: Examples of visual analogue scale from perceived air quality and SBS questionnaire.

The questionnaire was explained to the pupils by the investigators, but handed out by the teachers, and as with the performance tests, only data from pupils who had filled out the questionnaire both weeks were used in the analysis.

5.4.7 Statistical analysis

In the mathematics test and the reading and comprehension test, the dependent variables were the number of answers by each pupil during the time of the test and the number of errors made. To enable comparisons between the interventions, all results were normalized to answers per minute (speed) and percentage of incorrect answers (errors). Only data from pupils that had taken both tests were used in analysis and the results were adjusted for increased learning and familiarity with the tests. This was done by multiplying each pupil's result in week 2 by the ratio between the class average in the first week and in the second week.

Before the statistical analysis, the results of each test from the two classes were pooled together depending on the condition (ventilation or no ventilation). Using Shapiro-Wilk's W test, it was determined whether the results were normally ($p>0.05$) or not normally ($p<0.05$) distributed (Bluman, 2007). If the results were not normally distributed they were log-transposed and again tested for normality. All the statistical data analysis was performed in the software Statistica (Statistica, 2007). Table 9 shows the indication of whether the performance test data were normally or not normally distributed.

Table 9: Indication of whether the data from the performance tests were normally distributed or not.

	Mathematics		**Reading and comprehension**	
	Speed	Errors	Speed	Errors
Normal distribution				
Not normal distribution	X	X	X	X

None of the performance test results were normally distributed and therefore the non-parametric Wilcoxon matched pairs signed-rank test was used, showing statistical significance when $p<0.05$. The amount of data in the study was small and it is therefore not possible to make definitive claims based on the results. The results are presented in Paper III and the results of the mathematics test were in line with previous studies e.g. Wargocki *et al*. (2007), while the reading and comprehension test did not give the expected results. This was probably because the pupils found one of the tests more difficult, skewing the data and making it impossible to make comparisons between the weeks and conditions. Table 10 shows whether perceived indoor environment and SBS symptoms questionnaire data were normally distributed.

Table 10: Indication of whether the data from the perceived indoor environment and SBS symptoms questionnaire were normally distributed or not.

	Normal distribution	**Not normal distribution**
Temperature	X	
Draught	X	
Air freshness		X
Air dryness		X
Noise	X	
Illumination	X	
Nose congestion	X (log)	
Throat		X
Lips		X
Skin dryness	X	
Hunger	X	
Sleep at night	X (log)	
Enough sleep	X	
Fatigue	X	
Head ache	X	
Motivation	X	

For the data that were normally distributed, the one-way ANOVA test was used to determine significance $p<0.05$. Figure 11 shows the results of the perceived indoor environment and SBS symptoms parameters not presented in Paper III. The analysis found no significant improvements in the SBS symptoms or perceived indoor environment, with the exception of perceived air freshness. See Paper III for further analysis and discussion of the results.

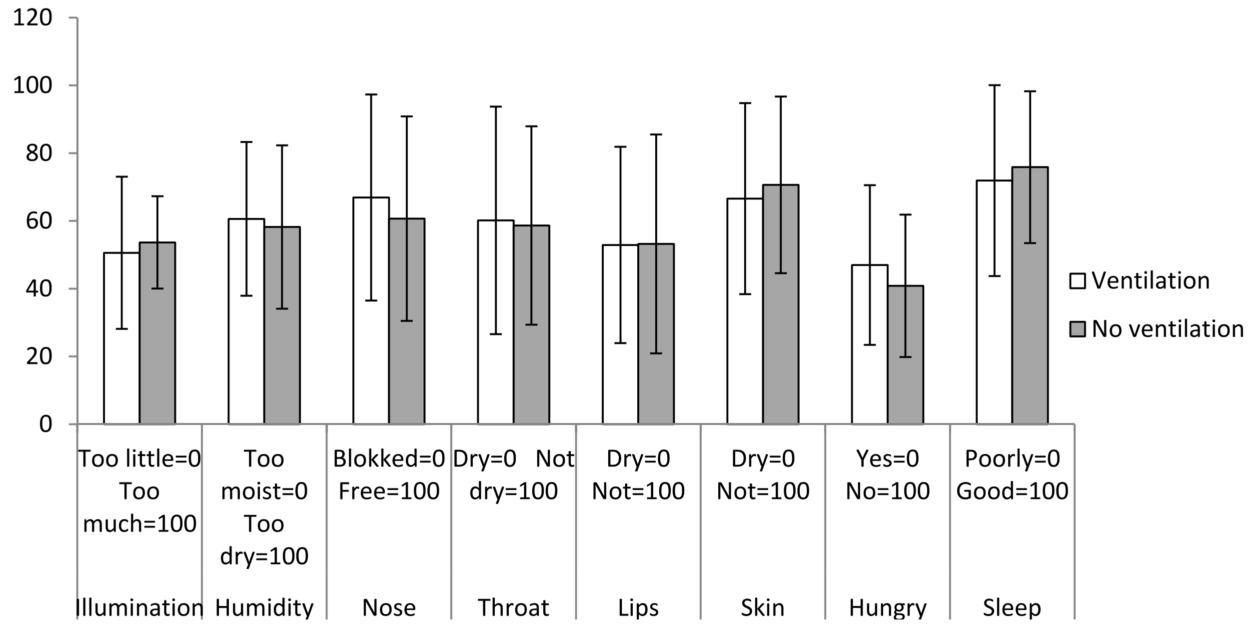

Figure 11: Results of perceived indoor environment questionnaire and SBS symptoms.

5.5 Control system

How a mechanical ventilation system is controlled has a great impact on its performance, both in terms of correct air distribution and energy consumption. It is well-established that VAV (Variable Air Volume) or DCV control (Demand Control Ventilation) can reduce energy consumption by up to 50% compared to CAV (Constant Air Volume) systems (Aktacir *et al*., 2006, Englander *et al*., 1992, Lorenzetti *et al*., 1992, Seppänen, 2007), but the actual magnitude of the reduction depends on the use of the building and the detailed operation of the control system (Yang *et al*., 2011). The aim of both control concepts is to reduce the air flow rate when demand decreases, simultaneously reducing pressure loss and thereby the fan power required to operate the system.

The control systems usually regulate the air flow using a fixed static pressure in the duct system maintained by the fan, which the actuator (damper, control box, terminal diffuser) can work against to adjust the flow rate to the demand. The actuators adjust the flow resistance in the duct system thereby controlling the air flow to the room under all operating conditions (Wang *et al*., 1998). When the demand decreases, the actuator closes and the static pressure in the duct system increases, the fan speed is then reduced to maintain the pre-set fixed pressure. In this way, the power consumption of the fans is reduced, but the full savings potential is not achieved. This static pressure set-point is usually determined according to the maximum ventilation demand for the system and in many cases even higher to ensure capacity at peak loads and fear of failure to meet demand (Liu *et al*. 1997). This results in unnecessary excess static pressure (i.e. power use) when the system is operating at partial load because the actuators are not fully open (Wei *et al*., 2004, Federspiel *et al*., 2005, Lui *et al*., 1997). The full savings potential can be achieved by using SPR (Static Pressure Reset) control, in which the static pressure set-point in the duct system is continuously adjusted (reset) in accordance with the actual demand, ensuring that one damper is always fully open. The principle and savings potential is well illustrated in Figure 12 from Wei *et al*. (2004).

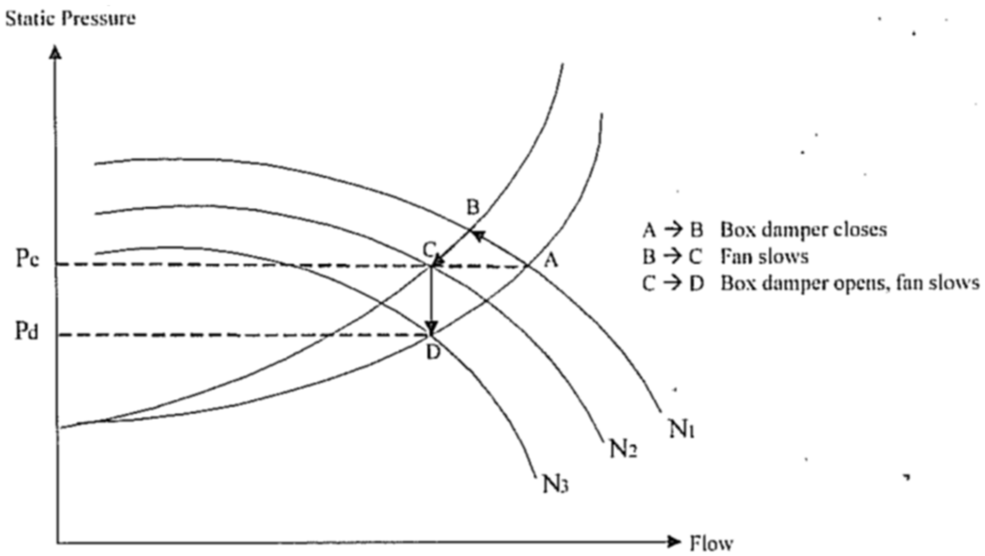

Figure 12: Principle of the relationship between flow and pressure in static pressure reset control systems.

Conventional control systems operate between point A, B and C. From point A to B, the static pressure increases as demand decreases and actuators close. The control system senses the increase and slows down the fan speed from N_1 to N_2 in order to maintain the fixed static pressure set-point P_c (B to C). The system is now operating at lower speed and power is saved in accordance with the previously mentioned 1st fan law (equation 4). However, the system is operating at unchanged high resistance because the actuators are partially closed. With SPR-control the actuators reopen and the fan speed is reduced to N_3 to maintain the same air flow. This reduces the pressure from P_c to P_d. This ensures minimal static pressure in the duct system at partial load and the power consumption is reduced in accordance with the 2nd fan law, while still providing sufficient air flow to all zones.

The concept was first introduced by Hartmann (1989), and is well-known and used in the US in ventilation systems with DDC (Direct Digital Control). There are several studies on SPR-control in the literature describing different approaches to modelling the control (Wei *et al*., 2004, Taylor, 2007, Wray *et al*., 2008, Federspiel *et al*., 2005, Liu *et al*., 1997, Warren *et al*., 1993). All of them achieve considerably lower static pressure in the system, resulting in reductions in power usage of up to 50%. No studies on SPR-control were found in Europe, but there are commercial products on the market (Belimo, 2013, Lindab, 2013). Common for the commercial products and the studies from the literature is the fact that they use actuators that induce relatively high pressure losses ((>>30 Pa) to maintain accuracy and control flow resistance. Though the SPR-control reduces fan power consumption, the high pressure loss in the actuators was not considered compatible with the low-pressure concept. A new SPR-control system employing a new droplet damper was therefore developed. The results are presented in Paper II, along with a detailed description of the control algorithm, method and experiments. The droplet damper has an aerodynamic design which reduces pressure loss, and measurements showed that 234 l/s flowed through the damper at 30 Pa pressure loss, while only 120 l/s could flow through a conventional flat plate damper. In the test set-up, the developed control algorithm could regulate the air flow to the rooms with a pressure loss down to 5 Pa. This is significantly lower than for previously reported system and something that is applicable for low-pressure systems.

6. Cases

Three case studies were carried out during the PhD project. The discussion at the level of components is dealt with in the respective chapters, while this chapter presents a summary of the results at the system level and further discussion of the results.

6.1 Intend Building

To evaluate the performance potential of the suggested low-pressure concept, components and design solutions, a case study was carried out for a test case office building, see Figure 13. The building was designed with the help of industrial partners including DTU.Byg as part of the project INTEND, which was initiated by the contractor MT Højgaard, to develop a standard office building that could meet the energy framework requirements for 2020 (Jørgensen *et al*., 2008, Mouridsen, 2008). The proposal for 2020 used a mechanical ventilation system with an SFP-value of 1.26 J/m^3. In the case study, a new ventilation system based on the concept was developed to further reduce the energy consumption for ventilation. The goal was to develop and design a system that could fulfil the ventilation and cooling demand all year so that only one system needs to be installed. The goal was to develop a system with an SFP-value of 250 J/m^3 resulting in an annual energy consumption of 4-5 kWh/m^2, corresponding to 20-25% of the energy framework for 2020 in Denmark. The main results, discussion and conclusions are reported in Paper I.

Figure 13: Suggested building design for INTEND building, Façade design (top), Floor plan (bottom).

The building was divided into two sections, each supplied by a separate AHU to reduce duct lengths and large duct dimensions. The duct system for each section supplies the individual floors through an installation shaft at each end of the building. A description of the design of the AHUs and components is given in Paper I, and their performance was evaluated through simulations of energy consumption and indoor environment. The simulation software iDbuild and IES Virtual Environment was used and, like other simulation software, only a single SFP-value can be assigned for the ventilation system (iDbuild, 2010, Nielsen, 2005, IES, 2011). The programs then calculate the power consumption based on the ventilation demand, but the SFP-value is not fixed and depends on the flow rate and pressure loss in the system. Average SFP-values can be assigned but they are difficult to determine without knowing the annual ventilation demand and this results in inaccurate power consumption estimates for ventilation systems. To achieve accurate results for the case system, the pressure loss characteristic of the system was determined. The pressure loss in the system can then be calculated based on the simulated hourly ventilation demand and then the hourly fan power needed to operate the system can be determined. By using Equation (2) in Paper I, the annual power consumption of the system and the average SFP-value of the system can be determined more accurately.

The case study showed that there is a huge energy-saving potential in reducing pressure losses in mechanical ventilation systems. Reducing the pressure loss was done by using low-pressure solutions, e.g. diffuse ceiling and ESPs and oversizing standard components like the duct system and heat exchanger compared to conventional dimensioning. Integration in the building required more space, but by placing the AHUs on the roof and putting vertical ducts in the installation shaft at each end of the building, no office space was used. Only the ceiling height in the corridors needed to be lowered to make room for horizontal ducts.

The average specific fan power is 330 J/m^3, resulting in an annual energy consumption of 3.7 kWh/m^2. This corresponds to 10-15% of the power consumption for conventional mechanical ventilation systems, which means the system can meet future energy requirements for buildings. The system was able to meet the indoor environment requirements by using night cooling to condition the building. The energy consumption for night cooling comprised 24% of the total energy consumption, representing an annual cost of DKK 1300 at an electricity price of DKK 2/kWh. This is a relatively small cost compared to the alternatives of installing a natural ventilation system for night cooling or an active cooling system for the supply air during the day. The ESPs used to filter the air comprised 57% of the power consumption, but whether this was the optimal solution is unclear, as discussed in Chapter 5.2. The conclusion from Paper I is that there is a great energy savings potential in using low-pressure mechanical systems and that they can help meet future energy requirements.

6.2 Vallensbæk School

To determine the performance of the low-pressure concept in practice, a pilot system was installed in two classrooms at Vallensbæk School. Paper IV gives a detailed description of the system along with the measurements and results for indoor environment and power use. The measurements and results for the diffuse ceiling ventilation concept used are presented in Paper

III and Chapter 5.4. One key priority in the study was to examine how the individual components performed at low pressure, to determine whether they were applicable for the concept. The indoor environment was examined in terms of CO_2-concentrations and air temperature, and subsequently the effect of improved air quality on the performance of pupils investigated. Finally, the LCC was calculated to determine whether it is feasible to expand the use of the concept to other schools.

The indoor environment in the classrooms was improved after the installation of the ventilation system, reducing CO_2-concentrations from approximately 1800 ppm to below 800 ppm during occupied hours. This showed that the system could meet the indoor environment requirements of maximum 1000ppm stated in the Danish Building Code and by the Danish Working Environment Authority (DBC, 2010, DWEA, 2012). The flow rate to the rooms was determined by the demand based on the CO_2-concentration and controlled accurately by droplet dampers. This was at air velocities of 1 m/s in the duct system and it showed that it is possible to control the air flow in duct systems at low air velocities and pressure losses. The pressure loss in the diffuse ceiling inlet and the ducts on the supply side was about 5 Pa and about 20 Pa across the damper. The limiting factor in reducing the static pressure further was not the dampers but the AHU fan control that had a minimum static pressure set-point of 25 Pa. If the static pressure set-point could be lowered, the pressure loss across the dampers could probably have been lower based on the performance of the damper measured in Paper II.

The attic was used for the AHU and the duct system, and this made it relatively easy to reduce the pressure loss in the duct system. The AHU was oversized compared to standard dimensioning and could just fit in the attic. The oversized AHU had its pros and cons. On the positive side the heat exchanger and filter pressure losses were reduced, bringing them into line with recommended pressure losses in the guidelines presented in Table 2. The negative aspect was that it resulted in the fan being oversized, leading to decreased efficiency at low air flow rates. Overall the system fulfilled the objectives set up, achieving an annual SFP-value of 611 J/m^3, and providing valuable information and experience of low-pressure mechanical ventilation systems.

The LCC calculation showed that the cost of the system was higher than the reference system. At component level, only the “oversized” duct system was cost effective; despite the relatively minor extra cost of DKK 10,000 for the dampers and the “oversized” AHU, the energy saving could make up for the increased cost. Whether the concept can be used in other schools depends on whether the ceiling needs changing; if not, the installation of a new ceiling will result in additional cost. However, if the ceiling is to changed anyway, diffuse ceiling ventilation can reduce the cost for ducts and diffusers, and the conventional preheating coil can be omitted without causing draught problems.

6.3 Hvidovre Community Centre

As a part of a major renovation of Hvidovre Community Centre, a new ventilation system was installed. The system only supplied the offices oriented west and located on the perimeter around the library hall, see Figure 14. The AHU unit was put in a small courtyard on the 1st floor and was optimized to reduce pressure losses within the space provided. A pressure gradient of 0.5 Pa at maximum load was used for the dimensioning of the duct system. Diffuse ceiling ventilation through conventional gypsum tiles in a suspension system was used to distribute the air in the offices. Measurements of the pressure loss across the ceiling are presented in Chapter 5.4.3, but draught and air distribution efficiency measurements have not yet been performed. The rooms are mainly single offices, so a simple 2-step control was chosen. A low flow rate of 20 m^3/h and high flow rate of 50 m^3/s when people are present, controlled by a motion sensor that also controls the general lighting. The meeting rooms, however, were equipped with CO_2-sensors to control the air flow.

Figure 14: Location of offices at Hvidovre Community Centre, marked in yellow. Floor plan – basement (left), floor plan - ground floor (Middle) and floor plan 1st floor (right).

The project was delayed and the system is expected to be fully installed in March 2013, so only the calculation and simulation results are presented. Table 11 lists the pressure losses of the components in the system, which are in the range of the recommended in Table 2, and the SFP-value at max load was 1.54 kJ/m^3. A simulation of the ventilation demand in the building gave an annual average SFP-value of 1.24 kJ/m^3 (Bsim, 2006). In the calculation of the average SFP-value, the pressure loss characteristic method described in Paper I was used. However, this method does not take into account where the air is needed in the building. In the calculation of the pressure loss characteristics, the reductions in flow rate are assumed to be evenly distributed according the max load in each room. This is rarely the case in practice because some rooms are used and some are not. So the method may not provide accurate power consumption estimates if one floor is not used and all the air has to be distributed on the other two.

Table 11: Pressure losses in ventilation components at Hvidovre Community Centre.

	Supply (8100 m^3/h)	**Exhaust** (8100 m^3/h)
Flow rate, Max [m^3/h]	8100	8100
Duct system incl. sound actuators [Pa]	164	199
Control/fire dampers [Pa]	96	124
Air intake/exhaust [Pa]	16	11
Air Terminal device [Pa]	2	22
External pressure loss [Pa]	278	356
Bag filter, F7 average [Pa]	104	104
Heat exchanger, rotary 85% [Pa]	108	108
Heating coil [Pa]	14	-
Sum [Pa]	504	568
Motor efficiency IE3 [%]	90	90
Fan efficiency [%]	85	85
Specific fan power [kJ/m^3]	0.73	0.81
Total SFP-value [kJ/m^3]	1.54	

Part of the optimization process was choosing the appropriate fan and motor and dimensioning them according to the pressure loss and flow rate. By changing from a centrifugal fan to an axial fan the efficiency was improved from 78 to 85%, and the motors were improved from IE2 to IE3 which increased the efficiency by 3%. As stated IE4 motors are already on the market, but the manufacturer could not deliver them before the air handling unit was to be installed. The improvement of the fans and motors increased the initial cost by DKK 14,950 and Table 12 shows a simple payback time calculation. The payback time is 9 years, and with the lifetime of the fan and motor being 15-20 years, the investment was assumed to be cost-effective.

Table 12: Calculation of simple payback time of improving motors from class IE2 to IE3.

	Supply (8100 m^3/h)	**Exhaust** (8100 m^3/h)
Pressure loss [Pa]	504	568
Operating hours [h]	2600	2600
EI2 efficiency [%]	0.867	0.867
EI3 efficiency [%]	0.9	0.9
Centrifugal fan efficiency [%]	0.782	0.787
Axial fan efficiency [%]	0.85	0.854
Power savings [kWh/year]	495	547
Electricity price [DKK/kWh]	1.62	1.62
Increased cost [DKK]	14,950	
Payback time [Years]	8.9	

It was therefore not possible to achieve the results described in Papers I and III due to the limited space for the AHU and its position that resulted in long distances to transport the air, and the conditions for integrating the system were far from ideal. The calculations on the system showed that the system should be able to fulfil the requirements and that it was in the range of the optimal pressure losses recommended in the guidelines, See Table 2.

7. Conclusion

The objective of this thesis was to develop a concept, solutions and components for low-pressure mechanical ventilation. The study of the literature on mechanical ventilation showed a large discrepancy between design guidelines, best practice, and the current design of mechanical systems. A combination of a lack of priority, focus, and knowledge of solutions for how to meet the guideline goals is probably the reason why they have not been achieved. The results in this thesis show that there is a great energy savings potential in the use of low-pressure mechanical ventilation, well beyond current guideline recommendations. The concept, solution and components presented are believed to help make it realistic to install the low-pressure mechanical ventilation system in buildings. Overall the following conclusions can be drawn:

- The literature study, investigation of components and the case study in Paper I showed a huge potential for reducing pressure losses and fan power consumption in conventional mechanical ventilation.
- The proposed concept was able to achieve an SFP-value of 330 J/m^3 theoretically, but it was not possible to reach that level in practice due to the use of conventional AHUs in the demonstration projects.
- Heat recovery is an established part of current ventilation systems, but the pressure loss can be reduced by using larger heat exchangers or customized units. This could also increase heat recovery efficiency to >90%.
- The concept was able to fulfil the indoor environment goals set up, both in theory and in practice.
- The demonstration system with a diffuse ventilation inlet proved that the concept works in practice, providing perfect mixing without any potential draught problems.
- The SPR-control system developed was able to operate at low pressure loss, thereby reducing the pressure loss and fan power needed.
- The indoor air quality can be improved by use of ESPs, but whether it is an energy-saving solution needs further investigation.
- The life cycle cost analysis showed that individual components and solutions were cost-effective, while at the system level it resulted in increased cost. Whether the reduced costs can make up for the increased cost is uncertain and needs further investigation.

To sum up, the overall hypothesis and individual goals of the thesis were achieved. Specific solutions were developed and tested for some of the focus areas while others were only described theoretically and some need further investigation to make definitive conclusions.

7.1 Concluding remarks and suggestions to further work

The work on low-pressure mechanical ventilation is far from over. There are several areas that are not fully disclosed and need further work. The following gives a list of the current status of the proposed components and suggestions for future work:

- LeanVent has carried on the torch on the static pressure reset control system and developed their own control algorithm that has been tested on fume cupboards in a laboratory facility. To further develop and improve the algorithm, LeanVent, in collaboration with DTU.BYG and the Danish Technological Institute, has received funding for a three-year project from the Danish Energy Agency.

- Diffuse ceiling ventilation is being installed in more and more buildings. Only the cement-bonded wood wool panels used at Vallensbæk School are to some degree designed for the purpose. Several suspended ceiling manufacturers have shown interest in developing and marketing products designed for diffuse ceiling ventilation.

- Develop and design AHUs for low-pressure systems, including optimal fan and motor type and size and heat exchangers with lower pressure losses without increasing size.

- Analysis of filtration methods to determine the optimal solution in terms of cost, energy use, pressure loss, maintenance, filtration efficiency and space requirements.

- Develop intake and exhaust components with reduced pressure loss and that perhaps can utilize wind effects to help drive the system.

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Part III:

Appended papers

Paper I

Performance Potential of Mechanical Ventilation Systems with Minimized Pressure Loss

S. Terkildsen, S. Svendsen

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Performance Potential of Mechanical Ventilation Systems with Minimized Pressure Loss

Søren Terkildsen and Svend Svendsen

Department of Civil Engineering, Section of Building Physics and Services
Technical University of Denmark, Brovej, Building 118, DK-2800, Kgs. Lyngby

Abstract

In many locations mechanical ventilation has been the most widely used principle of ventilation over the last 50 years but the conventional system design must be revised to comply with future energy requirements. This paper examines the options and describes a concept for the design of mechanical ventilation systems with minimal pressure loss and minimal energy use. This can provide comfort ventilation and avoid overheating through increased ventilation and night cooling. Based on this concept, a test system was designed for a fictive office building and its performance was documented using building simulations that quantify fan power consumption, heating demand and indoor environmental conditions. The system was designed with minimal pressure loss in the duct system and heat exchanger. Also, it uses state-of-the-art components such as electrostatic precipitators, diffuse ceiling inlets and demand-control ventilation with static pressure set-point reset. All the equipment has been designed to minimize pressure losses and thereby the fan power needed to operate the system. The total pressure loss is 30-75 Pa depending on the operating conditions. The annual average specific fan power is 330 J/m^3 of airflow rate. This corresponds to 10-15% of the power consumption for conventional mechanical ventilation systems thus enabling the system to fulfil future energy requirements in buildings.

Key words: pressure loss, mechanical ventilation, SFP-value, low-energy buildings, night cooling.

1. Introduction

In many climates, mechanical ventilation systems have become the most common method of ventilating buildings over the last 50 years (Dokka et al., 2003). However, the increasing need for energy-efficient solutions necessitates a revision of the traditional design of mechanical systems if they are to meet future energy requirements.

In 2004, the building sector represented 37-40% of the total energy consumption in the European Union and the USA. HVAC systems accounted for 48% of building sector consumption in the EU and 57% in the USA (Perez-Lombard et al, 2008). Of this, fans accounted for 15-50%, depending on the type, design and performance of the system (Wouters et al, 2001; Perez-Lombard et al, 2011). This indicates that there is a large savings potential if more energy-efficient ventilation systems can be developed. The energy performance of a ventilation system can be characterized by its Specific Fan Power (SFP) value which is defined as the energy required to drive the system, divided by the flow rate, see Section 3.7.

The range of SFP-values found in existing buildings, and even some new buildings, is very large, between 5500 J/m^3 and 13000 J/m^3 of airflow rate (Wouters et al, 2001). However, most new systems have typical SFP values of approximately 2500-3000 J/m^3 (Schild et al, 2009) and the requirement in the Danish Building code is 2100 J/m^3 (BR10). There are several ways to reduce the energy consumption for ventilation systems and numerous projects have introduced various approaches. Typically these have considered natural and hybrid ventilation, rather than mechanical ventilation (Heiselberg, 2002; Delsante et al, 2002). However, it is also possible to obtain significant reductions for mechanical ventilation. For example Berry (2000) considers a special design of the air handling unit (AHU) combined with good integration in the building. This resulted in an annual average SFP-value of 400 J/m^3. Hestad (1998) describes a design for minimal pressure loss which resulted in a pressure loss of only 50 Pa in the entire system and an SFP-value of 140 J/m^3. Currently ventilation design guidelines usually only aim to fulfil the requirements and do not consider methods to improve energy efficiency further. For example, a pressure gradient of 1 Pa/m for duct sizing is still being recommended (Wouters et al, 2001; Nilsson, 1995; ASHRAE, 2007). Furthermore, this has been introduced as a rule of thumb to avoid excessive noise generation rather than to achieve optimal energy efficiency. In practice, the SFP-value of a system depends on the efficiency of the fan combined with the volume airflow rate and pressure loss in the system.

The primary purpose of a ventilation system is to maintain an acceptable indoor air quality and thermal comfort in buildings. Therefore the ventilation system must be able to meet this requirement at all times as it otherwise loses its function. The efficiency of new fans today is around 80%, hence further savings potential in fan design is limited. Consequently, reducing system pressure loss is the only parameter that can be significantly improved. High pressure loss is the cause of the relatively high SFP-values in current systems. Currently the optimization of mechanical ventilation systems has focused on the development of efficient heat recovery – with success – and is a well-established part of current ventilation systems on the market. On the other hand the energy used to operate mechanical systems has been somewhat neglected. Future energy-saving potential is therefore in the fan power used to transport the air in the system. One reason for a lack of focus on fan power is that the market demand has been for HVAC systems that require minimal space in the building. If the SFP-value for mechanical systems is to be reduced, it will require a rethinking of traditional design solutions in order to reduce pressure loss. Relevant design aspects include component performance, system configuration, and integration in the building (Dokka et al, 2003; Wouters et al, 2001). Reducing the high pressure losses will also avoid the consequent high air velocities which can create noise and draught discomfort (Delsante et al, 2002; Malmstrom et al, 2002). Another key aspect in lowering the energy consumption is to ventilate only when necessary and in accordance with the actual demand. Various studies show that the control system has a significant effect on the energy consumption in terms of both operating hours and required fan power (Seppänen, 2007; Wei et al, 2004).

When developing, optimizing and designing a new ventilation system, the challenge is to find the optimal balance between economy, air quality, thermal comfort, energy consumption and environmental impact during heating and cooling periods (Heiselberg, 2002). This paper presents a concept for mechanical ventilation that provides comfort ventilation, increased ventilation to avoid overheating, and night cooling using minimum fan power consumption. This is a single system design aimed at fulfilling all ventilation needs using currently available components. The focus is to examine and display the performance potential of low pressure mechanical ventilation. This is exemplified by a case study office system that uses state of the art components and design solutions to fulfil future energy requirements without compromising the indoor environment.

The proposed ventilation system makes use of:

- A diffuse ceiling inlet (Nielsen et al 2009; Hviid et al, 2010a);
- Electrostatic precipitators;
- A static pressure reset control system (Wei et al, 2004; Hartmann, 1989);
- Standard components dimensioned to provide minimal pressure loss.

These improved components and design solutions lead to higher initial costs but, since ventilation systems can remain in service for 30 years, the lower operational costs for energy will outweigh the higher installation and purchase costs (Dokka,et al, 2003).

The performance of the case study system was simulated to evaluate energy consumption, indoor environmental conditions and cost. The goal was to achieve an SFP-value of 250 J/m^3, which is equal to reducing the requirement in the 2010 Danish Building Code by more than a factor of 8 and making the power consumption for night cooling negligible. The design also eliminates the need to install an automatic system for opening of windows. In this way, the annual energy consumption for the system could be reduced to 4-5 kWh/m^2, including a primary energy factor of 2.5 for electricity. This would correspond to 20-25% of the expected energy framework for 2020 in Denmark.

2. Concept

The ventilation system must provide an acceptable air quality and thermal indoor environment that meets the ventilation requirement all year round with minimal energy use. The concept was developed for a temperate climate similar to Denmark. This implies the use of heat recovery in the winter period with the option of increased airflow rate combined with a bypass of the heat exchanger in the summer, and night cooling in warm periods, to reduce the cooling demand during the day. In this instance, night cooling is defined as any ventilation required outside working hours to lower the temperature in the building. In this way, all ventilation needs can be met, and it is only necessary to install one system to obtain an acceptable indoor environment in the building. Fan power is reduced by optimizing the design as a whole and dimensioning every component to minimize pressure loss. The ventilation system was designed as a conventional mechanical system with an air handling unit and a duct system to distribute the air. As previously described, costs are reduced by using market available components.

3. Method

3.1 Case Study

To evaluate the performance of the developed ventilation concept and design solutions, a case study was carried out. A ventilation system for a typical office building was designed based on the concept described and its performance evaluated in simulations of its energy consumption and ability to maintain an acceptable indoor environment.

3.2 Test Case Office Building

The case study is based on a three storey building of plan dimensions 12x50 m intended for 150 occupants. Its long façades face north and south. The building is intended to fulfil the expected energy requirements for 2020, in the Danish Building Code, of 25 kWh/m^2. State-of-the-art solutions were therefore selected throughout the building. This means that the heat loads in the building from equipment, lighting and solar gains have been reduced to a minimum, to minimise the need for ventilation for cooling. The ventilation system is integrated with the building, with the intention of minimizing duct pressure loss. To reduce the duct lengths, and thereby pressure loss, the building is divided into East and West parts, with each supplied by separate AHU and duct systems. To allow for the use of natural driving forces and to avoid the take up space inside the building, the AHUs are located on the roof.

3.3 Ventilation Requirement

The system was designed to fulfil the requirements for thermal and atmospheric indoor environment Category II in EN 15251. This specifies an airflow of 7 l/s per person and 0.7 l/sm^2, assuming that the building materials are low polluting. The average occupancy is 12 m^2/person, which results in a minimum required ventilation rate of 1.1 m^3/s, corresponding to an air exchange rate of 1.5 h^{-1}. The ventilation rate required to remove heat loads and fulfil the thermal indoor environment was to be determined through simulations. The temperature range for Category II is 20 – 24 °C in winter and 23 – 26 °C in summer. However, in temperate climates, where the transition between summer and winter is long and vaguely defined, it is assumed that the occupants can adjust their clothing level during the working day, thereby expanding the comfort range to 20 – 26 °C for the summer period. A deviation from the temperature range is accepted for 5% of the occupied hours.

3.4 System Design

The AHU was designed and constructed as a conventional mechanical AHU, with the following components and capabilities to reduce the energy consumption and provide an acceptable indoor environment:

- A heat recovery unit with high efficiency to lower heating demand and low pressure loss to minimize fan power. A bypass option is included to avoid heat recovery in warm periods and to provide lower pressure loss;
- Air filters with minimal pressure loss, but able to remove harmful particles from the outdoor environment and avoid contamination of the AHU and duct system;
- A fan with high efficiency and demand control ventilation (DCV) control to minimize energy consumption

Heating and cooling coils were not included because their use in well-designed low-energy buildings under temperate climate conditions is limited and therefore dispensable. Recirculation was not included either, because it is rarely used in a temperate climate like that in Denmark

3.5 Heat Recovery Unit

Heat recovery is needed in temperate and cold climates to reduce heating demand. It is required in several national building codes and is standard in new mechanical ventilation systems. The efficiency of heat exchangers is up to 80-85% in new AHUs and the pressure loss is between 100-250 Pa (Nilsson, 1995). The potential for further energy savings for heating is therefore limited, but increasing the heat exchanger area can increase the efficiency to ≥ 90% and reduce the pressure loss to 30-60 Pa with standard components, and even further with special designs (Hoval, 2011).

3.6 Filtration

To obtain a healthy indoor environment, it is necessary to filter the air and thereby remove unwanted particles, fumes and odours from the indoor air. Filtration is also required to protect the AHU and duct system from contamination and thus impaired performance. In conventional ventilation systems, particles are usually trapped by bag filters and removed when the filter is changed. However, the trapped particles can still have a negative impact on the indoor environment (Bekö, 2008). The pressure loss across filters is around 100 Pa when clean and up to 250 Pa just before they need changing. The pressure loss in bag filters can be reduced to around 30 Pa by increasing the filter area (Tjelflaat et al, 1997; Delsante et al, 2002; Dokka et al, 2003), but this does not solve the problem of the negative impact of dirty filters. In the case study design, electrostatic precipitators (ESPs) were specified in place of bag filters. This is because they are more effective in removing ultrafine particles that can be harmful to humans and only have a pressure loss of 2 Pa. ESPs function by charging particles and aerosols in the airstream and removing them with an electrical field. Like bag filters, ESPs must be regularly cleaned to maintain efficiency. The negative aspects of ESP are high purchase price as well as high operating costs that are associated with power consumption. The power consumption depends on the number and size of ionized particles in the air. It is thus difficult to quantify the power required precisely because it will depend on the surrounding environment. For the component used in this paper, the producer specifies a power consumption of 60 W which is included in the SFP-value and total energy consumption of the ventilation system. (Peak, 2011).

3.7 Fan

A highly efficient fan is an essential part of a low-energy ventilation system. To achieve high efficiency, it is important that the fan is correctly dimensioned and well integrated to obtain appropriate flow conditions up- and downstream of the fan. This will give optimal running conditions for the fan and thereby high efficiency. Jensen et al (2003) showed that inappropriate design (dimensioning or integration) can decrease fan efficiency significantly and is one of the reasons for the high energy consumption of many ventilation systems today. There are many different fan types and models, each suitable for different purposes. The most commonly used in ventilation systems are radial fans with various blade shapes and angles. The efficiency varies between 55-85% depending on blade shape, angle, and required pressure increase and volume flow rate. For the case study an axial fan was specified because they operate more efficiently at low pressures than radial fans. The efficiency of the chosen fan was 78% and is powered by a frequency controlled motor with an efficiency of 82% (Novenco, 2011). To quantify the efficiency of the ventilation system, the Specific Fan Power (SFP) value is used (as given in Equation 1).

$$SFP = \frac{\sum P}{q_v} = \frac{\Delta p}{\eta_{tot}} \qquad (1)$$

Where:
$\sum P$ = sum of all the fan powers (kW)
q_v = gross amount of air circulated (m^3/s)
Δp_{tot} = total system pressure loss (Pa)
η_{tot} = total fan efficiency ($0< \eta_{tot}<1$)

SFP can also be expressed by the pressure loss divided by the total efficiency of fan, motor, converter, drive, etc. (Seppänen, 2007). The pressure loss, i.e. the fan power, depends on the airflow rate which varies over the year due to varying demand because of changing heat loads and the use of the building, so the SFP-value is not constant. To determine a representative SFP-value for the system, these variations must be taken into account, see Equation 2. Hourly data obtained by simulations give a good indication of the flow rate fluctuations over a year, and the appertaining pressure loss can be determined by setting up pressure loss characteristics for the system for the different operating conditions, such as night ventilation or bypass of heat exchanger. In this way, the pressure loss can be determined from the number of running hours and hence a representative SFP-value (Schild et al, 2009) can be obtained:

$$\overline{SFP} = \frac{\sum_{i=1}^{N}\left(\sum P_i \Delta t_i\right)}{\sum_{i=1}^{N}\left(q_{v,i} \Delta t_i\right)} \tag{2}$$

Where: $\overline{SFP}$ = average specific fan power
N = number of operation modes
Δt_i = duration of operation mode i

3.8 Duct Systems

In conventional mechanical ventilation systems, a duct system is used to transport and distribute the supply and exhaust air in the building. The design of the duct system and how well it is integrated in the building have considerable influence on the energy consumption, because inappropriate design will increase pressure losses in the system. To reduce the pressure loss, the duct system must be designed as aerodynamically as possible, with few bends and twists, short duct lengths, and as symmetrical as possible to limit the need for dampers (Liddament, 1996). A pressure gradient of 1 Pa/m is usually used to determine duct size. This is mainly applied to space needs against excess noise generation rather than optimising energy efficiency (Nilsson, 1995). However, by increasing the duct size one dimension, e.g. Ø200 to Ø250, the air velocity decreases by 35% and the pressure gradient by 60%, so increasing the duct 1 or 2 sizes results in significant reductions in pressure loss. For the case study building, the duct system was designed with a pressure gradient of 0.1 Pa/m.

3.9 Intake and Exhaust

There are countless types and designs of intake and exhaust components, and they typically have a design pressure loss of 40 Pa, however by choosing components 3-4 sizes larger, it is possible reduce the pressure loss to around 5 Pa for standard components on the market. This assumes no wind effects that could cause significantly increased pressure losses, so it is important to position the intake and exhaust components so that there are no negative wind effects. For optimum performance the exhaust terminal should be located where the prevailing pressure caused by wind is negative and the inlet terminal should be placed where the prevailing wind induced pressure is positive. Delsante (2002) describes various designs of intake and exhaust devices from other projects and he shows that it is possible to utilize wind and buoyancy effects positively by using specially designed intake and exhaust components. Wind effect and buoyancy are not taken into account for the case study system.

3.10 Diffusers – Supply and Extract

The supply and distribution of ventilation air in a room affects indoor air quality and thermal comfort. Conventional diffusers come in countless models and variations, but all require a pressure of 30-40 Pa to ensure effective mixing and avoid draught problems in the room. One alternative is to use diffuse ceiling inlets in which the air is supplied through the whole ceiling. These have been an area of recent research and are becoming more commonly used in commercial buildings (Nielsen et al, 2009; Jacobs et al, 2008). Diffuse ceiling inlets have several advantages; these include:

- Low pressure loss < 2 Pa;
- Large inlet area gives a good air distribution, lower air velocities and hence minimum draught problems;
- Ability to increase volume airflow rates without draught and noise problems (Hviid et al, 2010a; Nielsen, 2009).

The use of diffuse ceiling inlets also reduces cost because diffusers are not required and less ductwork is needed. This additionally enables increased floor to ceiling height because the space above the ceiling can be reduced. The plenum above the ceiling also increases the active thermal mass, because the supply is in contact with the upper concrete deck (Høseggen et al, 2009). In turn this assists with night cooling. Hviid et al (2010b) shows that significant free cooling can be obtained, but this effect is not be taken into account in the case study. Diffuse ceiling inlets cannot be used for extraction, because they will get filthy and unpleasant to look at. Therefore extraction was designed with standard extract diffusers that can be oversized and thereby limit pressure loss to 5 Pa. One concern about diffuse ceiling inlets is the potential soiling of the plenum that could lead to poor indoor air quality. However as the supply air is filtered the plenum is not expected to be contaminated. Only exhaust ducts require cleaning. Furthermore, investigations of reverse flow, where room air enters the plenum, show that it is limited, and polluted room air will therefore not contaminate the plenum (Hviid et al, 2010a).

3.11 Control Strategy

A key part of developing and designing a ventilation system is to use an appropriate control-strategy. The purpose of the control is to adjust the ventilation rate to the actual demand and avoid unnecessary energy consumption (Aggerholm et al, 2008). This strategy must focus on minimizing energy consumption for ventilation and provide an acceptable indoor environment without the use of mechanical cooling (Aggerholm et al, 2008). This can be quite a challenge, because there are many aspects to consider, such as energy consumption, thermal comfort, air quality, occupant satisfaction, security installation, running costs and integration with other installations, e.g. the heating system, solar shading (Heiselberg, 2002). With the use of demand control ventilation (DCV) the airflow can be varied according to the demand in each zone. This can provide significant energy savings of up to 50% compared to CAV-systems (Seppänen, 2007). DCV systems, however, increase initial costs while dampers or control boxes induce pressure loss in the duct system. Usually these systems are controlled by a fixed static pressure in the duct system maintained by the fan, which the damper or control box can work against to adjust the flow rate to the demand. This fixed pressure loss is usually set in accordance with the maximum ventilation demand in the building and in many cases is set higher for security reasons and fear of failure to maintain demand (Liu et al, 1997). This means the dampers never fully open and this results in unnecessary excess pressure loss in the system at all times. A solution is a static pressure reset (SPR) control, which varies (reset) the static pressure set point in the duct system continuously during operation in accordance with the demand, so that one damper is always fully open, Hartman (1989), Taylor (2007) and Wei et al (2004) show that using SPR gives energy savings between 30-50% compared to fixed pressure controls and Lui et al (1997) shows that the payback time of installing SPR control can be less than two months.

4. Simulation Input

Table 1 summarizes the pressure loss and data for the AHU components and duct system used in the

Table1. Pressure loss and data of principal components of the ventilation system.

	Supply		Exhaust	
	Pressure loss 1.1 m^3/s [Pa]	Pressure loss 1.9 m^3/s [Pa]	Pressure loss 1.1 m^3/s [Pa]	Pressure loss 1.9 m^3/s [Pa]
Electrostatic precipitator Size: 0.9x0.9m x2 Efficiency: EU10 Power: 60 W	2	2	2	2
Counter flow heat exchanger Size: 2.23x2.23m Efficiency: 89%	45	Bypass	45	Bypass
Axial Fan Size: d.900mm, stator. 578mm Fan efficiency: 78% Motor efficiency: 82% Power: 1.2 kW	-	-	-	-
Intake/exhaust Size: 1000x1000mm Type: LHPR	5	5	5	5
Diffuse ceiling inlet Type: Active/passive ventilation plates Material: Wood/concrete	2	2	5	5
Duct system Type: Circular Material: Steel	11	16	11	16
Flow dampers – control system Size: d. 250mm Type: LESX spherical shaped	10	10	10	10
Total	75	35	78	38

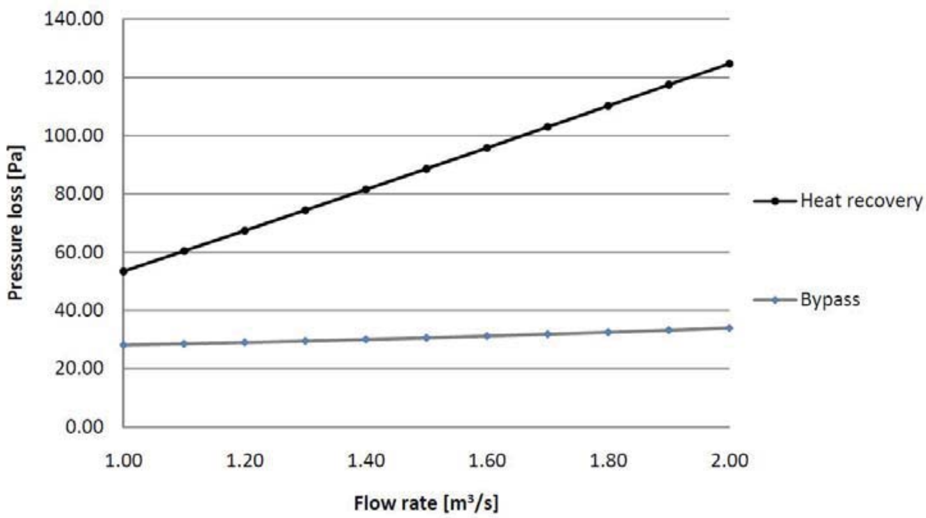

Figure 1. Pressure loss characteristics for the ventilation system in the case study building.

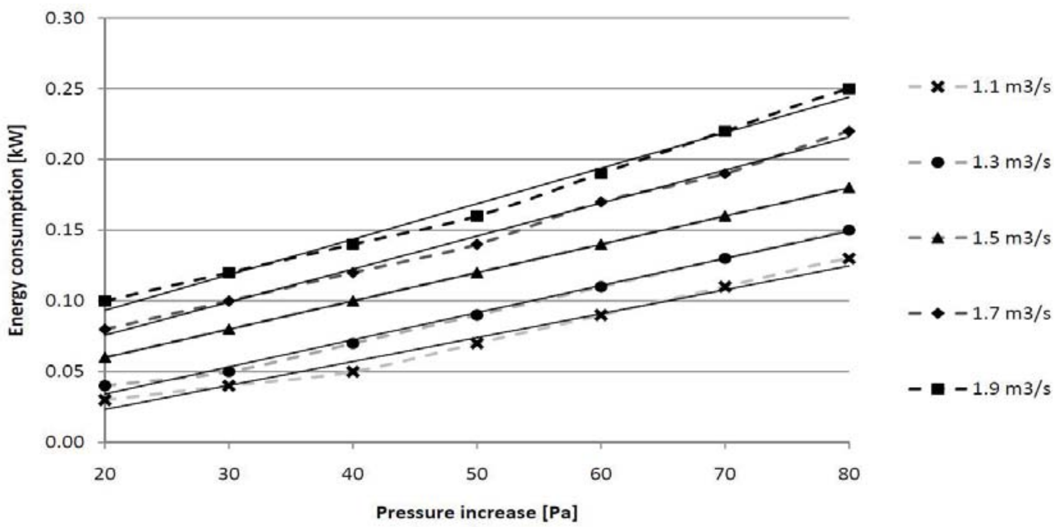

Figure 2. Power consumption of one fan at different flow rates as a function of pressure loss.

case study building for the minimum and maximum ventilation flow rate.

The total pressure loss in the system is between 75 and 78 Pa at minimum flow rate, with heat recovery for the supply and exhaust, and between 35 and 38 Pa at minimum and increased flow rate for bypass of the heat exchanger. Based on the pressure characteristics of all the components, a pressure loss characteristic for the system can be set up with and without heat recovery, see Figure 1. The pressure loss characteristics in Figure 1 show that the pressure loss increases to 125 Pa with heat recovery and to 35 Pa in bypass mode. The heat exchanger is bypassed in summer and during night cooling and whenever the ventilation demand is above 1.2 m^3/s in winter, because the increasing demand is caused by overheating, which obviates the need for heat recovery. Figure 2 shows the fan power consumption for coherent pressure losses and flow rates determined by Novenco (2011). The energy consumption for one fan is approximately 125 W for both the minimum flow rate with heat recovery and the maximum flow rate with bypass of the heat

Table 2. U-values of construction parts and internal loads in simulation models.

Constructions parts	U-value [W/m^{2}K]
Exterior wall	0.09
Roof	0.08
Deck	0.09
Window with exterior blinds dynamically controlled according to cooling set point (g-value = 0.55, T-value = 0.71)	0.82

Lighting, equipment and occupants	Heat load
Lighting system [W/m^2]	5.7
Equipment, 2 laptops + screens [W]	2 x 45
Occupants, 2 persons[W]	2 x 100

Table 3. Data for atmospheric and thermal indoor environment during occupant hours.

	South		North	
	iDbuild	IESVE	iDbuild	IESVE
Indoor environment Class I, [%]	22	29	13	18
Indoor environment Class II, [%]	78	71	87	82
Number of hours >26˚ C, summer [h^{-1}]	80	34	10	0
Number of hours >24˚ C, winter [h^{-1}]	20	0	5	0
Number of hours <20˚ C, all year	0	0	0	0

recovery system. To determine the annual energy consumption of the building and ventilation system, average SFP-value, and indoor environment, simulations were carried out in two simulation programs: iDbuild and IES Virtual Environment (iDbuild 2010, IES 2010). IDbuild is a Matlab-based simulation tool developed at the Technical University of Denmark for use in the early design process and IESVE is a commercial program that was used for a more detailed calculation of the building and provides comparison of results. In the simulations, the hourly ventilation demand to fulfil the indoor environment was calculated, so that it could be used to determine the hourly power consumption and annual average SFP-value of the system, using the data given in Figures 1 and 2. One representative room was modelled, assuming identical rooms throughout the building, and simulations were performed for two orientations (north and south). The hourly airflow was then added up to give the total airflow to be provided by one system. The winter and summer periods were defined as weeks 38-18 and 19-39, respectively, and the working hours were assumed to be between 08.00 and 17.00 Monday to Friday. Input data for construction parts and internal loads from installations and people are listed in Table 2.

5. Results

The hourly data from the simulations were analysed to investigate the indoor environment and the flow rate to fulfil the requirements. Table 3 shows the results of the indoor environment.

The atmospheric indoor environment was Class II or better during all working hours in both simulations. The indoor temperature exceeded 26 °C for maximum 80 hours in Summer and exceeded 24 °C for a maximum 20 hours in winter, respectively. This is within the acceptable deviation of 5%, so the requirements for the indoor environment were satisfied. The airflow rate needed to fulfil the indoor environment for every hour is listed in Table 4.

Table 4. Number of operating hours at different flow rate intervals from simulations in iDbuild and IESVE.

Flow rate [m^3/s]	<1.2	1.3	1.5	1.7	>1.8	Sum
			iDbuild			
South						
Summer [h]	470	4	3	15	417	909
Winter [h]	1370	5	9	15	50	1449
Night ventilation [h]	186	56	27	38	463	770
Sum [h]						3128
North						
Summer [h]	644	0	0	10	255	909
Winter [h]	1434	0	0	0	15	1449
Night ventilation [h]	240	0	0	0	142	382
Total [h]						2740
			IESVE			
South						
Summer [h]	378	32	15	25	450	900
Winter [h]	1353	17	19	35	34	1458
Night ventilation [h]	201	10	15	41	498	765
Sum [h]						3123
North						
Summer [h]	560	19	14	20	287	900
Winter [h]	1391	15	14	12	26	1458
Night ventilation [h]	216	8	7	3	165	399
Total [h]						2757

Table 5. System fan power at different flow rates and calculation of SFP-value based on iDbuild results.

Flow rate [m^3/s]	<1.2	1.3	1.5	1.7	>1.8
Average hourly fan power, winter [W]	218	102	166	208	226
Average hourly fan power, summer/night [W]	74	103	165	209	227
SFP-value, winter [J/m^3]	198	80	109	122	131
SFP-value, summer [J/m^3]	67	80	109	122	131

The number of operating hours from the two simulations is divided into the three operating conditions mentioned above. The results from the two simulations correlated quite well; there were some differences between the different flow rate intervals, but for the total number of hours the maximum deviation was below 1%. Based on the flow rate, the pressure loss in the system can be determined and thereby the energy consumption of the fan. The power needed to operate the system is shown in Table 5.

The SFP-value of the system was between 67 and 198 J/m^3, depending on the operating conditions. This does not, however, include the energy used for the ESPs, which was an additional 120 W during operation. The annual average SFP-value for the system calculated by Equation 2 is 329 J/m^3, including the power consumption for the ESPs, and 142 J/m^3 without. The power consumption is multiplied by a primary energy factor of 2.5, resulting in an annual energy consumption of 3.7 kWh/m^2 floor area, of which the ESPs account for 57%. The system was used for night cooling for 770 hours for the south facing offices and 338 hours for the north facing offices (see Table 4). The corresponding SFP-value during night cooling was between 245 and 285 J/m^3, with the power consumption for ESPs, resulting in an annual energy consumption for night cooling of 0.9 kWh/m^2 which represented 24% of the energy consumption.

6. Discussion

The case study building with balanced mechanical ventilation was simulated using two simulation programs. Overall the results from the two simulations correlate and the indoor environment results in Table 3 show that the requirement of Class II was fulfilled in both simulations. There was some deviation in the number of operating hours at the individual operating conditions, see Table 4, but the deviation in total number of operating hours was below 1%. There was hence no significant impact on the SFP-value whichever model was chosen. However the iDbuild results were chosen, because this simulation gave the highest number of operating hours with heat recovery and therefore gave the worst case annual average SFP-value. Of the annual energy consumption, night cooling accounted for 24% of the total, representing an annual operating cost of DKK 1300 at an electricity price of DKK 2/kWh. This is considered acceptable compared to a natural ventilation solution for night cooling.

A natural ventilation system would reduce the energy consumption and operating cost, but the acquisition cost would be greater than DKK 500,000. Natural ventilation could also cover most of the ventilation demand in the summer period, which would improve the investment, but it would cause draught problems and increase heating demand in the winter period, necessitating mechanical ventilation. The price of low-pressure ventilation systems will be higher than conventional systems, but how much is uncertain because the price increase of the larger components is unknown and it is difficult to put a price on the increased space use in the building. Overall, it is estimated that it will be economically feasible to use a low-pressure ventilation system, compared to installing a convention mechanical and a natural ventilation system, because the extra cost for larger components will be reimbursed by lower energy consumption (Dokka, 2003). Furthermore the low-pressure ventilation system ensures efficient filtration, improving the indoor air quality, especially in areas with high ultra-fine particle pollution from e.g. traffic pollution. The annual average SFP-value for the system is 330 J/m^3, 30% higher than the target, but this is mainly due to the power consumption of ESPs, which account for more than half of the total consumption. In the simulations, the ESP power consumption was set at 60 W, but this is not documented and could differ in practice. For example values ranging between 7 – 175 W were found from different producers. The pressure loss of ESPs, however, is only 2 Pa, which reduces the fan power needed. To find the optimal filter solution, an overall investigation of the relationship between power consumption, pressure loss and filtration efficiency is needed, but this was beyond the scope of this paper. The annual energy usage for the system of 3.7 kWh/m^2, however, met the target of reducing the consumption to less than 25% of the expected 2020 energy framework. This shows that by minimizing pressure loss it is possible to reduce the energy consumption for mechanical ventilation systems, so it can be used to help fulfil future energy requirements without renewable energy. The key element in achieving this result was reducing the pressure loss throughout the system to 30-75 Pa depending on operating conditions. An essential part of reducing the pressure losses is the use of state-of-the-art and oversized standard components.

The pressure loss in the heat exchanger was 30 Pa and it is not possible to reduce it much further with standard components, because lower air velocities could cause temperature separation in the air stream, thereby not utilizing the whole heat exchanger area. This would lead to impaired efficiency and an increased risk of freezing. The efficiency of the heat exchanger was 90%, which minimizes the heating demand for the ventilation air and obviates the need for a heating coil in the system under the simulated weather conditions. This was also aided by the diffuse ceiling that can handle a temperature difference of 8 °C between the room and inlet temperature without causing draught (Nielsen et al 2009). The diffuse ceiling inlet induces a minimal pressure loss of less than 2 Pa (Hviid, 2010a), far less than conventional diffusers that have pressure losses greater than 30 Pa. Nielsen et al (2009) shows that this method of distributing the inlet air is superior to conventional methods. No negative aspects of the method have been found in the literature, but it has to be investigated and assured that the pressure cavity and ceiling does not become dirty and pollute the inlet air.

The duct system in the case study was designed with a pressure gradient of 0.1 Pa/m, which gives a pressure loss between 11 and 16 Pa, depending on the flow rate. The duct sizes were 2 sizes larger than normal sizing, which will take up more space in the building and be a challenge in the design phase, especially for renovation projects. Hence proper integration and appropriate ductwork design is needed to reduce pressure loss in the duct system.

The other aspect in minimizing the pressure loss is to reduce the cooling ventilation demand by optimizing other installations, equipment and materials to minimise heat gains. This includes the use of solar shading, as well as low energy lighting systems and computers, etc. By reducing heat load, free night cooling becomes enough to maintain an acceptable thermal indoor environment in the case study building.

The maximum ventilation flow rate for the case study building was 1.9 m^3/s, corresponding to an air change of 2.5 h^{-1}. This is well below conventional systems, which can have flow rates up to 6-8 h^{-1}.

The last critical part of the ventilation system is the control system, because it has a huge effect on the pressure loss and energy consumption of a system. The use of static pressure set point reset control ensures that the fan power needed is minimized, because the static pressure is always reduced to a minimum. Several control strategies based on the static pressure reset concept have been developed, using different theoretical and practical approaches to determine the critical static pressure, ventilation demand, control airflows and to achieve a stable control system. Which approach is best will remain an open question here, but they all show that static pressure reset control can reduce the power consumption by up to 50% (Taylor, 2007; Wei et al, 2004).

The concept outlined in this paper is focused on applications in a temperate climate and therefore aspects concerning humidification, dehumidification, recirculation and mechanical cooling are not covered. Implementation of these components in hot and humid climates will increase the energy consumption as they use electricity and induce pressure loss that will increase the fan power needed. It will also require a more advanced control system to control those components and maintain an acceptable indoor environment. The design and control of such systems has a huge influence on the energy performance (Sekhar, 2004), but all aspects concerning those components are not part of the concept presented and therefore beyond the scope of this paper. There is however an increasing trend to move away from distributing large quantities of conditioned air from central air handling units and, instead, using decentralized air conditioning units. In such cases the proposed concept is assumed applicable for use under all climate conditions.

7. Conclusions

A low pressure drop concept for mechanical ventilation, based on the use of standard components, has been described in this paper in the form of an analytical case study. The results show that it is possible reduce the power and heating consumption of mechanical ventilation systems significantly and still maintain an acceptable indoor environment. The key points of the concept in the case study are the following:

- A single system is needed to fulfil all ventilation needs including night cooling with minimal power consumption;
- Low fan power is achieved by reducing pressure loss in the duct system and AHU components;
- Efficient heat recovery, combined with minimizing heating demand, avoids the need for heating coils in the MVHR system. Night cooling along with reduced internal and solar gains eliminates the need for cooling coils (mechanical cooling);
- Improved indoor air quality due to effective particle filtration can be accomplished using electrostatic precipitation filters (ESPs);
- a diffuse ceiling inlet provides effective mixing and minimizes draught and noise generation;
- An efficient demand control ventilation (DCV) control system with static pressure reset, that adjusts the airflow to the actual demand, decreases ventilation need to a minimum.

The chosen case study system was designed with, and is based on, known principles and components already available, so little development cost is needed to make the concept and system market ready.

No life cycle cost analysis or cost optimization of the system was carried out, so whether it will be economically beneficial to install such a system is uncertain under the current building requirements.

However, when the Danish energy requirements for 2020 are implemented, this concept will become competitive as it can fulfil all ventilation needs at less than 25% of the expected energy framework for 2020. Utilizing the low pressure mechanical ventilation concept will help make it possible to fulfil the 2020 energy requirements, so there is no need to install costly hybrid ventilation or combined natural and mechanical ventilation solutions to reduce energy consumption further.

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Paper II

A Static Pressure Reset Control System with a New Type of Flow Damper for Use in Low Pressure Ventilation Systems

S. Terkildsen, S. Svendsen

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A Static Pressure Reset Control System with a New Type of Flow Damper for Use in Low Pressure Ventilation Systems

Søren Terkildsen and Svend Svendsen

Department of Civil Engineering, Section of Building Physics and Services,
Technical University of Denmark, Brovej, Building 118, DK-2800, Kgs. Lyngby

Abstract

The control strategy for mechanical ventilation systems has significant impact on the performance of the system, in terms of energy consumption and correct air distribution. This paper presents a static pressure reset control system employing a new type of flow damper with lower pressure loss for use in low pressure ventilation systems. The flow damper has a droplet shape that minimizes turbulence generation and the resulting pressure loss. The performance of the damper was examined by making measurements of pressure loss and airflow. These were used to determine the required pressure loss for operation and the airflow accuracy. Results were compared to similar tests carried out with conventional flat plate dampers. A static pressure reset control algorithm was programmed and analyzed on a test system consisting of three dampers, representing three office rooms. The comparison of the dampers shows that the droplet shaped damper resulted in an airflow of 234 L/s at 30 Pa pressure loss, more than twice the airflow past a flat-plate damper at the same pressure loss (114 L/s). Also the droplet shaped damper could operate precisely down to 5 Pa pressure loss. The programmed control algorithm ensured that one damper was always fully opened, based on the flow demand to the zone, and that this reduces the static pressure loss under partial load by up to 53%. However, as the maximum static pressure during the tests was 15 Pa, the reduction was only 8 Pa, so it was not possible to document any significant fan power saving compared to a fixed static pressure control. A maximum static pressure of 15 Pa is a reduction of more than 50% compared to flat-plate dampers that required at least 30 Pa to operate precisely; this is expected to yield fan energy savings in practice.

Key words: control strategy, static pressure reset, energy consumption, low pressure ventilation, airflow damper.

1. Introduction

How a mechanical ventilation system is controlled has a large impact on its performance in terms of both correct air distribution and energy consumption. It is well established that VAV-systems (variable air volume systems) can reduce energy consumption compared to CAV-systems (constant air volume systems) (Aktacir et al, 2006; Englander et al, 1992a; Lorenzetti et al, 1992) but the magnitude of the reduction depends on the detailed operation of the VAV-control system (Yang et al, 2011). In VAV-systems the supply and exhaust flow rates vary according to the ventilation demand in the building. In this case the airflow is typically controlled by maintaining a preset static pressure in the duct system by adjusting the flow resistance of actuators in the duct system (i.e. dampers or terminal diffusers) to provide the desired airflow under all operating conditions (Wang et al, 1998). This static pressure set-point is based on full load conditions and in many cases set even higher to ensure capacity at peak loads. This approach results in excess static pressure (i.e. energy use) when the system operates at partial load (Wei et al, 2004; Federspiel et al, 2005; Liu et al, 1997). One way to reduce the static pressure is to employ an SPR-control (static pressure reset) that, based on continuous monitoring of zone demands, determines the critical zone (typically highest demand) and ensures that the actuator to that zone is always fully open. This ensures minimal static pressure in the duct system under partial load operation, while still providing sufficient airflow to all zones (Liu et al, 1997; Hartman, 1989; Englander et al, 1992b; Warren et al, 1993). This allows fans to operate at lower speed, thus reducing power usage by 30-50%. There are two overall methods to control the static pressure reset: a preset reset schedule based on parameters such as outdoor temperature, total supply airflow rate and supply air fan speed, or real-time monitoring of ventilation demand based on the airflow requirements at each control device (Wei et al, 2004). For real-time control, a feedback signal loop between the actuators and the fan is required, together with a control algorithm to determine the fan speed required. This requires the actuators to have an analog or DDC (Direct Digital Control) input, and output capable of transferring information about the ventilation demand, typically based on damper position or flow rate (Wei et al, 2004; Taylor, 2007). These actuators induce a relatively high pressure loss (>>30 Pa) that is required to maintain accuracy and control the airflow (flow resistance). It is however common to have higher static pressures. For example other studies on SPR-control systems report pressure ranges of 75-110 Pa (Wei et al, 2004), 35-325 Pa (Taylor, 2007), 145-325 Pa (Wray et al, 2008) and 120-400 Pa (Federspiel et al, 2005) up to 256-666 Pa (Liu et al, 1997) and 500-1000 Pa (Warren et al, 1993). These findings are still quite high. This is partly due to the inaccuracy and operating ranges of existing actuators and partly due to the fact that it makes systems easier to control and more robust if most of the pressure loss in the system is provided by the actuators. Hence actuators have priority in the system. In low pressure ventilation systems it is still preferable to have most of the pressure loss in the duct system across the actuators as the ductwork pressure loss can be in the range of 10 Pa (Hviid et al, 2010a; Tjelflaat, 2000). In addition, the pressure loss in the actuators can also be reduced. As mentioned above, conventional actuators do not operate accurately under such conditions. Current SPR solutions adjust the static pressure reset point with increments of 10-25 Pa (Federspiel et al, 2005; Taylor, 2007; Wray et al, 2008). These actuators are therefore not suitable for use in low pressure ventilation systems. This paper develops and analyzes a static pressure reset control system for use in low pressure ventilation systems. The control employs a new type of flow damper that can be controlled by both DDC and analog feedback. The damper has linear characteristic over its entire operational field which eases control and will, in theory, achieve accuracy even at very low pressure losses (<5 Pa). The damper has built-in flow measurement that enables the SPR control to adjust the airflow to each zone and maintain one damper fully open under all operating conditions. Experimental measurements made on a full scale test system were used to demonstrate the performance of the control system. These experiments were divided into two parts, (1) examining the performance of the new damper at low pressures and (2) determining the performance of a programmed SPR control algorithm based on the airflow demand to each zone. The objective was to determine the applicability of the new type of damper for low pressure ventilation systems and to develop an SPR-control system for use in such systems.

2. Method

To examine the performance of the damper and the SPR-control system, two test systems were installed in a test facility at the Technical University of Denmark. First, a preliminary setup was used to determine the performance of the damper when operating at low pressure. This was necessary because the damper was designed for use in process ventilation systems such as fume-hoods or welding stations and has therefore not been tested for low pressure ventilation purposes. Secondly, a mock-up of a small ventilation system was used to test the performance of the SPR-control system in terms of airflow accuracy, pressure loss, power consumption and stability. The power consumption was compared to measurements made when the system was operating with a preset static pressure control system.

2.1 Damper Design

The new damper design has a droplet shaped cylinder that can move forwards and backwards in the flow direction to control the flow resistance and thereby control the airflow (LeanVent, 2012) as shown in Figure 1. This aerodynamic design decreases turbulence generation within and down-stream of the damper so that pressure loss and noise generation are reduced. The airflow regulator box shown in Figure 1 measures the static pressure loss across the damper, and is used to determine the airflow through the damper according to Equation 1.

$$Q = k \cdot \sqrt{\Delta p} \qquad (1)$$

where:
Δp = static pressure loss across the damper [Pa]
K = pressure loss coefficient of the damper (k-factor) [L/s @ ΔP=1 Pa]
Q = flow rate through the damper [L/s]

The k-factor has a linear characteristic between opening position and flow rate, as may be seen in Figure 2. This makes the damper easy to control and is accurate over almost the entire opening range

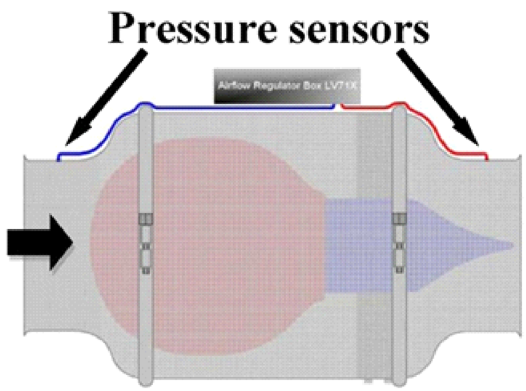

Figure 1. Drawing of LeanVent damper and placement of pressure sensors.

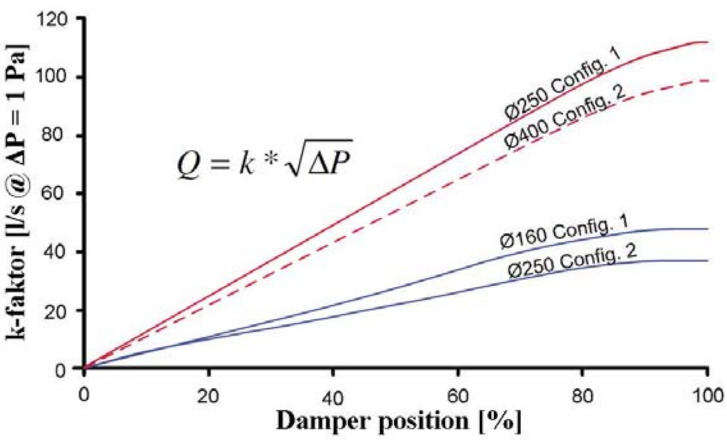

Figure 2. k-factor characteristic for various sizes of LeanVent dampers.

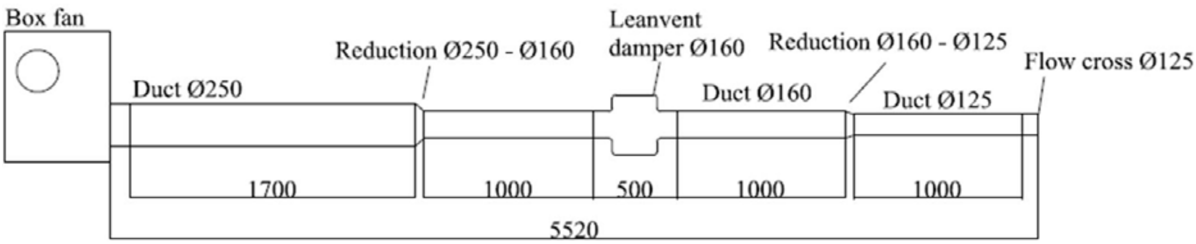

Figure 3. Schematic layout of the damper performance test stand (dimensions in mm).

from 0 to 100%. In contrast, conventional flat-plate dampers typically are limited to $45^o \pm 15^o$ angles (Lindab A/S, 2012). The k-factor characteristic is programmed in the airflow regulator box on each damper. The airflow regulator box uses the pressure loss measurements to calculate the position of the damper to match a given flow rate, and the flow rate is shown on the regulator box display. This correlation between pressure, position and airflow is used in the programming of the SPR-control. The damper has its own proportional-integral (PI) loop that must be tuned to ensure stable and robust control during operation.

2.2 Test Stand – Damper Setup

A schematic of the first setup is shown in Figure 3. The box fan was a model BESB 250-4-1 from Exhausto A/S, the damper size was 160Ø (mm) and the airflow and damper accuracy were measured using an MSD – Halton 125 flow measuring cross. The flow cross has an accuracy of ± 10% and the measuring results were corrected for the pressure loss between the damper and the measuring cross. The airflow was also measured using a hand-held Swema 3000 anemometer that has an accuracy of ± 0.05 m/s. Lastly the pressure loss across the damper was measured with a Retrotec DM-2 manometer to check the accuracy of the built-in pressure sensors. The fan was equipped with a frequency controlled variable speed drive (VSD) that was controlled by an analog signal of 0-10 V. In the experiments the damper accuracy was measured at fan speeds of '1', '3' and '5' corresponding to 1.0, 3.0 and 5.0 V. The damper position was fixed at every 10% interval from 0-100%. To validate the performance, the measurements were also performed with a conventional flat-plate damper that requires a minimum air velocity of 1.75 m/s, has a limited operating range of 45°±15° and requires a pressure loss of at least 30 Pa to function accurately (Lindab A/S, 2012).

2.3 Test Stand – Static Pressure Reset Setup

Figure 4 shows the schematic design of the test mock-up used to determine the performance of the SPR-control. The distribution of the air at each outlet was not investigated because room air distribution was not the focus of this paper. A pressure sensor was mounted in the duct system to adjust the fan speed and control the dampers. The placement of the pressure sensor is very important as improper placement can give imprecise control

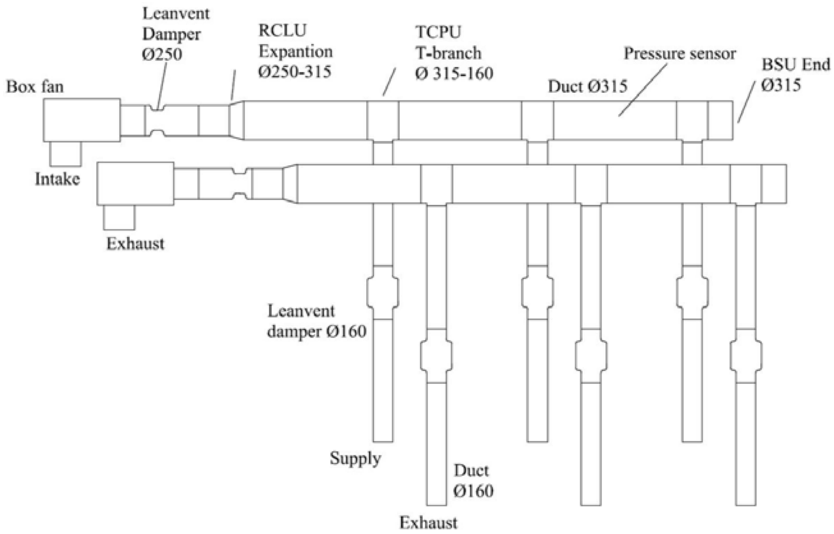

Figure 4. Schematic layout of SPR-control test stand.

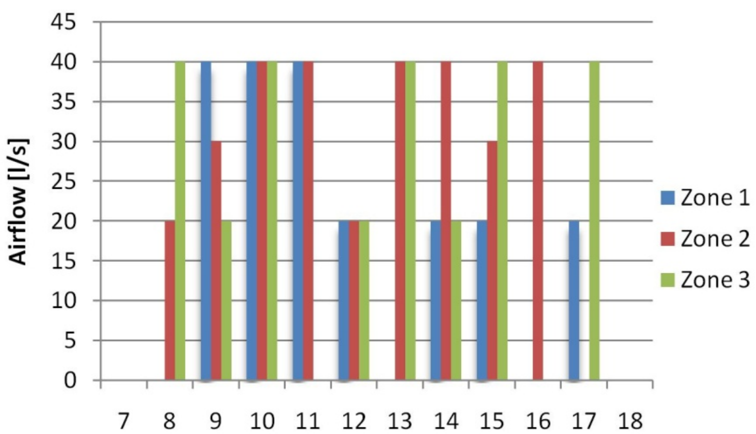

Figure 5. Fictive load schedule for SPR-control system.

that leads to either excessive energy use or insufficient airflow to certain zones in the system. The conventional rule of thumb recommends that the sensor is placed in the main duct 2/3 downstream from the fan (Nilsson, 1995; ASHRAE, 2011). Furthermore ASHRAE (2008) recommends a sensor placement between 75-100% from the first to the most remote terminal. However the airflow distribution before and after the sensor placement should be taken into account as well. In the test stand the pressure sensor was placed in the main supply duct between the 2nd and 3rd zone. This position does not completely comply with the guidelines in terms of distance from the fan or first terminal, but 2/3 of the airflow was distributed to zone 1 and 2. Therefore this position was used and no other sensor placements were investigated. The Ø250 dampers were inserted to represent a longer duct section and increase the pressure loss as the fans were oversized. Their position was fixed at 30% so the pressure loss varied with the flow rate. In the measurements a fictive load schedule was used to examine how the control algorithm performs under various transient load conditions. The airflow to each zone was varied from 20 to 40 L/s, corresponding to the ventilation demand in an office with 1 to 4 persons, as shown in Figure 5. This results in air velocities of 1-2 m/s at the dampers.

2.4 SPR Control Algorithm

As first stated in Hartman (1989), the concept of SPR-control is to always to ensure that one damper is fully open to minimize the pressure loss in the system. The challenge is to identify the correct damper to be fully open. In this paper the relevant damper is defined as the critical damper, under all operating conditions. The actual critical damper varies depending on the required airflow to each zone and pressure difference between the zones in the system. The control algorithm in this paper was developed for real-time monitoring of the actual demand in each zone. Therefore, the critical damper was identified according to the damper positions determined by the fictive load schedule. The control algorithm was modelled using the software tool Labview and the damper positions were transmitted to a computer by an analogue 0-10 V signal

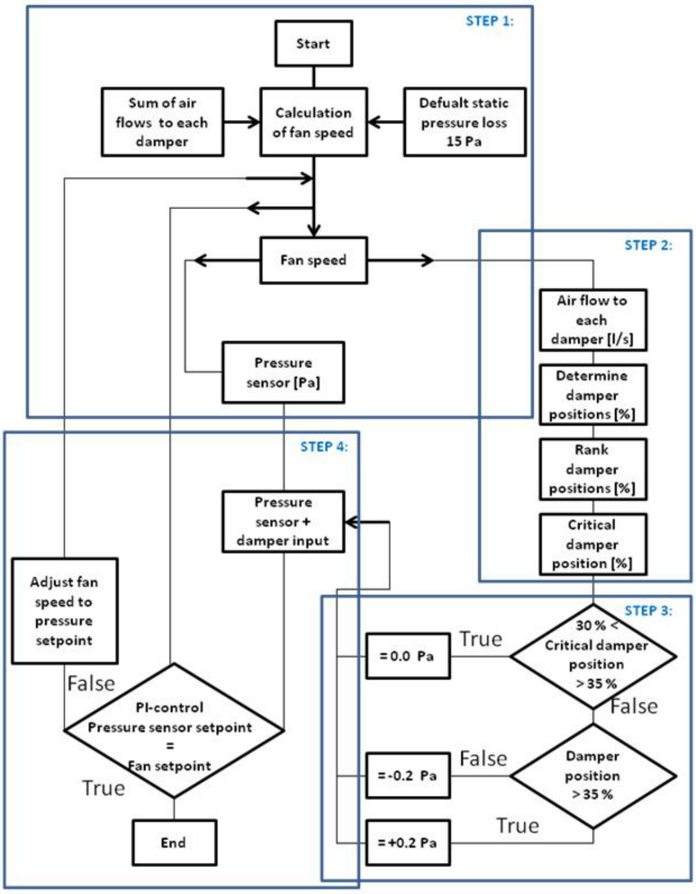

Figure 6. Schematic layout of the SPR-control algorithm.

(Labview, 2012). The structure of the control algorithm is similar to the one presented by Warren et al. (1993) and, in general, it can be divided into 4 main steps that are illustrated in Figure 6.

Step 1: The sum of the airflow to each damper and a default start pressure loss set to 15 Pa is used to determine the fan speed based on the fan characteristics. The fan then raises the static pressure to the default value and the dampers adjust to match the desired airflow.

Step 2: The position of the dampers are ranked according to their position to determine which one is critical (i.e. the critical damper is the one with the highest opening percentage). The position of the critical damper is then forwarded to check whether it is fully open.

Step 3: The position of the critical damper must be within a given range to be deemed fully open and if this criterion is met a signal of 0.0% is forwarded. (The range for a fully open damper was defined to be 30-35% in the experiments. This range was determined in preliminary tests showing that the damper could not operate at higher opening percentages because the pressure loss across it became too low.) If the damper position is outside the range a signal corresponding to ± 2.0 Pa is forwarded depending on whether the damper position must be increased or decreased.

Step 4: The values from Step 3 are added to the measured static pressure in the main duct thereby forcing the speed of the fan to be adjusted. The loop is then run until the critical damper meets the criterion, forwarding a signal of 0.0% in Step 3 when the pressure sensor and fan pressure set point are the same.

During the experimental work the measured airflow in the dampers was monitored and compared to the set point of the fictive load schedule to determine the accuracy. The airflow measurements in the dampers were validated by anemometer measurements of the air velocity. The performance of the SPR-control system was compared to the performance of a CSP-control system (constant static pressure) in terms of pressure loss and fan power consumption. For the CSP-control the static pressure was fixed at 15 Pa, corresponding to the static pressure required at full load. The same fictive load schedule was used in all experiments and the exhaust was operated as a slave of the supply.

3. Results

3.1 Damper Setup Results

The accuracy test of the LeanVent damper was performed, as explained in Section 2.2, at fan speeds ‘1’, ‘3’ and ‘5’ which resulted in air velocities of 2.5, 4.5 and 6.5 m/s in the duct system, respectively. Higher air velocities would be uncommon in duct systems due noise generation; thus measurements at higher fan levels were not performed. Figure 7 shows the results from the measuring cross, of anemometer measurements and average damper readings for the three fan speeds. The flow cross and anemometer readings gave steadily increasing curves while the damper reading were more

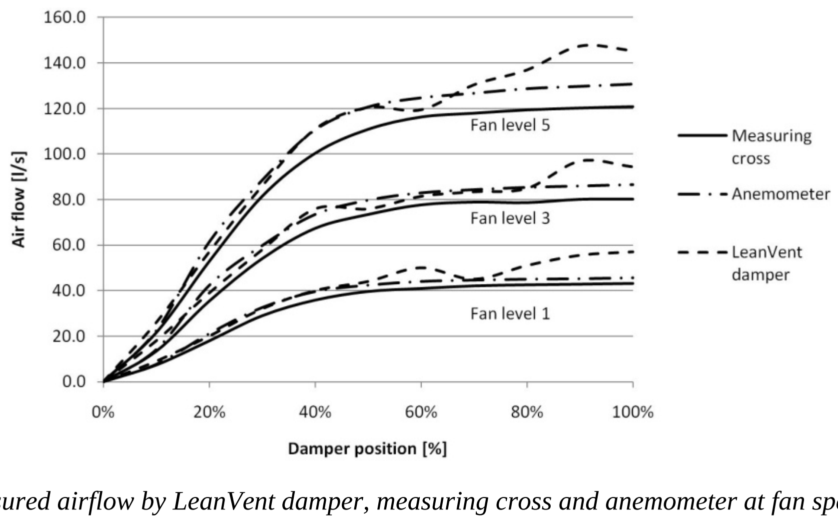

Figure 7. Measured airflow by LeanVent damper, measuring cross and anemometer at fan speeds 1, 3 and 5.

Table 1. Pressure loss across the LeanVent damper at fan speeds 1, 3 and 5.

Opening percentage [%]	0	10	20	30	40	50	60	70	80	90	100
Pressure loss [Pa] - fan level 1	20	18	16	12	8	6	5*	4*	4*	4*	4*
Pressure loss [Pa] - fan level 3	65	60	51	38	24	13	11	9	7	6*	6*
Pressure loss [Pa] - fan level 5	137	128	108	78	48	32	21	15	12	11	10*

* Pressure loss across the damper is too low for the pressure sensors to measure and thereby control the airflow accurately.

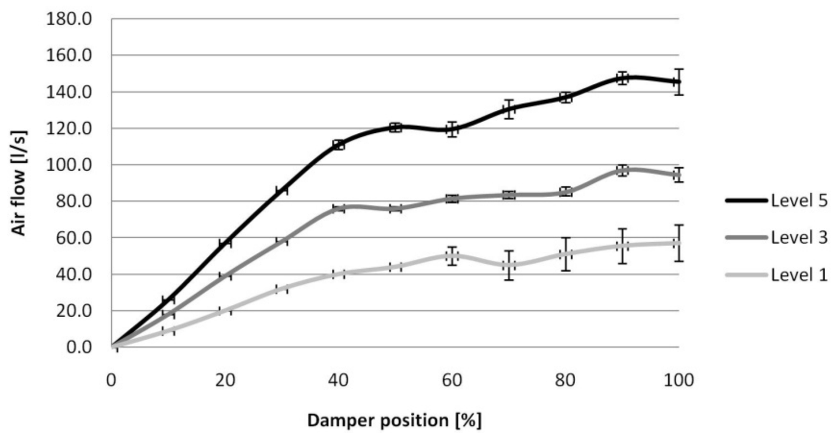

Figure 8. Deviation in damper airflow measurements at fan speed 1, 3 and 5.

unstable. The airflows measured by the flow cross were about 10% lower than the anemometer readings throughout the measurements. The airflow measured by the damper correlates within 5% deviation to the anemometer reading up to 50% opening percentage. Above that the deviation increases up to 25% as the pressure drop across the damper falls below 5 Pa, see Table 1.

Low pressure losses across the damper also resulted in unstable airflow readings from the dampers, although the position of the damper did not change. The range of airflow rate signal fluctuation for the damper airflow readings during the measurements are shown in Figure 8 for speeds 1, 3 and 5. At fan speed 1 the damper reading became unstable (above ± 3 L/s) at an opening percentage of 60%. This was when the pressure loss across the damper, shown in Table 1, dropped to 5 Pa. For fan speeds 3 and 5 the pressure loss never dropped to 5 Pa and therefore instability did not occur to the same degree. However, the readings had a tendency to become unstable when the damper was fully open and the fully open criterion was therefore defined as 30-35% open in all further work. Figure 9 shows coherent pressure loss and airflow readings for the LeanVent and the flat-plate damper over their opening range. The curves were fitted based on data retrieved from measurements performed at fan speeds 1, 3, 5 and 8. At a pressure loss of 30 Pa across the dampers, the airflow was 234 L/s and 114 L/s for opening percentages of 90% and 50% for the LeanVent and flat-plate damper, respectively. This means that, at 30 Pa, the airflow rate through the LeanVent damper was 105% higher than the airflow through the conventional flat-plate damper.

3.2 SPR-Control Setup Results

During the measurements, the exhaust system was operated as a slave of the supply system and therefore only the results for the supply system are presented. The accuracy of the damper airflow measurement presented in the previous section was assumed to be acceptable as the dampers will have been operating at opening percentages less than 50%. Therefore no additional airflow measurements were performed during SPR-control setup. Table 2 shows the measured airflows and damper positions for the three dampers under SPR-control operation

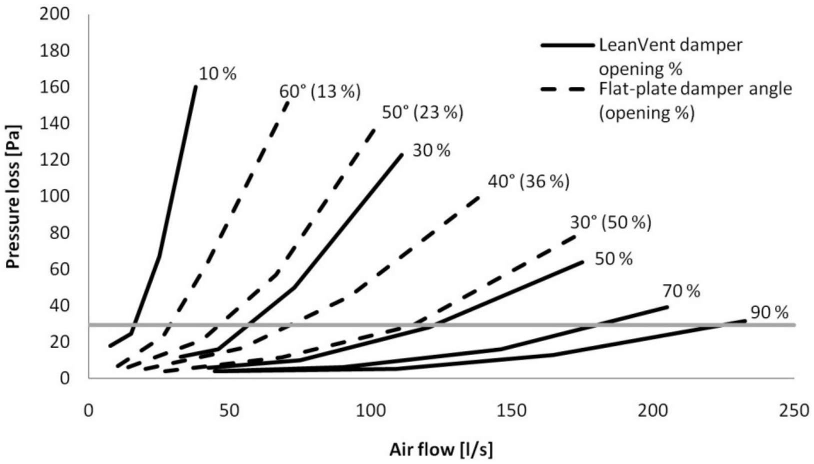

Figure 9. Coherent airflow and pressure loss results for the LeanVent and flat-plate damper measured with a flow cross and micro manometer.

Table 2. Accuracy of measured damper airflow compared to set point value under SPR-control operation and the opening percentage of the critical damper operating under the fictive load schedule.

	Time (h)	8	9	10	11	12	13	14	15	16	17
Damper 1	Set point [L/s]	0	**40**	**40**	**40**	**20**	0	20	20	0	20
	Damper position [%]	3	**35**	**33**	**35**	**32**	2	17	18	3	20
	Damper airflow [L/s]	3	**38**	**39**	**38**	**20**	2	17	17	3	20
	Deviation [L/s]	-3	**2**	**1**	**2**	**0**	-2	3	3	-3	0
	Deviation [%]	-	**5**	**3**	**5**	**0**	-	15	15	-	0
Damper 2	Set point [L/s]	20	30	40	40	20	**40**	**40**	30	**40**	0
	Damper position [%]	23	29	33	34	34	**35**	**35**	29	**35**	3
	Damper airflow [L/s]	19	31	39	40	23	**38**	**38**	28	**38**	3
	Deviation [L/s]	1	-1	1	0	-3	**2**	**2**	2	**2**	-3
	Deviation [%]	5	-3	3	0	-15	**5**	**5**	7	**5**	-
Damper 3	Set point [L/s]	**40**	20	40	0	20	40	20	**40**	0	**40**
	Damper position [%]	**35**	19	30	3	26	33	19	**35**	2	**35**
	Damper airflow [L/s]	**38**	21	36	3	20	36	20	**41**	2	**36**
	Deviation [L/s]	**2**	-1	4	-3	0	4	0	**-1**	-2	**4**
	Deviation [%]	**5**	-5	10	-	0	10	0	**-3**	-	**10**

and the numeric and relative deviation between the two.

The position of the critical damper (in bold) was 30 to 35% open during operation as specified in the control algorithm, the deviation between the set point and damper reading was within -3 L/s to 4 L/s and the relative deviation was within ±15%. The results of the static pressure measurements in the duct system and the pressure losses across the three dampers are shown in Figure 10 for both SPR-control and CSP-control operation. The pressure loss across the dampers varied from 11 to 14 Pa with CSP-control and the static pressure set point in the duct was always 15 Pa as specified in the control. With SPR-control the pressure loss across the dampers varied from 3 to 11 Pa and the static pressure set point varied from 7 to 15 Pa depending

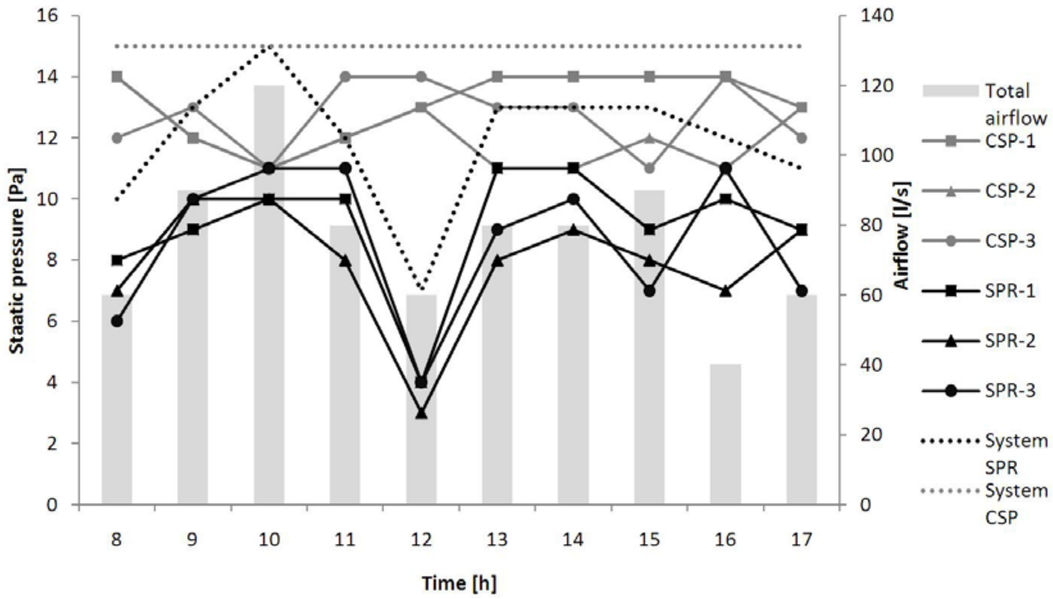

Figure 10. Measurement results of static pressure set point in the duct system and pressure loss across the dampers with CSP-control and SPR-control.

Table 3. Pressure loss in dampers with SPR-control and CSP-control and fan power consumption for one fan.

	8	9	10	11	12	13	14	15	16	17
Total airflow [L/s]	60	90	120	80	60	80	80	90	40	60
Fan power – CSP-control [W]	25	32	47	31	24	31	30	32	22	23
Fan power – SPR-control [W]	25	28	46	29	22	28	29	26	20	22
SFP-value – CSP-control [J/m^3]	417	356	392	388	400	388	375	356	550	383
SFP-value – SPR-control [J/m^3]	417	311	383	363	367	350	363	289	500	367
Fan power saving [%]	0	13	2	6	8	10	3	19	9	4

on the individual and total airflow, which is a reduction of up to 8 Pa or 53% in static pressure set point under partial load with the load schedule that was simulated. On average, the reduction in static pressure set point was 3 Pa or 20%. In general, the pressure losses across the individual dampers follow the curve of static pressure set point, which is expected as the static pressure set point is determined by the total airflow and the position of the dampers. When combining the results (see Figure 10 and Table 2), there is no clear correlation between the individual airflow and pressure loss across the dampers, as the damper position (i.e. the pressure loss) is influenced by the individual demand as well as the total airflow. The resulting fan power use during operation with the two control strategies are listed in Table 3 together with the SFP-value and energy saving for the supply system.

Fan power consumption was reduced with SPR-control and the savings varied from 0 to 4 W or 0 to 19% depending on the total airflow. On average the savings were 7.5% during the fictive load schedule. The fan power increased with increasing flow but was not affected by how the airflow was distributed between the three dampers. However, there was no clear correlation between fan power use and the reduction in static pressure set point shown in Figure 10.

4. Discussion

This preliminary investigation of the LeanVent damper shows that it can control and regulate the flow rate at pressure losses down to 5 Pa, while below that the readings became unstable and inaccurate. This is due to the accuracy limitation of the pressure sensors used in the damper, so the dampers could only be 30-35% open during SPR-control measurements, to maintain accuracy. It is possible to install more accurate pressure sensors but the producer did not have reason to do so as the current sensors are more than adequate for the applications for which the damper was originally designed. The pressure loss of 5 Pa is, however, still much lower and a significant improvement compared to the 30 Pa that conventional flat-plate dampers require to operate accurately. The production cost to install more accurate sensors would be higher and is therefore unlikely to be economically acceptable for a relatively small additional reduction in pressure loss with today's electricity prices.

At low pressure losses, the damper flow rate readings became unstable, as shown in Figure 8. Because of this the dampers do not adjust and this affects the precision of the control system. This can be avoided by increasing the sampling time of the damper readings. In the experiments the pressure loss across the damper was measured every 100 ms and every second the average pressure loss was calculated and recorded. However, if the averaging period was increased the stability of the measurements would also increase.

In the comparison of the LeanVent and flat-plate damper, Figure 9, the measurements gave a maximum flow rate of 234 L/s for the LeanVent damper and 114 L/s for the conventional flat-plate damper at the same pressure loss over the damper. Thus there is a 105% higher airflow through the LeanVent damper at the same pressure loss. This is due to the limited operating range of the flat-plate dampers from 13 to 50 % (45°±15°) while the LeanVent damper can operate from 0 to 90 %. The operating range of the flat-plate dampers is limited as they generate too much turbulence at lower opening percentages, and are inaccurate at low air velocities and higher opening percentages and so unable to precisely control the airflow.

The control of the static pressure set point functioned as intended i.e. providing a reduction in static pressure at partial airflow. The static pressure was reduced by up to 8 Pa or 53% under partial load, but there was no correlation between the reduction in static pressure set point or total airflow and fan power consumption, as would be expected. This is due either to inaccurate control of the fan, which seems most probable, or to inaccurate power readings. The fan control was based on the fan characteristic that was determined experimentally. However, as the fans were oversized and can provide an airflow up to 600 L/s, it was necessary to insert a damper after the fan to increase the pressure loss in the duct system. Only a small part of the fan curve was therefore used during operation, which made it difficult to control the fan, and this could have led to varying fan power readings. The fan power was measured with an electricity meter with an accuracy of ±1% that has been confirmed by calibration, but as the savings varied between 0 to 4 W an error of 1 W could strongly affect the results. Furthermore the measurements were performed using the fictive load schedule which was set up to determine how the control algorithm handled varying load conditions and its ability to determine the critical damper. They were not performed to document energy savings potential. The energy savings potential is therefore not clearly determined because it is very system dependent and a test facility with only three dampers is not a representative case to document the savings potential in practice. A main part of this system's energy savings potential in practice lies in the use of LeanVent dampers that reduce the pressure loss by around 25 Pa compared to conventional dampers.

The energy measurements were performed to document an energy saving using SPR-control compared to CSP-control and, in general, the SPR-control provided energy savings compared to CSP-control although the savings are somewhat limited. Small savings were expected as Wang (1998) has stated: i.e. the higher the proportion of static pressure set point in the total supply fan delivery head at full load, the greater the energy saving potential for static pressure set point optimization. However it was still expected that there would be a correlation between either static pressure set point or total airflow and fan power consumption.

The system operates well in the test setup but how the air is distributed at each outlet or how a diffuser will interact with the dampers was not investigated. A trial and error tuning will therefore almost always be required during commissioning to obtain appropriate fan characteristics and PID-control when the system is installed in practice (Taylor.

2007). The SPR-control algorithm developed here is expected to be appropriate for use with diffuse ceiling inlets (Hviid et al. 2010b, Nielsen et al. 2009) as it has no control device itself and is suitable for low pressure ventilation systems.

5. Conclusions

A preliminary investigation of the LeanVent damper showed that it is capable of controlling and regulating airflows at air velocities down to 1 m/s and a pressure loss of only 5 Pa. This is a significant improvement compared to conventional flat-plate dampers that require pressure losses of 30 Pa to function precisely. The improved performance is due to the new aerodynamic droplet design of the LeanVent and to built-in airflow measurement. The results show that, at a pressure loss of 30 Pa, the airflow through the LeanVent damper is approximately twice that of a flat plate damper. The built-in flow measurement makes it possible to control the airflows based on the demand in a zone and this was used to identify the critical damper in the SPR-control algorithm. The algorithm worked as intended and the static pressure set point was reduced by up to 53% (8 Pa), but due to the low fixed pressure set point of 15 Pa it was not possible to obtain a correlation with fan power savings. The results indicate that fan power consumption is reduced under partial load with SPR-control, but inappropriate sized fans made it impossible to document the energy saving potential precisely. However, overall the experiments documented that it was possible to develop a control system that can regulate the airflows accurately with less than 15 Pa pressure loss.

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Paper III

Performance of Perforated Suspended Ceilings as Diffuse Ventilation Air Inlet in School Classrooms

S. Terkildsen, C.A. Hviid, S. Svendsen

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Performance of Perforated Suspended Ceilings as Diffuse Ventilation Air Inlet in School Classrooms

Søren Terkildsen, Christian Anker Hviid and Svend Svendsen

Department of Civil Engineering, Section of Building Physics and Services

Technical University of Denmark, Brovej, Building 118, DK-2800, Kgs. Lyngby

Corresponding author: Terkildsen, S. (sterk@byg.dtu.dk)

Abstract

Diffuse ceiling ventilation is a novel air distribution device that combines the suspended acoustic ceiling with ventilation supply. Ventilation air is supplied above the acoustic tiles, effectively creating a plenum which distributes the supply air to the room through the ceiling. There is no distinct jet, and therefore the term diffuse. The concept has proven its performance in laboratory experiments, but to study the performance in real conditions a classroom was retrofitted with mechanical ventilation and a diffuse ceiling. The employed ceiling is comprised of active and passive panels that are permeable and non-permeable, respectively. The performance was studied with regard to risk of draught and ventilation effectiveness at two different air flow rates approx. corresponding to 5 and 10 l/s per person and the experiments were carried out during class to obtain realistic conditions. Furthermore was smoke visualization and infrared pictures performed to visualize the air flow and pupil tests and questionnaires to examine the perceived air quality and how improved air quality affects the pupils. At both air flow rates did the ceiling perform satisfactorily with air movements and temperatures within the requirements of European indoor climate standards. The ventilation effectiveness was comparable to conventional mixing ventilation. The overall results support the previously obtained experiences from laboratory experiments.

Keywords

Diffuse ceiling ventilation, ventilation efficiency, ventilation, Draught rate, tracer gas

1. Introduction

A recent study showed that 56 % of Danish schools have poor indoor environment with CO_2-concentrations above 1000 ppm and up to 4000 ppm in severe cases (Menå et al, 2010). Similar results have been reported for Dutch and Swedish schools and the poor air quality adversely affects the performance and wellbeing of the occupant (Jacobs et al, 2008, Smedje et al, 2000).School classrooms are also characterized by high occupancy and thus high thermal load that results in high ventilation demand which can be difficult to fulfill with conventional inlet diffusers with causing draught. Development of new concepts to ventilate school classrooms is therefore relevant and one promising solution is diffuse ventilation air inlet. The principle of diffuse ventilation air inlet is to inject the supply air into the plenum above a standard suspended ceiling that functions as a distribution chamber. A small overpressure is created in the plenum and the air is forced down into the room through cracks and perforations in the ceiling surface.

This gives a very large inlet area and hence low air velocities that reduces the risk of draught, noise generation and facilitates efficient mixing of the supply air resulting in an improved indoor environment. The air flow through the ceiling has random directions when it enters the room and thus the term diffuse. Furthermore is the concept economical beneficial if a suspended ceiling is to be installed for acoustics purposes. The plenum reduces the need for distribution ducts and diffusers thereby reducing investment cost, labour cost and installation time. Lastly is the pressure drop low compared to conventional diffusers, reducing the fan power required and hence operation cost and CO_2-emissions. The concept is commonly used in live-stock buildings and clean room industry, and is increasingly being used for comfort ventilation. The reported research in this area is however limited and mostly rely on laboratory experiments where results have been promising. In Nielsen et al (2009) diffuse ceiling inlet came out on top in comparison with 5 conventional air distribution systems in terms of ability to supply high flow rate and cooled air without causing draught. Tracer gas measurements in Hviid et al (2013) at a test facility office room showed that diffuse ventilation inlet provided perfect mixing while air temperature and –velocity measurements did not disclose any local discomfort in the occupied zone over a broad range of flow rates and inlet temperatures. Similar findings were reported in Fan et al (2013) where the same measurements were performed in the same test facility on a different ceiling type. In Jacobs at el (2008) measurements were carried out in a test facility resembling a small classroom, but the measurements included no quantifiable measurements of air temperature and velocity or ventilation efficiency.

Numerical analysis of diffuse ceiling ventilation by CFD (Computational Fluid Dynamics) have shown that thermal plumes from occupants or equipment can obstruct the supply air (Hviid et al, 2013; Jakubowska, 2008; Fan et al, 2013). The supply air is forced it to areas with no heat loads where it drops down and flows along the floor and this phenomena is also reported for life-stock buildings (Jacobsen et al, 2004). This could lead stagnant or short circuit depending on the placement of the exhaust diffuser, as well as draught problems at ankle height. Overall, the test results reported are promising, but the concept's performance in practice has not yet been documented.

Jacobs et al (2008) also report experiences and measurements from a pilot study where a diffuse ceiling inlet was installed in a school classroom were also reported. The measurements however only included the CO_2-concentration and a questionnaire to the occupants about perceived air quality, but did not investigate how the improved air quality and comfort affected the pupil's ability to perform school tasks. Previous studies on air quality effect on the occupants has mostly focused on adults performing office work, and most studies have been carried out in laboratories where the subjects knew they were not performing real office work only realistic examples of it. It is well established that adults performance can be improved up to 5 % by maintaining the CO_2-concentration below 1000 ppm (Mendell et al, 1993; Seppänen et al, 1999; Wargocki et al, 1999 and 2002). The effect on school children is less substantial, but there are several studies showing that the air quality have an increased effect on the pupils compared to adults and the performance can be improved by up to 15 % (Myrhvold et al, 1996; de Gids, 2006; Wargocki et al, 2007; Shaughnessy et al, 2007; Bakó-Biró et al, 2011).

To investigate the performance of diffuse ceiling inlets under real conditions the concept was installed in two classrooms at Vallensbæk primary school. The diffuse ventilation ceiling was made with cement bonded wood wool panels that consist of active panels that air permeable and passive panels that are non-permeable. The investigation encompassed several elements to document the thermal comfort and map the air distribution in the classrooms, including; air temperature and air velocity measurements, tracer gas, pressure drop across the ceiling panels, thermal camera pictures and smoke visualization to visualize the air movements in the room. Furthermore were learning performance tests and perceived air quality questionnaires given to the pupils to investigate how the new supply concept and improved ventilation affected their learning ability and well-being. The goal is to validate how diffuse ceiling inlet performs in practise document whether the diffuse ceiling ventilation concept is applicable for use in school classrooms.

2. Method – test setup

Two classrooms at Vallensbæk primary school was refurbished with a new mechanical ventilation system including a new suspended acoustic ceiling functioning as diffuse ceiling inlet. To determine the performance of the diffuse ceiling the following experiments was carried out;

- Tracer gas measurements to examine the air distribution and determine mean age of air and air change efficiency.
- Air temperature and -velocity measurements at different points in the room, to investigate local thermal comfort.
- Artificial smoke added to the supply air to visualize air flow patterns in the room.
- Infrared pictures to measure the ceiling temperature and determine discomfort for radiation asymmetry.
- Pressure drop across the suspended ceiling and single panels to determine the distribution of inlet air across the ceiling.
- Pupil tests to examine how improved ventilation effect their performance to do different tasks and questionnaires to determine perceived air quality and comfort.

The measurements were only conducted in one room, as the two rooms are almost identical, the pupil test and questionnaires were however carried out in both classrooms and the CO_2-concentration monitored during the tests.

2.1. Room description

Vallensbæk School has outer masonry walls and partitions of gypsum. The floor is a concrete deck with linoleum covering and the ceiling is a wooden roof structure, see Figure 1. The ceiling was suspended 0.2 m from the roof structure to create a plenum to distribute the air and resulted in a floor height of 2.5 m in the main part of the room, see Figure 1.

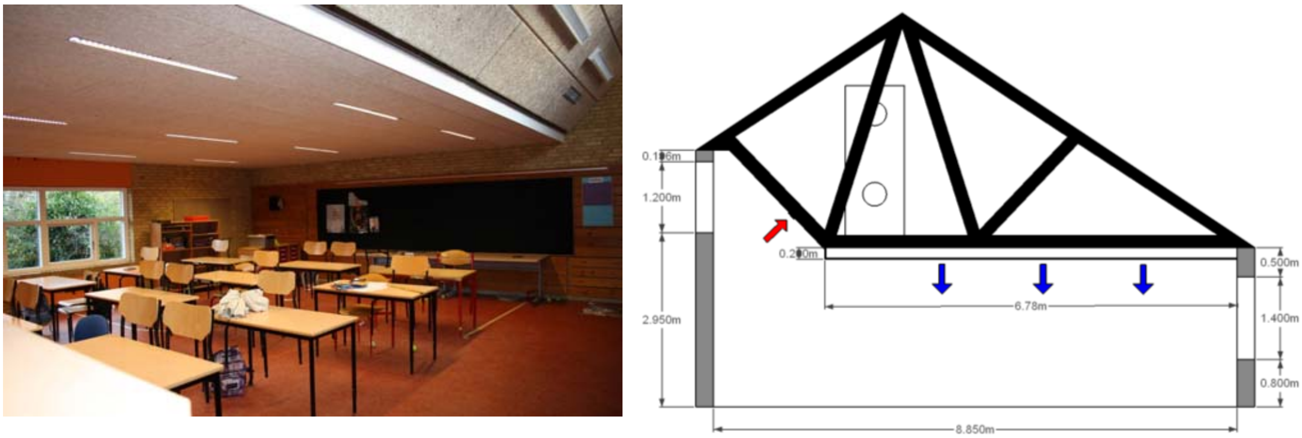

Figure 1: Left: Picture of classroom where measurements were performed, Right: Schematic sketch of classroom with dimensions and principle of air distribution (blue arrows inlet, red arrows exhaust).

The floor area was 9.15x8.85 m, see Figure 3 and the windows on the main facade is oriented North-West, the solar gain during the occupied hours was therefore limited and only fabric curtains were installed as shading device. The rooms were during the measurements occupied by a 6th grade (age 11-12) and there were 24 pupils in each class.

2.2. Diffuse ceiling layout

The ventilation ceiling consists of a metal frame suspension system to create the plenum on which the cement bonded wood wool panels were mounted. This is a commonly used product in Danish classrooms for its acoustic properties. For use as a diffuse ventilation inlet, two types of panels are needed - active and passive. The active panels are permeable so the supply air can penetrate and passive panels has 20 mm of hard painted mineral wool glued to the backside making them non-permeable, see Figure 2. The mineral wool improves the acoustic properties and permits control of the supply air distribution in the room. A small overpressure is created in plenum above the panels that ensures the supply air is equally distributed through the active panels. 6 active panels were used and placed in the room as shown in Figure 3, equalling an inlet area of 8.6 m^2 or 11 % of horizontal ceiling surface. The exact opening area was difficult to determine due to the random and unique configuration of every single panel and is therefore not known. The panels were not uniformly placed as intended, but this was discovered after the measurements were performed.

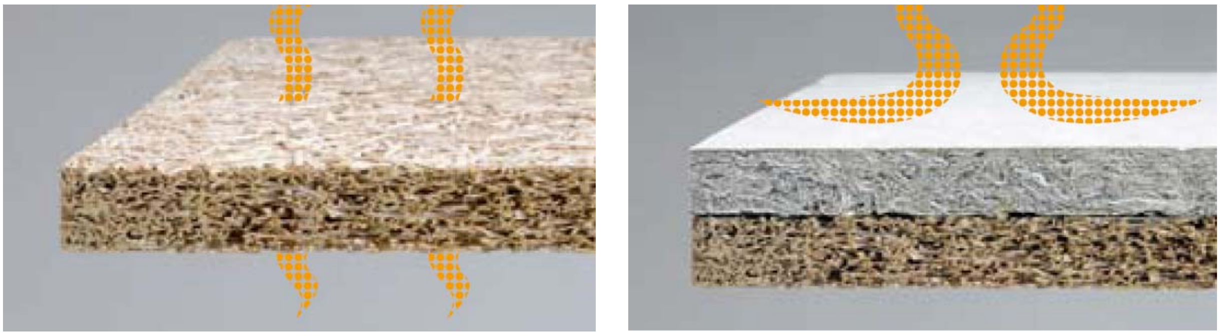

Figure 2. Left: Active plate of cement bonded wood-wool, Right: Passive plate has a layer of impenetrable mineral wool. Source: www.troldtekt.dk

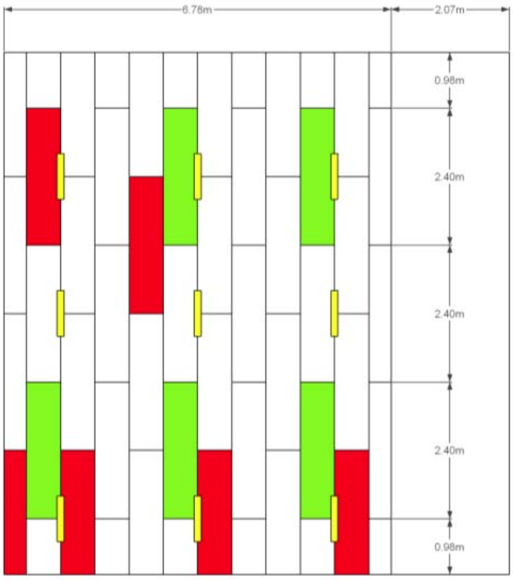

Figure 3: Placements of active cement bonded wood wool panels (green) and passive panels (white), green indicate the intended placement of the active panels.

The fresh air inlet into the plenum was placed slightly off-centre of the suspended ceiling, blowing vertically down against a passive panels thereby distributing the air uniformly in the pressure chamber. The extract air was removed through the existing diffusers located on the sloped part of the ceiling, see Figure 1. The supply and extract flow rate was balanced and controlled by dampers. The lighting fixtures were integrated in the ceiling for design reasons and to avoid vandalism, see Figure 1, and this caused leaks in the ceiling in addition to the active supply panels.

3. Measurements

Six scenarios of measurements were performed at airflow rates of 500 m^3/h and 1000 m^3/h and inlet temperatures of 10, 17 and 21 °C as stated in Table 1. The airflows correspond approximately to indoor environment category 3 and 1 in the European Standard EN15251 (EN 15251, 2007). The measurements with an inlet temperature of 10 and 17 °C was performed in the winter with an outside temperature of 2-5 °C and the measurements with an inlet temperature was performed in the summer with an outside temperature of 17 °C.

During the measurements 21 out of 24 pupils were present and in the winter one radiator below the windows where on and had a temperature of 55 °C corresponding to a heat load of approximately 1100 W. The only other heat load in the room was the lighting system with 9 fixtures of 2x18 W.

Table1: Overview of performed measurements

	Supply air temperature	**Supply flow**	**Air change**	**Air flow per pupil**	**Measurements**
Scenario 1	10 °C	500 m^3/h	2.45 h^{-1}	5.8 l/s	Velocity, temperature
Scenario 2	10 °C	1000 m^3/h	4.90 h^{-1}	11.6 l/s	Velocity, temperature
Scenario 3	17 °C	500 m^3/h	2.45 h^{-1}	5.8 l/s	Vel., temp., air change efficiency
Scenario 4	17 °C	1000 m^3/h	4.90 h^{-1}	11.6 l/s	Vel., temp., air change efficiency
Scenario 5	21 °C	500 m^3/h	2.45 h^{-1}	5.8 l/s	Velocity, temperature
Scenario 6	21 °C	1000 m^3/h	4.90 h^{-1}	11.6 l/s	Velocity, temperature

3.1. Air velocity and -temperature measurements

In common supply systems utilizing mixing ventilation, the mixing is achieved by inducing air at relatively high velocity above the occupant zone. The momentum of the air jet and buoyancy forces causes the supply air to be mixed with the contaminated air in the occupant zone. Efficient mixing is therefore a result of turbulent flow patterns, but also an indication of potential risk of discomfort by draught. In diffuse ceiling ventilation the supply air is induced into the room with relatively low air velocities, limiting the risk of draught caused by the supply air inlet. Mixing is achieved by movement, and buoyancy forces from occupants and other heat sources in the room. However can the thermal plumes obstruct the supply air and force the supply air to areas with no heat sources, leading to cold downdraft and risk of draught in those areas. Therefore velocity and air temperature measurements were carried out with two Brüel og kjær Indoor Climate Analyzer model 1213, to examine the risk of discomfort due to air movements in the room. The accuracy of the temperature sensor is ±0.2 °C and the velocity sensor has an accuracy of ±0.1 m/s. The measurements were performed in 8 points uniformly distributed and placed close to tables in the occupied zone of the room, marked with triangles in Figure 3.

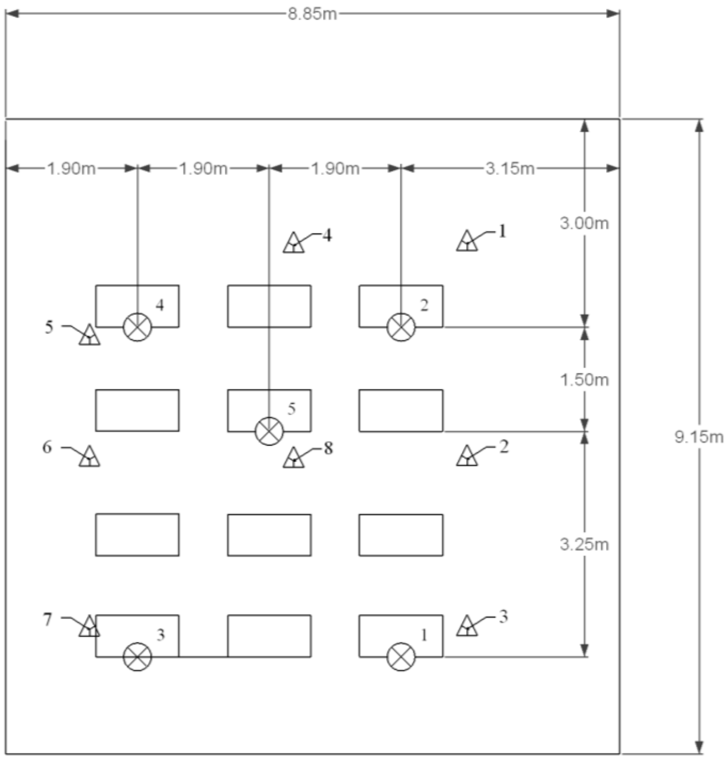

Figure 4: Location of measuring points in classrooms, triangles represent air temperature and –velocity measuring points and circles represent tracer gas measuring points.

At each point the air velocity and –temperature was measured at heights of 0.1 m and 1.1 m corresponding to ankle and neck level of a sitting person as specified in CR 1752 (1998) and each measurement was averaged over 3 minutes. According to CR 1752 (1998) maximum air velocities of 0.12, 0.15 and 0.18 m/s are allowed for category A, B and C, respectively. Category B is used as reference and therefore the air temperature must be between 20-24 °C for the winter period and 23-26 °C in the summer period which was representative for the conditions under which the measurements were performed.

From the measurements the percentage of people dissatisfied (PPD) due to draught can be determined by calculating the Draught rate as stated in CR 1752 (1998). The draught rate is an experimental determined and depends both on air velocity including fluctuation expressed as turbulence intensity and the air temperature However, at low air velocities the temperature becomes the more dominant term and the equation is only valid for air velocities above 0.05 m/s. The measured air velocities were expected to be around that limit and it is therefore more relevant to examine the raw air velocities.

3.2. Tracer gas

The concentration-decay method was used to determine the mean age of air. The equipment used was a photo-acoustic gas monitor model 1312 from Innova and a multipoint sampler and dozer from Brüel & Kjær model 1303. The gas used was Freon R-134a that has a density of 4.25 kg/m^3, about 3.5 times higher than air. The multipoint sampler had 6 tube connections that were divided with five in the room and one in the extract duct. In the room the 5 tubes were placed in the occupied zone with one tube at each corner table and at a table in the middle, all at a height of 1.1 m similar to the height of a sitting person. The points are marked with circles in Figure 4 and were chosen to give a representative picture of the mixing efficiency in the occupied zone and identify any stagnant zones. In the experiments the tracer gas was injected into the room until a constant concentration was reached and two swivel fans ensured fully mixing in the room. A constant concentration of approximately 5 and 10 ppm was achieved at respectively the low and high flow rate. Hereafter the swivel fans were turned off and the following decay at the 6 sampling points was recorded. From this the local mean age of air in each point and the air change efficiency in the room were determined. Any differences in local mean age of air between the sampling points can help identify potential problematic zones.

3.3. Air change efficiency and local air change index

The air change efficiency is a useful measure to check the air distribution in a room and identify any problematic zones. The definition of air change efficiency was first introduced by Sandberg et al (1983), and is the ratio of the shortest possible air change time in the room (nominal time τ_n) and the average time it takes to replace the air at a point (actual air change time $\bar{\tau}_r$).

$$\epsilon^a = \frac{\tau_n}{\bar{\tau}_r} \text{ x } 100\%$$

For a more thorough in depth description of the concept see Sandberg et al (1983). If the mixing of the supply air in a room is optimal the actual air change is twice the shortest possible time. Therefore complete mixing has a maximum air change efficiency of 50 %, lower efficiencies indicate incomplete mixing or short circuited flow and higher efficiencies indicate transition into displacement ventilation. Local air change index represents the ratio of mean age of air in the exhaust and local mean age of air at the point of interest. At full mixing the age of air in the exhaust is the same as the age of air in the room and the ratio is equal 100 %.

3.4. Pressure drop – flow distribution

The pressure drop across the ceiling was measured at three different positions at different flow rates to establish a characteristic between flow rate and pressure drop. The pressure drop was determined with a Retrotec DM-2 manometer measuring for one minute and recording the average. The pressure loss across a single panel was provided by the producer of the panels and the characteristic is shown in Figure 5.

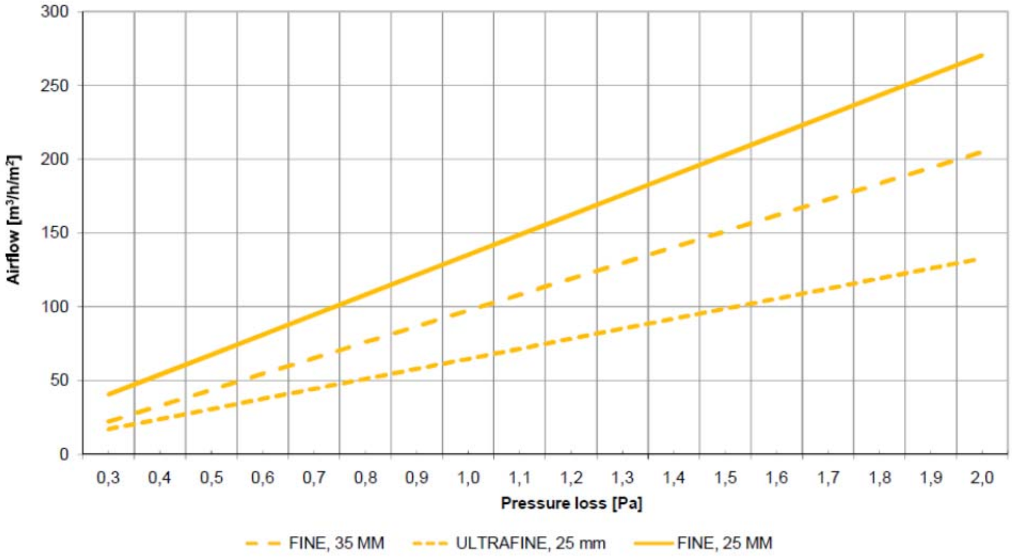

Figure 5: Correlation between flow rate and pressure loss of cement bonded wood wool panels, for different wood wool fibres sizes and plates thicknesses.

Fine and ultrafine refers to the size of the wood wool fibres which affects the permeability of the plates and at Vallensbæk school was used ultrafine 25 mm thick panels. Based on the pressure drop characteristic of a single panel and the measured pressure drop across the installed ceiling, the distribution of supply air between cracks and the active panels can be determined. This is important to know in order to predict and control the distribution of the supply air and investigate whether the concept of active and passive panels is applicable in practice.

3.5. Smoke

The tracer gas, air velocity and air temperature measurements focused on the occupied zone. To identify any potential problematic areas outside occupied zone or areas in the occupied zone not covered in the experiments a smoke visualization was performed. The smoke makes it possible to see the air propagation in room, determining any downdrafts and reveals whether the supply air enters the room through the active panels or through cracks and leaks. The smoke was injected into the supply duct before the plenum, so it was fully mixed before it entered the plenum and had a temperature of 18 °C slightly higher than the actual supply air of 17.1 °C. The smoke was generated by a smoke generator using a glycol mixture.

3.6. Infrared camera - thermovision

One of the great advantages of the diffuse ventilation inlet concept is the possibility to supply very cold air without causing draught. The reason for this is besides the low air velocities that the supply air gets heated when penetrating the ceiling panels, this also makes the ceiling act as a radiant cooling system (Hviid et al, 2013). To determine the ceiling surface temperature infrared camera pictures were taken. The pictures were also, as the smoke visualization, performed to investigate whether part the supply air entered the room through cracks and leaks instead of the active panels and to locate unintended leaks in the ceiling and room that could have affected the air velocity and -temperature measurements. The infrared pictures were taken with an inlet temperature of 10 °C to give a larger temperature difference and thereby clearer images.

3.7. Performance tests and perceived air quality

In the performance study the pupils occupying the two rooms were exposed to different air supply rates. The experiments were a so called crossover design where the two rooms were exposed to different air supply rates in the same week, and the conditions were then switched the following week. The two conditions were an air supply rate of 500 m^3/h corresponding to category 3 in EN15251 (2007) of 5 l/s per person and 0,35 l/s per m^2 and an air supply of 0 m^3/h corresponding to the conditions before mechanical ventilation was installed. The experiments was carried out in the first two weeks of September and the outdoor temperature during the day was around 20 °C and under both conditions the teachers and pupils were allowed to open the windows and doors as usual. The tests were given in the classes as part of their normal schedule to avoid changes in their routines and teaching environment. To give the pupils time to get acclimated to the specific condition that week the test were given on Thursday and Friday. The pupils were therefore not aware that the tests were part of an experiment and both the teachers and pupils were uninformed of the changes in air supply rate.

The tests were a math test where the pupils had to subtract two four digit numbers and a reading and comprehension text where the pupils out of three had to choose the correct word. All three words fitted in the context of the sentence but only one made sense in the context of the whole text. For a more thorough description of the test and the execution of the experiments see Wargocki et al (2007), where the tests and methodology used in this paper also were used.

To make comparisons between the interventions the results were normalized to answers per minute (speed) and incorrect answers per attempted (errors). Only data from pupils that had taken both tests was used in analysis. The results were adjusted for learning and increased familiarity with the test from week 1 to week 2 by multiplying each student´s week 2 results by the ratio between the class average week 1 and week 2. To determine whether there were a statistical significance ($P<0.05$) between the two conditions the Wilcoxon matched pairs test were used after determining that most of the data were not normally distributed. The pupils filled out a visual analogue scale questionnaire each Friday after they had taken the last test to indicate the perceived air quality and intensity of various sick building syndrome (SBS) symptoms. The questionnaire included 6 parameters regarding the indoor environment in the classroom; temperature, air movement, air dryness, air freshness, illumination and noise, and 10 questions concerning SBS symptoms and their ability and motivation to perform school work; nose congestion, throat, lip, skin dryness, hunger, sleep at night, fatigue, enough sleep, motivation and headache, see Figure 6.

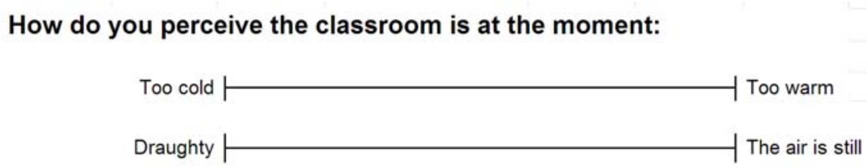

Figure 6: Examples of visual analogue scale from perceived air quality and SBS questionnaire.

The questionnaire was explained to the pupils by the investigators but handed out by the teachers. As for the tests only data from pupils who had filled out the questionnaire both weeks were used in the analysis. The CO_2-concentration in the rooms was continuously monitored over the two weeks with a VAISALA GM20 CO_2-transmitter connected an Onset HOBO U12-012 data logger.

4. Results

4.1. Air velocities

In Figure 7, 8 and 9 are shown the measurement results for a supply temperature of 10 °C, 17 °C and 21 °C, respectively. In the figures the average air velocity over the sampling period of three minutes is shown as columns with error bars displaying the standard deviation. The standard deviation shows negative air velocities has been recorded which are physically invalid, especially at low mean air velocities. The standard deviation assumes a Gaussian data distribution but this can be disturbed by relatively high outliers in the sampled data. These outliers are probably generated from turbulent flows caused by seated, yet active pupils during the measurements. The comfort threshold of 0.15 m/s is fulfilled except at point 3 and 7 with inlet temperatures of 10 and 17 °C. At point 7 the air velocity increases with increasing flow rate and only exceed the threshold at 1000 m^3/h, indicating potential draught issues at high flow rates. Though, at point 3 the air velocity was independent of the flow rate and surpasses the threshold at both flow rates.

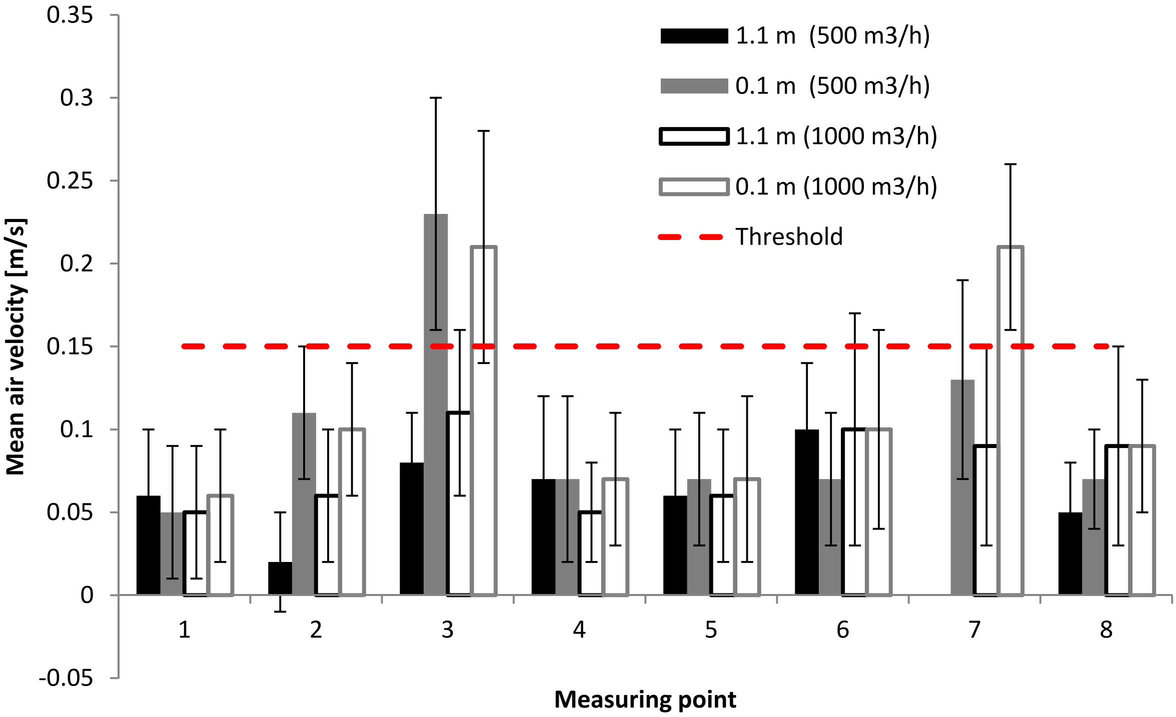

Figure 7: Measured air velocities at 0.1 and 1.1 m above the floor and flow rates of 500 and 1000 m^3/h with a supply temperature of 10 °C.

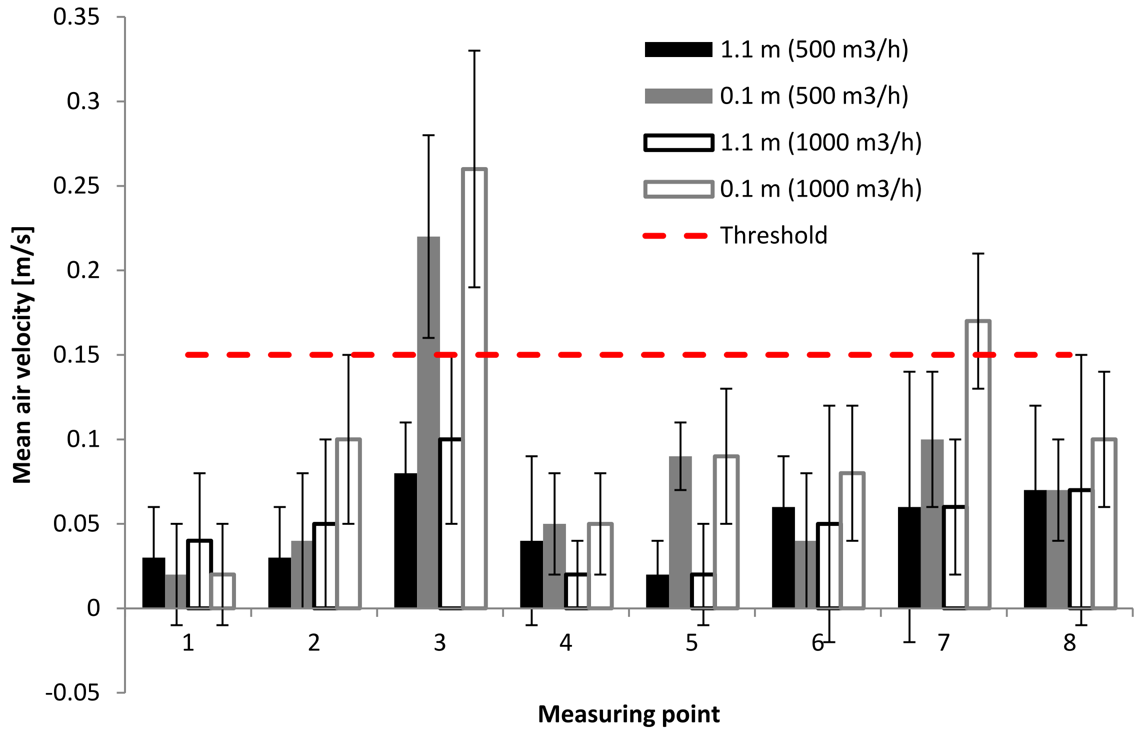

Figure 8: Measured air velocities at 0.1 and 1.1 m above the floor and flow rates of 500 and 1000 m^3/h with a supply temperature of 17 °C.

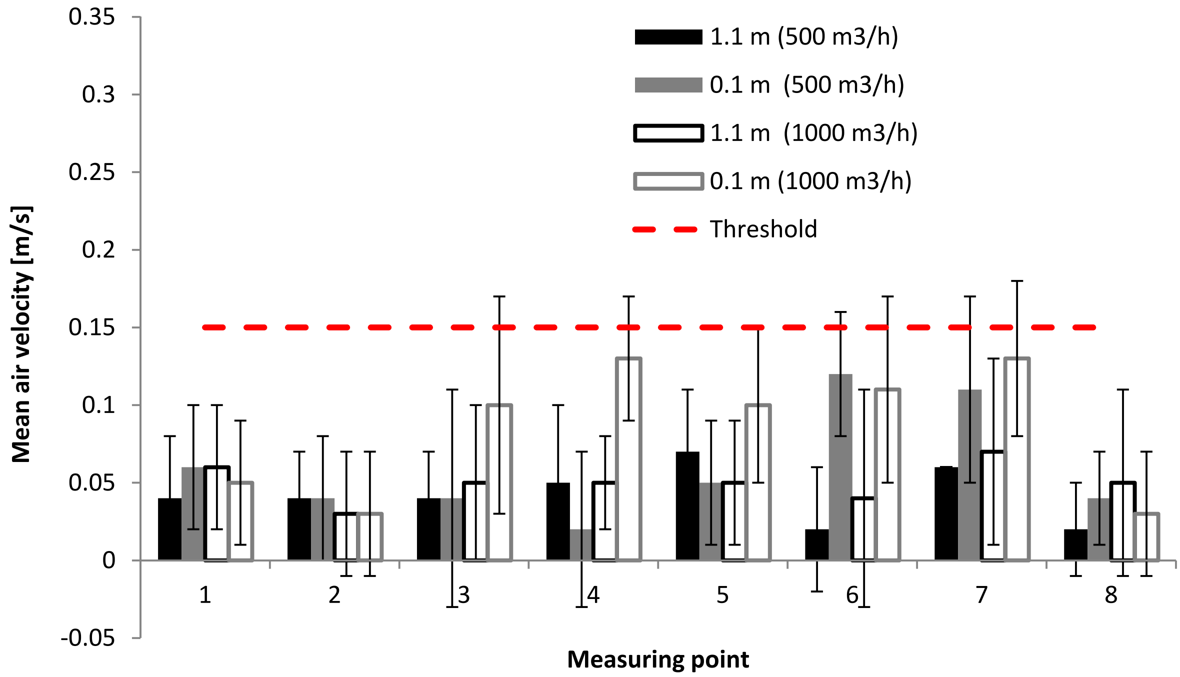

Figure 9: Measured air velocities at 0.1 and 1.1 m above the floor and flow rates of 500 and 1000 m^3/h with a supply temperature of 21 °C.

4.2. Air temperature

The measured air temperatures are shown in Figure 10, 11 and 12 for an inlet temperature of respectively 10, 17 and 21 °C. The results show that the air temperatures were quite uniform across the room, maximum 1.0 °C difference between the points when comparing the respective heights and flow rates. The results show clear temperature stratification of 0.5-1.0 °C between ankle and neck height at both flow rates and the three inlet temperatures. With an inlet temperature of 10 °C the measured air temperatures were below 20 °C in all the measuring points at the high flow rate, see Figure 10. At the low flow rate the temperature were around 0.5-1.0 °C higher except for point 8 where the temperature didn't change. In Figure 11 with an inlet temperature of 17 °C the room air temperature contrary increases at the high flow rate and the temperatures were above 20 °C at all times except for one measurement at point 1. In figure 12 the temperatures were between 21-22.5 °C with slightly lower temperatures at the high flow rate.

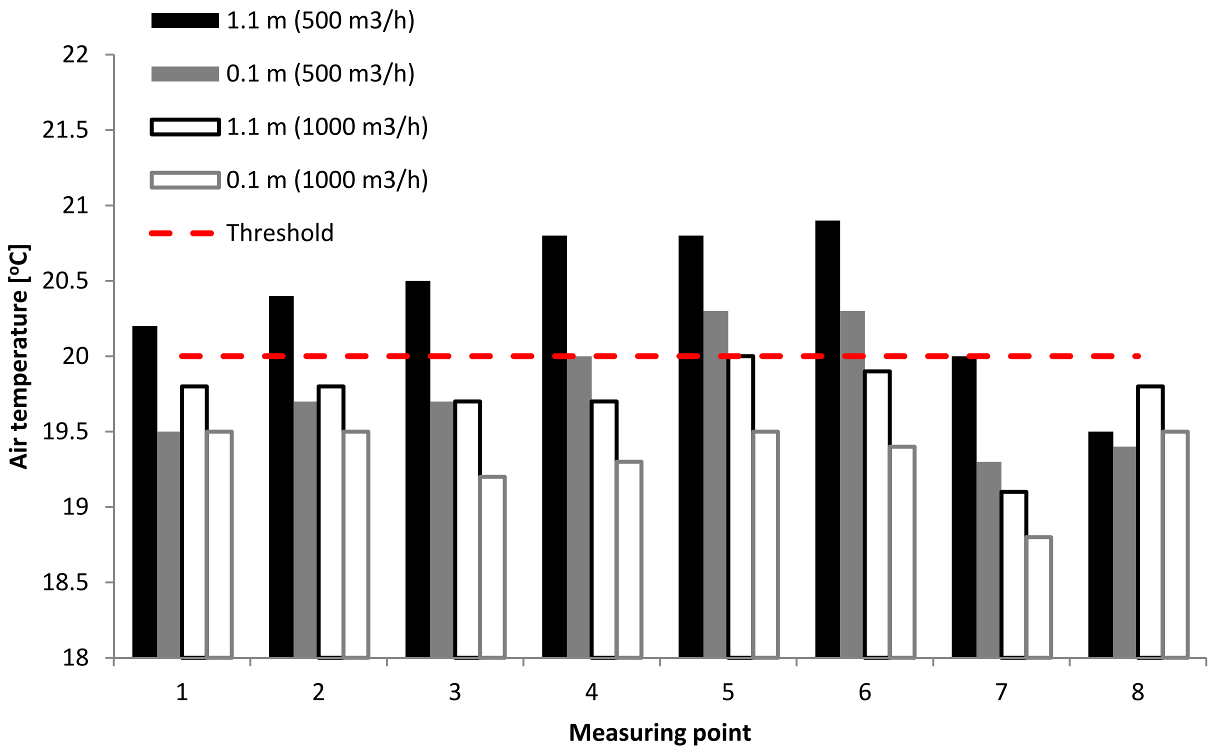

Figure 10: Measured air temperatures at 0.1 and 1.1 m above floor and flow rates of 500 and 1000 m^3/h with a supply temperature of 10 °C.

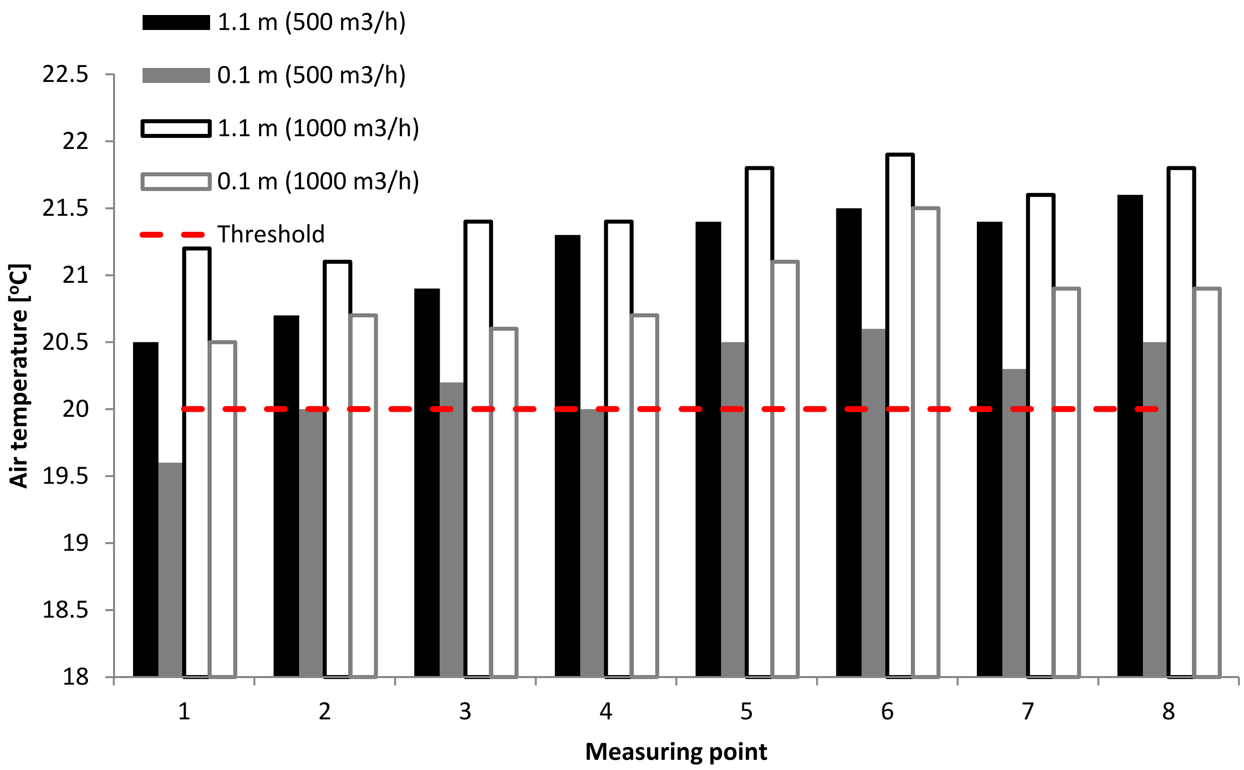

Figure 11: Measured air temperatures at 0.1 and 1.1 m above the floor and flow rates of 500 and 1000 m^3/h with a supply temperature of 17 °C.

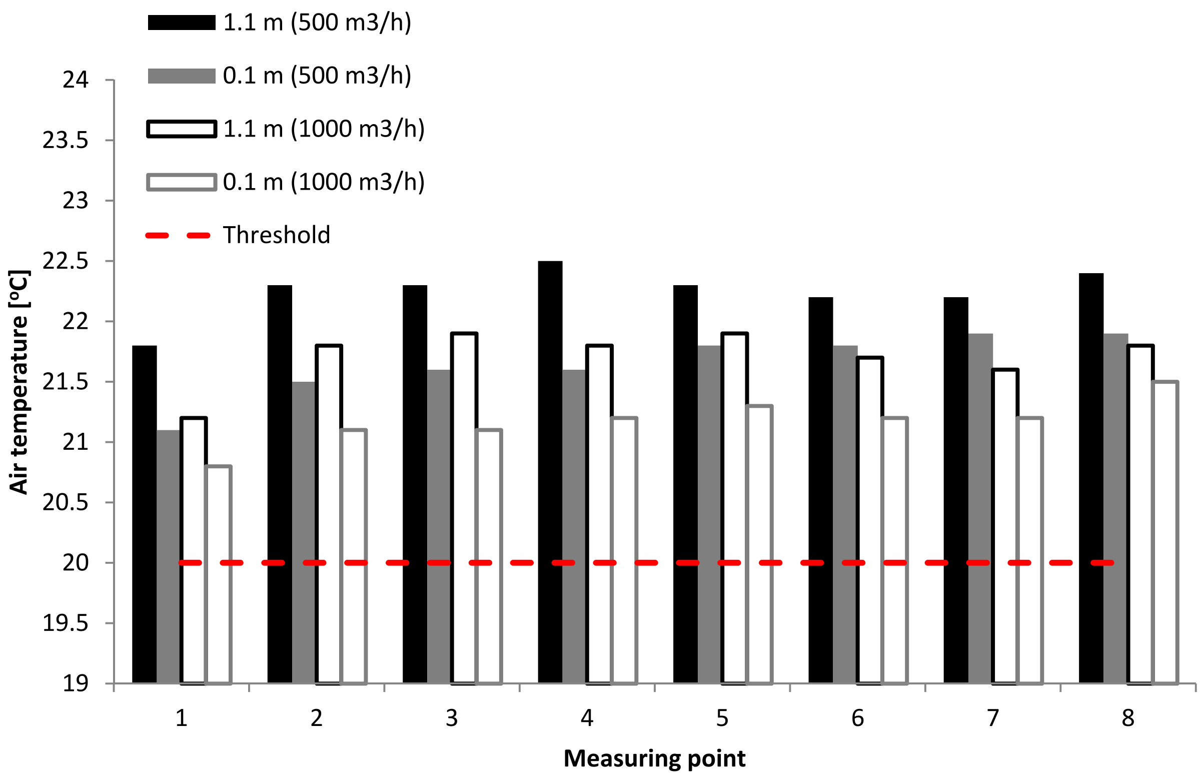

Figure 12: Measured air temperatures at 0.1 and 1.1 m above floor and flow rates of 500 and 1000 m^3/h with a supply temperature of 21 °C.

4.3. Air change efficiency

In Figure 13 (left) is the local air change index for the 5 sampling points in the classroom shown. The values were for all the points around 100 % at both flow rates indicating equal air quality at all the sampling points. The air change efficiency on room level shown in Figure 13 (right) is 49.5 and 53.0 % at the low and high flow rate, respectively, almost equal to perfect mixing, but however showing transition into displacement ventilation at the high flow rate.

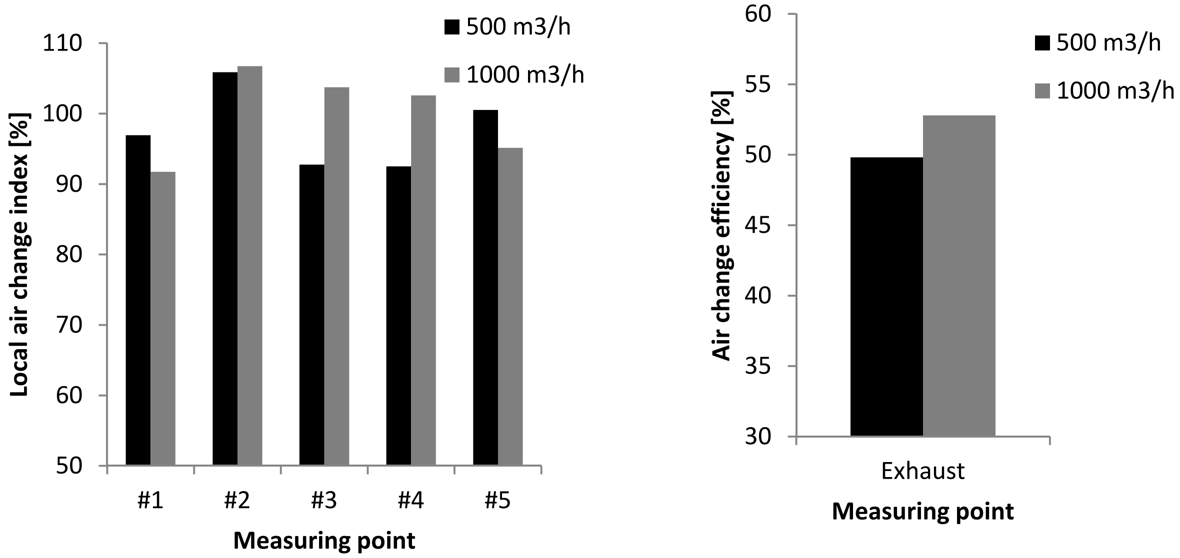

Figure 13: Local air change index at sampling points (left) and air change efficiency (right)

4.4. Pressure drop characteristic– flow distribution

The measured pressure drop across the ceiling at different flow rates are shown in Figure 14 together with the pressure drop characteristics of a single panel. The measured results had an almost linear characteristic that indicate that the airflow through the ceiling is laminar. The pressure drop across the panels varies from 0.48 to 0.98 Pa in the measured air flow range of 500-1000 m^3/h. The flow rate through the mounted ceiling was higher than for single panels at the same pressure loss, between 42-46 % higher. This indicates that not all the supply air enters the room through the active ceiling panels, but through unintentional cracks in the ceiling.

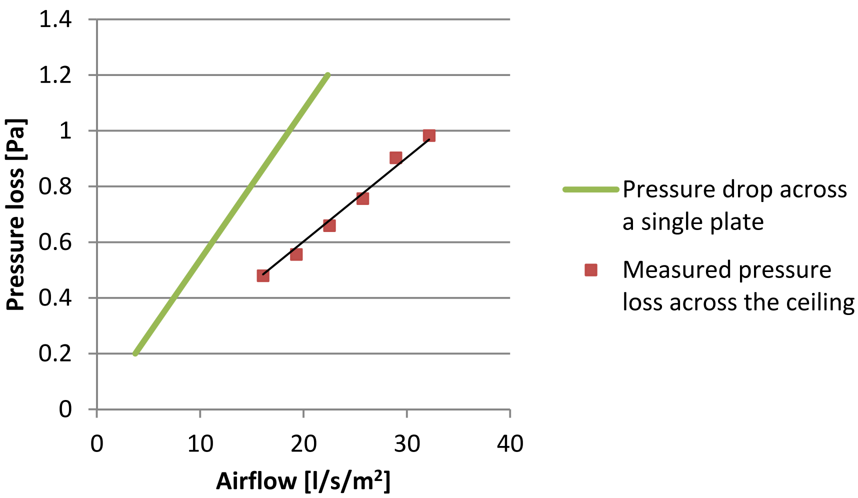

Figure 14: Measured pressure loss characteristics over the installed and a single active panel

4.5. Smoke visualization + Infrared pictures

The smoke visualisation revealed that the supply air not only entered the room through the passive plates but also through the lighting fixtures as shown in Figure 15.

Figure 15: Picture of smoke visualisation

This correlates well with the results from the pressure drop characteristics. The visualisation didn't show any noticeable leaks in the joint between the ceiling panels and the walls, but there were some noticeable air flow streams through the joint between the panels. (The exact distribution between the joints and lighting fixtures was not quantified).

The infrared pictures in Figure 16 clearly show that the active panels have a lower surface temperature than the passive panels. The surface temperature of the active panels is around 14 °C with a supply air temperature of 10 °C, while the passive panels' temperature was around 22 °C. The temperature was however lower in the joints between the panels. The images showed that the active panels weren't placed as prescribed/intended/described, see Figure 16 (left) and that a hatch in the ceiling to attic was leaky and had a temperature of 7 °C. The vertical inlet to the distribution plenum also creates an area on the ceiling with a temperature of 12 °C, indicating that the painted mineral wool on the back of the passive panels is not completely air tight.

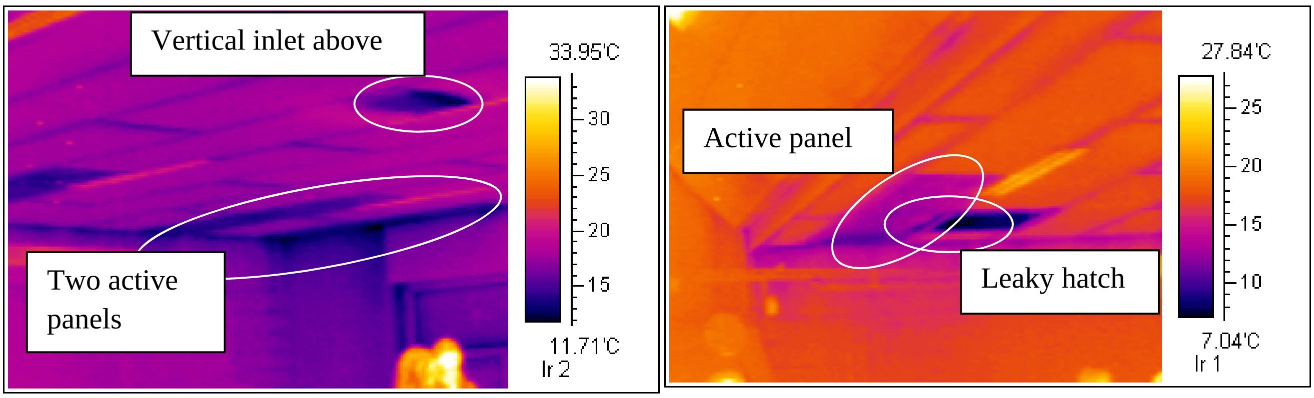

Figure 16: Infrared pictures of ceiling surface temperature in two corners of the classroom revealing placement of two active next to each other (left) and leaky ceiling hatch (right).

4.6. Performance and perceived air quality

In Table 2 are shown the results of the statistical analysis of the performance test. For subtraction the increased the ventilation rate did not significantly increase the speed of which the solved the calculations. The number of errors however decreased significantly (51 % fewer errors). By combining the speed and errors into number of correct answers the data also showed a significant improvement correspond to a 9 % increase in correct answers. The reading and comprehension test did not show an improvement due to increased ventilation rate, on contrary the speed decreased significantly and the number of errors also increased. The CO_2-concentration during the test was 600-750 ppm with ventilation and 1200-1500 without ventilation though 2000 ppm in room 42 during the reading and comprehension test in week 1.

Table 2: Statistical analysis of subtraction and reading and comprehension tests in regards to speed, errors committed and correct answers.

	Ventilation	Number of pupils	Mean (speed)	SD	Mean (errors)	SD	Mean (correct)	SD
Subtraction	Yes	36	2.857	0.98	0.111	0.126	2.578	0.97
	No	36	2.781	0.94	0.168	0.142	2.360	1.00
Wilcoxon p-value	P<0.05		0.278		0.001		0.013	
Reading and comprehension	Yes	38	1.434	0.725	0.160	0.129	-	-
	No	38	1.592	0.480	0.150	0.139	-	-
Wilcoxon p-value			0.001		0.492			

In Figure 17 is listed the results from the perceived air quality questionnaires. Only data for the parameters most likely to be influenced by the supply concept are presented, but there was found no significant improvements or decreases for air dryness, illumination, nose congestion, throat, lip, skin dryness, hunger, sleep at night and enough sleep.

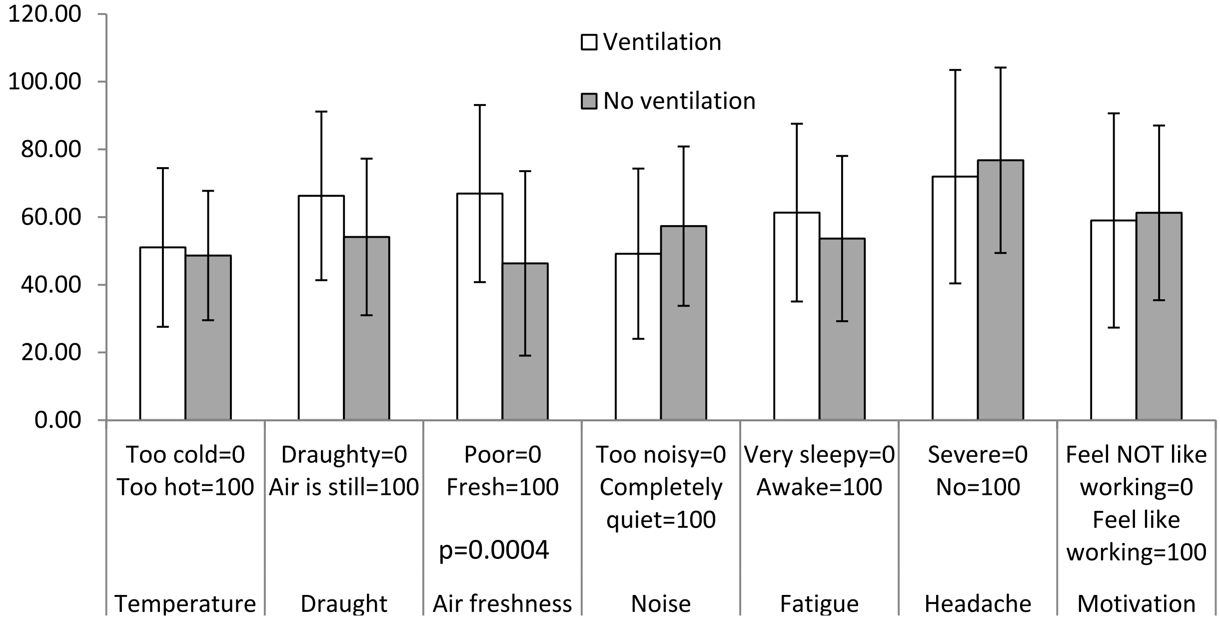

Figure 17: Parts of the results of the perceived air quality and SBS symptoms questionnaire.

The pupils perceived the air freshness significantly better while there was no significant difference for temperature, draught or noise. The improved air quality however did not result in any significant improvements in the SBS symptoms, but overall the pupils perceived the indoor environment acceptable and did not suffer from any SBS symptoms.

5. Discussion

The investigation of a diffuse ventilation inlet under real conditions in a classroom gave results that correlate well with previous investigations from experimental setups. The tracer gas measurements in Figure 13 showed efficient mixing at both the high and low flow rate correlating with the results in Hviid et al (2013) and Fan et al (2013), and this was achieved without causing draught, even with an inlet temperature of 10 °C. The measured air velocities in the occupant zone (Figure 7 and 8) generally fulfil the comfort threshold of 0.15 m/s with the exception of point 3 and 7. The infrared images revealed that the active panels was placed incorrectly, and the erroneous placement of two active panels near point 3 is probably the reason the air velocity increases above the threshold at the high flow rate. At point 7 the leaky ceiling hatch is probably the cause of the high velocities as the measured air velocities are independent of the flow rate, indicating that high air velocities are not caused by the supply air. This is further substantiated by the summer measurement in Figure 9 where the velocities at point 3 and 7 were below 0.15 m/s, due to reduced down draught because of the higher temperature in the attic.

The results in Figure 11 for an inlet temperature of 17 °C showed that the room air temperature increased at high flow rate, this is not intuitive as the inlet temperature is lower than the room air and should lead to a lower room temperature. A probable explanation for this is that the measurements with a high flow rate were performed after the measurements at low flow rate, where people and other internal heat loads had increased the room air temperature, despite having a fixed inlet temperature of 17 °C. With an inlet temperature of 10 °C (Figure 10) the room air temperature was below 20 °C and decreased at the high flow rate indicating that the heating system and internal gains was not able to compensate for the ventilation heat loss. Temperatures below 20 °C is not ideal from a comfort perspective, but besides the low inlet temperature it was partly due to one of the radiators not functioning and the heating system was therefore unable to maintain the desired minimum temperature. In practice inlet temperatures of 10 °C are rarely used, and if used, the heating system should be able to compensate. Furthermore were the measurements performed to examine the risk of cold down draught under real conditions, and here the concept performed satisfactorily and in correlation with reported results from experimental setups (Hviid et al, 2013, Fan et al, 2013; Nielsen et al, 2009). Overall the air velocities are of the same magnitude independent of the inlet temperature and there is limited thermal stratification. This indicates that the air distribution and mixing occurs despite limited buoyancy effect and most likely is caused by the movement and thermal plumes of the occupants in the room.

The local air change indexes calculated from the traces gas measurements were around 100 % at all the measuring points and the two examined flow rates, see Figure 13 (left). This shows that the diffuse ventilation inlet is able to distribute the supply uniformly in the room, with no stagnant zones or short circuits, contrary to the CFD simulations in Hviid (2010). The air change efficiency shown in Figure 13 (right) supports the conclusion on room level. However, at the high flow rate there is an indication the flow regimes transitioning into displacement ventilation, which increases the risk of downdrafts. This issue can be disclosed from the velocity measurements in Figure 7, 8 and 9, but is something to consider at even higher flow rates.

The surface temperature of the active panels was around 14 °C (Figure 16) even though inlet air temperature was 10 °C, and this shows that there is a considerable preheating of the air through the panels. This effect is probably due to radiation exchange between the active panels with the other surfaces in the room. The surfaces temperature of the passive panels was around 22 °C, the same as the other surfaces in the room. The panel joints were also clearly visible in the infrared pictures and had a temperature of 14-15 °C indicating that supply air is penetrating through the joints as well, but it could also be due to the heat transmission of the metal suspension system. This aspect cannot be determined from the measurements, but overall the ceiling had a surface temperature equal to the room air temperature and discomfort due to radiant asymmetry is therefore deemed negligible.

The pressure loss measured across the ceiling and a single active panel showed that the supply air not only penetrates through the actives plates but also through leaks in the ceiling, at an approximately 55/45 ratio. This ratio correlate well with the flow distribution between plates and leaks in Hviid et al (2013), however showed the smoke visualization that a large part of the supply air penetrates through the lighting fixtures. The exact share was not determined nor was the placement or share of other leaks e.g. panel joints, but the leaks ruin the concept of active and passive panels. The leaks caused by the lighting fixtures probably outweighed the incorrect placement of the active panels and misleadingly ensured uniform distribution in the room. There is however reason to believe that the concept of active and passive could work to a large extent, but in order to function optimally focus must be on limiting all unintended leaks in the ceiling.

The measured pressure loss across the ceiling varied from 0.5-1.5 Pa depending on the flow rate and matches the results presented in Hviid et al (2013) and Fan et al (2013). This is considerably lower than conventional inlet diffusers that induce pressure losses of 30 Pa minimum in order to ensure efficient mixing and avoid draught and this will reduce the fan power required to operate the ventilation system. Furthermore saves the plenum in diffuse ventilation inlet distribution ducts, diffusers, labour cost and installation time, as well as space that can be a constraint particularly in renovation cases.

The results of the performance tests gave mixed results, see Table 2. The subtraction test showed a significant improvement in the number of errors per answer while the speed was not affected and this combined lead to an increase of 9 % in number of correct answers. This correlates will with previous findings, most recently in Bakó-Biró et al (2011). The reading and comprehension test did not give the expected results as the performance decreased with ventilation. This is not logic intuitively and doesn’t match previous reported results in the literature (Myrhvold et al, 1996; de Gids, 2006; Wargocki et al, 2007; Shaughnessy et al, 2007; Bakó-Biró et al 2011). Analysing the data from week 1 vs week 2 independent of flow rate showed that the performance in terms of speed decreased by 35 % (p=0.0002) and the number of errors increased by 69 % (p=0.003). This strongly indicated that the pupils found the reading and comprehension test in week 2 more difficult despite the two texts had the same lix-number. The Lix number is a Danish readability index determined by the average number of words per sentence and the percentage of words with seven letters or more. A talk with the pupils and teacher later confirmed that they perceived the second text more difficult and this skewed the ruining the possibility to determine any effect of improved ventilation. The results from the perceived air quality questionnaire in Figure 17, showed that the pupils found the air quality significantly better with ventilation. This was also expected as the CO_2-concentration was twice as high and above the recommended level of 1000 ppm without ventilation. The pupils did not perceive any difference in terms draught and noise, showing they were not affected by air movements in the room or disturbed noise generated by the ventilation system. The improved air quality did not result in significant effects on the health and wellbeing of the pupils as found in other studies (Smedje et al, 2000).

This is most likely to SBS symptoms being a result of long terms effects any changes cannot be detected after one week.

6. Conclusion

The objective of the work in this paper was to validate the performance of diffuse ceiling ventilation in practise and document its applicability in school classrooms. Based on the investigation the following can be concluded on diffuse ceiling ventilation.

- The experimental results are in good agreement with previous results from test facilities and simulations.
- Virtually no risk of draught at both low and high flow rates.
- Uniform airflow field with little difference between ankle and head height.
- Uniform temperature distribution with little difference between ankle and head height at both high and low flow rates.
- Negligible radiant asymmetry.
- Perfect air mixing throughout the room independent of airflow rate.
- Enables high flow rates and low supply air temperature without causing discomfort by draught.
- The pupils performance improved 9 % (subtraction) with ventilation and they perceived the air significantly fresher without any discomfort

Overall the results are positive and no negative aspects were detected. It is therefore concluded that diffuse ceiling ventilation in the studied form is applicable for use in school classrooms.

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Paper IV

Performance and Life-Cycle Cost of Low-Pressure Mechanical Ventilation Concept for Renovation of School Classrooms – A Case Study

S. Terkildsen, S. Svendsen

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Performance and Life-Cycle Cost of Low-Pressure Mechanical Ventilation Concept for Renovation of School Classrooms – A Case Study

Søren Terkildsen[1] and Svend Svendsen
Department of Civil Engineering, Section of Building Physics and Services
Technical University of Denmark, Brovej, Building 118, DK-2800, Kgs. Lyngby
[1] Corresponding author. Tel: +45 45251927. E-mail: sterk@byg.dtu.dk

Abstract

Many Danish schools suffer from poor indoor environment because they have out-dated mechanical ventilation systems or rely on opening of windows to provide fresh air. This adversely affects the performance of the pupils and causes high energy consumption for fans and ventilation heat loss. This paper describes a low-pressure mechanical ventilation concept to reduce energy consumption and improve the indoor environment, developed for the renovation of school classrooms. A pilot system was installed in two classrooms and the performance was studied through computer simulations and measurements that comprised continuous monitoring of CO_2-concentrations, temperatures and fan power consumption. Furthermore, the life-cycle cost was estimated in terms of net present value and the cost of conserved energy. The ventilation system consists of a high efficiency heat exchanger, an oversized air handling unit and duct system, a diffuse ventilation inlet, and a new type of flow-control damper for demand control. The measurements showed a heat recovery efficiency of 84%, CO_2 concentrations below 1000 ppm and an SFP-value of 627 J/m^3 at the design flow-rate. This results in an annual fan power consumption of 5.79 kWh/m^2, which makes it applicable for implementation in future energy renovation projects for school classrooms.

Keywords

Low pressure, mechanical ventilation, diffuse ceiling, life-cycle cost, energy renovation

1. Introduction

Many primary schools in Denmark have poor mechanical ventilation systems or simply rely on opening of windows to provide fresh air. The majority of schools are from the 1960s-70s, so the mechanical systems are now outdated and the rooms are not designed to utilize natural ventilation efficiently. This results in high energy consumption for fans and/or ventilation heat loss along with poor indoor environment because in many cases the ventilation rate is insufficient. A recent study with data from 750 Danish classrooms showed that 58% of the schools in the Copenhagen region and 56% at the national level had CO_2-concentrations above 1000 ppm [1], the maximum level recommended by the Danish Working Environment Authority [2]. Similar results have been reported for Dutch schools. Moreover, the poor indoor environment adversely affects the performance and well-being of the pupils [3]. Another key priority is to reduce energy consumption to help fulfil the goal of a fossil-free building sector by the year 2035 [4]. Traditional mechanical ventilation system can have specific fan power (SFP) values of 5.5-13 kJ/m^3, while new systems have SFP-values of 2-2.5 kJ/m^3 [5]. The heat recovery units in traditional ventilation systems are usually cross-flow heat exchangers with efficiencies of around 50% if there is any heat recovery at all, while new systems have efficiencies of 80-85%. In the Danish building code, the maximum SFP value is currently 2.1 kJ/m^3 for VAV systems, and this will be reduced to 1.8 and 1.5 kJ/m^3 in 2015 and 2020, respectively. It is difficult to comply with these requirements in renovation cases and therefore not mandatory, but it is essential to upgrade the existing building mass to future levels as well to reach the goal of a fossil-free building sector in 2035. There have been numerous studies and projects with various approaches to reduce the energy consumption for ventilation. Most of them focus on natural or hybrid ventilation, but the designs are rarely applicable for renovation cases [6, 7, 8], and the story is the same for the projects focusing on mechanical ventilation. In Berry [9], Hestad [10] and Tjelflaat [11] significant reductions were obtained with SFP-values between 140-400 J//m^3, but it would require large structural changes to integrate the designs in an existing building and that would be costly. In general, we lack both solutions and knowledge about how to renovate the ventilation systems in existing schools and improve the indoor environment most efficiently. So new solutions and concepts are needed, and for the solutions to be implemented in practise they must be cost-effective compared to standard solutions. This paper describes a low-pressure mechanical ventilation concept that can reduce energy consumption and improve the indoor environment, and which has been developed for the renovation of school classrooms. Many schools have a general renovation backlog and solutions are needed for other aspects of school buildings, but this paper only focuses on the ventilation system. The system design is based on a low-pressure concept, and the objective was to determine the performance of the different design solutions and components used, including how a conventional oversized air handling unit operates at low pressure. Amongst other things, the system has a high efficiency heat exchanger, a diffuse ventilation inlet, and a demand-controlled flow rate in response to CO_2-concentrations. To examine the performance of the concept in practice, a pilot system was installed in two classrooms at Vallensbæk School. The performance of the concept was studied using computer simulations and measurements of energy consumption and indoor environment that included the continuous monitoring of CO_2-concentrations and temperatures. To reduce pressure loss, larger and special components were used, and this resulted in increased purchase cost. To determine whether it is feasible to extend the concept to other schools and other building types, the system must be cost-effective compared to standard design solutions. So the life-cycle cost (LCC) of the installed system was examined in terms of the cost of conserved energy (CCE) [12, 13] and net present value (NPV) [14]. The analysis looked at the entire system as well as the individual components and design solutions to determine the overall cost and identify the advantages and disadvantages of the concept. The aim of the pilot project was to validate the performance and overall applicability of the new ventilation concept. It had to provide an acceptable indoor environment by maintaining the CO_2-concentration below the maximum level recommended by the Danish Working Environment Authority of 1000 ppm, reach an annual average SFP-value of 650 J/m^3 (less than half of the value required for 2020 in the Danish building code of 1500 J/m^3), and determine whether the system and individual components are cost-effective.

2. Development of the concept

When designing, developing, and optimizing a new ventilation system, the challenge is to find the right balance between cost, air quality, thermal comfort, energy consumption and environmental impact in periods with cooling or heating over the year [15]. Maintaining an acceptable indoor air quality and thermal comfort in our buildings is the primary purpose of a ventilation system and it must be able to meet these requirements because otherwise it loses its function [4]. High efficiency heat recovery is a well-established feature of modern ventilation systems and only minor additional savings can be obtained in this area. The greatest energy savings potential lies in the fan power used to transport the air in the system. The fan power required depends on the volume flow rate and the total pressure loss in the system. Conventional mechanical systems have relatively high pressure losses because the focus in the industry has been to make systems as small as possible to minimize space use in buildings. To reduce the fan power needed, it is essential to improve the integration of ventilation systems in the buildings and to reduce pressure losses in diffusers, the duct system and the Air Handling Unit (AHU) [4, 6, 16]. Reducing the pressure losses will also decrease the air velocities in the system, reducing the risk of discomfort from noise and draught. Another key aspect in reducing energy consumption is to tailor the flow rate to the actual demand, so the control system has a significant impact on both operating hours and fan power [17, 18]. These changes will lead to higher capital costs, but since ventilation systems remain in service for 30-40 years, the lower operational costs for energy will outweigh the higher installation and purchase costs [16].

2.1 Concept proposal

The concept is also described in Terkildsen [19] and was developed for a temperate climate as in Denmark. This entails a need for heat recovery in the winter season and a limited need for cooling in the summer if high solar gains can be prevented. Active cooling is therefore dispensable and the ventilation system only needs to be designed to maintain an acceptable air quality and have highly efficient heat recovery. The use of school classrooms is highly irregular with varying day-lengths and classes in other school facilities. Demand control is therefore vital to reduce energy consumption and adjust for the peak loads that occur when classrooms are used for longer periods. To reduce the fan power needed, the entire system and every component must be designed, dimensioned and optimised to minimize pressure loss.

3. Method – Case study

To examine and evaluate the performance of the ventilation concept and design solutions in practice, a pilot system was installed in two classrooms at Vallensbæk School.

3.1 General description of Classrooms

The two classrooms are located in one wing of Vallensbæk School and denoted Rooms 42 and 44, see Figure 1. The outer walls are masonry with partition walls of plasterboard, and the windows on the main façade are oriented northwest. Room 42 has a floor area of 81 m^2 and Room 44 has 60 m^2, both with a floor height of 2.5 m in most of the room, see Figure 1.

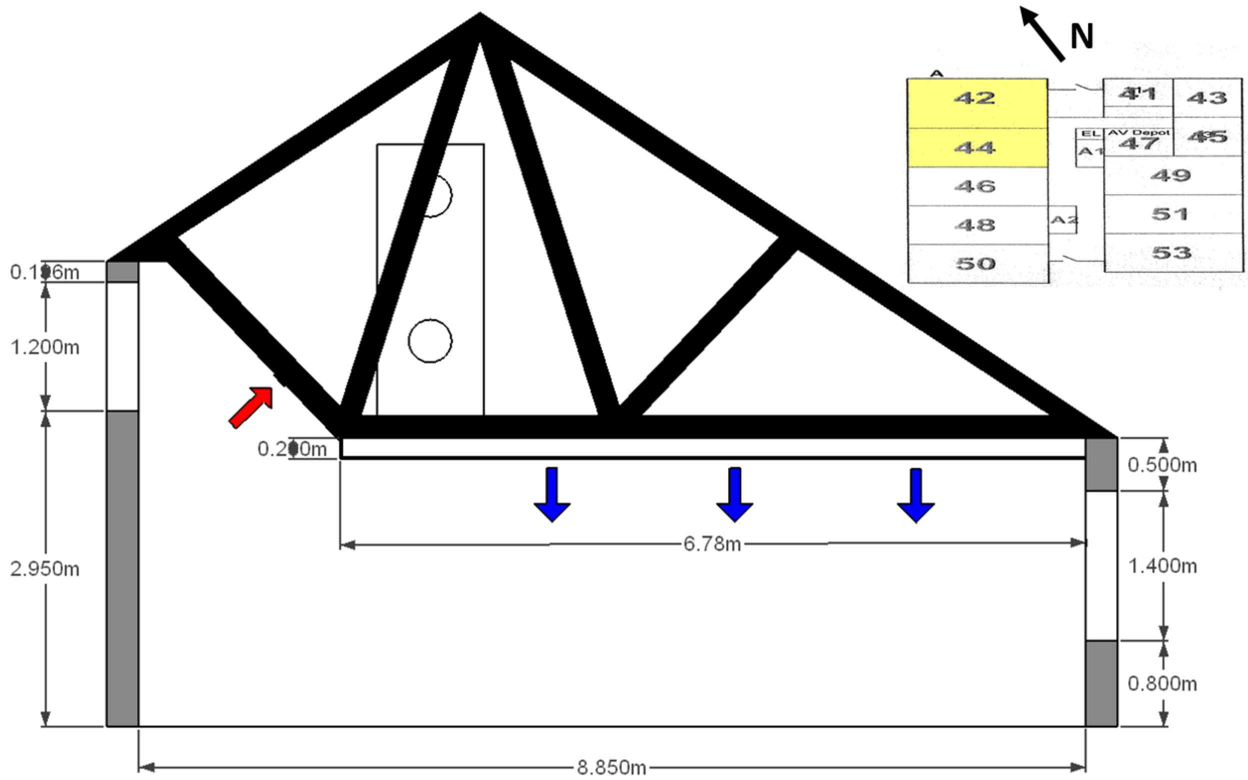

Figure 1. Section of the two classrooms at Vallensbæk School and a plan of the wing.

The AHU was positioned in the attic above Room 42 to reduce duct lengths, and fresh air for the rooms is supplied through a diffuse ventilation inlet in the horizontal part of the ceiling while the extraction was located in the sloped part of the ceiling.

3.2 Indoor environment requirements

The design criteria for the new ventilation system were to improve the indoor environment by maintaining CO_2-concentrations below 1000 ppm, which is the maximum recommended by the Danish Working Environment Authority. Furthermore, the room temperature was to comply with the temperature ranges recommended in EN 15251 Class II of 20-24 °C in the winter and 23-26 °C in the summer [20]. The heating demand was to be covered by the heating system, with the supply air preheated in the heat recovery unit to decrease ventilation heat loss and avoid draught problems. The design flow rate for the system was set to 5 l/s per person resulting in a flow rate of 450 m^3/h for one room with 24 pupils and one teacher. This corresponds to an air change rate of 2.0 h^{-1} and 2.6 h^{-1} for Rooms 42 and 44, respectively. The rooms are used for a maximum of 1.5 hours at a time before a break of at least 20 minutes, during which most of the pupils leave the room, which provides a break to recondition the room.

3.3 – System design

The ventilation system was designed as a conventional mechanical system with an air handling unit and a duct system to distribute the air, and using components available on the market to reduce development and purchase costs. The system was designed just to provide fresh air without active cooling because the cooling demand was expected to be limited. This was due to low internal gains, varying occupancy and the windows on the main façade oriented northwest that prevent high solar gains in the occupied hours. The system was intentionally designed with an oversized AHU and duct system to reduce pressure losses and the fan power required to operate the system. To minimize ventilation heat loss, the system employed two cross-flow heat exchangers providing highly efficient heat recovery. The supply air to the rooms was distributed through a diffuse ventilation inlet, so that the air is supplied over a large part of the ceiling area through small perforations. The supply method is relatively new for comfort ventilation, but test results have showed promise [3, 21, 22]) and on-site measurements from Vallensbæk School substantiate the results in practice [23]. The flow rate to the rooms was demand-controlled by the CO_2-concentrations in each room to avoid unnecessary energy use, and the flow rate was regulated by a new type of flow damper that induces lower pressure loss.

3.4 Air handling unit

The AHU was a conventional unit with a heat exchanger, radial fan and compact filter, but was intentionally oversized compared to standard dimensioning to reduce pressure losses in the filters and heat exchanger. According to the manufacturer, the unit needs a minimum flow rate of 450 m^3/h in order to function properly. This corresponds to the design flow rate for one room and would therefore be tested when only one room was occupied. The maximum flow rate of 2500 m^3/h was well above the expected demand of 900 m^3/h for the two rooms and the fan was therefore expected to operate at low speeds under all running conditions, reducing fan power consumption. The two counter-flow exchangers in series had a combined efficiency of 84% at the design flow rate. Higher efficiency was not expected to reduce the heating demand significantly because the ventilation heat loss is expected to be covered by internal gains from occupants, the lighting system and equipment [24]. A heating coil was omitted because previous studies of diffuse ventilation inlets have shown that it is possible to supply cold air ($\Delta T>8$ °C) without causing draught problems [21, 22, 25]. The efficient heat recovery therefore made a heating coil dispensable – an easy way to reduce cost. The unit had filters on both supply and extract sides with efficiencies of EU7 and EU5, respectively.

3.5 Diffuse ventilation inlet

Diffuse ventilation inlets have been used mostly for special applications such as clean rooms or livestock, and they represent a relatively new diffuser solution in comfort ventilation systems. The inlet air is supplied to the room through perforations in the ceiling via a distribution plenum above a suspended acoustic ceiling. This results in a very large supply area and thus at very low air velocities with random directions and hence the term diffuse. A new ceiling was installed in the two classrooms on the horizontal part of the ceiling surface, see Figure 2.

Figure 2. Picture during the installation of the diffuse ventilation inlet.

The ceiling was lowered 20 cm to create a plenum to distribute the supply air which consisted of a metal suspension system on which new cement-bonded wood wool acoustic panels were mounted. The panels are widely-used in Denmark due to their acoustic properties. Two types of panel were needed for the diffuse ventilation inlet: active and passive. The active panels were permeable so the supply air could penetrate and the passive panels had 20 mm of hard painted mineral wool glued on the back making them non-permeable. The panels created an overpressure of 1-2 Pa in the plenum above and ensured that the supply air was equally distributed through the active panels. The position of the active and passive panels allows control of the supply air distribution to specific part of a room. In the two rooms the active panels were to be uniformly distributed over the horizontal part of the ceiling and covered 11% and 17% in Rooms 42 and 44, respectively. The active panels were not uniformly distributed as intended, but this was discovered after the measurements had been made.

3.6 Duct system

Traditionally, a pressure gradient of 1 Pa/m is used to determine the duct size in order to avoid excessive noise generation and limit space use [26]. Reducing pressure loss in the duct system is relatively easy and straightforward and plays an essential part in reducing the total pressure loss and the fan power needed. By increasing the dimensions of circular ducts by one size (e.g. 250 to 315 Ø) the air velocity decreases ~35% and the pressure loss in straight circular ducts is reduced by ~60%. Even larger reductions can be obtained in bends, t-pieces, reducers etc. The duct system at Vallensbæk School was designed for a pressure gradient of 0.1 Pa/m, resulting in air velocities of 1-2 m/s at the design flow rate of 450 m^3/h to each room.

3.7 Demand control system

The air flow to the rooms was controlled by a new type of droplet-shaped damper [27], see Figure 3. The damper has an aerodynamic design that minimizes turbulence generation and pressure, and it can operate at low air velocities.

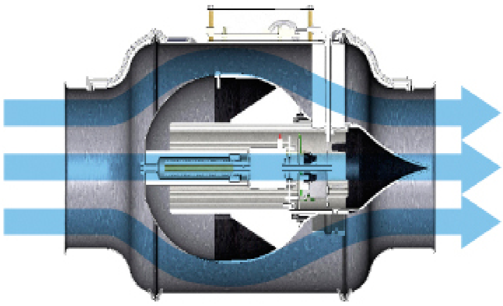

Figure 3. Sketch of airflow pattern through LeanVent® dampers (left) and picture of a damper installed at Vallensbæk School (right).

The control panel on the damper can calculate the flow rate by measuring the pressure loss across the damper and accurately regulate it by adjusting the position of the droplet-shaped head in the flow direction [28]. The ventilation demand was determined by a CO_2-sensor in each room that transmitted a signal to the respective flow control damper. The signals were in a range of 0-10 V corresponding to CO_2-concentrations of 0-2000 ppm. The dampers then regulated the flow rate in accordance with the diagram shown in Figure 4.

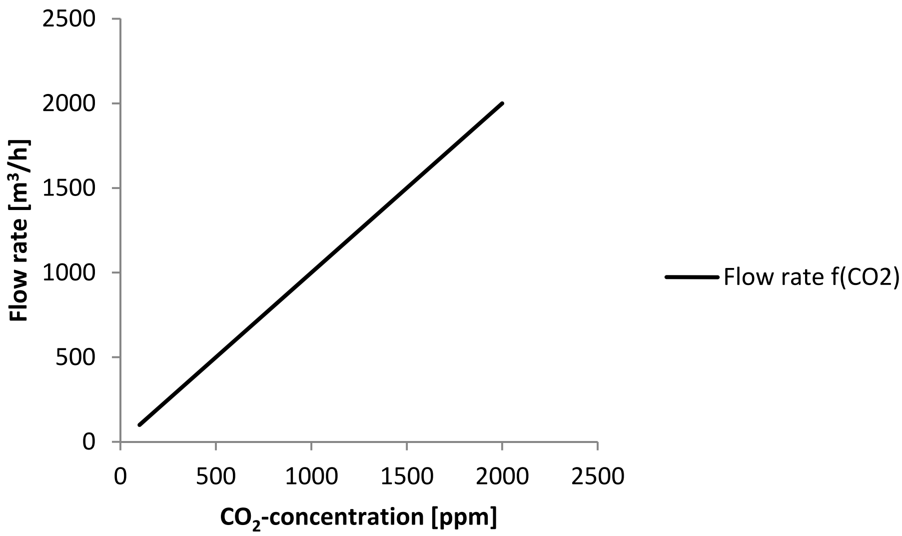

Figure 4. Theoretical ventilation flow rate to one room as a function of the total CO_2-concentration.

CO_2-concentrations in the rooms in the morning and after breaks were expected to be close to the outside concentration of 360-400 ppm, and the flow rate would therefore be lower than the design value. However, as the CO_2-concentrations build up, the flow would increase and maintain the concentration below the target value of 1000 ppm. The AHU had a time control, and motion sensors were installed in each room. The time control was set to start the system at 7.30 am and shut it down at 3.30 pm to avoid unnecessary ventilation outside occupied hours. The use of the classrooms is highly irregular because the pupils have classes at other locations in the school, so motion sensors were installed to shut down the flow to the room when it was not occupied. The fan was set to maintain a constant static pressure in the duct system controlled by a static pressure sensor positioned midway in the supply duct. The optimal set point for static pressure was determined during the system set-up process and the aim was to keep it as low as possible to reduce the fan power requirement.

3.8 Measurements and calculations

The performance of the ventilation concept and single components or design solution was evaluated on the basis of the indoor environment, energy consumption, pressure loss, general applicability and life-cycle cost. To determine the performance, the following calculations and measurements were carried out.

- Measurements and theoretical calculation of the pressure loss in the entire system and individual components to determine the pressure loss characteristics of the system.
- Measurements and theoretical calculation of energy consumption to determine the SFP-value of the system.
- Temperature and CO_2-concentration measurements to validate the indoor environment before and after installation.
- Calculations of indoor environment and annual fan power consumption in the simulation tool BSim [29].
- Investigation of the cost effectiveness of design solutions and components based on investment and running costs using the NPV and CCE methods.

During the measurements, the rooms were used by the 6th grade with 24 pupils in each class. The CO_2-concentrations were measured with a Vaisala GM20 CO_2-transmitter connected to a HOBO data logger that measured the air temperature. The energy consumption was measured with a Kyoritso 6305 energy analyser that can measure true RMS 3-phase power due to the non-sinusoidal line currents drawn by the frequency controlled variable speed drive (VSD).

3.9 Simulation model

The ventilation system was modelled in the simulation tool to determine the annual energy consumption, the average SFP-value, and the indoor environment. Weather data from the Danish design reference year was used in the simulations and Table 1 shows the input data for construction parts and internal heat loads.

Table 1. Input data for construction parts and internal heat sources in the simulation model

Constructions parts	U-value [W/m^{2}K]
Exterior wall	0.22
Roof	0.16
Floor	- (above heated basement)
Window with interior curtains (g-value = 0.56, LT-value = 0.73)	1.2

Lighting, equipment and occupants	Heat load
Lighting system [W/m^2]	7.3
Equipment, 1 laptop + smart board [W]	25 + 400
Occupants, 25 people [W]	25 x 100

The rooms were expected to be occupied for 1.5 hours with 0.5 hour breaks between 8 am and 3 pm Monday to Friday with the ventilation system starting 1 hour before and shutting down 1 hour after. The measured pressure loss characteristics and fan power use were used to simulate the annual energy consumption and the average SFP-value of the system.

3.10 Life-cycle cost

The CCE method calculates the cost per saved energy unit (DKK/kWh), which is directly comparable to the cost of energy supplied and therefore gives an immediate idea of the economic efficiency of the measure. The method can take the effect of inflation into account, but not the expected development in energy price as in the NPV method, so this method was also used. The CCE was calculated using Equation 1 [13]:

$$CCE = \frac{t \cdot a(n_r, d) \cdot I_{measure} + \Delta M_{year}}{f_1 \Delta E_{year} - f_2 \Delta E_{operation,year}} \qquad (1)$$

where I is the investment cost or, as in this case, the additional cost of the energy conserving measure, and ΔM is the increase in maintenance cost per year in DKK. ΔE_{year} and $\Delta E_{operation}$ are respectively the additional annual conserved energy and additional annual energy consumption of the measure compared to a reference system in kWh. f_1 and f_2 are the primary energy conversion factors for conserved and consumed energy for the energy conserving measure, which the Danish building code expects to be 0.6 for heating (district heating) and 1.8 for electricity in the year 2020. However, these were set to 1.0 in the calculations because the final cost for the owner depends solely on the unit price per kWh which is not affected by the primary energy factors. $a(n_r,d)$ is the capital recovery rate, where d is the real interest rate and n_r is the reference period or lifetime for the system:

$$a(n_r, d) = \frac{d}{1-(1+d)^{-n_r}} \qquad (2)$$

where d is the real interest rate taking inflation into account:

$$d = \frac{(1+d_{nominal})}{(1+d_{inflation})} - 1 \qquad (3)$$

The factor t is introduced to give a fair frame of reference for measures with different lifetimes. In this case, all the measures concern the ventilation system and the solutions were estimated to have similar lifetimes so it was set equal to 1.

The NPV method calculates the total cost of various energy saving measures over their lifetime. The measures can then be ranked, with the lowest NPV being the most cost-effective, and if the NPV is negative the measure is economically beneficial because the energy savings outweigh the investment. The NPV was calculated using Equation 4 [14].

$$NPV = I_0 + \sum_{J=x,y,z} \frac{I \cdot (1+r_I)^j}{(1+d)^j} + \sum_{i=1}^{n} \frac{E \cdot (1+r_E)^i}{(1+d)^i} + \sum_{i=1}^{n} \frac{M \cdot (1+r_M)^i}{(1+d)^i} \qquad (4)$$

Here again, I is the investment cost, but E and M are now the total annual energy consumption and maintenance costs and not the additional cost. However, the maintenance cost was assumed to be equal for all the systems and therefore left out of the calculations. r_I, r_E, r_M are the annual net increases in investment, energy and maintenance costs (net=above inflation) and d is the real interest rate determined by Equation 3. In the calculations, a nominal interest rate of 5%, inflation rate of 1.8%, electricity price of 1.62 DKK/kWh (ex. VAT), and an increase in electricity price of 2.62% per year was assumed. The increase in electricity price was the price development forecast by the Danish energy agency based on the IEA world energy outlook [30]. The lifetime of the system was set to 40 years with a reinvestment in the system after 20 years of 50% of the initial additional cost. The contractor gave tenders for installing a standard system and the low-pressure system specifying the additional purchase and installation cost of the specific design solutions and components and these prices are listed in Table 2. The installation cost was equal for all the components with the exception of the diffuse ventilation inlet where the additional labour cost was included; otherwise the prices are strictly additional purchase cost. The profitability of the low-pressure system was determined based on the additional cost and the energy savings of the solution and a standard system that had an annual average SFP-value of 1489 J/m^3 was used as reference.

Table 2. Cost of the individual parts in the standard system and the low-pressure system in DKK.

	Low-pressure system	Standard system	Additional cost
Air handling unit	82,600	72,600	10,000
Heating coil	0	20,000	-20,000
Duct system	38,000	36,000	2,000
Flow dampers	22,400	12,000	10,400
Supply system	94,000	20,000	74,000
Total	237,000	160,600	

When calculating the profitability of the supply solutions, the savings from the heating coil were subtracted from the additional cost of the diffuse ventilation inlet (supply system) because conventional diffusers would cause draught problems without preheating of the supply air. The energy consumption for the ventilation system was calculated using Equation 5 [13].

$$Q = SFP \cdot p \cdot k + c \cdot \rho \cdot (1 - \eta) \cdot D_v \cdot (t_{bal}) \quad (5)$$

The second term represents the heat loss from the ventilation in the heating season and depends on the heat recovery efficiency n. The efficiency only varied between 83-85% depending on the flow rate and external pressure loss, so this part was left out and only the power consumption of the fans was taken into account. The SFP-value depends on the flow rate and pressure loss and is therefore not a fixed value. In both the CCE and NPV calculations, the annual average SFP-value based on the simulated annual operation was used.

4. Results

4.1 Indoor environment

The air temperature and CO_2-concentration readings for a week in the winter period before the low-pressure system was installed are shown in Figure 5.

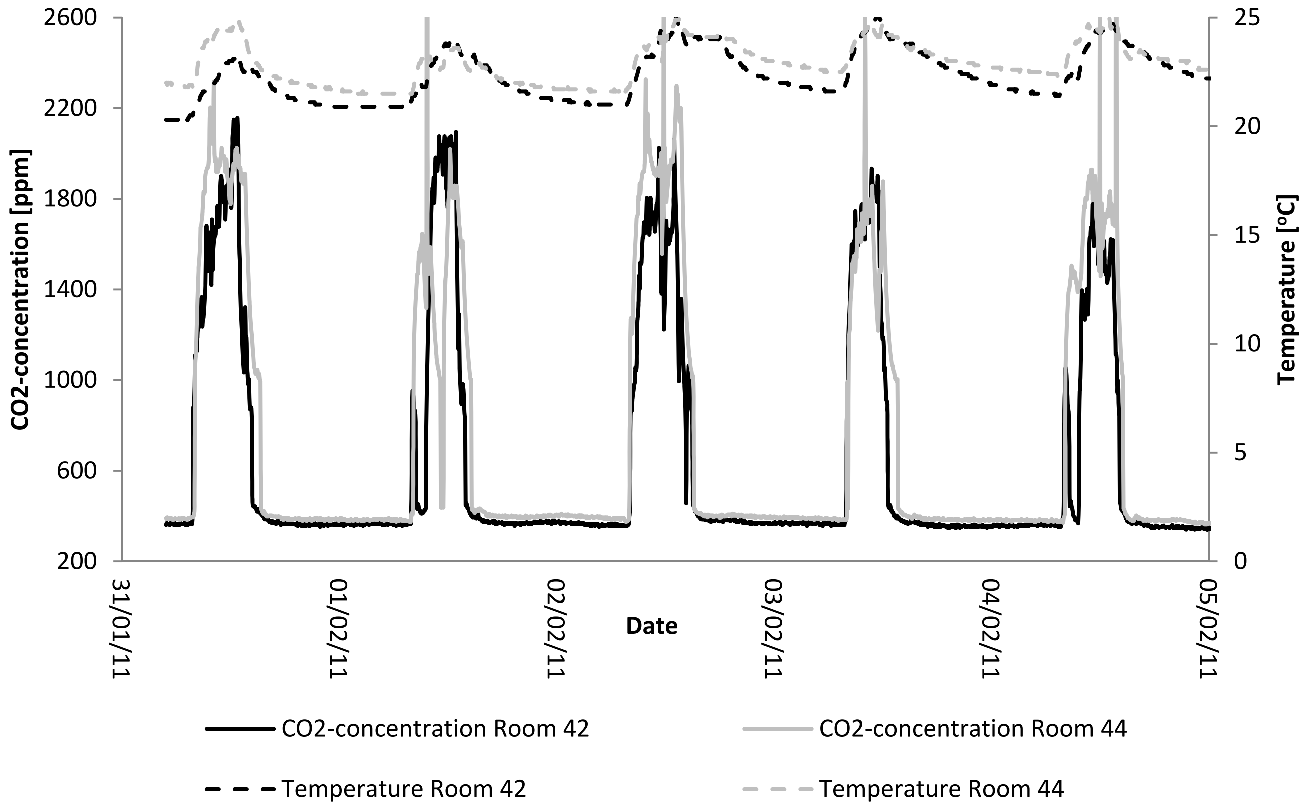

Figure 5. CO_2-concentration and air temperature measurements in the two classrooms before the installation of the new ventilation system.

The temperature varied between 21-25 °C in the two classrooms, exceeding the recommended temperature range for the winter period by 1 °C. The CO_2-concentration increased from approximately 400 ppm to around 1800 ppm during the occupied hours with peak values of up to 2600 ppm in Room 44, well above the 1000 ppm maximum recommended by the Danish Working Environment Authority. Figure 6 shows the air temperature and CO_2-concentration readings for a week in the summer period after installation of the new system.

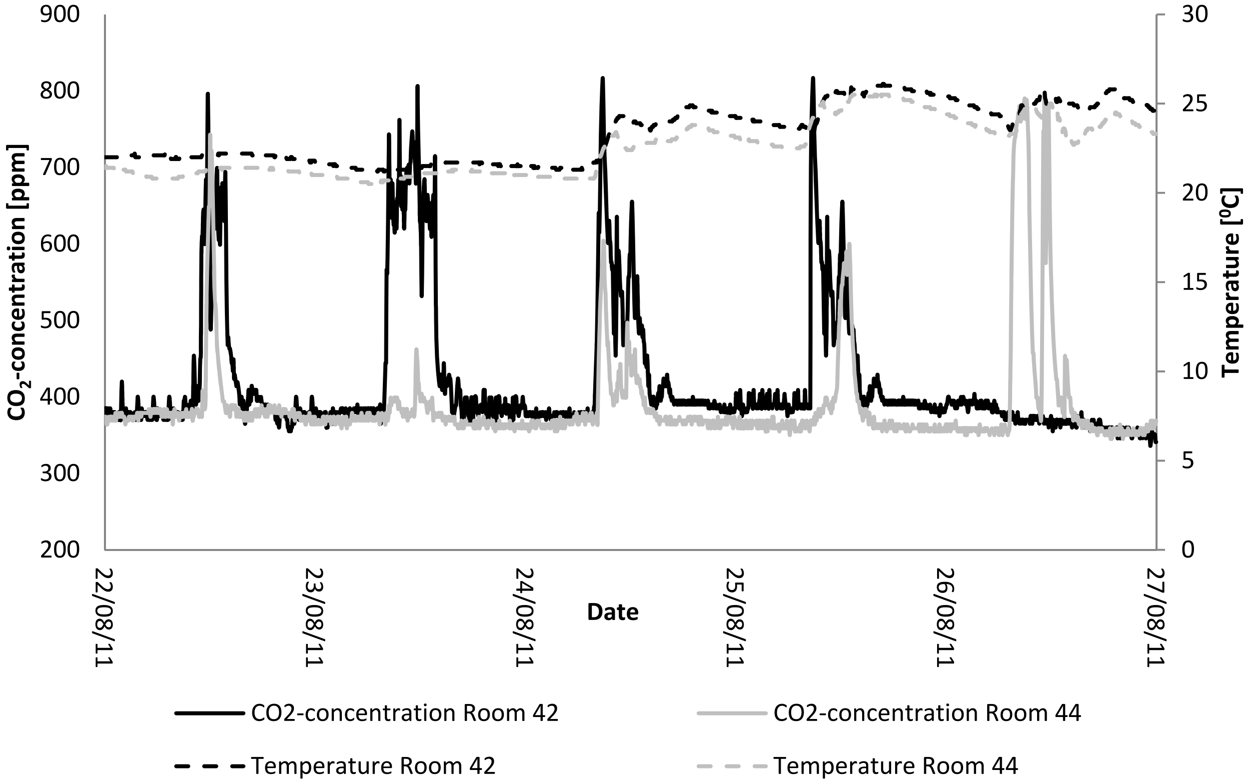

Figure 6. CO_2-concentration and air temperature measurements in the two classrooms after installation of the new ventilation system.

The temperature varied between 21-26 °C, exceeding the minimum recommended design temperature by 2 °C. The CO_2-concentration was around 500-700 ppm during the day with peaks of up to 800 ppm, a decrease of around 1000 ppm from before installation and well below the recommended maximum. The simulation gave identical CO_2-concentration fluctuations each day, see Figure 7, which does not reflect the measured CO_2-concentrations in the classrooms in Figure 5 and 6. However, the simulated CO_2-concentration peaked at 750 ppm, almost the same as the measured concentration, and the temperature in the rooms varied from 20-26 °C as defined in the design criteria.

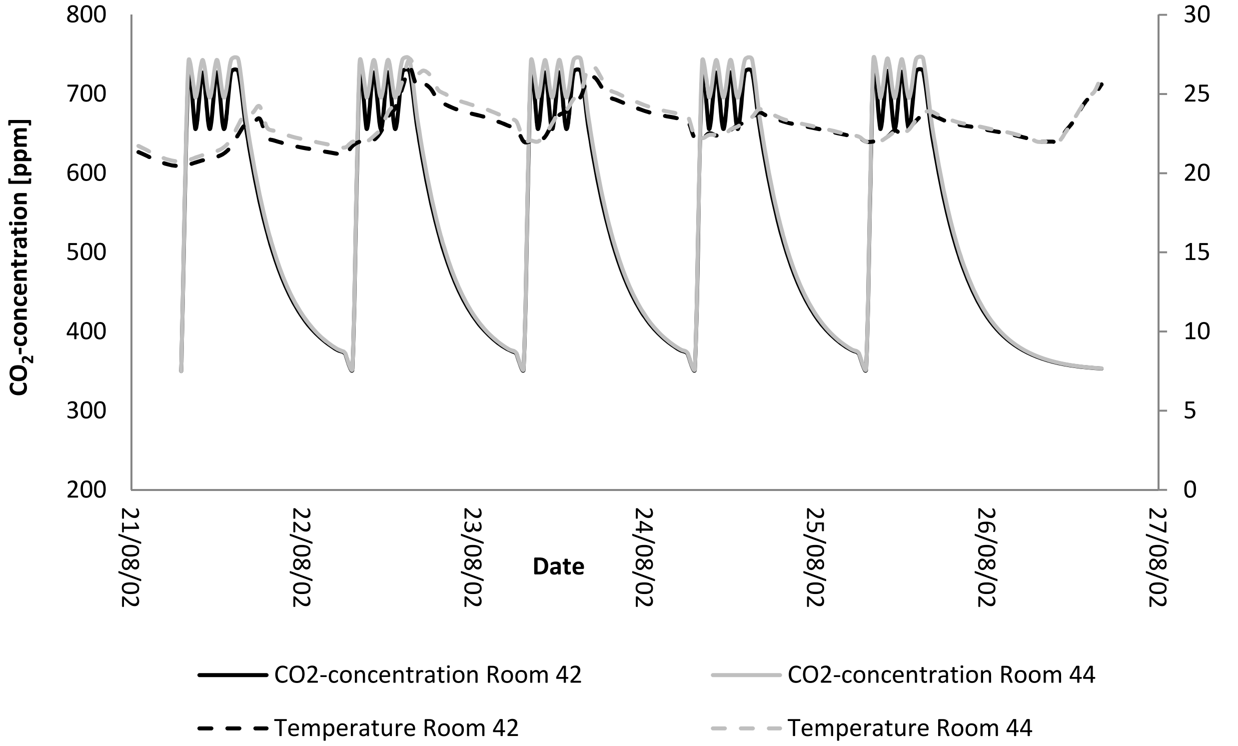

Figure 7. Simulated CO_2-concentrations and air temperatures in the two classrooms

4.2 Pressure loss characteristics and specific fan power

The design and measured pressure loss for the various components in the ventilation system with a flow rate of 900 m^3/h (corresponding the design flow rate to the two rooms) is listed in Table 3. The total measured pressure loss was respectively 36% and 28% lower than the design values for the supply and exhaust side. The difference was mainly due to lower pressure losses in the heat exchanger and duct system.

Table 3. Design and measured pressure losses for components in pilot ventilation system at a flow rate of 900 m^3/h.

Component	Design pressure loss – supply system	Design pressure loss – exhaust system	Measured pressure loss – supply system	Measured pressure loss – exhaust system
Intake/exhaust	5	5	-	-
Heat exchanger	55	55	32	32
Filter	34	28	31	29
Duct system	32	39	12	21
Droplet damper	20	20	19	23
Diffuse ceiling/ extract diffuser	2	10	1	8
Total	148	157	95	113

Figure 8 shows the pressure loss characteristics for the system along with the SFP-value characteristics depending on the flow rate. The measured total pressure loss was 100-150 Pa lower than calculated, and this was mainly due to the higher pressure losses in the AHU component calculated in the manufacturer's software tool. The lower than expected pressure loss did not result in a lower SFP-value; the measured values were within 4% of the calculated SFP-values. At the design flow rate of 900 m^3/h, the measured SFP-value was 611 J/m^3 and the heat exchanger efficiency was 84%. The total fan and motor efficiency indicated with arrows in Figure 8 shows that the efficiency decreased at low flow rates. In addition to the fan consumption, the system uses around 45 W for sensors, controls and dampers during operation that should be taken into account when determining the energy consumption of the system.

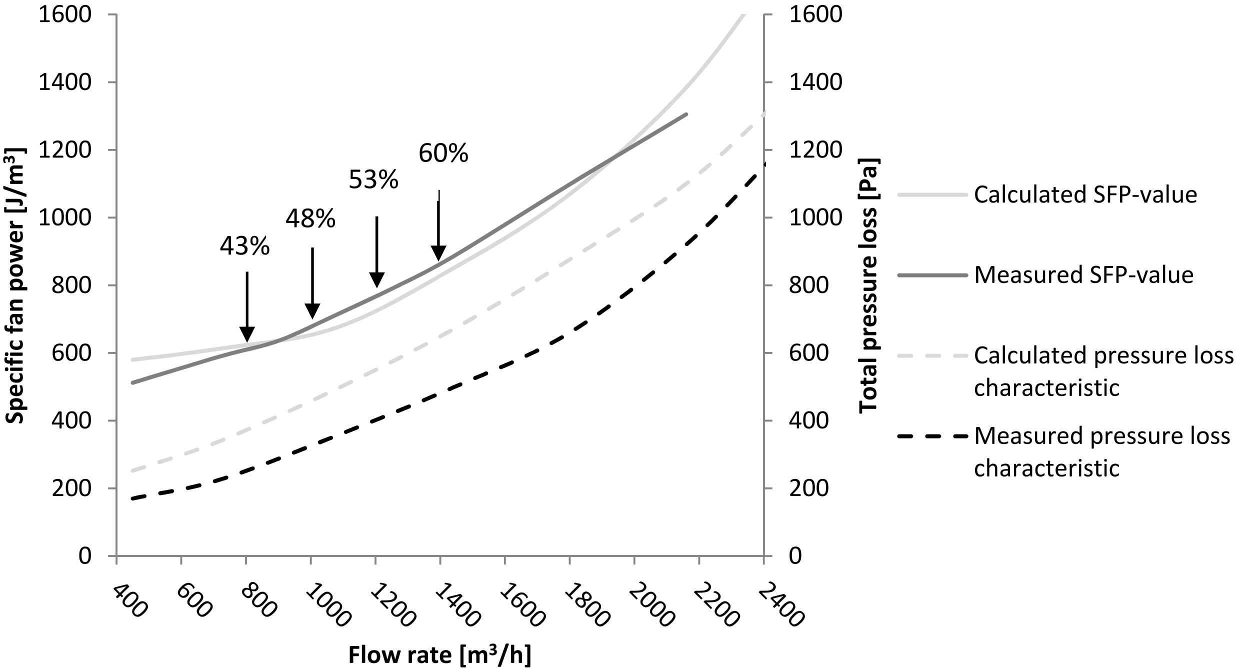

Figure 8. Calculated and measured SFP-values for the low-pressure ventilation system along the measured and calculated pressure loss characteristics of the system. Arrows indicate the total efficiency of the fans and motor at the respective flow rates.

In total, the annual power consumption was calculated to 349 kWh in the simulation tool, of which the controls accounted for 29%. This equals 4.48 kWh/m^2 including a primary energy factor of 1.8. The annual average SFP-value for the system was calculated to 627 J/m^3, including the frequency control but excluding sensors, controls and dampers following the method for variable flow rates described in [31]. Including sensors, controls and dampers, the annual power consumption and SFP-value increased to 450 kWh and 808 J/m^3, respectively.

4.3 Life-cycle cost

The results of the CCE and NPV calculations are listed in Table 4. The NPV calculations show that the droplet damper and the low-pressure duct system measures gave lower life-cycle cost, while the low-pressure AHU cost 3,000 DKK above the standard. The NPV of the diffuse ventilation inlet, however, was almost 2.5 times higher due to the high initial cost. The CCE calculation gives the same picture with the low-pressure duct system and droplet damper measures having lower values and the low-pressure AHU having slightly higher value than the electricity price of 1.62 DKK/kWh. The CCE for the entire system is 8 times higher than the energy price, again due to the initial cost. The high additional initial cost is mainly due to the cost of the diffuse ceiling. If that cost is taken out, the CCE decreases to 3.94 DKK/kWh and the NPV to 44.282 DKK.

Table 4. Results of NPV and CCE calculations for the individual measures, the standard system, and the entire low-pressure system.

	Additional initial investment [DKK]	Additional investment year 20 [DKK]	Annual energy consumption [kWh]	Net Present Value, total investment [DKK]	Net Present Value, energy use [DKK]	Net Present Value, total [DKK]	Cost of conserved energy [DKK/kWh]
Standard system	-	-	875	-	36.691	36.691	-
Low-pressure AHU	10.000	5.000	621	12.692	26.585	39.278	3.44
Diffuse ventilation inlet	54.000	27.000	501	68.539	21.008	89.547	12.64
Droplet dampers	10.400	5.200	452	13.200	18.954	32.154	2.15
Low-pressure duct system	2.000	1.000	440	2.538	18.451	20.989	0.40
Low-pressure system	76.400	38.200	378	96.970	15.851	112.821	13.44
Low-pressure system, without new ceiling cost	22.400	11.200	378	28.431	15.851	44.282	3.94

5. Discussion

The system was rather small, supplying just two rooms, but it was designed like larger systems with a duct system, and a central AHU and control system, and not as a decentralised ventilation unit. The measurements and experiences are therefore believed to provide valuable knowledge on how to reduce energy consumption that can be transferred to larger mechanical ventilation systems. The CO_2- measurements in Figure 5 before the installation of the low-pressure system showed that CO_2-concentrations increased rapidly and exceeded the recommended 1000 ppm when the rooms were occupied. This was due to the existing out-dated ventilation system that was not able to provide an acceptable ventilation rate, but the exact ventilation rate before installation was not determined. After installation, CO_2-concentrations were kept below 800 ppm, showing that the new system was able to improve the indoor air quality considerably. The measurements before and after were performed at different times of the year, respectively, winter and late summer. This could have affected the ventilation rate after installation positively because natural ventilation by opening windows is more likely to occur in the summer period, but in the measuring period it was ensured that the windows were kept closed during lessons and only opened during breaks. The measuring conditions before and after were therefore comparable and although the ventilation rate may have been increased by natural ventilation during breaks, the low CO_2-concentrations during classes proved that the CO_2-control worked as intended and was able to maintain good air quality in the rooms. The simulated CO_2-concentration progression in Figure 7 does not match the measured concentrations because of the assumptions and limitations in modelling the actual irregular use of the rooms in practice. However, the maximum CO_2-concentration of 750 ppm does correlate well with the measured maximum value and that is the most important aspect when predicting and assessing the performance of a ventilation system. The temperatures measured after installation did not fulfil the desired criteria because the temperature was below 23 °C in the mornings, but pupils would be dressed for colder outside temperatures (>20 °C) in the morning and are able to adjust their clothing level during the day. No problems with overheating were recorded, but the measurements were also carried out in a relatively mild part of the summer period. Further measurements will be necessary to determine the thermal indoor environment under warmer conditions and in the winter as well. If overheating arises in warmer periods, increased ventilation and/or night cooling could be used to solve the problem. The measured pressure losses of the system were respectively 36% and 28% lower for the supply and exhaust sides than calculated. This was due to lower pressure losses in the heat exchanger and duct system. For the duct system, the pressure loss was lower than calculated across the fire dampers and silencers, while the cause of the deviation for the heat exchanger has not been clarified. The calculated pressure loss was determined in the manufacturer's dimensioning software and measured by pressure sensors built in by the manufacturer to monitor the AHU, so the discrepancy most likely occurred due to either erroneous software or sensors. The droplet dampers were able to control the airflow precisely while inducing pressure losses of only 14-30 Pa depending on the flow rate, and they are the key to operating at a low static pressure. The static pressure set point to control the fan speed was set to 25 Pa, well below the standard values of 150-200 Pa recommended by the AHU manufacturer. The CO_2-concentration was mainly between 500-700 ppm, well below the 1000 ppm target. By altering the inclination of the control curve in Figure 4, the ventilation rate could be reduced while still fulfilling the design criteria. This would reduce the fan power needed and should be considered to optimize the control system, but it was not done during the measurements. The diffuse ventilation inlet induced a pressure loss less than 2 Pa and, moreover, Hviid et al [23] have shown that the method provides perfect mixing in the rooms without causing draught problems even at high flow rates and an inlet temperature of 10 °C. Potential soiling of the distribution plenum could contaminate the supply air and adversely affect the indoor air quality, and this needs further examination, but it is assumed not to be an issue if the supply air is efficiently filtered and will not be dealt with here. The annual average SFP-value of the system was 627 J/m^3 based on the annual simulation, so it fulfilled the target of reducing energy consumption to 50% of the 2020 requirements. Furthermore, the control system used 45W, which if included would increase the SFP-value to 808 J/m^3. This is not done in the current guidelines [31], but that extra consumption is something that should be considered when evaluating the energy consumption of a system. The calculated annual energy consumption of 5.79 kWh/m^2, of which controls, sensors and dampers account for 29%, will constitute 28% of the expected energy framework for new buildings in the year 2020 [4]. This shows that it is possible to make ventilation systems for renovation cases that can help meet future energy requirements, but their efficiency could be improved by optimising the AHU for low pressure, and especially the fan efficiency, which dropped to around 40% at low flow rates, could be improved by appropriate fan sizing. The life-cycle cost calculations showed that the investment in the low-pressure concept was more expensive than the standard system, see Table 4. The NPV and CCE calculations gave the same ranking of the individual measures, but the NPV method gave the most positive results. This is due to the difference in method because the NPV method takes the expected increase in energy price into account while CCE does not, but with the assumed increase in electricity price it will take 35 years before the price exceeds the CCE of the entire system (without the new ceiling) of 3.94 DKK/kWh. The uncertainty with regard to forecasting the discount rate, inflation, energy price and energy price increases over the lifetime of 40 years is high. It is therefore difficult to determine whether a particular measure or the entire system is cost effective, but it is possible to establish which measures are most likely to be cost effective. For the individual measures, only the low-pressure duct system was a good investment with both the CCE lower than the electricity price and the NPV lower than the reference system, showing that it can be a good investment to stray from the traditional guideline of 1 Pa/m pressure loss gradient that is normally used [26]. For the droplet dampers, the CCE was 32% higher than the electricity price while the NPV was 12% lower than the reference, so whether they constitute a profitable solution is uncertain, but if the cost of the dampers could reach market level it would clearly be a good investment also considering the improved air flow control [28]. The oversized AHU reduced the pressure loss in the heat exchanger and filters, but not enough make it a good investment, and this was mainly due to the fans also being oversized making the system inefficient at low airflows, as shown in Figure 7. The diffuse ventilation inlet was clearly not a good investment in a straight economic sense with a CCE 8 times higher than the electricity price and NPV 2.5 times higher than the reference system. This was due to the high initial cost and relatively limited energy savings, but it has clear advantages in terms ventilation efficiency and draught problems compared to conventional diffusers. However, it is difficult to determine the value of this increased efficiency and include it in the life-cycle cost of the system [32, 33] so it is not covered in this paper. It should also be added that the price of the cement-bonded wood wool panels was around 3 times (or 45,000 DKK) higher than gypsum panels like the ones used in Hviid 2012, and the lower initial cost would of course benefit the LLC. If the cost of the diffuse ventilation inlet was taken out of the calculations, as would be the case in a new building or larger renovation of building where a new a ceiling has to be installed anyway, it would clearly be a cost-effective solution because it saves the cost for conventional diffusers, reduces ducting and obviates the need for a heating coil. However, this scenario is not enough to make the whole system profitable; without the cost for a new ceiling, the CCE decreased to 3.94 DKK/kWh and NPV to 44,282 DKK equalling 204% and 21% above the reference. So while the oversized duct system and diffuse ventilation inlet were individually cost-effective and the larger AHU and droplet dampers were almost cost-effective, at least for the NPV method, the measures combined were not a good investment. This was because the individual energy savings do not compensate for the decreased fan efficiency at low pressure. So, to make the system competitive, it is necessary to redesign the AHU with larger heat exchangers and filters to reduce pressure loss and to dimension the fan accordingly to increase its efficiency at low pressure and not just oversize a standard unit. The reference used was the system that the contractor would have installed, and his bids were used to calculate the life-cycle cost because they gave a realistic estimate of market costs. The reference system was quite efficient and could fulfil the expected 2020 requirement, so the additional savings, though individually good, could not warrant the additional investment. If the existing system or a system that could just fulfil the current energy requirements had been used as reference, it would have made the CCE and NPV look more positive because the energy consumption constitutes a larger part of total cost.

6. Conclusion

The measurements on the pilot ventilation system showed that it is possible to make ventilation system renovations that improve indoor air quality while lowering energy consumption for ventilation to 50% of 2020 requirements. The average CO_2-concentration was lowered from around 1800 ppm to 800 ppm, which means that the atmospheric indoor environment now met the Danish Working Environment Authority's requirements. The desired criteria for the thermal indoor environment were not fulfilled, but the system was assumed acceptable because pupils would be dressed for the outside climate, which for a temperate climate like Denmark's entails temperatures below 20 °C in morning. If overheating problems were to arise, adjustments to the system could be made to abate or avoid overheating either by using night cooling or by increasing the flow rate during the day.

The use of new design solutions and "oversized" and special components led to decreased pressure losses, but also made it more difficult to fit the system into the building and increased initial costs. How the installation and manual work can be carried out should therefore always be considered. The system only supplied two rooms so it was almost like a decentralised unit, but the experience and the performance of the low-pressure concept and individual components proved valuable and can be transferred to larger systems.

- The increased size of the AHU reduced pressure loss in the heat exchanger and filters, but the fan and motor size was not reduced accordingly and that led to decreased efficiency. The LCC of choosing a larger AHU was almost cost neutral compared to the standard system.

- The oversized duct system with a pressure gradient of 0.1 Pa/m helped decrease the external pressure loss and the LCC results showed that it was a good investment. This is something that should be considered when dimensioning the duct system, instead of following the traditional rule of thumb of a pressure gradient of 1 Pa/m.
- The unique aerodynamic design of the droplet dampers made it possible to regulate the air flow rate to the rooms precisely while operating at air velocities of 1-2 m/s and inducing pressure losses of only 19-23 Pa at the design flow rate.
- The diffuse ventilation inlet is a unique way of distributing the supply air to the rooms with perfect mixing, no draught problems, and less than 2 Pa of pressure loss. The large inlet area and low inlet air velocities obviated the need for a heating coil, thereby reducing investment cost. The cost effectiveness of the method, however, depends on whether a new ceiling needs to be installed; if not, then the energy savings cannot outweigh the increased investment.

The solutions were all individually cost-effective or close to being cost neutral, but combined the system was not, due to decreased fan and motor efficiency at low air flow rates. If cost-effectiveness is to be improved, the fan and motor size must be optimized in accordance with the reduced internal and external pressure loss to improve the efficiency. The system fulfilled the energy goals set out, reaching an annual average SFP-value of 627 J/m^3 and the heat exchanger efficiency was 84%, but the annual energy consumption of 5.79 kWh/m^2 was higher than expected due to the power consumption of the control system. The control system accounted for 29% and that is something that should be taken into account when evaluating the performance the system.

Overall the results show that the low-pressure system concept with the design solutions and components used is capable of meeting future energy requirements. Furthermore, this can be done with limited additional cost and the concept is applicable for use in renovation cases as well as new buildings.

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Mechanical ventilation has been the most widely used principle of ventilation for the last 50 years, but the conventional system design must be revised in order to meet future energy requirement. The key parameter to design more efficient mechanical ventilation system is the pressure loss. This thesis examines the options and develops components and a concept for design of low pressure mechanical ventilation. The results are reported in four scientific papers that represent the main body work and shows that it is possible to reduce the fan power consumption for mechanical ventilation systems by 50 % compared to 2020 energy requirements.

DTU Civil Engineering
Technical University of Denmark

Brovej, Bygning 118
2800 Kongens Lyngby

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