A new combined ventilation and heating system enabling air temperature control on room level

Polak, Joanna

Publication date:
2020

Document Version
Publisher's PDF, also known as Version of record

A NEW COMBINED VENTILATION AND HEATING SYSTEM ENABLING AIR TEMPERATURE CONTROL ON ROOM LEVEL

BY
JOANNA POLAK

DISSERTATION SUBMITTED 2020

AALBORG UNIVERSITY
DENMARK

A NEW COMBINED VENTILATION AND HEATING SYSTEM ENABLING AIR TEMPERATURE CONTROL ON ROOM LEVEL

by

Joanna Polak

AALBORG UNIVERSITY
DENMARK

Dissertation submitted

.

Dissertation submitted: March 2020

PhD supervisor: Professor Alireza Afshari
Aalborg University

Assistant PhD supervisors: Senior Researcher Niels Christian Bergsøe
Aalborg University, Denmark

Adjunct Professor Göran Hultmark
Aalborg University, Denmark

PhD committee: Senior Scientist Ole Michael Jensen (chairman)
Aalborg University

Professor Jan-Olof Dalenbäck
Chalmers University

Associate Professor Marcus Keane
NUI Galway

PhD Series: Faculty of Engineering and Science, Aalborg University

Department: Department of Civil Engineering

ISSN (online): 2446-1636
ISBN (online): 978-87-7210-626-7

Published by:
Aalborg University Press
Langagervej 2
DK – 9220 Aalborg Ø
Phone: +45 99407140
aauf@forlag.aau.dk
forlag.aau.dk

Printed in Denmark by Rosendahls, 2020

ENGLISH SUMMARY

In order to reduce the overall energy and resource intensities of buildings and to support the shift towards more sustainable built environment, it is vital that state-of-the-art HVAC technologies evolve to address emerging challenges.

Airborne heating systems may offer a great potential in terms of flexible control of indoor environments, and thereby reducing the energy use. However, a number of existing constraints, mainly related to the provision of the desirable thermal conditions in individual rooms in a building, hinder successful implementation and operation of such heating systems in European residential buildings.

The objective of this PhD study is to develop a new combined ventilation and heating system for residential buildings, further referred to as heat valve ventilation (HVV) system, and to demonstrate the ability of the system to improve the control of thermal conditions in individual rooms in a building.

This thesis presents the development, analysis and performance evaluation of the proposed system in a laboratory environment and in a real-life building. In addition, numerical simulations were performed to evaluate the performance of two types of the supply air terminal devices intended for application in newly developed HVV system.

To enable the control of air temperature on a room level, the use of an air distribution box, called a manifold, containing a centralized heating coil and heat valves was proposed in the system. By the means of the heat valves the temperature of air could be controlled separately in each of the supply ducts in a range between 25°C and 52°C. Experimental results demonstrated that the HVV system was able to regulate independently the temperature of the air supplied to each of the served rooms at the constant supply airflow rate, thereby satisfying various demands regarding the desired air temperature in all rooms.

Concerning targeting of the HVV system energy use, the results showed that the total power used by the system could be minimized based on the operational parameters. For the reference single-family house simulated in the experiment, the minimum power use of the system occurred at the supply air temperature of 52°C and the supply airflow rate of 56 l/s, which corresponded to 0.6 air changes per hour.

Regarding the performance quantified in terms of the provided thermal conditions, the CFD simulations showed that the HVV system could avoid temperature stratification and draught inside the occupied zone for the wide range of the boundary conditions.

In terms of application in a real-life building, the main limitation of the HVV system is the capacity of the system, bounded by both the energy efficiency and the provision of the desirable indoor thermal conditions. In addition, the maintenance of the desirable air temperature in individual rooms is contingent on the internal door opening patterns.

Keywords: HVAC system, energy saving, thermal comfort, mechanical ventilation, air heating, valve

DANSK RESUME

For at bygningers samlede energi og ressource forbrug kan reduceres og understøtte den bæredygtige omstilling af det byggede miljø, er det vigtigt, at den nye HVAC-teknologi hele tiden udvikler sig for at imødekomme nye udfordringer.

Luftbårne varmesystemer har et stort potentiale i forhold til fleksibel kontrol af indeklima og derved reducering af energiforbruget. Imidlertid er der en række eksisterende begrænsninger, som hovedsagligt er relateret til de ønskede termiske forhold på rumniveau i en bygning, der hindrer vellykket implementering og drift af sådanne varmesystemer i europæiske boliger.

Formålet med dette phd-studie er at udvikle et nyt kombineret ventilations- og luftvarmesystem til boliger, som kaldes *heat valve ventilation (HVV) system* og at demonstrere systemets evne til at forbedre regulering af termiske forhold på rumniveau.

Denne afhandling præsenterer udviklingen, analysen og evaluering af ydeevnen af det foreslåede system. Systemet er undersøgt både i laboratoriemiljø og i en bolig i drift. Derudover er der udført numeriske simuleringer for at evaluere ydeevnen for to typer af tilluftsventiler beregnet til anvendelse i det nyudviklede HVV-system.

For at muliggøre regulering af lufttemperaturer på rumniveau er der brugt en luftfordelingsboks, kaldet en manifold, som indeholder et centralt opvarmningsbatteri og varmeventiler i systemet. Ved hjælp af varmeventilerne kan lufttemperaturen reguleres separat i hver af forsyningskanalerne i et område mellem 25° C og 52° C. Eksperimentelle resultater demonstrerer, at HVV-systemet er i stand til uafhængigt at regulere temperaturen af den luft, der tilføres til forskellige rum på grund af den konstante tilførselsluftmængde. På denne måde kan der opfyldes forskellige behov for lufttemperatur i forskellige rum.

Hvad angår målet om et minimalt energiforbrug viser resultaterne, at den samlede effekt, der bruges af systemet, kan minimeres baseret på de operationelle parametre. Vedrørende undersøgelsen i enfamiliehuset, der blev simuleret i eksperimentet, skete den minimale effektanvendelse af systemet ved indblæsningstemperaturen på 52° C og tilluftstrømning på 56 l / s.

Med hensyn til ydeevnen, der er kvantificeret i forhold til de tilvejebragte termiske betingelser, viser CFD-simuleringerne, at HVV-systemet kan undgå temperaturstratificering og træk inden i opholdszonen for det brede område af grænseforholdene.

Med hensyn til anvendelse af systemet, i den undersøgte bolig, er den primære begrænsning af HVV-systemets kapacitet afgrænset af både energieffektiviteten og tilvejebringelsen af de ønskede termiske forhold indendørs. Derudover er opretholdelsen af den ønskede lufttemperatur på rumniveau afhængig af de indvendige døråbningsmønstre.

Keywords: HVAC system, energibesparelse, termisk komfort, mekanisk ventilation, luftopvarmning, ventiler

ACKNOWLEDGEMENTS

The research work presented in the thesis was carried out at the Department of the Built Environment, Aalborg University, within a period February 2016 to March 2020. The experimental work in the research project was performed at Indoor Climate Solutions Competence Centre, Lindab. The financial support was provided by the Danish Social Housing Sector, the Danish Landowners’ Investment Association, and the Faculty of Engineering and Science at Aalborg University, and they are gratefully acknowledged for funding the research project.

My first sincere appreciation goes to my supervisor Professor Alireza Afshari, who provided the opportunity to me to pursue a doctoral degree. His guidance and unlimited support truly helped me during the course of my doctoral research and writing of the thesis.

I express my gratitude and thanks to my co-supervisor Senior Researcher Niels Christian Bergsøe for his continuous assistance and plenty of perceptive comments. I also would like thank my colleagues at the Department of the Built Environment for the pleasant working environment during the years of my PhD.

Special thanks must go to Göran Hultmark for all encouragement and unsparing help, and for bringing the opportunity to perform the experimental work at Lindab. Also, I deeply thank Dr Klemen Rupnik, who helped me a lot in my research, for his comprehensive advises and enthusiastic assistance. Furthermore, I would like to thank all the wonderful colleagues at Lindab for being so supportive and friendly to me.

I would like to thank Docent Sasan Sadrizadeh for providing me the opportunity to visit the division of Sustainable Buildings at KTH Royal Institute of Technology and for the fruitful collaboration. Furthermore, I would like to thank the colleagues from the division for the welcoming and stimulating environment, and for making my stay in Stockholm a positive and valuable experience.

I would like to thank every person who supported and encouraged me in this journey, especially my good friends.

I would like to dedicate the thesis to my parents, Beata and Leszek, and to my beloved sister Monika. For being the compass that guides me, for being the inspiration to reach ambitious targets, and for supporting me in any difficulties – my sincerest thank you.

Copenhagen, March 2020

Joanna Polak

NOMENCLATURE

ABBREVIATIONS

ACH	Air change per hour
ATD	Air terminal device
CAV	Constant air volume
CFD	Computational fluid dynamic
COP	Coefficient of performance
CO_2	Carbon dioxide
DCV	Demand controlled ventilation
EPBD	Energy performance of buildings directive
GHGs	Greenhouse gases
HVAC	Heating, ventilation and air conditioning
HVV	Heat valve ventilation
ISO	International organization for stansardization
PI	Proportional-integral control
SPF	Specific fan power
U-value	Heat transmittance coefficient
VAV	Variable air volume
ZEB	Zero energy building

LATIN LETTERS

$c_{p,a}$	Specific heat capacity of air [J/(kg·K)]
$c_{p,w}$	Specific heat capacity of water [J/(kg·K)]
k	Coverage factor
P	Heating power provided by heating system [W]
q_v	Volumetric flow rate [m^3/s]
h_{inlet_air}	Specific enthalpy of air at the inlet to the manifold [J/kg]
h_{outlet_air}	Specific enthalpy of air at the outlet to the manifold [J/kg]

h_{inlet_water}	Specific enthalpy of water at the inlet to the heating coil [J/kg]
h_{outlet_water}	Specific enthalpy of water at the outlet to the heating coil [J/kg]
P_{fan}	Energy used by the fan [W]
P_{heat_source}	Energy used by the heat source [W]
P_i	Heating power delivered to i-th room [W]
P_{system}	Total energy used by the HVV system [W]
$\dot{Q}$	Actual heat transfer rate [W]
$\dot{Q}_{air}$	Air heat transfer rate [W]
$\dot{Q}_{max}$	Maximum possible heat transfer rate [W]
$\dot{Q}_{water}$	Water heat transfer rate [W]
$\dot{q}_{m\_air,j}$	Air mass flow rate in j-th supply duct [kg/s]
$\dot{q}_{m\_water}$	Water mass flow rate [kg/s]
$\dot{q}_v$	Total airflow rate delivered by the fan [m^3/s]
$\dot{q}_{v,i}$	Airflow rate supplied to i-th room [m^3/s]
$\dot{q}_{v\_water}$	Water volumetric flow rate [m^3/s]
T_C	Temperature of cold reservoir [K]
T_H	Temperature of hot reservoir [K]
T_{inlet_water}	Temperature of water at the inlet to the heating coil [°C]
T_{outlet_water}	Temperature of water at the outlet to the heating coil [°C]
$T_{room,i}$	Temperature of air in i-th room [°C]
$T_{supply\_air,\ i}$	Temperature of supply air to i-th room [°C]

GREAK LETTERS

Δp	Total pressure increase in the fan [Pa]
ΔT	Temperature difference [°C]
ε	Effectiveness [-]
η	Fan efficiency [-]
ρ_a	Density of air [kg/m^3]
ρ_w	Density of water [kg/m^3]

TABLE OF CONTENTS

TABLE OF FIGURES

CHAPTER 1. INTRODUCTION

The introductory chapter presents the motivation underpinning the choice of research approach and design, along with the description of the research problems, the objectives of the study, the significance and the limitations of the study, the outline of the thesis and the list of papers published in relation to the study.

1.1. MOTIVATION

Buildings and climate: status and potential

Different sources estimate greenhouse gases (GHGs) emissions from the building sector to represent approximately 30% of emissions worldwide, according to the United Nations Environment Programme [1]. The International Energy Agency also estimates that buildings are the largest energy-consuming sector in the world, accounting for over one-third of total final energy use, and equally significant source of carbon dioxide (CO_2) emissions [2].

Recognizing the importance of energy efficiency improvements in the buildings sector, the European Union introduced the Energy Performance of Buildings Directive (EPBD) in 2002. The EPBD required all member states to improve building regulations by establishing minimum energy performance requirements in buildings. In 2010, the EPBD was revised with tougher requirements for buildings, including the requirement for member states to ensure that all new buildings will be nearly zero-energy buildings (ZEBs) by the end of 2020 [3]. According to a definition "a (net) zero-energy building (ZEB) produces enough renewable energy to meet its own annual energy consumption requirements, thereby reducing the use of non-renewable energy in the building sector" [4].

Achieving sustainable buildings: the role of HVAC systems

Buildings are crucial to a sustainable future as their design, construction, operation, and the activities in buildings are substantial contributors to energy-related sustainability challenges [5]. Nonetheless, there are several criteria, beside the energy use of the building, that need to be fulfilled in order for the building to be sustainable. It should be noted that even a zero-energy building can be unsustainable, if it does not provide a healthy and comfortable indoor environment for its users [6].

Buildings are primarily constructed to produce indoor environments, in which their occupants are comfortable, healthy, safe and productive. A complex combination of building engineering systems is used to achieve this purpose. The purpose of heating, ventilation and air-conditioning (HVAC) systems is to maintain the desired indoor thermal environment and air quality in various spaces of a building. Built environment is the results of a complex interplay between the building envelope and structure, the activities inside the building spaces, the outdoor climate, and the technical systems that are to provide the required conditions indoors [7]. That interplay is, as well, closely connected to the energy flows and conversions in the building. In fact, HVAC systems are found to be the most energy intense building services, representing approximately half of the final energy use in the building sector [8, 9].

Many of the essential sustainable building attributes, such as energy, resources, pollutions, noise, health and well-being of occupants, are directly or indirectly affected by the performance of the HVAC systems. Inadequately designed or malfunctioning HVAC systems can cause unnecessary use of energy and adverse effects on the indoor environment, such as presence of noise, draught, contaminants in indoor air or elevated relative humidity indoors [10].

HVAC systems: challenges and current advances

The occupant comfort and optimal management of the energy use play a crucial role in the research and development for emerging HVAC technologies. Energy demands as well as indoor environment requirements in the buildings can vary significantly from country to country depending on a number of factors, such as climate, economic development and building regulations [11].

A large share of old, energy-inefficient buildings located in regions where there is significant need for space heating, characterizes the buildings sector in Europe. The residential stock is the largest segment of the European building stock and constitutes 75% of the entire floor area [12]. Space heating is the most energy intense end-use in European residential buildings and accounts for around 70% of the total final energy use [2]. As a result, most of energy in buildings in Europe is used for heating and the largest potential for energy savings is through improvements in building envelopes and heating equipment. Building envelope improvements, including better wall insulation and highly insulated windows, offer significant potential energy savings and contribute to the reduction in the demand on space heating [13]. Nonetheless, in order to sufficiently reduce the overall energy intensities of buildings and to support the shift towards more sustainable built environment, it is crucial that the HVAC technologies evolve to address new challenges [14].

Looking at the past developments in the building stock, extensive changes have occurred in recent decades in the way the buildings are designed, constructed and used [15-22]. These emerging challenges comprise aspects, such as integration of the HVAC systems with other building services, for instance the demand management system [17, 18], renewable energy system and thermal storage system [19], increasing the flexibility in control of the indoor environments [20], and improving the response to the heating, cooling and ventilation needs [21]. Technical solutions that facilitate intelligent building control, such as smart control equipment and use of wireless sensor networks are becoming crucial for the future generation of buildings [22].

One way to address the challenge regarding enhancing the occupants' comfort as well as improving the response of an HVAC system to the building heating, cooling and ventilation demands is to provide the individual control of thermal conditions in each of the rooms served by the system. Such solution may better match occupants' preferences regarding thermal conditions, while avoiding unnecessary conditioning of unoccupied spaces.

The aforementioned challenges should be achievable without excessive increase of the HVAC systems complexity. Moreover, the novel HVAC technological solutions should enhance simple installation, maintenance and operation, and be cost-effective and time-efficient in order to attract all stakeholders. These emerging challenges and trends create a compelling need to provide a new and innovative HVAC concepts and technologies to be implemented in the built environment for advancing sustainable buildings and achieving the ambitious energy targets.

1.2. OBJECTIVE

The objective of this PhD study is to develop a new combined ventilation and heating system for residential buildings and to demonstrate the ability of the system to improve the control of thermal conditions in individual rooms in a building.

In order to achieve the research objective, the following tasks were accomplished in the PhD project:

- To design and develop the new ventilation and heating system and to examine the ability of the system to control the air temperature on a room level.
- To analyse the performance of the proposed system, in particular in terms of the energy use and the provision of the desirable indoor thermal conditions.
- To investigate the applicability of the systems in a real-life building through the assessment of the measured and perceived local thermal comfort.

1.3. LIMITATIONS

The focus in the thesis is on the control of indoor air temperature only. The control of other parameters relevant to the indoor climate, such as indoor air quality and humidity is not considered in the research work. Neither does the work include the aspects of filtration and cleaning of the supply air.

The study is limited to ventilation and heating. Cooling is not included in the scope of the thesis. The HVV system control strategy was limited to a constant air volume (CAV) system. Nonetheless, some considerations on a variable air volume (VAV) control strategies are given.

The present work has been undertaken by means of laboratory measurements and numerical investigations. In addition, the proposed heat valve ventilation (HVV) system was integrated in an existing building in order to study the measured and perceived local thermal comfort. The local thermal comfort is evaluated in terms of air temperature and air velocity patterns and local thermal discomfort due to draught.

With regard to the experimental work, a number of assumptions were made. Namely, the analysis in the experiment was focused on the Danish climate conditions. Thus, the assumptions in relation to the heat transmittance coefficients of building envelope and the calculated heating demands were made based on the Danish Building Regulations [23]. In addition, the building thermal mass and air passage through the building envelop were not addressed in the experiments.

Furthermore, since the HVV system has been operating in the real-life building for a short time, at the time of writing the thesis, the collection of representative data on the energy use of the HVV system was not possible.

This work focus on technical solutions and design, and does not further treats the subjects of economics.

1.4. OUTLINE OF THE THESIS

Chapter 1 presents the motivation for the choice of research approach and design, along with the description of the research problems, the objectives of the study, the significance of the study, the limitation of the study, the outline of the thesis and the list of papers published in relation to the study.

Chapter 2 provides the literature review on the existing heating and ventilation technology for application in residential buildings. In addition, this chapter presents the aspects of designing combined mechanical ventilation and air heating systems.

Chapter 3 presents in details the concept and the design of the proposed heat valve ventilation (HVV) system. Further, the prototype of the HVV system built in the laboratory hall is described in the chapter.

Chapter 4 contains the information on the research methodology applied in the study.

Chapter 5 includes the presentation and discussion of the results revealed from the analysis of the HVV system.

Chapter 6 presents conclusions outlined from the research study and recommendations for the future research directions.

1.5. LIST OF PUBLICATIONS

Following publications authored by Joanna Polak are part of the PhD project but are not part of the thesis:

I. **Paper I.** J. Polak, A. Afshari, N. C. Bergsøe, G. Hultmark, *Demand control on room level of the supply air temperature in an air heating and ventilation system*, In Proceedings of the International Conference on Indoor Air Quality and Climate, Healthy Buildings 2017, July 2-5, 2017 Lublin, Poland

II. **Paper II.** J. Polak, K. Rupnik, A. Afshari, N. C. Bergsøe, G. Hultmark, *Development on a novel temperature-based demand controlled ventilation system for residential buildings*, In Proceedings of the 15th International Conference on Indoor Air Quality and Climate, Indoor Air 2018, July, 2018, Philadelphia, USA

III. **Paper III.** J. Polak, A. Afshari, P. Sadeghian, C. Wang, S. Sadrizadeh, *Improving the performance of heat valve ventilation system: A study on the provided thermal environment*, Building and Environment 164 (2019) 106338

IV. **Paper IV.** P. Sadeghian, J. Polak, A. Afshari, S. Sadrizadeh, *Numerical investigation on the impact of different supply air terminal devices on the performance of the newly combined ventilation and heating system*, In Proceeding of the 10th International Conference on Indoor Air Quality, Ventilation and Energy Conservation in Buildings, September 5-7, 2019, Bari, Italy

CHAPTER 2. BACKGROUND AND STATE-OF-THE-ART

In order to mark out the limits of the existing HVAC solutions and to identify the main research gaps in the research context, a literature review on the existing heating and ventilation technology for application in residential buildings was performed. The focus is on the technology applied in cold European climates, with significant demand for heating during the cold season. Further, this chapter presents design aspects of a combined mechanical ventilation and air heating systems.

2.1. VENTILATING AND HEATING OF RESIDENTIAL BUILDINGS

Ventilation systems

Ventilation refers to a process of intentional exchange of air between the spaces inside a building and the outdoor environment [24]. The primary role of ventilation is to assists in maintaining acceptable indoor air quality by diluting and displacing airborne pollutants caused by unavoidable sources. Besides, ventilation supports maintaining satisfactory thermal comfort indoors. Ventilation can be given a secondary tasks, i.e. heating and cooling [25].

For the past decades, European residential buildings were mainly ventilated by means of natural ventilation, i.e. a process where airflows through a building are driven by pressure and buoyancy forces [26]. When using natural ventilation, the control of the air change rate in a building is limited. In fact, natural ventilation is the most uncontrollable form of ventilation [27]. Furthermore, natural ventilation does not allow for conditioning and filtrating of the supply air as well as for the heat recovery from the exhaust air [28]. In cold European climates (e.g. Scandinavian countries), significant heating requirements have led to a rigorous approach to building air tightness and, consequently, to a need for ventilation systems that offer sufficient control of air change rates [29]. In such highly-insulated residential buildings, natural ventilation may be unable to provide the adequate air change rate and the use of mechanical ventilation becomes necessary. Recent revisions in building regulations had led to a nearly exclusive implementation of mechanical ventilation in new as well as in energy retrofitted residential buildings in northern Europe [23, 30-33].

Mechanical ventilation systems have proven to be suitable for the provision of a steady and sufficient air change rate and able to respond to the varying requirements of occupants and pollutant loads [34-37]. Moreover, mechanical ventilation systems may incorporate exhaust air heat recovery techniques to reduce ventilation losses [35, 38-41]. Hamid et al. [41] have shown that a mechanical ventilation with heat recovery system does contribute to significant improvement of the buildings' energy efficiency as well as the perceived indoor environmental quality.

Different configurations of mechanical ventilation systems are applied in residential buildings and these are: mechanical supply ventilation, mechanical extract ventilation and balanced mechanical ventilation. Mechanical supply ventilation is not recommended for residential buildings in cold climates since it can force humid air generated by household activities, into the building fabric where condensation may occur. Contrarily, a slight under-pressure is preferred in residential buildings to prevent moisture from penetrating into the building fabric. Mechanical extract systems are widely used; however, in highly-airtight buildings, there may be insufficient inlet air paths, which can result in the rise of the pressure in the ventilation system, and hence, the increase of electrical energy use. Another disadvantage of such configuration is that the supply air cannot be filtered and conditioned. Balanced mechanical ventilation systems combine extract and supply systems as separately ducted networks. Balanced mechanical ventilation systems do not significantly affect the pressure of the indoor space with respect to outdoors and are not resistant to infiltration driven by wind and temperature effects. As a consequence, the building must be highly-insulated for achievement of an optimum performance [25].

Regarding the control strategy, mechanical ventilation systems are classified into constant air volume (CAV) or variable air volume (VAV). In advanced, balanced mechanical ventilation systems, the parameters of the supply air can be controlled according to the current demand; such systems are called demand controlled ventilation (DCV) systems. The 'demand' can refer to criteria related to indoor air quality and thermal comfort [42]. DCV systems have proven a significant potential for energy savings and for maintenance of satisfactory indoor air quality in residential buildings [43-45]. In terms of design, VAV systems are more complex than CAV systems since these systems add the demand on components such as air terminal devices, dampers, fans and sensors. Increased complexity in mechanical ventilation systems results in the increased installation cost and the maintenance needs. Thus, for such advanced systems to be considered there must be a significant reduction in energy use or other added value that makes the increased cost and complexity worthwhile. Typically, VAV is a solution suitable when the demand for airflow varies greatly without being constant for any long periods of time.

Hydronic heating systems

In residential buildings, heat is most commonly distributed by hydronic or air heating systems. Hydronic heating systems dominate in central and northern European climates. The main advantage of using water as a heat transfer medium is the thermal capacity of water, due to which the hydronic heating system require less working fluid comparing with air heating systems.

In hydronic heating systems, radiators, convectors and floor heating systems are utilized for the supply of heat to individual rooms. Radiators supply heat through a combination of radiation and convection, depending upon the type of radiator. The majority of the systems that utilizes radiators are sized with high supply water temperatures of 70-90°C. Nevertheless, during the last years the trend is to size the systems for lower temperatures, in order to reduce heat losses and to take advantage of the higher efficiencies of low temperature heating [46].

Floor heating systems have been highly regarded for comfort, and thus have become increasingly popular in residential buildings in the past years [47-50]. In such systems, the heat transfer takes place mainly by thermal radiation. Floor heating systems do not require a water temperature as high as radiators or convectors, the supply water temperatures is of about 30-40°C. This is an advantage when the building is provided with the heat from a heat pump or a similar heat source, which performance is strongly dependent on a relatively low supply water temperature. The relatively large thermal mass of floor heating, comparing to other heating systems, enables exploitation of the floors and slabs to store energy, and thus reduce the size of the heating equipment needed to meet the peak loads [49, 50]. On the other hand, the thermal inertia of a heating system influences the control of the room air temperatures. Switching on and off a floor heating system takes a relatively long time to cool down and heat up the spaces in a building. The controllability of a floor heating system should be carefully addressed, especially in highly-insulated buildings, in which the changes in internal loads (due to occupants, lighting, and equipment) or direct sunlight have a higher impact on the fluctuations of room air temperature than in buildings with standard insulation. For example, Knudsen [36] reported that due to the thermal inertia, the floor heating system was perceived slow in response to the dynamics of the indoor environment and therefore difficult to control. Besides, hydronic heating systems can only provide heating, meaning that a separate ventilation system is always required, which in turn increases expense and space requirements. Figure 2-1 presents a schematic diagram of a mechanical ventilation system combined with a hydronic heating system.

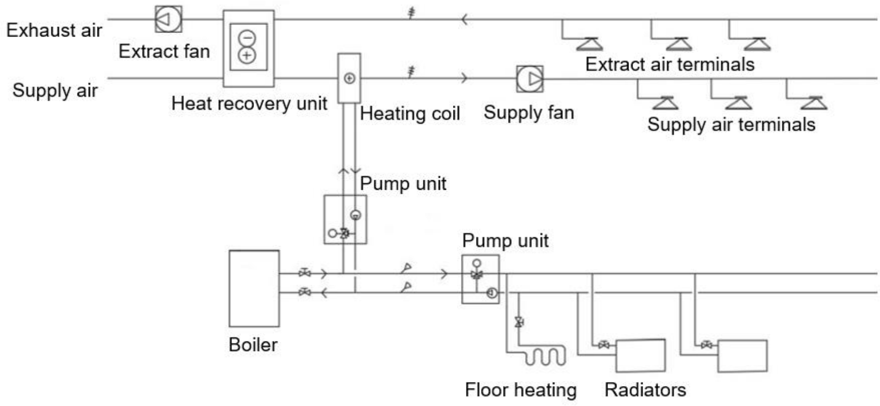

Figure 2-1. A scheme of mechanical ventilation with heat recovery combined with a hydronic heating system [53]. For the purpose of this work the original text in the Figure was translated to English.

Air heating systems

Air heating systems use air as a heat transfer medium. In such systems, a central air heater (e.g. gas furnace, electric furnace, a heat pump or a hydronic coil), equipped with a circulation fan and a heat exchanger, serves as a heat source. The heated air is distributed from the heat source to the rooms, which the system is designed to heat, by the means of ducts, plenums and air terminals. The air is returned from the heated spaces to the heater via a ceiling return grilles and a return duct. Air heating, also referred as to forced-air heating, are widely used in North America. In cold climates with significant need for heating, air heating systems require constant operation of the fan at a relatively large airflow rate to carry the heat from the heat source to the rooms [10]. This is the main reason for a limited implementation of air heating systems in northern Europe. In the literature, such heating systems has previously been linked to concerns in regard to local thermal discomfort due to the vertical air temperature difference [51], as well as increased air distribution duct heat losses [52].

Combining mechanical ventilation system and air heating system

In cold climates, where buildings need heating, the requirement for an air heating system is that the supply fan needs to operate constantly along to the heating season to provide heat. In residential buildings, on the other hand, outdoor air should be constantly supplied, at least at a minimum quantity, to maintain the indoor air quality.

In residential buildings with low heating energy demand, a supply of the required heating energy can be achieved at an airflow rate of a magnitude similar to an airflow rate required for indoor air quality purposes. The greater the reduction in space heating requirement is, the closer the two airflow rates become. Integrating the heating and ventilation into the same distribution system becomes economical from the perspective of both initial and operating costs as the separate space-heating distribution system in not required [53].

Air heating can be integrated in a mechanical ventilation system by post-heating of the supply air by the means of a heating coil. A ventilation system can contribute to space-heating from 0%, when no heating coil is installed in the ventilation system, to some air heating for comfort reasons, to 100% in case no supplementary heating source in installed and the space-heating demands are covered solely by a ventilation system.

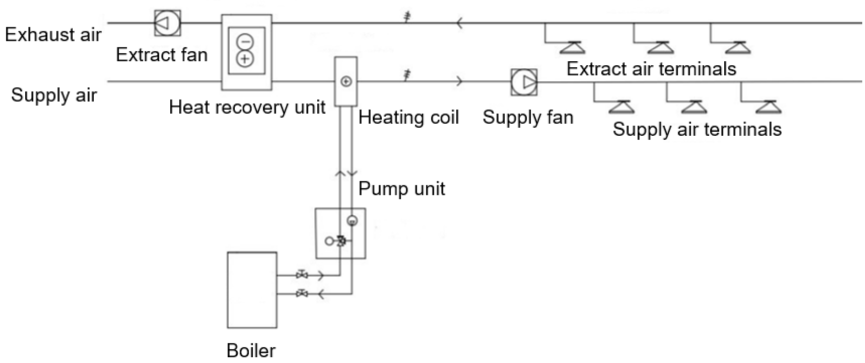

Figure 2-2. A scheme of mechanical ventilation with heat recovery combined with an air heating system [53]. For the purpose of this work the original text in the Figure was translated to English.

Figure 2-2 presents a schematic diagram of a mechanical ventilation system combined with air heating. This solution appears especially suitable for low-energy residential buildings, in which thermal comfort can be achieved solely by heating of the supply air, which is delivered at the minimum air change rates required to maintain a satisfactory indoor air quality. The substantial reduction in the space-heating demands and the presence of a balanced mechanical ventilation system make the use of an air heating system an efficient and economically viable option, in particular, in terms of initial cost because the separate heat distribution system is not needed (Figure 2-2). High performance of the building envelope is crucial for achieving satisfactory thermal conditions while using combined ventilation and heating systems. In highly-insulated buildings, the surface temperatures of outer walls and windows are close enough to the room air temperature to allow for thermal comfort without the need for the compensation of radiative asymmetries and cold air downdraught.

Feist et al. [54] discussed the feasibility of implementation of such a system in residential passive houses in moderate European climate. In the study, the aspects of design, i.e. heating power and heating load, heat distribution between the conditioned rooms, heat losses from ducts, and duct insulation were discussed. Further, the supply air distribution in the room in relation to the provided thermal comfort was addressed. The results showed the desired room air temperature, as well as the satisfactory thermal conditions with regard to a draught risk, could be maintained when using a combined ventilation and air heating system.

The feasibility of the combined mechanical ventilation and air heating in residential buildings was further considered in respect to cold climate conditions by Georges et al. [55]. The authors investigated the challenges in terms of thermal dynamics: the magnitude of the supply air temperature needed, the temperature difference between rooms, the impact of internal gains, the influence of thermal losses from ventilation ducts and the control aspects. Based on the results, the authors pointed out the limitations related an air heating system with a centralized heating coil as well as provided the guidelines for a consistent design of air heating in cold climates.

Both of the aforementioned studies emphasised that the main limiting factors for combined ventilation and heating systems in relation to the designing, application and performance are the supply air volume and the supply air temperature. In combined ventilation and heating systems, the control of room temperature can be achieved in two ways: through supplying a constant airflow with varying temperature (CAV systems) or through varying the supply airflow with a constant temperature (VAV systems). In practice, most CAV air heating systems are based on a centralized heating coil, and thereby supply the air at nearly uniform temperature to all of the served rooms. The temperature of the air measured in the main extract duct is used as the reference temperature for the heating coil output control. Such a control strategy provides little or no possibility to the system to suit various demands in the individual rooms. In a VAV system, the supply air is centrally treated to a suitable temperature and the supply airflow rate is typically varied by the means of a VAV box with a damper just upstream of the supply air terminal. Practically, CAV systems are more commonly applied than the VAV systems, as a VAV control could demand a relatively high airflow rates.

The lack of air temperature control on room level, in a consequence, the limitation in the provision of the desired thermal conditions in individual rooms in a building impose a great limitation to CAV air heating systems. One way to enable air temperature control on room level in CAV air heating systems is that the supplied air is heated in two-stages, first centrally and then locally in each zone by the means of reheat coil located upstream of the air supply terminal. This solution, however, is impracticable considering the maintenance and installation cost points of view.

Air temperature control on room level in combined ventilation and heating systems

Recently, an increased attention has been given specifically to the indoor thermal conditions in individual rooms in residential buildings [59-60]. Berge et al [56] presented a comprehensive literature review on the indoor air quality, thermal comfort and general experiences with the operation of combined mechanical ventilation and air heating systems in comparison to other heating strategies. In regards to the provision of the desired thermal conditions in individual rooms, the results indicated that occupants perceived relatively high temperatures in bedrooms and missed the possibility to adjust room air temperature individually in each room. Further research regarding the control of indoor air temperature in individual rooms exhibited a consistent feedback that an adjustability of supply air temperature on a room level is wanted [57-59]. Demands regarding different indoor air temperatures in individual rooms were identified, i.e. lower air temperature were desired in bedrooms in comparison to living rooms and bathrooms. Moreover, the results showed that the dissatisfaction with the indoor thermal conditions in individual rooms in residential buildings is a dominant driver of substantial window opening, which in turn, contributes to the increased space-heating demands [59]. Combined ventilation and heating systems should therefore be designed with the possibility to adjust at least the bedroom temperature independently from the temperature in other rooms.

In this respect, Berge et al. [59] proposed a two-zoned ventilation concept with two distinct supply air temperatures for the living areas and bedrooms. This strategy managed to reduce the influence of window opening but the increase in space-heating need was still substantial. Most recently, Georges et al. [60] investigated alternative ventilation strategies to reduce the impact of the supply air temperature on the increased space-heating needs. Among the alternative strategies, only decentralized ventilation provided that the supply air temperature could be adjusted specifically to individual rooms.

In this context, enabling the air temperature control on room level demonstrate a clear potential for contributing to the level of satisfaction with indoor thermal conditions in individual rooms as well as for the improvement of the energy performance in a residential buildings.

2.2. ON THE ASPECTS OF DESIGNING COMBINED VENTILATION AND HEATING SYSTEMS

Ventilation is regarded as a primary function in combined ventilation and heating systems. The ventilation rates are determined according to indoor air quality requirements.

An air heater may consist of a hot water coil connected to a heat storage or to a heat source. In this case, many different energy sources can be used. Direct electric heating is also possible, giving great flexibility concerning position and subdivision of the air heater. However, direct electric heating normally is associated with a high primary energy demand [54].

The total energy use of a combined ventilation and air heating system will depend on the efficiencies of the components, such as a specific fan power (SFP) value, a temperature efficiency of the heat recovery unit and a heat pump coefficient of performance (COP), in case a heat pump is used as a heat source in the system [10]. The efficiency of the components are influenced by the operational parameters of a combined ventilation and air heating system. Therefore, the operational parameters should be carefully addressed in designing combined mechanical ventilation and air heating systems. For example, the temperatures of the heat transfer fluids influence the efficiency and performance of a heat pump. Ultimately, due to Carnot efficiency limits, the heat pump's performance will decrease as the temperature difference between the hot reservoir (e.g. the water in the heating coil) and the cold reservoir (e.g. the outdoor air) increases. Therefore, the COP of a heat pump serving as a heat source in the system can be maximized by choosing a heating system requiring a low supply water temperature (~35-55°C) and by choosing a heat source with a high average temperature (e.g. the ground). Another example is the supply and exhaust airflow rates, which influence the energy use of the fans in a combined ventilation and heating system. In case the system is a CAV system, it is reasonable to keep the airflow rates at the minimum.

The design of the supply air distribution system is another aspect, significantly influencing the performance of combined ventilation and air heating systems. When a ventilation system is operating in the heating mode, the temperature of the supply air introduced to the room is higher than the room air temperature. The non-isothermal jet is affected by thermal buoyancy caused by air density difference. The buoyant forces may stabilize the air in the room by stratification [24]. The stratification inhibits the ventilation effectiveness in the occupied zone and causes that the supplied air will only circulate in the upper part of the room [61-64]. This, in turn, will give rise to large temperature gradients, which may cause a local thermal discomfort due to the vertical air temperature difference.

The insulation of the supply air ducts determines the share of the heat transferred through the duct casing. Extra insulation ensures to supply sufficient heat to exposed supply rooms. Intentionally left out insulation of a supply air duct leading through another room (e.g. the bathroom) allows transfer some extra heat to that room [54].

CHAPTER 3. DEVELOPMENT OF HEAT VALVE VENTILATION SYSTEM

This chapter presents the details on the concept and design of the heat valve ventilation (HVV) system. Further, the prototype of the HVV system built in a laboratory hall is described in this chapter.

3.1. CONCEPT AND DESIGN

The concept of the present study is to develop a novel ventilation and heating solution for the provision of adequate air change rates and the improvement of thermal conditions in individual rooms within a building. To that end, an innovative heat valve ventilation (HVV) system was proposed. The HVV system is a combined balanced mechanical ventilation and air heating system, which covers the space-heating demands solely by delivering the heated air to the conditioned spaces. The novelty of the HVV system is its ability to control the air temperature individually in each room. Thus, the HVV system is able to cover various space-heating demands and maintain desired air temperatures in all rooms.

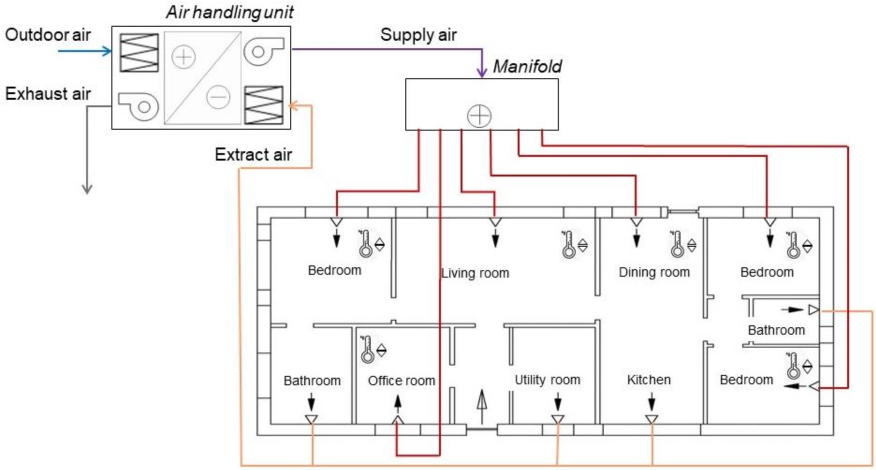

Figure 3-1. A scheme of heat valve ventilation (HVV) system.

The schematic diagram of the HVV system is presented in Figure 3-1. The HVV system consists of an air handling unit with basic elements, i.e. filters, supply and exhaust fans and a heat exchanger; a manifold with a built-in heating coil and heat valves; a supply and an extract ductwork.

The HVV system supplies the air at a constant airflow rate. Similar to a conventional CAV air heating system, the supply air is heated by the centralized heating coil. However, unlike to a conventional CAV air heating system, the use of the heat valves and separate supply ducts makes the room level air temperature control realizable without local reheating of the supply air, i.e. without the use of supplementary heating coils prior to each individual room. Furthermore, the room air temperature is controlled independently from the airflow rate as opposed to a VAV air heating system, in which the supply airflow rate is varied, by the means of a VAV box with a damper, to meet the heating load in the corresponding room. Thus, the concept of the HVV system allows the installation simplicity while providing the flexibility in control of the room air temperature.

Conditioning of the supply air takes place in two steps: first on a building level, and second on a room level. Namely, first the outdoor air passes through the air handling unit, where it is filtrated and heated in the heat recovery unit. Next, the air is distributed via the main supply duct to the manifold, where the room level conditioning takes place. The primary function of the manifold is to distribute the total supply airflow rate into separate ducts serving the corresponding rooms. A centralized heating coil is installed in the manifold to incorporate heating in the HVV system. The supply air is heated inside the manifold accordingly to meet the heating demands in individual rooms and then delivered to each room via the corresponding supply duct.

The conditioned outdoor air is supplied to habitable rooms, i.e. bedrooms, living and dining rooms, and the room air is extracted from moisture-polluted rooms, such as bathroom, kitchen, and utility room. In principle, this induces a flow pattern that inhibits the cross-contamination of air from moisture-polluted spaces to habitable spaces.

Manifold design

One of the main components of the HVV system is the manifold, i.e. an air distribution box from which a number of smaller ducts branch off. Figure 3-2 illustrates the design of the manifold. The manifold is split into two parts (see the vertical cross-section in Figure 3-2 c). The top part of the manifold contains the built-in heating coil and the bottom part is empty. Both parts are separated by the insulation layer in order to prevent the transfer of heat between the parts. The perforated plate is installed to ensure a uniform distribution of the air entering the manifold, prior to the air is heated and distributed to the particular supply ducts. Such a design ensures that the supply air can either pass through the heating coil or bypass it.

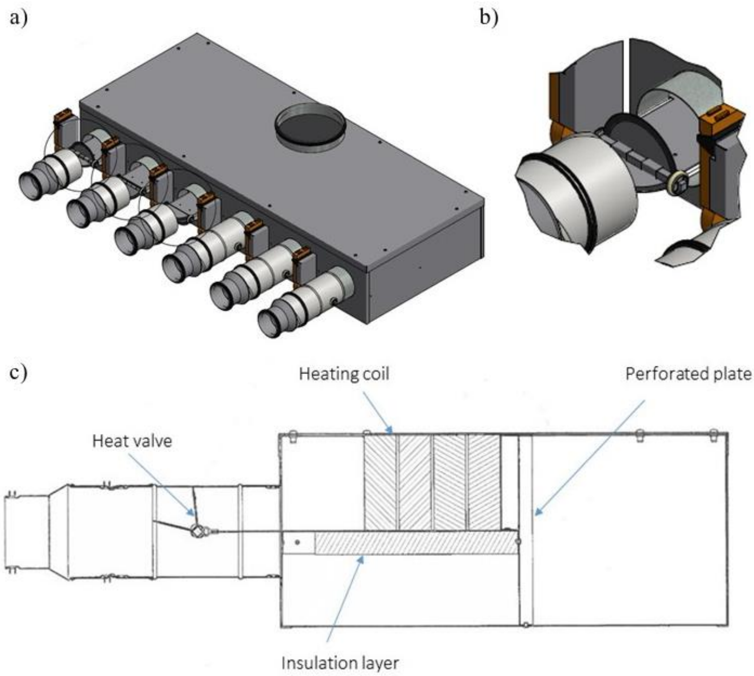

Figure 3-2. Illustration of the manifold – the 3D view (a), the heat valve detail (b) and the vertical cross-section (c).

The airflow pattern through the manifold is determined by the positions of the heat valves, which are installed at each of the outlets of the manifold. The detailed illustration of the heat valve is presented in Figure 3-2 b. The heat valve consist of two blades mounted on a rotary axle. The angle between the blades is fixed and of about 70°. The position of each heat valve determines the proportion of the air that either passes through or bypasses the heating coil. It should be noted, however, that the position of the heat valve does not have an influence on the total airflow rate in the particular supply ducts. The position of each heat valve can be adjusted from 0% to 100% open to heating, further referred as to position 0 and position 1, respectively. In case the heat valve is in position 0, the total amount of air distributed to the particular supply duct bypasses the heating coil. In case the heat valve is in position 1, the total amount of the air passes through the heating coil. In case the heat valve is adjusted between the positions 0 and 1, the air supplied to the particular duct is divided into two streams, heated and unheated, which are mixed together after passing through the heat valve. In this way the temperature of the supply air can be controlled independently at each of the outlets. Figure 3-3 illustrates the airflow pattern through the manifold in relation to the position of the heat valves.

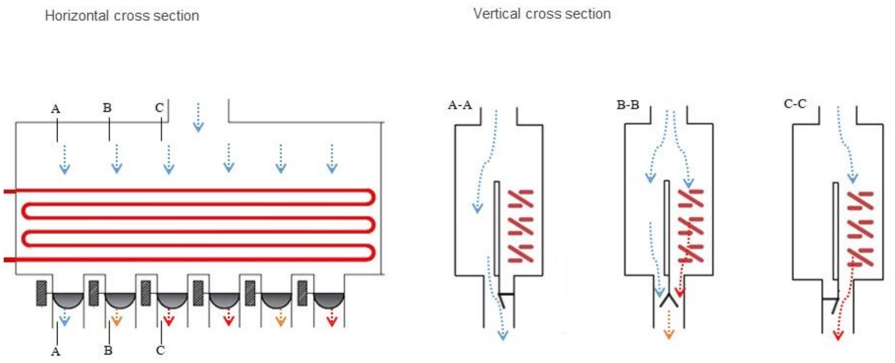

Figure 3-3. Illustration of the airflow pattern through the manifold.

Control

The provision of the adequate air change rate is the primary task of the HVV system. In residential buildings, the outdoor air should be constantly supplied, at least at the minimum quantity, required to maintain a satisfactory indoor air quality. According to the Danish Building Regulations [23], in residential buildings the supply of outdoor air must be present at any time at minimum 0.30 l/s per sq. metre heated floor area. The air should be extracted from kitchens, bathrooms and toilets at the minimum 20 l/s, 15 l/s and 10 l/s, respectively. In the HVV system, the supply and extract fans provide the necessary driving force to deliver the nominal airflow rate and the air distribution system is balanced to ensure the adequate airflow rate distribution to all rooms.

The heating power provided by the HVV system has to compensate for the transmission and infiltration losses, which occur in individual rooms, and the ventilation heat losses. The supply air capacity of transporting the heat to each room can be calculated as:

$$\boldsymbol{P_i = q_{v,i} \cdot \rho_{a,i} \cdot c_{p,a} \cdot (T_{supply\_air,i} - T_{room,i})} \qquad \textbf{(3.1)}$$

Where P_i (W) represents the heating power delivered to i-th room by the system, $q_{v,i}$ (m^3/s) is the airflow rate supplied to i-th room, $\rho_{a,i}$ (kg/m^3) is the supplied air density, $c_{p,a}$ (J/(kg·K)) is the air specific heat capacity, $T_{supply\_air,i}$ (K) is the supplied air temperature to i-th room and $T_{room,i}$ (K) is the air temperature in i-th room.

As the supply airflow rate ($q_{v,i}$) is constant, the control of the heating power delivered by the HVV system is achieved by regulation of the supply air temperature ($T_{supply\_air,i}$). The demand on heating power that has to be delivered to each room is determined by the actual air temperature in this room ($T_{room,i}$) and the heating temperature set point.

The position of each heat valve, and consequently the supply air temperature in the corresponding duct ($T_{supply\_air,\ i}$), is adjusted by an actuator, based on the signal from the corresponding room controller. The room controller constantly measures the current air temperature in the corresponding room and compares the measured value to the desired user-defined set points. Based on these two parameters the heating output is increased or decreased to meet the desired set points.

The air is supplied at the maximum temperature to cover the maximum heating load that occurs in the corresponding room. The maximum temperature of the supply air is limited by the heating capacity of the coil and the thermal loss occurring along the ductwork. The minimum supply air temperature is determined by the outdoor air temperature, extract air temperature and the efficiency of the heat recovery unit. The air is supplied at minimum temperature during sunny hours in winter, when the heat demand can completely be covered by solar and internal gains and the room air temperature is above or at the exact set point.

Supply air distribution system

The HVV system utilizes a radial duct system. The radial ducting air distribution system with the use of an air distribution box is commonly offered for application in residential buildings in Scandinavian countries [65, 66]. The supply ducts can be placed in attics and crawl spaces as well as they can be embedded in floor slabs or between floors of multi-storey structures. The radial ducting air distribution system offers benefits, such us providing acoustic separation between different rooms in a building as well as a relatively even distribution of the supply air as they eliminate multiple fittings, e.g. T-pieces and reducers, which affect the pressure. In addition, the installation of such systems is relatively simple comparing to other supply duct system types.

Duct thermal losses, which appear both from the manifold and along the ductwork, have to be taken into account in HVV system. One way to reduce the losses is to install the ductwork within building envelope. Alternatively, the system may be thermally insulated.

3.2. PROTOTYPE OF HVV SYSTEM

In order to analyse the performance of the proposed HVV system a prototype of the HVV system was constructed in a laboratory hall.

The HVV system prototype consisted of the following elements:

- The AHU HERU 100 S EC manufactured by Östberg [67], providing driving force to deliver the nominal airflow rate.
- The manifold with six outlets manufactured by Lindab.
- The electric water heater model 55 manufactured by Metro Therm [68], serving as a heat source.
- The supply air ductwork consisting of the main supply duct, connecting the AHU and the manifold, and six smaller ducts distributing the supply air between the manifold and the rooms.
- The extract ductwork removing the air from the rooms.
- The control system.

Figure 3-4 shows the constructed prototype of the HVV system.

In order to avoid undesirable thermal loss along the supply ductwork, the prototype of the HVV system was thermally insulated using 25 mm thick thermal insulation.

Simulated building

In order to demonstrate the ability of the HVV system to control individually the air temperature in different rooms, the prototype of the HVV system was connected to a number of compartments, simulating rooms in a residential building. A reference building consisting of ten rooms was considered. The compartments are further referred as to rooms. Each of the rooms measured 0.6 m length, 1.0 m width and 2.36 m height. Six of the rooms were connected to the supply ductwork of the HVV system to imitate habitable rooms in the building. These rooms are further referred as to habitable rooms. Four of the rooms were connected to the extract ductwork to imitate moisture-polluted rooms in the building. These rooms are further referred as to moisture-polluted rooms. For the ease of identification, the particular rooms were given the following names: bedrooms, living room, office, kitchen, bathrooms and utility room. To enable the airflow between the habitable and moisture-polluted rooms, the connections between the rooms were made. Besides, the rooms were sealed to minimize the influence of infiltration and exfiltration of air to the surroundings.

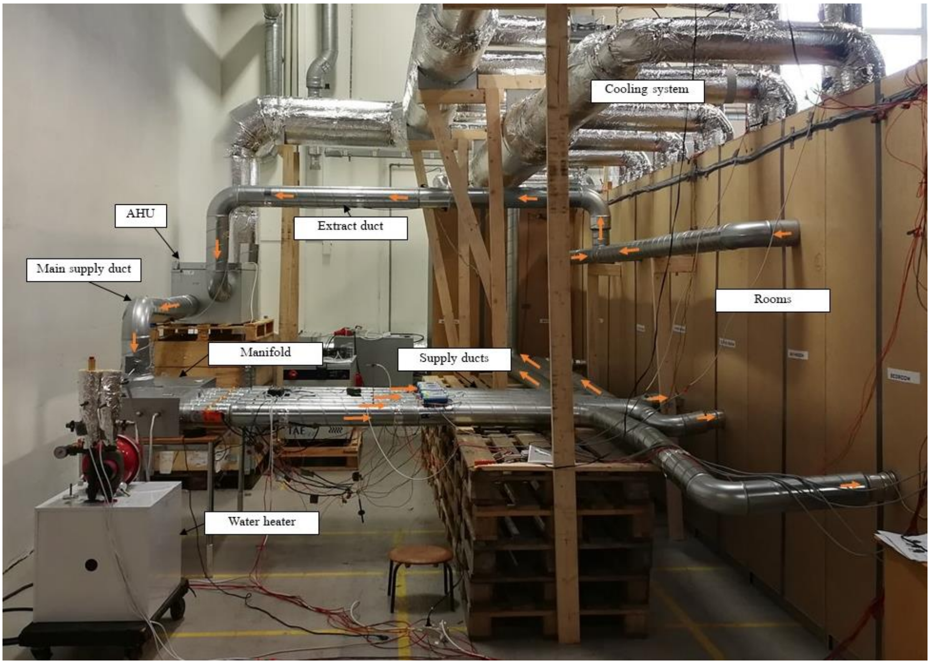

Figure 3-4. HVV system prototype in the laboratory hall.

Room air temperature control

A pre-programmed room temperature controller Regula Combi [69] with a built-in temperature sensor was installed in each of the rooms. Each of the room controllers, installed in the habitable rooms, was constantly modulating the heat valve actuator to satisfy the heating demand in the corresponding room. The reference value for the room controller was the room air temperature set point, while the measured value was the actual air temperature in the particular room. Since the HVV system did not deliver heating directly to the moisture-polluted rooms, the room controllers installed in these rooms were used to monitor the room air temperature only. The room air temperature in the control unit was maintained using PI control. The heat valve with actuator and the room controller are shown in Figure 3-5.

The temperature set points in all of the rooms were constantly set to the value of ambient air temperature measured in the laboratory hall. This was executed in order to minimize the uncontrollable heat transfer between the rooms and the surrounding laboratory hall.

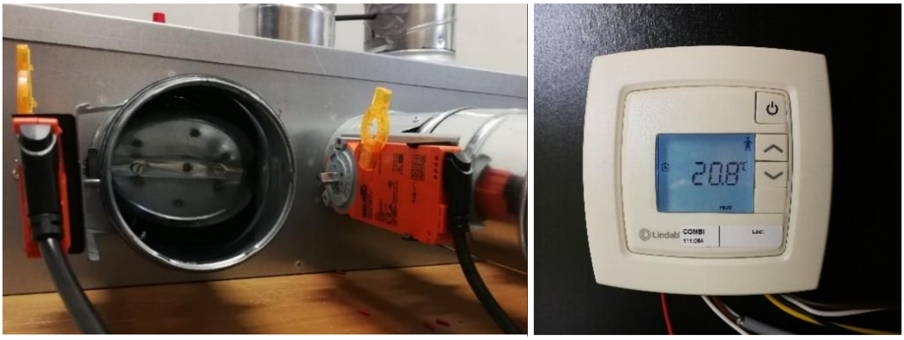

Figure 3-5. Heat valve with actuator to the left and room controller to the right.

Heating loads

To cause various heating demands in the rooms, an independent cooling system was integrated into the compartments simulating the rooms. The cooling system constituted a closed loop and consisted of a fan, a chiller, a heat exchanger and an air distribution ductwork. The air was constantly cooled down inside the heat exchanger and circulated inside the ductwork, which was leading through all of the rooms. Consequently, the rooms were cooled down as a result of the heat transfer occurring between the cold duct surface and the air inside the rooms. In order to control the amount of heat removed from the rooms via the cooling system, a flow regulator at the inlet to each room as well as temperature sensors at each inlet and outlet were installed. Figure 3-6 shows the connection of the cooling system to the rooms. Figure 3-7 illustrates a scheme of the complete experimental setup.

Figure 3-6. Cooling system integrated into the compartments in the laboratory hall.

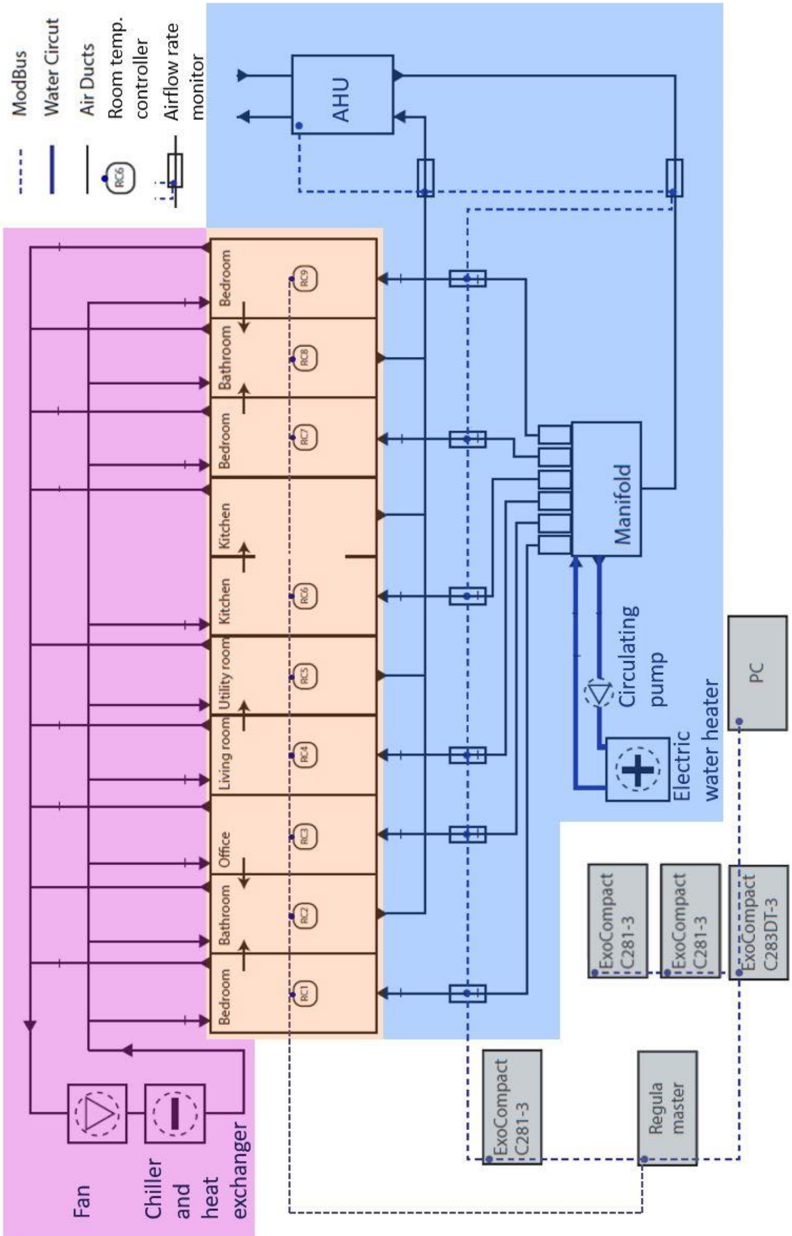

Figure 3-7. Schematic diagram of the experimental setup. Blue colour covers the area of the HVV system, pink colour covers the area of the cooling system, light orange colour covers the area of the buildings, and grey colour covers the area of the control system.

Residential Master program

A LabView based Residential Master computer program was developed for the purpose of control and data acquisition processes in the experiments. The measuring and control equipment installed in the HVV prototype were connected to a bus line enabling communication with the Residential master program via RS485. The interface of the Residential master program is presented in Figure 3-8. The measuring and control equipment is presented in Chapter 4.

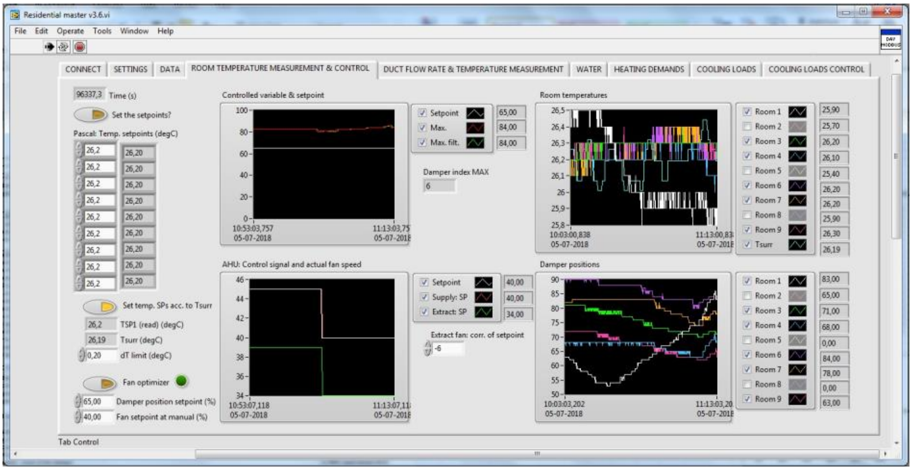

Figure 3-8. Residential Master computer program.

CHAPTER 4. RESEARCH METHODOLOGY

The research methodology in this thesis consists of laboratory experimental work, numerical simulations using Computational Fluid Dynamics (CFD) and real-life measurements in a single-family residential building.

4.1. LABORATORY EXPERIMENT

The laboratory experimental work consisted of two parts. The first part included performance measurement and analysis carried out on the prototype of the HVV system in a full-scale laboratory experiment. The second part included measurement and analysis of the provided thermal conditions by the HVV system integrated in an experimental chamber.

Part 1: The HVV system in the full-scale laboratory experiment

The experiment was performed with the aim to characterize the influence of the system design on the overall system performance and to identify possibilities and limitations for further implementation of the system in a building. This part includes the following:

a. Measuring of the supply air temperature, supply airflow rate, supply ductwork pressure losses and coil heat transfer effectiveness.
b. Measuring and analysis of the HVV system energy use.

Part 2: The HVV system integrated in the experimental chamber

The experiment was performed with the aim to characterize the influence of the supply air distribution system on the air distribution and thermal conditions in a room served by the HVV system. This part included performing experimental measurements to validate the numerical data. The measurements in this part included the following:

a. Measuring of the air velocity and air temperature patterns.
b. Measuring and analysis of the local thermal discomfort due to draught.

4.2. EXPERIMENTAL FACILITIES

The experimental facilities used in this study included the prototype of the HVV system and the experimental chamber.

Part 1: The HVV system in the full-scale laboratory experiment

The first part of the laboratory experiment on the system performance measurement and analysis was carried out on the prototype of the HVV system built in the laboratory. The details on the prototype of the HVV system are presented in Chapter 3, section 3.2.

Part 2: The HVV system integrated in the experimental chamber

The second part of the laboratory experiment regarding the thermal conditions provided by the HVV system was carried out in an experimental chamber representing a room in a residential building. The chamber measured 3.1 m (length) × 3.6 m (width) × 2.7 m (height) and consisted of a floor, a ceiling, three internal walls and one external wall with a window. The chamber was unoccupied, unfurnished and without any additional internal heat sources. The experimental chamber was located in a laboratory hall. The chamber walls were insulated and the temperature inside the hall was kept as close as possible to the temperature inside the chamber to minimize the heat transfer through the chamber walls and mimic the adiabatic condition. A cooling panel with a total area of 2.1 m^2 was located on one of the chamber walls to represent the window. The cooling output from the panel accounted for a total heat loss due to the transmission through the room envelope, which would occur in a real building during the coldest day in Copenhagen, Denmark. The experimental chamber was sealed in order to ensure the mass balance between the supply and the exhaust airflow rates. The geometry of the experimental chamber is shown in Figure 4-1.

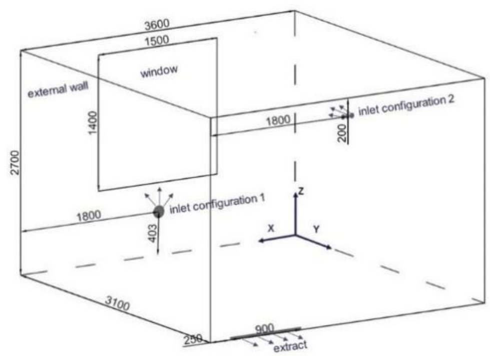

Figure 4-1. Experimental chamber geometry [70].

4.3. EXPERIMENTAL PROCEDURES AND EQUIPMENT

The experimental procedures and the measuring equipment used in the full-scale laboratory experiment as well as in the experiment performed in the chamber are described in the following sections.

Part 1: The HVV system in the full-scale laboratory experiment

In order to characterize the correlation between the heat valve position and supply air temperature in the corresponding supply duct, the supply air temperature was measured while changing the position of the heat valve from the position 0, i.e. fully closed to heating, to the position 1, i.e. fully open to heating. The heat valve position was changed in the following steps: 0°, 15°, 30°, 45°, 60°, 75°, and 90°, where 0° represents position 0, and 90° represents the position 1. The temperature of the supply air was measured inside the main supply duct (t_1), inside the top part of the manifold (t_2) and inside the corresponding supply duct (t_3). The air temperature was measured by the means of Pt1000/TG-K3 temperature sensors. The locations of the measuring points are presented in Figure 4-2.

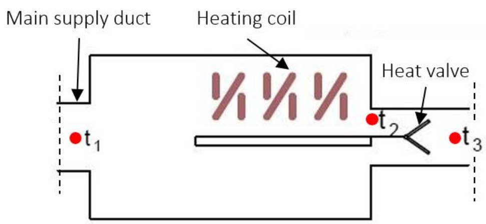

Figure 4-2. Location of the temperature sensors t_1, t_2 and t_3 presented in the vertical cross section of the manifold.

The static pressure measurements were performed, in order to identify the influence of the heat valve position on the total static pressure in the HVV system. The static pressure was measured by the means of Testo 400 measuring instrument. The static pressure was measured in the main duct (p_1), as well as in each of the supply ducts (p_2), while the position of each of the heat valves was adjusted as follows: 0°, 15°, 30°, 45°, 60°, 75°, and 90°. The placement of the pressure sensors in the ducts was chosen considering the fully developed air velocity profile in the duct. Therefore, the pressure sensors in the ducts was placed at a minimum distance of five times the duct diameter from an obstacle (e.g. heat valve). All of the heat valves were adjusted to the same position during the measurements. The pressure drop of the manifold was defined by measuring the pressure in the main supply duct and in each of the supply ducts serving the rooms and subtracting the two pressures to find the pressure difference.

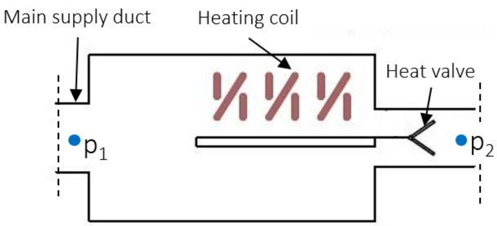

Figure 4-3. Location of the pressure sensors p_1 and p_2 presented in the vertical cross section of the manifold.

In the order to characterize the performance of the coil heat exchanger in the manifold, the coil heat transfer effectiveness was studied. The effectiveness ε (-) was defined as a ratio of the actual heat transfer rate $\dot{Q}$ (W) to the maximum possible heat transfer rate $\dot{Q}_{max}$ (W) between the air passing through the heating coil and the water circulating in the tubes of the coil and it was calculated as [71]:

$$\boldsymbol{\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}}} \tag{4.1}$$

The actual and the maximum possible heat transfer rates were defined by the equations (4.2) and (4.3), respectively:

$$\boldsymbol{\dot{Q} = \dot{Q}_{air} = \sum_{j=1}^{6} \dot{q}_{m\_air,j} \cdot \left(h_{outlet\_air,j} - h_{inlet\_ air}\right)} \tag{4.2}$$

$$\boldsymbol{\dot{Q}_{max} = \dot{Q}_{\,water} = \dot{q}_{m\_water} \cdot \left(h_{inlet\_water} - h_{outlet\_ water}\right)} \tag{4.3}$$

Where: $\dot{Q}_{\,air}$ (W) and $\dot{Q}_{water}$ (W) represent air and water heat transfer rates, respectively; q_{m_water} (kg/s) and $q_{m\_air,j}$ (kg/s) represent the mass flow rates of the water and the air in the j-th supply duct, respectively; $h_{inlet\_ water}$ (J/kg) and h_{outlet_water} (J/kg) are specific enthalpies of the water at the inlet and outlet of the coil in the manifold, respectively; $h_{inlet\_ air}$ (J/kg) and $h_{outlet\_air,j,}$ (J/kg) are specific enthalpies of the air at the inlet and outlet of the manifold, respectively.

During the continuous operation of the HVV system, the total airflow rate to different rooms remains constant. However, the amount of air that passes through or bypasses the coil heat exchanger may vary, depending on the current heating demand in the rooms that are served by the HVV system. Therefore, different scenarios in terms of the heating demands in the served rooms were examined in the experiment. First it was assumed that the heating is required only in one of the served rooms, therefore only one of the heat valves was set to position 1, i.e. fully open to heating. The remaining heat valves were adjusted to the position 0 directing the air to bypass the heating coil. Thus, the total airflow rate passing through the heating coil corresponded to the airflow rate measured in the duct serving the room with the heating demand (duct 1). Next, it was assumed that the heating is required consecutively in 2, 3, 4, 5 and lastly in all 6 of the served rooms. The number of the heat valves that were adjusted to the position 1, was consecutively: 2, 3, 4, 5 and 6. The total airflow rate passing through the heating coil corresponded to the sum of the airflow rates measured in the ducts serving the rooms with the heating demand. All measured cases are summarized in Table 4-1.

Table 4-1. Description of the cases examined in the coil heat transfer effectiveness measurements.

Case no.	**No. of rooms with heating demand**	**Position of heat valve**					
		Duct 1	Duct 2	Duct 3	Duct 4	Duct 5	Duct 6
(1)	1	1	0	0	0	0	0
(2)	2	1	1	0	0	0	0
(3)	3	1	1	1	0	0	0
(4)	4	1	1	1	1	0	0
(5)	5	1	1	1	1	1	0
(6)	6	1	1	1	1	1	1

In order to calculate the effectiveness, temperatures and flow rates of water and air at the inlets and outlets of the manifold were measured for each of the considered cases. The supply air temperature and the airflow rates were measured in the main supply duct as well as in each of the smaller ducts serving the individual rooms. The air temperature was measured by the means of Pt1000/TG-K3 temperature sensors. For the measurement of the supply air temperature in the main supply duct, the temperature sensor was placed after the air handling unit and before the manifold. Besides, the temperature sensors were placed in each of the ducts serving the individual rooms in the distance of 0.1 m from the manifold. The total supply airflow rate was measured using Lindab UltraLink Monitor. The airflow rate monitors were mounted in the main supply as well as in each of the supply ducts serving the individual rooms, respecting the mounting recommendations. The temperature of the water was measured in the supply and return pipes connected to the tubes of the heating coil. The water flow rate was measured in the supply pipe. The temperature of water was measured by the means of Pt1000/TG-D1 temperature sensor and the water flow rate was measured by the means of Kobolt MIK C34P magnetic inductive flow meter.

In addition to the aforementioned parameters, the energy use of the HVV system was studied in the full-scale experiment. The total energy use of the HVV system was defined as sum of the energy used by the fans (P_{fans}) and the energy used by the heat source (P_{heat_source}) and calculated as:

$$\boldsymbol{P_{system} = P_{fans} + P_{heat\_source}} \qquad \textbf{(4.4)}$$

Where: P_{system} (W) represents the energy demand of the HVV system; P_{fans} (W) is the energy used by the fans in the HVV system; P_{heat_source} (W) is the energy used by the heat source in the HVV system.

The total energy use of the HVV system is determined by the heating power that has to be delivered by the system to cover the ventilation and heating requirements in the served rooms. According to the equation (3.1) presented in Chapter 3, the heating power delivered by the HVV system depends on the supply air parameters, i.e. the supply air temperature (T_{supply_air}) and the supply airflow rate ($\dot{q}_v$). The supply air temperature is influenced by the temperature of water at the inlet of the heating coil (T_{inlet_water}). The parameters of the air and water affect the energy use of the HVV system. The same heating power can be delivered by the HVV system at various combinations of air and water parameters. In order to find the correlation between the air and water parameters and the energy use of the HVV system, the following assumptions were made:

Fan energy use. The energy used by the supply and exhaust fans (P_{fan} in eq. 4.4) was directly measured in the experiment. In addition, the efficiency of the supply and exhaust fans was calculated based on the following equation:

$$\boldsymbol{\eta = \frac{\Delta p \cdot q_v}{P_{fan}}} \qquad \textbf{(4.5)}$$

Where: η (-) represents the efficiency of the fan; Δp (Pa) represents the total pressure increase in the fan; q_v (m^3/s) represents the total airflow rate delivered by the fan; and P_{fan} (W) is the energy demand of the fan.

The total pressure increase in the fan (Δp), the total airflow rate (q_v) and the energy use of the fan (P_{fan}) were directly measured in the experiment.

Heat source energy use. The energy used by the heating source (P_{heat_source}) was defines as:

$$\boldsymbol{P_{heat\_source} = \frac{\dot{Q}_{water}}{COP}} \qquad \textbf{(4.6)}$$

Where P_{heat_source} (W) represents the total energy used by the heat source in the HVV system; $\dot{Q}_{water}$ (W) is the water heat transfer rate; COP (-) is coefficient of performance of a heat pump operating at the maximum theoretical efficiency, i.e. at the Carnot efficiency.

The heat supplied by the water to the HVV system ($\dot{Q}_{water}$) was defined as:

$$\dot{\boldsymbol{Q}}_{water} = \dot{\boldsymbol{q}}_{v\_water} \cdot \boldsymbol{\rho}_{w} \cdot \boldsymbol{C}_{p,w} \cdot \left(\boldsymbol{T}_{inlet\_water} - \boldsymbol{T}_{outlet\_water}\right) \qquad \textbf{(4.7)}$$

Where: $\dot{Q}_{water}$ (W) is water heat transfer rate; $\dot{q}_{v\_water}$ (m^3/s) is the volumetric flow rate of the water; ρ_w (kg/m^3) is the density of water; $C_{p,w}$ (J/(kg·K)) is the specific heat capacity of water; T_{inlet_water} (°C) and T_{outlet_water} (°C) are the temperatures of water at the inlet and outlet of the heating coil, respectively.

The volumetric flow rate of the water ($\dot{q}_{v\_water}$), and the temperatures of the water at the inlet (T_{inlet_water}) and the outlet (T_{outlet_water}) of the heating coil were directly measured in the experiment. The density of water (ρ_w) was considered as a function of temperature and the specific heat capacity of water was considered as a constant.

Heat source COP. The COP of the heat source was calculated according to the equation that represents the maximum theoretical coefficient of performance of any heat pump cycle operating between cold and hot reservoirs at T_C and T_H, respectively.

$$\boldsymbol{COP} = \frac{\boldsymbol{T}_H}{(\boldsymbol{T}_H - \boldsymbol{T}_C)} \qquad \textbf{(4.8)}$$

Where: COP (-) is the heat source coefficient of performance; T_H (K) and T_C (K) are the temperatures of the hot reservoir and cold reservoir, respectively.

The temperature of the hot reservoir (T_H) was defined as the temperature of water measured at the inlet to the heating coil. The temperature of the cold reservoir (T_C) was considered as of -5°C (268.14 K).

Air and water parameters. In the experiment, the parameters of the HVV system, i.e. the supply airflow rate and the temperature of water at the inlet to the heating coil, were adjusted to meet the heating demands in the served rooms. It was considered that the heating demands were met if the air temperature in all of the served rooms was maintained to the set point. The energy use of the HVV system was analysed for seven different combinations of the air and water parameters were studied.

Heating loads. By the means of the cooling system, different heating loads were generated in each of the rooms. The heating demands generated in the rooms are summarized in Table 4-2.

Table 4-2. Heating loads generated in the rooms in the full-scale experiment (Part 1).

Supply duct	**Served room**	**Heating load, W**
1	Bedroom	240
2	Office room	130
3	Living room	270
4	Kitchen	400
5	Bedroom	150
6	Bedroom	150

The measuring equipment used in the full-scale laboratory experiment (Part 1) is summarized in Table 4-2.

Table 4-2. Information on measuring equipment used in the full-scale experiment (Part 1).

Instrument	**Measured parameter**	**Measuring error**
UltraLink Monitor FTMU	Airflow rate	±5 % or ±1 l/s depending on which is the highest value
Pt1000/TG-K3	Air temperature	0.3 + 0.005 |t|°C
Testo 400	Air pressure	±1 Pa
Kobolt MIK C34P	Water flow rate	±2% of full scale
Pt 1000/TG-D1	Water temperature	0.3 + 0.005 |t|°C
Lutron DW-6163	Electric power	±5% + 5W

Part 2: The HVV system integrated in the experimental chamber

The following description of the experimental procedures is based on the published paper [70].

During the experiment, the air was supplied to the experimental chamber via a supply air terminal device. Two types of the supply air terminal device were examined: Type A – a circular valve, and Type B – three nozzles.

The Type A air terminal device is presented in Figure 4-4 a-b. The circular valve was placed in the external wall of the experimental chamber, below the window (X=1.8

m, Y=0.0 m, Z=0.40 m, see Figure 4-1). The inlet area of the circular valve was blanked off with a sector plate to 120° for directing the air stream upwards, along the window. The Type B air terminal device is presented in Figure 4-4 c-d. The air was supplied through three nozzles located in the wall opposite to the window in the upper part of the chamber (X=1.8 m, Y=3.1 m, Z=2.5 m, see Figure 4-1). The type and location of the supply air terminal devices are further referred to as Configuration I and II for the circular valve and nozzles, respectively. A gap below the door leaf served as the extract air terminal device in the experimental chamber. The location of the air extract remained unchanged throughout the entire experiment.

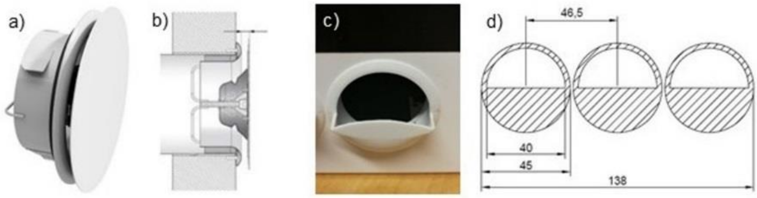

Figure 4-4. Supply air terminal devices used in the experiment: Type A) the circular valve (picture a-b) and Type B) the nozzles (picture c-d) [70].

In the experiment, the air temperature set point was set to 21°C. Two arrangements of the supply air parameters were investigated assuming two maximum supply air temperatures of 51°C and 36°C, correspondingly the supply-chamber air temperature differences (ΔT) were 30°C and 15°C. The airflow rates were set for the assigned supply air temperatures based on the maximum heating load in the chamber. The supply airflow rates were 7 l/s (ACH=0.84 h-1) and 14 l/s (ACH=1.67 h-1) for the supply air temperature 51°C and 36°C, respectively. For each of the two air inlet Configurations (I and II), the measurements were performed at both arrangements of the supply air parameters, in order to give understanding of the airflow behaviour within the chamber.

The steady-state condition in the experimental chamber was defined as the air temperature at all measurement points varied less than 0.5°C in a duration of 30 min. The sampling rate and measuring period were in accordance with EN 13182 [72], i.e. the thermo-anemometer recorded data for 3 min per each measurement. Within this measuring time, temperature and velocity were logged every 10 s. The measurement was repeated twice per each measurement set.

The measurements of the temperature and velocity were performed considering three vertical planes in the chamber, further referred as to Section A, B and C. In each section, the measurements were performed at three lines, further referred as to sampling lines L1, L2 and L3. The sampling lines L1, L2 and L3 were located in a distance of 0.25 m, 1.55 m and 2.85 m from the wall with the window, respectively. At each sampling line, a vertical bar equipped with eight SensoAnemo 5100SF sensors was placed. The sensors were distributed along the vertical bar at levels 0.10, 0.45, 0.80, 1.15, 1.50, 1.85, 2.20 and 2.45 m above the floor. The sections, sampling lines and location of the sensors are presented in Figure 4.5.

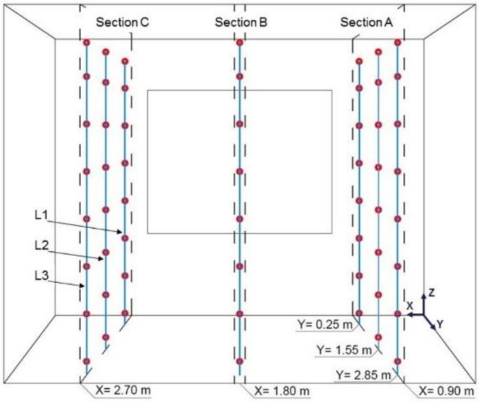

Figure 4-5. Sections (dashed outline), sampling lines (marked in blue) and sensors (marked in red) in the experimental chamber [70].

The supply airflow rate was monitored and controlled by a volume flow regulator consisting of a measuring unit and a damper, which was installed in the supply duct. The supply air temperature was measured prior to the supply air terminal inside the supply duct. The extract air temperature was measured in the middle of the gap below the door leaf. Both temperatures were measured using Pt100 resistance thermometers. The surface temperature was measured by means of infrared radiation thermometer AMiR 7811 in the middle and close the corners of each wall in the chamber. Additionally, the smoke visualization was performed in order to map the airflow distribution within the chamber.

Air velocity magnitude and air temperature distribution were measured using a Thermo-Anemometer transducer SensoAnemo 5100SF.

Detailed information on the measurement instruments used in the experimental chamber is presented in Table 4-3.

Table 4-3. Information on measurement instruments used in the experimental chamber (Part 2) [70].

Instrument	**Measured parameter**	**Measuring error**
Ametek KM 4 Pt100	Ambient, extract and supply air temperature	0.2°C ± 0.5% reading
AMiR 7811	Surface temperature	±1 % of measured value or ±1 K (whichever value is higher)
SensoAnemo 5100SF	Air temperature	±0.2°C
SensoAnemo 5100SF	Air velocity magnitude	0.02 m/s ±1.5% of readings
VRU	Supply airflow rate	±10% (1.2 - 3 m/s)

4.4. NUMERICAL SIMULATIONS

Numerical simulations were performed to evaluate the performance of two types of the supply air terminal devices intended for application in newly developed HVV system. The performance of the HVV system was quantified in terms of the provided thermal conditions. The results were analysed with respect to the air temperature and air velocity patterns and the local thermal discomfort due to draught. Further, a numerical parametric analysis was carried out for investigating the provided indoor thermal environment for a wider range of boundary conditions, including the influence of cold vertical surfaces, the influence of supply air temperature and supply airflow rate, and the influence of heating energy demand in the chamber.

The following description of the numerical simulations is based on a published paper [70].

Airflow modelling

The airflow was modelled based on the conservation equations of mass, momentum and energy. The present study employed the Realizable $k - \epsilon$ model to simulate airflow characteristics under turbulent airflow conditions. The model describes turbulence by means of two transport equations for turbulence kinetic energy k and dissipation rate ϵ. The Realizable $k - \epsilon$ model has been applied in many previous research works [73, 74] and proved to be appropriate for indoor climate simulations [75]. All these equations can be written in the general form of a transport equation as:

$$\frac{\partial(\rho\phi)}{\partial t} + \nabla \cdot (\rho\phi\vec{V}) = \nabla \cdot (\Gamma_\phi \nabla\phi) + S_\phi \tag{4.9}$$

Where ϕ represents the corresponding transported quantity, $\vec{V}$ is the air velocity vector, ρ is the air density, Γ_ϕ is the (effective) diffusivity and S_ϕ is the source term.

The computational domain was a replica of the experimental chamber, which was subdivided by unstructured grids that consisted of tetrahedral cells. The geometry of the supply air terminals were modelled in detail to obtain valid simulation results. In order to minimize false diffusion and dispersive errors in areas with high gradients, the grids were locally refined near the walls and regions with large gradients such as supply and extract air terminals. The grid independence was tested by considering three different grid resolutions. Coarse (0.57 million cells), medium (1.08 million cells) and fine (1.50 million cells) grids were generated and the results showed no considerable differences between the fine and medium grid in the predictions of airflow. Since increasing the grid resolution showed no significant improvement in the predictions, the medium grid was adopted.

The heat flux boundary condition was imposed on the window to simulate the heat loss, whereas the adiabatic conditions were assigned to all other solid boundaries, i.e. floor, ceiling and all walls. The Discrete Ordinate (DO) model was adopted for radiation modelling. The simulations were accomplished using ANSYS Fluent 18.2.

Model validation

The numerical model was validated by comparing the simulated air temperature distribution with data measured in the Part 2 of the experimental work. The validation was performed for both Configurations at the supply air temperature of 51°C (ΔT=30°C) and the supply airflow rate of 7 l/s (0.84 ACH). The comparisons are shown for Section C at the sampling lines L1, L2 and L3 in Figure 4-6.

For Configuration I, the predicted temperature profile well matches the measured results. For Configuration II, although the simulation slightly under-predicts the temperature level in the lower part of the chamber, the simulation results show reasonably good overall agreement with the experimental data. The discrepancy might be ascribed to the uncertainties in the boundary conditions (e.g. there might exist heat transfer through the walls of the chamber). The relative error between the predicted and measured values was on average 2.20 % and 4.61 % for Configuration I and II, respectively.

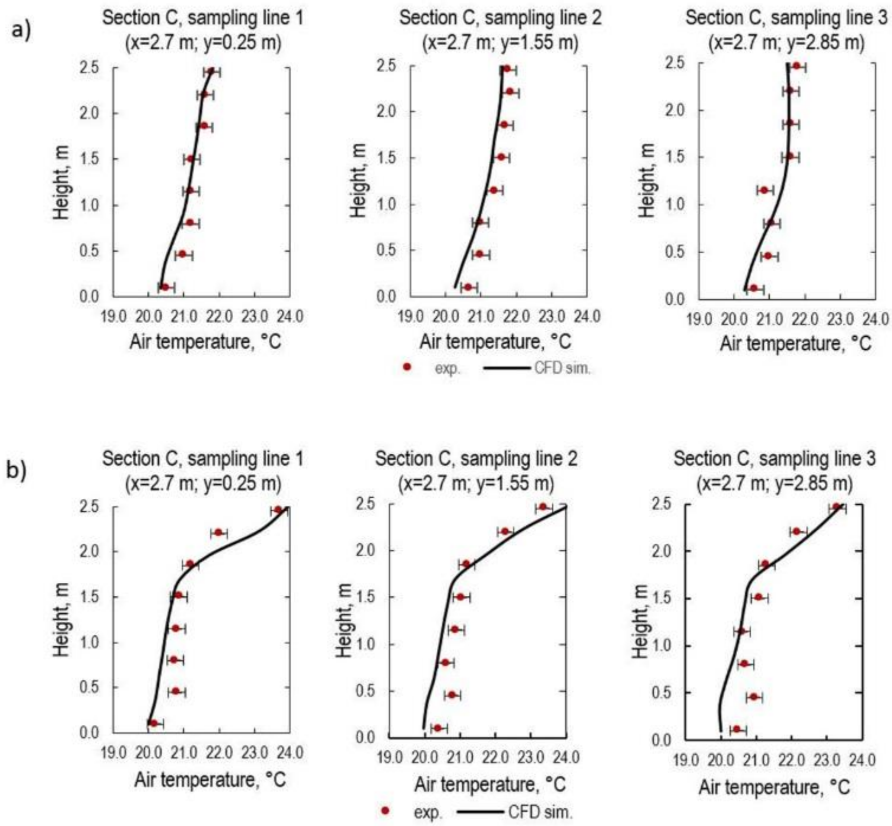

Figure 4-6. Comparison of vertical air temperature distribution by numerical simulations and experimental results for: a) Configuration I and b) Configuration II [70].

Parametric studies

The parametric studies focused on the flow behaviour under various conditions and their impact on the local thermal discomfort due to vertical air temperature differences and draught. The parametric studies were performed for both Configurations. The analysis started from a base model, which was the same for either Configurations. In the base model, both the supply air parameters and the boundary conditions were kept identical as those considered in the validation. Namely, the supply air temperature was 51°C (ΔT=30°C), the supply airflow rate was 7 l /s (ACH=0.84 h-1). The total transmission heat loss, assigned as a constant heat flux to the window surface was 234 W. Other solid surfaces were all set as adiabatic walls.

The parametric studies were carried out by altering the base model and performing numerical simulations. The study was carried out to predict the results for a wider range of operational parameters and boundary conditions and to illustrate the effect of these changes on the air distribution and thermal conditions in the investigated chamber.

The altered parameters were selected based on their potential influence on the air distribution. The analysis included the effect of the cold vertical surfaces, the supply air parameters and the heating energy demand in the chamber. Parametric studies were performed for both Configurations using identical sets of parameters. The input parameters for the parametric studies are summarized in Table 4-4.

Table 4-4. Input parameters used in the parametric studies [70].

Case	Analysed influence	Supply-chamber air temperature difference, °C	Air change rate, h-1	Surface with heat flux	Heating load, W	Internal heat gains, W
A	- (base model)	30.0	0.84	window	-234	0
B	cold surface	30.0	0.84	window, wall with the window	-234	0
C	cold surface	30.0	0.84	window, wall with the window, left site wall	-234	0
D	supply air parameters	22.5	1.19	window	-234	0
E	supply air parameters	15.0	1.67	window	-234	0
F	heating load	22.3	0.84	window	-234	60
G	heating load	10.7	0.84	window	-87	0

4.5. REAL-LIFE BUILDING MEASUREMENTS

The HVV system was installed in an existing single-family building located in Capital Region of Denmark. The building was one-storey detached building built in 1959. The floor area of the building was 145 m^2. The building was occupied by two persons. The average U-values of the external walls and windows were 0.7 $W/(m^2 \cdot K)$ and 1.6 $W/(m^2 \cdot K)$, respectively.

The real-life measurement activities consisted of:

a. Monitoring of the indoor air temperatures in a period January – April 2019.
b. Measuring of the local thermal comfort provided by the HVV system. The local thermal comfort was evaluated in terms of vertical air temperature difference and draught rate.

c. Evaluation of the perceived indoor climate based on an interview.

The building consisted of three bedrooms, kitchen, living room, one bathroom and a boiler room. The building layout is presented in Figure 4-7. The supply air terminal devices were installed in all rooms except the boiler room. The air was extracted in the bathroom and in the living room. Two types of the supply air terminal device were installed in the building. Namely, circular air valves were mounted in the ceiling in all bedrooms and floor mounted air terminals were installed in the kitchen and living room.

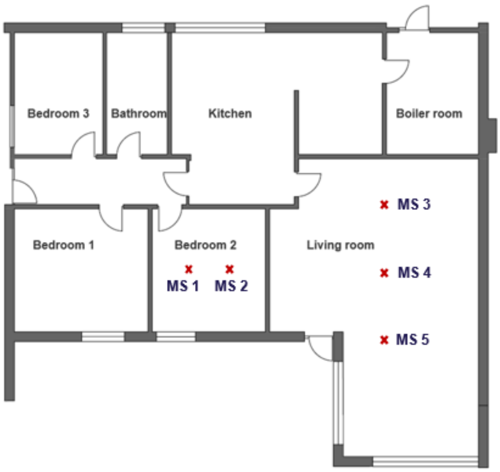

Figure 4-7. Layout of the building and the location of the measuring stands (MS 1-5) for the local thermal comfort measurements.

The air velocity and air temperature profiles were measured in the bedroom 2 and living room at the me at 0.1 m, 0.6 m, 1.1 m and 1.7 m above the floor. The location of the measuring stands is presented in Figure 4-7.

Detailed information on the measurement instruments used in the real-life measurements is presented in Table 4-5.

Table 4-5. Information on measuring equipment used in the real-life measurements.

Instrument	**Measured parameter**	**Measuring error**
SensoAnemo 5100SF	Air temperature	±0.2°C
SensoAnemo 5100SF	Air velocity magnitude	0.02 m/s ±1.5% of readings
IC meter	Air temperature	±0.3°C

4.6. MEASUREMENT UNCERTAINTY

The measured data were analysed in accordance with the ISO guide to the expression of uncertainty in measurement [76]. The reported expanded uncertainty is based on a standard uncertainty multiplied by a coverage factor k=2, providing a level of confidence of approximately 95%.

Part 1: The HVV system in the full-scale laboratory experiment

During the experiment, the air temperature was measured with the accuracy of ±(0.3 + 0.005| t |), the static pressure was measured with the accuracy of ±1 Pa, and the airflow rate was measured with the accuracy of ±5% of reading. The combined expanded measurement uncertainty of the coil heat transfer effectiveness was estimated to vary from 0.12 at the highest airflow rate passing through the heating coil (i.e. 68 l/s) to 0.34 at the lowest measured airflow rate passing through the heating coil (i.e. 11 l/s). The largest contribution to the uncertainty of the effectiveness is represented by the uncertainty of the difference in specific enthalpies of water at the inlet and the outlet of the heating coil, which can be ascribed to small differences in measured inlet and outlet water temperatures, and the accuracy of the temperature sensors. The maximum and average expanded uncertainty of the total energy use of the HVV system was estimated to 2.6 W and 2.1 W, respectively.

Part 2: The HVV system integrated in the experimental chamber

During the experiment, the air velocity probes were calibrated using a dedicated wind tunnel in the range of 0.05 to 5 m/s with the accuracy of 0.02 m/s ±1.5% of reading. The temperature probes were calibrated in the range of -10 to +50°C with the accuracy of ±0.2°C.

Real-life building measurements

During the measurements in the real-life building, the air velocity was measured with the accuracy of 0.02 m/s ±1.5% of reading and the air temperature distribution was measured with the accuracy of ±0.2°C. The room air temperature was measured with the accuracy of ±0.3°C.

CHAPTER 5. RESULTS AND DISCUSSION

The following chapter presents and discuss the results from the laboratory experimental work, numerical simulations using Computational Fluid Dynamics (CFD) and real-life measurements in a single-family residential building.

5.1. LABORATORY EXPERIMENT

The ability of the HVV system to control the air temperature on room level was demonstrated in the full-scale laboratory experiment (Part 1).

In the HVV system, the temperature of the supply air is controlled independently at each of the outlets of the manifold. By controlling the supply air temperature, the HVV system is able to satisfy various heating demands in all served rooms.

Figure 5-1 illustrates the variation of supply air temperature (*t2*) in relation to the position of the heat valve. The supply air temperature measured in the main supply duct (*t1*), was constant and of 22°C. In addition, the air temperature was measured inside the heated part of the manifold (*t2*). The temperature measured inside the manifold remained relatively constant and of 25 °C. An increase in the temperature inside the manifold (*t2*) could be observed between the measurements in the position of the heat valve 30° and 45°. This could be ascribed to the variation in the temperature of the water supplied to the heating coil in the manifold.

When the air passes through the manifold, undesirable heat transfer may occur inside the manifold. Therefore, the temperature in the supply duct (*t3*) was higher than the temperature in the main duct (*t1*). The temperature *t3* increased while changing the heat valve position. In the full open position (90°), *t3* was 52 °C, which relatively close to the temperature *t2* measured after the heating coil.

Figure 5-2 presents the total airflow rate and the measured static pressure difference in relation to the position of the heat valve. The difference of the static pressure measured in the main duct and the individual supply ducts varied between 43 Pa, for the heat valves position of 0°, i.e. when the air bypasses the heating coil, and 55 Pa at the heat valves position of 45°, when the air passes through both, heated and unheated parts of the manifold. This can be ascribed to the increased resistance created at the heat valve in mixing positions, i.e. 30°, 45°, 60° and 75°. Therefore, the total airflow rate dropped ±5%, from 56.4 l/s at full closed position (0°) to 53.5 l/s at mixing position (45°).

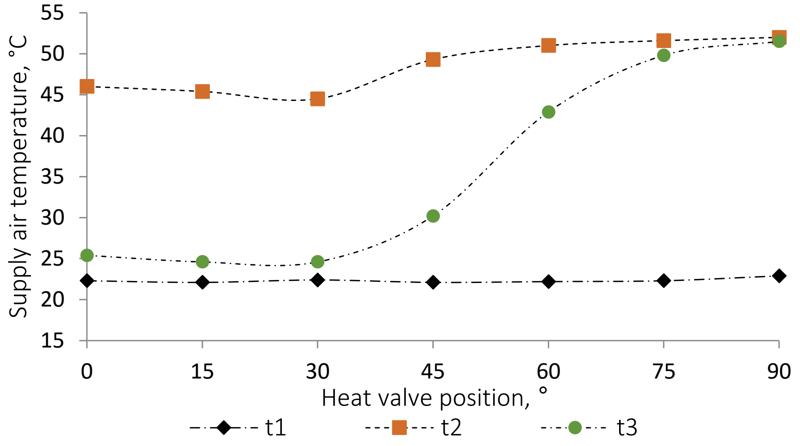

Figure 5-1. Supply air temperature in relation to the position of the heat valve.

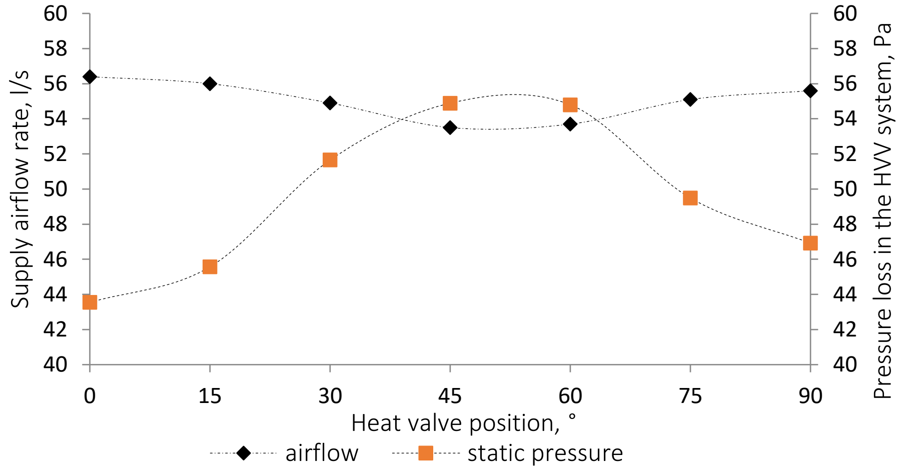

Figure 5-2. Total airflow rate and static pressure in relation to the position of the heat valve.

Figure 5-3 shows the effectiveness of the coil heat transfer between water and air in the manifold. The effectiveness varied between 0,79 at the lowest airflow rate, i.e. 11 l/s, and 0,94 at the highest airflow rate, i.e. 68 l/s. The exchange of heat can be carried out with high effectiveness due to the counter-flow nature. Thus, the system can operate efficiently both at a full-load and at a part-load. The lowest effectiveness of the heat coil transfer (0.79) might be ascribed to the thermal losses occurring along the manifold.

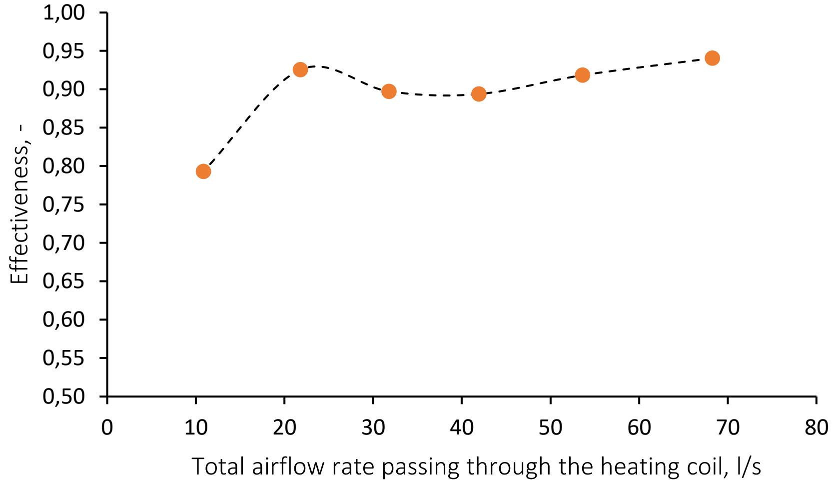

Figure 5-3. Effectiveness of the coil heat transfer in the manifold.

Figure 5-4 illustrates the energy use of the HVV system in relation to the water temperature at the inlet of the heating coil. Depending on the water temperature at the inlet to the heating coil, the energy used by the HVV system varied significantly. The results show that the minimum energy use of the system occurred at the supply air temperature of 52°C and the supply airflow rate of 56 l/s. A reason for the obtained results can be that the maximum theoretical COP of the heat source was consider in the calculation. Another reason contributing to the result was the low efficiency of the fans in the air handling unit used in the HVV system prototype. The efficiency of the fans was of 10%.

Figure 5-5 show the energy use of the HVV system, considering different thermal reservoirs resulting in different COP of the heat source. It can be seen that the minimum energy use of the HVV system was found at the temperature of water at the inlet to the manifold of 47.9°C, which corresponded to the total supply airflow rate of 65 l/s.

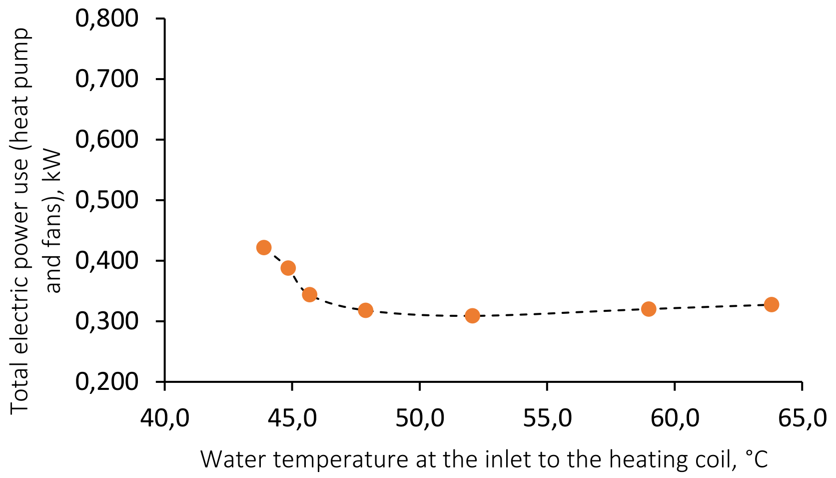

Figure 5-4. HVV system energy use in relation to the water temperature at the inlet to the heating coil, considering maximum theoretical COP of the heat source.

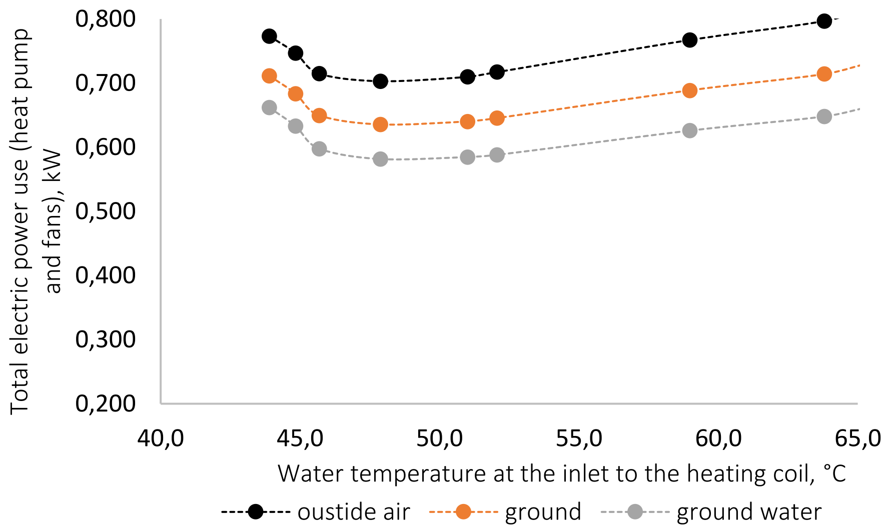

Figure 5-5. HVV system energy use in relation to the water temperature at the inlet to the heating coil, considering outside air, ground and ground water as the hot reservoir for the calculations of the COP of the heat source.

In the Part 2 of the experiment the HVV system was integrated in the experimental chamber.

The following description of the results from the experiment carried out in the chamber are based on [70].

Figure 5-6 presents the results from the smoke visualization for Type A and Type B of the supply air terminal device. For each Type, the visualization was performed for both arrangements of the supply air parameters, namely 51°C at 7 l/s (ACH=0.84 h-1, ΔT=30°C) and 36°C at 14 l/s (ACH=1.67 h-1, ΔT=15°C).

For Type A supply air terminal device (Figure 5-6 a), the smoke visualization test showed how the airstream accelerated upwards and moved in radial patterns along with the window. When the air reached the ceiling, it spread to the sidewalls and moved along the walls towards the floor. Then the air in the jet mixed completely with the air in the chamber within the entire space. This indicated that the air jet discharged from the air inlet below the window would be able to provide sufficient air mixing in the entire chamber. For both investigated arrangement of the supply air parameters, similar airflow patterns were observed. This proved that the parameters of the supply air had no significant influence on the flow pattern in the experimental chamber, when the air was supplied using Type A supply air terminal device.

When using Type B supply air terminal device (Figure 5-6 b), the warm air was introduced in the upper part of the chamber. For the supply air parameters arrangement 51°C at 7 l/s (ACH=0.84 h-1, ΔT=30°C), it was observed that the air in the jet moved along the ceiling toward the external wall and the window. When the air jet reached the opposite side of the chamber, the supplied warm air remained in the upper part of the chamber. For the supply air parameters of 36°C at 14 l/s (ACH=1.67 h-1, ΔT=15°C), the air in the jet initially tended to attach to the ceiling surface. As the buoyancy forces became greater than the inertia of the airstream, the jet detached from the ceiling approximately in the middle of the chamber. The visualisation revealed different airflow patterns in the chamber between the investigated supply air parameters arrangements. Nevertheless, for both arrangements of the supply air parameters, fully mixed conditions were not observed within the time of recording (~1min). Moreover, the observed airflow patterns indicated a possible risk of increased vertical air temperature differences inside the chamber.

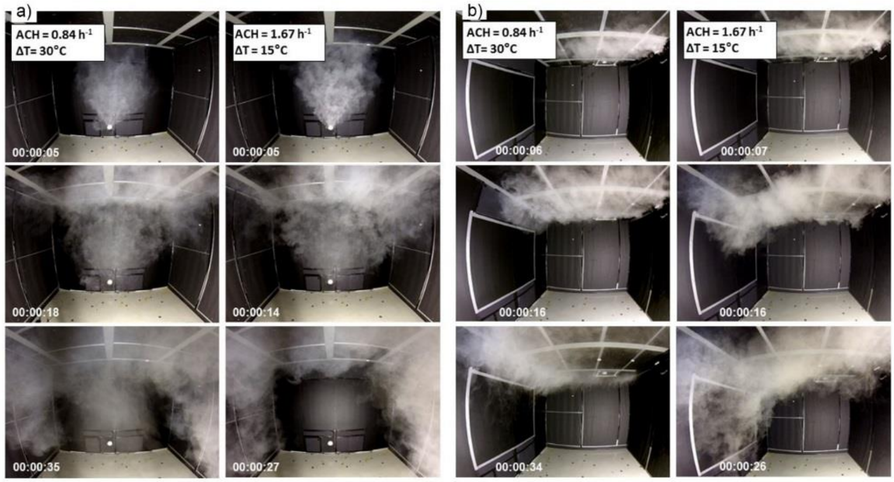

Figure 5-6. Smoke visualization of the airflow pattern for air terminal devices: (a) Type A, the view on the wall with the window; and (b) Type B, the view on the sidewall, the wall with the window is on the left side [70].

5.2. NUMERICAL SIMULATIONS

The following description of the numerical simulations results is based on a published paper [70]. In the following description, the experimental chamber utilized in the Part 2 of the experimental work is further referred to as room.

Vertical air temperature distribution

To evaluate the vertical air temperature difference in the room, two horizontal planes (XY-planes) were considered at 0.1 and 1.8 m above the floor, representing the boundaries of the occupied zone. The comparisons of the air temperature level between these two planes were made for each of the investigated supply air Configurations. The results are shown in Figure 5-7. When using Configuration I, the maximum air temperature difference between the level of 0.1 m and 1.8 m above the floor was 1.4°C. The room air temperature across both of the considered planes remained mostly uniform, at 20.4°C and 21.8°C at 0.1 m and 1.8 m, respectively. However, higher temperatures (22.4 – 23.0°C) could be observed near the window at the level of 1.8 m. The uniform air temperature distribution at the level of 0.1 m above the floor and the higher temperatures observed in the window vicinity at the level of 1.8 m above the floor indicate that the warm air supplied below the window counteracts the downdraught that is caused by the cold window surface.

In Configuration II, where the air was supplied in the upper part of the room, the maximum vertical air temperature difference was 2°C. The room air temperature remained uniform (21.8°C) at the level of 1.8 m above the floor. At the level of 0.1 m above the floor, the air temperature level varied between 19.6°C and 20.4°C. The lowest temperature value (19.6°C) was observed near the window and in the central part of the room. This may indicate a weakness in counteracting the cold downdraught caused by the window surface when using Configuration II. However, it should be taken into consideration that the total heat loss in the room (234 W) was only assigned as a heat flux to the window surface and thus could result in an increased downdraught and in consequence lower air temperatures below the window. For both Configurations, the results showed that the maximum vertical air temperature difference in the occupied zone did not exceed 3°C, which is a limit for the acceptable thermal environment for comfort recommended by (77, 78).

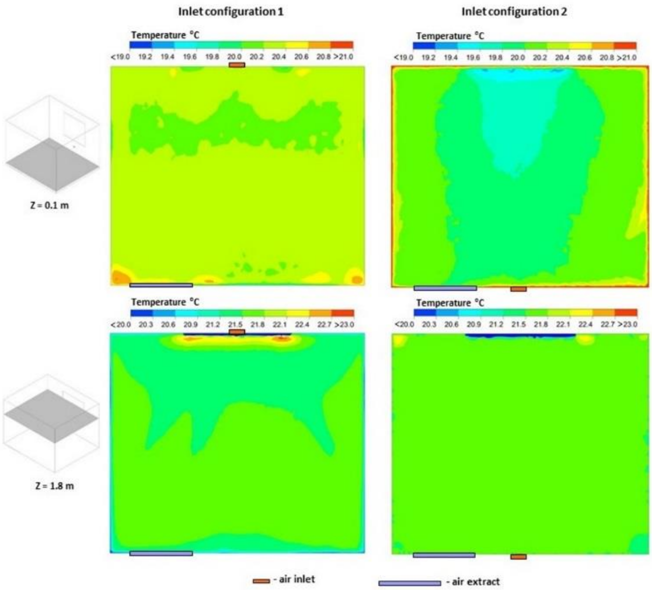

Figure 5-7. Temperature contour plot for Configurations I and II at the XY-planes at 0.1 m and 1.8 m above the floor.

Air velocity distribution

The air velocity distribution inside the room was analysed. The results are presented in Figure 5-8. The results are shown in a vertical plane in the middle of the room (YZ-plane, X=1.8 m). This location was found the most representative as the vertical plane cut covers a rather large part of the inlet air plumes. High air velocity (> 0.15 m/s) was observed along the air jet supplied to the room for both Configurations. Moreover, in Configuration I, increased air velocity was also observed along the wall with the window and near the ceiling. The air velocity inside the occupied zone was found between 0.0 – 0.08 m/s. In Configuration II, the results showed that higher air velocity (> 0.15 m/s) occurred in the area below the window and near the floor, which may indicate a draught risk. The air velocity within the occupied zone remained between 0.0 – 0.2 m/s.

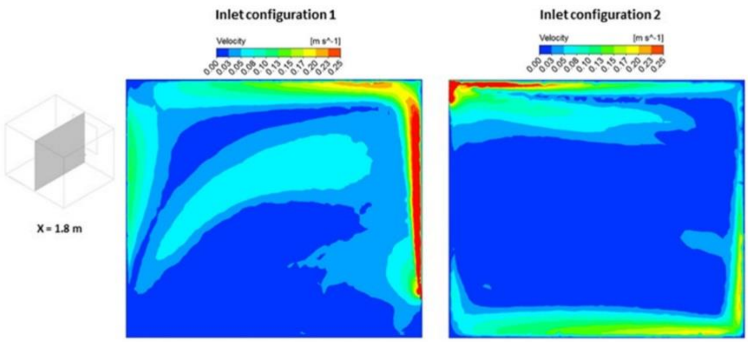

Figure 5-8. Velocity contour plot for Configurations I and II in the centre of the room at the YZ-plane.

Parametric studies

The following sections present the results of the parametric studies. The influence of the analysed factors on the local thermal discomfort, vertical air temperature difference and draught risk within the entire occupied zone were examined in accordance with ISO Standard 7730 [78]. The summary of all simulated in the parametric studies cases can be found in Table 4-4 in Chapter 4.

Cold vertical surfaces, such as windows, may generate thermal discomfort in rooms, due to radiation effects and cold downdraughts caused by natural convective flows from these surfaces. One of the factors influencing the air velocity in a room, especially close to the window and at the floor level, is the temperature difference between the inner window surface and the air in the room. The temperature of the inner window surface depends on the outside air temperature and window heat transmittance coefficient (U-value). Even high-performance windows should be considered as potential draught generators if the external temperature is lower than -5°C [79]. A heating source usually is placed beneath a window to compensate for radiative asymmetries and to avoid draught risk from cold air downdraught.

The influence of the cold vertical surfaces on the air distribution in the investigated room was examined in cases B and C. The base model (case A) considered the total heat loss in room 234 W, assigned as a heat flux to the window surface only. In case B and C, the total heat loss was distributed between the window surface and the surface of one (case B) or two walls (case C), considered as external walls. As the total heat loss was kept the same for all three cases, heat flux on the window surface decreased to 160 W in case B and 72 W in case C.

For Configuration I, in which the air inlet was placed below the window, both case B and case C gave results comparable to the base model case in terms of air velocity and temperature patterns, but with a higher air temperature gradient (Figure 5-9 a). The vertical air temperature difference in the room increased by 0.4°C and 0.8°C in case B and case C, respectively. For Configuration II (Figure 5-9 b), in which the warm air was supplied in the upper part of the room, no significant changes in the temperature distribution patterns and the air temperature gradient was observed. Similarly, the air velocity patterns in case B and C followed the base model case across the middle and upper part of the room (0.5 m – 2.5 m above the floor). However, at the level of 0.1 m above the floor, considerably lower air velocities compared to the base model case were noticed. Namely, the air velocity reduced from 0.14 m/s to 0.10 m/s in case B and to 0.05 m/s in case C. The temperature of the inner window surface significantly influences the air velocity close to the window and at the floor level in a room. The temperature of the inner window surface depends on the U-value of the window, which in case A was of 3.37 W/(m^2K). This value is greatly higher than the U-values of the windows currently available on the market (0.83 – 1.5 W/(m^2K)). Lower air velocities (0.05 - 0.10 m/s) are expected in a room with high performance windows, which are commonly used in low energy buildings.

The main criterion, when dimensioning the operational parameters of the HVV system, i.e. supply airflow rate and supply air temperature, is the maximum heating load that has to be covered by the system. A minimum supply airflow rate should be always ensured to maintain the necessary air quality in the space. Supplying the air at a relatively high supply air temperature and the minimum required supply airflow rate can save energy used by the supply fan in the system, but it may reduce the rate of air mixing inside the space. Increasing the supply airflow rate may improve air mixing; however, this would increase the energy used by the system and require larger space for the ductwork to avoid excessive pressure drop and air velocities inside the ducts. In this section, three combinations of the dimensioning supply air parameters were studied to identify whether decreasing the required maximum supply air temperature, and consequently, increasing the required supply airflow rate may benefit the indoor thermal environment in the room.

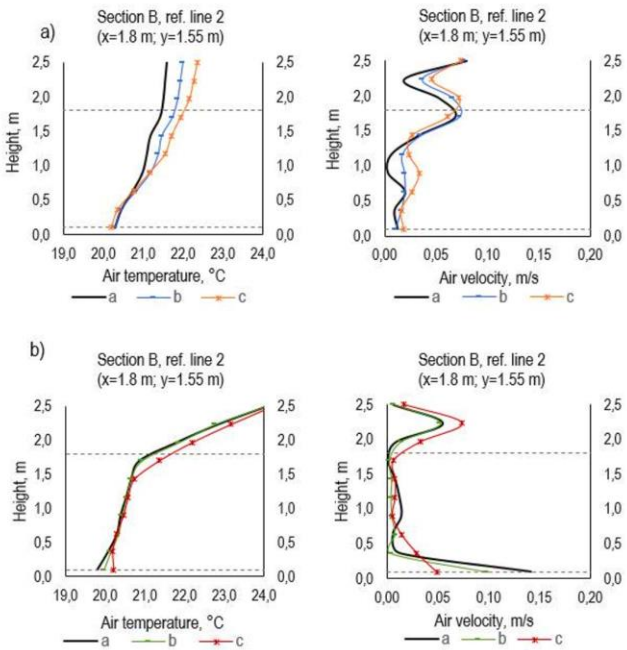

Figure 5-9. Air temperature (left) and air velocity (right) distribution for: a) Configuration I and b) Configuration II. Case A represents a base model case and it is marked in black. The dash lines indicate the occupied zone.

The influence of the supply air parameters on the air distribution was investigated in cases D and E. In the base model (case A) the air was supplied at the temperature of 51°C and airflow rate of 7 l/s (ACH=0.84 h^{-1}, ΔT=30°C). In case D, the supply air temperature was decreased to 43.5°C (ΔT=22.5°C). The supply airflow rate of 10 l/s (ACH=1.19 h^{-1}) was set for the assigned supply air temperature based on the total heat loss and the air temperature setpoint in the room. In case E, the supply air temperature was further decreased to 36°C (ΔT=15°C) and the supply airflow rate of 14 l/s (ACH=1.67 h^{-1}) was set by analogy with the case D. The total heat loss and the air temperature setpoint in the room remained unchanged in both cases (see Table 4- in section 3.3).

The results are shown in Figure 5-10. In Configuration I (Figure 5-10 a), the supply air parameters used in simulation of case E provided almost uniform temperature environment in the room, with a slightly lower temperature level at 0.1 m and 0.4 m above the floor. The results obtained in Case E also showed higher air velocity levels (~0.10 m/s) occurring in the occupied zone, indicating complete mixing of air in the room. Figure 5-10 b) shows the results from the simulations of Configuration II. It was observed that supplying the air at higher airflow rate (ACH=1.19 h^{-1}) and lower air temperature (ΔT=15°C) in case D and case E, resulted in higher vertical air temperature differences within the occupied zone (maximum 2.2°C and 2.6°C in case D and E, respectively) comparing to the base model case (1.2°C). At the same time, the maximum vertical air temperature difference within the entire room decreased. Both cases gave comparable results in terms of the air velocity distribution within most of the occupied zone; however, in the upper part of the room (1.7 m above the floor), the air velocity levels increased significantly (from 0.06 m/s to 0.12 and 0.15 m/s in case D and E, respectively). The increase in air velocities in the room appears to be due to the greater entrainment of the air surrounding the supply air jet. The greater the momentum the induced air achieves, the greater the penetration into the occupied zone there will be. This results in a larger proportion of the warm supply air being carried directly into the occupied zone and causing the higher vertical air temperature differences in the occupied zone.

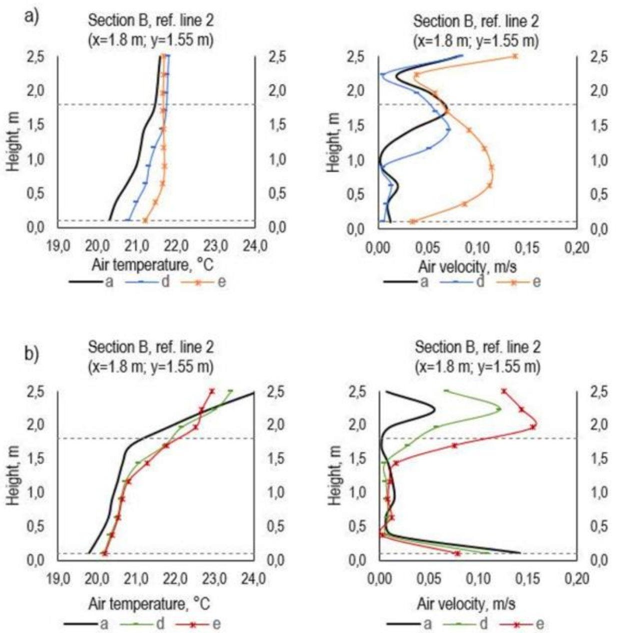

Figure 5-10. Air temperature (left) and air velocity (right) distribution for: a) Configuration I and b) Configuration II. Case A represents a base model case and it is marked in black. The dash lines indicate the occupied zone.

Influence of heating energy demand

Heating energy demand in a room decreases as the temperature difference between outdoor and the room air decreases or in case of the presence of internal heat gains, e.g. a presence of a person, in the room. As the control of the heating energy delivered via the ventilation system to the room is achieved through regulating the supply air temperature, reduction in heating demand will result in reduced supply air temperature. Moreover, in case of the presence of internal heat gains, a considerable amount of air is transported in thermal plumes above internal heat sources. Buoyancy forces, created as a result of the presence of internal heat sources, may have a major influence on the flow behaviour of the room air and, consequently, on the thermal environment in the room.

The airflow behaviour in the room and its influence on the air temperature and velocity distribution under different heating energy demands were investigated in cases F and G. In case F, the room was considered as occupied by one person, contributing to 60 W of internal gains in the room. Thus, the total heat loss of 234 W in the room was compensated partially by the internal heat gains from the person. The remaining 174 W of the heating energy demand was covered by the ventilation system. In case G, the total heating load in the room was decreased from 234 W to 87 W corresponding to the total heat loss due to the transmission that would occur in the room at the outdoor air temperature of 10°C. In case G, no internal gains were added to in the room, thus, the total heating energy demand of 87 W had to be covered by the ventilation system.

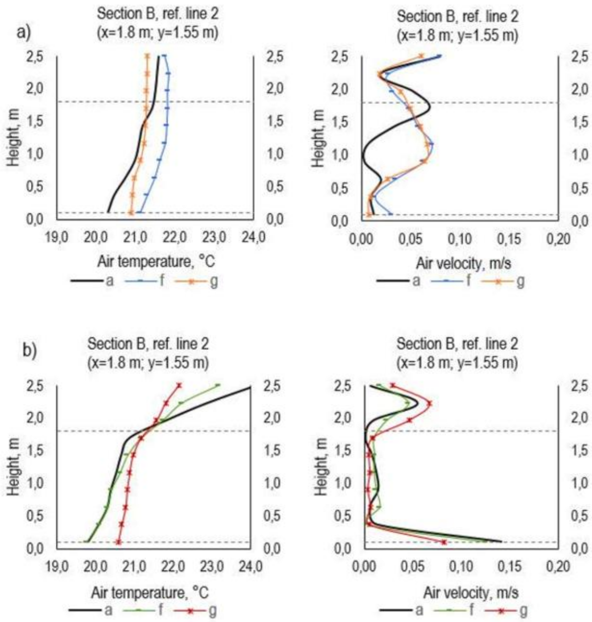

Figure 5-11. Air temperature (left) and air velocity (right) distribution for: a) Configuration I and b) Configuration II. Case A represents a base model case and it is marked in black. The dash lines indicate the occupied zone.

Figure 5-11 presents the results of the simulated Cases F and G. In general, decreased heating energy demand in the room resulted in more uniform temperature distribution for both of the considered Configurations. Specifically, the significant decrease in the heating energy demand in case G, and consequently the reduction in the temperature difference between supply air and room air, led to more uniform air temperature distribution and higher temperatures in the lower part of the room and, thus, to better thermal comfort in the room comparing to the base model case. When using Configuration I, the velocity patterns between 0.6 m and 1.7 m above the floor were different from the base model case in both cases G and F. The air velocity patterns were, however, similar to each other for these cases. In Configuration II, the air velocity at the level of 0.1 m above the floor decreased in Case G. Besides, no significant change on air velocity distribution was observed in Configuration II.

The predicted distributions of air velocity in case of presence of a person in the investigated room (Case F) are presented in Figure 5-12. In comparison to the base model case (Figure 5-8), different air velocity patterns could be observed in the middle of the room, due to the buoyancy-driven flow created by the warm surface of the mannequin. In addition, higher air velocities (~0.23 m/s) were observed below the window and at the floor level close to the window in Configuration II, indicating that the buoyancy forces, created as a result of the presence of person in the room may contribute to formation of downdraught below the cold surfaces in the room.

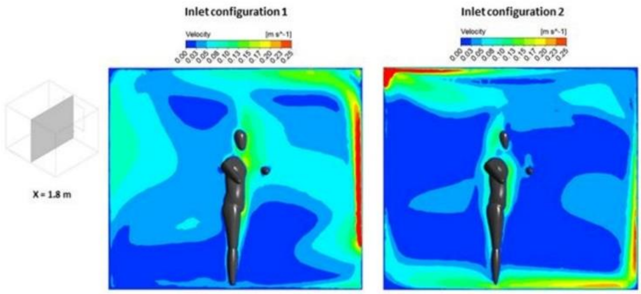

Figure 5-12. Velocity contour plot for Configurations I and II in the centre of the room at the YZ-plane for Case F.

The present investigation focuses on a low-energy building, defined as Building class 2020 in [23], thus, the study was limited to the relatively low heating loads. Moreover, the study does not focus on the influence of air infiltration on the air distribution and heat distribution in the room. Even though, a short discussion on the room heating load and the infiltration is further presented. In case a building with no sufficient insulation, in which the infiltration is of a significance and the heating loads are higher than in the present investigation, covering the heating demands would require either a significantly higher supply air temperature or a higher airflow rate, i.e. a higher volume of air than would be required for ventilating. This might significantly influence the indoor thermal environment by e.g. causing a draught in the occupied zone and changing the temperature distribution patterns inside the room. Besides, the required airflow rates might lead to an excessive energy use of a HVV system due to increased pressure drop and air velocities inside the ducts .Thus, the heating capacity of the HVV system may constitute a limitation when considering a building with a higher space heating energy demand.

Considerations on local thermal comfort

The maximum vertical air temperature difference inside the entire room was found in case C (2.45°C) for Configuration I and in case A (4.30°C) for Configuration II. The temperature difference in the occupied zone did not exceed 3°C in any of the investigated cases for both Configurations. The results are presented in Table 5-1.

Table 5-1. Local thermal comfort - results from the parametric studies [70].

Case	**Maximum vertical air temperature difference, °C**		**Draught rate, %**
	Inside the whole room (0.0 - 2.7 m)	Inside occupied zone (0.1 - 1.8 m)	Inside occupied zone (0.1 - 1.8 m)
Configuration I			
A	1.49	1.32	3.21
B	2.15	1.71	4.43
C	2.45	2.14	3.93
D	1.27	0.96	7.88
E	0.70	0.57	16.47
F	1.02	0.79	6.35
G	0.56	0.41	5.62
Configuration II			
A	4.30	1.25	18.09
B	4.10	1.09	12.23
C	4.11	1.27	3.29
D	3.30	2.21	15.76
E	2.83	2.60	17.05
F	3.38	1.84	20.63
G	1.61	0.72	9.92

The draught rate did not exceed 20% as recommended by [77, 78] in any of the investigated cases for Configuration I. For Configuration II, higher draught rate values (15 - 20%) were along the wall below the window and along the floor close to the wall with the window. These values may be an indication of local thermal discomfort due to the draught. However, the high value of heat flux (234 W) through the window and its contribution to the cold downdraught should be taken into considerations.

5.3. REAL-LIFE BUILDING MEASUREMENTS

Room air temperature

The indoor air temperatures monitored were monitored in six rooms in the building.

Figure 5-13 and 5-14 presents the measured indoor air temperatures during a winter and a spring day, respectively.

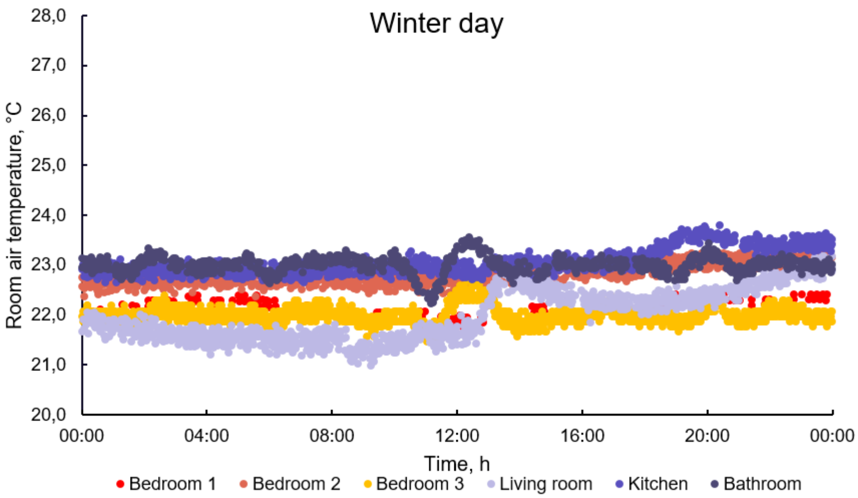

Figure 5-13. Indoor air temperatures monitored in the six rooms during a winter day (1-02-2019).

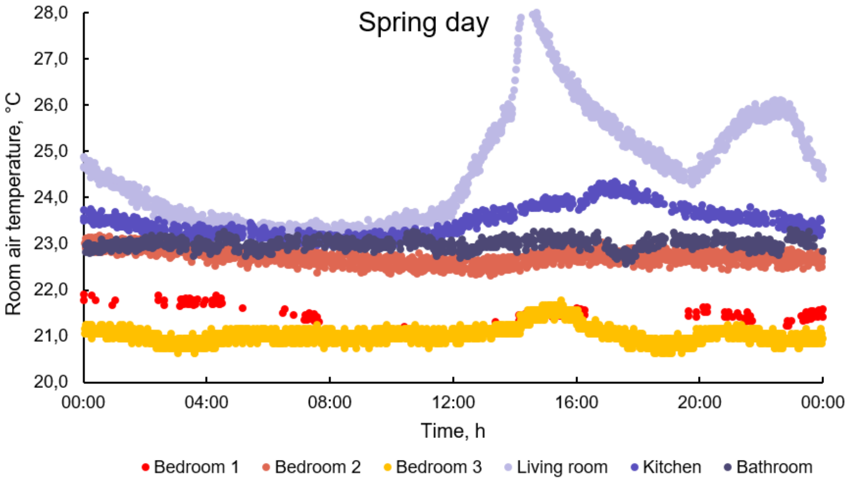

Figure 5-14. Indoor air temperatures monitored in the six rooms during a spring day (1-05-2019).

The desired temperatures (the set points) were of 23°C in all rooms during the period of monitoring the room air temperatures in the winter. The temperature set points were changed in bedroom 1 and bedroom 3 to 22°C and 21°C, respectively, in the spring since the rooms were mostly unoccupied. The internal doors to the bedroom 1 and bedroom 3 were kept closed. The results showed that it was possible to maintain various temperature set points in different rooms provided that the doors remained closed. Excessive indoor air temperature (of a maximum 28°C) were observed in the living room between 12:00 and 20:00. This might be due to the presence of the solar heat gains. That might also be caused by the changes in the HVV system settings done by the users.

Measured local thermal comfort

Figure 5-15 illustrates vertical air temperature difference measured in bedroom (MS1 and MS2) and living room (MS3, MS4 and MS5). The maximum measured vertical air temperature difference was of 2.0°C and 1.3°C in the bedroom and living room, respectively. Table 5-2 presents the maximum measured draught rate. The measured draught rate did not exceed 20% as recommended by [77, 78] in any of the measured points.

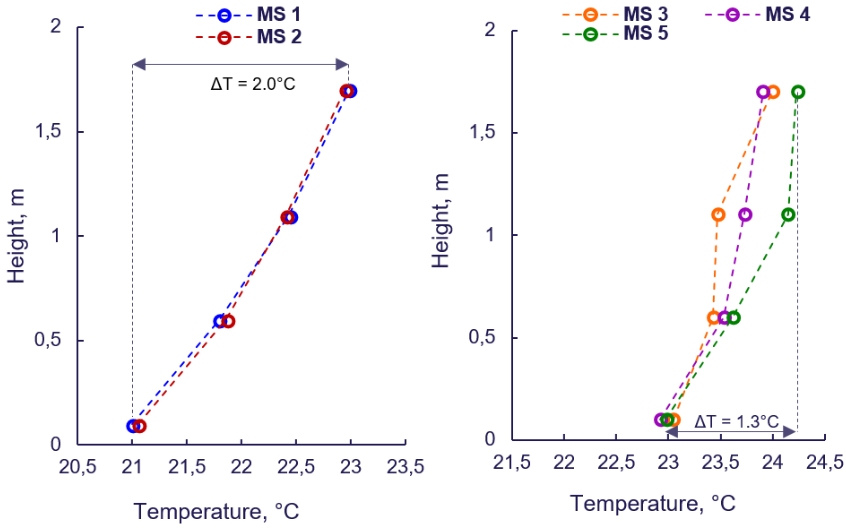

Figure 5-15. Vertical air temperature difference measured in bedroom (MS1 and MS2) and living room (MS3, MS4 and MS5).

Table 5-2. Maximum measured draught rates.

Height	MS 1	MS 2	MS 3	MS 4	MS 5
0.1 m	0 %	0 %	3.7 %	1.9 %	6.5 %
0.6 m	0 %	0 %	5.4 %	4.3 %	8 %
1.1 m	0 %	0 %	5.4 %	5.6 %	4.1 %
1.7 m	0 %	1.1 %	0 %	5.8 %	13.2 %

Perceived local thermal comfort

Two persons were living in the house at the time of the measurements. In general the thermal conditions in the house is perceived acceptable; however, a variation of indoor air temperatures in relation to the weather conditions reported. In addition, perceived draught rate was reported by one of the tenants. Despite the fact that the temperature set points were maintained in all rooms, the air temperatures in the living room and kitchen were perceived as too cold in the winter time. An intermittent heating source had to be used in the living room.

The occupants preferred the thermal conditions in the rooms, where the supply air terminal device was located in the floor. The occupants are more satisfied when the air was supplied to the rooms at higher air temperature and lower airflow rates, as during the hours when the heating demand is low (e.g. at the presence of solar heat gains), they occupants perceive that the airflow rates supplied to the rooms are too high, mentioning that '*it is difficult to adapt to new air all the time*'. The occupants would prefer a system where the both the supply air temperature and the supply airflow rate can be regulated individually in each room.

CHAPTER 6. CONCLUSION

In this work, a new heat valve ventilation system enabling air temperature control on room level was proposed. The performance of the HVV system was studied in the laboratory experiments performed on the prototype of the HVV system. Further, numerical simulations were performed to evaluate the performance of two types of the supply air terminal devices intended for application in newly developed HVV system. In addition, the HVV system was installed in a single-family house.

6.1. CONCLUDING REMARKS

Based on the results achieved is this work, the following conclusions can be outlined:

- The HVV system was able to achieve air temperature control on room level with the aid of the centralized heating coil and the heat valves installed in the manifold. It was possible to control the supply air temperature within a range of 25 - 52°C.

- HVV system can be used as a solution for residential buildings with low heating energy demand (~ 10 W/m2), where the heat losses are almost equally covered by solar gains, internal gains and the heating system.

- Neither a significant thermal stratification, nor the draught rate were revealed in the numerical analysis of the HVV system performance. The results showed that both of the applied air terminal devices could avoid temperature stratification within the occupied zone. The maximum air temperature difference between 0.1 and 1.8 m above the floor was 2.1 °C when using a circular valve placed in the external wall below the window and 2.6 °C in the case when the air was supplied through three nozzles located in the wall opposite to the window in the upper part of the room. In general, placing the air terminal device below the window provided more uniform air temperature distribution and contributed to the prevention of downdraught caused by a cold window surface.

- A successful operation of the HVV system in a real-life building was demonstrated. The measurements of the indoor air temperatures and the air temperature and air velocity profiles showed that the HVV system maintained the desirable thermal conditions in all of the rooms in the building. In addition, provided that the internal doors were closed, it was possible to maintain different temperatures in the rooms.

6.2. FUTURE WORK

Based on the research conducted in this study, the following recommendations for the future work are presented:

- Enabling supply airflow rate control on room level. Further development and analysis of the HVV system should be carried out in terms of applying different control strategies (e.g. water temperature and flow rate control), including implementation of the supply airflow rate control on room level.

- Annual energy performance. The annual energy performance of the HVV system should be further studied based on a real-life data.

- Implementation in a multi-apartment building. The HVV system may be implemented in a multi-apartment building in two configurations, i.e. centralized and decentralized.

- Improvement of the manifold design. In order to minimize the thermal loss along the manifold, the connection of the outlets in the manifold in two rows may be considered. By this, the heating coil part of the manifold (currently the top part) would be surrounded by the empty part (currently the bottom part). Thus, reducing the contact of the hot surface with the surrounding environment. Moreover, such a change in the design would result in more compact size of the manifold. Another possibility of further development of the system is to integrate a cooling coil in the manifold. This would enable the system both heating and cooling of the supply air.

- Implementation of the wireless control. Building automation and electronic monitoring of technical building systems have proven to be an effective replacement for inspections, in particular for large systems. The installation of such equipment should be considered to be the most cost-effective alternative to inspections in multi-apartment buildings.

LITERATURE LIST

[1] United Nations Environment Programme, Towards a zero-emission, efficient, and resilient buildings and construction sector, Global Status Report, 2017.

[2] International Energy Agency, Transition to Sustainable Buildings, Strategies and Opportunities to 2050, 2013.

[3] The European Parliament and the Council of the European Union, Directive 2010/31/EU, Energy Performance of Buildings, 2010.

[4] U.S. Department of Energy, The National Institute of Building Sciences, A Common Definition for Zero Energy Buildings, 2015.

[5] Building Performance Institute Europe, Europe's buildings under the microscope, A country-by-country review of the energy performance of buildings, 2011.

[6] U. Berardi, Moving to Sustainable Buildings: Paths to Adopt Green Innovations in Developed Countries, Versita, 2013.

[7] S. C. M. Hui, HVAC Design and Operation for Green Buildings, in: Proceedings of the Shaanxi-Hong Kong Refrigeration and HVAC Seminar, 11-13 June, Xian, China, 2001.

[8] A. Viswambharan, S. K. Patidar, K. Saxena. Sustainable HVAC systems in commercial and residential buildings, International Journal of Science and Research Publications, vol. 4, no. 4, 2014.

[9] A. GhaffarianHoseini, N. D. Dahlan, U. Berardi, A. GhaffarianHoseini, N. Makaremi, A. GhaffarianHoseini, Sustainable energy performances of green buildings: A review of current theories, implementations and challenges, Renewable and Sustainable Energy Reviews, vol. 25, 1-17, 2013.

[10] P. E. Nilsson, Achieving the Desired Indoor Climate, IMI Indoor Climate and Studentlitteratur, 2003.

[11] V. L. Erickson, A. E. Cerpa, Occupancy based demand response HVAC control strategy, in: Proceedings of the 2nd ACM Workshop on Embedded Sensing Systems for Energy-Efficiency in Buildings, 2 November, Zurich, Switzerland, 2010.

[12] European Commission Department of Energy, Energy Efficiency in Buildings, 2020. [Online]. Available:

https://ec.europa.eu/info/sites/info/files/energy_climate_change_environment/events/documents/in_focus_energy_efficiency_in_buildings_en.pdf

[13] W. Guo, M. Zhou, Technologies toward thermal comfort-based and energy-efficient HVAC systems: A review, in: Proceedings of the IEEE International Conference on Systems, Man and Cybernetics, 11-14 October, San Antonio, TX, USA, 2009.

[14] G. Brager, H. Zhang, E. Arens, Evolving opportunities for providing thermal comfort, Building Research and Information, vol. 43, no. 3, 274-287, 2015.

[15] V. Vakiloroayaa, B. Samalib, A. Fakharc, K. Pishghadamd, A review of different strategies for HVAC energy saving, Energy Conversion and Management, vol. 77, 738-754, 2014.

[16] A. Afram, F. Janabi-Sharifi, Theory and applications of HVAC control systems – A review of model predictive control (MPC), Building and Environment, vol. 72, 343-355, 2014.

[17] H. Wang, S. Wang, R. Tang, Development of grid-responsive buildings: Opportunities, challenges, capabilities and applications of HVAC systems in non-residential buildings in providing ancillary services by fast demand responses to smart grids, Applied Energy, Vol. 250, 697-712, 2019.

[18] A. Ghofrani, S. D. Nazemi, M. A. Jafari, HVAC load synchronization in smart building communities, Sustainable Cities and Society, Vol. 51, 101741, 2019.

[19] M. Fiorentini, P. Cooper, Z. Ma, Development and optimization of an innovative HVAC system with integrated PVT and PCM thermal storage for a net-zero energy retrofitted house, Energy and Buildings, Vol. 94, 21-32, 2015.

[20] W. Goetzler, M. Guernsey, J. Young, Research & Development Roadmap for Emerging HVAC Technologies, U.S. Department of Energy, Building Technologies Office, 2014. [Online]. Available:
https://www.energy.gov/sites/prod/files/2014/12/f19/Research%20and%20Development%20Roadmap%20for%20Emerging%20HVAC%20Technologies.pdf

[21] L. Erickson, Y. Lin, A. Kamthe, R. Brahme, A. Surana, A. E. Cerpa, M. D. Sohn, S. Narayanan, Energy efficient building environment control strategies using real-time occupancy measurements, In: Proceedings of the 1st ACM Workshop on Embedded Sensing Systems for Energy-Efficiency in Buildings, 4 November, Berkeley, California, 2009.

[22] E. Arens, C. Federspiel, D. Wang, and C. Huizenga, How Ambient Intelligence Will Improve Habitability and Energy Efficiency in Buildings. In: Ambient Intelligence, Springer, 2005.

[23] Executive Order on Building Regulations, Ministry of Transport, Building and Housing, The Building Regulations, Copenhagen, Denmark, 2018.

[24] D. Etheridge, M. Sandberg, Building Ventilation: Theory and Measurements, John Wiley, 1996.

[25] M. W. Liddament, A guide to energy efficiency ventilation, Air Infiltration and Ventilation Centre, 1996.

[26] C. Dimitroulopoulou, Ventilation in European dwellings: A review, Building and Environment, Vol. 47, 109-125, 2012.

[27] H. B. Awbi, Ventilation of Buildings, Routledge, 1995.

[28] V. Heide, L. Georges, A. G. Lien, H. M. Mathisen, Review of HVAC strategies for energy renovation of detached houses towards nZEB in cold climates, In: Proceedings of 1st Nordic conference on Zero Emission and Plus Energy Buildings 6–7 November, Trondheim, Norway, 2019.

[29] M. Kotol, C. Rode, G. Clausen, T. Rammer Nielsen, Indoor environment in bedrooms in 79 Greenlandic households, Building and Environment, Vol. 81, 29-36, 2014.

[30] Regulations on technical requirements for construction works: An unofficial English translation of the regulation "Forskrift om tekniske krav til byggverk (Byggteknisk forskrift - TEK17)" for information purposes, Ministry of Local Government and Modernization, 2017.

[31] Boverket's mandatory provisions and general recommendations, BBR, BFS 2011:6 with amendments up to BFS 2018:4, The Swedish National Board of Housing, Building and Planning, 2018.

[32] National Building Code of Finland, Decree of the Ministry of the Environment on the Energy Performance of New Buildings, The Ministry of the Environment, 2018.

[33] National Building Code of Finland, Decree of the Ministry of Environment on Amending the Decree of the Ministry of the Environment on Improving the Energy Performance of Buildings Undergoing Renovation or Alteration, The Ministry of Environment, Finland, 2018.

[34] P. Carrer, P. Wargocki, A. Fanetti, W. Bischof, E. De Oliveira Fernandes, T. Hartmann, S. Kephalopoulos, S. Palkonen, O. Seppänen, What does the scientific literature tell us about the ventilation–health relationship in public and residential buildings? Building and Environment, Vol. 94, 273-286, 2015.

[35] K. Engelund Thomsen, J. Rose, O. Mørck, S. Østergaard Jensen, I. Østergaard, H. N. Knudsen, N. C. Bergsøe, Energy consumption and indoor climate in a residential building before and after comprehensive energy retrofitting, Energy and Buildings, Vol. 123, 8-16, 2016.

[36] H. N. Knudsen, House owners' experience and satisfaction with Danish low-energy houses – focus on ventilation, In: Proceedings of CLIMA 2019 REHVA 13th HVAC World Congress, 26-29 May, Bucharest, Romania, 2019.

[37] M. Berge, H. M. Mathisen, Perceived and measured indoor climate conditions in high-performance residential buildings, Energy and Buildings, Vol. 127, 1057-1073, 2016.

[38] M. Fehrm, W. Reiners, M. Ungemach, Exhaust air heat recovery in buildings, International Journal of Refrigeration, Vol 25, no. 4, 439-449, 2002.

[39] A. Dodoo, L. Gustavsson, R. Sathre, Primary energy implications of ventilation heat recovery in residential buildings, Energy and Buildings, Vol. 43, 1566-1572, 2011.

[40] M. Morelli, L. Rønby, S. E. Mikkelsen, M. G. Minzari, T. Kildemoes, H. M Tommerup, Energy retrofitting of a typical old Danish multi-family building to a "nearly-zero" energy building based on experiences from a test apartment, Energy and Buildings, Vol. 54, 395-406, 2012.

[41] A. A. Hamid, H. Bagge, D. Johansson, Measuring the impact of MVHR on the energy efficiency and IEQ in multifamily buildings, Energy and Buildings, Vol. 195, 93-104, 2019.

[42] D. Mortensen, Demand controlled ventilation for multi-family dwellings, PhD Thesis, 2011.

[43] A. Afshari, N. C. Bergsøe, Reducing energy consumption for ventilation in dwellings through demand controlled ventilation, In: Proceedings of the 10th Indoor Air Conference, 4-9 September, Beijing, China, 2005.

[44] T. R. Nielsen, C. Drivsholm, Energy efficient demand controlled ventilation in single family houses, Energy and Buildings, Vol. 42, no. 11, 1995-1998, 2010.

[45] V. Pavlovas, Demand controlled ventilation: A case study for existing Swedish multifamily buildings, Energy and Buildings, Vol. 36, no. 10, 1029-1034, 2004.

[46] A. Hesaraki, Low-temperature heating and ventilation systems for sustainability in energy-efficient buildings, PhD Thesis, 2015.

[47] A. Hasan, J. Kurnitski, K. Jokiranta, A combined low temperature water heating system consisting of radiators and floor heating, Energy and Buildings, Vol. 41, 470-479, 2009.

[48] O. B. Kazanci, B. W. Olesen, Beyond nearly-zero energy buildings: Experimental investigation of the thermal indoor environment and energy performance of a single-family house designed for plus-energy targets, Science and Technology for the Built Environment, Vol. 22, no. 7, 1024-1038, 2016.

[49] B. W. Olesen, Radiant Floor Heating in Theory and Practice, ASHRAE Journal, Vol. 44, no. 7, 19–24, 2002.

[50] K-N. Rhee, B. W. Olesen, K. W. Kim, Ten questions about radiant and cooling systems, Building and Environment, Vol. 112, 367-381, 2017.

[51] A. Prozuments, A. Borodinecs, Indoor air stratification in warm air supply systems, ASHRAE Journal, Vol. 59, 54-65, 2017.

[52] Z. A. Babawale, L. Serive-Mattei, J. Litter, Domestic ducted forced-air heating: Duct system leakage and heating energy use, Building Services Engineering Research and Technology, Vol. 14, no. 4, 129-135, 1993.

[53] L. Hultmark, Kombinera ventilation och uppvärmning ur ett energiperspektiv, Master Thesis, 2016.

[54] W. Feist, J. Schnieders, V. Dorer, A. Haas, Re-inventing air heating: Convenient and comfortable within the frame of Passive House concept, Energy and Buildings, Vol. 37, 1186–1203, 2005.

[55] L. Georges, M. Berner, H. M. Mathisen, Air heating of passive houses in cold climates: Investigation using detailed dynamic simulations, Energy and Environment, Vol. 74, 1-12, 2014.

[56] M. Berge, H. M. Mathisen, The suitability of air-heating in residential passive house buildings from the occupants' point of view – a review, Advances in Building Energy Research, Vol. 9, no. 2, 175-189, 2015.

[57] M. Berge, J. Thomsen, H. M. Mathisen, The need for temperature zoning in high-performance residential buildings, Journal of Housing and Built Environment, Vol. 32, 211-230, 2017.

[58] M. Berge, H. M. Mathisen, Perceived and measured indoor climate conditions in high-performance residential buildings, Energy and Buildings, Vo. 127, 1057-1073, 2016.

[59] M. Berge, L. Georges, H. M. Mathisen, On the oversupply of heat to bedrooms during winter in highly insulated dwellings with heat recovery ventilation, Building and Environment, Vol. 106, 389-401, 2016.

[60] L. Georges, E. Selvnes, V. Heide, H. M. Mathisen, Energy efficiency of strategies to enable temperature zoning during winter in highly-insulated residential buildings equipped with balanced mechanical ventilation, In: Proceedings of 1st Nordic conference on Zero Emission and Plus Energy Buildings 6–7 November, Trondheim, Norway, 2019.

[61] H. Amai, S. Liu, A. Novoselac, Experimental study on air change effectiveness: Improving air distribution with all-air heating systems, Building and Environment, Vol. 125, 515-527, 2017.

[62] G. Gan, Evaluation of room air distribution systems using computational fluid dynamics, Energy and Buildings, Vol. 23, 83–93, 1995.

[63] E. Ridouane, Evaluation of thermal comfort in low load homes supplied by high sidewall air jets, ASHRAE Transactions, Vol. 118, 322-329, 2002.

[64] A. Prozuments, A. Borodinecs, Indoor air stratification in warm air supply systems, ASHRAE Journal, Vol. 59, 54-65, 2017.

[65] Intelligent and systematic residential ventilation, Ventilation units, duct systems, diffusers and accessories, System Air. [Online]. Available: https://www.systemair.com/support/media-center/residential-ventilation/

[66] Lindab InDomo, Integrated duct system for comfortable ventilation in residential buildings, Lindab. [Online]. Available: https://itsolution.lindab.com/LindabWebProductsDoc/PDF/Documentation/ADS/Lindab/Technical/InDomo.pdf

[67] HERU 100 S EC datasheet, Östberg. [Online]. Available: https://fsp.ostberg.com/products/4328/heru-100-s-ec/?region=se&lang=en

[68] El-varme manual, Metro Therm. [Online]. Available:

https://www.metrotherm.dk/download/18.74430b7416a10f4435f911/155721639388 6/Elvandvarmere-Model-55-60-110-160-Manual-08011-1805.pdf

[69] Regula Combi 1.5 control equipment. [Online]. Available: https://itsolution.lindab.com/LindabWebProductsdoc/pdf/Documentation/Comfort/Lindab/Technical/Regula_Combi_1_5.pdf

[70] J. Polak, A, Afshari, P. Sadeghian, C. Wang, S. Sadrizadeh, Improving the performance of heat valve ventilation system: A study on the provided thermal environment, Building and Environment, Vol. 164, 106338, 2019.

[71] F. P, Incropera, D. P. DeWitt, T. L. Bergman, A. S. Lavine, Fundamentals of heat and mass transfer Sixth Edition, John Wiley & Sons, 2011.

[72] DS/EN Standard 13182, Ventilation for buildings – Instrumentation requirements for air velocity measurements in ventilated spaces, European Committee for Standardization, 2002.

[73] S. Sadrizadeh, A. Afshari, T. Karimipanah, U. Håkansson, P.V. Nielsen, Numerical simulation of the impact of surgeon posture on airborne particle distribution in a turbulent mixing operating theatre, Building and Environment, Vol. 110, 140-147, 2016

[74] C. Wang, S. Holmberg, S. Sadrizadeh, Numerical study of temperature-controlled airflow in comparison with turbulent mixing and laminar airflow for operating room ventilation, Building and Environment, Vol. 144, 45-56, 2018.

[75] J. Srebric, V. Vukovic, X. Yang, CFD boundary conditions for contaminant dispersion, heat transfer and airflow simulations around human occupants in indoor environments, Building and Environment, Vol. 43, 294-303, 2008.

[76] JCGM 100:2008, Evaluation of measurement data – Guide to the expression of uncertainty in measurements, Joint Committee for Guides in Metrology, 2008.

[77] ANSI/ASHRAE Standard 55-2017, Thermal environmental conditions for human occupancy, American Society of Heating, Refrigerating and Air- Conditioning Engineers, 2017.

[78] DS/EN ISO Standard 7730, Ergonomics of the thermal environment – Analytical determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort criteria, European Committee for Standardization, 2006.

[79] J. Jurelionis, E. Isevicius, CFD predictions of indoor air movement induced by cold window surfaces, Journal of Civil Engineering and Management, Vol. 14, 29–38, 2008.

ISSN (online): 2446-1636
ISBN (online): 978-87-7210-626-7

AALBORG UNIVERSITY PRESS