THESIS FOR THE DEGREE OF DOCTOR OF PHILOSOPHY

Self-Regulating Active Chilled Beams

Peter Filipsson

Building Services Engineering
Department of Architecture and Civil Engineering
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2020

Self-Regulating Active Chilled Beams
PETER FILIPSSON
ISBN: 978-91-7905-263-8

Doktorsavhandlingar vid Chalmers tekniska högskola
Ny serie nr: 4730
ISSN 0346-718X

Department of Architecture and Civil Engineering
Chalmers University of Technology
SE-412 96 Gothenburg
Sweden
Telephone + 46 (0)31-772 1000

Chalmers Reproservice
Gothenburg, Sweden 2020

Self-Regulating Active Chilled Beams
PETER FILIPSSON
Department of Architecture and Civil Engineering
Chalmers University of Technology

Abstract

By designing comfort cooling systems to operate with higher chilled water temperatures, electricity driven chillers may be replaced by natural sources of free cooling. It also enables the possibility of self-regulation which, due to simplicity and robustness, brings cost-savings both during installation and maintenance of the building. Active chilled beams is one technology especially well suited for this high temperature cooling.

The objective of this thesis is to investigate the function of self-regulating active chilled beams in order to improve design and operation of such systems. Based on experience from existing systems, a hypothesis is that current models and knowledge do not fully capture the behavior of self-regulating active chilled beams.

The work has been carried out through measurements in an experimental facility, building performance simulations and analysis of operational data from an actual building. The purpose of the measurements was to develop a model of an active chilled beam and also to validate a modified zone model used to simulate the thermal climate in a room equipped with an active chilled beam. The purpose of the simulations was to determine the peak shaving effect of self-regulating active chilled beams and to study the influence of central control of the chilled water temperature. The purpose of the analysis of operational data was to study the performance of an actual self-regulating active chilled beam system with respect to energy use and indoor air temperatures.

The results show that a higher chilled water temperature generates more induction of room air in the active chilled beam, which is beneficial for the cooling capacity. Induction of room air also increases the internal convective heat transfer in the room, which reduces the required heat transfer area of the active chilled beam. As a consequence of self-regulation, buildings are precooled prior to the peak cooling load, further reducing the required heat transfer area. By increasing the chilled water temperature outside of the summer season, overcooling is avoided without causing thermal discomfort. The analysis of operational data demonstrates the ability to achieve the desired indoor temperatures with self-regulating active chilled beams without the use of a chiller.

Keywords: Active chilled beams, high temperature cooling, self-regulation, comfort cooling

This project was funded by SBUF
(The Swedish construction industry's organisation for research and development)

Acknowledgements

This work has been carried out in collaboration between the Division of Building Services Engineering at Chalmers University of Technology, CIT Energy Management AB and Skanska Commercial Development Nordic AB. The work was funded by The Swedish construction industry's organisation for research and development (SBUF).

I am forever grateful for all the support and guidance that have been given to me during the work of this thesis. First, I would like to thank my supervisors Jan-Olof Dalenbäck, Jonas Gräslund and Anders Trüschel. Your commitment, visions and expertise have been crucial for the progress and completion of this work.

I would also like to express appreciation to my dear colleagues at CIT Energy Management. You are an invaluable source of knowledge, experience and well-being.

Furthermore, I wish to acknowledge the supportive and pleasant environment at the Division of Building Services Engineering. Thank you all. Special thanks to Taha Arghand for support and fruitful discussions. Thank you Håkan Larsson, nothing of the practical work in the laboratory would have been completed or even started without you.

Thank you, Ewan Barr, for linguistic support and for teaching me how to use semicolon.

Nothing of this would have been possible without the unconditional love and support from my family. Your contribution to this work started long before I knew anything about it. Finally, my deepest gratitude is reserved for Elin. Thank you for your consistent encouragement, for keeping me aware of what is important and for bringing us the greatest joy in life. I love you both.

Peter Filipsson
Gothenburg, March 2020

List of publications

This thesis is based on the following appended papers. Paper I, II and IV are peer-reviewed journal articles, paper III and V are submitted for journal publication and paper VI is submitted to a conference.

Paper I	Filipsson P, Trüschel A, Gräslund J, Dalenbäck J-O (2016). Induction ratio of active chilled beams - Measurement methods and influencing parameters. *Energy and Buildings*, 129: 445-451.
Paper II	Filipsson P, Trüschel A, Gräslund J, Dalenbäck J-O (2017). A thermal model of an active chilled beam. *Energy and Buildings*, 149: 83-90.
Paper III	Filipsson P, Trüschel A, Gräslund J, Dalenbäck J-O (2020). Modelling of rooms with active chilled beams. Submitted to *Journal of Building Performance Simulation*.
Paper IV	Filipsson P, Trüschel A, Gräslund J, Dalenbäck J-O. (2020). Performance evaluation of a direct ground-coupled self-regulating active chilled beam system. *Energy and Buildings*, 109691.
Paper V	Filipsson P, Trüschel A, Gräslund J, Dalenbäck J-O (2020). Peak shaving effect of self-regulating active chilled beams. Submitted to *Journal of Building Performance Simulation*.
Paper VI	Filipsson P, Trüschel A, Gräslund J, Dalenbäck J-O (2020). Supply temperature control of self-regulating active chilled beams. Submitted to *BuildSim Nordic 2020* (Abstract accepted).

As the first author of all papers, Peter Filipsson has performed the studies and written the papers. The co-authors, Anders Trüschel (co-supervisor), Jonas Gräslund (co-supervisor) and Jan-Olof Dalenbäck (main supervisor and examiner) have contributed with feedback and comments during the studies and the writing process.

Following publications authored or co-authored by Peter Filipsson are also part of the PhD project but not included in the thesis.

Publication 1	Filipsson P, Gräslund J (2017) Självverkande kylbaffelsystem. Energi & Miljö 88 (5) 48-51.
Publication 2	Filipsson P, (2017) Termisk komfort med självverkande kylbafflar. Energi & Miljö 88 (6-7) 40-42.
Publication 3	Jangsten M, Filipsson P, Lindholm T, & Dalenbäck J-O. (2020). High temperature district cooling: Challenges and possibilities based on an existing district cooling system and its connected buildings. *Energy*, 117407.

Nomenclature

Symbols

$c_{p,w}$	Specific heat capacity of water	[J/kgK]
C_{min}	Minimum heat capacity rate	[W/K]
ΔP	Pressure difference	[Pa]
ΔT	Temperature difference	[K]
ε	Effectiveness	[-]
h	Convective heat transfer coefficient	[W/m^{2}K]
k	Empirical coefficient	[-]
$\dot{m}_w$	Water mass flow rate	[kg/s]
P_w	Waterside cooling capacity	[W]
$q"_{conv}$	Convective heat flux	[W/m^2]
t_a	Air temperature	[°C]
t_g	Globe temperature	[°C]
t_{op}	Operative temperature	[°C]
$\bar{t}_r$	Mean radiant temperature	[°C]
$t_{w,in}$	Chilled water supply temperature	[°C]
$t_{w,out}$	Chilled water return temperature	[°C]
t_s	Surface temperature	[°C]
v_a	Air velocity	[m/s]
$\dot{V}_{pri}$	Primary air flow rate	[l/s]
$\dot{V}_{sec}$	Secondary air flow rate	[l/s]

Abbreviations

ACB	Active chilled beam
ACH	Air changes per hour
AHU	Air handling unit
APE	Absolute percentage error
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
BSE	Building services engineering
COP	Coefficient of performance
COPA	Coil output to primary air flow ratio
HVAC	Heating, ventilation and air-conditioning
IR	Induction ratio
LW	Long-wave
NTU	Number of transfer units
P-control	Proportional control
PI-control	Proportional-integral control
PMV	Predicted mean vote
RMSE	Root mean square error
RTD	Resistance temperature detector
SW	Short-wave
TABS	Thermally activated building systems
VAV	Variable air volume
Z	Zone

Definitions

Cooling capacity	The rate at which heat is extracted from a space by the cooling equipment (term used when studying the cooling equipment).
Cooling load	The rate at which heat must be extracted from a space to maintain a constant room temperature.
Design cooling capacity	The rate at which heat is extracted from a space by the cooling equipment under specified conditions of operation.
Heat extraction rate	The rate at which heat is extracted from a space by the cooling equipment (term used when studying the heat balance of the space).
Heat gain	The rate at which heat is transferred to or generated within a space.
High temperature cooling	Operation at a chilled water supply temperature ≥ 20.0 °C in active chilled beams and ≥ 17.0 °C in air handling unit cooling coils.
Induction ratio	The volumetric flow rate of secondary air divided by the volumetric flow rate of primary air.
Internal convection	Convective heat transfer between the air and the inner surfaces of a room.
Latent cooling	Reduction of humidity by extraction of heat.
Operative temperature	The uniform temperature of an enclosure in which an occupant would exchange the same amount of heat by radiation and convection as in the actual nonuniform environment.
Overcooling	Excessive cooling capacity.
Oversizing	Excessive design cooling capacity.
Primary air	Air delivered from the air handling unit to the active chilled beam.
Reference temperature	The air temperature determining the internal convection of a zone model.
Room temperature	A general term covering both the air and operative temperature of a room.
Secondary air	Room air induced into the active chilled beam.
Self-regulating ACB	An active chilled beam without a conventional individual/zonal control system which instead relies on the self-regulating effect.
Self-regulating effect	The counteracting effect that a change of room temperature has on the cooling capacity.
Sensible cooling	Reduction of temperature by extraction of heat.
Set-point	The target value which a control system tries to maintain.
Space	Room (in a more general sense).
Stability	Absence of temporal variability.
Supply air	Air discharged from the active chilled beam to the room.
Uniformity	Absence of spatial variability.
Zone	Model of a room, a group of rooms or a part of a room with uniform air temperature.

The term *Heat transfer area* of the active chilled beams is used several times in this thesis. The correct interpretation of this term is *The product of heat transfer area and overall heat transfer coefficient,* also known as UA-value with the unit W/K. The shorter term is used for reasons of readability.

Unless otherwise specified, cooling capacity of the active chilled beam refers to cooling provided by the chilled water, excluding additional cooling provided by the primary air flow.

Table of contents

1 Introduction

1.1 Background

Comfort cooling is the fastest growing end-use of energy in buildings. While global demand for space heating remained stable during 1990-2014, the demand for comfort cooling doubled in OECD countries and almost quintupled in non-OECD countries (IEA, 2017). Recently, The International Energy Agency proclaimed the growth in global demand for comfort cooling as a blind spot of energy policy and one of the most critical energy issues of our time (IEA, 2018); this was just before the record-breaking hot summer of 2018.

The rising demand for comfort cooling is not a new issue though. Brown (1990) pointed out that as early as the 1960s it had become problematic to maintain comfortable thermal climate in Swedish offices due to larger windows, higher levels of artificial lighting, lighter structures and better insulated building envelopes. While similar factors are still pointed out as contributing to the growing demand for comfort cooling (proliferation of fashionable glass facades and thermal insulation (Aebischer et al., 2007)), there are also other, non-building specific factors in play. Population growth, intensification of urban heat islands, climate change and an increasing desire for (and ability to afford) thermal comfort are pointed out as major drivers for the growing demand (IEA, 2018).

Cooling is primarily generated by chillers, predominantly electricity driven compressor chillers and in some cases heat driven absorption chillers. Chillers usually generate cooling at temperature levels far below what is associated with human thermal comfort, often around 5 °C (versus 21-25 °C). The cooling is then distributed from the chiller to the occupied space by air and/or chilled water. Active chilled beams is a technology well suited for utilizing chilled water at relatively high temperatures. This is beneficial for the performance of the chillers but may also unlock the possibility of avoiding chillers altogether and instead using natural sources of free cooling such as from the ground, nearby bodies of water (rivers, lakes etc.) or the outdoor air.

Another opportunity unlocked by the use of relatively high temperature of chilled water is self-regulation. The cooling capacity of an active chilled beam is determined by the difference between the temperatures of room air and chilled water. If designing for a small temperature difference, even small changes in room temperature will have large counteracting effect on the provided cooling capacity. Self-regulating active chilled beams avoid control components (controllers, valves and thermostats) and associated costs during both installation and maintenance.

The commercial property development part of the construction and development company Skanska AB has more than 20 years of experience of high temperature cooling and since 2002 also of self-regulating systems. For Skanska, the main reason for using self-regulating active chilled beams is the robustness of avoiding control components that often malfunction and require maintenance or to be replaced. Jonas Gräslund, Technical Director at Skanska Commercial Development Nordic AB and adjunct professor at Chalmers University of Technology, experiences that these systems perform better in reality than expected based on simulations, which is the reason why he initiated and co-supervised the work presented in this thesis.

1.2 Objectives

The objective of this thesis is to investigate the function of self-regulating active chilled beams in order to improve design and operation of such systems. The work presented in this thesis aims at giving answers to the following three research questions.

- Are current models adequate for simulating self-regulating active chilled beams?
- Are self-regulating active chilled beams able to provide thermal comfort in Swedish office buildings?
- What are the consequences of different design and operation of self-regulating active chilled beams?

Current models refer to well-established models generally used by practitioners.

1.3 Scope and limitations

The type of active chilled beam analyzed in this thesis is mounted in the suspended ceiling and induces room air passing upwards through the cooling coil. This air flow is then mixed with primary air from the ventilation system to form the supply air. The supply air attaches to the ceiling by the Coandă effect when discharged into the room (this design is further explained in chapter 2.2). Other designs are not explicitly dealt with in this thesis. Following topics are closely related to self-regulating active chilled beams, and thereby relevant, but beyond the scope of this thesis.

- **Local thermal comfort** associated with active chilled beams, particularly draught, has been investigated in several studies. These studies deal with conventionally controlled low temperature active chilled beams. By comparison, self-regulating active chilled beams have higher chilled water temperature (hence higher supply air temperature), they are longer (hence lower supply air velocity) and the cooling capacity is continuous. It is reasonable to assume that these features reduce the risk of local thermal discomfort such as draught and radiant asymmetry.
- **Design of ground heat exchangers** and other topics related to the generation or seasonal storage of high temperature cooling are essential when working with direct ground-coupled systems. Design and operation of ground heat exchangers as well as chillers and district cooling are beyond the scope of this thesis.
- **Internal convective heat transfer** in rooms is complex to determine and an important part of building simulation. This thesis includes neither any measurements of convective heat transfer coefficients nor a comparison between existing models, but instead relies on models which were developed through previous research and are now widely established in building performance simulation software.
- **Physical design** of active chilled beams regarding geometry, nozzle configurations, fin spacing, tube circuitry etc. is not investigated in this thesis.

1.4 Method

The work presented in this thesis was carried out in a bottom-up manner where details were studied before investigating entire systems on a building level. The applied methods can be divided into following four categories.

- Literature review
- Experiments in a controlled laboratory environment
- Analysis of operational data from a real building
- Modelling and simulations

The purpose of the literature review was to gain insight into the previous research on active chilled beams, self-regulating systems and cooling load calculations. In this thesis, the literature review is not presented in a separate chapter. Instead, it is an integrated part of the theory and previous research presented in chapter 2.

The controlled laboratory environment refers to the Building Services Engineering Laboratory at Chalmers University of Technology and is further described in chapter 3. The purpose of the experiments in the laboratory was threefold:

- Comparison of measurement methods (Paper I)
- Investigation of influencing parameters (Paper I)
- Model validation, both on component level (Paper II) and on room level (Paper III)

The real building (described and analyzed in Paper IV) is the headquarters of Skanska AB, located in Stockholm and was taken into operation in 2014. The operational data originate from the permanent building management system. The data were used only for analyzing the performance of the building and not for model validation.

A thermal model of an active chilled beam (Paper II) was developed in the numerical computing environment MATLAB. MATLAB (version R2014a and version R2016b) was also used for analysis and visualization of data in all parts of this thesis.

Building performance simulations, included in Paper III, Paper V and Paper VI, were carried out with IDA Indoor Climate and Energy version 4.8 (henceforth referred to as IDA ICE) developed by EQUA Simulation AB. The rationale for choosing IDA ICE was the transparency of the models, the ability to adjust and implement self-developed models and a strong belief that IDA ICE is the predominant software for simulating active chilled beam systems in Sweden. IDA ICE is based on the heat balance method to determine conduction, convection and radiation in each zone, surface and part of a building. Validation has been carried out in several research projects, both validation of the heat balance method in general (Chantrasrisalai, 2003) and validation of IDA ICE specifically, both by measurements (Loutzenhiser, 2007) and by inter-model comparison through ANSI/ASHRAE Standard 140-2004 (EQUA, 2010).

While the literature review covered all levels, a schematic description of the scope of the other methods is presented in Figure 1.

	ACB	Room	Building
Measurements	BSE Laboratory		Skanska HQ
Modelling	Matlab	IDA ICE	

Figure 1 Categorization of scope and methods.

1.5 Thesis outline

The purpose of the thesis is to summarize the work carried out and to put the appended papers in context. This is done by providing a background upon which the papers are written and a discussion of the implications of the results.

Chapter 2 presents a theoretical foundation and previous research relevant for the appended papers. This chapter is intended to present basic concepts and to give the reader an understanding of why each paper was written.

Chapter 3 describes the experimental resources used in the studies presented in Papers I-III.

Chapter 4 presents the results obtained from each paper. The chapter is subdivided into papers and includes also a brief presentation of the purpose and method of each paper. For deeper insight into the details of each paper, readers are referred to the appended full-versions.

In chapter 5, the results are discussed with respect to previous research, partly presented in chapter 2. Practical implications of the results are discussed and some additional results, not included in the papers, are also presented. In contrast to chapter 4, it is subdivided into topics rather than papers.

Conclusions are presented in chapter 6 and suggestions for further work are listed in chapter 7.

2 Theory and previous research

This chapter presents basic theory, both as presented by others but also elaborated upon to demonstrate the principles of self-regulating active chilled beams. It also presents results and conclusions from previous research relevant for this thesis.

2.1 Comfort cooling

Comfort cooling refers to the extraction of heat from a space to prevent thermal discomfort. Globally, comfort cooling represents a considerable part of the total energy use. In 2010, annual energy use for comfort cooling of commercial (non-residential) buildings was around 560 TWh (mainly electricity). This was 6.7 % of the total energy use of commercial buildings. By 2050, this is predicted to increase to 1550 TWh, but due to uncertainties regarding population growth, efficiency of chillers, climate change etc. somewhere in the range of 900-2610 TWh is probable. Though focus of this thesis is on commercial buildings, global energy use for comfort cooling in residential buildings is both higher (687 TWh in 2010) and expected to grow faster (to around 5270 TWh in 2050) (Santamouris, 2016).

According to more recent statistics, global electricity use for comfort cooling was 2075 TWh in 2018, corresponding to a twofold increase since 2000, a threefold increase since 1990 and more than 10 % of the total global electricity use (IEA, 2019).

Comfort cooling is primarily associated with hot climates, but it is also common practice in commercial buildings in colder climates such as in Northern Europe. Due to high heat gains and low transmission losses, many commercial buildings require comfort cooling even at very low outdoor air temperatures. Core parts of an office building in Sweden may require comfort cooling all year round while the cooling load of perimeter parts are more affected by the seasonal weather variations.

The cooling is normally distributed throughout the buildings either only by air or by a combination air and water. These are often referred to as all-air systems and air-and-water systems respectively. A system using water for distribution is also often referred to as a hydronic system. All-air systems can be categorized based on the ductwork arrangement between the air handling unit and the conditioned space (single-zone, reheat, variable air volume (VAV), dual-duct, multizone etc.) with the common denominator that only air is used for distribution of the cooling (McQuiston et al., 2004).

The systems dealt with in this thesis belong to the category of air-and-water systems. In such systems, water is used for the sensible cooling load while the air flow ensures the required indoor air quality and provides latent cooling. However, the relatively cold air inevitably also provides sensible cooling. One benefit of using water instead of air is that the volumetric heat capacity of water is around 3500 times higher than that of air. As a consequence, the pipes in a hydronic system require much less space than the ducts in an all-air system and the pumps in the hydronic system require less electricity than the fans in the all-air system. The smaller space required may be utilized by lower floor-to-floor heights and larger rentable floor area.

Consequently, the main benefit of an all-air system is that it requires no pipework while the main benefits of an air-and-water system are that it requires less space for ductwork and air handling units as well as less electricity for distribution. An important difference is also that all-air systems are more influenced by the outdoor air temperature. This facilitates the use of free cooling when it is cold outside but requires a higher peak cooling capacity when it is hot outside. Many other factors also influence the choice between all-air and air-and-water systems, whereof one is the fact that the temperature of the chilled water in a hydronic system is usually higher than in the cooling coil in the air handling unit. Additional advantages of hydronic systems are lower risk of noise and draught compared to an all-air system (Vastyan, 2011).

While all-air systems are very common in North America and in many parts of the world influenced by the USA, hydronic cooling systems are more common in Europe. Furthermore, active chilled beams are more common in Northern Europe while thermally activated building systems (TABS) are more common in central Europe. Influencing factors may be outdoor climate, the power of traditions and different calculation periods affecting the valuation between investment costs and running cost. Characteristic parameters for comfort cooling in Northern Europe are low outdoor air temperature and humidity, high heat gain from (vertical) windows and well-insulated and airtight building envelopes as a consequence of minimizing heat losses during the winter season.

2.2 Active chilled beams

Chilled beams are divided into passive and active chilled beams. Active beams are connected to the ventilation ductwork while passive beams are not and the transfer of heat rely on natural convection. Active beams may be used also for heating. *Active beams* is a more general term including both active chilled beams and active beams used for heating. Practical guidance to design, commission and operate both active and passive beam systems are presented in the Active and Passive Beam Application Design Guide published by REHVA and ASHRAE (Woollett & Rimmer, 2015).

The operation of a very common type of active chilled beam is illustrated in Figure 2. Primary air is provided from an air handling unit while chilled water is circulated through a cooling coil (a finned tube heat exchanger) inside the active chilled beam. The primary air enters the active chilled beam through a pressure plenum and several nozzles along the beam. The high air velocity generated by the nozzles reduces the static pressure and induces an upward flow of room air (called secondary air). The secondary air cools down as it passes the coil before it mixes with the primary air. The mixed air is then discharged into the room through slots along both long sides of the beam and attaches to the ceiling by means of the Coandă effect. The flow or supply temperature of chilled water is usually controlled as a function of the room air temperature in order to match the cooling capacity to the actual demand.

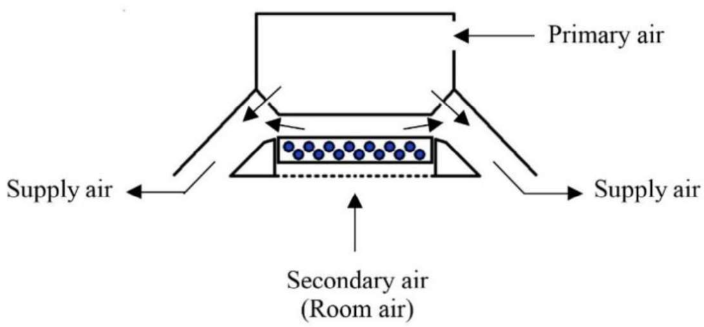

Figure 2 Schematic diagram of an active chilled beam

The secondary air flow rate divided by the primary air flow rate defines the induction ratio, which is an important parameter and hence further discussed later in this chapter. As a consequence of efficient heat transfer, active chilled beams are suitable for using chilled water of relatively high temperature. Conventionally in the range of 14-16 °C, but as investigated in this thesis, also around 20 °C. This provides benefits such as increased use of free cooling, better performance of chillers, reduced risk of condensation (reduced latent load), reduced thermal losses in the chilled water distribution system and a stronger self-regulating effect (explained in chapter 2.3). The disadvantage of a high chilled water temperature is that it requires more heat transfer area.

It shall be noted that a major part of the results presented in this thesis is applicable only to active chilled beams with an upwards flow of air through the cooling coil. Although being rare, there are other designs where the air flows horizontally or downwards through the cooling coil. Examples of these designs are illustrated in Figure 3.

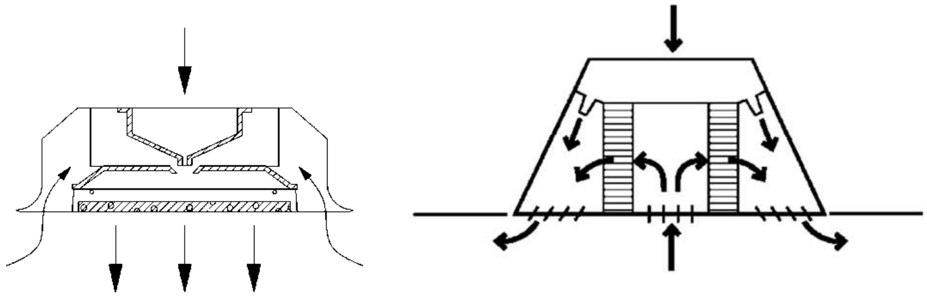

Figure 3 Active chilled beams with downwards (left) and horizontal (right) air flow through the cooling coil, figures published by Wang et al. (2019) and Rumsey & Weale (2007).

Active chilled beam cooling is often described as a technology well-established in Europe (and Australia) and during the last decades becoming more common in North America (and Asia). As a consequence, there are several articles comparing the "upcoming" active chilled beam technology with the more prevalent all-air VAV-technology. Rumsey (2010) concluded that active chilled beams was the better option regarding first cost, energy use, indoor air quality as well as floor-to-floor heights. A comparable comparison was made by Stein & Taylor (2013) who conversely concluded that all-air VAV was the better option regarding first cost, energy use, indoor air quality and possibly even floor-to-floor heights. It is obvious that the results of the comparisons depend on a multitude of conditions and assumptions. Other comparisons between active chilled beams and all-air systems have resulted in a better mix of advantages and drawbacks (Alexander & O'Rourke, 2008; Loudermilk, 2009).

Setty (2011) pointed out condensation on the coil surface, high maintenance and the inability to provide latent cooling as three potential limitations of active chilled beams. Due to the lack of a condensation drainage system (drain pan), any condensation on the coil surface will cause droplets of water falling down into the room and should strictly be avoided. A common strategy, referred to as dew point control, is to control the chilled water supply temperature to never be below 1 °C higher than the indoor air dew point. Obviously, this reduces the cooling capacity and calls for dehumidification of the primary air in the air handling unit. However, this issue is less relevant when designing for high chilled water temperature. Furthermore, the high maintenance is alleviated by reducing the amount of moving parts such as in a self-regulating system.

Livchak & Lowell (2012) pointed out that many active chilled beam systems are being oversized and as a consequence, all required cooling is provided by the primary air and the chilled water control valve is shut all the time. Hence, the potential benefits compared to an all-air system is not at all exploited. To address this, the authors introduced a parameter called the coil output to primary air flow ratio (COPA). The higher the COPA ratio, the more efficient use of primary air. While definitely being an important parameter, it is worth noting that the primary air temperature in the systems included in this thesis is 20 °C. In the example presented by Livchak & Lowell (2012), the primary air temperature was 12.8 °C.

Livchak & Lowell (2012) also presented a system of equations describing the cooling capacity of an active chilled beam including empirical equations for the coil heat transfer coefficient. However, Maccarini et al. (2015) concluded that manufacturers generally do not provide enough data to derive the parameters needed in the model presented by Livchak & Lowell (2012) and developed a new model with the programming language Modelica. The model was validated both in cooling and in heating mode and it was concluded that it corresponded closely with actual operation. Also Chen et al. (2014) developed an active chilled beam model, including dynamic behavior, and concluded that it performed satisfactorily during both static and dynamic operation. They noted that the model was useful in real-time control and optimization applications, e.g. to avoid condensation, but also called for simplifications of the complexity of the model. De Clercq et al. (2013) used the NTU-effectiveness (number of transfer units) method to model heating and cooling of active beams and concluded that the results showed very good compliance with measured data.

A review of how active chilled beams are modelled in five common building simulation software was presented by Betz et al. (2012). It was concluded that there are multiple issues with modelling active chilled beams regarding either accuracy, ease of use and/or cost. It was also concluded that software developers seem to receive too little feedback from the end-users of their products.

- ➢ Modelling of the cooling capacity of active chilled beams is the focus of Paper II.

Induction ratio

The induction of secondary air is a key to obtaining a high cooling capacity from an active chilled beam. A large flow of induced secondary air increases both the airside convective heat transfer coefficient and the driving temperature difference. Different manufacturers, researchers and associations use various definitions when quantifying the induction of secondary air. In this thesis, only the term induction ratio is used and it is defined as *the volumetric flow rate of secondary air divided by the volumetric flow rate of primary air* according to equation 1 (see Figure 2).

$$IR = \frac{\dot{V}_{sec}}{\dot{V}_{pri}} \qquad \text{(Equation 1)}$$

A phenomenon related to induction, and sometimes causing confusion, is entrainment. While induction is defined as *movement of space air into an air device*, entrainment is defined as *air drawn into an air jet because of the pressure differential caused by the airstream discharged from the outlet* (ASHRAE, 2017).

Though crucial for obtaining a high cooling capacity and necessary for determining the flow rate and temperature of supply air, the induction ratio generally gets limited attention. In the building performance simulation software IDA ICE, the induction ratio is not taken into account. In another major building performance simulation software, EnergyPlus, it is stated that good default values are supplied (the default induction ratio for active chilled beams is 2) and they should not be changed without information from the manufacturer (DOE, 2018). However, manufacturers of active chilled beams provide little information about induction ratio. This claim is based on reviewing technical data specifications of active chilled beams from five of the top manufacturers present on the Swedish market. While all of these include data about dimension, pressure drop, cooling capacity and sound level, only one includes any numeric specification of the induction ratio: *The volume of the room air is two to seven times that of the ventilation air* (Lindab, 2019). Of five additional manufacturers present on the global market, only one specifies the induction ratio of their product: *The induction ratio of [...] is typically 5 times that of the supply air (fresh air) rate* (Frenger, 2014). This lack of data may originate from a disinterest among the customers and/or the fact that the induction ratio is not one of the parameters required to be reported to achieve Eurovent certification.

The induction ratio of an active chilled beam is to a large extent influenced by its design regarding the size of the nozzles, the airside pressure drop characteristics of the cooling coil etc. (Ruponen, 2009). While the induction ratio benefits from a small nozzle area, this requires a higher plenum pressure and causes higher sound levels. A low cooling coil pressure drop is beneficial for the induction ratio but hard to obtain when a large heat transfer area is required. This implies some intricate trade-offs when designing active chilled beams, which is a process not included in this thesis. However, for a given design, the induction ratio is also influenced by the operating conditions.

Most previous studies on how the induction ratio is influenced by the operating conditions focus on plenum pressure and primary air flow. Some have reported a negative correlation (Ruponen & Tinker, 2009; Guan & Wen, 2016), some have reported a positive correlation (Chen et al., 2014), some have reported that the induction ratio peaks at a certain pressure (Hyun et al., 2014) and some have reported that the correlation is very weak (Cammarata & Petrone, 2008).

The relatively warm secondary air passing the cooling coil in an upwards direction is affected by buoyant forces. Livchak & Lowell (2012) took this into account by assuming that the induction ratio is influenced by the temperature difference between room air and chilled water. This is also how it is described in ANSI/ASHRAE Standard 200-2018 of methods of testing active beams (ANSI/ASHRAE, 2018). Freitag et al. (2016) conducted measurements and found that, at constant room air temperature, the induction ratio decreased with increasing water supply temperature. However, the active chilled beam was turned upside down which was compensated for by having the supply water warmer than the room air. It was concluded that partly oppositional behavior is expected when the chilled beam is used as intended.

There are several methods of measuring the induction ratio. The primary air flow rate can easily be determined by measuring the pressure drop over an orifice plate in the primary air duct. The secondary air flow is not as straightforward to determine. Rhee et al. (2015) measured the temperatures of supply air, primary air and induced secondary air downstream the coil (see the red circles in Figure 4A) to determine the induction ratio, henceforth called *the temperature method*.

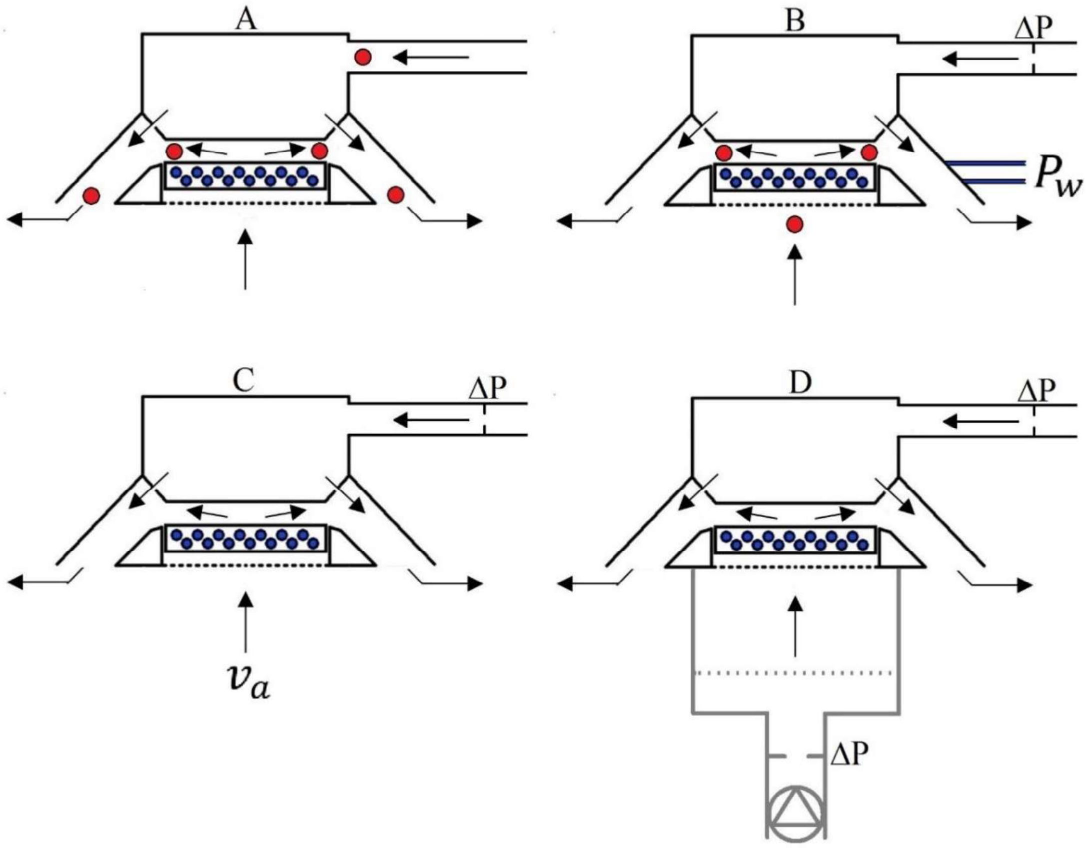

Figure 4 Illustration of measurements included in the temperature method (A), the capacity method (B), the velocity method (C) and the zero-pressure method (D).

Another common method referred to as *the capacity method* (also known as the thermal balance method) involves measuring the waterside cooling capacity (P_w) and the temperature drop of the air passing the coil (see Figure 4B). The secondary air flow rate is obtained by an energy balance of the coil and the induction ratio may be determined.

A third method, *the velocity method*, consists of measuring the average velocity of induced secondary air (v_a) and multiplying it with the area of the secondary air inlet (Figure 4C). Guan & Wen (2016) chose to measure the supply air velocity instead, since it is higher and thereby implies higher accuracy. Ruponen & Tinker (2009) used another approach and increased the velocity of secondary air by introducing a measurement venturi. This is similar to *the zero-pressure method* where the secondary air flow is determined by measuring the pressure drop over an orifice plate in a tightly sealed chamber connected to the secondary air inlet (see Figure 4D). In this case, a variable-speed compensation fan makes sure that the pressure drop through the chamber does not disturb the induction. British Standard 4954-1 (BSI, 1973) describes a similar method with the difference that the chamber is connected to the supply air outlet, which makes it more suitable for other types of induction units where the supply air is discharged through one single outlet.

Induction ratio is not at all dealt with in the European standard for testing and rating active chilled beams (CEN, 2008). In contrast, the North American counterpart includes four methods of measuring the induction ratio (ANSI/ASHRAE, 2018). The velocity method and the capacity method are described in a normative appendix while the zero-pressure method and the temperature method are included in an informative appendix.

- Different methods of measuring the induction ratio and the influence of operating conditions is the focus of Paper I.

2.3 Self-regulation

The cooling capacity of an active chilled beam is determined by the difference between temperatures of room air and chilled water. An increase of the room air temperature imply an increased temperature difference and consequently a higher cooling capacity. This is referred to as a *self-regulating effect*. Conventionally, the self-regulating effect is not strong enough to maintain thermal comfort under the conditions common in office buildings. Therefore, a control system is required. It usually consists of a room thermostat wired to a two-way control valve on the chilled water pipe. When the room air temperature is below the cooling set-point, the valve is shut and the cooling capacity is zero. When the room air temperature exceeds the cooling set-point, the valve opens, the chilled water flows and heat is extracted from the room. Another approach, less common, is to use a three-way valve and control the chilled water supply temperature.

If designing for a small temperature difference (high chilled water temperature), even small changes in room air temperature will have a significant impact on the cooling capacity, i.e. a strong self-regulating effect. At a high enough chilled water temperature, the self-regulating effect is strong enough to make the conventional control system redundant. An active chilled beam dependent on the self-regulating effect without a conventional control system is referred to as a *self-regulating active chilled beam*.

The cooling capacities of two active chilled beams are schematically illustrated in Figure 5. Both are sized to provide the design cooling capacity (100 %) at a room air temperature of 25 °C, but with different temperatures of chilled water.

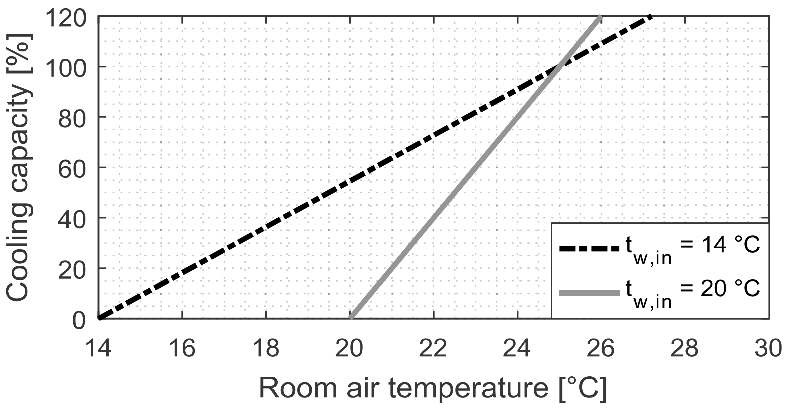

Figure 5 Schematic correlation between room air temperature and cooling capacity of two active chilled beams with different chilled water temperature.

As seen in Figure 5, the higher chilled water temperature makes the cooling capacity more sensitive to changes in the room air temperature, i.e. a stronger self-regulating effect.

The main benefit of self-regulating active chilled beams is the simplicity and robustness of reducing the amount of control equipment. In comparison to the cost of the active chilled beam unit alone, the cost of the control system adds around 25-55 % (B. Källkvist, Skanska, personal communication, November 23, 2019). The lower end of the range applies to larger zones where many active chilled beams share the same room controller while the higher end of the range applies to individually controlled active chilled beams. Drawbacks of self-regulating active chilled beams are the risk of overcooling and the increased circulation pump work required by the constant flow of chilled water.

Design of active chilled beam systems implies a trade-off between the required heat transfer area and the chilled water supply temperature. Conventionally, small heat transfer area is prioritized and a supply temperature of around 14 °C is chosen since lower temperatures would cause too much dew point control. Better performance of chillers and potential use of free cooling are factors that motivate a higher temperature.

While this trade-off is present in all active chilled beam systems, self-regulation adds important factors to the trade-off. One is that the self-regulating effect, hence the indoor temperature invariability, benefits from a higher chilled water supply temperature. This is schematically illustrated in Figure 6. The grey line represents the trade-off between chilled water temperature and required heat transfer area while the other lines represent the indoor temperature in rooms subject to cooling loads differing from their design cooling load.

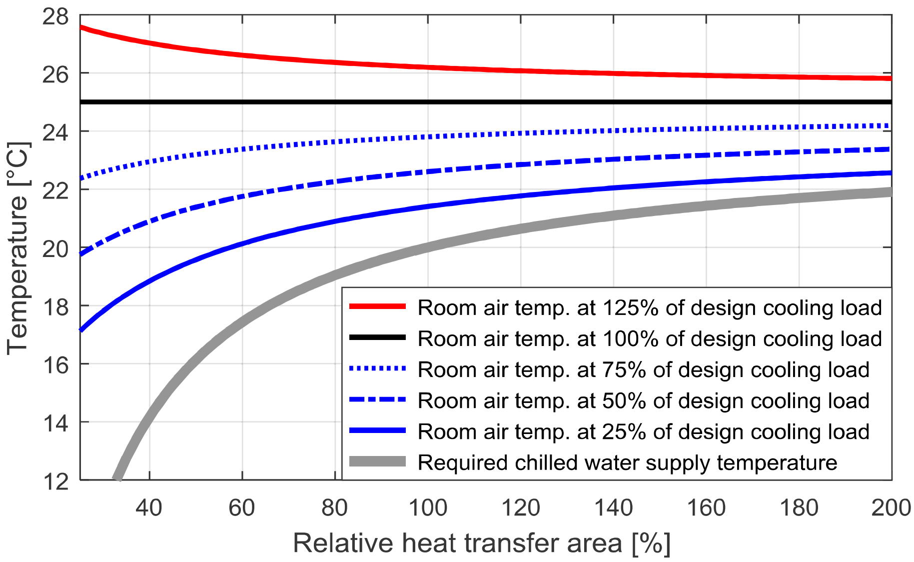

Figure 6 Schematic correlation between heat transfer area, chilled water temperature and room temperatures in a self-regulating system designed for a room temperature of 25 °C.

As seen in Figure 6, designing for a chilled water supply temperature of 20 °C implies that in a room subject to 25 % of its design cooling load, the temperature will be 21.4 °C. Designing for a chilled water supply temperature of 14 °C requires less than 40 % of the heat transfer area. A room subject to 25 % of its design cooling load will in this case be cooled down to 18.8 °C.

The schematic correlation in Figure 6 assumes static conditions not taking into account the dynamic nature of cooling loads and room temperature (further discussed in chapter 2.4). Taking dynamics into account, the trade-off between heat transfer area and chilled water temperature is further extended. A lower room temperature prior to the peak cooling load implies precooling and hence a reduced required peak cooling capacity.

- ➢ The peak-shaving effect by the precooling of self-regulating active chilled beams is studied in Paper V.

As long as excluding conventional individual/zonal control systems, an active chilled beam system is still defined as self-regulating although the chilled water supply temperature is controlled centrally. While the self-regulating effect handles short-term variations in cooling load, the possibility to increase the chilled water temperature is very valuable to handle seasonal cooling load variations.

- ➢ Avoiding overcooling by increasing the chilled water supply temperature during winter and mid-season is investigated in Paper VI.

Previous studies on self-regulating active chilled beams

The self-regulating effect is often referred to by researchers studying thermally activated building systems (TABS) (Olesen, 2012; Karlsson, 2010). Although less common, several studies on active chilled beam systems also refer to the self-regulating effect. Brister (1995) expressed skepticism towards the self-regulating effect and pointed out the need for control valves on each beam. However, these active chilled beams operated at a chilled water supply temperature of 14 °C. Schultz (2007) wrote, about passive chilled beams, that they may be designed without the use of room thermostats thanks to the self-regulating effect obtained when the design chilled water temperature is close to the room temperature. Henderson et al. (2003) mentioned the self-regulating effect as one of the advantages of active chilled beams.

Ruponen et al. (2010) simulated the energy use and indoor air temperatures in three active chilled beam systems. They included a conventional system (chilled water supply temperature of 15 °C), a high temperature system (20 °C) and a self-regulating high temperature system (20 °C). It was concluded that the self-regulating system required more cooling energy due to the lower indoor air temperature caused by that system. However, the authors concluded that this overcooling could be resolved by not having a constant year-round chilled water supply temperature of 20 °C. They also concluded that the simplicity of self-regulation is a big advantage and that self-regulating systems are likely to work as intended even after several years of operation.

Kosonen & Penttinen (2017) presented a simulation based investigation of the energy saving potential of four different chilled beam concepts, including a self-regulating system with active beams used for both cooling and heating. The water supply temperature was in the range of 20-22 °C and the indoor air temperature was allowed to vary between 21 and 25 °C. It was concluded that the high chilled water temperature gave energy-savings both due to better chiller performance and more utilization of free cooling. Furthermore, the high chilled water temperature implied no risk of condensation, hence there was no need for dehumidification of supply air nor increased chilled water temperature during humid conditions. Regarding adding self-regulation to the high temperature system, it was presented as simpler, more reliable and with less need for maintenance. However, guaranteeing thermal comfort in low occupied spaces was pointed out as a challenge. Indoor air temperatures were too low in empty and low occupied rooms during winter and mid-season. In addition to causing discomfort, the self-regulating system also caused slightly higher energy use. The authors called for compromising the simplicity of the self-regulating system by adding at least zonal-level control valves in order to solve these issues. However, night ventilation with a supply air temperature of 18 °C was operating all year round, which contributed to the overcooling and posed a challenge for the active beam system accounting for both cooling and heating.

Also Maccarini et al. (2017) presented a simulation based study of a self-regulating active beam system. The system was able to provide simultaneous heating and cooling with a water supply temperature of around 22 °C. Energy savings were obtained by the fact that heat was transferred through the active beam system from warmer to colder rooms. Regarding thermal comfort, the system was fully able to keep the indoor air temperature between 21 °C and 24 °C during operation. It was also concluded that inter-zone air flow (openings of doors) reduced the differences in air temperature between warmer and colder rooms and thereby also the transfer of heat through the active beam system.

Maccarini et al. (2019) presented the operation of an actual building with a self-regulating active beam system designed as the one simulated by Maccarini et al. (2017). It was concluded that the system performed as intended. Supply water temperatures were measured during one week in summer and one week in winter. Indoor air temperatures were measured at two positions in an open-plan office during a summer day and a winter day. Supply water temperature peaked at 23.2 °C during the winter week and was not below 20.0 °C during the summer week. The indoor air temperatures were between 21.2 °C and 23.0 °C during the winter day and between 21.0 °C and 23.2 °C during the summer day.

2.4 Cooling load calculation and zone modelling

The simulation work presented in this thesis was carried out in the building performance simulation software IDA ICE. IDA ICE applies the heat balance method to determine the thermal climate and the required amounts of heating and cooling of a building. The heat balance method is further presented later in this chapter.

Heat gain, cooling load and heat extraction rate

Cooling load calculation requires transient analysis due to the very variable behavior of heat gains (McQuiston et al., 2004). In cooling load calculations, it is important to differentiate between heat gain, cooling load and heat extraction rate.

The heat gain is the rate at which heat is transferred to or generated within a space. This includes both radiative and convective heat. As an example, around 80 % of the heat gain from an incandescent light bulb is by radiation while 20 % is by convection.

The cooling load is the rate at which heat must be extracted from the space in order to keep the room (air) temperature constant. Hence, the difference between heat gain and cooling load is due to storage of heat in the thermal mass. In other words, the radiative part of the heat gain heats the thermal mass of the room before it is further transferred to the air by convection. This detour via the thermal mass causes the difference between the heat gain and cooling load. The relationship between heat gain and cooling load at different radiative fraction and thermal mass is presented in Figure 7.

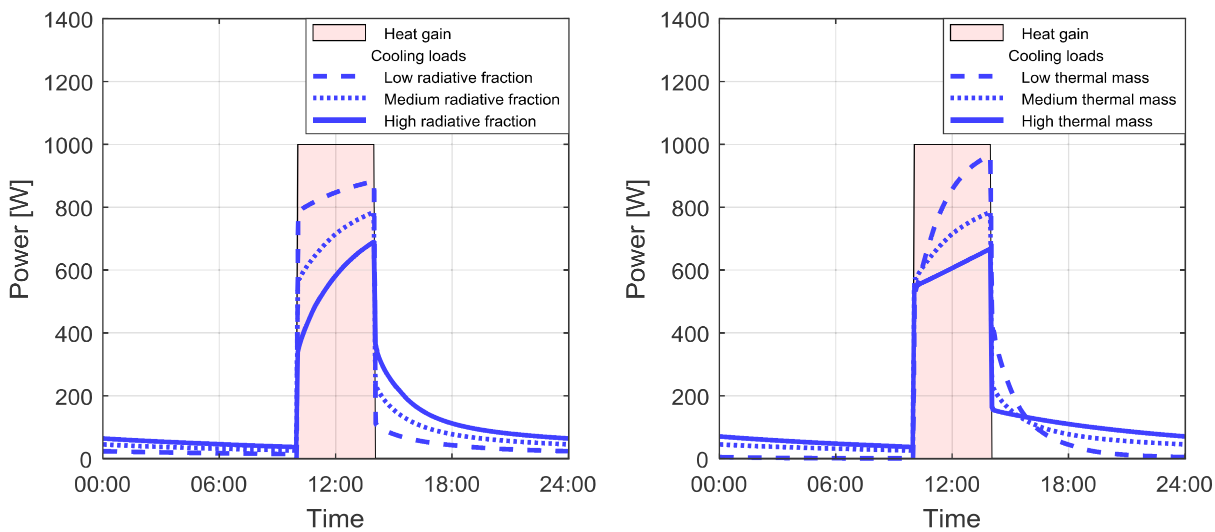

Figure 7 Heat gain and cooling load at different radiative fraction (left) and at different thermal mass (right).

As seen in Figure 7, higher radiative fraction as well as higher thermal mass imply lower cooling load.

The heat extraction rate is the rate at which heat is actually extracted from the room. Hence, any difference between cooling load and heat extraction rate causes a change of air temperature and to achieve constant temperature, they need to be equal. The relationship between heat gain and heat extraction rate, at three different levels of air temperature drifts, is presented in Figure 8.

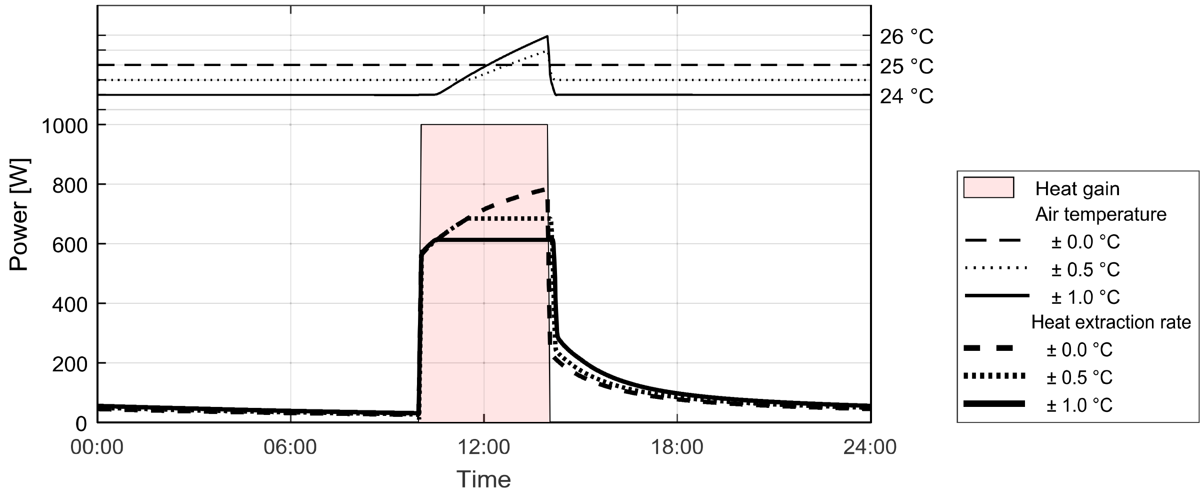

Figure 8 Heat gain, heat extraction rate and air temperature at three levels of air temperature drifts.

As seen in Figure 8, the heat gain of 1000 W active between 10:00 and 14:00 requires a peak heat extraction rate of almost 800 W in order to maintain a constant room air temperature. If allowing a non-constant room air temperature, the peak heat extraction rate is reduced. By allowing the air temperature to rise 2 °C, the peak heat extraction rate is reduced by almost 200 W. Or to put it another way, if limiting the heat extraction rate by almost 200 W, the air temperature rises 2 °C. Heat extracted by the active chilled beam is referred to as cooling capacity in the subsequent chapters of this thesis.

Figure 7 and Figure 8 represent the established definition of cooling load. That is that the room temperature is represented by the room air temperature. In this thesis, however, the purpose of the cooling system is to maintain a certain operative temperature. The operative temperature is not only affected by the convective heat transferred to the air, but also by the radiative heat transferred to the surfaces of the thermal mass. As a consequence, cooling loads and required heat extraction rates are closer to the heat gain. With both approaches, it is assumed that the heat extraction is purely convective. Figures corresponding to Figure 7 and Figure 8, with respect to the operative temperature instead of the air temperature, are presented in Appendix A (Figure A1 and Figure A2).

The heat balance method

The heat balance method involves determination of conduction, convection and radiation of each surface of a room as well as the convective heat balance of the room air. One of the main advantages of the heat balance method is that it is a detailed first principles model based on physical laws with no part of the calculations hidden from the user. The level of detail, however, can also be seen as a drawback since it requires more computational power than many other methods.

Although being very detailed and well-established, the heat balance method involves simplifications. The most fundamental simplifications are that the air temperature is uniform within each zone (generally assumed equal to the exhaust air temperature) and also that each surface has a uniform temperature. A schematic overview of the heat balance method is presented in Figure 9, details are found in ASHRAE (2017).

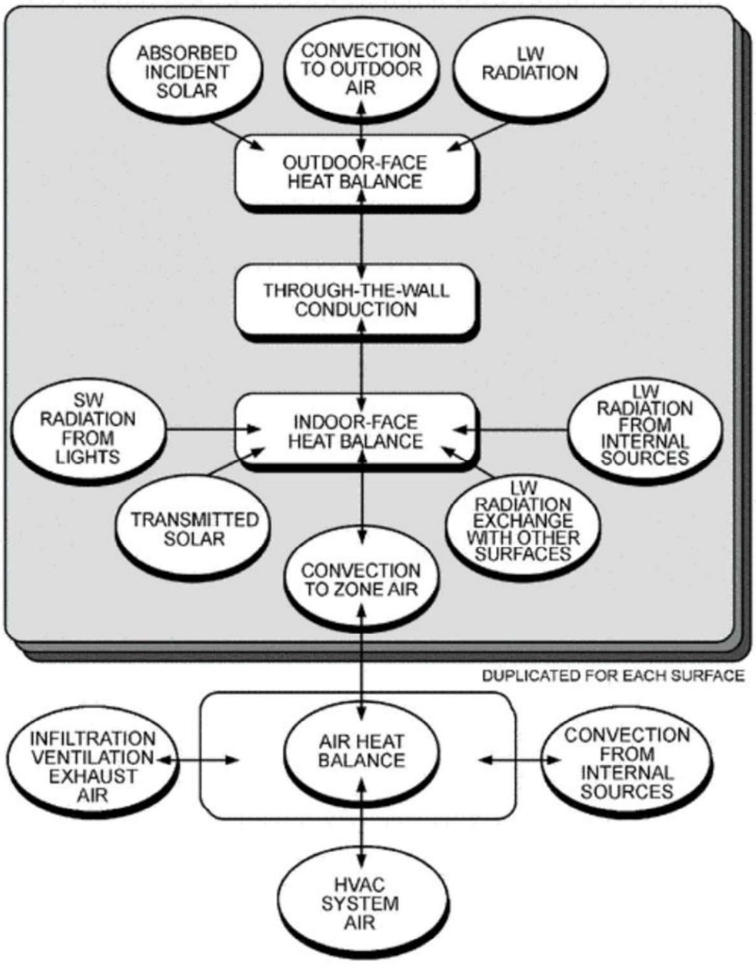

Figure 9 Schematic figure of the heat balance method (figure from ASHRAE, 2017)

A central part of the method is the inside surface (indoor-face) heat balance illustrated in the center of Figure 9. It involves six different heat transfer components which affect each other and need to be solved simultaneously. One of the important components is the convection to zone air, which will be further discussed.

Internal convective heat transfer

The internal convective heat transfer, referred to as convection to zone air in Figure 9 is henceforth referred to as internal convection. More generally, convective heat transfer is referred to simply as convection. There are three reasons why internal convection is relevant in this thesis.

- It is often considered the weakest link in building performance simulations (Peeters et al., 2011).
- It is highly influenced by the induction ratio of active chilled beams.
- High temperature cooling imply a small temperature difference driving the heat transfer. As a consequence, determination of the required heat transfer area is sensitive to errors in simulated room air temperature.

While simulation of conduction and radiation is based on well-established models (Fourier's law and the Stefan-Boltzmann law respectively), convection is more complex and not as straightforward. As a consequence, most building performance simulation software have a variety of convection models to choose between, an opportunity not utilized by the average user. In addition to being hard to model, it has also been shown that internal convection influences the results to a high degree. Beausoleil-Morrison & Strachan (1999) studied the heat demand of a room and concluded that the simulation results were far more sensitive to the uncertainties of internal convection modelling than to uncertainties of other factors such as air infiltration and the thermal properties of the envelope. Domínguez-Muñoz et al. (2010) studied uncertainties in peak cooling load calculations and found the internal convection to be one of the most significant factors influencing the results.

According to Newton´s law of cooling, convection is proportional to the difference between the temperature of the surface, t_s, and the temperature of surrounding air, t_a, in accordance with equation 2.

$$q"_{conv} = h \cdot (t_s - t_a) \qquad \text{(Equation 2)}$$

The convection coefficient h is influenced by airflow patterns and fluid dynamics not included in building performance simulation software. Furthermore, as previously mentioned, the heat balance method assumes uniform temperatures of both zone air and each surface. Consequently, in addition to the complex determination of h, the non-uniformity of room air temperature makes it necessary to find a temperature representative for t_a in equation 2, henceforth referred to as the reference temperature.

Several researchers have measured internal convection in rooms in order to determine how it is influenced by different circumstances. A comprehensive review of different models was presented by Peeters et al. (2011). In this thesis, only the default IDA ICE model is used which is presented in the following paragraphs.

The default convection model in IDA ICE originates from the work carried out in the Mechanical Engineering Laboratory at the University of Illinois around 30 years ago. Spitler (1990) conducted extensive experiments and presented convection coefficients for rooms with high ventilation air flow rates (ranging from 15 to 100 ACH). These were expressed as a function of the nondimensional jet momentum of the inlet flux and the room exhaust air temperature was chosen as the reference temperature. Fisher (1995) used the same test facility, but at air flow rates ranging from 3 to 12 ACH. Fisher (1995) used the supply air temperature as reference temperature and correlated the convection coefficients to the nondimensional "enclosure Nusselt number" and the "enclosure Reynolds number". Subsequently, Fisher & Pedersen (1997) combined the results presented by Spitler (1990) and Fisher (1995) and found that the convection coefficients could be correlated to the air change rate only. Furthermore, they found that using the supply air temperature as reference temperature resulted in lowest uncertainty and also significantly improved the correlations. When implemented in building performance simulation software, however, these correlations were reformulated to have the exhaust air temperature as the reference temperature. In IDA ICE, the default model has been further developed to take natural convection into account at low air flow rates. The default convection model for walls, floors and ceilings in IDA ICE is presented in equations 3-5.

$$h_{wall} = \max(1.208 \cdot \frac{\min(5, ACH)}{5} + 1.012 \cdot ACH^{0.604}, f(\Delta T)) \qquad \text{(Equation 3)}$$

$$h_{floor} = \max(3.873 \cdot \frac{\min(5, ACH)}{5} + 0.082 \cdot ACH^{0.98}, f((\Delta T)) \qquad \text{(Equation 4)}$$

$$h_{ceiling} = \max(2.234 \cdot \frac{\min(5, ACH)}{5} + 4.099 \cdot ACH^{0.503}, f(\Delta T)) \qquad \text{(Equation 5)}$$

In equations 3-5, ΔT is the temperature difference between the zone air and each surface while $f(\Delta T)$ is known as the BRIS model originating from a predecessor of IDA ICE described by Brown (1990). The BRIS model is shown in Paper III of this thesis.

Although being well-established and widely used, the default model is not undisputed. Le Dréau et al. (2015) measured internal convection in a room with an active chilled beam and concluded that existing correlations (as presented by Fisher & Pedersen (1997)) tend to overestimate the convection. New correlations were developed as a function of the modified Archimedes number to account for the jet behavior and the supply air temperature was used as reference temperature.

A high internal convection coefficient is equivalent to low thermal resistance between the zone air and the thermal mass. However, this does not necessarily imply that higher internal convection coefficients mean better utilization of the thermal mass. On the contrary, increased convection coefficients cause an increased cooling load, as illustrated in Figure 10. A higher convection coefficient accelerates the convection by which the radiative heat gain is transferred from the thermal mass to the air, hence the higher cooling load. Convection coefficients in Figure 10 are determined by equations 3-5 and the air change rates are 0 and 10 ACH in the natural and forced convection case respectively.

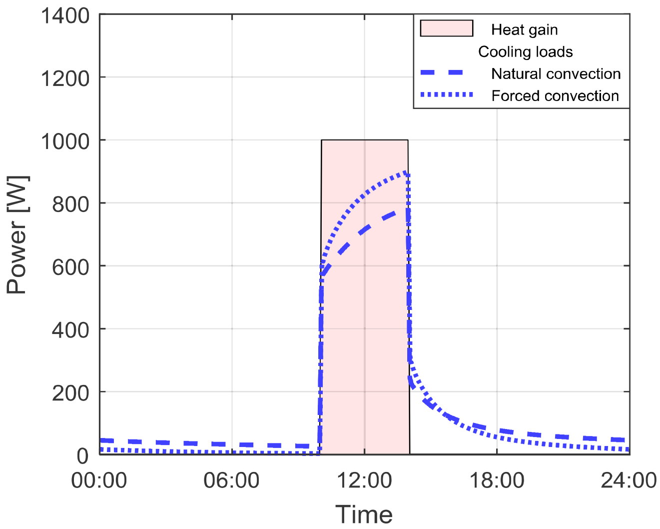

Figure 10 Heat gain and cooling load at different convection coefficients

However, although increasing the cooling load, higher convection coefficients do not necessarily imply higher required heat extraction rates. Higher convective heat transfer coefficients also increase the benefit of allowing the temperature to vary. As seen in Figure 11, which represent the case of forced convection, the heat extraction rate at a temperature drift of ±1.0 °C is slightly lower than in Figure 8.

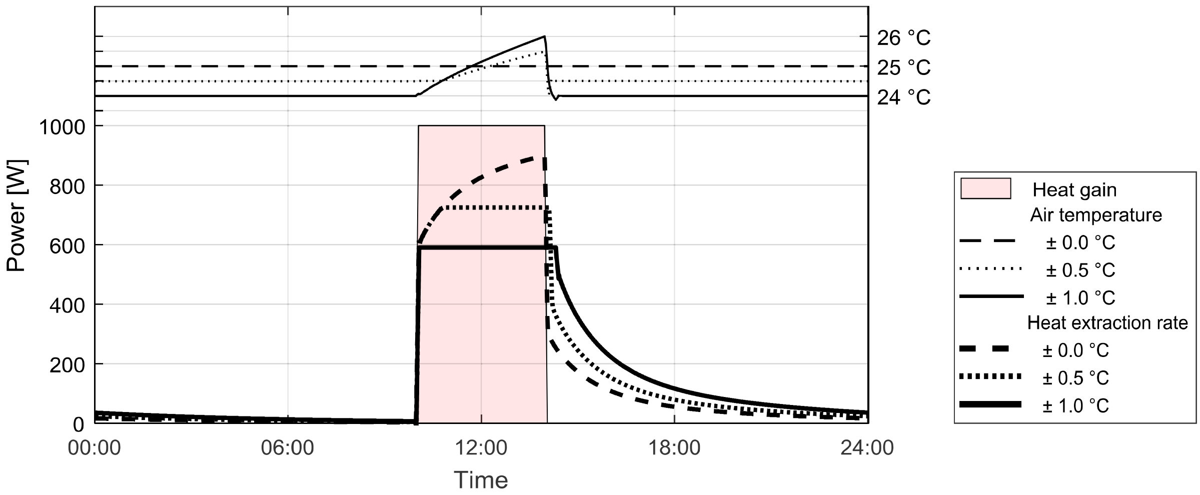

Figure 11 Heat gain, air temperature and heat extraction rate at high convection coefficients and at three levels of temperature drifts.

Just as with Figure 7 and Figure 8, corresponding figures for Figure 10 and Figure 11 with respect to the operative temperature, are found in Appendix A (Figure A3 and Figure A4). There are no fundamental differences, but the cooling loads and required heat extraction rates are generally higher when determined with respect to the operative temperature.

Control of the heat extraction rate is ideal in these examples, both in terms of speed, stability and its independence of the air temperature. With an active chilled beam, especially a high temperature active chilled beam, the heat extraction rate is dependent on the room air temperature (as shown schematically in Figure 5). This introduces yet another important aspect of the influence of internal convection. The convective heat transfer coefficients influence the difference between room air temperature and operative temperature. At design conditions, operative temperature generally exceeds the room air temperature. As a consequence, higher convection coefficients imply a higher room air temperature at fixed operative temperature. The higher room air temperature implies a higher temperature difference driving the heat transfer (cooling capacity) and hence decreasing the heat transfer area required to obtain a certain operative temperature.

Following is a summary of the discussed influences of increased internal convection:

- Higher cooling load
- Larger difference between cooling load and required heat extraction rate.
- Smaller difference between operative temperature and air temperature.

➢ The practical consequences of increased internal convection (and changed reference temperature) are presented in Paper III.

3 Experimental resources

The experimental work presented in Paper I-III was carried out during 2015 and 2016 in the Building Services Engineering Laboratory at Chalmers University of Technology.

3.1 The room

All measurements were carried out in a full-scale mockup of an office room, see Figure 12. The internal dimensions of the room were 3.0 m × 4.2 m × 2.4 m (width × length × height). The active chilled beam was positioned in the suspended ceiling in the middle of the room and two cylindrical thermal dummies (heat gain generators) were positioned on each side of the active chilled beam. The grey circles in Figure 12 represent the positions of the thermal dummies during the work presented in Paper I and II while the black circles represent their positions during the work presented in Paper III. An air gap of 10 cm between the floor and the cylinder allowed inflow of air while the height of the cap of the thermal dummies was adjustable in order to adjust the flow of air and thereby the radiative fraction of the heat gain. The cylinders housed six lightbulbs each (75 W) and the power was adjusted by a variable autotransformer.

The temperature outside the room was monitored and controlled to not deviate more than 2 °C from the temperature inside the room in order to reduce heat transfer through the envelope of the room. The walls of the room were made of 12 mm plasterboard attached on the inside of 100 mm expanded polystyrene. The floor was made of 22 mm fiberboard on top of 6 mm plasterboard and 30 mm expanded polystyrene.

The ceiling was a suspended ceiling of 15 mm fiberglass below an air gap of 270 mm under structural lightweight concrete.

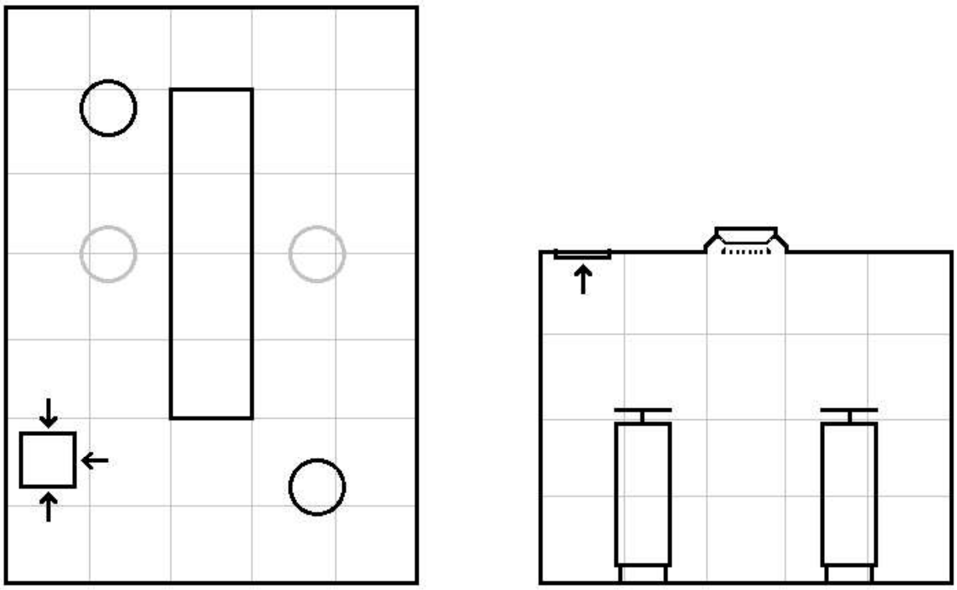

Figure 12 Layout of the room and positions of active chilled beam, exhaust air terminal device and heat gain generators. Top view to the left and front view to the right. Grey lines indicate a distance of 0.6 m.

All experiments were carried out with a Halton CBS active chilled beam with length and width of 2.4 m and 0.6 m. At a room air temperature of 25.0 °C, chilled water supply temperature of

20.0 °C, chilled water flow rate of 1.80 l/min, primary air temperature of 20 °C and primary air flow of 22.0 l/s the chilled water cooling capacity was 376 W and the sensible cooling from the primary air flow was 132 W. Data obtained from measurements in the laboratory was in accordance with specifications from the manufacturer (±1 %). Pressure drops of water and air flows were not measured in the laboratory, but correspond to 3.6 kPa (water) and 100 Pa (air) according to the technical specifications provided by the manufacturer.

3.2 The instrumentation

Sampling rate of all measurements was one per minute. Air temperature sensors were shielded from radiation. All relevant sensors used during the experiments are listed in Table 1.

Table 1 Summary of sensors used during the experiments.

Parameter	Unit	Label	Principle
Primary air flow rate	l/s	Klimatbyrån ZMC80 / Honeywell DPTM100D	Differential pressure
Exhaust air flow rate	l/s	Klimatbyrån ZMT160 / Honeywell DPTM100D	Differential pressure
Chilled water flow rate	l/min	Sharky FS 473	Ultrasonic
Secondary air velocity	m/s	Swema SWA31	Hot-wire anemometer
Room air velocity	m/s	Swema 03	Hot-wire anemometer
Primary air temperature	°C	Pentronic 7410000	RTD
Supply air temperature	°C	Atexis UK02	RTD
Secondary air temperature	°C	Atexis UK02	RTD
Room air temperature	°C	Atexis UK02	RTD
Globe temperature	°C	Swema 05	RTD in 150 mm globe
Ambient air temperature	°C	Schneider STD400	RTD
Chilled water supply temperature	°C	Schneider STP300	RTD
Chilled water return temperature	°C	Schneider STP300	RTD
Reference used for correction of the temperature sensors	°C	Dostman P650	RTD
Primary air CO_2 concentration	ppm	TAC SCD100-D	Infrared spectrometer
Supply air CO_2 concentration	ppm	Vaisala GM70	Infrared spectrometer
Secondary air CO_2 concentration	ppm	SenseAir 2001VTC2	Infrared spectrometer
Internal heat gain	W	Schneider PM800	-

During the work presented in Paper III, a globe thermometer was used to determine the mean radiant temperature and thereby the operative temperature. The mean radiant temperature, $\bar{t}_r$, was determined according to equation 6, as prescribed by ISO 7726 (ISO, 1998), valid for a matt black globe with a diameter of 150 mm.

$$\bar{t}_r = \left((t_g + 273)^4 + 2.5 \cdot 10^8 \cdot v_a{}^{0.6} \cdot (t_g - t_a)\right)^{1/4} - 273 \qquad \text{(Equation 6)}$$

The globe temperature is represented by t_g while t_a and v_a represent the temperature and velocity of air.

In Paper III, Paper V and Paper VI, the concept of operative temperature was used. The approximation of operative temperature presented in equation 7 is used consistently throughout this thesis.

$$t_{op} = \frac{t_a + \bar{t}_r}{2} \qquad \text{(Equation 7)}$$

Another frequently used approximation is presented in equation 8.

$$t_{op} = \frac{t_a \cdot \sqrt{10 \cdot v_a} + \bar{t}_r}{1 + \sqrt{10 \cdot v_a}} \qquad \text{(Equation 8)}$$

The rationale for choosing the simpler approximation according to equation 7 was twofold. At the air velocities and temperatures obtained during the measurements in Paper III, the difference between equation 7 and equation 8 was very small (less than 0.075 °C during 99 % of the time). Furthermore, equation 7 is the approximation used by the building performance simulation software IDA ICE with which the measurements were compared.

Uncertainties of measurements (presented in Paper I-III) refer to expanded uncertainties (a coverage factor of 2) and were estimated as the root sum square of the uncertainties caused by each source of uncertainty.

4 Results

This chapter gives a brief description of each appended paper and presents their main results. Details are provided in the appended full-versions of papers.

While the results of Paper I-III are applicable for active chilled beam systems in general, the results of Paper IV-VI are specific for self-regulating active chilled beam systems. Further categorization is made in Figure 13, where the appended papers are arranged regarding method and scope.

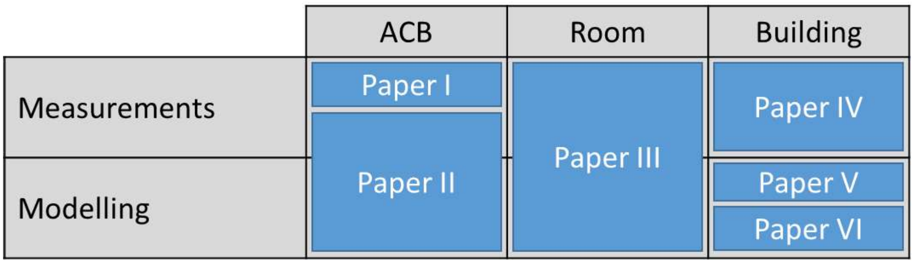

Figure 13 Categorization of the appended papers

Results from Paper I, knowledge about the induction ratio and how it is influenced by the operating conditions, were essential for the model presented in Paper II. The induction ratio was also valuable input to Paper III, Paper V and Paper VI.

The model presented in Paper II was not explicitly used in any subsequent papers. However, its accuracy supported the use of a simple and purely convective model in situations with constant flow of primary air as was the case in Paper III, Paper V and Paper VI.

The results from Paper III, modifications of the default zone model, were applied both in Paper V and Paper VI.

4.1 Paper I - Induction ratio of active chilled beams - Measurement methods and influencing parameters

The induction of room air is one of the most distinctive features of an active chilled beam. A high induction ratio is important in order to obtain a high cooling capacity and knowledge about the induction ratio is necessary to determine both the temperature and flow of supply air.

Paper I presents an experimental study where three fundamentally different methods of determining the induction ratio were compared. One of the methods was also used to investigate how the operating conditions influence the induction ratio. The three methods were based on measuring air velocity (*the velocity method*), tracer gas concentration (*the tracer gas method*) and cooling coil energy balance (*the modified capacity method*).

The investigated operating conditions were primary air flow rate, chilled water flow rate, chilled water supply temperature and room air temperature.

The main results from the comparison of methods were two novel methods of measuring the induction ratio of an active chilled beam. The tracer gas method has not been presented in previous literature. The modified capacity method is a combination of two previously known methods (the temperature method and the capacity method), both including the difficult determination of secondary air temperature downstream the coil. That issue was avoided by using the novel method. Compared to the velocity method and the tracer gas method, the induction ratio determined by the modified capacity method was around 15 % higher.

Regarding the influence of operating conditions, the results show that the induction ratio peaked at a certain primary air flow rate while both higher and lower flows gave lower induction ratio. In contrast to the previously published theory, the induction ratio was substantially less influenced by the room air temperature than by the chilled water temperature. Due to buoyant forces acting on the upwards moving secondary air, the induction ratio is counteracted by cold chilled water (see Figure 14). Regarding the room air temperature however, warmer air slows down more when chilled in the coil, but on the other hand, warm air is lighter and more easily elevated by the inductive force. The results indicate that the first phenomenon dominates at lower air flows while the second one dominates at higher air flows.

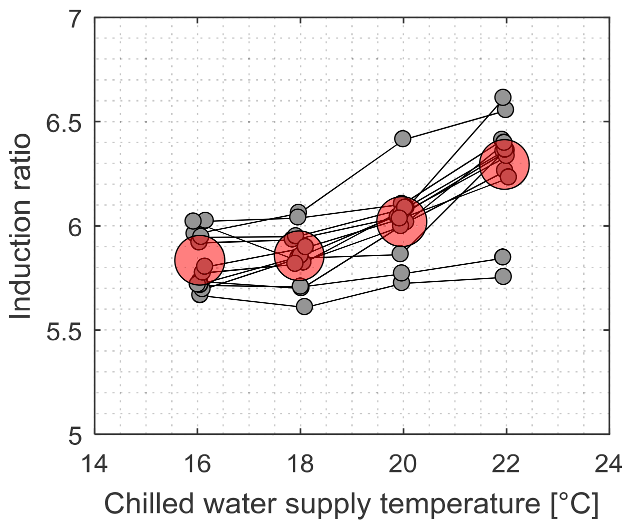

Figure 14 Induction ratio as a function of chilled water supply temperature. Grey circles represent individual cases and red circles represent averages. Lines connect cases with equal internal heat gain, chilled water flow rate and primary air flow rate.

4.2 Paper II - A thermal model of an active chilled beam

Conventional models of active chilled beams are to a large extent empirical and simulate the cooling capacity in a generic manner. Involved parameters are empirically derived at certain operating conditions (e.g. primary air flow rate) and determined by interpolation when simulating other conditions. Many conventional models do not, implicitly, involve the induction ratio and are thereby unable to take the buoyant forces (as presented in Paper I) into account as well as determining the flow rate and temperature of the supply air.

In Paper II, a thermal model of an active chilled beam based on NTU-effectiveness theory is presented. The work is based on measurements of 61 combinations of operating conditions. Thanks to the use of well-established heat transfer theory, few cases were needed for calibration of the model which then could be used to accurately determine the cooling capacity in a wide range of operating conditions, including laminar flow of chilled water.

Neglecting the influence of operating conditions on the induction ratio (investigated in Paper I) decreased the accuracy of the model slightly, but not more than what is generally accepted in building performance simulation. The accuracy of the model supports the often made assumption that the heat transfer of an active chilled beam is purely convective. Results from the model are presented in Figure 15. The bumps at around 1.3 l/min are a result of the transition between laminar and turbulent flow. The difference between the two temperature levels is primarily due to the correlation between chilled water supply temperature and induction ratio, presented in Paper I.

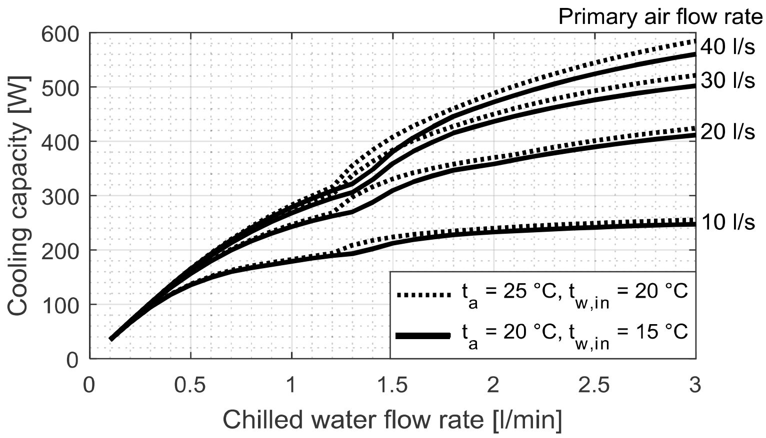

Figure 15 Cooling capacity as a function of flow rates of chilled water and primary air, at two temperature levels.

At turbulent flow of chilled water, the model had an average error of 2.7 % when only two cases were used for calibration and the induction ratio was assumed constant. Modelling a non-constant induction ratio required six cases for calibration and decreased the average error to 1.7 %. Accuracy of the model, expressed as absolute percentage errors, is presented in Table 2.

Table 2 Summary of accuracies

Flow type	Cases used for calibration	Cases used for validation	Induction ratio	Error (APE) [%] Avg.	Max
Turbulent	48	48	$f(\dot{V}_{pri}, t_{w,in})$	1.6	4.5
Turbulent	48	48	Constant	2.0	4.5
Turbulent	6	48	$f(\dot{V}_{pri}, t_{w,in})$	1.7	4.7
Turbulent	6	48	Constant	2.2	5.3
Turbulent	2	48	Constant	2.7	6.7
Nonturbulent	2 + 1	13	$f(\dot{V}_{pri}, t_{w,in})$	1.0	3.6

4.3 Paper III – Modelling of rooms with active chilled beams

When implemented in building performance simulation software, active chilled beams are generally modelled as purely convective cooling equipment, disregarding any radiative part. That is unproblematic, as demonstrated by the purely convective model presented in Paper II. However, other specific features of active chilled beams are often disregarded when modelling the convective part of the zone heat balance.

In Paper III, the convective heat transfer equations of the zone model were modified in order to take into account both the induction of room air and that the air supplied from an active chilled beam affects the ceiling surface temperature. This was done by adjusting the air change rate in the internal convection model and by using the supply air temperature as reference temperature for the internal convection at the ceiling. The modifications of the model resulted in a higher maximum air temperature and lower maximum operative temperature. Taking the induction ratio into account influenced the results to a larger extent than what the change of reference temperature did. Since the cooling capacity of an active chilled beam is determined by the air temperature, the modifications reduced the heat transfer area required to obtain a certain operative temperature. The modifications were validated by measurements in the room presented in chapter 3, exposed to rectangular pulses of internal heat gain. Simulation of both air and operative temperature corresponded more closely to measurements after the modifications. This is presented in Figure 16.

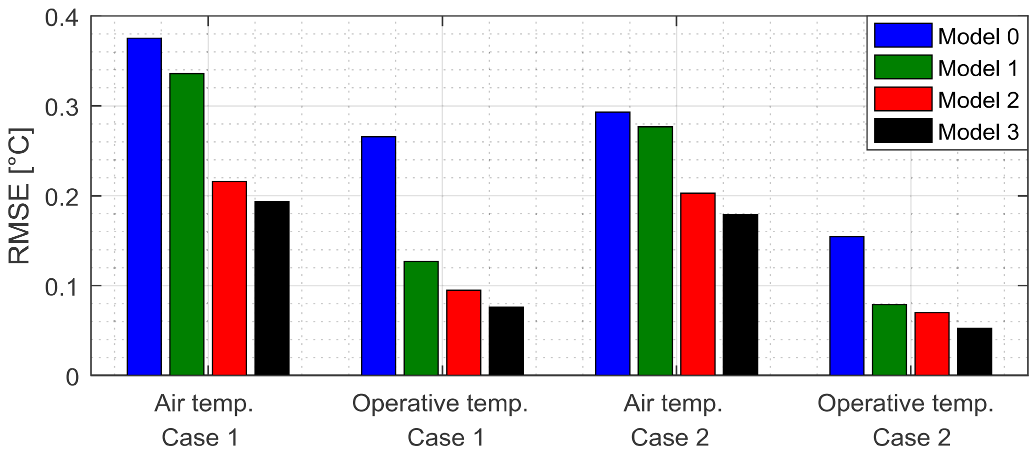

Figure 16 Root mean square errors of simulated temperatures. Model 0 is the original default simulation, Model 1 takes thermal inertia of the sensors into account, Model 2 also takes induction into account and Model 3 also uses the supply air temperature as reference temperature.

For a single office room located in Stockholm, with realistic thermal properties and internal heat gain, it was concluded that the modifications implied that the same maximum operative temperature could be achieved with 9 % less heat transfer area[1].

[1] In Paper III, it is concluded that the required *design cooling capacity* was reduced by 9 %. The *design cooling capacity* is the cooling capacity of the ACB at a specified room air temperature. Hence, the same reduction applies to the heat transfer area.

4.4 Paper IV - Performance evaluation of a direct ground-coupled self-regulating active chilled beam system

Paper IV presents an analysis of operational data from a real building equipped with direct ground-coupled self-regulating active chilled beams. The building is located in Stockholm, Sweden, the investigated part covers 22 144 m^2 and is dominated by open-plan offices. The analysis covers data since the building was taken into operation in 2014, but focus is on 2018, since that particular summer offered unprecedented conditions for stress-testing comfort cooling systems in Stockholm.

Cooling is obtained from pre-heating incoming air (free ACB cooling) and from the ground (ACB/AHU cooling) which is recharged during winter (ground charging). There is also a run-around coil recovery system in operation during both winter and summer (cool recovery). In 2018, for the first time, the system extracted more cooling from the ground during summer than was recharged during winter. This is presented in Figure 17.

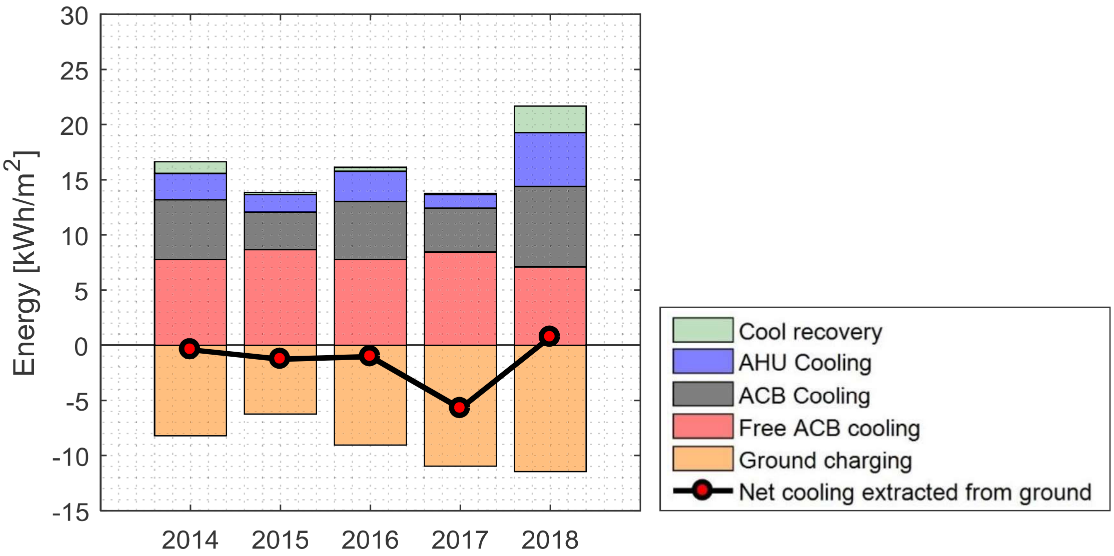

Figure 17 Total annual energy transferred in the cooling system during 2014-2018.

Though not involving a chiller, the cooling system uses electricity for pumps and also causes increased pressure drop in the air handling units. The useful cooling energy obtained was around 17 times higher than the electric energy required by pumps and additional fan work.

The results show that the system was fully capable of providing enough cooling to not exceed comfortable indoor air temperature levels. Indoor air temperatures in open-plan offices were analyzed and it was concluded that they were both uniform and stable. During working hours in 2018, the average difference between the highest and lowest air temperature was 1.9 °C and below 2.4 °C during 90 % of the time. Regarding stability, the average standard deviation was 0.5 °C while the least stable temperature had a standard deviation of 0.8 °C. The average indoor air temperature was 22.3 °C annually and 22.6 °C during summer. The maximum indoor air temperatures did not coincide with the maximum outdoor temperatures.

4.5 Paper V - Peak shaving effect of self-regulating active chilled beams

Due to thermal inertia of buildings and its content, required peak cooling capacity may be reduced by precooling the building prior to the peak cooling load. Due to the constant flow of chilled water, peak shaving by precooling is an inherent feature of self-regulation.

Paper V is based on simulations carried out in IDA ICE. It was investigated how the required peak cooling capacity is influenced by the supply chilled water temperature in a self-regulating system as well as by the cooling set-point in a conventional system. The studied scenario was one floor of an office building during a hot summer day in Stockholm. The design criterion was a maximum operative temperature of 25.0 °C.

The results show that the required peak cooling capacity in a self-regulating system with a chilled water supply temperature of 20 °C was 17 % lower than in a conventional system operating with a cooling set-point of 24 °C. Additionally, it was shown that the required heat transfer area was reduced slightly more (19 %), mainly because the precooling also reduced the difference between operative and air temperature.

Figure 18 illustrates the cooling capacity in the eight zones of the building in a self-regulating system operating with a chilled water supply temperature of 20 °C (left) and in a PI-controlled system with a cooling set-point of 24 °C (right).

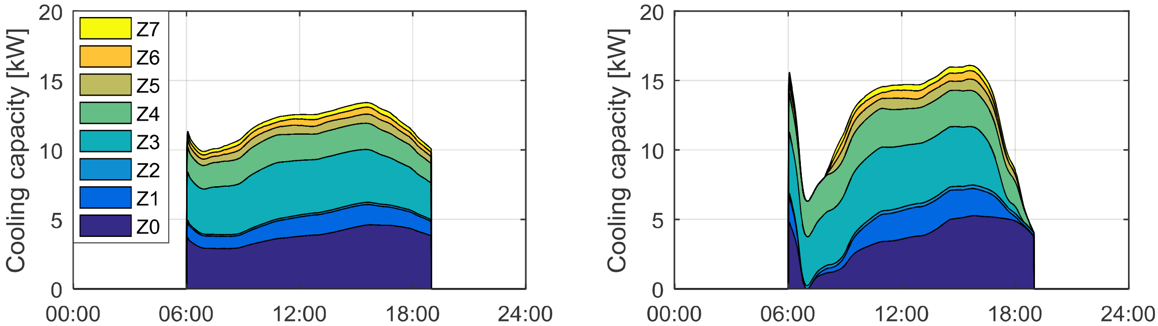

Figure 18 Cooling capacity in a self-regulating system (left) and in a PI-controlled system (right).

Precooling is associated with decreased indoor temperatures. During early morning, the average operative temperature was around 0.6 °C lower in the self-regulating system. Operative temperatures corresponding to the previous figure are presented in Figure 19.

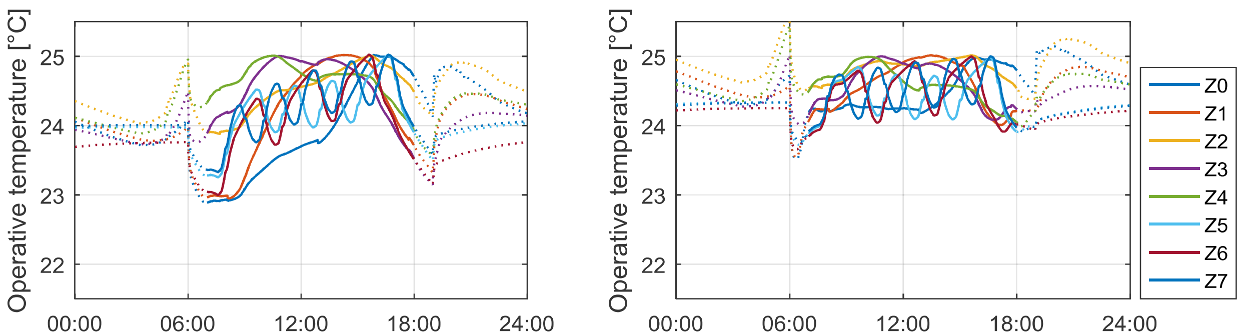

Figure 19 Operative temperatures in a self-regulating system (left) and in a PI-controlled system (right).

4.6 Paper VI - Supply temperature control of self-regulating active chilled beams

The title of Paper VI refers to central control of the chilled water supply temperature. Hence, all active chilled beams in the building are supplied the same temperature and the flow is constant. An inherent risk of constant chilled water flow rate is overcooling and consequently too low indoor temperatures and high energy use. In the worst case, cooling from the active chilled beam system is counteracted by heating from the radiator system, causing excessive use of both heating and cooling.

The purpose of this paper was to compare different strategies of controlling the chilled water supply temperature in terms of energy use and thermal climate. Individually controlled (non-self-regulating) active chilled beams were also included for comparison. The study was based on simulations in IDA ICE and the investigated model represented one floor of an office building located in Stockholm, Sweden. The floor was dominated by open-plan offices but included also three meeting rooms.

As in the real building presented in Paper IV, this model included the possibility of free active chilled beam cooling by preheating incoming air (when needed). In a system with constant chilled water temperature, this alleviated the consequences of overcooling regarding the energy use for active chilled beams and heating coil, while overcooling was still very detrimental to the heating demand for radiators.

However, the study also showed that overcooling is effectively avoided, without causing thermal discomfort, by increasing the chilled water supply temperature outside of the summer season. The operative temperature variability was higher in the self-regulating systems than with ideally and individually controlled active chilled beams, but still within comfort limits. With simple control of the chilled water supply temperature (with respect to the outdoor air temperature or the exhaust air temperature), a self-regulating active chilled beam system require less energy for heating and cooling than a conventionally controlled system. This is illustrated in Figure 20.

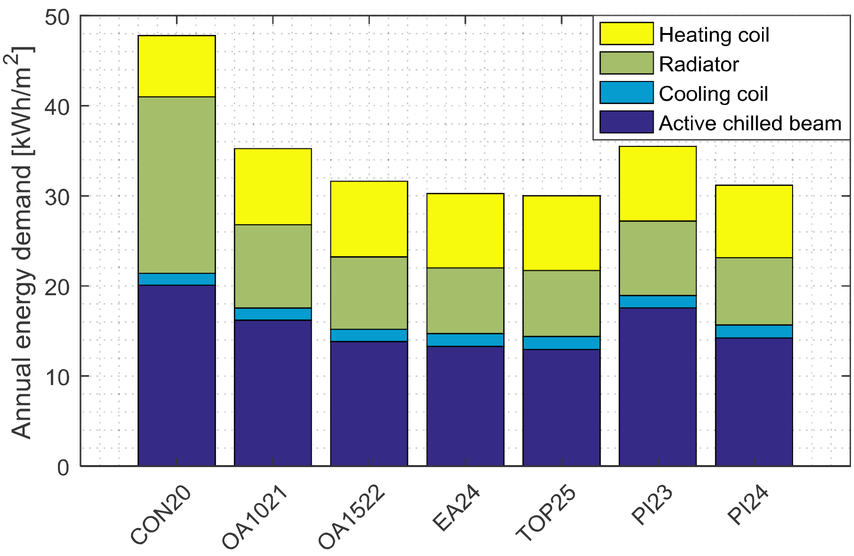

Figure 20 Energy use with different strategies to control the supply chilled water temperature (CON20: Constant, OA: As a function of the outdoor air temperature, EA: As a function of the exhaust air temperature, TOP: As a function of the maximum operative temperature, PI: Non-self-regulating control).

5 Discussion

5.1 Induction ratio

Measuring methods

In Paper I, three methods of measuring the induction ratio were presented. Two of these, the modified capacity method and the tracer gas method has not been presented in previous literature. An obvious question is why to present new methods when there already are several (illustrated in Figure 4).

The temperature method and the capacity method require measurement of the secondary air temperature downstream the coil. As also noted in previous studies (Chen et al., 2014), this temperature is very difficult to measure. The fact that it varies along (and across) the beam may be handled by averaging values from multiple temperature sensors. A greater concern is an observation that, even in a cross-section of one side of the beam, this air does not seem to be fully mixed with itself after leaving the coil before mixing with the primary air.

The velocity method involves measuring very low air velocity, which is associated with low accuracy. Furthermore, turbulence generated at the supply air outlet aggravates the measurement of induced secondary air. Average uncertainty of the velocity method due to instrument inaccuracies was 21 % under the conditions tested in Paper I. Corresponding uncertainties of the tracer gas method and the modified capacity method were 6 % and 5 % respectively.

The zero-pressure method requires bulky equipment to be attached underneath the active chilled beam. All studies on induction ratio of active chilled beams involving some kind of bulky equipment are made with the active chilled beam turned upside down (Ruponen & Tinker, 2009; Chen et al., 2014; Freitag et al. 2016; Guan & Wen, 2016). Based on this, it is assumed that it is difficult to use the zero-pressure method on a correctly mounted active chilled beam. The disadvantage of testing the active chilled beam upside down is that the influence of buoyancy gets the opposite effect.

Regarding the methods presented in Paper I, the tracer gas method and the velocity method may be suitable for on-site measurements aiming at fault detection while the modified capacity method is more suitable in laboratory facilities where temperatures and flow rates of air and water can be measured with high accuracy.

Influencing parameters

The induction ratio is often assumed to be constant. It is also often assumed being influenced by the primary air flow or by the plenum pressure (Chen et al., 2014). In some cases, buoyant forces are also taken into account by determining the induction ratio as a function of the temperature difference between the warm room air and the chilled water (Livchak & Lowell, 2012; ANSI/ASHRAE, 2018). As far as the author know, Paper I is the first study investigating whether an increased chilled water temperature influences the induction ratio differently than a decreased room air temperature. The results presented in Paper I indicate that this is true. In air flows driven only by buoyant forces, it is reasonable to assume that the air flow rate is determined by the temperature difference between the air and the surface. In the case of an active chilled beam though, the air flow is driven by the induction and only slightly counteracted by the temperature difference. Colder room air decreases the temperature difference but on the other hand, it is also heavier and less prone to flow upwards, a phenomenon present regardless of the temperature difference.

The practical consequences of taking this into account is generally small. However, testing the performance of an active chilled beam (calibrating a model) at certain conditions and erroneously assuming that the induction ratio is the same at other conditions gives error in the simulated cooling capacity (see the difference between the continuous and the dashed lines in Figure 15). Calibrating a conventional model at a chilled water temperature of 15 °C leads to underestimation of the cooling capacity by around 3 % at a chilled water temperature of 20 °C. This error is easily avoided by testing the performance of the active chilled beam at the same chilled water temperature as it is intended to be used at. Another option is to use a model that takes the correlation between chilled water temperature and induction ratio into account, e.g. the model presented in Paper II.

5.2 Active chilled beam modelling

Thermal models of active chilled beams have been developed by several researchers (Livchak & Lowell, 2009; De Clercq et al. 2013; Chen et al., 2014; Maccarini et al., 2015; Ji et al., 2019) What makes the model presented in Paper II unique, is that it takes into account the influence of buoyancy as presented in Paper I. Additionally, no other model known to the author is valid over such a wide range of operating conditions, including laminar flow of chilled water.

Modelling is typically a trade-off between empiricism and first principles. The model presented in Paper II involves first principles to a higher degree than any other active chilled beam model known to the authors. Only the air-side geometry, the induction ratio and Nusselt number at laminar water flow are required to be determined (by measurements of induction ratio and cooling capacity at a number of different operating conditions). As a consequence, very few cases are required for model calibration, before the model can be used over a wide range of operating conditions. It may be used to simulate conditions unfeasible to test in reality due to practical limitations in the laboratory or to investigate how fouling or alternative fluids influence the results. Empirical models are not useful for such analyses.

That being said, the practical benefits of this model should not be exaggerated. Manufacturers seldom have problems of testing their active chilled beams at numerous operating conditions and providing their customers with empirical data of the performance. Empirical data may also be considered more trustworthy than results from a model.

Many models of active chilled beams exist. Disregard the simplicity of the model, all agree that the cooling capacity is equal to the product of chilled water temperature rise, $(t_{w,out} - t_{w,in})$, mass flow rate, $\dot{m}_w$, and specific heat capacity, $c_{p,w}$, see equation 9.

$$P_w = \dot{m}_w \cdot c_{p,w} \cdot (t_{w,out} - t_{w,in}) \quad \text{(Equation 9)}$$

Since the outlet temperature is unknown, equation 9 is not sufficient to determine the cooling capacity. In Paper II, it was concluded that the cooling capacity was accurately determined according to equation 10.

$$P_w = \varepsilon \cdot C_{min} \cdot (t_a - t_{w,in}) \quad \text{(Equation 10)}$$

The effectiveness, ε, and the minimum heat capacity rate, C_{min}, are determined by the flow rates of air and chilled water. The parameter t_a is the temperature of the room air. A much simpler generic model is presented in equation 11, where k is an empirical constant determined by the primary air flow rate.

$$P_w = k \cdot \Delta T \qquad \text{(Equation 11)}$$

The temperature difference, ΔT, may be defined either as the arithmetic mean temperature difference presented in equation 12 or as the logarithmic temperature difference presented in equation 13. In both cases, disregarding the air temperature drop across the coil.

$$\Delta T_{am} = t_a - \frac{t_{w,in} + t_{w,out}}{2} \qquad \text{(Equation 12)}$$

$$\Delta T_{lm} = \frac{(t_a - t_{w,in}) - (t_a - t_{w,out})}{\ln\left(\frac{t_a - t_{w,in}}{t_a - t_{w,out}}\right)} \qquad \text{(Equation 13)}$$

In both cases, the temperature differences can be combined with equations 9 and 11 to yield an expression of the cooling capacity as a function of the difference between room air and chilled water supply temperature, see equations 14 and 15.

$$P_w = \frac{k_{am} \cdot 2 \cdot \dot{m}_w \cdot c_{p,w}}{k_{am} + 2 \cdot \dot{m}_w \cdot c_{p,w}} \cdot \left(t_a - t_{w,in}\right) \qquad \text{(Equation 14)}$$

$$P_w = \dot{m}_w \cdot c_{p,w} \cdot \left(1 - e^{-(k_{lm}/(\dot{m}_w \cdot c_{p,w}))}\right) \cdot \left(t_a - t_{w,in}\right) \qquad \text{(Equation 15)}$$

The empirical constants, k_{am} and k_{lm}, refer to using the arithmetic mean and logarithmic temperature difference respectively. The default active chilled beam model in IDA ICE uses the logarithmic temperature difference, hence represented by equation 15.

Equations 14 and 15 imply that, at constant flow of primary air and chilled water, the cooling capacity is proportional to the difference between room air and chilled water supply temperature. Assuming a constant induction ratio, this is true also for the more advanced model represented by equation 10. The only minor difference is that the advanced model takes into account that the thermophysical properties of air and water are influenced by the temperature. In the air temperature range between 20 and 25 °C, this influences the determined cooling capacity by less than 0.4 %. As discussed in chapter 5.1, the error from assuming a constant induction ratio may be substantially higher (around 3 %), but is avoided by calibrating the model at a chilled water temperature similar to its intended use.

If extending the comparison to include also non-constant flow of chilled water, e.g. during part-load conditions in a conventionally controlled system, it is obvious that there are larger differences between the models. Assuming an active chilled beam providing 380 W at a room air temperature of 25 °C and a chilled water supply temperature of 20 °C. Figure 21 presents the cooling capacity per degree temperature difference between room air and chilled water supply temperature obtained by equation 10 (Paper II), 14 and 15 respectively.

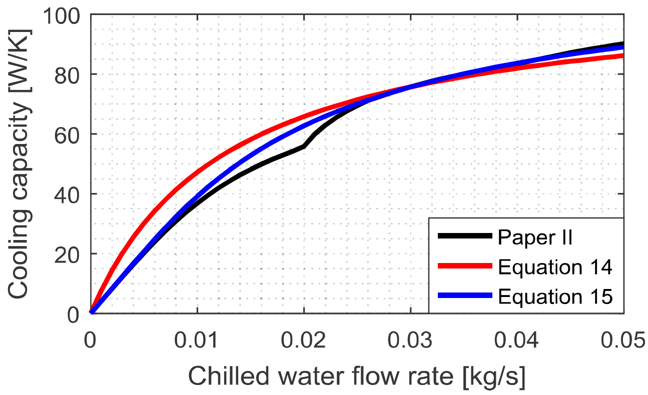

Figure 21 Cooling capacity per degree difference between room air and chilled water supply temperature, as a function of the chilled water flow rate.

However, when the chilled water flow rate is reduced to obtain a lower cooling capacity during part load, the error does not necessarily affect the simulated cooling capacity. If the system is PI-controlled, the cooling capacity will be what is required to obtain the cooling-set point, and the error, will manifest as an erroneously simulated chilled water flow rate (x-axis in Figure 21) and chilled water return temperature. In a P-controlled system on the other hand, the error will manifest in the simulated cooling capacity. With the most common control methods, simple on/off and more sophisticated pulse width modulation, the chilled water flow rate is always in one of the two points where all three equations yield equal cooling capacity, hence no difference.

If also including non-constant flow of primary air, an error is introduced by the interpolation of the empirical constant k. The magnitude of this error ultimately depends on how many different air flows k was determined at. For the active chilled beam used in this study, interpolation between primary air flows of 0 and 30 l/s yields an error of around 10 % in the mid-range. However, interpolation between 20 and 30 l/s yields an error of only 1 % in the mid-range. Errors are effectively minimized by avoiding too wide ranges of interpolation.

There are numerous of active chilled beam models, in this discussion, the one presented in Paper II is compared to very simple models. The purpose is to show the quality of the simple models rather than the necessity of the advanced one.

5.3 Zone modelling

In Paper III, following two modifications of the zone model were done in order to make the model more consistent with reality.

- Induction was taken into account when determining the internal convection.
- Convection at the ceiling was determined based on the supply air temperature.

The first one is uncontroversial, since it is just an implementation of an often neglected fact. The changed reference temperature is not as obvious and need more motivation. First of all, it shall be noted that neither the conventional assumption nor this modification is a correct representation of reality. The air at the surfaces (walls, floor and ceiling) is generally neither as warm as the exhaust air nor as cold as the supply air. In the general implementation of the heat balance method however, it is assumed that the room air temperature is uniform and equals the exhaust air temperature. A correctly installed active chilled beam should ensure that the supply air attaches to the ceiling by the Coandă effect. This motivates the assumption that the temperature of the air at the ceiling is better represented by the supply air temperature than by the exhaust air temperature. An even more correct approach would be to assume it to be somewhere in between the supply and exhaust air temperatures. Since this is true also for the walls and the floor, the chosen approach may be seen as a compromise.

In Paper III, it was not explicitly investigated to what extent the activation of the thermal mass was affected by the increased internal convection. Given the fact that the modifications lead not only to a decrease in maximum operative temperature, but also an increase in maximum air temperature, it is reasonable to assume that thermal mass activation was not significantly influenced. In other words, with reference to the bulleted list in the very end of chapter 2.4, the two first phenomena (increased cooling load and larger difference between cooling load and required heat extraction rate) seem to cancel each other out while the main influence is the reduced difference between air and operative temperature.

The reduced difference between the air and operative temperature may be considered negligible in conventional systems but is particularly noteworthy in a high temperature cooling system due to the small driving temperature difference between chilled water and room air. The high chilled water temperature makes the cooling capacity very sensitive to changes in the room air temperature (which is also the key to self-regulation).

In addition to influencing the activation of the thermal mass and the difference between air temperature and operative temperature, increased internal convection also influence the thermal transmittance of the building envelope. The convection coefficient increased by the induction (from 2.5 to 6.6 W/m^2K) implies that a wall with an overall U-value of 0.20 W/m^2K is increased to 0.21 W/m^2K.

In the room where the experiments of Paper III were carried out, the active chilled beam occupied 11 % of the total ceiling area. In rooms with considerably less ceiling area represented by active chilled beams, the use of supply air as reference temperature is less reasonable.

It shall be noted that the work presented in Paper III did not include any direct measurements of the convective heat transfer. In contrast, this was done by Le Dréau et al. (2015) who concluded that existing correlations (presented by Fisher & Pedersen (1997) and used in this thesis) tend to overestimate the convection. Though the results from Le Dréau et al. (2015) is a more comprehensive model based on direct measurement of internal convection in a room with an active chilled beam, Paper III presents simple modifications of how the internal convection is modelled in IDA ICE today. These modifications tend to improve the accuracy of the results.

The open-plan offices modelled in Paper V and Paper VI where divided into several zones as a way to capture spatial differences regarding air temperature. Air flow between these zones was driven by the buoyant forces caused by temperature differences. In reality, this air flow is also driven by the movements of people, supply air jets and unbalanced ventilation, but also counteracted by furniture and furnishings.

5.4 Self-regulation

In principle, self-regulation is equivalent to P-control with a proportional band ranging from the chilled water supply temperature without upper limit. The cooling acts instantaneously on a change in air temperature, but as all proportional control with wide proportional band, self-regulation is stable, cautious and always associated with an offset error.

Results from Paper V show that a self-regulating system may precool the building prior to the peak cooling load and consequently implies lower required cooling capacity. However, this is valid at a low enough chilled water temperature and compared to a high enough cooling set-point for a conventional system. Compared to a cooling set-point of 23 °C, a self-regulating system with a chilled water supply temperature of 20 °C required lower peak cooling capacity while a chilled water supply temperature of 22 °C required a higher peak cooling capacity.

Due to the fact that the difference between operative and air temperature was slightly lower in the self-regulating case, the required heat transfer area was reduced slightly more than the required cooling capacity. Another present phenomena was that the peak cooling capacity of different zones occurred more simultaneously in the self-regulating systems. Having the peaks more distributed over time is beneficial for total peak cooling capacity but does not affect the total required heat transfer area.

As seen in Figure 19, the peak-shaving of self-regulation is associated with lower indoor temperatures prior to the peak cooling load. Compared to a system with a cooling set-point of

24 °C the average operative temperature is 0.6 °C lower. This is not considered a too low operative temperature, especially since Bourdakis et al. (2018) found that people who commute to the office by foot or by bike prefer substantially lower indoor temperature during mornings. They concluded that the room air temperature at the beginning of the occupancy period could be 20.0-21.5 °C and should then increase at a rate of 1.5 K/0.5 h to reach the range of 23.0-26.0 °C. This implies that the temperature in the self-regulating system might be more preferable than in the controlled system.

Lower indoor temperature during summer is associated with a higher cooling demand. However, this is less detrimental in airtight buildings with low transmission losses and efficient ventilation recovery, which is common in Northern Europe as a consequence of minimizing heat losses during cold winters. In contrast to heat losses, the major part of heat gains in office buildings are independent of temperature.

There are countries where individual room temperature control is required by law (Babiak et al., 2013). The arguments for this are potential energy savings and improved comfort. These arguments are stronger in low temperature system where the cooling energy is more valuable and in single office rooms where one single person manages the thermostat. In adaptive comfort theory, it is well known that people are more forgiving of discomfort if they have control of alleviating it. Further, significant correlation between perceived control and perceived productivity has been reported (Leaman & Bordass, 2010). However, this correlation is stronger in bad performing buildings (performance regarding heating, cooling, ventilation, lighting and noise) and gets weaker as buildings perform better. Based on this, self-regulating active chilled beams fit better in well-performing buildings.

Self-regulation may cause overcooling and consequently excessive use of energy and too low room temperatures. This was concluded by Ruponen et al. (2010) and also observed in the strategy with a constant chilled water temperature in Paper VI. However, in Paper VI it was also concluded that overcooling may be avoided by increasing the chilled water supply temperature during winter and mid-season. Strategy OA1021 (see Figure 20) is close to how it is controlled in the real building presented in Paper IV. Both the investigation of the real building (Paper IV), and the simulation in Paper VI indicate that the chilled water temperature may be further increased without causing too high room temperatures.

Cooling system

In the building investigated in Paper IV, only slightly more cooling was extracted from the ground storage in the summer of 2018 than recharged during the winter (Figure 17). This was thanks to the fact that more cooling was recharged during this winter than any other year. 2018 did not only include an extremely warm summer but also the coldest winter since 2013. A more significant imbalance would occur if also the winter was extraordinary warm. To avoid ground storage imbalances in such cases, the set-points determining the recharging might be subject to changes. In 2018, Stockholm did not only experience the hottest summer ever recorded in Sweden. Also the number of hours of sunshine was the highest ever recorded in Stockholm, which had a significant influence on the cooling load.

An important feature of the building investigated in Paper IV was the fact that not only the supply temperature to the active chilled beams were high (at least 20 °C), but also to the cooling coil in the air handling unit (17 °C). This enabled avoiding a chiller. In more humid climates, it may be necessary to provide the cooling coil with colder water in order to dehumidify the supply air. In those cases, it is important to not use the same chiller for the cooling coil as for the chilled beams, hence losing major benefits of high temperature cooling.

One drawback of self-regulating systems is the constant full-speed operation of the chilled water circulation pump. As presented in Paper IV, the annual electricity use of the circulation pump was 0.28 kWh/m^2.

Thermal climate

In Paper IV, it was concluded that the indoor air temperatures in the investigated building were both stable and uniform, which indeed is a matter of definition.

Indoor air temperatures presented in Paper IV include open-plan offices only. Open-plan offices definitely facilitate the ability to maintain stable and uniform temperatures since spatial differences in cooling load are levelled off by the air movements. This is in contrast with meeting rooms which are separated by walls. On the other hand, occupied meeting rooms require additional air flow due to air quality reasons, and this provides cooling since its supply temperature is lower than the room air temperature. Air temperatures in meeting rooms have previously been measured in the investigated building and were presented in Publication 2 (see List of publications). Differences between open-plan offices and meeting rooms are presented in Figure 22 which include working hours between 1 November 2015 and 31 October 2016.

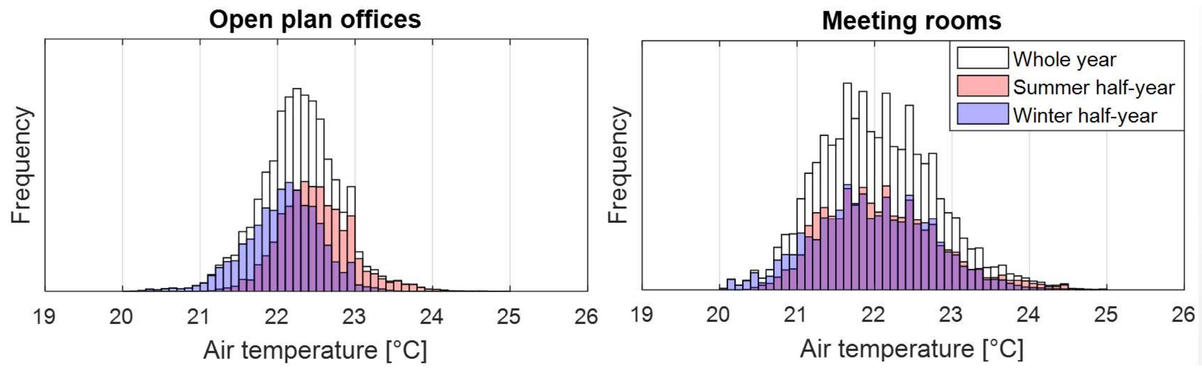

Figure 22 Indoor air temperatures in the open-plan offices (left) and meeting rooms (right) during working hours 1 November 2015 – 31 October 2016 (Purple color implies that red and blue bars overlap).

As seen in Figure 22, the temperatures vary more in the meeting rooms, but they are less affected by the season, which is explained by the fact that they are located in the core of the building. Private office rooms were not included in this thesis, neither measurements nor simulations. In private office rooms (and meeting rooms) without additional air flow during occupancy, the temperature variability is in all likelihood higher than presented in this thesis.

To get an impression of the magnitude of the variability of indoor air temperatures, they can be related to the variability of activity level through the theory of predicted mean vote (PMV) determined according to ISO 7730 (ISO, 2005). This theory predicts the mean vote, regarding the thermal climate, of a large group of persons on a seven-point scale ranging from cold to hot. Parameters required to calculate the predicted mean vote (PMV) are metabolic rate, clothing insulation, air temperature, mean radiant temperature, air velocity and relative humidity. In the following example, clothing insulation is determined as a function of the outdoor air temperature at 06:00 as described by ASHRAE Standard 55 (ASHRAE, 2013). The air velocity is 0.1 m/s in accordance with the average air velocity measured in the laboratory during the work presented in Paper III. The mean radiant temperature is set equal to the air temperature. Then, based on the air temperatures and the exhaust air relative humidity measured in the investigated building during 2018, PMV values are according to Figure 23. Three levels of metabolic rate are included.

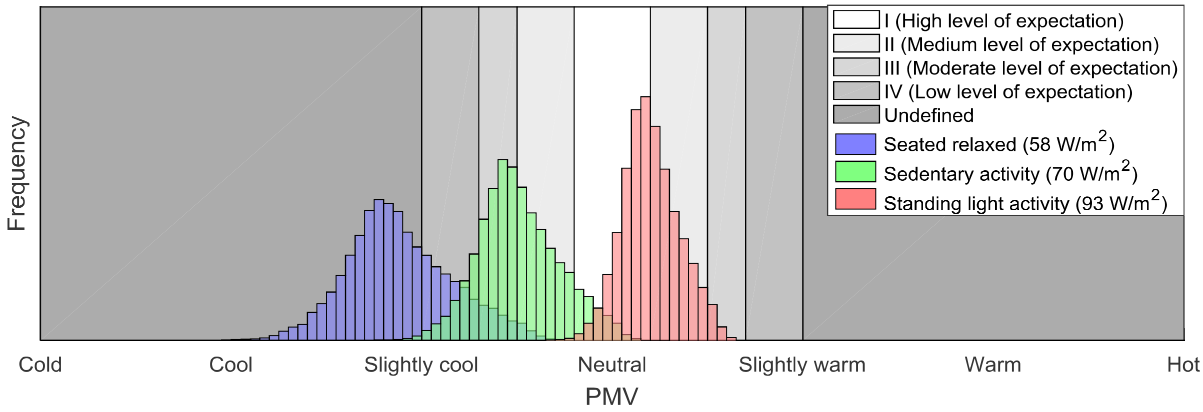

Figure 23 Predicted mean votes corresponding to all room air temperatures measured in the investigated building during working hours 2018.

The legends I-IV refer to the categories of thermal environment specified by European standard EN 16798-1:2019 (CEN, 2019). From Figure 23, it can be concluded that the indoor temperatures vary too much to always stay within the highest level of expectation. But also that people's activity level influences the PMV to a much larger extent. On one hand, it seems impossible to satisfy both a seated relaxed and a standing person in the same thermal environment. But on the other hand, adjusting your activity level to your thermal perception seems like a very efficient measure to find a neutral thermal sensation.

The assumption that the mean radiant temperature equals the air temperature is not undisputable. In several studies based on measurements, it has been concluded that the difference is very small (Dawe et al. 2020). On the other hand, simulations have shown differences of around 1.0 °C between air and operative temperature (equal to around 2.0 °C between air and mean radiant temperature). This refer to simulations carried out both internally by Skanska and by other researchers (Kosonen & Penttinen, 2017). However, based on the results from Paper III, it is reasonable to assume that simulations overestimate the difference between operative and air temperature in rooms equipped with active chilled beams.

Applicable standards and guidelines suggest indoor temperatures higher than measured in the investigated building, especially during summer. The measured temperatures, however, are in line with what several HVAC-designers in Sweden consider normal, in line with temperatures observed in other Swedish offices and have not caused excessive complaints. To clarify this discrepancy between theory and practice, a questionnaire based survey-study is recommended. If actually preferred, the low temperatures could be remedied by increasing the chilled water temperature, without detrimental effects on thermal uniformity.

6 Conclusions

The significant contributions of this thesis may be summarized as follows.

- Two novel methods of measuring the induction ratio of active chilled beams.
- Novel observation of how the induction ratio is influenced by temperatures of chilled water and room air.
- An accurate thermal model of an active chilled beam that takes into account the above-mentioned observation, valid in a wide range of operating conditions including laminar flow of chilled water.
- Identification of easily implemented modifications to improve how the building performance simulation software IDA ICE models internal convection in rooms with active chilled beams.
- Quantification of the peak-shaving effect of self-regulating active chilled beams.
- Simulations showing that overcooling in a self-regulating system is effectively avoided by increasing the chilled water temperature outside of the summer season.
- Evaluation of indoor air temperatures and performance of the cooling system in a building with direct ground-coupled self-regulating active chilled beams, including the unprecedented hot summer of 2018.

The numerical results of this thesis apply to the circumstances under which they were determined and should not be generalized without caution. These may be summarized as followed.

- The influence of chilled water supply temperature on induction ratio was $0.08\ K^{-1}$.
- The increased induction ratio caused by increasing the chilled water supply temperature by 5 °C resulted in 3 % higher cooling capacity.
- The internal convection model in IDA ICE was modified by taking induction into account and by using the supply air temperature as reference temperature for internal convection at the ceiling. If designing for a certain operative temperature, this decreased the required heat transfer area by 9 %, in the single office (Paper III) as well as in the office floor (Paper V).
- Self-regulation is associated with precooling prior to the peak cooling load. This reduced the required heat transfer area by 19 % and 5 % compared to a system with a cooling set-point of 24 °C and 23 °C respectively.

- By controlling the chilled water supply temperature aiming at an exhaust air temperature set-point of 24 °C, a self-regulating system requires less energy for heating and cooling than a system with individually controlled active chilled beams, without causing thermal discomfort.

- During working hours in 2018, average difference between the highest and lowest temperatures measured in the open-plan offices in the real building with self-regulating active chilled beams was 1.9 °C, and below 2.4 °C during 90 % of the time. The average standard deviation of individual sensors was less than 0.5 °C and the maximum standard deviation was 0.8 °C.

- During 2018, the cooling system in the real building with self-regulating active chilled beams supplied 19.3 kWh/m^2 of cooling. Electricity use allocated to the cooling system represented 1.12 kWh/m^2 which implies a figure comparable to the coefficient of performance (COP) of a chiller of 17.2.

Answers to the research questions presented in chapter 1.2 are formulated as followed:

- Are current models adequate for simulating self-regulating active chilled beams?
    - Current active chilled beam models accurately simulate the cooling capacity and are adequate for sizing the system and simulating the operation. This requires that empirical coefficients are determined at operating conditions close to the intended use and that interpolation over wide ranges is avoided.
    - When implementing an active chilled beam model in a zone model, induction of room air must be taken into account to accurately determine the internal convection of the room. In addition, using the supply air temperature as reference temperature when determining convection at the ceiling tends to correspond better with reality.

- Are self-regulating active chilled beams able to provide thermal comfort in Swedish office buildings?
    - Self-regulating active chilled beams are capable of providing a stable and uniform thermal climate in open-plan offices and meeting rooms. In open-plan offices, this is facilitated by the natural mixing of air. In meeting rooms, it is facilitated by additional air flow supplied for indoor air quality reasons.

- What are the consequences of different design and operation of self-regulating active chilled beams?
    - Designing for higher chilled water temperature implies more uniform and stable thermal climate. The drawback is larger required heat transfer area.
    - Operating with higher chilled water temperature implies better use of free cooling, less risk of overcooling, less need for dew point control, less thermal distribution losses and more induction of room air. The drawback is less peak shaving.

7 Future research

Following topics are suggested for future research.

- Investigating possibilities and limitations of applying high temperature cooling and specifically self-regulating active chilled beams in warmer regions. Global impact of the benefits of this technology requires it to be installed where comfort cooling demand is several fold higher than in Sweden. Key issues for such investigation is humidity, cooling load variability and generation of cooling.

- Investigating the magnitude of oversized active chilled beams in real buildings. Oversizing, conventionally compensated for by the control system, is a possible answer to why self-regulating system provide lower room temperatures than expected. HVAC systems are generally oversized by compounding safety factors at multiple levels in order to avoid problematic consequences of undersizing. The feedback generated from such investigation would not only lead to better design and performance of self-regulating active chilled beam systems but also improved sizing of HVAC systems in general.

- Investigating the economic consequences of applying high temperature self-regulating active chilled beams. Not only the initial savings from avoiding a certain amount of control valves, thermostats and pipe insulation (or thermal distribution losses) versus the larger active chilled beams, but also savings regarding maintenance of those components. The investigation may also include a life cycle analysis, weighing larger heat transfer area against the savings they enable.

- Conducting a thermal comfort survey in an office building with self-regulating active chilled beams. Objective measurements of thermal climate can only be used as an indication of the perceived thermal comfort. The subjective perceived thermal comfort of the occupants is more important and can only be identified by a questionnaire based survey.

- Investigating to what extent there is a discrepancy between theory and reality regarding preferred thermal environment. Indoor air temperatures observed during the work of this thesis are generally lower than suggested by standards based on the PMV-theory. This is not limited to buildings with self-regulating active chilled beams but may possibly be a general phenomenon in Swedish office buildings. Whether this is true and in that case whether it is due to erroneously assumed activity levels, clothing insulation or the theory itself, would be very valuable input to HVAC designers.

References

Aebischer, B., Catenazzi, G. & Jakob, M. (2007). Impact of climate change on thermal comfort, heating and cooling energy demand in Europe. In *Proceedings eceee Summer Study "Saving Energy – Just do it!",* pp. 859-870.

Alexander, D. & O'Rourke, M. (2008). Design considerations for active chilled beams. *ASHRAE Journal*, *50*(9), 50-58.

ASHRAE (2013). ANSI/ASHRAE Standard 55-2013: Thermal environmental conditions for human occupancy. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Atlanta, GA.

ASHRAE (2017). *ASHRAE Handbook, Fundamentals Volume*. American Society of Heating Refrigeration and Air-Conditioning Engineers, Inc., Atlanta, GA, 2017.

ASHRAE (2018). ANSI/ASHRAE Standard 200-2018: Methods of testing chilled beams, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Atlanta, GA.

Babiak, J., Olesen, B. W., & Petras, D. (2013). *Low temperature heating and high temperature cooling, Guidebook no 7*. REHVA.

Beausoleil-Morrison, I. & Strachan, P. (1999). On the significance of modeling internal surface convection in dynamic whole-building simulation programs. *ASHRAE Transactions*, 105, 929-940.

Betz, F., McNeill, J., Talbert, B., Thimmanna, H. & Repka, N. (2012). Issues arising from the use of chilled beams in energy models. In *Proceedings SimBuild*, *5*(1), pp. 655-667.

Bourdakis, E., Simone, A. & Olesen, B. W. (2018). An experimental study of the effect of different starting room temperatures on occupant comfort in Danish summer weather. *Building and Environment*, *136*, 269-278.

Brister, A. (1995). Chilled beams provide Perpetual cooling. *Building Services*, 17-20.

BSI (1973). BS 4954-1: Testing and rating induction units for air distribution systems – Part 1: Thermal and aerodynamic performance. British Standards Institution. London: BSI.

Brown, G. (1990). The BRIS simulation program for thermal design of buildings and their services. *Energy and Buildings*, *14*(4), 385-400.

Cammarata, G. & Petrone, G. (2008). A numerical investigation on active chilled beams for indoor air conditioning. In *Proceedings of the COMSOL Conference,* 2008, Hannover, Germany.

CEN (2008). EN 15116:2008. Ventilation in Buildings - Chilled Beams - Testing and Rating of Active Chilled Beams. European Committee for Standardization, Brussels.

CEN (2019). EN 16798-1:2019. Energy Performance of Buildings - Ventilation for Buildings - Part 1: Indoor environmental input parameters for design and assessment of energy performance of buildings addressing indoor air quality, thermal environment, lighting and acoustics – Module M1-6. European Committee for Standardization, Brussels

Chantrasrisalai, C., Fisher, D.E., Iu, I., & Eldridge, D. (2003). Experimental validation of design cooling load procedures: The heat balance method. *ASHRAE Transactions,* 109(2), 160-173.

Chen, C., Cai, W., Giridharan, K. & Wang, Y. (2014). A hybrid dynamic modeling of active chilled beam terminal unit. *Applied energy, 128*, 133-143.

Dawe, M., Raftery, P., Woolley, J., Schiavon, S. & Bauman, F. (2020). Comparison of mean radiant and air temperatures in mechanically-conditioned commercial buildings from over 200,000 field and laboratory measurements. *Energy and Buildings*, *206*, 109582.

De Clercq, B., Deltour, B. & Van Overloop, J. (2013). Measuring and modelling heat exchange capacity of active chilled beams. In proceedings *Clima 2013. 11th REHVA*

World Congress and 8th International Conference on Indoor Air Quality, Ventilation and Energy Conservation in Buildings, pp. 1097-1105
DOE (2018). EnergyPlus Engineering Reference 8.9.0. The Reference to EnergyPlus Calculations, Department of Energy, United States. Retrieved from https://energyplus.net/sites/all/modules/custom/nrel_custom/pdfs/pdfs_v8.9.0/EngineeringReference.pdf
Domínguez-Muñoz, F., Cejudo-López, J. M. & Carrillo-Andrés, A. (2010). Uncertainty in peak cooling load calculations. *Energy and Buildings*, *42*(7), 1010-1018.
EQUA Simulation AB. (2010). Validation of IDA Indoor Climate and Energy 4.0 build 4 with respect to ANSI/ASHRAE Standard 140-2004. Retrieved from http://www.equaonline.com/iceuser/validation/ASHRAE140-2004.pdf
Fisher, D. E. (1995). *An experimental investigation of mixed convection heat transfer in a rectangular enclosure.* Doctoral dissertation. University of Illinois at Urbana-Champaign.
Fisher, D. E. & Pedersen, C. O. (1997). Convective heat transfer in building energy and thermal load calculations. *ASHRAE Transactions, 103*(2), 137–48.
Freitag, H., Mathis, P. & Müller, D., Influence of thermal Boundary Conditions on the entrainment Behavior of an Active Chilled Beam. In proceedings Clima 2016. *12th REHVA World Congress*: *Volume 5*. Aalborg: Department of Civil Engineering, Aalborg University.
Frenger (2014). *Eco HQ Active chilled beam* [Product Sheet]. Retrieved from https://www.frenger.co.uk/pdfs/eco-hq-v1.1.pdf
Guan, Z. & Wen, C. (2016). Geometric optimization on active chilled beam terminal unit to achieve high entrainment efficiency. *Applied Thermal Engineering*, *98*, 816-826.
Henderson, M., Mech, B. E. & Mieaust, M. (2003). Application of chilled technology in modern office buildings. In Natural Hybrid or Air-conditioning Conference, Melbourne, Australia.
Hyun, J. S., Kim, J. Y., Shin, H. J. & Kim, J. S. (2014). Induction performance design of chilled beam system. In *Proceedings ACRA 2014 - The 7th Asian Conference on Refrigeration and Air Conditioning,* Jeju Island, South Korea.
IEA (2017). Energy Technology Perspectives 2017. International Energy Agency, Paris. Retrieved from https://www.iea.org/reports/energy-technology-perspectives-2017
IEA (2018). The Future of Cooling - Opportunities for energy-efficient air conditioning, International Energy Agency, Paris. Retrieved from https://www.iea.org/reports/the-future-of-cooling
IEA (2019). Tracking Buildings, International Energy Agency, Paris. Retrieved from https://www.iea.org/reports/tracking-buildings
ISO (1998). ISO 7726:1998. Ergonomics of the thermal environment - Instruments for measuring physical quantities. International Organization for Standardization, Geneva.
ISO (2005). ISO 7730:2005. 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. International Organization for Standardization, Geneva.
Ji, K., Cai, W., Zhang, X., Wu, B. & Ou, X. (2019). Modeling and validation of an active chilled beam terminal unit. *Journal of Building Engineering*, *22*, 161-170.
Karlsson, H. (2010). *Thermal Modelling of Water-Based Floor Heating Systems - supply temperature optimisation and self-regulating effects*. Doctoral dissertation. Chalmers University of Technology.
Kosonen, R. & Tan, F. (2005). A feasibility study of a ventilated beam system in the hot and humid climate: a case-study approach. *Building and environment*, *40*(9), 1164-1173.

Kosonen, R. & Penttinen, J. (2017). The effect of free cooling and demand-based ventilation on energy consumption of self-regulating and traditional chilled beam systems in cold climate. *Indoor and Built Environment*, 26(2), 256-271.

Le Dreau, J., Heiselberg, P. & Jensen, R. L. (2015). Experimental investigation of the influence of the air jet trajectory on convective heat transfer in buildings equipped with air-based and radiant cooling systems. *Journal of Building Performance Simulation*, *8*(5), 312-325.

Leaman, A., & Bordass, B. (2006). Productivity in buildings: the 'killer'variables. In *Creating the productive workplace* (p. 163). Taylor & Francis.

Lindab (2019). *Lindab Solus Active chilled beam*. Retrieved from https://itsolution.lindab.com/LindabWebProductsdoc/pdf/Documentation/Comfort/Lindab/Technical/Solus.pdf

Livchak, A. & Lowell, C. (2012). Don't turn active beams into expensive diffusers. *ASHRAE Journal*, *54*(4), 52-60.

Loudermilk, K. (2009). Designing chilled beams for thermal comfort. *ASHRAE Journal*, *51*(10), 58-60.

Loutzenhiser, P., Manz, H., & Maxwell, G. (2007). Empirical validations of shading/daylighting/load interactions in building energy simulation tools. *A Report for the International energy Agency SHC Task*, *34*.

Maccarini, A., Hultmark, G., Bergsøe, N. C., & Afshari, A. (2019). Full-scale operation of a novel two-pipe active beam system for simultaneous heating and cooling of office buildings. In *Proceedings E3S Web of Conferences* (Vol. 111). EDP Sciences.

Maccarini, A., Hultmark, G., Vorre, A., Afshari, A., & Bergsøe, N. C. (2015). Modeling of active beam units with Modelica. *Building Simulation, 8*(5), 543-550.

Maccarini, A., Wetter, M., Afshari, A., Hultmark, G., Bergsøe, N. C. & Vorre, A. (2017). Energy saving potential of a two-pipe system for simultaneous heating and cooling of office buildings. *Energy and Buildings*, 134, 234-247.

McQuiston, F. C., Parker, J. D. & Spitler, J. D. (2004). *Heating, ventilating, and air conditioning: analysis and design, Sixth Edition*, John Wiley & Sons Inc.

Olesen, B. W. (2012). Thermo active building systems using building mass to heat and cool. *ASHRAE Journal*, *54*(2), 44-52.

Peeters, L., Beausoleil-Morrison, I. & Novoselac, A. (2011). Internal convective heat transfer modeling: Critical review and discussion of experimentally derived correlations. *Energy and Buildings*, *43*(9), 2227-2239.

Rhee, K. N., Shin, M. S. & Choi, S. H. (2015). Thermal uniformity in an open plan room with an active chilled beam system and conventional air distribution systems. *Energy and Buildings*, *93*, 236-248.

Rumsey, P. (2010). Chilled Beams. *Heating/Piping/Air Conditioning Engineering*, *82*(1), 46-49.

Rumsey, P. & Weale, J. (2007). Chilled beams in labs: eliminating reheat & saving energy on a budget. *ASHRAE Journal*, *49*(1), 18.

Ruponen, M. (2009). *Operation and thermal comfort provision using induction units* (Doctoral dissertation, University of Leeds).

Ruponen, M., Itkonen, H., Mustakallio, P., & Jokisalo, J. (2010). Energy saving with very high temperature cooling water active chilled beam system. In *Proceedings of Clima 10th REHVA World Congress on sustainable energy use in buildings,* pp. 9-12.

Ruponen, M. & Tinker, J. A. (2009). A novel method to measure the air entrainment ratio of an active chilled beam. *International Journal of Ventilation*, *7*(4), 299-308.

Schultz, C. C. (2007). Next-generation cooling is looking up. *Engineered Systems*, 24(5), 64-67.

Santamouris, M. (2016). Cooling the buildings–past, present and future. *Energy and Buildings*, *128*, 617-638.
Setty, B. S. (2011). Application Issues for Chilled Beam Technologies. *ASHRAE Transactions*, *117*(1).
Spitler, J. D. (1990). *An experimental investigation of air flow and convective heat transfer in enclosures having large ventilative flow rates.* Doctoral dissertation. University of Illinois at Urbana-Champaign.
Stein, J., Taylor, S.T. (2013). VAV reheat versus active chilled beam & DOAS. *ASHRAE Journal*, 55(5), 18-32.
Vastyan, J. (2011). Chilled-beam basics. *Heating Plumbing Air Conditioning*, *83*(7).
Wang, Z., Li, J. & Wang, D. (2019). Performance test & analysis on fan coil unit and improved induction air unit system. In *IOP Conference Series: Earth and Environmental Science* (Vol. 371, No. 5, p. 052004). IOP Publishing.
Woollett, J. & Rimmer, J. (2015). *Active and Passive Beam Application Design Guide, Guidebook no. 21.* ASHRAE and REHVA.

Appendix A

Figure A1, A2, A3 and A4 corresponds to Figure 7, 8, 10 and 11 respectively. The only difference is the definition of the cooling load and the required heat extraction rate. In Figure 7, 8, 10 and 11, cooling load and required heat extraction rate are determined by the air temperature. In Figure A1, A2, A3 and A4, the cooling load and required heat extraction rate are determined by the operative temperature.

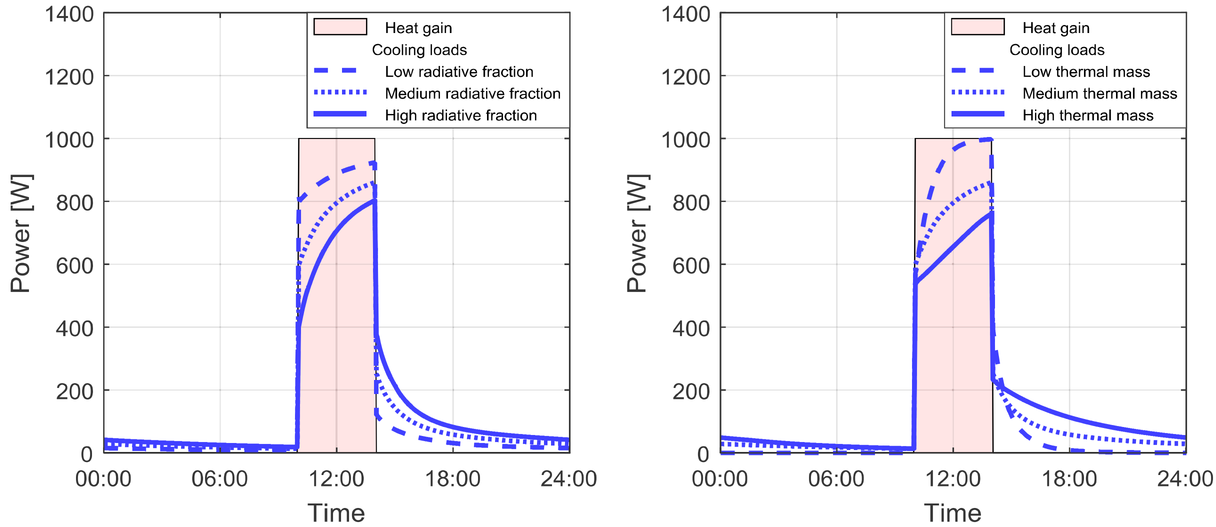

Figure A1 Heat gain and cooling load at different fractions of radiation (left) and different thermal mass (right).

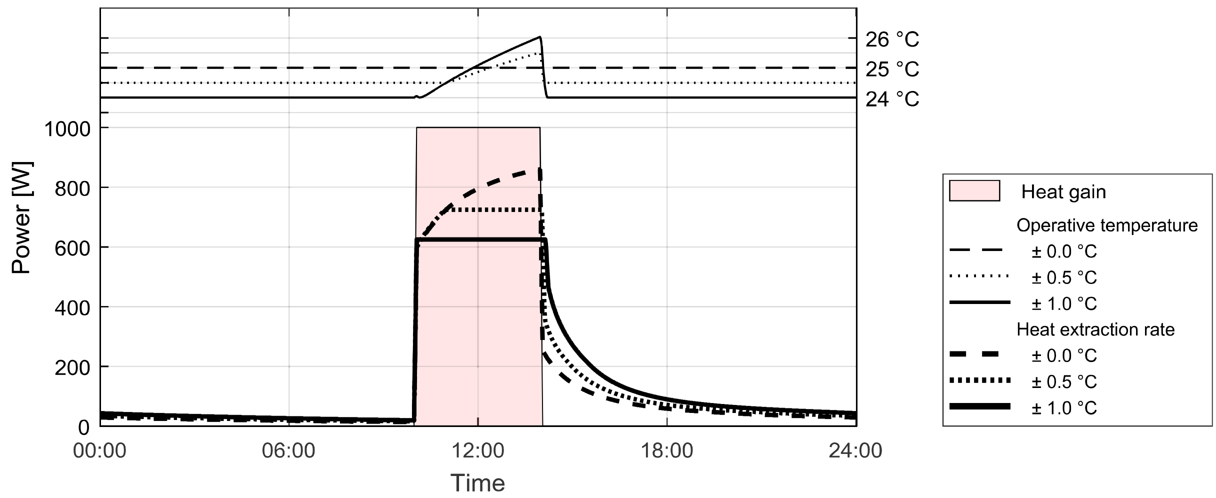

Figure A2 Heat gain, air temperature and heat extraction rate at three levels of temperature drifts.

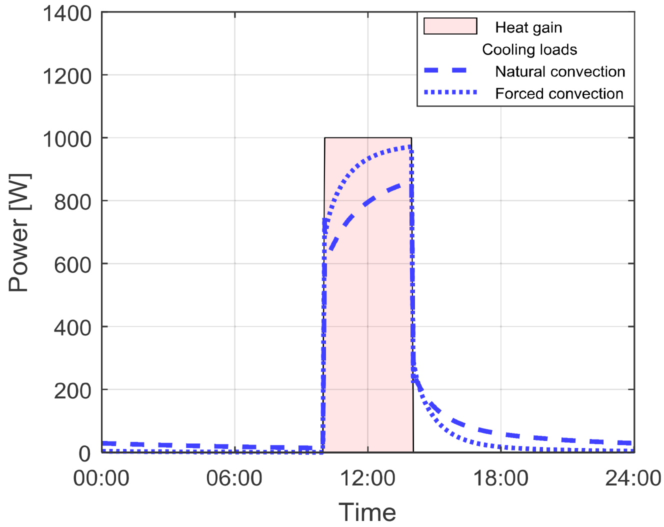

Figure A3 Heat gain and cooling load at different convective heat transfer coefficients

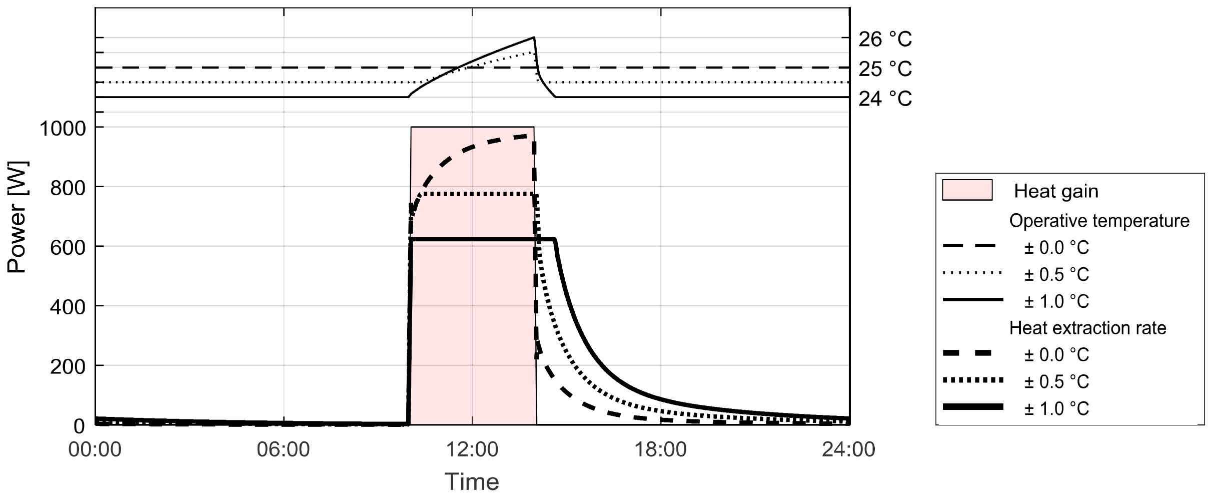

Figure A4 Heat gain, air temperature and heat extraction rate at high convective heat transfer coefficients and at three levels of temperature drifts.