THESIS FOR THE DEGREE OF DOCTOR OF PHILOSOPHY

Direct Ground Cooling Systems for Office Buildings

Taha Arghand

Department of Architecture and Civil Engineering

CHALMERS UNIVERSITY OF TECHNOLOGY

Gothenburg, Sweden 2021

Direct Ground Cooling Systems for Office Buildings
TAHA ARGHAND
ISBN 978-91-7905-447-2

Doktorsavhandlingar vid Chalmers tekniska högskola
Ny serie nr 4914
ISSN 0346-718X

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

Printed by
Chalmers Reproservice
Gothenburg, Sweden 2021

Direct Ground Cooling Systems for Office Buildings

TAHA ARGHAND

Department of Architecture and Civil Engineering

Chalmers University of Technology

Abstract

The solving of crucial global energy challenges hinges on improving energy efficiency in building energy systems. Accomplishing energy-efficiency targets often entails incorporating sustainable energy sources into the energy supply system. Direct ground cooling systems (DGCSs) are among the most sustainable technologies for comfort cooling in office buildings. With this technology, cooling is provided by the circulation of the working fluid through the ground heat exchangers. This technology is mostly used in cold climates where the underground temperature is low, and the building cooling loads are low enough to be offset by the ground cooling. Using only a modest amount of electricity to drive the circulation pumps, this technology is incredibly energy efficient. However, designing DGCSs presents some unique challenges, and only a handful of studies on this subject are available.

This work aims to develop knowledge about comfort cooling for office buildings using DGCSs and expand upon design and operation practices for this technology. The findings presented in this work are based on experimental and simulation results. The experimental results build upon existing knowledge for operating cooling systems and substantiate new operation methods for the DGCSs. The experimental results are also used to develop and validate simulations. The simulation results facilitate investigating the short- and long-term thermal and energy performance of the DGCSs for various design circumstances.

The borehole system design is usually performed independently from the building energy system design. In view of this work’s findings, considering the whole system (borehole, control system, terminal units) can enhance the design. A sub-system’s input design requirements can be aligned with the corresponding output of other sub-systems in a comprehensive design approach.

This work demonstrates and quantifies that terminal units with slow response, such as thermally active building systems (TABS), can smooth out the daily peak heat rejection loads

to the ground, resulting in shorter boreholes. Thus, the ground system can be much smaller than required for fast-response terminal units, such as active chilled beams.

This work analyses different operation practices for DGCSs. The results suggest that allowing the room temperature to rise somewhat during the “on-peak” cooling loads can reduce the ground heat rejection loads, for which shorter boreholes can be designed. If combined with precooling the space during the “off-peak” cooling loads, a further reduction in the ground loads is yielded.

This work also investigates the design and application of the DGCSs in existing office buildings. A systematic approach is provided to evaluate the influence of common renovation parameters on the design and energy performance of a DGCS. The systematic approach includes a step-by-step methodology to explain how sensitive the subsequent system design might be to the variations in the renovation parameters. Furthermore, the results quantify the potential electricity savings by using the DGCS instead of a chiller.

Keywords: Direct ground cooling; ground-coupled cooling; Geocooling systems; Passive ground cooling; Comfort cooling; Thermal comfort; Boreholes; Ground heat exchangers; Active chilled beams; Ceiling cooling panels; TABS; GeoTABS; Renewable energy; Energy renovation; Energy efficiency; High-temperature cooling; Optimization

Acknowledgements

It takes a village to raise a “PhD thesis”! This thesis bears the name of only one person but others also deserve credit for it.

I am deeply grateful to my supervisors Saqib Javed and Anders Trüschel and my examiner Jan-Olof Dalenbäck. Saqib, your knowledge, guidance and endless support for reading and re-reading my manuscripts have been priceless contributions to this work. Anders, your support, insights and experience have been crucial for me throughout my time as a PhD student. Jan-Olof, thanks for always being available, having endless patience and enthusiasm, and giving inspiring discussions. I consider myself very fortunate to have had the opportunity to learn from all of you.

I appreciate the invaluable assistance of Håkan Larsson for helping me with practicalities in the laboratory and being a nice lunch buddy. Many thanks to Peter Filipsson for his time, interest and valuable discussions. I would also like to thank Jonas Gräslund for his support in the early stages of this work.

To the Building Services Engineering family; Amir, Blanka, Despoina, Dixin, Jan, Jenny, Johan, Lars, Linda, Maria, Maria, Mohammad, Sara, Sarka, Theo and Torbjörn. Thanks for all the great moments. I couldn’t wish for better colleagues and friends.

I saved the best for last; I am deeply grateful to my parents for all their endless support. Mom, dad and my sister, thank you for continuing to encourage me to do what I like and to send me motivational messages. Our bond is stronger than the distance between us. None of this would have been happened without you, literally!

Taha Arghand

Gothenburg, February 2021

This work was financially supported by the Swedish Energy Agency (Energimyndigheten) through its E2B2 and EffsysExpand national research programmes. The in-kind contribution of laboratory facilities by Lindab, Swegon and Uponor is gratefully appreciated.

List of publications

The thesis is based on the following peer-reviewed publications.

I. Arghand, T., Javed, S., Trüschel, A. and Dalenbäck, J.O., 2019. Control methods for a direct-ground cooling system: An experimental study on office cooling with ground-coupled ceiling cooling panels. *Energy and Buildings*, *197*, pp.47-56.

II. Arghand, T., Javed, S., Trüschel, A. and Dalenbäck, J.O., 2021. Cooling of office buildings in cold climates using direct ground-coupled active chilled beams. *Renewable Energy*, *197*, pp.47-56.

III. Arghand, T., Javed, S., Trüschel, A. and Dalenbäck, J.O., 2021. Influence of system operation on the design and performance of a direct ground-coupled cooling system. *Energy and Buildings*, *234*, 110709.

IV. Arghand, T., Javed, S., Trüschel, A. and Dalenbäck, J.O., 2021. A comparative study on borehole heat exchanger size for direct ground coupled cooling systems using active chilled beams and TABS. Accepted in *Energy and Buildings*.

V. Arghand, T., Javed, S., Trüschel, A. and Dalenbäck, J.O., 2021. Energy Renovation Strategies for Office Buildings using Direct Ground Cooling Systems. Accepted in *Science and Technology for the Built Environment*.

Additional publications not included in this thesis are:

1. Zhang, C., et al., 2021. Resilient cooling strategies: A critical review and qualitative assessment. Submitted to *Energy and Buildings*.
2. Arghand, T., Dalenbäck, J.O., Trüschel, A. and Javed, S., 2019. Some aspects of controlling radiant and convective cooling systems. Proceedings of *13th REHVA World Congress CLIMA 2019, Bucharest, Romania*.
3. Arghand, T., 2019. Direct-Ground Cooling Systems for Office Buildings: Design and Control Consideration. Licentiate Thesis. Chalmers University of Technology, Gothenburg, Sweden.

4. Arghand, T., Trüschel, A., Dalenbäck, J.O. and Javed, S., 2018. Dynamic Thermal Performance and Controllability of Fan Coil Systems. Proceedings of *9th International Cold Climate HVAC Conference, Kiruna, Sweden*.

5. Arghand, T., Bolashikov, Z., Pastuszka, Z., Kosonen, R., Aho, I., Kaczmarczyk, J. and Melikov, A, 2016. Individually controlled localized chilled beam in conjunction with chilled ceiling: Part 2- Human response. Proceedings of *4th International Conference on Indoor Air Quality and Climate, Ghent, Belgium*.

6. Arghand, T., Bolashikov, Z., Kosonen, R., Aho, I. and Melikov, A, 2016. Individually controlled localized chilled beam in conjunction with chilled ceiling: Part 1- Physical environment. Proceedings of *4th International Conference on Indoor Air Quality and Climate, Ghent, Belgium.*

List of acronyms

ACB	Active chilled beam
BHE	Borehole heat exchanger
CC	Chilled ceiling
COP	Coefficient of performance
DGCS	Direct ground cooling system
GHE	Ground heat exchanger
GSHP	Ground source heat pump
HTC	High-temperature cooling
PMV	Predicted mean vote
PPD	Predicted percentage of dissatisfied
TABS	Thermally active building systems

Table of contents

1. Introduction

This chapter explains general concepts, reviews previous related literature and provides a comprehensive insight into how the DGCSs are used in practice. The chapter also outlines the research context by explaining the motivations, hypotheses and objectives of the thesis.

1.1. Background

Global energy demand in the building sector could double by 2050, mainly because of growing access to energy sources in developing countries [1]. Partly because of climate change, and partly because of increased access to cooling systems in warm climates, building cooling applications have the fastest-increasing share of growing energy demand [2]. The associated impacts of increased energy demand on global warming are also considered to be strong [3,4].

Chillers are the most widely-used system for cooling water in the building cooling application. Chillers are predominantly driven by electricity, and sometimes by heat in absorption-based systems. Chillers usually provide chilled water to the building within the temperature range of 6 ℃ to 13 ℃ [5]. However, research results and field surveys have consistently shown the possibilities of adopting high-temperature chilled water (> 14 ℃) for comfort cooling [6–11]. It is claimed that generating low-temperature chilled water degrades the energy performance of the cooling system [12] and increases the thermal losses [13,14]. Thus, research has sought to develop cooling technologies using high-temperature chilled water for cooling indoor spaces.

The cooling systems generating high-temperature chilled water have a higher coefficient of performance (COP) than the traditional systems, as the compressor work is reduced [8]. Furthermore, thermal losses in cooling and transporting the water are minimized [13]. Adopting the high-temperature chilled water systems can also allow the use of renewable cooling sources, including solar, outdoor air, ground, nearby water body heat exchange, etc.

The ground is among the most resilient and environmental-friendly sources can be utilized for comfort cooling in buildings. Ground temperature is seasonally closer to the room temperature than other sources [15], such as outdoor air. Besides, access to other sources, such as lakes or rivers, depends on the building's location and might not always be feasible.

A common ground-coupled technology widely used for building cooling applications is the ground-source heat pump (GSHP). GSHPs are systems that use ground or groundwater as a heat sink and/or source [16]. For cooling applications, ground-source heat pump systems use chillers or reversible vapour compression cycles to move heat from the building to the ground. GSHPs operate more thermodynamically efficient than air-cooled chillers, as they reject heat to a source with a comparatively lower temperature [17]. Moreover, heat between the GHSP and the ground loop is exchanged through the liquid (brine, water, aqueous antifreeze) with a relatively higher heat capacity than air. However, GSHPs still use mechanical refrigeration.

Another ground-coupled technology which can be mainly used in cold climates is the direct ground cooling system (DGCS). DGCSs rely solely on the ground thermal properties and ground thermal storage to cool the heat carrier fluid in the cooling system without the use of mechanical refrigeration. In this technology, cooling is provided by circulating the heat carrier fluid in an array of U-pipes, known as ground heat exchangers (GHEs), vertically drilled in the ground. Figure 1.1 shows a simple schematic of a DGCS.

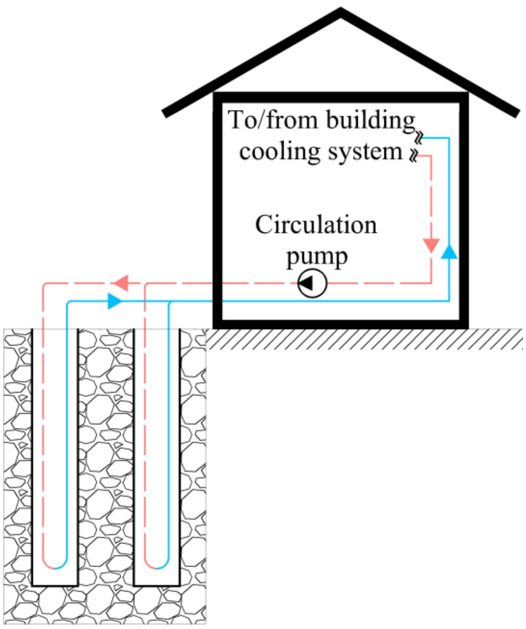

Figure 1.1. *Schematic of a DGCS using vertical GHEs.*

The term "direct" in DGCSs indicates that no energy conversion process is involved in cooling the heat carrier fluid. From a thermodynamics perspective, direct utilization is possible when the ground temperature is lower than the building temperature. The natural ground temperature below a few meters (< 10 m) from its surface is stable and is approximately equal to the average annual ambient temperature [18]. DGCSs are particularly feasible in cold to moderate climates where the ground temperature is low, and the building cooling loads are low enough to be offset by the ground. For instance, in Sweden, the ground temperature at a depth of 100 m varies from 3 ℃ in the north to 9 ℃ in the south [19], which is very suitable for high-temperature comfort cooling applications. Although the direct use of the ground might also be possible in warm climates, a small difference between the ground and indoor temperatures requires longer boreholes for which the economic and environmental benefits of using this system might be trivial.

Figure 1.2 shows a typical ground temperature distribution in the direct-cooling application. The upper and the lower limits are determined by the room temperature and the undisturbed ground temperature, respectively. The temperature difference between the lower and the upper limits determines the feasibility of using the DGCS [20]. The ground temperature increases in the short-term (daily) period and the long-term (seasonally/annually) period. The long-term temperature increase is associated with the yearly/long-term heat build-up in the ground with the highest impact on the thermal performance of the DGCSs at the end of the cooling period. Temperature increases in the short-term are related to the daily heat rejection loads to the ground. The daily loads affect the ground temperature in the borehole vicinity and hence, the fluid temperature leaving the borehole. However, short-term temperature increases are, to a great extent, recovered during the off-peak period on the same day, i.e. at night time, or in a few days [21,22].

Another feature shown in the figure is the terminal unit supply water temperature range. Generally, the lower the supply temperature, the higher the cooling capacity of a terminal unit. On the other hand, reducing the supply temperature requires longer/additional boreholes, resulting in higher drilling costs. Therefore, the choice of the supply temperature has a significant influence on the sizing of the boreholes.

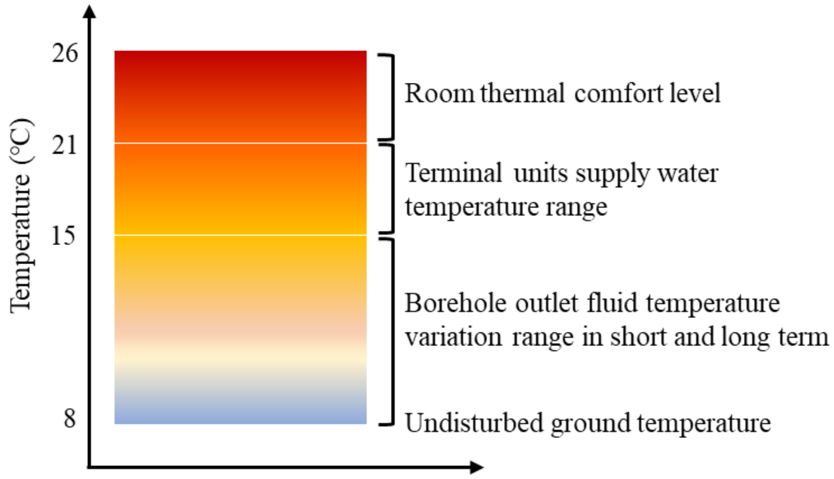

Figure 1.2. *Typical temperature levels in a DGCS in cold climates.*

In consideration of the above, an optimal design for the DGCSs entails smoothing out peak daily ground loads and reducing the annual ground loads. This can be done by minimizing the building cooling demand as much as possible during the design phase and also by considering effective measures in the design and operation of the DGCS. Therefore, design methodology of a DGCS is a package comprised of measures to reduce the building cooling loads, optimal design parameters for the selected terminal unit, and the cooling system's optimal operation.

1.2. Previous work

Earlier works on direct utilization of the ground generally concern heating applications [20,23–25]. Direct ground-cooling applications were mainly used for the pre-cooling of air in the ventilation systems [26–29] and the use of DGCS for comfort cooling was overlooked. This was partly because of high cooling demand in old premises, and partly because of underdevelopment of high-temperature cooling terminal units.

Direct ground-coupled cooling systems in Sweden are typically used to cool telecommunication stations, particularly in the northern part of the country [30]. Another common application is to use ground source heat pumps (GSHPs) for heating in winters and for free-cooling in summers. For example, the ground-coupled system at the Astronomy Centre at Lund University consists of 20 boreholes, each with a depth of 200 m and annually generates 300 MWh heating and 150 MWh cooling [31,32]. The system's thermal performance, defined as the proportion of delivered heat/cool to the electricity used, for heating and cooling is 4.8 and 50, respectively. Karlstad University has one of the largest ground-coupled heating and cooling systems in Europe, consisting of 204 boreholes, each of them 240 – 250 m deep [33]. The borehole system provides 5500 MWh of heating and 1000 MWh of cooling per year. The ground-coupled system for the new student centre at Stockholm University is equipped with 20 boreholes, drilled to a depth of 200 m, and has a heating and cooling performance of about 3.7 and 27, respectively [34]. The annual heating and cooling production is 200 MWh and 34 MWh, respectively. Note that in all cases above, the boreholes are sized for the GSHPs based on the buildings' required heating demand, because Sweden has a cold climate and heating is usually the dominant load. Therefore, the design of these cases is different from the typical design for the DGCSs.

There are only a handful of studies available on the DGCSs and they are mainly focused on analysing the energy performance of the system. Li et al. [35] used a DGCS to cool a student workroom in a cold region of China. The cooling performance of the system, expressed by seasonal performance factor (SPF), was 9.85. However, the measurements were performed for a short period (50 days) on a relatively small-scale office (~ 23 m^2). Eicker et al. [29] described a DGCS used for ventilation cooling and/or an embedded floor cooling system. The system's actual thermal efficiency was between 13 and 20 due to the small percentage share of the electricity demand for the pumps and the ventilation system. On a larger scale, Filipsson et al. [36] and Liu and Zhang [37] evaluated the thermal performance of a 21,200 m^2 large office equipped with a DGCS in Stockholm, Sweden. The SPF of the cooling system was calculated as high as 17, owing to the equipment's low annual operational electricity. The Ympäristötalo office building (~ 6800 m^2) in Helsinki, Finland uses a DGCS comprising 25 boreholes, each 250 m deep [38]. The annual cooling energy demand of the building was as low as 10.6 $kWh/m^2.y$.

Among the available building terminal units, only those designed to operate with high-temperature chilled water (> 14 °C) are functionally compatible with DGCSs. These terminals are known as high-temperature cooling (HTC) terminal units. HTC terminals include beams (active and passive), fan-coil units, embedded radiant cooling systems and radiant cooling panels [39–41].

Embedded terminals are the most widely used terminals in direct-cooling applications. These terminals have an intrinsic peak shaving effect due to their large thermal mass. Consequently, the ground peak hourly loads can be reduced, resulting in shorter boreholes. This cooling system has been used in different configurations, including embedded wall systems [42–44], radiant floor cooling systems [45] and thermally active building systems (TABS) [29,46,47]. Other HTC terminals incorporated with DGCS are active chilled beams (ACBs) [36,38] and fan-coil units [35].

1.3. Hypothesis

The provision of comfort cooling for commercial buildings is an integrated process often comprised of three main elements: the cooling plant, the distribution and the operational system(s), and the indoor terminal units. Traditionally, building energy simulations and borehole system (plant) simulations are performed separately. Performing separate simulations is time-consuming and does not give a comprehensive overview of the system’s design, challenges, advantages, and requirements. This work hypothesizes that energy-saving advantages offered by the DGCSs can be significantly improved by applying a holistic approach to design the system as a whole. The holistic approach entails the designing of each part of the system in relation to the requirements of the other parts. The hypothesis is based on the fact that the terminal units and operational strategies can influence boreholes’ sizing due to the differences made in the temperature levels of the system and the heat extraction rates from the building. The hypothesis also extends to the building design where a careful selection of the design parameters leads to a minimization of the building’s cooling loads and the increased energy performance of the DGCSs.

1.4. Research objectives

The general objective of this thesis is to develop knowledge about comfort cooling for office buildings using DGCSs. To achieve the general objective, the following specific objectives are defined:

- Demonstrating how indoor terminal units impact the sizing and thermal performance of the DGCSs.
- Demonstrating how system operation strategies impact the sizing and the thermal performance of the DGCSs.

- Expanding upon the possibilities of using the DGCSs for comfort cooling in existing office buildings in need of energy renovation and providing design improvement suggestions.
- Comparing a DGCS and a chiller-based cooling system in terms of building thermal performance and energy use, and providing quantitative information about the energy-efficiency benefits of using the DGCSs.

1.5. Limitations

This work only deals with comfort cooling for office buildings in cold climates. Other occupancy types, e.g. hotels, hospitals, shopping centres, residential premises, etc., are beyond the scope of this work. Furthermore, other climate zones are not used in the simulation studies.

This work is only concerned with the direct application of geothermal systems. Other mechanically-assisted cooling systems, e.g. air-cooled chillers and ground-coupled chillers, are not part of this dissertation, nor is the same for heating systems and ventilation systems for office buildings.

Another limitation is that the experimental and simulation studies reported in this work were performed using vertical closed-loop boreholes. The application of other types of GHEs for DGCSs are not investigated.

1.6. Research methodology

This thesis is initially developed based upon the existing fundamental principles related to ground-coupled cooling systems and building energy systems using HTC terminal units. Interviews with experts and group discussions, experimental laboratory studies and building energy simulation studies serve as the backbone of this work. The research methodology is discussed extensively in Chapter 2.

1.7. Thesis outline

This thesis includes seven self-contained chapters. Each chapter focuses on a topic relevant to the overall scope of the issues related to the DGCSs. The thesis is structured as follows:

Chapter 1 clarifies fundamental principles, summarizes previous related works and provides a background on how the technology is generally implemented. The chapter also outlines the research context by explaining the motivations, hypotheses and objectives of the dissertation.

Chapter 2 outlines the general methodology of the thesis.

Chapter 3 documents the main findings from interviews and discussions with practitioners. This chapter aims to highlight the main deficiencies in the design process and identify areas where research can be of help to designers' and consultants' needs.

Chapter 4 investigates how operational strategies influence the sizing of boreholes in DGCSs. The chapter includes experimental methodology and results, model development and validation, and simulation results, which address different operation strategies for the DGCSs. Chapter 4 draws from Papers I, II and III.

Chapter 5 focuses on studying the influence of different terminal units with their own specific controls on the design and sizing of DGCSs. Chapter 5 is based on Papers II and IV.

Chapter 6 deals with the application of the DGCSs for existing buildings in need of energy renovation. A systematic approach is developed and introduced to evaluate the influence of some renovation measures on the design and performance of a DGCS. Furthermore, the energy performance of a cooling system using a chiller or the DGCS is evaluated and quantified. Chapter 6 is based on Paper V.

Chapter 7 concludes the critical findings of this work and suggests possible future studies on this research area.

2. Methodology

The methodology of this thesis has been developed on the basis of interviews and discussions, experimental laboratory studies, and building energy simulations. The following outlines the rationale behind the methodology used in this work.

- The interviews and group/individual discussions aimed to ascertain the common design methods and the variations in the design methods from the perspective of design experts, manufacturers and research fellows. Indeed, these interviews and discussions formed the hypotheses of this thesis by identifying the areas in which research could be of help to designers' and consultants' needs.

- The experiments aimed to substantiate and advance the previously-used control methods for the HTC terminal units, with a particular focus on the DGCSs. The experimental studies also aimed to provide reliable data to develop and validate the simulation models of the DGCS. The experimental studies are performed in a climatically-controlled test room representing a mock-up of an office room. The cooling system is a DGCS consisting of a borehole, a terminal unit and a control system. For each experimental setup, standard control practices and design parameters for the terminal units are tested, and the possible implications of the results are evaluated.

- The simulation models facilitated the study of the system's performance over different time scales, e.g. daily, seasonally, and yearly. In addition, different operation strategies and configurations are developed and studied with simulations, which could not otherwise be investigated by the experiments. The simulation models of the DGCS for different office buildings are developed based upon literature review, standard practices, interviews with experts, and most importantly, experimental results.

2.1. Interviews and group discussions

The interviews and discussions were initiated to identify the gaps in the design of the HTC systems and the DGCSs. The target group was consultants, research fellows and industrial experts who demonstrate substantial practical experience in the design of the cooling systems for commercial buildings. The interviews were conducted in a semi-structured method. In addition, follow-up individual and group discussions were carried out to supplement and consolidate the interviews' outcomes, and were focused on the following topics:

- Early design considerations
- Comprehensive design approach
- Customized design approach for DGCSs
- Dynamic simulations of the whole system
- Use of different types of HTC terminal units
- Indoor temperature setpoints
- High-temperature cooling concept

2.2. Experimental studies and laboratory facilities

The borehole system consisted of a vertical single U-tube borehole heat exchanger (BHE) with an active depth of 80 m, Figure 2.1. The borehole was naturally filled with groundwater since the groundwater level on the site was high. The BHE was connected to the building loop via heat exchangers. Table 2.1 summarises the borehole specifications and thermal properties of the ground obtained by thermal response tests performed by Javed [48].

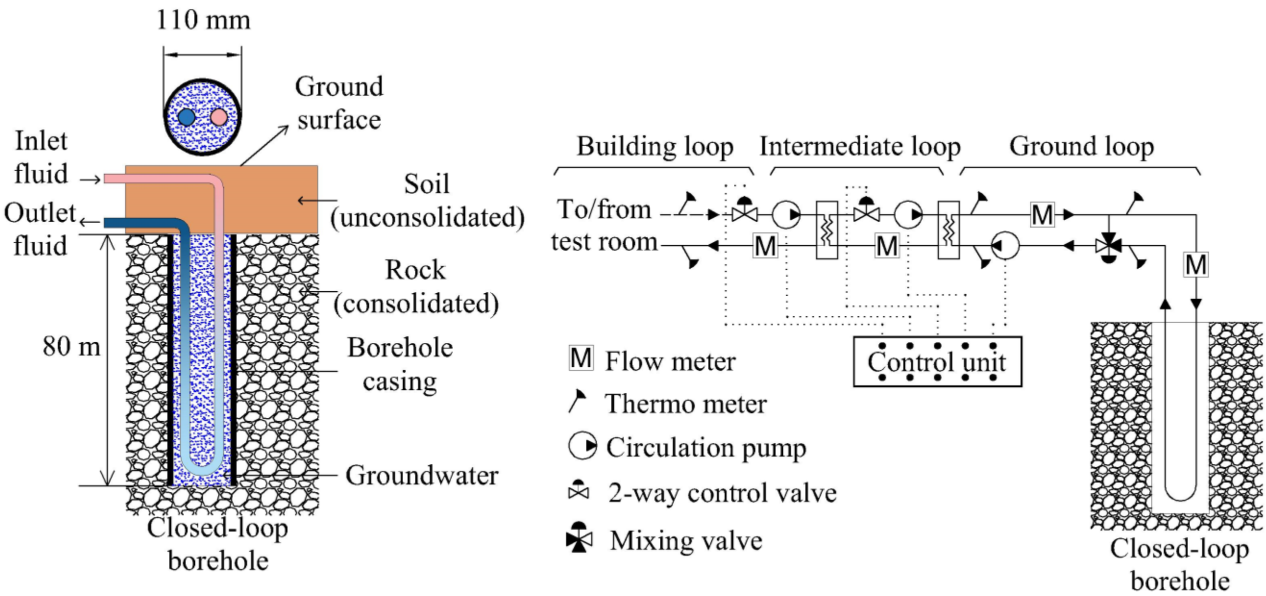

Figure 2.1. *Schematic of the groundwater-filled BHE with a single U-tube in the test facility and the laboratory borehole system.*

Table 2.1. *Ground and borehole system specifications for the experimental setup [48].*

Parameter (unit)	Specification
Borehole	
Active depth (m)	80
Diameter (mm)	110
Filling material	Groundwater
Thermal resistance (m.K/W)	0.059
Undisturbed ground temperature (°C)	8.3
Soil thermal conductivity (W/m.K)	2.88
U-tube	
Pipe type (-)	Polyethylene, PN8 DN40
Inner diameter (mm)	35.4
Outer diameter (mm)	40.0
Thermal conductivity (W/m.K)	0.42
Circulating fluid	
Type	Ethanol (29.5 %)
Thermal conductivity (W/m.K)	0.401
Specific heat capacity (J/kg.K)	4180

Experimental studies were performed in a laboratory test room located at the main campus of Chalmers University of Technology in Gothenburg, Sweden. The test room was a full-scale mock-up of a single-office room and was enclosed in a climatically-controlled room. Thus, the test room was protected from ambient air and solar radiation (see Figure 2.2A).

The test room was 12.6 m^2 large (4.2 m × 3.0 m) and had a suspended ceiling of 2.4 m high. The walls were 0.11 m thick and were made up of polystyrene panels with a finish of gypsum board panels. Three layers of an expanded polystyrene sheet (30 mm), plasterboard (6 mm) and a fibreboard (22 mm) separated the floor from the ground. The suspended ceiling consisted of 15 mm fibreglass panels.

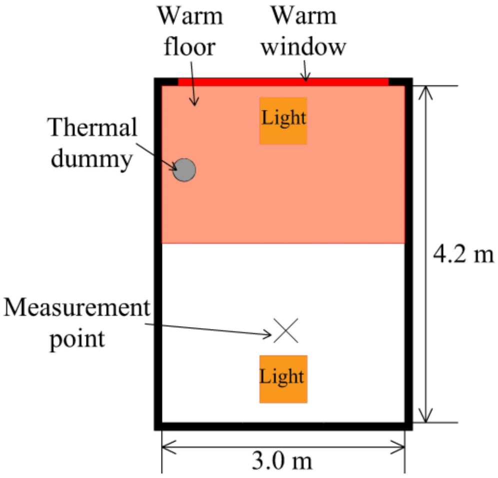

Figure 2.2. *Schematic of the test room.*

A schematic diagram of the room with the ACB and the experimental setup is shown in Figure 2.3. ACBs are integrated convective-based terminal units utilizing water and air for thermal conditioning and ventilating the room. The ACB used in the studies had a dimension of 1.8 m × 0.6 m and was positioned in the suspended ceiling in the middle of the test room.

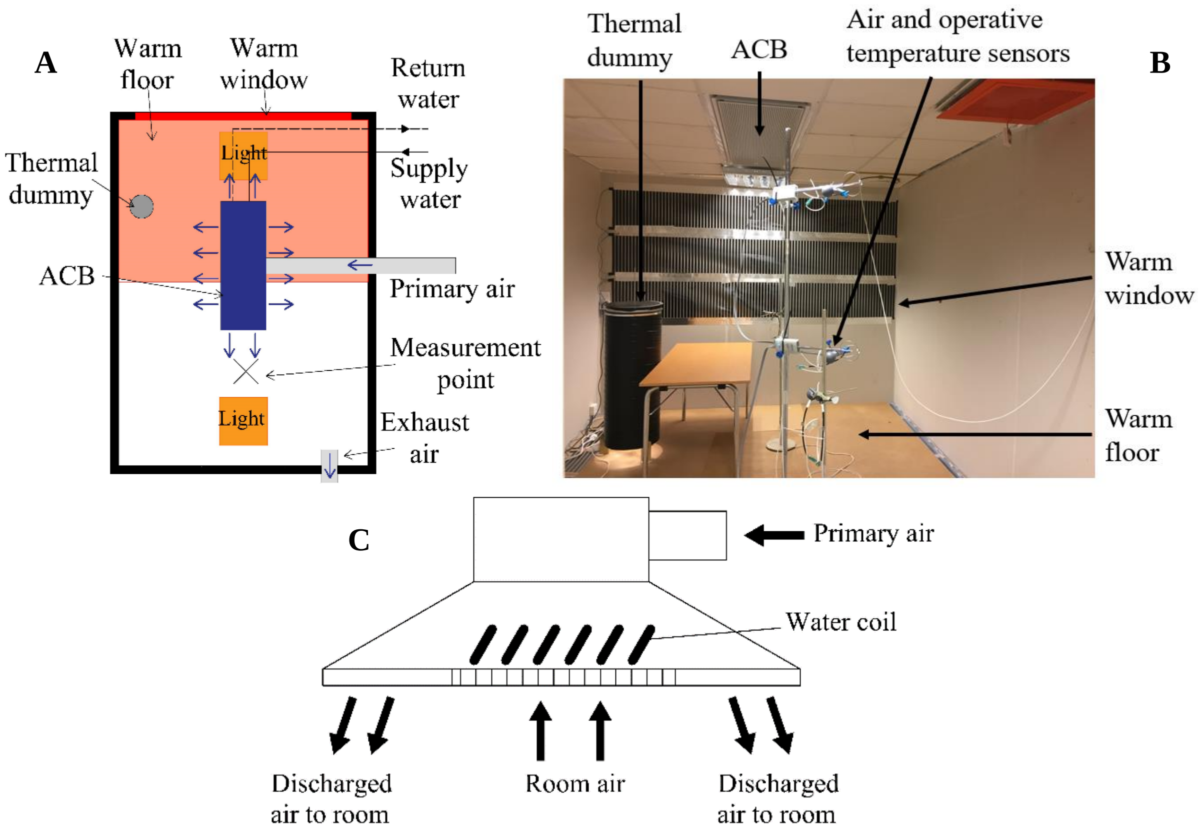

Figure 2.3. *A) Schematic of the test room using the ACB system, B) Photo of the test room, and C) layout of an ACB terminal unit.*

Figure 2.4 shows the experimental setup with ceiling cooling (CC) panels for the parametric study tests. The CC panels were made of copper pipes embedded into a layer of highly thermal conductive graphite with topside insulation [49]. Each panel had a dimension of 0.60 m × 0.60 m, which altogether covered approximately 70% of the ceiling area (see Figure 2.4).

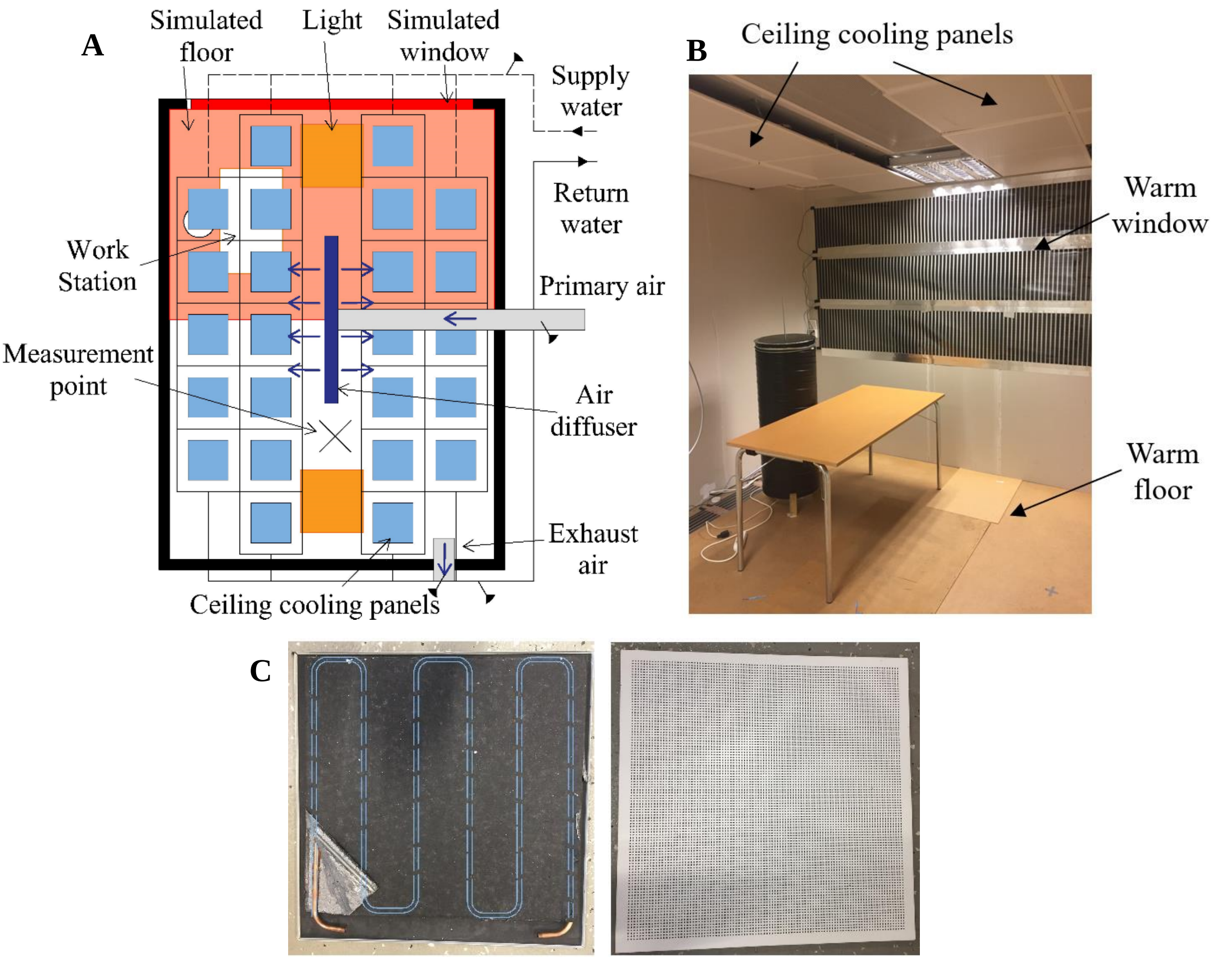

***Figure 2.4**. A) Schematic of the test room using the ground-coupled CC system, B) experimental facilities in the test room, and C) layout of a CC panel.*

The room air and the operative temperatures were measured using probe-type PT-100 sensors. The sensors were located at the measurement point in the test room (see Figure 2.2A). The sensors were installed at the heights of 0.6 m and 1.1 m above the floor, corresponding to the abdomen and head levels of a seated person, respectively. For operative temperature measurements, the measuring section of the PT-100 sensor was placed inside a grey-painted

Ping-Pong ball to account for the effect of the mean radiant temperature, as suggested in [50,51]. Table 2.2 lists the measurements instruments used in the experiments. All sensors were calibrated before the experiments.

Table 2.2. *List of measured variables, sensor types and their accuracies. Sensors' locations are shown in Figure 2.1 and Figure 2.4.*

Variable (unit)	**Accuracy**	**Type**
Room air temperature (°C) Room operative temperature (°C)	± (0.1 + 0.0017 × measured value)	Probe-type PT-100
Supply water temperature (°C) Return water temperature (°C) Primary air temperature (°C) Exhaust air temperature (°C) Intermediate loop supply water temperature (°C) Intermediate loop return water temperature (°C) Ground loop (borehole) outlet temperature (°C) Ground loop (borehole) inlet temperature (°C)	± (0.1 + 0.0017 × measured value)	Screw-in PT-100
Exhaust airflow rate (l/s) Primary airflow rate (l/s)	± 7%	Differential pressure transmitter
Intermediate loop flow rate (l/min)	± 1.5% of the full scale	Vortex-type
Building loop flow rate (l/min)	± 1.2% of the measured value	Ultrasonic
Internal heat gain (W)	± 0.075% of the voltage	Voltage measurement

2.3. Simulation resources

Building energy simulations were performed using IDA indoor climate and energy (IDA-ICE) version 4.8 [52]. This software has been validated against measurements under the framework of various standards, including CIBSE TM33 [53], ANSI/ASHRAE 140 [54] and EN 13791 [55]. In addition, using the IDA ICE borehole extension enables the modelling of various ground-coupled heating and cooling systems [56].

Borehole system simulations are carried out using IDA-ICE borehole extension [56]. The model performs transient heat conduction simulations within and in the boreholes' vicinity to calculate the upward and down-ward fluid temperature along the U-tubes. The calculations are carried out using a combination of the finite difference technique and the superposition method based on the following principles [57]:

- 1D heat transfer calculations for the inlet and outlet fluid in BHE along the U-tube's axial direction. The influence of the fluid flow rate in the BHE is considered by taking the Reynolds number into account. It should be noted that the thermal mass of pipe material is neglected.
- 1D heat transfer calculations between the fluid, filling material and ground along the U-tube' radial direction.
- 2D heat transfer calculations between the BHE wall and the surrounding ground in cylindrical coordinates.

The energy balance equations used to calculate the heat transfer in the fluid and the filling material are available in the Appendix B of Paper V.

The undisturbed ground temperature used in the model was the mean temperature over the active part of the borehole and includes the geothermal gradient effect, as suggested by Eskilson [58]. It is also worth mentioning that IDA ICE borehole model only uses conductive heat transfer equations and could not simulate groundwater-filled boreholes. Therefore, the grouted borehole thermal resistance in the model was set as the thermal resistance of the groundwater-filled borehole under turbulent flow condition (Reynolds number > 2300) based on the method presented by Spitler et al. [59].

3. Current practices and developments for designing DGCSs

DGCSs have the potential for improved energy efficiency, reduced electricity demand and low life-cycle energy costs. Existing projects using DGCSs have demonstrated outstanding performance in these regards [36,38]. However, the use of DGCSs is still not widespread in Sweden, and there are no well-established best practices for the design of these systems. This chapter summarizes some essential notes and barriers in the design and development of the DGCSs. The objective of this chapter is to highlight the main deficiencies in the design process and identify areas where research can be of help to designers' and consultants' needs.

The following notes were obtained during the research project via interviews and group/individual discussions with professionals, manufacturers and research fellows[1].

- **Early design considerations**

  The first and the foremost step in the building design is to minimize the building's thermal energy demand, considering energy efficiency and comfort requirements. In fact, thermal energy demand determines the cooling system type (plant and terminal units) that could be used to fulfil the cooling requirements. The use of some energy-efficient technologies, including DGCSs and HTC systems, is impossible for buildings with high cooling demands. When it comes to DGCSs, special considerations should be taken to avoid intense daily peaks. Decisions on the glazing area and type, the use of shading devices, occupancy layout, etc. can improve or diminish the cooling system's design.

- **Comprehensive design approach**

  A comprehensive design requires a consideration of the cooling system as a whole (cooling plant, control system and building cooling system). However, it is usually the case that each sub-system is designed by separate groups (geo-energy engineers and HVAC engineers) without a consideration of the mutual interactions between the sub-

[1] The interviews were prepared and conducted with Maria Jangsten from Chalmers University of Technology. The author would like to thank the following individuals for their time and interest to join the discussions and interviews (collaborators are listed by alphabetical order): Jose Acuna (KTH, Bengt Dahlgren), Göran Andersson (Gicon), Niklas Andersson (Andersson and Hultmark), Jonas Gräslund (Skanska), Göran Hultmark (Lindab), Jan Nordgren (WSP), Jakob Pontusson (Bengt Dahlgren), Julia Svyrydonova (Sweco), Carl-Ola Danielsson (Swegon), Peter Johansson (WSP), Tomas Utterhal (WSP) and Qian Wang (Uponor, KTH).

systems. Decisions on indoor temperature setpoints, types of control systems, terminal unit supply temperature, terminal unit types, etc. make significant differences in the cooling plant's design and the building's cooling system.

- **Customized design approach for DGCSs**

  The DGCSs' design differs from the GSHPs and chillers in the annual cooling energy, influence of peak loads, and ground energy balance. Unlike chiller-based systems which are sized based on the building's peak cooling loads, the sizing of the ground-coupled systems (GSHPs and DGCSs) is dependent on both peak and monthly building cooling loads. GSHPs in Sweden are mainly sized based on the heating loads and reducing the annual cooling loads may have an adverse impact on the thermal balance of the ground and is not prioritized. On the contrary, an optimized design for a DGCS is achieved by reducing the building's cooling demand. In some cases, it can also be beneficial to give up the heating demand to some extent in order to further reduce the cooling demand, for example, by reducing the windows' G-value. Note that although reducing the peak loads is essential for any cooling technology, the adverse consequences of the peaks are severe in terms of the increased borehole length and drilling costs for the DGCSs.

- **Dynamic simulations of the whole system**

  It is usually the case that the borehole design is considered more or less independently from the system design, though as shown in this thesis, there are advantages of considering them together. Combined simulations can reduce the simulation time, give a broad overview of the system operation and performance, and allow for the design of central proactive/reactive control systems, etc.

- **Use of different types of HTC terminal units**

  Among available cooling terminal units in the global market, only ACBs are commonly used in Sweden. The design and application of other HTC terminal units, such as ceiling cooling panels, embedded cooling systems and TABS are unfamiliar to many designers and practitioners. Using these terminals facilitates utilizing renewable sources, including DGCSs.

- **Indoor temperature setpoints**

  Strict indoor temperature setpoints impose a constraint on the borehole design. Strict temperature setpoints also have an unimportant outcome on the occupants' comfort since the human body is insensitive to small temperature variations over time. An adaptive indoor temperature setpoint plan which allows the room temperature to somewhat increase during the on-peak cooling demand can significantly reduce the peak loads and the required borehole length.

- **High-temperature cooling concept**

  Many designs with ground-coupled cooling systems (GSHPs and DGCSs) do not fully utilise the HTC concept. In most cases, the borehole outlet temperature is designed at a low temperature to provide dehumidification for the air handling units. In addition, the supply temperature of the building cooling system (in Sweden only ACBs) is set at about 14 °C – 16 °C. This design leads to additional borehole depth only to treat the most intense peaks, while the system operates at part-load conditions most of the time. Having a backup source (district cooling connection, chiller, particular GHEs designed for lower outlet temperature, etc.) to deal with the peaks allows for higher supply temperatures and reduces total borehole length.

The above-mentioned factors suggest aligning the design and operation of the building cooling system with the characteristics of the borehole system to maximize the efficiency of the DGCSs. Moreover, special considerations need to be taken when designing the building envelope to minimize the cooling loads. In the subsequent chapters, an effort has been made to integrate these factors into practice in order to improve the design of the DGCSs.

4. Operation strategies and DGCS

Indoor environmental control systems are key components to ensure a comfortable and healthy indoor environment. Control systems use various control methods to keep indoor environmental parameters, such as room temperature and humidity, in an appropriate range by adjusting the input parameters, such as temperature and flow rate of the working fluid. The flow control method is one of the most common methods for adjusting the terminal unit's cooling capacity. In this method, a terminal unit's cooling capacity is adjusted by regulating the water circulation rate. The flow control method can be implemented either by cycling pumps or two-position valves to get an intermittent flow rate, or by modulating valves or variable speed pumps to get a variable flow rate. In practice, intermittent flow operation mode is more common due to the simplified and relatively cheap setups [60]. The intermittent operation mode can be implemented using various controllers, such as two-position switching control, pulse width modulation, and model predictive intermittent control [61–63].

Intermittent flow control methods usually use feedback controllers to maintain the room temperature at the setpoint (desired) temperature. The feedback controller drives a control valve between the "open" state or "close" state, corresponding to "on-flow" or "off-flow" states, respectively. The time duration for the "open" and "close" states depends on the amount of heat that needs to be removed from the space and the terminal unit's cooling capacity.

The cooling capacity of the terminal units is dependent on various factors, including supply water temperature. For a given terminal unit, reduced cooling energy due to the increase of the supply temperature can be compensated by prolonging the "on-flow" duration. The time duration of the "on-flow" state is extended by increasing the supply water temperature. Further increases in the supply temperature make the control system impractical because the control valve is always in the fully-open position.

One operation strategy developed for HTC terminal units involves supplying high-temperature chilled water to the terminal units at a constant temperature (≥ 20 °C) and a constant rate. This operation strategy relies on the "self-regulation" effect of the HTC systems. In this operation strategy, heat transfer between the terminal unit and space intrinsically and instantaneously occurs as a response to changes in the room air temperature. In other words, the heat transfer rate of the terminal unit is naturally adjusted based on the variations in the room temperature. Using this method allows the room temperature to rise and fall somewhat during the on-peak and off-peak cooling load periods, respectively. This operation strategy is also known as “self-regulating”. Various studies showed that ACBs controlled by the self-regulating method could arguably provide a high level of indoor thermal comfort [36,64–66]. Adopting this operation strategy also allows for the removal of the control components (control valves, controllers and thermostats), making the system robust and cost-effective [65,67,68].

The aim of this chapter is to investigate how control and operation methods can influence the sizing of the boreholes and the thermal performance of the cooling system. The following tasks are carried out to reach this aim:

- A development and sensitivity study on an intermittent flow operation method. The experiments were carried out in the test room equipped with a DGCS using CC panels.
- Experimental comparison between the intermittent flow and the continuous flow operation methods for ground-coupled ACBs.
- Model development and validation with the operation methods in IDA ICE for a DGCS using ACBs.
- Simulation studies on an office model using a DGCS with ACBs controlled by the operation methods.

4.1. Parametric study on the intermittent flow control method

This section first outlines the experimental method and procedure to perform a parametric study on a ground-coupled ceiling cooling (CC) panel system. The parametric study focuses on the application of the intermittent flow control method for the CC panel systems. The results show how changing the maximum flow rate and the supply water temperature influence the pump energy use and the CC panels’ thermal output.

4.1.1. Experimental procedure

The experiments were carried out in the test room under a periodic heat gain condition to examine the performance of the CC panels under transient thermal condition. Schematic of the test room with the CC panels is shown in Figure 2.4. Electricity use by the electrical foils was changed intermittently between the low and the high heat gain cycles to simulate a periodic heat gain condition in the test room. The duration of each cycle was 120 min so that the room temperature could significantly change in accordance with the heat gain cycles. This duration was obtained based on the step-response test results to determine the time constant of the test room equipped with a fan-coil unit or CC panels [69,70]. The time constant is the time taken for the room air temperature to reach 63.2% of its final value after a step-change in the room internal gain intensity [71]. Designing the duration cycles in relation to the time constant of the room ensured that the changes in the internal gains had enough time to influence the room thermal mass and hence, the room temperature. The heat load was periodically cycled at 200 W (16 W/m^2) or 700 W (55 W/m^2) during the low and high heat gain cycles, respectively. Each experiment lasted 8 hours.

An on/off feedback controller was adopted to control the cooling capacity of the terminal unit, Figure 4.1. The controller simply drove the pump and the valve relative to the setpoint temperature at either “on” or “off” mode. When the room air temperature was above the setpoint, water was circulated in the building loop and the terminal unit at an “on” flow rate. As soon as the temperature dropped below the setpoint, the controller shut the valve and stopped the pump. The water in the intermediate loop was constantly circulated at its maximum value. The mixing valve in the ground loop had a fixed opening ratio for each experiment to keep the ground outlet temperature constant.

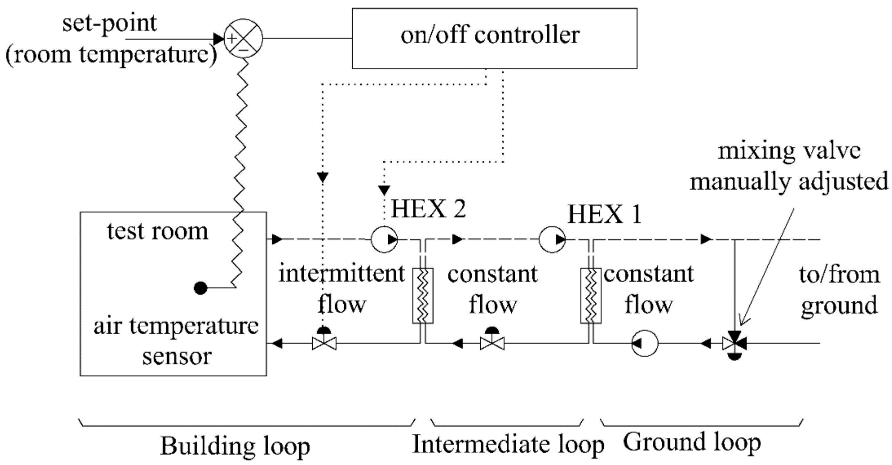

Figure 4.1. *Schematic of the intermittent flow control.*

The first set of experiments was performed to investigate the influence of the maximum water flow rate on the pump energy use and CC panels thermal output. The initial “on” flow rate to the CC panels was set at 1.1 l/min. The system operated under the periodic heat gain condition and the energy use of the pump in the building loop was measured for 8 hours. The experiments were repeated for other "on" flow rates at 2.1, 3.5 and 4.8 l/min. The borehole outlet temperature was kept constant at about 10.7 ℃ for all the flow rates tested. In addition, the flow rate in the intermediate and the ground loops were kept unchanged during this set of experiments.

The second set of experiments aimed to study the association between the supply water temperature, pump energy use and thermal performance of the CC panels. For this experimental setup, the "on" flow rate to the panels was kept constant at 4.8 l/min. Five supply water temperatures of 17.3 ℃, 18.1 ℃, 18.8 ℃, 19.5 ℃ and 20.3 ℃ were tested. The supply water temperature was increased due to the increase in the borehole outlet temperature.

4.1.2. Results

Heat flux delivered by the CC panels can be quantified as follows [72]:

$$q = K.\Delta T_m \tag{1}$$

where q is cooling output of panels (W/m^2), K is heat transmission coefficient (W/m^2.K), and ΔT_m is the cooling medium logarithmic temperature difference (K). ΔT_m can be calculated from the logarithmic difference between the mean water temperature in the panels and the room operative temperature (T_{op}) [72]:

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} = \frac{(T_r - T_{op}) - (T_s - T_{op})}{\ln \frac{T_r - T_{op}}{T_s - T_{op}}} = \frac{T_r - T_s}{\ln \frac{T_r - T_{op}}{T_s - T_{op}}} \tag{2}$$

where T_r and T_s are the return and supply water temperatures (℃) of the CC panels, respectively.

Figure 4.2 shows the "on-time" flow rate of the CC panels plotted versus ΔT_m and the pump energy use in the building loop. The "on-time" flow rate is the maximum flow in the CC panels during the "on" state mode. During the "off" state mode, the flow is zero. The borehole outlet temperature was constant at 11.8 ℃ for all conditions tested.

The pump energy use increases non-linearly with an increase in the maximum flow rate limit in the panels. On the other hand, increasing the maximum flow rate has a small influence on ΔT_m. Interestingly, increasing the flow rate from 3.5 l/min to 4.8 l/min reduces the ΔT_m. This could be due to the higher supply water temperature to the panels because of the faster water circulation in the heat exchanger between the loops, i.e. HEX 2 in Figure 4.1. It can be concluded that increasing the flow rate has a small influence on the panels' thermal output but a significant effect on pump energy use.

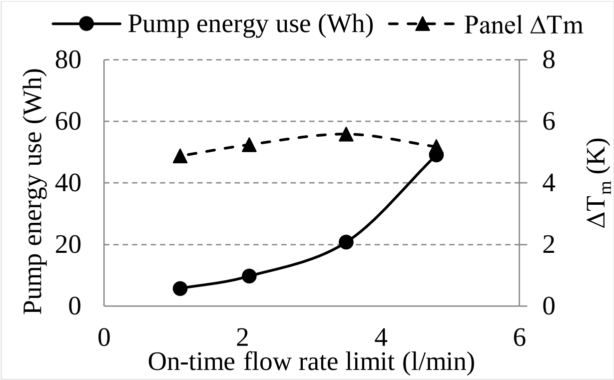

Figure 4.2. *Relation of "on-time" flow rate with on/off controller, ΔT_m of the CC panels and energy use of the pump installed in the building loop. The energy use was measured for a total period of 8 hours.*

Figure 4.3 shows plots ΔT_m and the pump energy use in the building loop against the supply temperature of the CC panels. The "on-time" flow rate in the CC panels was 4.8 l/min. ΔT_m shows a reduction by 28% (from 6.0 to 4.3) since the supply temperature increases by 3 K. Therefore, the pump needs to work for a longer period to remove the same amount of heat as the pump normally would at a low ground loop temperature.

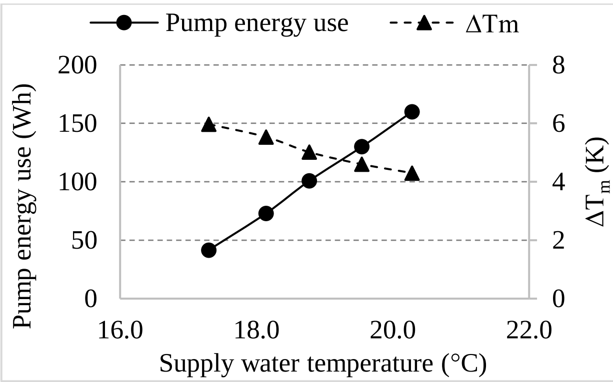

Figure 4.3. *Relation of the supply water temperature to the CC panels, the ΔT_m of the panels, and the pump's energy use in the building loop. Energy use was measured for a total period of 8 hours.*

Analysis on the pump operation period shows that the percentage ratio of the pump operating time to the total test time increases from 16% to 63% by increasing the supply temperature from 17.3 ℃ to 20.3 ℃. Although not performed, it is likely that further increases in the supply temperature make the pump operate continuously. Under this scenario, using the control system is trivial since the control valve is always open and has no influence on controlling the cooling capacity of the terminal unit. Indeed, the cooling capacity would be adjusted based on the self-regulation effect, i.e. it is adjusted naturally and intrinsically based on the room air temperature variations. This feature will be investigated and discussed in the following section.

4.2. Operation strategies and sizing boreholes

This section first presents the experimental methodology for developing the intermittent flow operation strategy and continuous flow operation strategy for ACBs. Then, a simulation model of a DGCS with ACBs for an office building is developed, and the operation strategies are used to control the cooling system. The results outline the potential influence of the operation strategies tested on the sizing of the boreholes.

4.2.1. Experimental methodology and results

ACB cooling capacity was controlled using the intermittent flow operation strategy and the continuous flow operation strategy, one strategy at a time. Experiments were carried out in the test room equipped with an ACB under a periodic heat gain condition (see Figure 2.3). The heat load was 110 W (16 W/m^2) and 440 W (35 W/m^2) during the low and high heat gain cycles, respectively. The duration of each cycle was 120 min. Other experimental conditions are listed in Table 4.1.

The intermittent strategy used a feedback control system to keep the room temperature at the setpoint value of 23 ℃. The feedback control system compared the instantaneous room air temperature and the setpoint temperature. Based on the error between the actual and the setpoint temperature, an on/off controller opened or closed the control valve, as explained in Section 4.1.1.

In the continuous flow strategy, supply water at a constant flow rate and temperature was continuously circulated in the building loop. Since no controller was used for regulating the beam's cooling capacity, the water flow rate and supply temperature needed to be carefully designed to maintain the room temperature within the desired range. The aim was to maintain the room air temperature within the range of 23.0 ± 0.5 °C for the low (110 W) and the high 440 W heat gains. The energy balance calculations were performed to estimate the supply water temperature for the water flow rate at 5.6 l/min. An on-site tuning of the system under the experimental conditions was required to modify the calculated supply temperature.

Table 4.1. *Experimental parameters for operation strategies assessment.*

Operation strategy	**Intermittent flow method**	**Continuous flow method**
Operating strategy principle	On/off supply water flow rate	Self-regulating effect
Primary air temperature (°C)	Same as room temperature	Same as room temperature
Primary airflow rate (l/s) (± SD)	25.2 ± 0.1	25.0 ± 0.1
Supply water flow rate (l/min) (± SD)	0/5.6 ± 0.2	5.6 ± 0.3
Supply water temperature (°C) (± SD)	19.8 ± 0.4	20.8 ± 0.1

When the intermittent operation is applied, the room air temperature represents a saw-tooth shaped pattern with small oscillations (± 0.1 K) around the setpoint temperature at 23.0 °C, Figure 4.4. The oscillations are due to the on/off control of the flow rate in the ACB. Using the continuous flow rate method results in a wave-shaped air temperature pattern in the room. The amplitude of the temperature drift is about 0.7 K. The rise and fall of the temperature is mainly dependent on the heat gain intensity. Since the water flow rate and the supply temperature are constant, increasing the heat gain intensity causes the room temperature to rise, and vice versa.

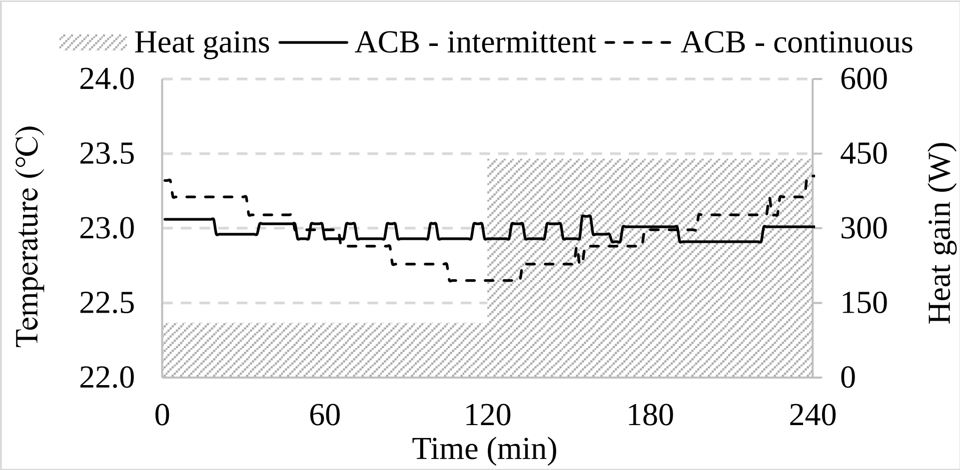

Figure 4.4. *Experimental room air temperature with two operation strategies. Heat gain refers to the consumed electrical power of the heat sources in the room.*

Figure 4.5 shows the heat extraction rates from the test room with the two operation strategies. The heat extraction rate with the continuous strategy varies between 113 W and 240 W, relative to the heat gain. Increasing the heat gain gradually increases the room temperature. Consequently, the ACB cooling capacity increases because the difference between the mean water temperature and the room temperature increases. With the intermittent method, the heat extracted throughout the experiment is controlled by the feedback controller to maintain the room temperature.

Figure 4.5 also shows a discrepancy between heat production by the sources and heat extraction by the ACB. The magnitude varies over time, as the experiment is performed under a transient heat gain condition. However, experiments performed under a steady-state condition reveal heat losses of about 130 - 140 W from the test room. The major part of the heat losses (~ 110 W) is thermal transmission through the test room envelope. A small part (~ 20 W) is due to infiltration losses. An even smaller part (~ 5 - 10 W) is due to ventilation losses.

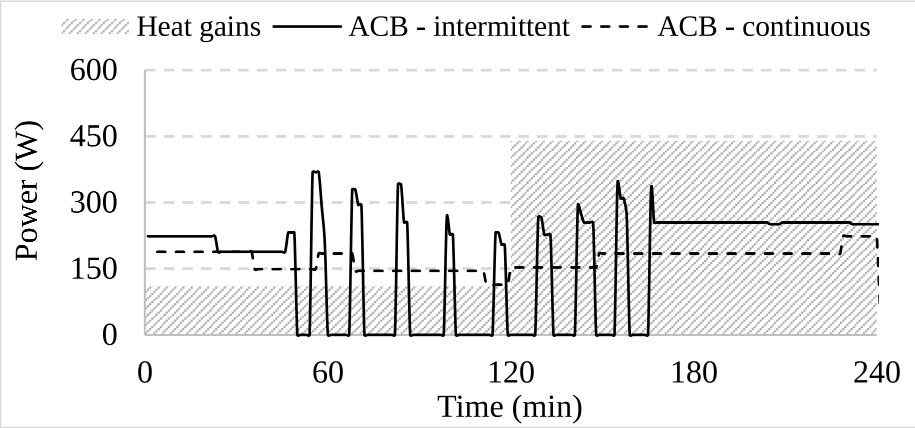

Figure 4.5. *Heat extraction rates by ACB with continuous flow strategy and intermittent flow strategy. The heat extraction rates are calculated based on the difference between the supply and return water temperatures obtained from the experimental data. Heat gain refers to the consumed electrical power of the heat sources in the room.*

An uncertainty analysis is carried out to estimate the impact of multiple systematic errors on the experimental studies' uncertainty. Systematic errors affect the measurements due to the use of faulty instruments, incorrectly-calibrated measurement equipment, etc. The overall uncertainty in the calculated ACB cooling capacity originates in water temperature and water flow measurements uncertainties. The uncertainties in water density and specific heat are negligible [73]. Based on the thermometers' accuracy of ± 0.1°C (Table 2.2), and the temperature difference across the ACB of about 0.7 K, the uncertainty in the temperature measurements can be estimated at ± 20%. The uncertainty in flow measurements is about ± 5%. The overall measurement uncertainty, calculated as the root sum of the squares of the uncertainties, is estimated at ± 20%.

4.2.2. Model development and validation

IDA ICE 4.8 models ACBs as pure convective terminal units. The ACB model includes an air diffuser and a water-to-air heat exchanger to simulate the ventilation and the hydronic parts of an ACB, respectively. The water coil's cooling capacity is calculated based on the difference between the mean water and the room air temperatures. The ventilation cooling capacity is calculated based on the temperature difference between the primary and the exhaust temperatures. The main input data used for the ACB simulation is summarised in Table 4.2. A detailed description of the modelling of the ACBs in IDA-ICE is given in Appendix B of Paper IV.

Table 4.2. *Simulation input data for validating the DGCS for the test-room model.*

Parameter (unit)	
Room	
Floor area (m^2)	12.6
Height (m)	2.4
Floor insulation thickness (m)	0.058
Floor U-value ($W/m^2.K$)	0.83
Wall thickness (m)	0.11
Wall U-value ($W/m^2.K$)	0.33
Suspended ceiling thickness (m)	0.015
Suspended ceiling U-value ($W/m^2.K$)	1.70
Main ceiling thickness (m)	0.27
Main ceiling U-value ($W/m^2.K$)	0.46
Internal heat gain	
Intensity during the low heat gain period (W)	110
Intensity during the high heat gain period (W)	440
Duration of each heat gain period (min)	120
Active chilled beam	
Primary airflow rate (l/s)	25
Primary air temperature (°C)	Room temperature
Exhaust airflow rate (l/s)	25
Supply water flow rate (l/min)	0/5.6 (intermittent strategy) 5.6 (continuous strategy)
Supply water temperature (°C)	19.9 (intermittent strategy) 21.0 (continuous strategy)
ACB cooling capacity control method	On/off water flow rate (intermittent strategy) Self-regulating (continuous strategy)

Figure 4.6 compares the test room and the simulation model. The simulation model consisted of the room model and the cooling system model. Input design parameters of the room model, such as dimensions, envelope material, internal heat gains, design parameters of the ACB, etc., were taken from the experimental study (Section 4.2.1). Other parameters regarding thermal characteristics of the heat sources, e.g. convective heat fraction of the lighting, U-value of the envelope material, etc., were taken from literature and adapted to the model to bring its simulations in line with the experimental observations. Table 4.2 lists the input data for simulating the test room. Details on the model development and validation of the results can be found in Papers II and III.

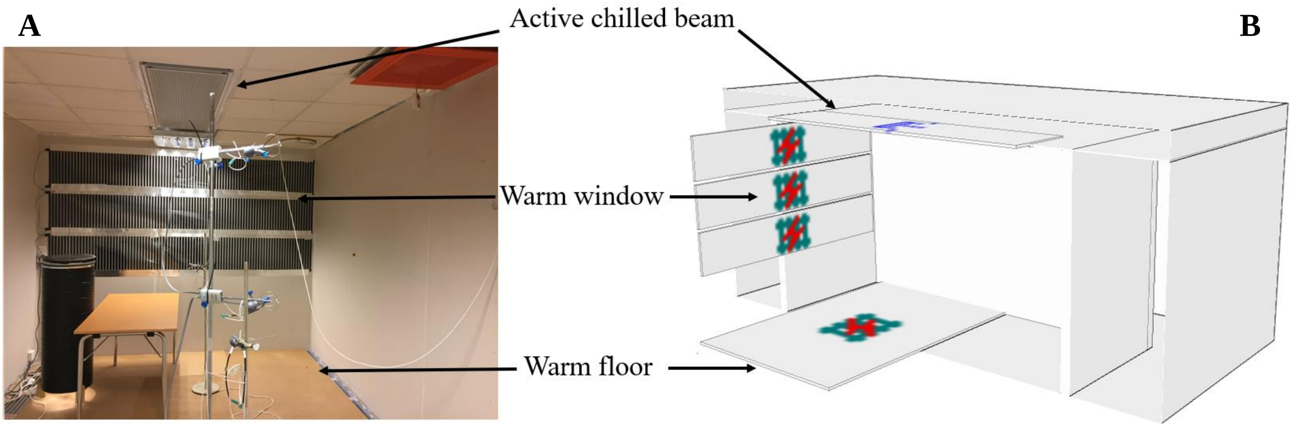

Figure 4.6. *A) experimental facilities and the test room and B) the test room model simulated in IDA ICE.*

Figure 4.7 compares the experimental and simulated room air temperatures with the control strategies tested. Heat gains shown in the figures correspond to the heat generated by the sources. A good agreement can be seen between the experimental and the simulated room air temperatures, with a discrepancy of about ± 0.1 K.

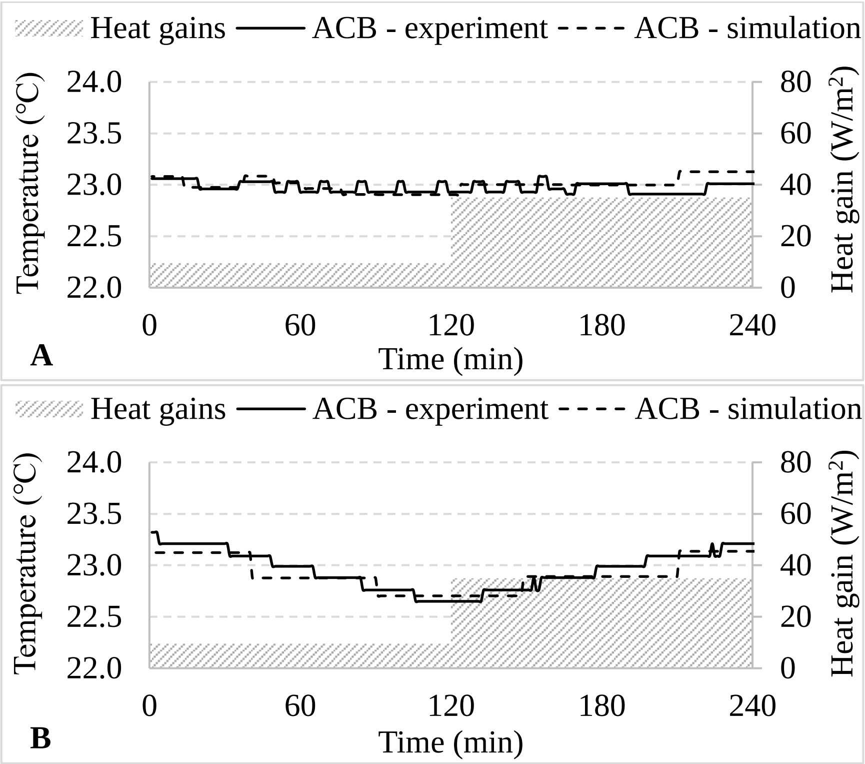

Figure 4.7. *Measurement data and simulation results of the room air temperature with A) the intermittent flow operation strategy and B) the continuous flow operation strategy. Heat gain refers to the consumed electrical power of the heat sources in the room.*

4.2.3. Extended simulation model

The extended model represented an open-plan office and was developed to investigate the long-term thermal performance of the DGCS under more realistic conditions. The extended model was an 88.7 m^2 large office zone with three external walls, see Figure 4.8.

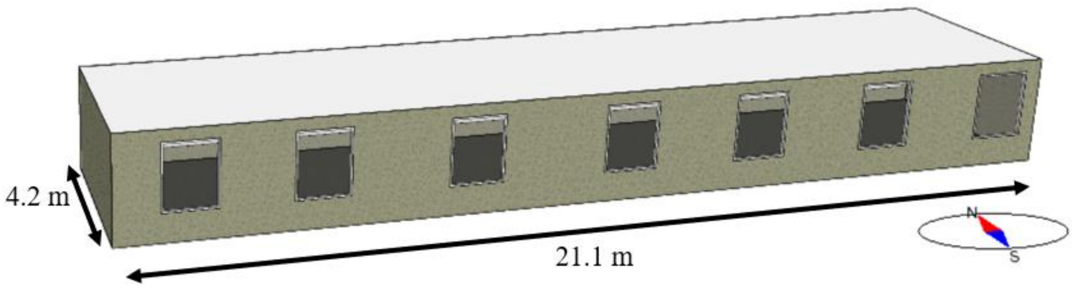

Figure 4.8*. Extended simulation model of an office zone in IDA ICE.*

The total internal heat gain from the occupants, lights and office equipment was 22 W/m^2 and was active on weekdays from 06:00 to 18:00. The model was mainly developed based on the test room model. Thermal characteristics of the lights and the electrical equipment, such as the convective fraction of heat gains for lights and long-wave radiation fraction of the equipment, were kept unchanged from the validated test room model. The simulated office was located in Gothenburg, Sweden, and the simulation period was from 14 May to 22 September. Summary metrics for the input design parameters of the extended model are listed in Table 4.4.

The office was equipped with direct ground-coupled ACBs. Table 4.3 summarizes the main operational parameters of the ACB system for the extended office model. With the intermittent control strategy, the cooling capacity of the ACBs was controlled by the on/off supply water flow controller to keep the room temperature constant at 23.0 °C. The cooling system only operated from 06:00 and 19:00.

The continuous strategy operated the system for 24 hours on weekdays. The operational parameters were different during the operating (6:00 - 19:00) and non-operating (19:00 - 06:00) hours. This control strategy allowed the room temperature to vary within the thermal comfort range and provided continuous cooling for the office zone.

Table 4.3. *Operational and design metrics for the extended office model cooling system.*

Operation strategy		Intermittent	Continuous
System specifications from 06:00 to 19:00	Primary airflow rate (l/s.m^2)	2	2
	Supply water temperature (°C)	16	19.5
	Total water flow rate (l/s)	0 / 0.5	0.5
	Pump operation mode	on	on
	Fan operation mode	on	on
System specifications from 19:00 to 06:00	Primary airflow rate (l/s.m^2)	0	0
	Supply water temperature (°C)	-	16
	Circulating water flow rate (l/s)	0	0.5
	Pump operation mode	off	on
	Fan operation mode	off	off

The preliminary DGCS design used a 200 m deep single U-tube BHE. The borehole depth was calculated to keep the maximum outlet temperature below 12.0 °C. The building's cooling loads used for sizing the borehole were simulated for the ACBs controlled by the intermittent operation strategy to maintain the room temperature at 23.0 °C. The DGCS was a thermally-balanced system where the annual ground heat rejection and extraction loads were approximately equal. The input design parameters of the DGCS are listed in Table 4.4.

Table 4.4. *Input design parameters for the extended model of the office zone.*

External walls	
Dimensions (m)	21.1× 2.4 (W × H), 4.2 × 2.4 (W × H)
U-value ($W/m^2.K$)	0.33
Thickness (m)	0.27
Internal walls	
Dimensions (m)	21.1 × 2.4 (W × H)
U-value ($W/m^2.K$)	0.54
Thickness (m)	0.11
Windows	
Number of windows	7
Dimensions (m)	1.5 × 1.2 (H × W)
U-value ($W/m^2.K$)	1.19
G-value (-)	0.43
Ground slab	
Dimensions (m)	21.1 × 4.2 (L × W)
Thickness (m)	0.46
U-value ($W/m^2.K$)	0.14
Borehole	
Depth (m)	200
Fluid mass flow rate (kg/s)	0.5
Other specifications	see Table 2.1
Active chilled beams	
Number of ACBs	7
Primary water flow rate ($l/s.m^2$)	2.0
Supply water temperature (°C)	See Table 4.3

4.2.4. Simulation results

Figure 4.9 compares the heat rejection loads to the ground for the extended model with the intermittent and the continuous strategy. The ground loads with the intermittent strategy are zero during the unoccupied hours and weekends, as the system is switched off. However, the ground loads never reach zero with the continuous strategy except on weekends.

It can be seen that both the peak daily and the average monthly ground heat rejection loads are smaller with the continuous strategy. Room temperature increase during the on-peak cooling load period provides a considerable potential for the reduction of hourly peaks. Pre-cooling the space during the off-peak cooling load period, i.e. unoccupied and part-load periods, also helps to reduce hourly peaks. However, pre-cooling has a minor influence, since the office is made of light-weight construction materials with low thermal mass.

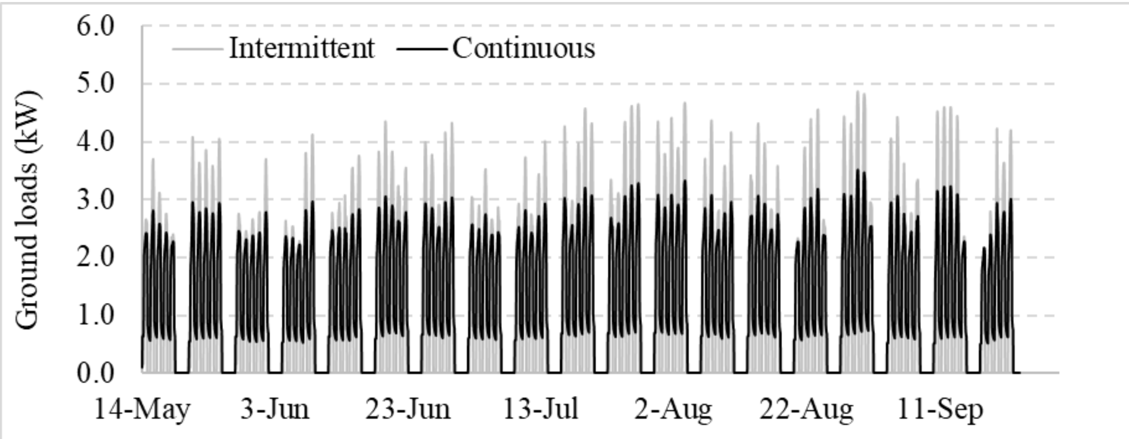

Figure 4.9. *Ground loads for the extended office model with the intermittent and continuous operation strategies. The ground loads are calculated based on the borehole's fluid flow rate and the temperature difference between the borehole inlet and outlet fluid.*

Figure 4.10 shows the outlet temperature of the existing borehole with the control strategies that were tested. Each central box of the boxplot presents the interquartile range, with a horizontal line at the median and the lower and the upper quartiles representing the 25th and 75th quartiles at the bottom and the top of each box. The whiskers define extra quartile values, and the dot symbols represent the outliers, if any. The results present the temperatures for the whole cooling season, including occupied and non-occupied periods and weekends.

The reference borehole was 200 m deep and was primarily sized based on the ground loads by the intermittent control strategy to keep the maximum borehole outlet temperature below 12 °C. Applying the continuous strategy reduces the peak daily and the average monthly ground heat rejection loads. Consequently, the maximum outlet fluid temperature lowers by about 0.8 K.

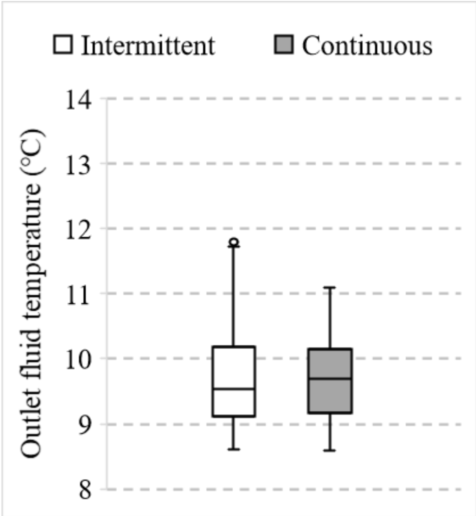

Figure 4.10. *Borehole outlet fluid temperature during the cooling period between 14 May 2018 and 22 September 2018 for the extended office model. The undisturbed ground temperature considered is 8.3 °C.*

As previously mentioned, the borehole depth was primarily calculated based on the building's cooling loads with the ACBs being controlled by the intermittent strategy. However, results in Figure 4.9 show that the ground loads are different when the continuous strategy is used to operate the cooling system.

Figure 4.11 shows the required borehole length calculated based on the actual ground loads shown in Figure 4.9. The sizing is performed to keep the maximum temperature of the fluid leaving the borehole below 12 °C. Applying the continuous strategy reduces the heat rejection loads to the ground for which a shorter borehole can be used. The required borehole length for the cooling system using the continuous operation strategy is about 18% smaller.

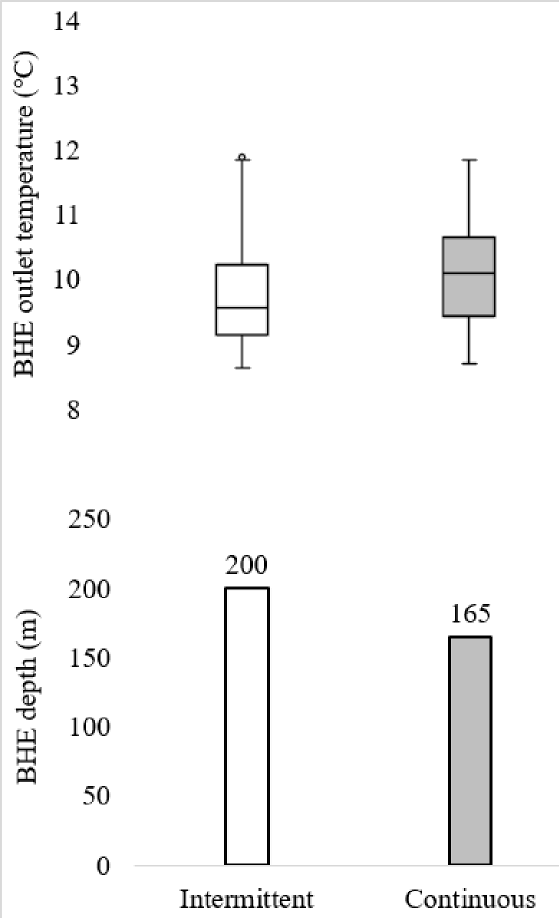

Figure 4.11. *Modified sizes of the BHE calculated based on the actual ground heat rejection loads to obtain similar maximum outlet fluid temperature. The undisturbed ground temperature is 8.3 °C.*

Using the continuous strategy has consequences for pump energy use. Compared to the intermittent strategy, the continuous strategy runs the circulation pump for a longer period. For the cooling period between 14 May to 22 September, the pump energy use for the intermittent and the continuous operating strategies are determined to be 0.63 kWh/m^2 to 1.0 kWh/m^2, respectively. Therefore, it can be concluded that using the continuous method reduces the investment costs but increases the running costs of the system. Given the low electricity tariffs in Sweden, it can be anticipated that reducing the borehole length achieved by applying the continuous method would have a more significant pay-off in the boreholes' drilling cost compared to the financial benefit of saving this much electricity.

4.3. Discussion

Supply water temperature has a key role in sizing and designing the cooling systems. In this chapter, very high-temperature supply water is utilized to establish the self-regulation effect for the ACBs. Incorporating the self-regulation effect in the control system makes the reduction of peak loads possible for which shorter borehole can be designed. On the other hand, the pump in the building loop must run for much longer hours and larger ACBs are needed to offset the impact of the increased supply temperature.

In this chapter, the boreholes are sized to reach the borehole outlet temperature as low as 12 ℃. The supply temperature to the ACBs in the intermittent case is 16 ℃ and it is obtained using a mixing valve. However, it is possible to increase the supply temperature to reach a higher borehole outlet temperature. In fact, increased supply temperature allows for an increasing of the borehole outlet temperature, resulting in shorter boreholes. In this case, special design considerations should be taken, as the borehole heat transfer rate per unit length (W/m) increases. In particular, the designs of the boreholes with higher heat transfer rates are sensitive to the design conditions that might not be fully understood, e.g. heat waves, compared to the boreholes with lower high transfer rates. This point is further investigated in Section 6.1.4.

In a part of this chapter, CC panels are used to develop the control principles for the direct ground-coupled HTC terminal units. Note that CC panels have a very short response time [74,75] and represent similar dynamic thermal behaviour as convective-based terminal units [69]. Therefore, the control methods developed for the CC panels can be used for the ACBs by making some custom-tuning modifications.

4.4. Conclusions

In the first part of this chapter, an experimental parametric study is performed on a ground-coupled CC panel system using an on/off feedback controller. The results show that increasing the “on” state flow rate has an insignificant influence on the CC panels’ ΔT_m but it causes the pump energy use to increase significantly. For a given flow rate, increasing the supply temperature causes the panels’ ΔT_m to decrease, and the pump operation time and the pump energy use to increase. Further increases in the supply temperature increases the proportion of the “on” flow time to the “off” flow time until the control valve always stays open. Under this scenario, the cooling capacity is regulated based on the self-regulation effect, i.e. it varies naturally and intrinsically by variations of the room air temperature.

The second part of this chapter studies the influence of the building cooling system's operation strategies on the sizing of the boreholes. Two operation strategies, continuous flow strategy and intermittent flow strategy, are experimentally examined for a ground-coupled ACB system. The experimental results are used to develop an extended simulation model of an office zone to investigate the cooling system's seasonal thermal performance. The continuous operation strategy provides pre-cooling by extending the cooling period and allowing room temperature to somewhat increase during the on-peak cooling load period. Both factors reduce the ground hourly/daily peak loads, resulting in a significant reduction in the required borehole length by about 18%. Figure 4.12 is a visual summary of the main findings presented in this chapter.

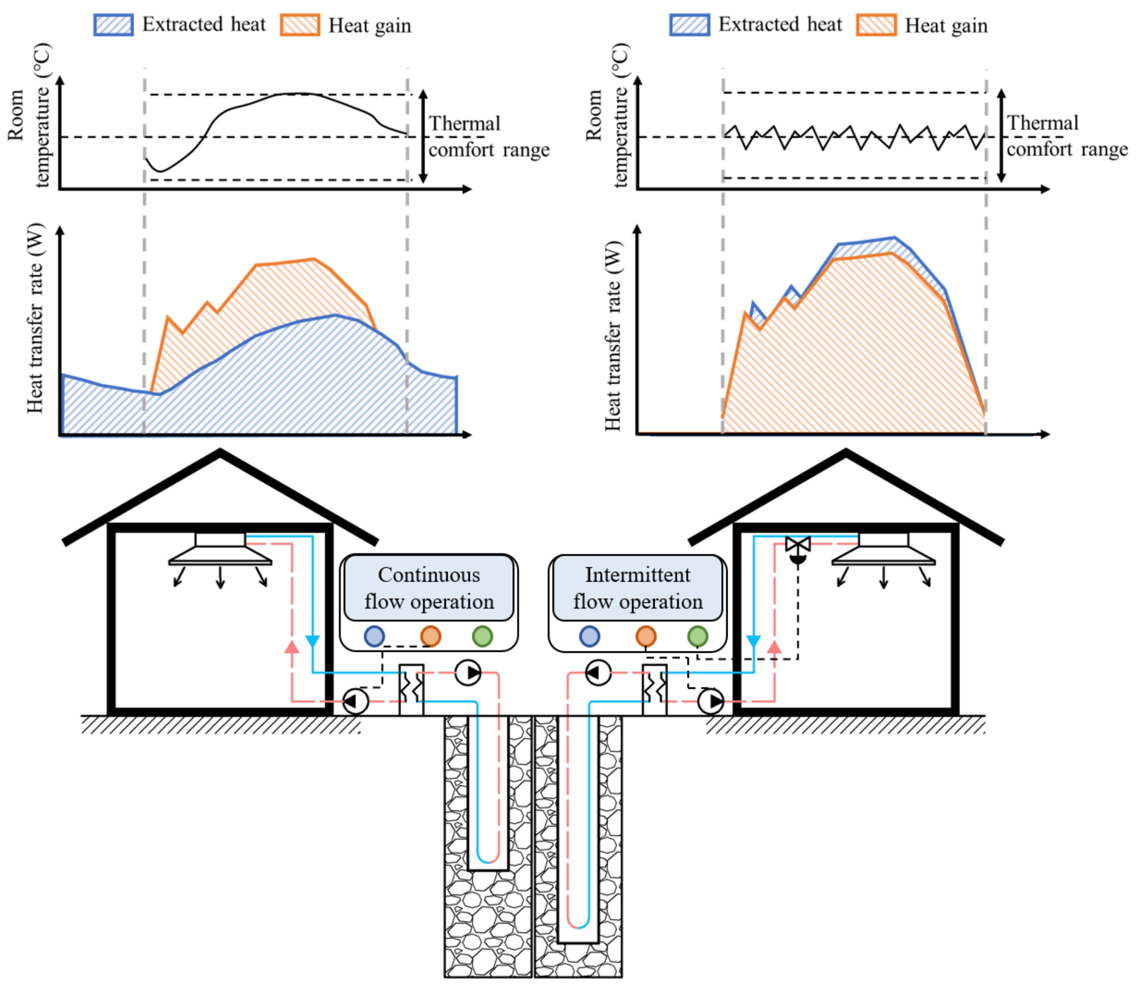

Figure 4.12. *A visual summary of the key findings presented in this chapter.*

5. Terminal units type and DGCS

The fluid temperature levels leaving the borehole system are of interest for designing the DGCSs. The increase or decrease in the borehole outlet temperature is relative to the peak daily and the annual ground heat rejection loads [76–78]. The annual loads gradually change the ground temperature over the years if the ground heat rejection and extraction loads are not balanced [79]. The daily loads usually cause a larger, but short-term, increase in the borehole outlet temperature. BHEs cannot tolerate intense peak loads due to the relatively slow thermal response of the nearby ground. Thus, reducing hourly/daily peak loads can strongly impact borehole temperature levels and sizes.

One way of reducing peak intensities is to use thermally active building systems (TABS). TABS have a high thermal mass and can absorb a substantial amount of heat during the on-peak cooling load period and gradually release it during the off-peak period [41]. The peak shaving effect of TABS benefits sizing the ground-coupled systems, because coping with intense hourly peaks requires additional BHEs.

It is well-understood that TABS and embedded terminal units have different heat extraction rates than ACBs due to the differences in the heat transfer method (radiation vs convection) [80–84] and/or their response time [40,41,75,85]. Thus, it is hypothesized that boreholes can be sized differently based on the terminal units' thermal characteristics. The key question addressed in this section is how using TABS and ACBs can influence the sizing of the BHEs.

In order to justify the hypothesis, the terminal units with their own control systems are applied to an identical office building for a cooling period from 14 May 2018 to 22 September. The case studies firstly use an identical BHE to investigate their thermal performance. Then, the BHE is sized for each case study to reach a prescribed borehole outlet temperature based on their actual cooling loads.

5.1. Simulation model

Building energy simulations and borehole heat transfer modelling presented in this chapter were carried out using IDA ICE software. The simulation model in this chapter uses the same office zone, as previously described in Section 4.2.3. However, the model is partly modified to meet TABS design requirements. The simulation model is parameterized and explained below.

5.1.1. Office zone

The simulated office zone is a south-facing open-plan office with a floor area of 88.7 m^2, see Figure 4.8. The office is on the first floor, has three external walls and windows only mounted on the southern wall. The total internal heat gain is 18 W/m^2 and is active on weekdays from 08:00-17:00. The simulated office is located in Gothenburg, Sweden, and the simulation period lasts from 14 May 2018 to 22 September 2018. Design specifications of the office zone are summarized in Table 4.4.

5.1.2. Cooling system

The DGCS utilized a single U-tube BHE to provide high-temperature chilled water for the building cooling system. The borehole had an active depth of 200 m and a diameter of 110 mm. The borehole was sized to maintain the maximum outlet fluid temperature at 11.0 ℃ based on the building's cooling loads. The building's cooling loads were simulated with an ACB system operated by an on/off control method to keep the room temperature at 23.0 ℃. The heat rejection loads to the ground during the cooling period were offset by the heat extraction loads from the ground to pre-heat the ventilation air in the heating period. Therefore, the under-study DGCS was a thermally-balanced system. Details about the borehole system are listed in Table 4.4.

The terminal units tested in the study were ACB and TABS. Seven ACBs, each having a nominal cooling power of 810 W, were installed for thermal conditioning the building. Two operating strategies were tested for the ACB system. The first strategy, denoted as ACB-A throughout this chapter, aims at keeping the indoor operative temperature at 23.0 °C during working hours, weekdays from 08:00 to 17:00. The cooling capacity of the ACBs was regulated using an on/off controller. Model development of the ACBs is described in Section 4.2.2. Design parameters of the ACB are summarized in Table 5.1.

In the second control strategy, referred to as ACB-B in this chapter, supply water to the beams was circulated at a constant rate and temperature. In this method, the beams' cooling capacity instantaneously and intrinsically varied with the room temperature based on the self-regulation effect. Principles of this control strategy were presented in detail in section 4.2. The operation period of the beams with the ACB-B strategy was also from 08:00 to 17:00 on workdays. The supply water temperature was approximately constant at 19.5 °C. The supply water temperature was obtained by iteratively adjusting the water temperature to reach the prescribed room operative temperature and thermal comfort range. Other design parameters for this control strategy are listed in Table 5.1.

Table 5.1. *Description of the input design parameters of the ACBs and TABS.*

Parameter (unit)	ACB	TABS
Operation time period (-)	08:00 - 17:00	00:00 - 24:00
Primary airflow rate (l/s.m^2)	2.0	1.2
Exhaust airflow rate (l/s.m^2)	2.0	1.2
Primary air temperature (°C)	Room temperature	Room temperature
Water flow rate range (l/min)	0 - 39.2	0 - 22.0
Supply water temperature (°C)	16.0 (ACB-A*) 19.5 (ACB-B*)	21.5
Cooling capacity control method	On/off water flow (ACB-A*) Self-regulating effect (ACB-B*)	Variable flow + operating schedule

* ACB-A and ACB-B are operating strategies for the ACB system.

The second terminal unit investigated in this study was TABS. Figure 5.1 is a schematic diagram of the TABS serving as a radiant floor cooling system in the simulation model. The TABS model in IDA ICE consists of an active part and a passive part. The active part includes the piping layer and was a proactive section of TABS determining the amount of heat removed from the building by the fluid in the pipes. The passive parts are the layers below and above the piping layer and behave like passive thermal masses. The passive parts absorb heat during the daytime and gradually reject it to the ground during night-time. A detailed description of TABS modelling in IDA ICE is provided in Appendix A of Paper IV.

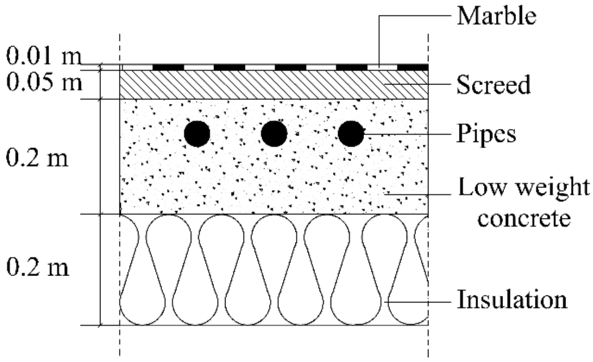

Figure 5.1. *Schematic layout of the TABS serving as a floor cooling system.*

The operation strategy for the TABS was an integrated method based on controlling the TABS surface temperature and an operating schedule. This operation strategy was among the common methods for controlling the TABS and was previously investigated in other studies [86–88]. The operating schedule was 24 h for the workdays. The schedule also considered pre-cooling the building from 15:00 on Sundays to reduce the accumulated heat in the space during the weekends. The supply flow rate was modulated to maintain the TABS surface temperature within the range of 22.6 ± 0.5 ℃. Water at the constant temperature of 21.5 ℃ was supplied to the TABS, and the cooling capacity was adjusted by altering the flow rate between 0.0 and 22.0 l/min (0 – 0.26 l/min.m^2). The targeted surface temperature provided the required cooling capacity for thermal conditioning the building while minimizing the overcooling risks. The design parameters of TABS are listed in Table 5.1.

5.2. Design and evaluation criteria

The design considerations used to develop the cooling system model are as follows:

- Total heat rejection loads to the ground during the cooling season are similar for all cooling systems, this is performed by adjusting the operation period and the cooling capacity of the terminals. Note that daily heat rejection loads are different since the cooling systems have different dynamic thermal behaviour.

- The DGCS is a thermally-balanced system where the annual ground heat rejection and extraction loads are equal. Therefore, the average annual ground temperature does not significantly change and thus, is not accounted for in the system design.

- The borehole is sized to get the borehole outlet fluid temperature below 11 ℃. The outlet temperature is calculated based on the required cooling capacity of the ACB-A system.

The thermal environment generated by the ACBs and TABS represent different features, and comparing them requires considering different thermal comfort criteria. The indoor temperature in the spaces handled by TABS varies somewhat during the occupied period, due to the large time lag for utilizing high thermal mass [89]. Thus, designing TABS aims to maintain thermal comfort within the acceptable range, instead of keeping the room temperature constant [90,91]. The evaluation criteria for the thermal environment with TABS is to keep the room operative temperature below 26 ℃, equivalent to a predicted mean vote (PMV) < + 0.5 on the thermal sensation scale and a predicted percentage dissatisfied (PPD) < 10%, to fulfil category B for office buildings, in accordance with ISO 7730 [92]. The room operative temperature with ACB is kept constant at 23 ℃. Differences in the thermal environment with the cooling systems are further discussed in Section 5.4.

5.3. Results

Simulations were carried out for daily and seasonal time frames. The daily simulations sought to investigate the room temperature and the ground heat rejection loads on an hourly basis on the peak day. The seasonal simulations aimed to investigate the borehole system's long-term performance and the borehole outlet temperature levels.

5.3.1. Peak day

Figure 5.2 shows the energy balance and operative temperature of the office zone served by the ACB system. The hourly simulations are performed on the design day to show the cooling system's thermal performance under peak condition. The total cooling load represents the amount of heat that must be removed from the building to maintain a constant temperature [93].

The building's maximum cooling load is about 5.1 kW and is reached at around 13:00. The ACB-A strategy manages to remove 4.7 kW of the cooling load and the remaining causes the room operative temperature to slightly increase (< 0.5 ℃) from the setpoint temperature (23 ℃), Figure 5.2A. The maximum heat extraction occurs approximately 1 hour after the peak cooling load.

The heat extraction rate of the beams operated by the ACB-B strategy is shown in Figure 5.2B. The fact that the beams' cooling capacity with this strategy varies with the room temperature can be seen in the figure. The heat extraction rate gradually increases as a response to the increase in the room temperature. Compared to the ACB-A strategy, the beams' heat extraction rate with the ACB-B strategy is clearly lower, and consequently, the room operative temperature is higher. Heat extraction rate peaks at 3.4 kW, which is 28% lower than that with the ACB-A strategy. The system is switched off at 17:00 and the excess heat causes the room temperature to increase. However, the room temperature gradually deceases overnight since the excess heat is transferred via the building envelope.

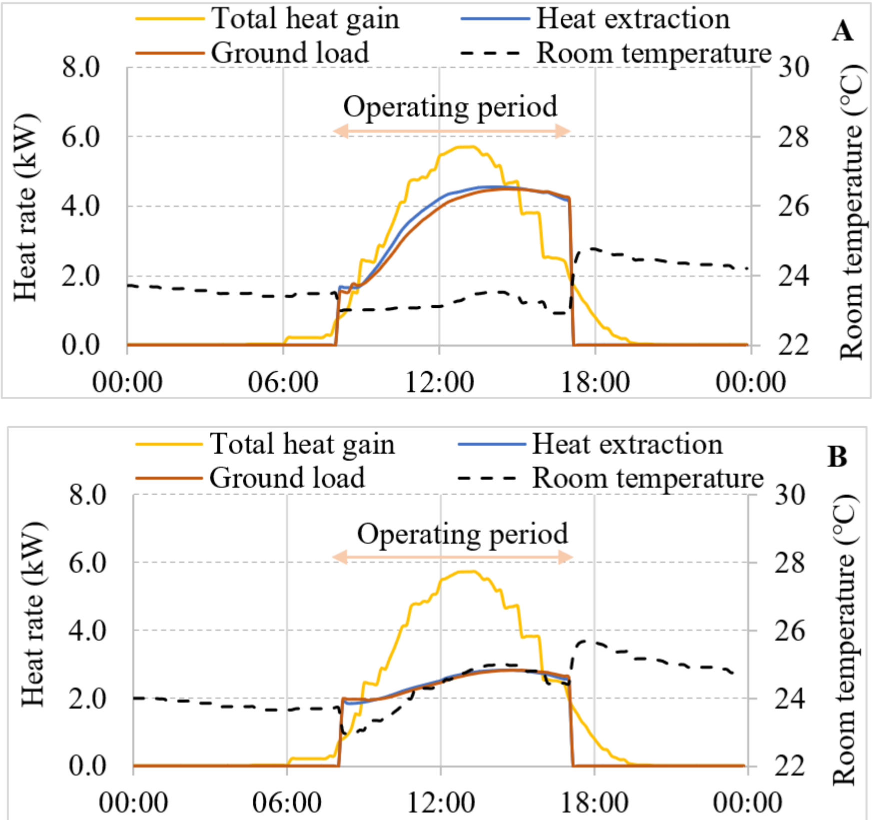

Figure 5.2. *Simulated heat flow rates and room operative temperatures of the office cooled by active chilled beams operated with A) ACB-A strategy and B) ACB-B strategy.*

Figure 5.3 shows the heat extraction rate and the room operative temperature with TABS. Due to the night cooling, the room temperature is 22.6 ℃ before occupancy begins and is comparatively lower than that with the ACB systems in Figure 5.2. However, the room temperature increases rapidly due to the low cooling capacity of the TABS. The figure shows a large difference between the cooling load and the heat extracted by the TABS. The maximum heat extraction is approximately 1.6 kW and takes place almost 3 hours after the peak in the building heat gain. Consequently, the room temperature rises to 26.1 °C.

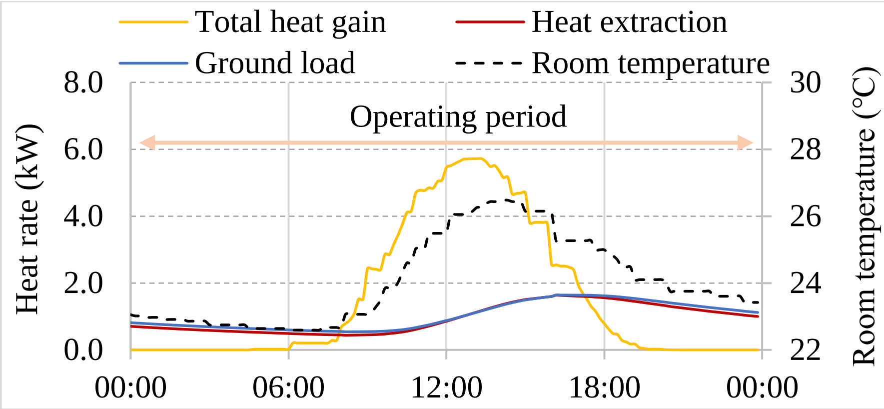

Figure 5.3. *Simulated heat flow rate and room operative temperature of the office cooled by TABS.*

5.3.2. Cooling period

Figure 5.4 compares the distribution of the ground loads for the ACB systems and TABS. The ground loads are calculated based on the temperature difference between the borehole inlet and outlet fluid temperatures and the fluid flow rate during their operation period. The operation period for both ACB strategies is between 08:00 – 17:00 and for TABS is 24 h. Note that the ground loads can be zero with the ACB-A strategy since the beam's cooling capacity is controlled intermittently. The maximum loads are significantly different for the ACBs and TABS, with the largest difference (~ 65%) between the ACB-A and TABS. Reduction in the ground loads is also apparent in the part loads, because the central box, containing 25% - 75% of the data, shows much smaller ground loads for TABS.

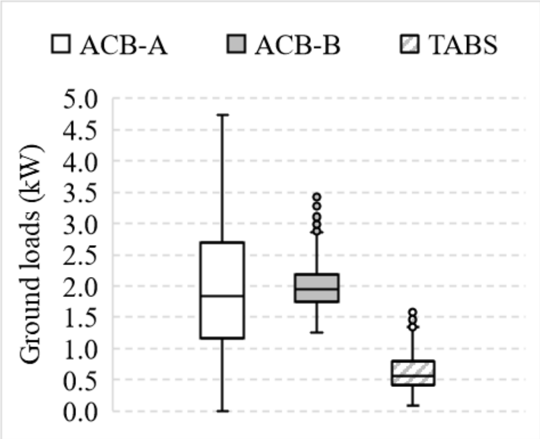

Figure 5.4. *Ground loads distribution for the cooling systems during the operation time for the simulation period between May 14th and September 22nd. The operation time for the ACB system is from 08:00 to 17:00 and for the TABS is 24 h.*

Figure 5.5 shows the fluid temperature distribution leaving the borehole over the operation period of 4.5 months. The borehole is primarily sized for the ACB-A strategy to get the maximum borehole outlet temperature of 11 °C. The undisturbed ground temperature is 8.3 °C. As expected, using the ACB-A strategy results in the highest outlet temperature, due to the highest heat rejection load to the ground. Applying the ACB-B strategy smooths out the hourly peak heat rejection loads and hence, the maximum outlet temperature. The outlet temperature is further reduced to 9.6 °C when TABS are used. Although TABS continuously reject heat to the ground, a reduction in the peak loads can effectively reduce the borehole outlet temperature.

Results shown in Figure 5.5 suggest that shorter boreholes can be designed for the ACB-B and TABS cases based on the actual heat rejection loads to get the same outlet temperature as delivered by ACB-A.

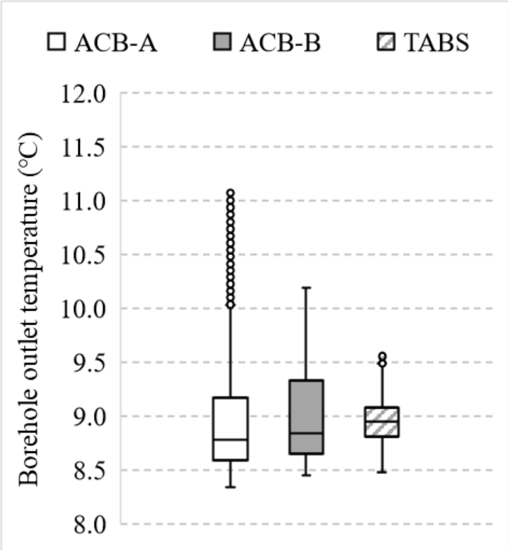

Figure 5.5. *Borehole outlet fluid temperature distribution for the simulation period between May 14th and September 22nd. The undisturbed ground temperature is 8.3 °C.*

So far, the results presented compare the thermal performance of the two cooling systems with different terminal units when using a borehole with the same length. Figure 3.6 shows the required borehole lengths sized based on the actual ground loads for the ACB-B strategy and TABS. The borehole sizing criterion is maintaining the maximum borehole outlet temperature at about 11 °C for the ground loads shown in Figure 5.4. For each case, the borehole depth is iteratively adjusted until the targeted outlet temperature is achieved. Moreover, room operative temperature is checked to fulfil the required thermal comfort levels specified in section 5.2.

From the operating strategy perspective, the ACB-B operating strategy effectively reduces the daily peaks by utilizing the self-regulating effect. The required borehole length is 30% shorter. From the terminal unit type perspective, using TABS instead of the ACBs can reduce the required borehole length by approximately 53% (from 200 m to 95 m). TABS gradually reject heat to the ground and can smooth out the peaks. Both longer heat rejection time and low-intensity peaks are favourable for the ground system since boreholes also have a slow thermal response.

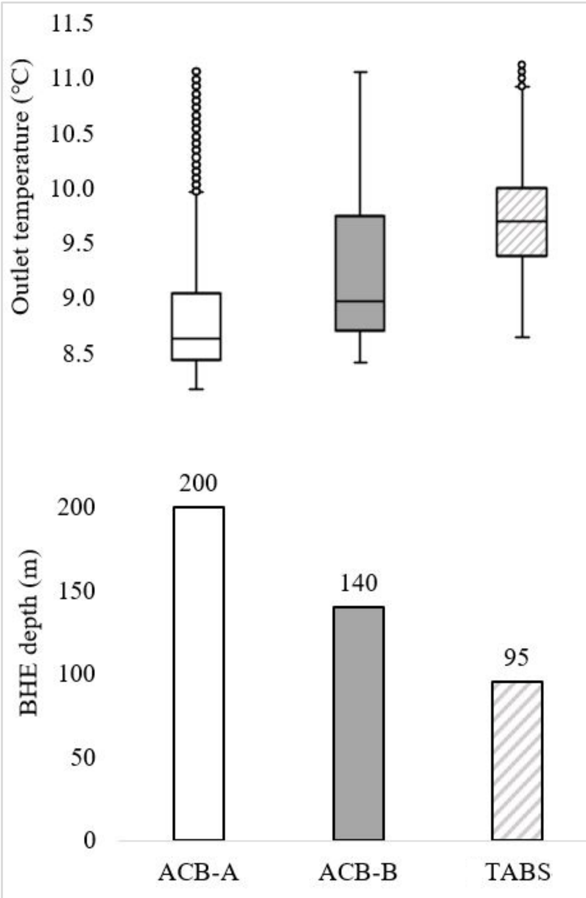

Figure 5.6. *Modified sizes of the BHE based on the actual heat rejection rate from each terminal unit to the ground to obtain similar maximum outlet fluid temperature. The undisturbed ground temperature is 8.3 ℃, and total heat rejection into the ground during the simulation period is similar for all cases.*

TABS have a longer response time than the ACBs because a large amount of thermal mass is involved in the heat transfer process. Thus, TABS have a slow heat extraction rate from the space for which the room temperature can increase during the peak hours. Figure 5.7 compares the duration curves of the room operative temperature with TABS and the ACBs. The room temperature profile with TABS is expanded over a range between 21.5 ℃ to 26.1 ℃. On the other hand, the room temperature with ACB hardly deviates from the setpoint temperature of 23 ℃. Based on the thermal comfort evaluation criteria described in Section 5.2, ACBs and TABS fulfil the thermal comfort categories A and B, respectively. Nevertheless, if maintaining the room temperature is important, a trade-off between stable room temperature and shorter BHE should be made when designing the cooling system.

Results shown in this figure are practical for designing indoor temperature levels. For instance, the Swedish building owner association (BELOK) suggests room temperature levels between 21 ℃ and 26 ℃ and allows overheating in office buildings for up to 80 h [94]. The maximum design temperature can therefore be 26.1 ℃ considering the permitted overheating hours with the TABS. Room temperature increases up to 26.1 ℃ for 80 h during the cooling period, which is acceptable.

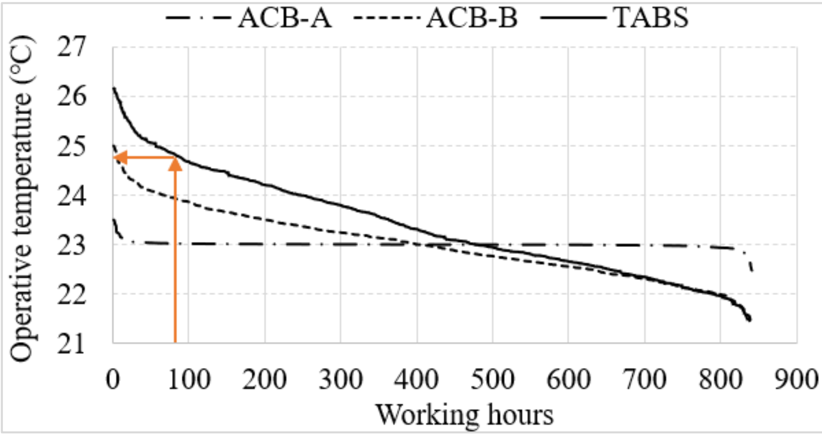

Figure 5.7. *Duration curves of operative room temperature for the working hours from 14 May to 22 September.*

Another feature regarding the comparison between the terminal units and the operating strategies is the pump electricity demand. The total pump electricity demand includes the electricity used by the borehole pump and building cooling system pump. For the cooling period simulated, the pump energy use for the ACB-A and ACB-B operating strategies are 0.21 kWh/m^2 and 0.33 kWh/m^2, respectively. The pump electricity use with the ACB-A is lower because the pumps are operated intermittently. Furthermore, the beams operated by the ACB-A require lower supply temperature, resulting in the reduced water flow rate. The pump energy use for TABS is 0.76 kWh/m^2, as the cooling system requires to work continuously for most of the period unless the surface temperature falls below the targeted temperature.

5.4. Discussion

In considering the results presented in this chapter, the terminal unit type is found to be an influential parameter in sizing and dimensioning the boreholes. In particular, those terminal units that can smooth out the hourly/daily peaks can effectively reduce the required borehole length. In other words, the findings presented in this chapter show that slow response systems could allow the ground system to be much smaller than what would be required for fast-response systems. This issue was previously studied by Bourdakis et al. [95] for chiller-driven TABS and embedded systems. Our findings confirm the results previously reported in other studies regarding the compatibility of slow-response systems, such as TABS and embedded systems, with ground-coupled systems [44,46,96].

Different thermal comfort levels may raise concerns over the fairness of the comparison made between the ACBs and TABS. As previously stated in Section 5.2, TABS are not suitable terminal units for places where precise room temperature or comfort level is a matter of concern. However, in practice, using a supplementary cooling system can alleviate the room temperature increase during the peak cooling loads. One method is to improve the occupants' thermal comfort by increasing the air movement using personalized ventilation [10,97] or ceiling fans [98,99]. Applying this method does not influence the cooling plant design. Another method is to use the ventilation system, which is already installed to ensure indoor air quality, to cover the peaks and unexpected loads and use the TABS to handle the main part of the cooling loads [100–102]. This method can influence the cooling plant if the same plant is used for both systems. Alternatively, a backup plant can be assigned for the ventilation and used under on-peak cooling demand periods.

5.5. Conclusions

This chapter describes the role of building terminal unit types in sizing the BHEs for the DGCSs. The terminal units investigated are ACBs and TABS. TABS can smooth out the daily ground heat rejection loads and allow the boreholes to be significantly shorter than what would be required for the fast-response terminal units, e.g. ACBs. On the other hand, using the ACB system requires longer BHEs since the daily peaks are intense.

The thermal performance of a terminal unit cannot be investigated without considering the influence of operation strategies. Using conventional on/off feedback controllers for the ACBs to maintain the setpoint temperature increases the daily peaks and has consequences for the BHE size. Adopting the "self-regulation" effect to reduce the daily peaks can alleviate the intense daily peaks and yield shorter boreholes compared to the typical control systems with an on/off feedback controller.

The findings presented in this chapter suggest considering a trade-off between the thermal comfort level, pump electricity use and reduced borehole length when choosing the terminal unit type. Thermal comfort with TABS varies and it is closer to the slightly warm level during the peaks. ACBs equipped with feedback controllers guarantee the maintenance of the desired thermal comfort level but need much longer BHEs, in this study approximately 53%.

The pump electricity use is mainly dependent on both the operation strategy and the terminal unit type. However, the findings show that using TABS leads to higher pump energy electricity use because they operate for much longer hours than the ACBs. Figure 5.8 provides a visual summary of the main findings presented in this chapter.

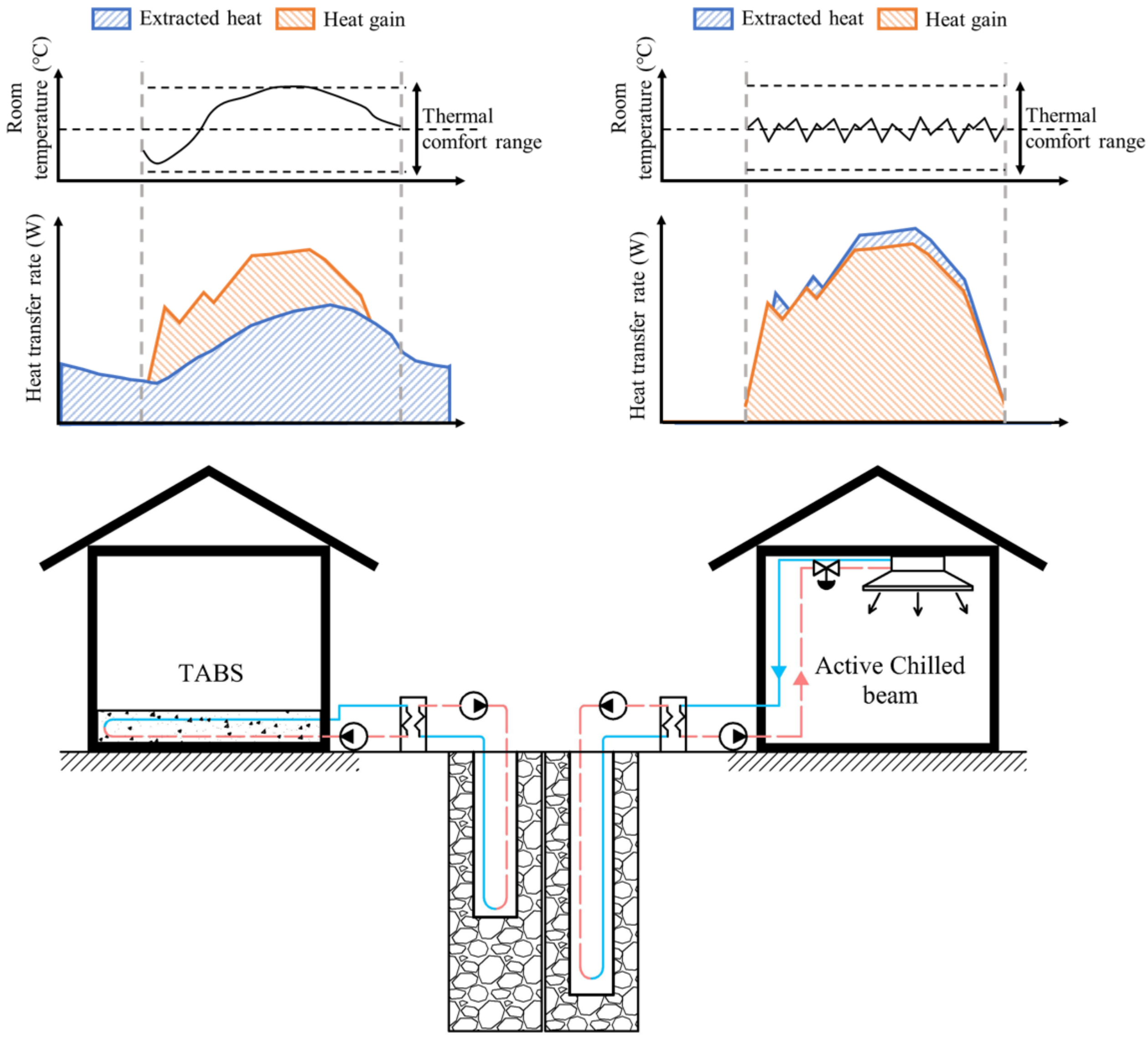

Figure 5.8. *A visual summary of the key findings presented in this chapter.*

6. Design and application of the DGCSs for existing buildings

DGCS is among the most energy-efficient and sustainable solutions for comfort cooling in newly constructed office buildings [38,65]. During the new buildings' design process, the building cooling load can be minimized by applying suitable measures to get a reasonably sized borehole system. However, using this technology for existing buildings may represent some difficulties. Existing buildings generally have high cooling loads and require a larger borehole system than those for the new buildings. The benefits of eliminating the chiller electricity and its initial cost likely become trivial if the DGCS installation costs become unreasonably high.

The present chapter aims to explore the possibilities of using DGCSs instead of chillers in existing office buildings. This chapter also presents a systematic approach to evaluate the influence of different renovation measures on sizing the boreholes. A typical Swedish office building using a chiller and ACBs and in need of energy renovation is taken as the reference case. In the first step, the chiller is replaced with a DGCS in the reference building. In the second step, a renovation package, consisting of some selected renovation measures, is applied to the building equipped with the DGCS. The final results compare the energy performance and the borehole design before and after the application of the renovation package.

6.1. Methodology

A medium-sized office building model developed by the U.S. Department of Energy (DOE) [103] is used to represent a simple yet realistic building. The model is modified to feature an old Swedish office building from the 1970s - 80s and it is assumed that it requires energy renovation. The original building uses an air-cooled chiller for comfort cooling. The following section summarizes the methodology used in the study. Further details are available in paper V.

6.1.1. Building model

The reference model is an office building with three floors and a total floor area of 4981 m^2. Each floor is divided into five zones, a large interior zone surrounded by four perimeter zones, see Figure 6.1. The building is assumed to be located in Gothenburg, Sweden.

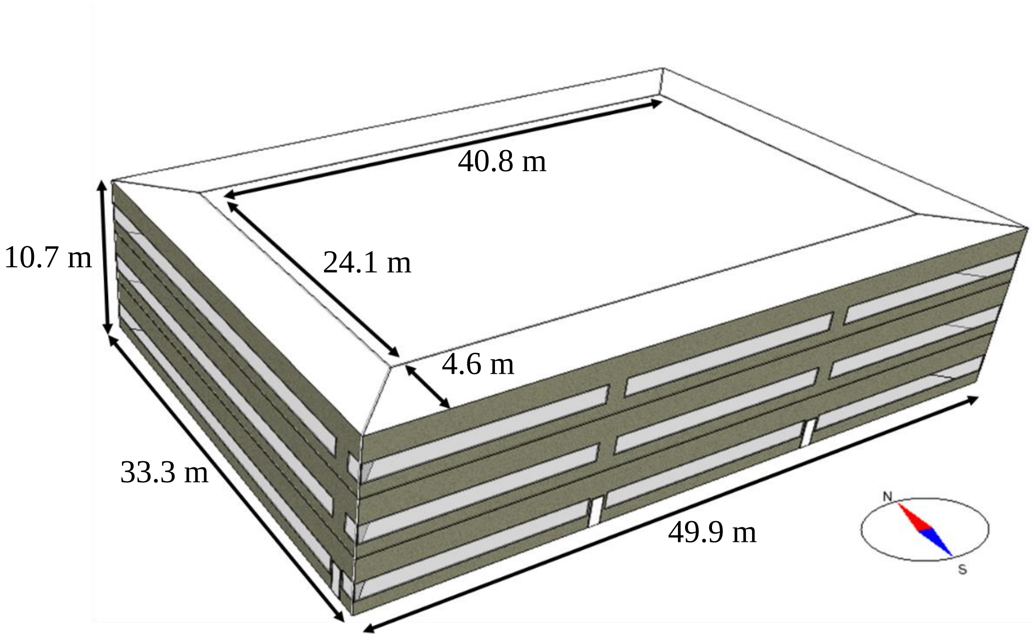

Figure 6.1. *Isometric view of the reference building showing the perimeter and the interior zones.*

Materials with low thermal insulation characterize the reference building envelope to represent an old Swedish office building. The building envelope has an overall U-value of 0.57 W/m^2.K. U-values are aligned with Swedish buildings constructions from the 1970 - 80s [104]. The internal heat gains comprise heat from office equipment (12.2 W/m^2), people (8.0 W/m^2), and lighting fixtures (10.6 W/m^2), according to ASHRAE Handbook of Fundamentals [93]. The occupancy period is between 8:00 and 17:00 on weekdays. Table 6.1 summarizes the main features of the geometry and the envelope characteristics of the reference building.

The reference building's cooling system uses a chiller and ACB terminal units. The chiller is air-cooled and has a nominal coefficient of performance (COP) of 2.7, providing a supply water temperature of 5 °C. The ACBs are designed for the supply water temperature of 17 °C, the primary air temperature of 20 °C and the primary airflow of 1.5 l/s.m^2. The cooling system operates on workdays from 06:00 to 17:00. Design parameters of the reference building's cooling system are provided in Table 6.1.

Table 6.1. *Main characteristics of the geometry, envelope and the cooling system of the reference building.*

Geometry	
Building occupancy type	Office
Number of floors	3
Total floor area (m^2)	4981
Window-to-floor area ratio (%)	13
Window-to-wall area ratio (%)	51
Thermal properties	
Average envelope U-value ($W/m^2.K$)	0.57
Windows G-value (-)	0.76
U-value of external walls ($W/m^2.K$)	0.4
U-value of roof ($W/m^2.K$)	0.27
U-value of windows ($W/m^2.K$)	2.86
U-value of the external floor ($W/m^2.K$)	0.3
Cooling system	
Operation time period (-)	06:00 - 17:00 weekdays
Primary airflow rate ($l/s.m^2$)	1.5
Exhaust airflow rate ($l/s.m^2$)	1.5
Primary air temperature (°C)	20.0
$T_{return,water} – T_{supply,water}$ at full power (K)	3
Supply water temperature (°C)	17.0
ACB cooling capacity control method	On/off water flow
Room temperature setpoint for cooling (°C)	24.0

6.1.2. Proposed renovations measures

Energy renovation measures in this study are performed in two steps. In the first step, the chiller is replaced with the DGCS. In the second step, a renovation package comprised of selected renovation measures is performed on the reference building equipped with the DGCS.

The renovation measures used in this study are listed in Table 6.2. Envelope U-values are taken based on the suggestions of the Swedish national board of housing (Boverket) and a database of Swedish commercial buildings constructed after 2010 [104,105]. Windows' G-values are chosen among the typical values often found in the literature [106,107]. Internal heat gains decreased from 30.8 W/m^2 to 18.0 W/m^2 due to reductions in internal gains from the lighting and equipment [93]. The room temperature setpoints are aligned with the recommended range in office buildings during the summer [94,105,108].

A night cooling system is designed to reduce the accumulated heat in the building and pre-cool the building structure during summertime between June 1st and August 31st. The ambient air is supplied at its natural temperature to cool the building temperature to 21.0 ℃. The system runs from 00:00 to 06:00 on weekdays and 18:00 to 00:00 on Sundays when the ambient temperature is below the room temperature setpoint.

Table 6.2. *A comparison summary of the renovation measures investigated in the individual measure analysis.*

Renovation measures	Variable	Range of variability
Building design	Window G-value (-)	From 0.76 to 0.3 with a step of 0.15
	Envelope U-value (W/m^2.K)	From 0.57 to 0.17 with a step of 0.1
	Internal heat gains (W/m^2)	From 30.8 to 18.0 with a step of 4.3
Cooling system operational setting	Temperature setpoint (℃)	From 26 to 22 with a step of 2
	Night cooling airflow rate (ACH)	0.75 and 1.5

6.1.3. Description of the DGCS

The DGCS in this study is designed as a thermally-balanced system. The annual heat rejection loads to the ground are offset by the heat extraction loads from the ground to pre-heat the ventilation air during the heating season, see Figure 6.2. For circumstances where balancing the ground loads cannot be fulfilled by preheating the ventilation air, a dry cooler is used for balancing the system. Using the dry cooler allows for the cooling of the borehole fluid for a more extended period, even outside the air handling unit’s operating hours. A control system is used to compare the annual ground heat rejection and extraction loads. The control system allows extracting heat from the ground until it becomes equal to the total heat rejection loads over a year.

When the outdoor temperature is under 12 ℃, the working fluid leaving the heat exchanger is directed towards the preheating coil in the air handling unit by the control valve, see Figure 6.2. The fluid preheats the primary air to the beams and gets cooled. It is then circulated through the BHEs to cool the ground. When the outdoor temperature is above 12 ℃, the control valve directs the working fluid from the heat exchanger directly to the BHEs. The working fluid is also sent directly to the BHEs if the borehole outlet temperature is < 2 ℃. This stops the circulation to the preheating coil and avoids freezing the ethanol-to-water heat exchanger.

When the dry cooler and air handling unit are simultaneously in use, the fluid first goes to the preheating coil and then to the dry cooler. During the non-working hours of the air handling unit, the fluid only circulates through the dry cooler coil as the valve to the preheating coil is shut.

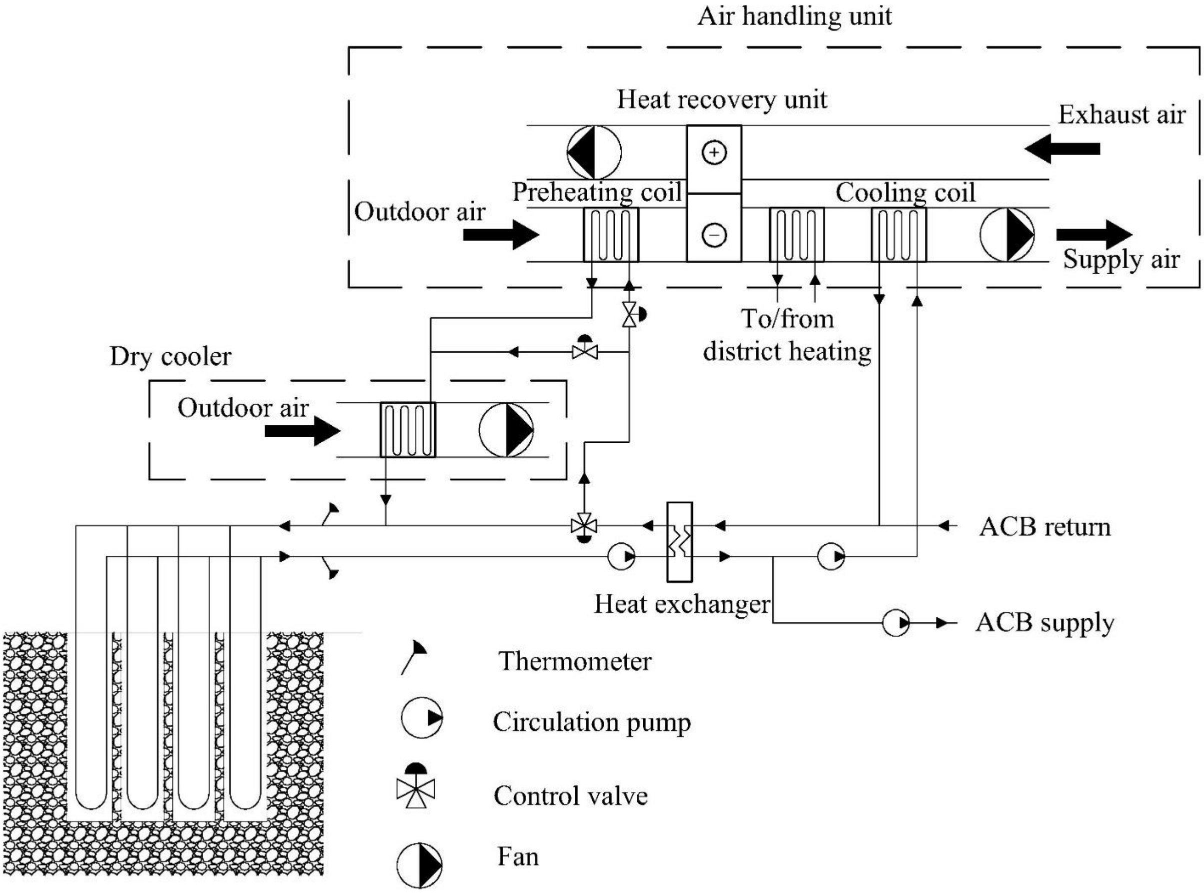

Figure 6.2. *Schematic of the DGCS.*

6.1.4. Borehole sizing criteria and initial design consideration

Using ACBs makes it possible to utilize supply water as high as just below the room temperature. Consequently, the borehole outlet fluid temperature can be aligned based on the required minimum temperature of the ACBs. For a given building with a specific cooling demand, reducing the required borehole length causes an increase in the fluid temperature leaving the borehole. This requires larger/additional ACBs to compensate for the increased supply water temperature.

Figure 6.3 shows the maximum borehole outlet fluid temperature plotted against the required borehole length for the reference building. There is a potential to reduce the required borehole length by 30% (from 4900 m to 3420 m) by increasing the outlet temperature from 16 ℃ to 20 ℃. To maintain the cooling capacity of the ACBs, the total coil length needs to be increased by 70%.

However, there are some challenges to implement this idea. Firstly, this study does not consider changing the ACBs design and dimensions. Secondly, reducing the borehole length for a given building cooling load increases the specific heat rejection rate per meter of boreholes. Reducing the borehole length from 4900 m to 3420 m increases the heat rejection rate from 53 W/m to 77 W/m. Having a high heat rejection rate to the ground is more feasible in places with high ground thermal conductivity (> 3.0 W/m.K) [19,109]. Furthermore, designing the boreholes with high heat rejection rates increases the risks related to the unpredicted circumstances and heatwaves.

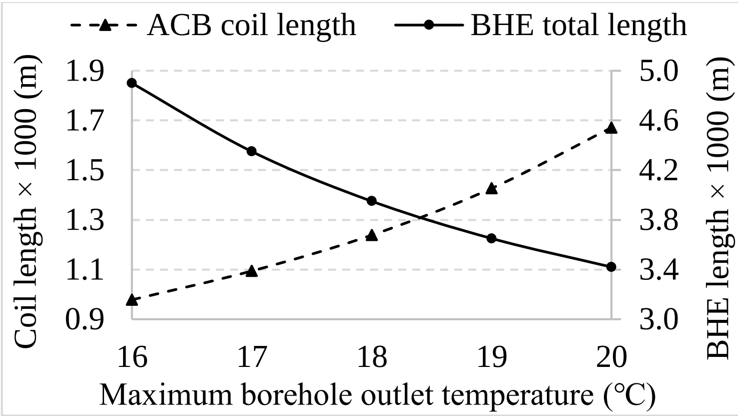

Figure 6.3. *Required BHE length and ACB coil length in relation to the maximum borehole outlet temperature for the reference building equipped with the DGCS.*

Figure 6.4 shows the design procedure for the borehole system used in this chapter. For each input design parameter set, the required borehole length is iteratively adjusted to get the maximum outlet fluid temperature of approximately 16 °C. This outlet temperature guarantees the required cooling capacity for the ACBs on the peak conditions to maintain the room temperature at 24 °C. In addition, the minimum inlet temperature is considered to avoid freezing the ethanol-to-water heat exchanger. The borehole length is kept between 200 m - 300 m, as per common practice in Sweden [30,59]. The borehole simulation results are the inlet and outlet borehole fluid temperatures, the borehole fluid flow rate, the required borehole length and the borehole configuration.

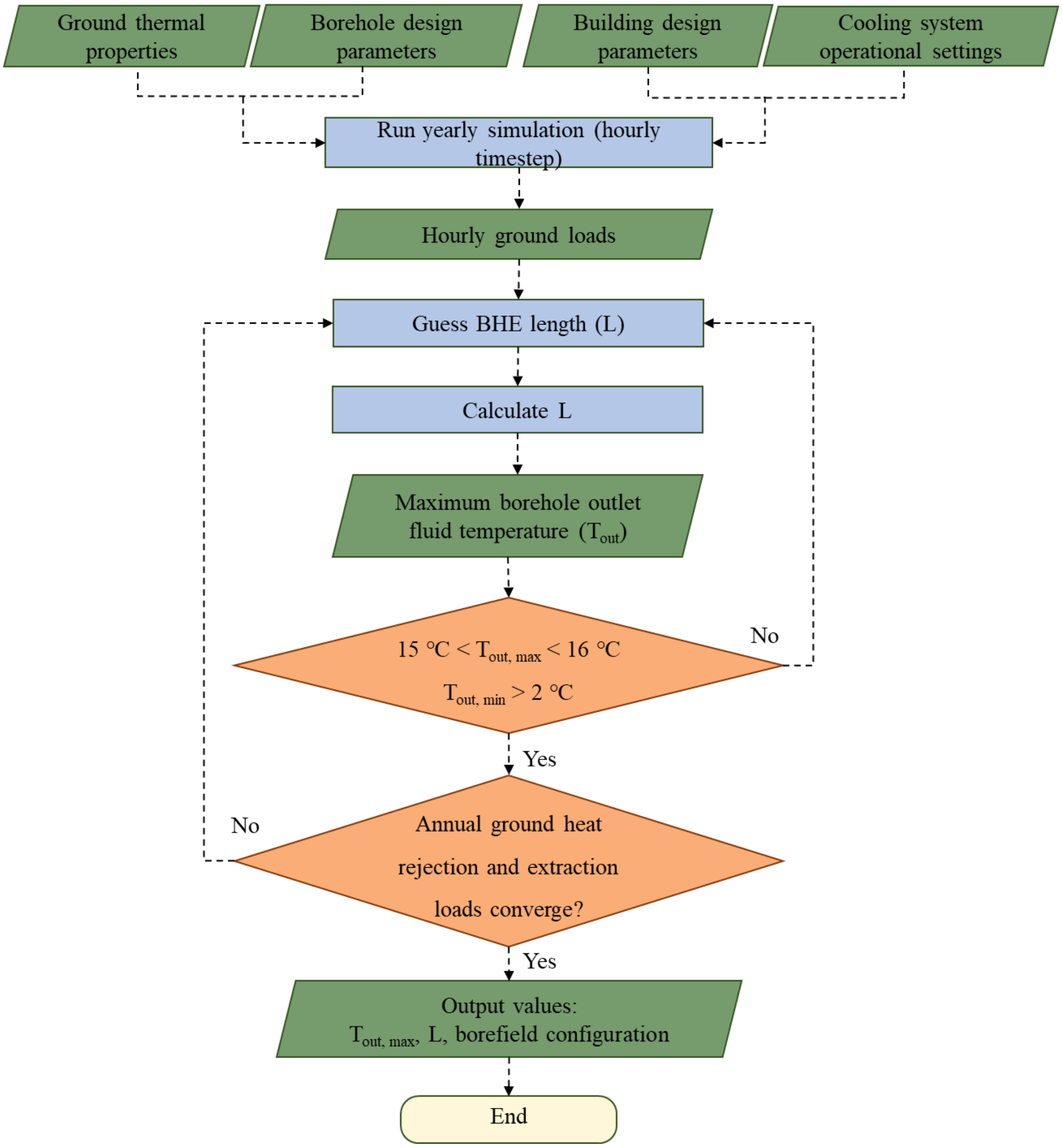

Figure 6.4. *Flow chart of the simulation process. "L" is the required borehole length, and "T_{out}" is borehole outlet fluid temperature.*

6.2. Results

This section first presents the results for the office building with the chiller system. The DGCS is then replaced with the chiller and results from the sensitivity study on the building loads and the borehole length are presented. The rest of this section deals with the results of a comparative analysis between the reference and the renovated buildings in terms of building thermal loads, energy demand and borehole design.

6.2.1. Reference building heating and cooling load profiles

Figure 6.5 shows the hourly heating and cooling loads of the reference building before renovation. The loads are assumed to be provided by the district heating and the chiller, respectively. Note that cooling provided by the outdoor air is not shown in this figure. Positive and negative values represent heating and cooling loads, respectively.

The reference building has a year-round cooling demand. In winter, the interior zone requires cooling to remove the internal gains. Cooling demand in summer increases to remove both the internal and external heat gains. The peak hourly and the annual cooling loads are 262.5 kW and 134 MWh, respectively.

The district heating system provides heating to the reference building. While heating is required all year round, the demand is small in summer and high in winter. Although the space heating system is off in summer, air heating is still needed to maintain the supply air at 20 °C.

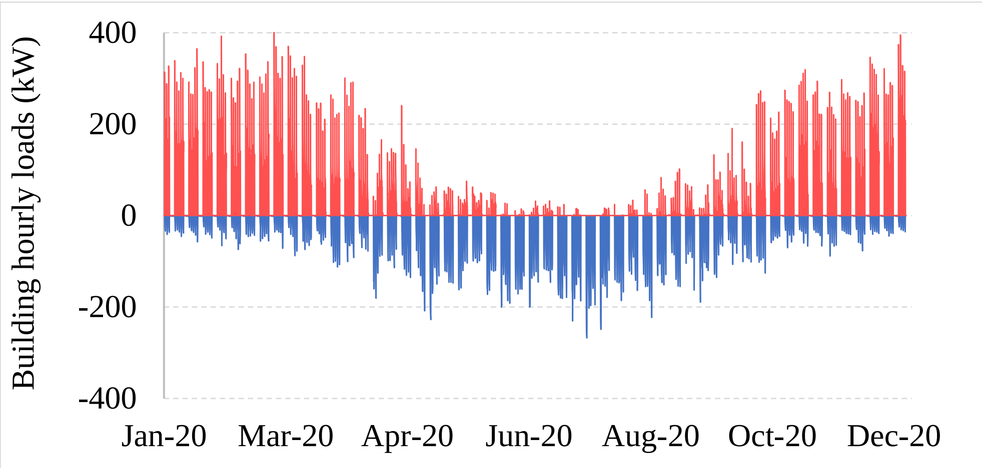

Figure 6.5. *Hourly cooling and heating load profiles for the reference building used for the preliminary design. The loads represent heating and cooling provided by the district heating system and the chiller, respectively.*

6.2.2. Sensitivity analysis on renovation parameters

Performing any renovation on a building influences the building's heating and cooling loads. Although the design of the DGCS is only dependent on the cooling load, the heating load cannot be disregarded. This is especially important in cold climates, such as Sweden, where heating is usually the dominant load. Therefore, the sensitivity analysis investigates the changes in both heating and cooling loads but sizing the boreholes is performed considering only the cooling loads.

Figure 6.6 shows the building cooling demand (peak and annual loads) and the required borehole length for changes in the renovation measures studied. Among the parameters investigated, reducing the G-value causes the highest reduction in the required borehole length. The required borehole length can be reduced by 35% (from 4900 m to 3200 m), see Figure 6.6C. Both the peak and the annual ground loads are significantly decreased with G-values. However, reducing the daily peak loads is expected to play a major role in sizing the boreholes, since boreholes cannot tolerate fast and intense ground loads.

Increasing the setpoint temperature over the range from 22 ℃ to 26 ℃ slightly decreases the peak load by 18% but dramatically decreases the annual load by 72%, see Figure 6.6J and Figure 6.6K. Significant decreases in the annual load can be mainly related to the building's geometry, having a large interior zone and small perimeter zones. The interior zone requires cooling all year round, and increasing the cooling setpoint causes a substantial reduction in the annual cooling demand.

Applying night cooling has a minor influence (~ 11%) on the required borehole length (see Figure 6.6O). Precooling the building at the airflow rate of 0.75 effectively reduces the peak load, but its influence on the annual load is insignificant. Further increase in the night airflow rate does not have tangible benefits in reducing ground loads.

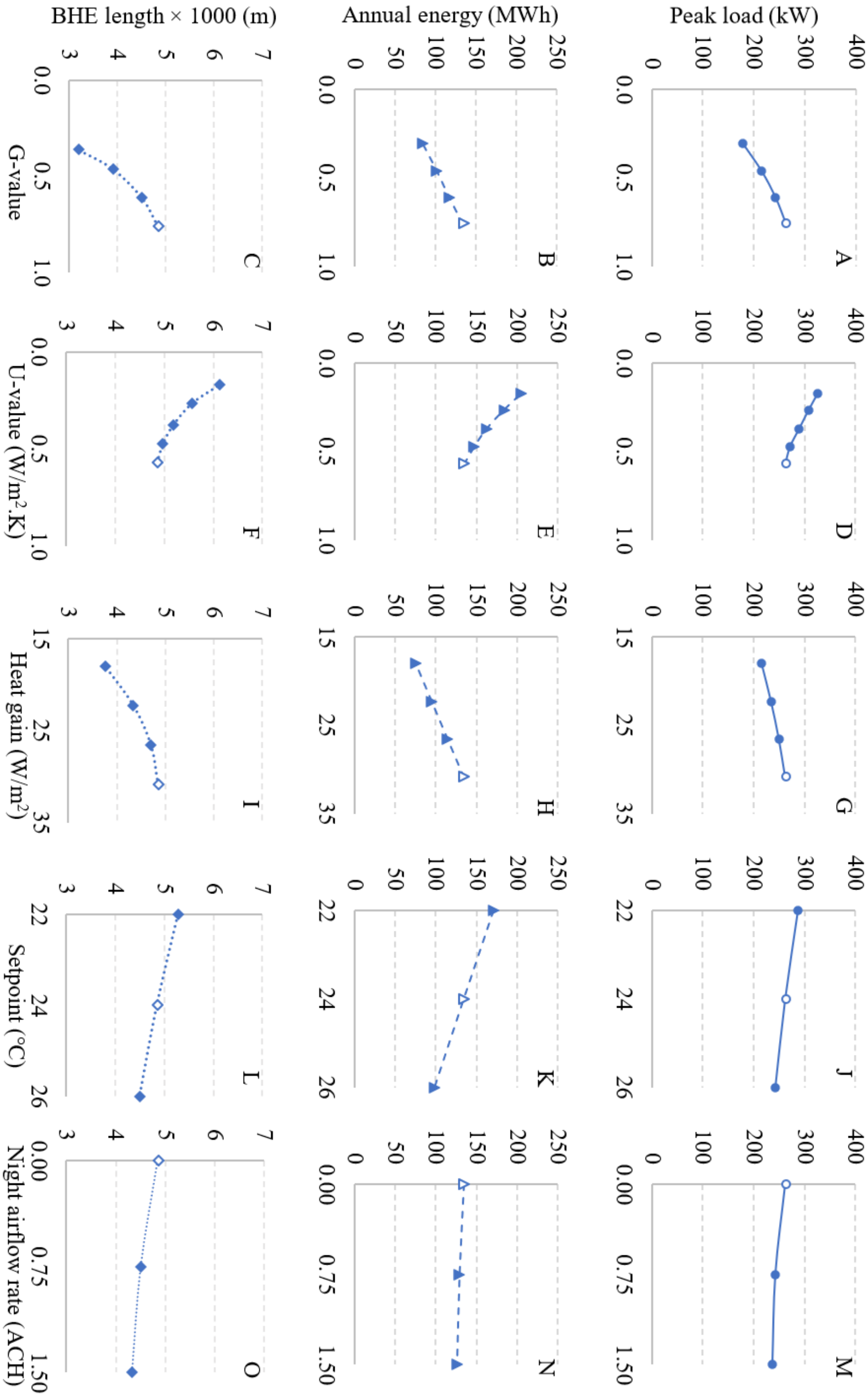

***Figure 6.6**. Peak daily and annual heat rejection loads to the ground, and the required borehole total length for different A-C) window G-values, D-F) Envelope U-values, G-I) internal heat gains, J-L) room temperature setpoints and M-O) night airflow rates for precooling the building. Values for the reference building are marked as empty markers.*

Figure 6.7 shows the hourly peaks and the annual heating loads plotted against the renovation values varying within the specified range. The G-value has an insignificant influence on the peak loads (see Figure 6.7A). Heating peaks happen on the cloudy days in the absence of solar radiation and hence, the G-value does not influence peak intensity. Therefore, the results suggest that changing the G-value has almost no effect on sizing the heating terminals. By comparing the results in Figure 6.6B and Figure 6.7B, a trade-off can be seen between the annual heating and cooling demands. The G-value influences the annual demands almost equally, but it has different consequences for the system design. A reduction in the cooling load leads to shorter boreholes. Changes in the annual heating load directly affect the building purchased energy, i.e. heating energy acquired from the district heating network. The influence of the G-value on the building peak and energy loads depends on other factors such as building geometry, glazing area, building geographical locations, etc. This will be further discussed in Section 6.3.

The U-value has the highest influence on the building's heating demand. A comparison of the building's heating and cooling demands (Figure 6.6E and Figure 6.7D) shows that the U-value has a more significant effect on the heating demand. Heating loads are highly dependent on the transmission losses while cooling loads are mainly determined by external and internal gains.

By default, IDA-ICE does not account for the internal gains for peak heating load calculations. Thus, internal gains have almost no influence (< 1%) on the peak loads, see Figure 6.7E. Comparing the annual heating and cooling demands in Figure 6.7F and Figure **6.6**I shows that internal loads have a higher impact on the annual cooling loads.

Changing the cooling setpoint temperature influences not only the cooling demand but also the heating demand. The interior zone always requires cooling and its temperature is controlled by the cooling setpoint. The heating setpoint is 21 °C and is always below the cooling setpoint temperature. Therefore, the interior zone heats the exterior zones and the rate is dependent on the cooling setpoint, see Figure 6.7H.

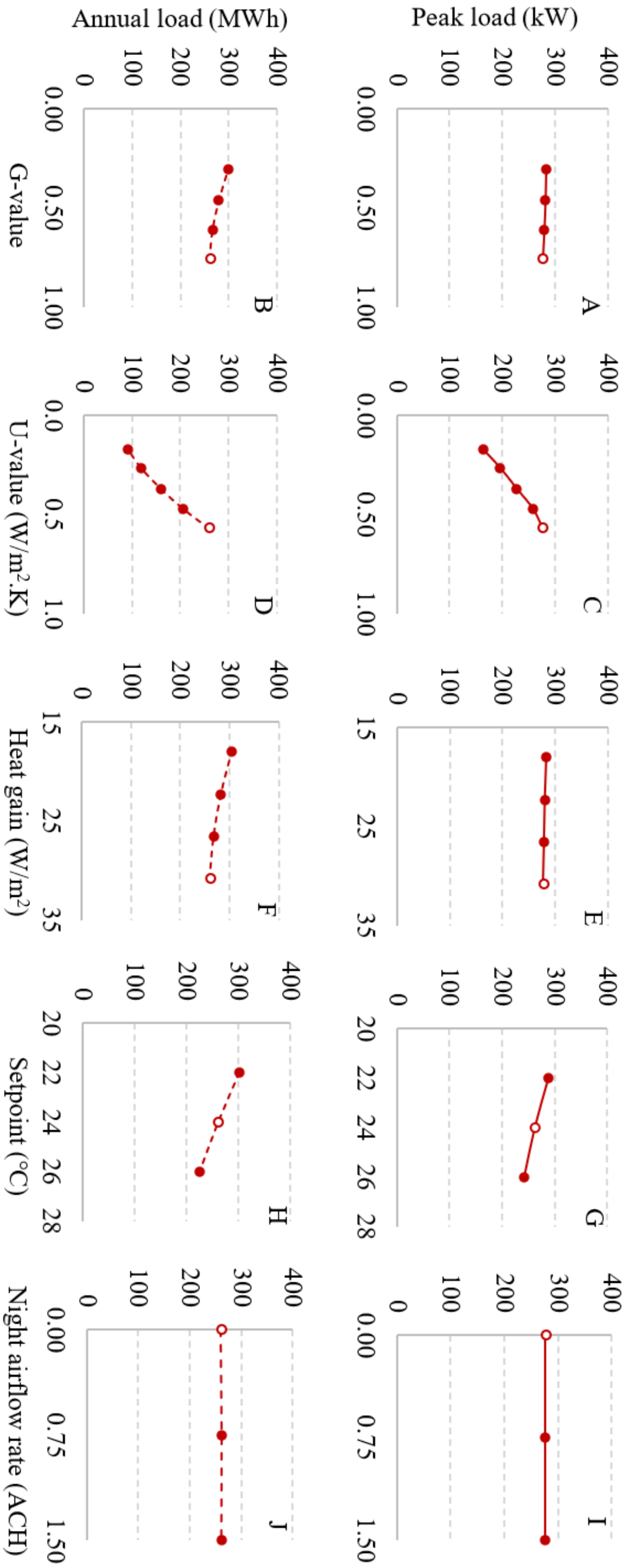

Figure 6.7. *Peak daily and annual heating loads for different A-B) window G-values, C-D) envelope U-values, E-F) internal heat gains, G-H) room temperature setpoints and I-J) night airflow rates for precooling the building. Values for the reference building are marked as empty markers.*

6.2.3. Combined measures

The reference building, denoted as "Ref-chiller", uses a chiller and has high electrical and thermal demands. Two major renovation packages are performed on the reference building to reduce its energy demand. The first package aims at reducing the electricity demand by replacing the chiller with the DGCS. This package is known as "Ref-DGCS". The second package consists of some renovation measures to reduce the building thermal demand and is referred to as "Renovated" in the text. The measures are selected based on the results of the sensitivity study presented in Section 6.2.2. Table 6.3 summarises the main characteristics of the building and the cooling system parameters in different office models in the combined measure analysis.

Table 6.3. *Design parameters used in the combined measures analysis.*

Variable	Ref-chiller	Ref-DGCS	Renovated
Envelope U-value ($W/m^2.K$)	0.57	0.57	0.42
Window U-value (-)	2.86	2.86	1.2
Windows G-value (-)	0.76	0.76	0.3
Internal heat gains (W/m^2)	30.8	30.8	18.0
Cooling system	Chiller	DGCS	DGCS
Night cooling airflow rate (ACH)	0	0	1.5

6.2.4. Building renovation and DGCS sizing

The DGCS is designed based on the cooling loads shown in Figure 6.5 to maintain the borehole outlet temperature <16 °C based on the design criteria explained in Section 6.1.4. Figure 6.8A shows the hourly heat rejection loads to the ground on the design day before and after implementing the energy renovation. Before the renovation, the ground loads increase rapidly and peak around midday, where they stabilize at approximately 262 kW. After the renovation, the ground loads are significantly reduced by 48% (from 262 kW to 126 kW).

As a result of the renovation, the total annual ground heat rejection load is reduced by 69%, Figure 6.8B. This substantial reduction makes it possible to remove the dry cooler and balance the ground loads by only pre-heating the primary air in the air handling unit (Figure 6.2).

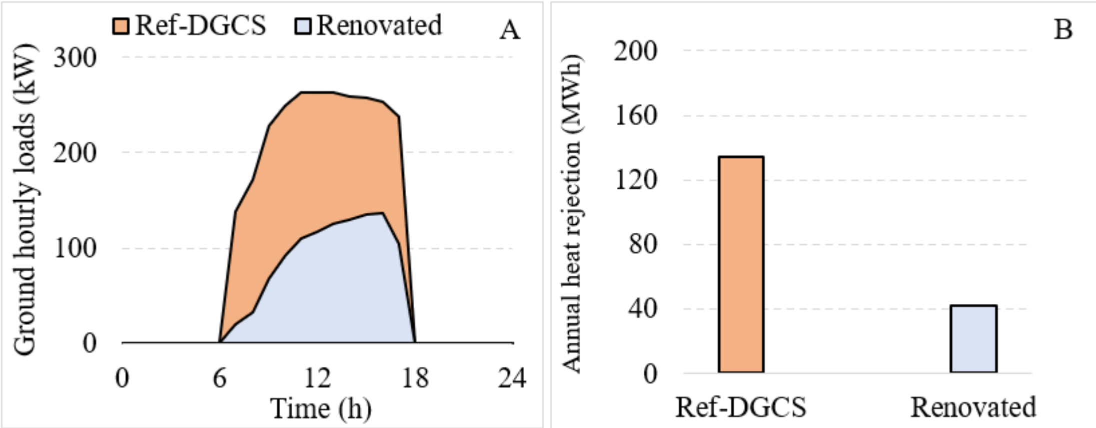

Figure 6.8. *A) Hourly ground heat rejection loads on the design day and B) the annual heat rejection loads for the reference building equipped with the DGCS (Ref-DGCS) and the renovated building.*

Figure 6.9 shows the required borehole length for the reference building before and after energy renovation. As previously stated, the sizing criteria keep the maximum outlet temperature at about 16 °C and the boreholes' depth between 200 m – 300 m. Due to the decreased ground heat rejection loads, the total borehole length for the renovated building is 56% smaller.

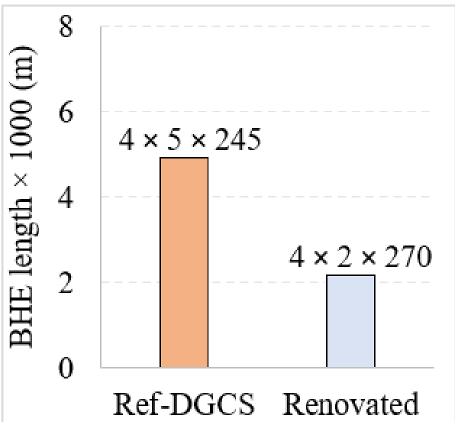

Figure 6.9. *Total required borehole length and borehole configuration for the renovated building and the reference building equipped with the DGCS (Ref-DGCS).*

Distribution of the borehole outlet temperature is presented in Figure 6.10. The maximum temperature for both cases is about 16 °C, as planned. The median temperature, represented by a horizontal line in the central box, is about 8.3 °C and is equal for both cases. This means that the DGCS is thermally balanced over the years of its operation. What causes the fluid temperature to become lower than the undisturbed ground temperature is the cooling of the borehole fluid in the air handling unit and in the dry cooler.

Considering the outliers appearing in the borehole outlet temperature for both cases, the borehole design for the renovated case is sensitive to the daily peak loads. The maximum outlet temperature for the renovated case, defined by the upper whisker, is 12.7 °C. Some intense peak daily loads extend the temperature distribution range to reach 16 °C. This temperature distribution pattern suggests using a supplemental cooling source to take care of the peaks in order to reduce the required borehole length.

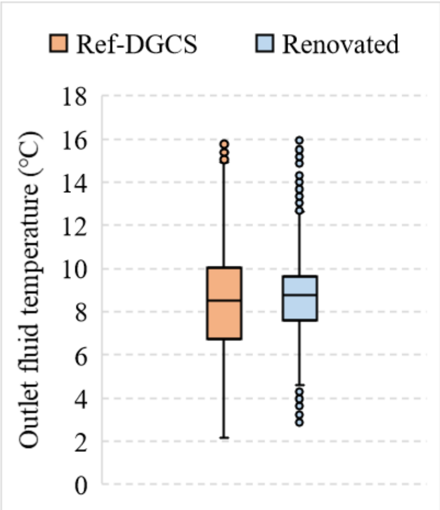

Figure 6.10. *Borehole outlet temperature for the renovated building and the reference building with the DGCS (Ref-DGCS). The undisturbed ground temperature is 8.3 °C.*

6.2.5. Energy demand analysis

Figure 6.11 compares the building energy use for the cases investigated. Building energy use is the total energy input to the building technical systems to fulfil the energy need for space heating, space cooling and ventilation. In other words, building energy use is the sum of the electrical, thermal and recovered energies supplied to the technical systems. The ground cooling represents the building's cooling demand provided by the borehole system, i.e. the annual heat rejection loads to the ground. The ambient air cooling deals with the free cooling by the outdoor air when the outdoor temperature is below 20 °C, i.e. the primary air temperature supplied to the beams. The electrical energy includes the electricity used to drive the chiller (if included), circulation pumps and fans in heating and cooling systems.

The annual building electricity demand for the reference building is 17.6 kWh/m^2.y, including auxiliary electricity at 3.5 kWh/m^2.y and chiller electricity at 14.1 kWh/m^2.y. As shown in Figure 6.11, using the DGCS instead of the chiller yields to a significant electrical energy decrease by 81% (from 17.6 kWh/m^2.y to 3.3 kWh/m^2.y). After performing the renovation, the electricity demand is further reduced from 3.3 kWh/m^2.y to 2.4 kWh/m^2.y. Reducing the annual cooling demand allows for balancing the ground loads without the need for the dry cooler, resulting in a further reduction in electricity use.

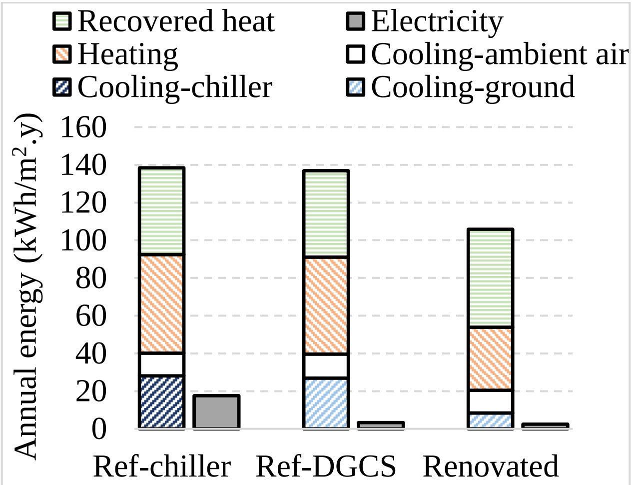

Figure 6.11. *Annual energy use (cooling, heating, recovered and electricity) for the reference building (Ref-chiller), the reference building equipped with the DGCS (Ref-DGCS) and the renovated building.*

Figure 6.12 presents the purchased energy use of the case studies. Purchased energy is defined as the energy supplied to the technical systems from the network. The purchased energy sources include electricity from the grid and heating from the district heating network. Auxiliary energy includes the purchased electrical energy to run the fans and pumps for the heating and the cooling systems.

Comparing the “Ref-chiller” and the “Ref-DGCS” cases shows that using the DGCS yields a substantial reduction in the purchased electricity by 81%. Comparing the “Ref-DGCS” and the “Renovated” buildings shows that implementing the renovation measures has a significant impact on the purchased heating demand (a reduction of 35%). It also contributes to reducing electricity demand by 27%.

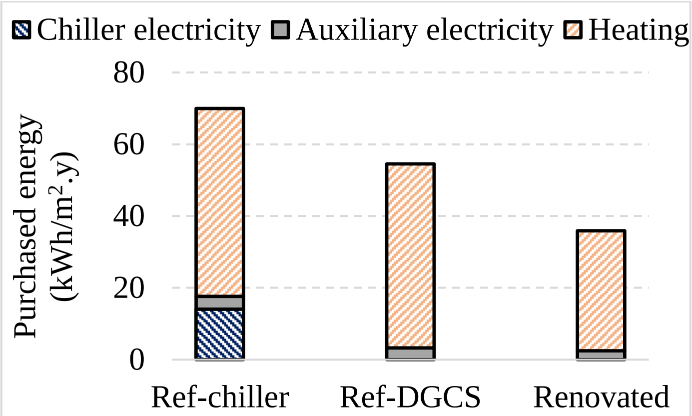

Figure 6.12. *Annual purchased energy use for the reference building (Ref-chiller), the reference building equipped with the DGCS (Ref-DGCS) and the renovated building.*

Having such a low electricity demand is an advantage in case of power outages. During the brownouts, when the electrical power supply is reduced, or blackouts, when the network provides no electricity, a small electric generator can keep the system operating smoothly.

6.2.6. Performance of the cooling system

There are various methods to define and quantify the performance of the cooling systems. The methods consider different system boundaries, cooling system types, plant type (heating and cooling), etc [34,110]. This study uses the seasonal performance factor (SPF) for the DGCS and the coefficient of performance (COP) in the case with chiller.

The SPF for the ground system (SPF_{DGCS}) is the proportion of the annual ground heat rejection loads to the annual electricity for the circulation pumps (for the boreholes and the ACBs), and the fan installed in the dry cooler. The SPF_{DGCS} values for the renovated and the Ref-DGCS cases are 71 and 23, respectively. The SPF for the renovated building is higher because the ground loads can be balanced without the aid of the dry cooler, resulting in a reduction of the fan's electricity use by about 3.8 MWh.

SPF can also be defined for the whole cooling system (SPF_{system}), representing the ratio of total cooling energy delivered to the building using the DGCS and the ventilation system to the total electricity demand used by these systems. The SPF_{system} values for the renovated building and the Ref-DGCS cases are 12 and 15, respectively. Most of the annual cooling load for the Ref-DGCS case is handled by the borehole system. The circulation pumps in the borehole system require less electricity compared to the fans in the air handling unit. However, in the renovated case, the ventilation air from the ACBs removes a larger proportion of the annual cooling load in comparison to the hydronic part of the ACBs. Therefore, it can be concluded that the SPF_{system} is highly influenced by the ventilation system and it does not clearly represent the borehole system's energy performance.

Energy performance for the Ref-chiller case can be defined in two ways. The chiller's performance can be determined by COP, as the proportion of the cooling delivered to the building by the electricity used. The calculated annual COP of the chiller is 2.1. Alternatively, using the SPF_{system} value for the Ref-chiller case gives a thermal performance of 2.4. The SPF_{system} is higher than the annual COP of the chiller due to the free-cooling provided by the ambient air. The SPF_{system} value is significantly lower than what is obtained for the ground-coupled cases and indicates the superiority of the DGCS in terms of energy use of the system.

6.3. Discussion

This chapter presents a systematic approach to evaluate the influence of the studied renovation measures on the building cooling demand and sizing the boreholes. The results suggest that performing energy renovations does not necessarily improve efficiency or reduce the required borehole length. For instance, renovating the building envelope to reduce the heating losses is common in building energy renovation plans in cold climates [111–113]. The sensitivity analysis shows that only reducing the reference building envelope U-value from 0.57 $W/m^2.K$ to 0.42 $W/m^2.K$ results almost in the same required borehole length (~5000 m) and the SPF_{DGCS} of 23. In comparison, as defined in Figure 6.9, the renovated building has the borehole length of 2160 m and the SPF_{DGCS} of 71.

The influence of many building design parameters, such as those studied in this chapter, are building dependent. Nevertheless, it is possible to generalize the findings reported in this chapter. The effectiveness of the G-value on the building load profile is associated with the building's geometry, the building's geographical location, the glazing area, the application of shading devices, etc. The G-value determines the external heat gains from solar radiation and causes large variations in the daily building cooling load. High-intensity daily loads are not favourable to the borehole system because of the low thermal conductivity and the long time constant of the ground surrounding the boreholes [46]. In this study, the G-value is shown to offer the highest effect on the peak hourly cooling loads. This is of particular importance since the building has a medium window-to-wall area ratio and a low window-to-floor area ratio. Therefore, it can be suggested that the G-value would have an enormous impact on the sizing of the boreholes regardless of the building's orientation and geometry, as also pointed out in [45].

The cooling setpoint in this study is another parameter offering considerable influence on borehole sizing. The importance of cooling setpoints is less than what we have observed in our previous publications [114–116]. This discrepancy is attributed to the building's geometry, which can enhance or diminish the significance of the cooling setpoint. In buildings where the cooling load is formed mostly by solar radiation, increasing the setpoint can significantly decrease the peak hourly loads and hence, the required borehole length. On the other hand, the impact of cooling setpoints on borehole size becomes less important in buildings with small window-to-wall ratio or large interior zones, as the cooling setpoints contribute considerably to the annual loads.

6.4. Conclusions

DGCS is among the most energy-efficient technologies for comfort cooling in new office buildings. However, using this technology for existing office buildings presents some challenges in designing and implementing the system. This chapter intended to evaluate the energy-saving possibilities of using a DGCS instead of a chiller-based cooling system for an existing building.

The DGCS investigated in this study required only a modest amount of electricity for the auxiliary demands. Therefore, a substantial reduction in the purchased electricity was yielded when the chiller was replaced with the DGCS. However, building energy demand was still higher than what recommended in the national guidelines, and the designed DGCS required a large borehole system. Therefore, a sensitivity analysis was performed to systematically evaluate the influence of various renovation measures on the building's thermal load profile and the borehole sizing. The sensitivity analysis showed that performing any renovation did not necessarily contribute to a reduction in the boreholes' length but could have a minor or even a reverse impact. It can be concluded that using the DGCSs can be a more feasible solution after implementing appropriate energy renovation measures, due to the reduction in the boreholes' length.

Figure 6.13 is a visual summary of the main findings presented in this chapter.

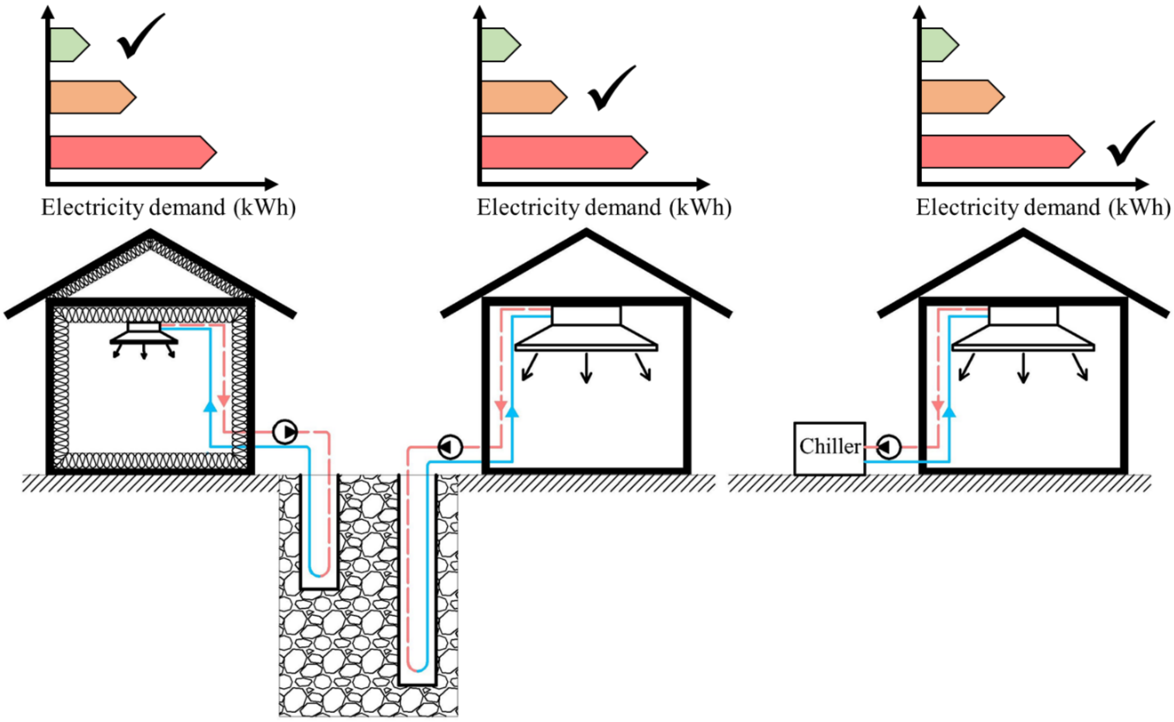

Figure 6.13. *A visual summary of the key findings presented in this chapter.*

7. Concluding remarks

The specific conclusions are given in the respective chapters and the articles. The following provides a general concluding discussion with regards to the research objectives.

7.1. Overall conclusions

Research objective 1: *Investigating how indoor terminal units impact the sizing and the thermal performance of the DGCSs.*

Boreholes are sized based on the ground loads and the temperature limits for the fluid leaving and entering the ground heat exchangers. HTC terminal units can affect both the ground loads and the temperature limits.

Terminal units with high thermal mass, such as TABS, have a long response time and can smooth out the peak cooling loads. This work shows and quantifies that slow response systems can allow the ground system to be much smaller than what would be required for the fast-response systems. The results agree with the previous findings regarding the compatibility of the slow response terminal units with the ground-coupled systems.

While slow response terminal units may offer a lower BHE size, it is important to consider the pump energy use and thermal comfort criteria before coming up with the final design. It is most likely that the slow response terminals require the pumps to run much longer hours than the fast response terminals, resulting in a higher pump energy use. Moreover, the room temperature in buildings using slow response terminals may drift during the day.

Sizing boreholes is directly associated with the borehole outlet fluid temperature and the supply temperature to the terminal units. The results show the advantage of designing the building terminal units for high-temperature supply water. Increased supply temperature to the terminals allows increasing the maximum borehole outlet temperature limit, resulting in shorter boreholes. Special considerations need to be taken regarding the borehole design as borehole heat transfer rates (W/m) increase when borehole length is reduced.

Research objective 2: *Investigating how system operation strategies impact sizing and thermal performance of the DGCSs.*

The results have shown and quantified that operation strategies and control methods can influence sizing the boreholes by changing the heat rejection loads to the ground. The operation strategy developed and used in this work relied mainly on the “self-regulation” effect to smooth out the daily peak cooling loads. Using this operation strategy yielded a much smaller borehole system. Based on the results, a reasonable strategy for operating the DGCSs is to allow the room temperature to rise somewhat during the “on-peak” heat gain periods. Using this operation strategy leads to higher pump energy use, since water needs to be circulated in the building loop at a constant rate and temperature.

Research objective 3: *Expanding upon the possibilities of using the DGCSs for comfort cooling in existing office buildings in need of energy renovation and providing design improvement suggestions.*

Using the DGCSs for comfort cooling is generally possible in cold climates, where the building’s cooling loads are low enough to be offset by utilization of the ground loads. The main challenge for using the DGCSs in existing buildings is their high cooling loads compared to the new buildings, resulting in larger boreholes. This work provides a systematic approach to evaluate the influence of various renovation measures on the building cooling and heating loads. The hypothesis is that if not chosen appropriately, the aggregation of the measures can inversely affect the sizing and thermal performance of the DGCSs. The hypothesis is proven by comparing the borehole size for the reference building (the case with ‘bad’ design parameters) and the renovated building. The results suggest performing a thorough analysis of the building load profile to determine an optimized configuration of the building’s renovation measures. For circumstances under which the office buildings in articles III, IV and V were simulated, the window’s G-value profoundly impacted the building’s cooling loads and thus, the borehole sizing.

Research objective 4: *Comparing a DGCS and a chiller-based cooling system in terms of thermal performance and energy use, and providing quantitative information about the energy-efficiency benefits of using the DGCSs.*

DGCSs require only a modest amount of electricity to drive the circulation pumps instead of the high electricity demand of mechanical refrigeration systems. The low electricity demand not only has obvious benefits in terms of energy costs, but it also makes the system remarkably resilient under power outage circumstances such as blackouts and brownouts. In this work, the most energy-efficient design is achieved when the annual ground loads are balanced by preheating the ventilation air only, without the use of a dry cooler, i.e. a supplementary system for balancing the ground loads. This design is possible when the building's cooling loads are minimized through a careful selection of building design parameters.

7.2. Future research

The following topics are worthy of future research.

- System design is a multi-objective optimization process. The present work primarily focuses on the design, thermal performance and energy performance of the DGCSs. Further work needs to be done to determine the importance of other factors on the design and performance of the DGCSs, e.g. initial costs, life-cycle costs, life-cycle energy use, environmental aspects, heatwaves and climate change.

- District cooling and air-cooled chillers are the cooling sources commonly used in Sweden. They are claimed to be superior to DGCSs on first cost and cooling capacity potential. A side-by-side comparison between these technologies regarding the initial costs, life-cycle costs, life-cycle energy use, and the primary energy demand can give useful insights into how to choose an appropriate cooling system.

- Increasing the supply temperature to the terminal units allows for raising the borehole outlet temperature and reducing the required borehole length. As the building peak cooling load is unchanged, reducing the borehole length results in an increase in the borehole heat transfer rate (W/m). In this scenario, unpredicted small changes in the building load could dramatically change the borehole outlet temperature. Thus, such a design makes the system sensitive to unpredictable circumstances, such as heatwaves. Likely circumstances should be simulated to investigate the possible risks.

- In line with the above, larger/more terminal units are required to compensate for the increased supply temperature to the terminal units. Analysis of cost trade-offs for different supply temperatures, the number of terminal units and the required borehole length may reveal the most cost-effective solution.

- Short-term storage, (such as storage tanks) buffer between the supply and demand processes and can smooth out the peaks. Investigating the use of the short-term storages on the design of the borehole systems is recommended.

References

[1] W. Graus, E. Blomen, E. Worrell, Global energy efficiency improvement in the long term: A demand- and supply-side perspective, Energy Efficiency. 4 (2011) 435–463. doi:10.1007/s12053-010-9097-z.

[2] IEA, The Future of Cooling: Opportunities for energy-efficient air conditioning, International Energy Agency (IEA), Paris, France, 2018. https://www.iea.org/reports/the-future-of-cooling.

[3] M. Santamouris, Cooling the buildings – past, present and future, Energy and Buildings. 128 (2016) 617–638. doi:10.1016/j.enbuild.2016.07.034.

[4] L.W. Davis, P.J. Gertler, Contribution of air conditioning adoption to future energy use under global warming, Proceedings of the National Academy of Sciences of the United States of America. 112 (2015) 5962–5967. doi:10.1073/pnas.1423558112.

[5] ASHRAE, ASHRAE Handbook - HVAC Systems and Equipment, American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta. (2012).

[6] R. Li, T. Yoshidomi, R. Ooka, B.W. Olesen, Field evaluation of performance of radiant heating/cooling ceiling panel system, Energy and Buildings. 86 (2015) 58–65. doi:10.1016/j.enbuild.2014.09.070.

[7] K.-N. Rhee, B.W. Olesen, K.W. Kim, Ten questions about radiant heating and cooling systems, Building and Environment. 112 (2017) 367–381. doi:10.1016/j.buildenv.2016.11.030.

[8] E.M. Saber, K.W. Tham, H. Leibundgut, A review of high temperature cooling systems in tropical buildings, Building and Environment. 96 (2016) 237–249. doi:http://dx.doi.org/10.1016/j.buildenv.2015.11.029.

[9] B.M. Shukuya, M. Saito, K. Isawa, T. Iwamatsu, H. Asada, ECBCS Annex 49: Low Energy Systems for High-Performance Building and Communities., IEA, 2009. http://www.ecbcs.org/Data/publications/EBC_Annex_49_HBE.pdf.

[10] T. Arghand, Z.D. Bolashikov, Z. Pastuszka, R. Kosonen, I. Aho, J. Kaczmarczyk, A.K. Melikov, Individually controlled localized chilled beam in conjunction with chilled ceiling: Part 2 – Human response, in: 14th International Conference on Indoor Air Quality and Climate, Ghent, Belgium, 2016. doi:978-0-9846855-5-4.

[11] T. Arghand, Z.D. Bolashikov, R. Kosonen, I. Aho, A.K. Melikov, Individually controlled localized chilled beam in conjunction with chilled ceiling: Part 1 – Physical Environment, in: 14th International Conference on Indoor Air Quality and Climate, 2016. doi:978-0-9846855-5-4.

[12] D. Schmidt, Low exergy systems for high-performance buildings and communities, Energy and Buildings. 41 (2009) 331–336. doi:10.1016/j.enbuild.2008.10.005.

[13] Y. Jiang, X. Liu, L. Zhang, T. Zhang, High Temperature Cooling and Low Temperature Heating in Buildings of EBC Annex 59, Energy Procedia. 78 (2015) 2433–2438. doi:10.1016/j.egypro.2015.11.222.

[14] X. Liu, T. Zhang, H. Tang, Y. Jiang, IEA EBC Annex 59: High temperature cooling and low temperature heating in buildings, Energy and Buildings. 145 (2017) 267–275. doi:10.1016/j.enbuild.2017.04.036.

[15] O.B. Kazanci, Low Temperature Heating and High Temperature Cooling in Buildings, DTU Civil Engineering, 2016. https://core.ac.uk/download/pdf/84000719.pdf.

[16] ASHRAE, ASHRAE Handbook - HVAC Applications, American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta. (2015).

[17] A.D. Chiasson, Geothermal Heat Pump and Heat, ASME Press and John Wiley & Sons, Ltd, West Sussex, United Kingdom, 2016.

[18] C.O. Popiel, J. Wojtkowiak, B. Biernacka, Measurements of temperature distribution in ground, Experimental Thermal and Fluid Science. 25 (2001) 301–309. doi:10.1016/S0894-1777(01)00078-4.

[19] B. Rosén, A. Gabrielsson, J.A.N. Fallsvik, G. Hellström, G. Nilsson, System för värme och kyla ur mark: en nulägesbeskrivning, Swedish Geotecnical Institute (SGI), Linköping, Sweden, 2001.

[20] K. Rafferty, Direct-use temperature requirements: A few rules of thumb, Geo-Heat Center Quarterly Bulletin, Oregon Institute of Technology: Klamath Falls, Oregon, USA, 2004.

[21] X. Cao, Y. Yuan, L. Sun, B. Lei, N. Yu, X. Yang, Restoration performance of vertical ground heat exchanger with various intermittent ratios, Geothermics. 54 (2015) 115–121. doi:10.1016/j.geothermics.2014.12.005.

[22] L. Liu, Z. Yu, H. Zhang, H. Yang, Performance improvements of a ground sink direct cooling system under intermittent operations, Energy and Buildings. 116 (2016) 403–410. doi:https://doi.org/10.1016/j.enbuild.2016.01.032.

[23] ASHRAE, Geothermal Energy, in: ASHRAE Handbook: HVAC Applications, American Society of Heating, Refrigerating and Air-conditioning Engineers, Atlanta, USA, 2015: pp. 34.1-34.43.

[24] P.J. Lienau, B.C. Lunis, Geothermal direct use engineering and design guidebook, Oregon Institute of Technology, Oregon, USA, 1989.

[25] J.W. Lund, D.H. Freeston, T.L. Boyd, Direct application of geothermal energy: 2005 Worldwide review, Geothermics. 34 (2005) 691–727. doi:10.1016/j.geothermics.2005.09.003.

[26] J. Gao, A. Li, X. Xu, W. Gang, T. Yan, Ground heat exchangers: Applications, technology integration and potentials for zero energy buildings, Renewable Energy. 128 (2018) 337–349. doi:10.1016/j.renene.2018.05.089.

[27] C. Matsuda, Y. Mino, Study on power-saving effects in direct-use of geothermal energy for datacenter cooling systems, in: International Telecommunications Energy Conference, IEEE, Austin, Texas, USA, 2016: pp. 1–6. doi:10.1109/INTLEC.2016.7749032.

[28] W. Lyu, X. Li, B. Wang, W. Shi, Energy saving potential of fresh air pre-handling system using shallow geothermal energy, Energy and Buildings. 185 (2019) 39–48. doi:10.1016/j.enbuild.2018.12.037.

[29] U. Eicker, C. Vorschulze, Potential of geothermal heat exchangers for office building climatisation, Renewable Energy. 34 (2009) 1126–1133. doi:10.1016/j.renene.2008.06.019.

[30] O. Andersson, S. Gehlin, State-of-the-Art: Sweden Quality Management in Design, Construction and Operation of Borehole Systems 2018, Svenskt Geoenegi Centrum, Lund, Sweden, 2018. http://media.geoenergicentrum.se/2018/06/Andersson_Gehlin_2018_State-of-the-Art-report-Sweden-for-IEA-ECES-Annex-27.pdf.

[31] H.Ö. Paksoy, Thermal energy storage for sustainable energy consumption: fundamentals, case studies and design, Springer Science & Business Media, Dordrecht, Netherlands, 2007.

[32] BELOK, Astronomihuset , Lunds Universitet, The client group for premises (BELOK), Gothenburg, Sweden, 2006. http://www.belok.se/docs/goda_exempel/astronomi_17_jan.pdf.

[33] J. Olsson, Anläggning i karlstad (Facility in Karlstad), GeoEnergi. 1 (2014) 12. http://media.geoenergicentrum.se/2014/05/SvGeo1_14_webb.pdf.

[34] J.D. Spitler, S. Gehlin, Measured Performance of a Mixed-Use Commercial-Building Ground Source Heat Pump System in Sweden, Energies. 12 (2019). doi:10.3390/en12102020.

[35] Z. Li, W. Zhu, T. Bai, M. Zheng, Experimental study of a ground sink direct cooling system in cold areas, Energy and Buildings. 41 (2009) 1233–1237. doi:10.1016/j.enbuild.2009.07.020.

[36] P. Filipsson, A. Trüschel, J. Gräslund, J. Dalenbäck, Performance evaluation of a direct ground-coupled self-regulating active chilled beam system, Energy and Buildings. 209 (2020). doi:10.1016/j.enbuild.2019.109691.

[37] H. Liu, H. Zhang, Performance Evaluation of Ground Heating and Cooling Systems-Long-term performance measurements of two case buildings, Department of Building and Environmental Technology, Lund University, Lund, Sweden, 2020.

[38] J. Kurnitski, nZEB office building Ympäristötalo in Helsinki, Finland, The REHVA European HVAC Journal. 49 (2012) 44–49. https://www.rehva.eu/rehva-journal/chapter/nzeb-office-building-ympaeristoetalo-in-helsinki-finland.

[39] H. Pieskä, A. Ploskić, Q. Wang, Design requirements for condensation-free operation of high-temperature cooling systems in mediterranean climate, Building and Environment. 185 (2020). doi:10.1016/j.buildenv.2020.107273.

[40] ASHRAE, Radiant heating and cooling, in: ASHRAE Handbook—HVAC Systems and Equipment, American Society of Heating, Refrigerating and Air-conditioning Engineers, Atlanta, USA, 2016: pp. 6.1-6.21.

[41] J. Babiak, B.W. Olesen, D. Petras, Low temperature heating and high temperature cooling, Federation of European Heating, Ventilation and Air Conditioning Associations (REHVA), Brussels, Belgium, 2013.

[42] J. Romaní, G. Pérez, A. de Gracia, Experimental evaluation of a cooling radiant wall coupled to a ground heat exchanger, Energy and Buildings. 129 (2016) 484–490. doi:10.1016/j.enbuild.2016.08.028.

[43] J. Romaní, L.F. Cabeza, P. Gabriel, A. Laura, A. De Gracia, Experimental testing of cooling internal loads with a radiant wall, Renewable Energy. 116 (2018) 1–8. doi:10.1016/j.renene.2017.09.051.

[44] A. Li, X. Xu, Y. Sun, A study on pipe-embedded wall integrated with ground source-coupled heat exchanger for enhanced building energy efficiency in diverse climate regions, Energy and Buildings. 121 (2016) 139–151. doi:10.1016/j.enbuild.2016.04.005.

[45] S. Javed, I.R. Ørnes, M. Myrup, T.H. Dokka, Design optimization of the borehole system for a plus-Energy kindergarten in Oslo, Norway, Architectural Engineering and Design Management. 15 (2019) 181–195. doi:10.1080/17452007.2018.1555088.

[46] D. Pahud, M. Belliardi, P. Caputo, Geocooling potential of borehole heat exchangers' systems applied to low energy office buildings, Renewable Energy. 45 (2012) 197–204. doi:10.1016/j.renene.2012.03.008.

[47] J. Liu, X. Xie, F. Qin, S. Song, D. Lv, A case study of ground source direct cooling system integrated with water storage tank system, Building Simulation. 9 (2016) 659–668. doi:10.1007/s12273-016-0297-0.

[48] S. Javed, Design of ground source heat pump systems: Thermal modelling and evaluation of boreholes, Chalmers University of Technology, Gothenburg, Sweden, 2010. http://publications.lib.chalmers.se/records/fulltext/122101/122101.pdf.

[49] Uponor, Build on Uponor with Varicool Carbon, Uponor Corporation, Vantaa, Finland, 2017. https://www.uponor.com/products/ceiling-heating-and-cooling/varicool-carbon-a-and-s-cooling-panel.

[50] A. Simone, J. Babiak, M. Bullo, G. Landkilde, B.W. Olesen, Operative Temperature for Control of Radiant Surface Heating and Cooling Systems, in: Clima 2007 WellBeing Indoors, FINVAC, Helsinki, Finland, 2007: pp. 233–237.

[51] C. Benton, F. Bauman, U. Fountain, M., A Field Measurement System for the Study of Thermal Comfort, ASHRARE Transaction. 96 (1990) 623–633.

[52] EQUA Simulation Technology Group, User Manual- IDA Indoor Climate and Energy version 4.8, EQUA Simulation Technology Group, Stockholm, Sweden, 2018. http://www.equaonline.com/iceuser/pdf/ICE48GettingStartedEng.pdf.

[53] S. Moosberger, Report: IDA ICE CIBSE-Validation: test of IDA Indoor Climate and Energy version 4.0 according to CIBSE TM33, issue 3, Lucerne University of Applied Sciences and Arts, Luzern, Switzerland, 2007. http://www.equaonline.com/iceuser/validation/ICE-Validation-CIBSE_TM33.pdf.

[54] EQUA Simulation Technology Group, Technical report: Validation of IDA Indoor Climate and Energy 4.0 build 4 with respect to ANSI/ASHRAE Standard 140-2004, EQUA Simulation Technology Group, Stockholm, Sweden, 2010.

[55] S. Kropf, G. Zweifel, Validation of the Building Simulation Program IDA-ICE According to CEN 13791 "Thermal Performance of Buildings - Calculation of Internal Temperatures of a Room in Summer Without Mechanical Cooling - General Criteria and Validation Procedures“, Lucerne University of Applied Sciences and Arts, Luzern, Switzerland, 2001.

[56] EQUA Simulation Technology Group, User Guide: Borehole 1.0, EQUA Simulation Technology Group, Stockholm, Sweden, 2014. http://www.equaonline.com/iceuser/pdf/UserGuideBoreholes.pdf.

[57] L. Eriksson, P. Skogqvist, Description of the IDA ICE borehole model, Internal Report,

EQUA Simulation Technology Group, Stockholm, Sweden, 2017.

[58] P. Eskilson, Thermal analysis of heat extraction boreholes. PhD thesis, Department of Mathematical Physics, University of Lund, Lund, Sweden, 1987.

[59] J.D. Spitler, S. Javed, R.K. Ramstad, Natural convection in groundwater-filled boreholes used as ground heat exchangers, Applied Energy. 164 (2016) 352–365. doi:10.1016/j.apenergy.2015.11.041.

[60] S.-H. Cho, M. Zaheer-uddin, Temperature regulation of radiant floor heating systems using two-parameter on-off control: An experimental study, ASHRAE Transactions. 103 (1997) 966–980.

[61] ASHRAE, Fundamentals of control, in: ASHRAE Handbook of Fundamentals, American Society of Heating, Refrigerating and Air-conditioning Engineers, Atlanta, USA, 2017: pp. 7.1-7.22.

[62] C.S. Kang, C.H. Hyun, M. Park, Fuzzy logic-based advanced on-off control for thermal comfort in residential buildings, Applied Energy. 155 (2015) 270–283. doi:10.1016/j.apenergy.2015.05.119.

[63] H. Tang, P. Raftery, X. Liu, S. Schiavon, J. Woolley, F.S. Bauman, Performance analysis of pulsed flow control method for radiant slab system, Building and Environment. 127 (2018) 107–119. doi:10.1016/j.buildenv.2017.11.004.

[64] A. Maccarini, G. Hultmark, N.C. Bergsøe, K. Rupnik, A. Afshari, Field study of a self-regulating active beam system for simultaneous heating and cooling of office buildings, Energy and Buildings. 224 (2020). doi:10.1016/j.enbuild.2020.110223.

[65] P. Filipsson, Self-Regulating Active Chilled Beams, Chalmers University of Technology, Gothenburg, Sweden, 2020. https://research.chalmers.se/publication/516194/file/516194_Fulltext.pdf.

[66] R. Kosonen, J. Penttinen, 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 (2017) 256–271. doi:10.1177/1420326X16683236.

[67] A. Maccarini, A two-pipe system for simultaneous heating and cooling of office buildings, Aalborg University Press, Aalborg, Denmark, 2017. https://www.researchgate.net/profile/Alessandro_Maccarini/publication/321670643_A_two-pipe_system_for_simultaneous_heating_and_cooling_of_office_buildings/links/5a2a57fc0f7e9b63e5356bc9/A-two-pipe-system-for-simultaneous-heating-and-cooling-of-office-buil.

[68] A. Maccarini, M. Wetter, A. Afshari, G. Hultmark, N.C. Bergsøe, A. Vorre, Energy saving potential of a two-pipe system for simultaneous heating and cooling of office buildings, Energy and Buildings. 134 (2016) 234–247. doi:10.1016/j.enbuild.2016.10.051.

[69] T. Arghand, J.O. Dalenbäck, A. Trüschel, S. Javed, Some aspects of controlling radiant and convective cooling systems, in: 13th REHVA World Congress, E3S Web of Conferences, Bucharest, Romania, 2019. doi:10.1051/e3sconf/201911105008.

[70] T. Arghand, A. Trüschel, J.-O. Dalenbäck, S. Javed, Dynamic thermal performance and

controllability of dry fan-coil unit, in: 9th Cold Climate HVAC Conference, Springer, Kiruna, Sweden, 2018: pp. 351–361. doi:10.1007/978-3-030-00662-4.

[71] W. Altmann, Practical process control for engineers and technicians, Newnes, Burlington, MA, USA, 2005.

[72] ISO, ISO 11855-2: Building environment design – Design, dimensioning, installation and control of embedded radiant heating and cooling systems – Part 2: Determination of the design heating and cooling capacity, International Organization for Standardization, Brussels, Belgium, 2015.

[73] J. Feng, Design and control of hydronic radiant cooling systems, University of California, Berkeley, 2014. http://escholarship.org/uc/item/6qc4p0fr Preferred.

[74] S.A. Mumma, Ceiling panel cooling systems, ASHRAE Journal. 43 (2001) 28.

[75] B. Ning, S. Schiavon, F.S. Bauman, A novel classification scheme for design and control of radiant system based on thermal response time, Energy and Buildings. 137 (2017) 38–45. doi:10.1016/j.enbuild.2016.12.013.

[76] J.D. Spitler, GLHEPRO-A design tool for commercial building ground loop heat exchangers, in: Proceeedings for the Fourth International Heat Pumps in Cold Climates Conference, Aylmer, Québec, 2000: pp. 1–15. http://www.hvac.okstate.edu/sites/default/files/pubs/papers/2000/08-HPCC_GLHEPRO.pdf.

[77] C. Yavuzturk, J.D. Spitler, Short time step response factor model for vertical ground loop heat exchangers, ASHRAE Transactions. 105 (1999) 475–485.

[78] J.R. Cullin, J.D. Spitler, A computationally efficient hybrid time step methodology for simulation of ground heat exchangers, Geothermics. 40 (2011) 144–156. doi:10.1016/j.geothermics.2011.01.001.

[79] J.D. Spitler, J.R. Cullin, Misconceptions Regarding Design of Ground-source Heat Pump Systems, in: World Renewable Energy Congress, Glasgow, Scotland, 2008.

[80] J. (Dove) Feng, S. Schiavon, F. Bauman, Cooling load differences between radiant and air systems, Energy and Buildings. 65 (2013) 310–321. doi:10.1016/j.enbuild.2013.06.009.

[81] J. Woolley, S. Schiavon, F. Bauman, P. Raftery, J. Pantelic, Side-by-side laboratory comparison of space heat extraction rates and thermal energy use for radiant and all-air systems, Energy and Buildings. 176 (2018) 139–150. doi:10.1016/j.enbuild.2018.06.018.

[82] J. (Dove) Feng, F. Bauman, S. Schiavon, Experimental comparison of zone cooling load between radiant and air systems, Energy and Buildings. 84 (2014) 152–159. doi:10.1016/j.enbuild.2014.07.080.

[83] J. Woolley, S. Schiavon, F. Bauman, P. Raftery, Side-by-side laboratory comparison of radiant and all-air cooling: How natural ventilation cooling and heat gain characteristics impact space heat extraction rates and daily thermal energy use, Energy and Buildings. 200 (2019) 68–85. doi:10.1016/j.enbuild.2019.07.020.

[84] J. Woolley, A multi-method investigation into design and control of radiant cooling and heating systems, University of California, Berkeley, USA, 2020.

https://escholarship.org/uc/item/98384265.

[85] L. Berglund, R. Rascati, M.L. Markel, Radiant heating and control for comfort during transient conditions, ASHRAE Transactions. 88 (1982) 765–775.

[86] D.O. Rijksen, C.J. Wisse, A.W.M. van Schijndel, Reducing peak requirements for cooling by using thermally activated building systems, Energy and Buildings. 42 (2010) 298–304. doi:http://dx.doi.org/10.1016/j.enbuild.2009.09.007.

[87] B.W. Olesen, K. Sommer, B. Duchting, Control of slab heating and cooling systems studied by dynamic computer simulations, ASHRAE Transactions. 108 (2002) 698.

[88] ISO, ISO 11855-6: Building environment design - Design, dimensioning, installation and control of embedded radiant heating and cooling systems - Part 6: Control, International Organization for Standardization, Brussels, Belgium, 2012.

[89] B.W. Olesen, Thermo Active Building Systems: Using Building Mass To Heat and Cool, ASHRAE Journal. 54 (2012) 44–52.

[90] CEN, EN 15377-3- Heating systems in buildings – Design of embedded water based surface heating and cooling systems – Part 3: Optimizing for use of renewable energy sources, European Committee for Standardization, Brussels, Belgium, 2007.

[91] ISO, ISO 11855-1: Building environment design - Design, dimensioning, installation and control of embedded radiant heating and cooling systems - Part 1: definition, symbols, and comfort area, International Organization for Standardization, Brussels, Belgium, 2018.

[92] CEN, EN 15251- Indoor environmental input parameters for design and assessment of energy performance of buildings addressing indoor air quality, thermal environment, lighting and acoustics, European Committee for Standardization, Brussels, Belgium, 2007.

[93] ASHRAE, Nonresidential Cooling and Heating, in: ASHRAE Handbook of Fundamentals, American Society of Heating, Refrigerating and Air-conditioning Engineers, Atlanta, USA, 2017: pp. 18.1-18.66.

[94] BELOK, Innemiljökrav för lokalbyggnader, The client group for premises (BELOK), Gothenburg, Sweden, 2015. http://belok.se/.

[95] E. Bourdakis, O.B. Kazanci, B.W. Olesen, Load calculations of radiant cooling systems for sizing the plant, Energy Procedia. 78 (2015) 2639–2644. doi:10.1016/j.egypro.2015.11.333.

[96] M. Krzaczek, J. Florczuk, J. Tejchman, Improved energy management technique in pipe-embedded wall heating/cooling system in residential buildings, Applied Energy. 254 (2019) 113711. doi:10.1016/j.apenergy.2019.113711.

[97] A. Lipczynska, J. Kaczmarczyk, A.K. Melikov, Thermal environment and air quality in office with personalized ventilation combined with chilled ceiling, Building and Environment. 92 (2015) 603–614. doi:10.1016/j.buildenv.2015.05.035.

[98] D.G. Leo Samuel, S.M.S. Nagendra, M.P. Maiya, Parametric analysis on the thermal comfort of a cooling tower based thermally activated building system in tropical climate – An experimental study, Applied Thermal Engineering. 138 (2018) 325–335. doi:10.1016/j.applthermaleng.2018.04.077.

[99] C. Karmann, F. Bauman, P. Raftery, S. Schiavon, M. Koupriyanov, Effect of acoustical clouds coverage and air movement on radiant chilled ceiling cooling capacity, Energy and Buildings. 158 (2018) 939–949. doi:10.1016/j.enbuild.2017.10.046.

[100] L. Helsen, Geothermally activated building structures, in: Advances in Ground-Source Heat Pump Systems, Woodhead Publishing, 2016: pp. 423–452. doi:10.1016/B978-0-08-100311-4.00015-7.

[101] M. Sourbron, L. Helsen, Slow Thermally Activated Building System and fast Air Handling Unit join forces through the use of Model based Predictive Control, in: Proceedings of the CLIMA 2013 Congress, Czech Society of Environmental Engineering (STP), Prague, Czech Republic, 2013: pp. 16–19.

[102] C. Zhang, P.K. Heiselberg, M. Pomianowski, T. Yu, R.L. Jensen, Experimental study of diffuse ceiling ventilation coupled with a thermally activated building construction in an office room, Energy and Buildings. 105 (2015) 60–70. doi:10.1016/j.enbuild.2015.07.048.

[103] M. Deru, K. Field, D. Studer, K. Benne, B. Griffith, P. Torcellini, B. Liu, M. Halverson, D. Winiarski, M. Rosenberg, M. Yazdanian, J. Huang, D. Crawley, U.S. Department of Energy commercial reference building models of the national building stock, National Renewable Energy Laboratory, Colorado, USA, 2011. doi:NREL Report No. TP-5500-46861.

[104] EU Building Stock Observatory, EU Buildings Database, European Commission. (2016). https://ec.europa.eu/energy/eu-buildings-database_en.

[105] Boverket, Boverket´s mandatory provisions and general recommendations , BBR, National Board of Housing (Boverket), Kalskrona, Sweden, 2018. https://www.boverket.se/en/start/publications/publications/2019/boverkets-building-regulations--mandatory-provisions-and-general-recommendations-bbr/.

[106] Y. Yildiz, K. Korkmaz, T. Göksal özbalta, Z. Durmus Arsan, An approach for developing sensitive design parameter guidelines to reduce the energy requirements of low-rise apartment buildings, Applied Energy. 93 (2012) 337–347. doi:10.1016/j.apenergy.2011.12.048.

[107] T. Østergård, R.L. Jensen, S.E. Maagaard, Early Building Design: Informed decision-making by exploring multidimensional design space using sensitivity analysis, Energy and Buildings. 142 (2017) 8–22. doi:10.1016/j.enbuild.2017.02.059.

[108] Swedish Work Environment Authority, Workplace design- Provisions of the Swedish Work Environment Authority on workplace design (AFS 2009:2), Arbetsmiljöverket, Stockholm, Sweden, 2009. https://www.av.se/globalassets/filer/publikationer/foreskrifter/engelska/workplace-design-provisions-afs2009-2.pdf.

[109] VDI, VDI 4640- PART 2: Thermal use of the underground - Ground source heat pump systems, The Association of German Engineers (VDI), Dusseldorf, Germany, 2019.

[110] H. Liu, H. Zhang, S. Javed, Long-Term Performance Measurement and Analysis, Energies. 13 (2020) 1–30. doi:10.3390/en13174527.

[111] F. Bonakdar, A.S. Kalagasidis, K. Mahapatra, The implications of climate zones on the cost-optimal level and cost-effectiveness of building envelope energy renovation and

space heat demand reduction, Buildings. 7 (2017). doi:10.3390/buildings7020039.

[112] T. Niemelä, R. Kosonen, J. Jokisalo, Cost-effectiveness of energy performance renovation measures in Finnish brick apartment buildings, Energy and Buildings. 137 (2017) 60–75. doi:10.1016/j.enbuild.2016.12.031.

[113] J. Rose, K.E. Thomsen, Energy saving potential in retrofitting of non-residential buildings in Denmark, Energy Procedia. 78 (2015) 1009–1014. doi:10.1016/j.egypro.2015.11.047.

[114] T. Arghand, S. Javed, A. Trüschel, J.O. Dalenbäck, Cooling of office buildings in cold climates using direct ground-coupled active chilled beams, Renewable Energy. 164 (2021) 122–132. doi:10.1016/j.renene.2020.09.066.

[115] T. Arghand, S. Javed, A. Trüschel, J.-O.O. Dalenbäck, Influence of system operation on the design and performance of a direct groundcoupled cooling system, Energy and Buildings. 234 (2021) 110709. doi:10.1016/j.enbuild.2020.110709.

[116] T. Arghand, S. Javed, A. Trüschel, J. Dalenbäck, A comparative study on borehole heat exchanger size for direct ground coupled cooling systems using active chilled beams and TABS, Energy and Buildings. (2021).