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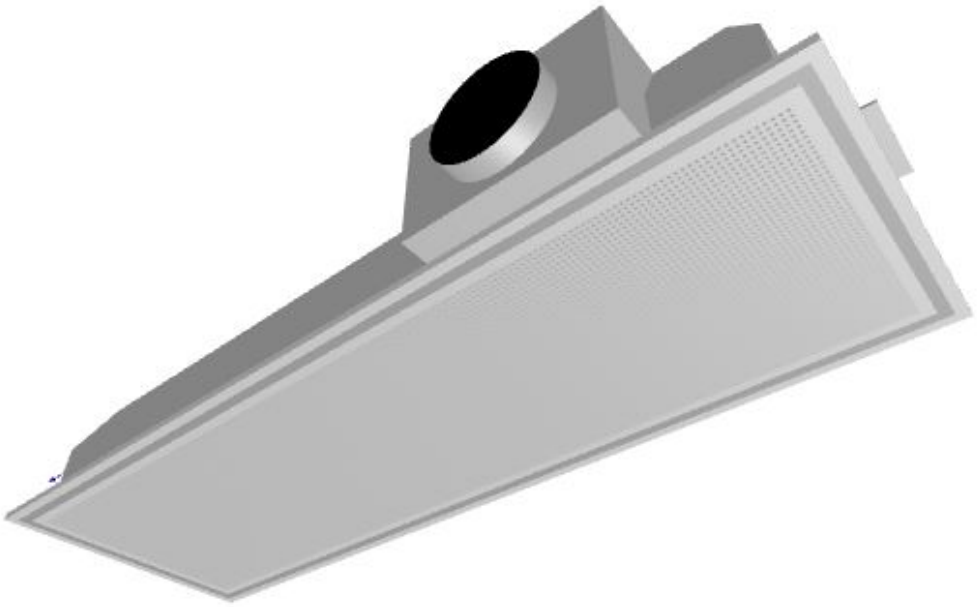

Design of Active Chilled Beam systems

Detailed economic analyses for different design temperatures and airflows

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CHALMERS UNIVERSITY OF TECHNOLOGY

Gothenburg, Sweden 2023

www.chalmers.se

Master's thesis 2023

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Cover: 3D model of active chilled beam manufactured by SWEGON.

Typeset in LaTeX
Printed by Chalmers Reproservice
Gothenburg, Sweden 2023

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Abstract

The predominant space-cooling system prevailing in Swedish commercial buildings today is the variable air volume (VAV) system. The system efficiently modulates the supplied air volume to meet changing cooling demands. An alternative cooling system is active chilled beams (ACB). ACB is a ceiling-mounted convection unit that decouples ventilation from thermal conditioning facilitating a potential for energy and cost savings.

This thesis aims to compare the energy and life-cycle costs (LCC) between the VAV and ACB systems, highlighting the impact of the cooling system choice on building energy demand and energy source design. The study grounds its investigations on the energy simulations of a case building located in Gothenburg, Sweden. The building is a six-story lightweight commercial building constituted primarily of open office landscapes. The energy is sourced from a combination of district energy and borehole heat-exchanger (BHE).

The study concludes that ACB systems can be designed at costs commensurate to comparable conventional VAV systems due to the savings attained from a smaller air-handling unit and ductwork, attributed to reduced airflows, exceeding the investment cost of ACB units and accessory equipment. Although the study can confirm energy savings achieved through reduced airflows, it could not validate other sources of energy savings. It was observed that VAV consumes less total energy but requires higher total power compared to ACB. Translated into annual costs, the higher power consumption of the VAV pertains to larger power costs which, combined with larger fan energy, results in higher operational expenses for the VAV system. Maintenance costs cause a further divergence between the systems, in favor of the ACB.

Investigating the different design parameters found that increasing the airflow of ACB systems generally leads to more balanced building loads. Contrarily, increasing supply air temperature cause an opposite effect on the specific component energies and the balance of building loads. Modulation of air temperature should conclusively be accompanied by proportional alterations to airflow rate to reduce the impact on investment costs; prioritizing a lower number of units over minimum airflow.

Keywords: space-cooling, high-efficiency, free cooling, active chilled beam, variable air volume, borehole, energy, investment cost, life-cycle cost

Acknowledgements

The thesis was conducted at the Bengt Dahlgren office in Gothenburg, and I would like to express my utmost gratitude to my supervisor, Taha Arghand, for the invaluable guidance and feedback throughout the process. I would like to extend my sincere gratitude to all the colleagues who generously devoted time to assist me with specific questions during the course of this work. I would like to express my appreciation to the exceptional individuals, with a special thank you to Carl-Ola Danielsson, at SWEGON for their support in providing insightful information and supplying the data utilized in this thesis. I would like to thank my examiner, Jan-Olof Dalenbäck, for the confidence bestowed upon me and my work. Lastly, I am grateful to my family and friends for their unwavering support and encouragement throughout my academic journey.

Andreas Fellegvari, Gothenburg, June 2023

List of Acronyms

Below is the list of acronyms that have been used throughout this thesis listed in alphabetical order:

ACB	Active Chilled Beam
AHU	Air Handling Unit
BHE	Borehole Heat-exchanger
COP	Coefficient of Performance
DGC	Direct-ground Cooling
GHE	Ground Heat-exchanger
GSHP	Ground Source Heat Pump
IR	Induction-ratio
LCC	Life cycle costs
NPV	Net Present Value
RH	Relative Humidity
VAT	Value-added Tax
VAV	Variable air Volume

Nomenclature

Below is the nomenclature of indices, sets, parameters, and variables that have been used throughout this thesis.

Indices

t	Index for time step

Parameters

CF	Cash flow
COP	Coefficient of performance
E	Energy
h_L	Latent heat
I_{vap}	Enthalpy of vaporization
NPV	Net pressent value
P	Power
Δp	Pressure differential
q	Ventilation flow rate
r	Discount rate
$\dot{V}$	Volume flow
Δx	Water content differential
η	Fan efficiency
ξ	Coverage factor
ρ	Density

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1
Introduction

This chapter declares the background and relevance of the study. The cooling systems of regard are introduced, the purpose and aim are defined, the research questions are formulated, the research methodology is developed, the limitations are outlined and the thesis structure is set forth.

1.1 Background

Conceding to the warming effects of climate change, developing economies, and growing urbanization, demand for global space cooling is rising out of necessity and desire. The European region has over the past four decades experienced close to a three-fold increase in cooling degree days[1] [1]. In conjunction with more extreme seasonal fluctuations and higher internal loads, the demand for mechanical cooling is growing at a rapid speed, accounting for an increase of 33 % between 2010-2018 [2]. Currently, economic growth drives an expanding possibility and desire for thermally regulated indoor environments. Developing cities, one driver of economic growth, further increases demand via more extreme temperatures due to heat-island effects[2] [3].

Constituting an estimated 60 % of cooled floor area in Europe, commercial buildings average a cooling demand twice that of residential buildings highlighting the significant influence of commercial buildings [4]. Moreover, the impact of commercial buildings is increasing. Over the last three decades, the portion of total energy consumption constituting space-cooling for commercial buildings reached 11.5 % up from 6 %. For comparison the same portion for residential buildings increased from about 2.5 % to 6 % [5].

[1] Occurrence and magnitude of days with average ambient air temperatures exceeding 18°C

[2] Higher ambient temperatures in urban environments due to thermal storage in the infrastructure

With the share of cooling in building's total energy estimated to more than double from the current 6 % to 14 % and the global energy for cooling in buildings projected to triple, reaching roughly 6 200 TWh, the world is destined for a cold crunch [5]. However, the extent and celerity of the inevitable growth of space-cooling can be mitigated and possibly curtailed by the effective employment of high-efficiency cooling systems and operative controls [3].

The most predominant space-cooling system prevailing in commercial buildings today is the variable air volume (VAV) system. VAV systems are HVAC systems that through a central air handling unit distribute conditioned outdoor air throughout the building. Terminal control boxes located in each zone regulate the supplied air volume at specific temperatures to achieve and uphold zone thermal comfort levels for various occupancy rates and heat loads. By continuously adjusting the heating and cooling supplied to the space, the VAV system can achieve great energy efficiency with the subsequent cost-savings. Utilizing unconditioned outdoor air for cooling whenever ambient temperatures fall below indoor temperatures, known as free cooling, provides additional energy savings for the system. However, VAV struggles in situations with unbalanced demands. Scenarios characterized by high cooling and low ventilation, or high ventilation and low cooling, often force the system to reheat the already-cooled air to prevent excessive cooling of the space. The process of reheat may account for as much as 20 % of energy consumption in extensive VAV installations, such as large office buildings [6].

An alternative system suggesting a potential for greater efficiency and operational cost savings is active chilled beams (ACB). ACBs are integrated HVAC convection units that decouple ventilation from thermal conditioning. A central air handling unit supplies constant high-pressure air, i.e. primary air, to the units. The primary air flows through a nozzle bank, creating a low-pressure zone above a high-temperature cooling coil. Room air is induced up through the coil where the air is cooled and mixed with the primary air before being diffused back into the room. Also called the entrainment effect, the induced air benefits the unit by enabling enhanced volumes of cool air at reduced air supplies yielding energy savings pertaining to air conditioning and fans.

Currently, no written comparisons of the two systems exist for operations in temperate climates. A reliable comparison between the two systems, considering both performance and life-cycle cost, would facilitate more optimal system selections based on prevailing building conditions and contractor requirements. This, in turn, would contribute to more cost and energy-efficient cooling solutions.

1.2 Purpose and aim

The purpose of this paper is to address the existing knowledge deficit in chilled beams' relative performance to traditional VAV systems in temperate climates. The differences in systems energy, investment costs, and life-cycle costs will be investigated and quantified on a comparative basis. Further, all systems will be analyzed in operation with two different energy sources whereby the effects on design will be evaluated and the pertaining costs will be assessed with regard to feasibility. Moreover, by sizing and optimizing the chilled beam systems with respect to various configurations of the ruling design parameters, the current conventional design and optimization procedures will be examined for any discordance.

The aim is to demonstrate how the choice of cooling system affects the overall energy demand of a building, how it affects the design of the cooling source, what optimization procedures can be implemented, and how economically feasible an ACB system is in comparison to VAV in temperate climates.

Additionally, the developed optimization strategies will be put forth in a comprehensive step-by-step procedure to enable further investigation and potential utilization.

1.3 Research questions

The following research questions have been formulated to resolve the prevailing uncertainties.

- What are the investment cost differences between the traditional VAV and the various ACB systems?
- What are the operational cost differences between the traditional VAV and the various ACB systems?
- What effects do different ACB system optimizations have on investment costs and overall building energy demand?
- What are the performance and cost differences between an ill-designed and a well-optimized ACB system?
- What effect does the ACB system design have on the cooling sources design?

1.4 Research methodology

The thesis will ground its work on a lengthy literature study and review whereby current design procedures for active chilled beam systems will constitute the framework for the design and optimization strategies investigated in this work. Each system will be evaluated in terms of annual energy consumption, including peak heating and cooling power, investment costs, and final life-cycle costs. System energies will be attained by simulations in the IDA ICE software. The life-cycle analysis will encompass all energy and power costs associated with the building and its energy source. In addition, the analysis will account for the costs of investment and general maintenance.

1.5 Scope and limitation

The study is constrained by the lone investigation of one case building. Thus the results are confined to the case buildings specific parameters. The building's thermal capacity, geometry, orientation, façade construction, operation, and occupancy all affect the final design and performance of the system. This implies that certain configurations of these parameters could render a building unsuitable for certain system applications or potentially yield greater benefits from alternative optimizations.

Likewise, the study is bound to temperate climate. Differences in ambient conditions, including solar insulation, and ground properties affect the potential for free cooling. Climates experiencing hotter annual temperatures may not attain conditions in which free cooling by means of direct-ground cooling becomes feasible due to higher undisturbed ground temperatures. Likewise, regions with higher humidity levels encounter increased difficulties with condensation and larger dehumidification demands.

Limitations in performing in-situ measurements of the case building preclude the validation of the simulation result. Consequently, the findings can only be reliably applied for comparative assessments of performance rather than the determination of the absolute system performance. Further limitations on the applicability of the results are imposed by the design of the systems. The systems are sized to ensure one singular measure of thermal comfort, room air temperature, as other aspects cannot be examined utilizing IDA ICE.

The thesis does not concern the design of VAV systems. The VAV system, used as a reference, was designed independently and prior to this thesis. The system is deemed complete and fully operational requiring no further modifications or optimizations. Subsequently, the theoretical aspects pertaining to VAV systems are kept concise.

The thesis does not originally encompass the design of the BHE systems. The systems employed in the study were designed adjunct to this thesis. Therefore the theoretical and methodological sections of this thesis concerning direct-ground cooling should be regarded as supplementary to the primary focus of this work and should not be relied upon for replication.

All selections made on the basis of cost are bounded by the costs available at that specific stage. Further, the accuracy of the life-cycle cost analysis is contingent upon the limited availability of cost information and thus relies extensively on cost estimations, particularly due to the rudimentary design of the ancillary systems that e.g. exclude detailed ductwork and control systems.

The study excludes optimization strategies pertaining to the particular designs of active chilled beam units, such as geometric shape and nozzle design. Additionally, it does not address the unit layout or the relative placement of the units with respect to the façade. Furthermore, a single unit is selected and employed for all spaces, potentially resulting in oversizing of certain zones. The exclusion of these factors collectively raises the likelihood of suboptimal and inefficient designs, affecting thermal comfort, initial system costs, overall energy demand, and ultimately the final life-cycle cost.

Lastly, the active chilled beams are exclusively utilized for cooling purposes, while heating is accomplished through the use of radiators located throughout each zone. Incorporating the chilled beams for both cooling and heating can yield advantages, particularly when combined with direct ground-coupled heating. This choice has implications for the final life-cycle cost, as it necessitates the inclusion of radiators and potentially results in higher energy and power consumption.

1.6 Thesis outline

This thesis is constituted of seven correlated chapters altogether answering the research questions. Each chapter is grounded in one of the procedural steps in the investigative study. The thesis is structured in the following manner:

Chapter 1 provides a comprehensive overview of the study, beginning with the rationale behind the research, the identified problem, and the specific inquiries being addressed. It outlines the purpose and objectives of the study, details the research methodology employed, and defines the scope and limitations of the investigation. Finally, the chapter provides an overview of the thesis structure.

Chapter 2 delves into the fundamental operations of the HVAC systems under investigation. It explains the current design parameters and system metrics that are relevant to the study. The chapter concludes with a literature review, summarizing the existing knowledge and research approaches pertaining to the investigated systems.

Chapter 3 introduces the case building that has been studied. It presents a summary of the relevant building metrics and provides a structured description of the study's methodology. This includes details on the simulation resources utilized, the system designs and optimization approaches, and the calculations involved in assessing the life-cycle cost.

Chapter 4 provides a detailed account of the system design process developed, offering an perspective on the steps and considerations involved. Furthermore, the finalized system designs for each case are showcased and discussed.

Chapter 5 presents the simulation results encompassing system energy and power consumption, the computed electricity usage, investment costs associated with the building and borehole systems, maintenance expenses, and the comprehensive assessment of life-cycle costs. The chapter concludes with a sensitivity analysis.

Chapter 6 constitutes a discussion of the overall findings obtained from the study, shedding light on potential errors and inadequacies that have been identified.

Chapter 7 presents the conclusions drawn from the research findings and offers suggestions for future research investigation.

2
Theoretical background

This chapter provides an overview of the cooling systems investigated in this thesis, outlining their general operation and prevailing design procedures. Additionally, it provides a brief description of energy sources and the factors that influence the final energy cost. Furthermore, a comprehensive literature review is conducted to explore and analyze current research papers that investigate and discuss the performance of these systems, both individually and in combination with other relevant systems.

2.1 Methodology

The thesis principally grounds its work on an extensive review of existing literature to establish a general foundation of knowledge and expectations, enabling a well-structured approach.

A comprehensive survey of theories, methodologies, and findings from various previous works formed the basis for developing an investigative framework. The literature review focused on several key aspects, including the design and performance of ACB systems in temperate climates, comparative analysis of ACB systems against traditional VAV systems, the design and performance of direct-ground cooling systems in temperate climates, and the potential synergies achieved by combining these systems.

The literature was gathered from two primary sources: the scholarly database Google Scholar and the Chalmers Library. To ensure the credibility of the thesis, each source was carefully evaluated for validity, potential bias, and relevance. Preference was given to peer-reviewed and institutional papers, while marketable and other non-reputable sources were only used for operational and factual information. In cases where specific information was not available elsewhere, personal communications were utilized.

The literature study followed a bell-curve pattern, starting with a limited and superficial exploration in the initial phase, followed by a more in-depth and comprehensive analysis in the latter half. As the thesis approached completion, the literature review gradually narrowed down to focus on the most relevant and significant papers.

2.2 Heating, ventilation, and air-conditioning

The heating, ventilation, and air-conditioning (HVAC) system refers to the different building systems managing the indoor environment by controlling air temperature, humidity, and purity. These variables together compose indoor air quality, of which thermal comfort is a predominant constituent. Thermal comfort may be described as the subjective satisfaction of each occupant with the thermal environment in the space. Due to the subjective nature of thermal comfort, many factors contribute to and influence the occupant's perception. The factors are metabolic rate, clothing insulation level, mean surface-radiant temperatures, air temperature, airspeed, and relative humidity. Regulating the variables of the indoor environment in a proper way, i.e. upholding conditions in accordance with regional building codes, is the primary function of an HVAC system.

Integrating the heating, ventilating, and air-conditioning functions into a centralized system offers the advantage of efficiently serving multiple spaces with similar space-conditioning requirements. The centralized HVAC system is generally established as a central air handling unit (AHU) situated outside the served zones. Central HVAC systems can be further categorized based on the mediums employed for thermal regulation in the serviced spaces, namely: all-air, air-water, and all-water systems.

2.2.1 Variable Air Volume

Variable air volume (VAV) systems are all-air HVAC systems that through a central air handling unit distribute conditioned ambient air throughout the building via fans and ductwork. Terminal control boxes located in each zone receive supply air at a constant temperature and velocity. Zone temperature sensors continuously monitor the space conditions and a damper in the terminal control box modulates the supplied air volume to achieve the desired cooling effect. This ensures thermal comfort is upheld during periods of varying occupancy rates and cooling loads. If a scenario would require the airflow to be reduced beyond the minimum airflow setpoint of the control box, a reheat component, in the form of a radiator or an integrated reheat coil, becomes necessary.

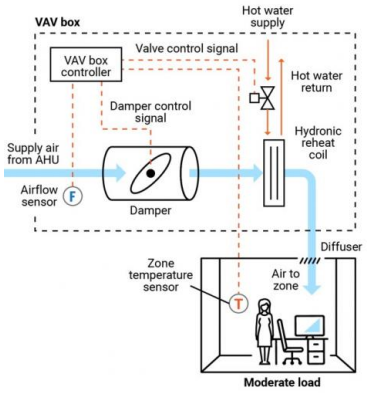


Figure 1. VAV terminal control box with integrated reheat principle image [7]

2.2.2 Active Chilled Beam

Active chilled beams (ACB) are integrated air-water convection units served by a central constant air volume (CAV) air handling unit. The unit is supplied with conditioned ambient air, primary air, at high pressure. This air is distributed and directed through a nozzle bank which discharges the air at high velocity into a mixing chamber creating a low-pressure zone above the cooling coil. This differential pressure facilitates the induction of room air, entrainment effect, up through the coil where the air is subsequently cooled. The induced air is then mixed with the primary air before being diffused back into the room via the coandă effects air distribution pattern.

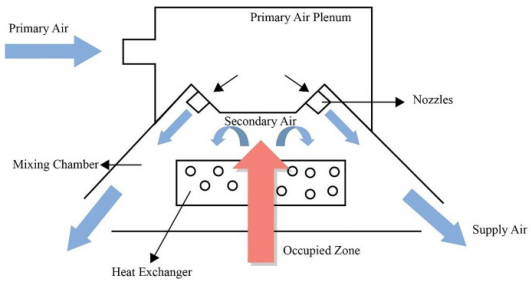


Figure 2. ACB principle image [8]

The ratio of induced air to primary air, referred to as the units induction ratio (IR), commonly ranges between 3-to-1 and 5-to-1 [9, 10]. IR benefits the unit by enabling large volumes of cool air at small air supplies, enhancing heat transfer efficiency [11]. Additionally, the air entrainment effect may greatly reduce the need for the energy-consuming reheat of the cooled air [10].

As prescribed by the REHVA guidebook the primary air ought to be supplied at temperatures reset by season; $18 - 20°C$ in the summer and $19 - 21°C$ in the winter [9]. Furthermore, to ensure adequate thermal comfort, the temperature difference between the supply air and the room air should not exceed $4,7°C$ [11].

The unit's utility of two thermal mediums—air and water— enables a mechanical decoupling of the ventilation load from the space load. Decoupled ventilation entails the opportunity of lower supply airflow rates, requiring only to account for latent heat loads and the minimum ventilation demand [12, 13]. The reduced airflow has reportedly achieved 17 % fan power reduction relative to conventional VAV and is argued to attain additional AHU-energy savings by reducing the volume of air in need of conditioning [14, 15, 16]. This indicates the importance of minimizing the primary airflow to achieve optimal performance [17, 18]. Furthermore, it suggests that the ideal operating conditions for ACB entails fulfillment of space-latent heat loads by minimum ventilation rate. The optimal conditions can be determined by a ratio of sensible cooling to latent cooling, which has been suggested to be approximately $0,7$ [19, 20].

The reduction in airflow entails further secondary effects relating to the air handling and distribution systems. Firstly, the lower airflow also lowers the duct resistance [6]. This implies additional benefits to fan energy due to the decreased system pressure. Furthermore, the reduced airflow allows for smaller AHU and may reduce the required ductwork by up to 50 % relative to conventional all-air systems [17, 21]. Moreover, the reduced duct dimensions result in reduced space requirement, enabling increases in floor-to-ceiling height of approximately 0,1 to 0,4 m for retrofits [10, 19]. For new constructions this instead entails lower floor-to-floor height which could potentiate the construction of e.g. six stories at the height typically required for five stories, offering substantial cost savings in construction.

The main source of sensible cooling is the unit-mounted coil which is supplied with high-temperature chilled water. The coils' cooling capacity is regulated by modulating the mass flow rate of the circulating water. To maximize thermal transfer, the flow rate should be selected to ensure turbulent flow during design conditions. Furthermore, the water temperature must be supplied above the room-air dew-point temperature to avoid condensation from forming on the surface of the coil [9, 14]. However, some studies suggest the potential for greater leniency, indicating that temperatures slightly below the dew point can be managed without condensation, and even lower temperatures, $1,5°C$, before water droplets form. Most commonly, chilled water is supplied within the range of $14 - 18°C$ [8, 9] although some applications may utilize supply temperatures of $\geq 20°C$ [22, 23]. The outlet water temperature is typically set $2 - 4°C$ higher than the inlet temperature [9]. The higher chilled water temperatures utilized for chilled beams may yield a $15 - 20$ % higher operational efficiency for a chiller dedicated to chilled beam cooling as opposed to a conventional system [10].

2.3 Energy-source

The selection of an energy source has significant effects on both economic and environmental aspects. Various factors influence the consumer cost of an energy source, many of which are inherent to the specific source itself. These factors include:

- Availability - specific resource availability governed by natural abundance and ease of exploitation
- Production - costs relating to extraction, production, and distribution
- Infrastructure - construction and maintenance of infrastructure necessary for extraction, production, and distribution
- Regulations and Subsidies - costs and benefits due from governmental and regulator bodies.

The availability of specific energy sources is determined by a) its natural abundance and b) its ease of accessibility. These factors together decide the initial feasibility of the energy source.

Cost for extraction and production of specific energy sources are influenced by the a) complexity of extraction, b) scale of production, c) conversion efficiency, and d) distribution network. These factors impact the specific demands placed on infrastructure.

Costs pertaining to infrastructure and its maintenance are dictated by the existence, extent, and efficiency of current infrastructure networks, the necessity and complexity of new infrastructure, and the requirement of infrastructure for utility at the end-user.

Rules and regulations established by governments and regulatory bodies impose specific requirements on the exploration, extraction, and production processes. These requirements append costs related to obtaining permits and complying with regulatory frameworks. Environmental regulations often pertain to the necessity of extraordinary infrastructure and processes.

Governmental financial support, incentives, and tax exceptions generally associated with renewable energy sources provide great cost benefits for the consumer.

2.3.1 Natural cooling

Natural cooling, or free cooling, refers to the process of cooling without the use of a mechanical refrigeration process or air conditioning system. Instead, it takes advantage of favorable ambient conditions to achieve space cooling either by air or water. The effective potential of free cooling is highly geographical. In Nordic climates, air temperatures typically remain below $14 - 18°C$ for $70 - 90$ % [24] of the year, while the soil and ground-water temperatures range between $2 - 9°C$ depending on latitude [25, 26].

Air-side free cooling leverages the use of cool ambient air to provide efficient space cooling for buildings. When the outdoor temperature falls below the desired indoor temperature, the mechanical cooling system is modulated to allow the entry and circulation of cool outdoor air within the building. Demonstratively more effective during cooler periods when the temperature differential between the outdoor and indoor environments is larger. During such periods, relying solely on the ambient air may be sufficient to meet all cooling needs.

Water-side free cooling leverages the use of cool ambient air, water, or ground to cool the circulating chilled water. Whenever the temperature of the cooling medium falls below the chilled water temperature, the outlet water is routed through an evaporative cooling tower, a conventional heat exchanger, or a ground heat exchanger, providing energy-efficient cooling.

In the case of ACB, increasing the water inlet temperature allows for the supply of higher outdoor air temperatures, thereby expanding the potential for free cooling. As a case example, raising the system inlet temperature from $15°C$ to $20°C$ can increase the free cooling duration from $2560\,h$ to $6200\,h$ [27].

2.3.2 Direct Ground Coupled Cooling

Direct-ground cooling (DGC) is a geothermal cooling source that exploits the near-constant subterranean soil and ground-water temperatures, undisturbed ground temperature, to provide consistent cooling to indoor spaces. The system circulates a heat transfer fluid through an array of ground heat-exchangers (GHE) whereby heat is dissipated into the surrounding soil and rock. The GHE system is coupled to the building's cooling system via a heat-exchanger. The method of geothermal cooling in utility with a heat pump has demonstrated savings in primary energy consumption of roughly $37\,\%$ compared to conventional air-to-water heat pumps [28].

2.3.2.1 Borehole heat-exchanger

A Borehole heat-exchanger (BHE) is a DGC system that consists of a network of single or double U-pipes inserted into vertical boreholes. The BHE system can be combined with a ground-source heat pump (GSHP) to enable the BHE to be effectively utilized for heating during winter months and further magnifies free cooling during the summer months. The borehole sizing is based on three main parameters: building loads, ground thermal properties, and borehole thermal resistance.

The borehole designs in this thesis accounted only for monthly energies and annual peak powers. Therefore, while not directly aligned with the primary focus of this work, the following descriptions of the design parameters serve as a short supplementary investigation into the subject matter.

Building cooling load: Defined as the ratio of the building's net cooling and heating loads to the sum of their absolute values. The load factor is a measure of the building-loads influence on the undisturbed ground temperature. The factor varies between -1 and +1 with either extreme representing a one-sided load pattern—performing only heating or cooling— that disturbs the ground temperature causing it to either rise or fall over years until an equilibrium is reached. This subsequently defines the system's long-term performance as the heating/cooling power is proportionally reduced over time. Preferably, the load factor is around zero entailing a balanced load ratio between cooling and heating leading to only small temperature fluctuations.

The required borehole heat transfer is determined by the building load intensity. As the depth of the boreholes is directly proportional to the heat transfer rate of the building, the aggregate borehole length is determined by the load intensity [29].

Ground thermal properties: The thermal signature of the ground is distinguished into three zones: Surface zone (from ground surface down to 1m depth), shallow zone (from 1 m to 10 m depth), and deep zone (below 10 m depth). At the depth of 10 m ground temperatures remain fairly stable over the year equaling the average annual air temperature. This temperature is called "undisturbed ground temperature" and is a determining factor of the potential thermal capacity of BHE systems and also constitutes the minimum achievable temperature in cooling applications.

The ground's ability to transfer heat, described as the ground's thermal conductivity, is determined by the ground's intrinsic properties relating to the mineralogy, soil size, gradation, and by environmental properties regarding soil moisture content and the undisturbed ground temperature.

Borehole thermal resistance: The borehole thermal resistance determines the thermal transfer rate between the circulating fluid and the ground surrounding the borehole. The resistances of the thermal transfer fluid, borehole casing, and backfill material influence the borehole's overall thermal resistance.

The choice of the thermal carrier fluid is mainly determined by the risk of freezing. If no such risk exists water may be used. If freezing would be of risk fluids mixed with ethanol or other non-freezing agents are used.

The backfill material's main purpose is to facilitate a proper heat transfer from the transfer fluid to the borehole casing. Apart from low thermal resistance, the material must fulfill the requirements of permeability, compression strength, and workability. The two types of backfill are grout and groundwater. In Scandinavia, groundwater is predominantly used as the groundwater level lies only a few meters below the surface.

The flow rate of the thermal transfer fluid affects the system's thermal capacity. Increasing the flow rate increases the thermal capacity. This is the case until the turbulent flow is achieved at which point further increases in flow rate cause negligible effects to the system's thermal capacity.

2.4 Literature Review

This section focuses on current literature concerning the performance of ACB systems compared to conventional VAV systems and the integration of ACB with BHE. The selected papers present relevant findings and highlight intriguing aspects of the studied systems.

2.4.1 Energy-saving potential of ACB vs. VAV

Latif et al. [30] conducted a comprehensive literature review of the active chilled beam technology, examining a final selection of 150 current articles, patents, and reports related to subjects including thermal comfort and building energy performance. The authors compiled an extensive list of reasons highlighting the energy-saving potential of ACBs compared to conventional all-air systems. These reasons will be investigated and validated within the scope of this thesis, aiming to provide concrete evidence for the following claims:

+ reductions in fan power due to decoupling of ventilation
+ higher chilled water temperatures yielding higher chiller efficiency
+ higher energy efficiency due to water's higher specific heat capacity

A newsletter published by TRANE, authored by Murphy. J, [31] provides a comprehensive examination of the advantages commonly attributed to ACB systems over traditional all-air systems. Although TRANE, as a supplier and manufacturer of HVAC systems, may hold a certain bias against ACB, the remarks are objectively accurate and serve the general discussion well.

The newsletter debated the relevance of the advantages associated with ACBs and their potential energy savings when compared to VAV systems. The energy savings are purportedly derived from three main effects: 1) decreased primary air flows, 2) higher chilled water temperatures, and 3) cooling at the zone level.

> 1) According to the author, when considering design conditions, a zone served by active chilled beams may exhibit a $60 - 70\,\%$ reduction in primary airflows compared to a conventional VAV system serving the same zone. However, the author emphasizes, design conditions do not accurately represent the annual operation characteristics of a VAV system. Specifically, VAV systems derive substantial benefits from a) reduced airflows during part load conditions, b) system load diversity, and c) unloading of the AHU supply fan.

a) In the case of an ACB, the unit relies on the primary airflow to induce room air for proper function. As such the airflow is typically constant regardless of load conditions. Contrarily, VAV systems regulate the airflow supplied to the zone in proportion to prevailing load conditions. As such, the system only rarely operates at full capacity. According to [32], a typical office building is on average occupied for $30-40\,\%$ of the time between 8 a.m. and 5 p.m., with individual workstations rarely being utilized for more than one-third of the core eight-hour business day. This supports the suggested advantage of variable airflow in potentially reducing fan energy consumption.

Declaring a minimum VAV terminal airflow of 30 % of design conditions, in accordance with ASHRAE 90.1 standard G3.2.3.13 [33], the author suggests that the VAV operates with lesser airflow when space loads are at and below roughly 65 % of design load. However, presuming 1) the building is suitable for ACBs—latent heat loads constitute less than 30 % of total cooling demand— and 2) the ACB system to be sized optimally—airflow equivalent to the minimum ventilation load— the scenario that VAV satisfies cooling loads up to 65 % of design the load with less airflow than the ACB becomes doubtful. For this statement to apply, the cooled space must exhibit high levels of occupancy, increasing the minimum ventilation airflow of the ACB, and resulting in a high rate of constant air volume. In this scenario, the VAV would definitely perform better during part-loads.

b) Load diversity in HVAC is the assumption that maximum demand of all zones will unlikely occur concurrently. This means a difference between the designed peak condition and the actual realized peak condition. As such the central supply fan in a multi-zone VAV system does not deliver peak air flows on a building-wide basis but rather on an individual zone level. This benefit does not affect the operation of ACB systems due to the system's utility of constant air volume. The effects of this include the possibility of smaller AHU and ductwork sizing than otherwise prescribed by the design conditions.

c) When the central supply fan reduces airflow during part-loads, called unloading, pressure loss through the air distribution system decrease. This results in an exponential reduction in fan energy during part-loads as reductions are yielded by both the lowered airflow and the pressure loss per equation (1).

$$P = \frac{\dot{V} * \Delta p}{\eta} \tag{1}$$

Where:

P	:	Fan power	[kW]
$\dot{V}$	:	Volume flow	[m^3/s]
Δp	:	Pressure differential	[Pa]
η	:	Fan efficiency	[%]

2) With reason to mitigate the risk of condensation ACBs are supplied chilled water at temperatures above dew-point. The hotter chilled-water temperature yields higher efficiency of the supplying chiller, measured as coefficient of performance (COP), compared to the lower water temperatures commonly supplied to central VAV air-handling units. This would entail energy saving on the part of the ACB system. However, the author notes that AHUs in an ACB system must dehumidify the primary air to offset space latent heat loads. Begot by the system's lower primary airflow rates, the AHU must deliver yet dryer air to compensate, which may require the dehumidifying component to be supplied chilled water cooler than the typical VAV system, counteracting the COP benefits.

Moreover, the author argues that the benefits of higher COP depend on the chiller plant configuration and that achieving the stated benefits requires two separate chiller plants. One supplying the AHU with sufficiently cold water for conditioning—cooling and dehumidifying— the ambient air and one supplying the ACBs with warmer chilled water benefiting from higher COP. However, as unmentioned, the same effect can be achieved by serving the ACBs through locally cooled water by means of e.g. water-to-air or water-to-water heat pumps. Moreover, the higher chilled water temperatures allow the use of cheaper and more sustainable energy sources further increasing the potential of cost savings.

3) By providing cooling on a local zone-level ACBs are believed to reduce the risk of over-cooling and the subsequent need to reheat, a common issue for many VAV systems. As sensible cooling loads decrease the ACB system reduces the water flow rate. In a scenario where space-sensible cooling loads drop below the point when the chiller-water valve is fully closed, the primary airflow risks providing more cooling than can be offset by heat loads resulting in over-cooling. To avoid over-cooling, heat must be added to the space by either a) if four-pipe ACB, open the hot-water valve, b) if two-pipe ACB, change water distribution to deliver hot water to all beams, or c) activate a separate heating system.

The same scenario for a VAV system follows a similar procedure. As cooling loads decrease, VAV terminals restrict the primary airflow supplied to the space. Upon reaching the minimum flow rate for the VAV terminal, reheat must be activated to avoid over-cooling the space.

Taking supply air temperature into account, whereby the ACBs cooling coil and utility of high-temperature chilled-water enables the supply higher primary air temperatures, a not-too-unlikely scenario where the ACBs constant airflow and the VAVs minimum airflow are commensurate, the ACB would inherently require slightly less reheat.

In summary, the discussed remarks indicate the validity of the benefits attributed to ACB, although the extent of these benefits may vary based on the specific system design. Overall, VAV systems demonstrate significant advantages in part-load conditions, while ACB systems primarily benefit from the potential of reduced airflows and high-temperature cooling.

2.4.2 Design and application of BHE

Arghand et al. [34] investigated the effects of thermal units on the sizing of BHE for DCG systems. It was concluded that the choice of thermal units affects the sizing of BHE by way of building extraction rates and ground injection rates.

Arghand et al. [35] studied the potential of combining DGC with district heating (DH) and ground source heat pumps by examining and comparing the required borehole depths and land area use of each. Results showed that cases with DGC and DH proved more balanced ground loads which enabled shorter boreholes compared to cases where DGC was combined with GSHP. Further, the authors noted that increasing borehole outlet temperatures only proved effective for cooling-dominated buildings.

2.5 Summary

To summarize, the existing body of literature agrees that ACB units offer certain energy savings compared to traditional all-air systems. These savings primarily stem from the inherent design features of ACB, such as the decoupling of ventilation and the utilization of higher-temperature chilled water. In addition, ACB systems offer advantages to the air distribution system. The reduced airflows enable the use of smaller AHUs and ductwork, resulting in more efficient and cost-effective designs.

Studies have previously investigated aspects of cost and performance independently and often in isolation from other systems. While some studies have explored the differences between VAV and ACB in terms of cost and performance, these investigations were often conducted in different climate contexts. Moreover, the investigation of one aspect often took precedence over the other. A limited number of studies have explored the practical applicability of DCG in combination with ACB, but these investigations primarily centered around the design of the energy source. Finally, no studies were found that convincingly investigated the differences between VAV and ACB in all aspects of energy, investment, and life-cycle costs.

3
Methodology

This chapter introduces the studied case building with a short summary of relevant building metrics. The methodology of the study pertaining to the simulation resource, system designs and optimizations, and life-cycle cost calculations are described in methodological order.

Figure 3 depicts the methodological scope and process.

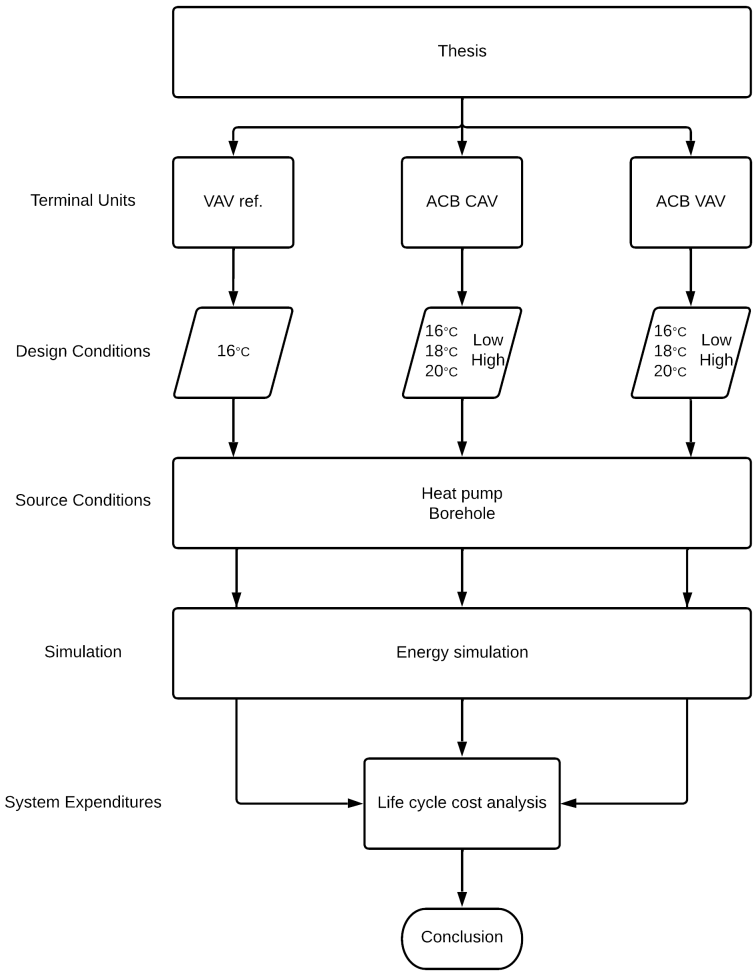


Figure 3. Thesis Flowchart

3.1 Modelling and simulation resources

Building energy simulations were performed through IDA indoor climate and energy (IDA-ICE) version 4.8 [36]. The simulation software has been extensively used for building energy and thermal comfort simulations in Sweden. It has been subjected to validation against measurements in accordance with the framework of the guidelines outlined in EN 13791 [37], ANSI/ASHRAE 140 [38], and CIBSE TM33 [39]. As such the software is ensured with accuracy and reliability.

The borehole heat-exchangers were designed through the software Earth Energy Designer 4.20 (EED) [40] with input data, see table 5 in appendix A, along with monthly building loads gathered from IDA ICE Energy-simulations. Based on parameter studies and a numerical simulation model, the software allows for analytical solutions for simple and complex vertical borehole systems.

3.1.1 Case building

The studied building, see figure 4, is a six-story commercial building located in Gothenburg, Sweden. The building houses restaurants, conference rooms, group rooms, and open office landscapes. The building has a north-south orientation with a glazed façade.

Figure 4. Case building: Johanneberg Science Park

The building footprint is of an irregular rectangular shape with an asymmetrically stacked and tapering architectural design. The total floor area amounts to $4458m^2$ distributed across six stories. The first floor, corresponding to 14% ($629m^2$), constitutes restaurant-, café- and, lunch-areas. The second to fifth floors, corresponding to 52% ($2323m^2$), is utilized for open office space with a few small group rooms, corresponding to 1% ($62m^2$). The middle section on all floors, corresponding to 13% ($580m^2$), functions as general open areas interconnecting all zones. The basement and sixth floor account for roughly 20% ($863m^2$) and house the building's mechanical room and other utilities.

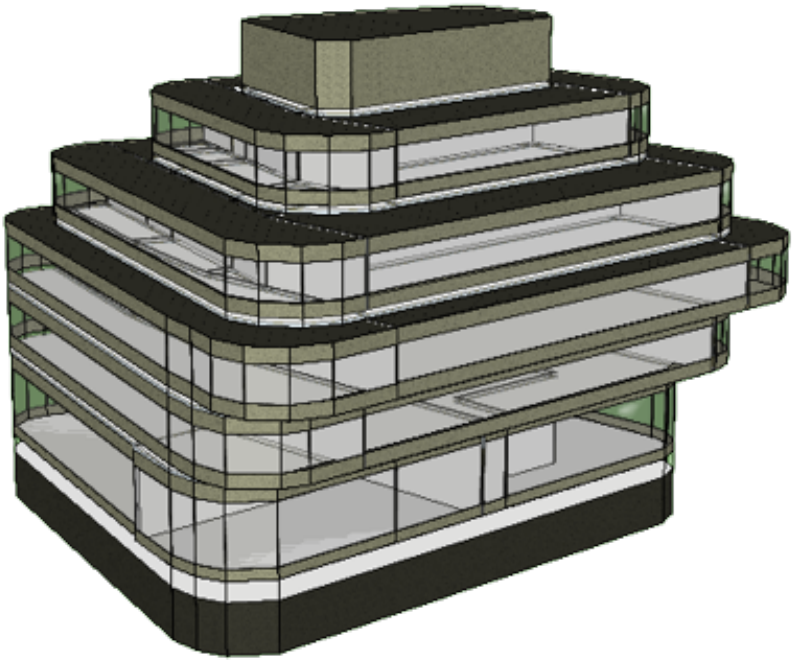


Figure 5. Case building: IDA ICE

Table 1 displays the modeled construction elements, from the building's as-built documentation, and table 2 shows the modeled internal loads of each zone. Occupants were modeled with an activity level of $1,2$ MET and clothing insulation level of $0,75 \pm 0,25$ CLO corresponding to sitting office work and traditional office apparel.

Table 1

Building construction

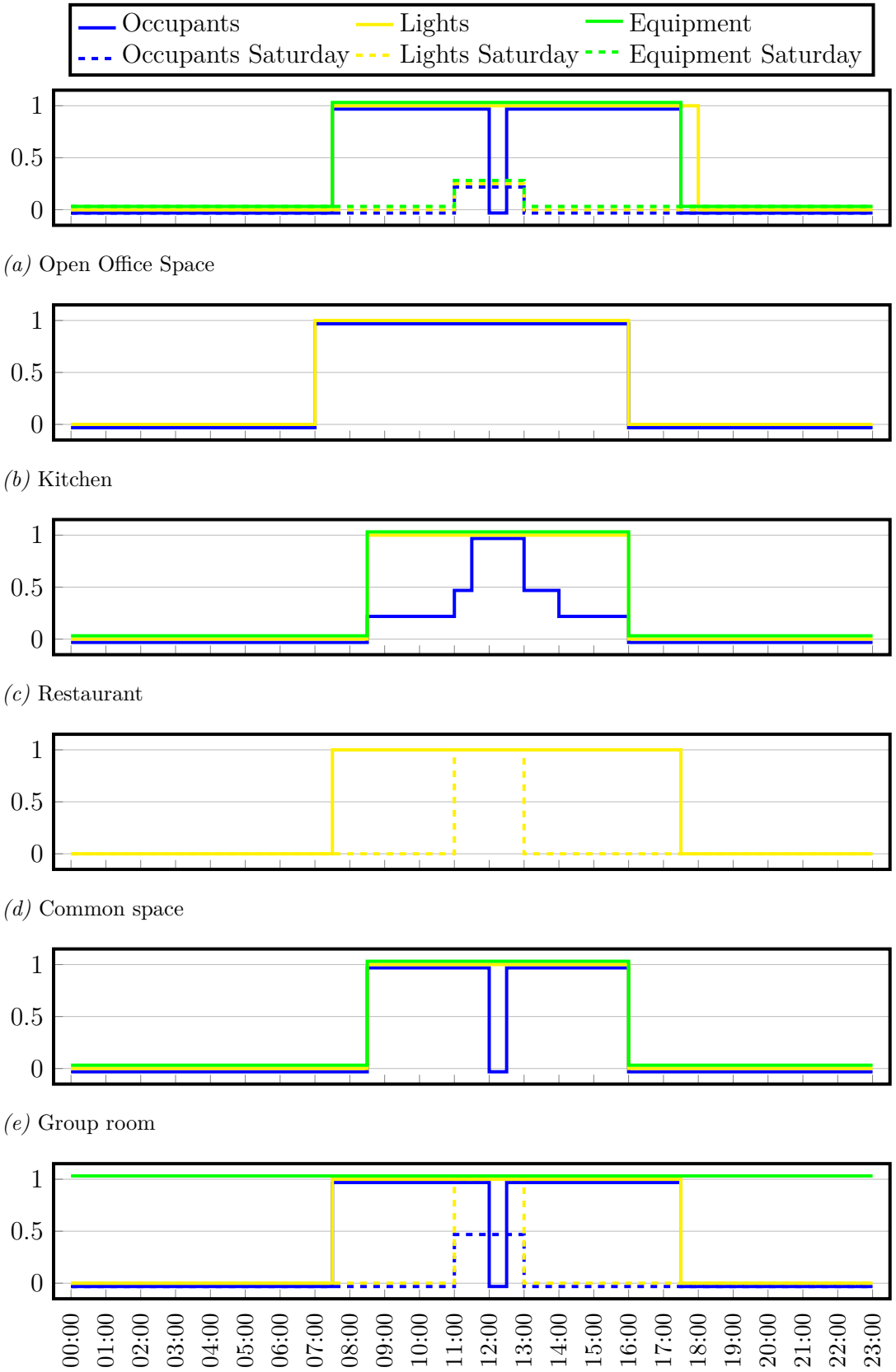
Element	U-value $W/(m^2K)$	G-value -	Solar transmittance -	Emissivity -
Glazing	0,54	0,4	0,32	0,837
	U-value $W/(m^2K)$	Thickness m		
External walls	0,15	0,256		
Internal walls	0,62	0,146		
External floors	0,061	0,919		
Internal floors	1,43	0,185		
Roof	0,14	0,531		

Table 2

Internal loads

Zone	Occupants	Lights		Equipment	
	no./m^2	W/m^2	kWh/m^2	W/m^2	kWh/m^2
Open office space	0,08	6,8	17,9	8,5	22,4
Kitchen	0,12	12,0	28,2	-	-
Restaurant	0,33	12,0	23,5	1,0	2,0
Common space	-	5,0	13,6	-	-
Group room	0,17	7,0	13,7	4,0	7,8
Mechanical room	0,003	5,0	13,6	1,0	8,8
	no.	W	kWh	W	kWh
Total	391	31 000	77 500	21 400	61 100

Figure 6 presents the scheduling of the internal heat loads.



(a) Open Office Space

(b) Kitchen

(c) Restaurant

(d) Common space

(e) Group room

(f) Mechanical room

Figure 6. Internal load scheduling

3.1.2 Climate and location

Gothenburg, Sweden has a temperate climate with the Köppen-Geiger climate classification: cfb [41, 42]. This climate is characterized by cool-to-cold winters and mild to warm summers. A weather file was sourced from The ASHRAE IWEC 2 [43] database for the location of Landvetter, Gothenburg.

3.1.3 Existing building heating and cooling systems

The case building is originally equipped with a VAV system featuring VAV controls for all zones, except the middle sections and mechanical rooms. In these areas, CAV is supplied at rates of $0,35\,l/s * m^2$. The VAV-covered zones are regulated based on temperature with minimum to maximum airflows of $0,5 - 5,50\,l/s * m^2$. The function of reheating is achieved using radiators located in each respective zone.

The air handling unit houses a heat-exchanger with a thermal efficiency of 80 % and minimum temperature of $-4°C$. The AHU operates between 7 a.m. and 6 p.m. with the heat-exchanger active at all times. The thermal components are supplied by a BHE supported by district heating and cooling with hot water at $60°C$ and chilled water at $5°C$ respectively.

The buildings setpoint for cooling is $24°C$ and for heating $22°C$.

3.2 Design parameters

To ensure an effective and efficient ACB system design, accurate determination of the cooling and ventilation demand parameters is essential. The cooling demand represents the difference between internal heat gains and heat losses. Oversizing or undersizing the system can lead to operational issues, such as inadequate cooling during peak load conditions or excessive cooling in part load conditions. The ventilation demand establishes the supply of fresh air to ensure adequate air quality by proper removal of contaminants and odors. The ventilation rate is determined by building size, occupancy rate, and activity. Insufficient ventilation may result in decreased performance and perceived discomfort caused by unpleasant smells and stagnant air.

3.2.1 Cooling demand

The primary parameter in the sizing of ACB systems is the maximum cooling capacity needed to maintain desired temperatures during critical design conditions.

The maximum cooling requirement was established using IDA ICE by deactivating ventilation—setting the ventilation rate to zero— and relying solely on unrestricted ideal coolers to provide cooling. Using this approach, any cooling resulting from mechanical airflow is eliminated, and the ideal coolers consume the power required to achieve the desired indoor air temperature setpoint. After conducting a 'Cooling load' simulation, the maximum power requirement is displayed for under 'Room unit cool', which regards all sensible and latent cooling performed by the room units while 'Local cooling unit' accounts for the sensible heat removed by the units. The difference the two indicates the latent heat required to be removed by the primary air. The cooling demand was extracted for each zone for the peak date in July corresponding to maximum heat removal.

3.2.2 Ventilation demand

To ensure optimal indoor air quality and thermal comfort, the units must deliver fresh outdoor air at a rate that adequately meets comfort conditions. These conditions are bound by the minimum ventilation requirement and the space latent heat loads. The larger of the two airflow rates was considered as the baseline for subsequent sizing calculations.

The minimum ventilation rate is determined as the hygienic airflow presented as equation (2) [44].

$$q = 0.35\, l/s * m^2_{floor} + 7\, l/s * occupant \tag{2}$$

The airflow rate required to offset the latent heat loads of each zone was determined by use of equation (3) [45], rewritten and solved for air volume flow, q. The enthalpy of vaporization of water was determined through an online calculator [46] at $20°C$ to equal $2454\, kJ/kg$. The density of air was $1,204\, kg/m^3$. The difference in moisture content between the setpoint, $24°C$ and 60% RH, and the delivered primary air, $16°C$ and 40% RH, was determined via a Mollier diagram to a humidity difference of $0,0067\, kg/kg$.

$$h_L = \rho_{air} * q * I_{vap} * \Delta x \tag{3}$$

Where:

h_L	:	Latent heat	[kW]
ρ_{air}	:	Density of air	[kg/m^3]
q	:	Ventilation flow rate	[m^3/s]
I_{vap}	:	Enthalpy of vaporization	[kJ/kg]
Δx	:	Water content differential	[kg/kg]

3.3 Active chilled beam sizing

An initial selection of three ACB products from SWEGON was made. The three products are presented in table 3. The selected units were deemed to possess sufficient capacities for both airflow and cooling, albeit in different proportions. The efficiency of each unit was assessed based on the discrepancy in the number of units required to satisfy the respective ventilation and cooling demand. The unit that measured the smallest disparity in the number of units required to meet the two demands was considered to have the most appropriate ratio of capacities for the case building.

Table 3

Maximum unit capacities for largest size of three selected products

Product	Cooling capacity W	Primary air flow l/s
PARASOL (Classic)	2055	85
Zenith	2135	100
PACIFIC	3400	75

After selecting a product, the next step involved deciding the optimal size of the unit. The chosen product was offered in three sizes: 600 mm, 1200 mm, and 1800 mm. Each size exhibits varying cooling and ventilation capacities and costs. The respective capacities of the beams were determined via a Product Calculator [47], provided by the manufacturer, for the parameters: indoor air temperature $24°C$, primary air temperature $16°C$ with a pressure drop of 75 Pa and circulating water temperature $14 - 17°C$. The resulting capacities are presented in table 4.

Table 4

ACB-Product Size

Product size mm	Cooling capacity W	Primary air flow l/s
600	865	45
1200	1 687	80
1800	2 255	100

The number of units required to meet the cooling and ventilation demands of each zone was determined. The demand which prescribed the largest number of units was selected as the design case. The design cases of each size were compared on the basis of total cost and the unit size yielding the cheapest system was selected.

3.4 Active chilled beam design and optimization

With a product and size selected, the process of system sizing and parameter optimization began with an combined focus on primary air temperature and airflow rate. Figure 7 illustrates the process.

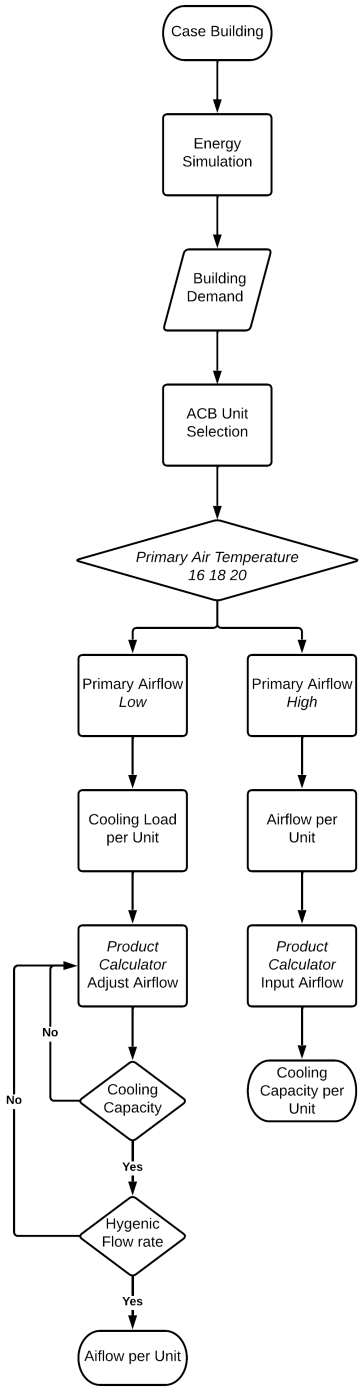


Figure 7. Active chilled beam design flowchart

The chilled beam units were modeled with "Active beam" elements in IDA ICE. These elements are supplied with CAV, and the supply temperature and airflow are adjusted according to the design case. Heating was provided by radiators located in each zone.

3.4.1 Primary air temperature and flow rate

To investigate the influence of primary air temperature and flow rate on system design, energy demand, and system costs, a total of six CAV design cases were established on the basis of three air temperatures and two ranges of airflow rate. Classified by temperature and sub-classified by airflow, each case presented a unique system design. The temperatures investigated were: $16°C$, $18°C$, and $20°C$. The airflow rates were classified and defined as: LOW or L) minimum design airflow required to achieve both ventilation and cooling demands and HIGH or H) the largest of LOW and $2,0\,l/s*m^2$. This airflow was selected as it roughly corresponds to two times the average hygienic flow rate. Zones constituting technical rooms and middle sections with no occupancy were sized with minimum ventilation flow, $0,35\,l/s*m^2$, regardless of the design case.

The design process began with establishing a system design for case 16-L. This served as the baseline design upon which each subsequent case was designed. At this stage, each calculation was performed at the zone level to ensure proper integration with the simulation model. The pressure drop and circulating water temperatures were kept constant at 75 Pa and $14-17°C$ respectively. The air velocity was assumed as $0,15\,m/s$.

Each sub-case L necessitated a unique minimum airflow to be determined. This was achieved by determining the number of units at full capacity required to meet the cooling demand, rounding up. Based on this, the cooling load per unit was computed. However, as a result of rounding up, each zone was slightly oversized with regards to cooling. To resolve this, the airflow was adjusted in the Product Calculator, keeping all other variables constant, till the unit reached the specified cooling capacity. The airflow rate was then cross-checked and compared to the required hygienic flow to ensure compliance. In cases of insufficient airflow, the hygienic flow rate was used instead.

The first step in the system design of sub-case H began with determining the sizing airflow rate as previously defined. From the airflow rate, the total zone airflow was computed. As this sub-case investigates the effects of higher than minimum airflow rates, the fulfillment of cooling capacity is guaranteed. As such the sizing demand becomes the airflow rate. Subsequently, the number of units required to meet the ventilation rate was determined. Thereafter, the airflow per unit was computed. The unit airflows were inputted into the Product Calculator which computed the unit's total cooling capacity.

3.4.2 Water temperature and flow-rate

Water-related variables were only analyzed from a theoretical and practical design standpoint. These variables were ineffective for simulations in IDA ICE. Water inlet temperatures could be adjusted, however, the software automatically adjusted the mass flow rate to achieve the specified cooling capacity.

To examine the influence of the circulating water temperature on the design of chilled beam- and subsequent borehole systems, a new design case was developed. Case 18-L constituted the base design upon which a high-temperature cooling case was established. The new case was classified as Case 18-L-HW and was established with both CAV and VAV. The circulating water temperature was set with an inlet-outlet temperature of $20 - 22°C$. The raised water temperature resulted in a significant reduction in the unit cooling potential, necessitating a complete redesign of the base case. Following the procedure described for case L in section 3.4 a new high-temperature system was designed.

3.4.3 Variable air volume

To investigate the potential synergies between ACB and VAV, cases 16-L, 16-H, 18-L, 18-L-HW and, 20-L were converted. For each case, the minimum airflow rate was determined based on the minimum ventilation requirement of $0,35\, l/s * m^2$ at zero occupancy. The maximum airflow rate was set to the maximum unit capacity for each case. This corresponds to the airflow supplied when each unit within the zone operates at full capacity at the specified pressure drop of 75 Pa.

The VAV function was implemented through the use of VAV dampers located in each zone. Meaning, VAV control was carried out at the zone level. The sizing of the dampers was performed based on the total airflow required for each zone.

In IDA ICE, VAV was implemented by changing the system control strategy to VAV, temp. control and assign each zone with the respective minimum and maximum airflow rates.

3.5 Heating and cooling source

The borehole design is not an originally intended section of this thesis. Therefore the procedures of system design are kept short and meager.

The BHE was designed to serve both heating and cooling purposes for two primary reasons. First, the dual functionality helped maintain a more balanced building load, resulting in smaller fluctuations in the undisturbed ground temperature compared to a system dedicated solely to heating or cooling. Second, it allowed for the variations in heating demand of different ACB designs to be accounted for in the BHE design and its associated costs. This enabled a more precise investigation of the relationship between building system design and energy source design.

The BHEs energy and power coverage factors were arbitrarily set, for both heating and cooling, to 70 % for power and 85 % for energy. A coverage factor of 100 % was deemed excessive as the resulting BHE fields were too large to be cost-effective. For the reference VAV case, the coverage factors necessitated an adjustment for the same reason. The borehole investment costs would place the VAV system at an unconducive disadvantage compared to the other systems. The coverage factors for heating were set to 70 % for power and 90 % for energy, and for cooling, 60 % for power and 75 % for energy.

For an initial one-factor design, the borehole depth was assumed and kept constant at 270 m. With the design parameters presented in table 5, the BHE systems were sized with regard to the number of required boreholes for the undisturbed ground temperature to remain below the setpoint temperature over the full-service life. The setpoint is bound by the selected inlet temperature of the building's cooling system. With an inlet temperature of $14°C$, the borehole outlet fluid should be at most $13°C$ due to thermal inefficiencies and losses in the heat exchanger connecting the ground loop and the building loop. To provide a slight margin of safety, $12,5°C$ was selected as a setpoint for the six primary design cases. The high-temperature cooling case enabled a setpoint temperature of $18,5°C$.

Table 5

Borehole Design Variables

Parameter	Value	Unit
Ground Properties		
Thermal conductivity	3,0	$W/(m*K)$
Volumetric heat capacity	2,4	$MJ/(m^3*K)$
Ground surface temperature	8,5	$°C$
Geothermal heat flux	0,04	W/m^2
Borehole thermal resistance		
Fluid/ground	0,09	$(m*K)/W$
Internal	0,00	$(m*K)/W$
Heat carrier fluid		
Thermal conductivity	0,582	$W/(m*K)$
Specific heat capacity	4192	$J/(Kg*K)$
Density	999,8	Kg/m^3
Viscosity	0,001308	$Kg/(m*s)$
Freezing point	0,0	$°C$
Borehole		
Spacing	8,0	m
Diameter	114,3	mm
Contact resistance pipe/filling	0,0	$(m*K)/W$
Filling thermal conductivity	0,6	$W/(m*K)$
Volumetric flow rate per borehole	0,65	l/s
U-pipe		
Outer diameter	45,0	mm
Wall thickness	2,6	mm
Thermal conductivity	0,420	$W/(m*K)$
Shank spacing	63,0	mm

Values source from [48]

3.5.1 Borehole depth

With the initial configuration and number of boreholes determined at the pre-established depth of 270 m, a rudimentary optimization was made with regards to the borehole depth. With the same number of boreholes and configuration, the BHE depth was incrementally reduced while ensuring the outlet borehole fluid temperature remained at the setpoint. The optimization was finished when the depth could not be reduced further without affecting the targeted outlet temperature.

3.6 Life-cycle cost analysis

The economic evaluation was done through a life-cycle cost (LCC) analysis which was performed via the net present value (NPV) method presented as equation (4). NPV represents the sum of all anticipated future cash inflows and outflows related to the investment and operation of the systems throughout their technical service life, discounted to present value. The calculation period was set to 20 years with an assumed equivalent service life for each component, meaning no account taken for reinvestment and residual values. For the analysis, an interest rate of 6 % was assumed.

The LCC analysis considered the initial investment costs, ongoing maintenance costs and all expenses related to energy and power demand during the operation of the systems. All costs were presented excluding value-added tax (VAT) for the purpose of easier comparison. The calculations were made in Swedish kronor (SEK).

$$NPV = \sum_{n=1}^{t_s} \frac{CF_{in,t} - CF_{out,t}}{(1 + r_i)^t} \tag{4}$$

Where:
- NPV : Net present value
- CF_t : Cash flow year t
- t : Year of cash flow
- t_s : Technical service life
- r_i : Discount rate

3.6.1 Investment and maintenance

The investment costs pertaining to the AHUs, ACBs, diffusers, and dampers were provided by the manufacturer. Material and installation costs for ductwork, borehole, piping and valves were estimated using typical cost estimates. Costs for radiators were sourced from [49].

The accounted investments costs for each component are presented in table 6. The component selection and cost of each respective design case were determined as follows:

Three AHU units of different sizes were provided by the manufacturer. Each case was assigned a unit based on the peak airflow reported by IDE ICE. In cases in which the maximum airflow exceeded the capacity of the largest AHU, two smaller AHUs were used.

The design of the VAV reference case was assumed to entail full coverage of VAV supply air diffusers. The VAV diffuser unit was arbitrarily selected and the number of diffusers was determined by dividing either cooling and airflow demand by the diffuser capacities. The largest number was selected.

For cases combining ACB with VAV, VAV dampers were assumed at zone level. This was primarily motivated by the zone-level control of the beam's water circulation. The VAV damper product was arbitrarily selected and sized based on each zone's total airflow.

Regarding ductwork, the duct size of $200\,mm$ was set as the baseline, representing a cost of $300\,SEK/m^2$. Each system was provided a basic ductwork sizing by calculating the airflow averaged across each zone of the building. Utilizing an online calculation tool [50], the required rectangular duct size for an air velocity of $6\,m/s$ was attained. The percent difference in the cross-sectional area was assumed to also pertain to the cost difference.

Each case utilized radiators as the heating component. A radiator was assumed to deliver $1000\,W$ of heating capacity. In each case heating demand was attained from IDA ICE 'Heating' simulations. The demand divided by the capacity yielded the required number of radiators.

Table 6

Component investment cost

Component	Total cost KSEK	Unit Cost SEK/n	Other Cost
AHU			
GOLD 070 RX	549 000		
GOLD 080 RX	585 000		
GOLD 100 RX	793 000		
ACB			
PARASOL Zenith 1800 mm		12 200	
VAV diffuser			
COLIBRI REACT ALS 250-600		7 750	
VAV damper			
REACT Va			
125		4 050	
200		4 150	
250		4 300	
315		4 350	
400		5 000	
500		5 350	
Radiator	5 030		
Pipes and control valves, SEK/m^2			350
Borehole, SEK/m			530

The maintenance costs accounted for all components except the boreholes. The costs were determined as a set percentage of the investment cost. Sourced from [51], the annual maintenance costs as a percentage of the respective initial investment costs are presented in table 7.

Table 7

Annual maintenance costs

Component	Percentage of initial investment %
Active chilled beam	4
Air handling unit	4
Damper	1
Diffuser	4
Ductwork	2
Pipes and valve	1,5
Radiator	1

3.6.2 Energy, power, and electricity

The energy rates were sourced from the supplier company in Gothenburg, Göteborg Energi [52]. Costs of district heating and cooling are constituted by separate costs for energy and power. The energy costs have monthly consumption rates. The power costs consist of a fixed rate determined by peak power draw and an accompanying variable cost for the total power consumption. Accounted energy costs are presented in table 8. Future energy costs were calculated with a price growth rate of 2 %.

Table 8

Energy, power and electricity costs

	Variable rate	Fixed rate
District cooling		
Energy, SEK/kWh		
Jan, Feb, Mar, Dec	0,145	
Apr, Oct, Nov	0,260	
May, Jun, Jul, Aug, Sep	0,340	
Power, SEK/kW		
0-100 kW	795	19 800
101-250 kW	630	36 300
251-500 kW	500	68 800
District heating		
Energy, SEK/kWh		
Jan, Feb, Mar, Dec	0,521	
Apr, Oct, Nov	0,377	
May, Jun, Jul, Aug, Sep	0,122	
Power, SEK/kW		
0-100 kW	911	9 510
101-250 kW	861	14 560
251-500 kW	808	27 863
Electricity, SEK/kWh	1,50	

Each system design was simulated in IDA ICE. 'Heating', 'Cooling', and 'Energy' simulations were run to attain the system energies and powers in peak seasonal conditions and in annual operations. IDA ICE accounts for periods of free cooling so no related adjustment had to be made.

A manual calculation method was used to obtain values for fan and pump energies. The fan energy consumption was calculated by multiplying the annual airflows with the specific fan power (SFP) of the AHU fan. The SFP values for three different AHU sizes were provided by the manufacturer. The annual airflows of the AHU were determined using IDA ICE and displayed in a duration diagram. The area under the curve was calculated and then multiplied by the SFP.

The pump energy for the ACBs was rudimentary calculated as 5 % of the cases' total heating and cooling energy.

Pump power for the boreholes was calculated with equation (5) assuming a pressure difference of $150 Pa$ and pump efficiency of 60%. The volumetric flow was determined by multiplying the number of boreholes with the per borehole flow rate of $0,65\, l/s$. To attain the final pump energy, the calculated pump power was multiplied with $2600\, h$ portraying the number of occupancy hours of the full year.

$$P_h = \frac{\dot{V} * dP}{1000} \tag{5}$$

Where:	P_h	:	Power	[kW]
	$\dot{V}$	:	Volumetric flow of fluid	[l/s]
	dP	:	Differential pressure	[kPa]

The electricity pertaining to the heat pump connecting the building's cooling system with the BHE was determined as shown by equation (6) with an assumed coefficient of performance (COP) of 3.

$$E_e = \frac{\xi * E_H}{COP} \tag{6}$$

Where:	E_e	:	Heat pump electricity	[kWh]
	ξ	:	Coverage factor	[-]
	E_H	:	Heating energy	[kWh]
	COP	:	Coefficient of performance	[-]

4

System design

This chapter aims to provide a detailed account of the system design process developed in chapter 3, offering a perspective on the steps and considerations involved. Furthermore, the finalized system designs for each case will be showcased and discussed.

4.1 Active chilled beam sizing

The number of units required to fully satisfy the building's hygienic ventilation flow rate and peak cooling demand respectively are presented in table 9.

Table 9

Number of units required for each sizing demand and the discrepancy between the two

	Required Units		
Product	Ventilation	Cooling	diff.
PARASOL (Classic)	51	67	(16)
Zenith	43	61	(18)
PACIFIC	58	41	(17)

A discrepancy in units between cooling and ventilating enables two scenarios, one of which may provide certain benefits in cases synergizing with VAV. If sizing for cooling, the unit discrepancy entails full utility of the cooling capacity and reduced utility of the airflow capacity. Thus, the airflow per unit in design conditions will be less than the unit's total capacity. This enables the possibility of increasing airflow which may be leveraged to provide a larger portion of free cooling during favorable ambient conditions by down-regulating the cooling coil and up-regulating the supplied air volume. It may also provide the opportunity to fully satisfy low-load conditions through an all-air operation, resulting in reduced pump and chiller energies, albeit with a slight increase in fan energy.

Observing that the PACIFIC system is sized on airflow rather than cooling signifies that the units in design conditions are primarily utilized for air diffusing and the cooling capacity is under-utilized. Therefore, it can be concluded that the product is not suitable for the current configuration of the case building. If the building system would allow for secondary supplies of ventilation air, via e.g. conventional air diffusers, whereby the unit would act as a secondary air supply, the unit could be sized in accordance with the cooling demand by which the large cooling capacity of the unit could benefit the system through lowering initial ACB costs, potentially reducing total investment costs.

Both PARASOL products are sized based on cooling demand. The Classic model exhibits the smallest discrepancy indicating it to be the most efficient unit for the case building in terms of ratio or capacities. This means that the unit has the highest utilization in both airflow and cooling under design conditions. However, the Zenith model measures the smallest number of units for both demands which concludes it to be the most cost effective model in terms of fulfilling demands.

Table 10 shows available sizes of the selected PARASOL Zenith and the number of units of respective size required to fulfill the building cooling demand. Only cooling needs to be assessed as cooling required a larger number of units in design conditions.

Table 10

ACB Product Size

Unit length mm	Number of units n	Unit costs SEK	Total system cost kSEK
600	171	7 850	1 342
1200	95	8 900	846
1800	77	12 200	939

The total unit costs for unit sizes 1200 and 1800 were deemed close enough to warrant further investigation about which unit size would result in the cheapest system and lowest energy costs. In a separate study, systems were designed for each size, and a life-cycle cost analysis was made. The short study yielded 1800 as the cheapest unit size, thus it was selected.

The 1200 mm system could be designed with a lower average airflow per unit. However, the larger quantity of units yielded a higher total airflow. This brought effects on the size and cost of the AHU and ductwork. Moreover, the additional units increased the potential of overcooling requiring a large addition of radiators. Collectively, these effects result in the 1200 mm systems investment costs being slightly more expensive than the 1800 mm.

In regards to the operation of the 1200 mm system, the addition of units pertain to lower cooling energies which are offset by larger heating energies while both powers increase. The rise in heating energy and power is the result of larger supplied airflows requiring air conditioning. Moreover, the increased airflows increase fan energy. In total, the 1200 mm consumes 3 % more energy. Accounting for the larger investment costs and subsequent maintenance, the final LCC is 6 % more expensive.

4.2 Active chilled beam system design

The seven CAV and five VAV systems designed in this thesis are presented in table 11 displaying the number of units, total cooling capacity, total airflow, and their respective coverage factors. The coverage factors illustrate the fulfillment of the design demands.

For all CAV cases designed with low airflow rates, the primary airflows are necessarily oversized relative to the hygienic flow due to the insufficient cooling capacity of the units at these minimum ventilation rates. This results in optimal coverage factors for cooling demand but oversized airflows. The oversizing of airflow could be mitigated at the cost of additional units. However, this approach is not recommended when regarding economic viability as it entails large extra investment costs for marginal savings in fan energy.

CAV systems designed with high airflow rates are inherently oversized regarding airflow. The additional units attributed to these cases are entirely for airflow purposes. The increased airflows result in greater cooling capacity yielding the subsequently larger coverage of cooling demand.

The disparity in the number of units and ventilation coverage factor between the two CAV sub-cases decreases as primary air temperatures increase. This highlights the significant increase in the additional units and per-unit airflow that is necessary to counteract the reduced cooling capacity.

The design of the VAV cases, specifically the determination of maximum airflow, gives the systems very high capacities of total airflow. Subsequently, the systems overall utilize air to a larger extent as the systems preferentially cool with air when ambient conditions enable free cooling. In other scenarios the VAV supplies minimum airflows, relying on the chilled beams to cool providing only supplementary cooling when needed. Moreover, the higher airflows naturally increase the system's total cooling capacity.

Table 11

ACB System Designs displaying each systems total capacities and the respective coverage factors relative to sizing parameters

Case	Number of Units	Total Cooling Capacity	Coverage of Cooling Demand	Air Flow	Coverage of Ventilation Demand
	n	W/m²	%	l/s*m²	%
CAV					
Case 16					
L	77	31,71	104	1,24	129
H	88	38,17	125	1,53	160
Case 18					
L	84	31,47	103	1,37	142
L-HW	118	30,78	101	2,24	236
H	93	36,28	119	1,61	168
Case 20					
L	92	31,28	102	1,53	159
H	99	34,44	113	1,71	178
VAV					
Case 16					
L	77	38,95	127	1,73	182
H	88	44,51	146	1,97	207
Case 18					
L	84	37,97	124	1,88	198
L-HW	118	34,52	113	2,65	279
Case 20					
L	92	36,63	120	2,06	217

Table 12 displays a comparison of the CAV and VAV cases 16-L, 18-L and 20-L in regards to peak power draw and total energy consumption for heating and cooling respectively. VAV exhibits a significant increase in cooling energy due to the larger volume of supply air in need of conditioning. However, this is offset by a slightly larger decrease in heating energy as the need of reheat decreases. In addition, great decreases in fan energy and heating power result in a moderate benefit to total energy consumption for all VAV cases. However, the discrepancy between the CAV and VAV systems decrease in magnitude with higher supply air temperature.

Table 12

Energy comparison between ACB with CAV and VAV

Case	Cooling		Heating		Electricity		Total
	kW_{max}	kWh	kW_{max}	kWh	kWh_{fan}	kWh_{pump}	kWh
Case 16-L							
CAV	185	76 814	74	70 626	32 717	8 540	188 552
VAV	174	90 996	41	53 109	14 300	8 373	166 747
	-5,9%	+18,4%	-44,6%	-24,8%	-56,1%	-2,0%	-11,6%
Case 18-L							
CAV	179	86 210	73	64 119	36 864	8 685	195 877
VAV	169	99 533	41	52 228	18 108	8 750	178 625
	-5,6%	+15,5%	-43,8%	-18,5%	-50,9%	+0,8%	-8,8%
Case 20-L							
CAV	172	103 818	74	69 508	32 717	9 834	215 876
VAV	165	114 207	40	57 201	20 289	9 738	201 435
	-4,2%	+10%	-45,9%	-17,7%	-38,0%	-1,0%	-6,7%

Note. Heating energies excluding domestic hot water.

4.3 ACB summary

The methodology of ADC design outlined in sections 3.3 and 3.4 are proven viable presuming building demands can be determined with relative accuracy. The size of units pertains to slight investment cost differences although not to the degree as changing airflow rate or supply air temperature. However, when accounting for the costs of energy and power over the systems entire technical service life, the effects of product size potentiate. In terms of final life-cycle costs, the cost disparity between the systems sized with 1200 mm and 1800 mm units is approximately equivalent to the difference between the cases 16-L and 16-H. Thus, the selection of product and size should be conducted with an attentiveness corresponding to choice of system parameters.

Regarding the selection of system parameters, airflow rate and supply temperature, lower is better. In terms of airflow, a minimized airflow pertains to the least number of units and lower fan energies. Selecting lower supply air temperature provides larger cooling capacity which might enable a reduced number of units.

Combining ACB with VAV is a great way of decreasing overall energy consumption and reducing peak power. Both result in lower energy costs. Reductions in fan and pump energy bring further cost savings compared to the system's CAV counterpart.

Further investigations will be limited to a selection of systems exhibiting properties as the most significant systems or enabling the comparison of different parameters and systems serving as a reference to those systems. The excluded systems will serve as verification of any conclusions formed during the course of subsequent chapters. The selected ACB cases are:

+ CAV 16-L — Investigates low, $16°C$, air temperature and low airflow rate
+ VAV 16-L — Investigates the potential of the VAV for case 16-L
+ CAV 20-H — Investigates high, $20°C$, air temperature and high airflow rate
+ CAV 18-H — Serves as reference to 18-L and 20-H
+ CAV 18-L — Represents current design guidelines: $18°C$ and low airflow rate
+ VAV 18-L — Servers as reference to high-temperature cooling
+ VAV 18-L-HW — Investigates high, $20°C$, temperature chilled water

4.4 Borehole sizing

Figure 8 presents the borehole systems sized for the investigated chilled beam design cases, illustrated by total borehole length. All designs achieve a $12,5°C$ borehole outlet temperature at the end of the service period. No definite cause-effect relationship between the borehole length and ACB system parameters can promptly be distinguished. Overall, increasing system airflow displays the possibility of a shorter total borehole length. It can be presumed that each supply temperature has an optimal airflow range that yields the shortest borehole length. Furthermore, increasing air temperature indicates a longer total borehole length. Comparing the reference case it can be concluded that ACB pertains to shorter total lengths. The high-temperature cooling case provides the largest discrepancy. This is exclusively due to the higher temperature rise of the borehole outlet temperature.

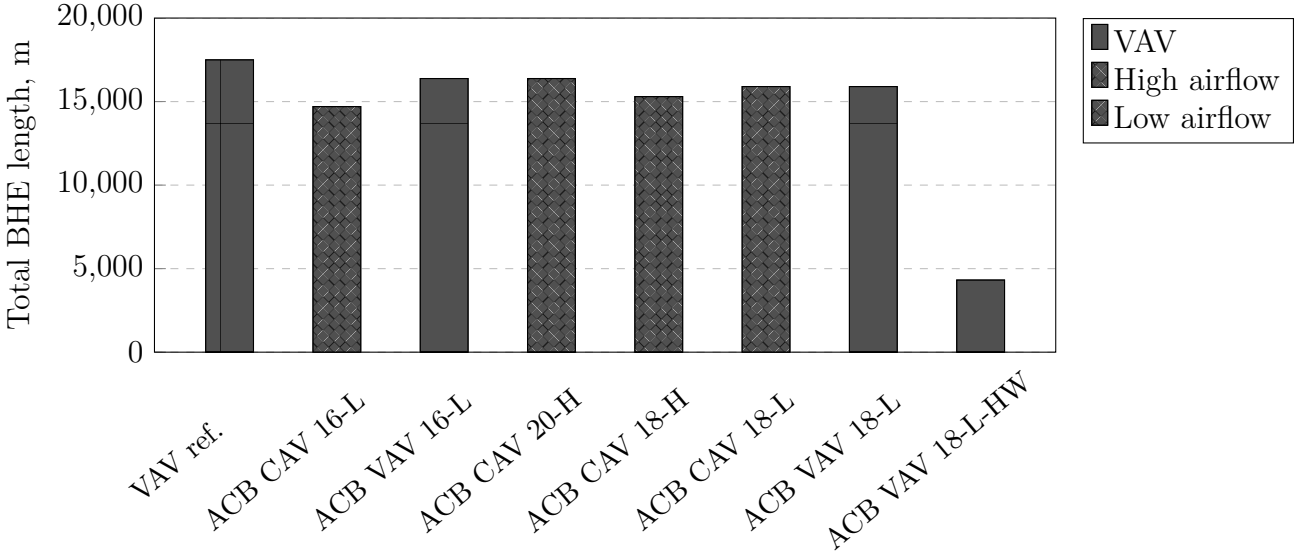


Figure 8. Total BHE length for the VAV reference and the investigated design cases.

5
Results

This chapter provides an overview of the simulation results pertaining to system energy and power consumption, calculated electricity usage, investment costs of the building and borehole systems, maintenance expenses, and the overall life cycle costs.

5.1 Energy, power, and electricity

Figures 9 and 10 display each case's respective building system energy- and power consumption covering the heating and cooling demand of the AHU and the zone units.

Overall, increasing primary airflow yields an increase in heating energy which is offset by a larger decrease in cooling energy resulting in a slight decrease in total energy. As the AHU cooling load slightly increases to accommodate the larger air volume, the total energy decrease is entirely attributed to zone cooling. The decrease of which is the result of a larger portion of the cooling load being handled by air. Moreover, increased airflow pertains to more balanced building loads. However, each primary air temperature exhibits different affinity towards this effect indicating that there is no distinct airflow rate that can be determined as universally optimal. Rather, each supply temperature pertains to a unique optimal airflow rate. Though it can be discerned that a general increase in airflow balance the building loads for this specific case building.

Increasing primary air temperature causes heating energy to slightly decrease from cases 16 to 18 and increase proportionally from cases 18 to 20. However, these minor fluctuations in heating are overshadowed by the effects of cooling. The AHU cooling is approximately halved with each increase in supply temperature, as a result of the decreased demand for cooling the ambient air. Furthermore, as the supply air temperature increases, the chilled beams must compensate for the additional heat supplied to the space, resulting in substantial (30 %) increases in zone cooling that are sufficient to counteract all other energy decreases, resulting in an overall increase in total energy.

In comparison to the VAV reference case, none of the ACB cases achieve energy demand levels equal to or below that of the all-air VAV system. Furthermore, the VAV reference case is the only scenario that results in a heating-dominated building, possibly indicating the higher demand for reheat functions.

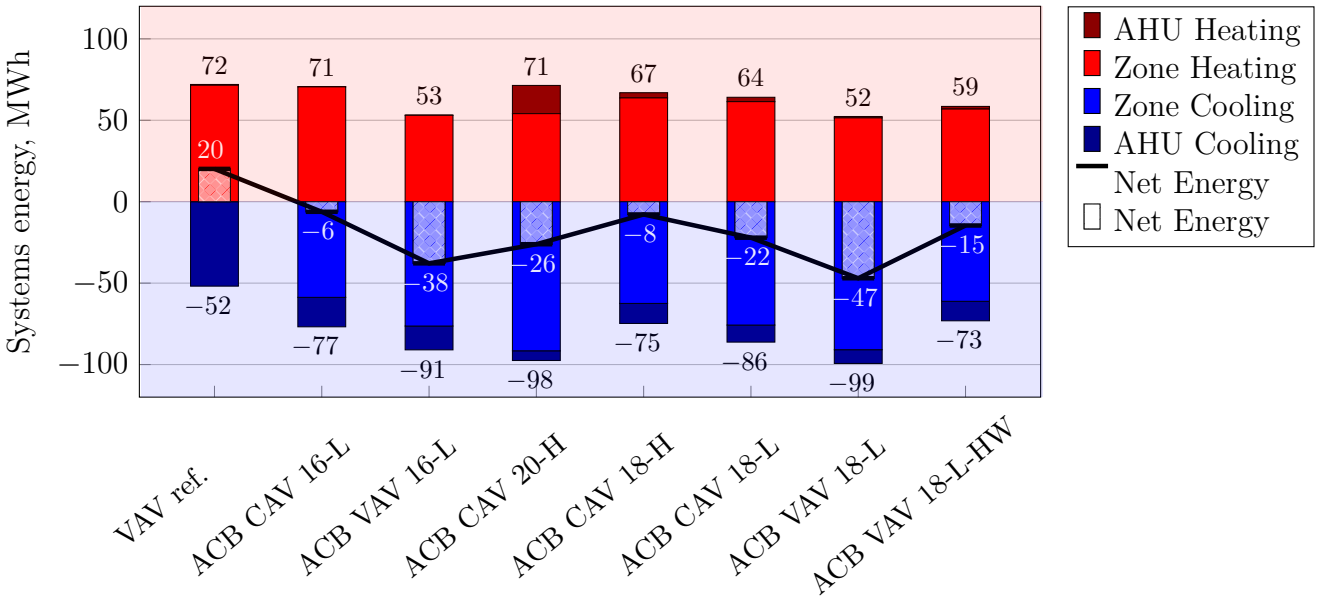


Figure 9. Annual heating and cooling energy and building energy balance.

Regarding system power, increasing primary airflow leads to a slight increase in both heating and cooling power. For heating, both AHU and zone heating powers increase as a result of the larger air volume requiring air conditioning, and the increased airflows enhance the space cooling, subsequently increasing the need for reheat. With the same reasoning, AHU cooling power experiences an increase while zone cooling slightly decreases.

Increasing the primary air temperature results in a marginal decrease in total power consumption. The AHU heating power increases as the ambient air is heated to higher temperatures. Moreover, the higher supply temperature reduces the potential for overcooling, leading to a decreased need for reheat and subsequently lowering the zone heating power. Conversely, these effects lead to a decrease in AHU cooling power and an increase in zone cooling power in order offset the higher supply temperatures.

In terms of total cooling, the impacts of increasing both airflow and temperature counterbalance each other, resulting in no significant change in cooling power. This can be observed in cases such as 16-L to 18-H and 18-L to 20-H. A similar effect is seen with total heating power when only the air temperature is increased, as illustrated in cases 16-L to 18-L or 18-H to 20-H.

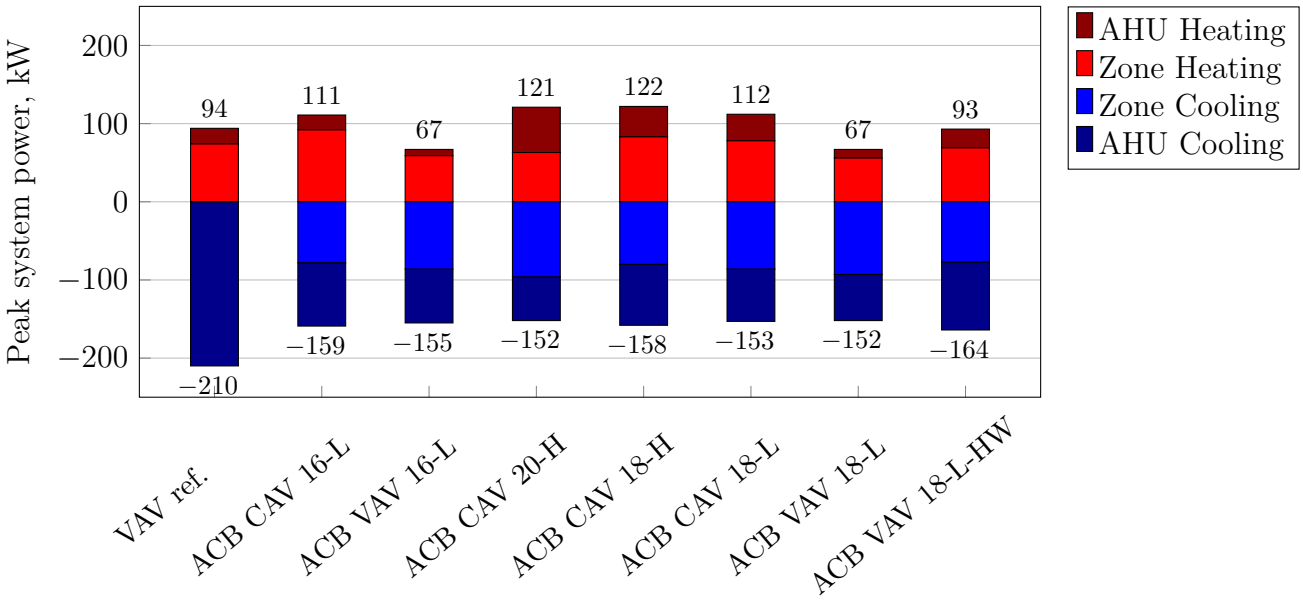


Figure 10. Annual heating and cooling peak power

Figures 11 and 12 depict the purchased energies and powers for each system when integrated with the BHE. Since the BHE coverage factor remains constant across all cases, except for the VAV reference, there are no further relative distinctions between the ACB cases. The coverage factor for the reference case is 5 % larger for heating and 10% smaller for cooling energy. While the subsequent disparity between the cases are relatively smaller, the reference case still maintains the lowest energy demand. In terms of power, the reference case has equivalent coverage for heating and a 10% smaller coverage for cooling. Considering the already large cooling power for the VAV system, the relative difference to the ACB cases is further amplified.

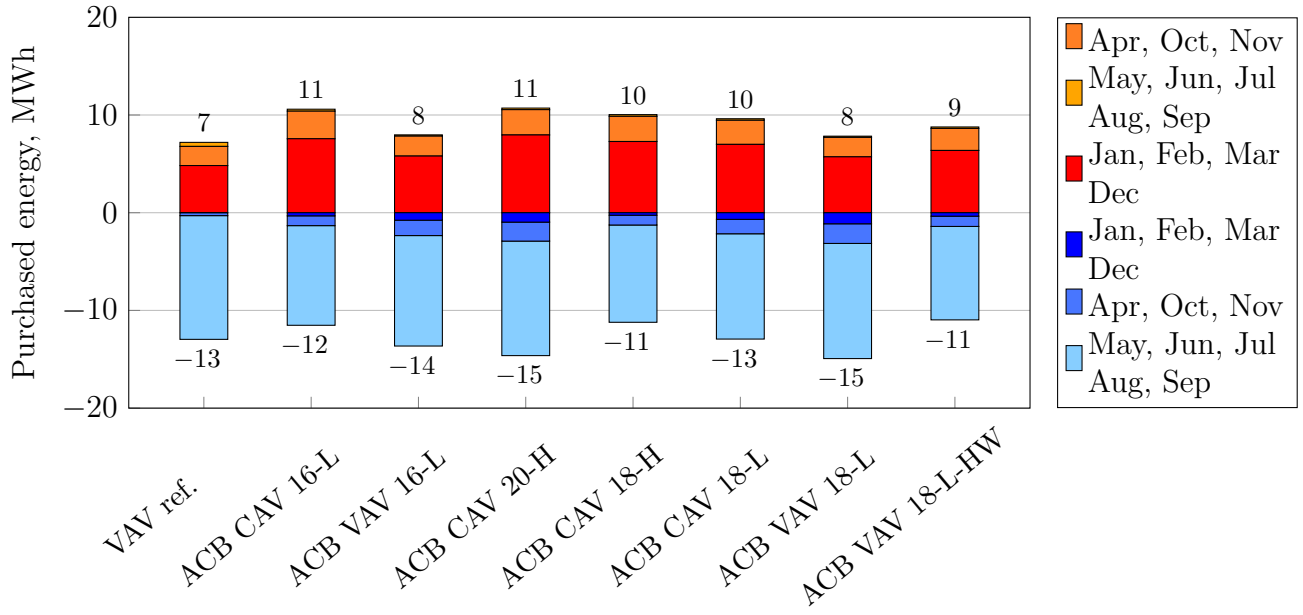


Figure 11. Purchased heating and cooling energy for systems synergized with BHE

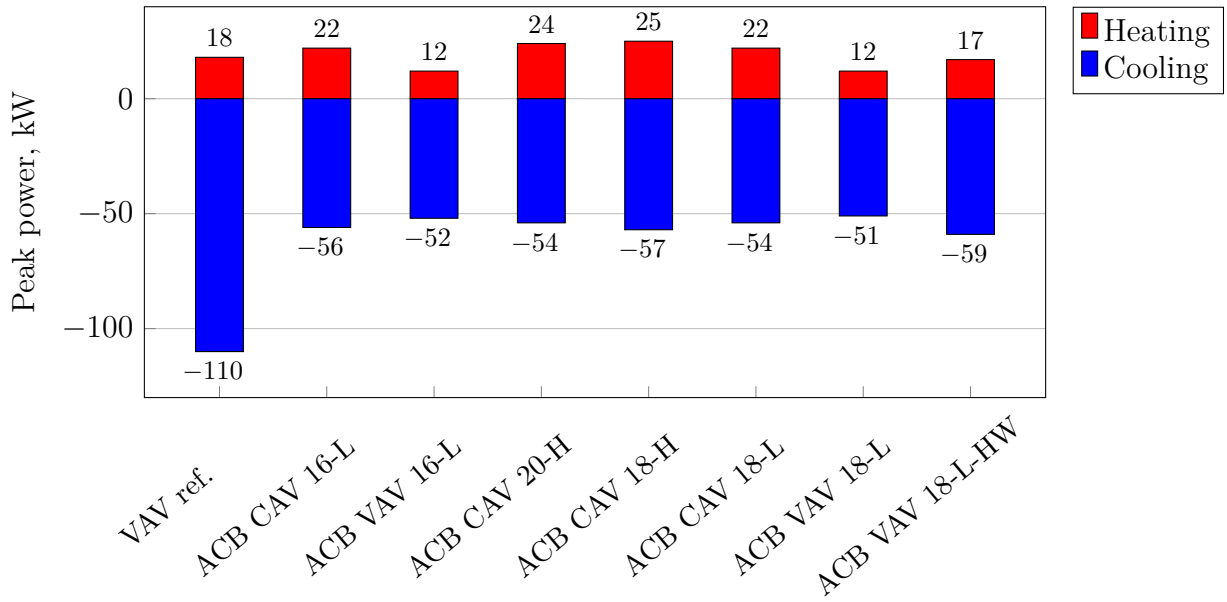


Figure 12. Purchased heating and cooling power for systems synergized with BHE

Figure 13 illustrates the purchased electricity for each system. In general, the slight advantages gained in heating and cooling energies by increasing airflow are mostly mitigated, and in some cases completely offset, by the increase in fan energies. However, not all cases of increased airflow result in proportionally larger fan energies. Some cases may even exhibit reductions. Implementation of higher airflows may enable the use of larger AHUs with more efficient fans, meaning lower SFP. For example, case 18-L to 18-H portray an airflow increase of approximately 17 %, equivalent to 4000 m^3/h. The corresponding rise in fan energy equates to only 3 % or 1200 kWh. This suggests that considering the size and efficiency of the AHU when selecting airflow rates, rather than solely focusing on minimizing, could yield potential benefits.

Comparing the VAV reference case with ACB CAV, reductions in fan energy can be discerned. These can be attributed to the effect of decoupled ventilation. Only case 20-H manages a slightly larger fan energy, 700 kWh, compared to the reference case. When comparing ACB systems supplied by CAV and VAV, additional significant savings in fan energy can be discerned. This is predominantly the result of the CAV operating at design airflows regardless of space loads while the VAV is able to modulate the supplied air volume. To quantify these effects on fan energy it can be noted that fan energy accounts for approximately 30 % of the total system energy in the reference case. In CAV ACB cases, it comprises around $20 - 25$ % of the total system energy, while in VAV ACB cases, it constitutes roughly 15 % of the total system energy. The resulting effect of these reductions is that VAV ACB outperform the reference VAV in terms of total energy consumption.

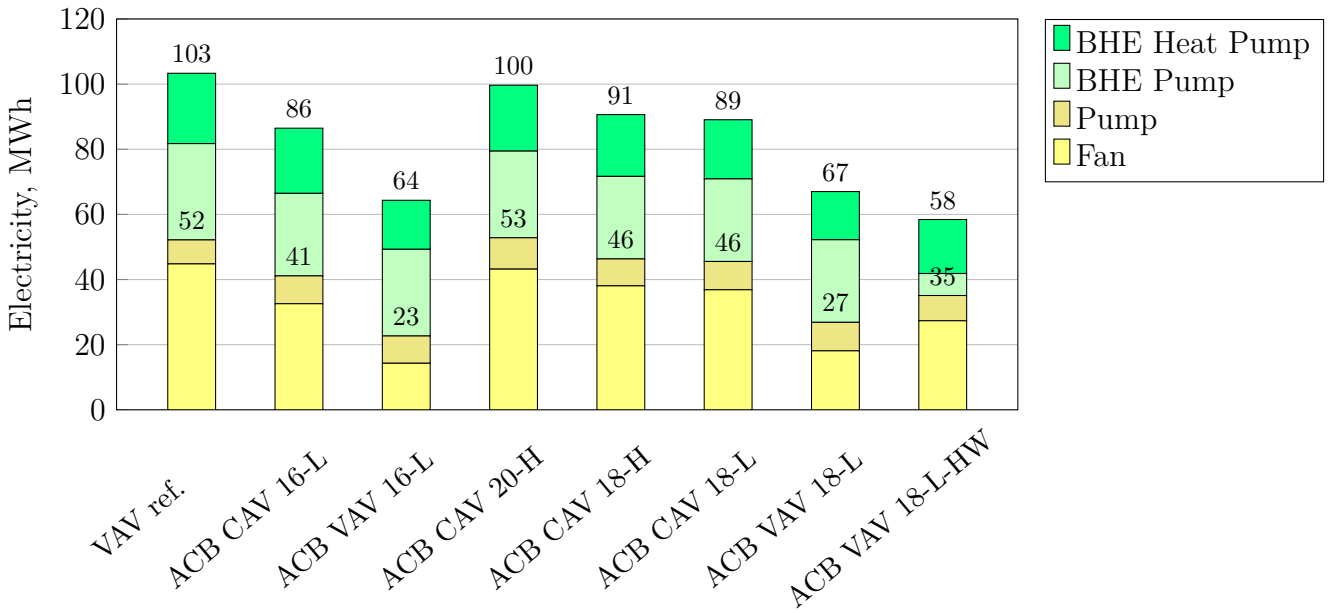


Figure 13. Purchased electricity for the investigated systems

5.2 Investment

5.2.1 Building System

Figure 14 presents the accounted investment costs for each studied case. Regarding the reference case, its high reliance on airflow requires two large AHUs along with extensive ductwork. The high cost of VAV diffusers could be mitigated by utilizing zone-level controlled VAV rather than on the diffuser level. Such a case would replace the diffuser cost with one equivalent to the VAV dampers for the ACB cases. With that, the reference case would cost approximately $750 - 880\, SEK/m^2$.

Overall, the ACB cases see savings to both AHU and ductwork compared to the VAV reference, as a result of the smaller airflows. The largest costs of the ACB system are the chilled beams and the required piping and valves.

Cases with higher airflows exhibit higher initial costs related to the need for larger AHUs, expanded ductwork, additional ACBs, and more radiators to compensate for the larger potential of overcooling. Changes to cost from increasing temperature portray similar effects. To mitigate the decreased cooling capacity of the supply air, higher airflows are necessary, leading to the requirement of larger AHUs, expanded ductwork, and an increased number of ACBs. However, a small benefit regarding radiators can be observed as the building shifts from heating dominated to cooling-dominated.

Combining VAV with ACB introduces the additional cost of VAV dampers, but the system may offer certain cost advantages compared to its CAV counterpart. As portrayed in case 16-L, VAV results in cost savings for the AHU due to reduced building airflow peaks benefited by load diversity. However, VAV requires slightly larger ductwork due to increased zone peak airflows. The primary cost benefit of VAV is the decreased need for radiators resulting from reduced reheat requirements.

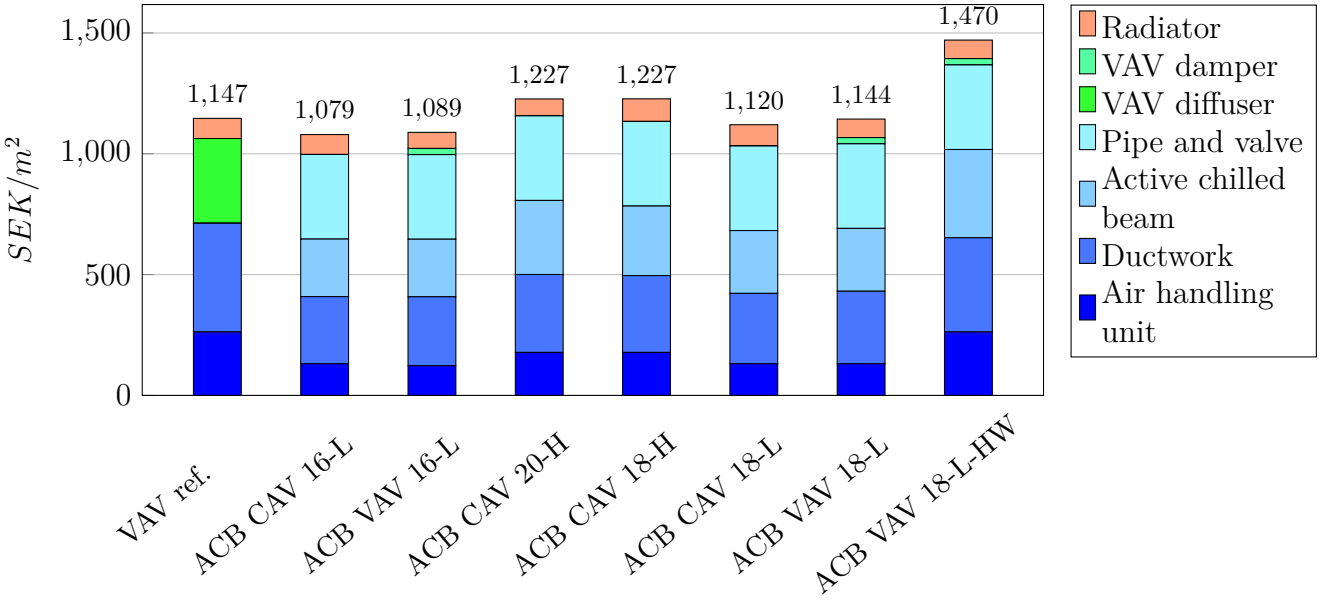


Figure 14. Investment costs comparison of VAV reference and ACB systems

5.2.2 Source

Figure 15 displays investment costs for the systems with the addition of BHE.

Due to the reduced peak power and generally more balanced building loads, ACB systems overall allow for smaller BHE designs compared to the reference case. However, this benefit decreases as the supply air temperature increases. For example, case 16-L exhibits 16 % smaller total BHE length compared to the reference case, while 18-L displays a 9 % smaller length. It can thus be concluded that increasing primary air temperature, raises the cost of both the building and the source systems.

In terms of airflow rates, the parameter's impact on building load balance provides advantages by allowing for shorter BHE lengths. The slight cost benefits from reduced BHE length, effectively mitigate the higher ACB system costs attributed to increased airflows. This means that increasing airflow, increases the cost of the building system but pertains proportional savings from the source. Thus, if utilizing BHE and the sole focus is on investment costs, the system airflow becomes inconsequential.

The case of $20°C$ chilled water temperatures allows for the largest increase in borehole outlet temperature decreasing the need for ground thermal mass of the BHE field which enables a smaller field with a lesser number of boreholes lowering costs by a large margin. Compared to its lower water temperature counterpart, the high-temperature cooling case benefits from a reduction in BHE length of 74 %.

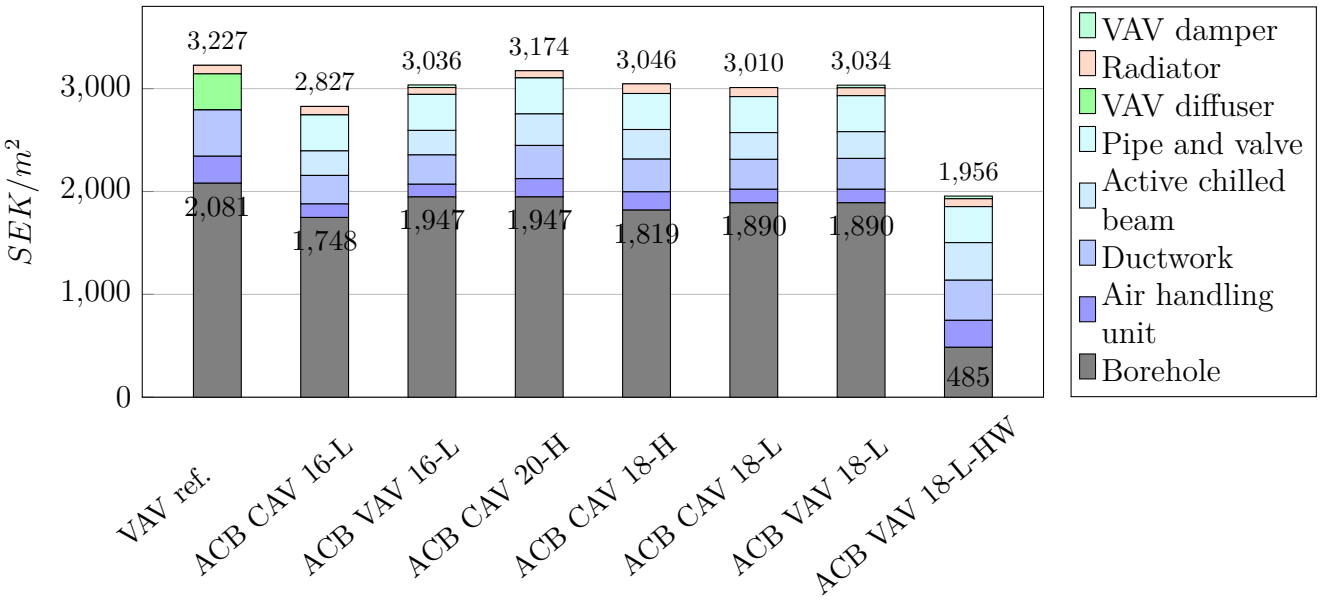


Figure 15. Investment costs comparison of VAV reference and ACB systems coupled with BHE

5.3 Life-cycle costs

Figure 16 shows the annual energy and power costs for each system. The cost of power in constituted by fixed annual subscription cost and an accompanying variable rate determined by the annual three day peak average draw. The sizable impact of the fixed subscription cost is evident from the reference case which exhibits close to double the subscription cost for cooling power. Aside from this cost, the reference case would have proven the cheapest case regarding cooling.

The fan energy savings attained by ACB systems are reflected by the smaller electricity costs. This highlights the benefits of utilizing ACB. The savings are further amplified in the cases of ACB VAV which display barely half the total electricity consumed by the reference case.

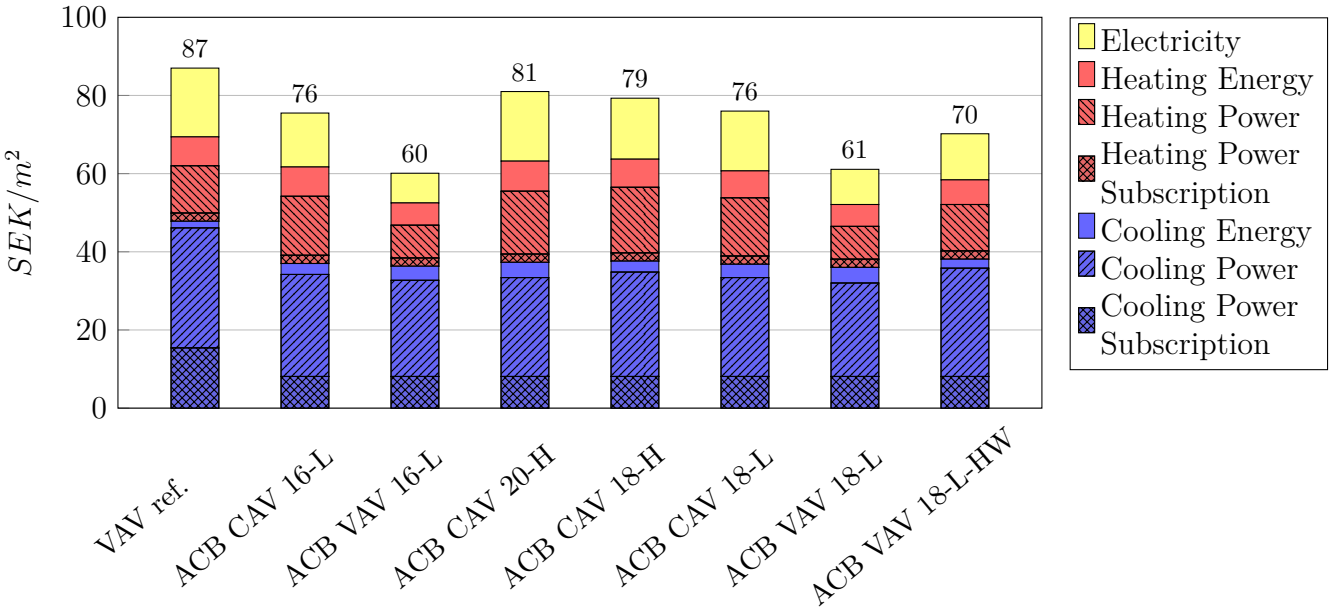


Figure 16. Annual energy and power costs for systems

Figure 17 shows the annual energy costs for each system coupled with BHE. In this scenario the majority of the system energies are handled by the BHE. The energy and power covered by the BHE are considered free of costs pertaining to the large reduction in costs. However, as the system utilizes a heat pump, a portion of the heating energy must be provided as electricity. An additional electricity cost pertaining to the operation of BHE systems is the pump. The pump electricity required for the BHE consumes approximately three times the electricity of the pump used for the chilled beams.

Regarding the reduction in annual costs by utilizing boreholes, ACB exhibits slightly larger total savings. The VAV reference case exhibits 70 % savings in heating and 50 % in cooling, while the average ACB case, exhibits 68 % savings in heating and 61 % in cooling. The gains in electricity are consistent across all CAV cases, including the reference case, with an average increase of 100 %. High-temperature cooling demonstrates the most significant advantages with the smallest increase in electricity consumption of only 66 %, compared to the 149 % of its lower-temperature counterpart. Overall, the case lowers the annual energy costs by 30 % more than the average ACB case.

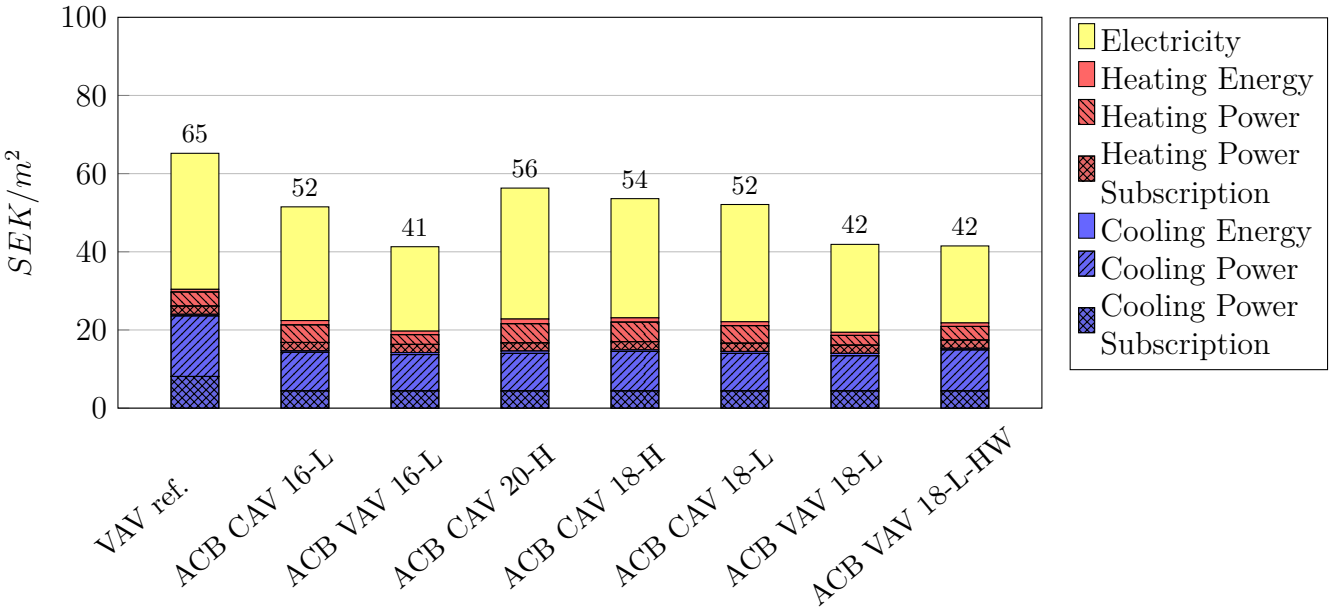


Figure 17. Annual energy and power costs for systems coupled with BHE

Figure 18 displays the various maintenance costs of each system associated with the different system components. A consequence of all costs being calculated as percentages of initial investment costs is that the higher investment yields larger maintenance costs which might not relate to actuality.

Even with only a small divergence in investment cost the slightly cheaper ACB systems yields much lower annual maintenance costs. From table 7 presented in section 3.6.1 it can be discerned that ACBs, AHUs and diffusers pertain to the largest costs. Followed by ductwork and pipes and valves. The necessity of large AHU and ductwork for the reference case yields maintenance costs that equate to the combined maintenance costs of AHU, ductwork and ACBs for the chilled beam cases 16 and 18.

The previous remark on the unnecessarily high costs of utilizing VAV diffusers for the reference case instead of zone-installed VAV dampers is further illustrated by the excessively large maintenance cost. A way to quantify the consequence of this effect is by determining the ratio of annual maintenance cost to initial investment cost. For the reference case, this yields $3,0\,\%$ with diffusers and $2,5\,\%$ with dampers. For comparison, in the ACB cases this ratio ranges between $2,4 - 2,7\,\%$ for all investigated cases. As such, a design of VAV tailing more towards demand-controlled ventilation (DCV) pertains to higher maintenance costs while a more reserved VAV system, with zone level control, is equal to ACB in terms of maintenance costs, judged by the introduced ratio.

The case of high-temperature cooling further distinguishes itself as the most costly building system. The near doubling of ACB units and the necessity of high airflow yields a case of exceptionally large maintenance costs. However, the discrepancy in cost is predominantly the result of the proportionally large investment costs. Analyzing the previously introduced ratio displays the high-end $2,7\,\%$, technically cheaper than the reference case.

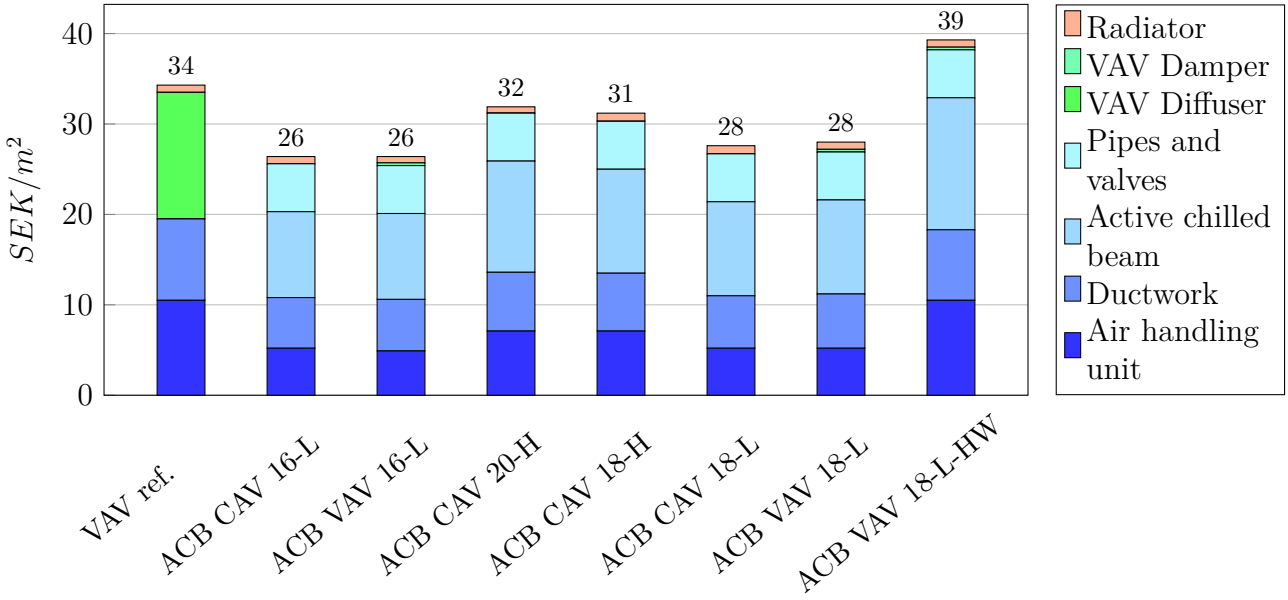


Figure 18. Annual maintenance costs

5.4 Total life-cycle cost

Figures 19 and 20 shows the final LCC costs for the ACB systems standalone and coupled with BHE. The final LCC includes all annual costs discounted over the full service life. Overall, BHE benefits the total LCC by reducing the operational costs of the building system. The costs saving are primarily due from reductions in purchased energy and power. With the reduced power subscription costs constituting a large portion of the savings.

The high-temperature case undoubtedly pertains the greatest potential in utilizing BHE, exhibiting a meager $3\,\%$ increase in final LCC. Comparatively, the lower temperature ACB cases exhibit cost increases between $55 - 77\,\%$ and the reference case raise LCC costs by $66\,\%$. The increased cost are exclusively due from the BHE investment costs.

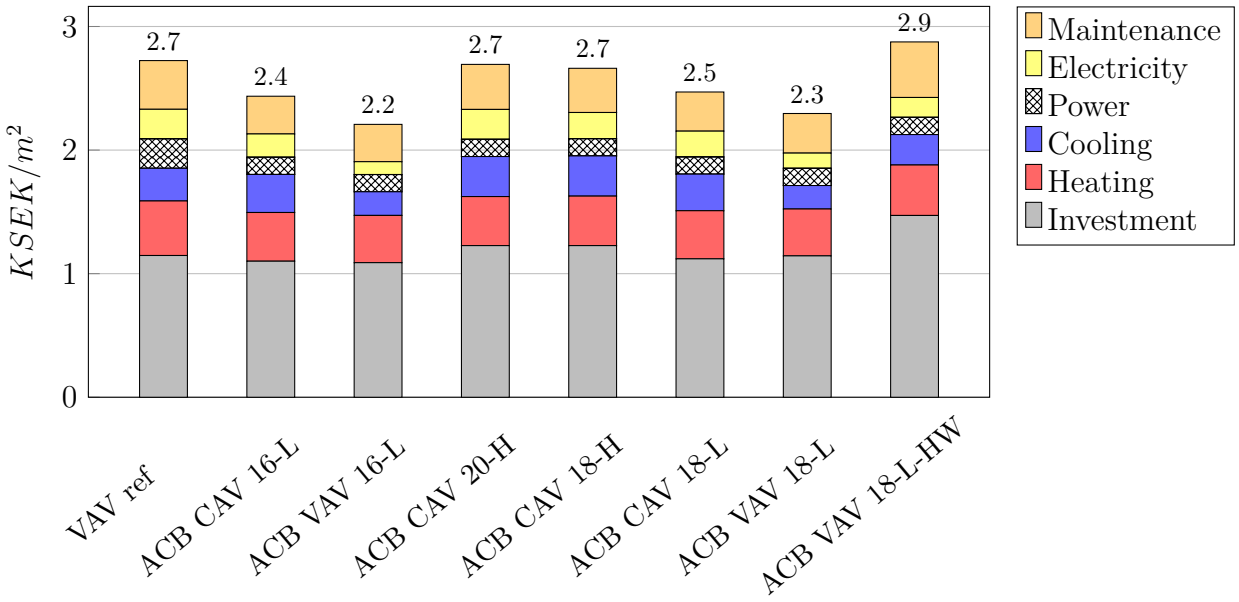


Figure 19. Life-cycle costs for systems

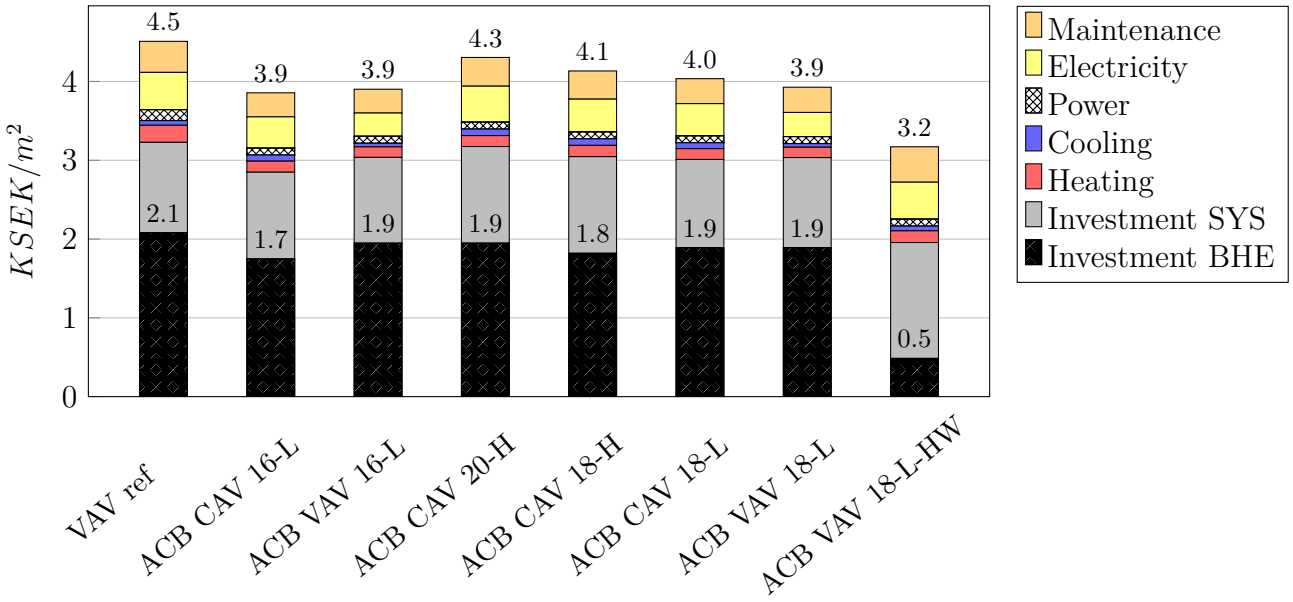


Figure 20. Life-cycle costs for systems coupled with BHE

5.5 Sensitivity analysis

The LCC analysis is subject to a degree of uncertainty due to the selection of discount rate. In figures 19 and 20, the presented LCCs were computed using a nominal discount rate of 6 %. The accuracy of the LCC heavily depends on this rate and thus the inherent uncertainties associated with the choice of rate necessitate a sensitivity analysis to determine its effect on the final LCC outcomes. Figure 21 presents the results of this sensitivity analysis performed for each studied case. The general trend of increasing discount rate is a decreasing total LCC and an overall reducing disparity between the cases.

The LCC for the building systems of the reference and the two high airflow cases tend towards a common point as the discount rate increases. A higher discount rate reduce the present value of future cash flows. This means that the life-cycle energy and maintenance costs constitute smaller parts of the LCC. As such, at high discount rates, the LCC mainly reflects the initial investments costs. The reverse of this becomes evident by observing VAV ref. and VAV 18-L-HW. With an discount rate of 1 %, the two systems exhibits equivalent LCC. In this scenario, the much larger investment cost of the HW-case is offset by the energy cost savings. As the discount rate increases, the significance of the future cost savings become less potent and the two systems diverge.

Regarding, the cases coupled with BHE, no convergence or divergence can be observed. This is similarly due to the dominant initial investment costs attributed by the BHE systems. However, the high-temperature cooling case exhibits interesting results. With low discount rates, when cash flows are material, the LCC equates with certain cases without BHE. Practically this implies that at low enough rates, a certain system design could result in a scenario where the cost of system and source, together, is cheaper than the cost of only the system for another case.

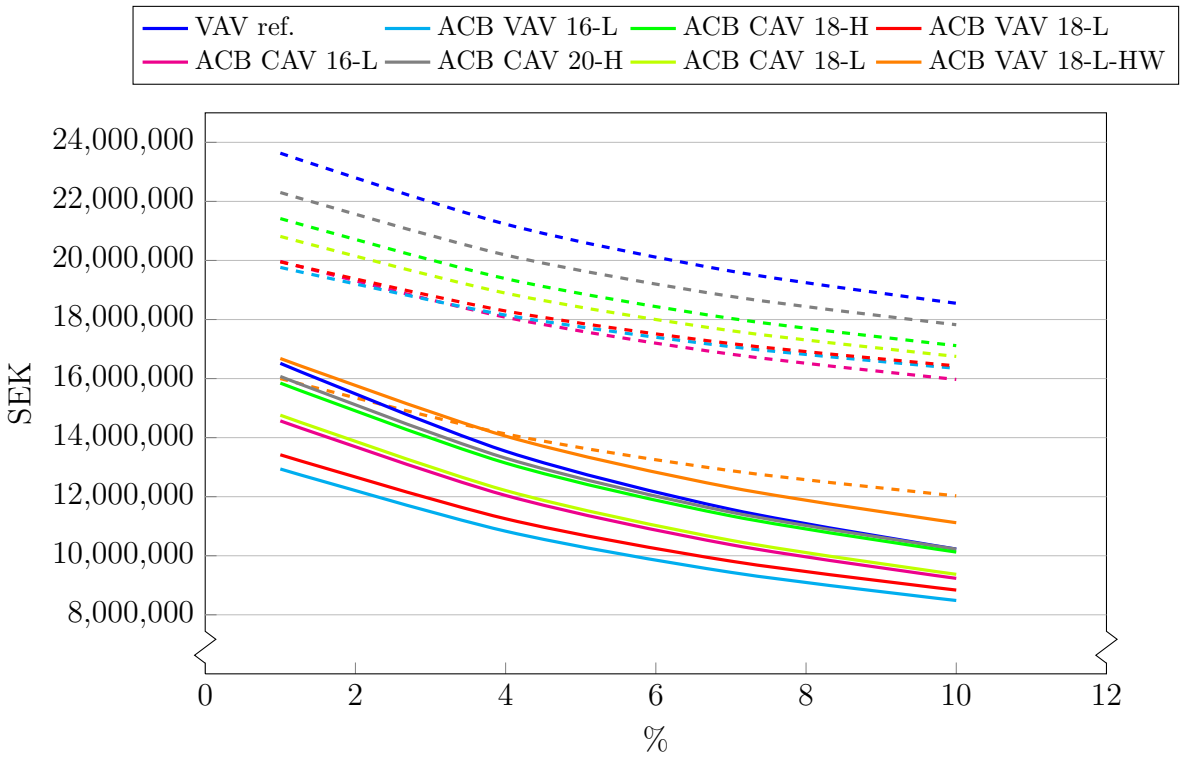


Figure 21. Sensitivity analysis: Total LCC as discount rates vary from 1 to 10 % for standalone system (solid line) and BHE coupled (dashed line)

6
Discussion

According to current literature, ACB systems are stated to pertain to energy savings compared to VAV [6, 10, 53, 54]. The savings are argued to stem from the decoupled ventilation, higher chilled water temperatures, higher thermal efficiency, and reduced demand for reheating. The energy savings have in part been validated through the results of this thesis. However, the potential for energy savings is highly reliant on system design. It can be discerned that CAV ACBs pertain to fan energy savings compared to the VAV reference. The best-performing case, 16-L, exhibits a fan energy reduction of 38 %, and the worst-performing case, 20-H, reports a marginal reduction of 4 %. The source of the energy savings pertaining to heating and cooling energy can not be distinguished due to the elementary nature of the investigation. Though, it is ascertained that ACBs, overall, consume less heating energy but more cooling energy, in total consuming more energy. Regarding power, ACBs draw more heating power but significantly less cooling power. The outcome for costs is a cheaper operation for ACB systems.

The rudimentary design of the ancillary systems causes a certain unreliability to the results. The primary culprit of this effect is the ductwork. The ductwork was not sized with regard to component compatibility or optimal system design. Pressure drops, air velocities, specific duct dimensions, and connections have all been exempt from consideration. This affects all studied systems in unmeasured direct performance factors e.g. noise, pressure loss, efficiency loss, and investment costs. Furthermore, the ductwork design might not vary as much as assumed in this thesis. By keeping all variables constant except duct size, the ductwork design is very limited and inefficient. For example, increases in size could be substituted by increases in air velocity thereby not affecting investment costs to the same degree. In total, these effects disadvantage design cases with higher airflow while benefiting cases with low airflow.

Similarly, the authors' interpretation of all sourced material pertains to the potential for variance and inconsistent application. For example, the assigned air handling units, which dominantly affect SFP, were attributed to each case based on peak system airflows while maintaining a percentage of headroom acting as a safety margin. Some systems possessed intermediate airflows not easily categorized into one of the defined AHUs. This was resolved by sizing up or roughly interpolating the SPF which naturally affects fan energy. The final consequence of this is a combination of higher AHU investment costs and higher or lower fan energies for a portion of the cases. Together affecting the final LCC.

To enable a more conclusive comparison, the systems should be investigated in part-load conditions which more accurately portray real-life operations [32]. This is primarily in favor of the VAV due to the benefits discussed in section 2.4. Although this would not have direct consequences for system design, it would yield a more accurate representation of operational costs.

Furthermore, the investigations should include and take great regard for control strategies and operating schedules. For example, allowing the ACB system to work within a range, allowing slight fluctuations around a room temperature setpoint, has proven benefits in reducing peak power [34,55]. In conjunction with BHE, this would result in either reduced peak outlet temperatures for existing systems or decreased BHE depth for new systems [56].

7
Conclusion

What are the investment cost differences between the traditional VAV and the various ACB systems?

The initial cost discrepancy between the two systems is highly dependent upon design choices. Disregarding the selection of design parameters, i.e. supply temperature and airflow rate, the main cost factor of active chilled beam systems is the selection of product and size. Accompanying pipes and valves are an additional large cost, however, these are less affected by direct design considerations. Concerning the traditional VAV, the predominant cost, and cost difference, is the AHU and ductwork. In summary, the two systems are possible to design within a similar budget.

What are the operational cost differences between the traditional VAV and the various ACB systems?

Operational costs vary greatly in-between the ACB cases. The primary factor pertaining to the cost differences between all cases, including the reference, is system airflow. Higher airflow means larger electricity costs, due to increased fan energy, and higher heating demand in terms of both energy and power. Compared with the reference case, all ACB systems manage lower operational costs.

What effects do different ACB system optimizations have on investment costs and overall building energy demand?

The investigations found that increasing ACB airflow in general leads to more balanced building loads while increasing the supply air temperature has opposite effects on the specific component energies and building load balance.

In terms of investment costs, the optimization of either parameter pertains to similar effects. Changes to airflow yield changes to the size of AHU and ductwork in addition to the number of units and radiators located in the zone. Changes in primary air- and chilled water temperature necessitate the modulation of airflow to compensate for the increase or decrease in cooling capacity. Chilled water temperature has the largest impact on cooling capacity and consequently pertains to the largest effect on investment costs.

What are the performance and cost differences between an ill-designed and a well-optimized ACB system?

The performance differences in total cooling and heating energy between the best and worst ACB case, concerning CAV, is approximately 14 %. For comparison, the difference between the reference case and the best ACB case is 19 %. The difference in power is practically negligible. Concerning fan energy, the worst ACB case consumes 32 % more. In total, the system differences yield a 7 % difference in annual energy and power costs and a 12 % difference in investment cost.

What effect does the ACB system design have on the cooling sources design?

Raising the circulating water temperature is observed to cause a significant effect on BHE design. Raising the chilled beam inlet temperature from $14°C$ to $20°C$ pertains to a 74 % reduction in total borehole length. Primary air temperature and airflow rate affect the BHE design by affecting the building load balance.

7.1 Further research

For further investigation, a more encompassing comparison is advised. For example: taking account of optimization possibilities relating to specific component designs of the ACB units, designing the systems with a broader focus on thermal comfort, investigating more combinations of the design parameters with more detailed increments, and optimizing control schemes and schedules. Such investigations would yield a more definite conclusion about the performance potential of well design ACB systems.

Bibliography

[1] Eurostat, "Heating of buildings decreasing, cooling increasing," 5 2022. [Online]. Available: https://ec.europa.eu/eurostat/en/web/products-eurostat-news/-/ddn-20220531-1

[2] International Energy Agency, "Global status report for buildings and construction," International Energy Agency, Paris, France, Tech. Rep., 12 2019. [Online]. Available: https://www.iea.org/reports/global-status-report-for-buildings-and-construction-2019

[3] International Energy Agency, "Insights brief: Space cooling," International Energy Agency, Paris, France, Tech. Rep., 2017. [Online]. Available: https://www.iea.org/reports/insights-brief-space-cooling

[4] S. Werner, "European space cooling demands," *Energy*, vol. 110, pp. 148–156, 9 2016, doi: 10.1016/j.energy.2015.11.028.

[5] International Energy Agency, "The future of cooling," International Energy Agency, Paris, France, Tech. Rep., 2018. [Online]. Available: https://www.iea.org/reports/the-future-of-cooling

[6] H. Sachs, W. Lin, and A. Lowenberger, "Emerging energy-saving hvac technologies and practices for the buildings sector," 12 2009.

[7] A. Mott, "Variable air volume (vav) systems operations and maintenance," 4 2021. [Online]. Available: https://www.pnnl.gov/projects/om-best-practices/variable-air-volume-systems

[8] J. Woollett and J. Rimmer, "REHVA Guidebook No. 21 - Active and Passive Beam Application Design Guide," REHVA, Ixelles, Belgium, 2015.

[9] M. Virta, D. Butler, J. Gräslund, J. Hogeling, E. L. Kristiansen, M. Reinikainen, and G. Svensson, "Rehva guidebook no. 5 - chilled beam application guidebook," REHVA, Ixelles, Belgium, 2004.

[10] K. Roth, J. Dieckmann, R. Zogg, and J. Brodrick, "Chilled beam cooling," *ASHRAE Journal*, 9 2007.

[11] W. Bingjie, "Air distribution and thermal comfort for active chilled beam systems," 2018, doi: 10.32657/10220/46679.

[12] K. J. Loudermilk and D. S. Alexander, "Efficient space humidity control with active chilled beam systems," *ASHRAE Journal*, vol. 54, pp. 28–38, 2012.

[13] A. Livchak and C. Lowell, "Don't turn active beams into expensive diffusers," *ASHRAE Journal*, pp. 52–60, 2012.

[14] D. Alexander and M. O´Rourke, "Design considerations for active chilled beam," *ASHRAE Journal*, vol. 50, 9 2008. [Online]. Available: https://www.researchgate.net/publication/237109317_Design_Considerations_For_Active_Chilled_Beams

[15] C. Chen, W. Cai, Y. Wang, and C. Lin, "Performance comparison of heat exchangers with different circuitry arrangements for active chilled beam applications," *Energy and Buildings*, vol. 79, pp. 164–172, 8 2014, doi: 10.1016/j.enbuild.2014.04.048.

[16] K. W. Roth, D. Westphalen, J. Dieckmann, S. D. Hamilton, and W. Goetzler, "Energy consumption characteristics of commercial building hvac systems - volume iii: Energy savings potential," 7 2002.

[17] P. Rumsey, "European technology taking hold in the u.s.: Chilled beams," *HPAC Engineering*, pp. 46–48, 1 2010.

[18] K. Loudermilk, "Chilled beams for thermal comfort," *ASHRAE Journal*, vol. 51, pp. 58–64, 10 2009.

[19] J. Scanes, "All-air VAV system versus chilled-beam-system energy consumption at design and part-load conditions," *Consulting-Specifying Engineer*, 12 2016. [Online]. Available: https://www.csemag.com/articles/all-air-vav-system-versus-chilled-beam-system-energy-consumption-at-design-and-part-load-co

[20] TROX, *Chilled Beam Design Guide*. TROX USA, 4 2014. [Online]. Available: https://www.yumpu.com/en/document/view/27289523/chilled-beam-design-guide-trox

[21] A. Katsnelson, "Chilled beam ceiling systems promise energy, maintenance savings," 8 2007. [Online]. Available: https://www.facilitiesnet.com/hvac/article/Chilled-Beam-Ceiling-Systems-Promise-Energy-Maintenance-Savings--7219

[22] P. Filipsson, A. Trüschel, J. Gräslund, and J. O. Dalenbäck, "Modelling of rooms with active chilled beams," *Journal of Building Performance Simulation*, vol. 13, pp. 409–418, 7 2020, doi: 10.1080/19401493.2020.1752801.

[23] P. Filipsson, "Self-regulating Active Chilled Beams," Ph.D. dissertation, Chalmers University of Technology, Gothenburg, Sweden, 2020.

[24] "Sveriges meteorologiska och hydrologiska institut."

[25] S. Gehlin and O. Andersson, "Geothermal energy use, country update for sweden." Strasbourg: European Geothermal Congress, 9 2016.

[26] B. Rosén, A. Gabrielsson, J. Fallsvik, G. Hellström, and G. Nilsson, "System för värme och kyla ur mark. en nulägesbeskrivning," 2001. [Online]. Available: https://www.diva-portal.org/smash/record.jsf?pid=diva2%3A1300412

[27] R. Kosonen and 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*, vol. 26, pp. 256–271, 2 2017, doi: 10.1177/1420326X16683236.

[28] J. Urchueguía, M. Zacarés, J. Corberán, Montero, J. Martos, and H. Witte, "Comparison between the energy performance of a ground coupled water to water heat pump system and an air to water heat pump system for heating and cooling in typical conditions of the european mediterranean coast," *Energy Conversion and Management*, vol. 49, pp. 2917–2923, 10 2008, doi: 10.1016/j.enconman.2008.03.001.

[29] L. R. Ingersoll, O. J. Zobel, and A. C. Ingersoll, *Heat Conduction: With Engineering, Geological and Other Applications*. McGraw-Hill Book Co., 1948.

[30] H. Latif, G. Hultmark, S. Rahnama, A. Maccarini, and A. Afshari, "Performance evaluation of active chilled beam systems for office buildings – a literature review," *Sustainable Energy Technologies and Assessments*, vol. 52, 8 2022, doi: 10.1016/j.seta.2022.101999.

[31] J. Murphy, "Understanding chilled beam systems," *Trane Engineers Newsletters*, vol. 38, 2009. [Online]. Available: https://www.trane.com/content/dam/Trane/Commercial/global/products-systems/education-training/engineers-newsletters/airside-design/adm_apn034en_1209.pdf

[32] J. Halvarsson, "Occupancy pattern in office buildings," 1 2012. [Online]. Available: http://hdl.handle.net/11250/234598

[33] "Energy standard for site and buildings except low-rise residential buildings," 2022.

[34] T. Arghand, S. Javed, A. Trüschel, and J.-O. Dalenbäck, "A comparative study on borehole heat-exchanger size for direct ground-coupled cooling systems using active chilled beams and tabs," *Energy and Buildings*, vol. 240, 6 2021, doi: 10.1016/j.enbuild.2021.110874.

[35] T. Arghand, S. Javed, and J.-O. Dalenbäck, "Combining direct ground cooling with ground-source heat pumps and district heating: Borehole sizing and land area requirements," *Geothermics*, vol. 106, 12 2022, doi: 10.1016/j.geothermics.2022.102565.

[36] EQUA Simulation Technology Group, "IDA ICE Getting Started," EQUA Simulation AB, Solna, Sweden, 2018. [Online]. Available: https://www.equaonline.com/iceuser/pdf/ICE48GettingStartedEng.pdf

[37] S. Kropf and G. Zweifel, "Validation of the Building Simulation Program IDA-ICE CEN 13791," Luzern, Switzerland, 2001. [Online]. Available: https://www.equaonline.com/iceuser/validation/ICE_vs_prEN%2013791.pdf

[38] EQUA Simulation Technology Group, "Validation of IDA Indoor Climate and Energy 4.0 build 4 with respect to ANSI/ASHRAE Standard 140-2004," Solna, Sweden, 2010. [Online]. Available: https://www.equaonline.com/iceuser/validation/ASHRAE140-2004.pdf

[39] S. Moosberger, "IDA ICE CIBSE - Validation Test of IDA Indoor Climate and Energy version 4.0 according to CIBSE TM33, issue 3," Luzern, Switzerland, 2007. [Online]. Available: https://www.equaonline.com/iceuser/validation/ICE-Validation-CIBSE_TM33.pdf

[40] G. Hellström, B. Sanner, T. Blomberg, and J. Claesson, "Earth Energy Designer," BLOCON Company, Lund, Sweden, 2022. [Online]. Available: https://buildingphysics.com/manuals/EED4.pdf

[41] M. Kottek, J. Grieser, C. Beck, B. Rudolf, and F. Rubel, "World map of the köppen-geiger climate classification updated," *Meteorologische Zeitschrift*, vol. 15, pp. 259–263, 6 2006, doi: 10.1127/0941-2948/2006/0130.

[42] W. B. Group, "Climate change knowledge portal." [Online]. Available: https://climateknowledgeportal.worldbank.org/country/sweden

[43] "International Weather for Energy Calculations, v 2.0," ASHRAE. [Online]. Available: https://www.equaonline.com/ice4user/indexIWEC2.html

[44] Arbetsmiljöverkets författningssamling, "Arbetsplatsens utformning," 6 2020. [Online]. Available: https://www.av.se/arbetsmiljoarbete-och-inspektioner/publikationer/foreskrifter/arbetsplatsens-utformning-afs-20201-foreskrifter/
[45] Engineering ToolBox, "Cooling and heating equations." [Online]. Available: https://www.engineeringtoolbox.com/cooling-heating-equations-d_747.html
[46] Engineering ToolBox, "Enthalpy of vaporization calculator." [Online]. Available: https://www.engineeringtoolbox.com/water-properties-d_1573.html
[47] SWEGON, "Product calculator." [Online]. Available: https://spc.rud.swegon.com/?productId=68869d3c-6485-47d6-bcb2-fd282d13c02e&language=en-GB
[48] T. Arghand, "Direct-ground cooling systems for office buildings," Master's thesis, Chalmers University of Technology, Gothenburg, Sweden, 2019.
[49] Åf Cost Management, *Åfs Lilla Prisbok*, 2018.
[50] Adicot, "Duct size calculator." [Online]. Available: https://www.adicotengineering.com/duct-size-calculator
[51] Swedish Standards Institute, "Energy performance of buildings-Economic evaluation procedure for energy systems in buildings-Part 1: Calculation procedures, Module M1-14," 6 2017. [Online]. Available: https://www.sis.se/std-8027041
[52] "Göteborg energi." [Online]. Available: https://www.goteborgenergi.se/foretag
[53] J. Vastyan, "Chilled-beam basics," *HPAC Engineering*, 7 2011. [Online]. Available: https://www.tacocomfort.com/documents/FileLibrary/ChilledBeamBasics_HPAC_Engineering_2011-7.pdf
[54] P. Rumsey and J. Weale, "Chilled beams in labs eliminating reheat saving energy on a budget," *ASHRAE Journal*, 2007. [Online]. Available: https://www.solaripedia.com/files/509.pdf
[55] T. Arghand, S. Javed, A. Trüschel, and J.-O. Dalenbäck, "Cooling of office buildings in cold climates using direct ground-coupled active chilled beams," *Renewable Energy*, vol. 164, pp. 122–132, 2 2021, doi: 10.1016/j.renene.2020.09.066.
[56] T. Arghand, S. Javed, A. Trüschel, and J.-O. Dalenbäck, "Influence of system operation on the design and performance of a direct ground-coupled cooling system," *Energy and Buildings*, vol. 234, 3 2021, doi: 10.1016/j.enbuild.2020.110709.
[57] Y. Yau and J. Tam, "A comparison study for active chilled beam and variable air volume systems for an office building," *Energy Procedia*, vol. 152, pp. 378–383, 10 2018, doi: 10.1016/j.egypro.2018.09.160.
[58] P. Filipsson, A. Trüschel, J. Gräslund, and J.-O. Dalenbäck, "Performance evaluation of a direct ground-coupled self-regulating active chilled beam system," *Energy and Buildings*, vol. 209, 2 2020, doi: 10.1016/j.enbuild.2019.109691.
[59] T. Lindholm, "Frikyla – Analys av Teknik och Systemlösningar," Energimyndigheten, Tech. Rep., 2004. [Online]. Available: https://varmtochkallt.se/wp-content/uploads/Projekt/effsys/4.%20Kylsystem/SR-Hs20-Frikyla.pdf
[60] H. Acül, "Energy efficient free cooling systems for air conditioning and process cooling systems," 10 2007. [Online]. Available: https://www.iklimsoft.com/public-uploads/ptax8EsbZO.251115.230125.pdf

[61] S. Gehlin and O. Andersson, "Geothermal energy use, country update for sweden." Haag: European Geothermal Congress, 6 2019.

[62] J. Stein and S. Taylor, "VAV Reheat Versus Active Chilled Beams and DOAS," UC Berkeley, Tech. Rep., 2013. [Online]. Available: http://escholarship.org/uc/item/1jh193x3

[63] P. Rumsey, N. Bulger, J. Wenisch, and T. Disney, "Chilled beams in laboratories: Key strategies to ensure effective design, construction, and operation." [Online]. Available: https://www.i2sl.org/documents/toolkit/bp_chilled-beam_508.pdf

[64] International Energy Agency, "Space cooling: More access, more comfort, less energy," International Energy Agency, Paris, France, Tech. Rep., 2017. [Online]. Available: https://iea.blob.core.windows.net/assets/9cff2dd2-ceba-4bf2-a77e-9def8a74b80c/SpaceCoolingEnergyEfficiencyInsightsBrief.pdf

[65] R. Norouzi, G. Hultmark, A. Afshari, and N. C. Bergsøe, "Study of a two-pipe chilled beam system for both cooling and heating of office buildings," 5 2013.

[66] S. Seyam, "Types of HVAC Systems," in *HVAC System*, M. S. Kandelousi, Ed. Rijeka: IntechOpen, 2018, ch. 4, doi: 10.5772/intechopen.78942.

A
Appendix 1

Table 13

Operating hours

	Monday-Friday		Saturday	
	hours	intensity	hours	intensity
Open office space				
Occupants	07:30 - 12:00	1,0	11:00 - 13:00	0,25
	12:30 - 17:30	1,0		
Lights	07:30 - 18:00	1,0	11:00 - 13:00	0,25
Equipment	07:30 - 17:30	1,0	11:00 - 13:00	0,25
Kitchen				
Occupants	07:00 - 16:00	1,0		
Lights	07:00 - 16:00	1,0		
Equipment				
Restaurant				
Occupants	08:30 - 11:00	0,25		
	11:00 - 11:30	0,50		
	11:30 - 13:00	1,0		
	13:00 - 14:00	0,50		
	14:00 - 16:00	0,25		
Lights	08:30 - 16:00	1,0		
Equipment	08:30 - 16:00	1,0		
Common space				
Occupants				
Lights	07:30 - 17:30	1,0	11:00 - 13:00	1,0
Equipment				
Group room				
Occupants	08:30 - 12:00	1,0		
	12:30 - 16:00	1,0		
Lights	08:30 - 16:00	1,0		
Equipment	08:30 - 16:00	1,0		
Mechanical room				
Occupants	07:30 - 12:00	1,0	11:00 - 13:00	0,50
	12:30 - 17:30	1,0		
Lights	07:30 - 17:30	1,0	11:00 - 13:00	1,0
Equipment	00:00 - 23:59	1,0		

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