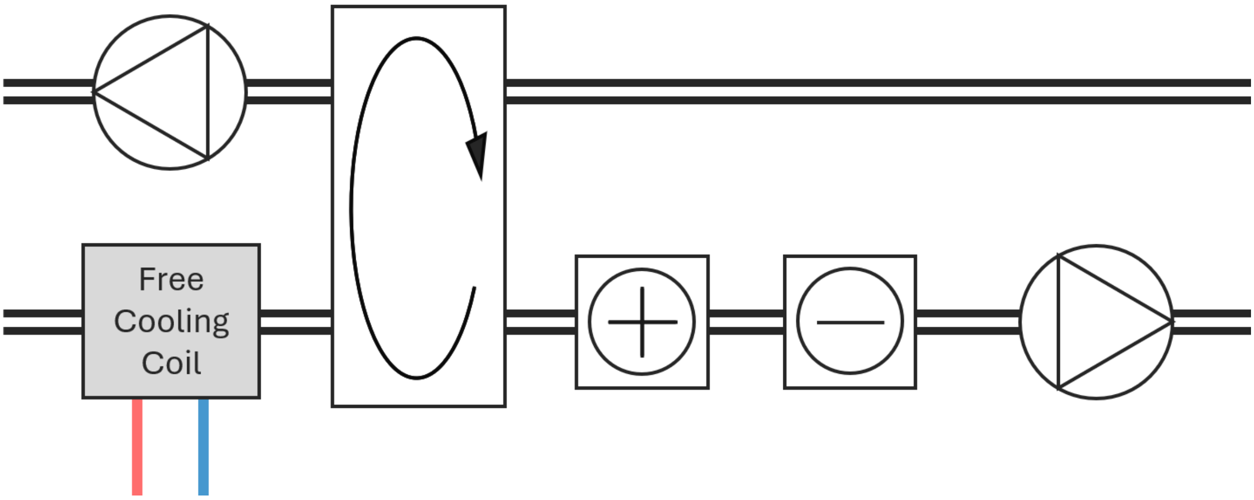

Space Cooling using Active Chilled Beams and Free Cooling by Outdoor Air

A Performance Assessment Based on Simulations in IDA ICE

Master's thesis in Structural Engineering and Building Technology

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Gothenburg, Sweden 2024
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Cover: Implementation of free cooling by outdoor air as a component in the air handling unit.

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Abstract

Comfort cooling is increasingly vital, particularly with rising ambient temperatures and growing demands for indoor environmental quality, significantly impacting a building's energy performance. Consequently, there is a need to reduce purchased cooling through the utilization of ambient free cooling sources. This study explores the option of free cooling by outdoor air in a space cooling system using active chilled beams (ACB).

The objective of this thesis is to develop a system model in IDA ICE that extract cooling from ambient air by utilizing the air flow of the existing air handling unit, substituting purchased district cooling with free cooling from outdoor air. A typical office building in Gothenburg was simulated with the developed system model to conduct a quantitative evaluation of the system's energy and economic performance.

This study concludes that the system's performance, both in terms of energy and economics, is not viable for the designed building. While the available cooling capacity during the cooler periods of the year is abundant, the cooling demand for a typical office during this period is not sufficient for the free cooling system to significantly reduce either specific or primary energy usage. A major part of the cooling demand occurs during the summer when the outdoor air temperature is too high for the free cooling system to be effective. This low reduction, combined with a high investment cost, results in an investment that does not show a foreseeable return.

Keywords: active chilled beams, comfort cooling, energy performance, free cooling, IDA ICE, life-cycle cost, office space, outdoor air

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We would like to express our sincere and utmost gratitude to our supervisor Giovana Fantin Do Amaral Silva, for her patience, guidance and feedback from start to finish. We would also like to extend our sincere appreciation to all the colleagues at Bengt Dahlgren's office in Gothenburg who provided us with their assistance and time while also making our stay incredibly welcoming and enjoyable. We want to thank our examiner Jan-Olof Dalenbäck, for his consistent support, prompt responsiveness, active engagement, and of course the banter. Lastly, we would like to thank Chalmers for an incredible experience over the past five years.

Axel Gunnemyr and Fabian Hofmann, Gothenburg, June 2024

List of Acronyms

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

ACB	Active Chilled Beams
AHU	Air Handling Unit
BBR	Boverkets Byggregler
CAV	Constant Air Volume
HVAC	Heating, Ventilation, & Air Conditioning
HX	Heat Exchanger
IEQ	Indoor Environmental Quality
IRR	Internal Rate of Return
LCA	Life Cycle Analysis
LCC	Life Cycle Cost
LCCA	Life Cycle Cost Analysis
NPV	Net Present Value
PCB	Passive Chilled Beams
PPD	Predicted Percentage Dissatisfied
PI	Proportional-Integral
PVA	Present Value of Annuity
RCP	Representative Concentration Pathways
SFP	Specific Fan Power
VAV	Variable Air Volume

Nomenclature

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

Indices

i	Index for energy carrier
t	Index for time step

Sets

$i \in \{1, 2, 3\}$	The set of energy carriers from 1 to 3
$t \in \{0, 1, 2, \ldots, T\}$	The set of time periods from 0 to T

Parameters

cp_{air}	Specific heat capacity of air
CF	Net cash flow
F_{geo}	Geographical adjustment factor
r	Discount rate
ρ_{air}	Density of air
T	Calculation time period

Variables

A_{floor}	Floor area
$E_{h,i}$	Energy for space heating

$E_{c,i}$	Energy for air conditioning
$E_{tw,i}$	Energy for hot tap water
$E_{f,i}$	Property energy
CF_t	Net cash flow (varying by year)
$n_{occupants}$	Number of occupants
PE_i	Primary energy factor per energy carrier
$T_{air,in}$	Temperature of air entering the free cooling coil
$T_{air,out}$	Temperature of air exiting the free cooling coil
q_{air}	Air flow
x	Yearly price increase

Contents

List of Figures

List of Tables

1
Introduction

This chapter's main purpose is to describe the study's relevance and intention. First, a background is composed to briefly describe the underlying motive of the thesis. Second, the study's purpose and objectives are summarized through research questions. Following, a section on scope and limitations is included to establish the study's boundaries. Finally, the study's structure and research approach are presented through the research methodology.

1.1 Background

The demand for comfort cooling in commercial buildings worldwide is expected to more than double by 2050 compared to today (International Energy Agency, 2018). This increase in demand is primarily fueled by factors such as increased standard of life, affordability for air conditioning, and the increasing temperatures globally (Santamouris & Kolokotsa, 2013). Innovative cooling solutions are going to be needed to address this escalating demand in a sustainable way.

The two main methods to provide comfort cooling are either through a variable air volume (VAV) system or a separate water-based (hydronic) system (Warfvinge & Dahlblom, 2010). Both methods increase the energy use since a VAV-system requires additional fan energy, and a hydronic system requires pump energy for the circulating chilled water. Both methods require cooling energy bought via district cooling or locally produced.

An effective strategy for lowering cooling costs involves utilizing ambient sources with naturally lower temperatures to cool the hydronic loop, a method commonly referred to as free cooling. Oftentimes, free cooling is extracted from the ground through boreholes. It is usually an effective strategy since the bedrock maintains a constant, low temperature throughout the year (SGU, 2023). However, there are cases where this method is unattainable due to extreme soil depths or drilling restrictions. A more versatile system, theoretically implementable in all scenarios, involves incorporating free cooling by outdoor air within the air handling unit (AHU). Such a system provides free cooling for the hydronic system while also preheating the supply air right after intake. Given its versatility, it is of interest to determine whether this system can provide sufficient cooling and be a profitable investment.

1.2 Purpose and Objectives

The thesis aims to extensively investigate the capabilities of utilizing free cooling by outdoor air in combination with active chilled beams (ACB). A case study is done on a fictional office building in Gothenburg, to evaluate the system's performance from different perspectives. Relevant objectives are established to fulfill the study's purpose. The objectives are connected to the three pillars of sustainability: economic, environmental, and social. First, to make the system economically feasible, the solution must be profitable. The system's profitability is thereby evaluated through a life-cycle cost (LCC) analysis. Second, the system must contribute to an improvement environmentally and its energy performance is therefore measured. Third, the free cooling system is required to provide an equally good indoor climate as a conventional system.

Since the outdoor temperatures are expected to rise substantially over the upcoming years, an evaluation of the free cooling system's ability to perform under those conditions is done. At last, the system is tested for different floor plans with unequal occupancy to evaluate the system's sensitivity to internal gains.

All performance evaluations are done using the simulation software IDA ICE 5.0. The evaluations are performed by comparing two identical cases, with the only difference being the inclusion of the free cooling system. By utilizing simulation software to conduct the study, the resulting model can serve as a valuable tool for future projects and research.

1.3 Research Questions

Based on the purpose and objectives, the following research questions are stated to emphasize the study's goal:

- Does the study's free cooling system improve the energy performance of a typical office space located in Gothenburg, Sweden?
- Is the study's free cooling system a profitable investment for a typical office space located in Gothenburg, Sweden?
- With the rising climate temperatures, what performance can be expected from the study's free cooling system in the future?
- To what extent do the internal gains affect the system's performance?

1.4 Research Methodology

The previously stated research questions acted as an indication when deciding on the methodological approach. A quantitative approach was mostly performed throughout the study, in line with the research questions. Quantitative research is defined as gathering and investigating numerical data, which is expected in this type of study (Sheard, 2018). A literature review was carried out to acquire relevant information about the subject. However, all decisions and conclusions in this paper were based on numerical results.

The mentioned numerical results were generated through data simulations. Several precautions were taken to ensure highly credible and reliable results. First, the selected simulation software IDA ICE is validated against standards such as EN 15255 and EN 15256 (EQUA, 2024). The program is also widely used in the industry, indicating high performance and accuracy. Second, the input data was carefully selected and assured to match real-world conditions. It was also ensured that all input data was identical in each test case. Last, modifications in the free cooling model were criticized in each iteration to eliminate potential issues. Additionally, the results were validated for physical plausibility through hand calculations.

Considering the amount of undefined parameters, several sensitivity analyses were conducted to ensure the robustness of the simulation outcomes. While every effort was made to ensure the validity and reliability of the results, it is essential to recognize that uncertainties and simplifications may have influenced the outcome.

1.5 Scope and Limitations

The paper's scope and limitations are stated to show the study's focus and the level of detail to be expected. Every aspect is described in the following sections.

This study's scope is limited to the two floor plans. Thus the results are limited to the specific parameters each building case is subject to. These parameters encompass occupancy, façade construction, geometry, thermal capacity, and operation. Changes to these parameters could render the case more or less suitable for the free cooling system. Additionally, it's worth noting that neither of the cases includes connections to the ground or roof, thereby excluding any energy losses to or from these elements in the simulations. For practical reasons, all simulations only account for one floor. However, all results are extrapolated for five equal floors, in line with a typical office building. This simplification is expected to marginally reduce the result's accuracy.

The case study is also bound by geographical temperate climate, as its location is set to Gothenburg. Changes in geographical location could affect the case study's results, due to differences in ambient conditions, solar intensity, and day cycles. Primary ambient conditions are outdoor temperatures and humidity levels, as both these can limit the possibility of utilizing free cooling.

One of the study's research questions is to determine the system's impact on energy performance. Although the system's design theoretically suggests only a minimal impact on heating, this aspect is disregarded since the study's focus is limited to cooling.

A limitation arises from the absence of empirical data in the case study, posing a challenge to validate the study's findings. In the absence of real case data, comparing simulations of a reference- and a free cooling case becomes the primary approach. While this method offers insights into the system's potential, the absence of empirical data precludes thorough validation.

Another limitation stems from the discrepancy between the design of the free cooling system in the simulation program IDA ICE and its real-world counterparts in terms of design, system, and installation performance. Consequently, the results offer insights into potential outcomes but may not accurately reflect real-world scenarios.

A further limitation arises from the absence of an advanced control system for the free cooling system, which could regulate flow based on cooling demand. In the simulated cases, a constant flow is maintained during periods where free cooling is applicable. This limitation suggests the potential for enhancing system efficiency further through dynamic flow regulation. This is however ignored due to the complexity of the control systems and implementation into the IDA model.

2 Theoretical Background

A theoretical background is written to contextualize the study. First, there is a description of comfort cooling. This information is presented to justify the cooling system chosen for the study. Second, the term *free cooling* is described to give an understanding of the concept and potential implementations. The working principle of different free cooling systems is presented to validate the relevance of this study's chosen system. A description of primary energy number and finance is also included to act as a foundation for the interpretation of the results.

2.1 Comfort Cooling

For commercial buildings, there is a minimum ventilation requirement often referred to as the hygienic air flow (Arbetsmiljöverket, 2020). The hygienic air flow's purpose is to ensure hazardous substances, moisture, and unpleasant odors are carried away properly (Boverket, 2021). Keeping indoor temperatures at the desired level remains challenging, particularly in commercial buildings due to large internal gains. The generated heat surplus necessitates additional cooling, usually provided via comfort cooling systems. Comfort cooling systems cool using one of two main principles, either through an air- or water-based (hydronic) system (Warfvinge & Dahlblom, 2010).

Air cooling is typically achieved by supplying variable air volume (VAV), much larger than the hygienic requirement. The air flow is designed to vary on factors such as indoor temperature, CO_2 levels, or presence. Larger fans are required to supply the increased air flow, resulting in increased electricity use. However, due to the variable element, the air flow can also be lowered depending on demand, promoting energy savings. Hydronic cooling generally operates as a separate system, in conjunction with a constant air volume (CAV) system that supplies hygienic air flow. Water-based cooling is based on the principle of transferring away heat using a circulating liquid. The running cost primarily comes from cooling the circulating refrigerant, which is often done via district cooling or a local chiller. Given that the cooling revolves around the water loop exchanging heat with a colder source, alternative cooling methods are considered suitable for implementation. Four common water-based systems are further discussed, see Figure 2.1 for an overview.

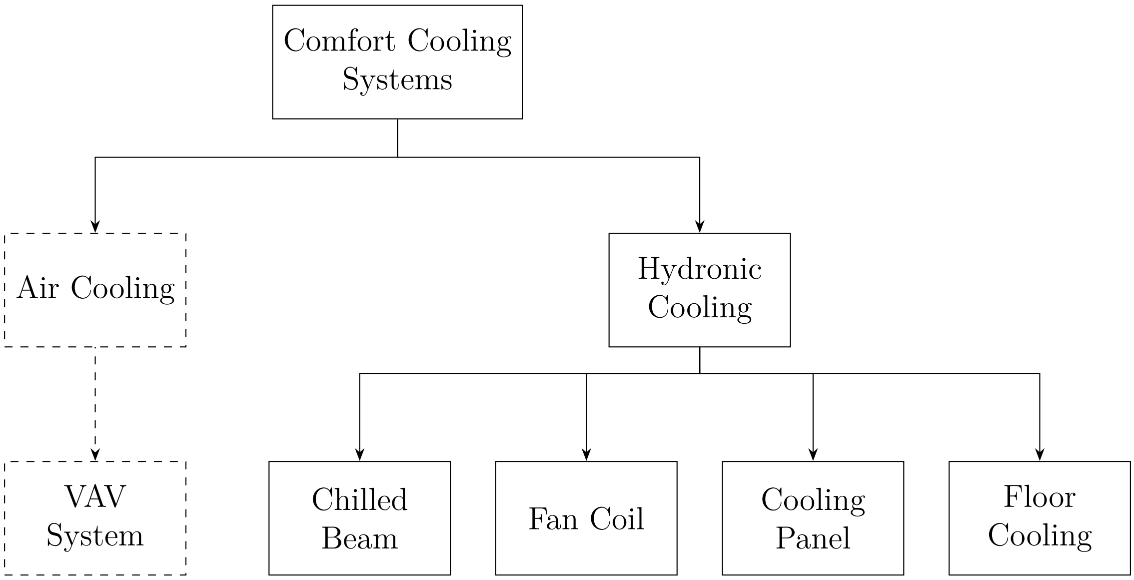


Figure 2.1: Schematic diagram for comfort cooling systems.

2.1.1 Chilled Beam

Chilled beams are roof-mounted units that utilize convective heat transfer to carry away heat. Convective heat transfer is defined as the movement of heat between a surface and a fluid in motion (Pinterić, 2017). Warm air has a lower density than cold, making it rise due to buoyancy, effectively acting as a fluid in motion. As heat is carried away in the chilled beam, the air temperature lowers and a circulation phenomenon occurs called natural convection, see Figure 2.2.

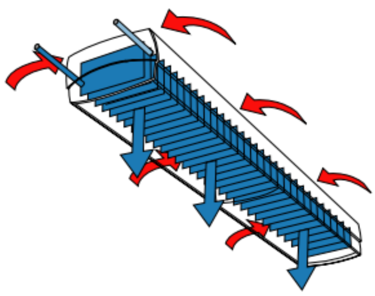


Figure 2.2: Natural convection for a passive chilled beam (SWEGON, 2007).

Beams utilizing natural convection are called passive beams. Passive beams can be a rather simple alternative since they do not require any additional installations. However, due to their passive nature, the cooling effect and controllability can be a limiting factor.

2.1.1.1 Active chilled beam

An active chilled beam (ACB) is characterized by having integrated supply air, resulting in an increased cooling effect driven by forced convection. Air is supplied to the primary air plenum with an overpressure of 30-120 Pa (Latif et al., 2022). The overpressure forces primary air to exit the plenum and enter the room at high velocity, resulting in an underpressure in the mixing chamber. Consequently, room air is forced to flow through the chilled water pipes, into the mixing chamber. Heat is transferred away and the mixed air is resupplied to the room at a lower temperature. The process is shown in Figure 2.3.

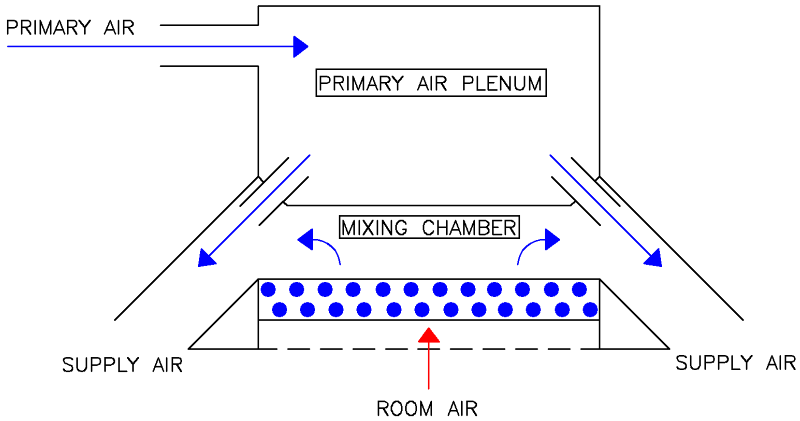


Figure 2.3: Working principle of an active chilled beam.

One important aspect of ACB is that it is a water-based system, highly influenced by its surrounding environment. A conventional ACB is designed to have as large of a temperature difference as possible between the supply and return water, maximizing the cooling ability. However, these temperatures are in reality limited for two reasons. First, due to the risk of condensation on the water pipes, the lowest possible supply water temperature must be kept above the room dew point temperature (Latif et al., 2022). For context, air which is 20°C and has an relative humidity (RH) of 70% has a dew point temperature of 14°C (Hagentoft & Sandin, 2017). Since ACBs are highly dependent on the room conditions, a control system adjusting the supply water temperature is required. Second, the return temperature must not be increased by more than 4.7°C to ensure a proper indoor environmental quality (IEQ) (Bingjie, 2018). A larger temperature difference will likely result in a low operative temperature in the room, affecting the indoor climate negatively.

2.1.1.2 Self-regulating active chilled beam

An alternative to the conventional ACB is the subvariant called *self-regulating active chilled beam*. The principle is that a constant water flow is used without any thermostat control or control valve (Filipsson, 2020). This allows the liquid in the active beams to have a higher temperature while still providing ample comfort cooling.

The cooling capacity of an active chilled beam depends on the temperature gap between the room air and chilled water (Filipsson, 2020). When the room air temperature rises, it leads to a larger temperature difference and thus increases the cooling capacity. This phenomenon is known as self-regulation. Self-regulating beams are dimensioned for water temperatures closer to the ideal indoor temperatures, approximately 20°C. The higher water temperature results in the beams being more sensitive to changes in room temperature and able to provide sufficient cooling through the mentioned self-regulation phenomenon.

Self-regulating systems offer simplicity and potential investment cost savings by reducing the need for control equipment. However, the lower temperature delta makes up a need for a greater heat transfer area. The greater heat transfer area means that larger beams must be used, which can lead to higher investment costs. Knowing this, detailed cost calculations must be done to evaluate its profitability. An additional drawback is that there might be an increased pump work compared to a conventional ACB system. As a conclusion, self-regulating ACB systems involves a trade-off between heat transfer area and chilled water supply temperature, with higher supply temperatures benefiting self-regulation.

2.1.2 Fan Coil

Fan coils are cooling units designed to regulate temperature within indoor spaces by circulating air over a coil. These units typically consist of a fan, which draws air from the room and passes it over the coil, transferring heat from the air (Warfvinge & Dahlblom, 2010). It operates much like an ACB, the main difference being that no supply air is directly connected to the unit. This opens up the possibility for flexible physical placement and usage in spaces without mechanical supply air. Figure 2.4 shows the unit's working principle.

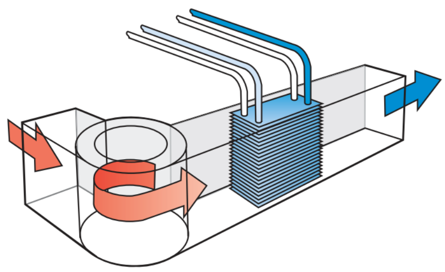


Figure 2.4: Working principle of a fan coil (SWEGON, 2007).

2.1.3 Ceiling Cooling Panel

Ceiling-mounted cooling panels carry away heat using radiation, see Figure 2.5. Radiation transfers energy through electromagnetic waves and does not require moving fluid (Pinterić, 2017). A typical example of radiant heat transfer is the sun heating the earth. Radiant cooling can be best explained as "negative" radiant heating, meaning the heat radiated by your body and any other objects in the room is absorbed by the panels, creating an overall sensation of comfort (Messana, 2024). Even though the main driving force is radiation, natural convection also contributes to part of the cooling effect.

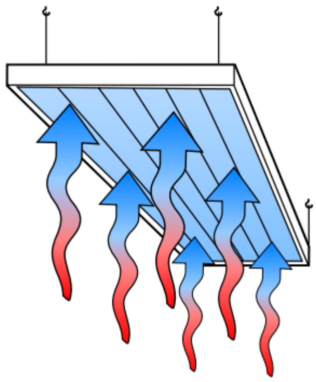


Figure 2.5: Working principle of a cooling panel (SWEGON, 2007).

The panels are often made from aluminum which is in contact with cooling coils that carry away the heat, creating a persistent, cold surface inside the room (SWEGON, 2007). A high enough water temperature must be kept to reduce the risk of condensation and prevent thermal discomfort through low operative temperatures.

2.1.4 Floor Cooling

Many are familiar with floor heating systems where circulating water pipes are placed underneath the floor, inside the slab (Olesen, 2008). By utilizing a lower water temperature, this type of system can be used for floor cooling as well. The circulating water makes the floor act like a large cooling panel. Due to the large surface area, there will be a cooling effect even when the water temperature is close to the desired indoor temperature. This is essential since a lower water temperature would result in a low floor temperature, certainly not desirable for a sufficient indoor climate. A temperature of 19/20°C is recommended for sedimentary or standing people wearing normal shoes (Olesen, 2008).

2.2 Free Cooling

The concept of free cooling is to reduce or eliminate the need to produce or buy cooling, by taking advantage of nearby naturally occurring cooling sources (SWEGON, 2024). A simple example is to fulfill a building's cooling demand by utilizing cool, ambient outdoor air, eliminating the need to purchase from the district cooling network. Free cooling can be gathered from different sources, and be utilized through different system principles. A handful of systems are described further to provide a deeper understanding of the concept.

2.2.1 Night Cooling

The concept of night cooling is to take advantage of the naturally low temperatures during the night, to proactively cool a given space (Warfvinge & Dahlblom, 2010). It is utilized when the temperature difference between the day and night is large, which is the case during summer. The idea is to reduce the mechanical cooling needed during the day, by having the existing ventilation system turned on during parts of the night. The fan's energy usage is most often marginal in comparison to the energy saved by not needing the cooling systems as much during the day. This is a simple and effective way to utilize free cooling since it relies on the already installed ventilation system.

Another concept of night cooling is to funnel the supply air through the hollow cores of the slab before entering the room (Termodeck, 2024). This way, the heavy slabs of the building are able to keep cool during warm periods, resulting in less cooling need during the day.

The effectiveness of night cooling is influenced by the building's thermal inertia. Thermal inertia, determined by the building's construction materials and mass, dictates its capacity to absorb, store, and release heat over time. Buildings with high thermal inertia exhibit slower temperature fluctuations, thereby enhancing the efficacy of night cooling by extending the duration of cooler indoor conditions. Likewise, structures with low thermal inertia may experience rapid temperature changes, potentially limiting the effectiveness of night ventilation.

2.2.2 Free Cooling by Local Water Source

Free cooling from a local water source utilizes ambient water sources such as lakes or seas (Warfvinge & Dahlblom, 2010). Cold water is retrieved from the source, cooling down the water pipes in the hydronic system via a heat exchanger (HX). The system has great potential to be efficient since the water temperatures are significantly lower than the air temperatures. However, the main drawback of this system is its limited use case since it relies on having close access to a sea or lake. It is therefore more often used in terms of district cooling (Energiföretagen Sverige, 2017). Large cooling facilities can be built close to lakes and distribute cooling via a district cooling network. District cooling can therefore be considered free cooling from an environmental point of view, but not from an economical one since cooling supplied by the district network must be paid for.

2.2.3 Free Cooling by Ground Heat Exchanger

Ground heat exchanger (GHX) commonly referred to as boreholes can be used as a free cooling- and heating source. Due to the massive thermal inertia provided by the soil, the ground maintains a stable temperature year-round, remaining nearly constant at a specific depth (Florides & Kalogirou, 2007). In essence, the ground acts as a reservoir for cooling: during summers, surplus building heat is directed into the ground via boreholes, slightly elevating its temperature. Conversely, in winters, the cold outdoor air cools the ground, offsetting the heat accumulated during summer. The ground temperature is determined by the annual mean temperature, making the system's efficiency vary a lot depending on the location (Hagentoft & Sandin, 2017). The annual mean temperature in Gothenburg is about 8°C, indicating that it has the potential to be an effective solution at this location.

Free cooling from boreholes is commonly implemented as a closed-loop system, where heat is exchanged between the building's hydronic system and the surrounding ground through a network of boreholes (Florides & Kalogirou, 2007). In this system, a GHX facilitates the transfer of heat from the building's hydronic system to a circulating fluid, typically water or a water-glycol mixture. The fluid then carries the heat to the boreholes, where it is released into the colder ground.

The implementation of free cooling via boreholes comes with challenges, primarily related to the drilling process. The drilling process may be constrained by factors such as site accessibility, geological conditions, and regulatory requirements. These limitations can affect the feasibility and scalability of boreholes, particularly in urban environments where space may be limited.

2.2.4 Free Cooling by Outdoor Air

In regions where the outdoor temperature is lower than the desired indoor temperature, the idea of using ambient outdoor air as a cooling source becomes applicable. Free cooling by outdoor air can be implemented as direct or indirect cooling. Direct cooling is when cold outdoor air is directly drawn into the building when below a setpoint temperature, sharing similarities with night cooling. Naturally, two benefits of this system are simplicity and efficiency. However, the drawbacks are possible humidity risks and high fan power usage (Zhang et al., 2014). The use case might be limited due to poor stability in indoor environmental quality when directly inducing cold air from outdoors.

Indirect cooling is a system that utilizes a closed chilled water loop, connected to the indoor cooling system. It can exchange excess heat with the outdoor air through an HX. This removed the drawbacks of humidity risks that the direct cooling system has at the cost of simplicity and efficiency. As Gothenburg has a temperature below or at 10°C approx. 57% of the year and below or at 15°C approx. 80% of the year (Sveby, 2016), the city has the potential to implement this free cooling system. The main issue with this method is that the largest cooling needs often occur during summer - when the outdoor temperatures are high. Therefore, it is likely best utilized in buildings with large internal gains year-round.

A possible implementation of an indirect free cooling system by outdoor air is by installing an air-to-liquid HX into the air handling unit (AHU). The concept is illustrated in Figure 2.6 and described by the following bullet points:

- A free cooling coil is placed inside the AHU, exchanging heat between the outdoor air and a hydronic loop.
- As heat is transferred away, the liquid's temperature in the hydronic loop is lowered.
- The liquid enters the cooling source at a lower temperature, effectively reducing the cooling demand.

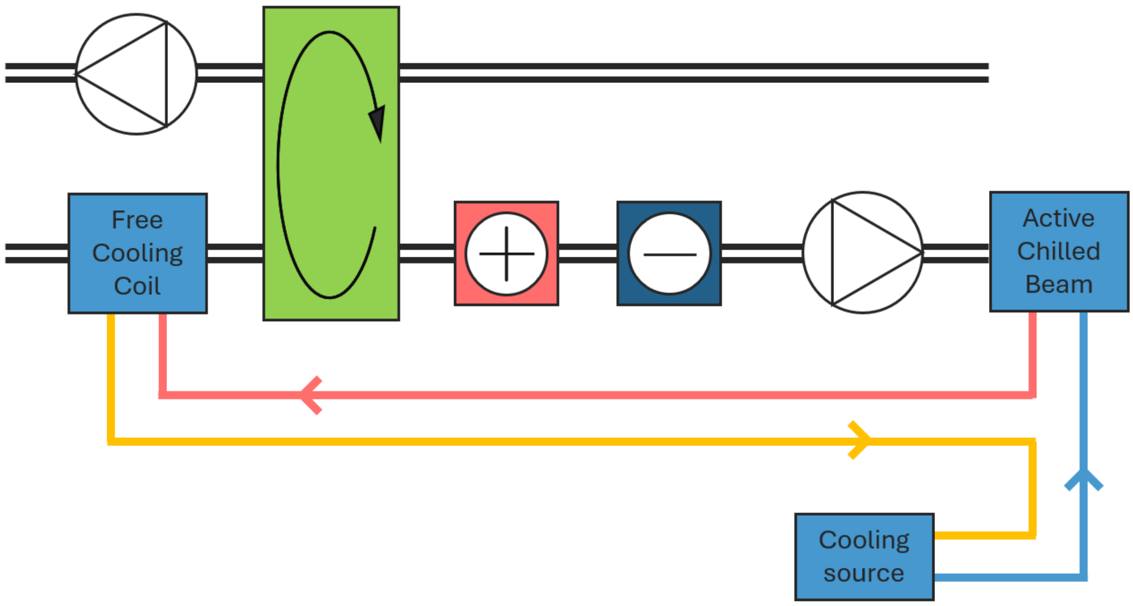


Figure 2.6: Working principle of indirect free cooling by outdoor air.

The outdoor air temperature must be lower than the return water temperature to gain a cooling effect. Consequently, the hydronic loop is monitored by a control valve, regulated by the ambient air temperature. The magnitude of effect that can be exchanged is dependent on the flow, temperature differences, and the air's & liquid's thermal capacity.

A side effect of the system is that the air-to-water HX cools down the liquid while also heating the incoming air. This is positive since free cooling only runs when the outdoor temperatures are low, and the air would have been heated anyway. Due to the increased inlet air temperatures, the heat recovery unit's efficiency is decreased. However, it is not considered negative since the higher air temperature is reached through previously recovered heat. One negative side effect is that the free cooling coil will contribute to an increased system pressure drop compared to a conventional solution, resulting in a larger fan power.

2.3 Primary Energy Number

While measuring a building's energy performance in Sweden, it is mandatory to evaluate and achieve certain BBR requirements for maximum primary energy, by calculating its primary energy number (EP_{pet}). In BBR, the primary energy number is defined as:

> "The value that describes the building's energy performance expressed as a primary energy number. The primary energy number is comprised of the building's energy use, where energy for space heating has been corrected with a geographical adjustment factor F_{geo}, multiplied by a primary energy factor for energy carriers and distributed over A_{temp}

(kWh/m^2 per year)."
The primary energy number is determined by multiplying the building's energy consumption for heating, cooling, hot tap water, and overall energy usage by a primary energy factor. This factor varies depending on the primary energy used. The equation calculating the primary energy number is depicted in Equation 2.1.

$$EP_{\text{pet}} = \frac{\sum_{i=1}^{3} \left(\frac{E_{h,i}}{F_{\text{geo}}} + E_{c,i} + E_{\text{tw},i} + E_{f,i} \right) \cdot PE_i}{A_{\text{temp}}} \tag{2.1}$$

where:

i = Index for energy carrier
$E_{h,i}$ = Energy for space heating
F_{geo} = Geographical adjustment factor
$E_{c,i}$ = Energy for air conditioning
$E_{\text{tw},i}$ = Energy for hot tap water
$E_{f,i}$ = Facility electricity
PE_i = Primary energy factor per energy carrier
A_{temp} = Heated floor area

Each primary energy factor is determined in Boverkets byggregler (BBR) 29 (Boverket, 2021) and those which are relevant for this study are presented in Table 2.1.

Table 2.1: Primary energy factors.

Energy carrier	PE_{i}
Electricity	1.8
District cooling	0.6
District heating	0.7

2.4 Finance

"Finance is the application of economic principles to decision-making that involves the allocation of money under conditions of uncertainty" (Fabozzi & Drake, 2009). Corporations leverage finance and its various evaluation methods to predict the profitability of different investments. These investments can take various forms, including both tangible and intangible assets.

Financial evaluation is a range of methods and techniques that aim at assessing the potential profitability and risk of various possible investments. One commonly used approach is financial modeling, wherein a mathematical representation is created of the financial performance of an investment over time. These models typically incorporate factors such as cash flows, discount rates, and growth projections to estimate the investment's net present value (NPV), internal rate of return (IRR), present value of annuity (PVA), and other relevant metrics.

Net present value can be defined as: "NPV is the value of all future cash flows (positive and negative) over the entire life of an investment discounted to the present"(CFI, 2024). As NPV is expressed in terms of cash value today, it provides a simplified method of decision-making. If the NPV is positive it presents a profitable investment (Berk, 2013). IRR is achieved when NPV is equal to zero.

$$NPV = \sum_{t=0}^{T} \frac{CF_t}{(1+r)^t} \tag{2.2}$$

where:

CF_t = Net cash flow (varying by year)
r = Discount rate
t = Index for time step
T = Calculation time period

The present value of annuity represents the current value of a series of equal cash flows received or paid at regular intervals over a specified period (Berk, 2013). The calculation of PVA is presented in Equation 2.3.

$$PVA = CF \cdot \frac{1-(1+r)^{-T}}{r} \tag{2.3}$$

where:

CF = Net cash flow
r = Discount rate
T = Calculation time period

Sensitivity analysis is another tool often employed to evaluate the resilience of an investment, by evaluating how changes in the investment variables affect the results. An example might be that the initial cost is more or less expensive, or that the cost of electricity is higher or lower than the current market portrays.

3
Methodology

This chapter examines the characteristics of the reference building, encompassing a detailed analysis of its layout, building envelope, occupancy, internal load profiles, and HVAC systems. Additionally, this chapter shows the design and implementation of the free cooling system, detailing the verification and sizing procedures undertaken. Furthermore, it expounds upon the methodology employed in conducting the life cycle cost (LCC) analysis.

3.1 Reference Building

Initially, the reference building underwent careful design and testing to ensure its functionality and applicability. It was an important step since all further calculations and iterations would rely on the reference building being properly designed. The building was designed to simulate a typical, modern office located in Gothenburg, Sweden.

3.1.1 Building Layout

The office has two different floor plans which are studied separately from each other, the main difference being the size and occupancy. Floor Plan A has an area of 960 m^2, with a floor-to-envelope ratio of 2.2, and is dimensioned for 52 employees. Floor Plan B has an area of 1870 m^2, a floor-to-envelope ratio of 3.1, and is dimensioned for 153 employees, resulting in 0.054 and 0.082 no/m^2 of occupants for each plan respectively. There is a smaller pentry on Floor Plan A and a larger lunch room on Floor Plan B. See Figure 3.1 for the two layouts. It can be concluded that most of the working space is open, while there are closed rooms for calls or meetings. This type of floor layout aligns with a typical modern office, making it an appropriate choice for this study. Detailed pictures of the floor plans, including naming for all zones can be found in Appendix A.

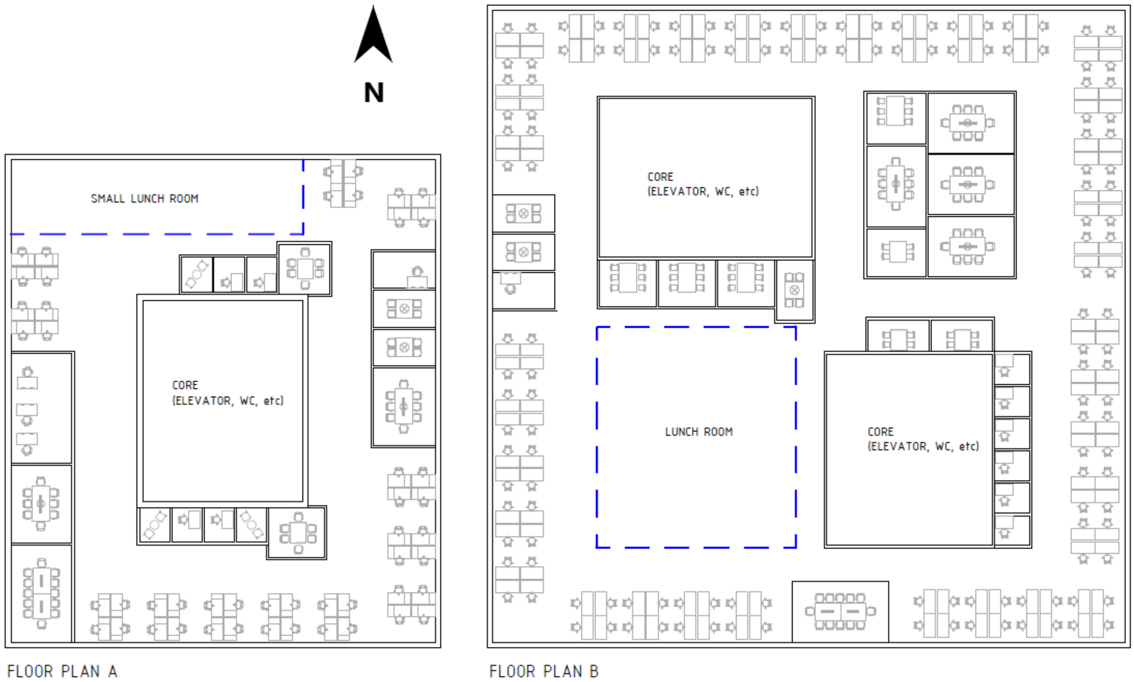


Figure 3.1: Building layout for Plan A and B.

3.1.2 Building Envelope

Both Floor Plan A and B are assumed to occupy the middle levels of the building, resulting in no heat transfer through the ground or roof. Therefore, only the exterior walls and windows are considered part of the building envelope. The properties of the chosen wall and windows are compiled in Table 3.1.

Table 3.1: Building envelope properties.

Element	Without shading		With shading	
	U-value W/m^2K	g-value -	U-value W/m^2K	g-value -
Glazing (North)	0.859	0.511	0.66	0.261
Glazing (South, East ,West)	0.803	0.317	0.624	0.163
	U-value W/m^2K	Thickness m		
External wall	0.15	0.4		

The windows used throughout the studied building differ between the cardinal directions, allowing a higher g-value to the north due to the marginal solar gains. However, both window types achieve a U-value of 0.95 W/m^2K when encompassing the glazing and frame. The glazing was designed in IDA ICE's detailed window function, with the selection process guided by a Pilkington glazing catalog for precision and reference to typical construction detail (Pilkinton, 2021). The windows are retrofitted with interior roller shading that considerably reduces the g-value and U-value. The shading is controlled by the sun, via a proportional-integral (PI) control set to activate at 12,000 lux. Both plans have an approximate window/envelope ratio of 46%.

The external walls are constructed to have low thermal storage capacity to prevent their thermal mass from impacting the cooling loads of the building, as it can reduce peak loads and flatten out the building's temperature fluctuations. Due to the focus of the study being the capabilities of the free cooling system such reductions and flattening are unwanted. As such the concrete is on the external side and the insulation is on the interior side, preventing the concrete's thermal mass from storing energy from the indoor climate and during night cooling. The designed wall has a U-value of 0.15 W/m^{2}K. Note that the building's floor slabs and cores are made from concrete. This is usually how a typical office building is constructed, for acoustic and structural reasons. Since concrete is a heavy material, it will contribute to a larger thermal inertia.

According to BBR (Boverket, 2021), the building envelope in commercial buildings should be designed to fulfill an average U-value of 0.50. Due to the limitation of only simulating the middle floors of the building, U-values for the roof and ground slab are unknown, and an average can not be calculated. Therefore, the U-values for the windows and exterior walls are ensured to at least fulfill the requirements for renovation of 0.18 and 1.2. It was assumed that a properly designed roof and slab in combination with the chosen wall and windows would reach the average U-value of 0.50.

The building's infiltration rate was set as 0.3 l/s/m^{2}ext.surf at 50 Pa, where 0.2 l/s/m^{2}ext.surf is considered ambitious (Byggal, 2017). During normal pressure and usage of the building, the infiltration rate translates to 0.015 l/s/m^{2}ext.surf. Thermal bridges were accounted for by increasing the overall thermal bridges to cover an approximate 23% of the U-mean-value for each individual floor plan.

3.1.3 Internal Gains and Scheduling

Internal gains were individually calculated for each zone. The following assumptions were made:

- Each occupant (126W) is equipped with a computer (60 W), the computer follows each occupant around into meeting rooms and calling rooms.

- All meeting rooms have a mounted projector (100 W). This projector is assumed to be in use when the room is occupied.

- In both plan's lunch rooms, there are coffee machines (1,400 W), microwaves (900 W), dishwashers (1,800 W), and fridges (17.1 W). A scheduled model for the equipment was made to simulate the following equipment usage:

- The fridge is assumed to be on constantly throughout the year.
- The coffee machine is assumed to be used during the scheduled breaks between 09.30-10.00 and 14.30-15.00.
- Microwaves are only used during lunch 12.00-13.00.
- Dishwashers are used before the first break 08.00-09.00 and after the lunch break 13.00-14.00.

For a detailed overview of each zone's internal loads, such as occupants, lighting, and equipment, see Appendix B. Due to the difference in occupancy between the two floor plans, there are different internal loads from both the occupants, lighting, and equipment.

For the occupancy schedules, two options were considered: employing a standardized office occupancy schedule per the ASHRAE standard 90.1-2004 or adhering to the ASHRAE standard 90.1-2007 and modeling occupancy schedules as deemed appropriate. According to a study by (Duarte et al., 2013), the expected occupancy levels in the ASHRAE standard 90.1-2004 are significantly higher than the actual occupancy. Notably this study was also conducted in 2013, predating the COVID-19 pandemic, which has since had a substantial impact on the occupancy level of commercial office spaces due to the widespread adoption of remote work practices (Motuzienė et al., 2022). As a result, significant differences in ASHRAE's 90.1-2004 recommended schedules can be expected. Due to this, a new occupancy level was modeled following C.Duarte's occupancy level data but with a further reduction considering the current post-covid working practices. Each occupancy level is presented in Figure 3.2.

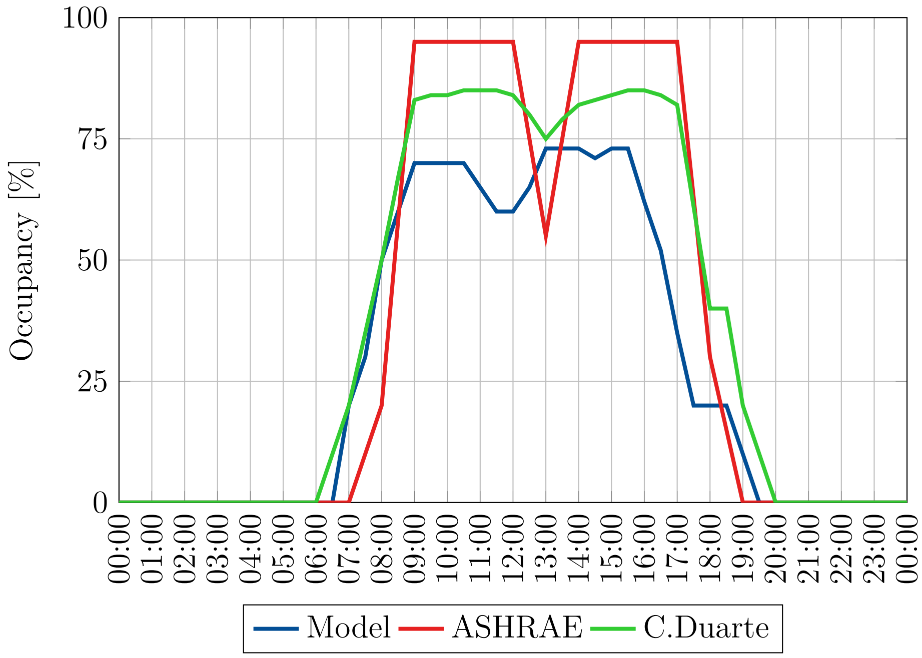


Figure 3.2: Occupancy schedules per a normal workday.

With this modeled occupancy level, a combination of schedules were created for the zones: office space, meeting, calling and lunch rooms. These schedules were created in such a way that it simulates that they are interlinked. So as one employee enters a meeting room, one occupant leaves the office space so they are not counted twice. Each zone's schedules for equipment, lighting, and occupancy are found in Appendix C. At first, a general schedule across all meeting rooms was used, resulting in an unrealistically low cooling demand. Individual schedules were therefore created for each meeting room to obtain viable results. All schedules are reduced by 50% to imitate summer vacations, during 1 July-17 August.

The difference in occupancy rate between Plan A and B is visualized in Figure 3.3, displayed in occupancy per square meter.

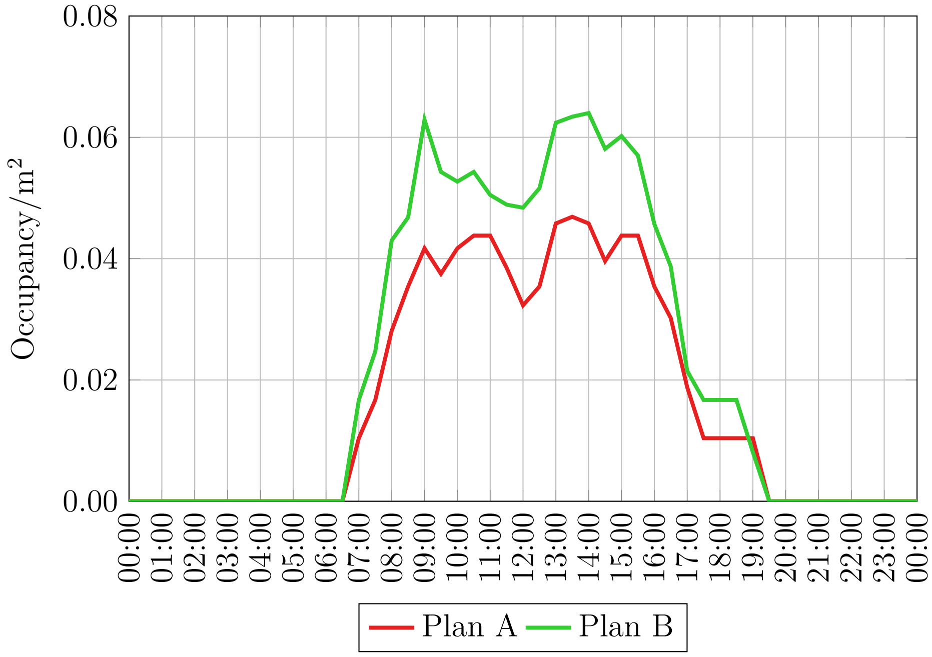


Figure 3.3: Occupancy per square meter during a typical workday.

3.1.4 HVAC

To achieve good indoor environmental quality (IEQ), appropriate heating, ventilation, and air conditioning are needed in each zone. Figure 3.4 displays the reference building's HVAC systems and their connections.

The indoor temperatures were assessed through an IDA ICE simulation, to determine whether the provided cooling power was sufficient to maintain a good indoor temperature and air quality. Each zone was inspected and reviewed for unacceptable temperatures during working days, with an interval between 21-23°C. During weekends and holidays exceeding the threshold was considered acceptable. The supply air temperature is set to a constant 18°C, with an additional degree added by the fan, resulting in a supplied air temperature of 19°C to all zones.

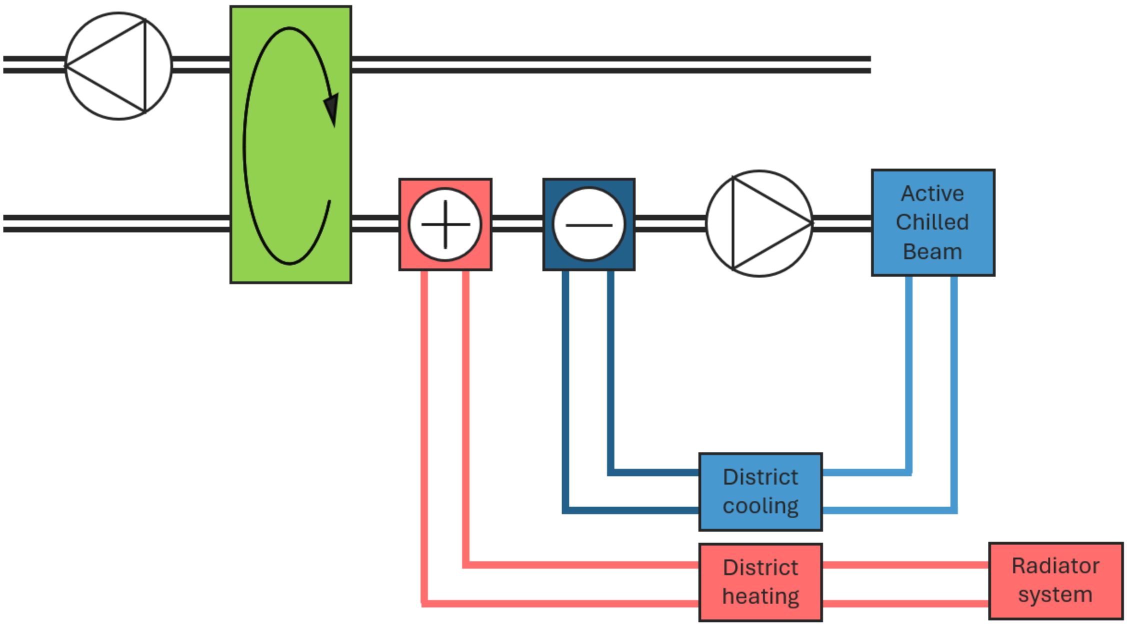


Figure 3.4: HVAC systems - reference case.

3.1.4.1 Heating

Heating is provided by water radiators connected to the building's heating systems, supplied by district heating. The simulation automatically calculates heating power for each zone through IDA ICE's "ideal heaters".

3.1.4.2 Ventilation

The reference case's key components of the AHU consist of a heat recovery system, chiller, and heater. An efficiency of 80% was used for the heat recovery system. The chiller and heater were connected to district heating and cooling as seen in Figure 3.4. A specific fan power (SFP) of 1.3 kW/m^3/s was determined. The AHU fans have an electricity-to-air efficiency of 70% and their pressure drops were designed to result in said SFP.

Each zone's air flow was designed to be above the BBR's recommended air flow rate for a commercial office building, calculated using Equation 3.1. This equation includes a base air flow per square meter, and an additional airflow per occupant designed to be present. For open office areas, corridors and calling rooms use CAV, these zones' airflow is constant at max capacity according to the designed occupancy and floor area per zone. All meeting rooms and lunch rooms use VAV as the occupancy varies a lot for these spaces, having an air flow rate starting at 0.35 $l/s/m^2$ to max capacity which includes all expected occupants. Designed occupancy per individual zone can be seen in Appendix B. Each zone's air flow is presented in Appendix D.

$$q_{air} = 0.35\, l/s \cdot A_{floor} + 7\, l/s \cdot n_{occupants} \tag{3.1}$$

The ventilation follows an operation schedule of being active between 06:00 and 20:00, turning on one hour before people arrive at the office. The night cooling is configured to activate during the months of April-September, during different times of the night depending on what day of the week. See Appendix C for specifics for both normal AHU operation and night cooling. The night cooling also only activates under certain circumstances, managed by IDA ICE's night cooling fan operation macro. The outdoor temperature must be above 15°C, and the return air must not reach below 23°C. This is to ensure the building is not excessively cooled and would require heating in the morning.

3.1.4.3 Cooling

Cooling is provided via active chilled beams (ACB). Three different beams with varying cooling powers were used to fulfill the cooling demand in each zone. The different beams have a power of 380 W, 645 W, and 1,025 W respectively, and were provided by Bengt Dahlgren. Each zone's designed cooling power is presented in Table D.1 and D.2. The beams are designed for a supply/return water temperature of 15°C / 19°C.

3.2 Free Cooling System

This chapter contains the creation, integration, verification, and sizing of the free cooling system model. From the original reference case the free cooling model system is integrated while keeping all previously stated properties and settings, only changing the AHU and plant. A simplified schematic for the free cooling system can be seen in Figure 3.5, where the free cooling coil is added to the reference case AHU before the heat recovery system. An additional pressure drop of 50 Pa was assumed due to the inclusion of the free cooling coil. This results in the free cooling model case having a slightly higher SFP of 1.37 kW/m^3/s compared to the reference model.

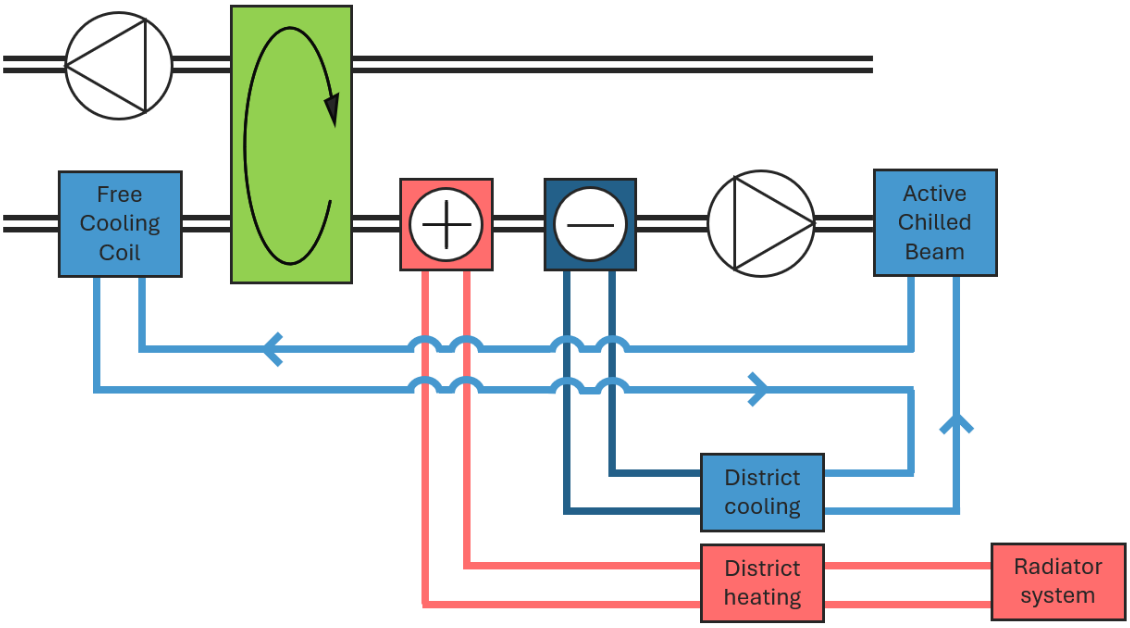


Figure 3.5: HVAC systems - free cooling case.

3.2.1 Free Cooling System Model

Figure 3.6 presents a simplified overview of the most vital parts of the free cooling system. The displayed components are described further:

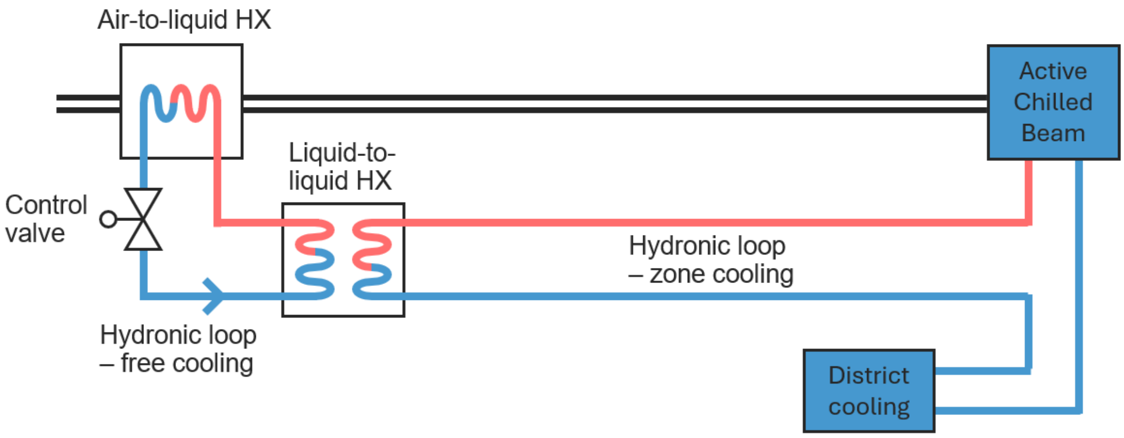


Figure 3.6: In-depth principle of the free cooling system.

- **Air-to-liquid HX:** The *air-to-liquid HX* (also referred to as the free cooling coil) effectively acts as a heating coil. Cold outdoor air enters the coil with the purpose of cooling the *hydronic loop - free cooling*. As a result, the air gains a slight temperature increase.

- **Hydronic loop - free cooling:** As the outdoor air cools the hydronic loop, it can effectively transfer away heat from the *hydronic loop - zone cooling* through the *liquid-to-liquid HX*.

- **Hydronic loop - zone cooling:** As the chilled water circulates through the active chilled beams, it absorbs heat from the surrounding room air. This process increases the temperature of the chilled water.

- **Liquid-to-liquid HX:** The *liquid-to-liquid HX* enables heat transfer between the two hydronic loops. As heat is transferred away from the *hydronic loop - zone cooling*, the demand for bought cooling is reduced.

- **Control valve:** If the outdoor ambient temperature exceeds a certain threshold, a control valve in the *hydronic loop - free cooling* is activated to halt the flow. This cessation prevents the free cooling system from operating under sub-optimal conditions. Without flow in the free cooling system, the *hydronic loop - zone cooling* does not engage in energy exchange within the *liquid-to-liquid HX*, thereby preserving system efficiency and preventing unnecessary energy consumption. A threshold value of 15°C was used throughout the simulations.

The model for the free cooling system was created in IDA ICE within the building model's dedicated plant and AHU schematics, which can be viewed, edited, and modified. In Figure 3.7 the modified AHU schematic with the included free cooling system can be seen. It's been modified from the default AHU system by adding a (1) air-to-water HX before the revolving heat recovery unit, as well as a (2) night cooling control system. The air-to-water HX is connected to the plant schematic by the two blue lines that go to the border of the schematic, which corresponds to the supply and return of the free cooling system. The night cooling system is modeled using the standard module in IDA ICE, with a slightly modified schedule.

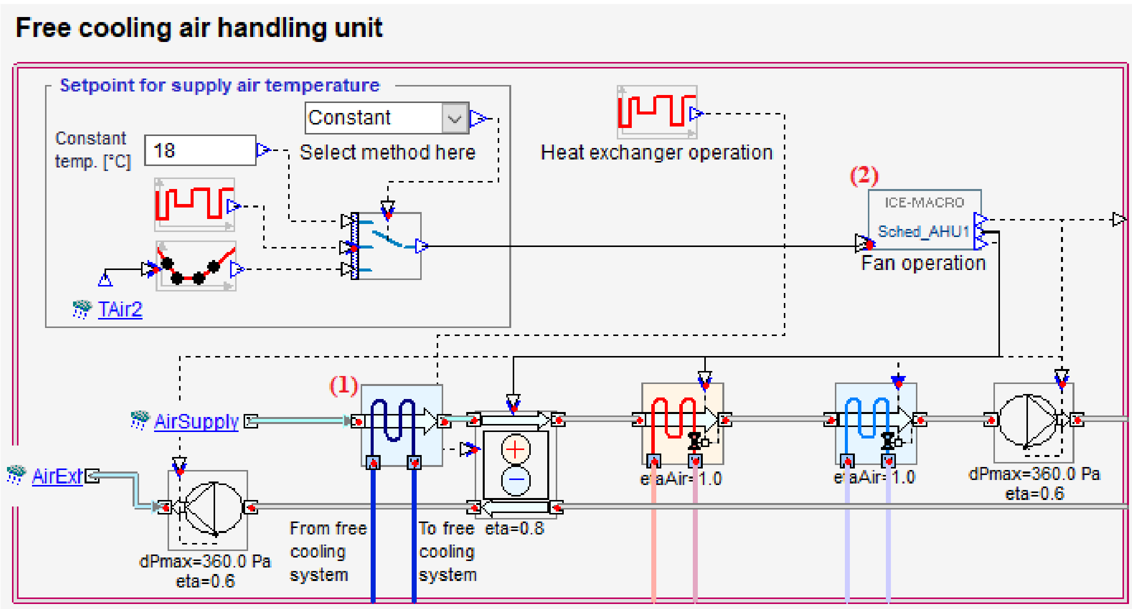


Figure 3.7: Detailed schematic of the AHU with the added free cooling system.

The modifications are more extensive in the plant than in the AHU, see Figure 3.8.

To ensure the simulation's functionality and ease of access to data, certain systems within the plant were separated. For instance, the district cooling system was divided into two distinct components: One for zone cooling and another for AHU cooling. In practical applications, these components are typically part of a single integrated system.

Situated at the center-left is the free cooling system, with (1) pipes leading from the AHU to the plant, subsequently connected to the zone cooling via a (2) brine-to-brine HX. Flow regulation within the system is managed by a (5) control valve, governed by a (4) proportional-integral (PI) controller regulated by the outdoor ambient air temperature, the setpoint for this controller is 15°C. Additionally, a (3) pump to provide mechanical flow.

Located at the top left corner is the district cooling system, connected to the zone cooling by a (6) brine-to-brine HX. The (7) chiller represents the source of district cooling, with flow regulated by a (8) control valve. This valve is regulated depending on a (9) PI control of the zone cooling system temperature at (10) a pump. The district cooling for the AHU is in the bottom-right section (15).

Finally, in the center is the zone cooling system which as stated before is connected both to the (2) free and the (6) district cooling by their respective HX. Along the path of the free and district cooling is a (10) pump and (11) control valve, this control valve is regulated by a (14) PI control of the outgoing zone cooling liquid temperature set at 15°C. The zone cooling leads in and out into the building's zones at (12) the bottom border, this is controlled by the (16) pump positioned right beforehand. This pump is scheduled to turn off during weekends and holidays. The zone cooling system is modeled to have multidirectional flows between the two (13a and 13b) brines (the blue circles). This allows the return water from the zone cooling to be directed back to the supply or into the path to be cooled by the two cooling systems (free and district).

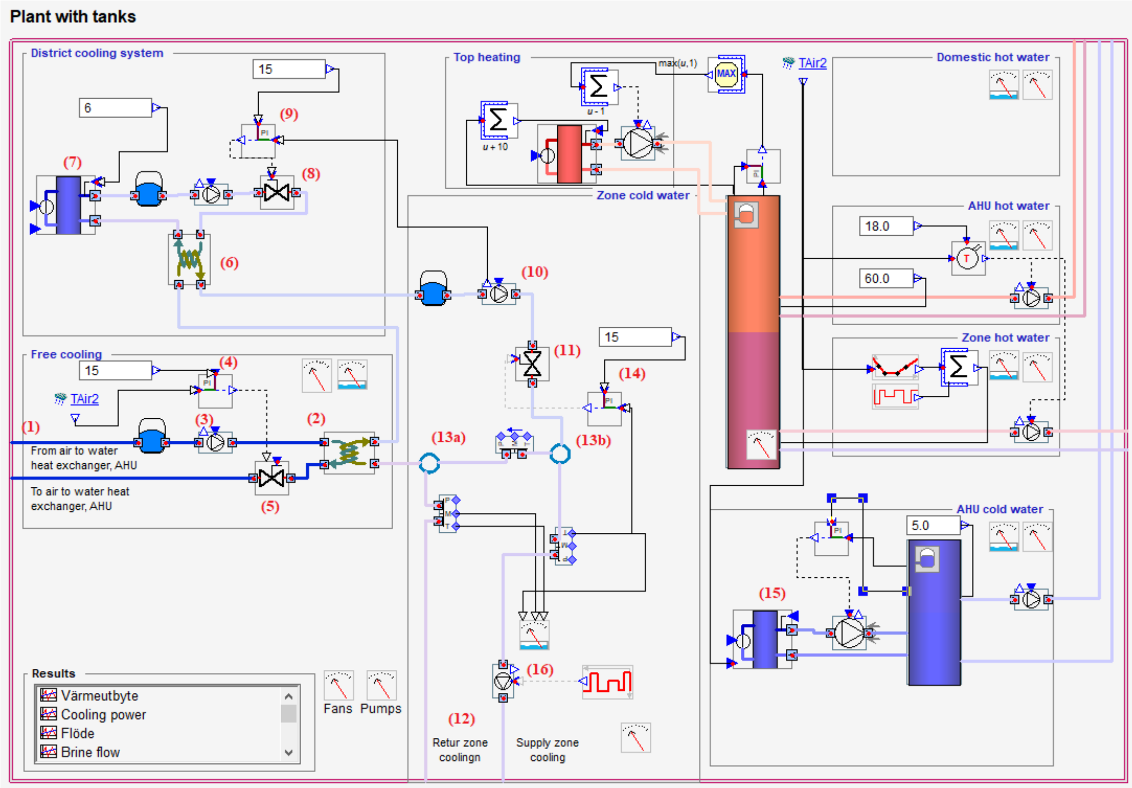


Figure 3.8: Detailed schematic of the free cooling plant.

3.2.2 Model Validation

The model went through numerous iterations and extensive testing throughout the study, resulting in the presented, final version. To prove the model's validity, results extracted from a selection of cases are presented in Figure 3.9, 3.10 and 3.11. Equation 3.2 was utilized to calculate the maximum allowed heat transfer between incoming air and the free cooling's hydronic system. $T_{air,out}$ was set to 18°C since that is air's setpoint after the free cooling coil. If the hydronic free cooling system heats the air further, it would need cooling before being supplied to the room, and the free cooling effect would be counteracted.

$$P = q_{air} \cdot \rho_{air} \cdot cp_{air} \cdot (T_{air,out} - T_{air,in}) \tag{3.2}$$

The maximum available heat transfer for each hour of the year could be calculated to ensure proper operation of the free cooling system and maximized utilization whenever possible.

Figure 3.9 displays a typical weekday in late March where the outdoor temperatures range between 2-5°C and the potential to use free cooling is constantly far above the demand. It can be seen that the entire cooling demand is supplied via free cooling, proving that the system works as intended.

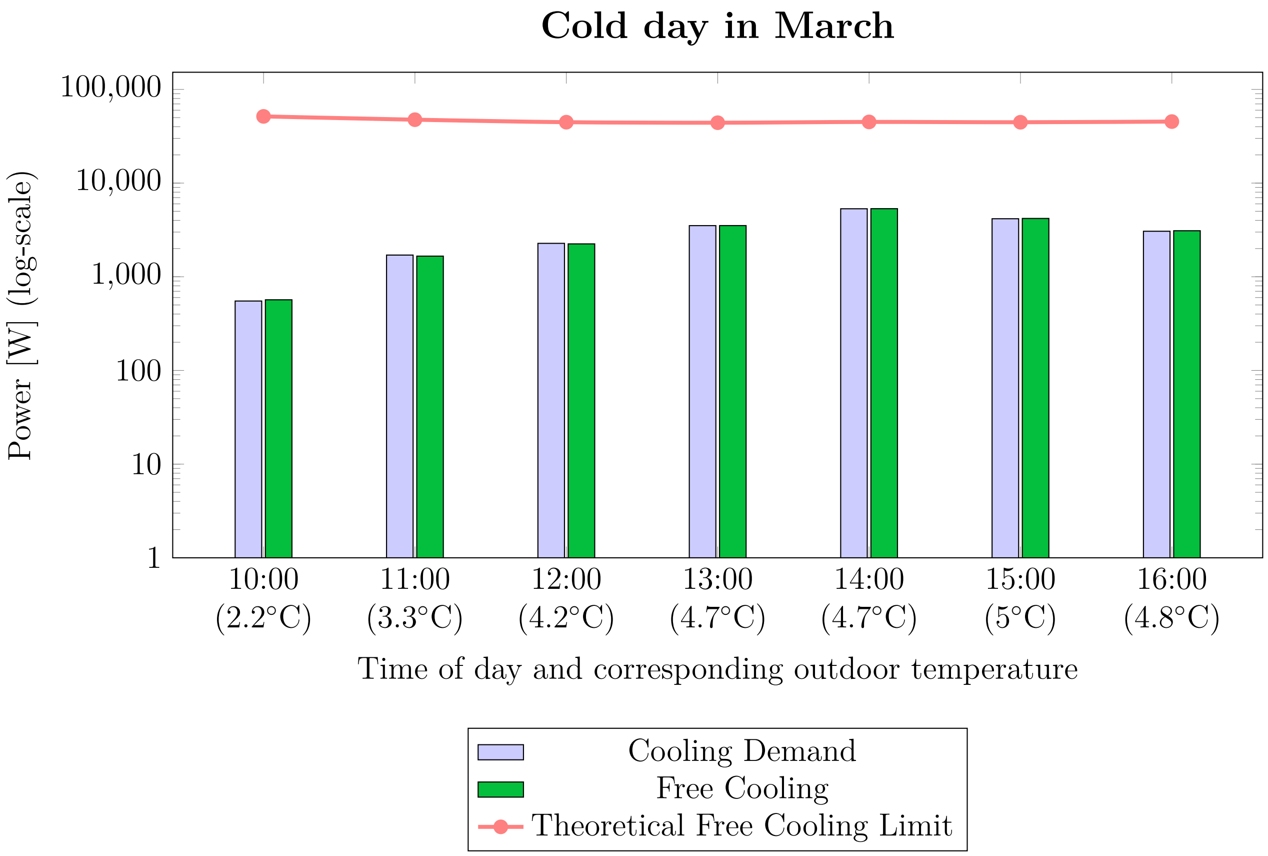


Figure 3.9: Cooling demand coverage and available cooling power during a cold day in March.

In Figure 3.10, the outdoor temperature ranges between 12-17°C, meaning the theoretical maximum heat transfer is a lot lower compared to the example shown in Figure 3.9. The example shows an important transition where the outdoor temperature surpasses the decided threshold of 15°C at 13:00. It can be seen that the bought cooling gradually increases between 12:00 and 14:00 even though there is theoretical potential for free cooling. This proves that the limiting threshold value of 15°C works as intended.

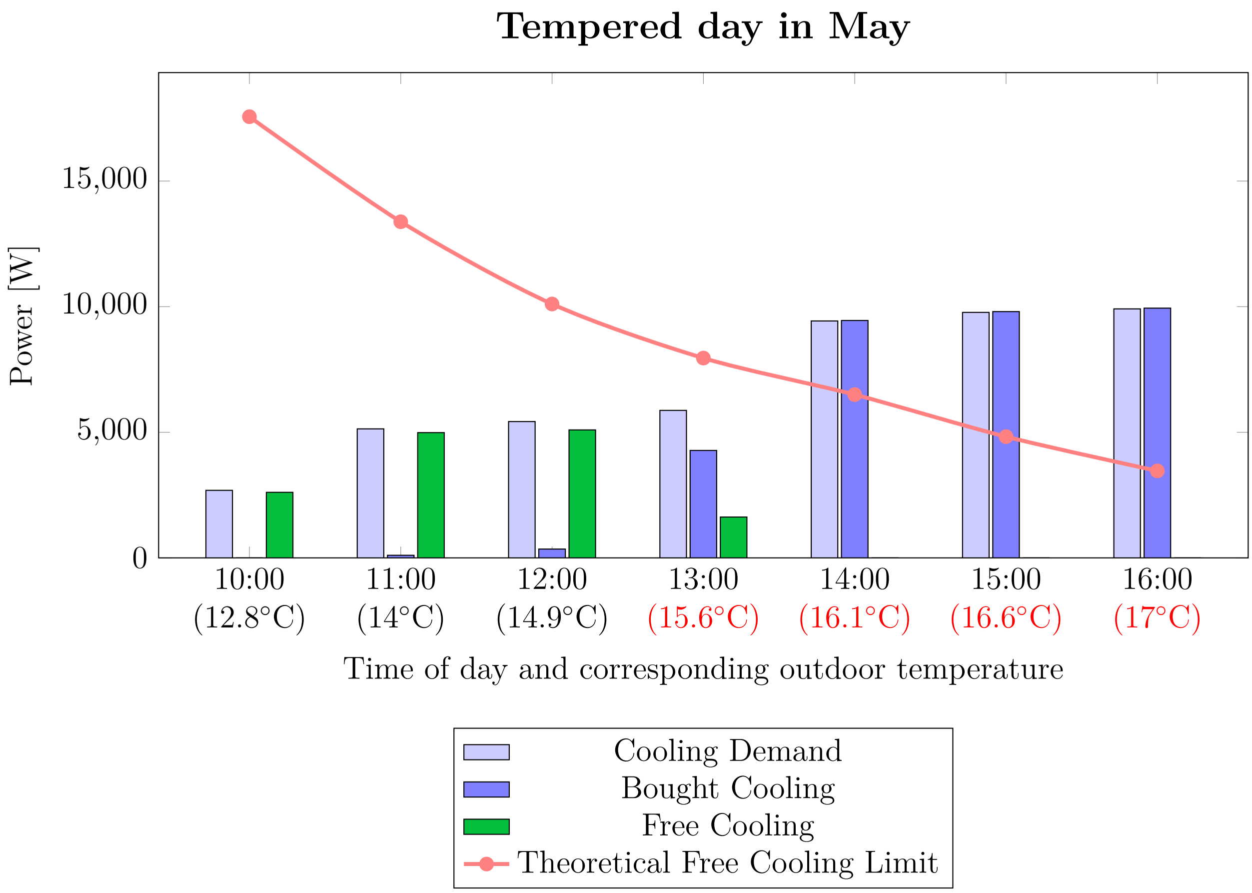


Figure 3.10: Cooling demand coverage and available cooling power during a tempered day in May.

Figure 3.11 shows an example of a warm summer weekday where the outdoor temperatures are constantly above 18°C. This addresses the potential risk of having negative free cooling since the theoretical heat transfer reaches large negative numbers. However, the diagram shows no negative free cooling, and the entire cooling demand is supplied through district cooling as intended.

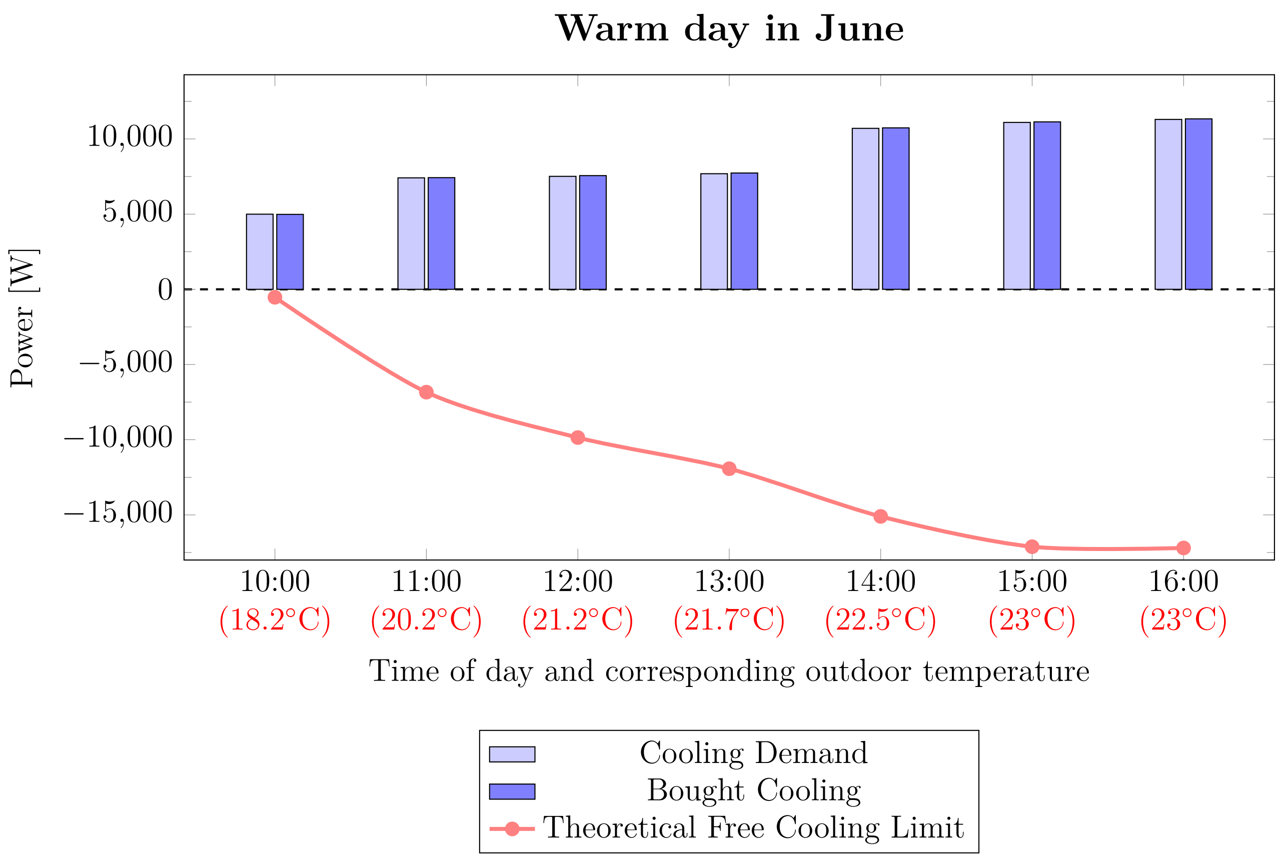


Figure 3.11: Cooling demand coverage and available cooling power during a warm day in June.

The ventilation and hydronic cooling system are supposed to be turned off during nights and weekends to not use excessive energy while the building is empty. As a result, high indoor temperatures are allowed to occur until Monday morning. This is often no issue since the temperatures naturally decrease during Sunday night, also emphasized by night cooling. However, from a modeling point of view, there is a risk of energy being stored in the hydronic system if not turned off properly. Figure 3.12 shows how the indoor temperature changes over a weekend in one of the more exposed rooms, reaching almost 30°C at times. Instead of constantly trying to cool, the system is turned off until 06:00 Monday morning, see the dotted line. This way, the temperature has naturally dropped to about 24°C due to the night cooling, and less cooling is required.

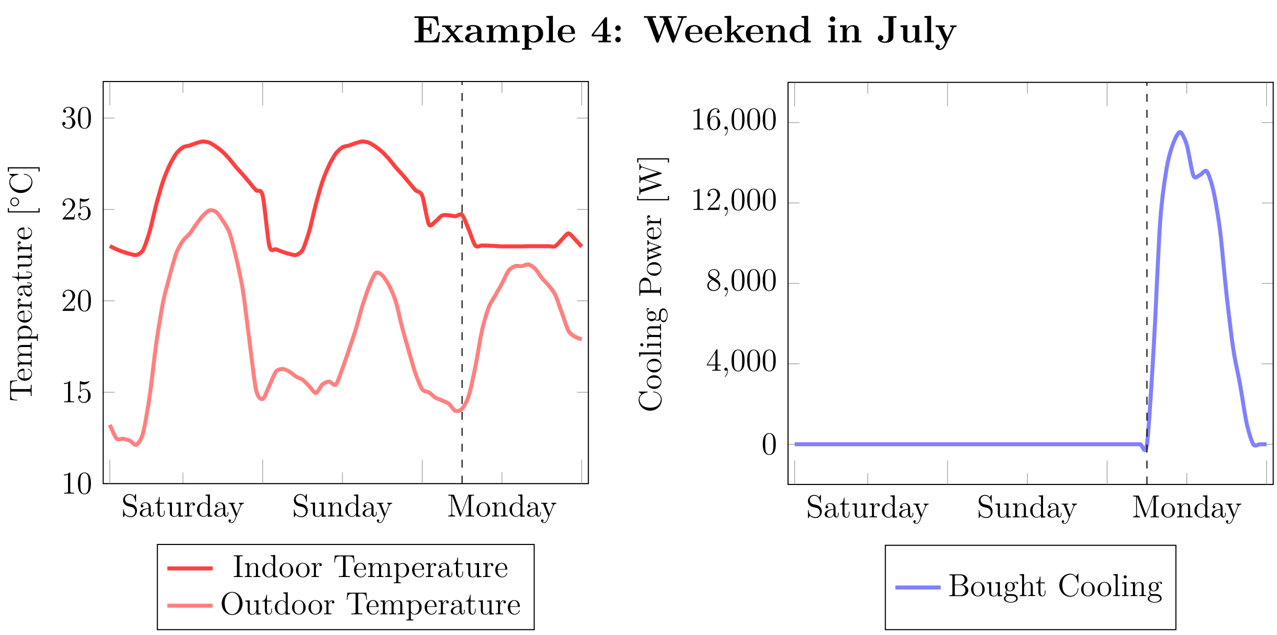


Figure 3.12: Night cooling performance during a weekend in July.

3.2.3 Sizing

Three parameters were sized according to the set conditions of the building. The sizing was performed for each floor plan separately while reviewing it as one story. Descriptions for each sized parameter are shown in the following list, see Figure 3.6 for reference.

1. **Liquid-to-liquid HX:** The variables for the liquid-to-liquid HX were adjusted based on the expected temperatures and maximum power requirements for each floor plan. The maximum power was determined by simulating each floor plan with an uncapped HX, then selecting a power level that covers approximately 98% of the free cooling effect. This approach ensures the component is not oversized. As a result, the heat exchanger for Plan A was set to 4 kW, while for Plan B it was set to 11 kW.

2. **Free-cooling systems liquid flow rate:** Flow rates were determined by performing a parametric analysis to assess how varying flow rates in the free cooling system impacted the overall energy performance of the model. This was evaluated by comparing the reference case with the model's heating, cooling, and electrical energy performance at different flow rates with and without BBR's primary energy factors. Both floor plans' results are presented in Figure 3.13 with a highlighted value for the optimal flow rate. The optimal dimensioned flow rate was 0.2 l/s and for 0.5 l/s for Plan A and B respectively.

3. **Air-to-liquid HX:** The maximum total air flow for each floor plan and a yearly average wet/dry bulb temperatures were used for the air-to-liquid HX's variables in IDA ICE, this pertains to the sizing of the air-to-liquid HX. It

also matches the power to be equal to the liquid-to-liquid HX power.

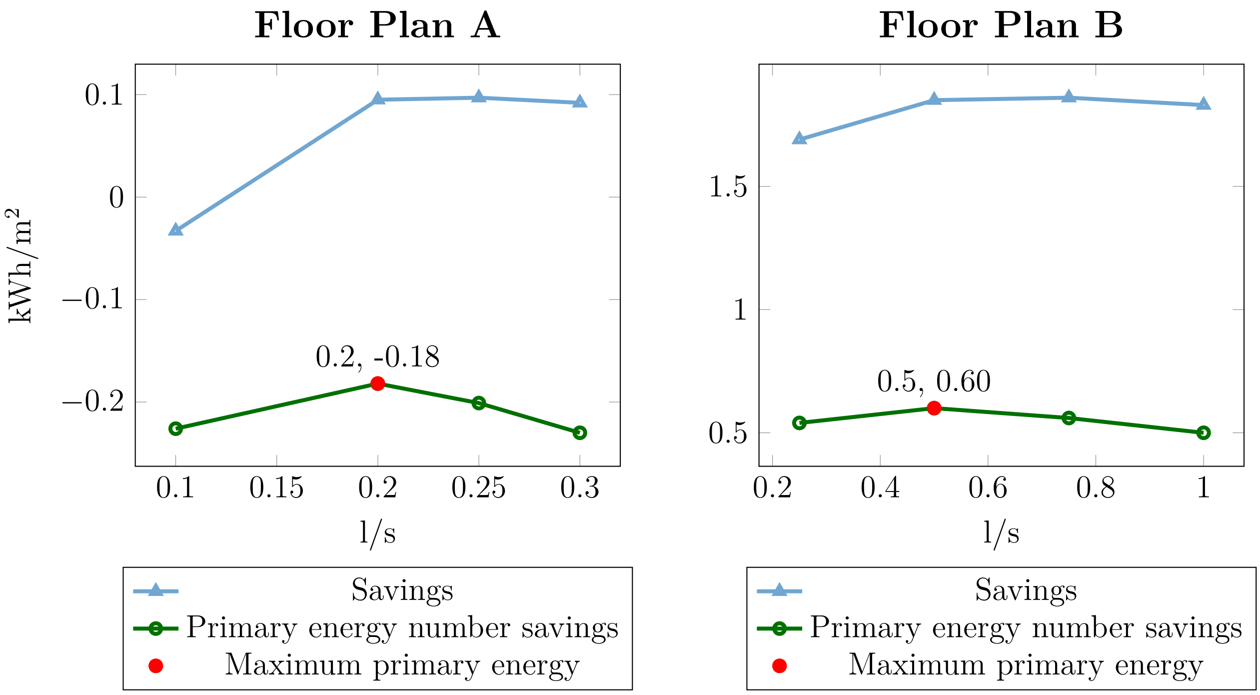


Figure 3.13: Sizing of liquid flow rate for the free cooling system's heat exchanger.

3.2.4 Future Climate

Using a modified climate file, the free cooling model's performance was tested in Gothenburg's future climate. The file was modified by a service provided by WeatherShift, which alters the original climate file according to selected greenhouse gas emission scenarios called RCPs (Representative Concentration Pathways). The file that was used was modified with an RCP option of 8.5. According to WeatherShift, these scenarios were created by the Intergovernmental Panel of Climate Change (IPCC) as a basis for the climate projections done for its fifth assessment (AR5) (IPCC, 2024) (WeatherShift, 2024).

3.3 Energy Performance Analysis

Once the model was verified to function as intended and each parameter was determined, data could be extracted for the energy performance analysis. Year-round simulations were done in IDA ICE for each case. The total energy use was extracted for bought cooling, heating, lighting, fan/pump electricity, and hot water use. A hot water use of 2 kWh/m^2 was estimated across all simulations sourced by district heating (Sveby, 2010). Although the study focuses on reducing bought cooling, the total energy use was also extracted to provide context for the energy savings and to calculate the total primary energy number for the different scenarios. Specifically for the free cooling scenarios, numbers for the additional pump- and fan energy were extracted to showcase the drawbacks of implementing free cooling.

3.4 Life-Cycle Cost Analysis

The economic evaluation was conducted employing an LCC analysis, utilizing the present value of annuity (PVA) method. The time frame selected for this calculation spans 20 years with an assumed equivalent longevity of all components, thereby omitting considerations for residual values and reinvestment costs. The investments included in the LCC are the initial investment, ongoing maintenance costs, and all expenses related to energy and power demand during operation time. A discount rate of 5.5% was used for the calculation.

3.4.1 Investment- and Maintenance Cost

The investment cost refers to the additional components and labor required for the free cooling system. Each price was provided by Bengt Dahlgren, including an estimate for labor cost. An investment of 425,000 kr is required for Floor Plan A, while the larger Floor Plan B requires an investment of 600,000 kr. For detailed data, see Table E.1 and E.2, in Appendix E.

Maintenance for each component was estimated through SS-EN15459-1:2017 (SIS, 2024). The expected maintenance cost per year was calculated as a percentage of the initial investment cost. In total, maintenance for Floor Plan A and B was calculated to 7,000 and 9,800 kr/year respectively. Each component's annual maintenance cost is presented in Appendix E, Table E.3 and E.4. An annual maintenance price increase rate of 2% is also included in the calculations.

3.4.2 Energy Cost

The energy cost calculations were done separately for each energy type. Calculation methods and costs were gathered from the local energy provider (Göteborg Energi, 2024).

District cooling costs are separated into three categories: energy, power, and flow. The energy cost was calculated by multiplying each month's simulated energy use, by the predetermined energy prices for 2024. Power cost is determined by the building's peak power demand. For this study, the maximum power demand occurs when the free cooling system is disabled. Accordingly, the peak demand is equal in both the reference- and free cooling case resulting in equal power cost. Due to them being equal, power cost was excluded from the LCC analysis. The flow cost was calculated by multiplying the yearly district cooling liquid flow by a fixed price of 0.8 kr/m^3.

Due to the free cooling system's added pumps, additional electricity costs had to be calculated. The energy use of the pumps was extracted for each month and multiplied with corresponding electricity prices from 2023, sourced from the same local energy provider as the district cooling (Göteborg Energi, 2024). All energy prices have an individual annual price increase rate, district cooling: 3%, district heating: 1.1%, electricity: 6.4%.

3.4.3 Sensitivity Analysis

The key values of the discount rate, annual energy price increase rate, and annual maintenance price increase rate provided by Bengt Dahlgren were compared with similar companies' key values (Eliasson & Virro, 2012). Which proved the values to be within the average range and thus acceptable.

A sensitivity analysis was performed after the initial calculations with a few of the aforementioned key values. During this analysis, discount rate and annual price increase of district cooling prices are altered individually in an interval to test the resilience and effects on the project from changes in the different rates. The interval minimum for discount rate is between 0% and 10%, and the annual price increase of district cooling interval is between 1% and 25%.

4
Results

This chapter aims to display the free cooling system's performance across different parameters. Initially, the system's yearly coverage is shown to act as a baseline for further results. The system's overall performance is displayed through graphs on energy performance and life-cycle cost (LCC). Energy performance was also measured and displayed for the future climate.

The results are presented for each floor plan as if it were a five-story building. This approach was chosen to evaluate the system against a typical office building size and to ensure an appropriate maximum power level for district cooling costs. Consequently, all energy results and the sizing of all components were multiplied by five.

4.1 Yearly Free Cooling Operation

Figure 4.1 shows the designed free cooling system's monthly average cooling coverage over a typical year, for Floor Plan B. It can be seen that during the period between October-March the free cooling system covers the whole building's cooling demand. During spring, the free cooling system starts showing signs of not having the capacity to cover the entire cooling demand, likely due to the increasing temperature in the outdoor climate, limiting the system's cooling capacity. The missing power needed for the required cooling demand is supplemented with additional bought cooling. During summer, most of the cooling is bought as the outdoor temperature is above 15°C a majority of the time.

Figure 4.2 presents a duration diagram. According to set parameters, the free cooling system's maximum cooling hours equals 3,640 hours per year. It is based on the ventilation system being able to turn on between 6:00 - 20:00, five days per week, year-round. For Floor Plan A, the graphs follow a similar pattern but at a consequently lower power. Graphs for Plan A are shown in Appendix F.

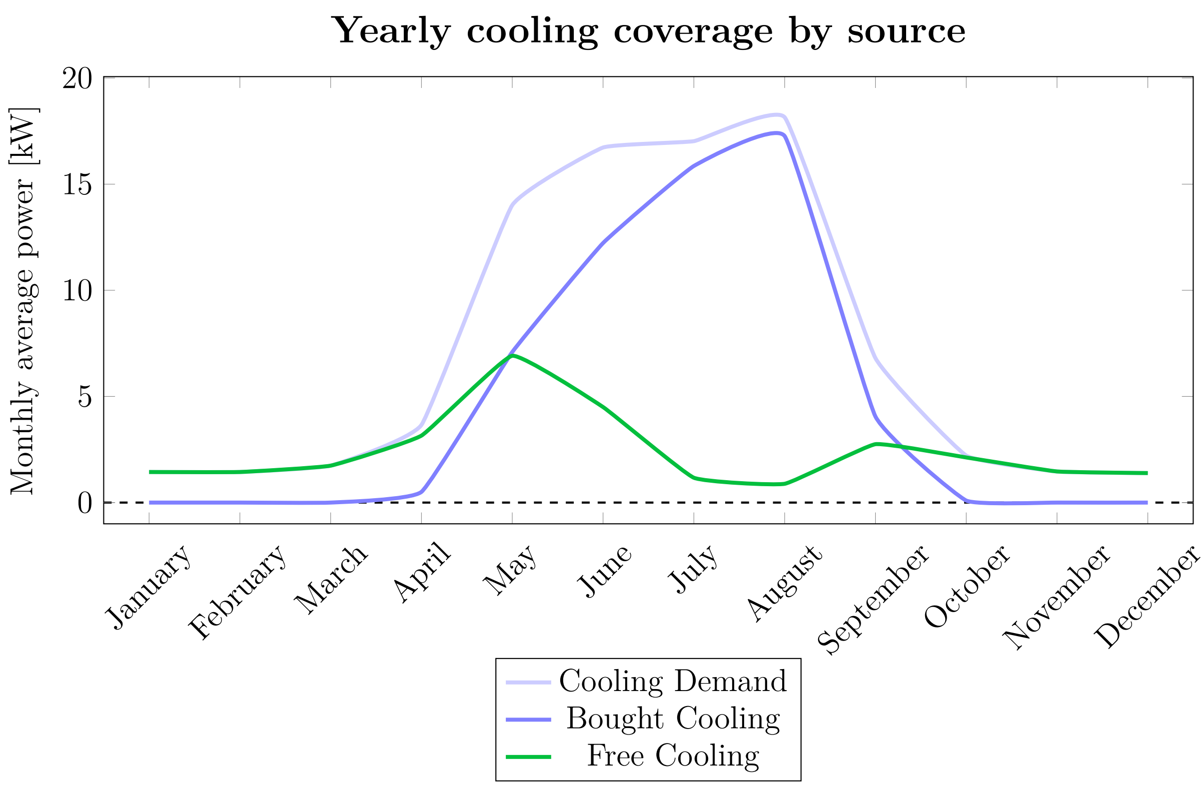


Figure 4.1: Monthly average cooling supply and demand (Floor Plan B).

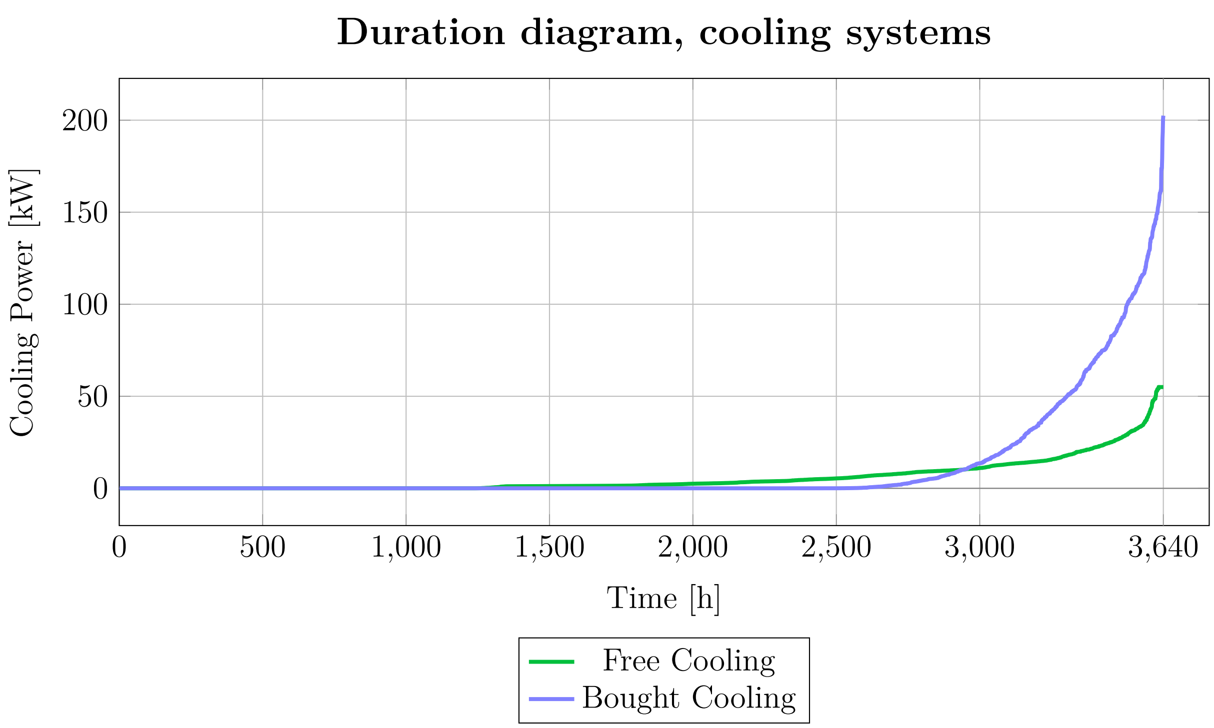


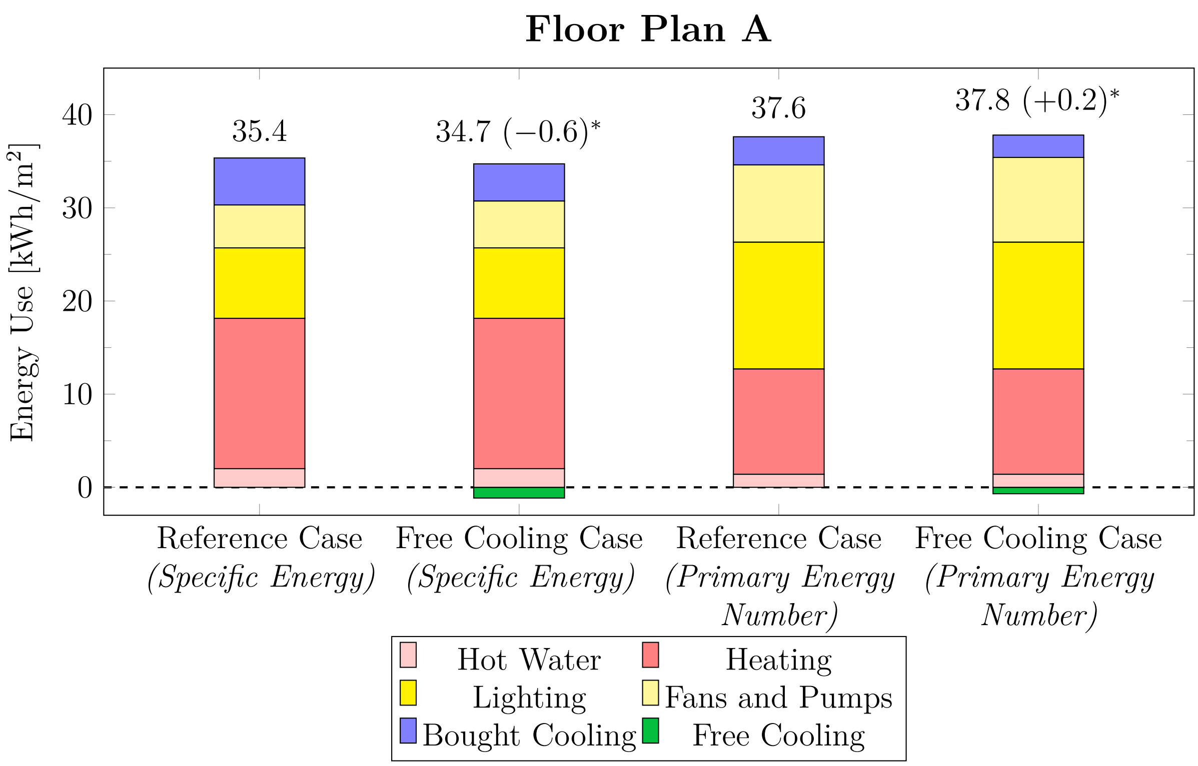
Figure 4.2: Duration diagram for free cooling- and district cooling operation (Floor Plan B).

4.2 Energy Performance

Figure 4.3 and 4.5 shows each plan's respective specific- and primary energy number results for the reference and free cooling case, encompassing the hot water, facility electricity, heating and cooling for both AHU and zone units. Facility electricity is the electricity needed for lighting, fans and pumps. Due to some of the categories being constant over both cases Figure 4.4 and 4.6 showcase only the affected variables. The category *Added Electricity* describes the higher electricity need, due to additional components in the free cooling system.

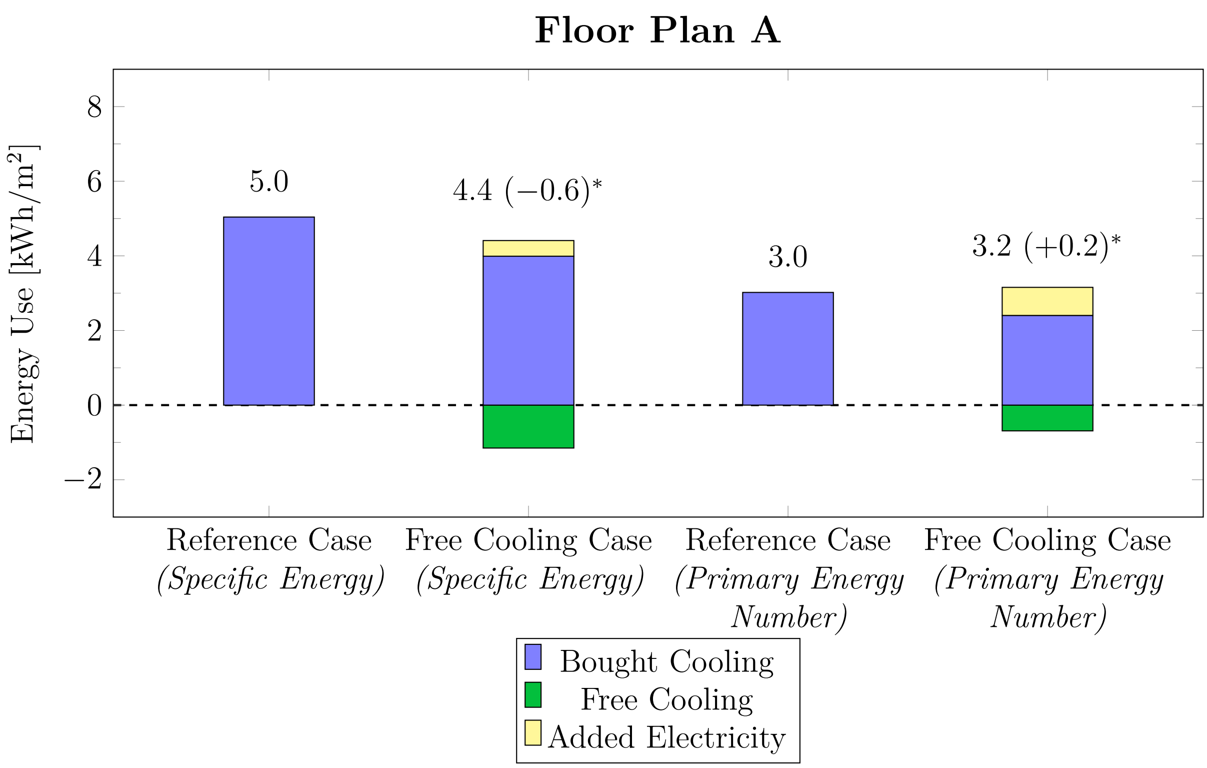
It can be seen in Figure 4.3 and 4.4 that Floor Plan A receives a decrease in specific energy of 0.6 kWh/m^2 and an increase in primary energy number of 0.2 kWh/m^2. This translates to -2% and +0.3% when accounting for the building's total energy use. As for Floor Plan B, the results are more noticeable. There is a decrease in specific energy of 1.8 kWh/m^2 and a reduction of primary energy number of 0.6 kWh/m^2. This translates to -6% and -2% when accounting for the building's total energy use.

The difference in heating demand for the two plans is due to the difference in occupancy and internal loads, with Plan A having a lower occupancy rate and thus higher internal loads per m^2.



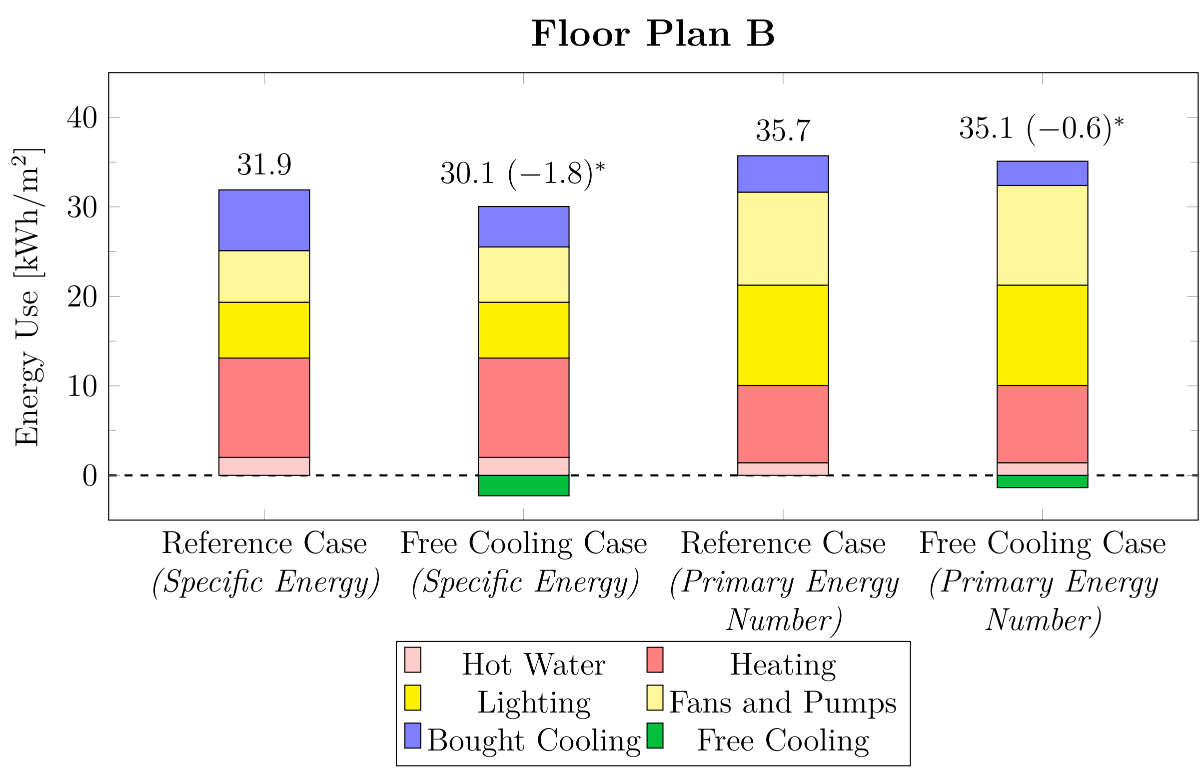
Increase or decrease compared to the reference case.

Figure 4.3: Energy performance comparison (Floor Plan A).



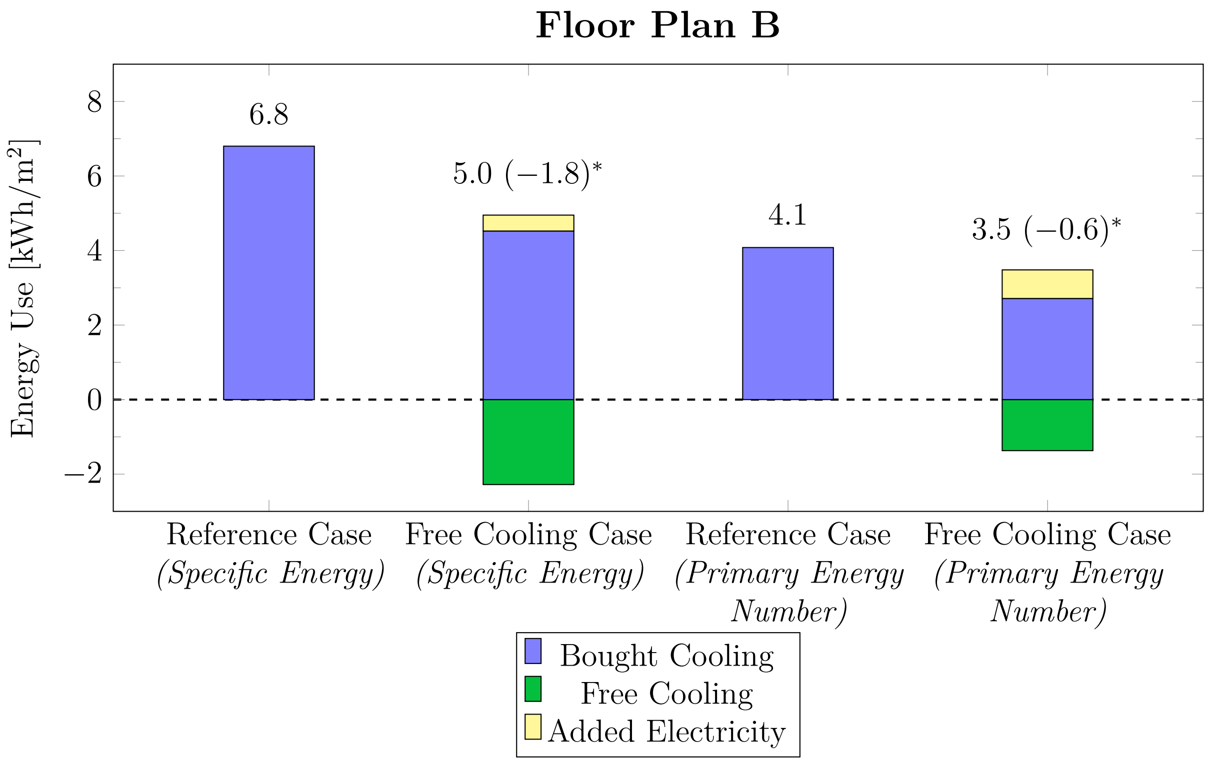
**Increase or decrease compared to the reference case.*

Figure 4.4: Cooling energy performance comparison (Floor Plan A).



**Increase or decrease compared to the reference case.*

Figure 4.5: Energy performance comparison (Floor Plan B).



**Increase or decrease compared to the reference case.*

Figure 4.6: Cooling energy performance comparison (Floor Plan B).

4.3 Life-Cycle Costs

Figure 4.7 shows the total LCC cost for both plans' reference case and free cooling system, calculated using the PVA-method. For both plans, there's a substantial difference in cost due to the additional investment needed for the free cooling system. The reduced cost of bought energy is approximately 31,300 and 110,000 SEK for Plan A and B respectively. However, the additional maintenance and electricity costs from the free cooling system add up to more than the cost saved on bought cooling for both plans.

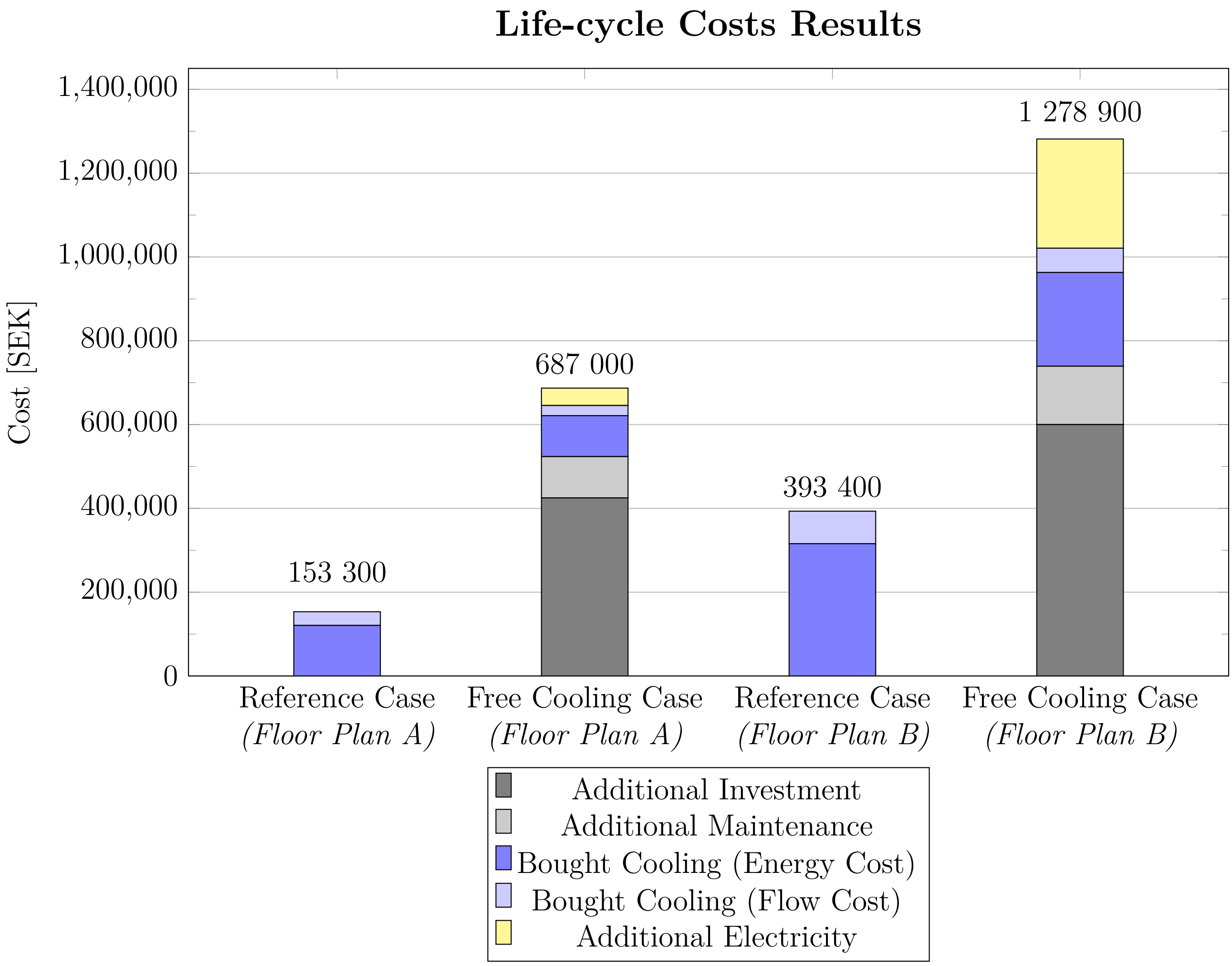


Figure 4.7: Life-cycle costs results.

4.3.1 Sensitivity Analysis

The analysis of LCC is inherently affected by the choice of discount rate, introducing a level of uncertainty. In Figure 4.7, the depicted LCCs were calculated using a discount rate of 5.5%. Given the critical role of this rate in determining LCC accuracy, conducting a sensitivity analysis becomes imperative to gauge its impact on final LCC outcomes. Figure 4.8 illustrates the results of this sensitivity analysis across the two cases for each plan. The results reveal that as the discount rate rises, there's a corresponding decrease in LCC, leading to a reduction in the gap between the reference case and the free cooling case for both floor plans. However, despite this trend, the difference between the two remains significantly large.

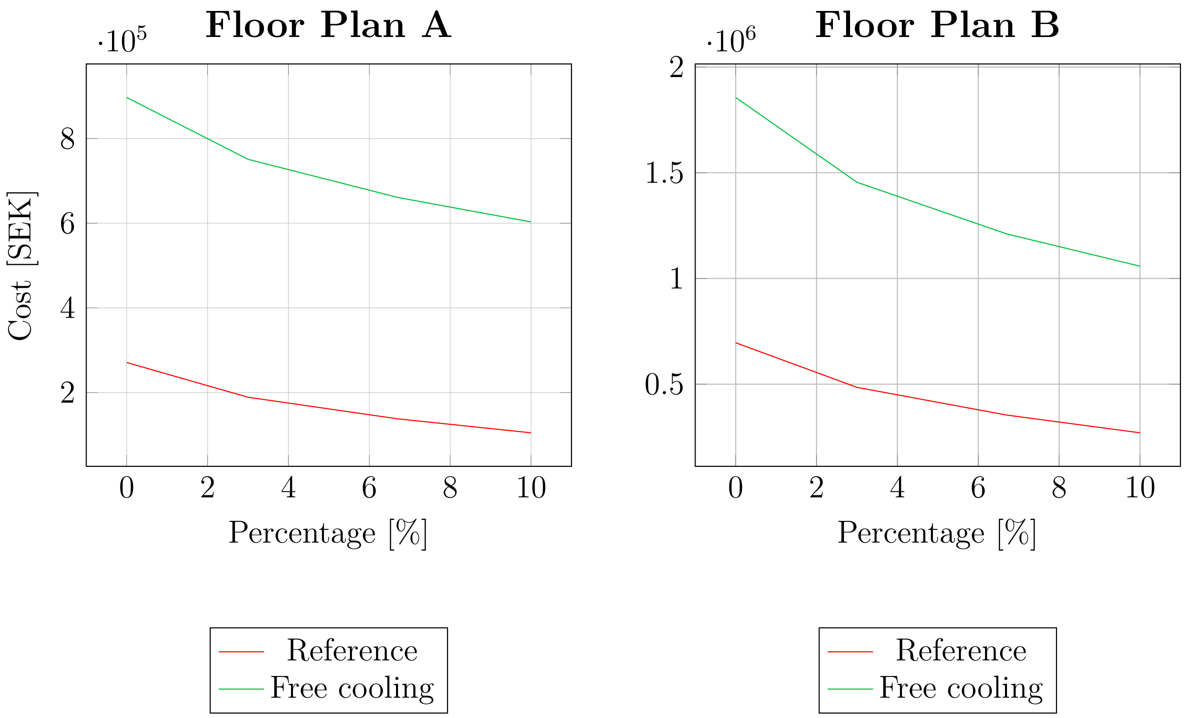


Figure 4.8: Sensitivity analysis - Discount rate.

The same uncertainty exists regarding the annual price increase rate for district cooling, As reducing the usage of bought cooling is the primary function of the free cooling system. In 4.7, the depicted LCCs were calculated using a district cooling annual price increase rate of 3%. For this sensitivity analysis, the rate's interval is very large, this is done to see at what rate is required for the free cooling system to turn profitable. Figure 4.9 illustrates the results of this sensitivity analysis across the two cases for each floor plan. The results show a trend of increasing rate results in a decrease in LCC. The disparity between the two cases reduces slightly in the 1-10% rate range, but as the rates increase past 10% a gradual reduction in disparity can be seen for both plans. At approx 19% Plan B turns profitable and at 25% for Plan A. Note that these levels of district cooling annual price increase rate are unrealistic, as such high rates would mean very high price increases.

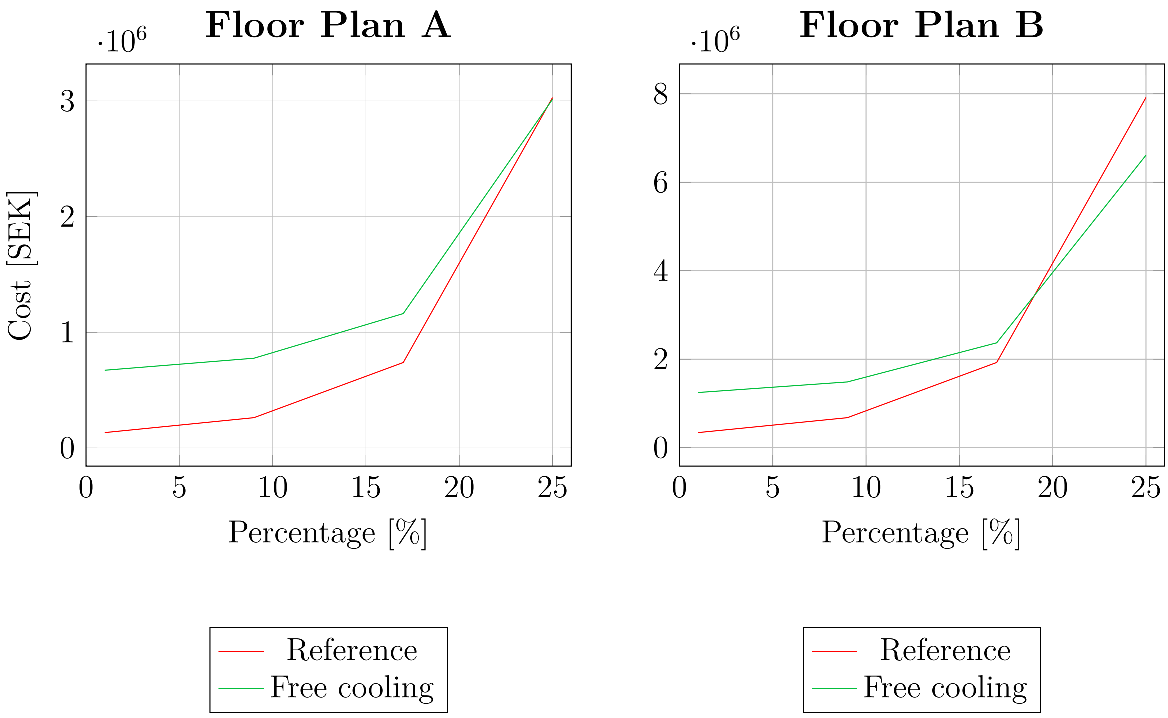


Figure 4.9: Sensitivity analysis - Annual price increase rate.

4.4 Future Climate

Results and comparison in energy performance between the free cooling model with present and future weather files is presented in Figure 4.10 for the two studied floor plans. The results show the building's energy demands that received varying results between the present and future weather scenarios.

The results show increased cooling demand across both AHU- and zone cooling for both plans in the future. An increase of 3.4 kWh/m^2 for both Plan A and B, an 85% and 75% increase in bought cooling demand for each plan respectively. The free cooling slightly decreases due to the increased outdoor temperature, affecting the system's yearly operation hours and the available cooling power. A decrease of 0.17 and 0.32 kWh/m^2 for each plan. That is approximately a 15% reduction of free cooling. Total bought heat demand for AHU- and zone heating has decreased for both plans due to the overall warmer climate of the future. The decrease in heating is approximately 5% for both plans. Minuscule increase in pump and fan electricity for both plans.

What can be interpreted from the results is that the expected future climate need for cooling will increase, while the effectiveness of free cooling from outdoor air will slightly diminish.

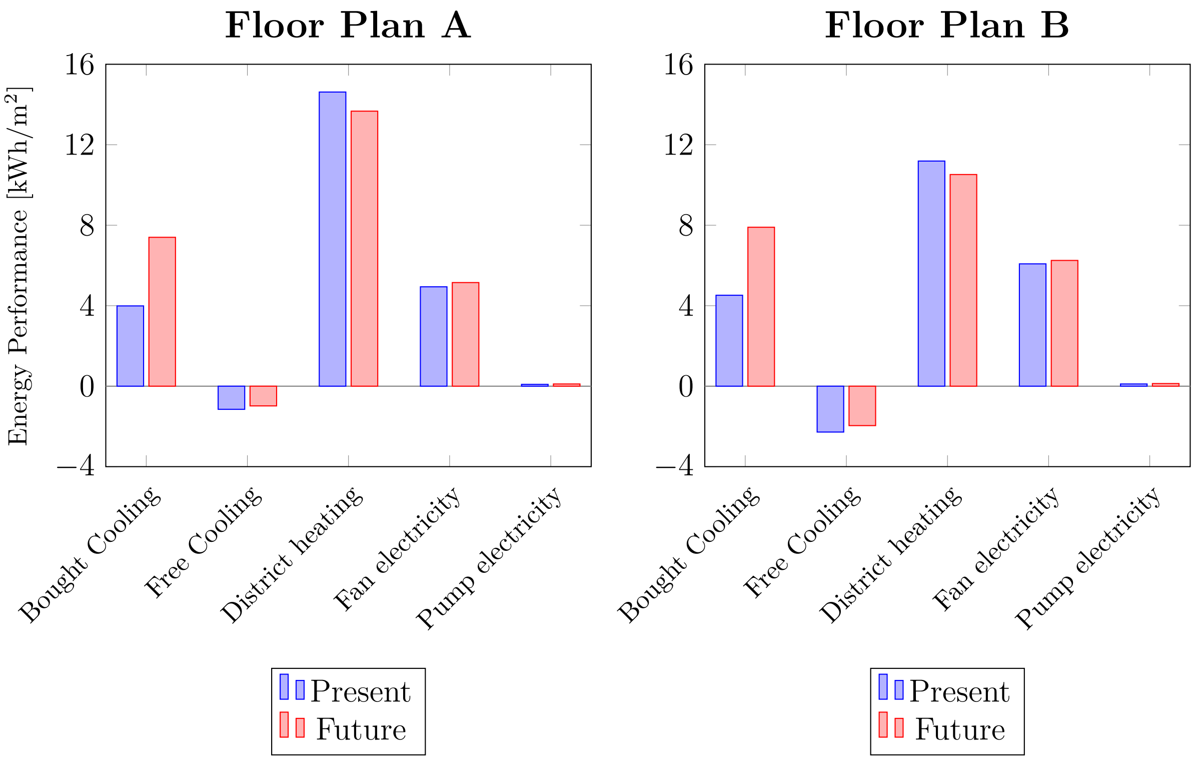


Figure 4.10: Differences in energy performance for present and future climate.

5

Discussion

One of the largest concerns regarding free cooling by outdoor air is the counteraction between free cooling potential and cooling demand. During winter, when the free cooling potential is high, the cooling demand is naturally low due to the cold outdoor conditions. The exact opposite occurs during summer, which the results showcase. It is clear from the results that the system contributes to a tiny benefit in energy performance at a large life-cycle cost over 20 years, which is also reinforced by the sensitivity analysis. The results for Plan B achieve a reduction in specific energy and a minor reduction in primary energy. As for Plan A, the results indicate an even smaller reduction in specific energy while even getting an increase in primary energy with the implementation of the free cooling system. Therefore, it can be concluded that the system is not ideal for any of the studied scenarios, with the decided properties and location. What is worth noting is that the designed building is really energy efficient, with a primary energy number just above 35 kWh/m^2 for both plans. It is therefore hard to get any substantial improvements, compared to a less efficient building. There could also be varying results if the floor plan were more enclosed since more heat could accumulate in smaller rooms. Regarding future performance, the results indicate that although cooling demand will increase, the cooling provided by the free cooling system will decrease. This is because the future is predicted to have higher average outdoor temperatures, meaning fewer hours per year will be below the threshold of 15°C.

Through the comparison of floor plans A and B, it has become evident that the free cooling system's efficiency is greater when there is a higher cooling demand. Also, the system is less effective in future conditions when the outdoor temperature is generally higher, due to the temperature being above the free cooling threshold for more hours per year. The continuing discussion includes simple testing of mentioned factors, to possibly determine more favorable conditions for the free cooling system.

As the occupancy and internal gains is one of the major factors affecting the varying results between the two planes, a brief test for Plan A was performed, where the occupancy was increased. Instead of using the custom-modeled schedule per Plan A, the ASHRAE schedule which can be seen in Figure 3.2 was assigned to all rooms except the lunchroom, core, and corridors. From July 1st to August 17th, the occupancy rate is halved, simulating the reduced activity typical of summer vacation. This provides a higher and constant occupancy for the building, which results in much higher cooling demand throughout the year. The overall energy demand of Plan A, with the same system sizing but higher occupancy, has a higher demand, due to increased cooling, fan and pump electricity, and lighting electricity. However, the free cooling system partially covers the higher cooling demand, which covers approximately 7.3 kWh/m^2 of the cooling demand. See Appendix G.1 for a comparison between the original scenario and high occupancy. This indicates that the free cooling system can provide and cover enough cooling to have a significant impact on energy demand when there's a high cooling demand throughout the colder periods due to a higher and constant occupancy or internal gains. Thus the free cooling system could be a viable option for buildings where there's a high and constant occupancy or internal gains, such as schools, factories, or data centers. The LCC for this scenario indicates that the free cooling system is a better investment than the original scenario, while still not being profitable. See Appendix G.2 for LCC.

As seen in Figure 4.1 and 4.2, the designed free cooling system in Plan B can satisfy the cooling demand from October to March and is in operation for a total of 2 450 hours/year. The main parameter influencing these results is the setpoint temperature in which the free cooling is turned off. This study has had a temperature setpoint of 15°C. A higher setpoint would allow the system to be usable throughout greater periods of the year. A simulation was done using the higher setpoint of 20°C to emulate the free cooling system's potential when using self-regulating active chilled beams. The concept of self-regulating beams is described in Section 2.1.1.2. Figure G.3 showcases an increased yearly duration of 2 860 hours when using a setpoint of 20°C. It is equal to 78% of the possible active hours, which is an increase from the previous 67%. Figure G.4 displays the increased usage of the free cooling system over the year. The energy savings were calculated to be 2.7 kWh/m^2, summarized in Figure G.5. That is a slight increase compared to the conventional system's 1.8 kWh/m^2. It is worth noting that this analysis was done to understand how a system with a higher setpoint temperature would perform. A properly designed self-regulating beam system's results might differ. Further research is needed for a more accurate analysis.

Lastly, the studied building was tested in a colder outdoor climate to see how much the free cooling potential would increase. Plan B had the location changed to Kiruna, a city located in the north of Sweden. Every parameter was kept the same as before, except the climate file. The colder climate results in the temperature being below the free cooling threshold of 15°C for more hours of the year. Results for the energy performance are shown in Figure G.5, showing a reduction of 2.3kWh/m^2 which is slightly higher compared to the original location's 1.8 kWh/m^2.

Figure 5.1 was created to illustrate the findings, found when experimenting with the different parameters. The original cases are represented by the conventional ACBs, which can utilize outdoor air of 15°C and the cooling demand originally calculated for the studied office building. As a simplification, these parameters are illustrated as sine- and cosine curves which pinpoint the issue of having a large cooling demand- and small free cooling potential during summer, and vice versa. The three studied parameters: setpoint temperature, internal gains, and outdoor climate affect the graph in different ways. A higher setpoint temperature and a colder outdoor climate both increase the free cooling potential. As seen in the graph, this results in the cooling demand being satisfied for a longer period of the year, as well as reducing the bought cooling slightly during summer. However, what is also apparent is that the free cooling potential during winter is still unchanged and far above the demand, which is the main reason for the rather low influence. The increased internal gains, as a result of the higher occupancy, do on the other hand affect the cooling demand. The graph illustrates a larger, year-round internal gain through the higher occupancy, which can be cooled using free cooling during the cold parts of the year. As a result, there are large savings percentage-wise, indicating that the system would be more suited towards activities with large internal gains year-round. This leads to the ideal cooling demand, which would be a building with a high internal load all year, but also a smaller peak demand during summer. This could be the case with activities such as a data center, where the servers constantly emit huge amounts of heat, and the building could keep a lower cooling demand during summer by not having windows.

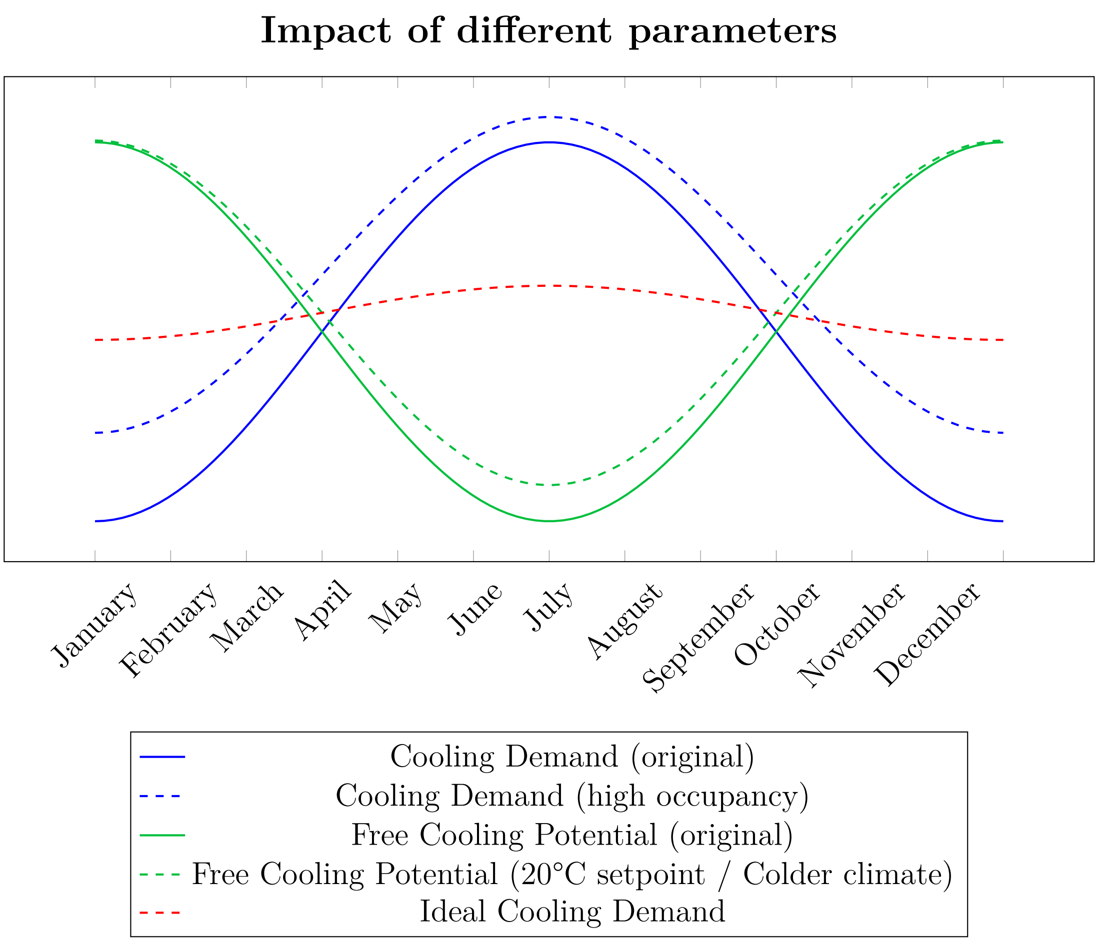


Figure 5.1: Illustration of how the cooling demand and free cooling potential are affected by different parameters.

It's also noteworthy to consider that the building envelope was designed in such a way that the wall's thermal mass did not affect the building's thermal and energy performance. Modifying the envelope to integrate the wall's thermal mass would likely yield different results. An increase in thermal mass would result in a delayed demand for cooling as heat would be stored inside the walls, reduce peak loads, and flatten out the building's temperature fluctuations. Additionally, the combination of night cooling and increased thermal mass could further extend the delayed cooling effect, thereby reducing the cooling demand during the mornings.

6
Conclusion

The previously stated research questions are addressed to serve as a conclusion.

- **Does the study's free cooling system improve the energy performance of a typical office space located in Gothenburg, Sweden?**

  It can be concluded that free cooling by outdoor air generally improves energy performance slightly. The improvement is almost negligible when considering the overall energy performance of a typical office located in Gothenburg.

- **Is the study's free cooling system a profitable investment for a typical office space located in Gothenburg, Sweden?**

  Given the marginal energy savings, free cooling by outdoor air is not a profitable investment for a typical office space in Gothenburg. Over a 20-year period, the lower energy costs are outweighed by the higher investment costs.

- **With the rising climate temperatures, what performance can be expected from the study's free cooling system in the future?**

  Predictions of future climates indicate an increase in outdoor temperatures, which will reduce the performance of the free cooling by outdoor air system as it is reliant on cold ambient air. Consequently, the system will be less effective compared to its performance under current climate conditions.

- **To what extent do the internal gains affect the system's performance?**

  Internal gains generally affect the system's performance more than a change of location or free cooling set point. An increase in internal gains increases the free cooling system's efficacy, as the system has an abundance of available cooling power during colder periods. Subsequently, high internal gains during summer reduce the system's efficacy as it is unable to cover any cooling due to the high outdoor temperature. However, if the cooling demand during colder periods is significant, it can offset the increased cooling demand during summer. This is because the colder climate lasts longer, allowing the free cooling system to meet the majority of the cooling needs.

6.1 Further Research

Suggestions for further investigation and research of the free cooling system:

- **Implementation of an advanced control system**

  Model and evaluate the impact of an advanced control system that regulates flow based on cooling demand rather than maintaining a constant flow. This approach could potentially lower pump electricity and further increase the system's efficiency.

- **In-depth analysis while using self-regulated active chilled beams**

  While the impact of a higher setpoint was examined in this thesis, a comprehensive and detailed implementation of self-regulating active chilled beams was not conducted and is thus suggested.

- **Data validation with real-life case**

  Due to no empirical data being available a challenge to validate data arises. A suggestion for further research is thus to study a building with an equivalent system comparable to the designed free cooling system, with the available empirical data. To provide a more detailed review, it is necessary to assess if the modeled system's results align with real-life cases.

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Appendices

A

Floor Plan A and B

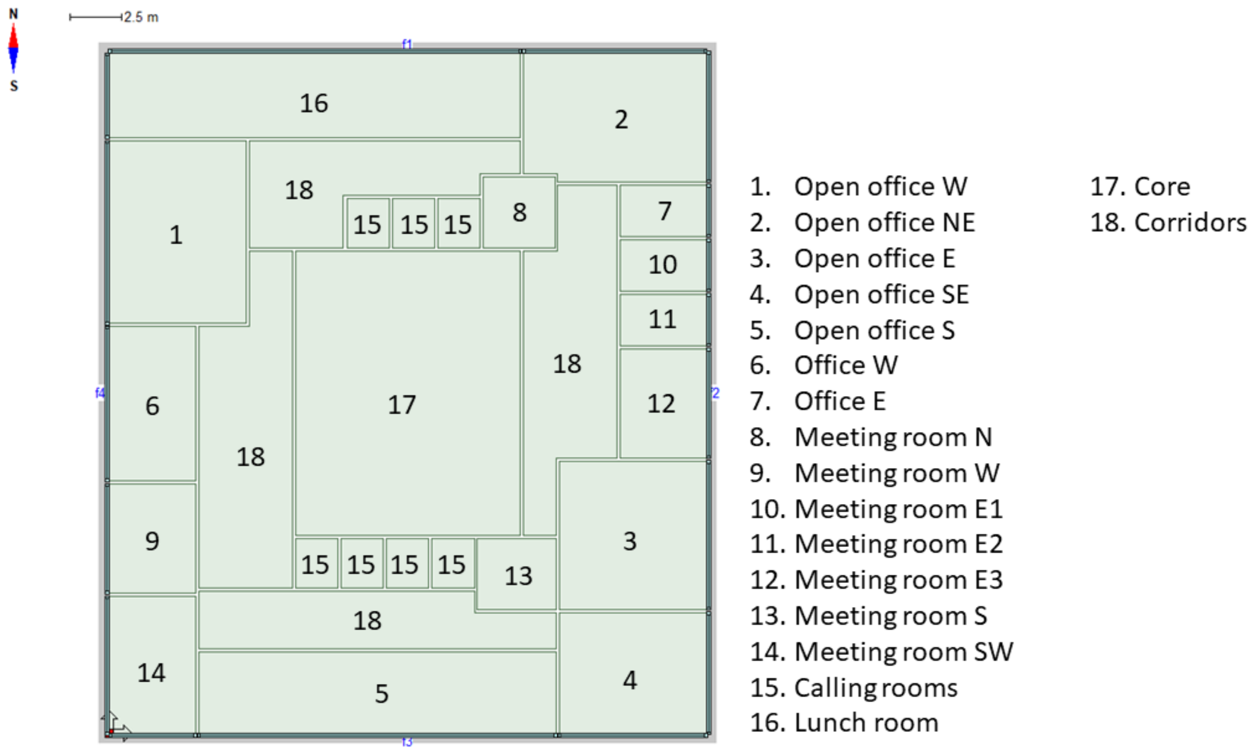

Figure A.1: Overview of Plan A including naming for all spaces.

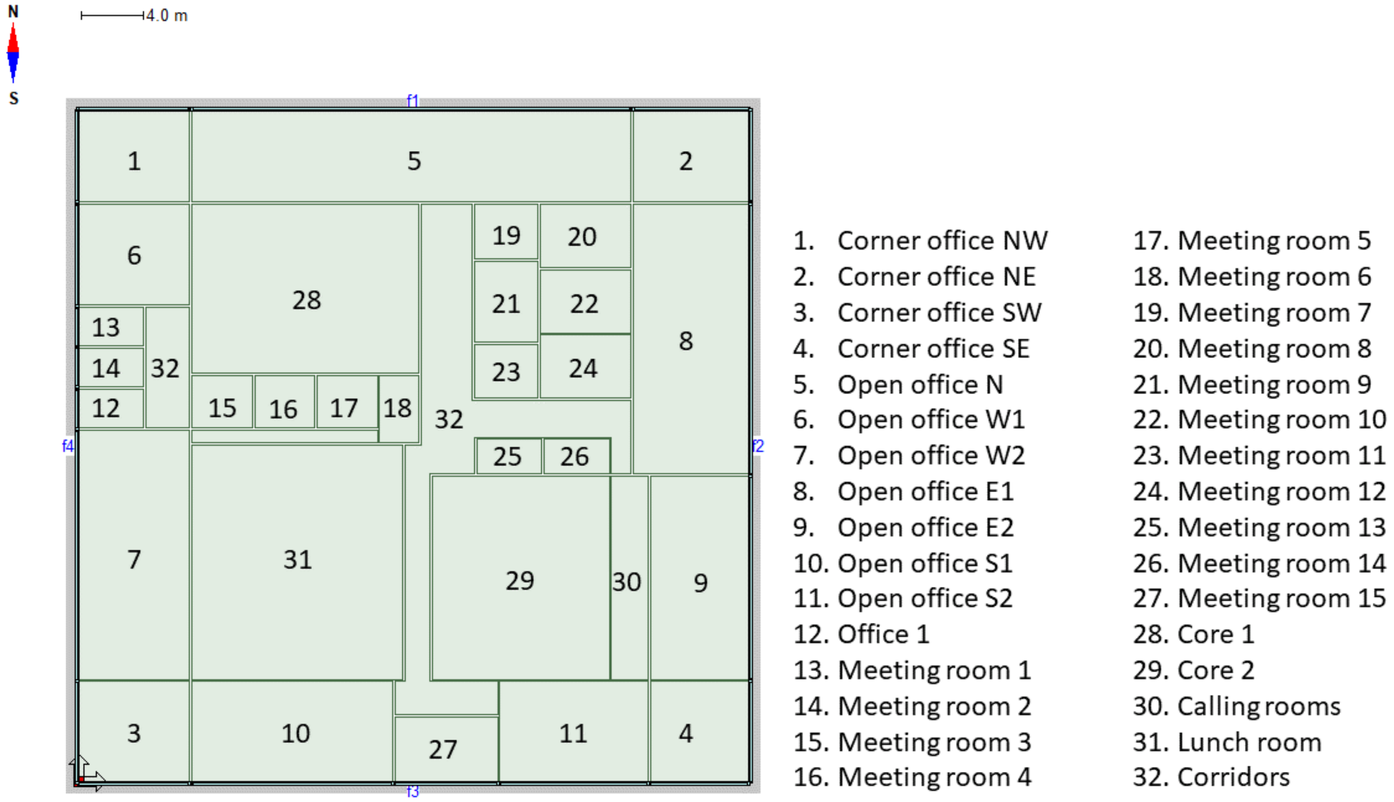

Figure A.2: Overview of Plan B including naming for all spaces.

B

Internal Gains

The equipment gains for the lunch room and dining area are summarized in Table B.1 and B.2. To explain the calculation, the coffee machine in Floor Plan A is used as an example. The coffee machine is assumed to have a power of 1,400 W. Each 30-minute break, it is assumed to be used by 27 people, 1 minute each. The total power is then calculated.

$$\text{Total Power} = 27 \cdot \left(\frac{1}{30}\right) \cdot 1,400 = 1,260\,\text{W}$$

Table B.1: Lunch room's equipment gains (Floor Plan A).

	Times used [n]	Time per use [min]	Time frame [min]	Power [W]	Total power [W]
Fridge	- (constant)	- (constant)	- (constant)	17.1	17.1
Dishwasher	1	60	60	1,800	1,800
Microwave	30	3	60	900	1,350
Coffee Machine	27	1	30	1,400	1,260

Table B.2: Dining area's equipment gains (Floor Plan B).

	Times used [n]	Time per use [min]	Time frame [min]	Power [W]	Total power [W]
Fridge	- (constant)	- (constant)	- (constant)	17.1	17.1
Dishwasher	3	60	60	1,800	5,400
Microwave	80	3	60	900	3,600
Coffee Machine	67	1	30	1,400	3,130

Table B.3: Internal loads Building A.

Zone	Occupants		Lights		Equipment	
	no	no/m^2	W/m^2	kWh/m^2	W/m^2	kWh/m^2
Calling Rooms	7	0.202	6	4.18	12.1	7.99
Core	0	0	4	12.54	0	0
Corridor	0	0	4	12.54	0	0
Meeting Room East No.1 & 2	4	0.382	6	4.02	32.5	25.2
Meeting Room East No.3	8	0.359	6	3.53	26	15.3
Meeting Room North	6	0.501	6	4.96	38.4	31.8
Meeting Room South	6	0.454	6	4.96	34.8	28.8
Meeting Room Southwest	12	0.432	6	3.06	29.5	15.1
Meeting Room West	8	0.358	6	3.53	26	15.3
Office Room East	1	0.095	6	4.94	5.7	4.7
Office Room West	3	0.095	6	4.94	5.7	4.7
Open Office East	8	0.156	6	4.94	9.4	7.7
Open Office Northeast	8	0.145	6	4.94	8.7	7.2
Open Office South	16	0.233	6	4.94	15.7	12.9
Open Office Southeast	6	0.143	6	4.94	8.6	7.1
Open Office West	8	0.136	6	4.94	8.2	6.8
Lunch Room	50	0.611	6	2.01	22.2	19.93

Table B.4: Internal loads Building B.

Zone	Occupants		Lights		Equipment	
	no	no/m^2	W/m^2	kWh/m^2	W/m^2	kWh/m^2
Calling Rooms	6	0.191	6	4.7	11.5	8.87
Core No.1	0	0	4	18.8	0	0
Core No.2	0	0	4	18.8	0	0
Corridor	0	0	4	12.5	0	0
Lunch Room	100	0.487	6	2.7	26.4	21.37
Meeting Room No.1	4	0.381	6	6.8	32.4	36.65
Meeting Room No.2	4	0.381	6	8.9	32.4	48.23
Meeting Room No.3	6	0.461	6	8.8	35.3	51.95
Meeting Room No.4	6	0.461	6	5.3	35.3	31.05
Meeting Room No.5	6	0.461	6	5.0	35.3	29.22
Meeting Room No.6	4	0.371	6	7.5	31.6	39.31
Meeting Room No.7	6	0.418	6	3.1	32.1	16.33
Meeting Room No.8	8	0.336	6	4.1	24.3	16.53
Meeting Room No.9	8	0.375	6	1.5	27.2	6.92
Meeting Room No.10	8	0.336	6	6.3	24.3	25.38
Meeting Room No.11	4	0.279	6	5.4	23.7	21.46
Meeting Room No.12	8	0.336	6	4.8	23.3	19.28
Meeting Room No.13	4	0.435	6	2.7	37	16.75
Meeting Room No.14	4	0.435	6	5.4	37	33.5
Meeting Room No.15	12	0.433	6	2.5	29.6	12.56
Office Room West	1	0.095	6	4.47	5.7	4.25
Open Office East No.1	18	0.139	6	4.47	8.3	6.21
Open Office East No.2	16	0.194	6	4.47	11.6	8.65
Open Office North	32	0.193	6	4.47	11.6	8.62
Open Office Northeast	8	0.183	6	4.47	11	8.19
Open Office Northwest	8	0.191	6	4.47	11.5	8.53
Open office South No.1	20	0.239	6	4.47	14.4	10.68
Open Office South No.2	12	0.194	6	4.47	11.6	8.65
Open Office Southeast	8	0.197	6	4.47	11.8	8.78
Open Office Southwest	8	0.173	6	4.47	10.4	7.74
Open Office West No.1	6	0.13	6	4.47	7.8	5.8
Open Office West No.2	16	0.139	6	4.47	8.4	6.22

C

Schedules

C.1 Plan A

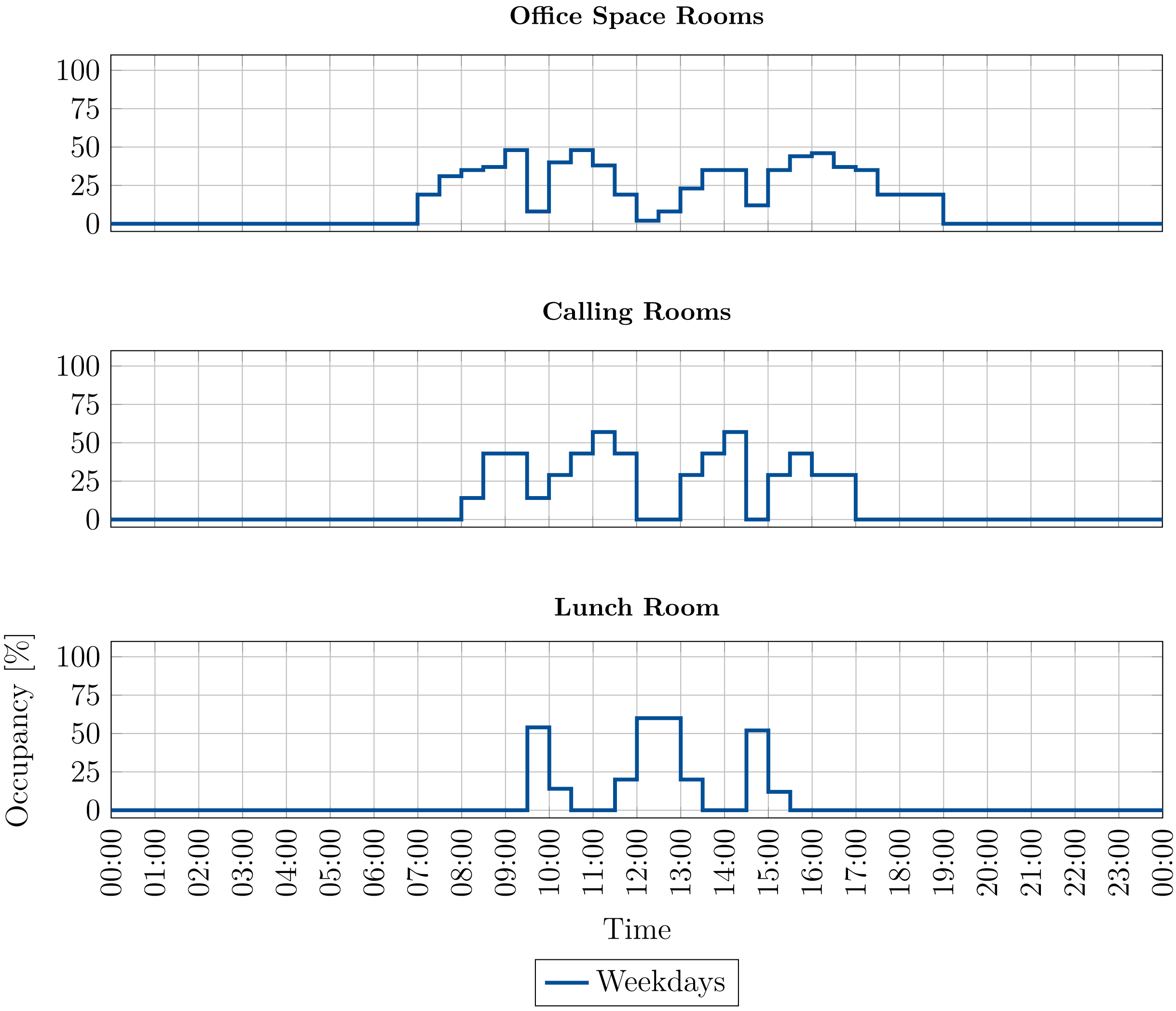

Figure C.1: Occupancy schedules Plan A (1-3).

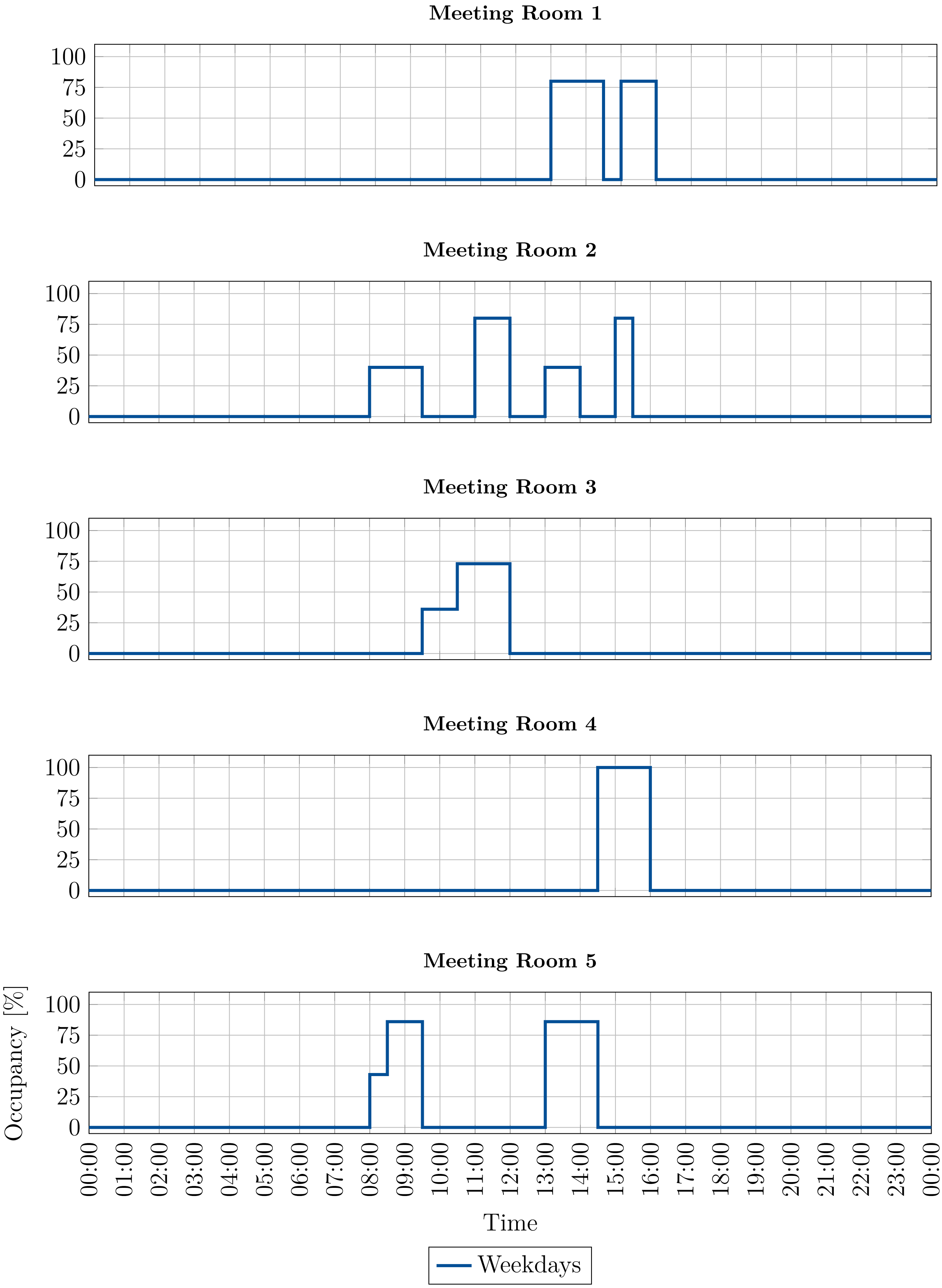

Figure C.2: Occupancy schedules Plan A (4-8).

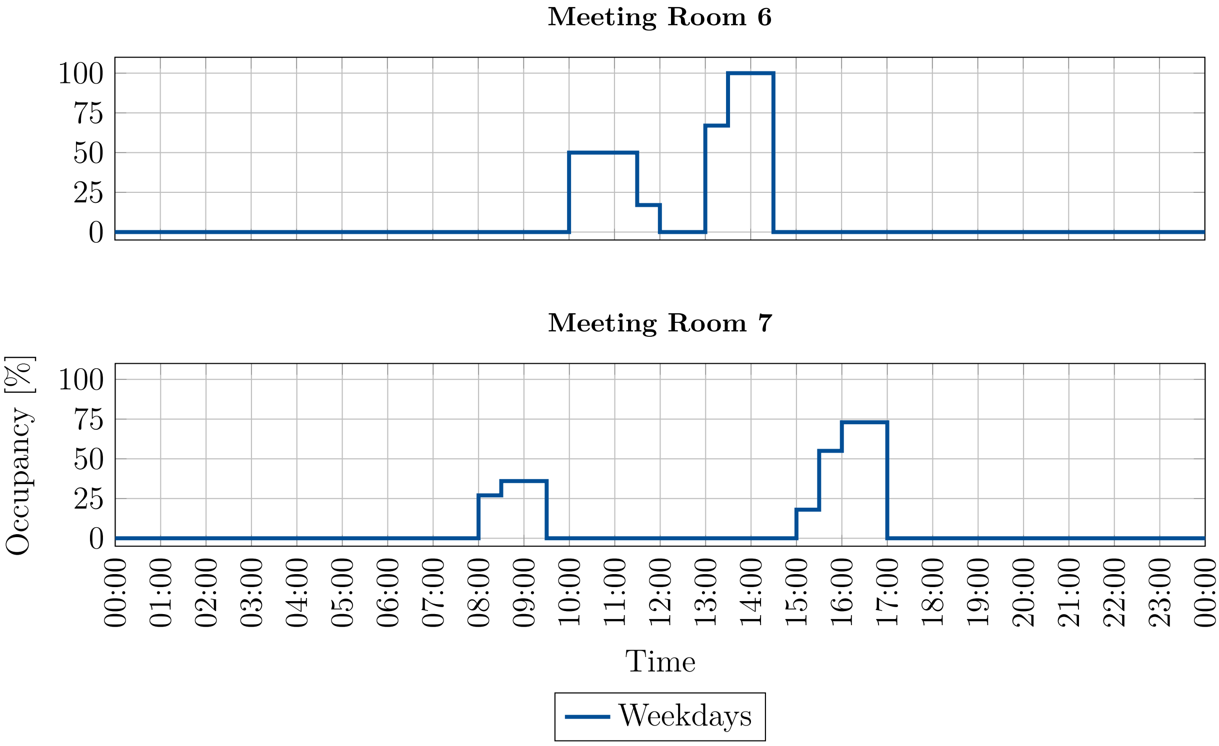

Figure C.3: Occupancy schedules Plan A (9-10).

C.2 Plan B

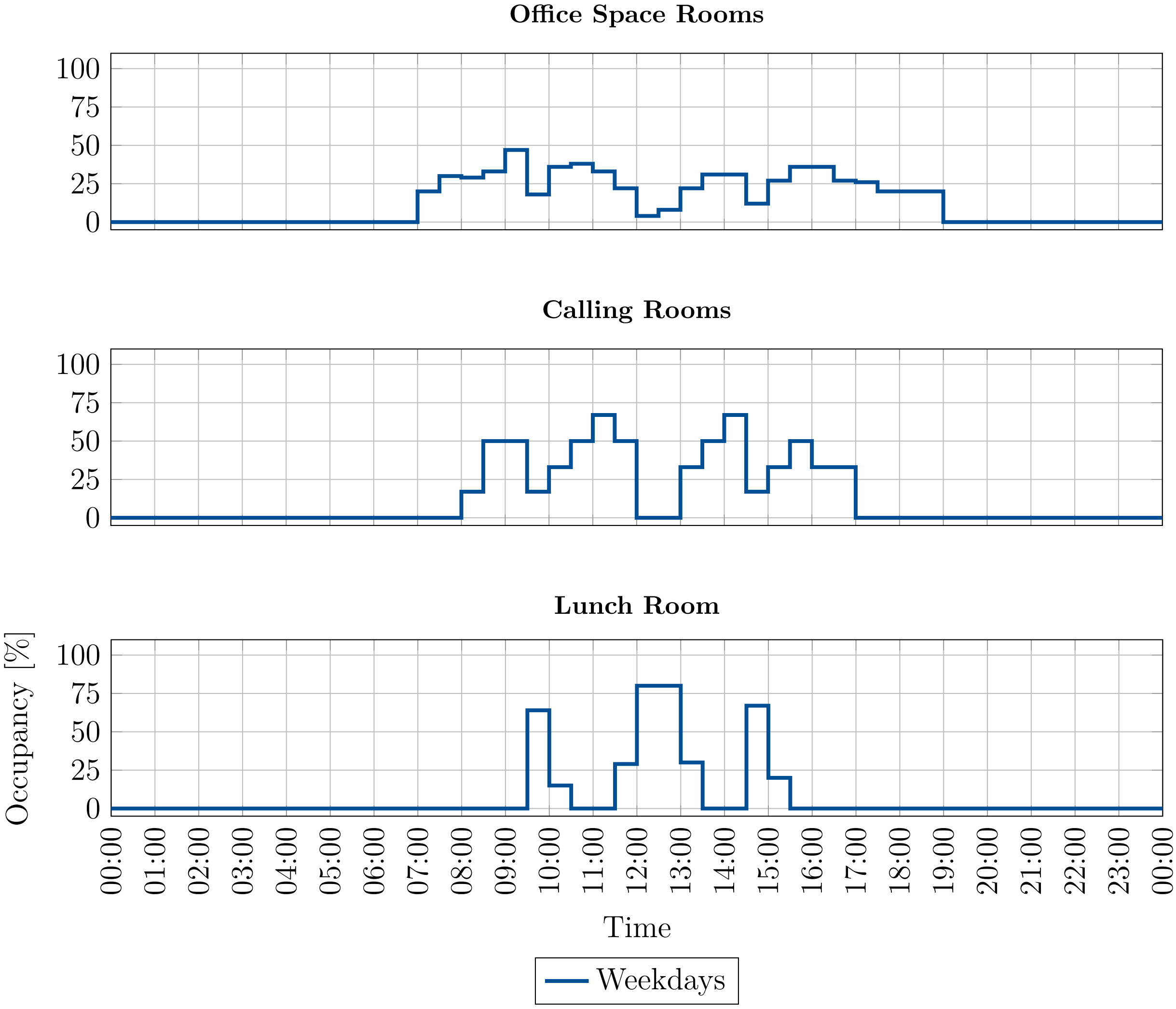

Figure C.4: Occupancy schedules Plan B (1-3).

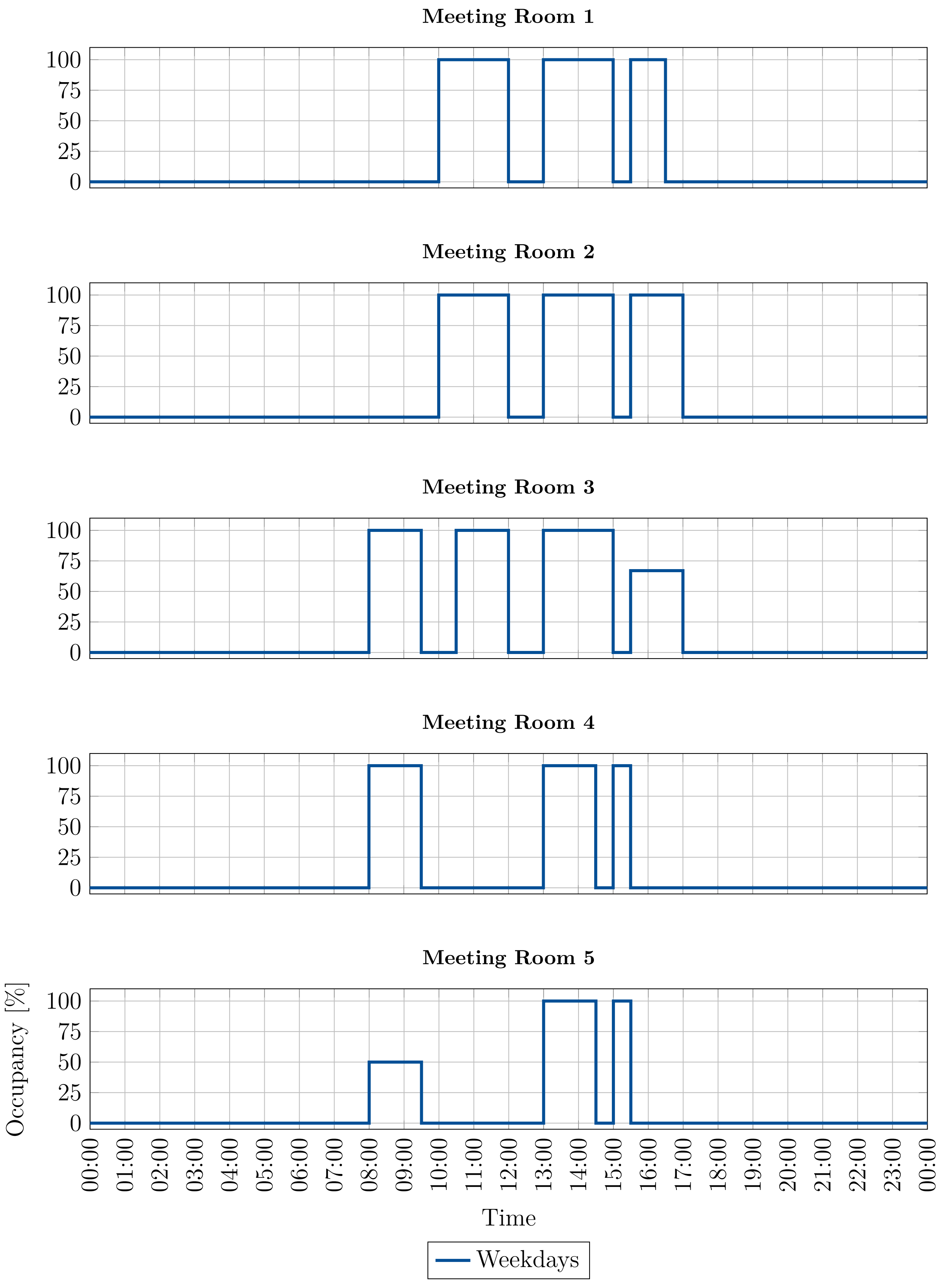

Figure C.5: Occupancy schedules Plan B (4-8).

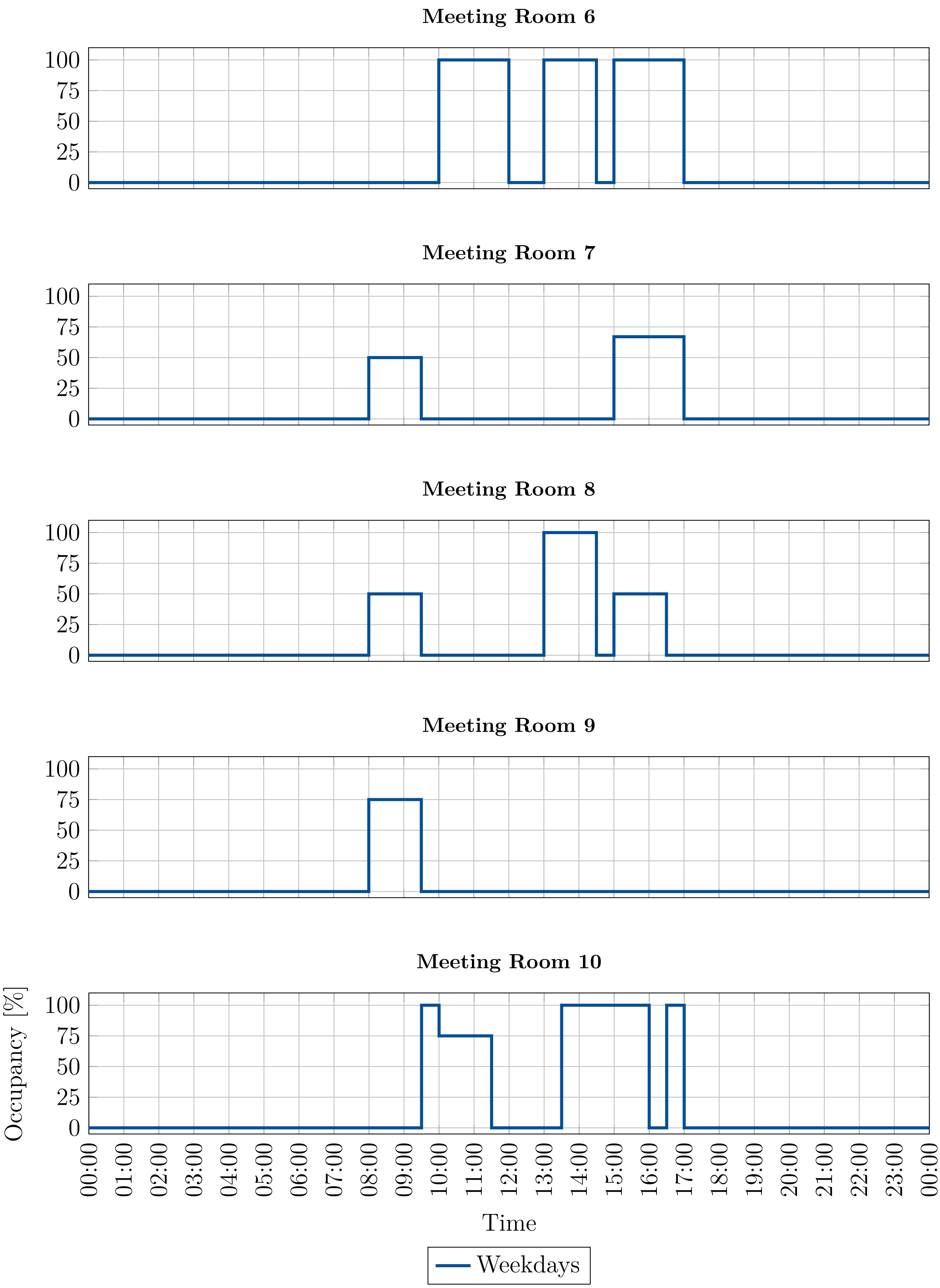

Figure C.6: Occupancy schedules Plan B (9-13).

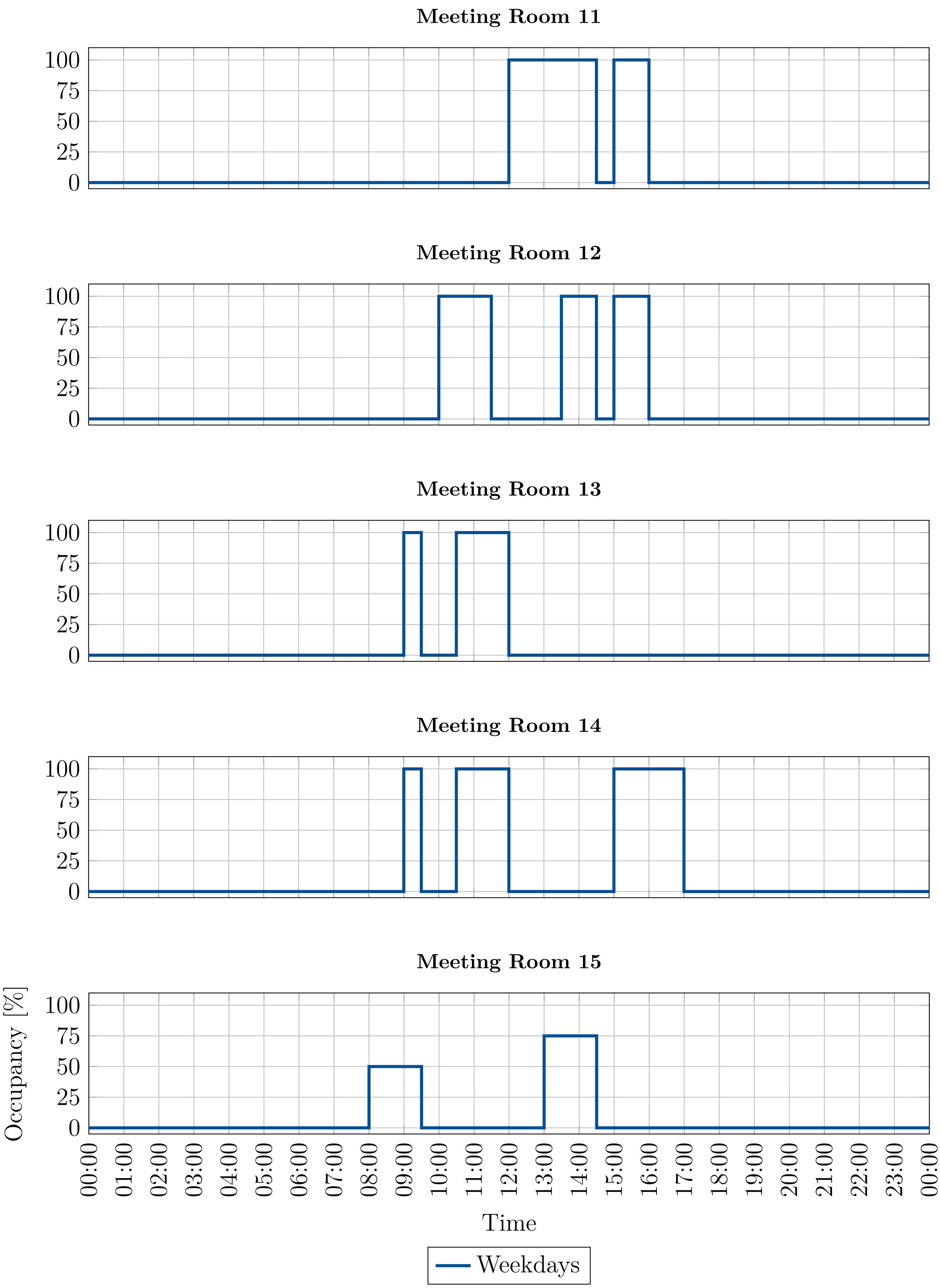

Figure C.7: Occupancy schedules Plan B (14-18).

C.3 AHU & Pumps

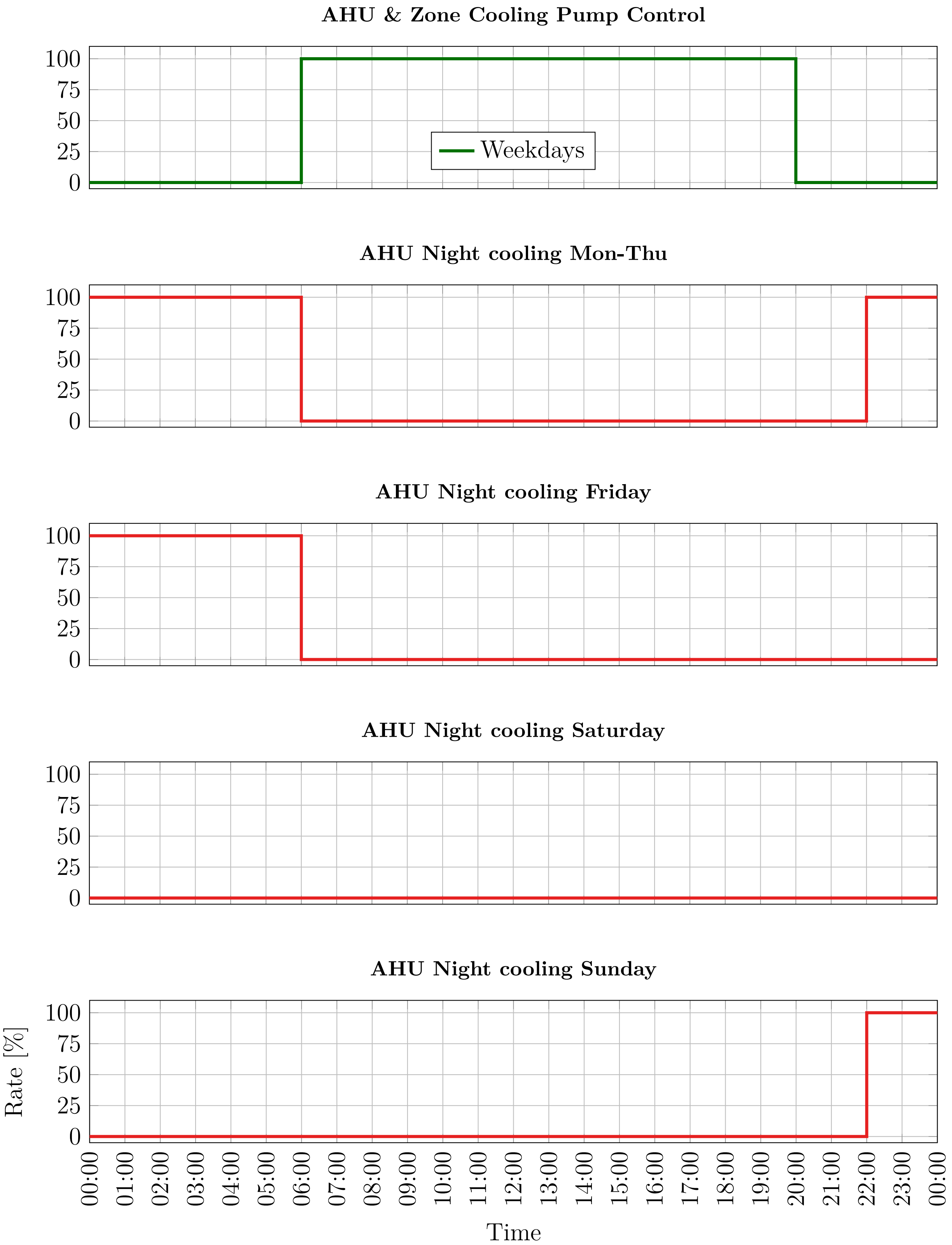

Figure C.8: AHU & zone cooling pump schedules.

D

Cooling Power & Air Flows

Table D.1: Cooling power & Air flows, Floor Plan A.

Zone	Cooling Power	Air flow	
	W	l/s	l/s/m^2
Calling Rooms	380	10	2
Core	0	30	0.35
Corridors	0	15	0.35
Meeting Room East No.1 & 2	1,025	20-32	1.9-3.8
Meeting Room East No.3	1,670	20-63	0.9-3.6
Meeting Room North	1,025	20-46	1.7-5.0
Meeting Room South	1,025	20-46	1.5-4.5
Meeting Room Southwest	1,670	20-94	0.7-4.3
Meeting Room West	1,670	20-64	0.9-3.6
Office Room East	645	20	1.9
Office Room West	1,290	30	1.0
Open Office East	2,150	68	1.3
Open Office Northeast	1,935	70	1.3
Open Office South	4,644	153	2.2
Open Office Southeast	1,591	51	1.2
Open Office West	1,935	70	1.2
Lunch Room	3,870	125-250	1.5-3.1

Table D.2: Cooling power & Air flows, Floor Plan B.

Zone	Cooling power	Air flow	
	W	l/s	l/s/m^2
Calling Rooms	380	10	1.9
Core No.1	0	55	0.35
Core No.2	0	52	0.35
Corridor	0	54	0.35
Dining area	6,450	250-500	1.2-2.4
Meeting Room No.1	1,025	20-40	1.9-3.8
Meeting Room No.2	1,025	20-40	1.9-3.8
Meeting Room No.3	1,025	20-60	1.5-4.6
Meeting Room No.4	1,025	20-60	1.5-4.6
Meeting Room No.5	1,025	20-60	1.5-4.6
Meeting Room No.6	645	20-40	1.9-3.7
Meeting Room No.7	1,025	20-60	1.4-4.2
Meeting Room No.8	1,670	20-80	1.7-3.4
Meeting Room No.9	1,670	20-80	1.9-3.8
Meeting Room No.10	1,670	20-80	1.7-3.4
Meeting Room No.11	645	20-40	1.4-2.8
Meeting Room No.12	1,670	20-80	1.7-3.4
Meeting Room No.13	645	20-40	2.2-4.4
Meeting Room No.14	645	20-40	2.2-4.4
Meeting Room No.15	1,670	20-120	1.4-4.3
Office Room East	645	20	1.9
Open Office East No.1	4,644	153	1.2
Open Office East No.2	4,128	136	1.7
Open Office North	8,027	272	1.6
Open Office Northeast	2,050	70	1.6
Open Office Northwest	2,114	70	1.7
Open Office South No.1	4,838	170	2.0
Open Office South No.2	3,612	102	1.7
Open Office Southeast	2,064	70	1.7
Open Office Southwest	2,000	70	1.5
Open Office West No.1	1,612	51	1.1
Open Office West No.2	4,128	136	1.2

E

Cost Calculations

E.1 Expenses

Table E.1: Investment costs: Floor Plan A.

	No. Units [n]	Unit cost [SEK/n]	Total cost [SEK]
Free Cooling Coil	1	40,000	40,000
Hydronic Heat Exchanger	1	16,500	16,500
Circulation Pump	2	26,180	52,360
Butterfly valve	5	3,860	21,535
Regulating valve	1	18,900	18,900
Safety valve	1	760	760
Control valve	1	15,855	15,855
Strainer	1	2,250	2,250
Differential pressure gauge	2	2,550	5,100
Expansion vessel	1	5,410	5,410
Mixing vessel	1	7,210	7,210
Pipe	30 [m]	1,715	51,825
Glycol	-	-	5,000
Control system	1	50,000	50,000
Labor cost	-	-	131,000
Verification / Adjustment	-	-	1,000
Total investment cost:			**425,000 SEK**

Table E.2: Investment costs: Floor Plan B.

	No. Units [n]	Unit cost [SEK/n]	Total cost [SEK]
Free Cooling Coil	1	60,000	60,000
Hydronic Heat Exchanger	1	38,130	38,130
Circulation Pump	2	32,830	65,660
Butterfly valve	5	8,700	45,735
Regulating valve	1	30,170	30,170
Safety valve	1	760	760
Control valve	1	28,280	28,280
Strainer	1	3,760	3,760
Differential pressure gauge	2	2,550	5,100
Expansion vessel	1	5,410	5,410
Mixing vessel	1	7,210	7,210
Pipe	30 [m]	2,432	73,335
Glycol	-	-	5,000
Control system	1	50,000	50,000
Labor cost	-	-	178,500
Verification / Adjustment	-	-	1,000
Total investment cost:			**600,000 SEK**

Table E.3: Maintenance costs: Floor Plan A.

	Maintenance cost [% of investment cost]	Maintenance cost [SEK/year]
Free Cooling Coil	3	1,200
Hydronic Heat Exchanger	3	500
Circulation Pump	2	1,050
Butterfly valve	1	215
Regulating valve	1	190
Safety valve	1	10
Control valve	6	950
Strainer	1	25
Differential pressure gauge	3	150
Expansion vessel	1	55
Mixing vessel	1	70
Pipe	1	520
Control system	4	2,000
Total maintenance cost:		**7,000 SEK/year**

Table E.4: Maintenance costs: Floor Plan B.

	Maintenance cost [% of investment cost]	Maintenance cost [SEK/year]
Free Cooling Coil	3	1,800
Hydronic Heat Exchanger	3	1,150
Circulation Pump	2	1,310
Butterfly valve	1	460
Regulating valve	1	300
Safety valve	1	10
Control valve	6	1,700
Strainer	1	40
Differential pressure gauge	3	150
Expansion vessel	1	55
Mixing vessel	1	70
Pipe	1	735
Control system	4	2,000
Total maintenance cost:		**9,800 SEK/year**

Table E.5: Bought cooling (energy): Floor Plan A.

		Reference Case		**Free Cooling Case**	
Month	Price [SEK/MWh]	Energy use [MWh]	Monthly cost [SEK]	Energy use [MWh]	Monthly cost [SEK]
January	145	0.0	7	0.0	0
February	145	0.0	7	0.0	0
March	145	0.0	28	0.0	0
April	243	0.7	166	0.1	31
May	326	3.8	1,253	2.2	709
June	344	4.5	1,546	3.5	1,202
July	344	5.5	1,896	5.5	1,875
August	344	6.2	2,127	6.0	2,054
September	344	1.8	613	1.1	390
October	291	0.3	75	0.0	3
November	247	0.1	15	0.0	0
December	145	0.0	7	0.0	0
Total cooling cost (energy):		**7,750 SEK/year**		**6,300 SEK/year**	

Table E.6: Bought cooling (energy): Floor Plan B.

Month	Price [SEK/MWh]	**Reference Case** Energy use [MWh]	**Reference Case** Monthly cost [SEK]	**Free Cooling Case** Energy use [MWh]	**Free Cooling Case** Monthly cost [SEK]
January	145	1.1	152	0	0
February	145	1.0	137	0	0
March	145	1.3	185	0	0
April	243	2.6	635	0.4	88
May	326	10.3	3,356	5.3	1,720
June	344	12.0	4,143	8.8	3,028
July	344	12.6	4,322	11.8	4,058
August	344	13.7	4,725	12.8	4,418
September	344	5.0	1,713	2.9	1,004
October	291	1.6	475	0.1	20
November	247	1.0	255	0	0
December	145	1.0	147	0	0
Total cooling cost (energy):		**20,250 SEK/year**		**14,350 SEK/year**	

Table E.7: Power cost (Göteborg Energi, 2024).

Peak power [kW]	Set price [SEK/year]	Variable price [SEK/kW,year]
0 - 100	**21,380**	**881**
100-250	**37,580**	**719**
250-500	75,830	566
500-1,000	112,830	492
$> 1{,}000$	137,830	467

The peak power demand was measured to be 98 kW and 205 kW for Floor plans A and B respectively. Table E.7 was then used to determine the set- and variable price. Corresponding values were used for the following calculations.

$$\text{Power cost}_A = 21,380 + 881 \cdot 98 = 108,000\,\text{SEK}$$

$$\text{Power cost}_B = 37,580 + 719 \cdot 205 = 185,000\,\text{SEK}$$

As stated in the Methodology, the measured peak power was measured to be equal in the reference- and free cooling case. Therefore, the calculated power cost was excluded from the LCC analysis.

Table E.8: Bought cooling (flow): Floor Plan A.

Month	Price [SEK/m³]	Reference Case		Free Cooling Case	
		Total flow [m³]	Total cost [SEK]	Total flow [m³]	Total cost [SEK]
Jan.-Dec.	0.8	3,406	2,700	2,540	2,000
Total cooling cost (flow):		**2,700 SEK/year**		**2,000 SEK/year**	

Table E.9: Bought cooling (flow): Floor Plan B.

Month	Price [SEK/m³]	Reference Case		Free Cooling Case	
		Total flow [m³]	Total cost [SEK]	Total flow [m³]	Total cost [SEK]
Jan.-Dec.	0.8	8,138	6,500	6,070	4,700
Total cooling cost (flow):		**6,500 SEK/year**		**4,850 SEK/year**	

Table E.10: Electricity cost: Floor Plan A.

Month	Price (Elområde 3) [öre/kWh]	Free Cooling Case	
		Energy use [kWh]	Monthly cost [SEK]
January	106.4	195	208
February	95.1	174	1,166
March	93.9	191	180
April	80	219	175
May	52.6	292	154
June	63.3	272	172
July	48.7	281	137
August	50.5	253	128
September	37.3	232	87
October	49.7	209	104
November	103.3	184	190
December	98.7	188	186
Total additional electricity cost:			**1,880 SEK/year**

Table E.11: Electricity cost: Floor Plan B.

Month	Price (Elområde 3) [öre/kWh]	Free Cooling Case: Energy use [kWh]	Free Cooling Case: Monthly cost [SEK]
January	106.4	370	1,466
February	95.1	364	1,163
March	93.9	348	1,215
April	80	385	1,108
May	52.6	419	819
June	63.3	389	887
July	48.7	347	629
August	50.5	359	674
September	37.3	387	519
October	49.7	387	715
November	103.3	357	1,327
December	98.7	356	1,305
Total additional electricity cost:		**11,830 SEK/year**	

E.2 Calculations

Equation E.1 was used to calculate the present value. The second part of the equation represents the present value factor. The equation was modified to account for both the discount rate and expected yearly price increases by the inclusion of (x).

$$PVA = CF \cdot \frac{1-(1+(r-x))^{-T}}{r-x} \tag{E.1}$$

Table E.12 and E.13 shows the calculated present value, also presented graphically in Figure 4.7.

Table E.12: Life-cycle cost calculation: Floor Plan A.

	x [%]	Present value factor [-]	Reference Case: CF [SEK/year]	Reference Case: Present value [SEK]	Free Cooling Case: CF [SEK/year]	Free Cooling Case: Present value [SEK]
Investment	-	-	-	-	425,000*	425,000
Maintenance	2	14.21	-	-	7,000	98,500
Cooling (energy)	3	15.59	7,750	120,700	6,300	97,700
Cooling (flow)	-	11.95	2,700	32,600	2,000	24,300
Electricity	6.4	22.02	-	-	1,880	41,500
Sum				**153,300**		**687,000**

** One time payment, [SEK]*

Table E.13: Life-cycle cost calculation: Floor Plan B.

	x [%]	Present value factor [-]	Reference Case CF [SEK/year]	Reference Case Present value [SEK]	Free Cooling Case CF [SEK/year]	Free Cooling Case Present value [SEK]
Investment	-	-	-	-	600,000*	600,000
Maintenance	2	14.21	-	-	9,800	139,300
Cooling (energy)	3	15.59	20,250	315,700	14,350	223,700
Cooling (flow)	-	11.95	6,500	77,700	4,850	58,000
Electricity	6.4	22.02	-	-	11,830	260,500
Sum				**393,400**		**1,278,900**

** One time payment, [SEK]*

F

Results

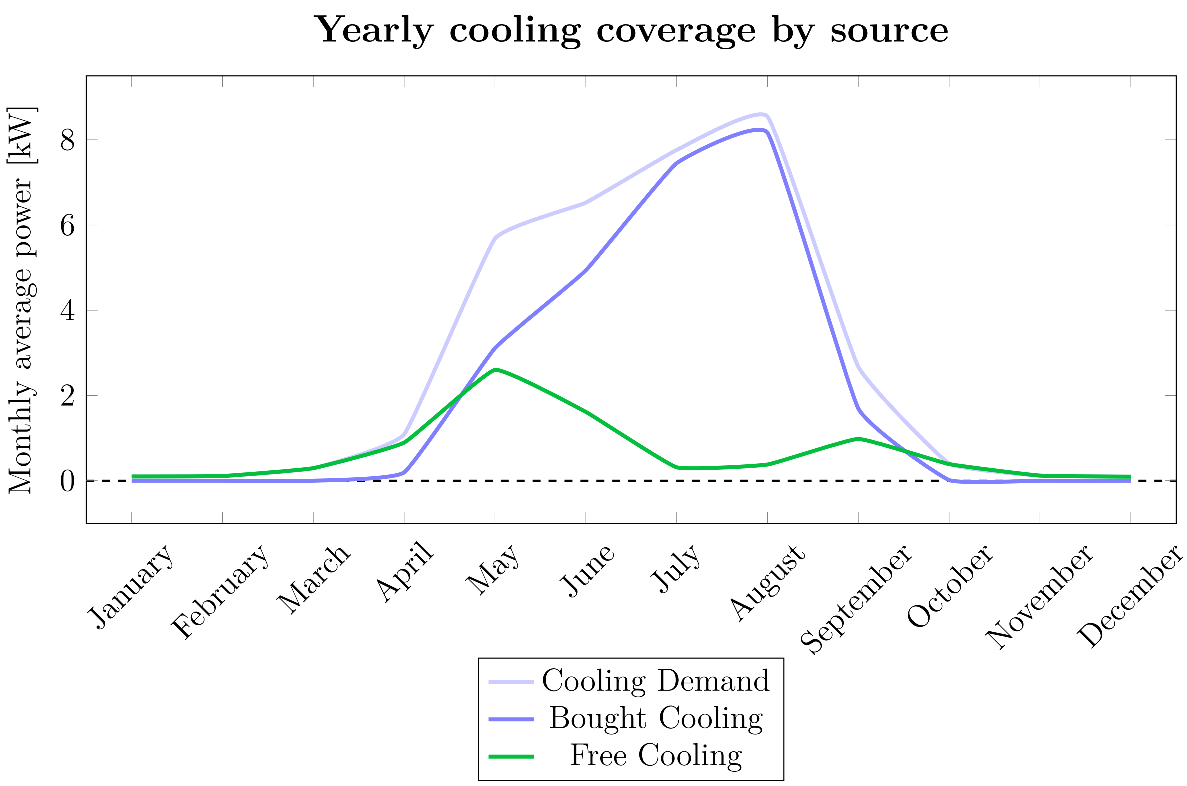

Figure F.1: Monthly average cooling supply and demand (Floor plan A).

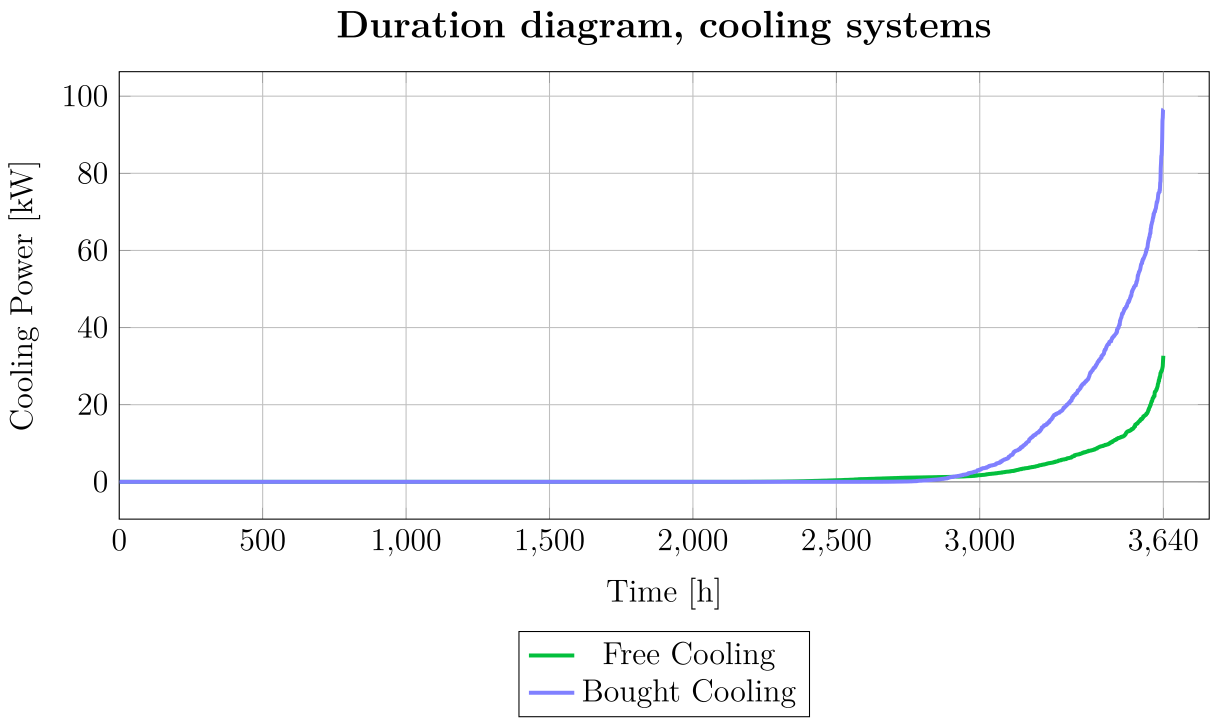

Figure F.2: Duration diagram for free cooling- and district cooling operation (Floor plan A).

G

Discussion

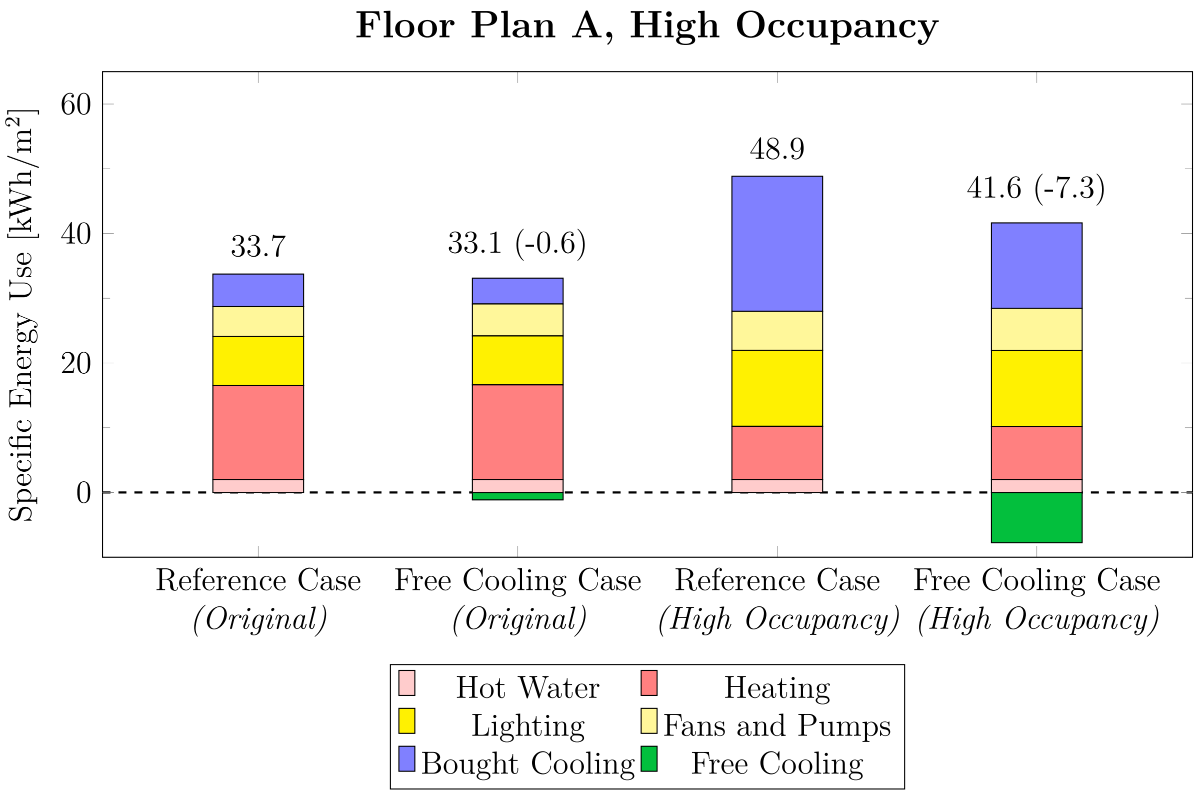

Figure G.1: Comparison in specific energy performance between original- and high occupancy cases.

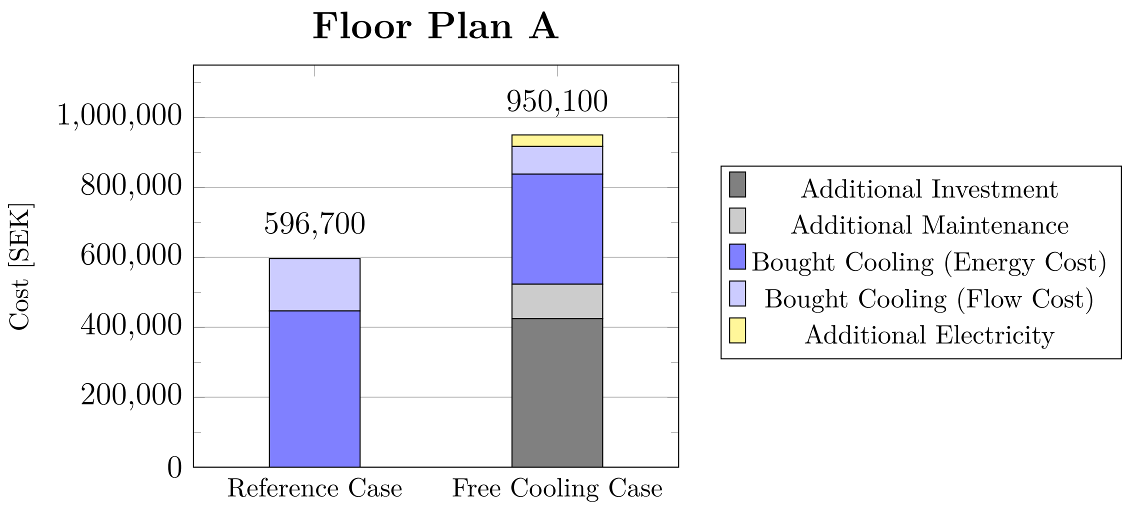

Figure G.2: Total life-cycle costs having high occupancy (Floor Plan A).

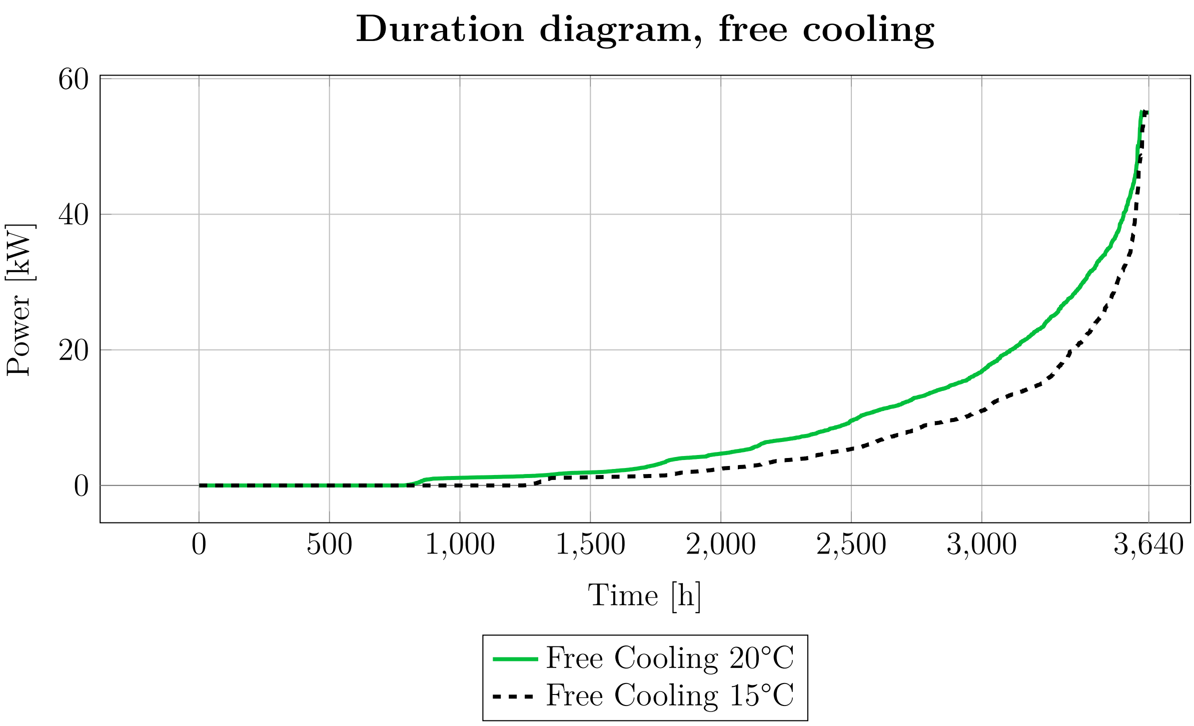

Figure G.3: Duration diagram for free cooling operation with a setpoint of 20°C (Floor Plan B).

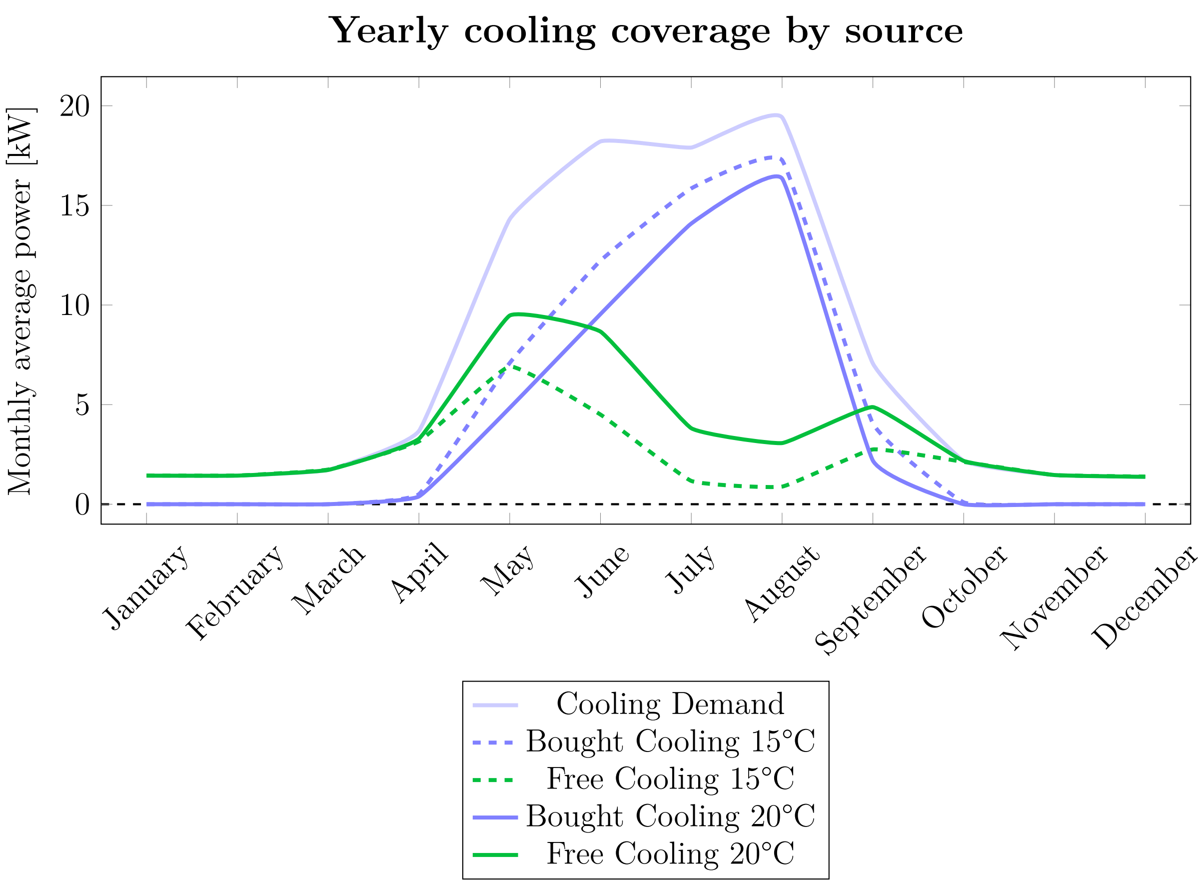

Figure G.4: Monthly average cooling supply and demand - comparison (Floor Plan B).

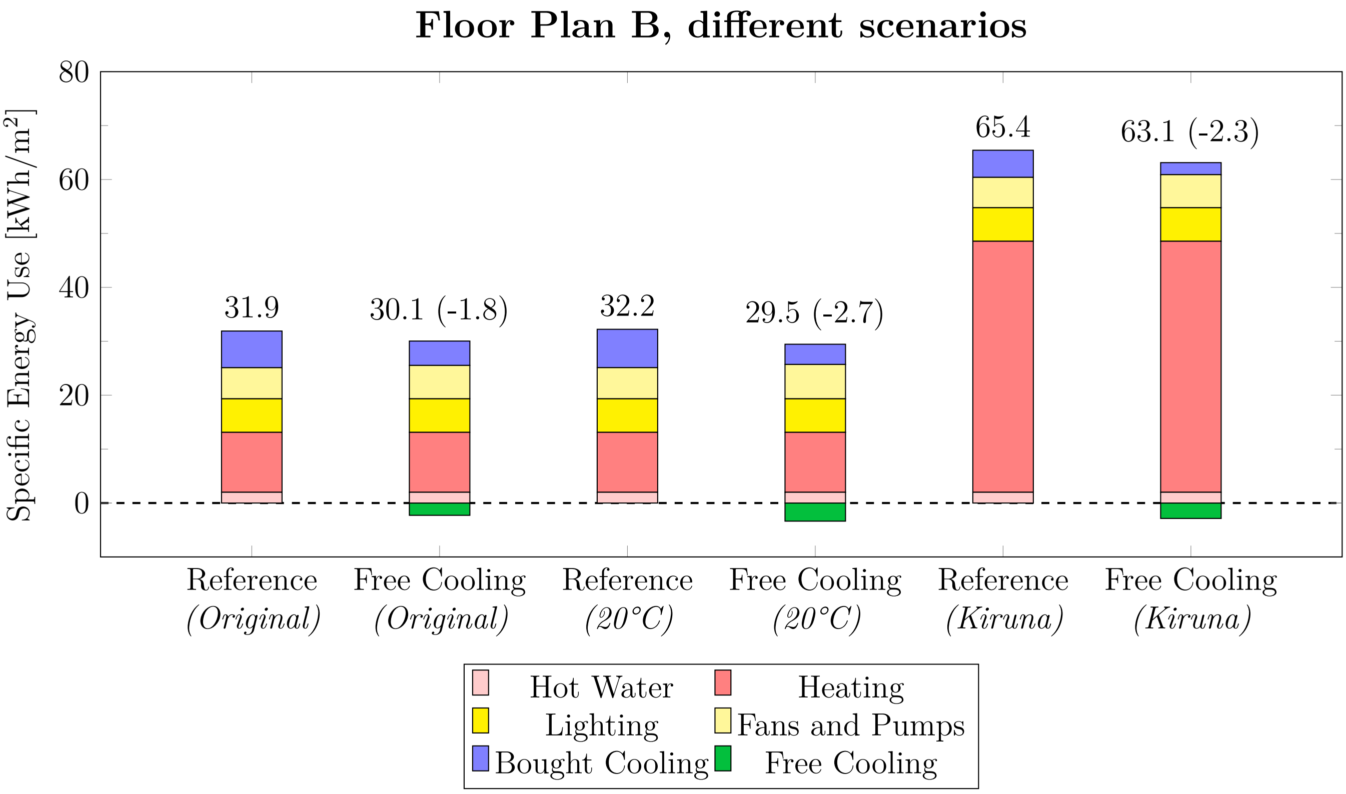

Figure G.5: Comparison in specific energy performance.

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