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Integrated Energy concepts for high performance hotel buildings

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MASTER THESIS

for

Student: Silje Marie Smitt
Spring 2017

Integrated Energy concepts for high performance hotel buildings

Integrerte energikonsepter for energikrevende hotellbygninger

Background and objective

Norway is a Tourist Destination and has a large number of hotels all across the country. Hotels do have a complex energy distribution network and requirements. Heat recovery and integrated energy concepts are not widely applied, i.e. there is a huge potential to further reduce the specific primary energy demand of these kind of building. The development and implementation of such concepts would reduce the cost of ownership and the environmental footprint of the sector significantly.

As an example for a new installation, Britannia Hotel in the city centre of Trondheim will be complete rebuild during the next two years. The surrounding buildings do have complementary heating and cooling demands with respect to the new hotel. Since these buildings are also owned by the same property owner, there is a large potential to develop an integrate energy system for the entire city block. This hotel will be one of the cases for the Master Thesis.

Current heat pumping systems are operated with non-natural working fluids, therefore focus will be given to develop a sustainable energy concept based on heat pumps with natural working fluids, providing the entire heating and cooling demands and utilizing surplus heat to minimize the purchase of thermal energy. Short term energy storage should be considered to be able to handle peak demands internally.

The objective of the Master Thesis is to develop/apply an energy flow simulation tool (Modelica), enabling the evaluation of different energy system concepts.

The following tasks are to be considered:

1. Literature review: energy systems for high performance buildings, surplus recovery heat exchangers and concepts, thermal energy storage devices, etc.
2. Describe and develop the sub-systems for the energy dirstribution and storage system, including interfaces to external suppliers or clients.
3. Develop/apply an energy flow (hour by hour) simulation tool (Modelica) to evaluate sytem alternatives. (Describe input parmeter matrix for the evalutation tool.
4. Analysis and discussion of simulated scenarios
5. Summary report
6. Draft scientific paper related to the findings of the Master Thesis
7. Proposal for further work

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Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analyzed carefully.

The thesis should be formulated as a research report with summary both in English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

The candidate is requested to initiate and keep close contact with his/her academic supervisor(s) throughout the working period. The candidate must follow the rules and regulations of NTNU as well as passive directions given by the Department of Energy and Process Engineering.

Risk assessment of the candidate's work shall be carried out according to the department's procedures. The risk assessment must be documented and included as part of the final report. Events related to the candidate's work adversely affecting the health, safety or security, must be documented and included as part of the final report. If the documentation on risk assessment represents a large number of pages, the full version is to be submitted electronically to the supervisor and an excerpt is included in the report.

Pursuant to "Regulations concerning the supplementary provisions to the technology study program/Master of Science" at NTNU §20, the Department reserves the permission to utilize all the results and data for teaching and research purposes as well as in future publications.

The final report is to be submitted digitally in DAIM. An executive summary of the thesis including title, student's name, supervisor's name, year, department name, and NTNU's logo and name, shall be submitted to the department as a separate pdf file. Based on an agreement with the supervisor, the final report and other material and documents may be given to the supervisor in digital format.

☐ Work to be done in lab (Water power lab, Fluids engineering lab, Thermal engineering lab)
☐ Field work

Department of Energy and Process Engineering, 15. January 2017

Prof. Dr.-Ing. Armin Hafner
Academic Supervisor

Research Advisor: Prof. Trygve M. Eikevik

Preface

This thesis is the written final work of my Master´s degree in Mechanical Engineering for the Department of Energy and Process Engineering at the Norwegian University of Science and Technology (NTNU), spring 2017.

The thesis describes an integrated energy system and distribution network for a hotel located in Trondheim. The aim is to investigate the thermodynamic potential of a heat pumping system based on natural working fluids and heat recovery from available sources. Simulations of the system have been conducted and the main results are listed and discussed.

The simulation tool (Dymola) is based on Modelica, which I previously had no experience with. I have learned a lot from using this tool, both in regards to simulation setup and system control.

A team of supervisors supported the work for my project. My main supervisor was Professor Armin Hafner, with whom I discussed different concepts and all the details of the system. I benefited immensely from his experience with combined heating and cooling systems. Postdoctoral fellow Ángel Álvarez Pardiñas has been a great support in regards to system modeling and getting familiar with the simulation tool. At the same time I was working on the thesis my colleague Håkon Selvnes was working with the tool as well. He has been a great support and my go to person when I needed to bounce around ideas. I would also like to thank Yacine Lakel for his help with data processing. Last but not least, a special thanks to all my fellow students for their encouragement and patience during these past months.

Silje Marie Smitt Trondheim, July 1st 2017

Abstract

On a national level, buildings are responsible for roughly 40% of the primary energy consumption, where domestic hot water and space heating are major contributors. By introducing an energy system that employs natural working fluids to cover all thermal demands, one is able to induce both economical and environmental benefits. The aim of this report is to reveal the energy related potential of a combined heat pump and refrigeration system intended for a hotel in Trondheim. Optimizing elements, such as heat recovery and thermal storage, have been included in the energy system.

The building simulation software *Simien* was utilized to create a model of the hotel, in order to estimate annual demands for space heating and cooling. Due to limitations in the software regarding transient operations, additional demand curves had to be constructed for domestic hot water, refrigeration and freezing loads. A model of the energy system was created in *Dymola*, where several simulations were carried out in order to investigate the performance and potential of the CO_2 and propane heat pumps.

The simulations revealed that the purposed energy system is able to cover all thermal demands for the seasonal scenarios. The annual average energy efficiency for the CO_2 system was found to be 5.55. By implementing thermal storage for space heating and cooling, the loads were reduced considerably in comparison to actual demand. The consequences of this being peak load reduction and a delayed of a few hours. The magnitude of the storage tanks limits the number of stops for the propane heat pump and enables a high efficiency for the system. A yearly average energy efficiency of 5.64 was found for the propane heat pump.

The energy system is highly efficient compared to more traditional heat pump solutions, and a considerable reduction in operational costs can be achieved. The current system design can be improved by reducing heat rejection to the ambient. In addition, the CO_2 mid-pressure level should be increased for better performance.

Sammendrag

På landsbasis kan omlag 40% av det totale energiforbruket knyttes til driften av bygninger. De største bidragsyterne innen denne kategorien er henholdsvis arealoppvarming og produksjon av tappevann. Ved å implementere et energisystem som tar i bruk naturlige arbeidsmedium for å dekke alle termiske behov, vil en kunne redusere kostnader på en miljøvennlig måte. Denne oppgaven har som formål å undersøke det energimessige potensialet ved et kombinert kjøle- og varmepumpeanlegg for et hotell i Trondheim. Optimaliserende elementer som energilagring og varmegjenvinning har blitt inkludert i systemet.

Bygningssimuleringsverktøyet *Simien* ble brukt til å konstruerer en modell av hotellet for å estimere årlige behov for romkjøling og oppvarming. Grunnet begrensingen i programvaren ble tidsavhengige laster for varmtvannsforbruk, kjøle- og frysebehov beregnet og inkludert i modellen. Energisystemet ble modellert og simulert i *Dymola*, for å avdekke både ytelsen og potensialet til CO_2 og propan varmepumpene.

Resultatet fra simuleringene viste at energisystemet var i stand til å dekke alle termiske behov for de simulerte scenarioene. Gjennomsnittlig årlig energieffektivitet for CO_2 system ble beregnet til å være 5.55. Ved å implementere termisk lagring for romkjøling og oppvarming, minket lasten betydelig i forhold til faktisk etterspørsel. Konsekvensene av dette er at topplasten ble redusert og utsatt med et par timer hver dag. Størrelsen på de termiske lagringstankene viste seg å påvirke antall stopp av propan varme- og kjøleanlegget. Den sjenerøse lagringskapasiteten tillatte færre stopp og dermed en høy systemeffektivitet. Gjennomsnittlig årlig energiytelse for propananlegget ble gjennom simuleringene funnet til å være 5.64.

Energisystemet viste seg å være mer effektivt sammenlignet med tradisjonelle kjøle-og varmepumpeanlegg for hotell. Beregninger demonstrerte at store reduksjoner i operasjonelle utgifter kan oppnås. Nåværende systemdesign kan forbedres ved å redusere mengden varme som blir forkastet til omgivelsene. I tillegg bør det nåværende mellomtrykket i CO_2-anlegget økes for å oppnå bedre systemytelse.

Nomenclature

List of abbreviations

AC	Air Conditioning
COP	Coefficient of Performance
DHW	Domestic Hot Water
E.E.	Energy Efficiency
GW	Grey Water
H.E.	Heat Efficiency
HFC	Hydrofluorocarbon
HX	Heat Exchanger
IW	Ice Water
LP	Low Pressure
LPHX	Low Pressure Heat Exchanger
MIMO	Multiple Input, Multiple Output
MP	Mid Pressure
PCM	Phase Change Material
PED	Pressure Equipment Directive
SH	Space Heating
SIMO	Single Input, Multiple Output
SISO	Single Input, Single Output
SPF	Seasonal Performance Factor
TEK	Norwegian building regulations
TEV	Thermostatic Expansion Valve
U-value	Overall heat transfer coefficient
VSD	Variable Speed Drive

List of Symbols

A	Area	[m^2]
$C_{\text{freez-rooms}}$	Chiller load in freezer rooms	[W]
$C_{\text{ice-water}}$	Chiller load CO_2-ice water interface	[W]
$C_{\text{ref-rooms}}$	Chiller load in refrigerated rooms	[W]
C_{space}	Chiller load for propane-ice water interface	[W]
c_p	Specific heat capacity	[kJ/kgK]
E	Energy	[kWh]
h	Entropy	[J/kg]
H_{GCs}	Heating load CO_2 gas coolers	[W]
$H_{\text{LP-SH}}$	Heating load CO_2 low pressure-space heating heat exchanger	[W]
H_{space}	Heating load propane-space heating interface	[W]
$\dot{m}_R$	Mass flow of refrigerants	[kg/s]
n_i	Number of air shifts	[/h]
$\dot{Q}_0$	Heat of evaporation	[W]
$\dot{Q}_c$	Heat of condensation	[W]
Q_{gain}	Heat gain	[W]
Q_{inf}	Infiltration heat loss	[W]
Q_{net}	Net heat load	[W]
Q_{tran}	Transmission heat loss	[W]
Q_{ven}	Ventilation heat loss	[W]
R_{con}	Conductive resistance	[K/W]
R_{cov}	Convective resistance	[K/W]
R_{tot}	Total thermal resistance	[K/W]
T_{amb}	Ambient temperature	[K]
T_f	Temperature of fluid	[K]
T_i	Inner temperature	[K]

Δ T	Temperature difference	[K]
Δ T_{lm}	Log mean temperature difference	[K]
U	Overall heat transfer coefficient	[W/m^2K]
V	Volume	[m^3]
$\dot{V}$	Volume flow rate	[m^3/h]
$\dot{W}$	Compression work	[W]
$\dot{W}_{is}$	Isentropic compression work	[W]
W_{LPcomp}	CO_2 low pressure compression work	[W]
W_{MPcomp}	CO_2 mid pressure compression work	[W]
$W_{\text{Parallelcomp}}$	CO_2 parallel compression work	[W]
$W_{\text{Propane-comp}}$	Propane compression work	[W]

Greek letters

η	Efficiency	[-]

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Chapter 1

Introduction

Britannia Hotel was built in 1897 and is a five star hotel located in the city center of Trondheim. The building contains 247 hotel rooms, 4 restaurants, a conference room for 450 people and a spa treatment center. The combined demand for domestic hot water, space heating and cooling is thus substantial. The hotel was closed in June 2016 for a full-scale renovation that was estimated to last until late 2018, with a cost over a billion NOK. With the implementation of a heat pump based system to cover the heating and cooling demands, the potential energy savings are tremendous.

The EU F-gas regulation aims for a major phase out of hydrofluorocarbons (HFCs) by the year 2030, which grants natural working fluids a strong position in regards to application in new systems. CO_2 and propane are both natural working fluids widely employed in both heat pump and chiller units. Due to its heats rejection at gliding temperatures in the supercritical region, CO_2 is exceptional for domestic hot water heating. Propane, on the other hand, is suitable for both low and medium temperature applications. If combined in an integrated system, excessive heating and cooling loads can be efficiently covered.

1.1 Objectives

The objective of this report is to design an energy system for the hotel and investigate the thermal potential through simulations. A combined heat pump and chiller unit with heat recovery from waste energy is purposed to maximize the use of available sources. Energy storage units have been implemented in order to evaluate possible peak reduction and load shifting.

1.2 Thesis structure

Chapter 2: This chapter provides a small overview of the current energy consumption in Norway. Heat losses and gains in buildings are elaborated.

Chapter 3: This chapter includes an overview of heating and cooling systems, where auxiliary heating and storing devices are investigated. Optimizing measure with regards to system control and recovery from available sources are considered.

Chapter 4: Includes a description of the heat pump design and operation. Subsystems for domestic hot water, space heating and cooling with their distribution networks are outlined in this chapter.

Chapter 5: The simulated model and assumptions are explained in this chapter. Operational restrictions, control of the system and the different simulation scenarios are defined.

Chapter 6: The simulation results are described and discussed. A parameter study together with an economical evaluation are also presented.

Chapter 7: Discussions regarding the restrictions of the simulation model. Strengths and weaknesses of the purposed system are assessed in this chapter.

Chapter 8: Includes suggestions for further work on the simulation model and future improvements of the energy system.

Chapter 2

Thermal loads in buildings

This chapter is a brief introduction to energy management in buildings, presenting current practice and the major contributions to thermal energy consumption in households. The dominant mechanisms of heat transfer in buildings are presented, and their influence is discussed.

2.1 Energy management

The term *energy management* covers a wide range of strategic and operational measures related to building systems. The specter includes everything from architectural considerations and technical installations, to energy consumer patterns. Conscious management of energy utilization will primarily provide tradeoffs for both the consumer and the environment. Reduced operational costs and increased indoor comfort are traditionally the main motivations for implementing energy saving measures. In later years, however, the increased focus on environmentally friendly concepts has been an incentive for consumers to embrace futuristic solutions.

In contrast to most other European countries, Norway is mainly supplied with energy from hydropower production. This source is renewable and relatively cheap, which in turn has resulted in generally low electricity prices that have made electrical heating preferable. A solution with an electrical heating element in each room, in addition to a wood fired oven, is considered the typical Norwegian practice. Figure 2.1 illustrates the electricity consumption for an ordinary Norwegian household.

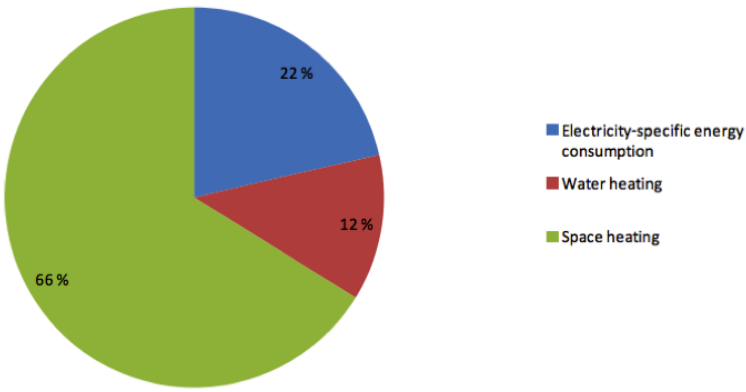


Figure 2.1: *End use energy consumption in households (Bergesen et al., 2013).*

This practice has resulted in high electricity consumption whereof 66% of the total energy use is related to space heating. If one includes domestic hot water, this amounts to a staggering 78%. On a national level, buildings are responsible for about 40% of the primary energy use, and thus, the related greenhouse emissions (Bergesen et al., 2013). Hence, the potential to reduce both consumption and emission are tremendous if the right design, strategy and technology are implemented.

2.2 Heat transfer in buildings

In order to determine the thermal demand of a building, it is necessary to account for all major contributions in the energy balance. For high-rise commercial buildings, heat gains from equipment, lighting and people play an imperative role in the energy balance. The function of the building is therefore essential when investigating the thermal demand. In hotels, for instance, the dominant contributor to the demand is domestic hot water (Krarti, 2011). The net heat demand of a building is expressed in equation 1.1 and can be calculated as the sum of losses subtracted by the heat generation (Eikevik, 2016).

$$Q_{net} = Q_{tran} + Q_{inf} + Q_{ven} - Q_{gain} \tag{1.1}$$

Energy demand can vary considerably during the year. Large variations are due to thermal factors like heating, ventilation and refrigeration. Naturally, the demand for heating is greatest during the winter, and the need for refrigeration is largest during the summer. The contribution from other electrical equipment is generally balanced throughout the year, but will differ a great deal with regards to the specific type of building and its function.

Heat transfer by transmission, $\mathbf{Q_{tran}}$

The dominant mechanisms of heat transfer in buildings are conduction and convection through walls and roofs. Fourier´s law states that the driving force of conduction is the temperature difference between the inner and outer climate, as heat will spontaneously flow from a warmer to a colder body. One-dimensional conduction is considered adequate to describe the heat transfer unless there are significant thermal bridges. The total thermal resistant R-value, shown in equation 2.1, is a function of both the conductive and convective resistances (Frank et al., 2007).

$$R_{total} = \sum R_{cond}, R_{conv} = \frac{1}{UA} \tag{2.1}$$

The transmission heat transfer can thus be characterized by the overall heat transfer coefficient, as illustrated in equation 2.2

$$Q_{tran} = UA\Delta T = \frac{\Delta T}{R_{total}} \tag{2.2}$$

where ΔT represents the difference between the indoor and outdoor temperature.

Heat transfer by infiltration, Q_{inf}

Unlike ventilation, which is air moved by mechanical systems, infiltration is the uncontrolled flow of air through leaks in the bulding structure. Generally, air infiltration takes place in all types of buildings, but is more relevant for smaller buildings as the leakage-area-to-indoor-volume ratio is much higher (Krarti, 2011).

$$Q_{inf} = c_p n_i V \Delta T \tag{2.3}$$

The infiltration heat transfer is expressed by the temperature difference, ΔT, the specific heat capacity of air, c_p, the number of air shifts due to infiltration, n_i, and the air volume in the building, V (Eikevik, 2016).

Heat transfer by ventilation, Q_{ven}

The purpose of ventilation is to maintain a constant airflow in the building and thereby insure human comfort and health. For large buildings it is most common to use a mechanical ventilation system that can supply large air volumes. Due to the temperature difference, it is necessary that a recovery system heats input air before it is released in the room. The regenerative heat exchanger transfers heat through a surface that is alternately in contact with supply and exhaust air. This system is typically a rotary wheel where one half of the wheel is in contact with each stream. Both heat and humidity are released to the supply air as the surfaces of the heat exchanger continuously turn. This heat exchanger has a generally high efficiency of 80-90%, and is thus commonly used for larger buildings when there is no requirements regarding particle transfer (Energiforskning, 2007).

Heat loss from the ventilation system represents the heat that is not being transferred in the ventilation heat exchanger. The magnitude of the exchanger efficiency, η, therefore plays a central role in calculations, which can be expressed through equation 2.4 (Eikevik, 2016).

$$Q_{ven} = c_p \dot{V}(1 - \eta)\Delta T \tag{2.4}$$

The parameter $\dot{V}$ represents the airflow through the ventilation system.

Heat gain in buildings, Q_{gain}

Casual heat sources in buildings are usually a byproduct from electrical equipment like lighting and home appliances. For residential buildings, these loads are generally not noticed, as there are few major contributors and a limitation in people giving off heat. Commercial buildings, on the other hand, often have difficulties with removing the heat accumulated.

It is a complex matter to precisely estimate the casual heat gain, and therefore important to pinpoint the major and continuous sources. In a commercial kitchen, for instance, loads can be readily estimated and will not diverge considerably during operational hours. A hair dryer, on the other hand, will accumulate a relatively large load, but has a small operational time that will deviate greatly. It is therefore prudent to neglect unpredictable sources from the heat balance (Shemmeri, 2011).

Chapter 3

Heating and cooling systems

Norway is the number one country in the world with regards to electrical heating. Wall-mounted panels and radiators are the most common electrical heating equipment, and accounts for more than 80% of heating devices (Energiforskning, 2007). To be able to meet the increasing energy demand, new and better heating alternatives must be implemented. Heat pump systems are such a solution, as they can provide a relatively high seasonal performance factor (SPF) and, in combination with innovative distribution and storage systems, cover the entire thermal load. This chapter presents theory on the concept of heat pumping cycles, with a focus on utilizing CO_2 as a refrigerant. The advantages with incorporating thermal storage devices are emphasized through a literature review of possible solutions and studies within the field. Optimizing measures in relation to both heat recovery and the control of the system are also explained in this chapter.

3.1 Heat pumping systems

Heat pumps move thermal energy from a source at low temperature, and delivers it as heat to a sink at higher temperatures. As stated in the second law of thermodynamics, this can only be accomplished by the use of external power. For heat pump cycles where heat is absorbed and delivered at essentially constant pressure and temperature, the *reversed Carnot Cycle* is used to describe the process (Stene, 2009). Figure 3.1 illustrates a subcritical heat pump cycle, which is representative for several refrigerants, such as propane.

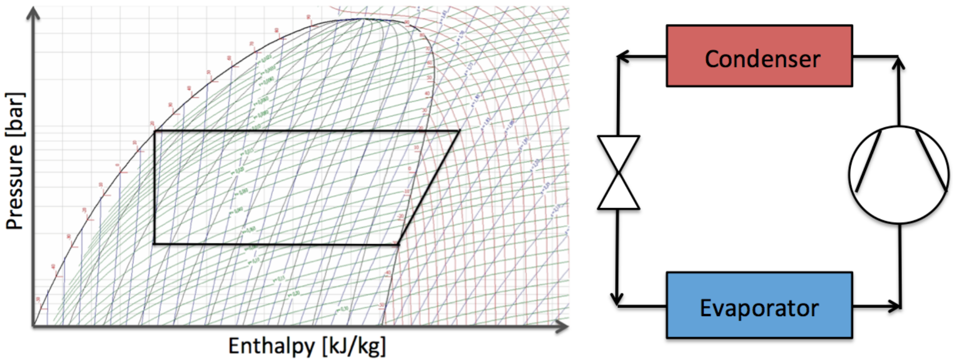


Figure 3.1: *Principle heat pump sketch with corresponding logarithmic pressure-enthalpy diagram.*

The working fluid enters the compressor in vapor phase and is isentropically compressed to the high-pressure side. The refrigerant is in the superheated phase when exiting the compressor. Thenceforth, the working fluid enters the condenser, and the heat transported from the low-pressure side is released. The fluid starts to condense at constant pressure as heat is released from the refrigerant. When all the gas is condensed, the valve expands the liquid from high to low pressure in an isenthalpic process. This brings the refrigerant in to the liquid-vapor region, where the gas quality is decided by the pressure difference over the expansion valve. The pressure in the evaporator is regulated to a fluid temperature below that of the heat source. Heat will then be transferred to the fluid that will start to evaporate at constant pressure. The low pressure in the evaporator is maintained by the compressor, which continuously removes fluid that has reached the vapor phase.

The evaporation process can be described by the heat collected, $\dot{Q}_0$, which is a function of the refrigeration mass flow and the enthalpy of evaporation.

$$\dot{Q}_0 = \dot{m}_R(h_1 - h_4) \tag{3.1}$$

The real work performed by the compressor is described by the theoretical work, $\dot{W}_{is}$, and the isentropic efficiency of the compressor.

$$\dot{W} = \frac{\dot{m}_R(h_{2s} - h_1)}{\eta_{is}} = \frac{\dot{W}_{is}}{\eta_{is}} \tag{3.2}$$

The heat rejected by the condenser, $\dot{Q}_c$, is thus a function of the heat collected, $\dot{Q}_0$, and the compression work, $\dot{W}$.

$$\dot{Q}_c = \dot{m}_R(h_2 - h_3) = \dot{Q}_0 + \dot{W} \tag{3.3}$$

The expansion process takes place at constant enthalpy, in that $h_3 = h_4$.

For systems with several stages of evaporation it is necessary to estimate the heat that can be recovered at different pressures, e.g. $\dot{Q}_{01}$, $\dot{Q}_{02}$, ..., $\dot{Q}_{0n}$. Thenceforth, the mass flow for each stage is calculated based on the corresponding enthalpy difference, $\dot{m}_1 = \frac{\dot{Q}_{01}}{\Delta h_{01}}$. The amount of condensing heat will then be determined by the total mass flow in the system and the magnitude of the enthalpy range.

3.2 CO_2 heat pump systems

CO_2 differs from other working fluids, as it has a low critical temperature and a high critical pressure of respectively 31.1 ^oC and 73.8 bar. Practically, the condensation temperature limit is in the range of 28 ^oC. For heat rejection above this temperature, the heat pump will operate in the transcritical region. In the supercritical region, the gas temperature is dependent on gas cooler pressure, where the heat is rejected at gliding temperatures. CO_2 is therefore suitable when heating domestic hot water, as it offers a good adaptation to the relatively wide temperature span. In order to achieve a high coefficient of performance (COP) for the transcritical cycle, it is imperative that useful heat is rejected over a broad temperature range, resulting in a large enthalpy difference in the gas cooler. Hence, it is essential with sufficient cool down to achieve a relatively low temperature before throttling to reduce losses. CO_2 operations in warmer climates, where there are scarce access to cold sources, thus present a challenge, as illustrated in figure 3.2

(Stene, 2009).

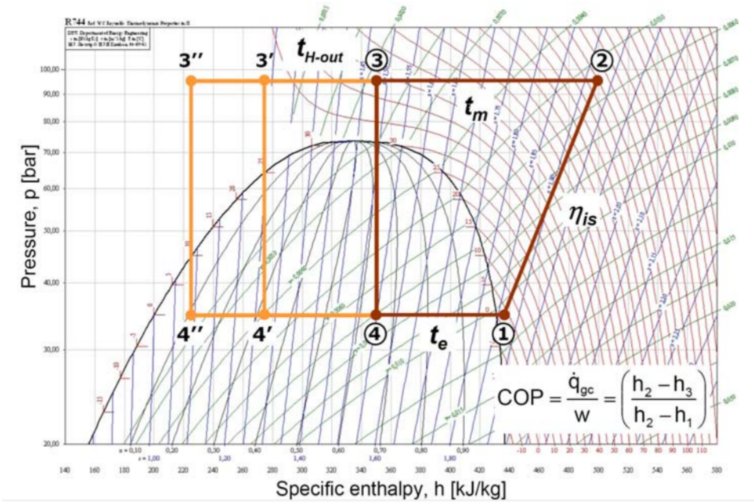


Figure 3.2: *CO_2 cycle in logarithmic pressure-enthalpy diagram at different gas cooler outlet temperatures (Stene, 2009).*

A major advantage with the transcritical cycle is that one is able to include different heating functions according to their appropriate temperature level. A typical example of this is a tripartite gas cooler with heat exchangers in series, where the third and first heat exchangers are used for preheat and reheat of domestic hot water, respectively. The mid gas cooler rejects heat to a space heating circuit, which typically operates with a small temperature difference. Figure 3.3 illustrates such a configuration.

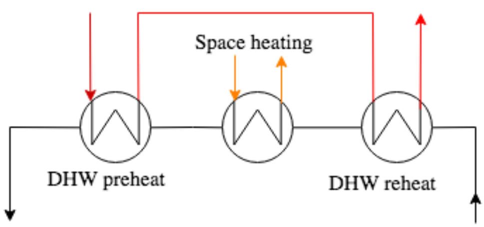


Figure 3.3: *CO_2 gas cooler including interface for space heating, preheat and reheat of domestic hot water.*

Theoretically, one can implement as many functions as desired, with the only limitation being the temperature fit in the gas cooler and the amount of available heat. Nonetheless, the available heat is highly restricted by the temperature glide at the particular level. This presents great difficulties when dealing with large heating loads and small temperature differences. The magnitude of heat provided for a specific function influences the others, e.g. space heat and domestic hot water. A proportional heating demand is therefore optimum, but not the case in most circumstances. Hence, it is imperative to thoroughly investigate combinations of potential loads if planning to utilize the CO_2 cycle to cover multiple heating functions.

3.3 Integrated systems for space heating and hot water heating

Heat can be distributed to domestic hot water and space heating circuits separately, as shown in figure 3.3, or supplied to an integrated system that combines both functions. Several configurations are possible, as illustrated in figure 3.4. However, they all share the common factor that the domestic hot water tank transfers heat to the space heating circuit.

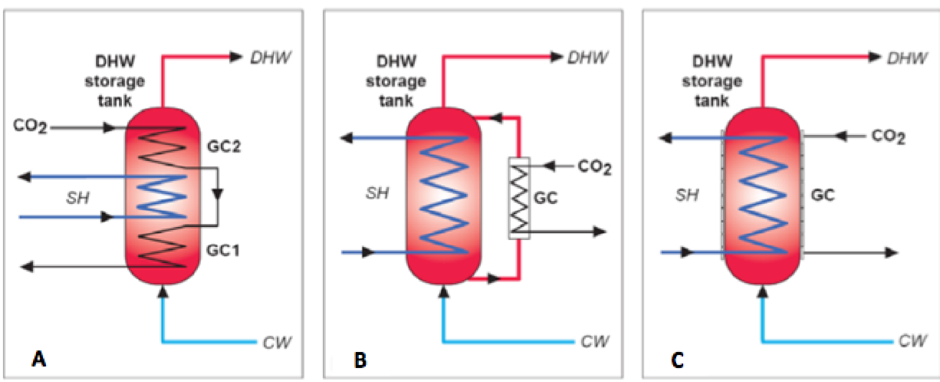


Figure 3.4: *Concepts for gas cooler interface to integrated heating systems (Stene, 2009).*

Figure 3.3A shows a dual interface to the gas cooler by means of preheat and reheat of the domestic hot water, located in the lower and upper part of the tank, respectively. The tube coil interface to space heating is located in the middle of the tank, due to the appropriate temperature level. When operating in domestic hot water mode, the middle heat exchanger is bypassed from the space heating side. This will increase the temperature in the tank, as no heat is lost to the space heating circuit. Consequently, the outlet temperature of the gas cooler will increase, resulting in a reduced COP (Stene, 2009).

Figure 3.3B illustrates how the gas cooler is connected to an external thermosyphon loop, where the water flow rate is induced by the temperature and density difference across the tank. For 3.3C, the gas cooler consists of multiport extruded micro-channels that are mounted around the outside of the tank. Both figures 3.3B and 3.3C have a coil tube interface to the space heating circuit. Similarly to figure 3.3A, these can be bypassed with the same results. All concepts presented will have a lower COP than the tripartite gas cooler due to the secondary heat transfer from domestic hot water to space heating. This could, however, be a viable solution when dealing with difficulties regarding heat rejection from the CO_2 gas cooler to several circuits (sec 3.2) (Stene, 2009).

3.4 Auxiliary heating and storage devices

In order to ensure full heating coverage, auxiliary systems providing backup heating must be in place. Electrical backup coverage, such as heating panels and boilers, can make the consumer sensitive to fluctuations in the electricity prices. Therefore, alternative and environmentally friendly backup heating solutions will be investigated.

3.4.1 District heating

Moving thermal heat efficiently within a region is the main goal of district heating and cooling. By connecting consumers to available hot and cold sources, one is able to meet demands with a relatively low use of resources. The heat load can either originate from renewable natural resources, like geothermal heat extraction, or from a source where the resource otherwise would be lost.

The hot and cold loads are moved across distribution pipelines containing a suitable working fluid, usually pressurized water. In analogue with electrical power, where voltage is lowered before reaching the user, the temperature and pressure is reduced in district heating. Essential elements of the substations are heat exchangers, valves and mixing systems, conducting the reduction of temperature and pressure. The reasoning behind the latter is to eliminate operational risk and malfunction on the consumer's side. In addition, less expensive equipment can be utilized in the domestic system as the pressure and temperature is decreased (Frederiksen, 2013).

For hotels, it is beneficial with a custom-made system with flexibility in regards to layout, equipment and accessibility for maintenance. An indirect interface to district heating is favorable, as it becomes less critical whether the network fluid contains corrosive substances, like dissolved oxygen, which can cause severe equipment damage. Also, the indirect system is shielded against pressure surges that may occur in the district heating network (Frederiksen, 2013).

There are many possible customer-side configurations, but the one thing they have in common is the category of equipment being used. They all need some types of heat exchangers, valves and control equipment.

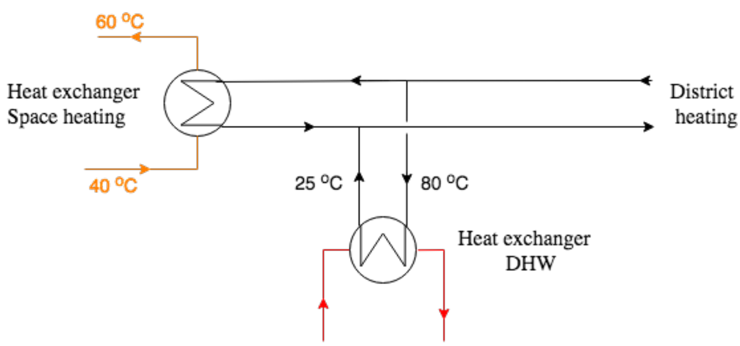


Figure 3.5: *Layout of interface for district heating at Scandic Hotel Bakkelandet (Stene, 2016).*

Today, the most common heat exchanger installed in substations in northern Europe is the plate heat exchanger, mainly being the brazed plate heat exchanger. This is due to the fact that they are very reliable, compact and cheap (Serth, 2014).

Scandic Hotel Bakkelandet is an example of a high performance building that utilizes district heating to cover peak loads. The installed heat pumping system covers 80% of the total heat demand, both in regards to domestic hot water and space heating. The system interface for district heating consists of two plate heat exchangers in parallel, as illustrated in figure 3.5. This solution is quite flexible as it enables separate heat distribution to domestic hot water and space heating. The minimum return temperature to the district heating network is 25 ^{o}C, and thus a large portion of the available heat has been utilized (Stene, 2016).

3.4.2 Thermal energy storage tanks

If there is access to a source that remains more or less constant during the year, a considerable share of both heating and cooling demands can be covered. This is achieved by rejecting heat when in surplus, and extracting when in need. When no natural source is present, water tanks are an economical and simple energy storage concept. Sensible heat is utilized as the tanks are charged and discharged by changing the temperature of the liquid. The amount of stored heat depends on the specific heat of the fluid, the change in temperature and the amount of mass available, as defined in equation 3.4 (Stritih et al., 2015).

$$Q = \int_{T_i}^{T_f} mc_p dT = mc_p(T_f - T_i) \tag{3.4}$$

Various storing medias can be used in such systems, but water appears to be the most suitable liquid. This is due to its availability, high specific heat and the fact that water can be used directly, e.g. domestic hot water. With thermal energy storage one can achieve peak shaving and increase the overall efficiency. Storage tanks are widely used in heat pump based heating systems, and can reduce the number of compressor starts/stops, as the capacity of the heat generated is adapted to the demand (Floss and Hofmann, 2015). *Rema 1000 Koppanmarka* is an example of a heat pump system that utilizes such a solution. With its storage tanks of 2700 liters, the floor heating loop is supplied, when demand is high, through heat exchangers in the domestic hot water tanks (Jorschick, 2014).

3.4.3 Phase changing materials

It is desirable to maximize the storing potential of the thermal energy tanks, which can be achieved by utilizing phase changing materials (PCMs). The materials change phase, e.g. from solid-to-liquid or liquid-to-gas, dependent on the temperature. Energy is thus absorbed and released by the mechanism of latent heat. Most applications in storage tanks use the solid-to-liquid phase transformation due to the vast increase in volume when entering the gas phase. The phase change takes place at an almost constant temperature, resulting in a stable thermal environment around the elements. The implementation of PCMs enables the possibility of peak shaving by shifting load demands to a later time (Fleischer, 2015).

A study conducted by Nkwetta et al. (2014) concludes that a PCM consisting of 90% sodium acetate and 10% graphite can greatly increase the energy storing potential of domestic hot water tanks. The material has a melting point of 58 ^{o}C and a latent heat of fusion of 180-200 kJ/K. The amount of stored energy is proportional with the amount of PCM placed in the tank. By implementing a PCM of 39.6 kg to a 120 liters storage tank, the maximum storing capacity was found to be 60,946 kJ. Considering the energy is distributed during the time span of an hour, this amounts to 16.93 kWh.

Another study was conducted by Cabeza et al. (2006) on the effects of PCMs in stratified hot water tanks. The PCM was placed in the top of a 146 L tank and had a melting point of 58 ^{o}C. During cooldown, it was found that the energy density was considerably increased (40-66%) when having a 1 K temperature difference. This resulted in an extra 35 min of hot water due to the heating from the PCM elements, as can be observed in figure 3.6.

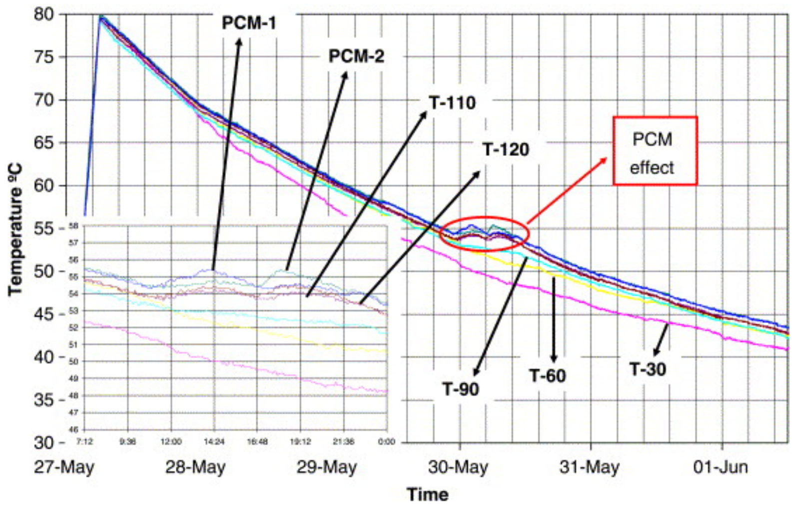


Figure 3.6: *Cool down process of the tank with two PCM modules, where T-x illustrates the stratification layer x cm from the tank bottom (Cabeza et al., 2006).*

The newly constructed building for Bergen University College is equipped with 4x60 m^3 cold storage tanks, making it the largest installation of its sort in Europe. The storage tanks are equipped with 47,000 PCM elements, consisting of salt hydrates (transition temperature of 10 ^oC). The PCMs are able to store 11,200 kWh. In comparison, 180 energy wells, a compressor of 700 kW and a cooling capacity of 3000 kW would have to be installed to provide equal cooling (Stavset and Kauko, 2015). The producers of the cooling tanks for Bergen University College, *PCM Products*, offer PCM implementation for several different temperature levels, as illustrated in table 3.1.

Table 3.1: *PCM products and storing potential*

Type	Transition temperature [C^o]	Capacity [kWh/m^3]
TubeICE S34	34	50
FlatICE S32	32	57
TubeICE S8	8	45
FlatICE S8	8	44

PCMs can also be implementend in building construction such as wallboards, concrete and insulation. A review conducted by Baetens et al. (2010) describes how

PCM enchanced wallboards are easily installed and can provide thermal storage thoughtout the building. Numerical simulations of coated wallboards[1] have been done for a conventional house situated in Helsinki of 120 m^2. The results revealed that the annual auxiliary heating was reduced with 2 GJ or about 6% (Peippo et al., 1991). Another possibility for applying PCMs in building structure is PCM enchanced concrete. Test cubicles with such materials were installed in buildings in Spain[2] for cooling purposes, which resulted in a reduced indoor temperature and a peak temperature delay of about 2 hours (Castellón et al., 2006). The implementation of PCMs in both thermal storage tanks and building construction could potentially reduce peak demand and thus the overall energy consumption of the building.

3.5 Optimizing measures

There are several other measures, in addition to thermal storage, that could optimize the system. Using accessible heat, such as grey water, would provide the heat pump with a relatively high temperature source. However, without optimized control, the efficiency can decrease. Control strategies for heat pump systems and models for energy behavior are therefore reviewed.

3.5.1 Grey water heat recovery

Optimizing measure related to thermal energy production involves covering demand by utilizing available and easily accessible sources. For a combined heat pump and chiller unit, such functions are already provided, as the different parts of the system act as both sinks and sources. For buildings with large domestic hot water demand, grey water is a potential heat source that will remain fairly constant throughout the year.

Grey water is defined as drainage water from showers, bathtubs, sinks, dishwashers and washing machines, which usually has a generally high temperature when

[1]Transition temperature 21.5^oC and ΔH= 540kJ/m^2.
[2]Transition temperature 26^oC and ΔH= 110kJ/m^2.

entering the sewage system. The potential energy to be recovered from the fluid is dependent on the supply temperature of the grey water and the temperature approach in the heat exchanger, expressed by equation 3.5 (Frank et al., 2007)

$$Q = UA\Delta T_{lm} \tag{3.5}$$

where ΔT_{lm} is the logarithmic mean temperature difference.

Shell-and-tube heat exchangers are most commonly implemented in grey water heat recovery processes (Kleven, 2012). These are robust and can be equipped with a "self-cleaning system", which is unavoidable when dealing with water containing impurities. According to Assoc. Prof. Jørn Stene (NTNU), it is necessary for larger facilities to have several heat exchangers in parallel for redundancy during cleaning periods. The amount of impurities in the grey water, especially grease, is a central parameter with regards to design and operation of the heat exchanger. Limiting the grade of contamination in the water increases the operational time of the grey water recovery system. It can therefore be practical to evaluate which drainage sources to use. When employing grey water heat recovery in a hotel, the drainage manifold system should be kept separate from restaurants, as they will accumulate large amounts of grease.

Ni et al. (2012) conducted a study of a multiple-function heat pump for domestic hot water, space heating and cooling, which utilized grey water as an energy source. The recovery concept, illustrated in figure 3.7, is a storage tank with coils for closed circulation to the evaporator.

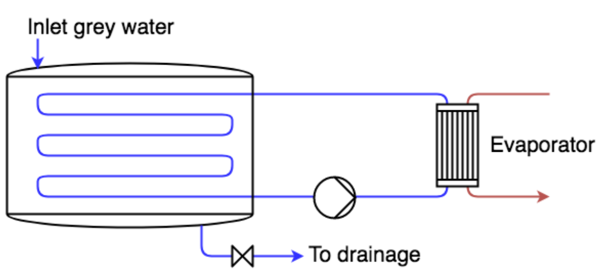


Figure 3.7: *Grey water heat recovery with secondary fluid circulation to evaporator.*

This solution is beneficial as the water impurities are limited to the grey water tank. There will, however, be an additional heat transfer stage, because the water in the coils act as a liaison between the grey water and the evaporating fluid. For a typical residential house with four family members and three bedrooms in New York, the simulation results show that the energy consumption decreased with 76.0%, 23.5% and 2.7% for hot water heating, space heating and cooling, respectively. The statistical results revealed that the total energy savings were increased by 33.9%, compared to a conventional system.

3.5.2 Control strategies

Control strategies are an important factor in the optimization of energy systems. Naturally, this concerns the operation of heating and cooling systems, the subsystems, ventilation and more. Selecting the optimum control structure for the energy system and tuning of the required algorithms can be a complex matter, as the relationship between the various functional components (valves, compressors, pumps, fans) and the measured outputs (temperature, pressure, heating/cooling load) are fully dependent. Moreover, the dynamics for such a system are distinctly nonlinear, and the responds varies according to the operational conditions.

Wang et al. (2017) investigated energy-optimum control strategies for heat pump systems. Three methods were compared in order to reveal the best tracking performance and control of the heating capacity, cooling capacity and superheat. The strategies reviewed were single-input single-output (SISO), single-input multiple-output (SIMO) and multiple-input multiple-output (MIMO) control systems. The results revealed that all methods had good tracking performance. However, MIMO provided better control of the pressure and superheat. Nevertheless, the different control strategies have their strengths and weaknesses, which must be considered before deciding on a control structure. SISO has a low complexity, and thus is less accurate, but sufficient for simple heat pump cycles. MIMO, on the other hand, provides better control during dynamic operations, but is highly complex.

It is, however, a complicated matter to accurately control the various systems and account for how the different regulations will influence the total energy behavior of the building. Zhao and Magoulès (2012) conducted a review of prediction models for building energy systems, including relatively simple engineering models to highly complex artificial intelligence models. The predictions can be performed by carefully analyzing each influencing factor, or by approximating the usage by considering several major factors.

The engineering methods are most widely used to calculate energy performance. The energy behavior of the building is based on input values and step-by-step calculations of physical principles to estimate the thermal dynamics. These methods are accurate in practice, but require extremely detailed input values regarding everything from building structure and environmental data to occupant behavior. The accuracy of the engineering models are limited, as detailed input data can be difficult to obtain, and in some cases behave highly dynamic.

In contrast to the engineering methods, *the artificial intelligent models* predict the energy behavior of the building based on historical data regarding everything from heating/cooling loads, to sub-level component operation and optimization. Hence, it becomes possible to estimate particular variables during different operational conditions. A practical example is that it could be possible to estimate the water consumption for hotel guests based on, e.g., the time of the year, room temperature, age and gender.

Chapter 4

System design and operation

The energy system, consisting of a CO_2 and propane heat pump working in parallel, will primarily provide domestic hot water, space heating and cooling through indirect subsystems. The heat pumping system concept and distribution subsystems for thermal energy are explained in this chapter. The main motivation behind the system design is to utilize excess heat, that would otherwise be disregarded, for useful purposes. By recovering heat from freezers, refrigerated rooms and space cooling one is able to simultaneously provide multiple services. In regards to the dimensioning of the system, it is first necessary to investigate the contributions from these sources, as well as heating and cooling demands for the hotel.

4.1 Heating and cooling demand

A model of Britannia Hotel had to be constructed in order to estimate the annual demand for domestic hot water, space heating and cooling. The engineering method based (ref. 3.5.2) simulation software *Simien* was used to create a model of the building body. Due to insufficient data regarding the hotel, the construction is assumed to meet minimum standards in accordance to the *Norwegian Building Regulations* (TEK10), §14-3. Central building and operational properties are listed in table 4.1, whereas additional input data can be found in appendix A.4.

City plan drawings from Trondheim County, together with data from Britannia's website, were used to set the magnitude of the construction to approximately 8400 m^2. The building body is modeled as a single temperature zone with 24-hour operation of the heating and cooling systems. Climate data for Trondheim was used in the simulation.

Table 4.1: *Central input values and energy requirements.*

Description	Value	Requirement
Combined glass, window and door area divided by floor area [%]	4.8	20.0
U-value exterior walls [W/m^{2}K]	0.17	0.18
U-value roof [W/m^{2}K]	0.13	0.13
U-value floor to ambient [W/m^{2}K]	0.15	0.15
U-value glass/windows/doors [W/m^{2}K]	1.20	1.20
Normalized thermal bridge value [W/m^{2}K]	0.06	0.06
Leakage value (air density at ΔP= 50 Pa) [Air change pr. h]	1.50	1.50
Average heat recovery in ventilation system [%]	80	80
Specific fan power (SFP) [kW/m^2/s]	2.00	2.00

Due to limitations in Simien regarding transient operations, additional demand curves for conference rooms, refrigerated rooms and freezers had to be constructed. This was achieved by using the refrigeration system modeler *CoolPack*, to which input data are listed in appendix A.1. As infiltration losses due to high activity will be sporadic, it is assumed that the load from refrigeration rooms and freezers are constant. The cooling demand for refrigeration rooms and freezers were found to be 5 kW and 10 kW, respectively.

The domestic hot water consumption for the hotel has been estimated based on *Standard Norway* SN/TS 3031:2016, attachment A. Cooling curves for the conference rooms have been assessed based on maximum capacity of 450 guests and a typical user pattern. After discussions with the Prof. Armin Hafner, a 24-hour cooling demand pattern for the kitchens was assumed based on the number of restaurants and the average heat gain from food service facilities (American Society of Heating and Engineers, 2001). Calculations and assumptions for the constructed load curves mentioned above can be found in appendix A.2 and A.3. Demand curves for domestic hot water, conference rooms and kitchen cooling are illustrated in figures 4.1 and 4.2. To simplify the energy flow tool, a constant operational load per hour is assumed.

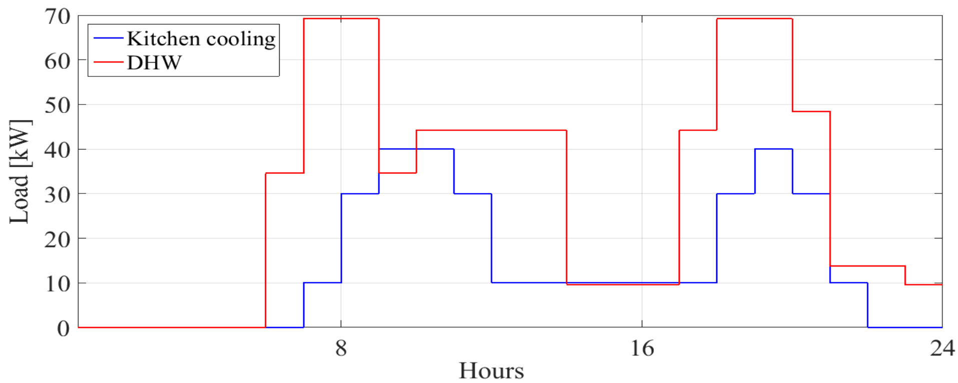


Figure 4.1: *Energy demand pattern for domestic hot water and kitchen cooling.*

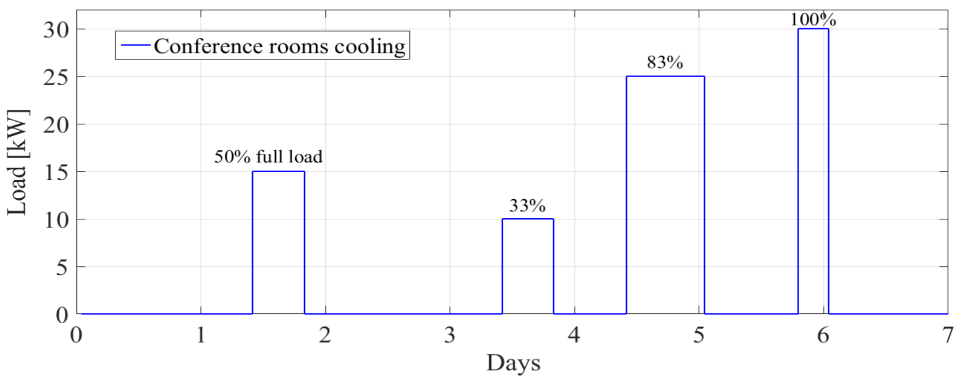


Figure 4.2: *Energy demand pattern for cooling of conference rooms.*

The annual domestic hot water, space heating and cooling energy demands are represented in accumulated order in figure 4.3. The daily average domestic hot water demand is considered constant throughout the year.

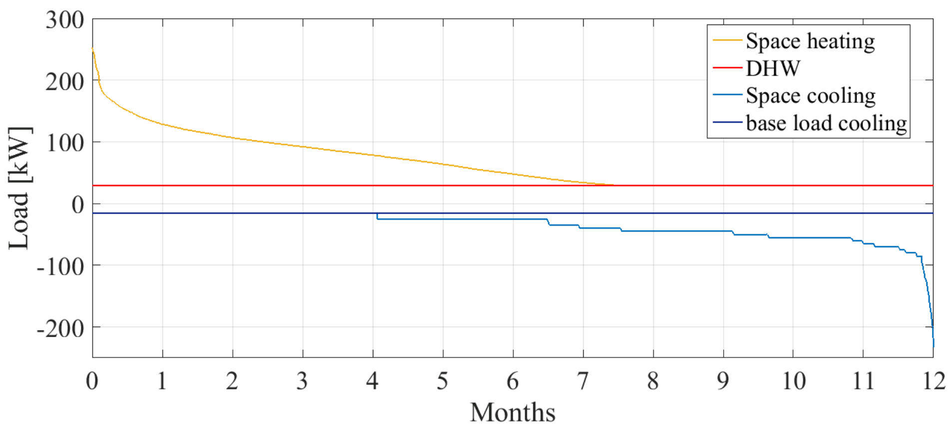


Figure 4.3: *Load-duration curve for Britannia Hotel.*

The heating demand is naturally largest during the winter season and reaches a peak of 252 kW, where 223 kW are accredited to space heating. The space cooling load includes all functional cooling, and is thus fairly stable at a low level, with the exception of peak loads during the warmest days of the year. Dimensioning of the system for heating, including both the CO_2 and propane heat pumps, should be determined based on the investment cost and possible energy savings. The system should be able to cover peak cooling demands, while peak heating loads can alternatively be handled by district heating. However, in order to investigate the full potential of the energy system, full heating coverage will be integrated.

4.2 Heat pump system

The combined system for the hotel consists of two separate cycles working in parallel, as presented in figure 4.5. The main concept of the system is to provide flexibility according to the demands for domestic hot water, space heating and cooling. The main heat pump uses CO_2 as a working fluid to mainly produce domestic hot water from continuous sources, such as refrigerated rooms and freezers. The propane backup heat pump can provide both heating and cooling according to demand. This configuration provides redundancy and reliability in case of cycle shutdown. The solution is also flexible, as it grants the possibility of only running the CO_2 cycle during parts of the season when space heating and cooling demands are sufficiently low, for instance during late spring.

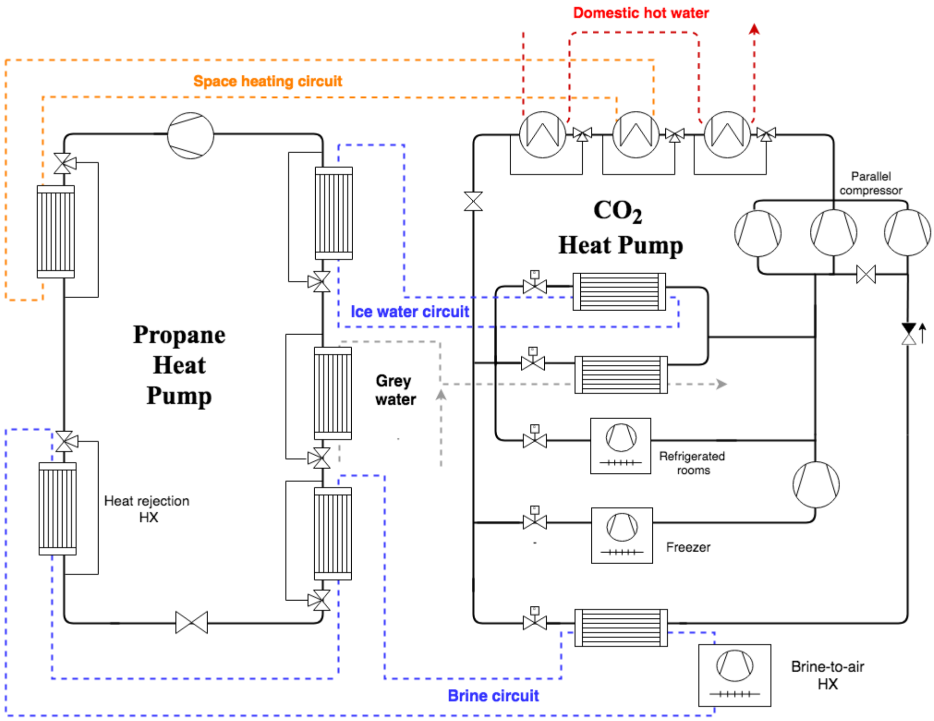


Figure 4.4: *Principle sketch of heat pump systems.*

The heat pumps are provided with interfaces to indirect subsystems for distribution and collection of heat, where only the refrigerated rooms and freezers are implemented with direct cooling. Indirect systems are beneficial as they enable the possibility of thermal storage, which can sever the close connection between thermal energy production and instant demand, as discussed in section 3.4. The domestic hot water subsystem is implemented with thermal storage, as it necessary to meet instantaneous demand and distribution of water. For the space heating and cooling subsystems, on the other hand, the purpose of the thermal storage is to shift peak demands and grant a higher degree of continuous operation of the combined heating and cooling system.

Rather than having a direct heat exchange with ambient air, a brine circuit operates as a liaison between both heat pumps and the ambient. The circuit provides an indirect interface to the ambient, as heat is transported to and from the air-to-brine heat exchanger. Dependent on necessity, the circuit can function both as a heat source and a heat sink within a single loop. Such a solution makes it possible to utilize the roof of the building for placement of heat exchangers, which in turn will facilitate the dimensioning of the system with respect to available space.

Another advantage with the indirect system is that it will ease the placement of the machine room, which can be especially beneficial in regard to the propane heat pump. As propane is classified as an A3 working fluid (flammable, non-toxic), in the *Pressure Equipment Directive* (PED), certain additional safety measures are required in comparison with CO_2[1]. An independent fail-safe ventilation system must be installed for the propane heat pump in case of leakage. The machine room must be placed above ground in order to expose the ventilation drainage in a secure outdoor location. With an indirect interface to the propane heat pump, one is relieved from additionally accounting for the location of sources/sinks when placing the machine room.

4.2.1 CO_2 system

The CO_2 system is designed for operations at low pressure, mid pressure and a parallel pressure level that is dependent on ambient temperature, as illustrated in figure 4.5. As it is assumed that external sources, such as energy wells, are unavailable, the system concept incorporates available sources to provide domestic hot water. The refrigerated rooms and freezers require continuous operation of the heat pump, resulting in round-the-clock generation of domestic hot water, which is stored for later use.

[1] A1 classification (non-flammable, non-toxic).

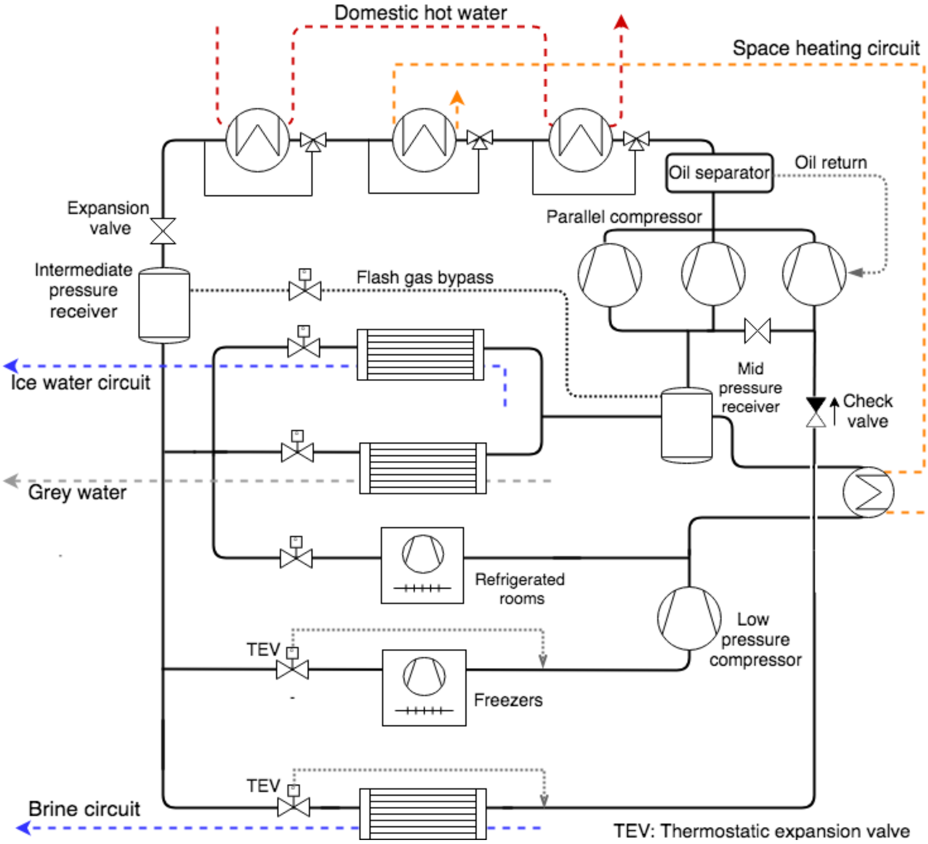


Figure 4.5: *CO_2 heat pump system.*

The gas cooler configuration consists of three heat exchangers in series, which delivers heat to the domestic hot water and space heating circuits. The lower range of the CO_2 temperature glide is utilized for preheating domestic hot water from 5 ^oC to 25 ^oC, as illustrated in figure 4.6. The mid heat exchanger covers the temperature span of the space heating circuit from 25 ^oC to 35 ^oC. The reheating of domestic hot water takes place in the first heat exchanger on the CO_2 temperature glide, where the water is heated from 25 ^oC to 70 ^oC. In the gas cooler, heat is rejected at relatively constant pressure until it reaches a temperature in the proximity of the inlet domestic hot water. This temperature is typically 2-5 ^oC above the inlet water temperature, depending on the season. During seasons with no demand for space heating, the mid heat exchanger will be bypassed and the entirety of the heat will be allocated to domestic hot water.

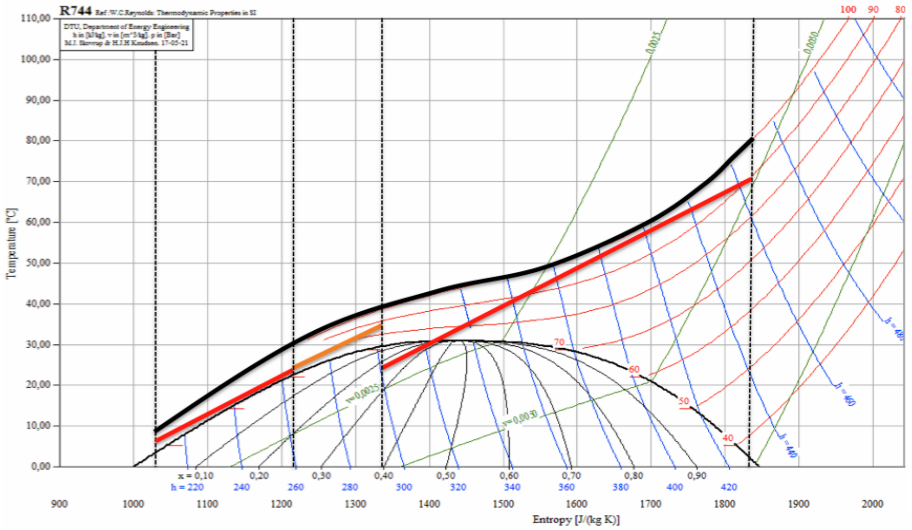


__Figure 4.6:__ Temperature-entropy diagram illustrating CO_2 gas cooler temperature fit.

The pressure-enthalpy diagram in figure 4.7 illustrates the operation of the CO_2 heat pump when bypassing the brine interface, and thus the parallel compressor. From the gas cooler, the fluid is expanded from the high-pressure side to an intermediate pressure receiver at 40 bar (3-4). The receiver functions as a flash tank where vapor is removed through a bypass to the mid pressure receiver (4-5). This configuration is beneficial as only liquid is transported further. Hence, low vapor quality will provide superior heat transfer in the evaporators.

The CO_2 cycle collects thermal energy from the ice water for space cooling and the refrigerated rooms at 34 bar and -0.9 ^oC (6-7). Expansion and flow control valves regulate pressure and mass flow according to load. After evaporation, the gas proceeds to the mid pressure receiver, which serves several purposes, such as preventing liquid carryover to the second stage compressors. Consequently, it enables the possibility of running the evaporators flooded, which will greatly improve heat transfer. The mid pressure receiver functions as an intercooler for the gas from the low-pressure stage (10-1), and assists in evaporating liquid present in the tank. The pressure in the tank is controlled by the second stage compressors (1-2), which continuously remove gas and increase the pressure. The compressor unit is comprised of several piston compressors working in parallel. Capacity control is achieved by cylinder unloading and on/off regulation. This is preferable to a single compressor unit with a variable-speed drive (VSD) motor, as there will be great variations in capacity. The system configuration and storage tanks allow the heat pump to run with low capacity for several hours. It is therefore not beneficial with great speed reduction, as this is closely tied to decreased compressor efficiency. As the focus of this report is on energy flow, and thus the total work, the actual required number of piston compressors for the system has not been determined.

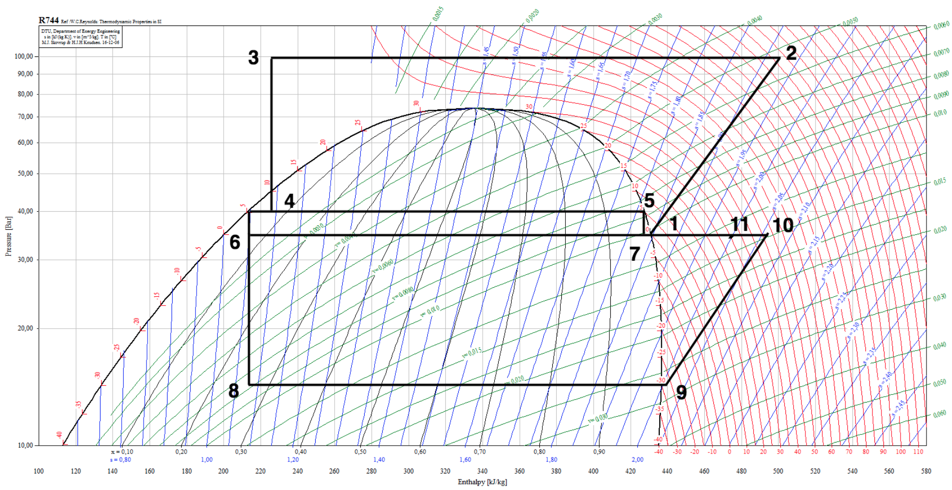


Figure 4.7: *CO_2 Pressure-enthalpy diagram without parallel compression.*

It is within the mid-pressure level possible to collect heat through grey water from the hotel drainage system (6-7). This heat circuit will only be included when the domestic hot tanks require additional water production, e.g. to maintain a certain temperature. While above this temperature level, the grey water heat exchanger is bypassed. The amount of available heat is tied to the domestic hot water consumption. By assuming that grey water has a ΔT of 18 ^oC in the heat exchanger (20-2 ^oC), the potential maximum heat recovery is 275 kWh/day[2]. In reality, this source will be smaller due to peak load supply of grey water, which would require an unreasonably large heat exchanger working with fluctuating loads. The solution to this obstacle is to implement a storage system for grey water that can function

[2] Assuming domestic hot water is delivered at 50 ^oC, and thus a mass mixing ratio 9:4 between water at 70 ^oC and 5 ^oC, with the domestic hot water consumption in figure 4.1.

as a buffer. The required magnitude of the storage tanks will be investigated in the simulations.

The low pressure evaporator operates at a pressure and temperature of 14.2 bar and -30 ^oC (8-9), and collects heat in the freezer rooms, where a thermostatic valve ensures gas phase before compression. To reduce the superheat before entering the mid pressure receiver, a heat exchanger interface to the space heating circuit is implemented (10-11). Similar to the mid compressor arrangement, the low pressure compressor is a reciprocating unit.

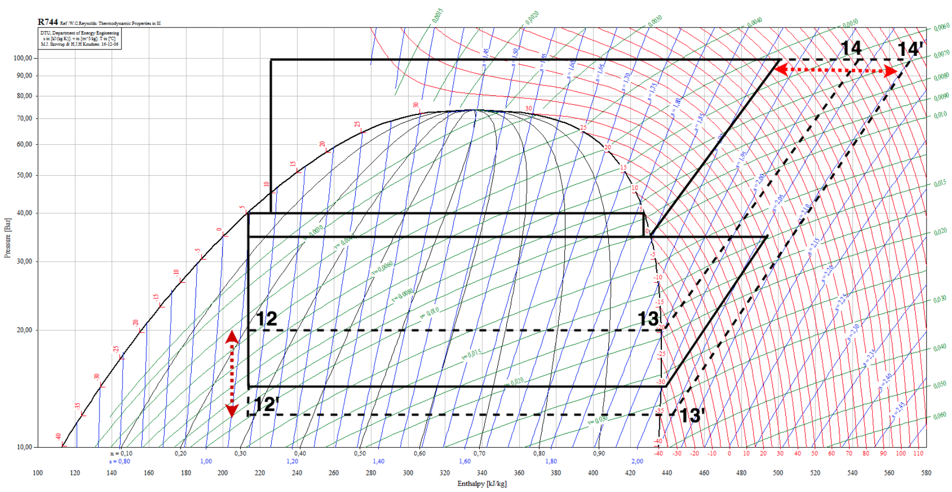


Figure 4.8: *CO_2 Pressure-enthalpy diagram with parallel compression.*

When heat recovery from grey water is not sufficient to cover the domestic hot water demand, the interface to the brine circuit is employed and the parallel compressor is activated. The current parallel circuit implementation enables the possibility for the compressor to function in several different ways, as illustrated in figure 4.8. When the ambient temperature, and thus brine temperature, provides the means of evaporation close to the mid-pressure level, it is possible to run the parallel compressor incorporated in mid compressor arrangement (Hafner, 2017). This is achieved by opening the valve between the compressors, making the parallel compressor accessible for the remainder of the mid-pressure level. The compressors will then work in a single stage, as illustrated in figure 4.7, stage (1-2).

During cold seasons, the ambient reaches a temperature close to or below that of the CO_2 mid level evaporating temperature (-0.9 ^oC). This makes it impossible to run the parallel compressor integrated in the mid compressor arrangement due to the lack of sufficient temperature difference in the brine heat exchanger, or plainly a negative temperature gradient. The valve between the compressors will then be shut and the parallel compressor will be dismantled from the mid-pressure level. The brine evaporator circuit then operates with a parallel compressor to the rest of the system (12-14). This enables evaporation at low temperatures, e.g. during winter season.

4.2.2 Propane system

The propane heat pump functions as a backup system for heating and cooling, which is activated if the demand is lager than the thermal energy produced by the CO_2 system. The cycle can be operated purely as a heat pump, refrigeration unit or as a combination unit, depending on the application of the different heat exchangers.

The propane system is a single stage heat pump with several piston compressors working in parallel. The compressors are controlled analogously to the CO_2 compressors, by on/off regulation or by means of cylinder unloading. On the condenser side, a plate heat exchanger provides an interface to the space heating circuit. The propane system will primarily utilize ice water as a heat source and bypass the other evaporators. This will be the case when there is demand for cooling or during charging of the cold storage. If this source is not sufficient to cover the heating demand it will be bypassed and an interface to the brine circuit or grey water will be applied. When the demand for cooling surpasses that of heating, the brine circuit is utilized for heat rejection, as the space heating heat exchanger is bypassed. The propane heat pump can operate at various pressures, both in the condenser and evaporator, dependent on the source and sink temperatures. When the interface to the space heating circuit is active, the condensing temperature and pressure is approximately 36.1 ^oC and 12.5 bar.

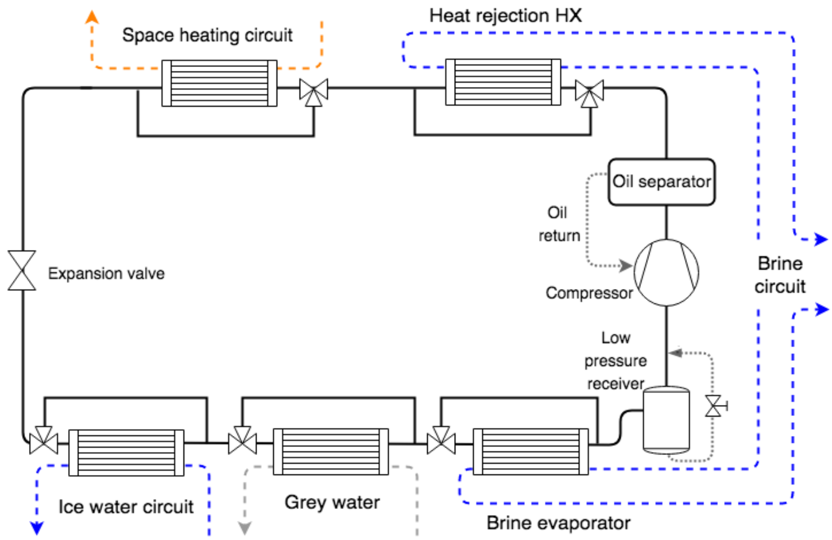


Figure 4.9: *Propane heat pump system.*

The temperature in the space heating circuit varies between heating return temperature and 35 ^oC as the thermal storage is being charged. This will influence the propane vapor quality at the outlet of the condenser. When the space cooling circuit is connected, the evaporator operates at 4 ^oC and 5.3 bar to produce ice water at 7 ^oC. Similar to the space heating circuit, the inlet temperature to the ice water heat exchanger changes according to the return temperature of the cooling circuit and the level of the storage unit.

4.2.3 Demand based system operation

The main challenge with the system is that it must be controlled according to both heating and cooling demands. This becomes especially evident during parts of the year when neither heating or cooling demand is in vast dominance, but still of a certain magnitude. When this occurs, both the CO_2 and propane heat pumps are active, and the operational strategy must be coordinated according to both contributions. Due to the shared sources and sinks, e.g. space heating and cooling, an intricate control system is unavoidable. The CO_2 system will produce maximum possible space heat within the temperature glide, but at the same time, the magnitude of heat produced is dependent on the cooling provided. Whether or not this heat is sufficient to meet the next peak demand is thus dependent on variables beyond the instantaneous charge in the thermal storage. The control strategy for the system is presented for the following two scenarios.

Heating mode

When the combined heating and cooling system is running in heating mode, all available sources are utilized for such purposes. As illustrated by the green path in figure 4.10, both heat pumps are activated and it is assumed that there is no demand for cooling. The CO_2 cycle collects heat from grey water and the brine circuit, in addition to the base load from refrigeration rooms and freezers. If space heating demand predominates that of domestic hot water, the gas coolers are controlled for maximum rejecting to this circuit. This is managed by controlling the water flow through the heat exchangers and reducing the pressure in the gas coolers. This ensures a more leveled heat rejection curve, and thus more potential heat to the mid heat exchanger.

Similar to the CO_2 system, the propane heat pump utilizes the brine circuit as a heat source. This brine loop is, however, in parallel with the CO_2 interface, as a series solution would result in low grade energy for one of the heat exchangers. As grey water is a limited source, it is in this scenario only assigned to the CO_2 cycle. This will ensure continuous and stable operations for the propane system, which would not be the case if it were necessary to switch sources during operations.

Cooling mode

During cooling mode, it is assumed that there is no space heating demand, and that the thermal storage is fully charged, as can be observed in figure 4.11. The CO_2 heat pump only includes the base load functions from refrigeration rooms and freezers to produce domestic hot water. Hence, the heat exchanger interface to the space heating circuit is bypassed. The domestic hot water tanks must be of a certain magnitude to avoid heat rejection to the ambient. Ice water is produced and the surplus heat is rejected when the propane system is working in refrigeration mode.

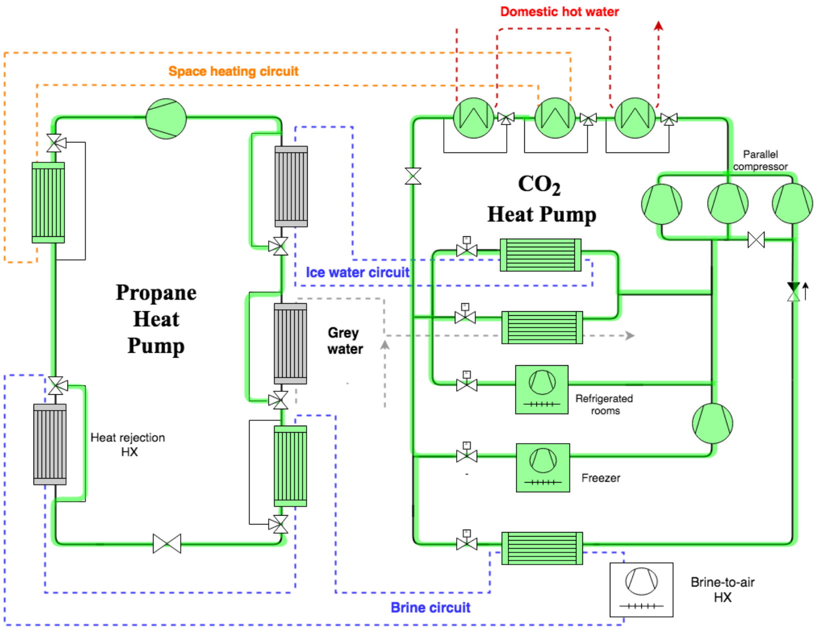


Figure 4.10: Heat pump operation with high heating demand.

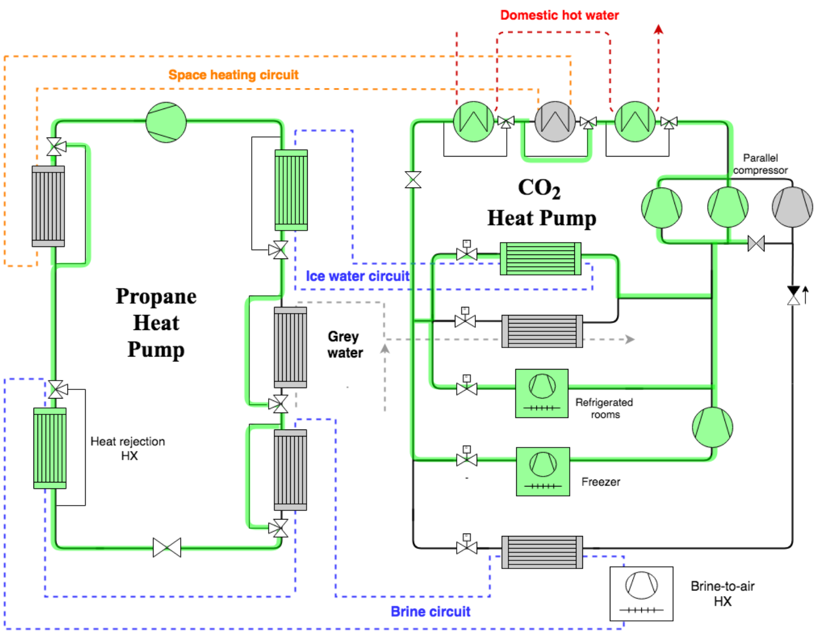


Figure 4.11: Heat pump operation with high cooling demand.

4.3 Subsystems

4.3.1 Domestic hot water subsystem

One of the primary functions of the CO_2 system is to produce domestic hot water. As seen in figure 4.12, the domestic hot water circuit is connected in parallel to the CO_2 cycle, making bypass possible. Before entering the hotel water distribution system, city water is filtered to remove bacteria and particles. The city water is moved to the tanks, directly to the CO_2 system for heating or to the mixing tap for distribution. When the CO_2 system is in operation, water is either removed from the tanks or supplied from the return line, depending on the tank levels and demand. Water is then heated in two series heat exchangers, first preheat and then reheat, as discussed in section 3.2. After reaching the desired temperature of 70 ^oC, water is supplied to the tanks for storage. If demand is to increase, domestic hot water can be supplied directly from the CO_2 gas coolers to the consumer supply line.

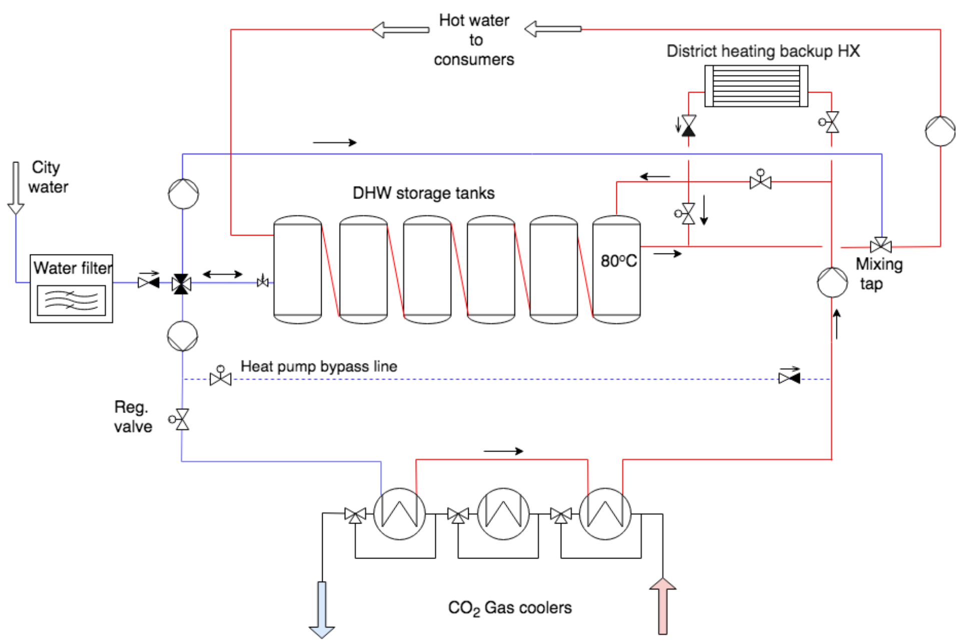


Figure 4.12: *Subsystem for distribution of domestic hot water.*

Even with a low demand, domestic hot water must be circulated through the CO_2 gas coolers. This is a challenge when the tanks are full and, as a result, the inlet water temperature to the heat exchangers increases. When lacking sufficient cool down before throttling, cooling capacity is reduced drastically, as can be observed in figure 3.2. Whether or not this will result in significant losses will be investigated in the simulations.

Taking into consideration the consumption curve presented in figure 4.1, the magnitude of the domestic hot water storage system must be in the range of 10 m^3. This equals an energy reservoir of 755 kWh and insures that the tanks alone can act as a buffer that is able to supply the hotel with hot water for at least 24 hours. This is a safety measure in case of cycle shutdown over short periods of time. A district heating backup heat exchanger has been included in the design as a redundancy. When utilized, the CO_2 system will be bypassed and water will be heated in a single step from 5-70 ^oC.

As the high water temperature provides sufficient thermal storage, it is deemed unnecessary with PCM implementation in the tanks. However, if the magnitude of the storage tanks proves problematic, it is possible to implement PCMs and ensure the same thermal capacity with reduced water volume.

4.3.2 Space heating subsystem

The space heating system utilizes the mid section of the CO_2 temperature glide to heat water to 35 ^oC. The water is then supplied to the buffer storage tanks or directly distributed for heating purposes. After heat has been rejected across the building, the fluid enters the return line and is recirculated to the tanks or to the CO_2 system for reheating, depending on demand. When tank levels are low and the demand cannot be covered solely by the CO_2 system, the propane heat pump is activated. As the tanks are charged, water from the return line is mixed with water from the tanks. Thenceforth, water is distributed in parallel lines to the CO_2 and propane heat pump, as illustrated in figure 4.13. Similar to the domestic hot water circuit, the inlet water temperature to the gas cooler will increase if the demand is low and the thermal storage is to reach full potential. This will result in increased water mass flow through the heat exchanger(s). The space heating circuit is also implemented with a district heating backup heat exchanger for redundancy purposes. This can be utilized alone to cover the entire heating demand or work in parallel with the CO_2 and propane heat pumps.

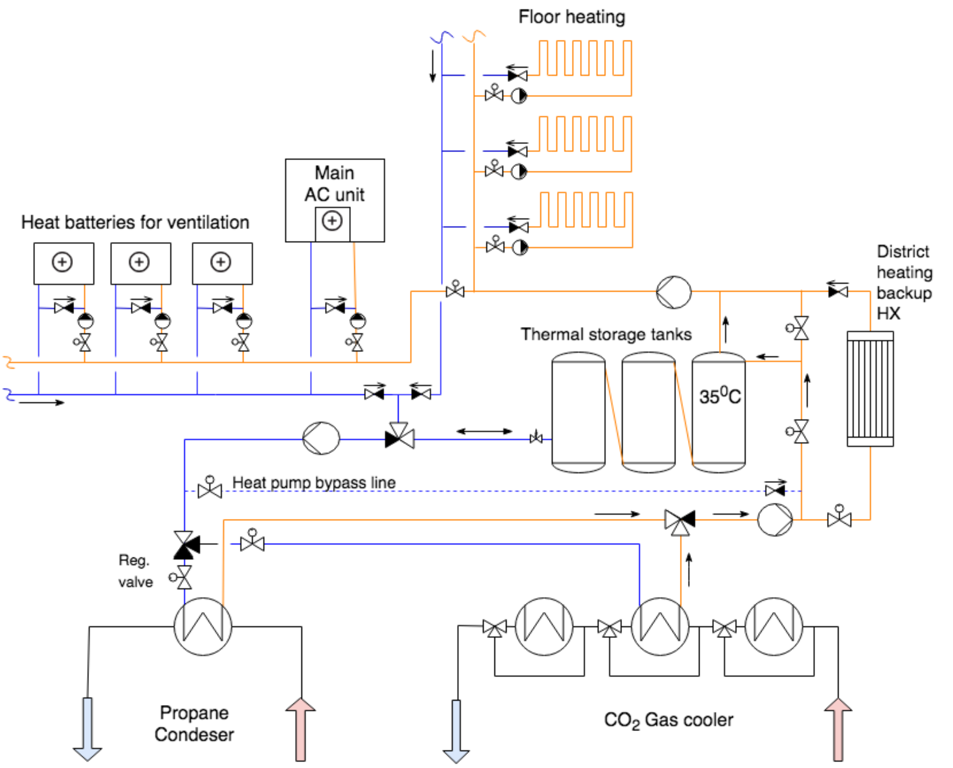
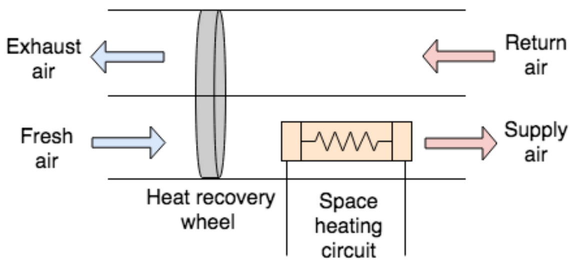


Figure 4.13: Subsystem for distribution of space heating.

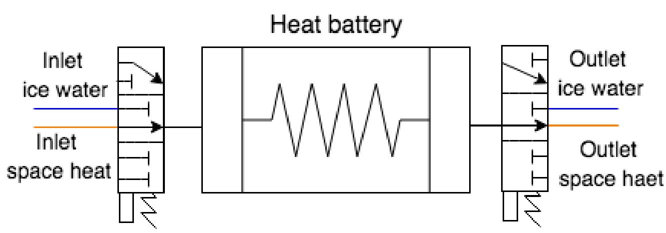
If the space heating system were to have the same energy storage capacity as domestic hot water, the tanks would have to be in the range of 72.3 m^3, which is both unpractical and unrealistic. It is, however, necessary with a thermal storage of a certain magnitude due to the high space heating demand during winter operations. The buffer provided would, in addition, grant more stability in the system, as discussed in section 3.4.2. By installing a PCM tank of 10 m^3, equipped with a similar capacity to that of TubeIce S34, the hotel would obtain a reservoir of 500 kWh.

Compared to a water tank of equal size, this would be a 481% increase in capacity[3], which is equivalent to a space heating water volume of 47.9 m^3. However, the PCMs should have a transition temperature equal to the setpoint temperature of 35 ^oC. This would require a greater maximum tank temperature in order to charge the PCM modules. The opposite is true if utilizing PCMs for cold storage.

Contrary to the domestic hot water subsystem, the space heating circuit is closed as fluid within the system recirculated. Heat is primarily distributed as floor heating through parallel configurations for temperature insurance. Heat is also provided to the air condition (AC) supply air after primary heat recovery has transpired, as seen in figure 4.14a. For separate temperature regulation in each room, ventilation-heating batteries are installed in the supply air shafts. The temperature in each room can be regulated according to the thermal comfort of that particular guest.



***(a)** Heat interface to AC after recovery.*



***(b)** Heating and cooling battery.*

***Figure 4.14:** Space heating and cooling interface to air condition.*

[3]Assuming constant tank inlet/outlet temperature(25 ^oC-35 ^oC) and no heat losses.

The batteries consist of a four-way piping system, with a return line for both heating and cooling circuits, as illustrated in figure 4.14b. The flow through the battery is controlled by valves with three positions, enabling flow from only one circuit at any time. If the distribution pipes are adequately isolated, this configuration will provide a simple and reliable system for both comfort heating and cooling. When there is no thermal demand, the valves are set in a closed position.

Building structural PCMs should be installed to reduce the peak demands for space heating and cooling. Even a simple solution, such as concrete blocks, would enable both hot and cold storage, depending on the season. Another possibility is installing PCM wallboards. These are, however, treated by materials with a specific transition temperature, making them difficult to use for both heating and cooling due to the different setpoint temperatures.

4.3.3 Space cooling subsystem

The ice water subsystem is a closed circuit, as illustrated in figure 4.15. Water at 7 ^oC is supplied to ventilation cooling batteries in the hotel rooms, main AC unit and the kitchen areas. The configuration of the batteries is as described in the previous section.

In the return line, water is circulated to the high temperature side of the tank arrangement or supplied directly to the CO_2 system. The ice water circuit utilizes the CO_2 mid-pressure evaporator, depending on cooling demand and tank charge. When tank levels are low and cooling demand exceeds the refrigeration load provided by CO_2, the propane heat pump compressor is turned on. If the heat pump is already activated to provide space heating, the brine heat exchangers will be bypassed and the ice water heat exchanger will become the evaporator.

The storing potential in the water tanks is limited due to the low temperature difference between the tank inlet and outlet. By installing the same type of PCM tanks as Bergen University College (sec. 3.4.3), the cooling capacity would increase immensely.

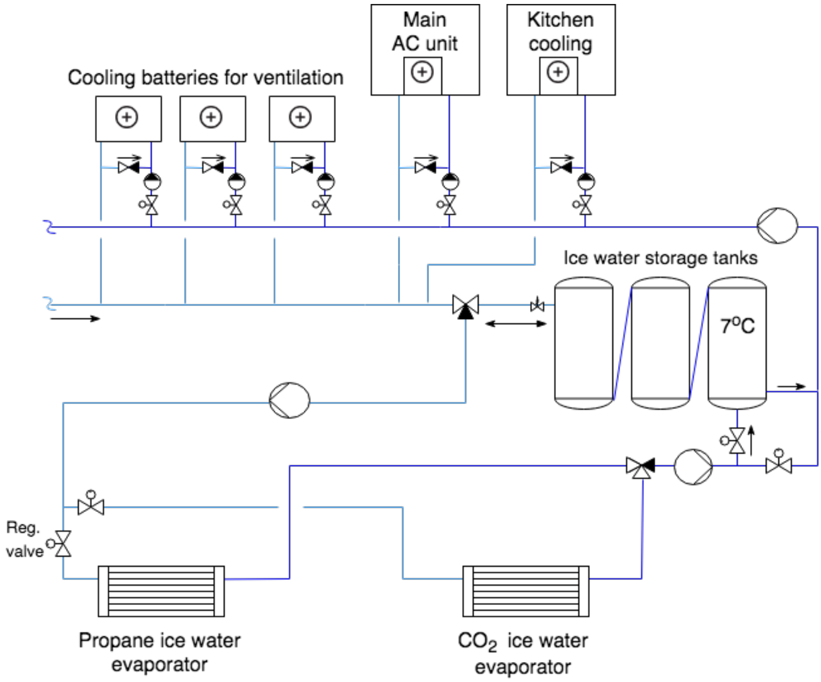


Figure 4.15: *Subsystem for distribution of ice water for cooling.*

A 10 m^3 tank filled with PlateIce S8 would insure a thermal reservoir in the range of 440 kWh. Regarding placement, it could be possible to install the tank outside the building below ground. If such a solution is to be considered, inquiries regarding the piping scheme, ground temperature, composition, etc. must be made.

Chapter 5

Simulation platform and model architecture

This chapter describes the simulation tool and main assumptions for the model, where the structure and control strategy are reviewed for different parts of the system. Seasonal scenarios and input values for simulations under different operational conditions are also explained.

5.1 Simulation platform

The physical simulation tool, *Dymola*, was applied for the modeling and simulation of the energy system. The software utilizes an orientated physical modeling approach, where large and complex systems are composed of component elements. What this means in practice is that each component in the diagram structure represents a real physical element in the system. When the elements are connected in the model structure, the mathematical equations in all components constitute the dynamic behavior of the model. Dymola is based on the *Modelica* modeling language and covers multiple engineering domains. Different elements for mechanical, electrical and hydraulic operations are among the disciplines included in the Modelica libraries. *TIL-Media* is a library designed by *TLK-Thermo*, and is especially made for efficient and accurate calculations of thermophysical properties. This package provided the components necessary to build the energy system.

An important factor for simulation performance is choosing which integration method to use, as the solvers behave differently according to the particular type of problem. For instance, explicit integration methods work well for systems with a narrow range of time scales. The energy system for the hotel is large, complex and contains a wide range of dependent dynamic time scales, causing model *stiffness* (Tiller, 2001). The consequence of this occurrence is that certain integration methods become numerically unstable unless the simulation step size is reduced to an extremely small value. Implicit solvers, on the other hand, handle stiff problems more efficiently. The *Radau IIa* integration solver is an implicit *Runge-Kutta* method that was selected to handle the energy system model. This method uses single steps, meaning that the solution at the current time step is only influenced by data from the previous time step (Jorissen et al., 2015). It is therefore very effective when handling simulations where many *events* are generated. Events are system changes that will cause the solver to reset and reduce step size in order to locate the precise time of occurrence, resulting in decreased simulation speed. The integration tolerance describes the acceptable error in each time step and is set to the value of 1e-6 for all simulation scenarios.

The simulation of a Dymola model generally proceeds as follows. First, the state variables are initialized based on the initial equations and start values. Then, the continuous time integration starts and results are saved at the time intervals until completion. The selection of correct initial values is therefore key for simulation success. When dealing with dynamic models, it can often be an advantage to initiate the simulation using constant input values. A constant initialization stage of 14400 seconds (4 hours) has been selected for all simulation.

5.2 Description of the Dymola model

The hotel energy system has been modeled in a single model structure, containing over 250 components, and can be found in its entirety in appendix B.1. The Dymola model has been simplified in design for simulation purposes. The brine circuit interfaces to the CO_2 and propane heat pumps have been replaced by cross-flow air heat exchangers working with separate air loops. The simplification will have an influence on the efficiency of the heat exchange, as an additional heat transfer step has been eliminated. The influence from the simplification will have minor impact on the energy flow in the system, but will, on the other hand, considerably reduce the number of dependent variables in the model. While still present in the CO_2 system, the grey water heat recovery interface to propane system has been excluded. This was done in order to investigate the necessary grey water storage volume and determine if it is within a reasonable range when available to only one heat pump system.

The following assumptions have been made for the overall system:

- No unintentional pressure loss in components.
- No unintentional heat transfer between components and the ambient.
- No tube elements between components; the influence of tube dimension neglected.

The heat exchangers have been dimensioned to cover the required load with a temperature pinch of maximum 5 ^{o}C. The CO_2 grey water and ice water heat exchangers are working on the same pressure level as the refrigeration rooms (34 bar), and are therefore restricted by the required low temperature in these rooms (3 ^{o}C). The temperature pinch in the ice water heat exchanger is therefore larger as the ice water circuit has a setpoint of 7 ^{o}C. Heat exchangers with variation in stream temperature or pressure will also have a pinch variation dependent on these variables.

Overall assumptions for all heat exchangers are listed below and their specification can be found in appendix B.2.

- Geometry based wall conduction model for tubes in all heat exchangers.
- 1-D efficiency approximation (Schmidt) for fins in fin-and-tube heat exchangers.
- Constant heat transfer in all heat exchangers:
  - U-value for propane and CO_2 in heat exchangers is set to 2500 W/m^{2}K.
  - U-value for moist air in heat exchangers is set to 150 W/m^{2}K.
  - U-value for water in heat exchangers is set to 1000 W/m^{2}K.

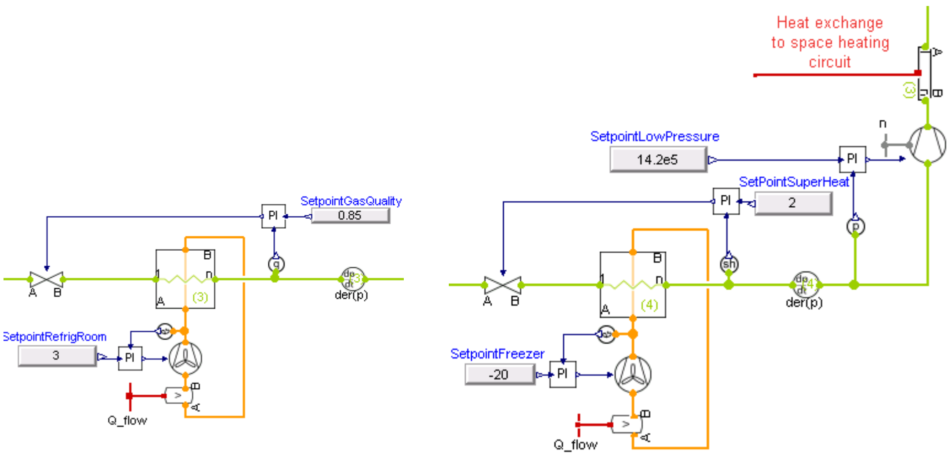
All compressors are modeled as efficiency compressors and regulates the displacement of fluid with a proportional-integral (PI) controller. Assumptions for all compressors are listed below and detailed specifications can be found in appendix B.3.

- Constant volumetric and isentropic efficiency of 0.70 for all compressors.
- Constant speed of 50 Hz for all compressors.

5.2.1 Model structure and control strategy

Cold storage rooms

The refrigerated rooms are modeled as a single volume of 75 m^3, where a continuous heat source of 10 kW is applied to the air volume, expressing the heat accumulated due to infiltration and cooling of goods. A fan regulates the airflow through the fin-and-tube cross-flow heat exchanger to keep the room temperature at the setpoint of 3 ^{o}C, as illustrated in figure 5.1a. The area regulation of the expansion valve controls the flow of CO_2 through the heat exchanger to maintain a gas quality of 0.85 at the outlet.



(a) Refrigeration rooms interface. *(b) Freezer rooms interface.*

Figure 5.1: *CO_2 evaporators supplying cooling to refrigeration and freezing rooms.*

The freezer rooms are modeled and controlled much like the refrigerated rooms, with a setpoint temperature of -20 ^{o}C and a room air volume of 60 m^3, as depicted in figure 5.1b. The valve is controlled to ensure a superheat of 2 ^{o}C at the outlet of the heat exchanger, and thus no liquid carryover to the compressor. The compressor regulates the displacement of fluid in order to maintain a pressure of 14.2 bar. A tube at the compressor outlet transfers heat to the space heating circuit before the vapor enters the mid pressure receiver.

CO_2 gas coolers and domestic hot water circuit

Figure 5.2 depicts the model for the CO_2 gas coolers and domestic hot water circuit. The compressor rack is simplified to a single unit that has the capacity to regulate the displacement in order to maintain the mid-pressure level of 34 bar. On the high-pressure side, CO_2 is cooled down and heat is distributed to domestic hot water and space heating, as described in section 4.2.1.

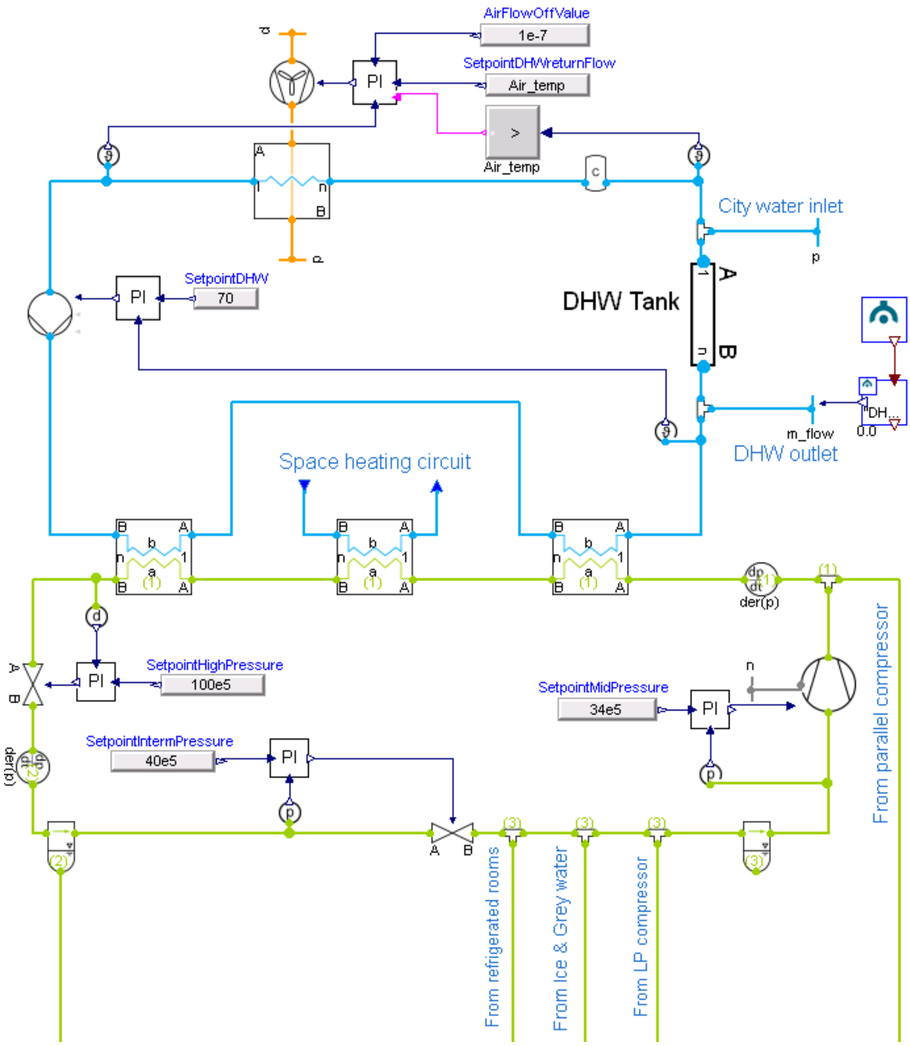


__Figure 5.2:__ CO_2 intermediate, mid and high pressure control with the domestic hot water circuit.

The high pressure is controlled by area regulation of the expansion valve to uphold the setpoint of 100 bar. The liquid is let down in the intermediate pressure receiver, where vapor is removed through the bypass. The bypass valve controls the amount removed as the valve area is regulated to maintain 40 bar in the receiver. CO_2 removed through the valve is mixed, in junction elements, with the streams from the mid and low levels, before entering the mid pressure receiver.

The domestic hot water mass flow through the gas coolers is regulated by the pump in order to reach a temperature of 70 ^{o}C at the inlet of the storage tank. If water is in demand, it is removed directly to the consumer, represented by a outlet mass flow boundary. Input values for the mass flow is read from an *Excel* work sheet to the boundary, according to the hourly demand presented in figure 4.1. The domestic hot water tanks are modeled as a single tube with an inner diameter of 1 m and height of 12.73 m. The tube model pushes water through the volume in both directions, depending on demand. When demand is greater than supply, water moves from the bottom to top (A to B), and water at a constant temperature of 5 ^{o}C is supplied both to the bottom of the tank and to preheat. When demand is less than supply, water is pushed in the opposite direction (B to A), and mixes with water from the inlet boundary, which is set to have a constant pressure of 1.013 bar (1 atm). During time intervals when the tank is full and demand is low, the water supply temperature to the gas cooler increases. To maintain the cooling capacity in the CO_2 system, heat is removed through a cross-flow air heat exchanger. To simplify the model, a bypass of the heat exchanger is replaced by an on/off regulation of the air volume flow. As Dymola has difficulties with singularities, the off-value is set to 1e-7 m^3/s. The fan is activated when the water supply temperature to the gas cooler exceeds 10 ^{o}C and the air volume flow is then regulated to chill the water stream down to this temperature level. When $T_{amb} >$ 10 ^{o}C, the fan activation and supply water setpoint temperature is set to that of the ambient air. The input temperature to the air boundary is fixed to a constant average for the simulated time interval in order to limit the dynamics in the model.

Domestic hot water backup supply

If the heat supplied from the CO_2 evaporators is not sufficient to maintain a temperature of 70 ^oC in the top of tank, backup evaporators will collect additional heat. The grey water heat exchanger will first try to solely handle the load. When activated, the valve regulates the area in order to obtain a superheat of 1 ^oC, as illustrated in figure 5.3. Contrary to the intended system presented in chapter 4.2, the evaporator is not flooded. This is due to liquid accumulation in the mid pressure receiver during instances when all evaporators are active. The liquid in itself will in reality not cause problems, but affects the simulations greatly as the receiver filling level starts fluctuating between a small value and 0, which generates events.

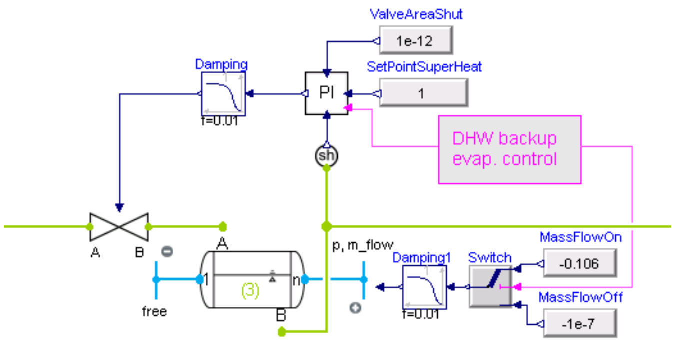


Figure 5.3: *CO_2 grey water backup heat exchanger.*

When the valve is active, the grey water mass flow through the shell-and-tube heat exchanger is 0.106 kg/s, based on the average domestic hot water produced over 24 hours. The grey water inlet boundary is overdetermined, and thus specifies both temperature and pressure. The latter is set to 1.013 bar (1 atm) and the temperature is assumed to be constant at 20 ^oC. When deactivated, the valve area is set to 1e-12 m^2 and the grey water mass flow to 1e-7 kg/s. First order damping filters are included to smoothen the on/off transition of the signals.

Figure 5.4 depicts the parallel flow path and compressor in the CO_2 system. For simplification, the compressor is connected to a separate circuit and only handles the fluid from this line. When activated, the fan airflow through the evaporator is set to 10 m^3/s and the superheat at the outlet is controlled to 2 ^oC. The compressor regulates the fluid displacement in order to maintain a heat flow of 25 kW in the heat exchanger. The pressure at this stage is therefore not set, but determined based on the constraints previously mentioned.

The evaporator acts as a ”last resort” in circumstances when heat recovery from grey water is not sufficient to maintain an acceptable temperature level in the domestic hot water tank. The magnitude of the heat flow is set such that the tank can quickly be charged.

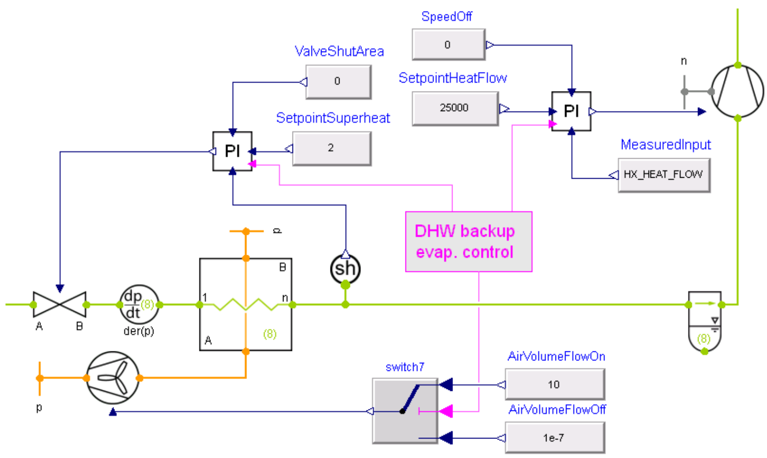


Figure 5.4: *CO_2 backup air evaporator.*

The fluid is compressed directly to the high-pressure side, where it is mixed with fluid from the mid compressor, before entering the gas coolers. As seen in figure 5.4, a liquid receiver has been included in the model due to fluid entrapment during deactivation.

As the valve and compressor act as fluid blockers in both ends of the flow path, it is possible to set their deactivation value to 0. When this occurs, the fan is deactivated and the volume flow is fixed to 1e-7 m^3/s.

As seen in figures 5.3 and 5.4, the activation of components is regulated a control box. The decision tree, depicted in figure 5.5, illustrates the control strategy for the production of domestic hot water. Details of all logical elements included in the control can be found in appendix B.1.

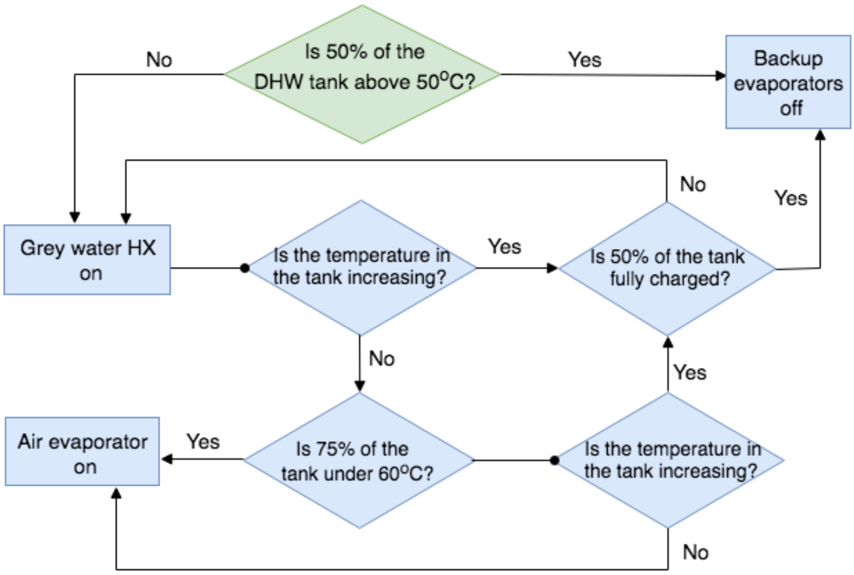


Figure 5.5: *Decision tree illustrating the control logic for the domestic hot water backup evaporators.*

Propane heat pump and interfaces to space heating and cooling circuits

The heating and cooling systems receive contributions from both heat pumps, as depicted in figure 5.6. Similar to the domestic hot water tank configuration, the space heating and cooling tanks are modeled as tube elements. The thermal storage tanks are simplified to plain water tubes with a storing capacity equal to 10 m^3 PCM tanks, as discussed in section 4.3. The space heating and cooling tubes have an inner diameter of 1 m and height of 60.93 m and 60.15 m, respectively. When activated, the pumps in both circuits regulate the flow through the heat exchangers to obtain the setpoint temperature, which is 35 ^oC for space heating and 7 ^oC for ice water. The deactivation value for all pumps are set to 1e-7 kg/s.

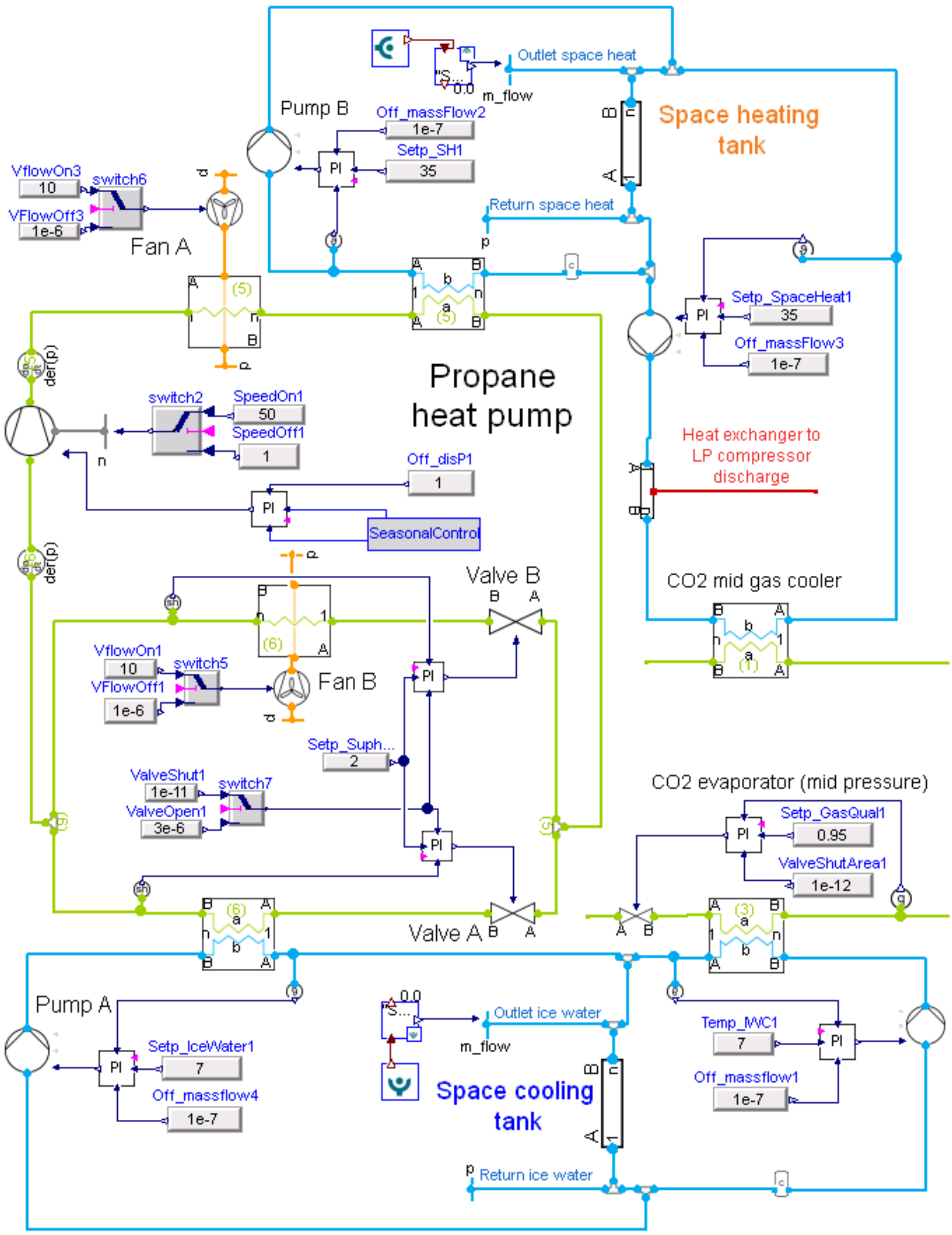


Figure 5.6: *Propane heat pump with the space heating and ice water circuits.*

The space heating system is supplied with heat from a tube interface from the low-pressure level before entering the mid gas cooler. Heated water is returned to the tank or distributed to the building through the space heat outlet boundary. The hourly demand is read from an excel file to the outlet boundary, and an equal amount of water is returned through the inlet boundary, at a constant temperature of 25 ^{o}C. The ice water circuit collects heat from the mid-pressure level in the CO_2 heat pump, where a valve controls the CO_2 mass flow to maintain a gas quality of 0.95 at the outlet of the heat exchanger. Cooled water is returned to tank or to the outlet boundary, which is configured in the same manner as the space heating circuit with a constant return temperature of 15 ^{o}C.

If the CO_2 system is not able to solely handle either heating or cooling demand, the propane heat pump is switched on. The heat pump is simplified to two condensers and two evaporators, which are activated according to demand. An air condenser is installed to reject heat during time intervals when the space heating tank is fully charged. The bypass of the air condenser is replaced by an on/off regulation of the air volume flow. During instances with heating demand, the air condenser volume flow is turned to the deactivation value of 1e-6 m^3/s. Heat is then distributed to the space heating circuit, as the pump controlling the water mass flow (pump B) is activated. Table 5.1 shows the components activated for different operation modes.

Table 5.1: *Propane heat pump operation modes and component values.*

Operation mode	**Pump A** [kg/s]	**Pump B** [kg/s]	**Valve A** [m^2]	**Valve B** [m^2]	**Fan A** [m^3/s]	**Fan B** [m^3/s]
Heating and cooling	A	A	A	D	D	D
Heating only	D	A	D	A	D	A
Cooling only	A	D	A	D	A	D
Propane heat pump deactivated	D	D	D	D	D	D

A = Activated, D = Deactivated

Dependent on cooling demand, heat is collected either through an air evaporator or directly from the ice water circuit. If the cooling tank is not fully charged, the valve to the air evaporator (valve B) is shut, and propane flows through the ice water evaporator. The expansion valve (valve A) controls the flow to obtain a superheat of 2 ^{o}C. If the ice water tank is fully charged, the evaporator operation is reversed. The valve to the ice water evaporator (valve A) is then shut, while the air evaporator valve (valve B) controls the flow to reach a 2 ^{o}C superheat. The propane heat pump is deactivated when there is no thermal demand. The compressor speed is then reduced to 1 Hz while both expansion valves are opened.

Similar to the CO_2 compressors, the propane compressor consists of a single unit that regulates the displacement of fluids. During seasons with high heating demand, the displacement is controlled according to the space heating tube's top fluid layer temperature. If this temperature drops below the setpoint temperature, the tank is almost empty. The compressor will then displace more fluid in order to charge the tank. The displacement controller insures that at least 25% of the tank is fully charged before normalizing the displacement. When cooling demand is dominant, the displacement is regulated in the same manner with the bottom layer temperature of the ice water tank as the control parameter. In reality, a number of piston compressors in the rack would be employed to regulate the flow.

To prevent the propane heat pump from frequently switching on and off, the temperature levels in both tanks are evaluated before deactivation. If the space heating tank becomes fully charged while the ice water tank is charging, the control system checks whether or not the ice water tank is at an "acceptable level". If this is the case, the propane pump is switched off. An equal evaluation is performed should the ice water tank become fully charged while the space heating tank is charging. The control logic for the space heating and cooling system is illustrated in the decision tree in figure 5.7, and the logical components can be found in appendix B.1.

When classifying the control system for heating and cooling, it is difficult to distinguish between SIMO and MIMO, which were discussed in chapter 3.5.2. The system consists of multiple control blocks that receive input from one or more parameters and return a single output to one or several components. Combined, the control logic is categorized as MIMO control, but separately act as a multiple-input single-output (MISO) regulation of different components in the heat pump.

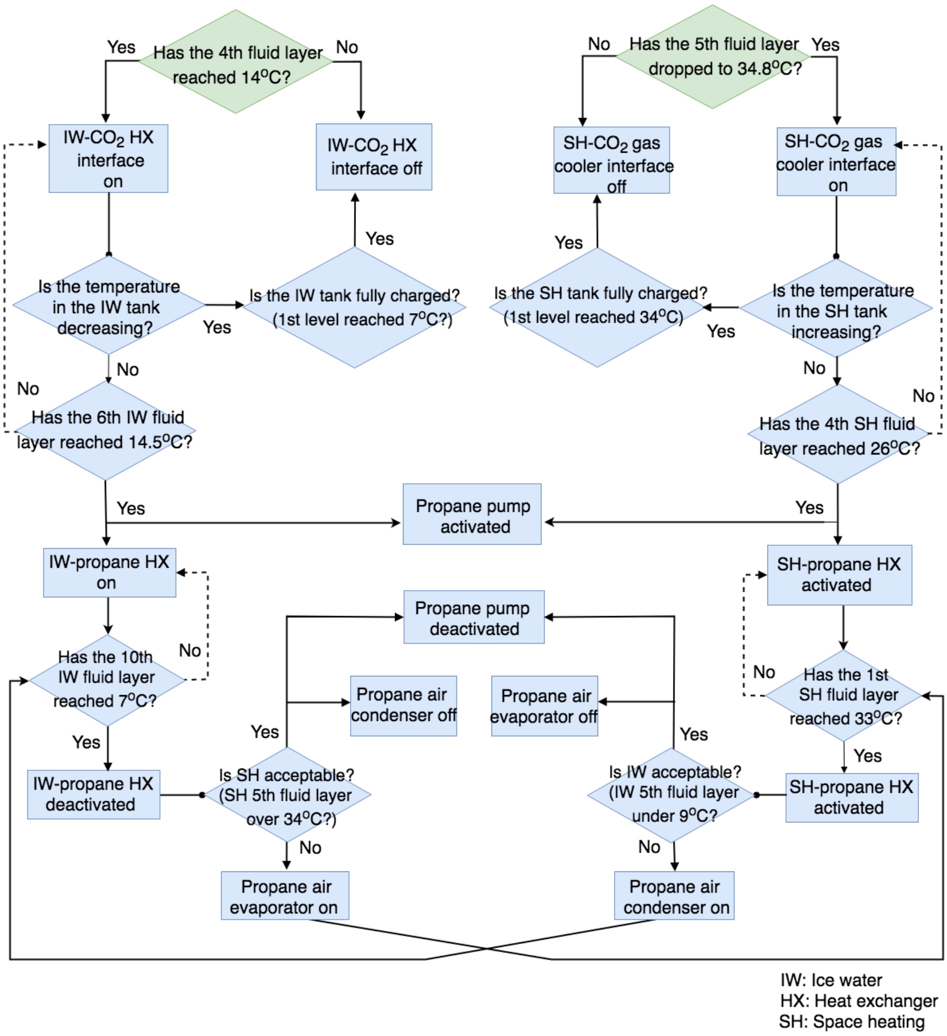


Figure 5.7: Decision tree depiction the control logic for the space heating and cooling systems.

The objective of the heat pump control is primarily to cover all thermal demands and determine the energy efficiency within these operational criteria. As the control system has not been optimized, the simulations can only outline the full potential of the combined heating and cooling system.

5.2.2 Simulation scenarios

All simulations are conducted during the time-span of a week to best evaluate effects that the buffer systems have on the operation of the heat pumps. To reveal possible operational limitations, peak values have been selected from the annual demand data created in Simien and presented in figure 4.3. This includes the week of the year with highest cooling demand, which is used for the summer scenario. On the other end of the specter, the week of the year with largest space heating demand has been chosen for the winter scenario. For the fall/spring simulation, a "typical" week with a average demand for both space heating and cooling has been selected. All simulations are based on hour-to-hour demand inputs with constant values for that particular hour. The average temperature for the week is used as input for the ambient air inlet boundaries, e.g. for the propane air evaporator and condenser. The domestic hot water demand and the cooling demand for refrigeration and freezer rooms are equal for all scenarios. The CO_2 gas cooler pressure is set to 100 bar for all simulations.

Fall/spring scenario

The week (May 7th-13th) has been selected based on fluctuations in the ambient temperature and the corresponding heating demand, which is representative for both fall and spring.

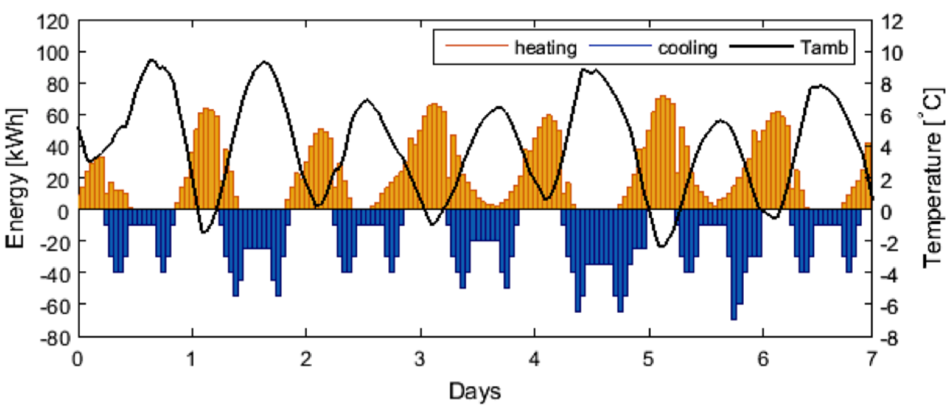


Figure 5.8: *Space heating and cooling demand with T_{amb} for fall/spring scenario.*

Figure 5.8 illustrates the space heating and cooling demand with the ambient temperature for the week. The peak heating demand occurs on the forth day at 67 kW, while the space cooling peak demand of 70 kW transpires on the fifth day. The only contribution to space cooling is due to heat generation from kitchens and conference rooms. The propane compressor displacement is controlled according to the temperature level in the space heating tank, as this demand is dominant for the week. The average temperature for the week is 4.1 ^{o}C, which is only 1 ^{o}C below the normalized average temperature for the location.

Winter scenario

Figure 5.9 illustrates the thermal demand during the week of the year with largest heating demand (Jan. 11th-17th). The heating peak value of 222 kW occurs when the ambient temperature is -19.5 ^{o}C, on the second day.

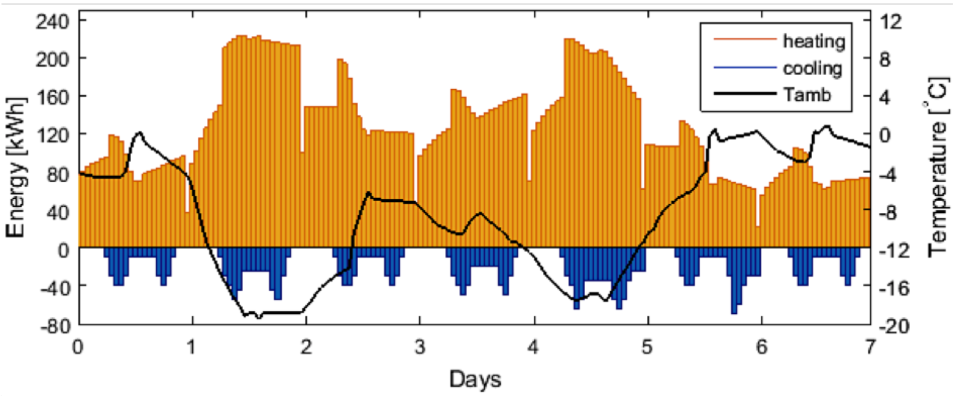


Figure 5.9: *Space heating and cooling demand with T_{amb} for winter scenario.*

The winter and fall/spring scenarios have many similarities with respect to input data and system control. The space heating demand is dominant for both scenarios and the space cooling demand pattern is identical, as can be observed. The only remarkable difference is the magnitude of the space heating demand, where the peak is 331% larger than for the fall/spring week. The average temperature for the week is -9 ^{o}C.

Summer scenario

From the summer input data, illustrated in figure 5.10, it can be observed that the space heating demand is nonexistent for the week (Aug. 14th-20th). The propane compressor displacement is therefore controlled according to the temperature of the bottom fluid layer in the ice water tank. Space cooling peaks occurs in relation to high ambient temperature, and reach a maximum of 218 kWh on the 6th day. The average temperature for the week is 17.2 ^oC.

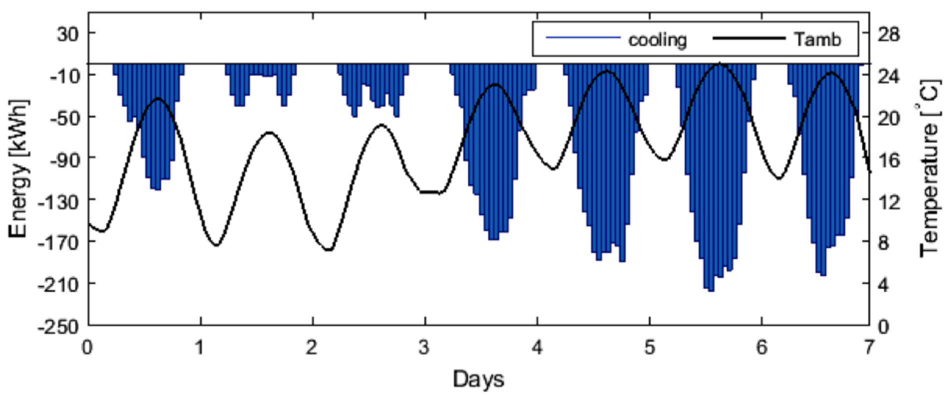


Figure 5.10: *Space heating and cooling demand with T_{amb} for summer scenario.*

Chapter 6

Results

This chapter presents the results from the simulations performed in Dymola. The main focus for the seasonal results is to review how the heat is distributed and collected within the energy system under different operational conditions. The energy consumption in relation to pumps and fans are not considered in the results, but only the work from the compressors. Additional results that are not reviewed in this chapter can be found in appendix C. A parameter study has been performed to evaluate how certain input parameters and system arrangements influence the overall performance. Due to a high degree of disturbance, the MATLAB filter *hampel* has been used to remove unwanted behavior from the results. It is important to emphasize that this does not affect the overall results in themselves. To conclude the chapter, an economical evaluation reviews the benefits of the energy system compared to conventional systems.

6.1 Seasonal operations

The energy system has been simulated for three different seasons, which are presented in this section. The results are concentrated around the fall/spring simulation, as this scenario is representative for the largest period of the year.

6.1.1 CO_2 heat pump performance

Figure 6.1 shows the CO_2 heat pump energy efficiency for all seasons. The energy efficiency (E.E.) is defined as all useful thermal contributions divided by the total compressor work within the heat pump cycle, as stated in equation 6.1

$$E.E. = \frac{H_{GCs} + H_{LP-SH} + C_{freez-rooms} + C_{ref-rooms} + C_{ice-water}}{W_{MPcomp} + W_{LPcomp} + W_{Parallelcomp}} \tag{6.1}$$

where H and C represent heating and cooling contributions, respectively.

The yearly average of 5.55 is calculated based on the average energy efficiency values for the simulated weeks, with the summer and winter results weighted at 25% each, while the fall/spring result is weighted 50%.

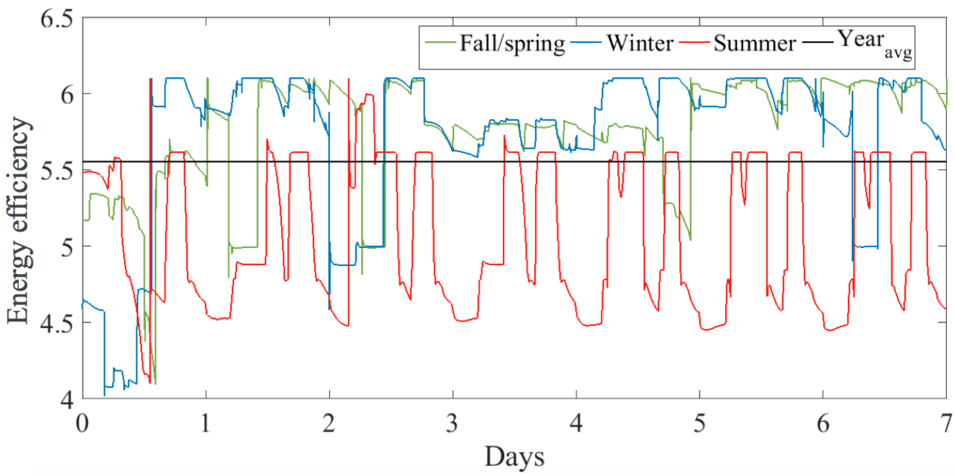


Figure 6.1: *CO_2 heat pump energy efficiency.*

The behavior of the energy efficiency is similar for the winter and fall/spring scenarios, due to the likeness in the input values presented in section 5.2.2. Intervals with highly reduced efficiency are caused by either a bypass of the second gas cooler connected to the space heating circuit, or bypass of the ice water evaporator. The result from the summer simulation shows a considerably lower energy efficiency, which is partially due to the nonexistent space heating demand. When the ice water tank is fully charged and the evaporator is bypassed, the efficiency is at its minimum of 4.5, as two contributions are eliminated (eq. 6.1). Another factor that influences the efficiency of the summer simulation is the high ambient temperature, which prevents sufficient cool down in the gas cooler before expansion. Figure 6.2 depicts the dependency between the energy efficiency and gas cooler outlet temperature. The majority of the scattered results from the fall/spring and winter simulations are placed in the upper left corner, representative for high efficiency and low CO_2 gas cooler outlet temperature. The summer scenario data accumulates in the temperature range 25-35 ^oC, yielding an efficiency of 4.5-5.5. The regression line illustrates the linear dependency between the variables within the data range. However, many values are outside the proximity of the regression line. Some of these points are a result of quick changes in the simulations, which generate system unbalance.

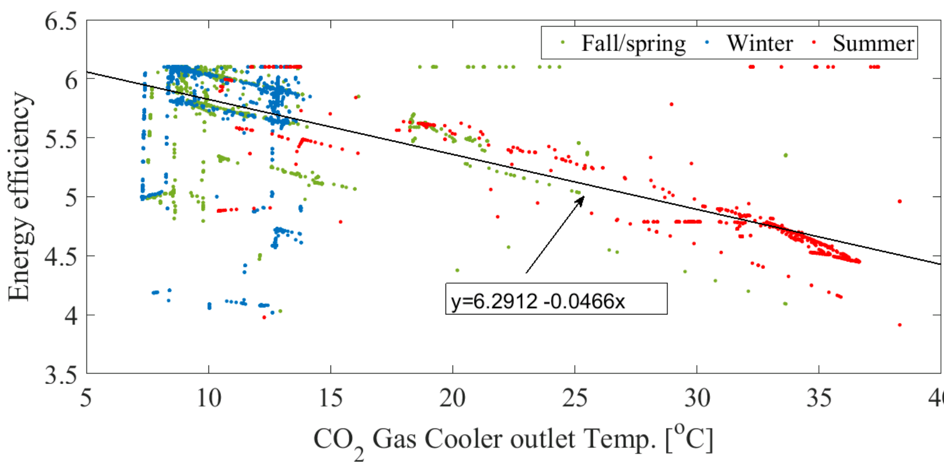


Figure 6.2: *Energy efficiency related to CO_2 gas cooler outlet temperature.*

Another explanation is that the plot does not contain any information about how the system is arranged, and does not account for the influence from activation and deactivation of different parts of the system. The CO_2 gas cooler outlet temperature is dependent on the return temperature of the domestic hot water. As explained in section 5.2.1, an ambient air heat exchanger is installed to remove heat in case the water return temperature is too high. Figure 6.3 illustrates the heat rejection to the ambient for different seasons. Naturally, the summer scenario dominates the rejection as all heat from the gas cooler is allocated to domestic hot water, independent of actual demand. The total amount of energy rejected during the weeks can be found in table 6.2.

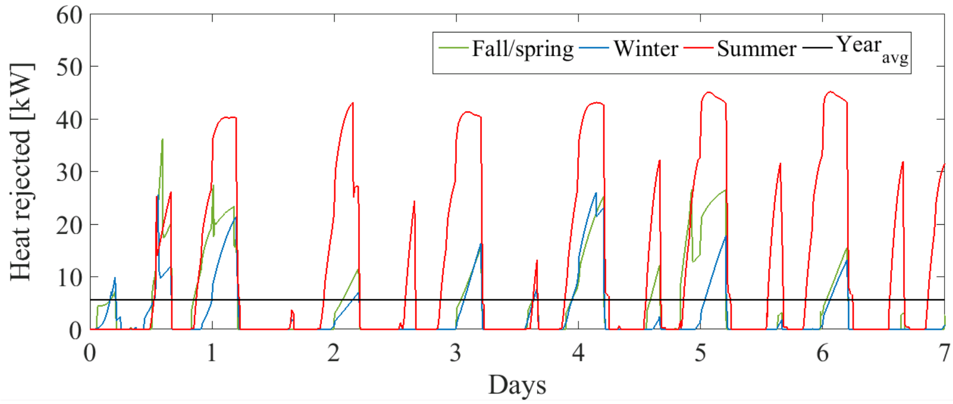


Figure 6.3: *Heat rejected from the domestic hot water curcuit for different seasons.*

It is evident that there is a relation between heat rejection and total heating demand for the hotel. The amount of energy rejected to the ambient is at its lowest for the winter simulation, which is the week of the year with largest heating demand of 446.7 kWh, or about 64 kWh per day. However small, the disregarded heat represents a loss in the system and should be assessed.

Table 6.1: *Total amount of energy rejected during the time span of a week.*

	Fall/spring	**Winter**	**Summer**	**Yearly average**
Energy rejected [kWh]	713.6	446.7	1993.9	967.0

*Winter and summer weighted 25 %, fall/spring weighted 50 %.

Due to restrictions in the control system for the model, heat rejection within the tripartite gas cooler is not optimized. As the simulation operates with a constant high pressure of 100 bar, it is within reason to assume that the heat rejected in reality would be smaller. By adjusting the high-pressure side for maximum heat rejection to the space heating circuit, one would reduce rejection during winter and fall/spring scenarios. As a larger amount heat from the CO_2 system is allocated to space heating, the need for heat supply from the propane heat pump diminishes. This would reduce the utilization of this secondary heating system. There are several other solutions that could decrease the amount of heat rejected. An increase in the domestic hot water demand would eliminate this problem entirely. The hotel has a spa treatment center that could utilize the excess domestic hot water to replace the water in the pool. Another possibility is supplying neighboring buildings with heat, which could also generate profit.

6.1.2 Domestic hot water production

Figure 6.4 depicts the stratified fluid layers in the domestic hot water tank during the fall/spring simulation. The top layer of the tank is at a relatively constant temperature of 70 ^oC for the simulated week, while the bottom fluid layer fluctuates as the tank is being charged. As the top fluid layer is required to keep the setpoint, the tank will never be at a low uniform temperature. Because of this, the stratification in the tank insures that a certain amount of useful energy is present, even when the tank is considered empty. As can be observed in the figure, such an occurrence takes place between day two and three.

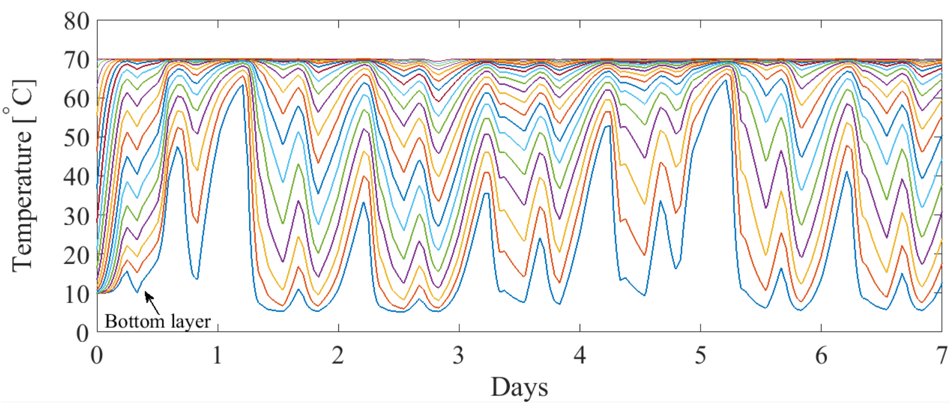


Figure 6.4: *Domestic hot water tank stratification for fall/spring simulation.*

When the fluid layer temperature drops sufficiently, the backup evaporators are activated. Figure 6.5 illustrates how the grey water and parallel loop heat exchangers are deployed to cover the domestic hot water demand. For all simulations, the parallel loop heat exchanger is only utilized at the beginning of the simulated week. This is because the domestic hot water tank is empty at the start of each simulation and needs quick charging. If not taking this circumstance into consideration, it can be questioned whether or not this evaporator is dispensable under normal operational conditions.

For the fall/spring simulation, the grey water heat exchanger is in use for about two days, between day three and five. In total, this period amounts to about 410 kWh and a constant flow of grey water that would require a storage of 20 m^3. Because of the domestic hot water consumption, drainage water would replace some of the fluid in the tank, decreasing the required volume. Roughly 18 m^3 of water at 70 ^oC is utilized during the time period. If assuming the same mixing ratio as presented in chapter 4.2.1 for delivery at 50 ^oC, this would amount to a total of 26 m^3 drainage water available for recovery. Due to the charge still present in the domestic hot water tank, the grey water heat exchanger would in reality not be deployed for such a long period of time. Nevertheless, the grey water tank must be of a certain magnitude to ensure the temperature integrity in the domestic hot water tank.

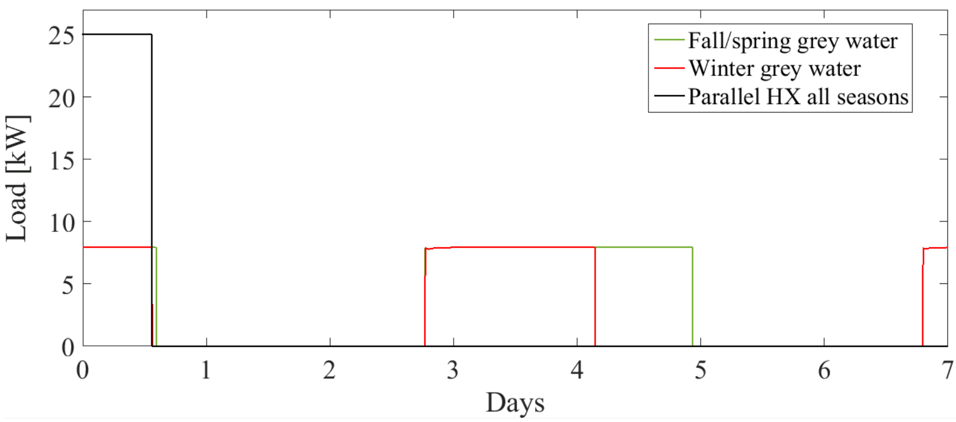


Figure 6.5: *Utilization of domestic hot water backup evaporators.*

By increasing the domestic hot water tank volume, one could increase the thermal buffer with heat that otherwise would be rejected, and thus decrease the need for grey water heat recovery. This will, however, be a trade-off between available space, investment cost and thermal buffer. The mass flow of grey water from the boundary is set to be a constant value of 0.106 kg/s, restricting the heat transfer in the heat exchanger to approximately 8 kW. Alternatively, the grey water could be employed in a different manner. A possibility is controlling the mass flow of grey water, and thus heat recovery, according to the load variations in the CO_2 mid-pressure level. If regulating the mass flow of grey water in order to stabilize the mid-pressure level, one can insure fewer variations in compressor mass flow. This will, however, require more or less continuous recovery. Propane heat pump configurations with respect to heat recovery from grey water will be evaluated in the discussion (ch. 7.2).

6.1.3 Propane heat pump performance

Figure 6.6 shows the activation and deactivation of the propane compressor during the time span of a week. Due to the possibility of employing compressors of different sizes, represented by displacement control in the simulations, the base loads for both heating and cooling are quite low.

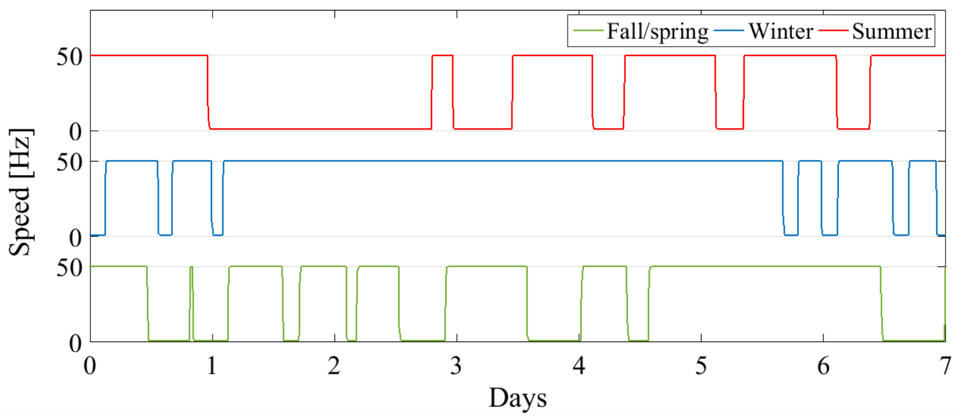


Figure 6.6: *On/off control of the propane compressor.*

The propane heat production base load of 28 kW enables a higher degree of contentious operation, as both tanks are being charged during periods with low demand. Only when there is no heating or cooling demand is the propane heat pump deactivated, as illustrated with the decision tree in chapter 5.2.1.

As can be observed from the figure, the number of stops is highest for the fall/spring scenario. This is due to the almost equally balanced heating and cooling demands, which affects the heat pump control accordingly. During the winter and summer simulations, when one demand is in vast dominance, the number of stops are limited. The propane heat pump is controlled to primarily utilize space heat and ice water before activating the air condenser or evaporator. For instance, if space heating demand is dominant, ice water is used as the primary heat source. In theory, as the ice water tank is charged contentiously, both tanks should be fully charged when the propane compressor is switched off. Even though this type of control was not fully implemented in the Dymola model, one can observe the effects as the number of compressor stops are reduced.

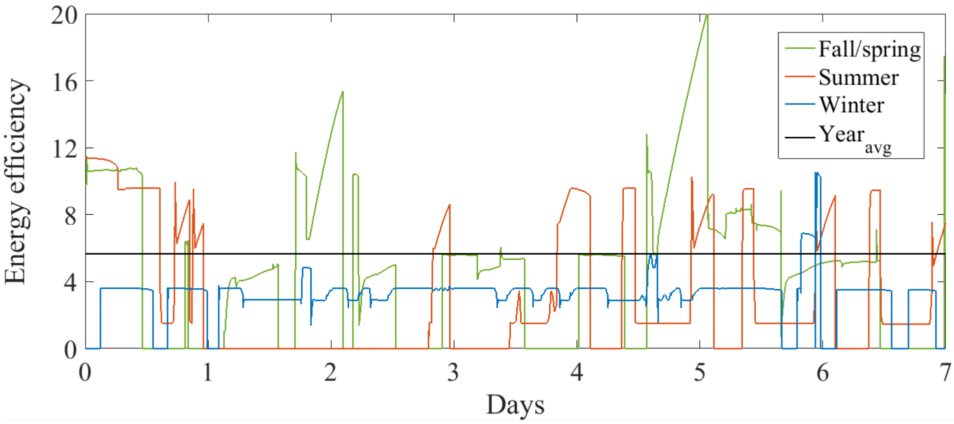


Figure 6.7: *Propane heat pump energy efficiency for all simulated seasons.*

Figure 6.7 shows the energy efficiency of the propane heat pump, which is defined as contributions to space heating and cooling divided by the compressor work, as stated in equation 6.2.

$$E.E. = \frac{H_{space} + C_{space}}{W_{Propane-comp}} \tag{6.2}$$

The energy efficiency yearly average of 5.64 is calculated in the same manner as for the CO_2 heat pump.

When both space heating and cooling heat exchangers are employed, the efficiency reaches an average maximum of around 10. During the fall/spring simulation, there are two incidences when the efficiency is well above this value. As can be observed in figure 6.8, this is caused by the low pressure difference in the heat pump, which occurs when the air condenser is in use.

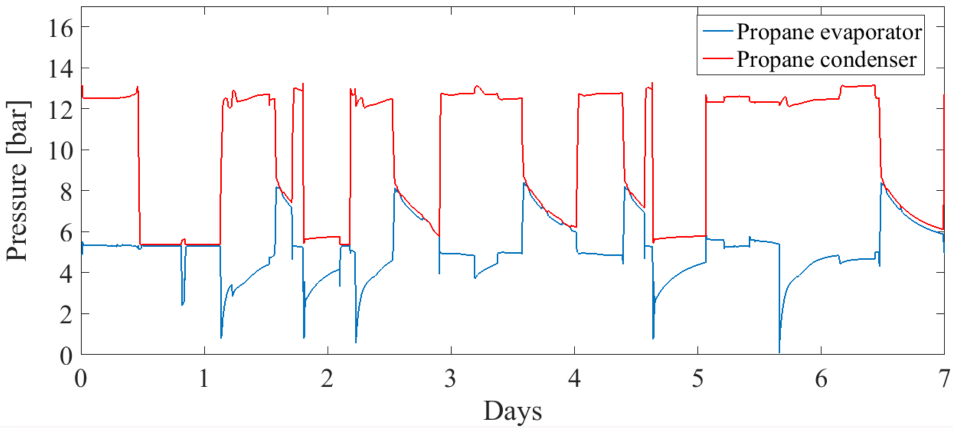


Figure 6.8: *Propane heat pump evaporating and condensing pressure for fall/spring scenario.*

The heat pump is controlled to adjust the pressure in order to reach fluid condensation at the outlet of the air evaporator, with a minimum temperature pinch in the heat exchanger. When the air condenser is activated and the ambient temperature is low, the high-pressure level is reduced. In reality, there would be a pressure regulation insuring a minimum difference between high and low pressure. In instances like this, the air evaporator fan would be turned off to reduce the heat transfer in the heat exchanger. As the convection is switched from forced to natural, a higher evaporating pressure would be induced in order to reach condensation.

6.1.4 Space heating and cooling systems

Figures 6.9 and 6.10 show the instantaneous space heating and ice water production versus the actual demand. For both cases, the load is reduced significantly due to the magnitude of thermal storage tanks. For space heating and cooling, the maximum peaks are reduced with 22.3 kW and 31.4 kW, respectively.

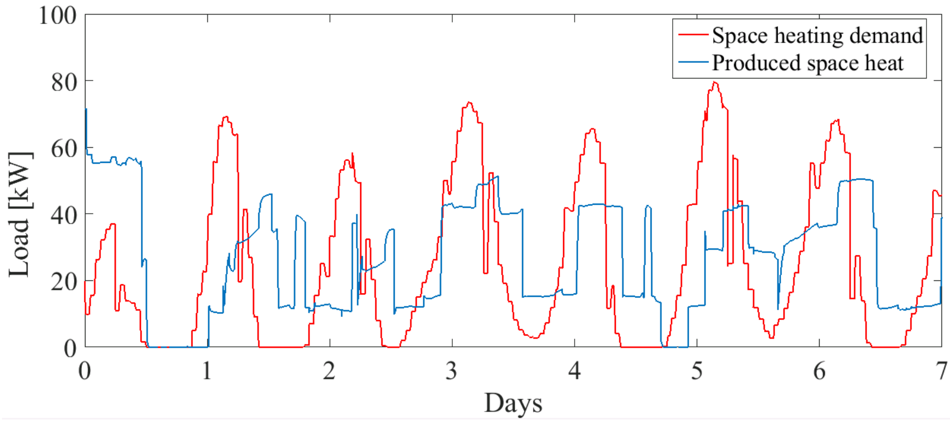


Figure 6.9: *Space heat produced versus demand for fall/spring scenario.*

The heating system peak load is delayed with a few hours each day, representing the effects of the PCM modules, as discussed in chapter 3.4.3. As the PCM modules are modeled as water volumes with the same storing capacity, the delay depicted in figure 3.6 is not shown in the tank stratification.

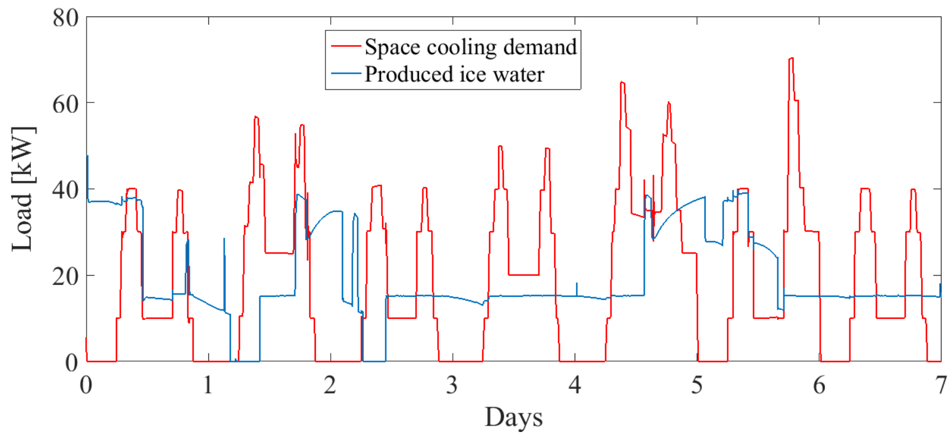


Figure 6.10: *Ice water produced versus demand for fall/spring scenario.*

The fluid layer stratification in the space heating and cooling tanks are shown in figures 6.11 and 6.12, respectively. As the space heating tank is supplied with heat from both heat pumps, the top layer temperature error is increased, which also is the case for the ice water tank bottom layer.

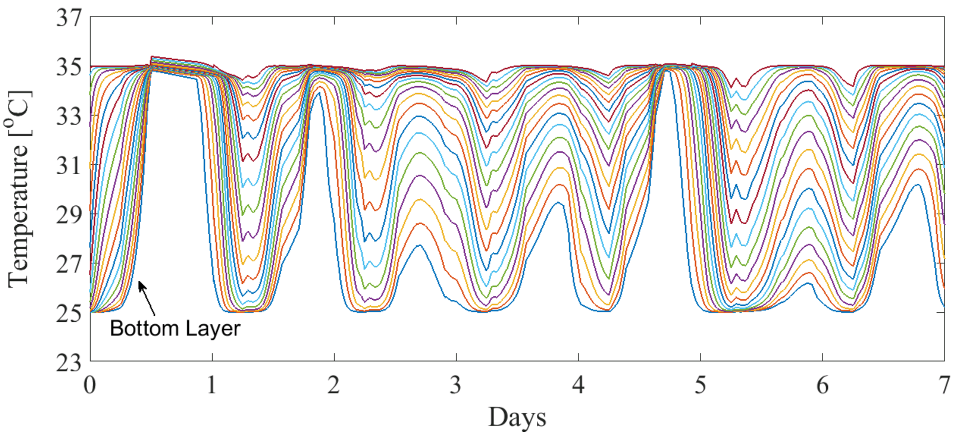


Figure 6.11: *Space heating tank fluid layer temperature for fall/spring simulation.*

Between the fourth and fifth day, the ice water bottom temperature layer increases to 9 ^oC due to limitations in propane compressor displacement for the simulated week. A certain deviation from the setpoint temperature in the tank is expected, and does not have a large effect on the temperature of the fluid delivered to the hotel building. This is due to the arrangement of the Dymola model, which enables supply of ice water directly from the heat exchangers to the AC unit. The same is true for the domestic hot water and space heating circuits.

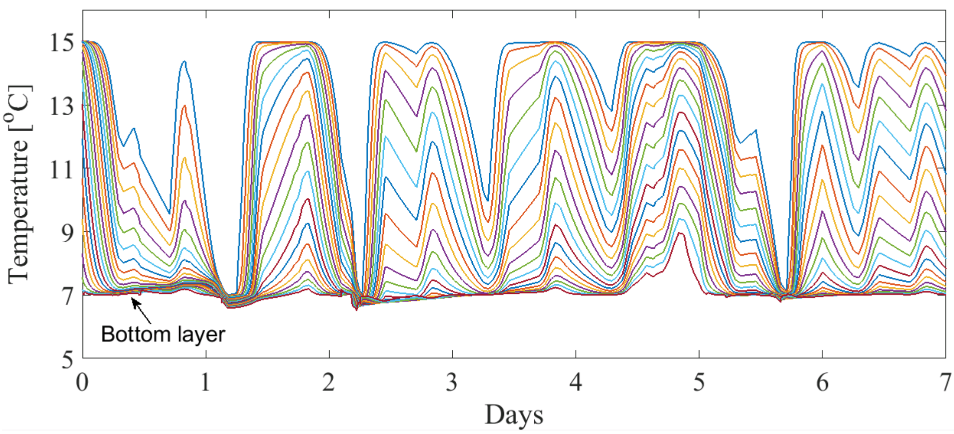


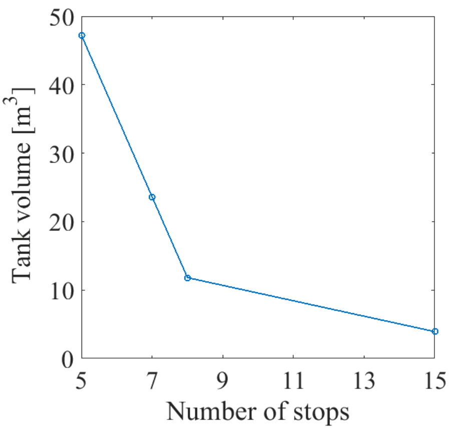
Figure 6.12: *Ice water tank fluid layer temperature for fall/spring simulation.*

6.2 Parameter study

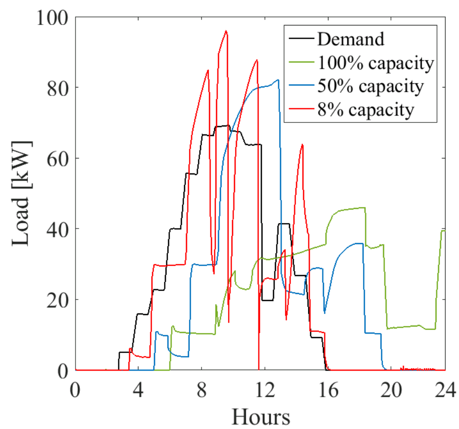
In this section, certain parameters have been investigated to show their influence on the overall system. The base case for the study is the fall/spring scenario.

6.2.1 Variation in the size of the storage tanks

In order to evaluate how the size of the space heating and cooling tanks influence the performance of the system, simulations with different storage volumes have been performed. Over a three day period, the storage volume has been reduced to 50%, 25% and 8% for both tanks. Figure 6.13a illustrates the relationship between the number of stops of the propane heat pump and the size of the space heating and cooling tanks. From this figure, it can be observed that the number of stops of the propane compressor increases with reduction in tank volume. From 100% to 25% volume, a near linear relationship exists. 15 stops occur over the three day period when the volume of the tanks are reduced to 8%. This is due to the fact that the volumes are so small that the circuits can no longer use one another as sink and source.



***(a)** Number of stops of the propane compressor over a three day period.*



***(b)** Space heat demand versus production load for different tank sizes.*

***Figure 6.13:** The influence of space heating and cooling tank volumes on system operations.*

The propane heat pump works to cover both space heating and cooling, which now function nearly independent of each other. Consequently, combined heating and cooling for thermal comfort without storage tanks will not be effective. The thermal buffers must be of a certain magnitude to provide a positive influence on the overall system.

Figure 6.13b depicts the space heating production load versus the actual demand over an interval of 24 hours. The base load case of 100% capacity is equivalent to the plot presented in figure 6.9, where the peak is considerably lower than the demand. As the sizes of the tanks decrease, the peak load required to cover the demand increases. For the 50% capacity case, the load is well above the actual space heating demand. This is due the low charge in the space heating tank, which triggers a high compressor displacement as the system attempts to cover the demand while charging the tank. The 8% capacity case illustrates the production load when there is nearly no thermal buffer. As the space heating demand is approaching its peak, the compressor overcompensates the displacement, similar to the 50% capacity case.

One major difference between the cases is that the peak production load is not delayed when the volume is reduced to 8%. As the compressor attempts to cover the load, it turns on and off sporadically. In comparison with the original case of 100% capacity, the compressors and heat exchangers would have to be dimensioned for a greater load. The energy efficiency of the system would be reduced with at least 50%, as combined heating and cooling operations are highly limited.

6.2.2 Variation in the CO_2 mid-pressure level

The current system design is made on the basis that the refrigeration rooms govern the mid-pressure level of the CO_2 heat pump. Included in this pressure level are naturally the refrigeration rooms themselves, but also the ice water and grey water evaporators. As the contributions from the ice and grey water are larger than from the refrigerated rooms, it should be evaluated if the current system arrangement can be improved.

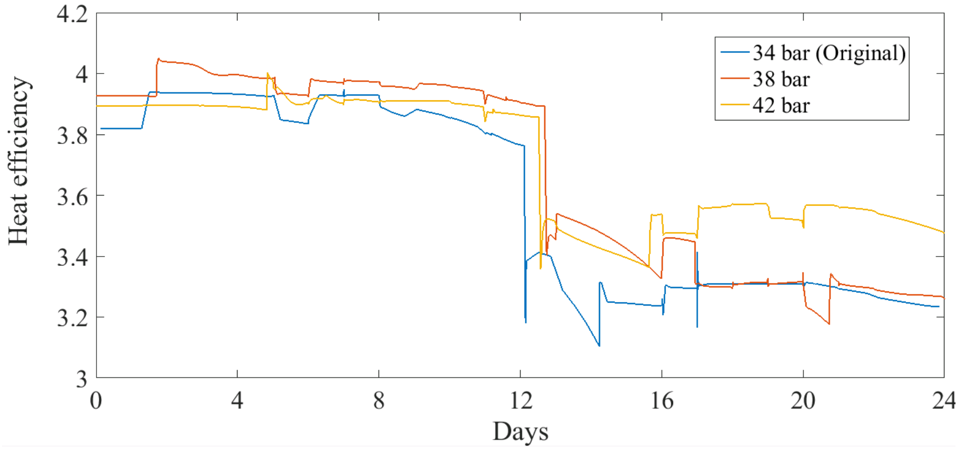


Figure 6.14: *Ice water tank fluid layer temperature for fall/spring simulation.*

Figure 6.14 shows the heat efficiency for three different mid-pressure levels. As the components in this level have been optimized with regards to the original pressure, only the effect on the heat delivered is evaluated. The heat efficiency is defined as heat delivered in the gas coolers divided by the total compressor work, as stated in equation 6.3.

$$H.E. = \frac{H_{GCs}}{W_{MPcomp} + W_{LPcomp} + W_{Parallelcomp}} \tag{6.3}$$

As can be observed in the figure, the heat efficiency increases with the pressure. This relationship is expected, as the compressor work is reduced with the decrease in pressure difference. How to interpret these results in future system design will be discussed in chapter 7.

6.3 Economical evaluation

An economical evaluation of operational costs for three different energy solutions has been performed. Table 6.2 shows the annual thermal energy budget for the hotel, where the largest contributions are allocated to domestic hot water and space heating. The annual thermal energy demand is based on the Simien model and additional demand curves.

Table 6.2: *Hotel annual thermal energy budget.*

	Annual thermal demand [kWh]
Domestic hot water at 70 ^oC	311,334
Space heating at 35 ^oC	252,546
Space cooling at 7 ^oC	174,617
Refrigerated rooms at 3 ^oC	43,800
Freezer rooms at -20 ^oC	87,600

Three different heat pump systems have been compared, where *Case 1* represents a traditional heat pump solution for a hotel. *Case 2* is an improved solution and *Case 3* is the energy system presented in this report.

CASE 1: Traditional solution where 80% of the space heat is covered by a heat pump with a COP_H of 4. The remaining 20% of the space heat, as well as 100% of domestic hot water, are covered by electric heating. All refrigeration demands are covered with separate chillers with a COP_C of 3.

CASE 2: Improved solution where separate heat pumps cover all thermal demands. 80% of the space heat and 100% domestic hot water are covered by a heat pump with a COP_H of 4. The remaining 20% of the space heat is covered with electric heating. Separate chillers with a COP_C of 3 cover refrigeration demands.

CASE 3: The purposed energy system is utilized to cover all thermal demands. The yearly average heat energy efficiency is 3.41 and 4.15 for the CO_2 and propane heat pumps, respectively. The coverage of cooling functions are assumed to be a byproduct of the heat produced. Is is assumed that 20% of the annual space heating demand is covered by the CO_2 system. Heat rejection is accounted for.

Table 6.3 shows the results from the analysis, where an average electricity cost for commercial buildings of 0.925 NOK/kWh is used in the calculations (Statistics Norway, 2017). More details on the economical evaluation can be found in appendix C.3.

Table 6.3: *Yearly operational costs and % improvement from case 1.*

	Yearly cost [NOK/year]	**Reduction in cost [%]**
Case 1	487,461	0
Case 2	249,268	48.9
Case 3	154,552	68.3

The results show that case 3, the energy system, is superior with its operational cost of 154,552 NOK/year. This is an improvement of 68% in relation to case 1, even with the generous COP_H of 4. However, expenses in relation to maintenance and electricity to additional equipment are not included. The COP and energy efficiency for all cases will deviate from the average given the season. This analysis is therefore not representative in regards to the actual expenses, but rather the trend of cost reduction.

Chapter 7

Discussion

This chapter will discuss the simulation model validity, in addition to strengths and weaknesses of the purposed system. Some discussions have already been presented with the simulation results in chapter 6.

7.1 Validity of the simulation model

The simulation results presented in the previous chapter are based on a standard year Simien simulation, with a time step of one hour. Though many aspects are taken into account in the building and Dymola models, several assumptions deviate from a real life situation, such as accounting for evaporator defrosting.

Input values and model structure

The model input values are partially based on the results from the Simien simulation. Due to limited information regarding the hotel, it is assumed that the building construction and materials are in accordance with TEK10. Whether or not this is representative for Britannia Hotel can only be determined by analyzing and comparing actual user data to the current model. However, altering the hotel building data will essentially only influence the magnitude of the different thermal loads. The load-duration pattern for the hotel (fig. 4.3) would behave in the same manner but have different values. It would therefore not be necessary to alter the current Dymola model, only the input values for the different demands. The results obtained can therefore be deemed valid for a hotel with equal building energy flows, within the boundaries of the assumptions. The assumptions made in relation to the Dymola model affects the results from the simulations in a positive manner. Pressure loss, variation in heat transfer properties and inconstant compressor efficiencies would particularly influence the system performance. Furthermore, the work in relation to fans and pumps is not taken into consideration in the results. The energy efficiency would undoubtedly be reduced if the aforementioned factors were accounted for.

Modeling of essential equipment

The heat exchangers have been dimensioned to cover their specific load with a reasonable minimum temperature pinch. Whether or not this temperature approach is appropriate for the particular heat exchanger has not been assessed in detail, and the area has not been adequately optimized with regards to load and operational conditions. Accounting for these factors would mainly influence the investment cost of the system, which is not evaluated in this report. Nevertheless, optimization of the heat exchangers would have a positive influence on the results, as certain evaporating pressure levels could increase due to a smaller heat exchanger temperature pinch.

The compressor racks are modeled as single units with variation in fluid displacement. In reality, would the larger units, such as the compressors for CO_2 mid pressure and the propane heat pump, consist of several compressors in parallel. As the load differs greatly for these compressors, a solution that provides high efficiency for both low and high loads is preferable. A possible solution is installing a rack with compressors of different sizes, which can be deployed alone or in combination.

Constant temperature boundaries

The distribution of domestic hot water, space heat and ice water for space cooling have been modeled as boundaries. For space heating and cooling, an additional heat exchange between the fluids and air has therefore been neglected. This was done in order to gain some stability in the simulations, as the temperatures of the return water streams are constant. The flow of water through the heat exchangers is controlled to meet the setpoint temperature of that particular storage tank. Greater variations in the inlet water temperature to the heat exchangers would therefore induce the same behavior in the control parameter of the PI-controllers. This assumption was therefore deemed acceptable, as the alternative would have a great impact on simulation performance. When comparing the heating demand from figures and 5.8 and 6.10, one can observe the consequence of this assumption. The heating demand is slightly larger in the simulations than for the Simien model. Correcting for this error would have a positive effect on a system with similar behavior, as discussed in chapter 6.2.1.

7.2 Design of the energy system

The proposed combined heating and cooling system for the hotel has shown potential in regards to energy efficiency. The economical evaluation revealed that the operational costs of the hotel are reduced drastically, in comparison to more conventional systems.

Load coverage

The system has been dimensioned to cover the entire heating load, as discussed in section 4.1. An argument for this solution is that the extra functions needed to achieve full coverage would be limited to increasing the size of the heat exchangers. The compressors, must after all, be able to cover the maximum space cooling demand of 220 kW. In comparison to the total investment cost for a system of this magnitude, the extra expenses in relation to the heat exchangers would in all likelihood be inconsequential. Another consideration to be made is that the space heating peaks in question occur during the winter season when alternative heat sources, such as electric and district heating, are at their most expensive. Only a thorough economical analysis can reveal if complete heating coverage is beneficial over time. However, when only considering the heat efficiencies of the system and the price district heating, the energy system proves superior. The average costs for district heating and electricity are 0.652NOK/kWh and 0.925 NOK/kWh, respectively (Statistics Norway, 2017). The heat efficiency of the energy system would therefore have to be greater than 1.5, in order to be profitable, which is true for all results presented in this report.

Evaluation of secondary heat pumping system

Propane was selected as the working fluid for the secondary heat pump, based on its environmentally friendly and convenient properties, which makes it suitable for low-pressure systems. As discussed in the previous section, the magnitude of the loads can be changed without having to alter the simulation model. It should, however, be considered replacing propane with ammonia in the future system if the heating and cooling demands prove larger than anticipated. Ammonia has superior heat transfer properties and a high enthalpy of evaporation. Due to its high volumetric heating capacity, a minimum load of approximately 50 kW is necessary, which makes it unsuitable under current conditions. Because of ammonia's classification as a B2 working fluid (Toxic, slightly flammable), strict safety measures are necessary in regards to the machine room. Changing working fluids would, however, have a minor influence on the particular arrangements surrounding the heat pump, as similar requirements must be fulfilled when using propane.

In the purposed system, presented in chapter 4.2, a propane interface to the space heating circuit was provided. The utilization of grey water as a heat source in the propane heat pump is highly possible when only considering the amount of heat available. When accounting for the water storage volume and the fact that this source is used by the CO_2 system, it is necessary to evaluate how a shared limited source would affect both systems. Additionally, a control system would have to be in place to successfully distribute the source between both systems.

Raising CO_2 mid-pressure level

Another design aspect that should be assessed is the arrangement of the CO_2 mid-pressure level. Results from the parameter study show that the efficiency increases with the pressure of this level. In the current design, the refrigeration load of 5 kW and the room temperature requirement of 3 ^oC are governing the CO_2 mid pressure. As the space heating load is decidedly larger, the mid pressure should be raised to a level where production of ice water is optimized. The refrigeration rooms could alternatively run on a separate lower pressure level with its own compressor. In addition, direct evaporation rather than by use of ice water, should be considered for large and continuous space cooling loads, such as accumulated by the commercial kitchens. Hence, a secondary energy transfer would be eliminated and ice water pump work would be reduced.

The setpoint temperatures for the space heating and cooling subsystems should be evaluated in light of the results and potential optimization of the heat pump performance. As the heating system already has a setpoint of a reasonably low temperature, a reduction is limited. Increasing the setpoint temperature of the cooling circuit to 12-15 ^oC would make it possible to raise the mid-pressure level with 10 bar, depending on the size of the ice water heat exchanger.

Reducing heat rejection
The amount of heat rejected from the energy system accounts for approximately 9% of the annual operational costs, which is a system flaw that must be assessed. As discussed in the previous chapter, proficient regulation of the high pressure enables a larger amount of heat rejection to the space heating circuit. It should, however, be discussed if employing all three gas coolers at the same time is the best strategy. Even with an adjusted high-pressure level will domestic hot water be produced continuously, as a byproduct of the chillers. An elimination of the heat rejection from this circuit can therefore not be guaranteed. A possible strategy is to only supply heat to one circuit at a time. The second gas cooler would then be bypassed during charging of the domestic hot water tanks. Only when the tanks are fully charged would the first and the third gas coolers be bypassed. All heat would be allocated to space heating when the interface to this circuit is reconnected. This procedure would repeat itself when the domestic hot water tanks are close to empty.

Improved control system
An intricate control system is necessary to optimize the operation of the energy system. The CO_2 system will attempt to cover the heating and cooling demands to the best of its ability before activating the propane heat pump. One major challenge is determining when and if it is necessary to activate the propane heat pump, in regards to both heating and cooling. This pinpoints the essence of the difficulties

surrounding the control system. The entirety of the energy system must be able to cover all demands, but at the same time, be regulated according to the heat pump configuration that provides maximum energy efficiency.

The number of variables the control system algorithm accounts for, decides the accuracy and thus efficiency of the energy system. Nevertheless, whether or not the CO_2 system is able to handle the next peak cannot solely be predicted based operational data. The heating and cooling demands in the hotel are influenced by many factors, such as weather conditions, building structure and layout, heat gain due to electrical equipment, number of occupants and their behavior. As discussed in section 3.5.2, Zhao and Magoulès (2012) investigated artificial models that could predict energy performance of buildings. If such a model is implemented in the heat pump control system, predictions would be made regarding the outcome and effect on the system behavior. Based on previous experiences, the system control could be optimized, and thus determine the right course of action related to the CO_2 and propane heat pump configuration.

Chapter 8

Conclusion

A combined heating and cooling system, consisting of a CO_2 and a propane heat pump, has been purposed for the hotel. Sufficient thermal storage was installed for domestic hot water, space heating and cooling. Several simulations were carried out using Dymola, in order to investigate the performance and potential of the energy system during different seasons. The economical aspects were not the main focus of this report, though some comparisons were made. The results from the simulations revealed a high efficiency for the energy system, as the thermal buffers make it possible to utilize the heat pumps simultaneously for both heating and refrigeration. However, the parameter study affirmed that there is still room for improvements in regards to system design. The following list presents the most important discoveries made in this report:

- The purposed system is able to cover all thermal demands for the simulated scenarios. This includes domestic hot water, space heating, space cooling and refrigeration of cold storage rooms.
- The energy efficiency for the CO_2 system proved high for both fall/spring and winter scenarios. The results from the summer scenario revealed a somewhat lower efficiency. The average annual energy efficiency for the CO_2 system was calculated to 5.55.

- There is a strong relationship between ambient temperature and CO_2 energy efficiency. For optimal heat pump performance, the CO_2 gas cooler outlet temperature should be less than 15 ^oC.
- An weekly average of 1000 kWh is rejected from the energy system, which means that about 6 kWh is wasted per hour of operation. This value will be reduced with sufficient gas cooler pressure control. Increasing the domestic hot water storage tank will reduce the amount of heat rejected and the need for grey water heat recovery.
- The magnitude of the storage tanks limit the number of stops for the propane heat pump and enables a high energy efficiency for the system. A yearly average of 5.64 was found for the propane heat pump.
- By implementing thermal storage for space heating and cooling, the loads are reduced considerably in comparison to the actual demands. The consequence of this is that peak loads are delayed with a few hours, which was the same effect expected from the PCM modules. More stable operations with a 20-30 kW peak reduction is achieved in relation to both space heating and cooling.
- The number of stops of the propane compressor increases with the reduction of space heating and cooling tank volumes. As the magnitude of the tanks decrease, the peak load necessary to cover the demand increases.
- Increasing the CO_2 mid pressure level would improve the efficiency of the system. The refrigeration rooms should be operating at a separate pressure level, while the CO_2 mid pressure is adapted to the temperature of the ice water circuit.
- The purposed energy system is more profitable in regards to operational costs, compared to more conventional hotel heat pump solutions. With an optimized energy model and a detailed economical analysis, the operational profitability of the system is expected to increase considerably.

Chapter 9

Suggestions for further work

The scope of this project is large, with its focus on combined heat pump systems, energy recovery and thermal storage. Hence, it has not been possible to thoroughly investigate all the influencing factors. The simulation model includes assumptions and approximations of uncertain magnitude, which leads to deviations in performance from a real hotel energy system. Future simulations should be based on physical data from the hotel. Sufficient input data is necessary in order to obtain results that reflect the actual demand of the hotel, which in turn will facilitate accurate dimensioning of the energy system.

Improvements on the Dymola model presented in this report should focus on eliminating assumptions regarding heat loss, pressure loss and constant heat transfer coefficients in heat exchangers. The compressors should be modeled as separate units with efficiency models in accordance with reality. The brine system should be implemented to reveal how a combined heat collection circuit will affect both heat pumps and the overall efficiency. Controlling the system has proven very difficult and placed strict limitations on performance optimization. The control system must be improved and tested under various circumstances. An optimizing algorithm accounting for all variables should be considered and further investigated.

The system design should be evaluated in the light of the findings in this report. One of the design aspects that should be assessed is the arrangement of the CO_2 mid-pressure level. The refrigeration rooms should be installed on a separate pressure level, making it possible to raise the pressure in accordance with the ice water temperature. It should also be evaluated if the temperature of the ice water could be increased. How this will affect both the efficiency of the heat pumps and the ability to provide thermal comfort must be evaluated.

A large amount of heat is rejected from the energy system, and thus represents wasted potential. It should be investigated if there are any neighboring buildings that could utilize this heat. Alternatively, the size of the domestic hot water tanks could be increased. Gas cooler pressure control to maximize the contribution to space heating should be assessed, as this would increase efficiency and reduce the amount of heat rejected. Different operational strategies of the gas cooler heat distribution should also be evaluated.

Potential savings from an economical point of view should be evaluated to reveal the long-term profitability of the system, as the economical evaluation performed in this report is limited. More work should be put into estimating both operational and investment cost in order to find the optimum solution with relation to life-cycle cost (LCC). The determination of suitable components for the heat pumping systems is thus a premise. The magnitude of the storage tanks in regards to energy efficiency and investment cost should be investigated, reveling the relationship between tank size and possible economical savings. Though the system utilizes natural working fluids, should a life-cycle assessment (LCA) be performed in order to reveal the associated environmental impact of the system.

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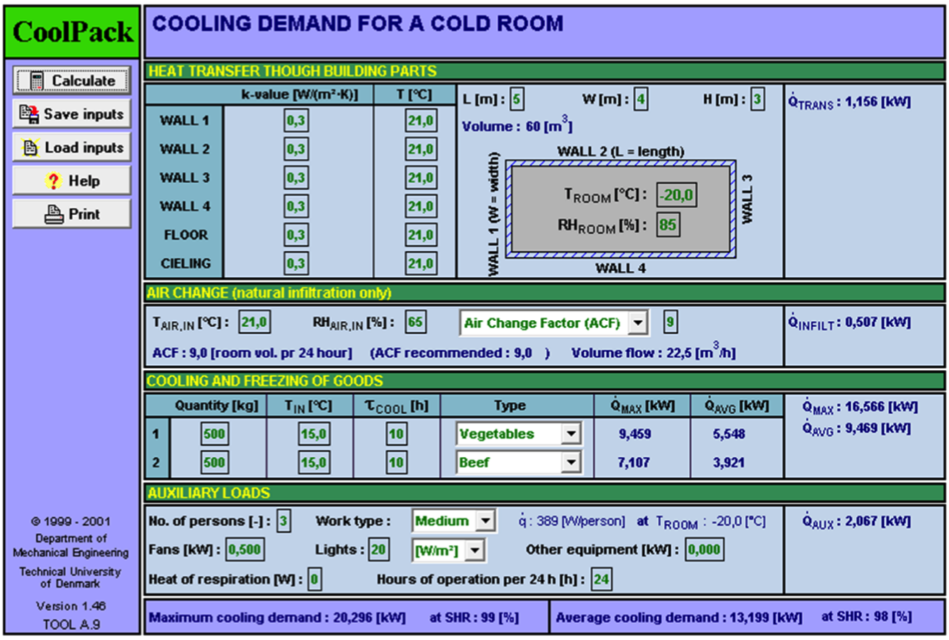
Appendix A: Hotel modeling and load calculations

A.1 CoolPack simulations for freezers, refrigerated rooms and conference room

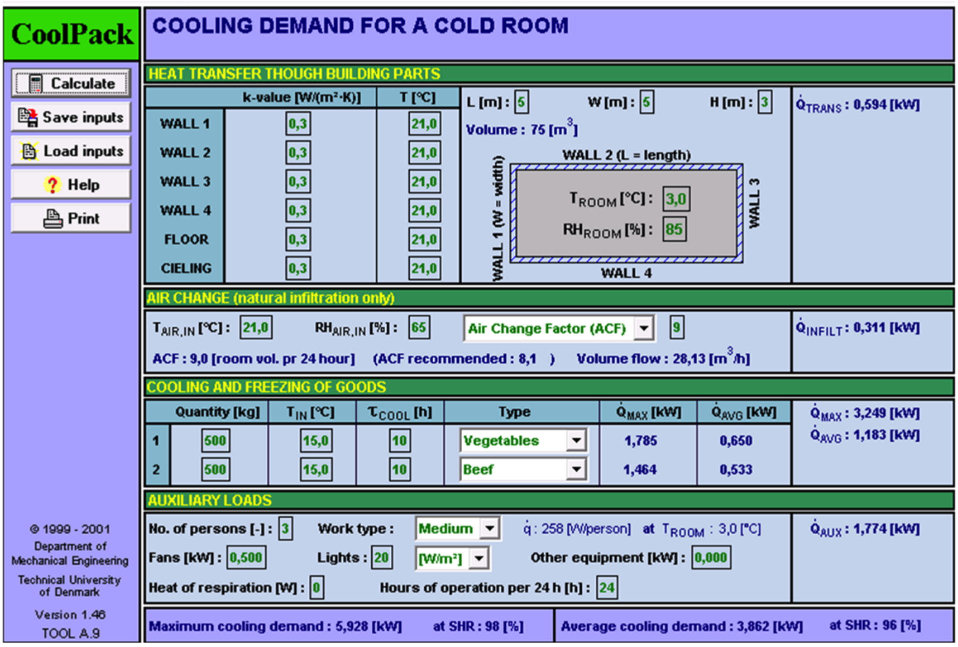
Hotel freezing load

Simulation performed in CoolPack for estimation of freezing load in the Hotel. Freezer rooms assumed to be a total of LxBxH = 5x4x3 = 60 m^3 and at a temperature of -20^oC.



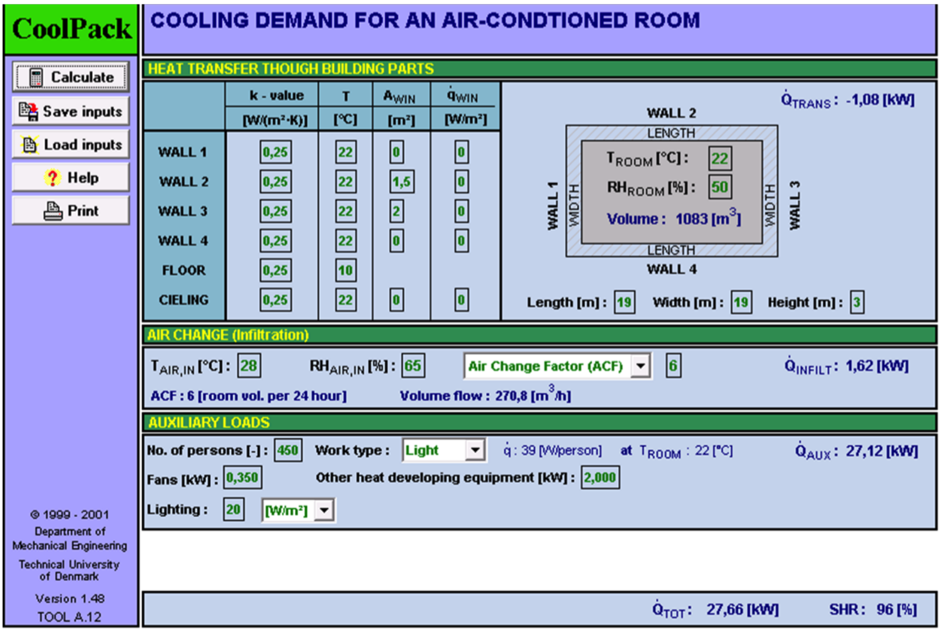
Hotel refrigeration load

Simulation performed for estimation of refrigeration load for cold storage rooms in the Hotel. Refrigeration rooms assumed to be a total of LxBxH = 5x5x3 = 75 m^3 and at a temperature of 3^oC.



Hotel conference rooms space cooling load

Simulation performed to estimate the maximum space cooling load in the conference center located in the Hotel. Conference rooms assumed to be a total of LxBxH = 19x19x3 = 1083 m^3 and at a temperature of 22^oC. Maximum load occurs when there is 450 people present.



A.2 Heat gain calculations for commercial kitchen

Hotel kitchen space cooling load

Due to difficulties regarding precise calculations of the heat gain from kitchen appliances, the overall heat gain from all four restaurants have been assumed to be an average of 30 kW during operational hours (7.5 kW each). This value is based on a heat budget for typical kitchen appliances, shown in the table below. The heat gain values are from the American Society of Heating and Engineers (2001) handbook, and is presented in appendix A.3.

Appliance	**With/Without hood [w/wo]**	**Heat gain [W]**
Oven (roasting), 0.4 m^2	w	468
Dishwasher (hood type, water sanitizing), 300 dishes/h	wo	537
Toaster (small conveyor)	wo	468
Range (hot top/fry top), 0.2 m^2	w	1720
Griddle/grill (small), 0.4 m^2	w	504
Coffee brewing urn (small), 5 L	wo	650
Hot plate (single buner, high speed)	w	2200
Steam kettle (small) 10 L	wo	350
Total heat gain per kitchen		**7479**

A.3 ASHRAE Recommended Rates of Heat Gain From Typical Commercial Cooking Appliances

Table 5 Recommended Rates of Heat Gain From Typical Commercial Cooking Appliances (*Concluded*)

		Energy Rate, W		Recommended Rate of Heat Gain,[a] W			
				Without Hood			With Hood
Appliance	Size	Rated	Standby	Sensible	Latent	Total	Sensible
Toaster (bun toasts on one side only)	1400 buns/h	1500	—	800	710	1510	480
Toaster (large conveyor)	720 slices/h	3200	—	850	750	1600	510
Toaster (small conveyor)	360 slices/h	2100	—	560	490	1050	340
Toaster (large pop-up)	10 slice	5300	—	2810	2490	5300	1700
Toaster (small pop-up)	4 slice	2470	—	1310	1160	2470	790
Waffle iron	0.05 m^2	1640	—	700	940	1640	520
Electric, Exhaust Hood Required							
Broiler (conveyor infrared), per square metre of cooking area	0.19 to 9.5 m^2	60800	—	—	—	—	12100
Broiler (single deck infrared), per square metre of broiling area	0.24 to 0.91 m^2	34200	—	—	—	—	6780
Charbroiler, per linear metre of cooking surface	0.6 to 2.4 m	10600	8900	—	—	—	2700
Fryer (deep fat)	15 to 23 kg oil	14000	850	—	—	—	350
Fryer (pressurized), per kilogram of fat capacity	6 to 15 kg	1010	—	—	—	—	38
Griddle, per linear metre of cooking surface	0.6 to 2.4 m	18800	3000	—	—	—	1350
Oven (full-szie convection)		12000	5000	—	—	—	850
Oven (large deck baking with 15.2 m^3 decks), per cubic metre of oven spacer	0.43 to 1.3 m^3	17300	—	—	—	—	710
Oven (roasting), per cubic metre of oven space	0.22 to 0.66 m^3	28300	—	—	—	—	1170
Oven (small convection), per cubic metre of oven space	0.04 to 0.15 m^3	107000	—	—	—	—	1520
Oven (small deck baking with 7.7 m^3 decks), per cubic metre of oven space	0.22 to 0.66 m^3	28700	—	—	—	—	1170
Open range (top), per 2 element section	2 to 10 elements	4100	1350	—	—	—	620
Range (hot top/fry top), per square metre of cooking surface	0.36 to 0.74 m^2	22900	—	—	—	—	8500
Range (oven section), per cubic metre of oven space	0.12 to 0.32 m^3	40600	—	—	—	—	1660
Gas, No Hood Required							
Broiler, per square metre of broiling area	0.25	46600	190[b]	16800	9030	25830	3840
Cheese melter, per square metre of cooking surface	0.23 to 0.47	32500	190[b]	11600	3400	15000	2680
Dishwasher (hood type, chemical sanitizing), per 100 dishes/h	950 to 2000 dishes/h	510	190[b]	150	59	209	67
Dishwasher (hood type, water sanitizing), per 100 dishes/h	950 to 2000 dishes/h	510	190[b]	170	64	234	73
Dishwasher (conveyor type, chemical sanitizing), per 100 dishes/h	5000 to 9000 dishes/h	400	190[b]	97	21	118	38
Dishwasher (conveyor type, water sanitizing), per 100 dishes/h	5000 to 9000 dishes/h	400	190[b]	110	23	133	41
Griddle/grill (large), per square metre of cooking surface	0.43 to 1.1 m^2	53600	1040	3600	1930	5530	1450
Griddle/grill (small), per square metre of cooking surface	0.23 to 0.42 m^2	45400	1040	3050	1610	4660	1260
Hot plate	2 burners	5630	390[b]	3430	1020	4450	1000
Oven (pizza), per square metre of hearth	0.59 to 1.2 m^2	14900	190[b]	1970	690	2660	270
Gas, Exhaust Hood Required							
Braising pan, per litre of capacity	102 to 133 L	3050	190[b]	—	—	—	750
Broiler, per square metre of broiling area	0.34 to 0.36 m^3	68900	1660	—	—	—	5690
Broiler (large conveyor, infrared), per square metre of cooking area/minute	0.19 to 9.5 m^2	162000	6270	—	—	—	16900
Broiler (standard infrared), per square metre of broiling area	0.22 to 0.87 m^2	61300	1660	—	—	—	5040
Charbroiler (large), per linear metre of cooking area	0.6 to 2.4 m	34600	21000	—	—	—	3650
Fryer (deep fat)	15 to 23 kg	23500	1640	—	—	—	560
Oven (bake deck), per cubic metre of oven space	0.15 to 0.46 m^3	79400	190[b]	—	—	—	1450
Griddle, per linear metre of cooling surface	0.6 to 2.4 m	24000	6060	—	—	—	1540
Oven (full-size convection)		20500	8600	—	—	—	1670
Oven (pizza), per square metre of oven hearth	0.86 to 2.4 m^2	22800	190[b]	—	—	—	410
Oven (roasting), per cubic metre of oven space	0.26 to 0.79 m^3	44500	190[b]	—	—	—	800
Oven (twin bake deck), per cubic metre of oven space	0.31 to 0.61 m^3	45400	190[b]	—	—	—	810
Range (burners), per 2 burner section	2 to 10 burners	9840	390	—	—	—	1930
Range (hot top or fry top), per square metre of cooking surface	0.26 to 0.74 m^3	37200	1040	—	—	—	10700
Range (large stock pot)	3 burners	29300	580	—	—	—	5740
Range (small stock pot)	2 burners	11700	390	—	—	—	2290
Range top, open burner (per 2 element section)	2 to 6 elements	11700	4000	—	—	—	640
Steam							
Compartment steamer, per kilogram of food capacity/h	21 to 204 kg	180	—	14	9	23	7
Dishwasher (hood type, chemical sanitizing), per 100 dishes/h	950 to 2000 dishes/h	920	—	260	110	370	120
Dishwasher (hood type, water sanitizing), per 100 dishes/h	950 to 2000 dishes/h	920	—	290	120	410	130
Dishwasher (conveyor, chemical sanitizing), per 100 dishes/h	5000 to 9000 dishes/h	350	—	41	97	138	44
Dishwasher (conveyor, water sanitizing), per 100 dishes/h	5000 to 9000 dishes/h	350	—	44	108	152	50
Steam kettle, per litre of capacity	12 to 30 L	160	—	12	8	20	6

Sources: Alereza and Breen (1984), Fisher (1998).
[a]In some cases, heat gain data are given per unit of capacity. In those cases, the heat gain is calculated by: q = (recommended heat gain per unit of capacity) × (capacity)
[b]Standby input rating is given for entire appliance regardless of size.

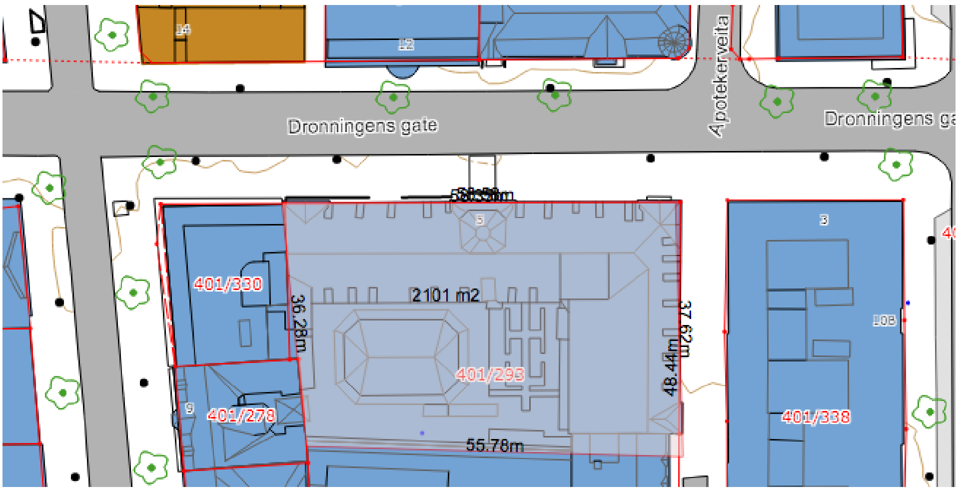
Table 5 Recommended Rates of Heat Gain From Typical Commercial Cooking Appliances

		Energy Rate, W		Recommended Rate of Heat Gain,[a] W			
				Without Hood			With Hood
Appliance	Size	Rated	Standby	Sensible	Latent	Total	Sensible
Electric, No Hood Required							
Barbeque (pit), per kilogram of food capacity	36 to 136 kg	88	—	57	31	88	27
Barbeque (pressurized) per kilogram of food capacity	20 kg	210	—	71	35	106	33
Blender, per litre of capacity	1.0 to 3.8 L	480	—	310	160	470	150
Braising pan, per litre of capacity	102 to 133 L	110	—	55	29	84	40
Cabinet (large hot holding)	0.46 to 0.49 m^3	2080	—	180	100	280	85
Cabinet (large hot serving)	1.06 to 1.15 m^3	2000	—	180	90	270	82
Cabinet (large proofing)	0.45 to 0.48 m^3	2030	—	180	90	270	82
Cabinet (small hot holding)	0.09 to 0.18 m^3	900	—	80	40	120	37
Cabinet (very hot holding)	0.49 m^3	6150	—	550	280	830	250
Can opener		170	—	170	—	170	0
Coffee brewer	12 cup/2 brnrs	1660	—	1100	560	1660	530
Coffee heater, per boiling burner	1 to 2 brnrs	670	—	440	230	670	210
Coffee heater, per warming burner	1 to 2 brnrs	100	—	66	34	100	32
Coffee/hot water boiling urn, per litre of capacity	11 L	120	—	79	41	120	38
Coffee brewing urn (large), per litre of capacity	22 to 38 L	660	—	440	220	660	210
Coffee brewing urn (small), per litre of capacity	10 L	420	—	280	140	420	130
Cutter (large)	460 mm bowl	750	—	750	—	750	0
Cutter (small)	360 mm bowl	370	—	370	—	370	0
Cutter and mixer (large)	28 to 45 L	3730	—	3730	—	3730	0
Dishwasher (hood type, chemical sanitizing), per 100 dishes/h	950 to 2000 dishes/h	380	—	50	110	160	50
Dishwasher (hood type, water sanitizing), per 100 dishes/h	950 to 2000 dishes/h	380	—	56	123	179	56
Dishwasher (conveyor type, chemical sanitizing), per 100 dishes/h	5000 to 9000 dishes/h	340	—	41	97	138	44
Dishwasher (conveyor type, water sanitizing), per 100 dishes/h	5000 to 9000 dishes/h	340	—	44	108	152	50
Display case (refrigerated), per cubic metre of interior	0.17 to 1.9 m^3	1590	—	640	0	640	0
Dough roller (large)	2 rollers	1610	—	1610	—	1610	0
Dough roller (small)	1 roller	460	—	460	—	460	0
Egg cooker	12 eggs	1800	—	850	570	1420	460
Food processor	2.3 L	520	—	520	—	520	0
Food warmer (infrared bulb), per lamp	1 to 6 bulbs	250	—	250	—	250	250
Food warmer (shelf type), per square metre of surface	0.28 to 0.84 m^3	2930	—	2330	600	2930	820
Food warmer (infrared tube), per metre of length	1.0 to 2.1 m	950	—	950	—	950	950
Food warmer (well type), per cubic metre of well	20 to 70 L	37400	—	12400	6360	18760	6000
Freezer (large)	2.07 m^3	1340	—	540	—	540	0
Freezer (small)	0.51 m^3	810	—	320	—	320	0
Griddle/grill (large), per square metre of cooking surface	0.43 to 1.1 m^2	29000	—	1940	1080	3020	1080
Griddle/grill (small), per square metre of cooking surface	0.20 to 0.42 m^2	26200	—	1720	970	2690	940
Hot dog broiler	48 to 56 hot dogs	1160	—	100	50	150	48
Hot plate (double burner, high speed)		4900	—	2290	1590	3880	1830
Hot plate (double burner stockpot)		4000	—	1870	1300	3170	1490
Hot plate (single burner, high speed)		2800	—	1310	910	2220	1040
Hot water urn (large), per litre of capacity	53 L	130	—	50	16	66	21
Hot water urn (small), per litre of capacity	7.6 L	230	—	87	30	117	37
Ice maker (large)	100 kg/day	1090	—	2730	—	2730	0
Ice maker (small)	50 kg/day	750	—	1880	—	1880	0
Microwave oven (heavy duty, commercial)	20 L	2630	—	2630	—	2630	0
Microwave oven (residential type)	30 L	600 to 1400	—	600 to 1400	—	600 to 1400	0
Mixer (large), per litre of capacity	77 L	29	—	29	—	29	0
Mixer (small), per litre of capacity	11 to 72 L	15	—	15	—	15	0
Press cooker (hamburger)	300 patties/h	2200	—	1450	750	2200	700
Refrigerator (large), per cubic metre of interior space	0.71 to 2.1 m^3	780	—	310	—	310	0
Refrigerator (small) per cubic metre of interior space	0.17 to 0.71 m^3	1730	—	690	—	690	0
Rotisserie	300 hamburgers/h	3200	—	2110	1090	3200	1020
Serving cart (hot), per cubic metre of well	50 to 90 L	21200	—	7060	3530	10590	3390
Serving drawer (large)	252 to 336 dinner rolls	1100	—	140	10	150	45
Serving drawer (small)	84 to 168 dinner rolls	800	—	100	10	110	33
Skillet (tilting), per litre of capacity	45 to 125 L	180	—	90	50	140	66
Slicer, per square metre of slicing carriage	0.06 to 0.09 m^2	2150	—	2150	—	2150	680
Soup cooker, per litre of well	7 to 11 L	130	—	45	24	69	21
Steam cooker, per cubic metre of compartment	30 to 60 L	214000	—	17000	10900	27900	8120
Steam kettle (large), per litre of capacity	76 to 300 L	95	—	7	5	12	4
Steam kettle (small), per litre of capacity	23 to 45 L	260	—	21	14	35	10
Syrup warmer, per litre of capacity	11 L	87	—	29	16	45	14

A.4 Simien simulation: Relevant input values

The simulations were conducted in Simien, in order to accumulate a reasonable load curve for the hotel building, which is of older construction. The building is simulated as a single zone for the three main floors and an additional zone for the basement. Results and relevant input data from the simulations are presented in this appendix. Due to limited data regarding the building, assumptions have been made based on TEK10.

Trondheim county map of Britannia Hotel



Climate data

Variable	**Value**
Location	Trondheim
Latitude	63^o 30’
Longitude	10^o 22’
Time zone	GMT + 1
Normalized year average temperature [C^o]	5.1
Sun radiation horizontal surfaces [W/m^2]	102
Normalized wind speed [m/s]	4.6

Energy system operational strategy

Variable	Value
Space heating setpoint temperature [C^o]	21.0
Space cooling setpoint temperature [C^o]	22.0
Operation hours heating system [h]	24
Operation hours cooling system [h]	24
Operation hours ventilation system [h]	16
Operation hours lighting [h]	16
Operation hours equipment [h]	16
Residence time people [h]	24

Energy system specification

Variable	Value
Ventilation heat recovery temperature efficiency [%]	80
Specific fan power (SFP) [$kW/m^2/s$]	2.00
Air flow during operational hours [m^3/hm^2]	10.00
Air flow outside operational hours [m^3/hm^2]	5.00
Heat generation efficiency [%]	90
Room radiators efficiency [%]	85
Installed power heating batteries [W/m^3]	80
Installed power cooling batteries [W/m^3]	70
Power factor heating/cooling system [-]	2.50
Efficiency heating/cooling batteries [%]	88

Energy system specification

Variable	Value
Lighting power and heat gain during operational hours [W/m^2]	8.00
Equipment power and heat gain during operational hours [W/m^2]	1.00
Heat gain from people [W/m^2]	2.00

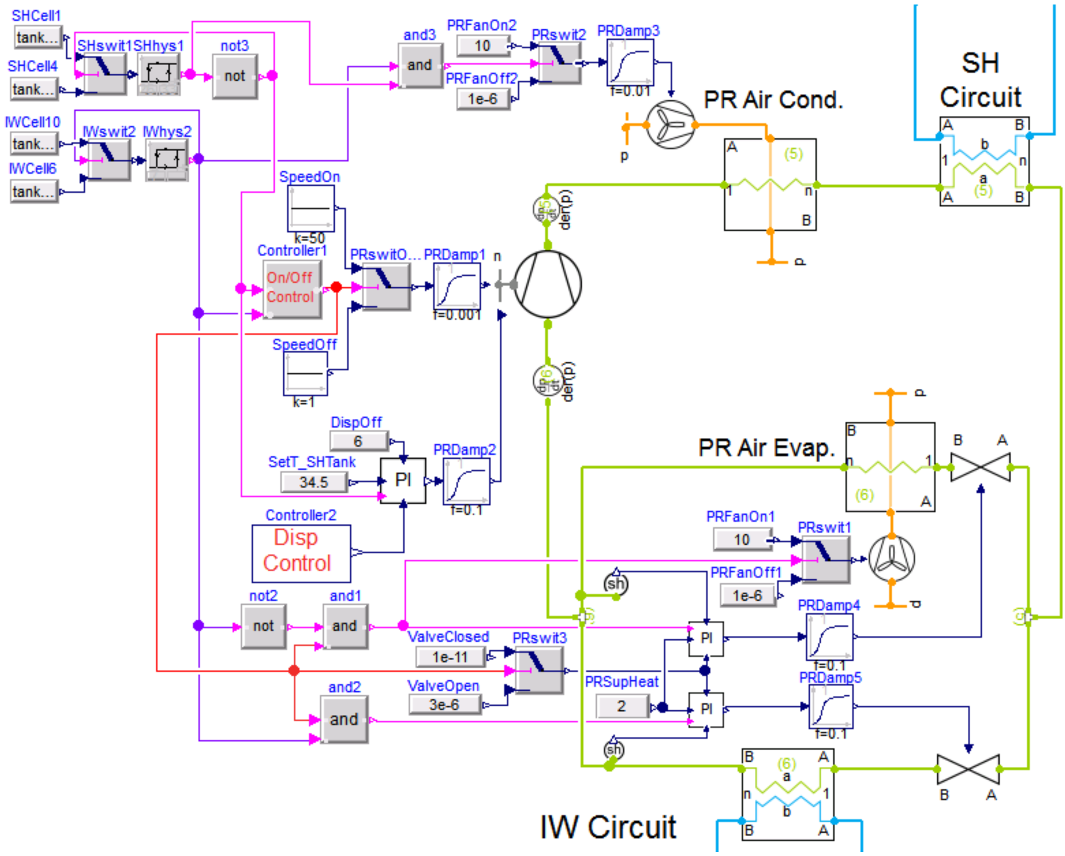
Building structure specifications

Variable	**Value**
Exterior wall area [m^2]	1015
Ground floor area [m^2]	2100
Roof area [m^2]	2100
Number of floors [-]	4
Heated floor area [m^2]	8400
Heated building air volume [m^3]	25200
U-value exterior walls [W/m^2K]	0.17
U-value roof [W/m^2K]	0.13
U-value floor to ambient [W/m^2K]	0.15
U-value glass/windows/doors [W/m^2K]	1.20
Combined glass, window and door area divided by floor area [%]	4.8
Normalized thermal bridge value [W/m^2K]	0.06
Normalized heat capacity [Wh/m^2K]	104
Leakage value (air density at ΔP= 50 Pa) [Air change pr. h]	1.50
Sun factor for windows [-]	0.75
Average window sill factor [-]	0.20
Shading factor windows (North/East/West/South) [-]	0.60/1.00/0.64/1.00

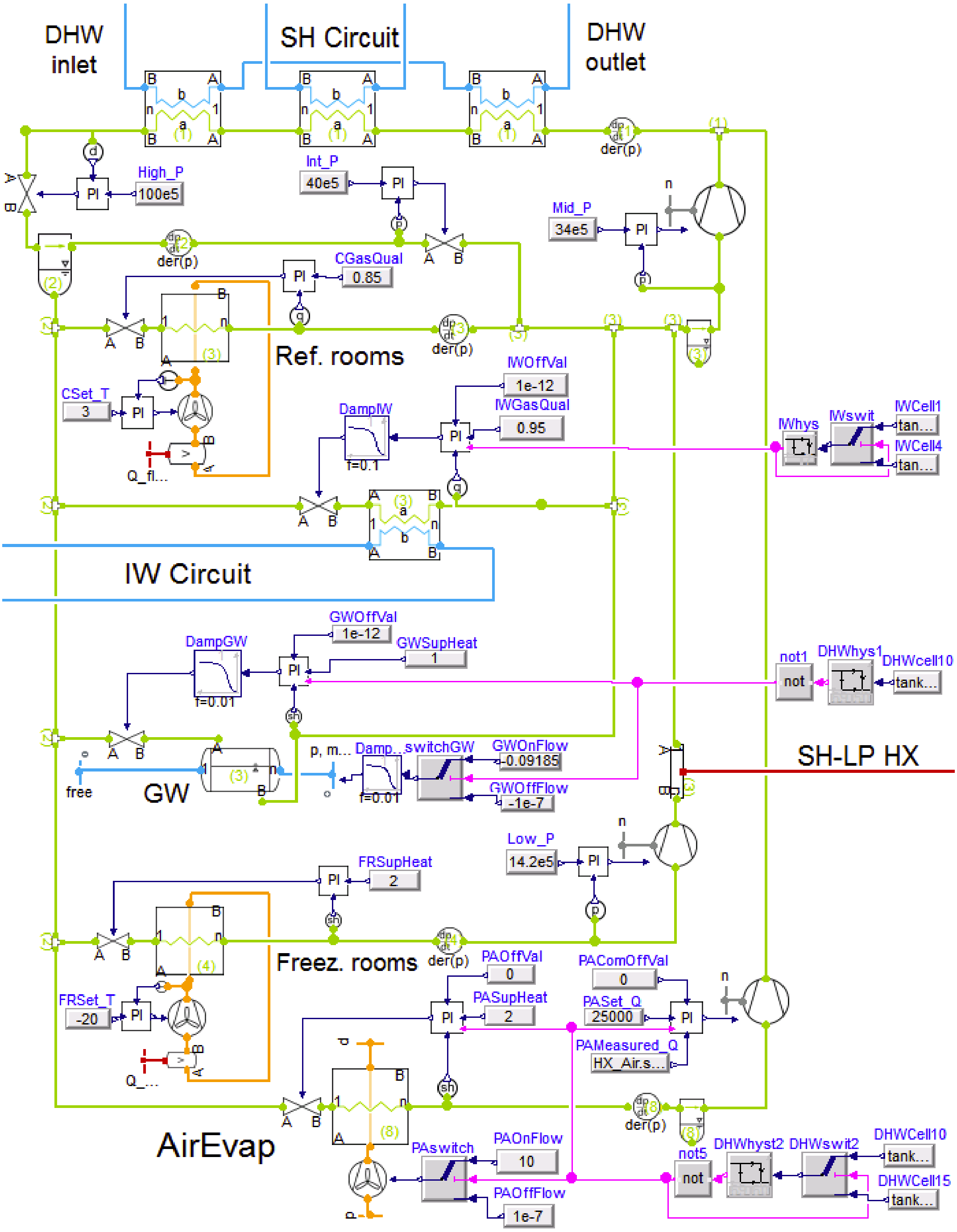
Appendix B: Dymola energy model

B.1 Complete Dymola model and control logic

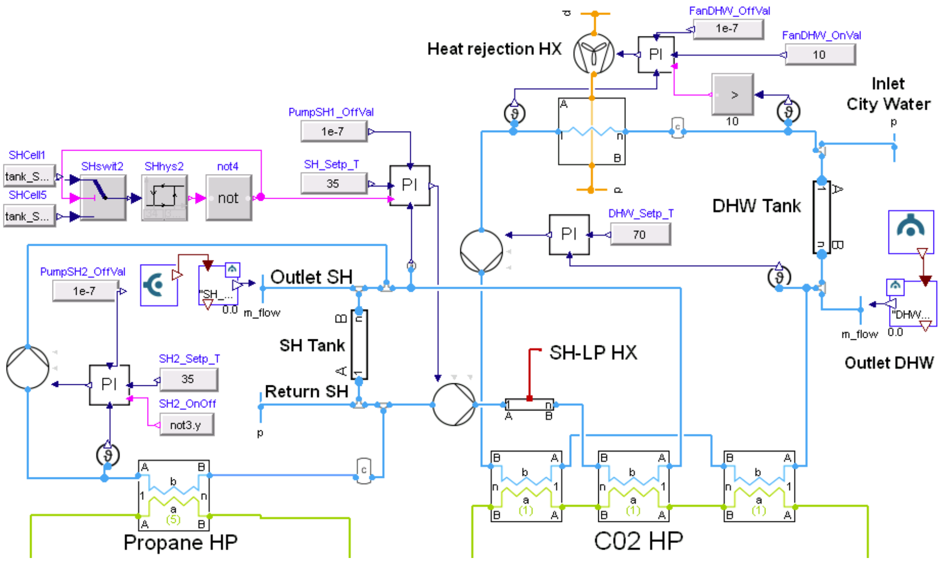
Propane system



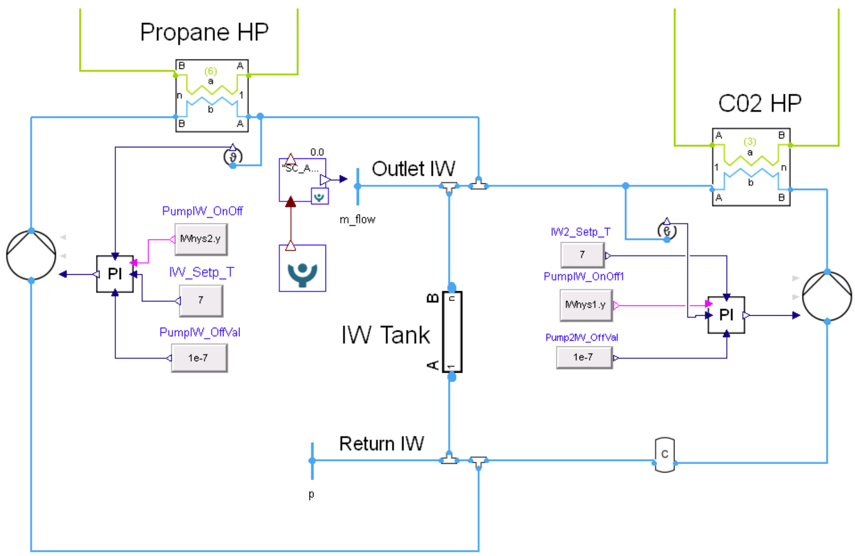
CO_2 system


DHW inlet
SH Circuit
DHW outlet
High_P
100e5
Int_P
40e5
Mid_P
34e5
CGasQual
0.85
der(p)
Ref. rooms
IWOffVal
1e-12
IWGasQual
0.95
DampIW
f=0.1
CSet_T
3
IWhys
IWswit
IWCell1
IWCell4
IW Circuit
GWOffVal
1e-12
GWSupHeat
1
DampGW
f=0.01
not1
DHWhys1
DHWcell10
SH-LP HX
free
GW
p, m...
Damp...
switchGW
GWOnFlow
-0.09185
GWOffFlow
-1e-7
Low_P
14.2e5
FRSupHeat
2
Freez. rooms
PAOffVal
0
PAComOffVal
0
FRSet_T
-20
PASupHeat
2
PASet_Q
25000
PAMeasured_Q
HX_Air.s...
AirEvap
PAswitch
PAOnFlow
10
PAOffFlow
1e-7
not5
DHWhyst2
DHWswit2
DHWCell10
DHWCell15

DHW and space heating system



Space cooling system



B.2 Input values heat exchangers

Plate heat exchangers

	CO_2-Ice water HX	Propane-Ice water HX	Propane-Space heating HX	1st Gas cooler	2nd Gas cooler	3rd Gas cooler
Number of plates	80	100	300	144	144	144
Length [m]	0.3	0.3	0.4	0.6	0.6	0.6
Width [m]	0.1	0.2	0.25	0.2	0.2	0.2
Pattern angle [deg]	35	30	30	30	30	30
Wall thickness [m]	0.75e-3	0.9e-3	0.9e-3	0.9e-3	0.9e-3	0.9e-3
Pattern amplitude [m]	2e-3	2e-3	1.1e-3	1.1e-3	1.1e-3	1.1e-3
Pattern wave length [m]	12.6e-3	12.6e-3	12.6e-3	12.6e-3	12.6e-3	12.6e-3

Specifications grey water shell-and-tube heat exchanger

Number total tubes	50
Shell inner diameter [m]	0.4
Top clearance [m]	0.01
Bottom clearance [m]	0.01
Tube outer diameter [m]	10e-3
Tube wall thickness [m]	1.5e-3
Tube active length [m]	1.5

Specifications fin-and-tube heat exchanger (KKI001)

	Freezers HX	Refrigeration room HX	CO_2 backup air evaporator	DHW heat rejection HX	Propane air evaporator	Propane air condenser
Finned tube length [m]	0.6	0.6	2	0.6	2	2
Number serial tubes	8	14	10	20	12	12
Number parallel tubes	10	14	25	20	30	30
Serial tube distance [m]	22e-3	22e-3	22e-3	22e-3	22e-3	22e-3
Parallel tube distance [m]	25.4e-3	25.4e-3	25.4e-3	25.4e-3	25.4e-3	25.4e-3
Fin thickness [m]	0.2e-3	0.2e-3	0.2e-3	0.2e-3	0.2e-3	0.2e-3
Fin pitch [m]	2.2e-3	2.2e-3	2.2e-3	2.2e-3	2.2e-3	2.2e-3
Tube inner diameter [m]	7e-3	7e-3	7e-3	7e-3	7e-3	7e-3
Tube wall thickness [m]	1.5e-3	1.5e-3	1.5e-3	1.5e-3	1.5e-3	1.5e-3
Number parallel tube side flow	4	4	3	4	4	4

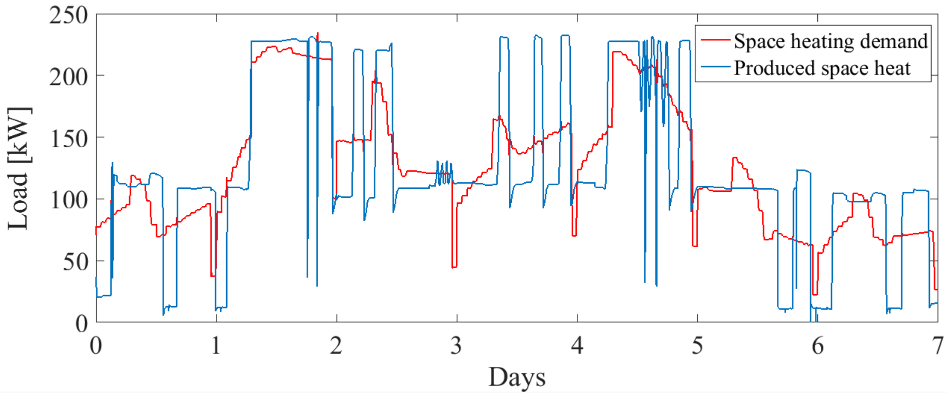
B.3 Input values compressors

Specifications efficiency compressor

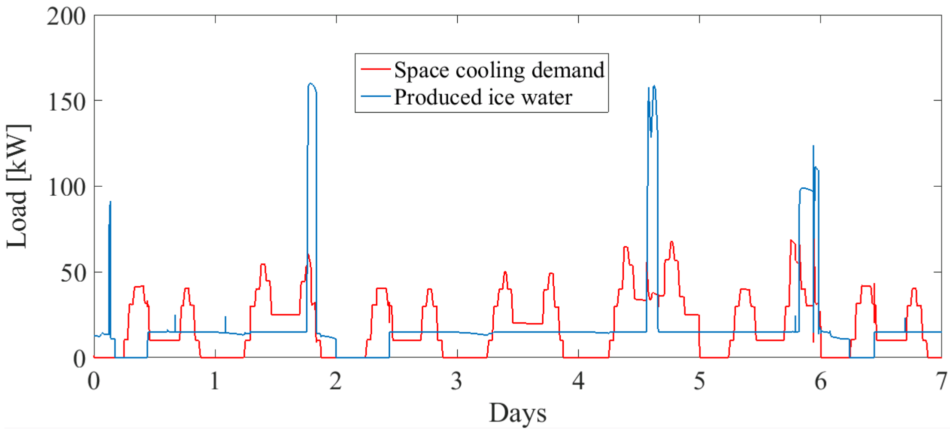
	CO_2 MT compressor	CO_2 LT compressor	CO_2 parallel compressor	Propane compressor
n. Fixed (Speed) [Hz]	50	50	50	50
Displacement [m^3]	2e-5	2e-5	2e-5	20e-5
Relative displacement operational range	1 - 10	1 - 1.5	1 - 2	1 - 18
Volumetric efficiency [%]	70	70	70	70
Isentropic efficiency [%]	70	70	70	70
Effective isentropic efficiency [%]	70	70	70	70

Appendix C: Additional results

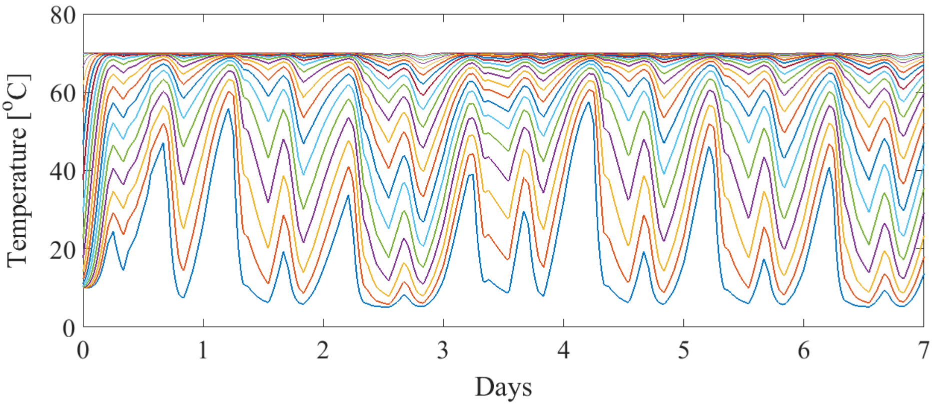
C.1 Winter scenario



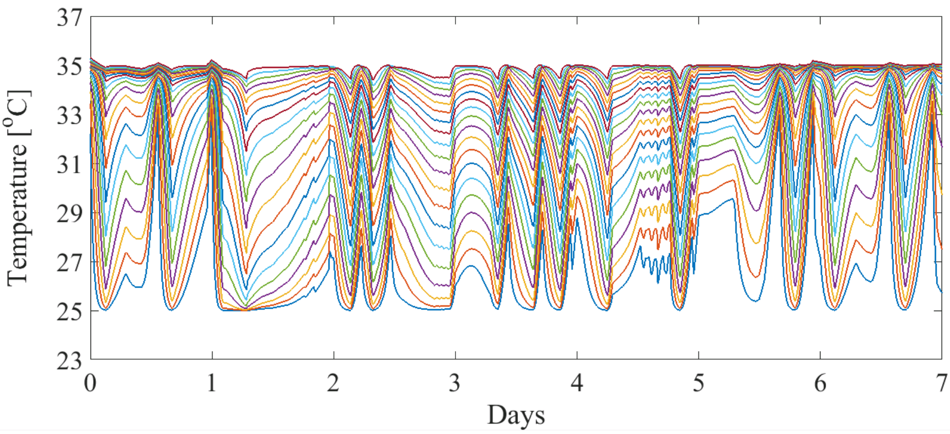
Space heating demand vs. produced space heat.



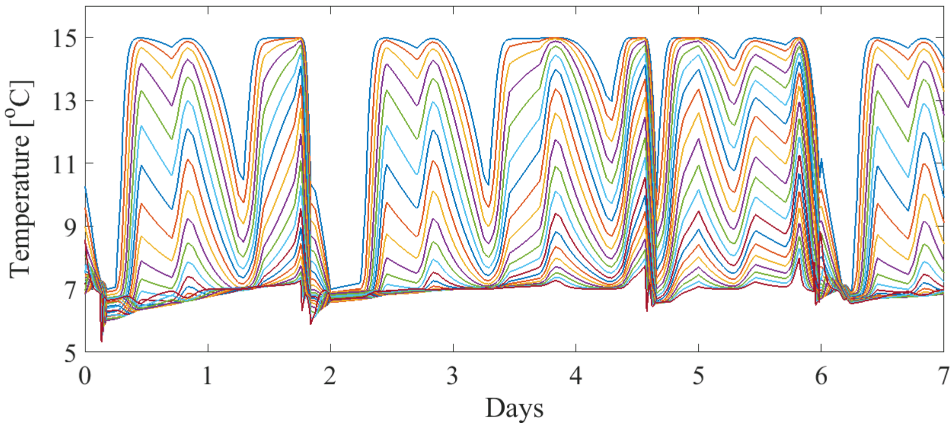
Space cooling demand vs. ice water produced.



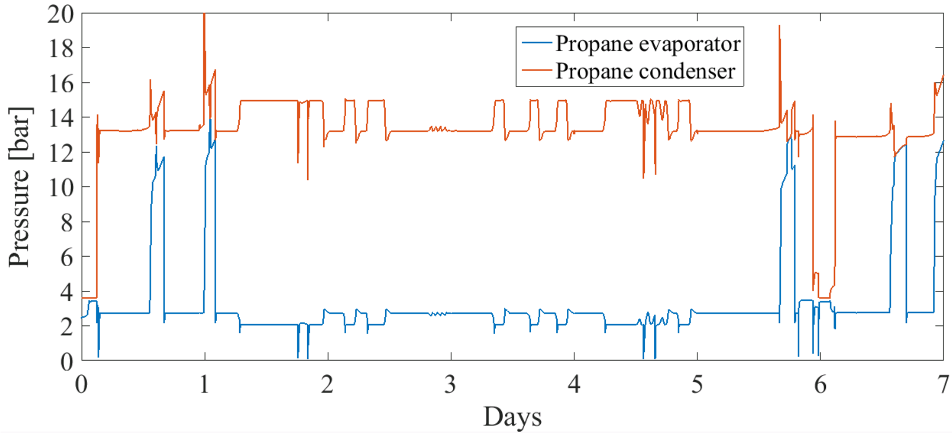
Domestic hot water tank fluid layer temperature.



Space heating tank fluid layer temperature.

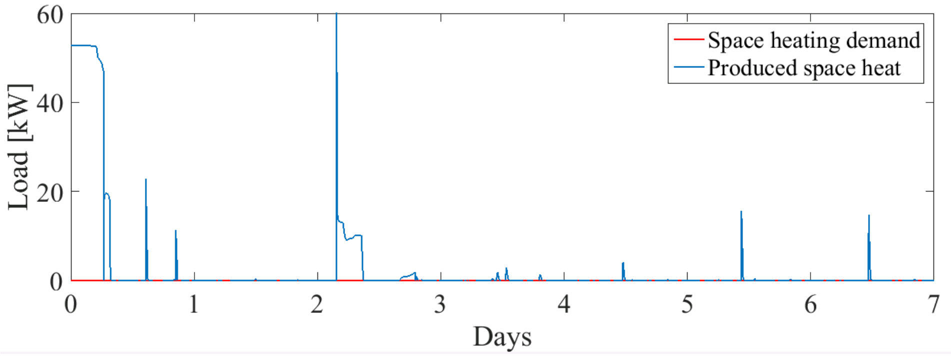


Ice water tank fluid layer temperature.

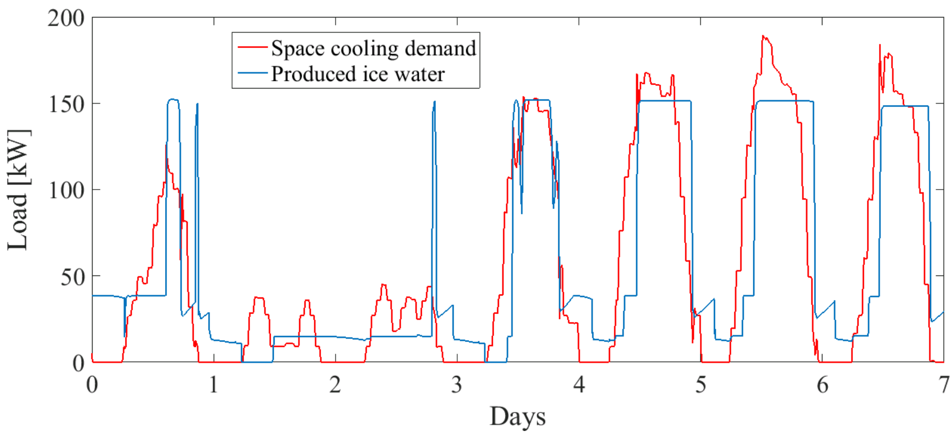


Propane heat pump evaporating and condensing pressure.

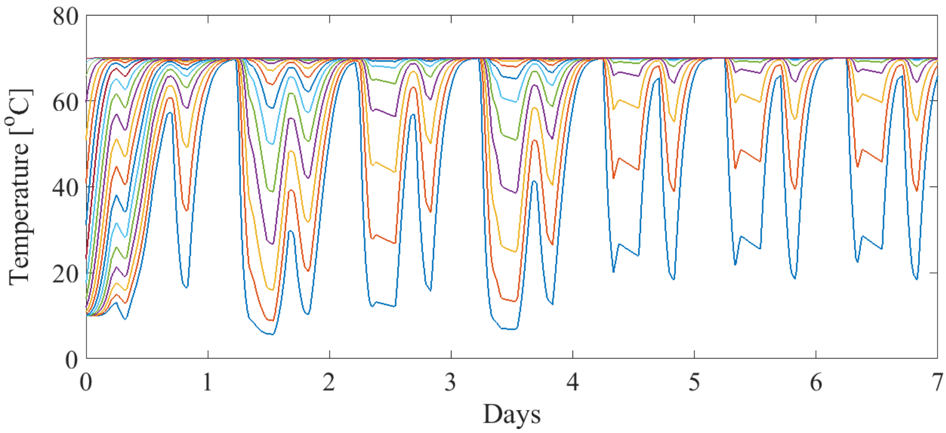
C.2 Summer scenario



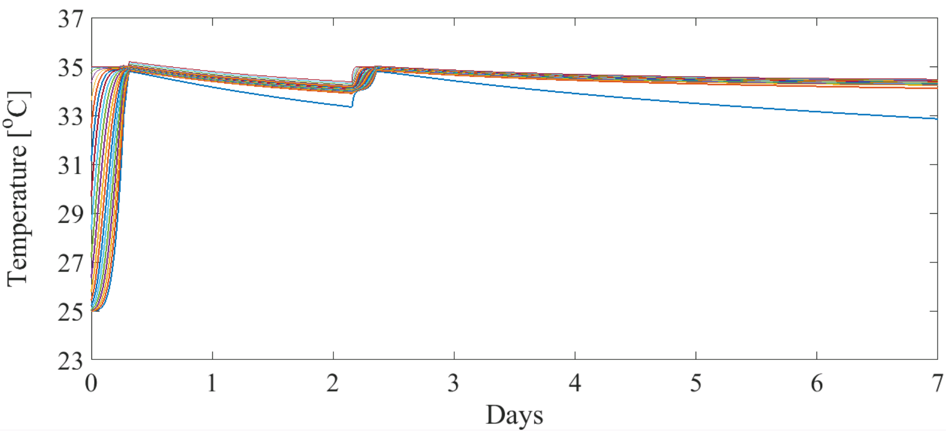
Space heating demand vs. produced space heat.



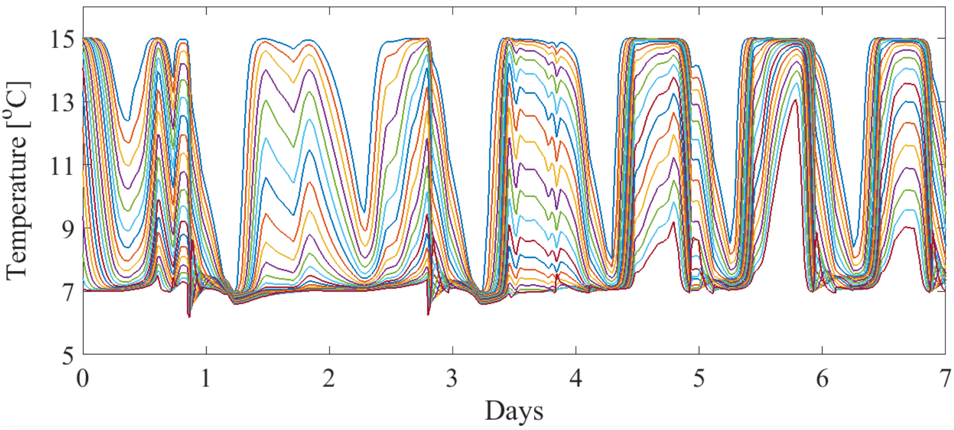
Space cooling demand vs. ice water produced.



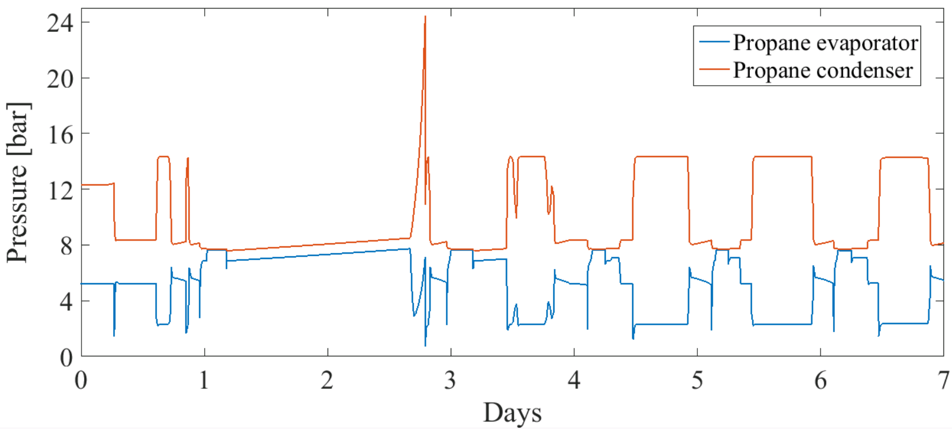
Domestic hot water tank fluid layer temperature.



Space heating tank fluid layer temperature.



Ice water tank fluid layer temperature.



Propane heat pump evaporating and condensing pressure.

C.3 Economical evaluation

Case 1

	Yearly cost [NOK]	% contribution
Space Heating	287983,7	59,08
Domestic hot water	105122,4	21,57
Space cooling	53840,1	11,05
Refrigeration rooms	13505,0	2,77
Freezer roms	27010,0	5,54
TOTAL	487461,3	

Case 2

	Yearly cost [NOK]	% contribution
Space Heating	115193,5	46,2
Domestic hot water	58401,4	23,4
Space cooling	53840,1	21,6
Refrigeration rooms	13505,0	5,4
Freezer roms	8328,1	3,3
TOTAL	249268,1	

Case 3

	Yearly cost [NOK]	% contribution
Space Heating	72405,5	46,8
Domestic hot water	68506,0	44,3
Space cooling	0,0	0,0
Refrigeration rooms	0,0	0,0
Freezer roms	0,0	0,0
Heat dump	13640,1	8,8
TOTAL	154551,5	

Appendix D: Abstract for article

INTEGRATED ENERGY CONCEPTS FOR HIGH PERFORMANCE HOTEL BUILDINGS

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ABSTRACT

Integrated energy concepts with heat recovery are not widely applied for hotels, i.e. there is a huge potential to further reduce the specific primary energy demand in this kind of buildings. The development and implementation of such concepts would reduce the cost of ownership and the environmental footprints in the sector, significantly.

A combined heat pump and refrigeration system employing CO_2 and propane is presented for the main and backup system, respectively. Grey water usage and thermal energy storage are included in the energy system. A model of the system has been developed and several simulations were carried out in order to investigate the demand in several scenarios.

The integrated energy system can efficiently provide heating, Air Conditioning (AC), domestic hot water and refrigeration capacities at various temperature levels. The implementation of thermal storage resulted in both reduction and delay of peak loads.

Keywords: CO_2 heat pump and refrigeration, hot and cold thermal energy storage, integrated energy systems