Norwegian University of
Science and Technology

Evaluation of different concepts for the utilization of the cooling potential from the regasification of natural gas

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Master of Energy and Environmental Engineering
Submission date: June 2018
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Department of Energy
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EPT-M-2018-110

MASTER THESIS

for

student Tone Øverby

Spring 2018

Evaluation of different concepts for the utilization of the cooling potential from the regasification of natural gas

Evaluering av ulike anleggskonsepter for utnyttelse av kjølepotensialet fra regassifisering av naturgass

Background and objective

At producing countries, natural gas is cooled down to -162ºC to become Liquefied Natural Gas (LNG), which is then transported to the consuming countries. This allows transportation of gas over long distances, without the need of pipelines, typically in specially designed ships or road tankers. At the import terminal, the LNG is regasified and distributed through gas networks.

Regasification of LNG is a very energy intensive process: high amounts of cold are given off within the regasification process. However, despite its high exergetic value, the high amounts of cold energy that are available in the 37 LNG terminals in Europe are poorly recovered. In most of the cases, it is wasted to the sea.

The objective of this Master Thesis will be to analyse the potential for conversion of the ultra-low thermal energy available in natural gas into low-, medium- and high-temperature refrigeration applications applying R744 (CO_2) or other natural working fluids and possible auxiliary power generation. In addition, a proposal and evaluation of several designs for this conversion should be made.

The following tasks are to be considered:

1 Literature review on state of the art regasification technologies for LNG terminals
2. Describe some case scenarios and develop system configurations enabling the utilisation of cold thermal energy during the regasification towards refrigeration applications and power generation
3. Develop a calculation tool with Modelica, to perform the analysis of energy saving potentials for the various system configurations and types of heat exchangers
4. Perform the data processing and analysis of simulation results
5. Develop a cost analysis tool to estimate the operational costs compared to conventional systems
6. Investigate the challenges when transferring the technology from LNG to hydrogen terminals in the future.
7. Discussion, conclusions and proposal for further work
8. Make a draft scientific paper based on the main results

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 2018

Prof. Dr.-Ing. Armin Hafner
Academic Supervisor

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Preface

This thesis represents the final work of my Master's degree, carried out spring 2018, and concludes my degree in Energy and Environmental Engineering at the Norwegian University of Science and Technology (NTNU), Department of Energy and Process Engineering.

The thesis evaluates different concepts for utilizing the cooling potential available from the regasification of natural gas. It provides the reader with proposed designs for conversion into multiple temperature level refrigeration applications, applying a natural working fluid, as well as auxiliary power generation. Additionally, a developed calculation tool in Dymola serves the purpose of analyzing the associated potential energy savings, which has constituted as the main part of the work this spring.

A project work, serving as a feasibility study for this Master Thesis carried out autumn 2017, granted me with important knowledge of the present regasification technologies for LNG terminals and has hence been a supportive toolbox through this final work.

I will like to thank my main supervisor, Professor Dr.-Ing. Armin Hafner at NTNU, for offering good discussions and great knowledge of energy efficient cold recovery systems. My co-supervisor Dr. Ángel Álvarez Pardiñas has been an essential asset in the development of the Dymola simulation models, and I will also like to express my gratitude for his availability during this semester. To Ph.D. Candidates Håkon Selvnes and Silje Marie Smitt, thank you for your time and expertise with Dymola. I am additionally grateful for the opportunity to participate at the Modelica & TIL training course organized by TLK-Thermo GmbH, carried out in Braunschweig, Germany, August 2017. This granted me with important knowledge during startup with the used simulation tool, Dymola. At last, but not least, I will like to thank my family and friends for support and motivation during my five years study here at NTNU.

Trondheim, 11.06.2018

Tone Øverby, MSc. student

Abstract

The increasing interest in energy efficient solutions, and working towards a more sustainable future, give rise to the importance of utilization and thus reduction of the current wastage of excess energy today. As the total energy demand in the world increases, both renewable energy and the use of more environmentally friendly fuels compared to coal and oil, have become significant. Natural gas can be considered as both purer and more environmental friendly and serves as an important energy carrier for the future. However, the intensive energy use during its production and distribution can be made more efficient. As 50% of the natural gas today is provided by liquefied natural gas at about -162 °C, a large cold energy potential is introduced. Nonetheless, it is poorly recovered at the regasification terminals, mainly wasted to the sea. The aim of this Master Thesis is thus to reveal and evaluate different concepts for utilizing this potential.

Japan is proven to be at the forefront of exploiting the mentioned excess cold energy. At the Senboku terminal, an installation of multiple cold recovery systems recovers 100% of the available cooling potential. However, at the 37 LNG terminals in Europe, the utilization level is rather low. Only a few installations exist, including a 4.5 MW Rankine cycle at the Huelva plant and 5.5 MW turboexpanders at the Barcelona terminal in Spain. For that reason, system configurations for cold energy recovery into refrigeration applications, with CO_2 as working fluid, as well as auxiliary power generation were designed in this study. In addition, a calculation tool with the dynamic simulation software Dymola was developed, serving the purpose of analyzing the potential energy savings related to the system designs.

Moreover, a scenario analysis was performed, looking into variable regasification demand at the two configured cold recovery systems: a supermarket refrigeration system and an Organic Rankine Cycle. The conducted simulations confirmed the huge available cooling potential for further usage. A number of 435 supermarkets, with the designed refrigeration system, were found necessary to cover a regasification demand of 10 kg/s, indicating that multiple cold recovery systems are substantial for an efficient utilization of the LNG cold. A closer temperature fit between the heating of LNG and the cold recovery systems for the inclusion of an auxiliary power generation, confirmed the observation.

Additionally, a cost estimate and emission analysis were carried out based on the simulated scenarios. A large reduction in operational costs could be observed for both the refrigeration system and the included auxiliary power generation, even an estimated annual income of about 664,561 NOK for the latter case. Regarding the emission analysis, an assumption of no cold recovery at all for an annual global natural gas trade of 346.6 billion m^3 in 2016, resulted in a total potential reduction of about 213,759 ton CO_2.

Sammendrag

Den økende interessen for energieffektive løsninger, samt å arbeide mot en mer bærekraftig fremtid, viser til viktigheten av en bedre utnyttelse av dagens tilgjengelige overskuddsenergi, og dermed en reduksjon av dagens sløsing. Ettersom det totale energibehovet i verden øker, har overgangen til både fornybar energi og bruk av mer miljøvennlig brensel, sammenlignet med kull og olje, blitt av betydelige interesse. Naturgass kan betraktes som både renere og mer miljøvennlig og utgjør dermed en viktig energibærer for fremtiden. Imidlertid kan det intensive energiforbruket gjennom produksjon og distribusjon gjøres mer lønnsomt. Siden 50% av naturgassen som leveres i dag distribures som flytende naturgass på rundt -162 °C, introduseres et stort energipotensial ved lav temperatur. Allikevel er dette potensialet svært dårlig utnyttet ved regassifiseringsterminalene, det er hovedsakelig sluppet ut i havet. Formålet med denne masteroppgaven er derfor å avdekke og evaluere ulike systemer for å utnytte dette potensialet.

Japan har vist seg å være i forkant ved å utnytte den nevnte overskuddskulden. En installasjon av flere kuldegjenvinningssystemer utnytter til sammen 100% av det tilgjengelige kjølepotensialet ved Senboku terminalen. Ved de 37 LNG terminalene i Europa er derimot utnyttelsesgraden lav. Bare noen få innstallasjoner eksisterer, inkludert en 4,5 MW Rankine syklus ved Huelva anlegget og 5,5 MW turboekspandere ved Barcelona terminalen i Spania. På bakgrunn av dette ble systemkonfigurasjoner for kuldegjenvinning til kjøleapplikasjoner, med CO_2 som arbeidsmedium, samt hjelpekraftproduksjon utformet i denne studien. I tillegg ble et beregningsverktøy med det dynamiske simuleringsprogrammet Dymola utviklet, noe som muliggjør analyse av de potensielle energibesparelsene knyttet til systemdesignene.

Videre ble en scenarioanalyse utført, basert på varierende regassifiseringskrav for de to utviklede kuldegjenvinningskonfigurasjonene: et kjølesystem for et supermarked og en organisk Rankine syklus. De utførte simuleringene bekreftet det store tilgjengelige kjølepotensialet for videre utnyttelse. Hele 435 supermarkeder, med det utformede kjølesystemet, ble funnet nødvendig for å dekke et regassifiseringskrav på 10 kg/s, hvilket indikerer at flere kuldegjenviningssystemer er nødvendig for en effektiv utnyttelse av den tilgjengelige kalde energien. Dette ble i tillegg bekreftet ved en tettere temperatur tilpasning mellom oppvarmingskurven for LNG og gjenvinningssystemene ved inkludering av konfigurasjonen for termisk energi produksjon.

Det ble i tillegg utarbeidet en kostnadsberegning og utslippsanalyse basert på de simulerte scenariene. En stor reduksjon i driftskostnader ble observert både for kjølesystemet og den medfølgende energi produksjonen. I forbindelse med energiproduksjonen ble det estimert en årlig inntekt på om lag 664 561 NOK. Ved å anta ingen kuldegjenvinning for den årlige globale LNG distribusjonen i 2016 på 346,6 milliarder m^3, ble dessuten en total potensiell reduksjon på cirka 213 759 tonn CO_2 estimert.

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Chapter 1

Introduction

In the last decades, the fuel share of natural gas has increased and made natural gas a very important energy carrier for the future. Energy statistics produced by the International Energy Agency presents that the world fuel share of natural gas has increased from 16% to 21% from 1973 to 2014 [16]. Compared to other petroleum products natural gas is purer and more environmental friendly. Use of natural gas is therefore preferable due to the lower amount of pollution, the ease of production, together with its availability and lower costs. As the total energy demand in the world will likely increase in the following years, one should thus prepare for more use of natural gas in the future.

When transporting natural gas over long distances the use of pipelines is no longer favorable, and the use of specialized ships or road tankers is considered for future investments. To obtain a profitable solution, the differences in volumes related to liquid and gaseous states of natural gas should be evaluated. As 1 m^3 of liquefied natural gas (LNG) corresponds to 600 Sm^3 of natural gas, the liquefaction process seems very promising [29]. This process is already widely used today, around 50% of the natural gas is provided by LNG, as seen by Figure 1.1.

The value chain of LNG can be divided into four different stages: exploration and pretreatment, liquefaction and storage, transportation, and regasification with further distribution and storage [25]. At the liquefaction stage, the natural gas is cooled down to about -162 °C at atmospheric pressure. This process, the actual production of LNG, requires a huge amount of energy. The consumed energy can further be referred to as a low-temperature energy that now is stored in the LNG, representing a cooling potential for further usage.

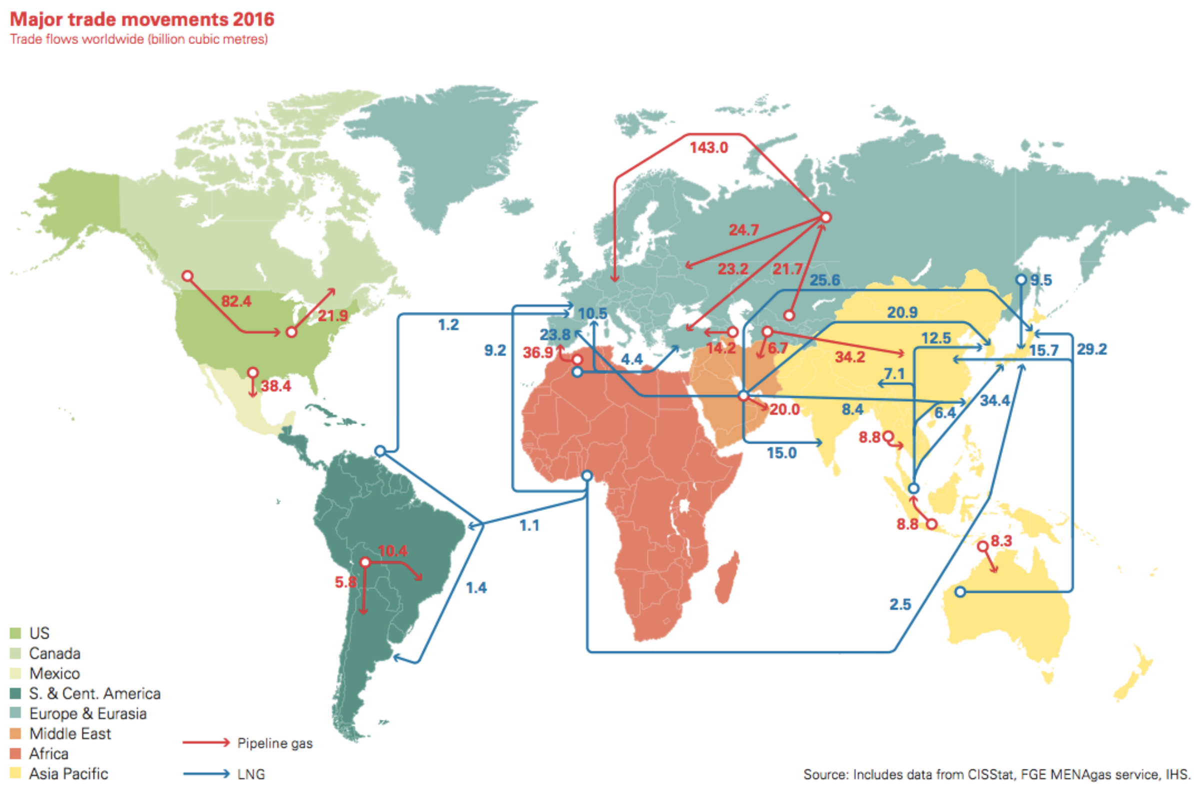


Figure 1.1: Major trade movements of natural gas in 2016 [4].

At the regasification stage, adding of heat is necessary, which today is mainly taken from the environment or waste heat. The energy-intensive process during both liquefaction and regasification, introduce a disadvantage with the provision of natural gas by LNG compared to pipelines; There is a large amount of energy needed to complete the cycle of LNG. In that sense, there is even more important to look into concepts for a better usage of the excess cold energy. Utilization of the cooling potential will most likely result in a significant increase in energy efficiency, as well as increased sustainability related to environmental impact. However, this potential is poorly recovered today, mainly wasted to the sea. This means not only wastage of energy, but also disturbance of the natural environment in the sea [3]. The objective of this Master Thesis is hence to establish and analyze different concepts for utilizing this cooling potential, as well as to develop analytical tools to assess and prove the capabilities of the developed system configurations. An evaluation of the energy savings potential related to the various system solutions is to be carried out by employing the calculation tool, together with a cost estimate and emission analysis reflecting the possible energy savings.

1.1 General characteristics of LNG

In this section, some general characteristics of LNG are presented and a few assumptions for simplicity are made. The assumptions stated here are more or less consistent throughout this study.

First, a typical composition of LNG is presented in Table 1.1. A mole fraction of about 95% indicates that LNG consists of mainly methane, and is thus for convenience considered as purely methane further in this thesis.

Table 1.1: Typical LNG composition for Atlantic LNG [24].

Component	Mole percent [%]	Molar mass [kg/kmol]
Methane	95	16.04
Ethane	4.6	30.07
Propane	0.38	44.10
Butane	0	58.12
Nitrogen	0.02	28.01

Second, natural gas is cooled down to about -162 °C through the liquefaction process at producing countries, and further regasified and heated to about 0 °C at the receiving terminals, dependent on the terminal's site. Hence, the cooling potential from the regasification process is evaluated based on these given temperatures. However, the sales gas specifications varies slightly between receiving countries, and the temperature used in this report does not necessarily coincide with a specific regasification terminal.

Chapter 2

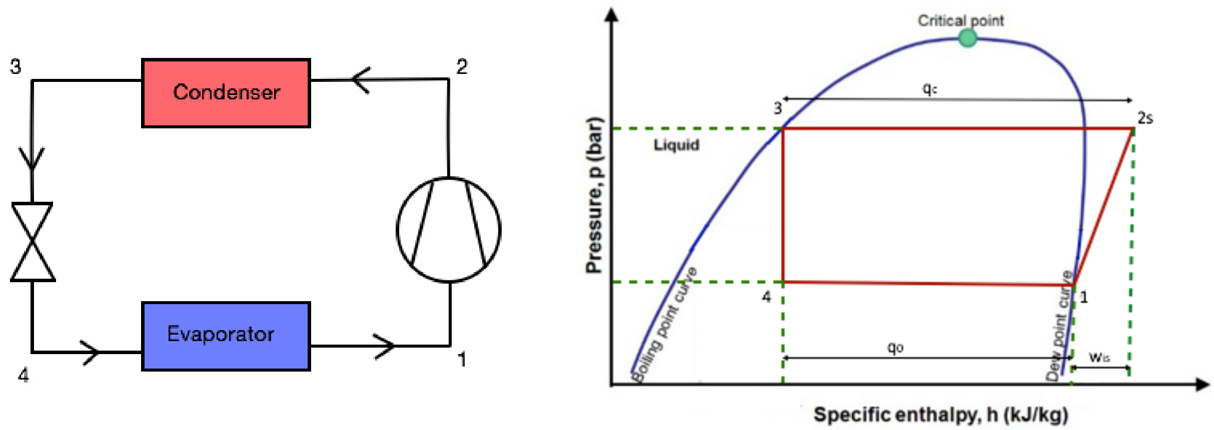
Cold energy recovery

To be able to utilize the cold energy stored in the LNG, a suitable cold energy recovery system should be evaluated. A general consideration of such a system is based on the low-temperature LNG as a heat sink for thermodynamic systems. This include for instance refrigeration systems and thermal power production cycles. Both of these applications require a heat sink to function, and to get this for free by using the LNG cold can be considered as highly beneficial. Implementation of one of these methodologies will thus perform and achieve two purposes in one, regasification of natural gas, and distinct refrigeration applications or power generation.

In this chapter, some useful theory is described starting with the ideal refrigeration system, and further working through CO_2 as a working fluid. There is additionally presented a general exergy analysis based on LNG, which is useful for evaluating the possibility for thermal power production. Furthermore, some basics about thermal power generation, considering ideal cases, are presented. And finally, technologies for thermal energy storage are studied for cold energy recovery purposes.

2.1 Refrigeration systems

The purpose of a refrigeration system is to remove heat from a refrigerated space to achieve a temperature lower than the ambient. A system that fulfills this is based on a so-called vapor compression cycle that consists of four basic components: a compressor, a condenser, an expansion valve, and an evaporator. The vapor compression cycle can generally be described as a closed loop circuit which works to absorb heat from the cold side and deliver it to the warm side; The condensation process will release heat and the evaporation will absorb heat [8]. Thus, the evaporation is the key process in a refrigeration system. When assuming that heat is absorbed and released at constant pressure and temperature, the refrigeration process can be described by considering *the reversed Carnot cycle* [35]. This represent an ideal refrigerator which provides an useful tool for comparison and evaluation of a real refrigeration system.



(a) A principle sketch.

(b) System visualization in a logarithmic pressure-enthalpy diagram.

Figure 2.1: A simple refrigeration system based on the vapor compression cycle.

Figure 2.1 illustrates a refrigeration cycle that operates in the subcritical area, which is the case for most common refrigerants, for example propane and ammonia. In this cycle the working fluid (refrigerant) is circulating and changing phase along the way. When approaching the compressor the refrigerant is in vapor phase and is further isentropically compressed through the compressor. The refrigerant is then exiting in the superheated region, which means that the vapor is at a higher temperature than its saturation temperature. In the condenser the refrigerant condenses and releases heat to the surroundings, as the ambient temperature is lower than the temperature of the refrigerant. The liquid generated by the condenser is then further being expanded through an expansion valve to the low pressure region by an isenthalpic process. The expansion process brings the refrigerant into the two-phase area, where the quality is related to the pressure difference between high an low operating pressures. Further approaching the evaporator, the refrigerant evaporates and absorbs heat from the surroundings, as the surrounding temperature is higher than the temperature of the refrigerant. This is regulated by the chosen pressure maintained by the compressor, which continuously removes the evaporated refrigerant (vapor).

By using Figure 2.1(b), relations for the exchanges of heat and work in the refrigeration system are obtained. The refrigeration capacity, $\dot{Q}_0$, which also can be described as the absorbed heat in the evaporator, is a function of the mass flow of the refrigerant, $\dot{m}_R$, and the enthalpy difference, Δh.

$$\dot{Q}_0 = \dot{m}_R \cdot q_0 = \dot{m}_R \cdot (h_1 - h_4) \tag{2.1}$$

The isentropic (theoretical) work, $\dot{W}_{is}$, of the compressor is as well a function of the massflow of the refrigerant and the enthalpy difference.

$$\dot{W}_{is} = \dot{m}_R \cdot w_{is} = \dot{m}_R \cdot (h_{2s} - h_1) \tag{2.2}$$

This can further be used together with the isentropic efficiency, η_{is}, to express the real work of the compressor.

$$\dot{W} = \frac{\dot{W}_{is}}{\eta_{is}} = \dot{m}_R \cdot (h_2 - h_1) \tag{2.3}$$

It can be seen that the released heat in the condenser, $\dot{Q}_c$, is a function of the refrigeration capacity, $\dot{Q}_0$, and the compression work, $\dot{W}$.

$$\dot{Q}_c = \dot{m}_R \cdot (h_2 - h_3) = \dot{Q}_0 + \dot{W} \tag{2.4}$$

Finally, the expansion process is characterized as an isenthalpic process, $h_3 = h_4$.

Heat exchanger analysis
To predict or design the performance of a heat exchanger, for instance the condenser and evaporator in the refrigeration system, the logarithmic mean temperature difference is an applicable tool. The method assesses the essential relationship between fluid temperatures, heat transfer area (A), and the heat transfer coefficient (U) in order to analyze the efficiency of the heat exchanger. Equation 2.5 expresses the mentioned parameters' relation to the total rate of heat transfer, q, where ΔT_{lm} is the logarithmic mean temperature difference. This tool assume no heat transfer with the surroundings as well as negligible changes in potential and kinetic energy. Further, the calculation of the logarithmic mean temperature difference is shown in Equation 2.6.

$$q = UA\Delta T_{lm} \tag{2.5}$$

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{ln(\Delta T_2 / \Delta T_1)} \tag{2.6}$$

, where ΔT_2 and ΔT_1 represent the temperature difference between the heat exchanging fluids at the inlet and outlet of the heat exchanger. This statement is true for both a parallel-flow and a counterflow designed heat exchanger.

2.1.1 CO_2 as working fluid

Carbon dioxide (CO_2) was reintroduced as a working fluid for heat pumping systems in the late 1980's, after being replaced by synthetic refrigerants for several years [8]. The reason for reintroducing may be justified by the increased awareness of the environmental impact the synthetic refrigerants represent. In comparison, CO_2 is a natural working fluid, which is non-toxic and non-flammable.

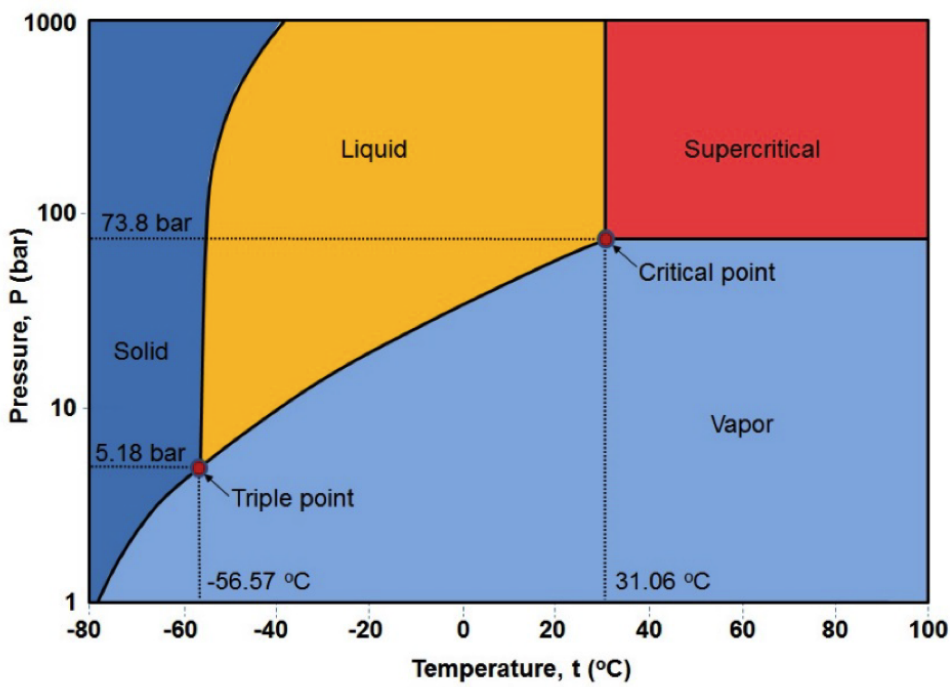


Figure 2.2: A phase diagram for CO_2 presented in a pressure-temperature format [8](Chapter 7, Figure 7.3). Reused with permission granted by Trygve Magne Eikevik.

By comparing CO_2 with other conventional working fluids, such as ammonia, one will notice its low critical temperature and high critical pressure, respectively 31.1 °C and 73.8 bar. It has also a high triple point, which implies that CO_2 will turn into solid state already below -56.6 °C and above 5.18 bar. Figure 2.2 illustrates the different phases of CO_2 with the stated points of transformation. This means that the conventional heat pump cycle is particularly limited when using CO_2 as a refrigerant; The evaporation temperature is practically limited to about -55 °C, and the condensation temperature to 28 °C. However, a transcritical cycle will enable gas cooling at higher temperatures. Nevertheless, due to its low-temperature range, CO_2 has become a widely used working fluid in cryogenic refrigeration, in addition to the well-used hydrocarbons.

2.2 Exergy analysis

The energy potential stored in LNG, because of the highly energy intensive liquefaction process, can be expressed in terms of exergy. Exergy describes the maximum amount of work that can be achieved for a given form of energy, when the system change from a certain state to a state where it is in equilibrium with the surroundings [13]. The properties of the environment/surroundings are often considered as $T_0 = 25$ °C and $p_0 = 1$ bar, and is applied here as well. As exergy is described as the maximum amount of work, it can be evaluated by considering the two laws of thermodynamics, and can be expressed as shown in Equation 2.7 when neglecting the change in kinetic and potential energy.

$$W_{max} = T_0(S - S_0) - (H - H_0) \tag{2.7}$$

The flow exergy is by definition the maximum work a flowing medium can generate when approaching the state of equilibrium with the surroundings, and can be expressed as shown in Equation 2.8.

$$E_x = -W_{max} = (H - H_0) - T_0(S - S_0) \tag{2.8}$$

Moreover, exergy can be decomposed into different types related to different energy forms. For the case of LNG one can consider the total exergy as a combination of low temperature exergy, $E_{x,th}$, and pressure exergy, $E_{x,p}$ [22]. The available exergy is treated as completely chemical and physical of nature, and the kinetic and potential parts are negligible in comparison. During the regasification process the physical aspect of exergy, which is due to the imbalance in temperature and pressure compared to the surroundings, is the only part that is able to be externally utilized [14]. This part of exergy, the physical, can also be considered as the thermo-mechanical part of exergy, which means that it consists of a temperature part and a mechanical part. Each of these parts can be considered by holding the other variable as a constant. That means that the pressure exergy, the mechanical part, describes the change in pressure from the system pressure to the surrounding pressure, while holding the temperature constant and equal to the surrounding temperature. The procedure for the low temperature exergy, the temperature part, is equal, but now the pressure is held constant and equal to the surrounding pressure [13]. A way of expressing these two terms is shown in Equation 2.9 and 2.10, where LNG is heated from its storage state (T_s, p_s) to the state of being in equilibrium with the surroundings.

$$E_{x,th} = E_x(P,T) - E_x(P,T_0) = (\frac{T_0}{T_s} - 1)h_{fg} + \int_{T_0}^{T_s} C_p(1 - \frac{T_0}{T})dT \tag{2.9}$$

$$E_{x,p} = E_x(P,T_0) - E_x(P_0,T_0) = \int_{P_0,T_0}^{P,T_0} v\,dp \tag{2.10}$$

The total low temperature exergy is based on two different terms, based on the two different processes that occurs when heating LNG from the storage temperature, T_s, to the surrounding temperature, T_0. These are the vaporization latent heat exergy and the apparent heat exergy, which are respectively shown as the first and second term on the right hand side of Equation 2.9 [22].

2.3 Thermal power generation

Thermal power generation is based on a process where an energy conversion from heat to work occurs. This is basically done by using a turbine that further is connected to a generator where the produced mechanical work can be converted to electric power. For the purpose of this thesis, three distinct power cycles are of importance: the Rankine cycle, direct expansion systems, and the Brayton cycle. Additionally, the Rankine cycle is one of the possibilities evaluated for utilization of cold energy later in this report, and a general presentation with focus on the ideal case will be given here. How the different power cycles are used for cold energy recovery during regasification of LNG are presented in Section 3.2.2.

The conservation of energy principle requires that net work generated by a power cycle is equal to the net input of heat, $W_{net} = Q_H$. When this is the case, an idealized power generation system with a thermal efficiency of 100% is present. In real life irreversibilities and losses occur in the system and cause a reduction of the thermal efficiency. This can be seen as a deduction of the second law of thermodynamics. For any system that carries out a power cycle operating between two reservoirs, only a portion of the added heat, Q_H, can be converted to work. The remaining part, Q_C, must then be discharged to the cold reservoir. By this, the relation $W_{net} = Q_H - Q_C$ is true, and the thermal efficiency, η_t, can be expressed as shown in Equation 2.11.

$$\eta_t = \frac{W_{net}}{Q_H} = 1 - \frac{Q_C}{Q_H} \tag{2.11}$$

When further considering a reversible power cycle operating between two reservoirs, there are the temperatures that provide the driving force for heat transfer and thereby also the production of work. The thermal efficiency thus depends only on the reservoir temperatures, T_C (cold) and T_H (hot), and the thermal efficiency can be expressed as shown in Equation 2.12. This relation represent the maximum thermal efficiency of power cycles and is known as the *Carnot efficiency*.

$$\eta_{max} = 1 - \frac{T_C}{T_H} \tag{2.12}$$

Rankine cycle

The Rankine cycle is based on an idealized version of a heat engine that converts heat to work by circulating a phase changing working fluid. For the simple case it consists of four interconnected components: A turbine, a condenser, a pump, and a boiler. A basic schematic of the simple Rankine cycle is shown in Figure 2.3.

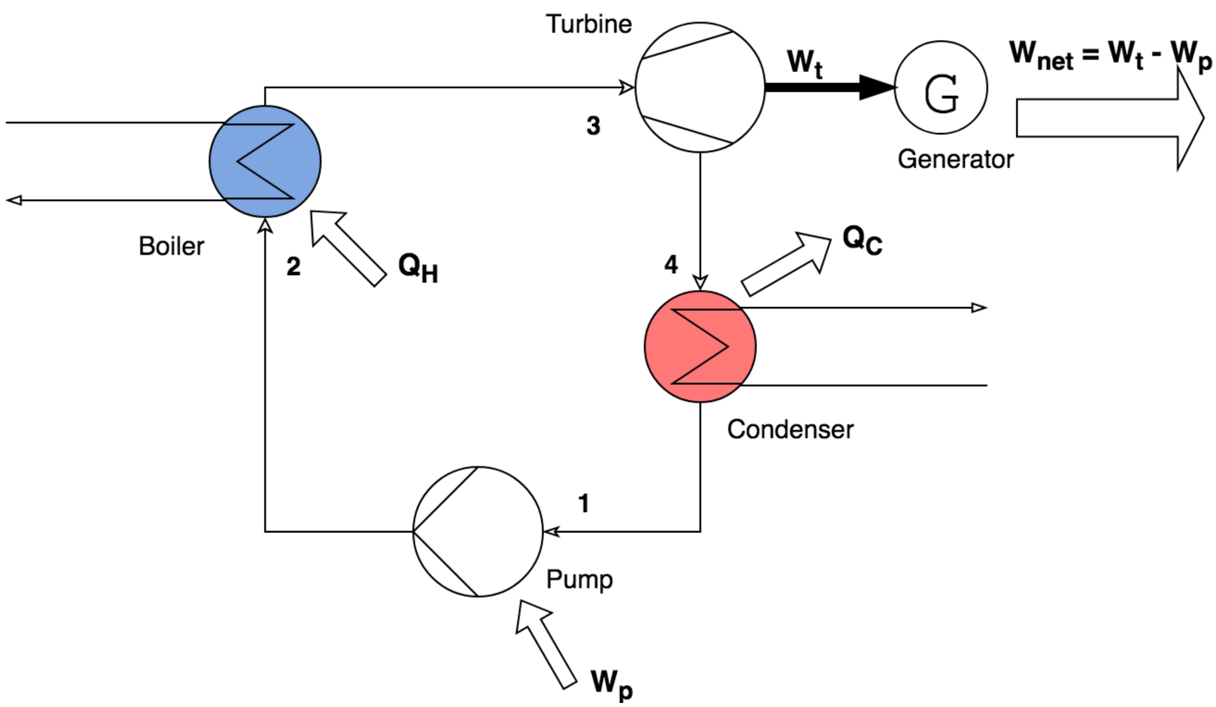


Figure 2.3: A simple sketch of a Rankine Cycle.

When considering a Rankine cycle where the working fluid passes through the cycle without irreversibilities, the flow through the condenser and the boiler operate at constant pressure. In the absence of irreversibilities, together with considerations of isothermal conditions, both the turbine and pump operate isentropically. Application of these considerations, or idealizations, give rise to the ideal Rankine cycle. The ideal Rankine cycle consists of only internally reversible processes, and one can therefore consider average temperatures when developing an expression for the thermal efficiency. Thus, the maximum thermal efficiency can be expressed as in

Equation 2.12. The overall efficiency can be increased by increasing the heat input temperature, and several improvements can be done to achieve this; Both implementation of superheat and reheat, as well as regeneration will improve the performance of the power cycle.

2.4 Thermal energy storage

The ability to store thermal energy is of highly importance in order to recover heat, or cold, from the system in times when it's not needed. Thermal Energy Storage (TES) technology can thus play a significant role for correcting mismatches between energy supply and demand for a thermal energy system, and by that ensuring a more efficient and environmental use of the available energy. The storage system will also increase the overall system's reliability, as well as better the economics by reducing both investment and running costs [6].

There exists several storage technologies, where the main three types are categorized as sensible heat, latent heat, or thermochemical storage. Sensible heat storage is related to the energy required to change the temperature of a substance without phase change, and can be exemplified by storing hot water in tanks. Latent heat storage is based on the energy available in a phase change process, most common between solid and liquid. Thermochemical storage on the other hand, can be described by using a thermochemical material which can utilize both sorption and chemical reactions to produce heat, and further store it in for example a hot water tank. By comparing these three types, the latent heat storage has multiple benefits. It has much higher energy density compared to sensible heat storage, and is easier to work with than thermochemical storage. These comparisons result in less required material mass, as well as smaller tank volumes and less needed support equipment for the latent heat storage [10]. One of the most beneficial options for latent heat storage is phase-changing materials (PCMs), and this will be the main focus in this section.

2.4.1 Phase-changing materials

By utilizing the latent heat stored in a material, energy can either be absorbed or released at constant temperature in a system. Materials that undergo this process can then be described as phase-changing materials (PCM), and is thus a great option for thermal storage. The quantity of energy released or absorbed is characterized by either the latent heat of fusion, representing the sold-liquid phase change, or the latent heat of vaporization, representing the liquid-gas phase change. In PCM units, the latent heat of fusion is the most used. Basically because this phase change does not require as high amount of support equipment to function, compared to the latent heat of vaporization. On the other hand, the vaporization part generally release/absorb more heat than the fusion part [10].

For cold storage purposes, thermal energy is absorbed by the PCM during periods with high energy supply, and released in periods with high energy demand. In that regard, the PCM cold storage unit can be deemed as a heat exchanger. Such applications can for example be considered for refrigeration systems, where the PCM storage unit is implemented as a shell and tube heat exchanger providing the necessary condensation at high energy demands. The PCM is typically placed on the shell side of the exchanger, and the type of PCM is chosen based on the necessary temperature level. Such an implementation will establish a quick charge and discharge of thermal energy between the PCM and a working fluid flowing in the tubes. In the study done by Beck et al. [2], the cold storages for domestic chilling and freezing are modeled in this way, using a cylindrical tube filled with a PCM installed around an evaporator. In that sense, such designs are of increasing interest within refrigeration applications, for example in supermarkets.

Chapter 3

Regasification technologies for LNG terminals

When the LNG enters the receiving terminal, it needs to be unloaded from the ships (or the road tankers) and stored in highly insulated tanks before the regasification process can start. A typical LNG receiving terminal is shown in Figure 3.1. Unloading arms are used to unload the LNG and further transferring it through an unloading line into the storage tanks. Due to safety features, the system also consists of several emergency shutdown valves, which are designed to close quickly when needed. Different kinds of storage tanks, depending on the facility's needs and requirements, exists as an assessment aspect [24]. Further, the LNG reaches the vaporization part of the terminal, which is the concerning part of this Master Thesis, as the objective entails evaluation of different concepts for utilizing the LNG cold. Therefore, this chapter presents the state of art of vaporization technologies for LNG terminals, and further looks into the different concepts evaluated, and maybe even used, for cold utilization today.

3.1 LNG vaporization

The LNG regasification process is mainly based on a heat transfer process between LNG and another working fluid, and there exists different technologies used for this purpose, including different heat exchangers, working fluids and types of heat source. This heat transfer process is commonly called an LNG vaporization system, and the type is project specific, depending primarily on the terminal's location, environmental conformity, regulatory limitations, and energy efficiency. The most common vaporizers used today are the Open Rack Vaporizer (ORV) and the Submerged Combustion Vaporizer (SCV), which are respectively used at about 70% and

20% of the base load regasification terminals [24]. Other used vaporizers are for instance Shell and Tube Vaporizer (STV), including Intermediate Fluid Vaporizer (IFV), and Ambient Air Vaporizer (AAV). Pu et al. [30] additionally state that the Intermediate Fluid Vaporizer is one of the four most commonly used vaporizers today, and is likely the most represented vaporizer for the remaining 10% of the base load regasification terminals.

A typical regasification power plant has an annual regasification capacity of about 3 million ton, indicating an average massflow entering the vaporizers of approximately 100 kg/s [27].

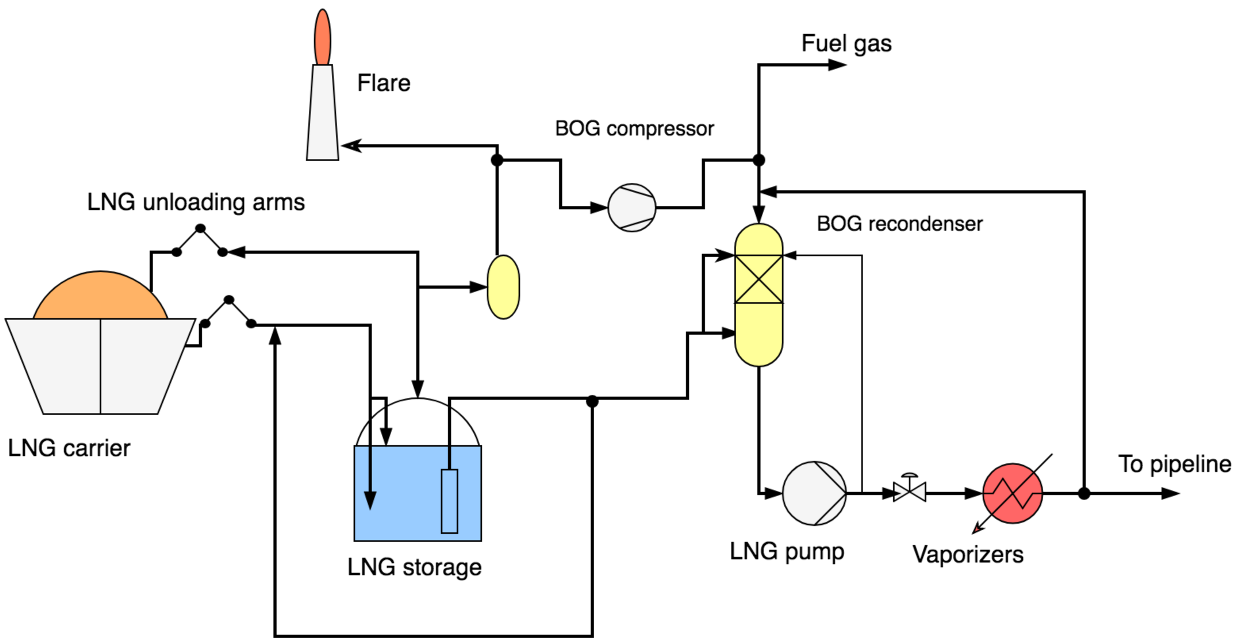


Figure 3.1: A typical LNG receiving terminal. Adapted with permission from Elsevier Inc. [24]

The Open Rack Vaporizer (ORV)

The ORV is a heat exchanger that employs seawater as the source of heat and is therefore favorable when the receiving terminal is located close to the sea, which it normally is. The construction is based on finned aluminum alloy tubes that are arranged in panels, where LNG flows through the tubes and vaporizers as seawater is sprayed over the panel. Figure 3.2a illustrates a simple sketch of an ORV.

As the LNG has a very low temperature, the main problem with this vaporizer is the water's tendency to freeze on the tube surfaces. The potential ice formation on the tubes causes a thermal resistance of huge significance and hence reduces the heat transfer efficiency. In order to reduce this tendency, a prediction of both the ice layer thickness and the distribution of it were performed by Jeong et al. [18], and from that result, an improved configuration named the super ORV was developed. The super ORV is structured similar to the ORV, but it consists of a double-tube structure that effectively can suppress ice formation in the lower part of the panel of tubes. It was shown in a comparative analysis that the vaporization capacity of the super ORV was about three to five times higher than for the original ORV. Together with lower operating costs and energy consumption, a major advantage using the super ORV was detected. [30]

The Submerged Combustion Vaporizer (SCV)

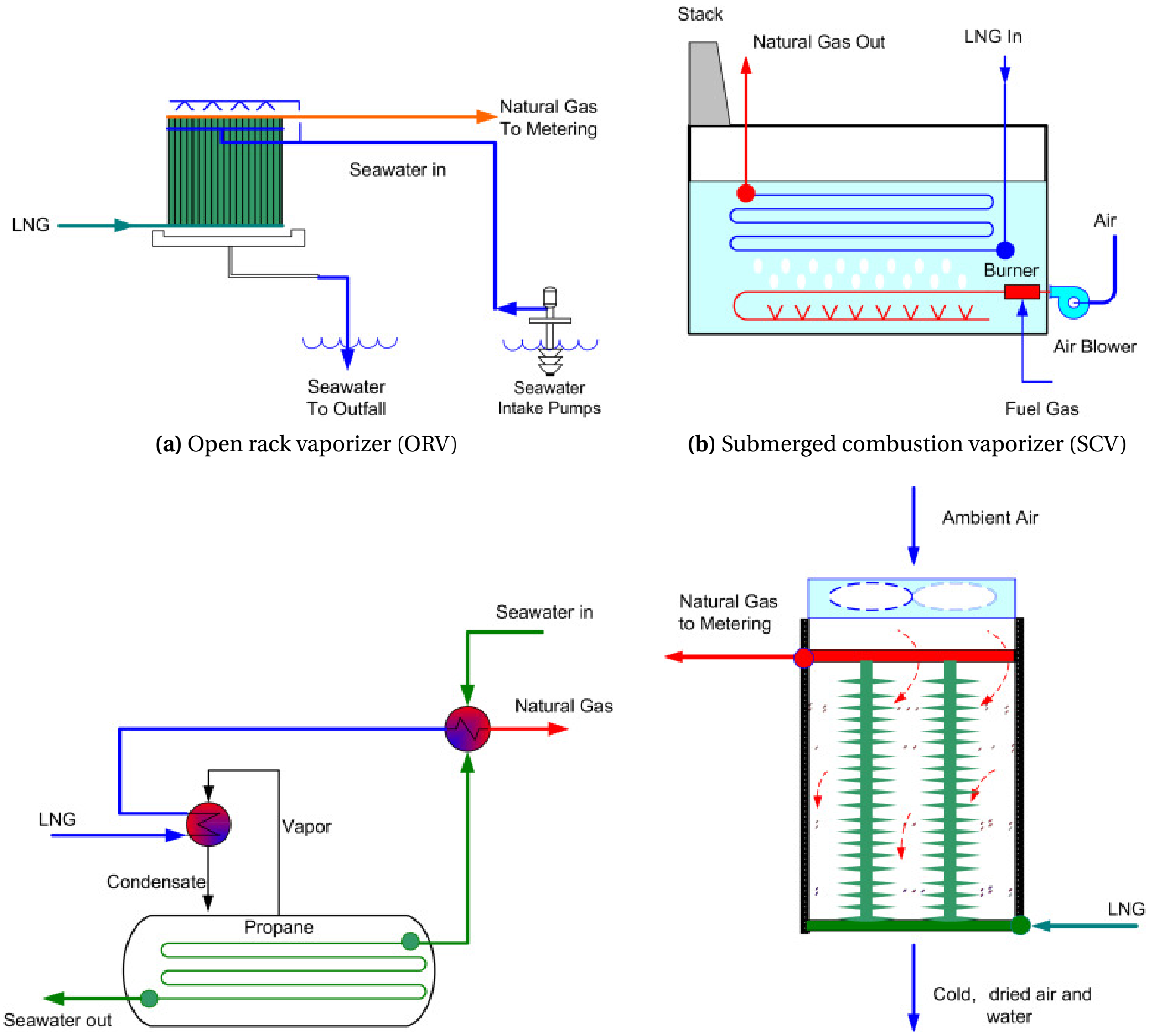
The SCV, the other commonly used vaporizer, exploits hot exhaust gases from an installed gas burner as the source of heat. As a consequence, the vaporizer often requires some of the vaporized LNG as fuel, which leads to a reduction in the total efficiency of the LNG supply chain as well as higher operational costs. This vaporizer is therefore preferable in sites where there is no other free source of heat available. In comparison to the ORC, the SCV has no impact on the marine life, however, it will increase the number of emissions due to its combustion process. [12]

The construction consists of a tube coil that is submerged in a water bath, where the water is heated by the exhaust gases from the combustion, and the LNG vaporizes by flowing through the tube coil [24]. A simple schematic is shown in Figure 3.2b. Because the water bath is maintained at a constant temperature, the SCV is able to start up quickly and thus responds rapidly to a change in load. These vaporizers are therefore very reliable and offer great safety features. Due to the efficient control systems, it will detect possible gas leakages and quickly shut down the unit to restore safety against explosions. However, it is normally no risk of explosion, as the water bath temperature stays below the ignition point of the natural gas.

The Intermediate Fluid Vaporizer (IFV)

Basically, the IFV is a shell and tube vaporizer that uses an intermediate heat transfer fluid suitable for low operating temperatures, which includes different glycols and hydrocarbons such as propane and butane [24]. For the case of using propane, seawater is used as the source of heat. The system consists of an evaporator, a shell and tube exchanger where the seawater heats up the propane such that it vaporizes on the shell side, and a condenser where the propane vapor condenses as it heats up the LNG [30]. A simple sketch of the described working principle is shown in Figure 3.2c.

The main advantages of an IFV include no icing on racks, lower operating costs, and higher environmental friendliness. By comparing it with the most commonly used ORV, the IFV has additionally a lower requirement for seawater quality and a more compact volume. These advantages, in comparison to the two previous presented vaporization technologies, make the IFV a more energy efficient and reliable vaporizer. [30]



(a) Open rack vaporizer (ORV)

(b) Submerged combustion vaporizer (SCV)

(c) Intermediate fluid vaporizer (IFV)

(d) Ambient air vaporizer (AAV)

Figure 3.2: Simple schematics of the presented vaporization technologies for the regasification of natural gas. Reused with permission granted by Elsevier Inc. [24]

The Ambient Air Vaporizer (AAV)

The AAV is characterized, as the name indicates, by the use of ambient air as the source of heat. Hence, it avoids both the use of seawater and fuel gases, which makes this vaporizer more environmentally friendly than both the ORV and SCV. The use of ambient air additionally reduces the fuel consumption and operational costs. A typical AAV consists of long vertical heat exchange tubes that assist the downward stream of ambient air fed in at the top. The actual vaporizer is constructed as a cross-flow heat exchanger, where the temperature is higher at the top and lower at the bottom where LNG is injected. A simple schematic of an AAV is shown in Figure 3.2d.

Due to the use of ambient air, this vaporizer is most favorable in hot climate regions where the ambient air remains consequently at a high temperature. If used in colder climates, a supplementary heating may be required in order to reach the given gas specifications. Ambient temperature and humidity of the air highly influence the performance of the vaporizer, and one can thus consider the vaporizer as a bit unstable as these parameters are not changeable. These factors induce both frost growth and deposition on the vaporizer surface, which will decrease the heat transfer dramatically. [20, 24]

3.2 LNG vaporization with integrated cold utilization

During the liquefaction of natural gas, a huge amount of power is consumed, more accurate about 6.5 GW per Sm^3/d of natural gas [24]. In theory, some of the consumed power can be recovered at the LNG receiving terminal, either by using LNG as a refrigerant for different cooling purposes, as a heat sink in power generation, or as a combination of the two, a combined heat and power system (CHP). In the previously presented vaporization technologies, the low temperature exergy of LNG is destroyed without any use. Therefore, vaporization technologies utilizing this exergy will be of further importance in the future.

The present status of research on integrated cold utilization technologies include several types of research areas, however, the implementation level is still low. Japan serves as the forefront of these technologies in the world, where the Senboku LNG terminals for Osaka Gas can be considered as one of the most efficient regasification terminals in the world in regards of cold utilizations. At Senboku Terminal 1 there exists an advanced utilization system with several different cold recovery strategies, which have been implemented in steps since 1978. This includes air conditioning, carbon dioxide liquefaction, warm water chilling, brain chilling, power generation by an expansion turbine, air separation, and an ethylene plant, given in the implemented order. Together these strategies represent a utilization level of approximately 100%. [38]

Further in this Chapter some basics about these cold recovery strategies are presented, and some guidelines on what a good cold recovery system for the LNG cold should include are given.

3.2.1 Refrigeration applications

Several refrigeration applications are suited to utilize the LNG cold, either directly or indirectly. A direct approach is mainly industrial and hence located close to the terminal. However, for the indirect case, using the low temperature of LNG to cool down another working fluid, the cold can also be distributed to locations away from the terminal. The potential cold recovery systems by refrigeration are based on typical heat exchange processes, where the LNG is being vaporized at the same time as the end user fulfills its cooling demand. If considering the simple refrigeration cycle, presented in Section 2.1, the LNG will work as the heat sink for the condenser making the refrigerant condense.

In order to utilize the LNG cold efficiently, the cooling curves of the end users should match the heating curve of LNG. This makes it quite difficult to fully utilize the cooling potential, according to the need of suitable end users for the several available temperature levels, and can only be possible in cases where the regasification terminal is located in an industrial complex. Figure 3.3 shows some typical LNG heat release curves for three different pressure levels, and one can observe that at a lower pressure the available low temperature refrigeration capacity is a bit higher. The best solution for utilization as much of the cold as possible, at locations away from the terminal, is thus to consider an end user with a need of multiple temperature levels. Such end user, or industries, can for instance be supermarkets, agro-industrial processes, or other refrigeration applications combined with a power generation system in the bottom. [7]

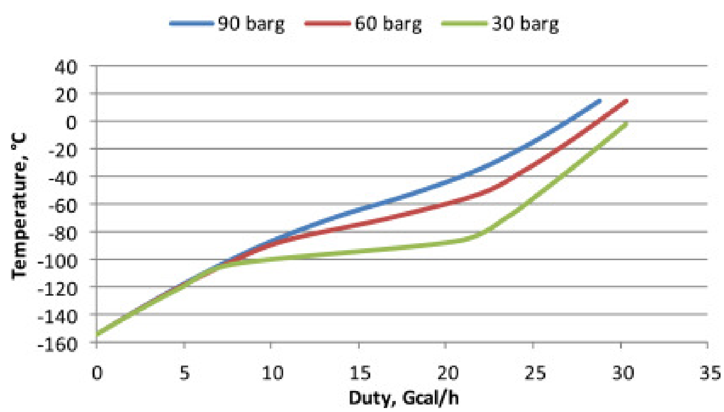


Figure 3.3: Typical LNG heat release curves. Reused with permission from Elsevier Inc. [24]

When the LNG cold should be utilized at a distance from the regasification site, a secondary fluid that can transfer the thermal energy from the terminal to the end user needs to be considered. Due to the ultra-low temperature range, there is a lack of suitable secondary fluids. Commonly used working fluids, such as alcohols or glycols, requires a large mass flow rate to circulate, which can give rise to an increase in the needed pumping power because of high viscosity in their low temperature range. With reference to Section 2.1.1, the use of CO_2 should be evaluated because of its suitable temperature range and its environmental advantages. The study by Vincenzo La Rocca [31] found the use of carbon dioxide suitable for refrigeration applications exploiting the LNG cold, and developed system configurations for both an agro-food industry and a hypermarket with the use of CO_2. The CO_2 is hence liquefied at the regasification terminal, recovering the LNG cold, and further distributed to the end users where it's fed to the evaporators of the cold utilities. Moreover, the evaporated CO_2 is returned to the regasification facility in gaseous phase, and the process is repeated.

To summarize this section, the main cases that have been evaluated for LNG-based co-generation for refrigeration applications today are itemized below.

- Desalination processes, for example of seawater [37, 5]
- Agro-industrial processes for deep freezing purposes or commercial space conditioning [31]
- Application in a cold warehouse [21]
- Cryogenic air separation for production of oxygen and nitrogen with high purity [23]
- Cryogenic energy storage system for production of energy [27]

3.2.2 Power generation

By using the LNG cold as a heat sink in a power generation system, both a utilization of the exergy content in the LNG and a reduction in the need of an external heat source for vaporization can be obtained. In a power generation system, a low level waste heat is available and can hence be utilized as the heat source for the vaporization of LNG, which will result in a more compact and efficient system. Moreover, a heat source for the power generating system is needed. This can typically be seawater, waste heat or a small gas turbine using some of the natural gas as fuel. The use of seawater may be considered as inappropriate because of serious environmental impact. The need of chlorination to avoid fouling in pipes due to marine growth will cause severe pollution of seawater, and even also a destruction of marine life. In [12] the use of a small turbine is therefore evaluated for different power generation cycles. However, the advantages related to the sea water's availability and cost often knock out the other options.

In Japan LNG power plants have been tested and operated for several years, namely from the 1970s. In 2004, Kaneko et al. [19] stated that the total generated power output was about 85 MW in Japan. Both Kaneko et al. [19] and Mokhatab et al. [24] have investigated the cryogenic power plants in Japan, and by comparison, an increase in both the number of facilities and the amount of power generated can be observed from 2004 to 2014. The type of power generation system used in the mentioned cryogenic power plants is either a Rankine cycle, a direct expansion cycle or a combination of the two. An efficiency between 13-23% is accomplished, where around 20% of the exergy stored in the LNG can be recovered [19].

In Europe the large natural gas nation Spain, with Enagás in the lead, has investigated multiple options for increasing the efficiency and reducing emissions within their regasification terminals. For the Huelva regasification plant an installation of a 4.5 MW Rankine cycle is installed, which have both increased the energy autonomy and reduced the related greenhouse emissions. Enagás has also investigated the utilization of the pressure potential generated by the necessary pressure increase for the vaporization process by installing 5.5 MW turboexpanders at the Barcelona LNG terminal. [14]

Which type of power generation system to be used can be evaluated considering an exergy analysis. By comparing the two parts of the thermo-mechanical exergy stored in LNG, as described in Section 2.2, a simple criterion for whether the Rankine cycle or the direct expansion cycle is most suitable is obtained. Using Equation 2.9 and 2.10, the criteria can be formulated as follows [22]:

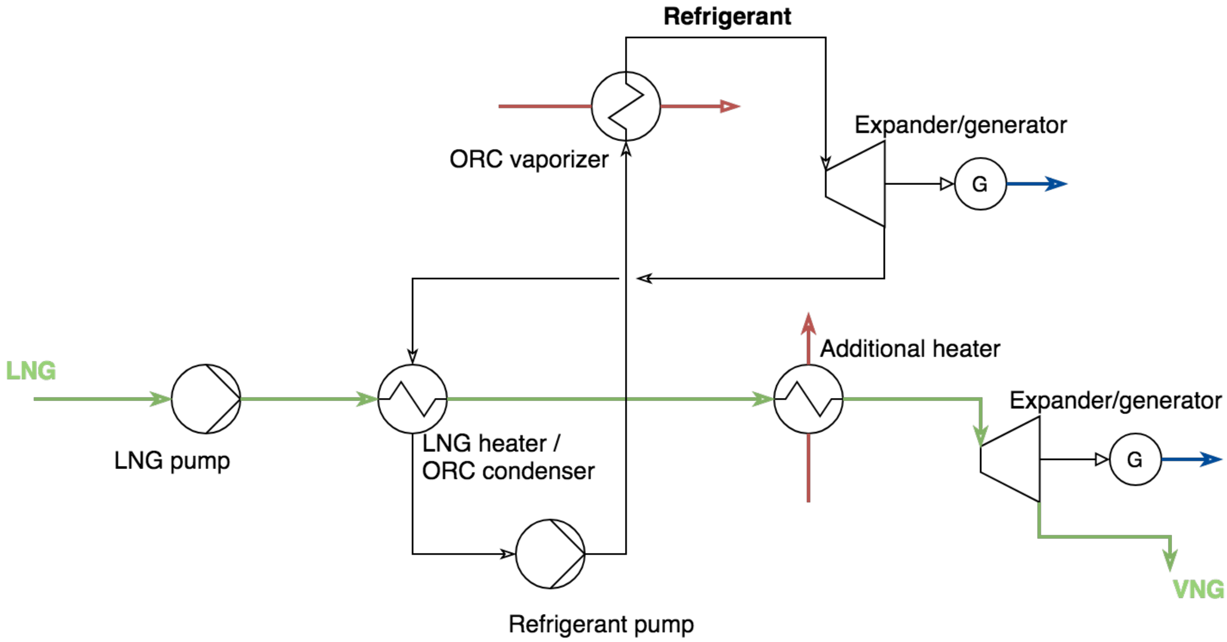
1. When $E_{x,th} >> E_{x,p}$, the Rankine cycle is the most suitable.
2. When $E_{x,th} << E_{x,p}$, the direct expansion cycle is the most suitable.
3. When $E_{x,th} \approx E_{x,p}$, a combination of the two cycles may be used.

The Organic Rankine Cycle (ORC)

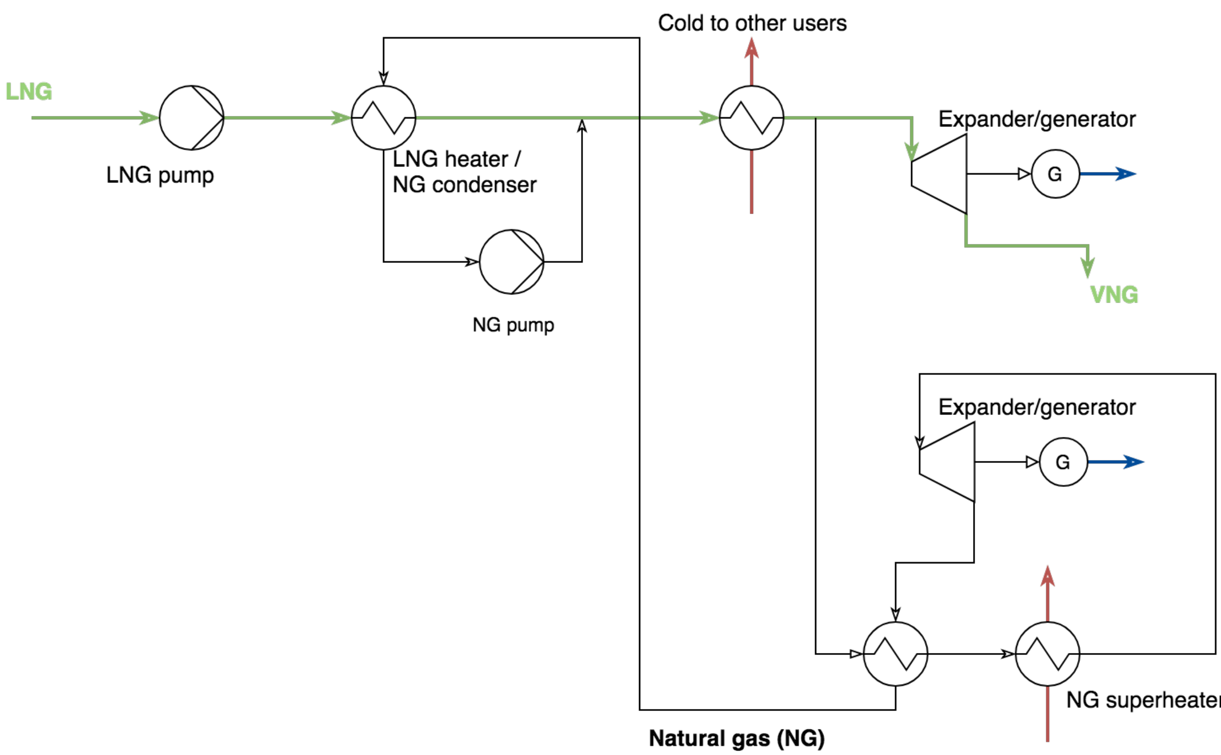
One of the most common cycles used for power generation with LNG today is based on the Rankine cycle, specifically the Organic Rankine Cycle (ORC). It operates by using the cold released during LNG vaporization to liquefy the working fluid of the ORC. The working fluid, which can either be an intermediate fluid or LNG itself, goes through a pumping section, a vaporizing section, and a superheating section before it is expanded through an expander. In the expander, the pressure and temperature are reduced and mechanical power is generated, which further can be converted to electrical energy in a generator [24]. The working fluid should be chosen in words of non-flammable, non-toxic and non-corrosive properties, and should neither increase the risk of danger to the environment. With reference to Section 2.1.1 and [12], CO_2 fulfills the listed criteria and can be considered as a potential candidate. Other possible working fluids are different hydrocarbons, most likely propane, butane or a mixed hydrocarbon, as their availability at the regasification terminal is high.

As mentioned, the working fluid can either be an intermediate fluid or LNG itself. In that sense, the ORC can either be operated as a closed cycle or an open cycle, which is respectively shown in Figure 3.4a and 3.4b. The reason for the extra heater and expander on the natural gas stream is to secure complete vaporization and fulfill sales gas specifications.

By considering the ideal Rankine cycle, an observation of the Carnot cycle efficiency, as stated in Equation 2.12, shows that the lower the temperature of the heat sink is, the higher the efficiency will become. The use of this concept can thus generate a significant amount of power, according to the ultra-low temperature level of LNG.



(a) Closed cycle



(b) Open cycle

Figure 3.4: Typical schematics of a closed and an open ORC. Adapted from [24], with permission granted by Elsevier Inc.

Direct expansion systems
In direct expansion systems, the LNG is used as the refrigerant itself, where the vaporized LNG is used to drive a turbine to produce mechanical power, which further can be converted to electrical energy in a generator. This scheme is often considered as the most simple configuration for power generation and is suitable for small LNG regasification terminals which provide low pressure natural gas. However, the principle of direct expansion is often used to reduce the vaporization pressure to the sales specified pressure after the regasification, as the turboexpanders at the Barcelona LNG terminal.

A compression process is used to increase the pressure of the LNG. Further, the LNG goes through a heat exchange process where it is heated and regasified, often with the use of seawater as the source of heat. Thereafter, the vaporized LNG, high pressure natural gas, expands through a gas turbine which drives the generator. At the final stage, natural gas is reheated to obtain the surrounding temperature, or the given properties for the further distribution of the now vaporized natural gas (VNG) [11]. A simple sketch of a direct expansion system, with the described processes, is shown in Figure 3.5.

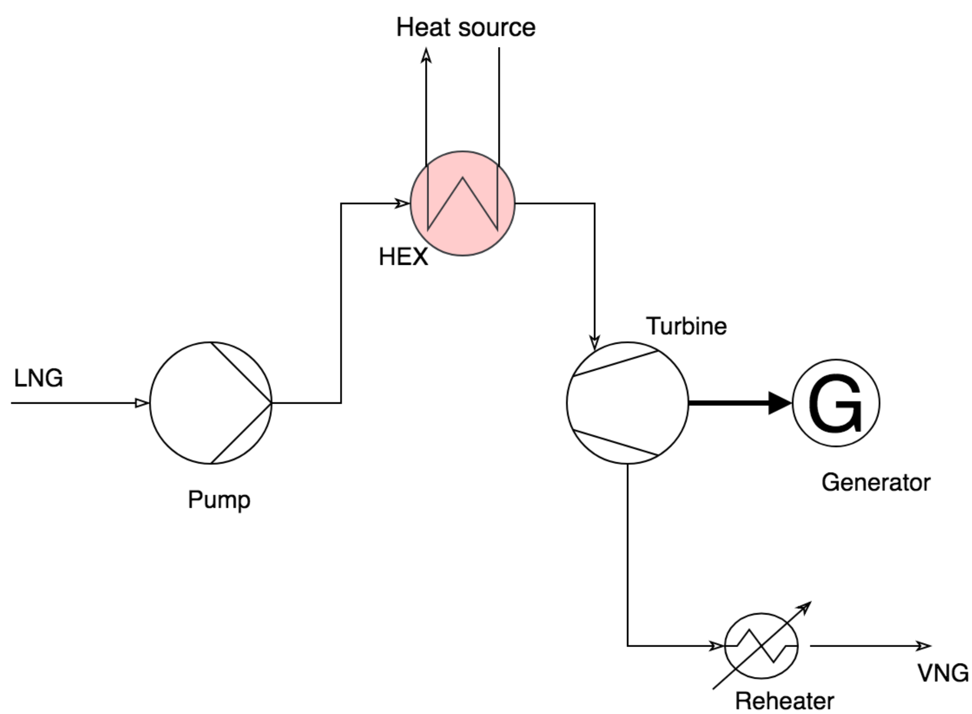


Figure 3.5: A simple direct expansion system for power generation with LNG as the refrigerant.

As mentioned, this type of system is suitable for regasification terminals with low supply pressure. However, it can also be used for higher feed gas pressures by modifying the simple cycle shown in Figure 3.5. This includes the use of multistage expansion and internal heat recovery.

The Brayton Cycle

Research on the use of another thermal power cycle, the Brayton cycle, has become of increasing interest the last years. It has however not been implemented on the same level as the two other described power generation systems. The Brayton cycle is constructed in almost the same way as the ORC, but operates with a compressor rather than a liquid pump. This difference introduces the main advantage with the Brayton cycle; It is more suitable for medium or high grade heat source. The use of a compressor will allow higher temperatures of the working fluid, and the performance will thus increase at higher temperature heat sources compared to the ORC. [14]

3.3 Comparison of present regasification technologies

To summarize this chapter, a table including main characteristics, advantages, and disadvantages of the presented regasification technologies is given in Table 3.1.

Table 3.1: Comparison of the presented LNG vaporization technologies.

Regasification technology	Main characteristics	Advantages	Disadvantages
Open Rack Vaporizer (ORV)	• Mostly used vaporizer in the world • Commonly uses seawater as the source of heat	• Well known and proven technology • Low operational costs	• High investment costs • Pollutes marine life • Requires high quality water • Ice formation on tubes
Submerged Combustion Vaporizer (SCV)	• Exploits hot exhaust gases as the source of heat • The second most used vaporizer in the world	• Lower investment costs • Good control systems and safety features	• Decreases the efficiency of the LNG supply chain • Introduces high operational costs and an increase in emissions (NO_x and CO_2)
Intermediate Fluid Vaporizer (IFV)	• A shell and tube vaporizer using an intermediate heat transfer fluid suitable for low operating temperatures	• Lower requirement of sea water quality and a more compact volume • Lower operating costs and higher environmental friendliness • No icing on racks	• The heat source of the intermediate fluid is limiting
Ambient Air Vaporizer (AAV)	• Utilizes ambient air as the source of heat	• More environmentally friendly, avoids the use of seawater and fuel gases • Low fuel consumption and operating costs	• Highly dependent on unchangeable parameters such as ambient air temperature and humidity • Requires significant heat transfer area and air volumes
LNG vaporization with integrated refrigeration applications	• Utilizing the LNG cold for desalination, air separation, and freezing, chilling and AC applications	• Increases energy efficiency and reduces environmental impact	• Unbalances in energy supply and demand enhances the need for backup systems
LNG vaporization with integrated power generation	• Utilizing the LNG cold as a heat sink for thermal power generating systems	• Increases the efficiency of the total LNG supply chain and reduces the environmental impact • Reduces the need of an external heat source	• Higher investment costs and system complexity

Chapter 4

System design and operation

In this chapter, system configuration proposals for utilizing the LNG cold are presented and described. One basic configuration is designed for end users with refrigeration capacity needed at multiple temperature levels, performing the function of a commercial refrigeration system. The system design is evaluated in regards of a supermarket case, however, it can be applied in several other applications only by changing the temperature levels, making it an applicable system design. Agro food industries serve as an additional candidate, as investigated by La Rocca [31]. There is additionally implemented an auxiliary power generation system to further increase the energy and exergy efficiency of the total cold recovery system. Finally, to evaluate the applicability of the system designs, a transfer of these configurations into regasification of liquefied hydrogen is assessed. But first, an evaluation of the available cold energy stored in the LNG is performed.

4.1 Evaluation of available cold energy

The amount of energy available in the LNG is an important factor for further development and analysis of concepts for cold utilizations. An evaluation was done by using a simplified process of the LNG regasification, considering a pressure rise conducted by a liquid pump followed by a heat exchanger which increases the temperature and vaporizes the LNG. If the heat source for the vaporization process in any way becomes too cold, an additional heater was installed as a backup. The sales gas specifications, or the specifications of the vaporized natural gas, is site dependent, and an expander was therefore installed at the end to optionally reduce the pressure to the preferred level. Figure 4.1 illustrates the described simple regasification process.

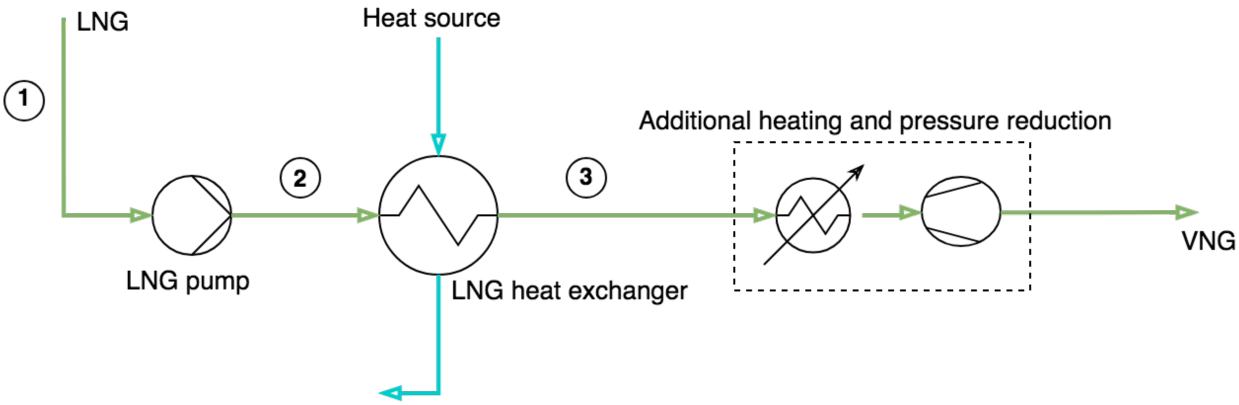


Figure 4.1: A simple schematic of the regasification process.

The process may be presented in a thermodynamic diagram to evaluate the available cooling potential. In Figure 3.3 some typical heat release curves related to the vaporization process of LNG are presented, and as mentioned, both the available cold energy and temperature level varies with the vaporization pressure. The simplest way of showing the mutual dependency between vaporization pressure and available cold energy is by using a logarithmic pressure-enthalpy diagram. Figure 4.2 illustrates the process scheme from Figure 4.1 with corresponding numbering for three different pressure levels. For convenience, the pump was assumed to operate isentropically, and by making use of the chart, this implies isothermal and isenthalpic operation as well.

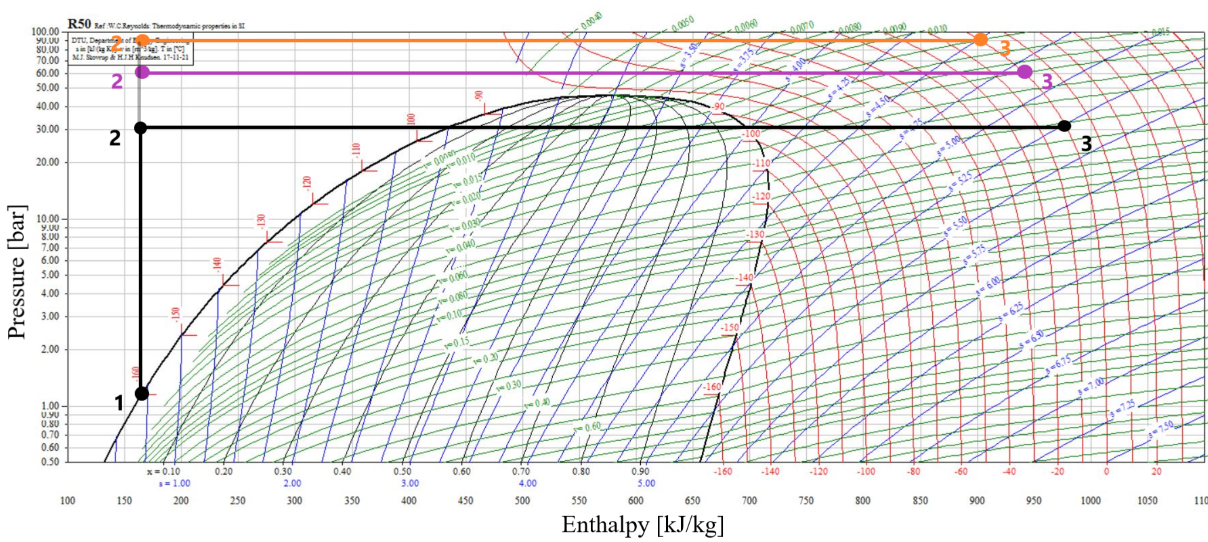


Figure 4.2: The regasification process of LNG illustrated in a logarithmic pressure - enthalpy diagram for $p = 30$ bar (black), $p = 60$ bar (purple), and $p = 90$ bar (orange).

The available cold energy for cold utilization purposes is represented by the change in enthalpy, the horizontal length of the vaporization process 2-3 shown in the diagram. Table 4.1 indicates the change in enthalpy, or the available specific energy, for the presented pressure levels.

Table 4.1: The amount of specific energy available for each presented pressure level, extracted from REFPROP[1].

Pressure, p [bar]	**Change in enthalpy, Δh [kJ/kg]**
30	818
60	776
90	732

Further in this chapter, some case scenarios for utilizing this available cold energy is described and evaluated. The basis for these cases is the conversion of the cold thermal energy presented in this section into low-, medium-, and high-temperature refrigeration applications and auxiliary power generation.

4.2 Commercial refrigeration system

A supermarket is a complex commercial refrigeration system in means of the need for different temperature levels for refrigeration of goods, as well as space conditioning. In a supermarket there are both storages and display units that will need refrigeration systems to function and fulfill the demands for safe storing of food. Like in any buildings there is also a need for air conditioning to ensure a good indoor environment. In general, there are therefore three different temperature levels needed in a supermarket.

- $T \approx -22$ °C, for the case of frozen food, either located in a storage room or in a display unit.
- $T \approx +3$ °C, for the case of perishable food, which needs a given temperature level to obtain their shelf life. The food is stored either in a storage room or in display units.
- $T \approx +21$ °C, for the case of air conditioning to obtain a good indoor climate for both employees and customers in the supermarket.

[1]NIST Reference Fluid Thermodynamic and Transport Properties Database, `https://www.nist.gov/srd/refprop`.

4.2.1 System design for a supermarket

By designing a system that utilizes the excess cold from the regasification of LNG to drive a secondary circuit to distribute the cold into these three temperature levels, one can obtain several benefits. Every utilization of LNG cold can be considered as beneficial compared to the high level of wastage occurring today, and one can assume that this will lead to a higher energy and exergy efficiency, as well as an increase in environmental friendliness. The heat exchanger that combines the two circuits, regasification of LNG and the refrigeration applications, have two different functions; It vaporizes the LNG and works as the condenser unit for the refrigeration system.

When designing such a system there are multiple cares to be taken to ensure system performance and reliability. A possible system solution is based on dividing the vaporization of LNG into several heat exchangers, one for each needed temperature level in the coupled refrigeration system, as shown in Figure 4.3. Each circuit is configured in the same way, where the working fluid of the refrigeration system is condensed through the LNG heat exchanger by the low-temperature heat of LNG. Further, it approaches a liquid receiver, which works as a storage vessel to ensure saturated liquid and vapor when critical. This is the case for the next component, the liquid pump; Gas bubbles occurring in the pump could cause pump cavitation and potential breakdown. The pump is used to ensure a pressure difference in the system and thus establish the needed circulation of refrigerant, and can by this be characterized as the driving force of the system. When the pressure is raised to the appropriate pressure level, the aim of the refrigeration system can be fulfilled by the evaporators. These absorb heat from the refrigerated storages or displays, and from the medium used for the air conditioning system, to evaporate the working fluid. As shown in Figure 4.3, there can be several end users at almost the same pressure level, illustrated by multiple evaporators.

Ahead of each evaporator, there is a valve controlling both the refrigerant flow and the pressure level. There will be timeframes where less cooling capacity is needed at one or more of the end users, and the valve can then close and stop the refrigerant flow. Cautions should hence be taken to the possible appearance of overflow in the system, which can cause backflow and potential destruction of the liquid pump. Therefore, a pressure controlled bypass line is installed with an additional feeding valve to be opened at times with no requirement of refrigeration. Furthermore, a bypass of the natural gas of the associated heat exchanger is as well installed for situations of either no energy demand at the specific temperature level or in case of failure in the system. These situations will most likely not occur simultaneously as a decrease in the needed vaporization capacity of LNG. A backup, as well as a storage system for the opposite imbalance, is thus necessary to accomplish the regasification demand and avoid wastage of cold energy.

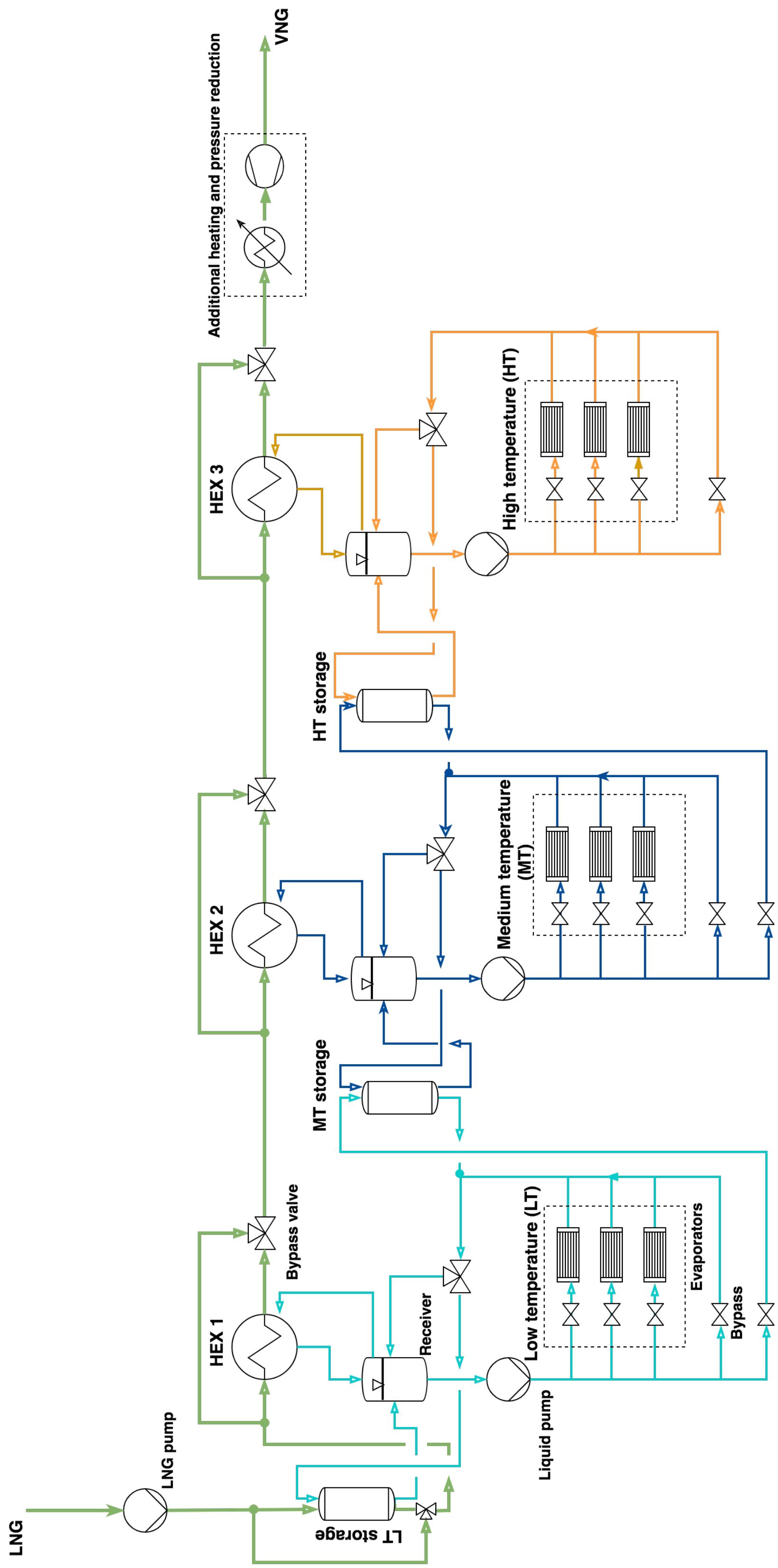


Figure 4.3: A principal sketch of the proposed system solution for cold recovery by refrigeration applications.

Selection of working fluid

Suitable working fluids for supermarket refrigeration should fulfill demands like non-flammable and non-toxic, and most important it must have the needed operating abilities regarding temperature and pressure distribution. Since this case is combined with a cryogenic heat sink, the working fluid should additionally have a low temperature range. One can consider hydrocarbons like ethane and propane, but these may be characterized as both highly flammable and even explosive. They are thus mainly used in smaller or more compact systems where control is easier achieved according to a reduced amount of working fluid. With reference to Section 2.1.1, CO_2 is a good candidate. Also according to the state-of-art of these technologies today, Section 3.2.1, the use of CO_2 is promising. This case scenario is hence performed with CO_2 (R744) as the working fluid.

Storage and backup systems

For this system solution proposal the phase-change material (PCM) technology, described in Section 2.4, is implemented between the three different refrigeration applications, temperature levels, to obtain the necessary storage and backup capacity. The placement of these storage tanks can be seen in Figure 4.3. Different types of PCMs should be chosen based on the necessary temperature level in the storage unit, and some proposed temperature levels with appropriate PCMs are presented in Table 4.2 for the three implemented storage units.

Table 4.2: Proposed temperature levels and appropriate PCMs for the storage units.

Application	Temperature level [°C]	Appropriate PCM
LT storage unit	-55	Carbon dioxide, CO_2
MT storage unit	-30	A water-glycol solution
HT storage unit	0	Water

The proposed cold thermal energy storage (CTES) system is configured as a shell and tube heat exchanger, where the shell side of the exchanger is fully filled with the chosen PCM, and by that works as a storage tank. For the fully charged condition the expansion of the PCM caused by solidification can cause space issues, thus, a liquid supply tank is connected to the storage tank. A schematic of the working principle of these CTES systems is shown in Figure 4.4, where also the special design of the tubes is illustrated. The cold fluid is flowing through the inner part of the tube when charging the storage tank, while the hot fluid is flowing in the small pipes arranged around the inner one when discharging. An evaporation process (heating process for the LT storage unit) of the cold fluid will occur in the inner tubes when charging. This process will absorb heat from the PCM on the shell side and thus formate an ice layer on the surface of the tubes. Further, when discharging, the flow of the hot fluid will make the ice melt while itself condenses due to the heat released from the phase change from solid to liquid of the PCM. Based on the described procedure, the charging takes place at times when the refrigeration demand is lower than the regasification demand, and the discharging at times with the opposite imbalance.

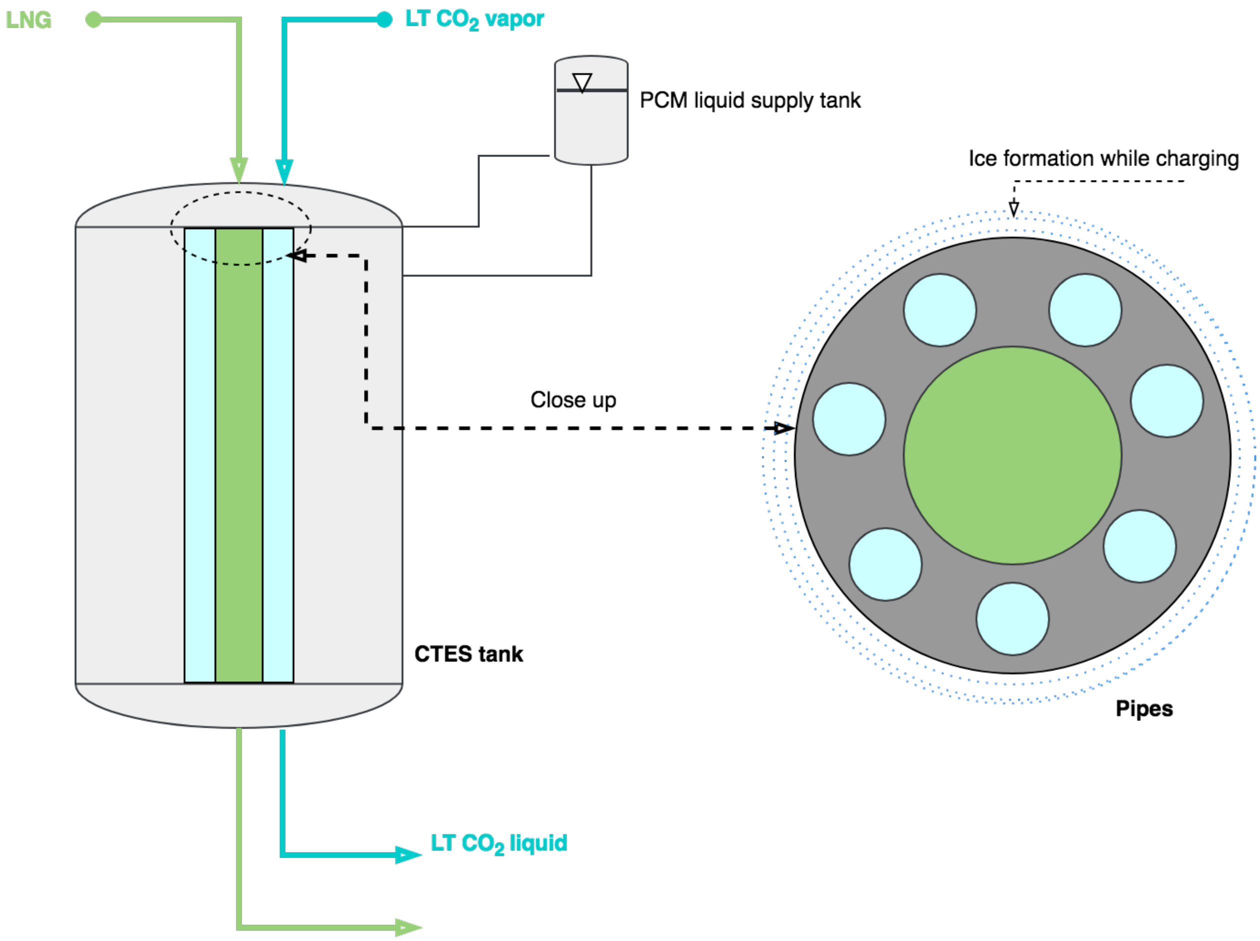


Figure 4.4: A schematic of the working principle of the PCM units, illustrated by the LT storage unit.

4.2.2 System operation

As described in Section 3.2.1, the cooling curves of the end users should match the heating curve of LNG in order to utilize the available cold energy efficiently. In that sense, a temperature-enthalpy diagram showing the heating curve of LNG was made. For this case, a vaporization pressure of 30 bar was chosen, and the heating curve is shown as the green curve in Figure 4.5.

Each of the three refrigeration applications in the system design is chosen to operate with equal condensation and evaporation temperatures, $T_c = T_o$. The liquid receiver operates as the heart of the system at the appropriate pressure level, sending saturated liquid to the evaporating part and saturated gas to the condensing part. As initial values, some proposed pressure levels for the three different applications are presented in Table 4.3.

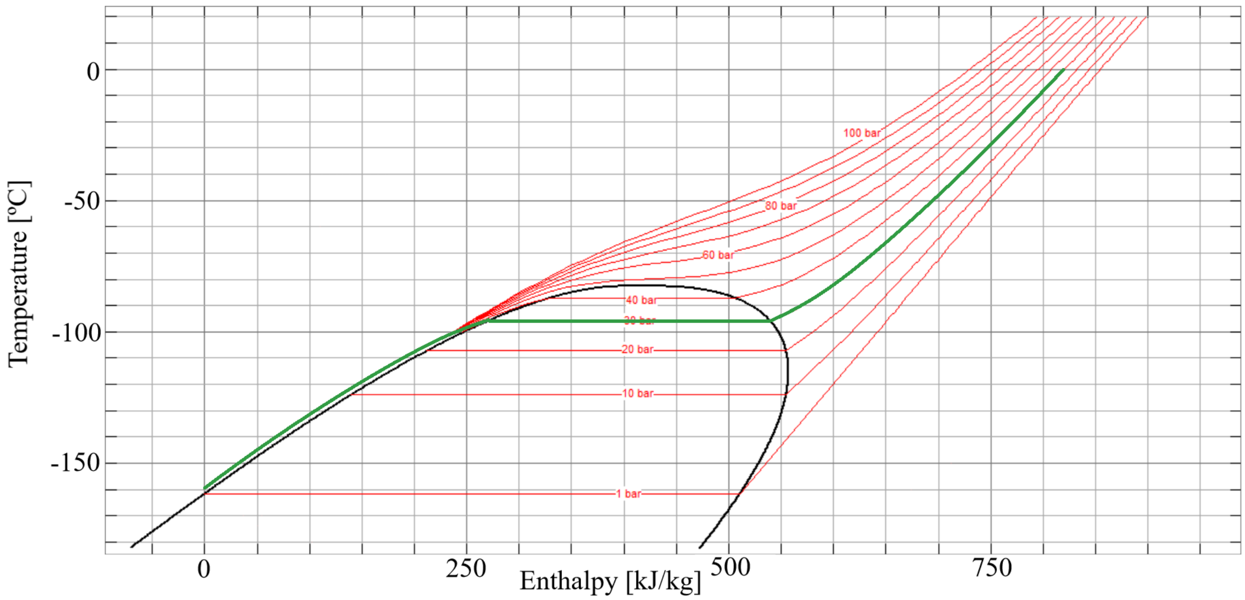


Figure 4.5: The regasification of LNG shown in a temperature - enthalpy diagram.

Table 4.3: Pressure level proposal, with corresponding saturation temperatures and heat of vaporization, for the selected working fluid, CO_2, for the supermarket design.

Application	Pressure level [bar]	Saturation temperature [°C]	Heat of vaporization [kJ/kg]
LT refrigeration	10	-40	322,4
MT refrigeration	26	-10	258,6
HT refrigeration	45	+10	197,2

In the designed supermarket refrigeration system, the vaporization of LNG is mainly divided into four heat exchanger units. First the low-temperature (LT) storage unit, followed by the low-, medium- (MT) and high-temperature (HT) condensers. In all of these four units the working fluid, CO_2, or the PCM is changing phase and thus remains at a constant temperature. The system is designed to make the lowest needed temperature match the lowest temperature part of the LNG, to be able to obtain the closest possible temperature fit. Another important feature to consider is the heat exchange through the two-phase area. It is preferable to have pure liquid or gas at the inlet of every heat exchanger to easily obtain operation reliability and simplify the needed calculations. Hereby the complete phase change of natural gas from liquid to gas should occur in one of the condensers, and according to the temperature fit this is chosen to be the MT condenser. The available cold energy presented in Table 4.1 is divided into four stages based on this evaluation, and a conceptual proposal of this energy distribution is stated in Table 4.4.

Table 4.4: Proposed distribution of the available cold energy from the regasification process at 30 bar for the supermarket design.

Characteristic	Available specific energy [kJ/kg]
LT storage unit	71
LT refrigeration	137
MT refrigeration	391
HT refrigeration	219
Total	818

By this evaluation, the temperatures and specific capacities of the LNG heat exchangers are known, and a temperature fit profile is visualized in a temperature - enthalpy diagram in Figure 4.6. Assuming adiabatic operation, the heat absorbed by the LNG is equal to the heat released by the CO_2 in the heat exchangers, making the massflows the only remaining optimizing factors. These assessments will hence serve as the starting point for the development of the simulation models.

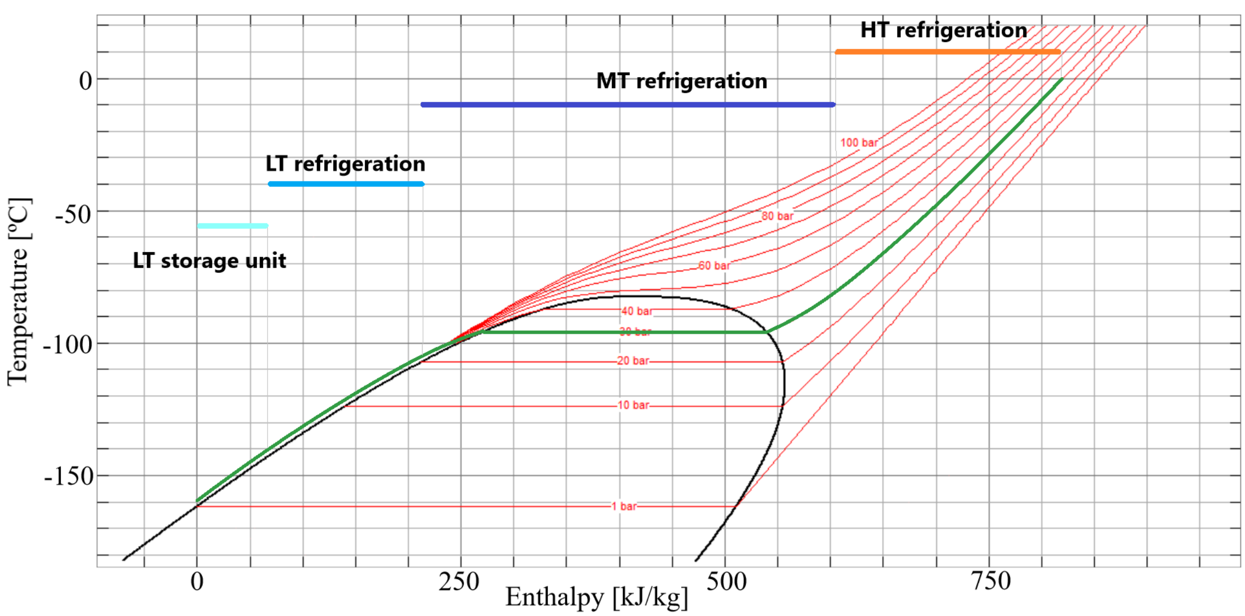


Figure 4.6: Temperature - enthalpy diagram illustrating the conceptual regasification temperature fit for the supermarket design.

4.3 Inclusion of auxiliary power generation

Inclusion of auxiliary power generation can utilize even more of the available LNG cold, exploiting the high amount of thermo-mechanical exergy, as explained in Section 2.2. This will likely increase the efficiency of the total LNG supply chain, as well as reduce the environmental impact and the need of an external heat source.

For this purpose, the Organic Rankine Cycle (ORC) was considered, and the possibility of including this into the previous cases for distribution of the LNG cold was evaluated. The suitable placement of the ORC to exploit most of the available exergy is at the lowest temperature level of the system. At that state, the temperature part of the physical exergy is larger than the pressure part, which verifies the choice of an ORC as the chosen power generation system according to [22], as described in Section 3.2.2. According to the *Carnot efficiency*, Equation 2.12, this also introduces the largest temperature difference between the heat source and - sink and thus the highest efficiency and power output. The ORC condenser, or the LNG heater, as seen in Figure 3.4, was placed after the LNG pump, as the first heat exchanger in the line. Figure 4.7 illustrates the described inclusion.

Thermodynamic considerations
Several options for the type of working fluid for the closed ORC were investigated in means of the criteria listed in Section 3.2.2, and the applicability at the present temperature range. CO_2 was evaluated as the most appropriate working fluid in means of all of the listed criteria. However, it came out short when looking into efficiently heat transfer, as it solidify at a temperature much higher than the approach temperature of LNG as seen by Figure 2.2. As the efficiency of the heat transfer processes highly coincide with the objective of this thesis, a hydrocarbon was seen as a better choice. Methane was hence introduced as the working fluid for this ORC system, which in addition has a obvious reliable availability at the regasification terminal.

Appropriate pressure levels for the ORC depends on the available heat source and heat sink. The heat sink for this case is LNG, and the temperature, T_C, is set to approximately -160 °C. Moreover, the initial choice of the heat source is seawater, which is commonly available at the regasification site. The standard temperature of seawater is 7 °C, which sets the temperature of the heat source, T_H. Assuming a minimum temperature approach in the condenser and evaporator of 5 K states that the standard evaporation process is carried out as a gas heating process for this case, as the critical temperature of methane is about -82.6 °C. This ORC can hence be considered as a transcritical cycle. The condensing temperature is chosen based on how much of the available LNG cold it shall utilize, and should thus be evaluated for each particular case. An evaluation for the purpose of this thesis is conducted as a part of the simulation model and described in Section 5.2.2.

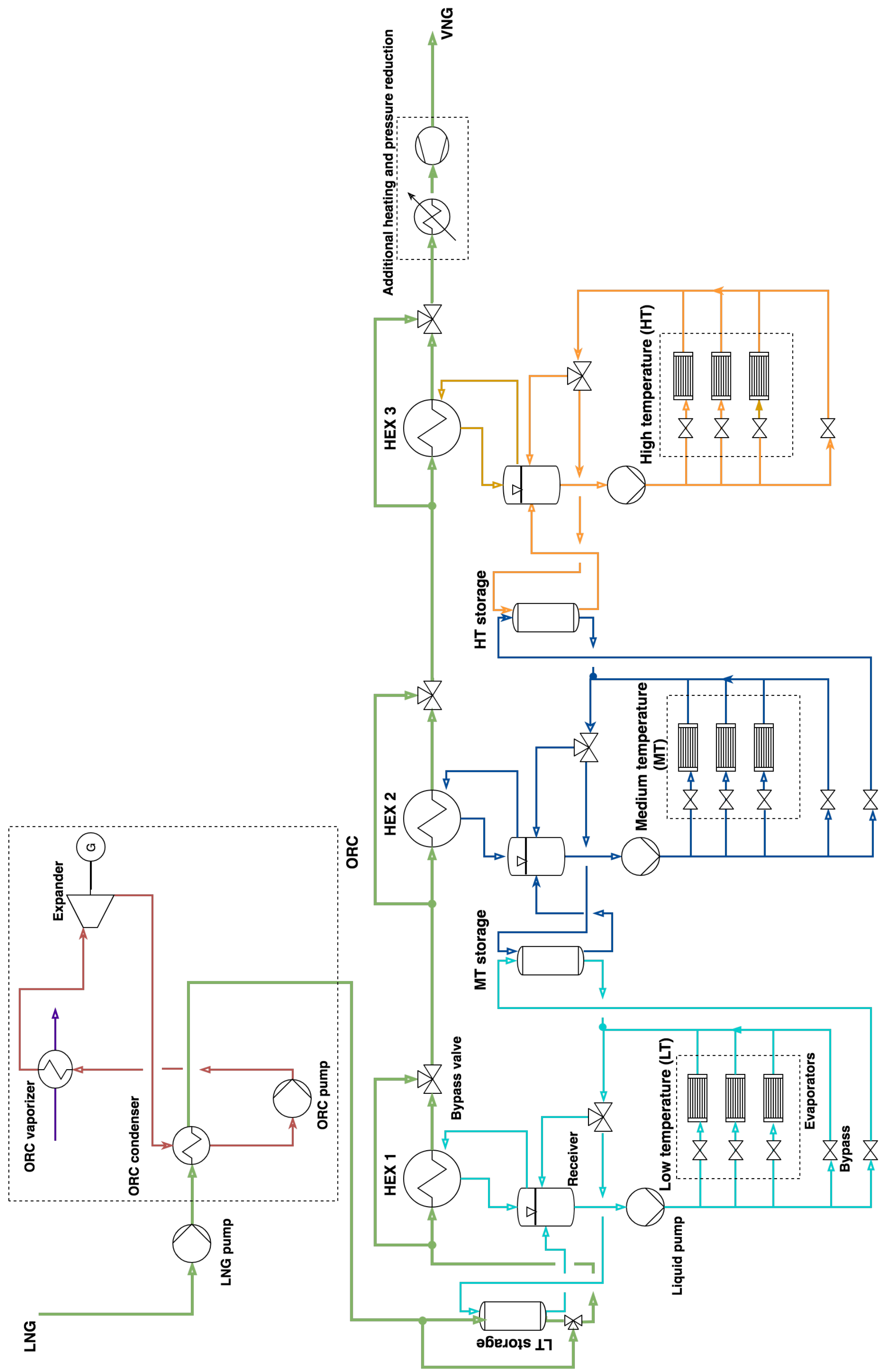


Figure 4.7: A principal sketch of the proposed system solution for cold recovery by refrigeration applications including auxiliary power generation.

Based on the chosen heat source and heat sink, the maximum thermal efficiency (*Carnot efficiency*) of the power cycle can be calculated using Equation 2.12 and is performed in Equation 4.1.

$$\eta_{max} = 1 - \frac{T_C}{T_H} = 1 - \frac{273,15 - 160}{273,15 + 7} \approx 60\% \tag{4.1}$$

4.4 Evaluation of transferring the developed technology into hydrogen terminals

In the development of more carbon constrained energy systems, liquefied hydrogen has become of increasing interest both as an energy carrier and as a fuel. The use of hydrogen in the transportation sector, as well as in energy production, will likely reduce the overall emission density, as it can replace the use of carbon-based fuels and energy sources [1]. However, the liquefaction of hydrogen requires a huge amount of energy, likely an even more intensive amount than for natural gas, as the vaporization temperature at atmospheric conditions is considerably lower, about -253 °C. Thus, a large cooling potential is available at the regasification terminals, and the application of cold recovery systems, as the ones designed in this chapter, is substantial for future investments. In order to compare the regasification processes of natural gas and hydrogen, and to analyze the possibility of transferring the configured cold recovery system to hydrogen terminals in the future, Table 4.5 provides relevant thermodynamic properties. Thus, this section tests the applicability of the developed system configurations.

Table 4.5: Thermodynamic properties related to the liquefaction and further regasification of methane and hydrogen.

	Methane, CH_4	**Hydrogen, H_2**
T_{vap}, at 1 atm [°C]	-161.60	-252.87
h_{fg}, at 1 atm [kJ/kg]	511	447
Critical point [°C / bar]	-82.59 / 45.99	-240.00 / 13.03

As liquefied hydrogen exists at an extremely low temperature level, with a deviation of almost -100 °C from LNG, the temperature fit between the heating of hydrogen and the developed cold recovery systems are not preferable for a complete exploitation of the excess cold. For the supermarket case, with its lowest demanding temperature of -50 °C, a significant temperature difference is present. A change of function and chosen working fluid may have served a more fitted solution for utilizing the excess cold from hydrogen regasification, however, power generation systems and possible air separation can be considered as the best possible cold recovery systems in this remarkably cold temperature range. In case of multiple recovery systems, as found beneficial at the Senboku terminal in Japan, the refrigeration system design could exploit the excess cold at the higher temperature range of the regasification of hydrogen.

Chapter 5

Simulation model

5.1 Simulation platform

For the type of systems handled in this Master Thesis, which can appear as large and complex, a fitted modeling and simulation environment that serves the needed objective is significant. The choice of an appropriate tool fell on the dynamic simulation software Dymola [2], which uses the basis of the object-oriented modeling language Modelica. This serves the goal of modeling the dynamic behavior of technical systems related to multiple domains, such as thermal and mechanical, which is needed for this work.

In Dymola, components from different domains are put together in Modelica libraries. Important libraries for the development of the simulation models of the system designs described in Chapter 4, are mainly TIL Media and TIL 3.5, which is designed by TLK-Thermo GmbH, together with some features packed in the standard Modelica library. TIL Media contains thermodynamic properties of several common refrigerants, gases, and liquids, whereas TIL 3.5 is a library with predefined components for construction of thermal systems. These libraries together fulfill the necessary building equipment for the presented system designs and make the evaluation of their dynamic behavior possible.

The TIL and Modelica libraries have a set of default colors for different features, providing easy recognition of media types and signals. These features with their accompanying description and color code are presented in Table 5.1. However, the calculation method of Dymola do not include the essential pipelines between the thermodynamic models; The conditions at the outlet of one component is exactly the same as at the inlet of the next component. Thus, the connections are only a visualization of the appearance of pipelines.

[2] Dynamic modeling software by Dassault Systèmes, `https://www.3ds.com/products-services/catia/products/dymola/`.

Table 5.1: Description and color code of available medium types in TIL Media and other relevant connections in the TIL and Modelica libraries.

Feature	Description	Color code
Gas	Ideal gases / gas mixtures	Orange
Liquids	Incompressible single-phase fluids and mixtures	Blue
VLE Fluids	Compressible two-phase fluids, vapor/liquid, pure substances / mixtures	Green
SLE Mediums	Solid, liquid or solid-liquid equilibrium	-
Heat flow	-	Red
Signals	Transfer of a real expression	Dark blue
Logic signals	Transfer of a boolean expression	Pink

The TIL library is developed with the intendancy to construct and simulate thermodynamic systems, both for stationary and transient situations. Versatile components ensure detailed calculations of even large and complex systems, making the simulation both precise and fast. However, it is highly dependent on the chosen integration method and its interval time and tolerance, which serve as the most important optimizing measures for an efficient simulation model. The available solvers act differently for a given type of model and should be chosen based on the range of time scales, size, and complexity of the system. Explicit solvers works best for a narrow range of time scales, while for a mixed range an implicit solver would be the preferred choice. Model stiffness is an effect one should be aware of in a large and complex system, as the one handled in this thesis, which occurs as the chosen numerical method becomes numerically unstable [36]. Two main choices exist to avoid this; The step size should be reduced to an extremely small level if using an explicit solver, or an implicit solver should be used [9].

The overall advice from TLK-Thermo is to use the implicit and multi-step method Dassl, and the advice is followed up in this thesis. However, different interval lengths and tolerances have been investigated in order to achieve the best performance, as a single-step integration method could have handled the large and complex model even better. For such a method, the solution at the current stage is only influenced by the the previous step, which makes such solvers more effective when handling simulation with many events [9]. An event is the terminology used for a system change, which causes the solver to reset and further reduce its step size to located the exact time of the incident. As this reset reduces the simulation speed, an acceptable tolerance of the integration method should be assessed. For the simulation scenarios carried out in this thesis, a tolerance of 1e-4 was set, together with a number of 500 output intervals independent of the simulation time. This also coincides with the default settings provided by TIL 3.5.

5.2 Description of the Dymola model

As seen in the system design proposal, the complete system consists of three almost equal loops with different temperature levels to fulfill the refrigeration demand in the particular case. The simplest way of modeling this was thus to use a part by part procedure starting from the left by the LT loop, making a basis for further duplications into the two other refrigeration loops. Furthermore, the related connections between the loops were attached, introducing a valuable tool to evaluate the performance of the complete refrigeration system. Troubleshooting and tuning along the modeling process are of highly importance, and the process didn't proceeded before the components and parameters operated correctly in the considered matter.

Several cares was taken to make the model as close to real life as possible in order to evaluate the potential energy savings in a real perspective. However, multiple assumptions and simplifications had to be made in order to obtain a workable model within the given timeframe. The most tellingly may be no pressure drop in pipelines and components, as well as adiabatic operations. In the heat exchangers, constant heat exchange coefficients were chosen, and the pump efficiencies were set to a constant value of 0.4, based on the defined default values in the TIL library. Additionally, the drive, or motor, efficiencies of the pumps were set to 1, assuming no inefficiencies in the conversion of supplied electric energy into mechanical energy.

Table 5.2: Heat transfer coefficients used in the heat exchangers in the Dymola model.

Fluid	Plate [$W/m^2 \cdot K$]	Fin and tube [$W/m^2 \cdot K$]	Storage tanks [$W/m^2 \cdot K$]
Refrigerant (CO_2)	2000	2000	1500
LNG (methane)	2000	-	1500
Gas (moist air)	-	100	-

5.2.1 The commercial refrigeration system

The Dymola model of the LT refrigeration loop is shown in Figure 5.1, which has served as the basis for the other two refrigeration loops. Later in this section some chosen parts of the system is illustrated and described in more detail, however, the complete Dymola model with all of the related control strategies can be found in the Appendix B. First, an overall description of the development of the LT refrigeration model together with its related control strategies is performed.

The connecting component, the condenser, which performs two of the main functions in the system solution, regasification of natural gas and condensation of refrigerant, may be considered as the most important component in the system. Therefore, this component started out as the basis for the rest of the simulation model. The initial values stated in Chapter 4 was set as boundary conditions, such that the heat exchanger operated in the right area in the phase envelope according to the design. Further, plate heat exchangers were chosen for all three condenser units, based on advice from experienced Dymola users. This type of heat exchanger can also be categorized as compact, which makes it economically favorable for this particular usage that may contain large transfers of heat.

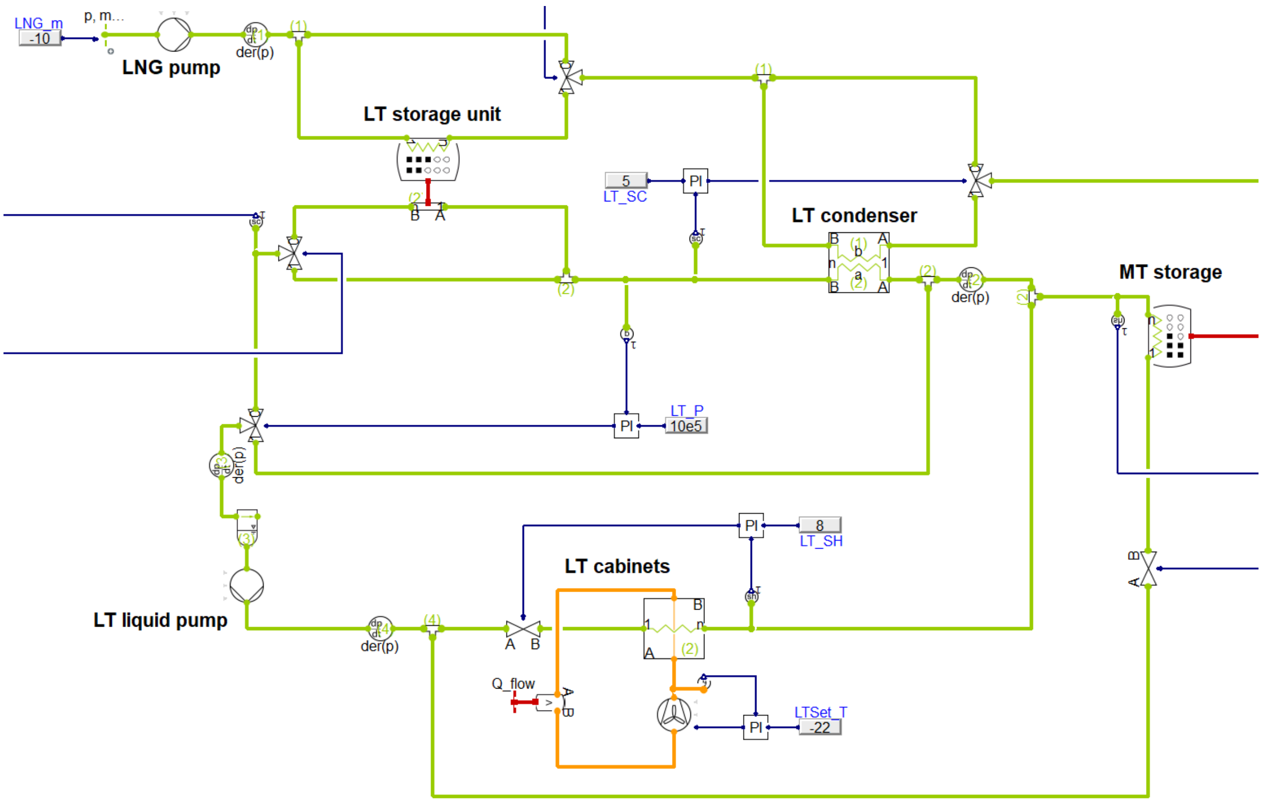


Figure 5.1: Dymola model of the LT refrigeration loop.

Furthermore, some adjustments were applied to the separator as the designed version shown in Figure 4.3 has as many as five ports, which may cause problems for the simulation as the separator stored in the TIL library only has three ports. For that reason, either an adapted separator model had to be designed or an alteration of the system design had to be done. The last option was chosen, and the insurance of complete evaporation was achieved by a superheat control and the additional condensation process conducted by the LT storage tank was placed in series with the main condenser. The separator now serves only the function of ensuring complete liquid refrigerant into the liquid pump, as well as a storage vessel for times of service or failure in the system.

For controlling the pressure level, a bypass of the condenser was introduced in order to stabilize the condensing pressure. The control strategy is based on a PI controller that can change the position of the three-way valve, operating as the bypass valve, in order to hold the condensing pressure stable and equal to 10 bar. An issue that occurred during tuning of this controlling strategy was the amount of subcooling after the condenser. As the fraction of bypass increased, the amount of massflow available for the heat transfer in the condenser decreased. In order to keep the load balance in the refrigeration loop, the refrigerant side of the condenser compensated by a higher specific energy. Hence, a large amount of subcooling was introduced. Due to the very low temperature range of LNG, CO_2 dry ice is an issue one should be aware of at this point as Dymola doesn't notify this. A solution to the subcooling problem became another bypass, now at the LNG side of the heat exchanger, with a bypass valve controlling the amount of subcooling. A PI controller fulfills this function and sets the subcooling to a value of 5 K. The implemented bypass lines can also be considered as a realistic approach, as there will be necessary to close the refrigeration loop in cases of service or failure in the loop.

The liquid pump sets the needed circulation of refrigerant by introducing a pressure rise of 4 bar before it approaches the evaporating part of the system. Furthermore, feeding valves control the amount of refrigerant approaching the refrigerated cabinets and the cold energy storage for the next refrigeration loop, and reduces the pressure to the correct level set by the pressure control system. The feeding valves are controlled by a PI controller varying the effective area of the feeding valves in order to obtain a superheat of 8 K after the evaporative equipment. This is done to ensure a complete evaporation as a compensation for the alteration of the separator unit, as described earlier.

LT storage unit

The function of the LT storage tank is multiple; It serves a part of the regasification process at the same time as it stores cold energy for further usage (charging), and it can operate as a condenser for the LT refrigeration loop (discharging). In order to control the charging and discharging of the tank, the flows conducting these functions need to be controlled, and a bypass line on each side of the tank made it possible. The Dymola model of the LT storage system is shown in Figure 5.2.

For the charging part of the tank a switch control based on the liquid amount of the total SLE medium mass was applied, making the flow of natural gas present in the line of the tank (position 1 of the three-way valve). Whenever this amount reaches 10% or less of the total mass, the position of the valve changes to 0, with a first order damping with a cut-off frequency of 0.001 Hz, and the storage tank is considered as fully charged.

The placement of the additional condenser tube connected to the storage tank (backup) deviates from the system design because of the alteration of the separator. Several possible options were evaluated; It could be placed in parallel, in series, or partly in series with the main condenser. The latter option was chosen, where the partly series connection is accomplished by a bypass structure as seen in Figure 5.2.

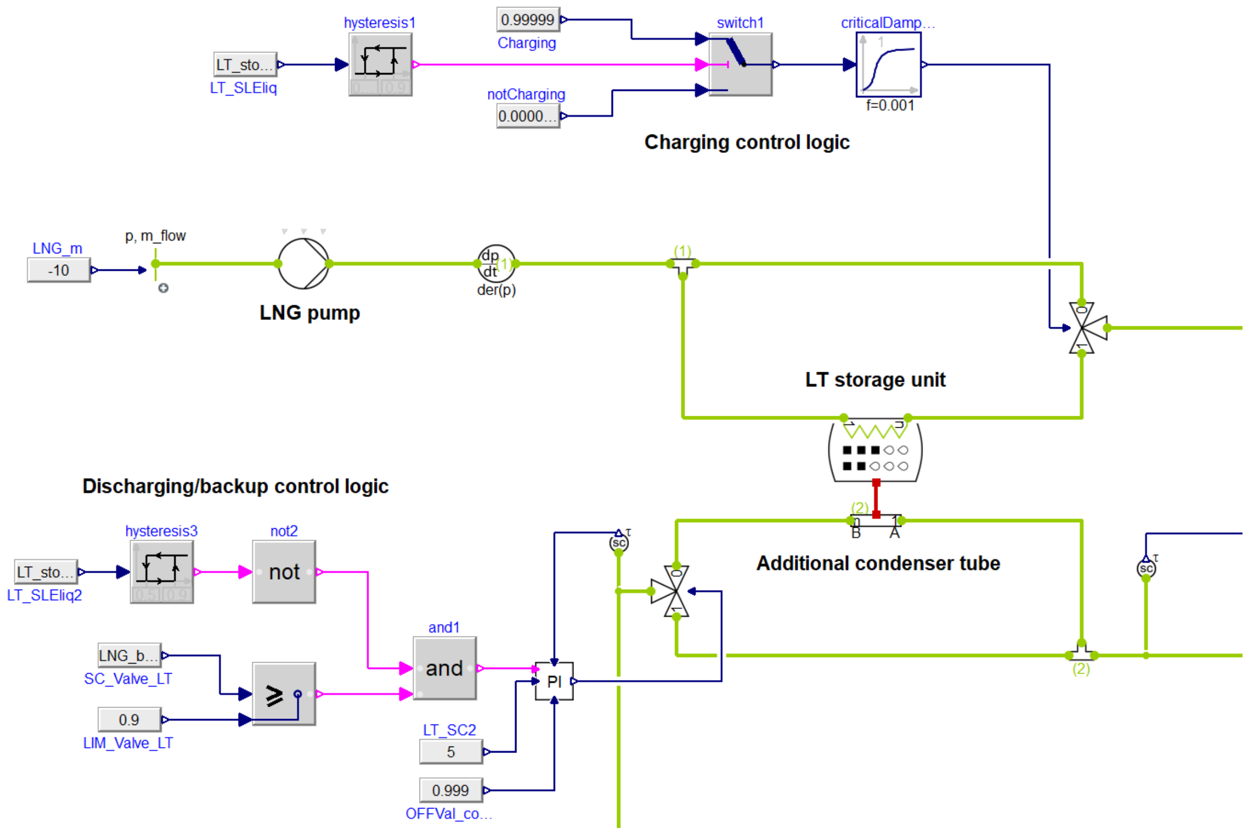


Figure 5.2: Visualization of the LT storage placement and control strategy.

The discharging of the tank only occurs in situations where the regasification process can't cover the complete refrigeration capacity of the LT refrigeration loop. When the regasification rate is high enough, the bypass valve controlling the flow into the tube representing the additional condenser is set to complete bypass (position 1 of the three-way valve). A PI controller is controlling the valve position, and is activated when the subcooling of required 5 K after the main condenser isn't reached. The logical activation strategy of this PI is twofold; The storage tank needs to have obtained 50% or more of the fully charged condition, and the position of the bypass valve of the main condenser controlling the needed subcooling should be close to 1, zero bypass.

The PCM used in this storage tank is an eutectic PCM designed by PCM Products Ltd with a design phase change temperature of -49.8 °C called E-50 [28]. This deviates from the design solution, which stated that CO_2 should be used as the PCM with a phase change temperature of -55 °C. However, as the solidification of CO_2 and its behavior around the triple point is not yet adequately investigated [15], and for simplification purposes for the simulation process, an eutectic water-glycol solution from a verified producer was chosen.

The LT storage tank was dimensioned to cover the complete LT refrigeration capacity for a period of 24 hours. A simple calculation gave a storage capacity of about 480 kWh, which implies a needed tank volume of 6 m^3. In order to get a fast and reliable charging of the tank, the heat transfer area based on the number of tubes in the tank should be optimized. For this case a selected minimum spacing between the tubes of 5-10 cm was considered, and a dimensioning charging time at a regasification rate of 10 kg/s of roughly 12 hours were set. Applying both of these considerations gave a heat transfer area for the LT storage tank of approximately 3.7 m^2.

Cold storage rooms and display units

A preferable way to model the cold storage rooms and displays is to gather them into one single volume. Hence, only one evaporator was necessary for the model solution, compared to the visualization of multiple ones in Figure 4.3. This overall volume is represented by an air volume tank, where the total heat load of the refrigerated spaces is applied as a continuous heat source. Moreover, a control strategy was necessary, and the obvious setpoint of such a system is the required storage temperature. Thus, the fan operating in the storage rooms or displays is regulated by the storage temperature fulfilled by a PI controller, as illustrated in Figure 5.3. The fan is hence controlling the flow of air through the CO_2 evaporator in order to exchange the right amount of heat to obtain the given setpoint. This way of modeling the cold storage rooms and displays is inspired by [32, 33] and based on an example given in the TIL library. Table 5.3 presents the characteristics for the refrigerated spaces for each refrigeration loop.

Table 5.3: Design capacities and temperatures of the cold storage rooms and display units.

	Design refrigeration capacity [kW]	**Design temperature [°C]**
LT	17.5	-22
MT	65.0	3
HT	100.0	21

As mentioned, the evaporation process of CO_2 is controlled by the effective area of the feeding valve in front of the evaporator in order to obtain an amount of superheat of 8 K. However, as this refrigeration loop is pump circulated and do not contain a compressor, the amount of superheat is not a critical value.

For this design, direct evaporators are used, which are conducted by crossflow fin and tube heat exchangers. This is a good solution for the freezers and chillers, however, a more applicable approach for the HT evaporator is to use an ice water cycle as the source of heat, which can further distribute the cold out to the air conditioned spaces. Introducing this indirect refrigeration system, the ice water loop can additionally fulfill other demands at the end user or the regasification terminal. A potential application, which is discussed in Section 7.2.2, is to supply the attached ORC with the necessary heat source for its vaporization of refrigerant. By making this assessment, which increases the level of interconnections in the system design, a reduction in necessary external utilities can be expected. However, for simplicity, this was not applied in the simulation model.

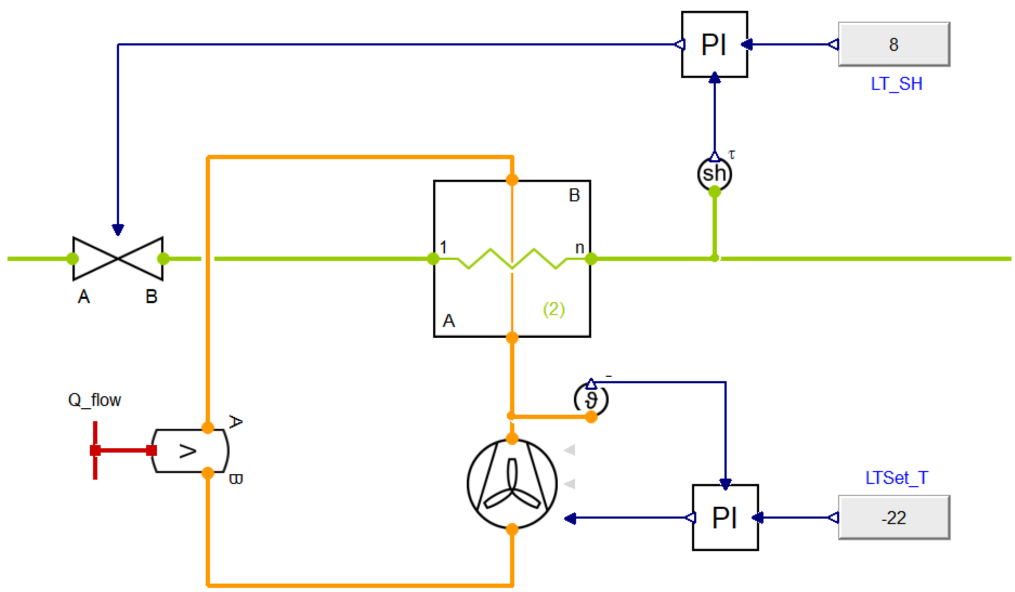


Figure 5.3: Model solution of the cold storage rooms and display units, illustrated by the LT refrigeration loop.

The other storage units

The other two storage units, MT and HT storage, are the connecting parts between the refrigeration loops as described in Section 4.2.1. These operates in the same way as described for the LT storage, but have some other features for the control strategies.

As the storage units are not infinitely large they will at a time become fully charged and the feed should be closed. By applying a control logic that shuts down the PI controller, which controls the amount of superheat after the storage, and sets the opening of the feeding valve to a fixed, very small value, this was solved. Using a hysteresis, which transforms a real signal to a boolean one, with a liquid fraction of 10% of the PCM present in the storage unit as the triggering factor, served this purpose. The placement and connection between the refrigeration loops together with the control strategies are shown in Figure 5.4 by the MT storage system.

The discharge process of the storage tank is controlled in the same way as for the LT storage tank, where the available cold energy in the regasification part operating in the related range is too low to cover the needed refrigeration. The dimensioning of the tanks was also based on the same decisive considerations as for the LT storage, and an overview of the dimensioning factors of all three storage tanks are summarized in Table 5.4.

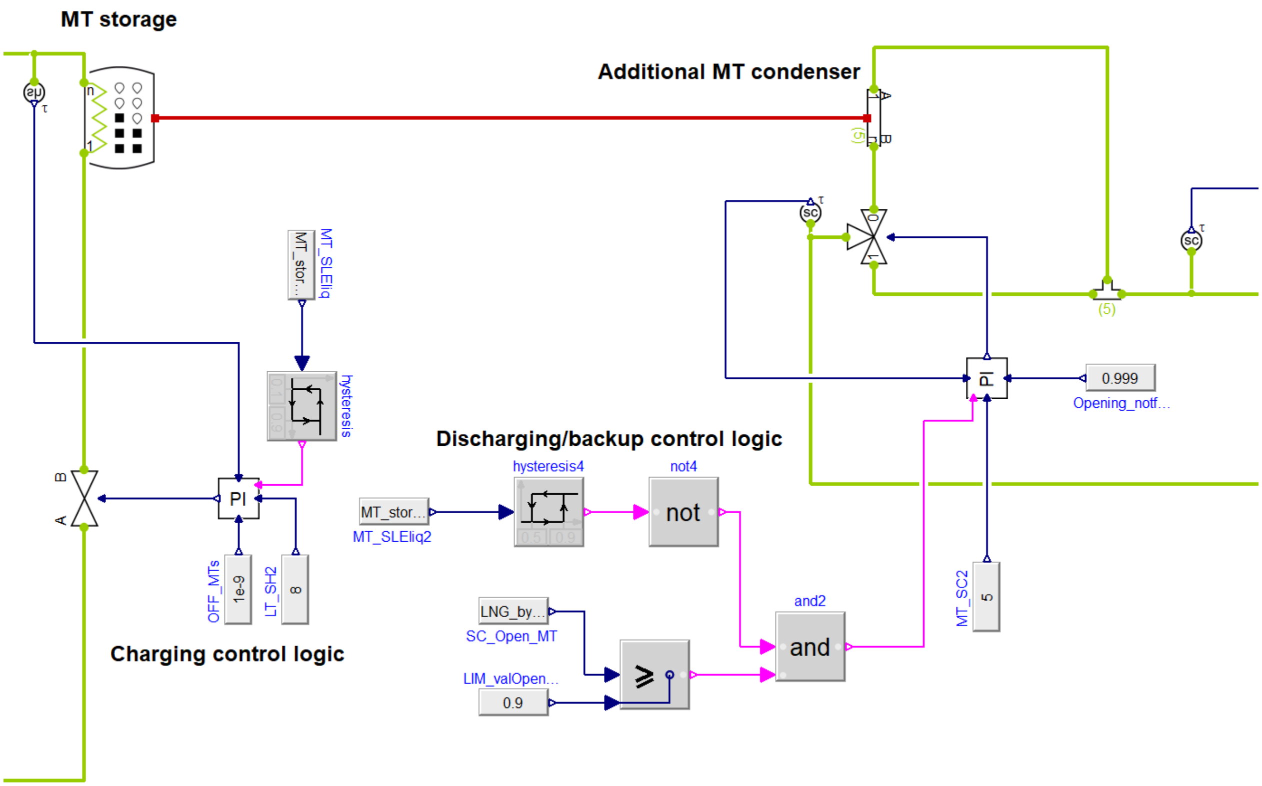


Figure 5.4: Modeling principle of the MT and HT storage systems, illustrated by the MT storage system.

Table 5.4: Dimensioning factors of the storage units.

	Storage capacity [kWh]	Tank volume [m^3]	Heat transfer area [m^2]	PCM [-]
LT	480	6	3.7	E-50 [28]
MT	1440	16.5	262.4	E-29 [28]
HT	2400	28	408.2	Water

5.2.2 Auxiliary power generation

For the inclusion of the auxiliary power generation system described in Section 4.3, a simplified model in Excel using the REFPROP database for accurate thermophysical properties was developed. The output of this model, the outlet properties of the regasification process part, was set as the inlet conditions for the connected refrigeration system, as seen in Figure 4.7, in the

Dymola model. Table 5.5 states the chosen features and assumptions used in the development of the ORC model.

Table 5.5: ORC model features and assumptions.

Feature	**Value**	**Unit**
Refrigerant	Methane	-
Condensing pressure	36.4	bar
Gas heater pressure	172.1	bar
Heat source temperature	12 - 7	°C
Isentropic efficiency - pump	0.9	-
Isentropic efficiency - turbine	0.85	-
Minimum temperature approach in HEX	5	K
Regasification demand	1	kg/s

The idea of including the auxiliary power generation system was to make use of the maximum available cooling capacity. For that reason an analysis of the total claim by the refrigeration system was done and compared to the total available cooling capacity from the regasification process at 1 kg/s. The remaining cooling potential was thus designed to be conducted by the ORC, which gave an outlet condition of T = -95.88 °C and h = 434.9 kJ/kg for the regasification process. By Figure 4.5 one can observe that this state is located in the two-phase area, and it is, as mentioned, not favorable to design a system that operates with two-phase flow in the pipes between the heat exchangers in regards to necessary calculations and operation reliability. However, for this case the aim is to cover the whole regasification process of LNG with related cold energy recovery systems to investigate the energy savings potentials. According to that, the stated outlet conditions was applied.

The thermodynamic process for the modeled ORC with the given features described in Table 5.5 is visualized in a logarithmic pressure - enthalpy diagram in Figure 5.5.

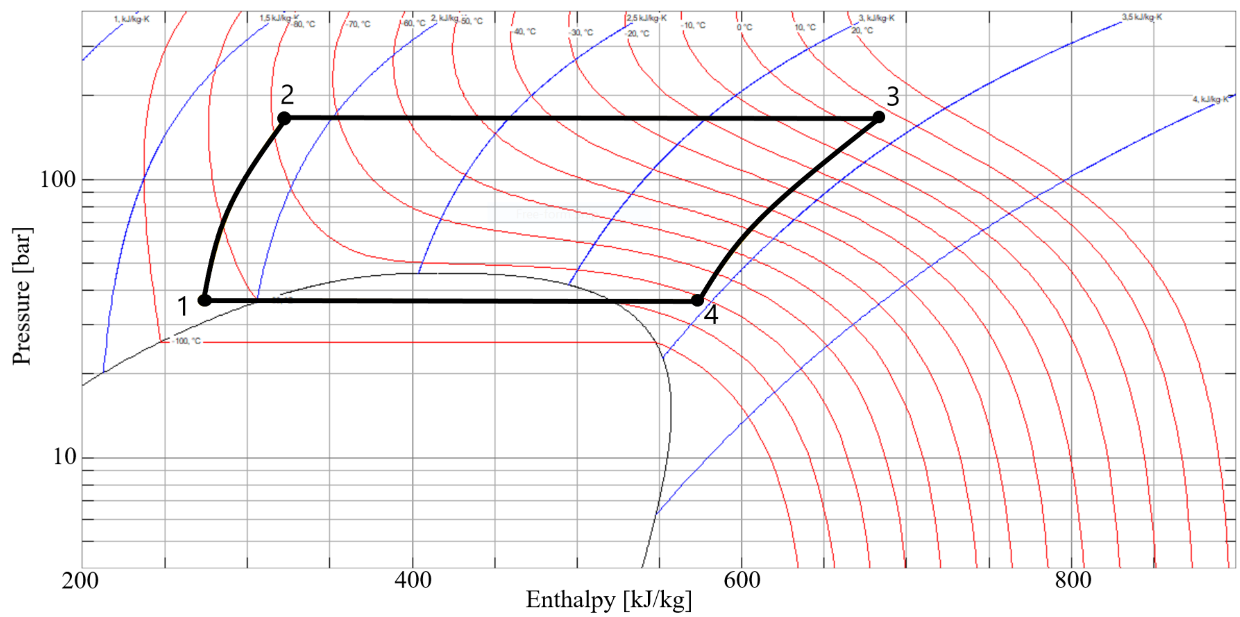


Figure 5.5: The designed ORC model presented in a logarithmic pressure - enthalpy diagram.

5.3 Simulation scenarios

To investigate the potential of cold energy recovery from the regasification of natural gas, a scenario analysis of the designed system based on variable regasification demand was proposed. Three various regasification demands were chosen based on three different criteria. First, a partly realistic demand was elected (Case 1). Further, a massflow that simplifies the related calculations and the ability to upscale or downscale the system design was proposed (Case 2). Finally, a regasification demand near the best performance state of the designed cold recovery system was included (Case 3). The ORC was designed in order to cover the whole regasification process of LNG, together with the refrigeration system, at a regasification demand of 1 kg/s, and hence operates as Case 4. The cases considered are rendered in Table 5.6.

Table 5.6: Simulation scenarios applied in the thesis.

Case	Regasification demand [kg/s]	Cold recovery system
1	10	Refrigeration
2	1	Refrigeration
3	0.53	Refrigeration
4	1	Refrigeration and ORC

Chapter 6

Results

In this chapter, relevant results from the Dymola simulations are presented, with focus on the recovery potential of the LNG cold. This includes the connecting heat exchangers and storage tanks, and the performance of these components are emphasized in order to verify the reliability of the developed simulation model. The inclusion of the ORC model is as well analyzed by means of the utilization level and potential energy savings. In addition, eventual savings in operational costs regarding necessary electricity supply for functioning pumps, together with the related reduction of CO_2 emissions, are evaluated.

The presented cases are based on the scenarios described in Section 5.3.

6.1 Cold energy recovery potential

The main objective of this thesis entails how much cold energy that can be recovered with the configured system designs. As found in the literature survey, there is a small degree of cold energy recovery at existing regasification terminals, and the recovered cold energy in this thesis will thus be compared to the worst case scenario, that all of the cold energy is wasted to sea.

The maximum available cooling capacity at each of the four temperature levels present in the refrigeration system, at various regasification demands, are presented in Figure 6.1. A huge potential at each refrigeration temperature can be observed, implying that there is an enormous wastage of cold energy today. Each of the recovery heat exchange processes presented are assumed to have a minimum temperature approach of 5 K, which serves as the limitation for how much of the available cooling capacity that can be utilized at each temperature level. It may be noticed that the available cooling capacity for the HT refrigeration coincides with the total available potential, as the HT condensing temperature is more than 5 K higher than the evaluated VNG outlet temperature of 0 °C. A wastage of about 80 MW cold energy at a typical regasification terminal, with a regasification demand of 10 kg/s, is hence true for existing terminals.

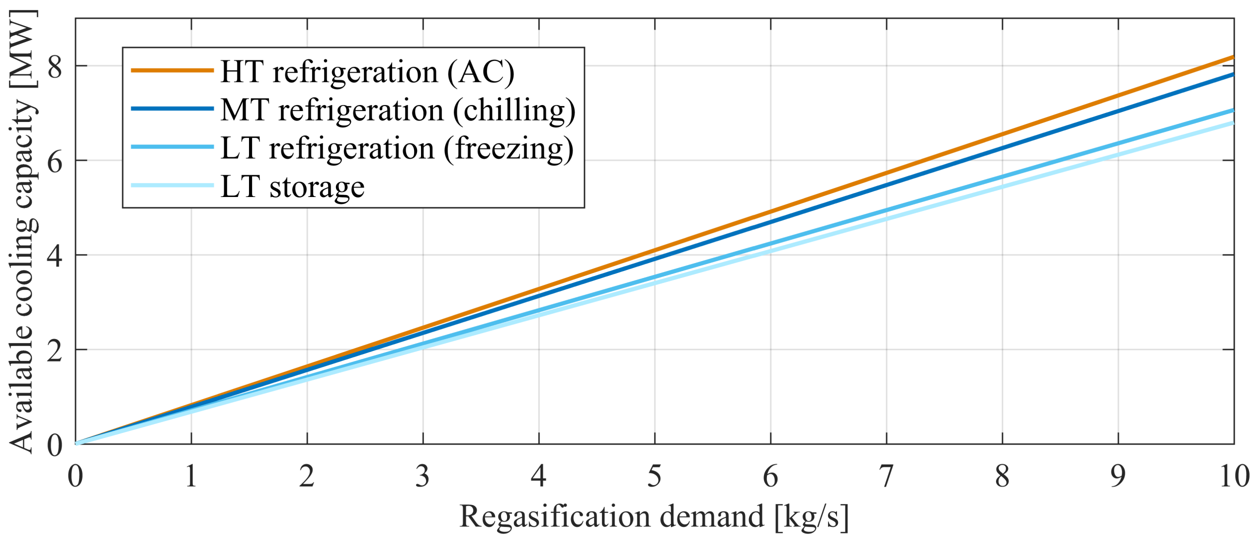


Figure 6.1: Visualization of the total available cooling capacity at each refrigeration application for various regasification demands.

6.1.1 Refrigeration system

The designed refrigeration system consists of four different subsystems recovering the available cold energy from the regasification process and hence vaporizes the LNG. First, the LT storage unit releases heat as the PCM located in the storage tank solidify. Second, three heat exchanger units, operating as condensers for the related refrigeration system, either for freezing, chilling or AC for the evaluated supermarket, releases heat as the refrigerant condenses. Figure 6.2 shows how much of the total available cooling capacity that is utilized at each recovery subsystem, and in total, for the three chosen regasification demands stated in Section 5.3 (Case 1, 2 and 3). The assessed situation includes charging of all three storage tanks, which provides the maximum utilization level of the designed refrigeration system.

A total utilization level of approximately 92% can be observed for Case 3. By comparison, the most realistic scenario, Case 1, only utilizes about 5% of the available LNG cold. Therefore, the designed refrigeration system is considered too small in order to cover a realistic regasification demand of 10 kg/s, or even 100 kg/s as stated in Chapter 3. More end users or more refrigeration load should hence be present in order to fulfill the necessary heating at a typical regasification terminal today. By making use om Figure 6.1, which shows a total cooling capacity of about

80 MW at 10 kg/s, approximately 435 supermarkets, with the same refrigeration capacities as designed, must be present for a complete utilization of the LNG cold. For this specific assessment, the increased load caused by the charging of the three storage tanks is neglected.

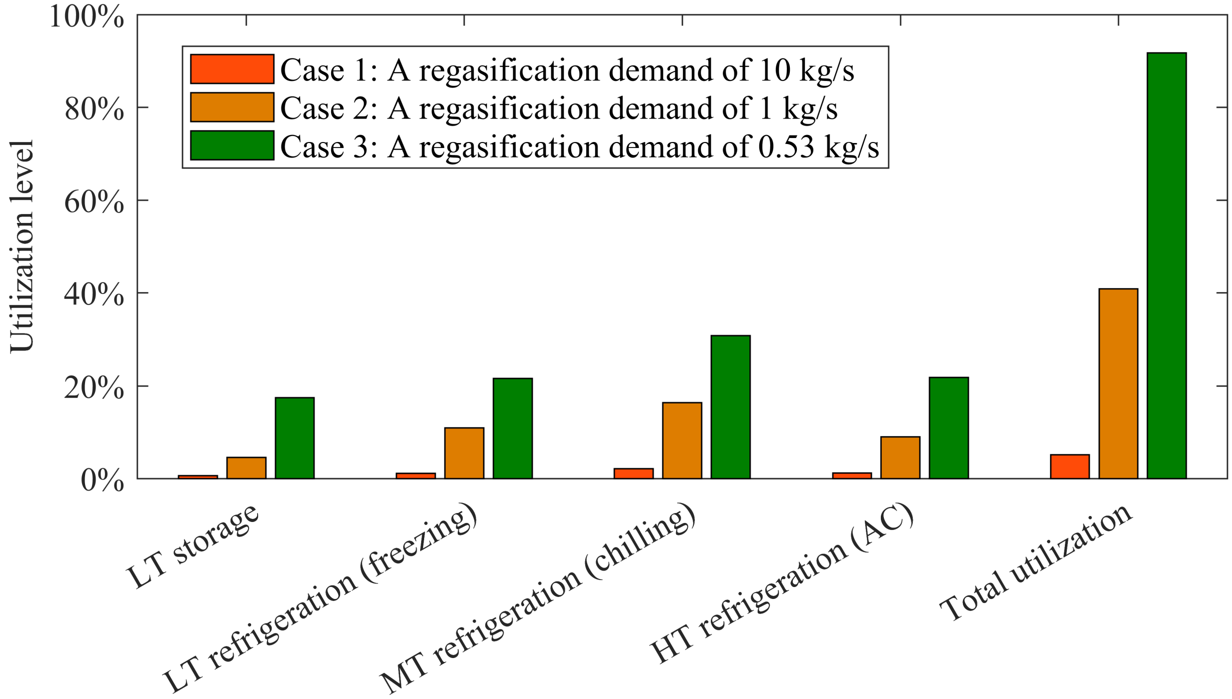


Figure 6.2: The cold utilization levels at different regasification demands.

The specified temperature fits for Case 1, 2 and 3 are drafted in temperature - enthalpy diagrams in respectively Figure 6.3, 6.4 and 6.5, and are showing the temperature fit between the heating curve of LNG and the applied cold recovery system. By comparing these three cases, in the given order, a large reduction in the need for backup heating can be observed. The temperature fit is also greatly improved as more cold energy is recovered, which indicates that the multi-temperature refrigeration system is applicable for cold energy recovery from the vaporization of LNG. This coincides well with findings in literature, which specifies that the best solution for a complete cold energy recovery entails end users demanding cold at multiple temperature levels. Another strategy already introduced at the Senboku terminal in Japan, as mentioned in Section 3.2, is the implementation of multiple cold energy recovery systems at different temperature levels. This will likely introduce a higher total cooling demand, as well as the ability of an even closer temperature fit, than the ones presented in Figure 6.3, 6.4 and 6.5, recovering more of the huge amount of available exergy. An auxiliary power generation system is thus analyzed in Section 6.1.3.

Another aspect to consider is that these results, showing the total amount of cold utilization, includes the charging of all three storage tanks. The charging demands introduce a considerable increase in the possible utilization, compared to only the main refrigeration applications in the supermarket. When the charging time of all three storage tanks have passed, the total utilization will drop to the sum of the freezing, chilling, and AC capacity in the supermarket, 182.5 kW. This will introduce an increase of the necessary backup capacity of about 57% for the most promising case, a refrigeration demand of 0.53 kg/s. An inclusion of another cold recovery system to be present at times with no charging demand would thus be favorable, which could have avoided the extra commissioning of the seawater heat exchanger.

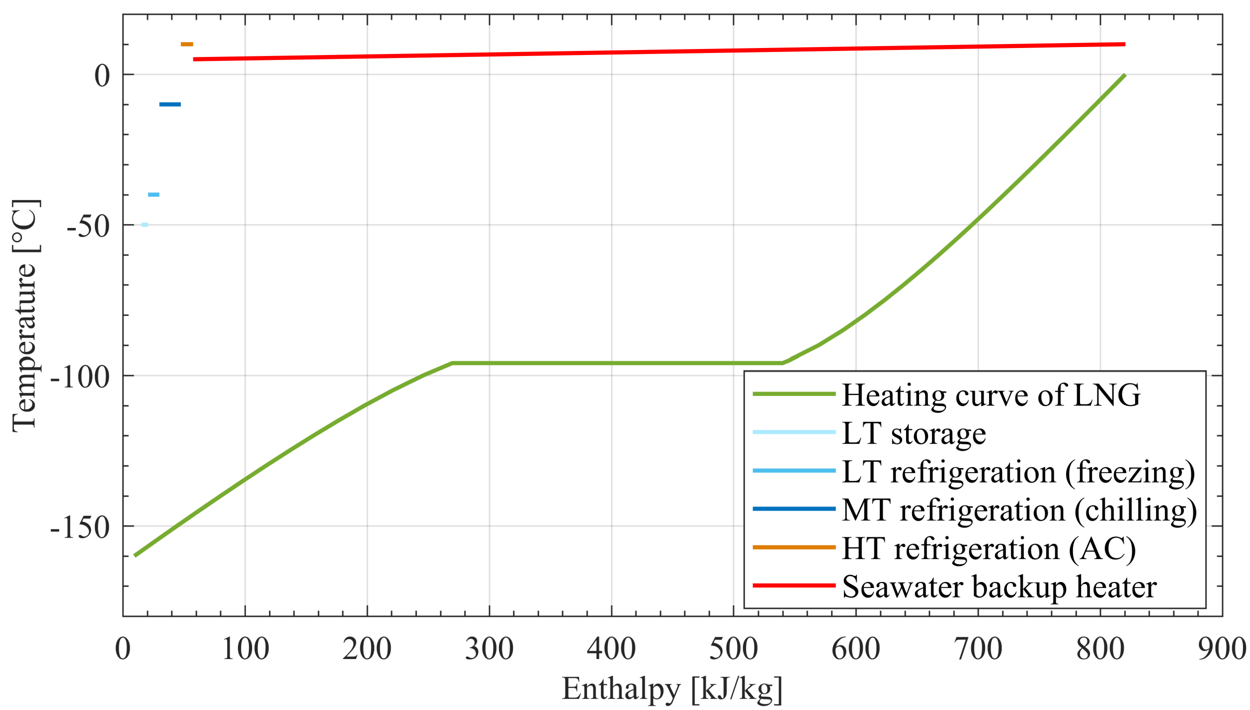


Figure 6.3: Temperature fit for Case 1, a regasification demand of 10 kg/s.

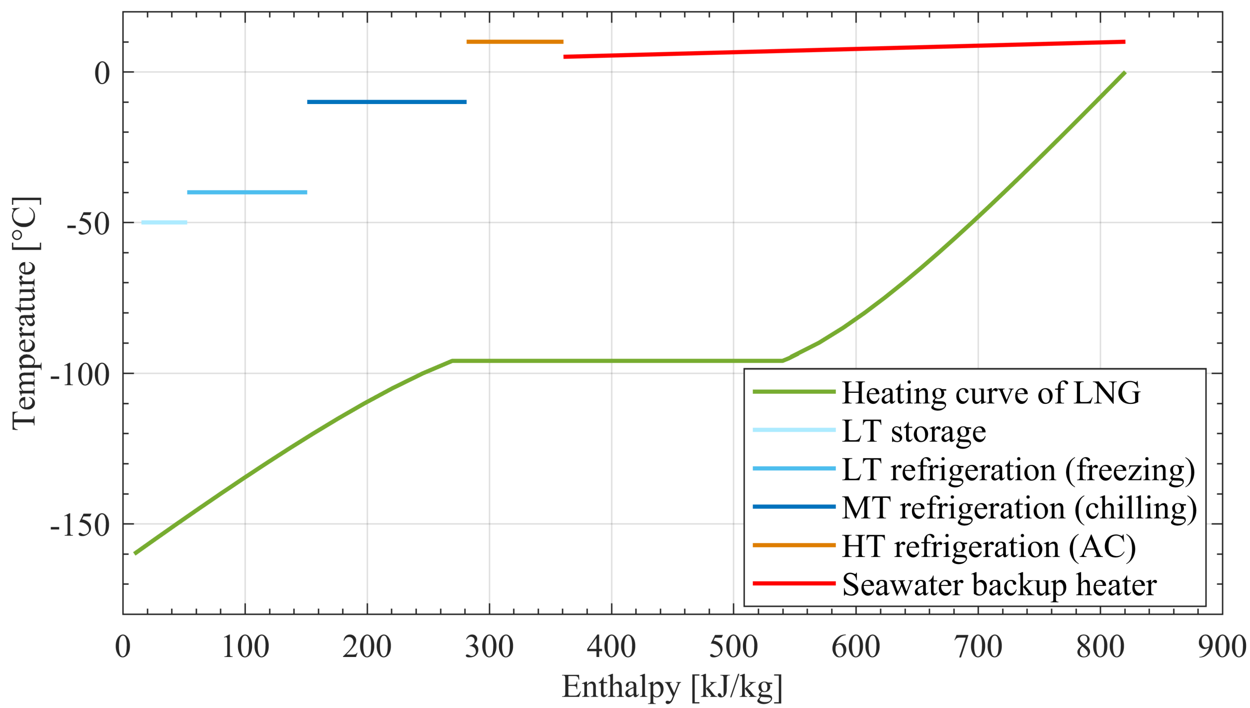


Figure 6.4: Temperature fit for Case 2, a regasification demand of 1 kg/s.

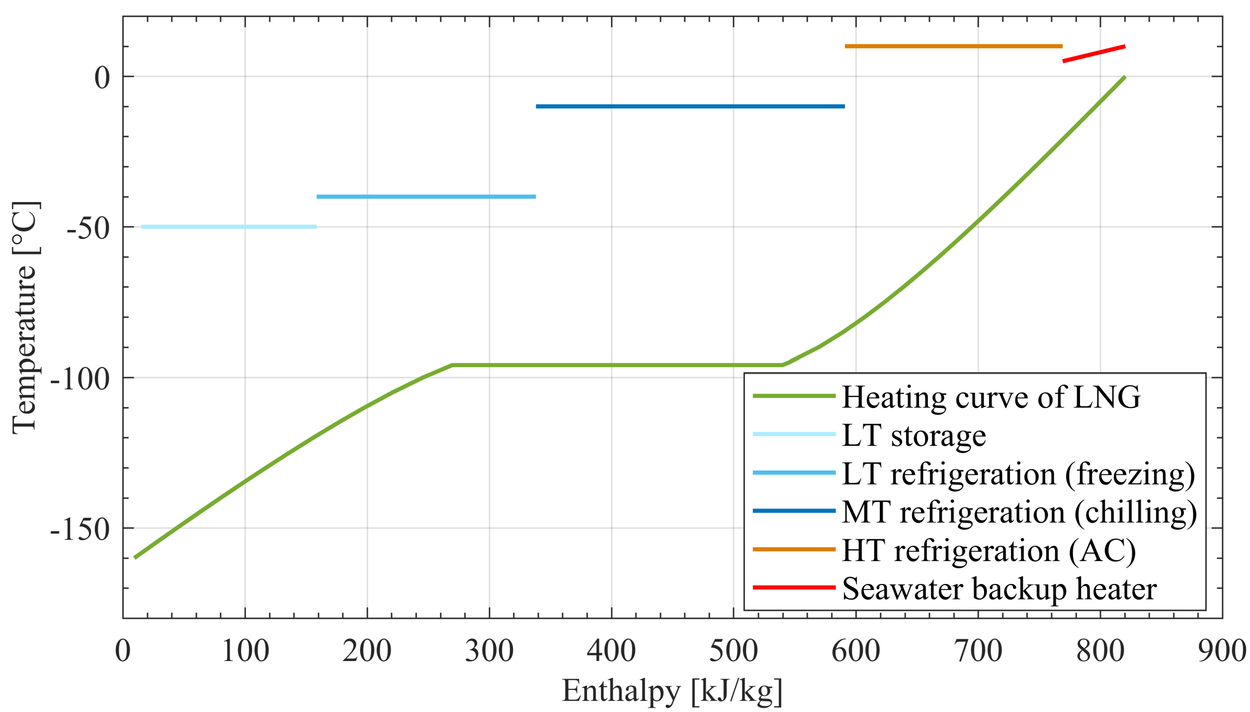


Figure 6.5: Temperature fit for Case 3, a regasification demand of 0.53 kg/s.

Performance of the connecting heat exchangers

In order to investigate the performance of the cold energy recovery system, the properties of the connecting heat exchangers represent an important part. The temperature approach in the heat exchangers depicts the behavior according to temperature driving forces, and a close temperature approach is thus essential for an efficient heat exchange. Additionally, it decreases the need of flow in order to achieve the same transfer of heat, hence, a reduction of the required pumping power for the liquid pumps in the related refrigeration loop can be expected. However, a closer temperature approach will most likely require a larger heat transfer area, introducing a higher investment cost. For the given objective, a minimization of the operational costs was emphasized. A further discussion, considering the heat exchanger design, is followed up in Section 7.1.

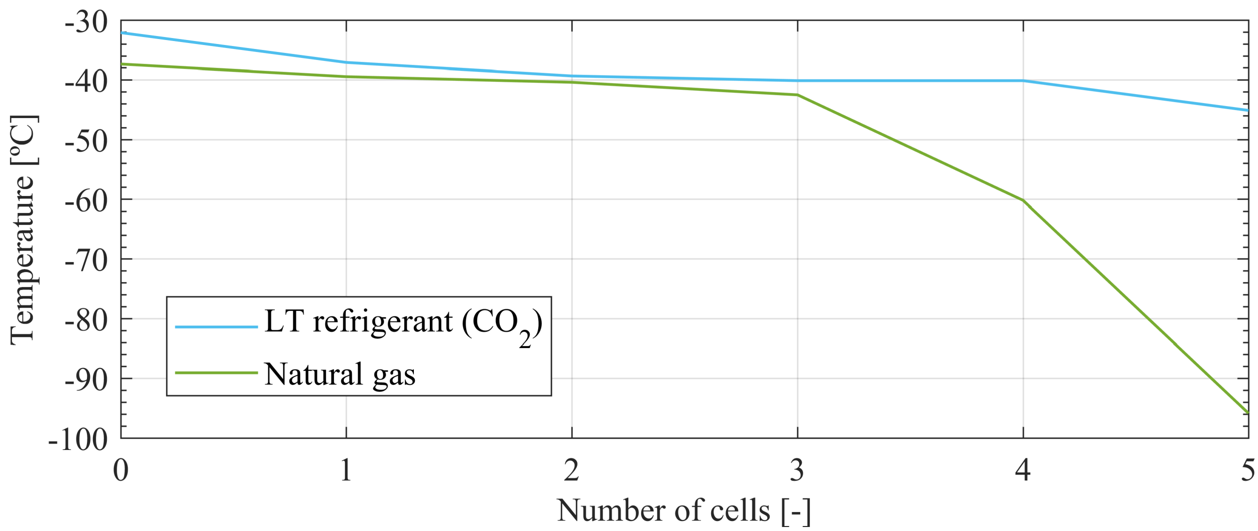


Figure 6.6: Temperature approach for the LT condenser at a regasification demand of 1 kg/s.

To serve the purpose of this analysis, the temperature approach for the condensers at each refrigeration loop in Case 2 are extracted and shown in Figure 6.6, 6.7 and 6.8. The heat exchangers are divided into five cells in order to inspect the temperature approach in steps through the heat exchange process. As the amount of LNG through each heat exchanger is controlled by the subcooling on the refrigeration side, the temperature approach in the heat exchangers is closer than observed in the related temperature fit diagram in Figure 6.4. In all three heat exchangers, the LNG outlet temperature is quite near the inlet temperature of the refrigerant, introducing a close temperature approach. As a consequence of the minimum temperature approach, the temperature difference between the two flows of natural gas approaching the downstream bypass valve is large, and hence initiates a decrease in the exergetic efficiency. In addition, the temperature approach diagrams show that the natural gas experiences vaporization, or are already vaporized, in all three condensers, which differs from the process shown in Figure 6.4. It can be observed that the inlet temperature of the natural gas stream is at saturation in every condenser. Presumably, this is caused by the control strategy using the bypass structure, and the validity of this concept is further discussed in Section 7.1.1.

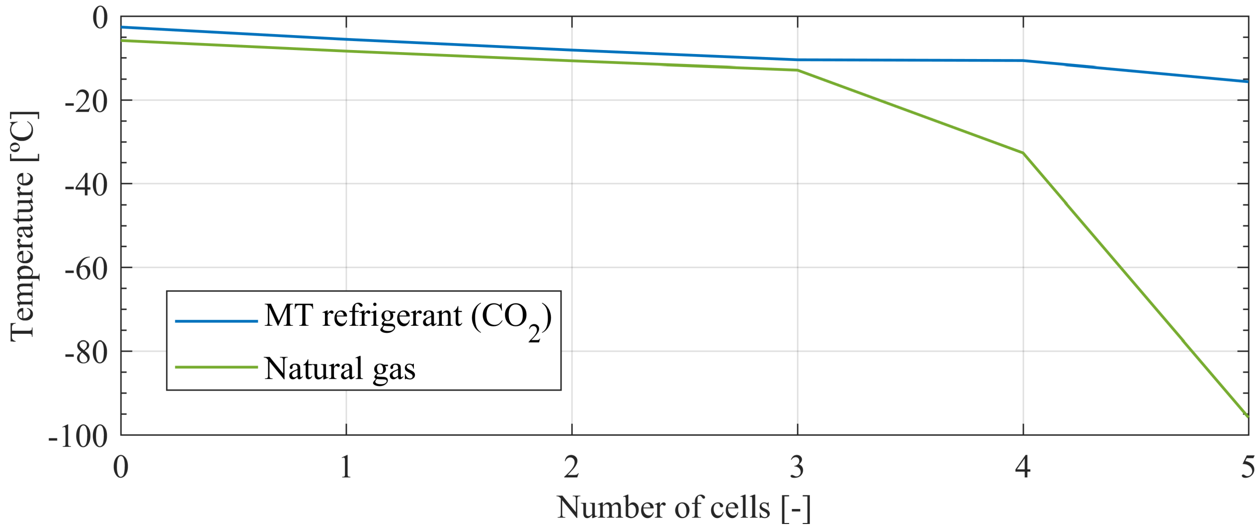


Figure 6.7: Temperature approach for the MT condenser at a regasification demand of 1 kg/s.

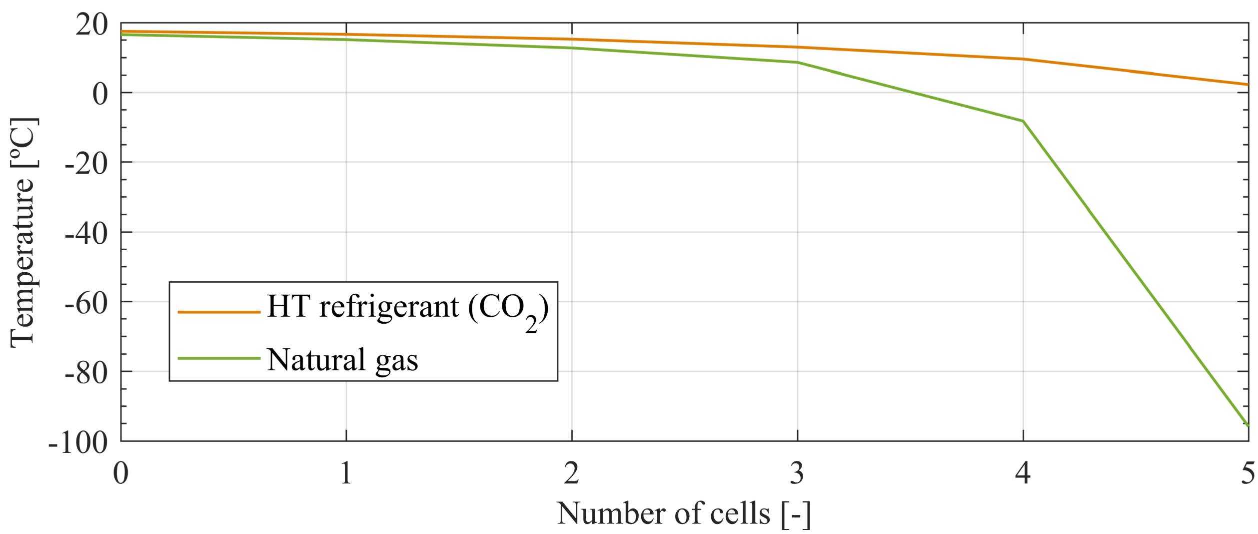


Figure 6.8: Temperature approach for the HT condenser at a regasification demand of 1 kg/s.

6.1.2 Cold thermal energy storage

The performance of the system design must also be evaluated in regards of the CTES systems, especially the LT storage unit as it handles a part of the regasification process while charging. These storage tanks serve as a backup capacity for the refrigeration loops when the regasification demand is zero or too small to cover the whole refrigeration demand, and the working principle of the LT storage tank is visualized in Figure 6.9. Results from Case 1 are extracted in order to compare the actual performance with the design choices stated in Section 5.2. First, the storage tank is in charging mode, and the liquid mass fraction of the PCM stored in the tank is decreasing. The actual charging time of about 12 hours matches the design, although a maximum solidification level of 10% liquid mass fraction was set due to the presence of expansion. Further, it can be observed that the regasification demand covers the LT refrigeration demand for the next 24 hours, before the discharging mode is switched on at 36 hours after startup. By Figure 6.10 the discharging mode is active when the PCM is absorbing heat, meaning a positive heat transfer. For this case the discharging load is equal to the total LT refrigeration capacity, meaning that the regasification demand is completely absent and that the MT storage is fully charged. The last mentioned condition is not implicit, as it does not exist a control system that shuts off the charging of the storage tanks at situations with no coverable regasification capacity. Therefore, the discharging of the LT storage may also have to cover the charging of the MT storage tank if it has not reached fully charged condition.

It can be observed by Figure 6.10 that the heat transfer in charging mode is peaking before it more or less stabilize at 55 kW. The peak can be related to the necessary subcooling of the PCM before the solidification can initiate, as an initial value of a few degrees above the temperature for freezing is set due to simulation performance. However, as the control system has a idle response time before it reaches the setpoints, the described peak may be out of range and should hence be reviewed with a grain of salt.

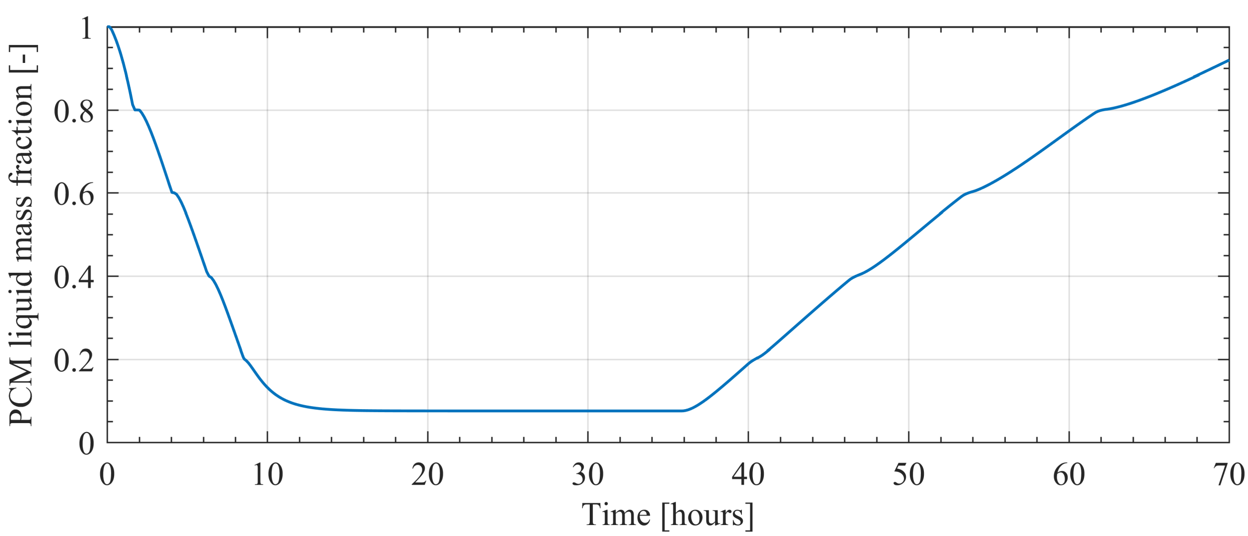


Figure 6.9: Charging and discharging pattern of the LT storage unit at a regasification demand of 10 kg/s.

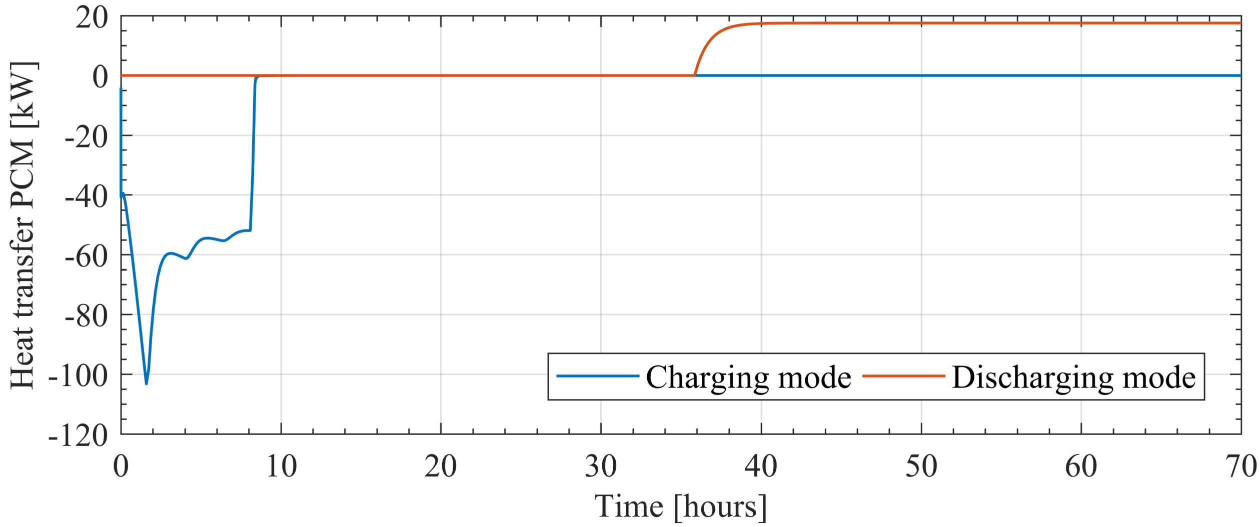


Figure 6.10: Charging and discharging of the LT storage unit illustrated by the heat transfer of the PCM at a regasification demand of 10 kg/s.

6.1.3 Organic Rankine Cycle

The auxiliary power generation system, conducted by an ORC, is designed to fully utilize the cooling potential together with the refrigeration system at a regasification demand of 1 kg/s. When the LT storage is in charging mode this implies a complete regasification of natural gas without any use of an external utility, compared to the previously analyzed cases. Figure 6.11 displays the distribution of the cold recovery systems for vaporization of natural gas for Case 4. The graph shows that a complete regasification of natural gas was carried out for the given design. This configuration also introduces a closer temperature fit at each temperature level present, which increases the performance of the total cold recovery system and confirms the preferable choice of multiple cold recovery concepts. Additionally, the possible power production related to this configuration of an ORC was detected to approximately 154.4 kW, assuming an idealized generator efficiency of 1.

Energy savings

The overall goal of the ORC is to make the regasification terminal self-sufficient with energy. A power accounting, considering the power consuming equipment in the cold recovery system, was thus conducted for Case 4. The result is shown in Table 6.1. After supplying all of the power consuming equipment in the cold recovery system illustrated in Figure 4.7, an amount of 69.84 kW remained as an exportation potential. This excess power may also be used in other power demanding processes located at the terminal or in the supermarket refrigeration system, as, for instance, the fans operating in the cold storage rooms and displays or the actuators needed for the control of the valves.

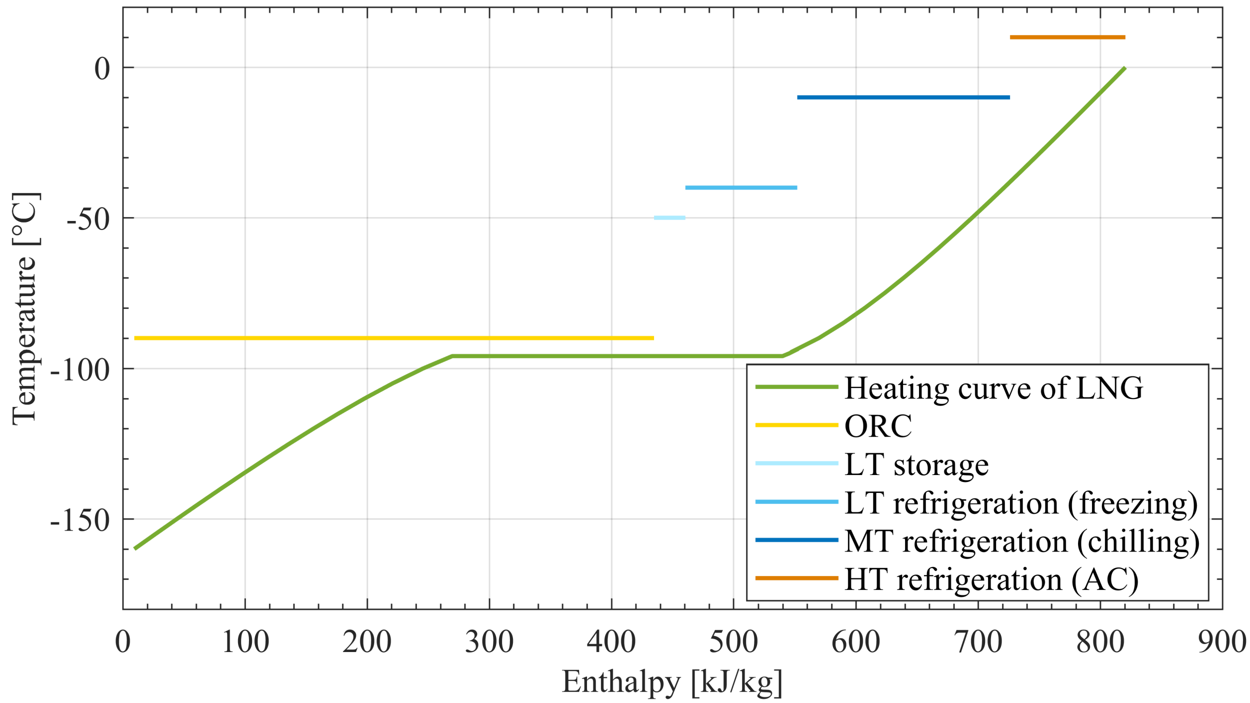


Figure 6.11: Temperature fit for Case 4, a regasification demand of 1 kg/s.

Table 6.1: Power accounting for Case 4.

Equipment	**Power consumption/production [kW]**
LNG pump	-16.96
LT liquid pump	-0.25
MT liquid pump	-0.65
HT liquid pump	-0.54
ORC pump	-65.17
ORC turbine	154.41
Summarized	69.84

It should be mentioned that the efficiencies used for the liquid pumps in the simulation model are not in the most realistic range. The pump efficiency is modest, and the inefficiencies caused by the conversion of electrical energy into mechanical energy should be accounted for. A standard motor efficiency between 0.65 and 0.75 indicates the actual pump power consumption of the pumps in relation to electricity supply, as this is not assessed in these results.

6.2 Cost estimation and emission analysis

A cost estimate analysis comparing two of the simulated cases, Case 2 and 4, to the conventional solution was carried out. For comparison purposes, the regasification rate considered in the two cases, 1 kg/s, was as well applied for the conventional system, even though it's just about 1% of a typical regasification demand. Therefore, the calculated estimates only depict the trend of the potential cost reduction by utilizing the LNG cold.

The annual thermal energy demand for the regasification process is presented in Table 6.2, together with the thermal demands for a conventional supermarket refrigeration system with the same refrigeration capacities as applied in the simulation model. As the refrigeration capacities have been considered constant throughout the simulated cases, the same is applied here for simplicity. The power consumption by the LNG pump is neglected in the following analysis, as it can be regarded as a necessary consumption anyhow.

Table 6.2: Annual thermal energy budget for a constant regasification demand of 1 kg/s.

	Annual thermal energy demand [kWh]
Regasification	7,165,680
Supermarket freezers	153,300
Supermarket chillers	569,400
Supermarket AC	876,000

The conventional regasification process was chosen to operate with an ORV, using seawater as the source of heat. Considering that the terminal is located close to the sea, the operational costs related to the use of seawater is mainly caused by the pumping process supplying the ORV with the necessary flow of seawater. An evaluation assuming a necessary pressure rise of 4 bar at an operating sea water temperature from 12 °C to 7 °C, conducted by a pump with an isentropic efficiency of 0.9, was hence carried out for an annual energy demand estimate. For the conventional supermarket, with the same refrigeration capacities as the ones used in the simulation model, a COP of about 2.2 was considered.

Moreover, in Case 2 and 4, the results obtained from the Dymola simulations were used to estimate the operational costs in the supermarket, which is caused by the power consumed by the liquid pumps in the refrigeration cycles. The amount of necessary backup heating by the ORV was as well estimated by the simulation results, which in addition includes the charging capacities of the three storage tanks. For that reason, the cost estimates for Case 2 and 4 show the best performance state of the cold energy recovery systems described in Chapter 4.

Although the regasification terminal is most likely not located in Norway, an average price of electricity was sourced from Statistics Norway. For the first quarter of 2018, an average electricity price for commercial buildings was found to be about 0.874 NOK/kWh, inclusive taxes and grid rent [34].

Table 6.3: Annual operational costs and % improvement of the conventional case.

	Annual cost [NOK/year]	Cost reduction [%]
Conventional systems	732,627	0
Case 2	57,638	92.1
Case 4	-664,561	190.7

Table 6.3 presents the estimated annual operating costs for the three compared cases. The calculated cost for the conventional systems represents both the regasification process and the supermarket refrigeration, without any integrated recovery systems. A large reduction in cost can be observed for both cases that include cold energy recovery. For Case 4 even a possibility for a yearly income of 664,561 NOK is estimated. However, as the charging of the storage tanks highly contributes to the reduced need for backup capacity by the ORV, the cost estimates should be considered with a critical eye.

Related CO_2 emissions

An analysis of the CO_2 emissions associated with the regasification of natural gas, based on the necessary electricity consumption, was additionally carried out. In order to look into the remarkable savings obtainable with cold energy recovery, a regasification terminal located in Spain was assessed. Here the energy mix is quite complex, consisting of both renewables, coal, and gas, thus introducing a substantial amount of CO_2 emissions related to the power generation. The energy mix used in this analysis is given in Figure 6.12 and is sourced from the IEA [17]. For this energy mix, an emission rate of about 265 gCO_2/kWh can be assumed, which is calculated based on a procedure described by NVE [26].

By considering a realistic 30 MW regasification train, an annual thermal energy demand of 262.8 GWh was detected. For the conventional regasification process conducted by an ORV, the electricity consumption was, as before, associated with the seawater pumps. A similar estimation as the one carried out for the cost analysis was performed, which resulted in an annual electricity consumption of 4.1 GWh. Further, the best case scenario was applied, meaning that all of the LNG cold related to the regasification capacity of 30 MW was utilized, implying a non-existing annual electricity consumption for the actual vaporization process. What kind of cold energy recovery system used is not of particular interest for this analysis, as it most likely would be a mix of several technologies in order to obtain the most efficient solution. The main aim of this analysis is hence to investigate the potential energy savings of the regasification terminal, and how much CO_2 emissions that can be avoided related to the decrease in necessary power consumption. The described assessment resulted in Table 6.4, showing an expected potential reduction of about 1,085 ton CO_2. In addition, large savings associated with the cold recovery systems is anticipated, as a considerable electricity consumption most likely is present in their conventional system configurations as well.

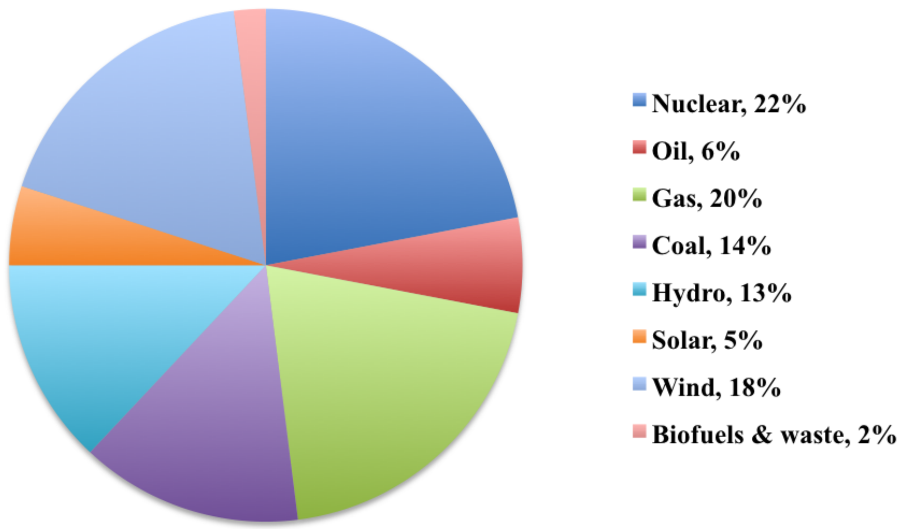


Figure 6.12: Energy mix in Spain 2016, sourced from the IEA [17].

Table 6.4: CO_2 emissions associated with the electricity consumption of a 30 MW regasification train.

Scenario	**Annual thermal energy demand [GWh]**	**Related annual electricity consumption [GWh]**	**Annual CO_2 emissions [ton]**
No cold recovery	262.8	4.1	1,085
Complete cold recovery	262.8	0	0

By making use of Figure 1.1, an extension of this evaluation into at worldwide perspective was carried out. The annual natural gas trade distributed by LNG in 2016 was 346.6 billion cubic meters [4], implying a regasification demand of about 7,210 kg/s. This initiates an annual thermal energy demand of 51,663 GWh. Applying the same simplifications regarding the electricity consumption as before, an annual needed supply of 804 GWh was detected. Further, the energy mix was assumed constant and equal to Spain for all worldwide terminals, and the ORV was considered as the standard vaporizer. Thereby, the annual CO_2 emissions related to the energy consumption of the seawater pumps in 2016 was 213,759 ton.

The evaluation of CO_2 emissions, that can be related to the conventional vaporization of natural gas, shows a striking potential for an environmental improvement of the distribution of natural gas by LNG. As known, the total energy demand in the world will likely increase in the following years, and a further raise in the fuel share of natural gas can hence be expected. However, the extreme amount of necessary energy supply and related emissions can be considered as a limiting factor. The investigated cold energy recovery systems, together with other assessed utilization technologies described in Chapter 3, is thus unavoidable implementations for a more sustainable future with the use of natural gas.

Chapter 7

Discussion

In this chapter an overall discussion, looking into the performance of the proposed system designs and developed simulation models, is performed. Most of the result oriented discussion have already been carried out in Chapter 6, however, a discussion focusing on the larger picture is conducted here.

7.1 Validity of the simulation model

As the simulation model is a tool to predict the real performance of the cold energy recovery system, the validity of the developed model, including input values, assumptions, and other simplifications, is important to assess. Therefore, in light of the analyzed simulation results, the substantial equipment and control system are appraised.

Input values and overall assumptions

The most tellingly overall assumptions made in the development of the simulation models, stated in Section 5.2, make the performance of the simulated systems more efficient than realistic. If both pressure loss and heat loss to the surroundings had been present, a considerable reduction in the overall energy efficiency of the system could be noticed. Variations in heat transfer properties and pump efficiencies would additionally have limited the performance of the system. More realistic efficiencies of the pump motors, as stated in Section 6.1.3, would have caused an increased total power consumption. However, a change in the motor efficiency would most likely not affect the CO_2 liquid pumps to a large extent, as their necessary pumping power is continuously low. Regarding constant heat transfer coefficients, occurrence of fouling on the heat transfer surfaces after a period of operation causes a considerable thermal resistance and thus reduces the heat transfer efficiency. An example of such fouling is the ice formation detected on the open rack vaporizer in Section 3.2, however, this cannot be considered as a permanently fouling condition.

Another assumption, which has been consistent throughout the thesis, simplifying natural as 100% methane, introduces uncertainties related to the amount of available cold energy and the heat transfer properties. However, as this thesis only serve the objective of proving the concept of multiple cold energy recovery methods, the simplification is considered appropriate.

The chosen refrigeration capacities have proven to be too small compared to the available cooling potential from the regasification of natural gas. It is not likely that 435 supermarkets, which potentially could use the designed refrigeration system, are located nearby the terminal, as found necessary for a complete regasification of natural gas at a typical demand of 10 kg/s in Chapter 6.1.1. Additionally, the load of the supermarket system will vary on a daily basis, where peaks are occurring related to loading of fresh goods and the presence of customers in the shop. Variable refrigeration load in the supermarket is hence a potential case scenario for a more realistic evaluation of the cold recovery potential by a supermarket. In the simulated scenarios in this thesis, the refrigeration loads are held constant and equal to the maximum installed capacities, looking into the cold recovery system's best point of performance. According to Selvnes [32], a typical base load of a similar supermarket is only about 20% of the maximum installed capacity, which would have remarkably limited the potential cold recovery. Therefore, a further research on cold recovery by larger hypermarkets or agro-food industries, which probably have a considerable larger base load of freezing and AC requirements, should be carried out. Larger system configurations with greater refrigeration capacities may have served the objective of this thesis to a larger extent. However, the developed system configurations in Chapter 4 are designed to be applicable for several possible cases, making a change in the type of system or related refrigeration capacities a simple task for further investigation.

Heat exchanger performance

In the modeling part of this work, the main aim was to obtain a workable model. For that reason, optimization measures considering the heat exchangers were not emphasized to an optimum extent. The heat transfer area of the plate heat exchangers, operating as the condensers, have therefore been set as a large estimate for simplicity. If considering the LT condenser unit, with the temperature approach shown in Figure 6.6, an logarithmic mean temperature difference (LMTD) of about 20.97 K can be expected. Using the assumption of an overall heat transfer coefficient of 2000 W/m^2·K, stated in Table 5.2, and a necessary heat transfer of about 90 kW in charging mode, a required heat transfer area of 1.50 m^2 is found by making use of Equation 2.5. However, the heat transfer area used in the simulations was approximately 25.57 m^2. This introduces an excess area of as much as 24 m^2, which causes a severe extent of pointless investment costs and reduction of performance. Withal, the temperature approach in the condensers is a result of the bypass structures of the units, which will be further discussed in Section 7.1.1.

Another aspect to assess, is as mentioned earlier, the occurrence of fouling on the heat exchanger plates. Practically speaking, a larger total heat transfer area should be emphasized in order to avoid the large costs related to the necessary cleaning of the heat exchangers due to fouling, introducing a trade-off between investment and operational costs. An optimal heat transfer area does hence consist of several dependent factors, making the optimization process a time consuming task. Therefore, the optimum performance of the heat exchangers serves as a suggestion for further work, which are summarized in Chapter 9.

Performance of the CTES systems

The PCM storage technology applied in the proposed system design have melting temperatures significantly lower than an average outdoor temperature. As these storage units are designed to store cold thermal energy over a longer period when fully charged, as shown by the working principle in Figure 6.9, cold leakage to the surroundings is an important factor to assess. In the simulation model, all operations were considered adiabatic for simplicity, introducing a best performance state of the system. Nonetheless, this assumption deviates considerably from the actual operation of the storage tanks when bearing in mind the large temperature differences. A proper insulation of the storage tanks is hence highly important to avoid a countable heat transfer by conduction. For the TIL's SLE unit used to model the cold storage tanks, an insulation thickness is not possible to include. The conduction through the wall of the storage tank is only influenced by the chosen wall material and its geometry. As the optional wall material types only consist of different metals, a usual insulation material can't be included in the calculations. A proper choice of wall material should have a low thermal conductivity, and the lowest obtainable in the TIL's SLE heat exchanger is stainless steel and its thermal conductivity of 14.6 W/m·K. For that reason, a realistic consideration of the modeled cold storage tanks would have included a large cold leakage to the surroundings, creating a large loss of possible storage capacity with time after charging.

Realistically speaking, an insulation layer is also quite necessary for other equipment, like heat exchangers and pipelines, especially if the pipelines are transporting large amounts of fluids with sub-zero temperatures over longer distances. However, as the simulation model do not include the presence of pipelines or appropriate insulation materials, evaluations of these concepts can't be carried out with the developed calculation tool in Dymola.

The operational change between charging mode, storage mode and discharging mode of the storage tanks is not working properly in the simulated scenarios. A deficient tuning of the controllers affecting these changes of operation, induces both oscillations and a non responsive control, increasing the simulation time remarkably. By setting a linearly decrease in the regasification demand in the complete Dymola model, the stiff model effect initiated termination of the simulation, as the solver was not able to detect the time of the event. In order to show the working principle of the storage tanks, only the LT refrigeration loop, with simplified control strategies of the LT storage unit, was simulated. As this reduced the number of coinciding control strategies, a usable result was obtained, implying that the complete package of controllers are in desperate need of further improvements and testing. However, by not including the two other refrigeration loops, as well as their interconnections, the results presented in Figure 6.9 and 6.10 deviates from their actual performance. As both MT and HT storage tanks are much larger in size and capacity than the LT storage, the assumption initiated by the simplified model, considering the other storage tanks fully charged (storage mode), is not valid. The discharging of the LT storage should as well have charged the MT storage, as a control strategy to shut off the charging at times with no coverable regasification demand is not installed. Such a control would have limited the unnecessary relocations of cold energy between the storage tanks, and hence retained the designed backup capacities of the storages. A more precise discussion about the control system of the Dymola model is performed in Section 7.1.1.

7.1.1 Control system

The limited timeframe, together with a lack of experience in control technology, have highly restricted the tuning of the control system. Thus, oscillations and poor response of the regulators may have diminished the performance of the simulation model. Additionally, the designed control system of the Dymola model consists of a set of reliant controllers that are affecting each other, inducing a fluctuating response. A more experienced control engineer had presumably seen the larger context and applied a more suitable overall control strategy, increasing the reliability of the cold recovery system. Nonetheless, the lack of appropriate tuning has not influenced the evaluation of the cold energy recovery, the main objective of this work. Further, the selected control strategies are discussed in detail. But first, changes in the design of system operation caused by the absence of particular controls are surveyed.

A preference of the operational design, described in Section 4.2.2, was the avoidance of two-phase flow between the multiple heat exchangers vaporizing the LNG. The complete vaporization process was thus designed to be completely conducted in one of the refrigeration condensers, presumably the MT condenser due to the desire of a close temperature fit. However, for simplicity, this design choice was disregarded in the development of the simulation models, and no control system was set to regulate the outlet temperature of the natural gas after each heat exchanger. Practical issues related to the design of pipelines and equipment, as flow patterns and operational challenges as further phase change, cavitation, and other types of erosion can hence be expected as a consequence of the two-phase flow.

The seawater heat exchanger was not implemented in the complete Dymola model, simply because of controlling issues that caused trouble for the integration solver. Huge amounts of seawater is necessary to fully regasify the LNG without freezing, and a suitable control system regulating the massflow of seawater with both the outlet temperature of seawater and the natural gas as restrictions/setpoints was found to be difficult. A cascaded control strategy would have served as the proper solution, however, the investigation of this logic was not carried out. The necessary heating capacity by the seawater was hence calculated based on the remaining regasification to obtain an outlet temperature of 0 °C for the VNG.

Bypass control structures

As mentioned in Section 6.1.1, the temperature approach in the condensers, presented in Figure 6.6, 6.7, and 6.8, are closer than observed by the associated temperature fit diagram in Figure 6.4. The inlet temperature of natural gas in each of the condensers is at saturation, which differs considerably from the actual inlet temperature of both the LT and MT condensers. A division of the natural gas flow is conducted by a junction element in order to obtain the bypass structure, however, the division calculations in Dymola seems to struggle. Several factors can be recorded related to this, associated to both the model design and the working method of Dymola.

Although the temperature of natural gas increases as it pass through the condensers, the temperature difference the heat exchangers must handle is large for each condenser unit for the applied cold recovery system. The heat exchanger model in Dymola would like to obtain a close temperature match as possible. Therefore, it keeps trying to find an appropriate state which provides this. Dymola would also like to simplify its calculations, and as the vaporization process introduces both constant pressure and temperature, the saturation state serves as a proper choice. Whenever Dymola finds an appropriate state, which doesn't affect the overall purpose of the model, the solver is satisfied. In that regard, whatever division of the natural gas stream Dymola finds appropriate doesn't matter, as long as the conditions upstream and downstream of the bypass structure coincides with the heat exchanged in the condenser. To improve the physical validity of this flow division, the calculation procedure of the junction element should be investigated in detail.

The chosen control structure may also induce challenges for the solver, as most of the controllers affect each other. For this part of the model, considering the control system for the condenser units, the subcooling and pressure control works simultaneously. The obvious solution, in means of theory, would be to implement these two controllers in a cascaded control system. This would have set the controlling measures in order, where one setpoint had been reached before the other, and increased the stability and response of the system. By setting the condensing pressure prior to the necessary subcooling, fluctuations in the system caused by this coherent control could have been considerably decreased.

Other factors that may cause a part of this unrealistic behavior are the absence of pipelines in the simulation approach and the available range of properties collected in TIL media for methane. The deficient optimization of the condenser units may also lead to this matter. Based on this discussion together with the considerations on the performance of the condensers, a few tests with less heat transfer area were performed. These confirmed a remarkable improvement of the division of streams, making the inadequate optimization of the heat exchanger units the largest uncertainty for the current discussion.

Feeding valves control

The control system controlling the valves feeding refrigerant to the evaporative equipment, have the ability to close when there is no cooling capacity requirement. In that case, they have also the ability to close simultaneously, which means that there will be a stoppage of refrigerant flow, or even backflow, in the refrigeration system. The liquid receiver works as a storage vessel for such situations, however, a pressure controlled bypass line or valve, as the one installed in the system design in Figure 4.3, should be installed to manage the overflow (backflow). A bypass will thus ensure stable conditions at shutdown and avoid potential breakdown of the pumps. Nevertheless, the absence of bypass in the simulation model has not affected the results obtained in this thesis, it serves only as a practical consideration.

7.2 System design evaluation

Several choices and assumptions made in the development of the system configurations can be questioned with respect to an optimal performing design. The most significant choices influencing the overall operational efficiency is hence evaluated in this section.

Regasification pressure level

The chosen regasification pressure decides several parameters, such as the available cold energy that can be utilized and the practical operation of system components. A suitable choice is therefore important, and one can question if the initial choice of 30 bar is sufficient. Study of Figure 4.2 shows that a higher pressure level, such as the two other shown, will improve the heat exchange process by avoiding the two-phase area. This can make the operation of the heat exchangers easier, and increase the options in the division of available cold energy. On the other hand, a small reduction in the available cold energy can be noticed, and if considering the refrigeration demand as a constant, it means that the mass flow of LNG has to be continuously larger. With only a small change in the regasification demand, the refrigeration will suffer, and the PCM units will likely be more active at a higher pressure level. In addition, a higher vaporizing pressure would have caused a higher energy consumption related to the LNG pump, and thus a higher operational cost and emission density.

7.2.1 Refrigeration system

The developed refrigeration system is the basis for the cold energy recovery investigated in this report, and the performance of this design is thus an important assessment factor for further optimization of the utilization of LNG cold.

Backup systems

A functioning performance of the PCM storage tanks was confirmed in Figure 6.9 and 6.10. However, a situation with both no regasification demand and no available cold storage capacity will cause severe problems for the refrigeration loop. With no active condensation system the refrigeration capacities cannot be fulfilled. An accumulation of gas in the liquid receiver can hence be expected, which may induce an excessive hydraulic pressure in the receiver. A conventional refrigeration system should hence be implemented as an external backup system, operating at times where the cold energy recovery systems cannot supply the necessary cooling capacity. The danger of accumulation of gas in the liquid receiver, which may also occur in other situations than the one considered here, lead to another necessary feature; A pressure relief valve should be installed to avoid the described pressure buildup.

Operational improvements

Operational improvements of the configured system design include type of evaporators for low temperatures, number of pumps, and further interconnections to increase compactness.

A flooded evaporator is a possible implementation for low-temperature evaporation to increase performance and to enable a closer temperature fit through the entire heat exchanger. In the system design, as well as in the simulation model, the evaporators are constructed as direct expansion components with 8 K superheat. Superheating ensures that the refrigerant is completely vaporized at the outlet of the evaporators, however, as the configured system do not include a compressor the amount of superheat is not critical. For that reason, the level of superheat could have been preferable decreased as the superheating of the refrigerant additionally uses a substantial share of the total heat exchanger area due to large differences in heat transfer coefficients. A flooded evaporator would hence be an appropriate choice, as it introduces a closer temperature approach throughout the heat exchanger and further increases the overall performance.

In order to increase the operational reliability of the refrigeration system, the number of pumps in parallel is a crucial factor. Service or failure in one of the pumps will not affect the total operation of the refrigeration cycle with a parallel installation of multiple pumps, which will as well avoid unnecessary stoppages of the refrigeration system.

Further interconnections could have increased the energy efficiency, as well as reduced the amount of related emissions, as the necessary external power supply probably would have decreased. The possible interconnections are highly related to auxiliary cold recovery systems, and the potential solutions for the auxiliary power generation design is proposed in Section 7.2.2.

7.2.2 Auxiliary power generation

The vaporizer of the ORC has about the same operating properties as the evaporators in the refrigeration systems. An implementation of an additional evaporator in one of the refrigeration loops to serve the function of the ORC vaporizer will hence result in both higher level of interconnections and removal of the need for an external heat source. Conforming to the achievement of a thermal efficiency as high as possible, the temperature of the heat source of the ORC vaporizer should be as high as possible. By means of the stated possible interconnection, implementation of an additional evaporator at the HT refrigeration loop is the most favorable. However, in comparison with the initially chosen external heat source, seawater, this will only result in a temperature rise of about 3°C. Thus, this interconnection will not result in any higher power output from the ORC.

Another question related to this interconnection is where the ORC should be placed. Most likely it will be favorable to locate it at the regasification terminal, to be able to use the generated power within the facility. Such an interconnection will hence be infeasible if the supermarket is located at a distance from the terminal. The largest amount of interconnections can thus occur when the end users are located in an industrial complex together with the regasification facility. If, however, the process scheme can be treated as an industrial complex, the ability to move the ORC circuit between different stages in the regasification line should be examined. The refrigeration demand of the different loops may not change corporately, and with a skew distribution, the available cold energy could be higher at a different stage than the initial placement of the

ORC. For instance, if the refrigeration demand of the LT and MT refrigeration loops are high, but the demand of the HT is low, the available cold energy at the last part of the regasification process will become excess. The backup heat exchanger at the end of the regasification line is thus necessary to complete the regasification process. This could have been avoided if the ORC condenser now was operating between the MT and HT condensers. However, this would have caused a lower maximum temperature difference over the ORC, and thus a significant reduction in the power output. But on the bright side, it would have reduced the need for external heating and the wastage of cold energy in total.

An extension of the discussion on the interconnection of ORC vaporizer and HT evaporator should include evaluation of other external heat sources. For the case of seawater, described in Section 4.3, Equation 4.1 demonstrates the maximum achievable thermal efficiency, found to be 60%. The temperature of the heat source cannot be considered as high, and another external heat source can result in a significant improvement of the thermal efficiency. Exhaust from a gas turbine or other combustion processes may be a proper source. However, this can cause a reduction in the total efficiency of the LNG supply chain, if the fuel used is parts of the vaporized natural gas, as well as increased environmental impact. If it can be characterized as waste heat from an unavoidable industrial process, the use can nonetheless be considered as beneficial. For comparison, the maximum thermal efficiency for an ORC using waste heat of about 400 °C is 83%, an improvement of 23%.

Chapter 8

Conclusion

The energy-intensive process occurring in both liquefaction and regasification of natural gas, introduce a high amount of cold energy serving as a potential energy carrier for further usage. Together with the increasing total energy demand in the world, research on the potential for use of this cold energy in suitable recovery systems is thus a necessity for a sustainable future. For that reason, cold energy recovery systems utilizing the huge amount of excess cold from the regasification of natural gas were developed. An applicable refrigeration system solution, including cold thermal energy storage systems, was designed and investigated in terms of a single normal-sized supermarket. In addition, an auxiliary power generation system was included for a possible increment of cold utilizations.

The literature survey revealed a huge potential for utilizing the LNG cold, however, the implementation level at existing regasification terminals are rather low. It was found that Japan serves as the forefront of such cold recovery technologies today, and has already installed multiple types of cold utilization concepts at the Senboku terminal, recovering 100% of the available cooling potential. However, the total cold utilizations in the world, considering an annual natural gas trade by LNG of 346.6 billion cubic meters, is still at a disproportionately low level. By assessing no exploitation of the excess cold at all, a potential reduction of 213,759 ton CO_2 emissions was detected.

In order to investigate the performance of the designed cold recovery systems, a model in the dynamic simulation software Dymola was developed. Various scenarios, based on varying regasification demands, were hence simulated; Three scenarios considering the refrigeration system of a supermarket and one scenario assessing the inclusion of an auxiliary power generation system. The simulation results confirmed the huge potential for converting the ultra-low thermal energy available in the natural gas into refrigeration applications and auxiliary power generation; A number of 435 supermarkets, with the designed refrigeration capacities, was found necessary to cover a regasification demand of 10 kg/s. Multiple cold recovery systems were additionally found substantial for an efficient exploitation of the LNG cold, observed by a closer temperature fit between the heating curve of LNG and the cold utilizations. The included ORC was in addition associated with a potential power generation of 154.4 kW at a regasification demand of 1 kg/s. As a result, a remarkable reduction in operational cost related to the vaporization of natural gas was observed for cold utilizations both by refrigeration applications and auxiliary power generation. The latter revealed as well an exportation potential introducing a possible yearly income of 664,561 NOK.

The work of this Master Thesis has thus produced applicable system designs for cold energy recovery together with a useful calculation tool in Dymola, which may simplify further research of improving the sustainability of production and distribution of natural gas.

Chapter 9

Suggestions for further work

In this chapter, the suggestions for further work are presented. Several possible improvements of both the developed system configuration and the simulation model have been observed, however, the most limiting factors are emphasized in these suggestions for further work.

For future investments, an analysis of multiple cold recovery systems exploiting the complete cooling potential at a typical regasification terminal should be carried out. The inclusion of an ORC, in addition to the refrigeration system, indicated an improvement of the performance of the total cold recovery system by introducing a closer temperature fit. Hence, more complex cold recovery systems should be investigated.

As emphasized in the methodology, observed by the results, and further discussed in the discussion, the design of the essential heat exchangers is of importance. However, the limited timeframe and applied simplifications, in order to obtain a workable simulation tool, restricted the amount of optimizing measures carried out for the multiple heat exchangers. Energy efficient recovery systems are highly affected by the heat exchanger performance, making this absence a limitation of the developed system design and simulation model. Further adaptations of the configurations should thus include optimization features of the entire rack of heat exchangers, by evaluating both the type of heat exchanger and the design.

To obtain a workable Dymola model within the given timeframe, several assumptions were applied. Hence, improvements to the developed model should focus on eliminating these assumptions for a more realistic approach. Heat loss to the surroundings and pressure drop in pipelines and equipment should be accounted for, and the constant heat transfer coefficients should be evaluated. As described, fouling and other factors can considerably reduce the heat transfer abilities with time, and should hence be accounted for. Additionally, several limitations of the applied control system were found, including both the strategies used and the amount of tuning. A more realistic perspective should thus be applied for further work, making the control strategies congruent with its practical limitations. In order to achieve a more responsive system, the tuning of the control system should as well be carried out to a larger extent, as it has highly restricted the possibility for optimization of system performance.

Moreover, the described district cooling should be investigated, both for air conditioning purposes and possible interconnections with other cold recovery strategies, as discussed in Section 7.2.2. The potential operational improvements, discussed in Section 7.2.1, may also be investigated by means of increasing the performance and compactness of the configured system.

A more accurate economic and environmental evaluation should be carried out, as the cost estimate and emission analysis carried out in this thesis are strictly limited. The profitability of the cold recovery systems should be investigated by means of both operational and investment cost. A suitable payback procedure should be applied for essential equipment, as the heat exchangers, in order to look into the advantages and challenges related to the exploitation of the excess cold.

Finally, as indicated in Section 4.4, the ability to transfer the developed technology into other similar problem scenarios, as for liquefied hydrogen, should be investigated for future investments.

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Appendix

Appendix A
Nomenclature

List of abbreviations

AAV	Ambient Air Vaporizer
AC	Air Conditioning
COP	Coefficient of Performance
CTES	Cold Thermal Energy Storage
HEX	Heat Exchanger
HT	High-Temperature
IFV	Intermediate Fluid Vaporizer
LMTD	Logarithmic Mean Pressure Difference
LNG	Liquefied Natural Gas
LT	Low-Temperature
MT	Medium-Temperature
NG	Natural Gas
ORC	Organic Rankine Cycle
ORV	Open Rack Vaporizer
PCM	Phase-Changing Material
SCV	Submerged Combustion Vaporizer
STV	Shell and Tube Vaporizer
TES	Thermal Energy Storage
VNG	Vaporized Natural Gas

List of symbols

A	Area	[m^2]
C_p	Heat capacity	[kJ/K]
E_x	Flow exergy	[kJ]
h	Specific enthalpy	[kJ/kg]
h_{fg}	Latent heat of vaporization	[kJ/kg]
$\dot{m}_R$	Refrigerant mass flow	[kg/s]
η_{is}	Isentropic efficiency	[-]
η_t	Thermal efficiency	[-]
η_{max}	Maximum thermal efficiency (*Carnot efficiency*)	[-]
q	Specific heat	[kJ/kg]
$\dot{Q}_0$	Refrigeration capacity	[kW]
$\dot{Q}_c$	Condensation capacity	[kW]
Q_C	Discharged heat - heat to the cold reservoir	[kW]
Q_H	Added heat - heat from the hot reservoir	[kW]
p	Pressure	[bar]
S	Entropy	[kJ/K]
T	Temperature	[°C]
ΔT	Temperature difference	[°C]
T_0, p_0	Surrounding properties	[°C, bar]
T_s, p_s	Storage properties	[°C, bar]
T_C	Temperature of the cold reservoir	[°C]
T_H	Temperature of the hot reservoir	[°C]
T_{vap}	Vaporization temperature	[°C]
U	Overall heat transfer coefficient	[$W/m^2 \cdot K$]
v	Specific volume	[m^3/kg]

$\dot{W}_{is}$	Isentropic (theoretical work)	[kW]
W_{net}	Net generated work	[kW]
W_{max}	Maximum achievable work	[kW]

Appendix B

Complete Dymola model

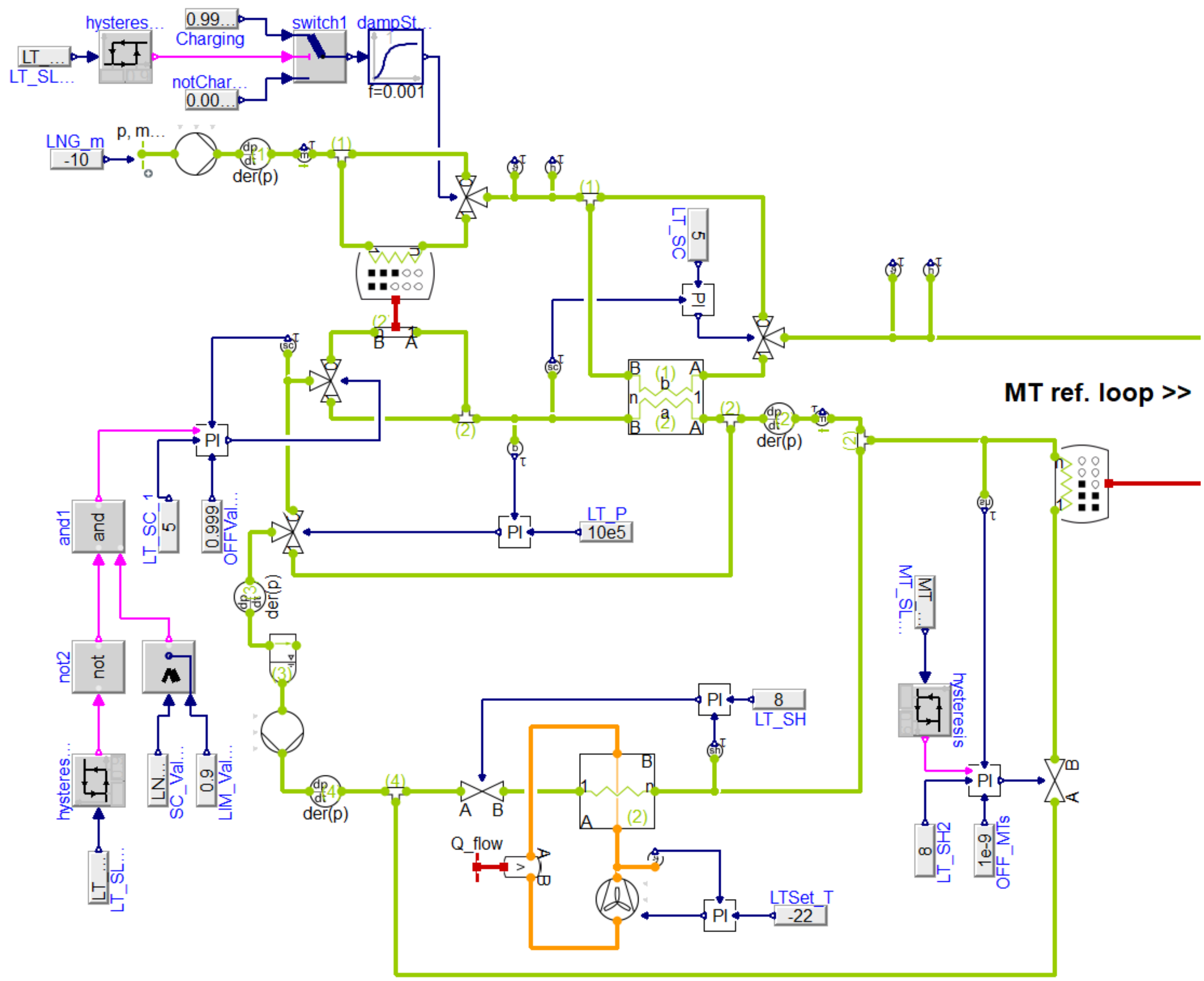

Figure B.1: Complete Dymola model of the LT refrigeration loop.

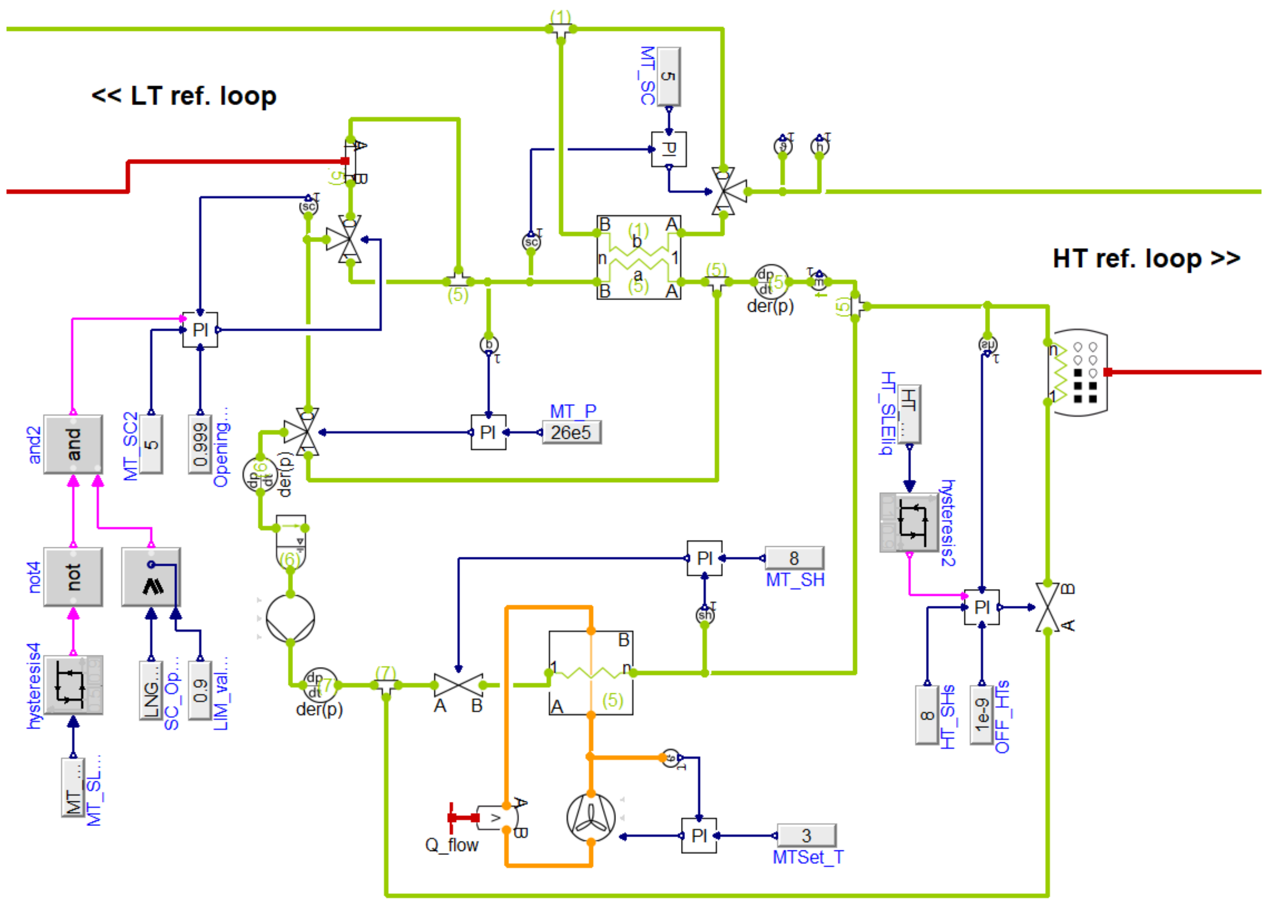

Figure B.2: Complete Dymola model of the MT refrigeration loop.

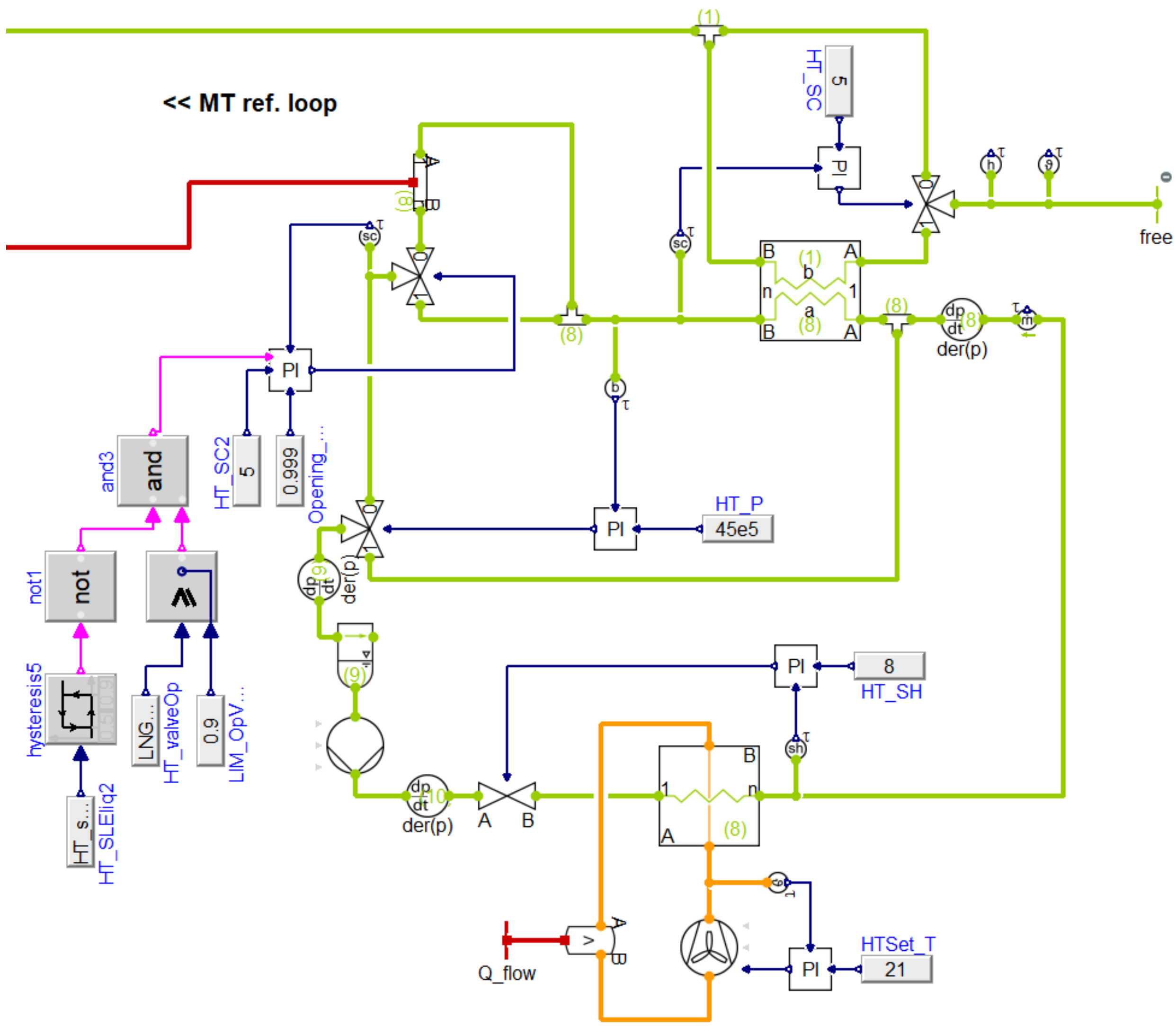

Figure B.3: Complete Dymola model of the HT refrigeration loop.

Appendix C
Abstract for article

EVALUATION OF DIFFERENT CONCEPTS FOR THE UTILIZATION OF THE COOLING POTENTIAL FROM THE REGASIFICATION OF NATURAL GAS

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ABSTRACT

The increasing interest in energy efficient solutions, and working towards a more sustainable future, give rise to the importance of utilization and thus reduction of the current wastage of excess energy. Natural gas is an important energy carrier for future investments, however, its production and distribution is not efficient. As 50% is provided by LNG at -162 °C, a large cold energy potential is introduced, currently mainly wasted to sea.

The importance of efficient cold recovery systems is emphasized, and an applicable refrigeration system, including CTES systems, is presented. Additionally, an auxiliary power generation system is included for further cold utilizations. A calculation tool has been developed and used to investigate potential energy savings.

The huge potential for exploiting the excess cold associated with the regasification of natural gas was confirmed, and the necessity for multiple cold recovery concepts for the most efficient utilization of the LNG cold was accentuated.

Keywords: Cold energy recovery, LNG regasification, CO_2 refrigeration, cold thermal energy storage (CTES), auxiliary power generation (ORC)