Master's thesis

NTNU
Norwegian University of Science and Technology
Faculty of Engineering
Department of Energy and Process Engineering

Tina Alise Martinsen

Investigation of advanced supermarket refrigeration units

Master's thesis in Mechanical engineering
Supervisor: Armin Hafner
Co-supervisor: Eirik Rødstøl and Erik Hoksrød
October 2021

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ABSTRACT

This thesis consists of analysis of two case supermarkets provided by Kelvin AS, which is one of the leading suppliers of refrigeration equipment towards the Norwegian supermarket chains. The systems are transcritical refrigeration systems with CO_2 as the refrigerant, but with two different system solutions. One of them have a low-pressure ejector system and the other one is equipped with a parallel compressor. The main focus of this thesis is directed towards case supermarket Spar Røyken which is the low-pressure ejector system that was implemented in autumn of 2020. This system was analyzed in two phases, as there were some challenges regarding ejector operation. In the first phase which lasted from autumn 2020 to the end of June 2021, there were done no changes from the initial setup. In phase one there were only a few dates with ejector operation, and a 40 minute period was analyzed showing an average pressure lift of 3.01 bar. As a result of the analysis in phase one there were made a few changes in an effort to improve the operation conditions for the ejector. The control settings of the medium temperature (MT) compressors was changed for faster response, and a driver update for the ejector was done. In phase two of the evaluation of the low-pressure ejector system, one could see the benefits of the changes done after phase one in terms of variable stability. There were two dates to choose from during the summer period, where one of the dates had three separate ejector-operating periods which was chosen for analysis. The average pressure lift done by the ejector during these periods were between 4.33 and 4.77 bar, which was an increase in pressure lift by at least 1.32 bar compared to phase one. Comparing the ejector operation period with a period without pressure lift from the ejector was also conducted with the same date as basis. With the common run capacity on the MT-compressors, the energy demand was compared as well as several other variables. This comparison shows that the period with ejector operation demands less energy with the same run capacity on the MT-level, even though there was an elevation in the low temperature compressors run capacity due to the need to match the pressure lift from the ejector.

Spar Snarøya was also analyzed in terms of energy performance, due to few measuring points in the refrigeration system, the analysis was conducted by use of Bitzer online software. The average coefficient of performance (COP) with operation of the parallel compressor on the hottest day in 2020 was found to be 2.68.

SAMMENDRAG

Denne masteroppgaven omhandler analyse av to kjølesystemer gitt av Kelvin AS, som er en av de ledende firmaene innen kjølesystemer til norske matbutikker. Systemene er transkritiske kjølesystemer med CO_2 som kuldemedie, men de har to ulike systemløsninger. Det ene har lavtrykks-ejektor og det andre har parallell kompressor.

Hovedfokuset i oppgaven har vært rettet mot kjølesystemet til Spar Røyken. Detter er et nylig rehabilitert kjølesystem med en lavtrykks-ejektor, som ble satt i gang høsten 2020. Systemet ble analysert i to faser, grunnet utfordringer med ejektor-drift. I første fase som varte fra høsten 2020 og ut juni 2021 ble det ikke gjort noen endringer etter igangkjøring. I fase en var det kun noen få datoer med ejektor-drift der èn periode på 40 minutter ble analysert. Denne viste et gjennomsnittlig trykkløft på 3.01 bar utført av ejektoren. Etter fase en ble det gjort noen endringer i et forsøk på å forbedre driftsvilkårene til ejektoren. Reguleringshastigheten på MT-kompressorene ble økt og det ble gjennomført en driveroppdatering på ejektoren.

I fase to av evalueringen kunne man se forbedringer sammenlignet med fase en på flere av variablene. Det var to datoer å velge fra i sommerperioden, der en av datoene hadde tre ulike tidsintervaller med ejektor-drift. Det gjennomsnittlige trykkløftet gjennomført av ejektor denne datoen var mellom 4.33 og 4.77 bar, noe som var en økning på minst 1.32 bar sammenlignet med fase en. Sammenligning av systemet på Spar Røyken med og uten ejektor-drift ble gjort på samme dato der lik «run capacity» på MT-kompressorene ble brukt som grunnlag for valg av periode. Energibehov ble sammenlignet i tillegg til andre variabler. Dette viste at perioden med ejektor-drift hadde lavere energibehov sammenlignet med perioden uten ejektor-drift og en økning i «run capacity» på LT-kompressorene ble observert grunnet økt utløpstrykk for å løfte til samme trykknivå som ejektoren.

Kuldesystemet hos Spar Snarøya ble også analysert, men på grunn av få målepunkt rundt parallell kompressoren ble Bitzer online software benyttet for å estimere COP. COP ble simulert med verdier fra den varmeste dagen i 2020 med påslått parallell kompressor. Gjennomsnittlig COP ble estimert til 2.68.

PREFACE

This master thesis summarize the work done in spring and autumn 2021. It's a continuation of the project report written in autumn 2020, both of these at Norwegian University of Science and Technology at the Department of Process Engineering.

The work revolves around refrigeration systems with CO_2 as the sole refrigerant, and energy efficiency in two case supermarkets. The project work consisted of gaining knowledge about refrigeration systems with CO_2 and the different possibilities for improvement of the energy efficiency, as well as working on understanding the case supermarket system solutions. Calculation, evaluation and improvements was conducted during the master thesis work.

I would like to thank my supervisor Professor Dr.Ing Armin Hafner for his guidance and his ability to share his knowledge and interest in CO_2 refrigeration system. I would also like to thank my co-supervisors at Kelvin AS, Eirik Rødstøl and Erik Hoksrød, for providing the case supermarkets and their involvement in problem-solving during the work. My family and boyfriend have provided great support during this work, and this has been much appreciated. Finally, a special thanks to Monica and Atlas for support, quality time and for creating the perfect work environment during this master thesis work.

Tina Alise Martinsen

Trondheim 13.10.2021

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4 ABBREVIATIONS AND NOMENCLATURE

4.1 Abbreviations

LT	Low temperature
MT	Mid temperature
R744	CO_2 refrigerant
FGBV	Flash gas bypass valve
VSD	Variable speed drive
IHX	Internal heat exchanger
PC	Parallel compressor
COP	Coefficient of performance

4.2 Nomenclature

ΔT	Temperature difference [K]
Δp	Pressure difference [Bar]
$\dot{m}$	Mass flow [kg/s]
C_p	Specific heat capacity [kJ/kgK]
η_{is}	Isentropic efficiency
W	Work
h_{2is} and h_{is}	Isentropic enthalpy [kJ/kg]
Δh	Enthalpy difference [kJ/kg]

5 INTRODUCTION

5.1 Background

In a world that's experiencing climate change and global warming, the chase to slow down and prevent depletion of the ozone-layer is on. Among preventative measure for this is phasing out harmful working fluids with high global warming potential. As this is done, natural working fluids are one of the replacement solutions. The refrigerant R744 is one of the natural refrigerants that have been making advances in the market, but the transition toward natural working fluids for all sectors is challenging since each energy concept has its pros and cons. There have been made many developments in the refrigeration systems utilizing CO_2 as the refrigerant in the last years, both regarding components and system configuration. And the suppliers and end-users are continuously interested in increasing the energy efficiency of the systems.

The most widely used CO_2-system in supermarket refrigeration is the transcritical booster system, and the efficiency have been improved by introducing new components and technologies. Two of the newest system solutions are introducing an ejector to perform a pressure lift or inserting a parallel compressor which can compress the flash gas without needing to throttle it down to suction gas level for the MT-compressors, both of these solutions are used to prevent throttling losses as can be big due to the nature of the refrigerant. The parallel compressor have been proven to enhance efficiency in several studies. The ejector solution have been proven efficient, especially in warmer climates where there is a lot of transcritical operation.

However, the ejector technology have not been researched a lot in Scandinavian climates where the amount of transcritical operation are significantly lower. The information regarding how well it works is limited, and it is an interesting field to look further into. The potential to reduce throttling losses is important when a more energy efficient system is desirable.

5.2 Task and scope

The main title on the master thesis is: **Investigation of advanced supermarket refrigeration units for Scandinavian climates.**

As given in the main title of the thesis, the purpose of this project to examine two different transcritical system given by Kelvin AS and evaluate their system performance. This should be done by identifying different days to evaluate the system performance for each of the refrigeration units. In addition to analyzing the two systems given by Kelvin, the master thesis should give an answer to the question: Is a low-pressure ejector system a good solution in Nordic climates?

The following tasks are to be considered in the master thesis:

1. Literature review of energy systems in supermarkets, including HVAC systems, hot water production and heat reclaim.
2. Describe the refrigeration system of the case supermarkets delivered by Kelvin AS.
3. Develop basic skills in modelling environments and develop simplified model(s) representing the supermarket refrigeration units (Modelica or Excel), enabling to estimate the energy demand for the systems (also in part load operation).
4. Calculate system performance based on measured date from the shops at different operational scenarios and analyze and discuss the results based on the measured data.
5. Prepare a scientific master thesis report including discussions.
6. Make proposal for further work.
7. Prepare a draft version of a scientific paper based on the main results of the work. The work will be edited as a scientific report, including a table of contents, a summary in Norwegian, conclusion, an index of literature etc.

5.2.1 SCOPE OF THE WORK

The field of research for both the preliminary project and the master thesis is transcritical refrigeration systems with CO_2 as the sole refrigerant, with the system solutions in the case supermarkets as the main focus. The duration of the study will be from autumn 2020 to autumn 2021.

A literature review of energy systems in supermarkets including energy efficiency measures will be conducted, with the main focus on ejector systems and parallel compression. The case supermarkets will be studied and described for further analysis. Basic skills in modelling environment will be developed for analysis of the case supermarkets. Excel will be the main modelling tool for estimating energy demand and system performance.

By using IWMAC, the online surveillance system utilized by both case supermarkets, the appropriate dates for evaluating the system will be identified. The needed variables from each refrigeration system will be exported to Excel and processed with the appropriate relations. The system performance including part load operation will be calculated based on the values from IWMAC.

These results will then be presented in the master thesis in a structured manner, analyzed and discussed. The results will also give a basis for the proposal for further work. A draft version of a scientific paper based on the main results will also be prepared and delivered as an Appendix with the master thesis.

6 LITERATURE REVIEW

The literature review is based on expanding the knowledge regarding refrigeration systems with CO_2 as the working fluid. This review is divided into sections to give a systematic introduction to the knowledge basis used for this master thesis.

6.1 CO_2 as a working fluid

There are several different ways to provide cooling and freezing in a supermarket, depending on size of the supermarket and the food that needs to be stored. Carbon dioxide [CO_2, R744] is one of the refrigerants that are mostly used today. It is categorized as a natural refrigerant due to its existence in the environment. It has a global warming potential [GWP] equal to zero when it is used as a working fluid, it is non-toxic, non-flammable and odorless. R744 is stable, inert and compatible with regular construction materials. It has a low critical temperature, 31.1°C, and a high critical pressure, 73.8 bar, which gives a significantly higher operating pressure than conventional working fluids. Because of the relatively high gas density of CO_2 operating with high pressure, lower compressor volume is needed due to high volumetric refrigeration capacity [kJ/m^3]. The dimensions for piping and valves are relatively small because of low viscosity and low $\frac{\Delta T}{\Delta p}$. The high pressure level in systems with R744 gives a lower pressure ratio across the compressor, which contributes to higher isentropic and volumetric efficiency compared to conventional configurations. Furthermore, the properties for CO_2 contributes to high efficiencies for heat exchanging in the evaporator and gas cooler (Stene & Hjerkinn, 2010).

Because of the relatively low critical temperature of CO_2 compared to other refrigerants, there exist two different types of refrigeration systems. The subcritical system, which has operation pressures between 5.7 bar and 73.6 bar and the corresponding temperature of -55°C to 31.1°C. The second type of refrigeration system is running in ambient temperatures above 31.1°C. It will therefor operate transcritically as the heat rejection will take place above the critical point. In these cases, the refrigerant above the critical point is a supercritical fluid where there is no clear distinction between liquid and gas phase. At this state the R744 won't condense and there is no correlation between pressure and temperature, and heat rejection is done by a gas cooler.

The gas cooler rejects heat by cooling the gas and not by phase change as is the case with a condenser. Depending on the ambient temperature, any direct CO_2 system have the possibility to operate in subcritical and transcritical mode. The figures below show an example of the subcritcal and transcritical refrigeration process given in a pressure-enthalpy diagram (Shecco, 2020).

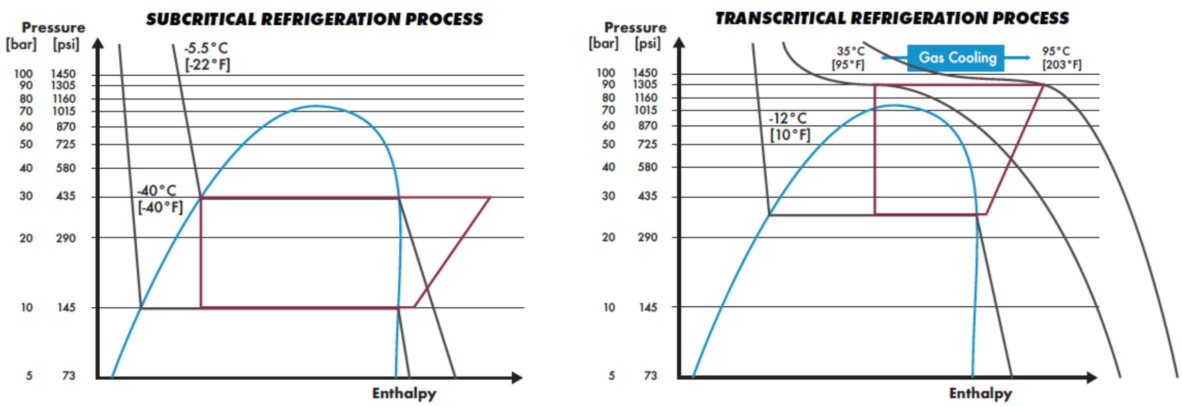

Figure 6-1: Pressure-enthalpy diagram of a subcritical and transcritical refrigeration process (Shecco, 2020)

6.2 Transcritical booster system

The newest configurations with CO_2 as a working fluid are the transcritical booster systems as shown in the previous section. In these systems R744 covers the entire cooling and freezing need for the supermarket. The booster system gets its name from the separate compressor that elevates the pressure in the low-temperature [LT] level to the pressure at the medium-temperature [MT] level. Downstream of the LT compressor rack, the refrigerant is mixed with the CO_2 from the MT-evaporators and goes through the MT compressor, heat recovery and the gas cooler. The refrigerant is then throttled by a high-pressure control valve to a receiver. This receiver separates the flash gas and the liquid, as well as stores excess refrigerant. The flash gas is produced spontaneously when the liquid is subjected to boiling, and this happens in any refrigeration system due to the pressure drop in the two-phase region. With CO_2 as the refrigerant, the flash gas occurs due to the pressure drop in the high-pressure valve. The higher vapor percentage in transcritical systems results in a larger amount of flash gas in these types of systems compared to subcritical ones (Shecco, 2020). The flash gas is transported to the MT compressor through a flash gas by-pass valve [FGBV], which plays an instrumental role in controlling the pressure level in the receiver. The liquid from the receiver is further throttled down to MT- and LT-level. Figure 6-2 shows an example of such a system, as well as a log p-h-diagram of the system.

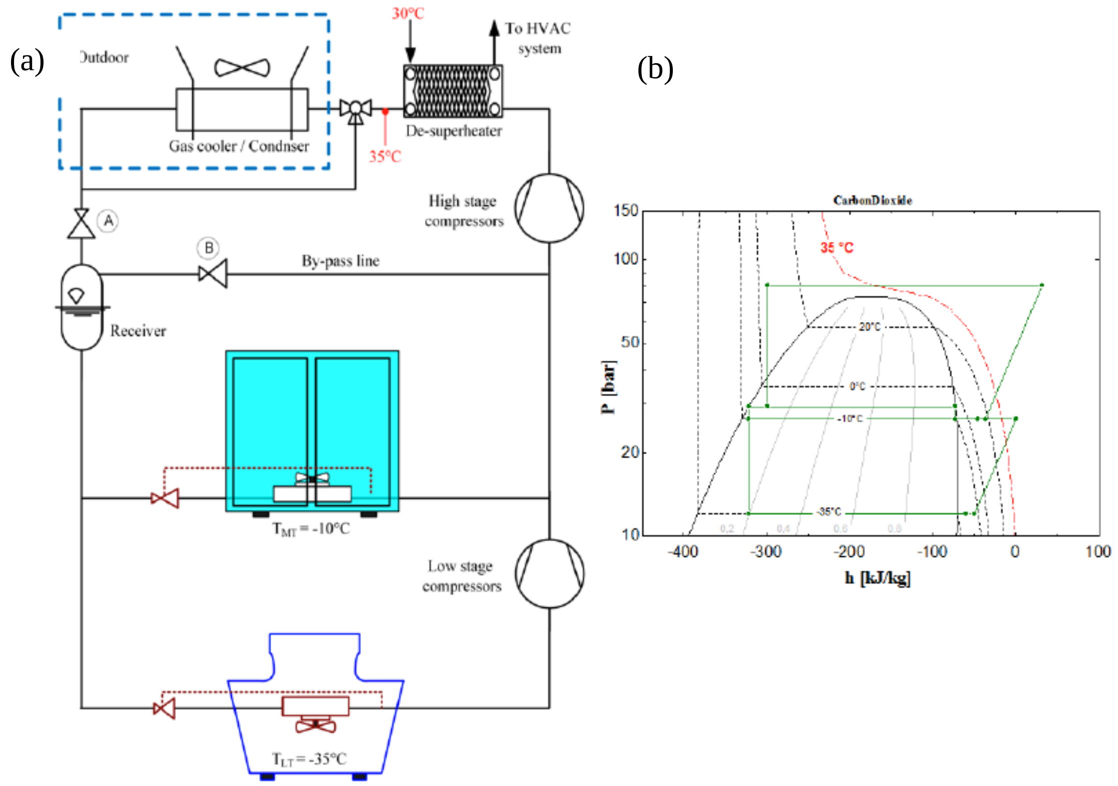

Figure 6-2: (a) Schematic of a transcritical R744 booster system. (b) Log p-h-diagram (Karampour, et al., 2016)

One of the factors that influence the system efficiency is the output temperature from the gas cooler. In warmer climates especially because of higher expansion losses and increased heat rejection losses, and in colder climates where too low outlet temperature may be an issue. If the temperature in the high-pressure control valve is too low, the consequence is a pressure drop in the receiver. This pressure drop may cause problems with delivering liquid refrigerant to the evaporators. Ways to avoid this can be bypassing the gas cooler with parts the refrigerant or by fewer heat recovery solutions (Kauko, et al., 2016).

6.2.1 MEASURES TO INCREASE ENERGY EFFICIENCY

The transcritical systems utilizing CO_2 as the refrigerant have seen several different upgrades in an effort to increase the energy efficiency. These configurations stem from an effort to make use of the heat that is produced, utilize the residual cooling, to minimize losses and reduce compressor work. Amongst these configurations, heat recovery, HVAC, parallel compression and ejector solutions have been introduced. Figure 6-3 shows a schematic where a few of these configurations have been integrated.

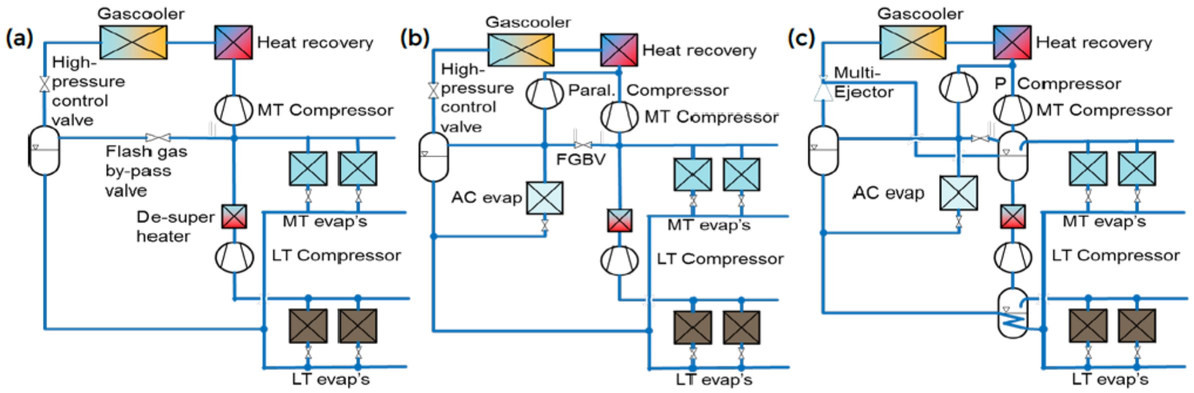

Figure 6-3: (a) Regular transcritical booster system. (b) Transcritical booster system with parallel compression. (c) Transcritical booster system with ejector (Kauko, et al., 2016)

6.2.1.1 Heat recovery and HVAC systems

A great way to increase the total energy efficiency of the refrigeration system is by recovering the heat that would be released into the atmosphere. There are several ways to utilize the heat that is reclaimed from a CO_2 refrigeration system. Heating of tap water is especially suited for heat recovery in these system due to the temperature glide at heat rejection which results in a good temperature adaption (Nekså, et al., 1998). Preheaters in the ventilation system is also a way to increase the total energy efficiency of the system. It has become increasingly popular to take advantage of the possibility of having an integrated system for energy saving, and there are several ways to do this. For CO_2 as the refrigerant, a good solution is by introducing one or more heat exchangers before and possibly after the gas cooler because of the high discharge pressure of the refrigerant. A transcritical booster system with R744 as the refrigerant is one of the most energy efficient systems regarding heat recovery. The high discharge pressure and the transcritical operation increases the available heat considerably (Karampour, et al., 2016).

Because of the high discharge temperature of these system and the high enthalpy when compared to traditional HFC refrigerants, a higher portion of the rejected heat can be recovered, or heat can be reclaimed at a higher efficiency. Higher ambient temperature gives the possibility of a larger heat reclaim without compromising the COP of the refrigeration. In the wintertime, if more heat is needed for floor heating for example, the pressure in the gas cooler can be increased so that the compressor work increases which will increase the amount of heat that can be reclaimed. Due to high COP-values this can be a preferred solution to other heat sources. Occasionally it is possible that 100% heat reclaim from the refrigeration system will be sufficient to fulfill the demand for heating and hot water in a supermarket. This way it is possible to reduce investment cost and running cost for other heat sources. But reclaiming the last percentages is not very efficient, so it has to be evaluated against the cost. The gas cooler needs to be bypassed to be able to reclaim the last portion of heat and with this, increased amount of compressor work is required (Danfoss, 2015).

The tap water heating are one of the large benefits with this refrigerant, due to the high discharge pressure. Because of the high de-superheating heat or the continuous temperature change of CO_2 during supercritical cooling, it's a good way to heat water coming from a very low temperature up to 70°C (Shecco, 2020). CO_2 does not undergo a phase change during heat exchange with the water, so the temperature of the refrigerant drops gradually as the water is heated. Due to this the temperature difference between the water and the CO_2 is almost unchanged as the heat is being transferred. This results in a smaller irreversible loss in the heat exchanger (Taira, u.d.). The temperature glide for the CO_2 can be seen in the figure below, which compares heat transfer between water and an HFC to water and CO_2. This is shown in a T-Q diagram.

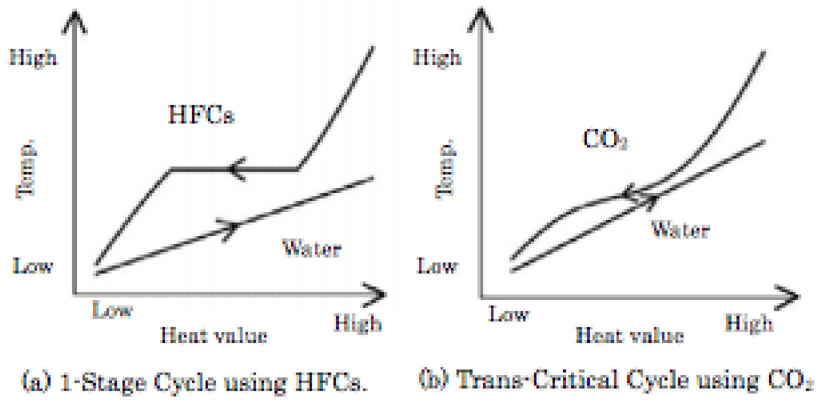

Figure 6-4: Temperature-heat value diagram with comparison of heat transfer between water and refrigerant (Taira, u.d.)

It is possible to integrate the heating and air conditioning (AC) system into the CO_2 refrigeration unit. For the heating system, a de-superheater can be installed before the gas cooler. This is a suitable system solution for refrigeration units with a refrigerant that has a relatively high discharge temperature, such as CO_2. There is a regulating valve after the gas cooler that adjusts the discharge pressure, and therefore the capacity of the de-superheater.

Compared to an isolated HFC system, research has shown that the integrated solution with AC has a higher COP for ambient temperatures lower than 25°C. This gives a compact solution as well as being environmentally friendly. However, such a system might require a higher degree of fine-tuning of the control system due to the conversion to a multi-function system. Reports have shown that this kind of system have the ability to deliver the entire or a great share of the heating and AC demand. Integrating the AC system in the transcritical booster system is done by adding a heat exchanger before the receiver, as shown in the figure below (Karampour, et al., 2016).

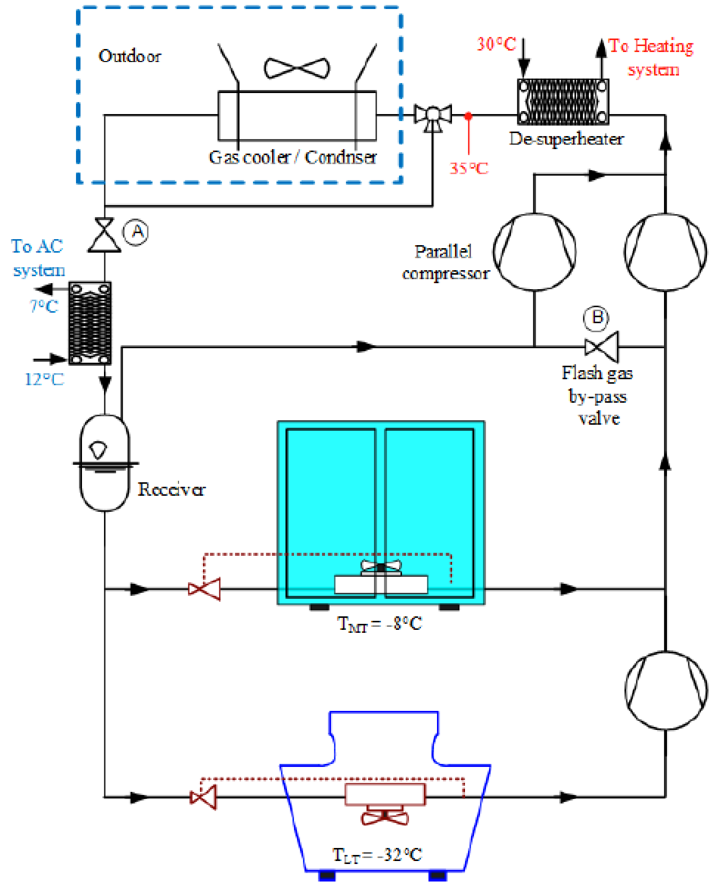

Figure 6-5: Schematic of a CO_2 refrigeration system with heat recovery, AC and parallel compression (Karampour, et al., 2016).

6.2.2 PARALLEL COMPRESSION

As mentioned in previous section, the amount of flash gas in the refrigeration system is larger in transcritical operation, and this increases further with higher external temperatures. The specific cooling capacity is reduced and leads to a larger amount of refrigerant that needs to be compressed by the MT compressor. To take care of this extra amount of flash gas an extra auxiliary compressor can be installed in the system. This compressor will suck parts of the entire volume of flash gas and compress it to the pressure level for the gas cooler. By utilizing a parallel compressor, the losses due to flashing is reduced. Instead of throttling the flash gas by a flash gas by-pass valve down to the pressure level required before the MT compressor, the auxiliary compressor can compress the flash gas directly which will reduce the amount of compressor work needed. The parallel compressor [PC] will only be operating if the amount of flash gas is sufficient, to ensure that the compressor is operating under appropriate conditions. So, if the amount of flash gas is low enough, the flash gas is throttled through the FGBV as in the standard booster system mentioned in the previous section (Kauko, et al., 2016). Below a log p-h diagram for the booster system with parallel compression is shown.

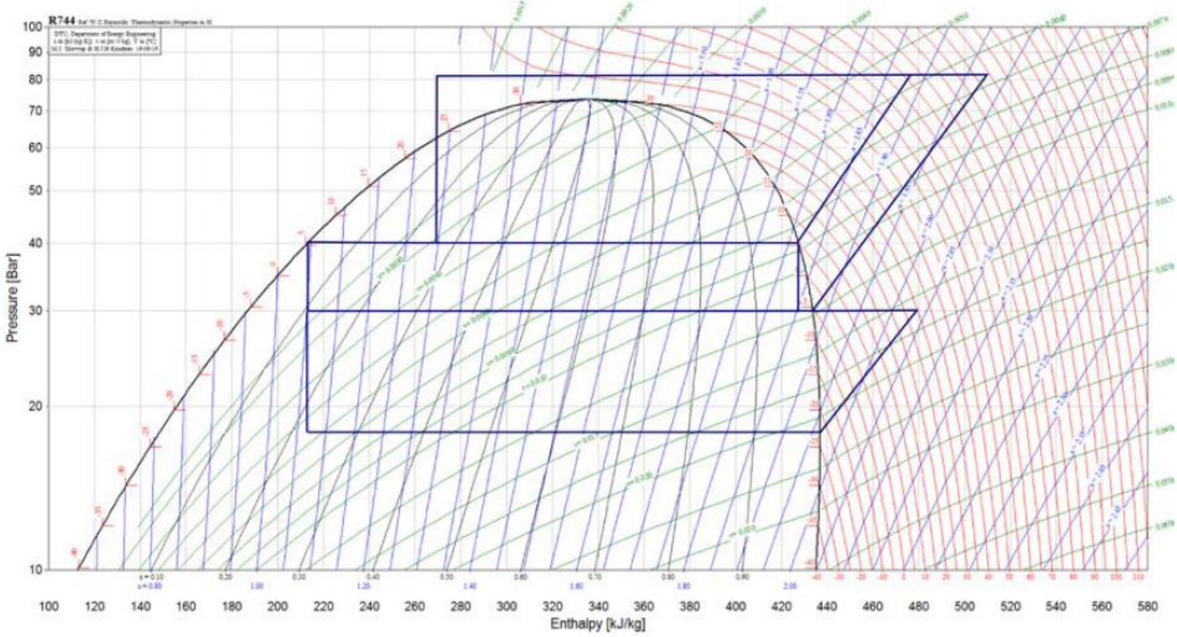

Figure 6-6: Log p-h diagram for a transcritical booster system with parallel compression (Kauko, et al., 2016).

The energy efficiency of a transcritical booster system is raised by use of a parallel compressor due to the saved compressor work. The auxiliary compressor operates with a lower pressure lift than the MT-compressor, and thus require less energy. However, the auxiliary compressor should compress the total amount of flash gas to accomplish the best performance. Therefore, a need for a large parallel compressor would increase the total investment cost, and an analysis of cost vs energy trade-off should be executed (Gullo, et al., 2016).

6.2.3 EJECTOR

When the refrigeration system utilizes R744, one of the major contributors to losses in the system is throttling losses. This is due to the high $\frac{Cp}{\Delta h_f}$-value, and the throttling loss is particularly large in CO_2-systems operating in warm climates because of a higher exiting temperature from the gas cooler which gives higher specific enthalpy. But even in colder climates such as in Norway, the throttling losses are significant. One of the newest technologies to recover this loss is by replacing the high-pressure control valve with ejectors. The ejector operates by partly entraining the suction fluid downstream of the MT evaporators, by means of the motive fluid that is exiting the gas cooler. The motive fluid is accelerated in the motive nozzle (Kauko, et al., 2016). A principal sketch of an ejector is found below.

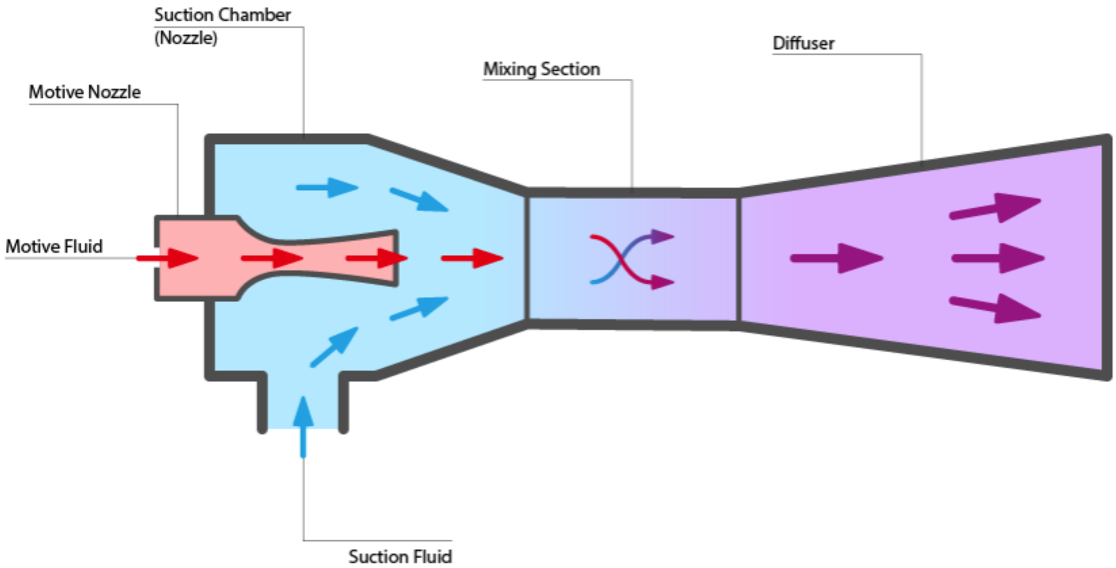

Figure 6-7: Principal sketch of an ejector (Carel, u.d.).

The ejector converts expansion energy with a given temperature and pressure into increased suction pressure for the compressor, which results in the ejector doing compressor work without the need for extra power consumption. The refrigerant is then discharged to the receiver. The amount of refrigerant available for the ejector depends on the available expansion work. Using a system with an ejector enables the possibility to operate the evaporators without superheat, flooded mode, which provides higher heat transfer rate and better utilization of the heat exchanger area. As a consequence, the refrigeration system can be operated with higher evaporation temperature which leads to increased overall energy efficiency (Kauko, et al., 2016).

The pressure profile of the mixing process can be seen below. Here, the gas enters from the gas cooler at a high pressure (denoted P_H), and it flows through the throat which accelerates the flow. The gas is now at supersonic speed, which creates the low pressure (P_s). The pressure P_s is lower than the suction fluid pressure (P_L) so the suction fluid now flows into the suction port. As the flows are combined in the mixing section, the pressure gradually increases. At the diffuser, the flow velocity decreases causing the pressure to rise further. When the gas leaves the ejector through the diffuser, the pressure (P_D) is higher than the pressure of the suction fluid (P_L) (Danfoss, 2018).

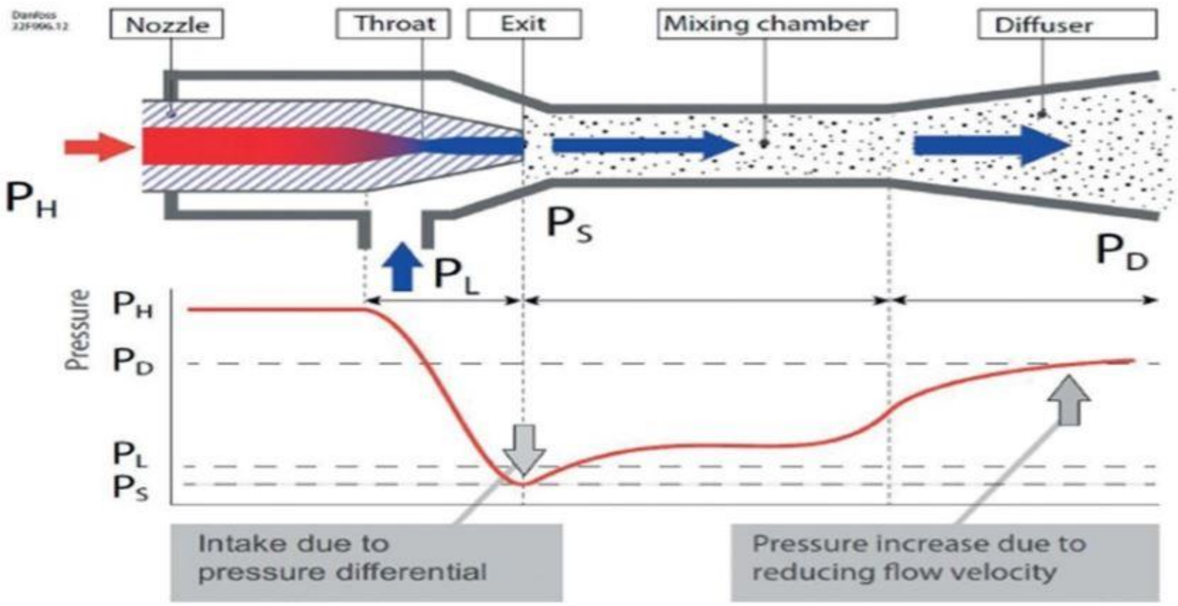

Figure 6-8: Pressure profile ejector (Danfoss, 2018).

A p-h diagram and a schematic of a transcritical booster system with an ejector is included below. The system schematic includes numbered points which one can locate in the p-h diagram.

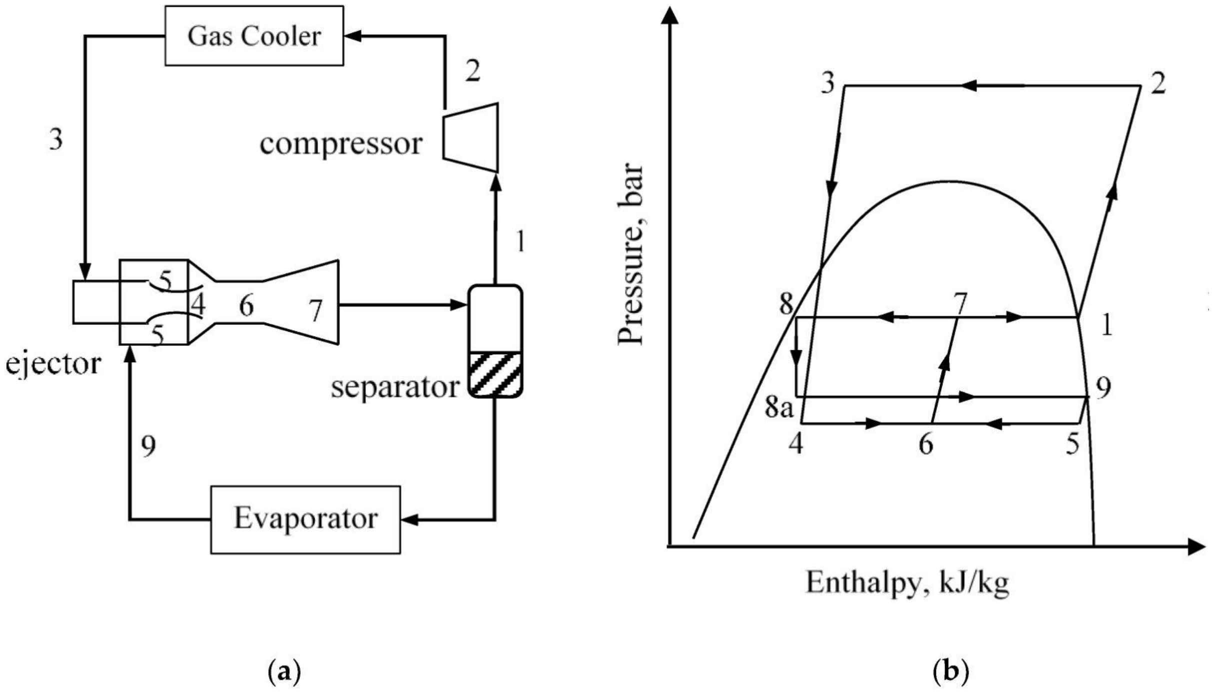

Figure 6-9: (a) Schematic of system with ejector. (b) P-h diagram of a transcritical CO_2 refrigeration system (Bruno, et al., 2019).

The efficiency of incorporating the ejector to the cycle depends on the performance of the ejector with evaluation from both expansion and compression functions (Fornasieri, et al., 2009). The COP of a refrigeration system utilizing R744 with an ejector is found to depend on a few crucial points. Critical entrainment ratio of the ejector, optimal heat rejection pressure, and critical outlet temperature of the gas cooler. By implementing an ejector into the transcritical CO_2 system an increase in COP of 30 % and reduction of exergy loss by 25 % was found in warmer climates (Fangtian & Yitai, 2011).

7 METHODOLOGY

To answer the stated problem, several approaches were used for different parts of the work. The methods section is therefore divided into multiple parts to give an overview of the work.

7.1 Literature review

Several sources were used to gather information about CO_2 as a working fluid and the refrigeration systems operating with this refrigerant. Old lecture notes from the course TEP20, TEP4260 and TEP4255 was used to provide a basis to conduct the literature review. These lecture notes gave a basic understanding of how CO_2 operated as a working fluid, and the common losses that can be present in a cycle that utilizes R744 as the working fluid.

Obtaining the information for the literature review was done by using Google, Google Scholar and Oria as search engines. Search words such as “banning of working fluids”, “CO_2 refrigeration”, “transcritical systems”, “natural working fluids”, “and CO_2 refrigeration ejector”, “CO_2 refrigeration parallel compression” were used to develop the literature review. Parts of the research is not shown in the theory section because some of it doesn’t fit the theme of the project work, and some of it fits the work that is going to be done in the master thesis and is therefore going to be included in the thesis. When searching for relevant literature, a wide search was conducted in the beginning and was then narrowed during the building of the knowledgebase. Started out with general publications on refrigeration and CO_2 as a refrigerant, and slowly focused in on different configurations of the refrigeration system and evaluation of these. Published articles from science direct and journal refrigeration was prioritized when choosing literature to base this project work on, as well as report from comprehensive analysis of CO_2 systems. Because of the rapid development in the field, the newest articles were focused on to ensure that the information was up to date. Further on, the curriculum presented for the course TEP4525 on CO_2 was used as support literature and as a way of verifying the articles that were found in the literature search. Both in terms of the trustworthiness and if the information was up to date.

7.2 Analyzing Spar Snarøya and Spar Røyken

For the part with description of the different systems that are going to be analyzed in the master thesis, several different sources were used. Firstly, oral communication in several meetings with Armin Hafner, supervisor from NTNU and co-supervisors Eirik Rødstøl and Erik Hoksrød from Kelvin AS. A previous master thesis involving one of the two systems that is going to be analyzed in this project work and in the master thesis was obtained (Hoff, 2020). This master thesis was used as support when describing the system at Spar Snarøya. Kelvin AS provided system drawings for the two refrigeration systems that are going to be analyzed in addition to access to IWMAC, which is used to get an overview of the two refrigeration systems as well as obtaining measurements. Technical descriptions were also provided in a folder.

The calculation procedure for each of the system have some similarities as well as a few differences. The similarities are presented first, followed by the specifics for each refrigeration system.

7.2.1 SHARED FEATURES

When evaluating the system performance of the two systems in question there were a few common features in calculation and data processing. These commonalities are discussed in this section of the methodology chapter.

7.2.1.1 IWMAC

For analyzing of the data the selected parameters were chosen in the settings section, shown with a cogwheel in IWMAC. A screenshot of the menu with the cogwheel is added in Figure 7-1 with the settings section is marked with a red circle.

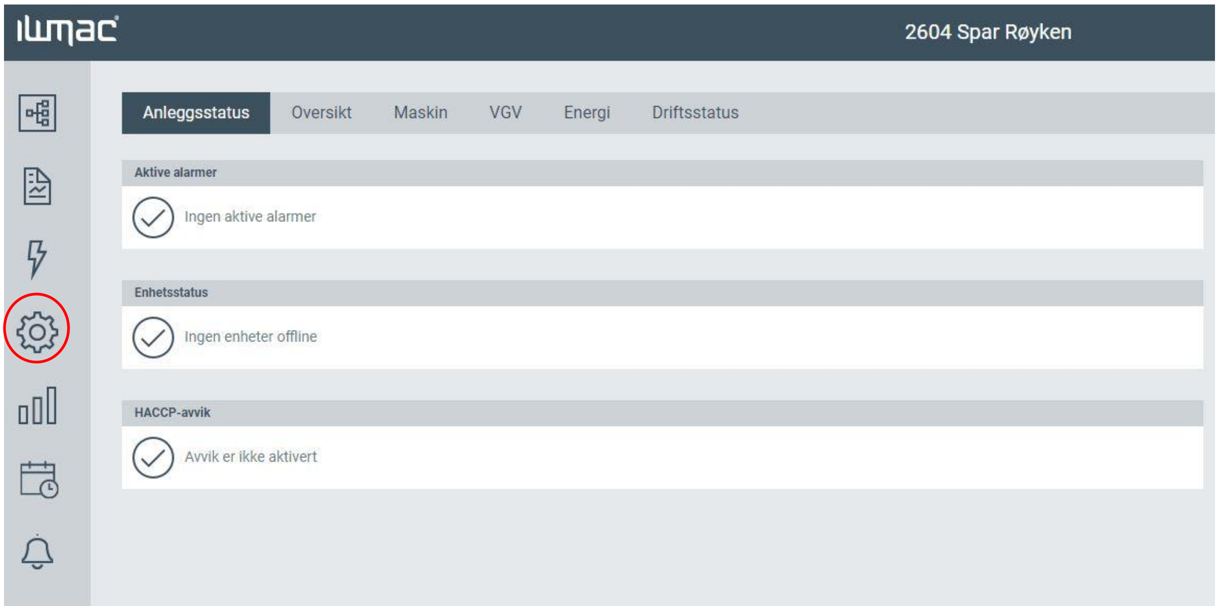

Figure 7-1: Menu IWMAC

The graph section were then chosen, and the given dates were selected in the menu. When exporting the values of the chosen variables from IWMAC to Excel, the timeslot had to be set to two hours earlier than the timeframe that was intended. This was due to a fault in the web page (i.e. the time slot had to be set from 08:00-20:00 if one wished for the exported values to range from 10:00-22:00). The parameters and given values were then exported to an excel-file, where the calculations were conducted. An example of the export-file can be found in Appendix A.

The energy surveillance system that both Spar Snarøya and Spar Røyken utilize, log the different values for the parameters only when there is a change in the given parameter. As a result, the parameters have different timestamps, and needed to be normalized in order to analyze them. One of the co-supervisors from Kelvin stated that even if two values are timestamped at the same time, doesn't necessarily mean that the value changes at the same time. So some offset from the timestamp may have to be taken into account when evaluating the results. As a result of this information, it was deemed reasonable to either use average values, use a fixed time period for collecting values, or collect values every x minute.

For Spar Røyken, the time interval chosen to compare the parameters were two minutes. The value of each variable was counted every two minutes when developing the variable graphics in 0, and the average value over the given time period was used in the comparison section. When collection values every two minutes, some values may be left out, if the parameters changes several times during the two minute period. A selection of parameters can be seen below with the deviation from the export value. The example involves the parameters that change the most during the period that is analyzed. As one can see in the table below, the deviation is between 0% and 2%. The variables shown below is used for comparing the system with and without ejector operation[1].

Table 7-1: Overview of deviation for parameters

Parameter	**Date**	**Time**	**Export value avg.**	**2min-value avg.**	**Deviation**
Vrec OD [%]	27.04.2021	10:10-11:10	67,99	69,99	2,00
Vrec OD [%]	15.07.2021	13:00-14:30	88,80	88,65	0,15
MT run cap [%]	27.04.2021	10:10-11:10	52,83	53,85	1,01
MT run cap [%]	15.07.2021	16:14-18:00	57,47	57,48	0,01
MT suction gas [°C]	27.04.2021	10:10-11:10	7,34	7,34	0
MT suction gas [°C]	15.07.2021	18:54-20:00	12,49	12,42	0,07

When the comparison between the chosen period with the ejector on and the period with the ejector off was conducted, a 30 minute average of each chosen variable was used to create an impression of the state of the variables in that time interval.

[1] Vrec OD: Used in system graphics. MT run capacity and MT suction gas: used in system graphics and comparison between operation mode one and two at Spar Røyken

For Spar Snarøya, the first part of the calculations were done with an hourly average. This was chosen because of the long duration of parallel compressor operation. To get an overview of the entire 24 hour period, the hourly average was deemed appropriate. The second part which consist of comparison between the system in operation mode one and operation mode two, a smaller time interval was chosen to make sure that the values of the variable didn’t hide as many fluctuations as one might find with an hourly average. Here the average was calculated over a 30 minute period for comparison.

7.2.1.2 Refrigeration load

The approach for calculation the refrigeration load for the systems were similar. The given period was chosen, and excel was used to do the calculation. The average run capacity and suction temperature on both the MT and LT compressor rack was calculated, as well as outlet temperature from the compressors. The evaporation pressure on MT and LT level was obtained and the average value in each time interval was calculated. From the average evaporation pressure and suction gas temperature an online software [2] was used, where these two variables were the input values. An example can be found in Appendix B. Here the density and specific heat capacity for each time interval was collected and entered in the Excel sheet, there is an example of the Excel-calculations in Appendix C. To calculate the mass flow through the compressors the following equation was used:

$$\dot{m}[\frac{kg}{s}] = \frac{(Run\ capacity\ [\%] * Stroke\ volume[\frac{m^3}{h}]) * Density[\frac{kg}{m^3}]}{3600}$$

(1)

After calculation of these values, the mass flow, delta T over the compressor and specific heat capacity were used to calculate the load with equation (2).

$$Q[kW] = \dot{m} * Cp * \Delta T$$

(2)

To then calculate the refrigeration load, the specific load was divided by total installed cooling capacity for each of the pressure levels. The following equation was used:

$$Refrigeration\ load\ [\%] = \frac{Q_{Interval}}{Q_{Total}}$$

(3)

[2] Online software: https://www.peacesoftware.de/einigewerte/co2_e.html

7.2.2 SPAR SNARØYA

As a starting point, the temperature profile at Snarøya was examined for the year 2020. The date 19th June 2020 stood out as the day with the highest temperature and as this was a Friday where the supermarket was open, this day was chosen for closer evaluation. After exporting the values, one could see that the parallel compressor was running the duration of the opening hours of the supermarket.[3] This was also the case for the AC-loop due to the high temperatures that day. Several variables was obtained from IWMAC and exported to excel, and with excel commands the average values for the selected time interval was calculated.

7.2.2.1 Air condition calculations

To calculate the AC-demand, there were to different approaches. The expansion valve that is located upstream of the cooling coil is a Danfoss expansion valve, with the product name CCM10-1. The initial approach was by using the free cooling calculation software from Danfoss, "Coolselector" version 4.5.2. [4] The goal was to use the software with the properties obtained from IWMAC to obtain the mass flow through the valve and use this to calculate the AX-demand. However the support team for Danfoss Coolselector informed that the software does not support the CCM10-1 valve on the transcritical line. This method for calculating the AC-demand was as discarded as a result of this information.

The second solution for this calculation was to use excel, Mollier diagram and equations to obtain the AC-demand. This solution increased the degree of uncertainty, but was the next best thing when Coolselector no longer was an option. To do this, the values upstream and downstream of the cooling coil in combination with values obtained from the ventilation system was utilized. The volumetric flow of air in the ventilation system, the air temperature entering and exiting the cooling coil and the inlet and outlet temperature of the refrigerant was exported from IWMAC to excel. The inlet temperature of the CO_2 was used as the temperature of the cooling surface, and a Mollier diagram was used to estimate the enthalpy difference between the measuring points in the ventilation system. To calculate the demand, the volumetric flow was converted to mass flow and added to the enthalpy difference as seen in equation (4).

---

[3] Opening hours Spar Snarøya: 08:00-22:00
[4] https://www.danfoss.com/en/service-and-support/downloads/dcs/coolselector-2/

$$Q[kW] = \dot{m}[\frac{kg}{s}] * \Delta h[\frac{kJ}{kg}]$$

(4)

These results were in turn used to calculate the mass flow of refrigerant through the cooling coil. The enthalpy for each point of the refrigerant was obtained by use of online software. [5] The calculation was done by the equation (5).

$$\dot{m_{CO2}} = \frac{Q_{AC}}{\Delta h_{CO2}}$$

(5)

7.2.2.2 P-h diagram

For the drawing of the P-h diagram, the different values were obtained from IWMAC and processed in excel. The average values for the given time interval was calculated and used as markers for points in the diagram. The outlet temperature for the compressor racks are measured a distance downstream of the compressors, so the discharge temperature was calculated by the use of isentropic efficiency. The Bitzer simulation software[6] for CO_2 systems was utilized for this, the average input values was used and the software gave a higher discharge temperature than the one that was stated in IWMAC. This discharge temperature in combination with the input power from Bitzer was in turn used to calculate the isentropic efficiencies by equation (6). This calculation was done to be able to account for the heat loss in the compressors.

$$\eta_{is} = \frac{\dot{m} * (h_{2is} - h_1)}{W}$$

(6)

The isentropic enthalpy was obtained by using online software for properties [7] with the suction gas temperature and the suction pressure as input for the specific entropy. Discharge temperature from the Bitzer software and discharge pressure from IWMAC was then used to

[5] Online software: https://www.peacesoftware.de/einigewerte/co2_e.html
[6] https://www.bitzer.de/websoftware/
[7] https://www.peacesoftware.de/einigewerte/co2_e.html

acquire h_{is} with the same online software. An example of input and output values from the software can be found in Appendix B.

When using the software from Bitzer the compressor models weren't an option as they have been replaced by newer models. As a result of this, the newer models had to be selected for the simulations Table 7-2 show an overview of the compressors used in the Bitzer software and the corresponding compressors in the system.

Table 7-2: Compressors used in Bitzer simulations Spar Snarøya

Compressors in the system	**Compressors in Bitzer software simulation**
4PTC-7K	4PTE-7K
4MTC-10K	4MTE-10K
4KTC-10K	4KTE-10K
4JTC-15K	4JTE-15K
2KME-1K	2KME-1K
2KME-1K	2KME-1K

7.2.2.3 Parallel compressor

A few approaches was used to estimate the mass flow through the parallel compressor as this isn't logged in IWMAC. The only variable available from the online monitoring system is whether the parallel compressor is running or not, this was simply states as "1" for off and "2" for on. The closest variables that are logged are the receiver pressure and on the high pressure side it is downstream of the point where the refrigerant from the MT-compressors and the refrigerant from the parallel compressor mixes.

Solution one was to utilize the Bitzer software[8] with the given values from IWMAC. This approach was used when calculating the mass flow for the refrigeration load, as well as ΔT over the compressor. The hourly average for each of the input variables was calculated by use of Excel and the mass flow was returned. One example of the Bitzer simulation can be seen in Appendix D. For the Bitzer software, the same choices had to be made regarding compressor models as mentioned in section 7.2.2.2 and compressor number three on MT-level (4KTC-10K) were not included in the simulation as this compressor was off the entire simulated period. The air condition demand that was calculated as explained in the previous section had to be used during this simulation. If the simulation was executed without that load, the error "too large/too many compressors in the parallel-stages" was displayed in the Bitzer software. However, the placement of the air condition is a bit different than in the actual system. In Bitzer it is displayed in connection to the receiver, but in the system at Spar Snarøya it is located upstream of the receiver. Figure 7-2 shows screenshot of location of the AC in the system schematic in Bitzer as well as input for air condition, both are marked in red.

[8] Bitzer online software: https://www.bitzer.de/websoftware/

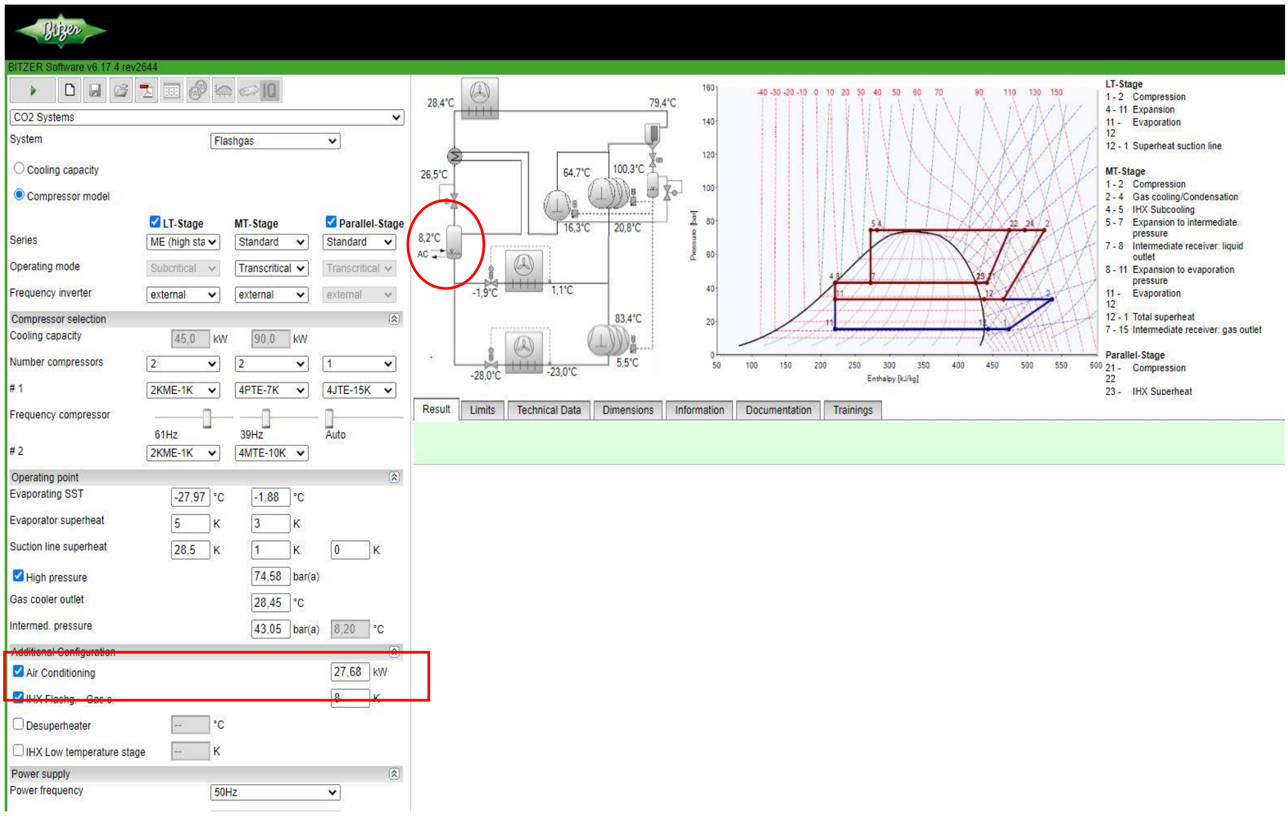

Figure 7-2: Screenshot of input AC-demand in Bitzer

The gas cooler pressure was used as the high pressure value, as a result of this, the three first hours of runtime for the parallel compressor could not med estimated. The average pressure was below 73.77 bar, and the Bitzer software could not estimate the values with the input variables.

The input variables for the simulation software are listed below, and the input field can be seen in Figure 7-2.

- Frequency MT compressor 1
- Frequency LT compressor 2
- Evaporation temperature MT
- Evaporation temperature LT
- Suction gas temperature MT
- Suction gas temperature LT
- Temperature gas cooler outlet
- Pressure gas cooler
- Receiver pressure
- Compressor models as stated previously
- Calculated AC demand

The second solution was to conduct the same simulations, but without the high pressure value from the gas cooler set to "Auto". This made it possible to run the simulation, and it gave an estimate of the mass flow.

7.2.2.4 Date selection for comparison with and without parallel compressor

When choosing the period to compare the system at Spar Snarøya with and without the parallel compressor, the first criteria was choosing a period with similar run capacity on the MT compressor rack. Second it was deemed as an advantage to choose the same weekday as the day with the parallel compressor to try and eliminate some of the differences. As a result, 29th May 2020 was chosen as the date. More specifically the period between 09:00 and 23:00 were originally chosen because that's the period where the Bitzer simulation software was able to simulate with the gas cooler pressure as well. Further on the time interval with the most similar run capacity on the MT compressors were chosen.

7.2.2.5 System efficiency

When evaluating the system efficiency at Spar Snarøya with parallel compression, the same Bitzer-simulations as done in the previous section was used. When running the simulations the software also estimated a COP-value, which was deemed the best solution to use as one could compare the COP-values by running simulations on the same system without parallel compression.

7.2.3 SPAR RØYKEN

7.2.3.1 Phases

The work regarding Spar Røyken is split into two phases, the reasoning behind this is that there were done a few changes to try to improve the working conditions for the ejector. In the first phase, no changes were done and the system was operating as installed. The duration of this phase lasts from the start of the project work (i.e. autumn 2020), to 28^{th} June 2021. Phase two was initiated as a consequence of the results in phase one. A driver update and a change in the PI control of the MT compressors was implemented after 28^{th} June 2021. During ejector operation the reference point for MT T0 was changes from evaporation temperature to receiver temperature as well. The variables displayed in diagrams for each phases are the same to give an equal basis for evaluation. All the values was exported from IWMAC, and treated in Excel.

7.2.3.2 Dates and time periods

When the dates were chosen for the two phases for analysis of Spar Røyken, the first priority was duration of ejector operation. The advantage of longest possible running time was considered one of the most important factors to ensure that the data pool was large enough to observe a trend in each parameter. As a result of this the date in phase one and two were chosen.

Furthermore, the baseline for comparing the system during ejector operation with the system running without the pressure lift from the ejector, the MT running capacity was the major factor as well as the weekday in an effort to analyze the data with similar conditions. As this part of the analysis was done in phase two, the date pool only consisted of days in July and August, the chosen date, 15th July 2021, described in section 9.2.2 was the only date that matched with the chosen criteria. To compare the system with and without ejector operation, two 30-minute intervals were picked from the chosen date. As there were several options with similar run capacity on MT-level, the two periods were chosen as the second time interval after the ejector either turned on or off. This was to ensure stability in the system.

7.2.3.3 Variables for analyzing of the data

When the data from Spar Røyken was evaluated, some variables were chosen to give a basis for comparison and evaluation of the refrigeration system. The variable “Evap K23” was chosen to represent evaporation temperature at MT-level, as the reference point for MT T0 is changed during ejector operation. Below is a table of the chosen variables, as well as their name in IWMAC. The names from IWMAC are used in chapter 0.

Table 7-3: Variable overview and the corresponding name in IWMAC

Variable	**Name in IWMAC**
Discharge pressure MT compressors	Pc MT comp
Run capacity MT compressors	MT run cap
Requested capacity MT compressors	MT req cap
Reference pressure receiver	Prec ref
Pressure receiver	Prec
Temperature receiver	Trec
Opening degree of the FGBV	Vrec OD
Ejector on/off	Voltage input
Evaporation temperature MT cabinet K23	Evap K23
Reference temperature suction gas	Suction ref
T0 for MT compressors	MT T0
Evaporation pressure MT	Pevap MT
Run capacity LT	LT run cap

7.2.3.4 P-h diagram

When calculating the values for each compressor point in the P-h diagram, two approaches were used. Initially the solution was to use energy input for each compressor to do the calculation, as it is logged in IWMAC, as can be seen in Figure 7-3.

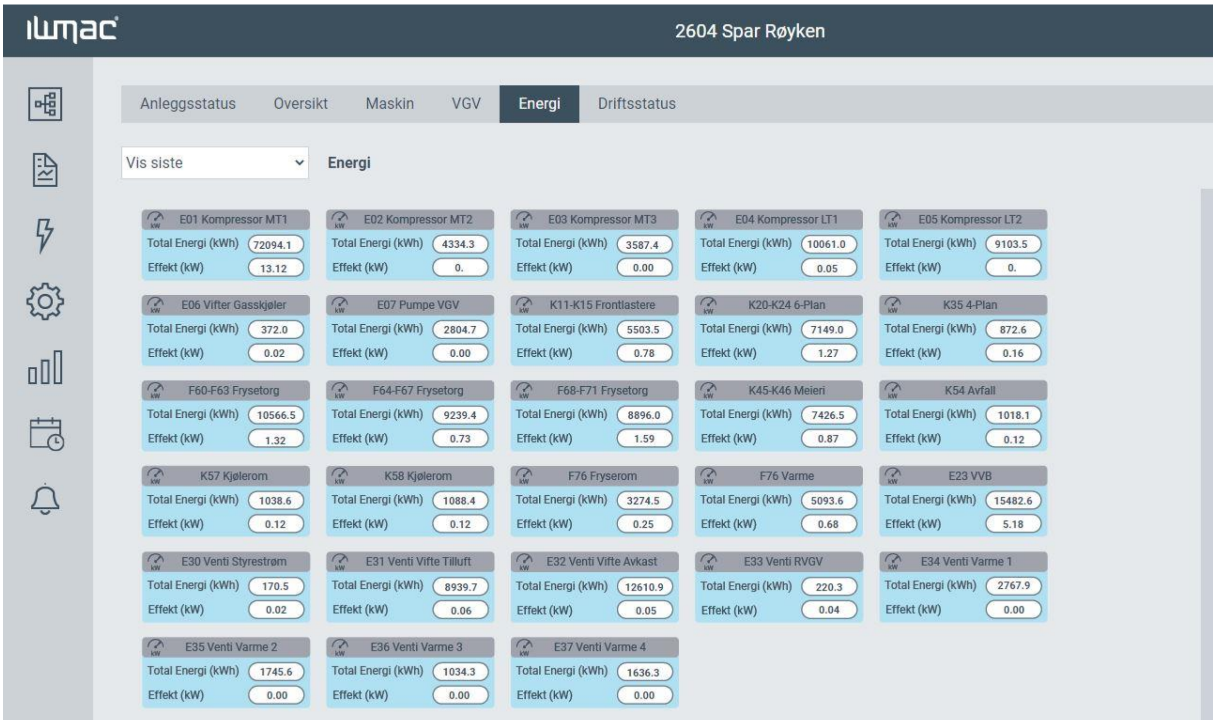

Figure 7-3: Energy overview from IWMAC, Spar Røyken

However, when exporting values from IWMAC there were no values in the export-file. With a closer examination of the values directly in IWMAC, there didn’t seem to be a change in the variable during the chosen time period. This can be seen in Figure 7-4 and in Figure 7-5.

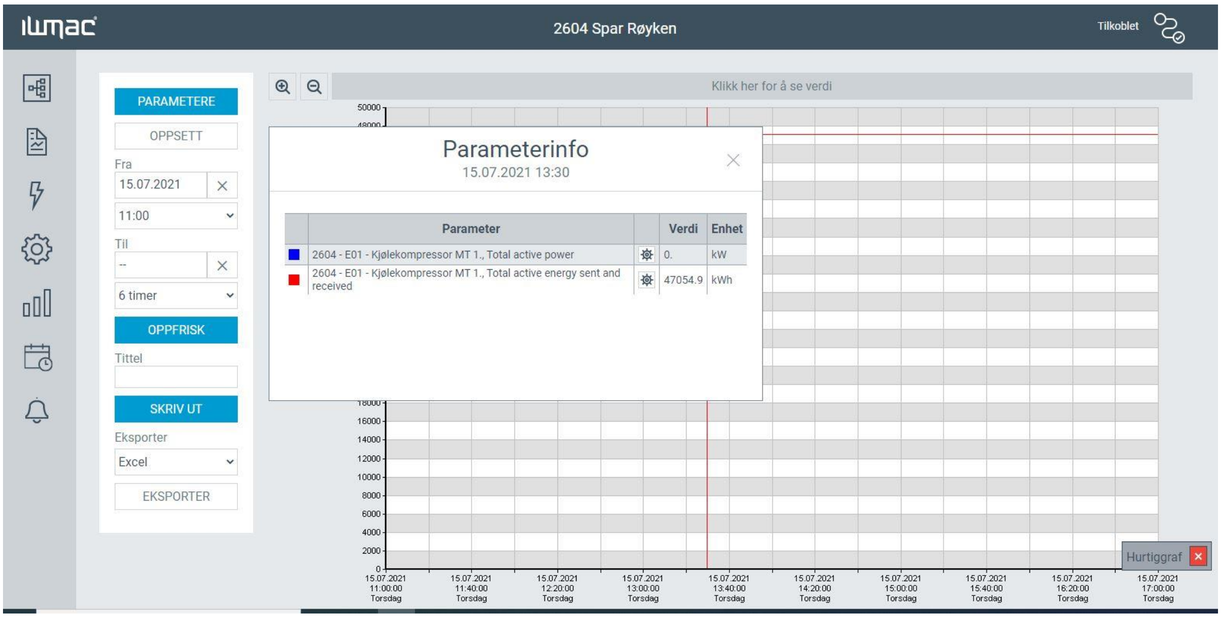

Figure 7-4: Parameter value 13:30 at Spar Røyken

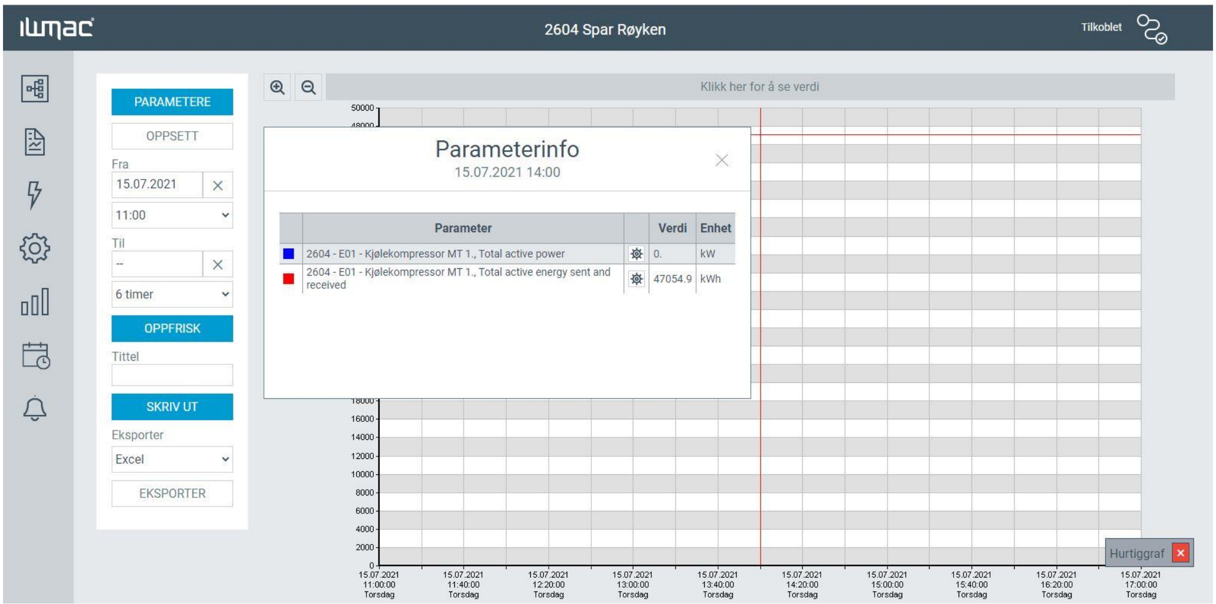

Figure 7-5: Parameter value 14:00 at Spar Røyken

As a result of this, the Bitzer simulation software had to be used to calculate discharge temperatures and isentropic efficiencies with fixed run capacity and evaporation temperature. Each compressor in the system was simulated separately, which gave the opportunity to select older models when executing the simulation. The average calculated isentropic efficiency for each compressor rack was then used for calculation. An example of this simulation can be found in Appendix E.

The points regarding ejector operation in the diagram was estimated by the mass balance seen in equation (7), it states that the total mass flow through the ejector consists of refrigerant from the gas cooler and the mass flow from the MT evaporators. In addition equation (8) was used to estimate vapor quality after the mixing section in the ejector.

$$\dot{m}_{ejector} = \dot{m}_{gas\ cooler} + \dot{m}_{MTevap} \tag{7}$$

$$x = \frac{\dot{m}_{vapour}}{\dot{m}_{liquid} + \dot{m}_{vapor}} \tag{8}$$

8 CASE SUPERMARKETS

There are two systems that are going to be analyzed in this master thesis. They are both been designed and implemented by Kelvin AS, and they are both located in the same area in Norway. The systems operates with R744 as the sole refrigerant, and they are both transcritical booster systems.

8.1 Weather data

Spar Snarøya is located in the municipality of Bærum, and Spar Røyken is located in the municipality of Asker. The two stores are located 19.7 km apart according to Sartopo.[9] The distanced was measured by use of coordinates supplied by the website 1881.no for both Spar Snarøya[10] and Spar Røyken[11]. A screenshot of the input values can be found in Appendix F.

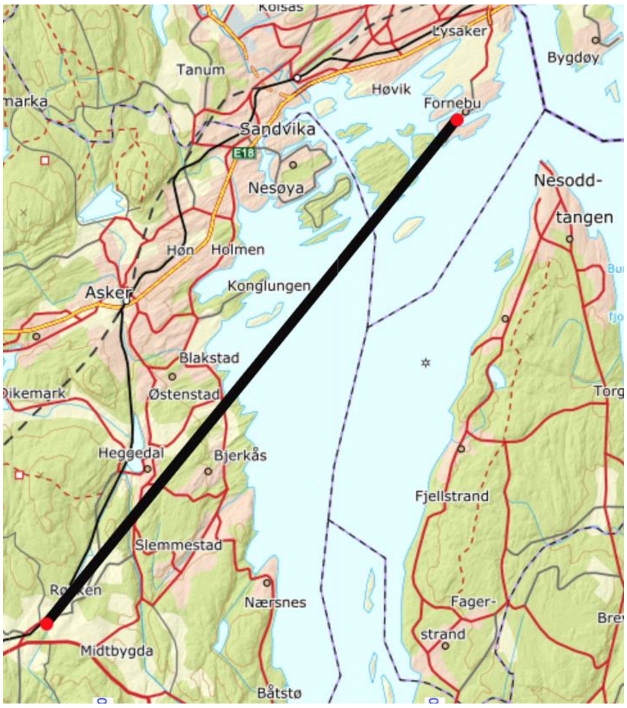

Figure 8-1: Screenshot from Sartopo with Spar Snarøya and Spar Røyken

[9] Sartopo is a collaborative online and offline mapping tool that is used for search and rescue.
[10] https://kart.1881.no/baerum/1367-snaroeya/snaroeyveien-139
[11] https://kart.1881.no/asker/3440-roeyken/braasetveien-3

Climatic conditions are classified by Köppen climate classifications Dfb, which is warm summers and cold winters (Zimmerman, et al., 2018). The closest weather station to both supermarkets is Bygdøy observation station, which is 5km from Snarøya. During 2020 the temperature varied from -9.3°C (27th February) to +31.4°C (19th June) at this weather station (Norwegian Meteorological institute and the Norwegian Broadcasting Corporation, 2021) and for 2021 the lowest temperature was -17.0°C (14th February) and the highest measured temperature was 30.7°C (3rd July) (Norwegian Meteorological Institute and the Norwegian Broadcasting Corporation, 2021). Figure 8-2 shows the temperature fluctuations in 2020 and Figure 8-3 shows the temperature fluctuations in most of 2021.

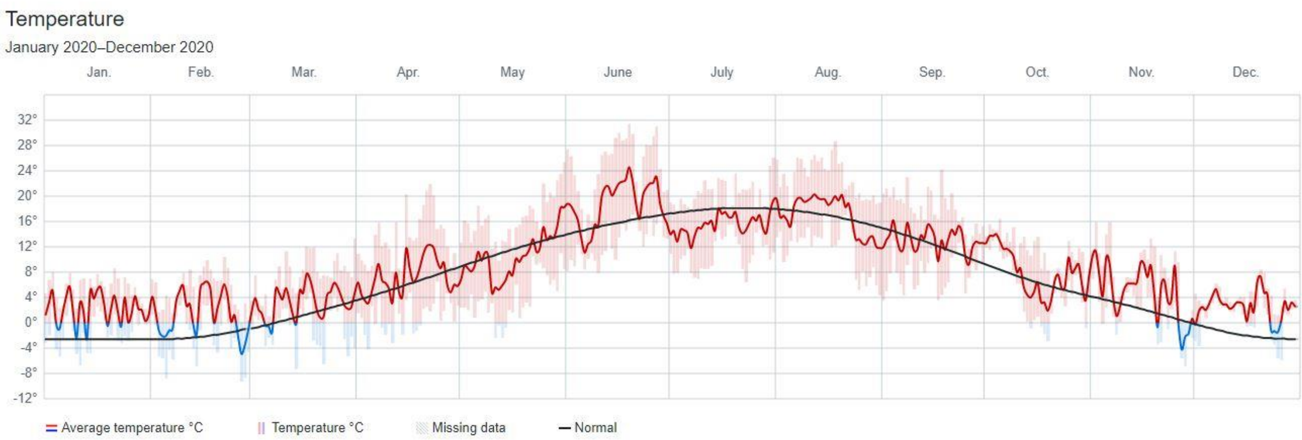

Figure 8-2: Temperature fluctuations at Bygdøy oberservation station in 2020 (Norwegian Meteorological institute and the Norwegian Broadcasting Corporation, 2021).

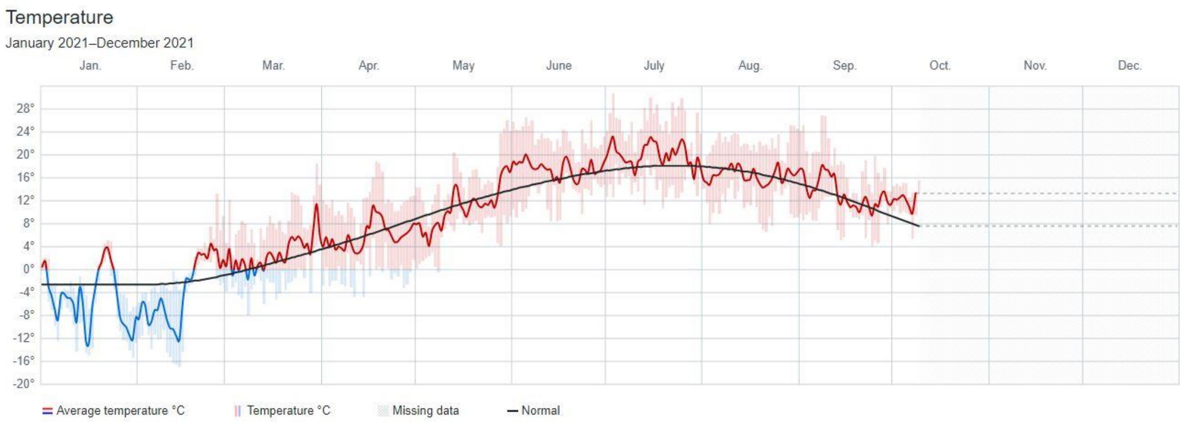

Figure 8-3: Temperature fluctuations at Bygdøy observation station in 2021 (Norwegian Meteorological Institute and the Norwegian Broadcasting Corporation, 2021).

8.2 IWMAC

The two system utilizes an energy surveillance system that is called IWMAC. It's a web monitoring system with the possibility to download values for different parameters. IWMAC is the online management system for both of the stores, the system is utilized to monitor and control the refrigeration systems. Real time information regarding temperature, pressure, operation of compressors and so on can be found here. Historic information can also be exported as an excel-file. The values for the available parameters is updated when a change happens, therefore each of the variables have different timestamps when exported to an excel-file.

In IWMAC there are several different pages where one can get an overview of values in the system. Typically there is one page with a status update, where alarms and blockages are stated. There is an overview of temperatures in the different cabinets, cooling rooms and so on. This is seen in a screenshot in Figure 8-4. By use of the cogwheel mentioned in section 7.2.1.1 one can obtain more values than the ones that are stated on the screenshots as well.

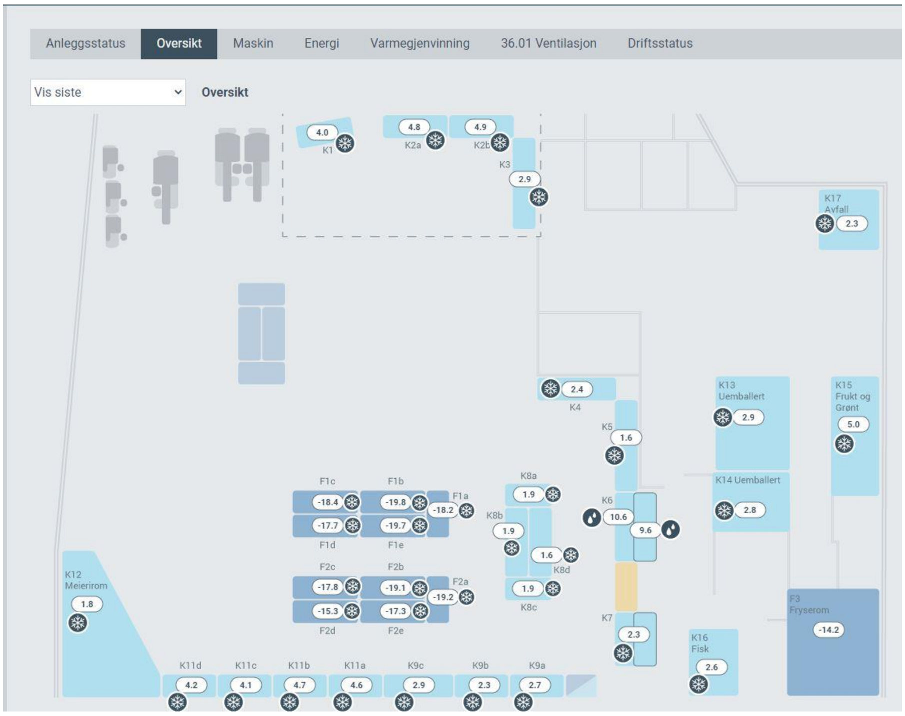

Figure 8-4: Screenshot IWMAC overview of the cabinets

There is also an overview of the machine, the heat recovery and the ventilation system as seen in Figure 8-5 to Figure 8-7.

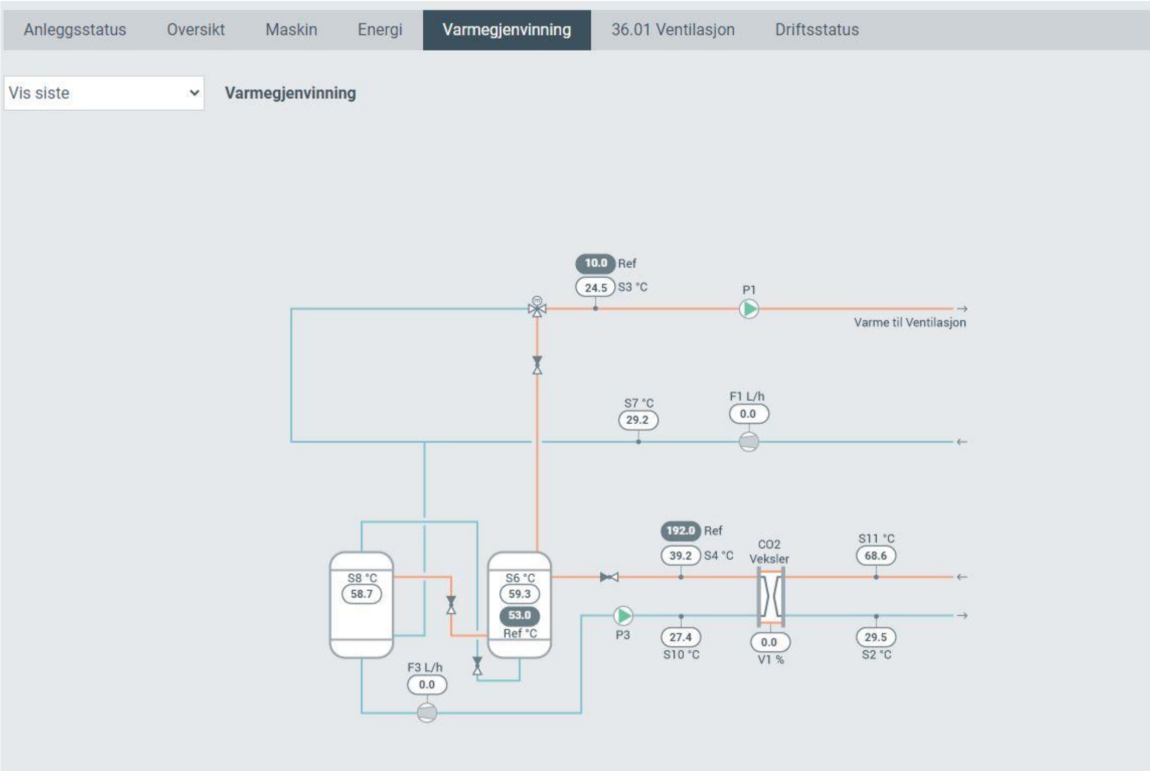

Figure 8-5: Screenshot IWMAC heat recovery unit

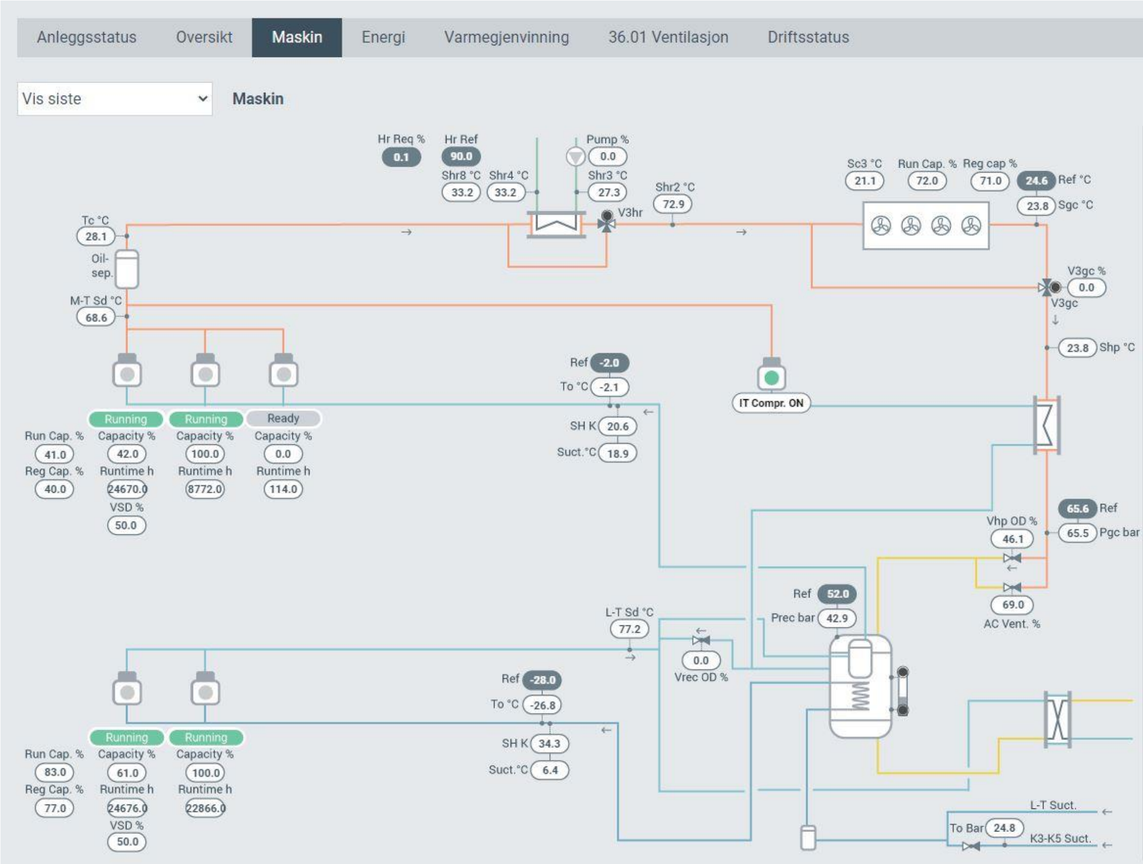

Figure 8-6: Screenshot IWMAC machine

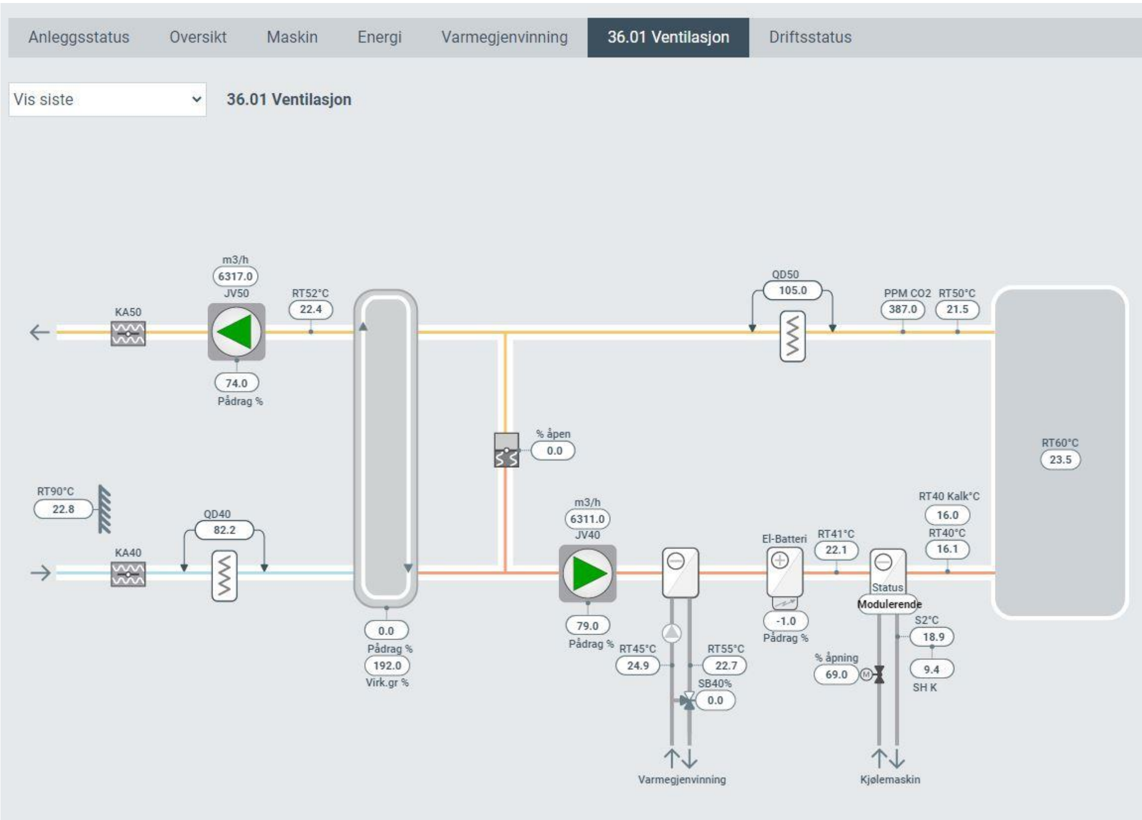

Figure 8-7: Screenshot IWMAC ventilation system

8.3 Spar Snarøya

The refrigeration system at Spar Snarøya is a transcritcal booster system with R744 as the only refrigerant, which was implemented in 2018. This system utilizes parallel compression, according to the concept explained in the literature review. The system at Spar Snarøya has a capacity of 45kW/-2.5°C and 9.3kW/-27°C, and is charged with 140kg R744.

Table 7-1 states the cooling and freezing appliances at Spar Snarøya with the corresponding suction group. Two of the cooling cabinets operate at a different pressure level than the rest, due to a demand for lower storing temperature for the food in question. The refrigerant is throttled by a Danfoss CCM-10 valve to LT level as can be seen in the system schematics in subsection 8.3.3.

Table 8-1: Chiller and freezing appliances at Spar Snarøya

Appliance	Amount	Suction group
Freezers	6	LT
Freezer rooms	1	LT
Coolers	15	MT/LT
Cold rooms	6	MT
Fruit and vegetables cabinets	4	MT

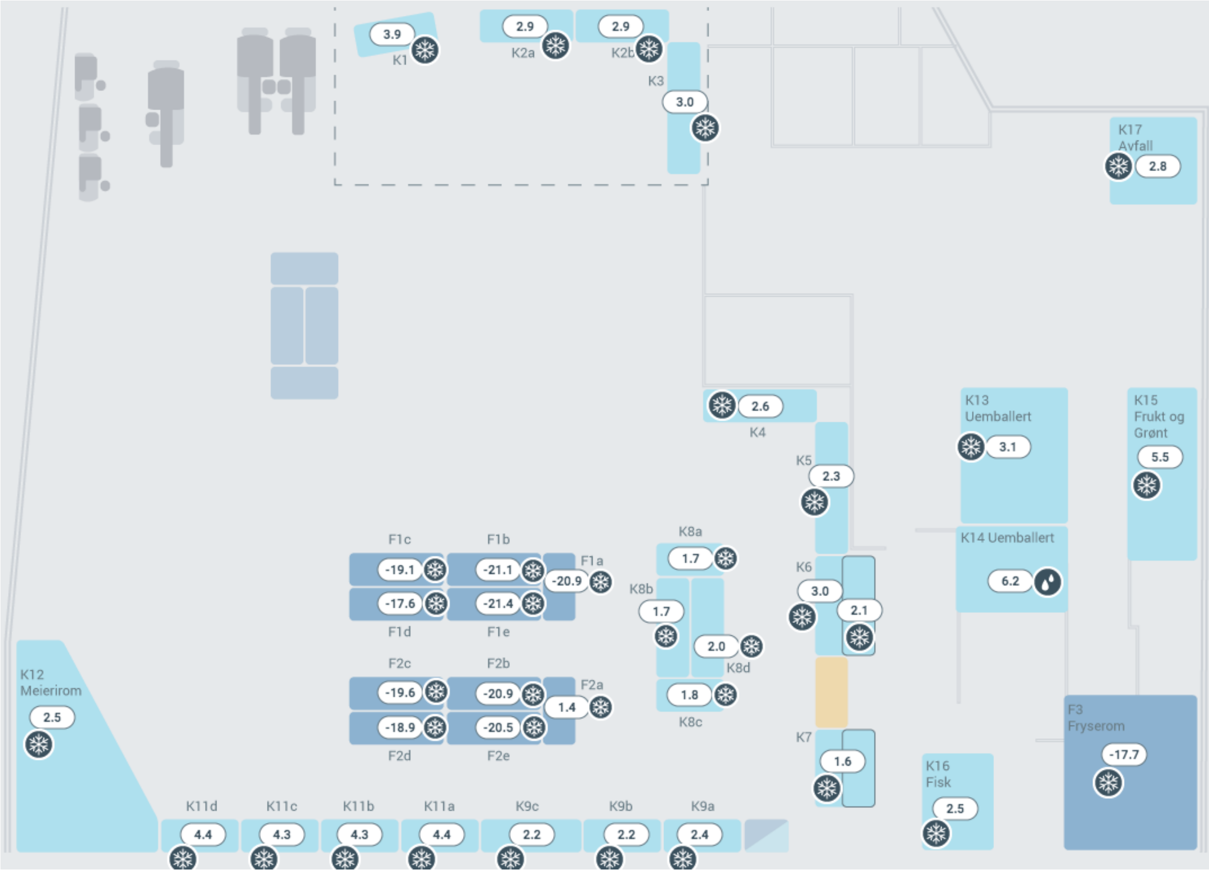

Figure 8-8: Outline of appliances at Spar Snarøya

There are three compressors serving the cooling section, where one of the compressor has variable speed drive [VSD], as well as the auxiliary compressor which also is a VSD-compressor. The parallel compressor is activated by the opening degree of the flash gas bypass valve. When the FGBV opening degree is 26% and above for more than 280 seconds, the parallel compressor is turned on. So 26% opening degree of the FGBV correlates to the lowest capacity of the compressor. When the parallel compressor is activated it is internally regulated directly in the frequency inverter, which is suction pressure/receiver pressure regulated. In the low temperature circuit, there are two compressors, where one of them is a VSD-compressor. An overview of compressors and location can be found in Table 8-2.

Table 8-2: Compressor overview Spar Snarøya

Compressor	Capacity [kW]	Stroke [m^3/h]	Suction group	Frequency [Hz]	VSD
4PTC-7K	9,6	4,3	MT	70	Yes
4MTC-10K	15,4	6,5	MT	50	No
4KTC-10K	22,8	9,6	MT	50	No
4JTC-15K	31,2	9,2	MT/PC	70	Yes
2KME-1K	4,8	2,71	LT	70	Yes
2KME-1K	4,8	2,71	LT	50	No

Figure 8-9 shows the runtime for MT and LT compressors pr. 15th July 2021 at 13:30.

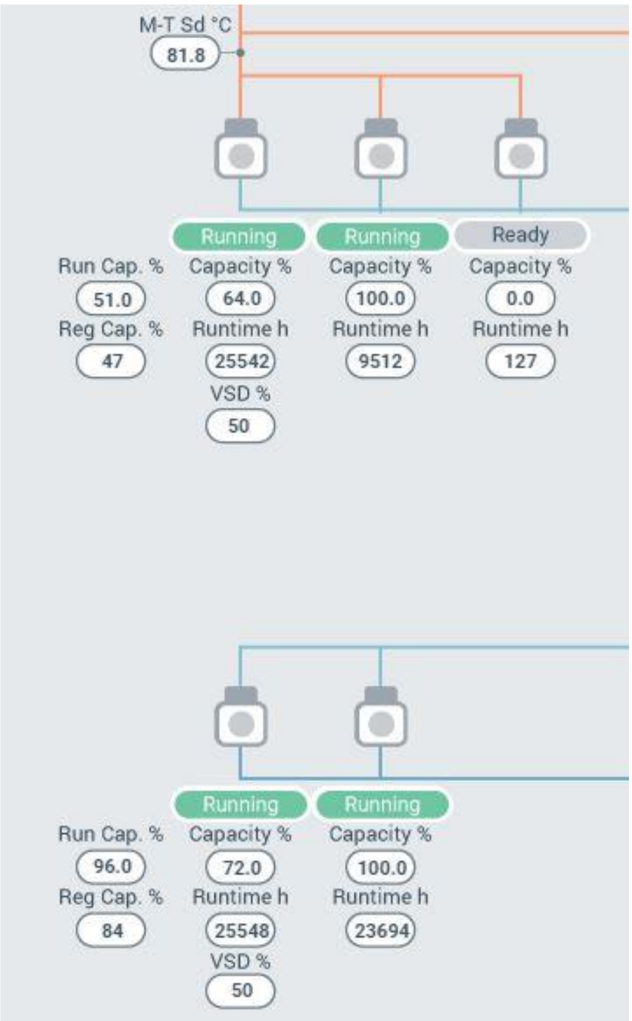

Figure 8-9: Runtime for MT and LT compressors at Spar Snarøya

8.3.1 HEAT RECOVERY

The heat recovery system at Spar Snarøya utilizes the reclaimed heat for heating of water and heating of air. This is done by a heat exchanger where there is CO_2 on one side and pure water on the other side. There are two storage tanks in the heat recovery system with a capability of storing 500 L each. Pure water is circulating between the water tanks and a heat ventilation battery at the air-handling unit and a heat exchanger transferring heat for preheating of tap water. A schematic of this system is included in Figure 8-10.

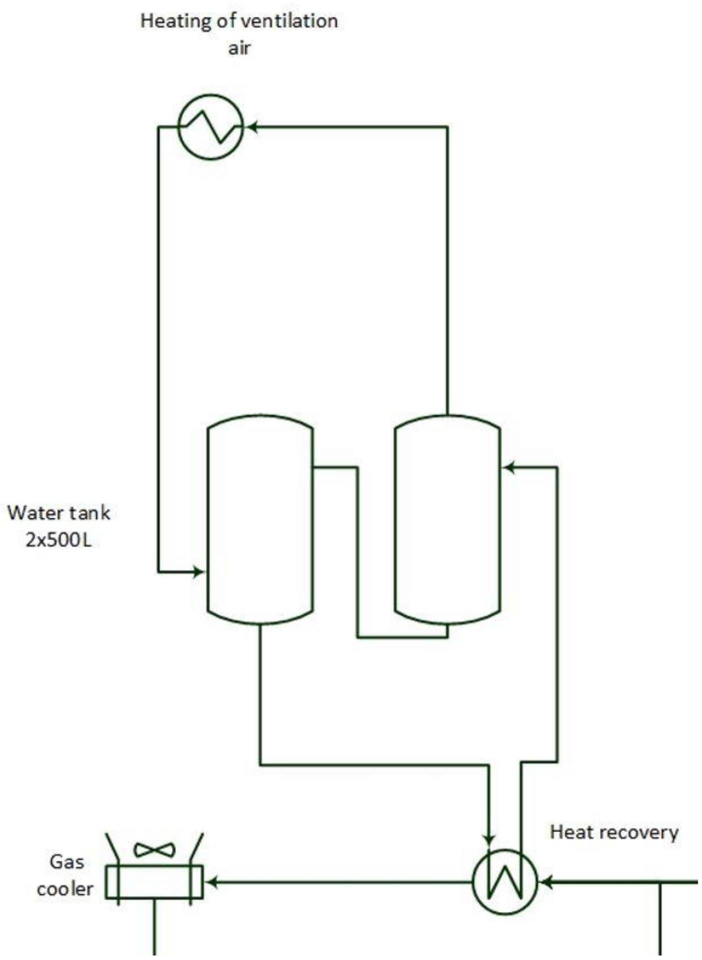

Figure 8-10: Schematic of the heat recovery system at Spar Snarøya

8.3.2 AIR CONDITION

The air condition at Spar Snarøya is integrated into the system. It is regulated by valves on the high pressure side upstream of the receiver. The AC flow expanded by a Danfoss CCM10 valve, as mentioned in section 7.2.2.1. The cooling coil is displayed with the expansion valve opening in the machine overview in IWMAC, and the temperature of the refrigerant is displayed in the ventilation section. The ventilation air temperature and mass flow can also be seen in real time in this section, among other variables. A schematic of the ventilation system is seen in Figure 8-11, with an example of values.

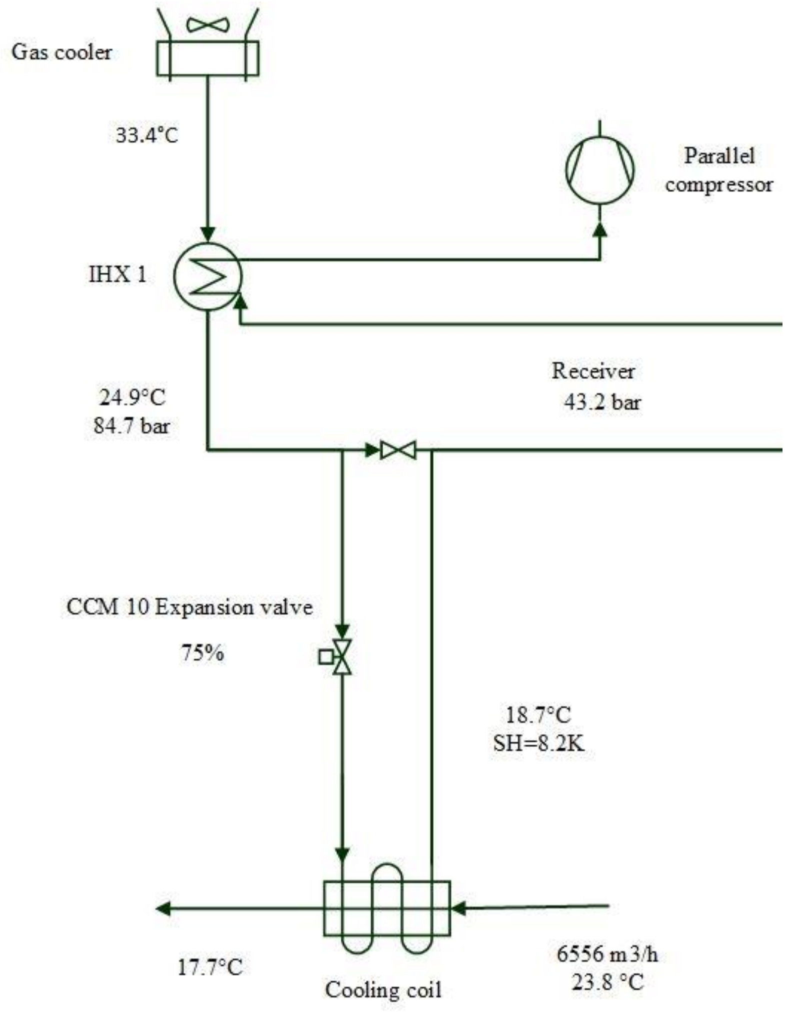

Figure 8-11: Schematic of the air condition solution at Spar Snarøya

8.3.3 OPERATION MODES

Spar Snarøya have two different operation modes, depending on whether the parallel compressor is running or not. Both modes are described in the following subchapters.

At Spar Snarøya there are three internal heat exchangers as can be seen in the system schematics. Internal heat exchanger [IHX] one is located downstream of the gas cooler, internal heat exchanger two can be found in the outer shell of the receiver and internal heat exchanger three is located downstream of the MT-evaporators.

8.3.3.1 Operation mode one

In operation mode one the parallel compressor is running. In this case, the FGBV have been open for 26% or more for 280 consecutive seconds. The flash gas bypass valve then closes, and the gas is supplied to the parallel compressor via IHX 1, which exchanges heat between the flow from the gas cooler and the flash gas from the receiver. The LT-evaporators are supplied refrigerant from the receiver, the refrigerant then travels through IHX 2 in the outer shell of the receiver before it enters the LT-compressors. The refrigerant is then supplied to the inner shell of the receiver.

The MT-evaporators are supplied with refrigerant from the outer shell of the receiver through IHX 3, where there is heat exchange between the supply flow to the MT-evaporators and the refrigerant leaving the MT-evaporators. The CO_2 is then expanded to the correct evaporation pressure. The two coolers that have lower evaporation pressure than the rest of the MT-evaporators have an expansion valve downstream of the evaporators in question, which throttles the pressure down to LT-level. The other part of the MT refrigerant exits the evaporators, goes through IHX 2, mixes with the flow from the LT-compressors and is supplied to the inner shell of the receiver. The MT-compressors is supplied from the inner shell of the receiver where the pressure is elevated, the refrigerant then travels through the heat recovery heat exchanger to the gas cooler where the temperature is lowered. Downstream of the gas cooler is IHX 1, where the flow transfers heat to the supply refrigerant to the parallel compressor. Part of the refrigerant travels through the AC loop if this valve is opened before it is supplied to the outer shell of the receiver.

A system sketch is included in Figure 8-12, where one can observe the location of each component and where the refrigerant flows.

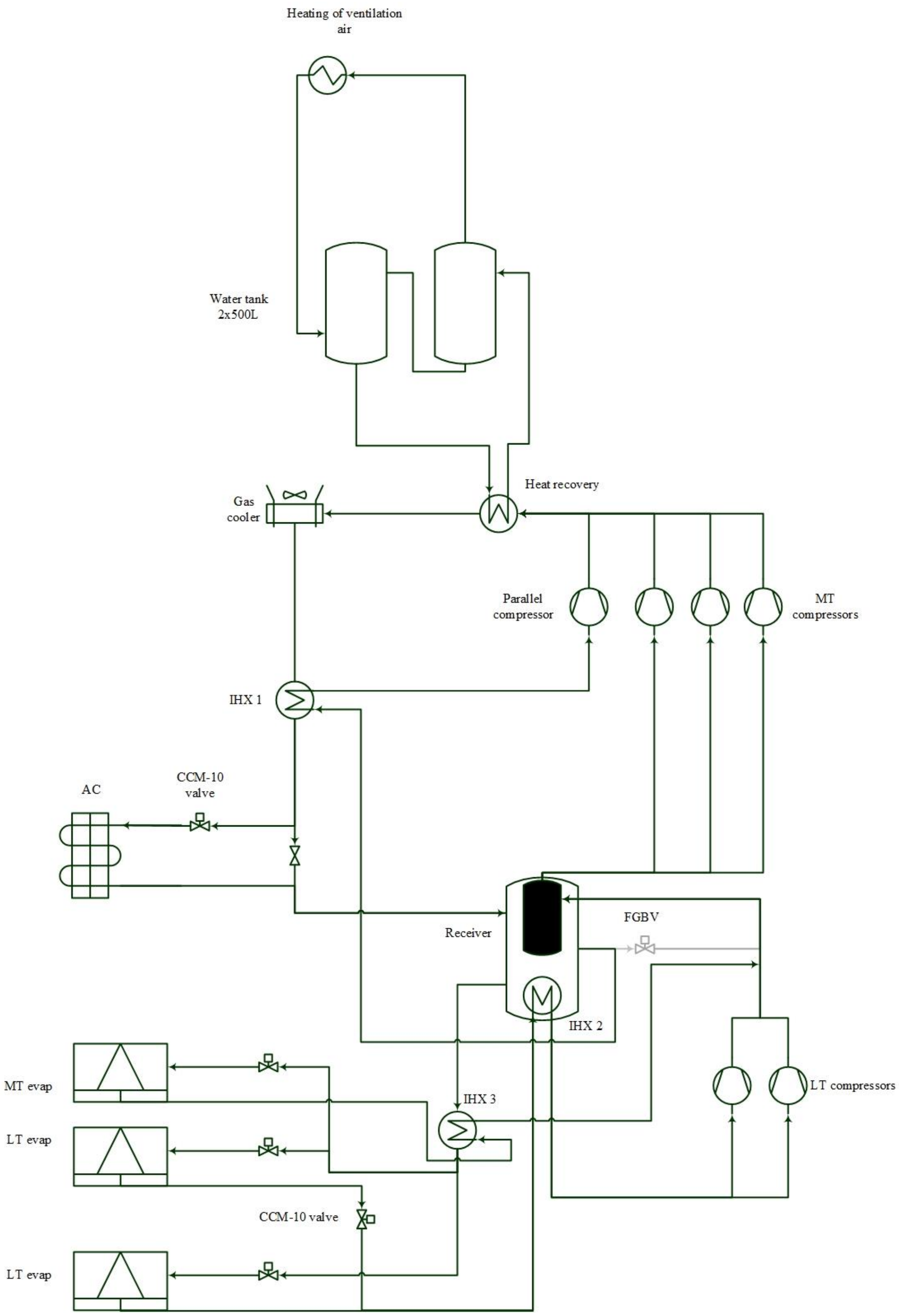

Figure 8-12: Schematic of Spar Snarøya, operation mode one

8.3.3.2 Operation mode two

The system operates similarly to mode one in many aspects. The difference is that the parallel compressor is not activated, so the flash gas bypass valve is partly open. The flash gas goes through the flash gas bypass valve and is mixed with the flow from the LT-compressors, and the refrigerant is supplied to the inner shell of the receiver.

A system sketch is included in Figure 8-13, where the location of the components can be seen as well as the pipes that are in use during this operation mode. The piping between the receiver and the parallel compressor is in this case marked in grey, as it is not being used.

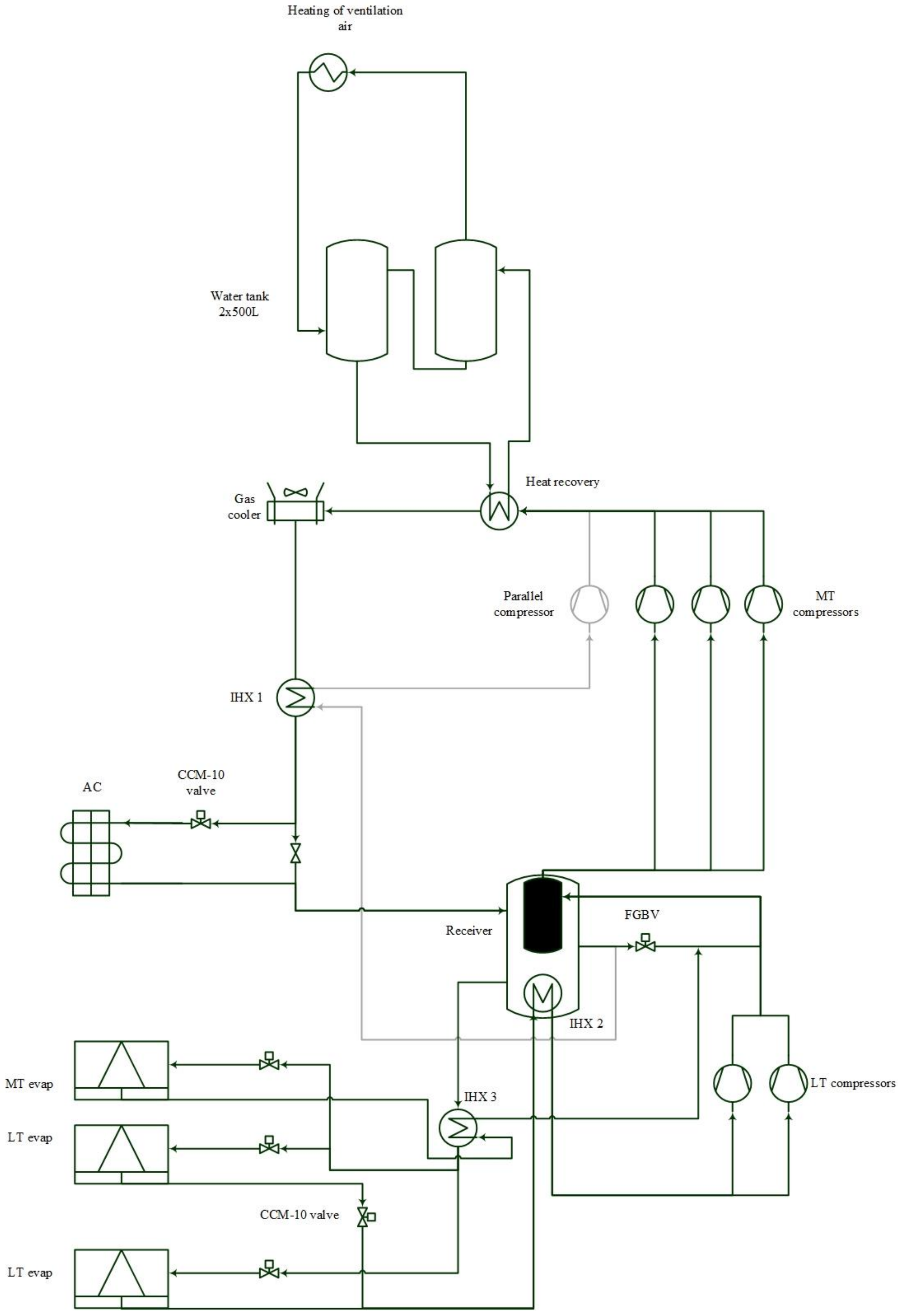

Figure 8-13: Schematic of Spar Snarøya, operation mode two

8.4 Spar Røyken

This is a system which had its start-up in the autumn of 2020. The system was refurbished by Kelvin AS, originally it was a subcritical refrigeration system with R744 as the refrigerant from 2008. The rehabilitation turned the system into a transcritical refrigeration system, with a low pressure ejector and a gas cooler. The system at Spar Røyken have a capacity of 56kW for cooling and 13kW for freezing and is charged with 120kg R744. Table 8-3states the cooling and freezing appliances at Spar Snarøya with the corresponding suction group, as well as a Figure 8-14 showing a screenschot from IWMAC with an outline of the appliances.

Table 8-3: Chiller and freezer appliances Spar Røyken.

Appliance	Amount	Suction group
Freezers	13	LT
Freezer rooms	1	LT
Coolers	12	MT
Cold rooms	6	MT
Fruit and vegetables cabinets	2	MT

Figure 8-14: Outline of appliances at Spar Røyken

There are two MT compressors, where one is VSD, and the same goes for the LT section. The compressor models, stroke volume, capacity and suction group is stated in table 8-4. In addition there is a low pressure ejector system, Multi Ejector LP 1935, delivered by Carel. The placement of this can be seen in the schematic of the system in section 8.4.2.

Table 8-4: Compressor overview Spar Røyken

Compressor	Capacity [kW]	Stroke [m^3/h]	Suction group	Frequency [Hz]	VSD
4HTC-20K	39	12	MT	70	Yes
4JTC-15K	29,7	9,2	MT	50	No
2JME-2K	8,6	3,48	LT	70	Yes
2JME-2K	6,1	3,48	LT	50	No

Figure 8-15 shows the runtime for each compressor at Spar Røyken pr. 15th July 2020 at 13:30.

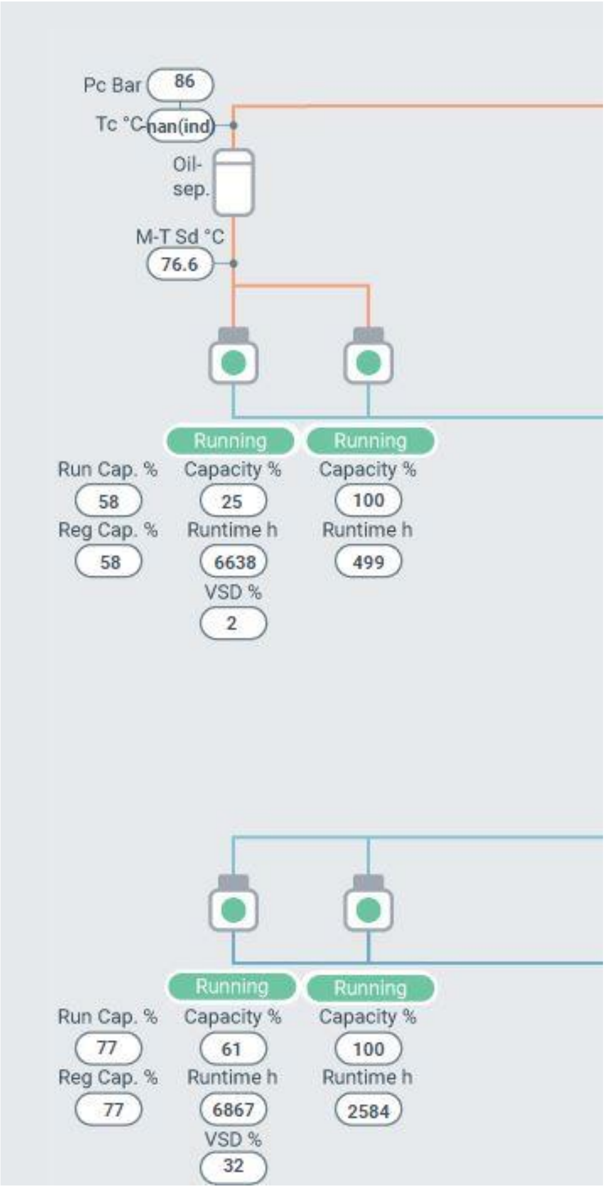

Figure 8-15: Runtime MT anc LT compressors Spar Røyken

8.4.1 HEAT RECOVERY

The heat recovery unit at Spar Røyken is located upstream before the gas cooler. The heat exchanger operates with ethylene glycol on one side and CO_2 on the other side, and is a plate heat exchanger. The reclaimed heat is used for heating of water as well as transferring heat to the ventilation system. The part of heat recovery that supplies heat to the ventilation system is kept from the previous installment.

8.4.2 OPERATION MODES

The system at in Spar Røyken have two different operating modes depending on whether the ejector is being utilized for a pressure lift or not. The ejector is triggered when the opening degree of the FGBV is above 80%.

8.4.2.1 Operation mode one

In operation mode one the ejector works solely as a high pressure valve. The system runs as a regular transcritical booster system. The mass flow from the LT-evaporators travel from the evaporators through an internal heat exchanger, via the LT-compressors and is deposited in the inner tank in the receiver. This is where it is mixed with the flow from the MT-evaporators as well as the flash gas that have gone through the flash gas bypass valve. From this point the refrigerant is compressed by the MT-compressors before it travels through the heat recovery upstream of the gas cooler. Both the heat recovery and the gas cooler have the option of bypass. After going through the gas cooler, the stream travels via the high pressure valve to the outer tank of the receiver. The outer tank is responsible for feeding the evaporators with refrigerant as well providing subcooling via the internal heat exchanger for the flow from the LT-evaporators to the LT-compressors. From the tank the mass flow travels to the evaporators and are throttled down to the correct evaporation-level.

Figure 8-16 shows a schematic of operation mode. The different components are marked, as well as the piping that is being used. As one can see, the piping to the ejector as well as the piping through the additional valve parallel to the FGBV is marked in grey as it is not in use.

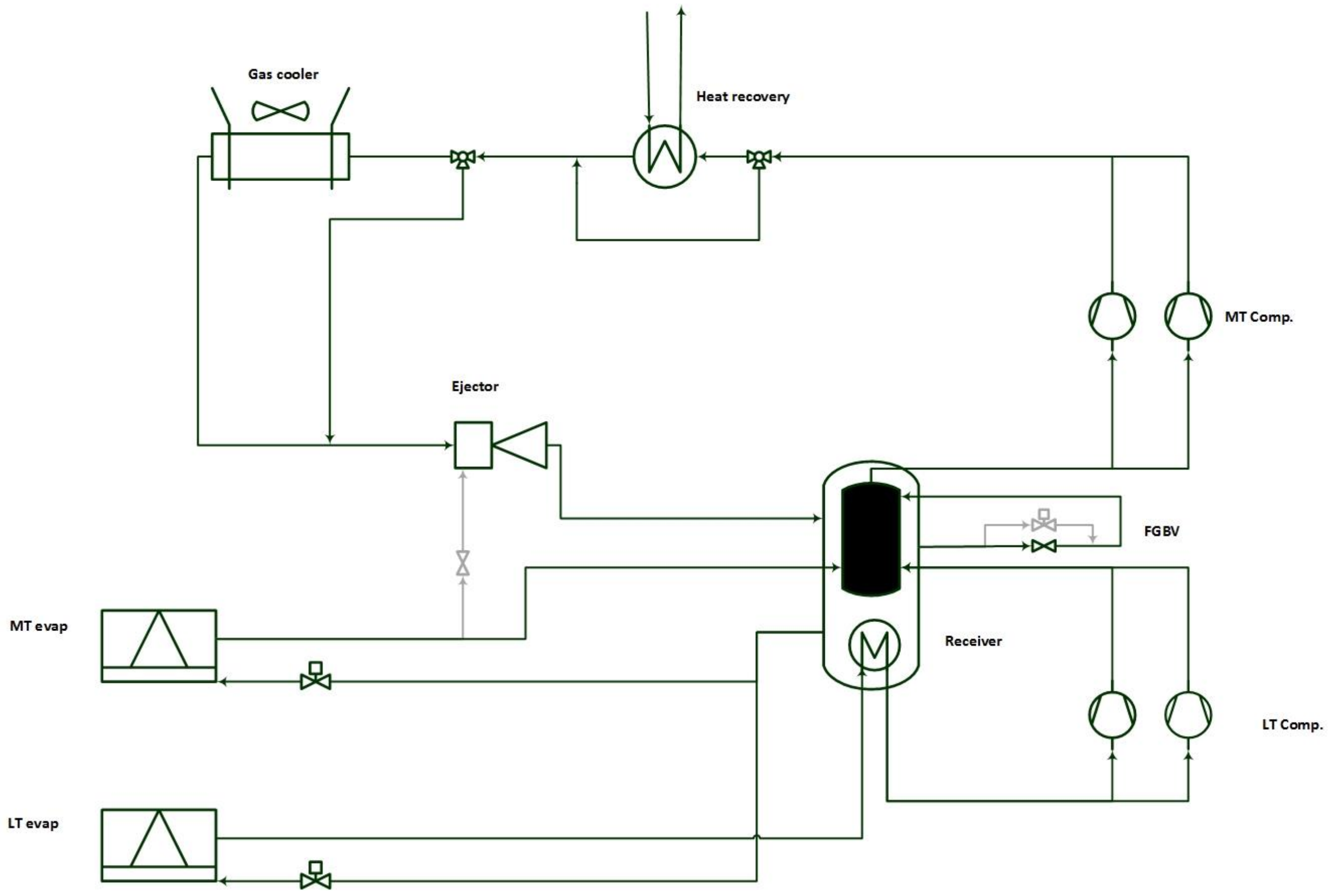

Figure 8-16: Spar Røyken operation mode one

8.4.2.2 *Operation mode two*

In operation mode two the flow from the LT-evaporators coincides with the flow in operation mode one. However, where these two operation modes differ, is in the MT-section. The flow exiting the MT-evaporators is now the suction fluid for the ejector, it is mixed with the flow from the gas cooler which is the motive fluid. The flow is then mixed, and travels through the diffuser part of the ejector. It is then fed to the outer shell of the tank. When the ejector function is active, the parallel valve to the flash gas bypass valve is also open as an addition to the FGBV. And the refrigerant passes through these two valves to the inner tank in the receiver. An outline of the system in operation mode two can be seen in Figure 8-17. The location of each component is marked, as well as the pipes that the refrigerant travels through.

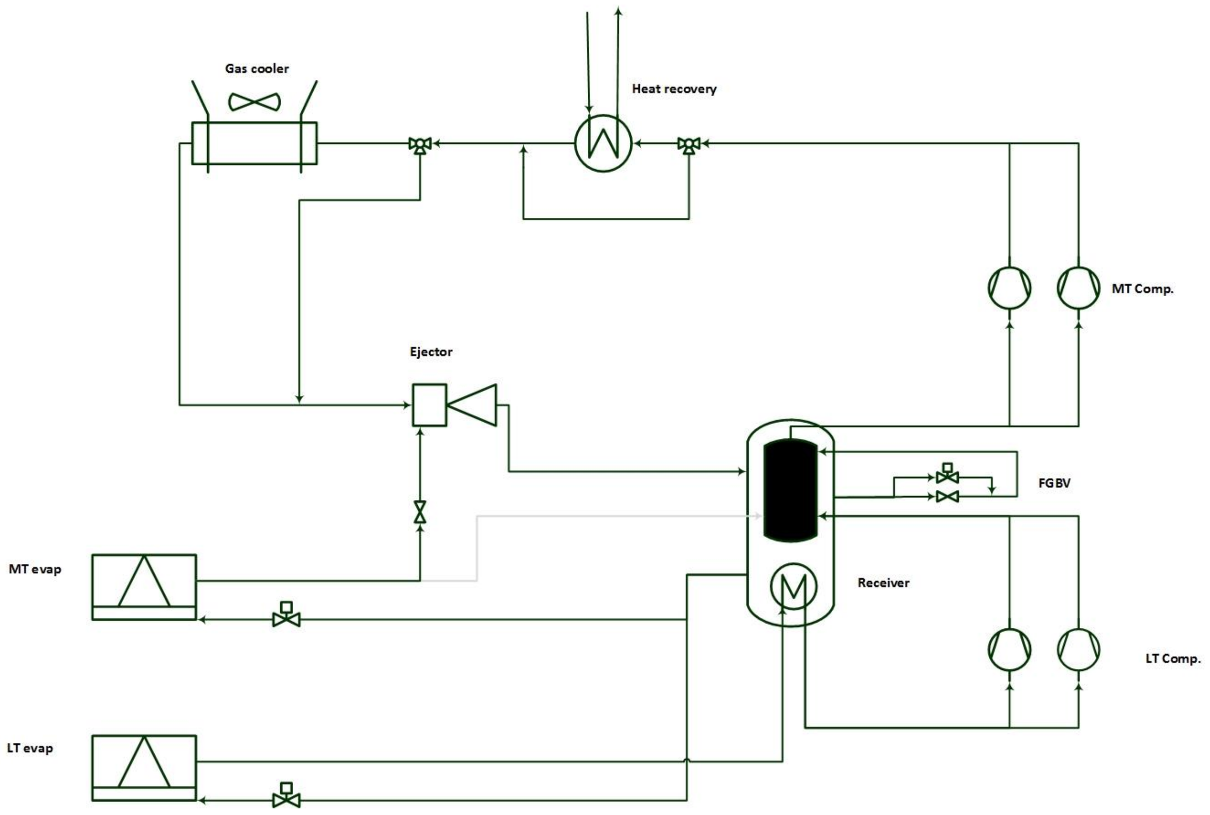

Figure 8-17: Spar Røyken operation mode two.

9 RESULTS

9.1 Spar Snarøya

The results at Spar Snarøya is divided in two sections. The first section is the chosen day where the parallel compressor runs the entire duration of the opening hours. The second section is a day that has similar average run capacity on MT-level, but no activation of the parallel compressor. This is done to be able to make a comparison for the system with and without parallel compressor operation.

9.1.1 FRIDAY 19.06.2020

Friday the 19th has one of the highest average temperatures in 2020 over a 24hour period, with the highest maximum temperature measured in 2020. The temperature curve during the day is displayed below (Norwegian Meteorological institute and the Norwegian Broadcasting Corporation, 2021). At this day, the parallel compressor turned on at 06:10 and stayed on until 23:08.

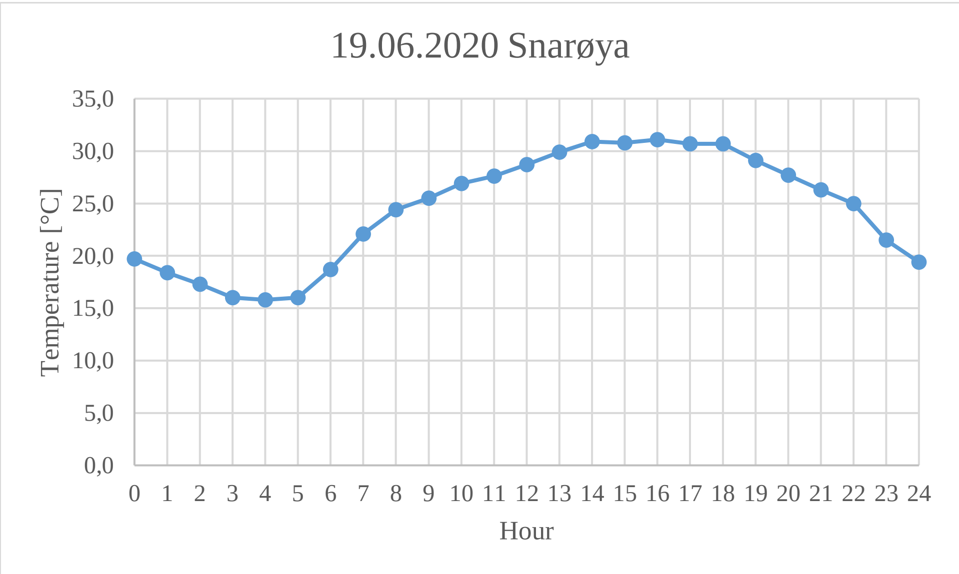

Figure 9-1: Temperature variation 19th June 2020 at Snarøya

9.1.1.1 Air condition

The graph below show the average opening degree of the AC-valve each hour during the 19th June 2020. With a maximum opening degree of 81.16 % and a minimum value of 62.94 %.

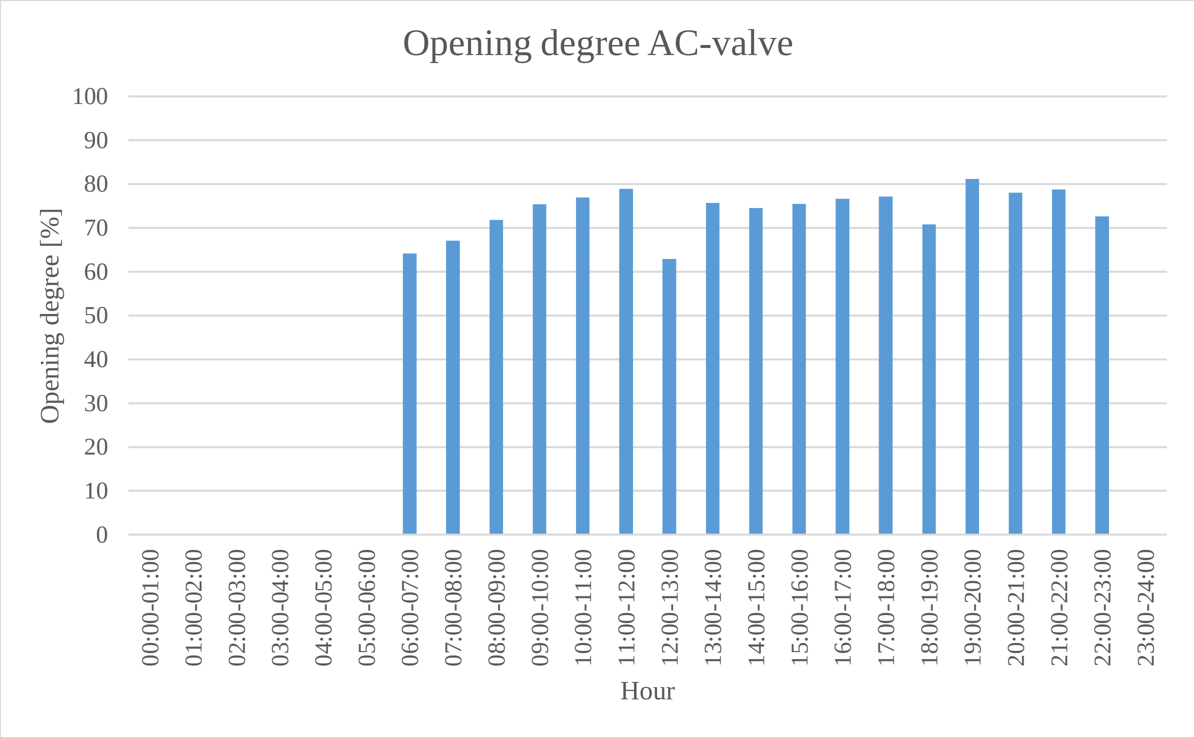

Figure 9-2: Opening degree AC-valve at Spar Snarøya

The average air condition demand was calculated on the basis of the values exported from IWMAC as explained in section 7.2.2.1. The highest value was 34.34 kW and the lowest value was 12.27 kW.

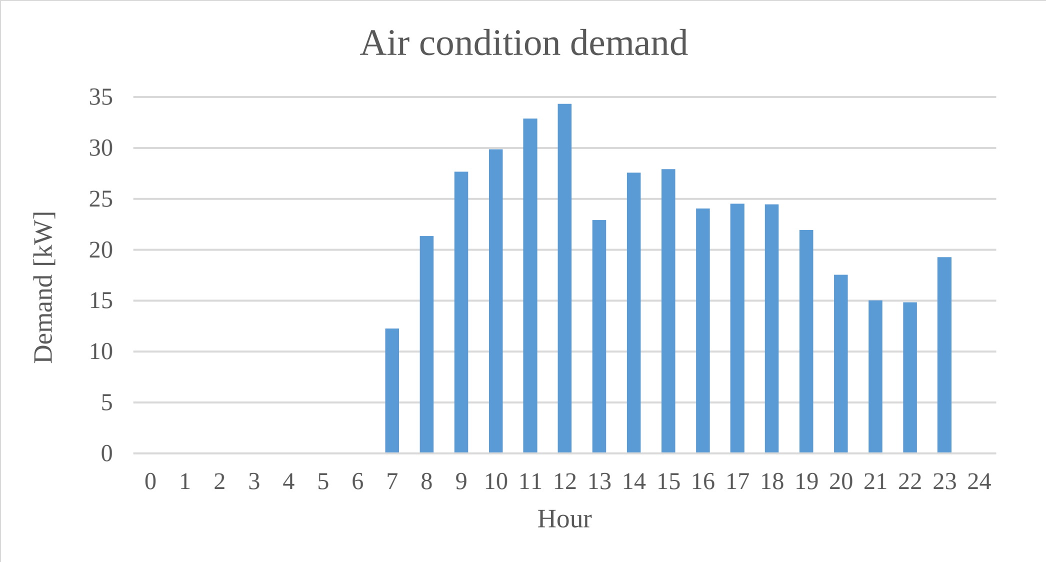

Figure 9-3: Air condition demand at Spar Snarøya

9.1.1.2 *Load LT and MT compressors*

In the graph in Figure 9-4, the running capacity for LT and MT compressors is displayed. These are average values each hours. For the LT compressors the run capacity ranges from 69 % to 99.5 % during the 24 hour period, and for MT compressors the range is 37 % to 51 %.

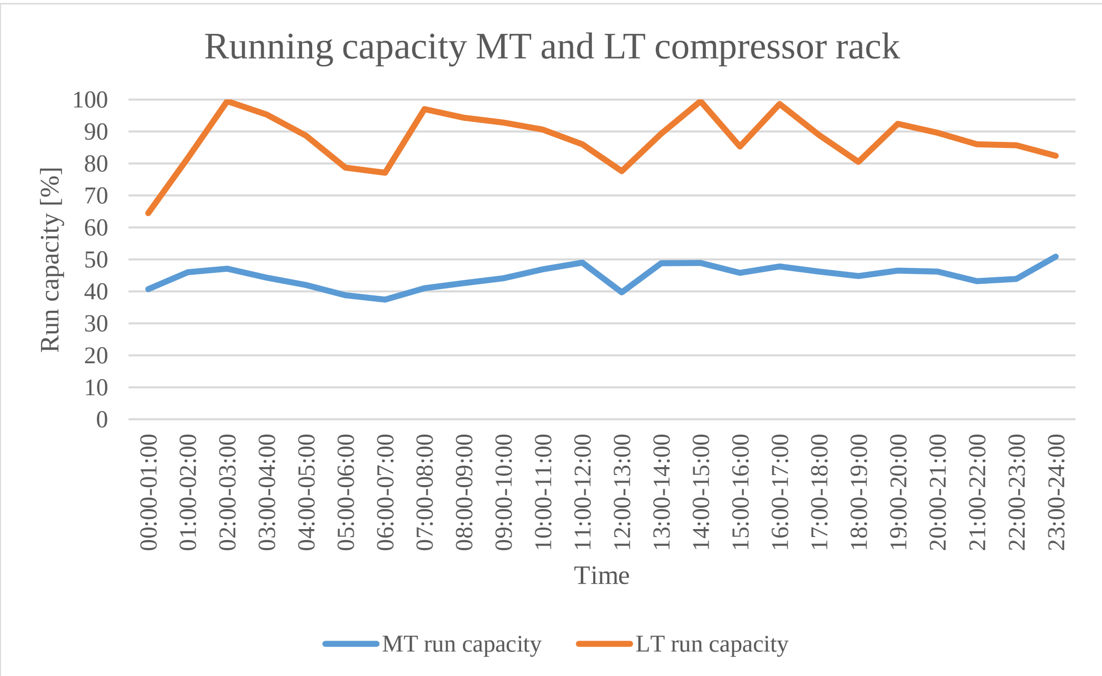

Figure 9-4: Running capacity LT and MT compressors

The refrigeration load is shown in Figure 9-5, and the development during the 24 hour period. For the LT compressors the refrigeration load ranges from 27 % to a maximum value of 41 %, for the MT compressors the corresponding values are 19 % as minimum and 50 % as maximum.

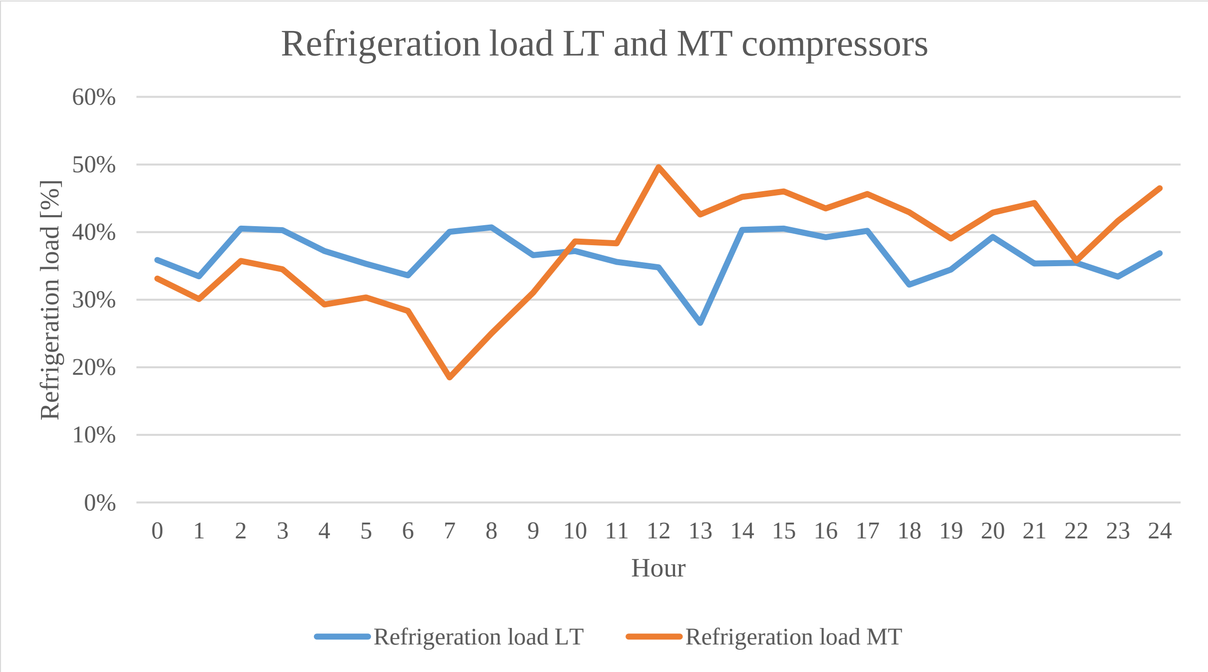

Figure 9-5: Refrigeration load LT and MT compressors

The load on the parallel compressor was calculated as stated in chapter 7.2.2.2. As shown in Figure 9-6, the parallel compressor is running from 06:00 - 23:00, and the corresponding load varies from 10.76 kW as the minimum value to 30.05 kW as the maximum value.

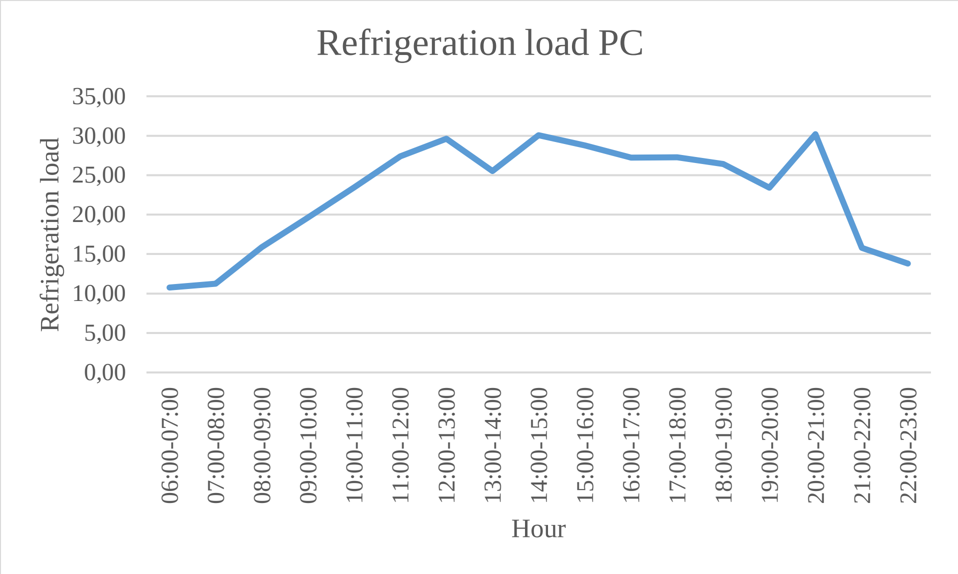

Figure 9-6: Refrigeration load PC at Spar Snarøya

9.1.1.3 Total refrigeration load

The total refrigeration load for the system is displayed below. The graph shows the load contributed by LT, MT and parallel compressors. The total load ranges from 17.6kW to 54.55kW as the maximum value.

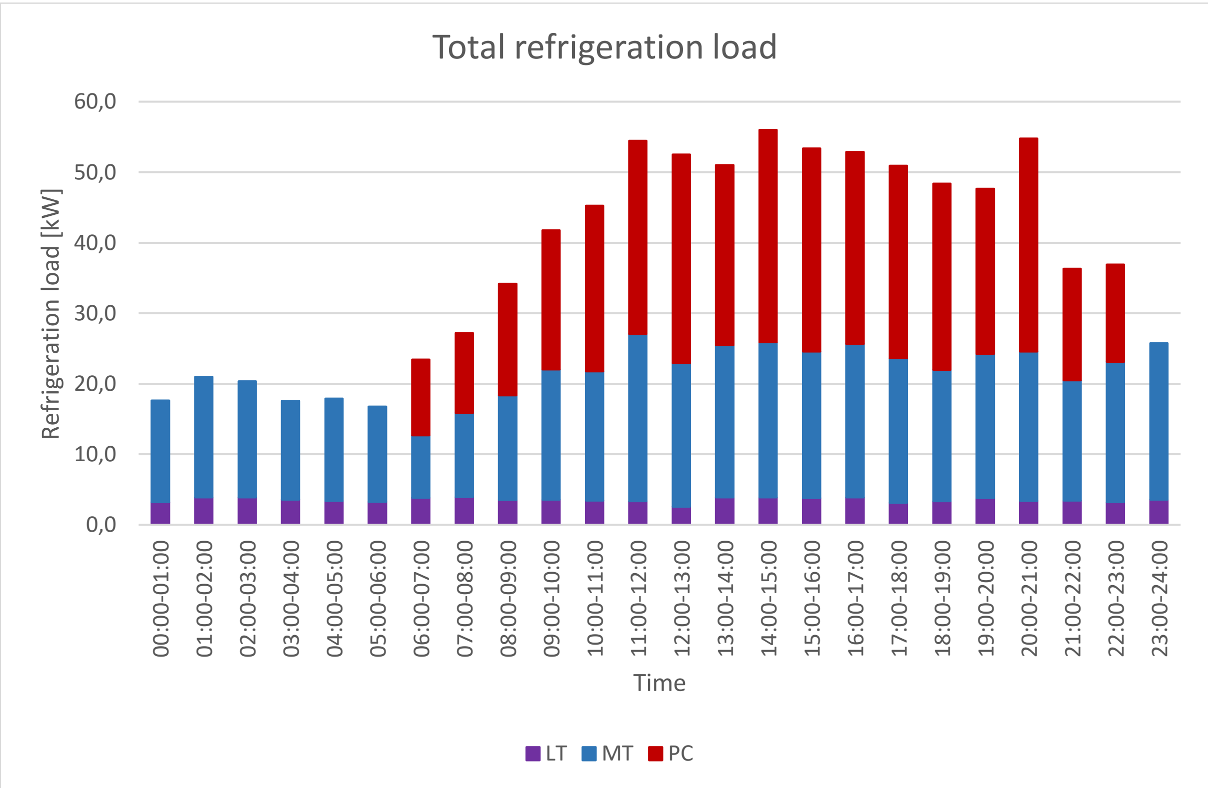

Figure 9-7: Total refrigeration load Spar Snarøya.

9.1.1.4 Bitzer software COP

As mentioned in subchapter 7.2.2.5, the three first time intervals with parallel compression during the day in question has a gas cooler pressure below the 73.77 bar. As a result of this the high pressure side was set to “auto” in Bitzer, which made it possible to conduct the simulation. In Figure 9-8 one can see the P-h diagram supplied by Bitzer, showing that it simulates the system with a higher pressure than it actually is.

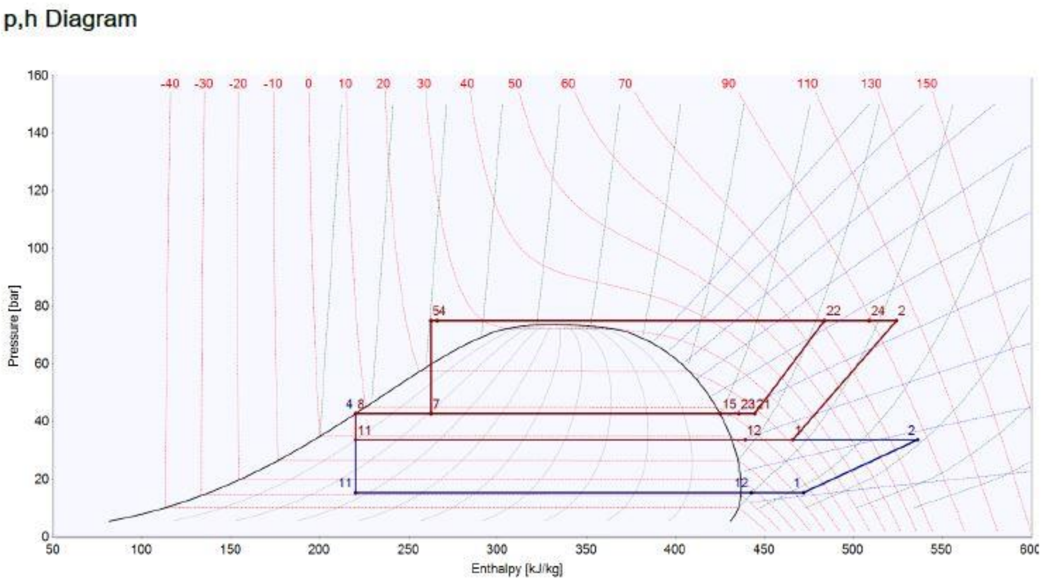

Figure 9-8: P-h diagram from Bitzer simulation

In Table 9-1 there is an overview of the gas cooler pressure that have been simulated transcritically.

Table 9-1: Average pressure gas cooler during subcritical operation

Hour	**Average pressure gas cooler [Bar]**
06:00-07:00	65.8
07:00-08:00	68.08
08:00-09:00	71.34

In Figure 9-9 there is graphic displaying the different COP-values from the Bitzer software with the hourly average values as input. The COP ranges from 2.34 to 3.13.

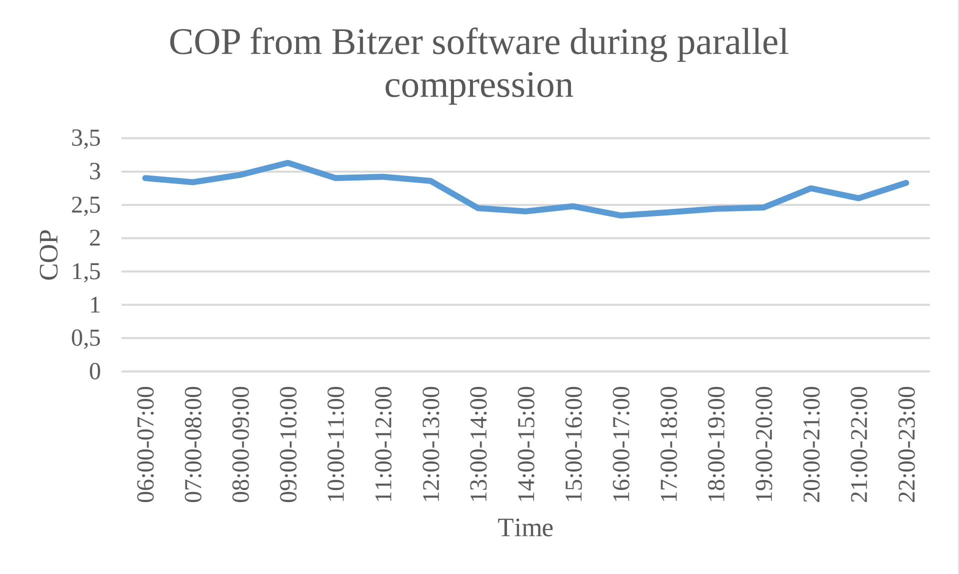

Figure 9-9: COP from Bitzer software during parallel compression

9.1.1.5 P-h diagram

Figure 9-10 displays the p-h diagram for Spar Snarøya with the average hourly values from 15:00-16:00. It includes the pressure levels in the gas cooler, receiver, MT evaporators and LT evaporators. The compressors are marked with suction and discharge values as well as isentropic enthalpy.

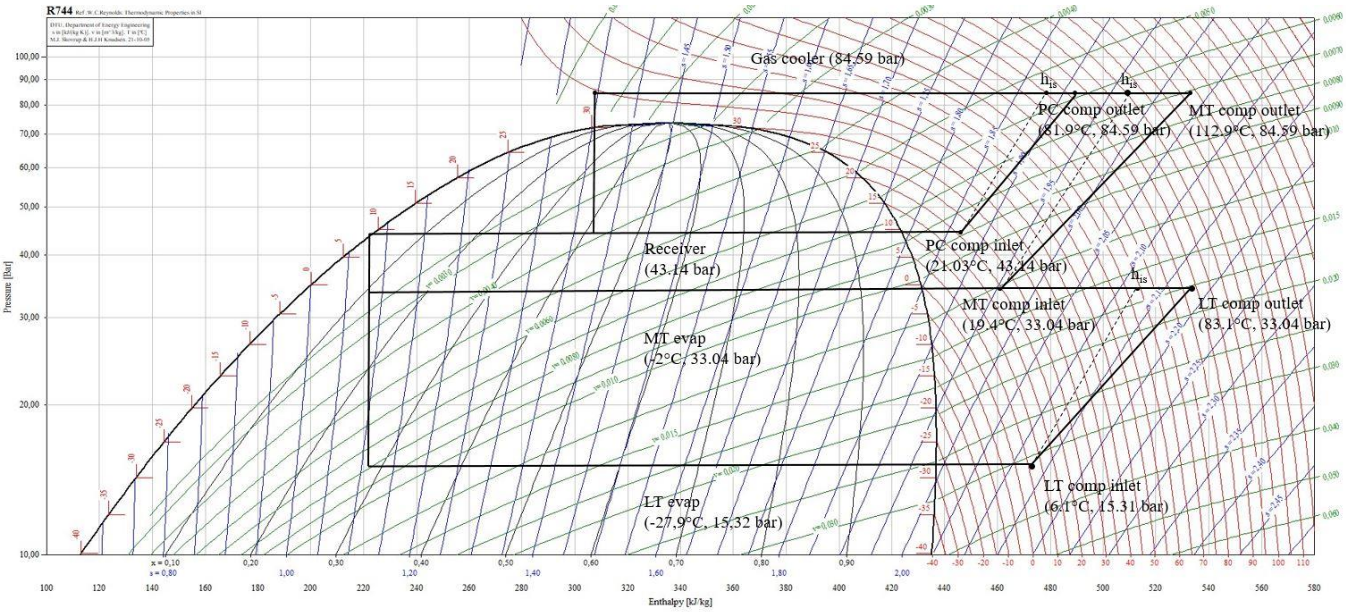

Figure 9-10: P-h diagram Spar Snarøya

9.1.2 FRIDAY 29.05.2020 COMPARED TO FRIDAY 19.06.2020

The day chosen to compare with the parallel compressor period have a similar hourly average run capacity for the MT compressor racks during the day, as can be seen in the figure 9-6.

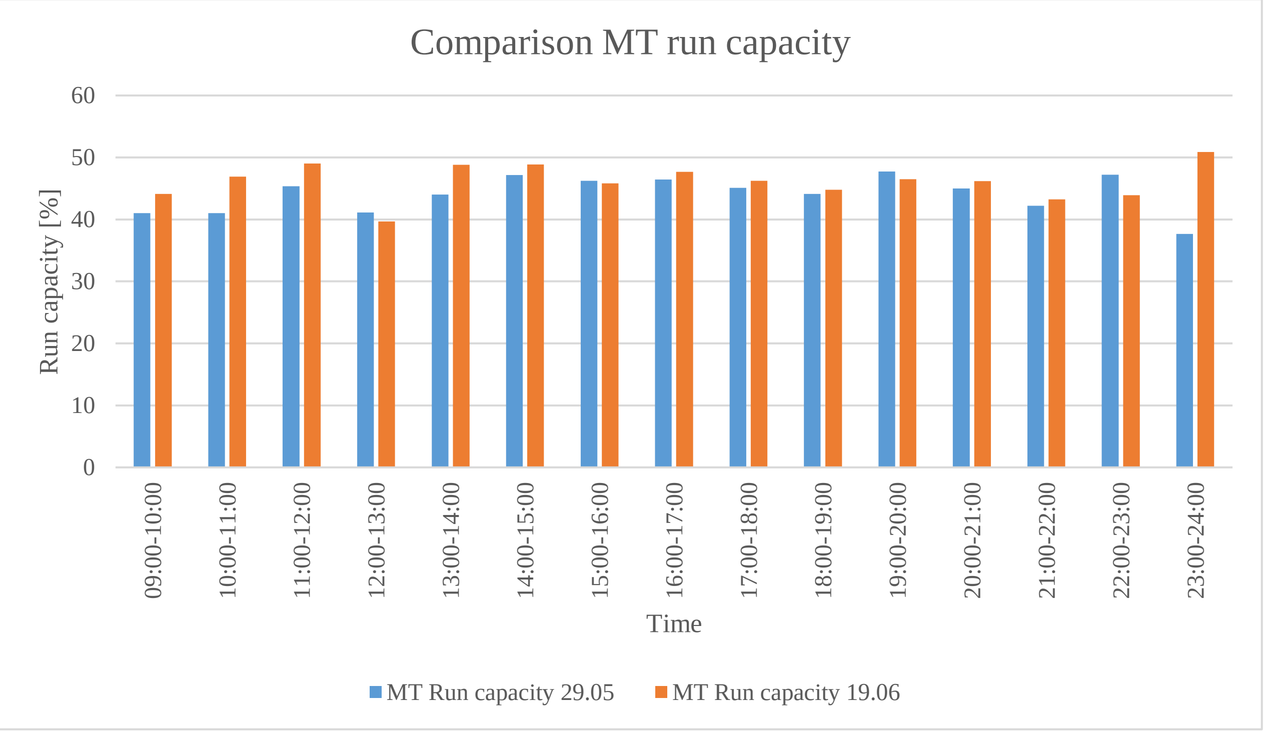

Figure 9-11: Comparison, MT run capacity Spar Snarøya

In the graphics the MT run capacity 29th May ranges from 37.67 % to 47.75 %, and the same variable at 19th June have its lowest value as 39.67 % and the highest value is 50.88 %. The time period 15:00-16:00 have a 0.45 % difference in the average hourly value. In figure 9-7 the different variables in the same time period is displayed.

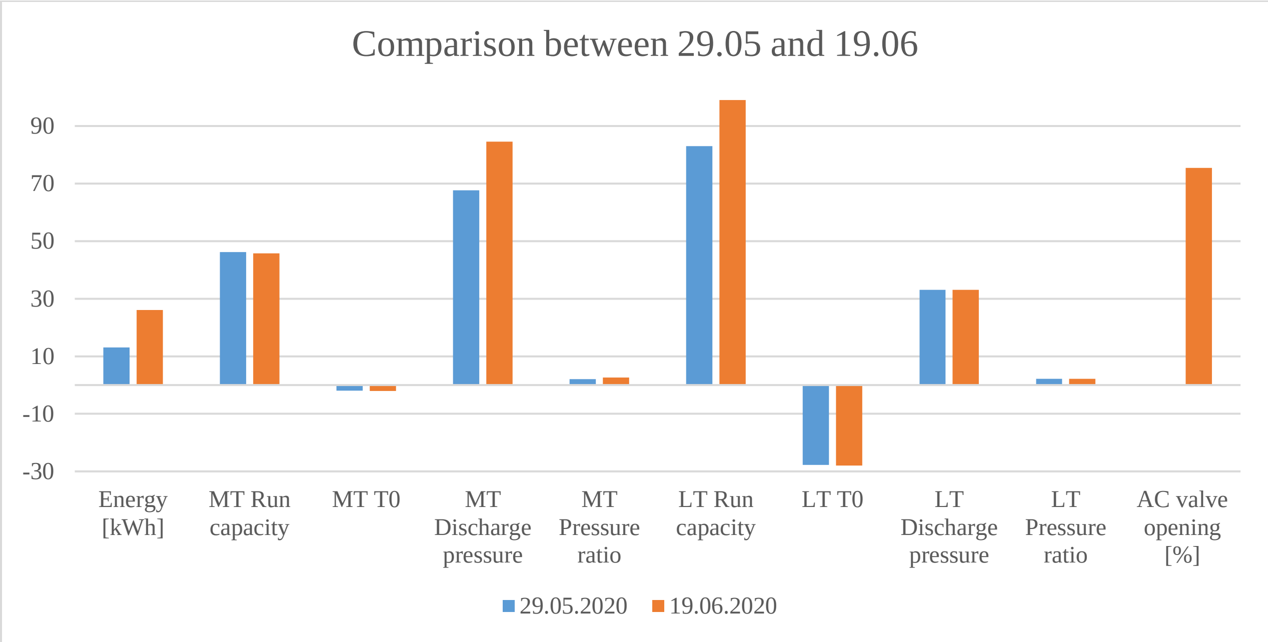

Figure 9-12: Comparison between variables 29th May 2020 and 19th June 2020 at Spar Snarøya

9.2 Spar Røyken

As mentioned in chapter 7, the first part of the results from Spar Røyken is split into two phases due to the changes that were made. The variables chosen for evaluation in this section is described in chapter 7.2.3.3.

9.2.1 PHASE ONE

In the first phase there were a few periods where the ejector was operating, the date 27.04.2021 were chosen because it was the longest continuous running period before any changes were made. At this day, the ejector operated from 10:25 to 10:56, a total of 31 minutes.

The outside temperature during this day can be seen in the graphics in Figure 9-13.

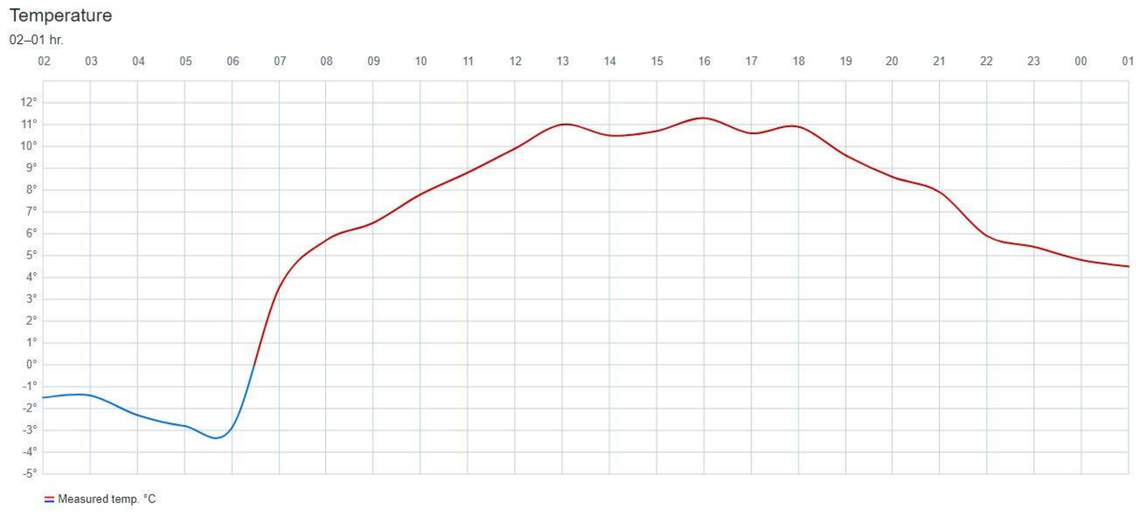

Figure 9-13: Outside temperature at Bygdøy observation station 27th April 2020 (yr.no, 2021).

9.2.1.1 Tuesday 27th April 2021 10:10 – 11:10

The first diagram displays the development of the chosen variables in a 60 minute period. When voltage input is shown as 1, the ejector is running, and 0 when the ejector is off. The ejector was on from 10:25-10:58 at this date.

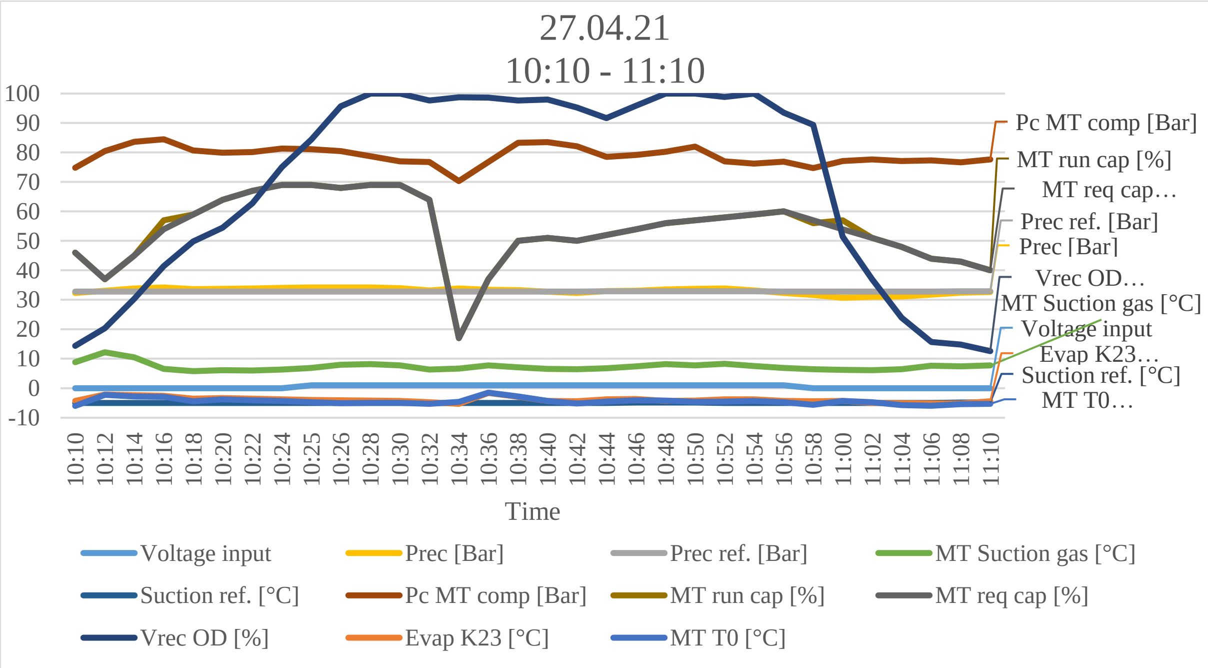

Figure 9-14: Diagram 27th April 10:10 - 11:10.

The following figures shows the upper and lower part of the graph displayed in figure 9-7, to enhance the development of the variables in the 60 minute period. The upper half shows the discharge pressure of the MT compressors, the run and request capacity of the MT compressor rack, the pressure in the receiver as well as the opening degree of the flash gas bypass valve.

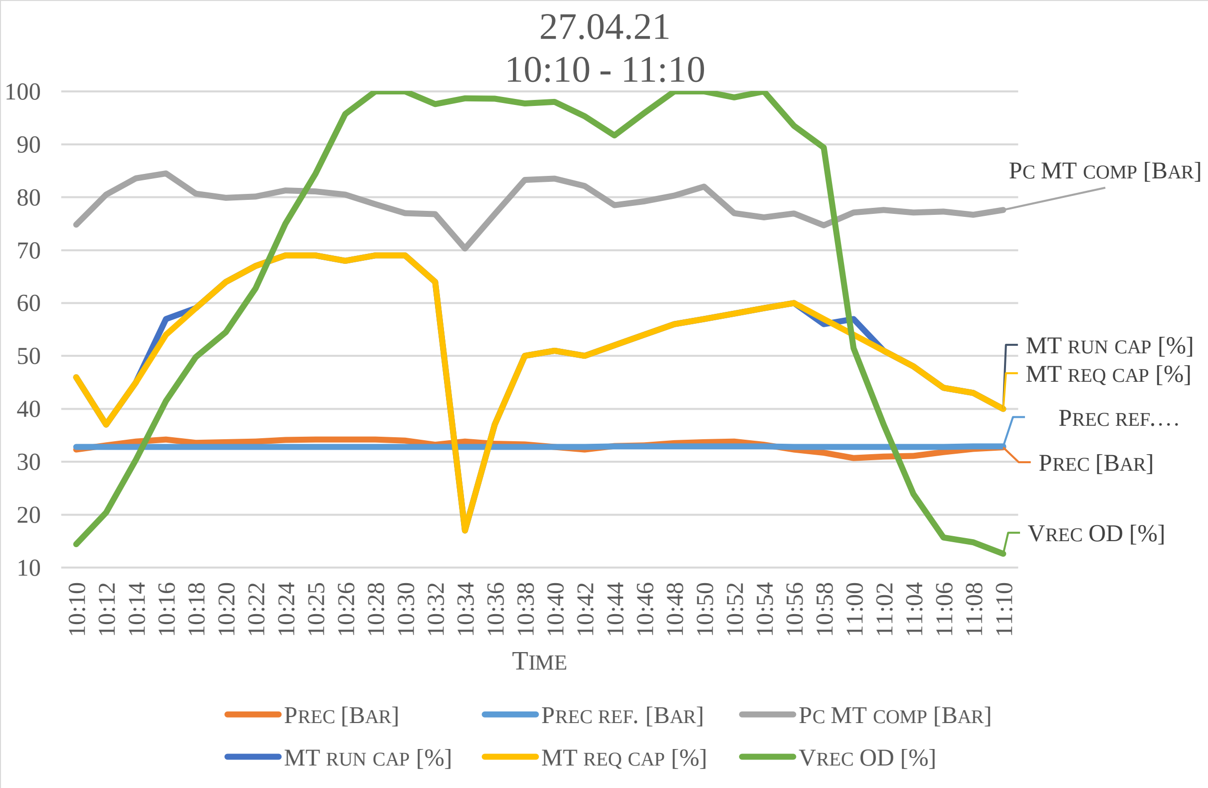

Figure 9-15: Diagram upper half 27th April 10:10 - 11:10.

For the lower part of the diagram, the suction gas temperature for MT compressors are displayed, ejector status 0/1, T0 for MT compressor rack, the suction reference for the MT section as well as the evaporation temperature for unit K23.

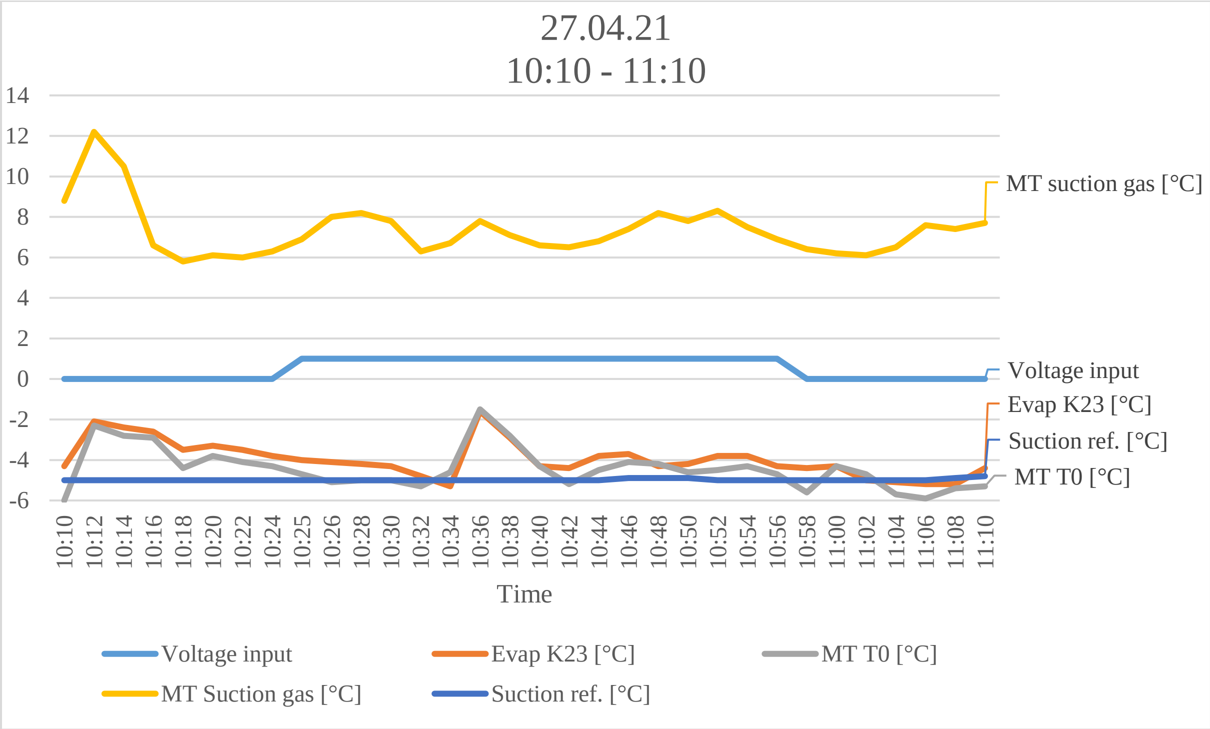

Figure 9-16: Diagram lower half 27th April 10:10 - 11:10.

As a result of the runtime of the ejector, a certain pressure and temperature lift was obtained, which is stated in table 9-1. The period with ejector operation is marked in green in the table below. The pressure lift ranges from 0.6 bar to 4.6 bar, with an average of 3.01 bar. The temperature lift has its lowest value as 1.3 °C and maximum value as 4.7 °C, with an average of 3.57 °C temperature lift.

Table 9-2: Pressure and temperature lift 27th April 10:10 - 11:10.

Time	Voltage input	Evap K23 [°C]	Trec [°C]	Pevap [Bar]	Prec [Bar]	Pressure lift [Bar]	Temperature lift [°C]
10:10	0	-4,3	-1,6	30,8	32,3	1,5	2,7
10:12	0	-2,1	-0,7	32,4	33,1	0,7	1,4
10:14	0	-2,4	0,1	32	33,8	1,8	2,5
10:16	0	-2,6	0,5	31,4	34,2	2,8	3,1
10:18	0	-3,5	-0,3	30,6	33,6	3	3,2
10:20	0	-3,3	-0,2	30,9	33,7	2,8	3,1
10:22	0	-3,5	0,1	30,9	33,8	2,9	3,6
10:24	0	-3,8	0,4	30,4	34,1	3,7	4,2
10:25	1	-4	0,5	30,3	34,2	3,9	4,5
10:26	1	-4,1	0,5	30,3	34,2	3,9	4,6
10:28	1	-4,2	0,5	30,2	34,2	4	4,7
10:30	1	-4,3	0,1	30,1	34	3,9	4,4
10:32	1	-4,8	-0,7	29,5	33,2	3,7	4,1
10:34	1	-5,3	-1,5	29,2	33,8	4,6	3,8
10:36	1	-1,6	-0,3	32,8	33,4	0,6	1,3
10:38	1	-2,9	-0,7	32,1	33,3	1,2	2,2
10:40	1	-4,3	-1	30,1	32,8	2,7	3,3
10:42	1	-4,4	-1,3	29,8	32,3	2,5	3,1
10:44	1	-3,8	-1	30,1	33	2,9	2,8
10:46	1	-3,7	-0,6	30,5	33,1	2,6	3,1
10:48	1	-4,3	-0,3	30,1	33,5	3,4	4
10:50	1	-4,2	0	30,2	33,7	3,5	4,2
10:52	1	-3,8	0,5	30,5	33,8	3,3	4,3
10:54	1	-3,8	-0,4	30,5	33,2	2,7	3,4
10:56	1	-4,3	-1,4	30,5	32,3	1,8	2,9
10:58	0	-4,4	-1,5	29,9	31,7	1,8	2,9
11:00	0	-4,3	-3	30,4	30,7	0,3	1,3
11:02	0	-5	-3	29,4	31	1,6	2
11:04	0	-5,1	-3	29,4	31,1	1,7	2,1
11:06	0	-5,2	-2,1	29,2	31,8	2,6	3,1

9.2.2 PHASE TWO

In phase two the ejector runs in three periods during Thursday 15th of June, all three of these periods are shown in their own subchapter below.

The development of the outside temperature during this day can be seen in Figure 9-17.

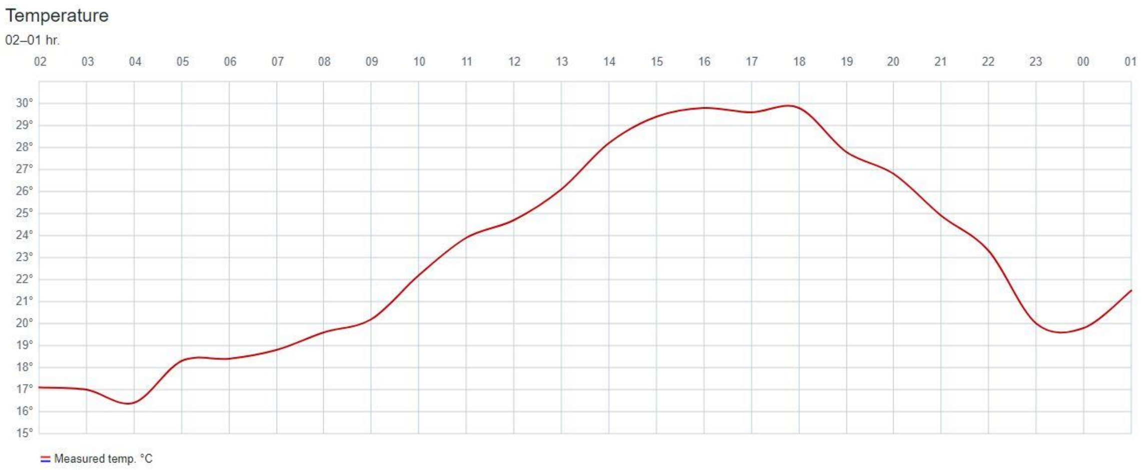

Figure 9-17: Temperature profile at Bygdøy observation station 15th July 2021 (yr.no, 2021).

9.2.2.1 1st period with ejector on, 15th June 2021 13:00 – 14:30.

In this period the ejector runs from 13:06 to 14:26, a total of 80 minutes. The 90 minute period shown in the graphics below includes the same variables as used in phase one, and as in the table in subchapter 7.2.3.3.

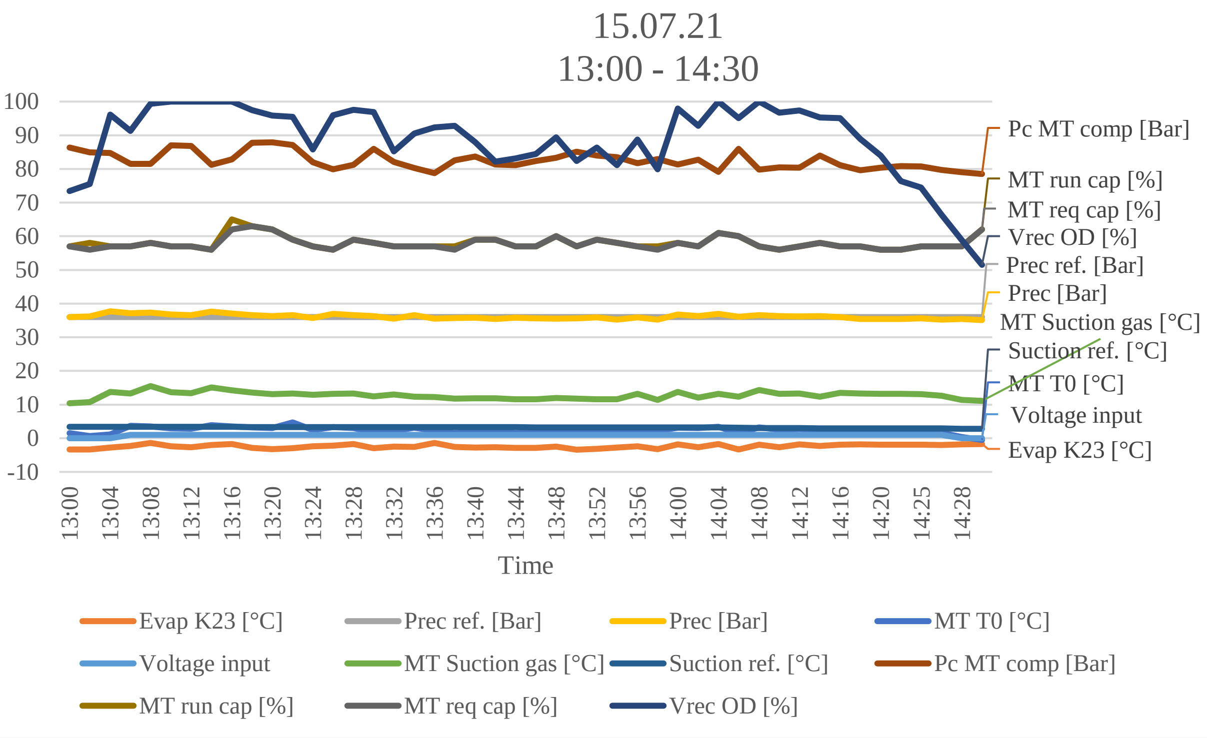

Figure 9-18: Diagram 15th June 13:00 - 14.30

The following charts show the upper and lower part of Figure 9-18, to get a better understanding of the different variables at each time stamp.

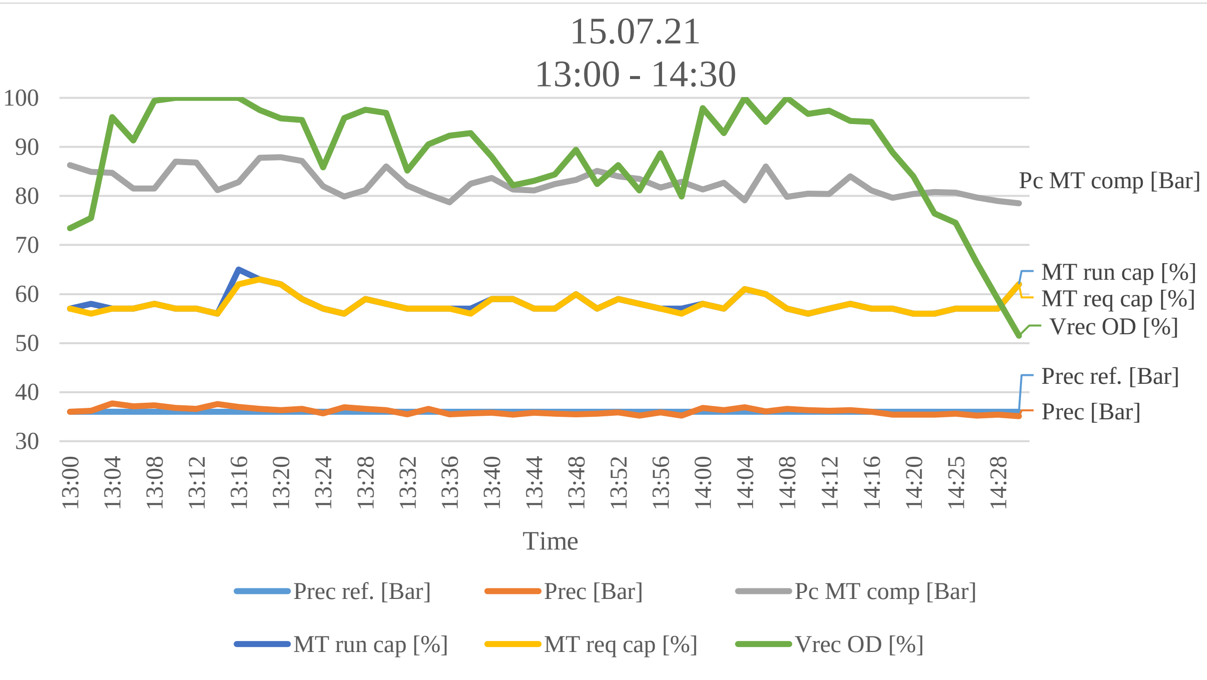

Figure 9-19: Diagram upper half 15th June 13:00 - 14.30

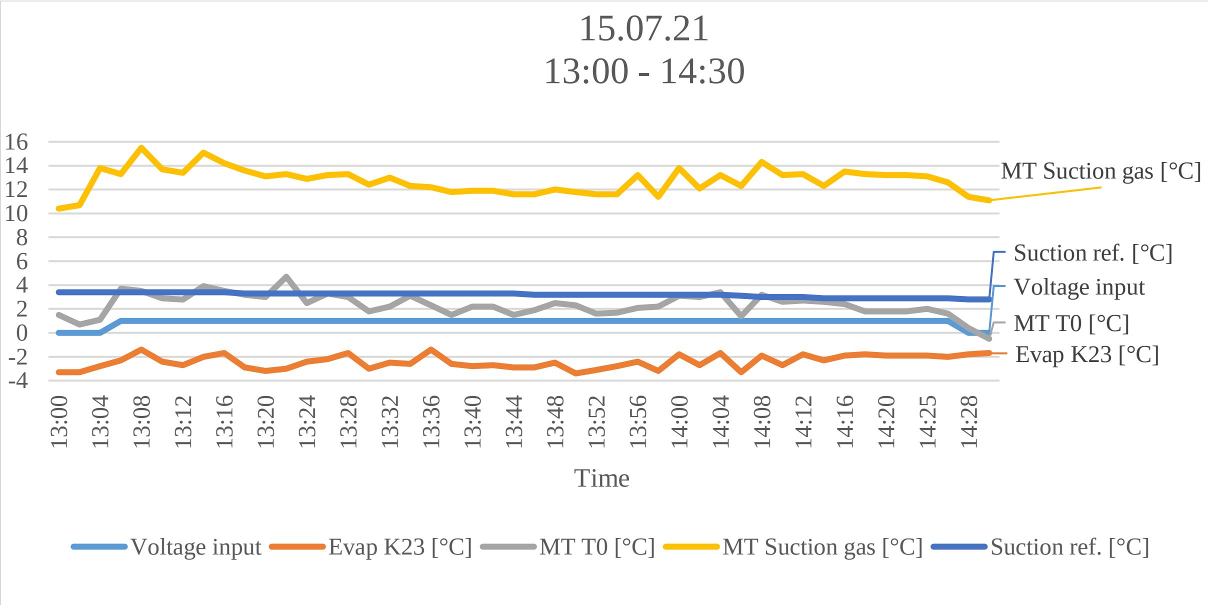

Figure 9-20: Diagram lower half 15th June 13:00 - 14.30

Table 9-3 shows the pressure lift and temperature lift during this 90 minute period, the part of the table marked in green shows the period where the ejector is in operation. The pressure lift ranges from 5.6 bar as the highest value to 3.1 bar as the lowest, with an average pressure lift of 4.41 bar. The temperature lift has its maximum value as 5.9 °C, and minimum value as 3.5 °C, with an average temperature lift of 4.91 °C.

Table 9-3: Pressure and temperature lift 15th June 13:00 - 14.30

Time	Voltage input	Evap K23 [°C]	Trec [°C]	Pevap [Bar]	Prec [Bar]	Pressure lift [Bar]	Temperature lift [°C]
13:00	0	-3,3	2,4	30,9	36	5,1	5,7
13:02	0	-3,3	2,4	30,9	36,2	5,3	5,7
13:04	0	-2,8	3,8	31,3	37,7	6,4	6,6
13:06	1	-2,3	3,3	31,9	37,1	5,2	5,6
13:08	1	-1,4	3,8	32,6	37,3	4,7	5,2
13:10	1	-2,4	3,1	31,7	36,8	5,1	5,5
13:12	1	-2,7	2,9	31,5	36,6	5,1	5,6
13:14	1	-2	3,8	31,1	37,6	6,5	5,8
13:16	1	-1,7	3,4	32,2	37	4,8	5,1
13:18	1	-2,9	2,9	32,3	36,6	4,3	5,8
13:20	1	-3,2	2,7	31,3	36,3	5	5,9
13:22	1	-3	2,9	31	36,6	5,6	5,9
13:24	1	-2,4	2,6	31,2	35,7	4,5	5
13:26	1	-2,2	3,2	31,7	36,9	5,2	5,4
13:28	1	-1,7	3,1	31,4	36,6	5,2	4,8
13:30	1	-3	2,6	32	36,3	4,3	5,6
13:32	1	-2,5	2,4	32,4	35,5	3,1	4,9
13:34	1	-2,6	2,8	31,2	36,6	5,4	5,4
13:36	1	-1,4	2,1	31,6	35,5	3,9	3,5
13:38	1	-2,6	2,1	31,6	35,7	4,1	4,7
13:40	1	-2,8	1,7	32,6	35,8	3,2	4,5
13:42	1	-2,7	1,9	31,3	35,4	4,1	4,6
13:44	1	-2,9	1,9	31,4	35,8	4,4	4,8
13:46	1	-2,9	2	32,4	35,6	3,2	4,9
13:48	1	-2,5	2,2	31,3	35,5	4,2	4,7
13:50	1	-3,4	2,1	31,6	35,6	4	5,5
13:52	1	-3,1	2,2	31,1	35,9	4,8	5,3
13:54	1	-2,8	1,8	31	35,2	4,2	4,6
13:56	1	-2,4	2,8	31,1	35,9	4,8	5,2
13:58	1	-3,2	1,8	31,4	35,2	3,8	5

14:00	1	-1,8	3,3	31,7	36,8	5,1	5,1
14:02	1	-2,7	2,3	31,1	36,3	5,2	5
14:04	1	-1,7	3,3	32,3	36,9	4,6	5
14:06	1	-3,3	2,4	31,5	36,1	4,6	5,7
14:08	1	-1,9	3,1	32,4	36,6	4,2	5
14:10	1	-2,7	2,7	31	36,3	5,3	5,4
14:12	1	-1,8	2,6	32,2	36,2	4	4,4
14:14	1	-2,3	2	31,4	36,3	4,9	4,3
14:16	1	-1,9	2,4	32,3	36	3,7	4,3
14:18	1	-1,8	1,8	31,8	35,4	3,6	3,6
14:20	1	-1,9	1,8	32,2	35,4	3,2	3,7
14:22	1	-1,9	1,8	32,3	35,4	3,1	3,7
14:25	1	-1,9	2	32,2	35,6	3,4	3,9
14:26	1	-2	1,5	32,1	35,2	3,1	3,5
14:28	0	-1,8	1,7	32,2	35,4	3,2	3,5
14:30	0	-1,7	1,4	32,3	35,1	2,8	3,1

9.2.2.2 2nd period with ejector on, 15th June 2021: 16:14 – 18:00.

The second period with ejector operation on 15th June lasts for 92 minutes. The values of the chosen variables is visualized in Figure 9-21.

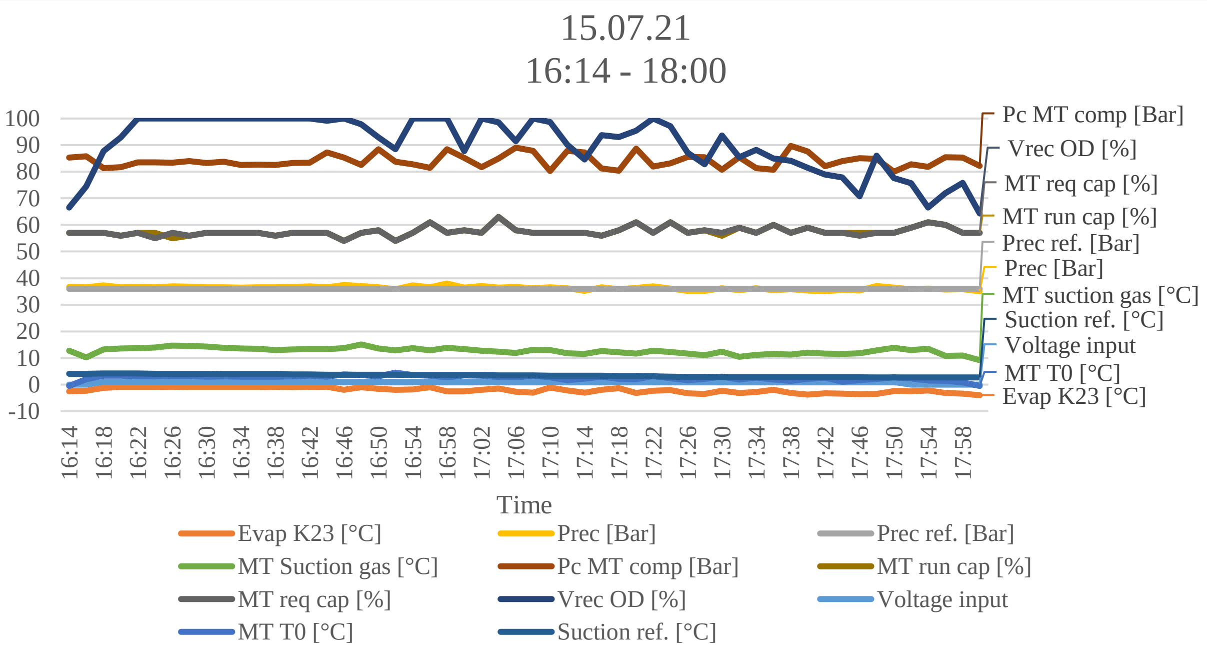

Figure 9-21: Diagram 15th June 16:14 - 18:00

Again, the graph from the given period is divided into an upper and lower half, to make sure that the development of the values are displayed properly.

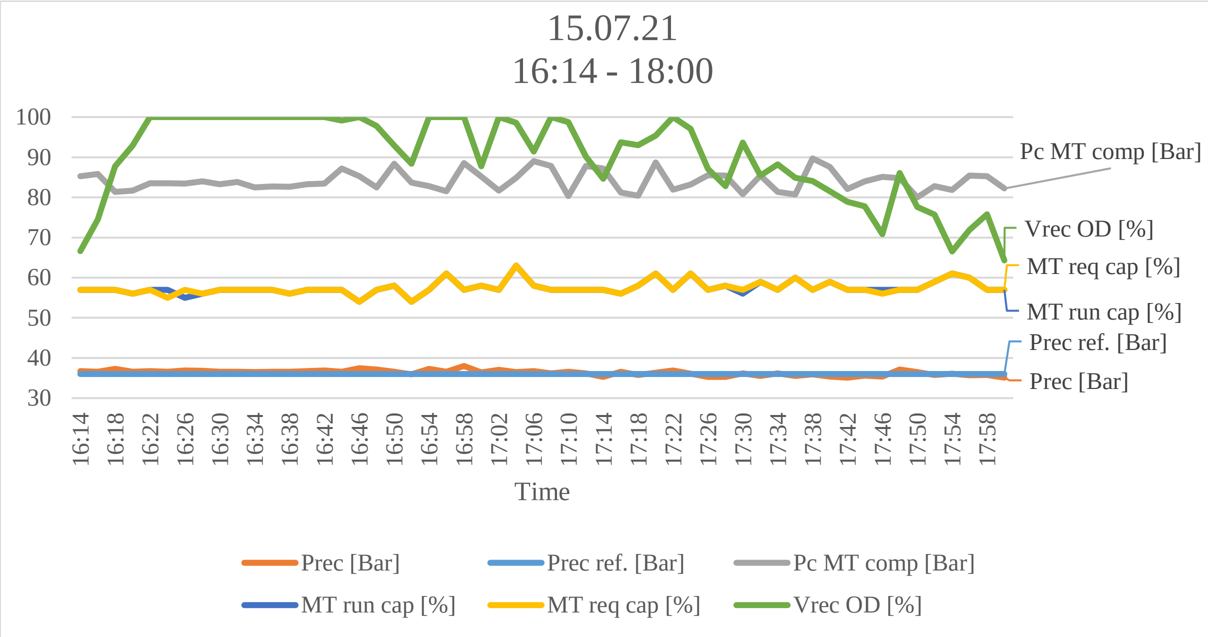

Figure 9-22: Diagram upper half 15th June 16:14 - 18:00

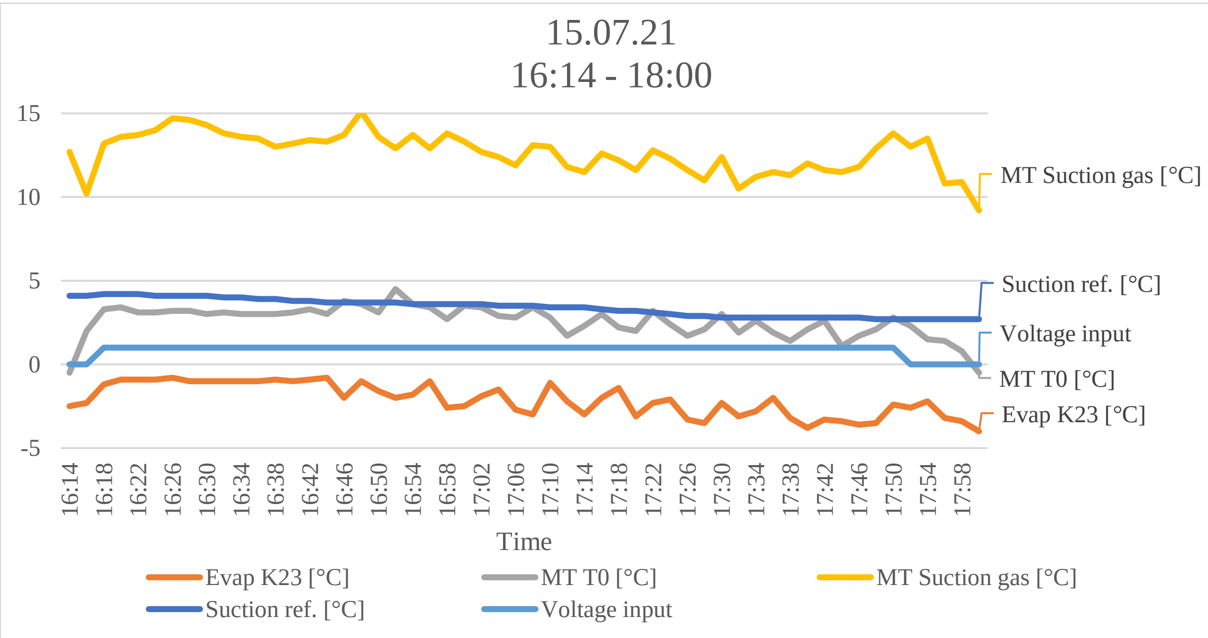

Figure 9-23: Diagram lower half 15th June 16:14 - 18:00

Table 9-3 displays the pressure and temperature lift in the second period of 15th June, the ejector run time is highlighted by the green color. The max pressure lift in this period is 6.4 bar and the minimum value is 3.5 bar, with an average pressure lift of 4.33 bar. The temperature lift ranges from the highest value of 6.6 °C to the lowest value of 3.8 °C, with an average of 4.78 °C.

Table 9-4: Pressure and temperature lift 15th June 16:14 - 18:00

Time	Voltage input	Evap K23 [°C]	Trec [°C]	Pevap [Bar]	Prec [Bar]	Pressure lift [Bar]	Temperature lift [°C]
16:14	0	-2,5	4,5	31,6	36,7	5,1	7
16:16	0	-2,3	2,8	31,8	36,6	4,8	5,1
16:18	1	-1,2	3,7	32,9	37,3	4,4	4,9
16:20	1	-0,9	3	33,1	36,6	3,5	3,9
16:22	1	-0,9	3,1	33	36,7	3,7	4
16:24	1	-0,9	3	33,1	36,6	3,5	3,9
16:26	1	-0,8	3,1	33,2	36,9	3,7	3,9
16:28	1	-1	3,3	33	36,8	3,8	4,3
16:30	1	-1	3	32,9	36,6	3,7	4
16:32	1	-1	3	33	36,6	3,6	4
16:34	1	-1	2,9	33	36,5	3,5	3,9
16:36	1	-1	3	33	36,6	3,6	4
16:38	1	-0,9	3	33	36,6	3,6	3,9
16:40	1	-1	3	33	36,7	3,7	4
16:42	1	-0,9	3	33,1	36,9	3,8	3,9
16:44	1	-0,8	3	32,4	36,6	4,2	3,8
16:46	1	-2	3,6	32,1	37,4	5,3	5,6
16:48	1	-1	2,9	33	37,1	4,1	3,9
16:50	1	-1,6	3	31,6	36,6	5	4,6
16:52	1	-2	2,6	32,1	35,9	3,8	4,6
16:54	1	-1,8	3,6	32,3	37,3	5	5,4
16:56	1	-1	2,8	33	36,6	3,6	3,8
16:58	1	-2,6	3,9	31,6	38	6,4	6,5
17:00	1	-2,5	2,8	31,6	36,4	4,8	5,3
17:02	1	-1,9	3,4	32,3	37	4,7	5,3
17:04	1	-1,5	2,9	32,5	36,5	4	4,4
17:06	1	-2,7	3	31,4	36,7	5,3	5,7
17:08	1	-3	3,3	31,2	36,2	5	6,3
17:10	1	-1,1	2,9	32,9	36,6	3,7	4
17:12	1	-2,2	2,5	31,9	36,2	4,3	4,7

17:14	1	-3	2	31,5	35,3	3,8	5
17:16	1	-2	3	32,1	36,6	4,5	5
17:18	1	-1,4	2,3	32,7	35,8	3,1	3,7
17:20	1	-3,1	2,6	31,1	36,3	5,2	5,7
17:22	1	-2,3	3,2	31,9	36,9	5	5,5
17:24	1	-2,1	2,5	32	36,1	4,1	4,6
17:26	1	-3,3	2,2	30,9	35,3	4,4	5,5
17:28	1	-3,5	1,8	30,7	35,3	4,6	5,3
17:30	1	-2,3	2,9	31,9	36,2	4,3	5,2
17:32	1	-3,1	1,7	31	35,5	4,5	4,8
17:34	1	-2,8	2,5	31,4	36,2	4,8	5,3
17:36	1	-2	2	32,1	35,5	3,4	4
17:38	1	-3,2	2,2	31,2	35,9	4,7	5,4
17:40	1	-3,8	2	30,5	35,4	4,9	5,8
17:42	1	-3,3	1,7	30,9	35,1	4,2	5
17:44	1	-3,4	1,8	31	35,6	4,6	5,2
17:46	1	-3,6	1,8	30,6	35,4	4,8	5,4
17:48	1	-3,5	3,1	30,7	37,1	6,4	6,6
17:50	1	-2,4	2,8	31,8	36,5	4,7	5,2
17:52	0	-2,6	2,2	31,4	35,8	4,4	4,8
17:54	0	-2,2	2,6	31,9	36,1	4,2	4,8
17:56	0	-3,2	2	31	35,7	4,7	5,2
17:58	0	-3,4	2,2	30,8	35,8	5	5,6
18:00	0	-4	1,5	30,5	35,1	4,6	5,5

9.2.2.3 3rd period with ejector on, 15th June 2021: 18:54 – 20:00.

The third and final period with ejector operation at 15th June happens from 19:00 to 19:50, lasting a total of 50 minutes. As the previous subchapters, the variables are displayed in Figure 9-24. First all of the chosen variable and the development of the values during the 3rd period is shown.

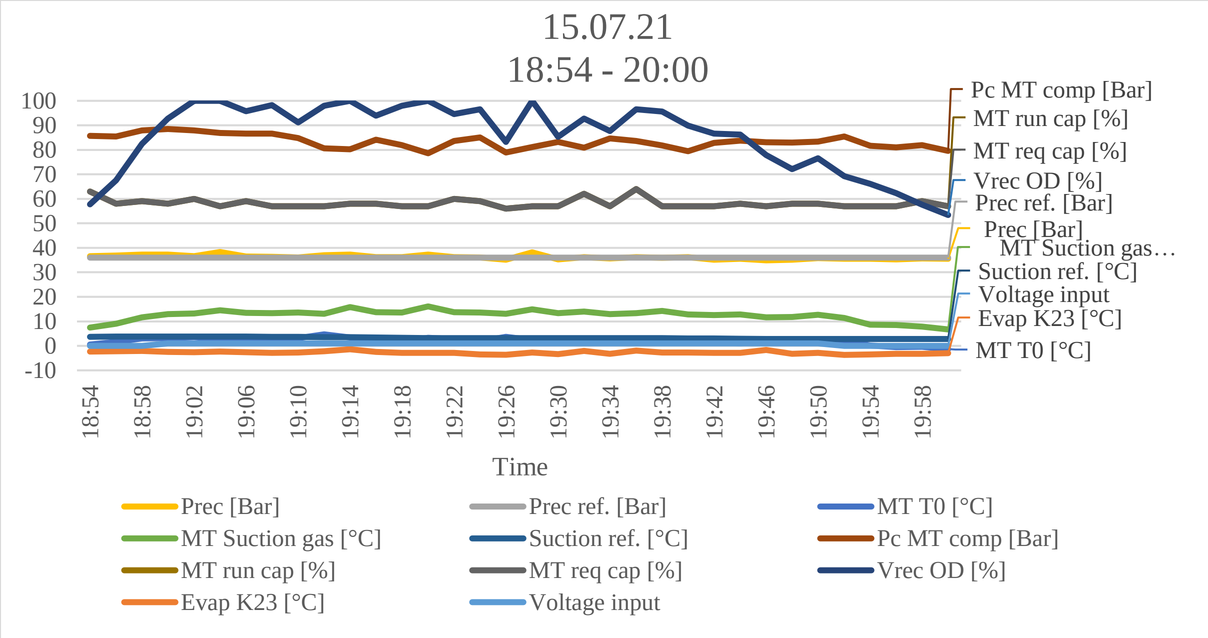

Figure 9-24: Diagram 15th June 18:54 - 20:00

Below the figure 9-17 is split into an upper and lower part to enhance the development of the variables with a different value range on the y-scale.

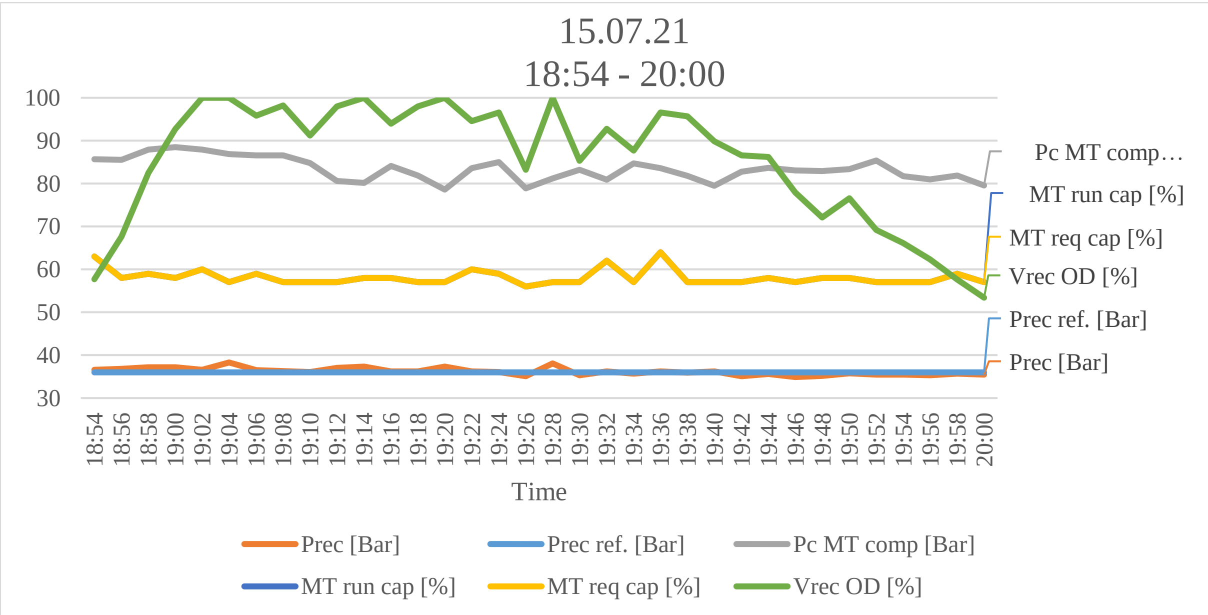

Figure 9-25: Diagram upper half 15th June 18:54 - 20:00

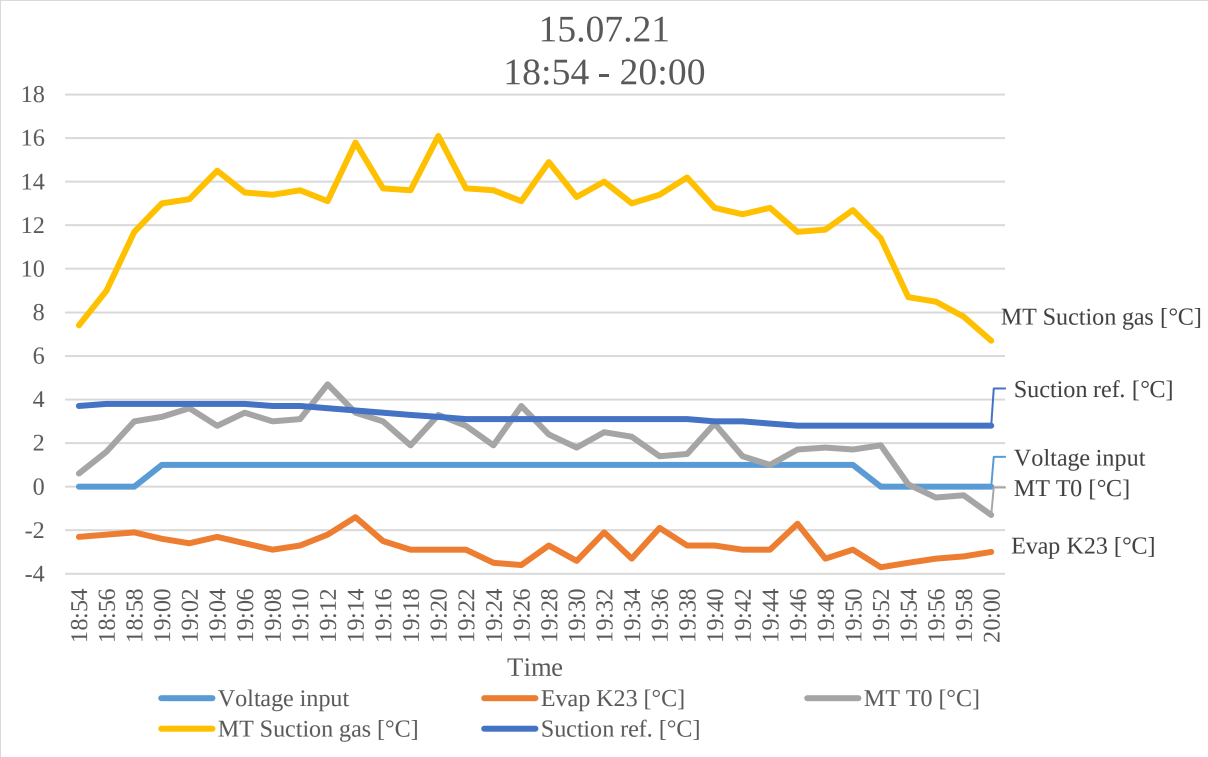

Figure 9-26: Diagram lower half 15th June 18:54 - 20:00

In table 9-4 the pressure lift and temperature lift during the 3rd period is displayed. The pressure lift range from the lowest value of 2.6 bar to 6.7 bar with a mean value of 4.77 bar. The temperature lift has it minimum at 4.6 °C and maximum value as 6.5 °C. The average temperature lift is 5.36 °C.

Table 9-5: Pressure and temperature lift 15th June 18:54 - 20:00.

Time	Voltage input	Evap K23 [°C]	Trec [°C]	Pevap [Bar]	Prec [Bar]	Pressure lift [Bar]	Temperature lift [°C]
18:58	0	-2,1	3,6	32	37,2	5,2	5,7
19:00	1	-2,4	3,4	31,7	37,2	5,5	5,8
19:02	1	-2,6	3,2	31,6	36,6	5	5,8
19:04	1	-2,3	4,2	31,9	38,3	6,4	6,5
19:06	1	-2,6	3,1	31,5	36,5	5	5,7
19:08	1	-2,9	2,9	31,3	36,3	5	5,8
19:10	1	-2,7	2,8	31,5	36,1	4,6	5,5
19:12	1	-2,2	3,2	31,9	37	5,1	5,4
19:14	1	-1,4	3,8	32,6	37,3	4,7	5,2
19:16	1	-2,5	2,7	31,6	36,2	4,6	5,2
19:18	1	-2,9	2,5	31,5	36,2	4,7	5,4
19:20	1	-2,9	3,6	31,3	37,3	6	6,5
19:22	1	-2,9	2,5	31,3	36,2	4,9	5,4
19:24	1	-3,5	2,4	30,8	36,1	5,3	5,9
19:26	1	-3,6	1,9	30,7	35,1	4,4	5,5
19:28	1	-2,7	3,5	31,4	38,1	6,7	6,2
19:30	1	-3,4	1,9	30,8	35,3	4,5	5,3
19:32	1	-2,1	2,5	32	36,2	4,2	4,6
19:34	1	-3,3	2,2	30,9	35,7	4,8	5,5
19:36	1	-1,9	2,5	32,2	36,2	4	4,4
19:38	1	-2,7	2,8	31,5	35,9	4,4	5,5
19:40	1	-2,7	2,5	31,5	36,2	4,7	5,2
19:42	1	-2,9	1,7	31,3	35,1	3,8	4,6
19:44	1	-2,9	2	31,3	35,6	4,3	4,9
19:46	1	-1,7	1,5	32,3	34,9	2,6	3,2
19:48	1	-3,3	1,7	30,9	35,2	4,3	5
19:50	1	-2,9	2,3	31,3	35,8	4,5	5,2
19:52	0	-3,7	1,8	30,6	35,5	4,9	5,5
19:54	0	-3,5	1,9	30,8	35,5	4,7	5,4
19:56	0	-3,3	1,9	30,9	35,3	4,4	5,2

9.2.2.4 *Comparison between periods with the ejector on and off, 15th June 2021*

Figure 9-27 and Figure 9-28 show the refrigeration load both in kW and in percentage of total load. The poles marked with green are periods where the ejector is running, and the orange poles denotes when the ejector is only operation as a high pressure valve (operation mode one).

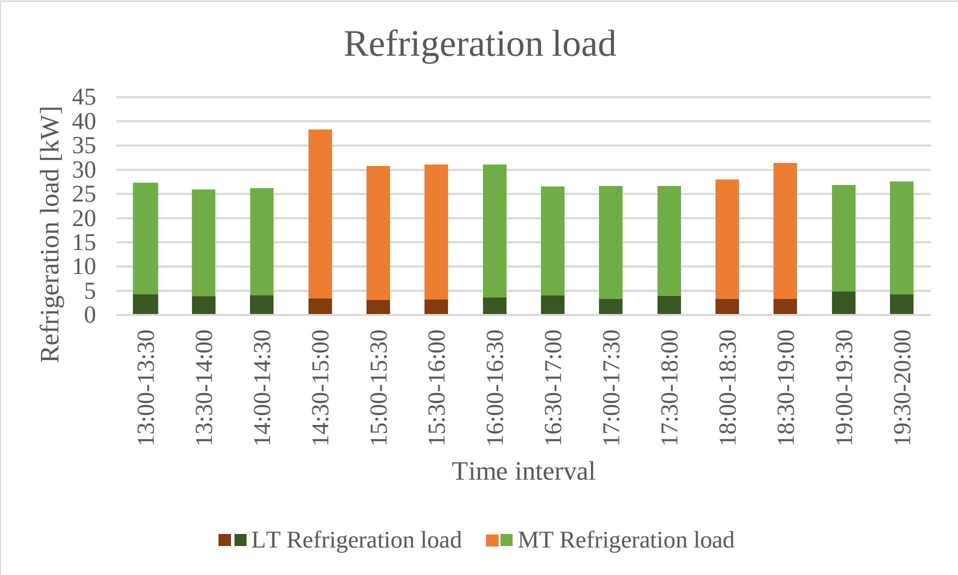

Figure 9-27: Refrigeration load [kW] 15th June 2021.

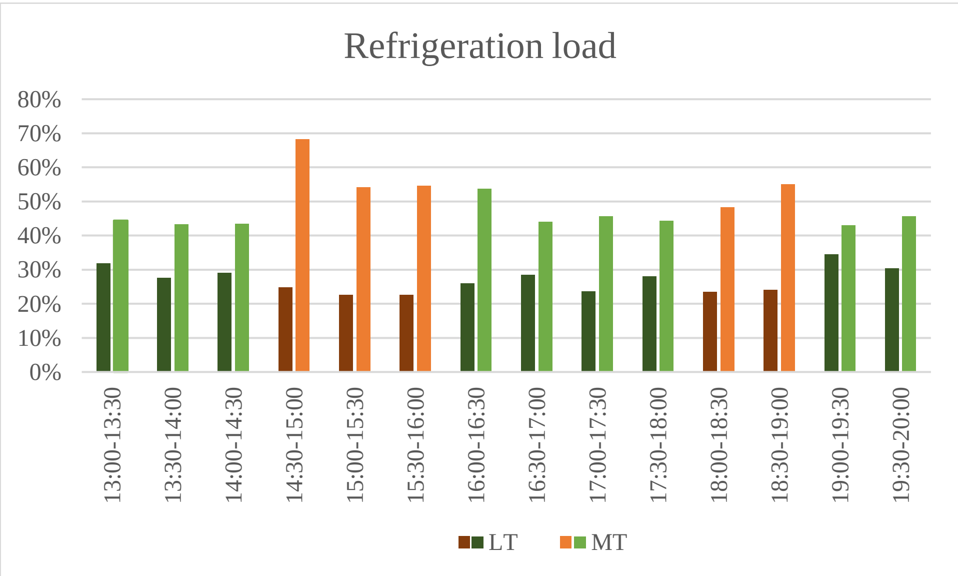

Figure 9-28: Refrigeration load 15th June 2021.

The average run capacity for the MT compressor rack is displayed in the graph below, the ejector runtime periods are marked in green, and the off periods are marked in orange. The energy demand for each period is also shown, to give an impression of the trend when the ejector is running and when it is off.

As stated in subsection 7.2.3.2, the goal was to compare the system in the two operation mode with similar run capacities on the MT compressors. As one can note from Figure 9-29 that the MT run capacity ranges from 57.3 % to 59 % when the ejector is in operation, and from 55 % to 70.0% when the ejector is operation solely as a high pressure valve.

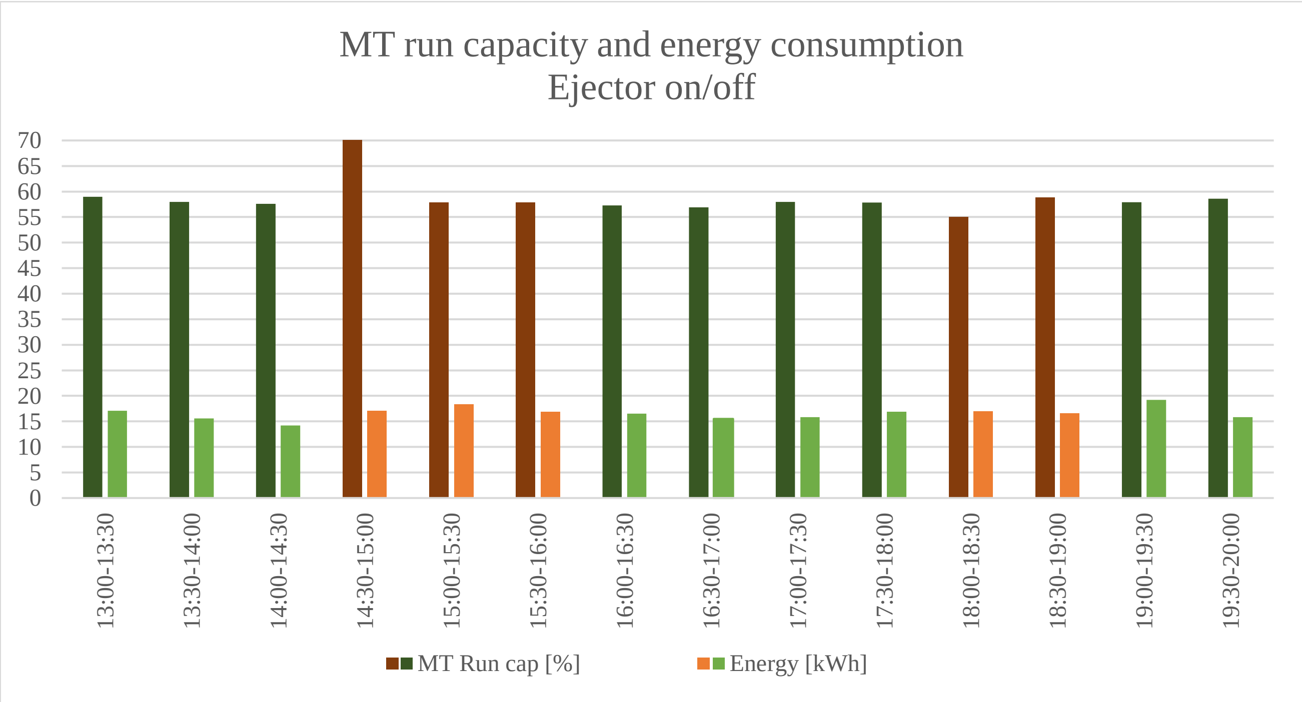

Figure 9-29: Average MT Run capacity 15th June 2021.

The periods 13:30-14:00 (marked green for ejector on) and 15:00-15:30 (marked orange for ejector off) where chosen from Figure 9-29, as they have similar MT run capacity (i.e. 58 % for the first period where the ejector is on and 57.89 % in the last period where the ejector operates as a high pressure valve). The variables are displayed in Figure 9-30 to show the difference in working conditions for the compressor racks.

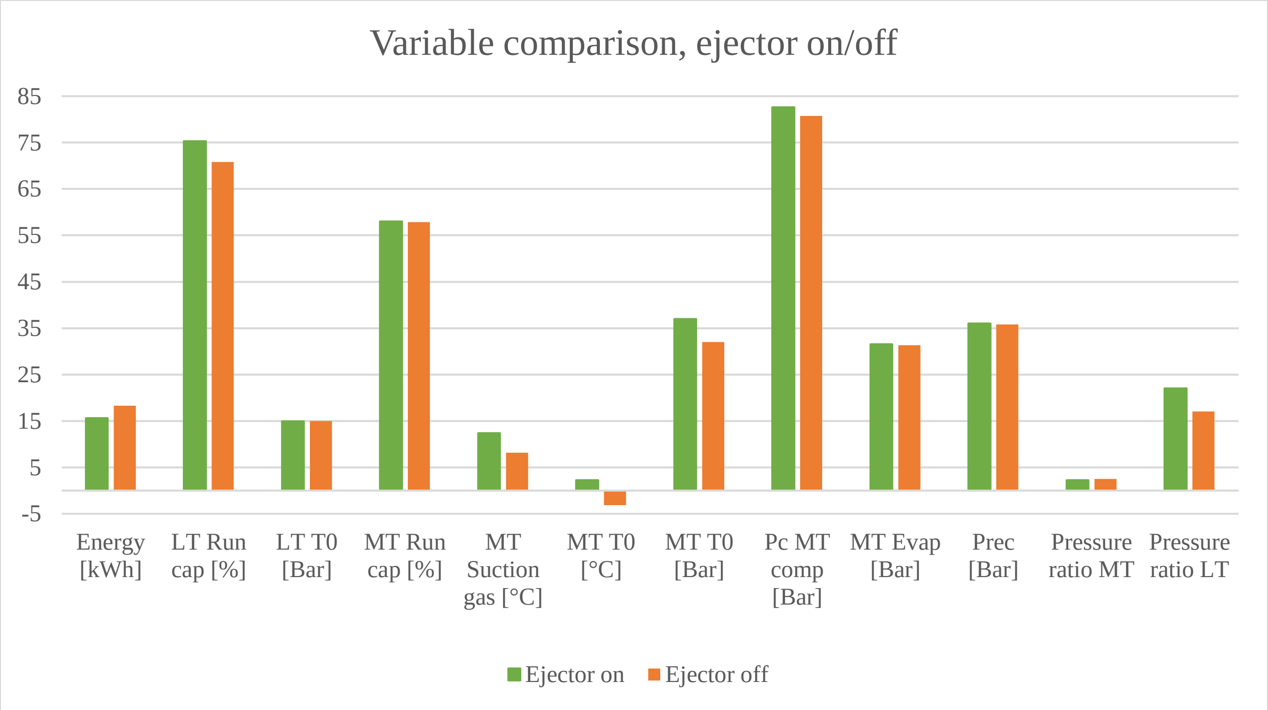

Figure 9-30: Variable comparison, ejector on/off.

As one can see demonstrated, the energy demand is somewhat higher for the period where the ejector is not performing a pressure lift, the energy need is 15.6kW when the ejector is running, compared to 18.3 kW when the ejector is off. The MT suction gas is higher with the ejector running, 12.37°C compared to 8.19 °C when the ejector function is off. One also notes the shift in MT T0 as a result of the change in reference point between the operation modes. The discharge pressure is higher during ejector operation with 82.59 bar compared to 80.75 bar when the ejector is off, which gives a pressure lift of 45.63 bar when the ejector is on and 48.71 bar when it is off. The pressure ratio over MT compressor racks is 2.23 when the ejector is running, and 2.52 when the ejector is in operation mode one. LT compressor run capacity is slightly elevated in ejector operation compared to when the ejector is off, 75.31 % in contrast to 70.8. The LT compressor rack also have a higher pressure ratio when the ejector is in operation, 22.03 bar, compared to 17.06 when the ejector is off.

9.2.2.5 P-h diagram

Figure 9-31 shows the p-h diagram with average values from the period 13:30 to 14:00, with ejector operation. This includes the LT evaporation pressure at 15.51 bar, MT evaporation pressure at 31.53 bar and gas cooler pressure at 82.59 bar. The receiver pressure and exit temperature from the gas cooler is notes, as well as the isentropic enthalpy.

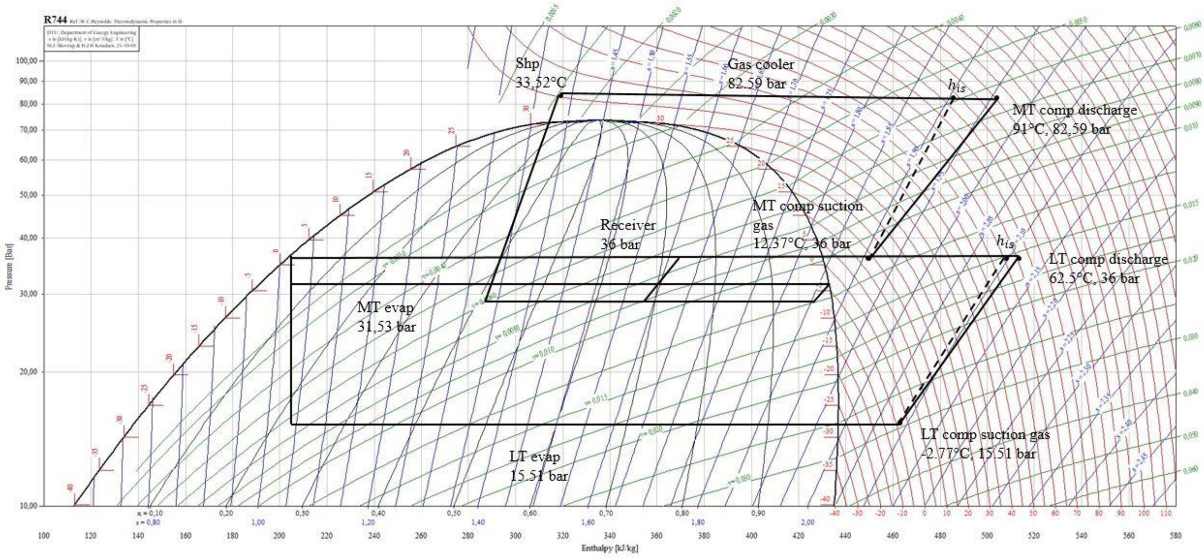

Figure 9-31: P-h diagram for Spar Røyken

9.3 System comparison, Spar Snarøya and Spar Røyken

The analyzed periods in the previous sections is put together in Figure 9-32. For Spar Snarøya the values correspond to 19th June 2020, with the time interval 15:00-16:00 with parallel compressor operation. The values from Spar Røyken are collected from 15th July 2021, in the time period 13:30 to 14:30 with ejector operation.

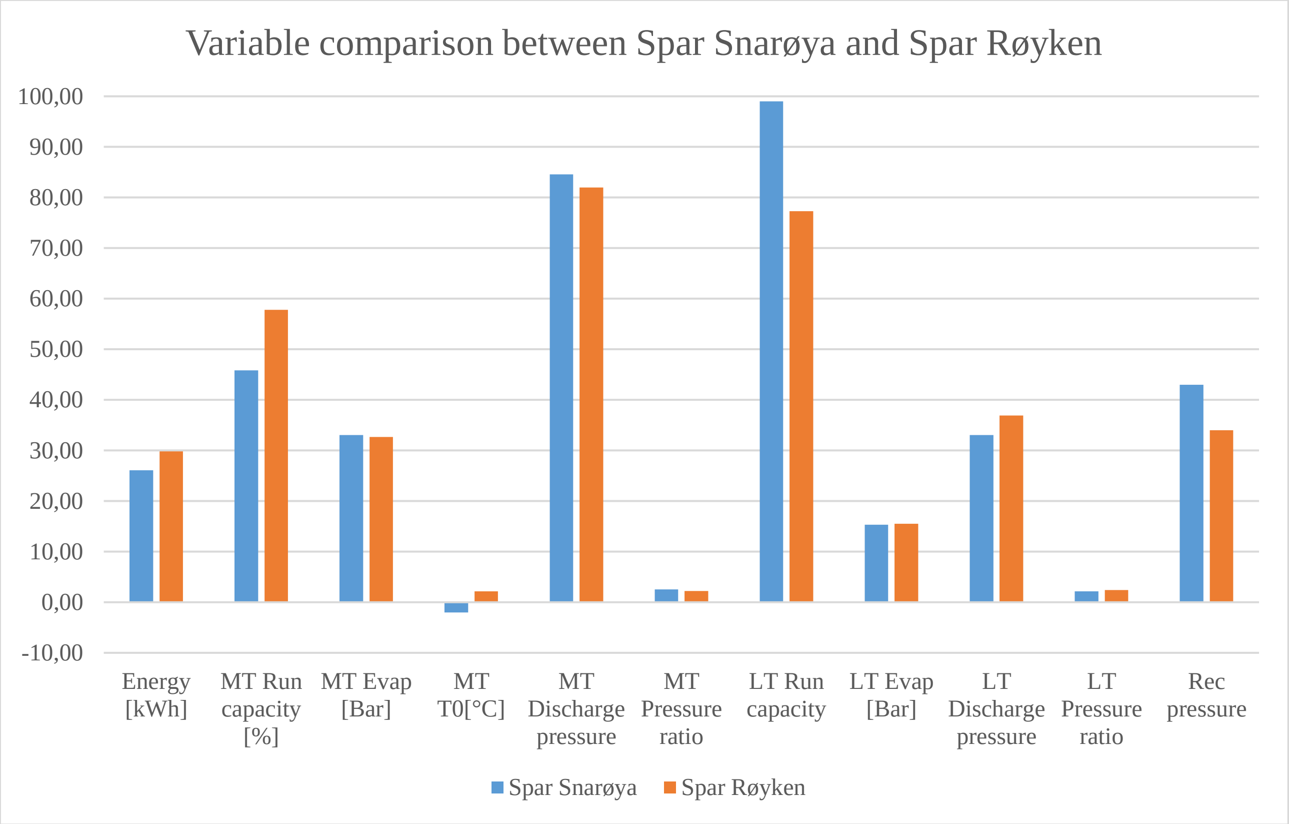

Figure 9-32: Variable comparison between Spar Snarøya and Spar Røyken

The system comparison states differences in variables between the two refrigeration units. One can observe a higher energy demand in the average hourly value for Spar Røyken with 26.9 kWh compared to 26.1 kWh for Spar Snarøya, but also a higher MT run capacity at 57.82 % at Spar Røyken, with Spar Snarøya at 45.8 %. The pressure ratio is somewhat higher on the MT compressors at Spar Snarøya, at 2.56 compared to 2.22 at Spar Røyen. One can also note the difference in receiver pressure, which is 43 bar at Spar Snarøya and 34 bar at Spar Røyken. Further on, the systems have similar evaporation pressures both on the LT level and the MT level.

10 DISCUSSION

In general there are a few points to note regarding the results in this thesis. The formerly noted discrepancy between timestamps and actual variable change may affect the results in varying manners. For the calculation part there were made a few assumptions that might influence the result, as well as the Bitzer simulations might not be correct. All of these factors might influence the result, and give a different conclusion than the one reached in this thesis.

10.1 Spar Snarøya

Ideally the system COP should have been calculated by values given from sensors in the refrigeration system, but due to the missing values surrounding the parallel compressor this was not possible. The high pressure value used by Bitzer when it was set to "Auto" to simulate the three first time intervals are a bit too high compared to the values from IWMAC. According to the attached P-h diagram the pressure is set to be borderline transcritical as one can see from figure Figure 9-8, as a result of this the high pressure side is estimated a bit higher than the actual value.

10.2 Spar Røyken

10.2.1 27TH APRIL 2021

This date corresponds to phase one as mentioned several times during this thesis. This was the initial analysis of the system during ejector operations. From the graphs displayed in section 9.2.1.1 one can note a few things. One of the major things were that it takes the compressor racks 10 minutes to respond to the ejector being turned on by changing the requested capacity. As a result of this a drop in suction pressure is observed. With the slow response from the MT compressor rack, the receiver pressure also declines without response. The receiver pressure is also set lower to create a more flash gas and in turn a larger opening degree in the flash gas bypass valve. The receiver pressure declines, and the MT compressor rack isn't responding fast enough. The team from Kelvin AS could also inform that there was some issues with the evaporators on MT level due to the low evaporation pressure.

The initial impression of the system layout was also a part of phase one. The ejector pushes the refrigerant via the receiver, through piping with valves and bends before it supplies the refrigerant to the inner shell of the receiver. This extra piping may cause a pressure loss that doesn't benefit the ejector, when the goal is to achieve a pressure lift. The system also appears to be too large, as the mass flow in the system isn't large enough for the ejector to operate.

These observations resulted in a meeting with the manufacturer and Kelvin. The issues that had been uncovered during the analysis of the results from phase one were discussed and a few things were changed. The manufacturer informed that there had been some issues with the ejector, where it loses count of opening degree after being open for a while. As a solution to this, a driver update was scheduled. The PI control of the MT compressor rack was also adjusted to ensure a quicker response, which was thought to help with ejector operations.

10.2.2 15TH JULY 2021

In phase two there was only one date to choose from, but there were three longer periods where the ejector was in operation. In these periods one could see the difference from the previous phase in a few aspects. Firstly, the MT-compressors responds faster than in phase one which causes the system to operate with a higher degree of running stability. The decline in receiver pressure is also absent in this phase as a result of this.

One can also note a change in the variable called MT T0 due to reference changes from phase one. In phase one MT T0 refers to evaporator temperature and pressure, but in this phase it is changed during ejector operation. The reference point for MT T0 is now receiver pressure and temperature. The evaporation temperature for unit K23 is also more stable during ejector operation in this phase, there are fewer high spikes in temperature values.

A higher pressure lift is also noted during phase two, and the difference between the minimum and maximum pressure lift is also smaller. The average pressure lift is increased by at least 1.32 bar during ejector operation.

The decline in MT request capacity is not observed as expected, a slight decline can be seen when the ejector is running but not the amount as expected. However there is an increase in LT run capacity because the LT compressors have to match the pressure lift done by the ejector. So the decline in the requested capacity for the MT compressor rack might be masked by the increased demand on the LT-level.

10.2.3 15TH JULY 2021, ON VS OFF

The 15th July was used as both baseline and as ejector operation date. The comparison between two periods with the same run capacity on the MT level gives an opportunity to see the differences in the rest of the system. As one can see, energy demand is a bit higher when the ejector isn’t running, due to the pressure lift done by the ejector. The LT run capacity is slightly higher in the period where the ejector is operation, due to need to match the pressure lift done by the ejector as the refrigerant streams are being fed to the receiver.

The change in reference point for MT T0 is also noted in the figure, showing that the reference point shifts from evaporation temperature to receiver temperature during ejector mode. The suction gas temperature to the MT compressors are also elevated as a result of the pressure lift done by the ejector. The discharge pressure is slightly higher when the ejector is in operation, but due to the higher suction pressure the pressure ratio across the compressor is lower when the system is operation in ejector mode. This is contributor to the lower energy demand when the ejector is lifting the pressure. One can also note a higher pressure ratio across the LT compressor, but this doesn’t equalize the lower energy demand on MT-level.

10.3 System comparison

The systems are paired together in subsection 9.3, and there were a few differences as well as similarities. However, there are too few variables available during this master thesis to conduct a thorough comparison of the system and their performance.

11 CONCLUSION

The goal with designing the ejector system is to install a smaller machine, and let the ejector deal with the top load. The few hours where the conditions are sufficient for ejector operation should cover about 16 hours in the southern part of Norway. As a result of this, the refrigeration systems could be built 15-16 % smaller if the ejector works as one would want. However, this is not done at Spar Røyken, and this does not benefit the system. To ensure that the supermarket didn't suffer under the trial of this new solution, the system was dimensioned for full capacity in the compressor racks, and the ejector was added on top. As a result of this the system is too large, and the ejector doesn't get sufficient mass flow to work properly. One can also note from the screenshot from IWMAC in Figure 8-15 that compressor number two doesn't have many run hours. The ejector is also in itself too big, an as a result it struggles to get a large enough mass flow to execute the pressure lift.

As one can see from the results stated in chapter 9.2., there is a decrease in energy demand during the period of ejector operation. When the ejector is running it also performs a pressure lift between 4 and 5 bar on average, which definitely helps the system performance. However, in some aspects the system design works against the ejector. With the additional valves and piping the ejector has to push the refrigerant through, a pressure loss can occur. This doesn't benefit the ejector operation where the goal is to achieve a pressure lift. A shortcut might have been a solution to this predicament, a sort of bypass when the ejector is executing the pressure lift to ensure that there is as small as possible pressure loss before the refrigerant enters the MT compressor rack.

As a conclusion from the collected data, the ejector system can be beneficial in Scandinavian climates. The ambient temperatures in the eastern part of Norway are high enough to facilitate transcritical operation of a refrigeration system with CO_2 as the refrigerant. With a well dimensioned system one can utilize the ejector to perform a pressure lift to lower the energy demand. If the ambient temperature is one of the variable that influence whether the ejector solution is appropriate for Nordic climates, there are several days with transcritical operation hours. These are viewed as potential operation hours for the ejector, but this isn't confirmed as the ejector only operated once during the summer period of 2021. However, further studies have to be conducted to determine how beneficial the system solution can be, preferably from a larger data pool.

The trend in the COP-estimation at Spar Snarøya shows promise, however, it is hard to make a conclusion based on the presented data. The results are estimated in several aspects, and this increases the degree of uncertainty. Further studies have to be conducted to make a definite conclusion regarding the system performance.

12 FURTHER WORK

For the system at Spar Snarøya, it would be beneficial for further work to involve a larger amount of variables, especially surrounding the parallel compressor. To be able to fully evaluate the system performance with less degree of uncertainty a larger amount of measuring points should be introduced.

For further work on this topic observation of an ejector unit over time would be beneficial. As a result of the dimensions at Spar Røyken, there were few dates to choose from and the runtime pool was limited. Analyzing a system with ejector operation on should be conducted with multiple periods and over time. Preferably longer ejector operation intervals should be analyzed to be able to observe a trend in variables that might not show up in the limited operation time that was used in this thesis. A further insight into why the ejector stops during the operation period should be obtained, as a lot of the parameters that should facilitate ejector operation mode are in place even though the ejector stops or doesn't run at all. Solutions to this should also be explored, there might be control changes that can be done to ensure increased operation periods.

Observing an ejector system that is dimensioned a bit smaller compared to cooling demand might be a way to approach the thesis main focus, to see if the size of the refrigeration system and the ejector affect the performance in the same degree as the impression. In general it would also be beneficial to study a system with more measuring points. Temperature, pressure and mass flow measurements in more places would help decrease the degree of uncertainty in the calculation and evaluation. A further study should also involve a cost-analysis to determine the magnitude of potential cost savings when implementing the ejector solution in form of fewer/smaller compressors.

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APPENDIX

Appendix A

04, 001:101, Advans	Comp. capacity MT	04, 001:101, Advans	Pc
15.07.2021 08:00:52	55	15.07.2021 08:02:25	77,8
15.07.2021 08:02:14	56	15.07.2021 08:03:58	77,7
15.07.2021 08:03:47	57	15.07.2021 08:05:31	78
15.07.2021 08:11:09	66	15.07.2021 08:06:52	78,1
15.07.2021 08:12:55	58	15.07.2021 08:08:26	78,2
15.07.2021 08:14:38	62	15.07.2021 08:09:49	83,1
15.07.2021 08:16:14	57	15.07.2021 08:11:20	80,1
15.07.2021 08:30:16	55	15.07.2021 08:13:06	78,9
15.07.2021 08:31:38	56	15.07.2021 08:14:49	78,3
15.07.2021 08:32:59	55	15.07.2021 08:16:24	74,4
15.07.2021 08:34:20	53	15.07.2021 08:17:55	75,9
15.07.2021 08:35:47	52	15.07.2021 08:19:16	77,3
15.07.2021 08:38:30	53	15.07.2021 08:20:39	77,9
15.07.2021 08:41:16	51	15.07.2021 08:22:22	78
15.07.2021 08:42:37	52	15.07.2021 08:24:04	77,9
15.07.2021 08:43:59	53	15.07.2021 08:25:36	77,7
15.07.2021 08:45:23	57	15.07.2021 08:27:07	78,1
15.07.2021 08:46:44	55	15.07.2021 08:28:50	78
15.07.2021 08:49:51	56	15.07.2021 08:30:27	78,2
15.07.2021 08:51:36	57	15.07.2021 08:31:49	77,8
15.07.2021 08:53:08	66	15.07.2021 08:33:10	77,7
15.07.2021 08:54:40	73	15.07.2021 08:34:31	76,9
15.07.2021 08:56:02	70	15.07.2021 08:35:58	77,3
15.07.2021 08:57:47	63	15.07.2021 08:37:20	77,8
15.07.2021 08:59:22	74	15.07.2021 08:38:41	77,9
15.07.2021 09:01:07	71	15.07.2021 08:40:03	77,8
15.07.2021 09:02:50	63	15.07.2021 08:41:27	77,9
15.07.2021 09:05:50	57	15.07.2021 08:44:10	78,4
15.07.2021 09:07:33	58	15.07.2021 08:45:34	78,3
15.07.2021 09:09:04	57	15.07.2021 08:46:55	77,9
15.07.2021 09:10:35	64	15.07.2021 08:48:40	78
15.07.2021 09:12:07	63	15.07.2021 08:51:47	80,3
15.07.2021 09:13:50	57	15.07.2021 08:53:19	82,7
15.07.2021 09:15:23	61	15.07.2021 08:54:51	81,8
15.07.2021 09:17:01	57	15.07.2021 08:56:13	78,8
15.07.2021 09:18:33	56	15.07.2021 08:57:58	78,9
15.07.2021 09:20:04	57	15.07.2021 08:59:33	77,4
15.07.2021 09:24:13	58	15.07.2021 09:01:18	76,2
15.07.2021 09:25:38	68	15.07.2021 09:03:01	77,7
15.07.2021 09:26:58	57	15.07.2021 09:04:27	78,3
15.07.2021 09:28:31	61	15.07.2021 09:06:00	77,4
15.07.2021 09:30:03	62	15.07.2021 09:07:44	77,8
15.07.2021 09:31:25	63	15.07.2021 09:09:15	77,9
15.07.2021 09:32:46	61	15.07.2021 09:10:46	79
15.07.2021 09:34:08	57	15.07.2021 09:12:18	78,5
15.07.2021 09:36:53	62	15.07.2021 09:14:02	74,1

15.07.2021 09:38:17	60	15.07.2021 09:15:34	77,4
15.07.2021 09:39:40	57	15.07.2021 09:17:12	76,6
15.07.2021 09:41:03	63	15.07.2021 09:18:44	75,3
15.07.2021 09:42:26	57	15.07.2021 09:20:15	76,7
15.07.2021 09:43:48	62	15.07.2021 09:21:37	77,6
15.07.2021 09:45:11	57	15.07.2021 09:22:57	78,8
15.07.2021 09:50:46	56	15.07.2021 09:24:23	81,9
15.07.2021 09:52:08	57	15.07.2021 09:25:48	79,9
15.07.2021 09:57:38	56	15.07.2021 09:27:09	79,3
15.07.2021 10:02:16	54	15.07.2021 09:28:42	79,1
15.07.2021 10:03:41	52	15.07.2021 09:30:14	76,9
15.07.2021 10:06:38	57	15.07.2021 09:31:36	78,2
15.07.2021 10:09:34	64	15.07.2021 09:32:57	79,7
15.07.2021 10:10:58	74	15.07.2021 09:34:19	74,5
15.07.2021 10:12:18	64	15.07.2021 09:35:41	78,2
15.07.2021 10:14:03	62	15.07.2021 09:37:04	80,5
15.07.2021 10:15:35	73	15.07.2021 09:38:28	79,6
15.07.2021 10:16:56	58	15.07.2021 09:39:51	78,8
15.07.2021 10:18:17	62	15.07.2021 09:41:14	79,3
15.07.2021 10:19:39	61	15.07.2021 09:42:37	78
15.07.2021 10:21:02	57	15.07.2021 09:43:59	79,5
15.07.2021 10:22:22	59	15.07.2021 09:45:22	78,4
15.07.2021 10:23:45	57	15.07.2021 09:46:45	75,7
15.07.2021 10:25:09	62	15.07.2021 09:48:07	76,2
15.07.2021 10:26:30	54	15.07.2021 09:49:30	76,8
15.07.2021 10:28:03	50	15.07.2021 09:50:57	77
15.07.2021 10:29:35	51	15.07.2021 09:52:19	78
15.07.2021 10:30:56	46	15.07.2021 09:53:40	78,3
15.07.2021 10:32:21	26	15.07.2021 09:55:00	78,1
15.07.2021 10:34:04	25	15.07.2021 09:56:23	78,4
15.07.2021 10:35:37	32	15.07.2021 09:57:48	77,5
15.07.2021 10:37:10	31	15.07.2021 09:59:08	77,7
15.07.2021 10:38:43	29	15.07.2021 10:00:48	78
15.07.2021 10:40:15	48	15.07.2021 10:02:27	77,6
15.07.2021 10:41:46	56	15.07.2021 10:03:52	77,3
15.07.2021 10:44:38	51	15.07.2021 10:05:27	77,1
15.07.2021 10:46:26	46	15.07.2021 10:06:49	78,8
15.07.2021 10:48:09	39	15.07.2021 10:08:21	83
15.07.2021 10:49:43	16	15.07.2021 10:09:45	83,1
15.07.2021 10:51:25	26	15.07.2021 10:11:09	80,3
15.07.2021 10:52:50	47	15.07.2021 10:12:29	81
15.07.2021 10:54:25	57	15.07.2021 10:14:13	79,4
15.07.2021 10:57:30	56	15.07.2021 10:15:46	76,9
15.07.2021 11:00:31	55	15.07.2021 10:17:07	78,4
15.07.2021 11:02:09	50	15.07.2021 10:18:28	78,9
15.07.2021 11:03:43	48	15.07.2021 10:19:50	78
15.07.2021 11:05:05	46	15.07.2021 10:21:13	72
15.07.2021 11:06:38	51	15.07.2021 10:22:33	77,3
15.07.2021 11:08:17	53	15.07.2021 10:23:56	75,1
15.07.2021 11:09:50	41	15.07.2021 10:25:20	77,3
15.07.2021 11:11:12	16	15.07.2021 10:26:41	76,1
15.07.2021 11:12:32	40	15.07.2021 10:29:46	77,2
15.07.2021 11:14:04	16	15.07.2021 10:31:07	77,3
15.07.2021 11:15:27	37	15.07.2021 10:32:32	78
15.07.2021 11:16:48	40	15.07.2021 10:34:15	77,3
15.07.2021 11:18:31	57	15.07.2021 10:35:48	76

15.07.2021 11:21:29	70	15.07.2021 10:37:21	77,5
15.07.2021 11:23:13	75	15.07.2021 10:38:53	80,5
15.07.2021 11:24:51	74	15.07.2021 10:40:25	78,4
15.07.2021 11:26:13	63	15.07.2021 10:41:57	78,8
15.07.2021 11:30:56	62	15.07.2021 10:43:17	78,7
15.07.2021 11:32:27	64	15.07.2021 10:44:49	77,9
15.07.2021 11:34:13	62	15.07.2021 10:46:37	76
15.07.2021 11:35:34	65	15.07.2021 10:48:20	76,3
15.07.2021 11:36:57	58	15.07.2021 10:49:53	76,7
15.07.2021 11:38:19	70	15.07.2021 10:51:36	78
15.07.2021 11:39:41	71	15.07.2021 10:53:01	79,1
15.07.2021 11:41:05	62	15.07.2021 10:54:36	80,4
15.07.2021 11:42:26	66	15.07.2021 10:56:04	79,5
15.07.2021 11:43:48	74	15.07.2021 10:57:41	78,6
15.07.2021 11:45:22	71	15.07.2021 10:59:01	77,9
15.07.2021 11:47:06	74	15.07.2021 11:00:42	77,4
15.07.2021 11:48:25	77	15.07.2021 11:02:20	77,1
15.07.2021 11:50:04	79	15.07.2021 11:03:54	76,4
15.07.2021 11:51:24	81	15.07.2021 11:05:16	76,9
15.07.2021 11:52:57	83	15.07.2021 11:06:49	78,6
15.07.2021 11:54:20	86	15.07.2021 11:08:28	78,1
15.07.2021 11:55:41	91	15.07.2021 11:10:01	76
15.07.2021 11:57:13	92	15.07.2021 11:11:22	74,6
15.07.2021 11:58:50	94	15.07.2021 11:12:43	76,7
15.07.2021 12:00:25	93	15.07.2021 11:14:15	77,5
15.07.2021 12:01:45	91	15.07.2021 11:15:38	77,4
15.07.2021 12:03:08	90	15.07.2021 11:16:59	79,1
15.07.2021 12:05:52	89	15.07.2021 11:18:42	80,1
15.07.2021 12:07:13	92	15.07.2021 11:20:16	79,3
15.07.2021 12:08:47	94	15.07.2021 11:21:39	82,7
15.07.2021 12:10:20	88	15.07.2021 11:23:24	81,8
15.07.2021 12:11:52	90	15.07.2021 11:25:02	79,5
15.07.2021 12:13:29	91	15.07.2021 11:26:23	80,5
15.07.2021 12:15:03	89	15.07.2021 11:28:06	78
15.07.2021 12:18:03	79	15.07.2021 11:31:07	77,7
15.07.2021 12:19:30	73	15.07.2021 11:32:38	78
15.07.2021 12:21:03	65	15.07.2021 11:35:45	77,9
15.07.2021 12:22:31	58	15.07.2021 11:37:08	77,6
15.07.2021 12:25:42	59	15.07.2021 11:38:30	76,9
15.07.2021 12:27:13	60	15.07.2021 11:39:52	77,2
15.07.2021 12:28:51	65	15.07.2021 11:41:16	77,9
15.07.2021 12:30:23	75	15.07.2021 11:42:37	78,3
15.07.2021 12:31:55	70	15.07.2021 11:43:59	77
15.07.2021 12:33:34	62	15.07.2021 11:45:32	78,1
15.07.2021 12:36:45	57	15.07.2021 11:47:17	78,9
15.07.2021 12:38:24	58	15.07.2021 11:48:36	80,4
15.07.2021 12:39:55	57	15.07.2021 11:50:15	80,7
15.07.2021 12:44:28	58	15.07.2021 11:51:35	80,2
15.07.2021 12:45:58	57	15.07.2021 11:53:08	81
15.07.2021 12:52:10	58	15.07.2021 11:54:31	81,5
15.07.2021 12:53:34	57	15.07.2021 11:55:52	82,9
15.07.2021 13:02:25	58	15.07.2021 11:57:23	83,9
15.07.2021 13:03:58	57	15.07.2021 11:59:01	84,6
15.07.2021 13:07:12	58	15.07.2021 12:00:36	85,8
15.07.2021 13:08:46	57	15.07.2021 12:01:56	85,7
15.07.2021 13:10:24	60	15.07.2021 12:03:18	85,5

15.07.2021 13:11:55	57	15.07.2021 12:04:41	83,7
15.07.2021 13:13:18	56	15.07.2021 12:06:03	83,8
15.07.2021 13:14:44	62	15.07.2021 12:07:24	85
15.07.2021 13:16:05	65	15.07.2021 12:08:58	84,6
15.07.2021 13:17:40	63	15.07.2021 12:10:31	84,5
15.07.2021 13:19:11	62	15.07.2021 12:13:39	85,4
15.07.2021 13:20:33	59	15.07.2021 12:15:14	86,4
15.07.2021 13:22:16	57	15.07.2021 12:16:42	86,1
15.07.2021 13:25:31	56	15.07.2021 12:18:14	85,4
15.07.2021 13:27:04	59	15.07.2021 12:19:41	84,9
15.07.2021 13:28:42	58	15.07.2021 12:21:14	82,4
15.07.2021 13:30:28	57	15.07.2021 12:22:42	80
15.07.2021 13:38:39	59	15.07.2021 12:24:19	78,8
15.07.2021 13:40:14	57	15.07.2021 12:25:52	78,9
15.07.2021 13:41:52	59	15.07.2021 12:27:24	78,7
15.07.2021 13:43:24	57	15.07.2021 12:29:02	81,2
15.07.2021 13:48:04	60	15.07.2021 12:30:34	82,5
15.07.2021 13:49:38	57	15.07.2021 12:32:06	85,8
15.07.2021 13:51:14	59	15.07.2021 12:33:45	84,4
15.07.2021 13:52:45	58	15.07.2021 12:35:19	83,3
15.07.2021 13:54:06	57	15.07.2021 12:36:55	84,9
15.07.2021 13:59:48	58	15.07.2021 12:38:34	84,4
15.07.2021 14:01:15	57	15.07.2021 12:40:06	83,5
15.07.2021 14:04:07	61	15.07.2021 12:41:38	85
15.07.2021 14:05:34	60	15.07.2021 12:43:07	84,9
15.07.2021 14:06:55	57	15.07.2021 12:44:39	85,1
15.07.2021 14:09:49	56	15.07.2021 12:46:09	83,6
15.07.2021 14:11:11	57	15.07.2021 12:47:44	83,3
15.07.2021 14:12:33	58	15.07.2021 12:49:18	83,7
15.07.2021 14:14:01	57	15.07.2021 12:50:50	83,9
15.07.2021 14:18:39	56	15.07.2021 12:52:21	85,1
15.07.2021 14:23:09	57	15.07.2021 12:53:45	85
15.07.2021 14:30:49	62	15.07.2021 12:55:06	84,4
15.07.2021 14:32:21	57	15.07.2021 12:56:29	84,7
15.07.2021 14:33:55	63	15.07.2021 12:57:53	85,4
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15.07.2021 14:40:10	70	15.07.2021 13:04:09	84,7
15.07.2021 14:41:43	57	15.07.2021 13:05:43	81,5
15.07.2021 14:43:11	70	15.07.2021 13:08:57	87
15.07.2021 14:44:33	62	15.07.2021 13:10:35	86,8
15.07.2021 14:45:57	83	15.07.2021 13:12:07	87,8
15.07.2021 14:47:29	94	15.07.2021 13:13:29	81,2
15.07.2021 14:50:12	96	15.07.2021 13:14:55	82,8
15.07.2021 14:51:36	94	15.07.2021 13:16:16	88,5
15.07.2021 14:52:58	89	15.07.2021 13:17:51	87,8
15.07.2021 14:54:22	79	15.07.2021 13:19:22	87,9
15.07.2021 14:55:44	64	15.07.2021 13:20:44	87,1
15.07.2021 14:57:23	57	15.07.2021 13:22:27	80,4
15.07.2021 14:58:54	58	15.07.2021 13:24:00	82
15.07.2021 15:00:34	57	15.07.2021 13:25:42	79,9
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15.07.2021 15:06:36	46	15.07.2021 13:30:38	84
15.07.2021 15:08:10	40	15.07.2021 13:31:58	82,1

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15.07.2021 15:10:59	55	15.07.2021 13:34:44	78,7
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15.07.2021 15:16:56	60	15.07.2021 13:40:25	81,3
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15.07.2021 15:21:46	61	15.07.2021 13:45:06	82,4
15.07.2021 15:25:00	57	15.07.2021 13:46:44	83,3
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15.07.2021 15:27:56	61	15.07.2021 13:49:49	85,1
15.07.2021 15:29:18	57	15.07.2021 13:51:25	84
15.07.2021 15:30:53	61	15.07.2021 13:52:56	83,5
15.07.2021 15:32:15	58	15.07.2021 13:54:17	82,9
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15.07.2021 15:37:39	56	15.07.2021 13:58:38	78,3
15.07.2021 15:41:46	57	15.07.2021 13:59:59	81,3
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15.07.2021 15:51:50	57	15.07.2021 14:02:58	79,1
15.07.2021 15:57:45	64	15.07.2021 14:04:18	82,6
15.07.2021 15:59:09	58	15.07.2021 14:05:45	86
15.07.2021 16:00:32	61	15.07.2021 14:07:06	79,8
15.07.2021 16:01:55	59	15.07.2021 14:08:27	84,5
15.07.2021 16:03:17	58	15.07.2021 14:10:00	80,5
15.07.2021 16:04:40	57	15.07.2021 14:11:22	80,4
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15.07.2021 16:21:48	57	15.07.2021 14:14:12	82,7
15.07.2021 16:24:51	55	15.07.2021 14:15:45	81,1
15.07.2021 16:26:15	57	15.07.2021 14:17:18	79,6
15.07.2021 16:27:42	56	15.07.2021 14:18:50	80,4
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15.07.2021 16:36:55	56	15.07.2021 14:23:20	80,7
15.07.2021 16:38:17	57	15.07.2021 14:25:11	79,7
15.07.2021 16:45:08	54	15.07.2021 14:26:54	79
15.07.2021 16:46:45	57	15.07.2021 14:29:39	78,5
15.07.2021 16:48:17	58	15.07.2021 14:30:59	79,9
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15.07.2021 16:52:52	57	15.07.2021 14:34:06	81,2
15.07.2021 16:56:04	61	15.07.2021 14:35:37	82,2
15.07.2021 16:57:45	57	15.07.2021 14:37:15	81,9
15.07.2021 16:59:07	58	15.07.2021 14:38:38	80,1
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15.07.2021 17:03:25	63	15.07.2021 14:43:22	79,7
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15.07.2021 17:06:24	57	15.07.2021 14:46:08	85,3
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15.07.2021 17:19:45	61	15.07.2021 14:51:46	87
15.07.2021 17:21:22	57	15.07.2021 14:53:09	86,6
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15.07.2021 17:25:57	57	15.07.2021 14:57:34	81,6

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15.07.2021 17:34:31	56	15.07.2021 15:06:47	75,8
15.07.2021 17:35:58	60	15.07.2021 15:08:20	75,5
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15.07.2021 17:40:14	59	15.07.2021 15:11:10	79,7
15.07.2021 17:41:59	57	15.07.2021 15:12:32	80,7
15.07.2021 17:51:08	59	15.07.2021 15:14:06	86,1
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15.07.2021 17:53:50	61	15.07.2021 15:17:07	83,3
15.07.2021 17:55:18	60	15.07.2021 15:18:50	81,7
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15.07.2021 18:04:21	56	15.07.2021 15:23:28	83,3
15.07.2021 18:07:06	57	15.07.2021 15:25:11	82,8
15.07.2021 18:15:21	56	15.07.2021 15:26:45	83,1
15.07.2021 18:19:52	57	15.07.2021 15:28:07	82,5
15.07.2021 18:22:58	56	15.07.2021 15:29:29	79,8
15.07.2021 18:24:39	54	15.07.2021 15:31:04	83
15.07.2021 18:29:05	51	15.07.2021 15:32:26	80,5
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15.07.2021 18:33:40	51	15.07.2021 15:36:29	79,6
15.07.2021 18:35:03	53	15.07.2021 15:37:50	80,7
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15.07.2021 18:40:33	57	15.07.2021 15:40:34	81,1
15.07.2021 18:46:53	77	15.07.2021 15:41:57	80,6
15.07.2021 18:48:32	63	15.07.2021 15:43:24	81,1
15.07.2021 18:50:02	71	15.07.2021 15:44:44	80,7
15.07.2021 18:53:16	63	15.07.2021 15:46:24	80,5
15.07.2021 18:54:54	58	15.07.2021 15:47:58	80,3
15.07.2021 18:56:18	59	15.07.2021 15:49:19	80,9
15.07.2021 18:59:11	58	15.07.2021 15:50:40	81,2
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15.07.2021 19:02:05	57	15.07.2021 15:53:51	80,6
15.07.2021 19:04:48	59	15.07.2021 15:55:11	81,1
15.07.2021 19:06:11	57	15.07.2021 15:56:34	81,3
15.07.2021 19:13:46	58	15.07.2021 15:57:56	84,5
15.07.2021 19:16:41	57	15.07.2021 15:59:20	85,5
15.07.2021 19:21:34	60	15.07.2021 16:00:43	85
15.07.2021 19:22:58	59	15.07.2021 16:02:06	84,5
15.07.2021 19:24:30	57	15.07.2021 16:03:28	83,7
15.07.2021 19:25:52	56	15.07.2021 16:04:51	83,5
15.07.2021 19:27:13	57	15.07.2021 16:06:37	84,2
15.07.2021 19:31:43	62	15.07.2021 16:08:11	85,3
15.07.2021 19:33:05	57	15.07.2021 16:09:31	85,8
15.07.2021 19:35:49	64	15.07.2021 16:12:49	85,3
15.07.2021 19:37:22	57	15.07.2021 16:14:22	80,2
15.07.2021 19:43:03	58	15.07.2021 16:15:44	85,8
15.07.2021 19:44:25	57	15.07.2021 16:17:06	81,4
15.07.2021 19:47:33	58	15.07.2021 16:18:40	81,7
15.07.2021 19:52:16	57	15.07.2021 16:20:23	82,5
15.07.2021 19:58:21	59	15.07.2021 16:21:59	83,5

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		15.07.2021 16:27:53	84
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		15.07.2021 16:30:46	83,8
		15.07.2021 16:32:17	82,7
		15.07.2021 16:34:00	82,5
		15.07.2021 16:35:33	82,7
		15.07.2021 16:37:06	82,6
		15.07.2021 16:38:28	83,4
		15.07.2021 16:39:50	83,3
		15.07.2021 16:41:14	83,4
		15.07.2021 16:42:35	83,6
		15.07.2021 16:43:57	87,2
		15.07.2021 16:45:19	85,3
		15.07.2021 16:46:55	82,5
		15.07.2021 16:48:28	85,5
		15.07.2021 16:50:03	88,4
		15.07.2021 16:51:23	83,7
		15.07.2021 16:53:02	82,8
		15.07.2021 16:54:33	81,5
		15.07.2021 16:56:15	88,9
		15.07.2021 16:57:56	88,5
		15.07.2021 16:59:18	85,2
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		15.07.2021 17:08:16	80,6
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		15.07.2021 17:11:14	87,8
		15.07.2021 17:12:46	87,6
		15.07.2021 17:14:09	87,2
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		15.07.2021 17:16:52	80,4
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		15.07.2021 17:19:55	88,7
		15.07.2021 17:21:32	81,9
		15.07.2021 17:23:12	83,2
		15.07.2021 17:24:45	87,8
		15.07.2021 17:26:08	85,5
		15.07.2021 17:27:41	85,4
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		15.07.2021 17:30:23	80,5
		15.07.2021 17:31:43	85,4
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		15.07.2021 17:34:42	80,7
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		15.07.2021 17:37:31	89,7
		15.07.2021 17:39:04	87,6
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		15.07.2021 17:42:10	82,1
		15.07.2021 17:43:41	84
		15.07.2021 17:45:03	85,1
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		15.07.2021 17:58:30	84
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		15.07.2021 18:05:54	78,8
		15.07.2021 18:07:17	79,8
		15.07.2021 18:10:01	79,6
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		15.07.2021 18:12:48	79,5
		15.07.2021 18:14:10	79,4
		15.07.2021 18:15:32	79,6
		15.07.2021 18:17:15	79,7
		15.07.2021 18:18:41	79,8
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		15.07.2021 18:21:36	79,9
		15.07.2021 18:23:09	80
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		15.07.2021 18:29:15	78,4
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		15.07.2021 18:33:51	79,3
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		15.07.2021 18:36:38	80
		15.07.2021 18:38:01	80,3
		15.07.2021 18:39:23	79,9
		15.07.2021 18:40:44	80,4
		15.07.2021 18:42:26	80
		15.07.2021 18:43:59	80,5
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		15.07.2021 18:47:04	84,7
		15.07.2021 18:48:43	83,7
		15.07.2021 18:50:13	82,9
		15.07.2021 18:51:55	85,6
		15.07.2021 18:53:27	85,7
		15.07.2021 18:55:05	85,5
		15.07.2021 18:56:29	87,3
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		15.07.2021 19:04:59	86,6
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		15.07.2021 19:13:56	80,2
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		15.07.2021 19:23:09	85
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		15.07.2021 19:40:27	77,5
		15.07.2021 19:41:49	82,8
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		15.07.2021 19:44:36	77,5
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		15.07.2021 19:47:44	82,9
		15.07.2021 19:49:16	83,4
		15.07.2021 19:50:49	85,4
		15.07.2021 19:52:27	83,4
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		15.07.2021 19:55:23	81
		15.07.2021 19:57:09	81,9
		15.07.2021 19:58:32	81,1

04, 001:101, Advans	**Shp**	**04, 001:101, Advans**	**Voltage input 1**
15.07.2021 08:05:29	29,6	15.07.2021 13:05:34	1
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15.07.2021 08:09:47	30,1	15.07.2021 17:52:31	0
15.07.2021 08:11:18	30,7	15.07.2021 18:59:14	1
15.07.2021 08:13:04	31,1	15.07.2021 19:52:18	0
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15.07.2021 08:16:23	30,3		
15.07.2021 08:17:53	29,9		
15.07.2021 08:19:14	29,2		
15.07.2021 08:20:37	28,8		
15.07.2021 08:22:20	29,3		
15.07.2021 08:24:03	29,2		
15.07.2021 08:25:34	29,3		
15.07.2021 08:28:48	29,5		
15.07.2021 08:30:25	29,8		
15.07.2021 08:31:47	29,9		
15.07.2021 08:34:29	29,7		
15.07.2021 08:35:56	29,5		
15.07.2021 08:40:01	29,6		
15.07.2021 08:42:46	29,8		
15.07.2021 08:44:08	29,9		
15.07.2021 08:45:32	30,1		
15.07.2021 08:48:39	30		
15.07.2021 08:51:45	30,1		
15.07.2021 08:53:17	30,7		
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15.07.2021 08:56:11	31,3		
15.07.2021 08:57:56	31,1		
15.07.2021 08:59:31	31		
15.07.2021 09:01:16	30,8		
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15.07.2021 09:04:25	30,2		
15.07.2021 09:07:42	29,9		
15.07.2021 09:09:13	30,1		
15.07.2021 09:10:44	30,2		
15.07.2021 09:12:16	30,4		
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15.07.2021 09:15:32	29,9		
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15.07.2021 09:25:47	30,6		

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15.07.2021 09:32:55	30,2		
15.07.2021 09:34:17	29,9		
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15.07.2021 09:38:26	29,9		
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15.07.2021 09:54:59	29,6		
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15.07.2021 13:40:20	1,9
15.07.2021 13:41:58	2,2
15.07.2021 13:43:30	1,5
15.07.2021 13:45:00	1,9

15.07.2021 13:46:39	2,5
15.07.2021 13:48:10	1,8
15.07.2021 13:49:44	2,3
15.07.2021 13:51:20	1,6
15.07.2021 13:52:51	1,7
15.07.2021 13:54:12	2,1
15.07.2021 13:57:09	2,2
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15.07.2021 14:01:21	3
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15.07.2021 14:04:13	2,7
15.07.2021 14:05:39	1,4
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15.07.2021 14:08:22	2,3
15.07.2021 14:09:55	2,6
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15.07.2021 14:14:06	2,6
15.07.2021 14:15:40	2,4
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15.07.2021 14:32:27	-4,4
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15.07.2021 14:35:31	-4,5
15.07.2021 14:37:10	-4,8
15.07.2021 14:38:32	-4,4
15.07.2021 14:40:16	-3,5
15.07.2021 14:41:49	1
15.07.2021 14:43:17	-3,5
15.07.2021 14:44:39	-3,6
15.07.2021 14:46:03	-3,9
15.07.2021 14:47:35	-3,7
15.07.2021 14:48:56	-3,4
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15.07.2021 14:57:29	-0,9
15.07.2021 14:59:00	-2,9
15.07.2021 15:00:40	-4,9
15.07.2021 15:02:13	-3,1
15.07.2021 15:03:45	-3,8
15.07.2021 15:05:09	-3,6
15.07.2021 15:06:42	-3,3
15.07.2021 15:08:15	-3,9
15.07.2021 15:09:39	-2,6
15.07.2021 15:11:05	-2,8
15.07.2021 15:12:27	-2
15.07.2021 15:14:01	-3,7
15.07.2021 15:15:43	-3,5

15.07.2021 15:17:01	-3,3
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15.07.2021 15:26:40	-3,9
15.07.2021 15:28:02	-3,8
15.07.2021 15:29:24	-1,7
15.07.2021 15:30:59	-3,6
15.07.2021 15:32:21	-1,5
15.07.2021 15:33:42	-4,5
15.07.2021 15:35:02	-2,4
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15.07.2021 15:39:07	-2
15.07.2021 15:40:29	-1,9
15.07.2021 15:41:52	-1,6
15.07.2021 15:43:18	-1,5
15.07.2021 15:44:38	-1,6
15.07.2021 15:46:19	-1,8
15.07.2021 15:47:52	-2,1
15.07.2021 15:49:13	-2,2
15.07.2021 15:50:34	-2
15.07.2021 15:51:56	-1,7
15.07.2021 15:53:46	-1,5
15.07.2021 15:55:05	-1,2
15.07.2021 15:56:29	-1
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15.07.2021 16:00:38	-2,9
15.07.2021 16:02:01	-3,4
15.07.2021 16:04:45	-2
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15.07.2021 16:08:06	-0,8
15.07.2021 16:09:26	-0,2
15.07.2021 16:11:10	0,7
15.07.2021 16:12:44	-0,5
15.07.2021 16:14:17	3,4
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15.07.2021 16:24:57	3,2
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15.07.2021 16:43:52	3
15.07.2021 16:45:13	3,8
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15.07.2021 16:49:58	3,1
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15.07.2021 16:52:57	3,6
15.07.2021 16:54:28	3,4
15.07.2021 16:56:10	3,1
15.07.2021 16:57:51	2,7
15.07.2021 17:00:46	3,5
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15.07.2021 17:03:31	2,9
15.07.2021 17:04:56	2,8
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15.07.2021 17:08:11	3,4
15.07.2021 17:09:48	2,8
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15.07.2021 17:12:41	2,4
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15.07.2021 17:18:08	2,2
15.07.2021 17:19:50	2
15.07.2021 17:21:27	3,2
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15.07.2021 17:24:40	1,7
15.07.2021 17:26:03	2,3
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15.07.2021 17:31:38	1,9
15.07.2021 17:33:04	2,6
15.07.2021 17:34:37	2,5
15.07.2021 17:36:04	1,9
15.07.2021 17:37:26	1,4
15.07.2021 17:38:59	2,1
15.07.2021 17:40:20	2,2
15.07.2021 17:42:05	2,6
15.07.2021 17:43:36	1,1
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15.07.2021 17:48:16	1,7
15.07.2021 17:49:50	2,8
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15.07.2021 17:58:25	0,2
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15.07.2021 18:03:04	-0,1
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15.07.2021 18:07:12	-2,4
15.07.2021 18:09:56	-2,6
15.07.2021 18:11:20	-2,8
15.07.2021 18:12:42	-2,6
15.07.2021 18:14:05	-2,4

15.07.2021 18:15:27	-2,2
15.07.2021 18:17:10	-2
15.07.2021 18:18:36	-2,2
15.07.2021 18:19:58	-2,3
15.07.2021 18:21:31	-2,4
15.07.2021 18:24:44	-2,9
15.07.2021 18:27:49	-3
15.07.2021 18:29:10	-3,1
15.07.2021 18:30:44	-2,8
15.07.2021 18:32:15	-3
15.07.2021 18:33:46	-2,2
15.07.2021 18:35:09	-2,9
15.07.2021 18:36:33	-2,4
15.07.2021 18:37:56	-2,6
15.07.2021 18:39:18	-2,4
15.07.2021 18:42:21	-2
15.07.2021 18:43:54	-1,5
15.07.2021 18:45:16	-4,1
15.07.2021 18:46:59	-2,5
15.07.2021 18:48:38	-2,3
15.07.2021 18:50:08	-0,6
15.07.2021 18:51:50	-3
15.07.2021 18:53:22	0,6
15.07.2021 18:54:59	1,6
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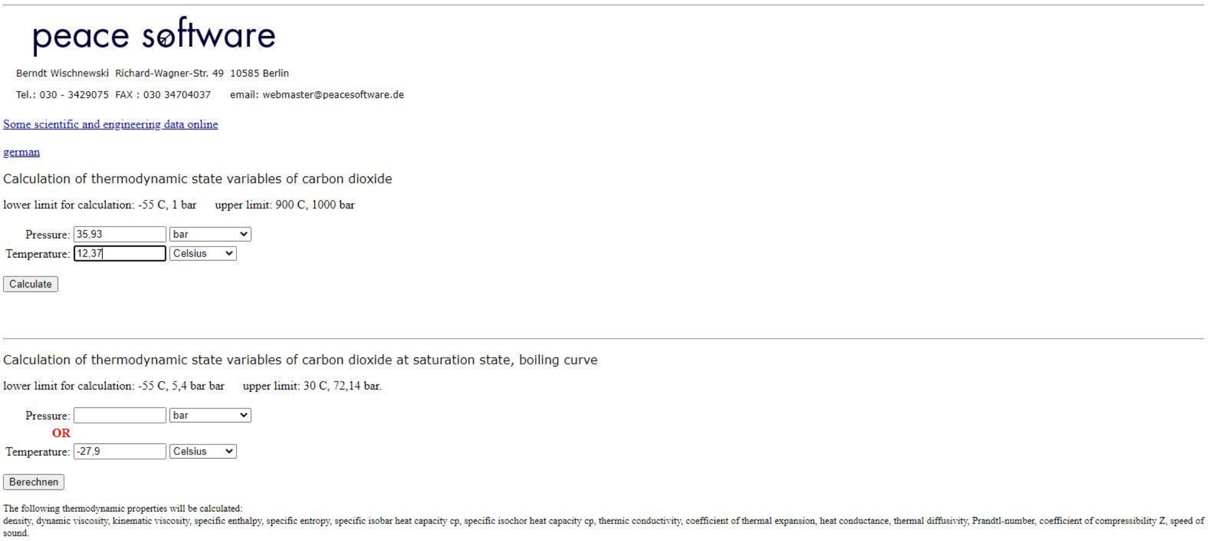
15.07.2021 18:59:17	3,2
15.07.2021 19:00:40	3,6
15.07.2021 19:02:11	3,2
15.07.2021 19:03:33	2,8
15.07.2021 19:04:54	3,4
15.07.2021 19:06:16	2,5
15.07.2021 19:07:39	3
15.07.2021 19:09:01	3,1
15.07.2021 19:10:22	4,7
15.07.2021 19:12:07	3,6
15.07.2021 19:13:51	3,4
15.07.2021 19:15:12	3
15.07.2021 19:16:47	1,9
15.07.2021 19:18:30	3,1
15.07.2021 19:20:09	3,3
15.07.2021 19:21:40	2,8
15.07.2021 19:23:04	1,9
15.07.2021 19:24:35	2,1
15.07.2021 19:25:58	3,7
15.07.2021 19:27:18	2,4
15.07.2021 19:29:03	1,8
15.07.2021 19:30:25	3,2
15.07.2021 19:31:48	2,5
15.07.2021 19:33:11	2,3
15.07.2021 19:34:34	3,3
15.07.2021 19:35:54	1,4
15.07.2021 19:37:27	1,5
15.07.2021 19:39:00	2,9
15.07.2021 19:40:22	2,7
15.07.2021 19:41:44	1,4

15.07.2021 19:43:09	1
15.07.2021 19:44:31	3,7
15.07.2021 19:45:53	1,7
15.07.2021 19:47:39	1,8
15.07.2021 19:49:11	1,7
15.07.2021 19:50:44	1,9
15.07.2021 19:52:22	0,5
15.07.2021 19:53:56	0,1
15.07.2021 19:55:18	-0,5
15.07.2021 19:57:04	-0,4
15.07.2021 19:58:26	-2,7
15.07.2021 19:59:59	-1,3

Appendix B


peace software

Berndt Wischnewski Richard-Wagner-Str. 49 10585 Berlin

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Some scientific and engineering data online

german

Calculation of thermodynamic state variables of carbon dioxide

lower limit for calculation: -55 C, 1 bar upper limit: 900 C, 1000 bar

Pressure: 35,93 bar

Temperature: 12,37 Celsius

Calculate

Calculation of thermodynamic state variables of carbon dioxide at saturation state, boiling curve

lower limit for calculation: -55 C, 5,4 bar bar upper limit: 30 C, 72,14 bar.

Pressure: bar

OR

Temperature: -27,9 Celsius

Berechnen

The following thermodynamic properties will be calculated:
density, dynamic viscosity, kinematic viscosity, specific enthalpy, specific entropy, specific isobar heat capacity cp, specific isochor heat capacity cp, thermic conductivity, coefficient of thermal expansion, heat conductance, thermal diffusivity, Prandtl-number, coefficient of compressibility Z, speed of sound.

peace software

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Property	Value	Unit
Medium :	**carbon dioxide**	
state of aggregation :	**gas**	
Pressure :	35.93	[bar]
Temperature :	12.37	[Celsius]
Density :	91.18114154	[kg / m^3]
Specific Enthalpy :	448.0064099	[kJ / kg]
Specific Entropy :	1.904521574	[kJ / kg K]
Specific isobar heat capacity : cp	1.559387701	[kJ / kg K]
Specific isochor heat capacity : cv	0.814445142	[kJ / kg K]
Isobar coefficient of thermal expansion :	9.965188833	[10^{-3} (1 / K)]
Heat conductance	20.36698277	[10^{-3} (W / m * K)]
Dynamic viscosity :	15.26441885	[10^{-6} (Pa s)]
Kinematic viscosity :	0.16740763048359	[10^{-6} m^2 / s]
Thermal diffusivity :	1.549690426	[10^{-7} m^2 / s]
Prandtl-Number :	1.1628249522	
Coefficient of compressibility Z :	0.7404789561	
speed of sound :	224.9023017	[m / s]

Appendix C

		LT comp										
		Running capacity LT		T_inn	T_out	delta T	P_evap	ro[kg/m3]	massflow [kg/s]	cp[kJ/kgK]	Q[kW]	Refrigeration load
13:00-13:30	1	86,18	86,18 %	-3,46	73,13	76,59	14,99	33,92	0,057	1,03	4,46	31,84 %
13:30-14:00	1	75,31	75,31 %	-2,77	73,63	76,40	14,93	33,77	0,049	1,03	3,87	27,64 %
14:00-14:30	1	78,75	78,75 %	-3,38	73,29	76,67	14,97	33,92	0,052	1,03	4,08	29,13 %
14:30-15:00	0	75,53	75,53 %	-3,39	64,92	68,31	14,93	33,92	0,050	1,03	3,49	24,89 %
15:00-15:30	0	70,80	70,80 %	-2,56	64,40	66,96	14,98	33,55	0,046	1,03	3,17	22,62 %
15:30-16:00	0	69,50	69,50 %	-3,13	65,36	68,49	14,85	33,5	0,045	1,03	3,18	22,68 %
16:00-16:30	1	75,00	75,00 %	-2,85	68,77	71,62	15,12	34	0,049	1,03	3,64	25,98 %
16:30-17:00	1	76,70	76,70 %	-2,96	74,66	77,61	14,95	33,69	0,050	1,03	3,99	28,53 %
17:00-17:30	1	65,56	65,56 %	-2,44	74,27	76,71	14,88	33,3	0,042	1,03	3,32	23,70 %
17:30-18:00	1	77,13	77,13 %	-2,47	73,78	76,25	15,02	33,79	0,050	1,03	3,94	28,13 %
18:00-18:30	0	70,64	70,64 %	-3,19	67,91	71,09	14,74	33,23	0,045	1,02	3,29	23,50 %
18:30-19:00	0	72,50	72,50 %	-2,22	66,19	68,41	15,17	34,12	0,048	1,03	3,37	24,07 %
19:00-19:30	1	92,12	92,12 %	-3,18	73,95	77,13	15,13	34,22	0,061	1,03	4,84	34,58 %
19:30-20:00	1	82,47	82,47 %	-3,75	73,22	76,97	14,85	33,62	0,054	1,03	4,25	30,37 %

	MT comp										
	Running capacity MT		T_inn	T_out	delta T	P_evap	ro[kg/m3]	massflow [kg/s]	cp[kJ/kgK]	Q[kW]	Refrigeration load MT
1	59	59,00 %	13,28	77,95	64,67	31,64	75,23	0,261	1,34	22,65	44,41 %
1	58	58,00 %	12,37	75,95	63,58	31,53	75,36	0,257	1,35	22,09	43,32 %
1	57,6	57,60 %	12,87	75,46	62,59	31,98	76,68	0,260	1,36	22,14	43,41 %
0	70,79	70,79 %	7,96	86,07	78,10	31,36	76,98	0,321	1,39	34,84	68,31 %
0	57,89	57,89 %	8,19	84,18	75,99	31,32	76,7	0,261	1,39	27,62	54,16 %
0	57,9	57,90 %	11,35	84,35	73,00	32,48	79,35	0,271	1,41	27,85	54,60 %
1	57,3	57,30 %	10,32	81,18	70,86	32,48	80,08	0,270	1,43	27,38	53,69 %
1	56,9	56,90 %	13,63	76,52	62,89	32,46	77,9	0,261	1,37	22,49	44,10 %
1	58	58,00 %	12,27	76,67	64,40	31,89	76,71	0,262	1,38	23,29	45,66 %
1	57,85	57,85 %	11,68	78,12	66,44	31,23	74,67	0,254	1,34	22,65	44,40 %
0	55	55,00 %	11,52	83,04	71,52	31,85	77,02	0,249	1,38	24,62	48,27 %
0	58,85	58,85 %	10,39	84,05	73,65	32,07	78,51	0,272	1,4	28,05	55,01 %
1	57,91	57,91 %	13,82	78,71	64,88	31,54	74,56	0,254	1,33	21,94	43,02 %
1	58,6	58,60 %	11,76	78,49	66,73	31,32	74,95	0,259	1,35	23,30	45,68 %

BITZER Output data

Created on : 10.10.2021 15:29:45

Table of content

Project survey

Selected compressors

CO2 Systems	2x	2KME-1K
	1x	4PTE-7K
	1x	4MTE-10K
	1x	4JTE-15K

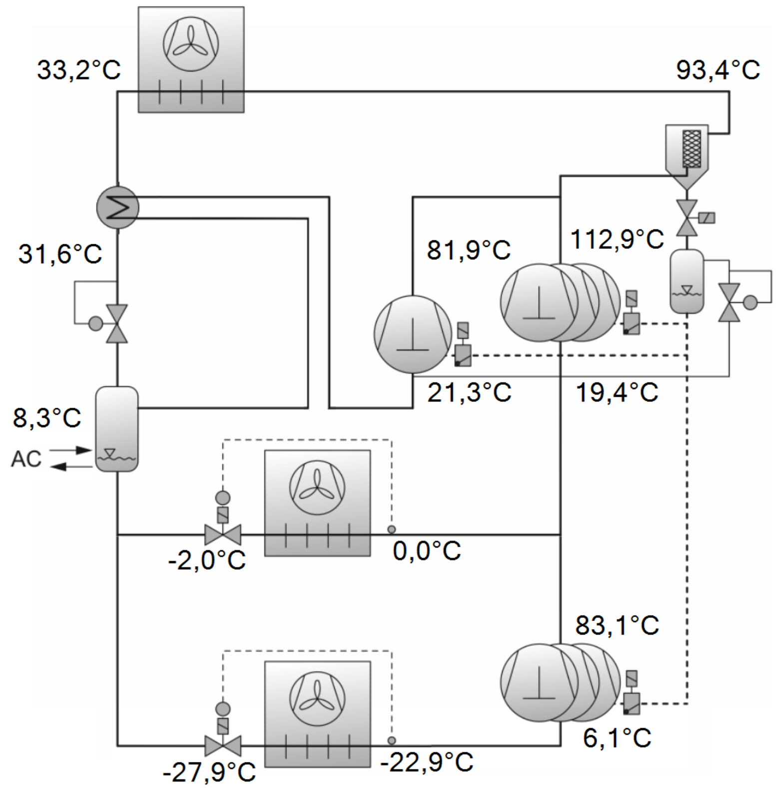
Chosen accessory

VARIPACK	1x	FNY+30-4
	1x	FMY+14-4
	1x	FMY+6-4

Selection: CO2 Systems

COP/EER Evaporator: 2,48

Input Values	LT-Stage	MT-Stage	Parallel-Stage
System		Flashgas	
Series	ME (high standstill pressures)	Standard	Standard
Operating mode	Subcritical	Transcritical	Transcritical
Number compressors	2	2	1
Evaporating SST	-27,91 °C	-2,01 °C	
Evaporator superheat	5,00 K	2,00 K	
Suction line superheat	29,0 K	2,10 K	5,00 K
High pressure		84,6 bar(a)	
Gas cooler outlet		33,2 °C	
Intermed. pressure		43,1 bar(a) / 8,30 °C	
Air Conditioning			27,9 kW
IHX Flashg. - Gas c.			8,00 K
Power frequency	50Hz		
Power voltage	400V		



Result

Compressor	LT-Stage	2KME-1K	2KME-1K
Frequency compressor	--	54,0 Hz	--
Evaporator capacity	9,37 kW	4,86 kW	4,50 kW
Ratio	--	51,9 %	48,1 %
Power input	2,62 kW	1,37 kW	1,26 kW
Current	4,15 A	2,23 A	1,92 A
Voltage range	--	380-420V	380-420V
Mass flow	152,1 kg/h	79,0 kg/h	73,1 kg/h
Total superheat	33,9 K	33,9 K	33,9 K
Discharge gas temp. w/o cooling	83,1 °C	83,4 °C	82,9 °C

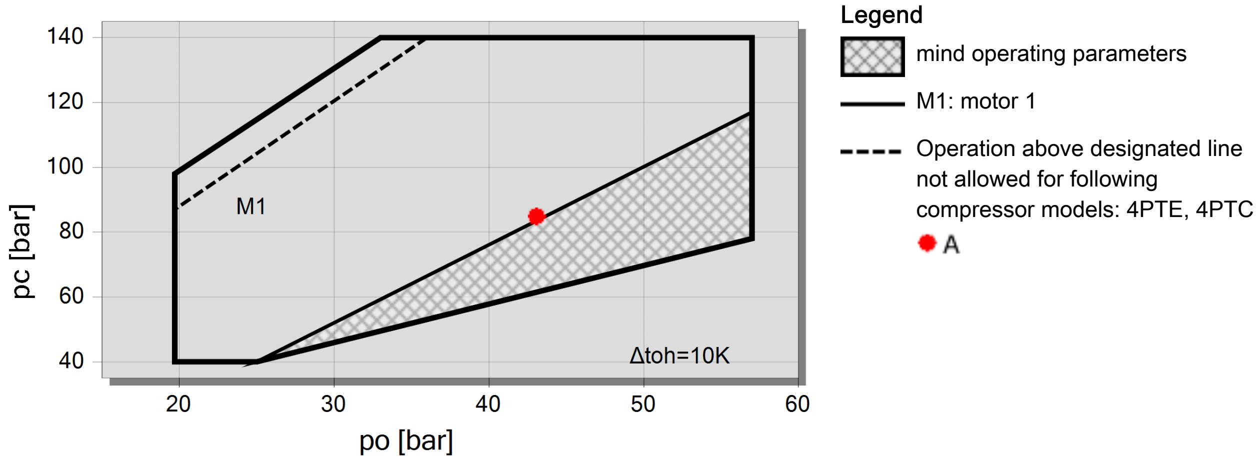
Compressor	MT-Stage	4PTE-7K	4MTE-10K
Frequency compressor	--	43,0 Hz	--
Evaporator capacity	26,5 kW	9,05 kW	17,47 kW
Ratio	--	34,1 %	65,9 %
Gas cooler capacity	93,8 kW	13,44 kW	25,9 kW
Power input	11,65 kW	4,12 kW	7,52 kW
Current	22,4 A	9,09 A	13,35 A
Voltage range	--	380-420V	380-420V
Mass flow	597 kg/h	204 kg/h	393 kg/h
Total superheat	21,4 K	21,4 K	21,4 K
Discharge gas temp. w/o cooling	112,9 °C	114,9 °C	111,8 °C
optimal high pressure	82,0 bar(a)	--	--

Compressor	Parallel-Stage	4JTE-15K
Frequency compressor	--	55,0 Hz
Evaporator capacity	27,9 kW	--
Ratio	--	100,0 %
Power input	11,42 kW	11,42 kW
Current	19,48 A	19,48 A
Voltage range	--	380-420V
Mass flow	1011 kg/h	1011 kg/h
Total superheat	13,00 K	13,00 K
Discharge gas temp. w/o cooling	81,9 °C	81,9 °C

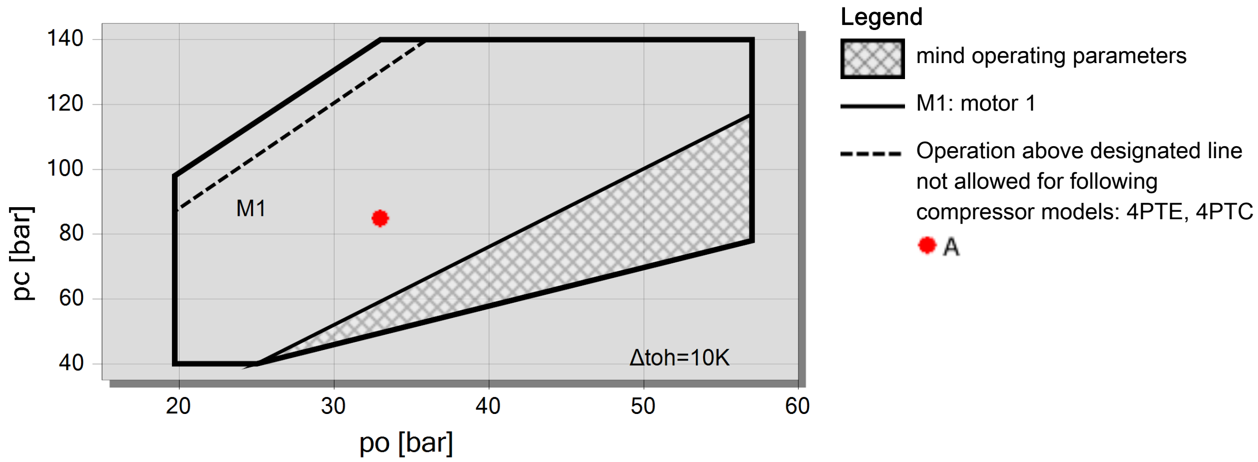
: Tentative Data.
: Power consumption at compressor inlet.

MT-Stage: Tentative Data.
MT-Stage: Power consumption at compressor inlet.
Parallel-Stage: Tentative Data.
Parallel-Stage: Attention, consider operating parameters. See KP-130 or consult BITZER.
Parallel-Stage: Power consumption at compressor inlet.

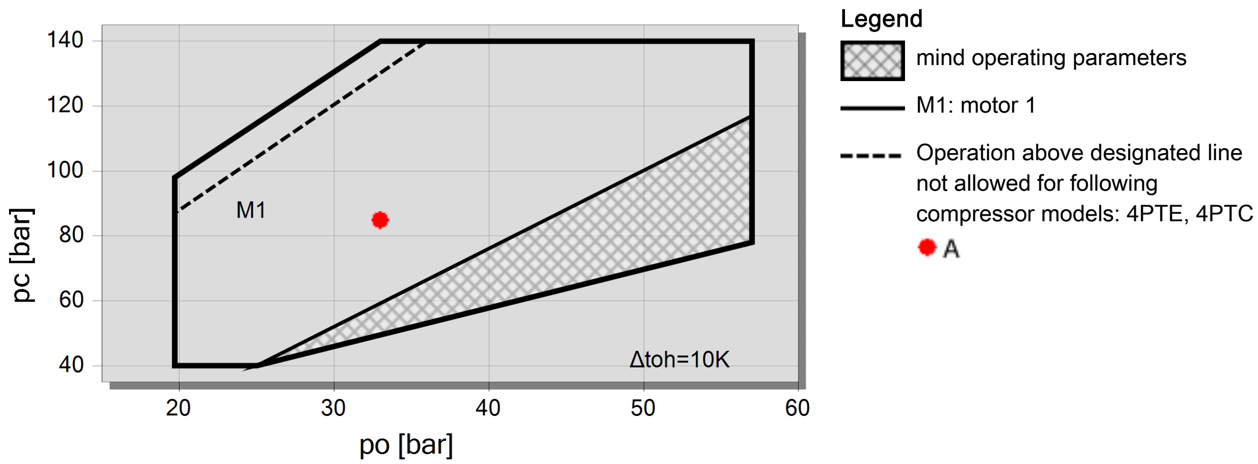
Application Limits 4JTE-15K



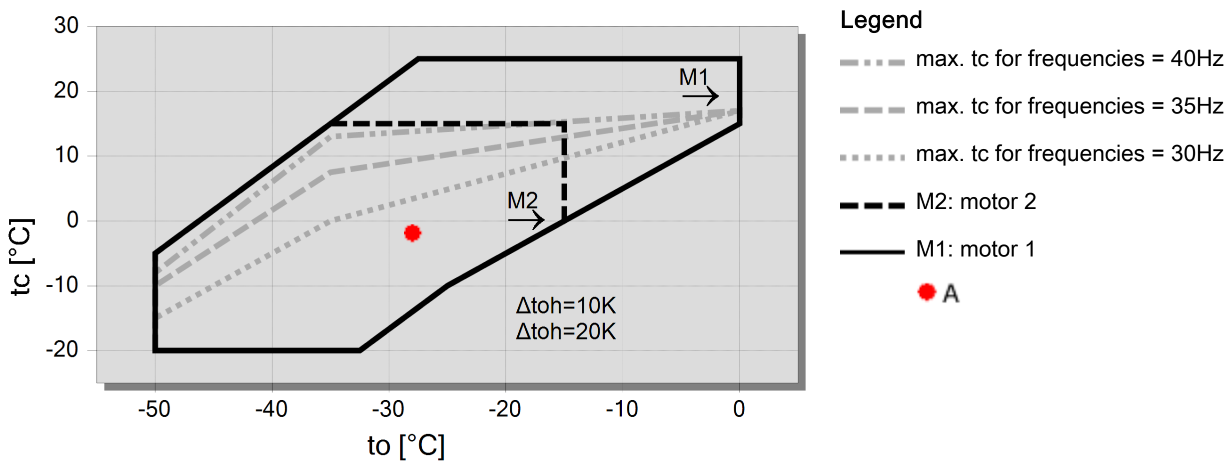
Application Limits 4MTE-10K



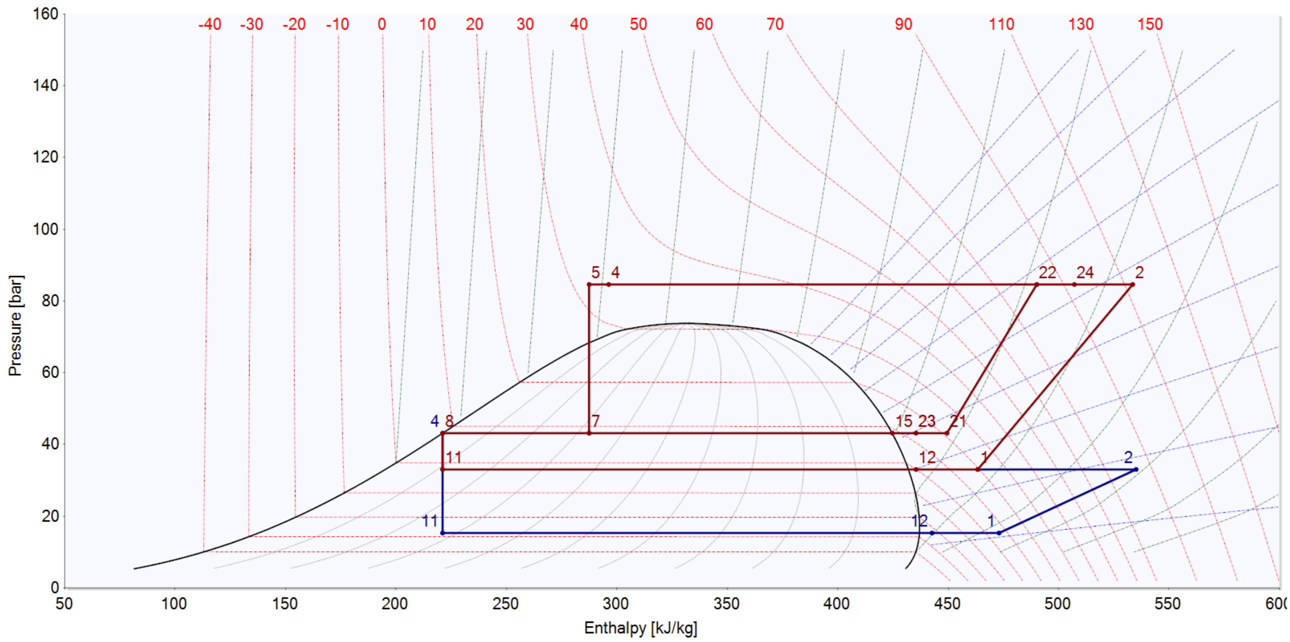
Application Limits 4PTE-7K



Application Limits 2KME-1K



p,h Diagram



LT-Stage

1 - 2	Compression
4 - 11	Expansion
11 - 12	Evaporation
12 - 1	Superheat suction line

MT-Stage

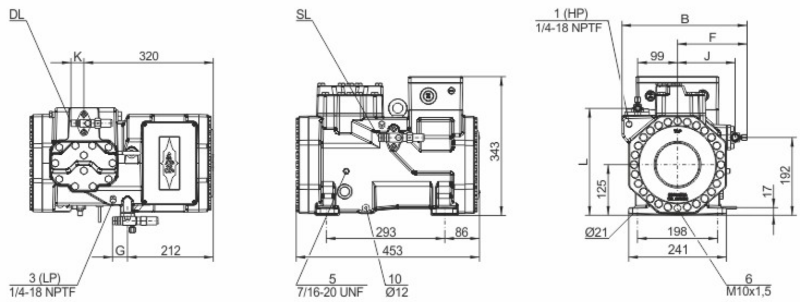
1 - 2	Compression
2 - 4	Gas cooling/Condensation
4 - 5	IHX Subcooling
5 - 7	Expansion to intermediate pressure
7 - 8	Intermediate receiver: liquid outlet
8 - 11	Expansion to evaporation pressure
11 - 12	Evaporation
12 - 1	Total superheat
7 - 15	Intermediate receiver: gas outlet

Parallel-Stage

21 - 22	Compression
23 - 21	IHX Superheat

Technical Data: 2KME-1K

Dimensions and Connections



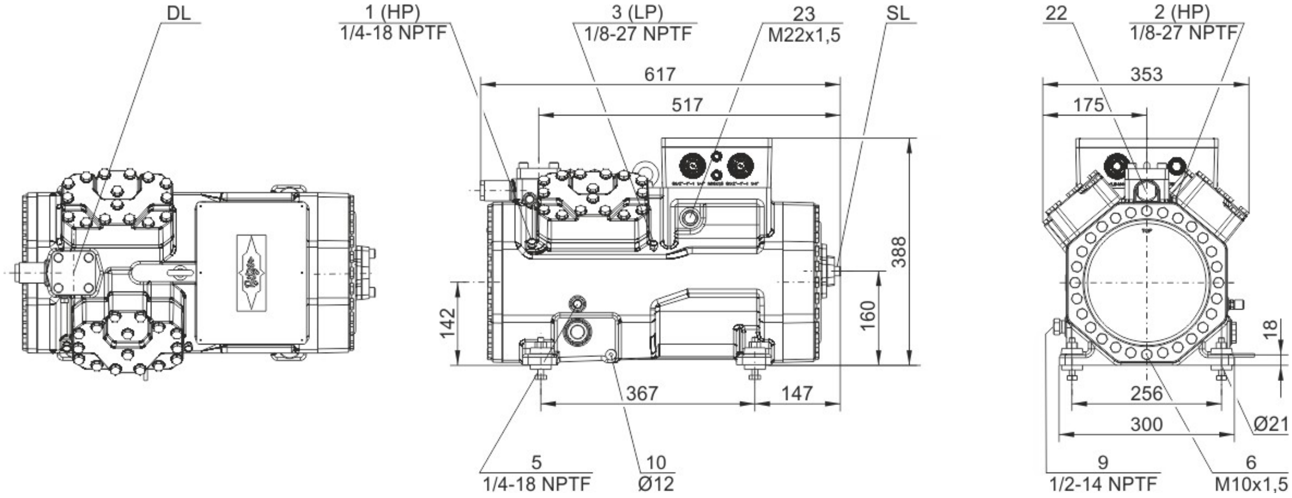
	B mm	F mm	G mm	J mm	K mm	L mm
2MME-07K..2FME-5K	311	174	37	145	32	264
2EME-4K..2DME-7K	319	182	58	149	37	268

Technical Data

Technical Data	
Displacement (1450 RPM 50Hz)	2,71 m3/h
Displacement (1750 RPM 60Hz)	3,27 m3/h
No. of cylinder x bore x stroke	2 x 30 mm x 22 mm
Weight	81 kg
Max. pressure (LP/HP)	100 /100 bar
Connection suction line	16 mm - 5/8''
Connection discharge line	12 mm - 1/2''
Oil type R744 (CO2)	BSE60K (Standard), BSE85K (Option), BSG68K (Option)
Motor data	
Motor version	2
Motor voltage (more on request)	380-420V Y-3-50Hz
Max operating current	3.7 A
Starting current (Rotor locked)	26.0 A
Extent of delivery (Standard)	
Motor protection	SE-B3(Standard), SE-B2(Option)
Enclosure class	IP65
Vibration dampers	Standard
Oil charge	1,2 dm³
Discharge shut-off valve	Standard
Suction shut-off valve	Standard
Available Options	
Crankcase heater	0..120 W PTC (Option)
Sound measurement	

Technical Data: 4JTE-15K

Dimensions and Connections

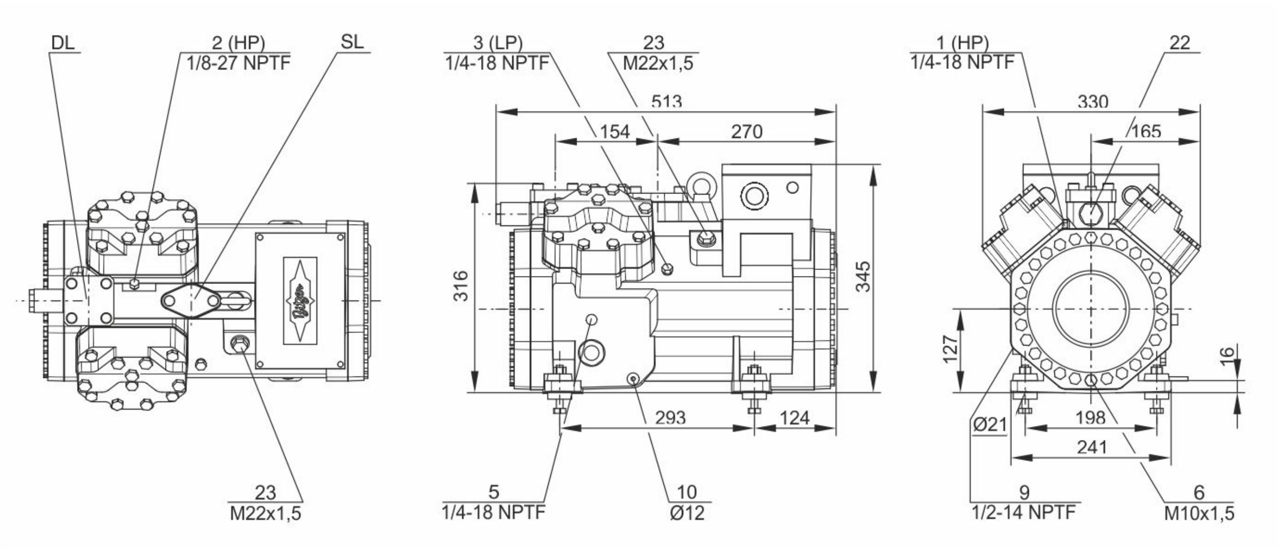


Technical Data

Technical Data	
Displacement (1450 RPM 50Hz)	9,3 m3/h
Displacement (1750 RPM 60Hz)	11,2 m3/h
No. of cylinder x bore x stroke	4 x 30mm x 38mm
Weight	182 kg
Max. pressure (LP/HP)	100/160 bar
Connection suction line	28 mm - 1 1/8"
Connection discharge line	18 mm - 3/4"
Oil type R744 (CO_2)	BSE85K (Standard), BSG68K (Option)
Motor data	
Motor version	1
Motor voltage (more on request)	380-420V PW-3-50Hz
Max operating current	32.0 A
Winding ratio	50/50
Starting current (Rotor locked)	81.0 A Y / 132.0 A YY
Extent of delivery (Standard)	
Motor protection	SE-B3(Standard), SE-B2(Option)
Enclosure class	IP65
Vibration dampers	Standard
Oil charge	2,60 dm³
Crankcase heater	0..140 W PTC (Standard)
Available Options	
Connection suction line	Option
Discharge shut-off valve	Option
Oil level monitoring	OLC-K1 (Option)
Sound measurement	
Sound power level (-10°C / 90bar)	79 dB(A) @ 50Hz
Sound pressure level @ 1m (-10°C / 90bar)	71 dB(A) @ 50Hz

Technical Data: 4MTE-10K

Dimensions and Connections

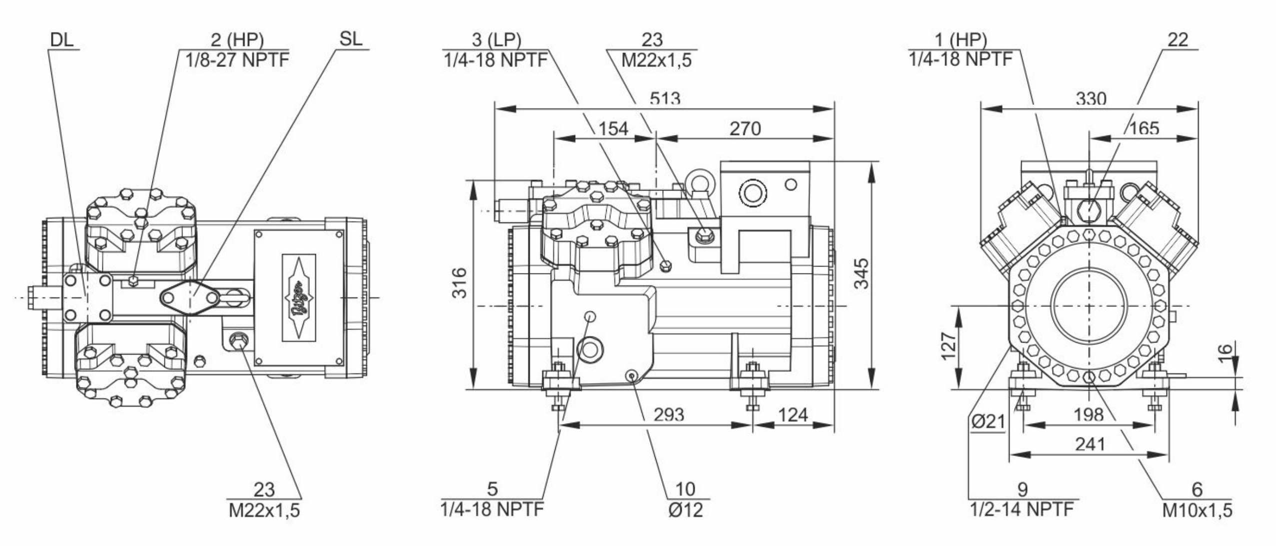


Technical Data

Technical Data	
Displacement (1450 RPM 50Hz)	6,5 m3/h
Displacement (1750 RPM 60Hz)	7,8 m3/h
No. of cylinder x bore x stroke	4 x 30mm x 27mm
Weight	120 kg
Max. pressure (LP/HP)	100/160 bar
Connection suction line	22 mm - 7/8''
Connection discharge line	18 mm - 3/4''
Oil type R744 (CO2)	BSE85K (Standard), BSG68K (Option)
Motor data	
Motor version	1
Motor voltage (more on request)	380-420V Y-3-50Hz
Max operating current	21.9 A
Starting current (Rotor locked)	97.0 A
Extent of delivery (Standard)	
Motor protection	SE-B3(Standard), SE-B2(Option)
Enclosure class	IP65
Vibration dampers	Standard
Oil charge	2,00 dm³
Crankcase heater	0..120 W PTC (Standard)
Available Options	
Connection suction line	Option
Discharge shut-off valve	Option
Capacity Control - infinite	100-25% (Option)
Oil level monitoring	OLC-K1 (Option)
Sound measurement	
Sound power level (-10°C / 90bar)	70 dB(A) @ 50Hz
Sound pressure level @ 1m (-10°C / 90bar)	62 dB(A) @ 50Hz

Technical Data: 4PTE-7K

Dimensions and Connections



Technical Data

Technical Data	
Displacement (1450 RPM 50Hz)	4,3 m3/h
Displacement (1750 RPM 60Hz)	5,2 m3/h
No. of cylinder x bore x stroke	4 x 30mm x 17,5mm
Weight	118 kg
Max. pressure (LP/HP)	100/160 bar
Connection suction line	22 mm - 1 1/8"
Connection discharge line	18 mm - 3/4"
Oil type R744 (CO_2)	BSE85K (Standard), BSG68K (Option)
Motor data	
Motor version	1
Motor voltage (more on request)	380-420V Y-3-50Hz
Max operating current	15.3 A
Starting current (Rotor locked)	82.4 A
Extent of delivery (Standard)	
Motor protection	SE-B3(Standard), SE-B2(Option)
Enclosure class	IP65
Vibration dampers	Standard
Oil charge	2,00 dm³
Crankcase heater	0..120 W PTC (Standard)
Available Options	
Connection suction line	Option
Discharge shut-off valve	Option
Capacity Control - infinite	100-25% (Option)
Oil level monitoring	OLC-K1 (Option)
Sound measurement	
Sound power level (-10°C / 90bar)	70 dB(A) @ 50Hz
Sound pressure level @ 1m (-10°C / 90bar)	62 dB(A) @ 50Hz

Design remarks of CO2 Booster Systems

The design of CO2 booster systems is influenced by many factors. Different system configurations and the operating conditions especially at part load are the main factors that influence the system performance and determine the adequate amount and size of the compressors. In the following, the most important remarks regarding the design of such a system are listed.

Flash Gas Bypass Booster System

In a CO2 booster system, the refrigerant is expanded by means of a high pressure control valve into a liquid receiver on an intermediate pressure level. The liquid receiver mainly acts as a phase separator and buffer. The saturated liquid refrigerant from the liquid receiver is used to supply the medium- and low-temperature evaporators. The, ideally saturated, flash gas is further expanded to MT-stage pressure level by means of a back pressure control valve and routed to the MT compressors. This flash gas bypass operation reduces both the operating pressure inside the receiver and adjacent components in the liquid line plus the mass flow rate from the intermediate pressure vessel to the evaporators.

Flash gas Bypass

The flash gas that is generated during the expansion process in the intermediate receiver has to be bypassed to the MT suction line in order to maintain the liquid receiver on a constant pressure level. Depending on the pressure difference of the intermediate receiver to the MT stage, a certain amount of liquid is generated while bypassing the saturated flash gas to a lower pressure level.

Attention: Liquid operation can occur!

In order to minimize the risk of liquid slugging, it is recommended to use a flash gas bypass heat exchanger in order to evaporate the generated liquid during expansion of the flash gas. This heat exchanger can exchange heat between any other refrigerant stream that contains warmer fluid with a sufficient heating capacity (e.g. liquid refrigerant out of the intermediate receiver or gas cooler outlet – Attention: Patents have to be considered).

Mixing Point

In the suction line of the MT compressors, three different refrigerant streams are mixed to one MT suction mass flow

* Bypassed flash gas out of the liquid receiver
* Discharge gas from the LT stage compressor(s)
* Superheated gas from the MT evaporators

Mixing those streams can result in either insufficient or excessive suction gas superheat, depending on the load ratio of the system (MT-Load / LT-Load), the ambient temperature (amount of produced flash gas) and the intermediate pressure level (amount of liquefied flash gas). Therefore, it is mandatory to also check the worst case scenarios in part load of both, the LT and MT stage as well as the operation at low ambient temperatures.

Scenario 1: Low load at MT stage, high load at LT stage, low ambient temperatures

* Low ambient temperature: Low amount of flash gas– less “cold” flash gas at mixing point
* Low MT load: Less “cold” gas from MT evaporators
* High LT load: More “hot” discharge gas from LT compressors

→ Tendency for higher suction gas superheat
→ Motor cooling can be insufficient! It has to be ensured that the suction gas superheat is within the limits (max. 40K resulting superheat, VARISPEED: 20K). A discharge gas desuperheater downstream of the LT compressors reduces the total suction gas superheat in the MT stage.

Scenario 2: High load at MT stage, low load at LT stage, high ambient temperatures

* High ambient temperature: Large amount of flash gas– more “cold” or even liquid flash gas at mixing point – liquid refrigerant has to be evaporated before mixing point.
* High MT load: More “cold” gas from MT evaporators
* Low LT load: Less “hot” discharge gas from LT compressors

→ Lower suction gas superheat, liquid slugging can occur
→ Liquid operation will harm the compressors (low oil temperature, reduction of oil viscosity, foaming, liquid hammer, oil washout). It has to be ensured that the suction gas superheat is within the limits (min. 10K). Flash gas superheating via internal heat exchanger is recommended.

Scenario 3: Low load at MT stage, low load at LT stage, low ambient temperatures (winter operation)

* Low overall load (night time, winter)

→ On / off cycling reduces system stability and compressor lifetime due to less effective lubrication.
→ Compressor selection has to be made for the lowest possible load ratio in order to ideally operate at least one compressor per stage continuously.

Compressor selection

Compressors can be selected manually or automatically by the software. Generally, the first compressor (lead compressor) is operated with a frequency inverter (selection of VARIPACK or VARISPEED) to ensure a stable operation point. For smooth operation and stepless capacity control, it is recommended that the displacement of the lead compressor within its approved frequency range is comparable to the displacement of the next fixed speed compressor. When switching off the fixed speed compressor, this capacity gap should be compensated by the lead compressor. Also a combination of uneven compressor displacements will help to adapt the cooling capacity as good as possible. It is recommended to choose a proper amount of compressors in order to reach the lowest cooling capacity possible without excessive cycling of the compressors.

Please keep in mind, that the application limits shown in the BITZER Software are valid for a fixed superheat of 20K for LT compressors and 10K for MT compressors. The total superheat at each stage should be in between the allowed limits (10K – 40K, VARISPEED 20K). Too high superheat or the use of a frequency inverter can restrain the application limits. Possible frequency inverter limitations can be observed in the VARIPACK selection in the accessories menu.

Due to the generally low pressure ratio of the LT compressors, it is highly recommended to use an internal suction gas heat exchanger in order to maintain a proper superheat to increase the discharge- and oil temperatures. This is also a measure to protect the compressors against liquid operation at high load fluctuations, hot gas defrost or malfunctioning expansion valves.

Oil management

Compound systems are typically equipped with an active oil management system. This consists of a high pressure oil separator, low pressure oil reservoir, and oil level regulators on the individual compressors in order to ensure a reliable compressor operation.

Please mind oil supply and oil distribution challenges in systems with more than 6 compressors per stage. Important is to carefully dimension the oil return pipe. Especially, excessive pressure drop in the oil return line has to be avoided as otherwise degassing effects due to pressure reduction can adversely affect the oil supply to the compressors. Furthermore, the suction collector has to be installed perfectly horizontal. Consultation with BITZER is recommended.

More information regarding the safe and reliable operation of CO2 systems can be found in the documentation KB-130, KP-130 and the publications: Designing, calculation and simulation of booster refrigeration systems with CO2 and Operating behavior of CO2 booster systems. Furthermore, BITZER offers trainings for designing and operating sub- and transcritical CO2 systems.

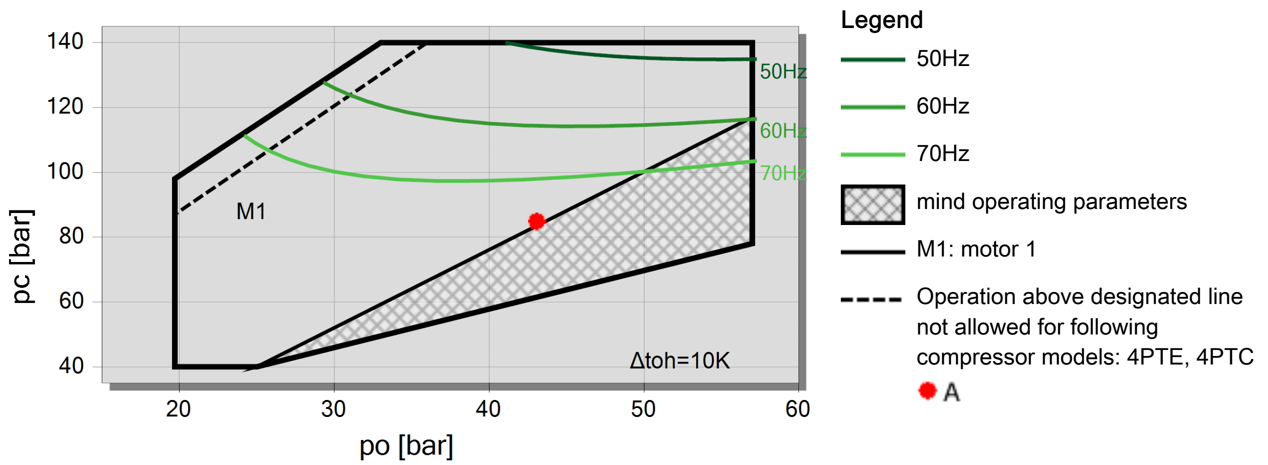
Selection: VARIPACK

Result

Compressor:	4JTE-15K (Parallel)	4PTE-7K (MT)	2KME-1K (LT)
Recommendation:	FNY+30-4	FMY+14-4	FMY+6-4
Selection	**FNY+30-4**	**FMY+14-4**	**FMY+6-4**
Frequency compressor:	55 Hz	43 Hz	54 Hz
Recommended operating point:	A	A	A
Selected operating point:	A	A	A
Power input	11,85 kW	4,25 kW	1,42 kW
Current (400V)	17,27 A	6,37 A	2,07 A
Max. current	25,30 A	11,40 A	3,42 A
min. cooling capacity	12,25 kW (25 Hz)	5,03 kW (25 Hz)	3,22 kW (30 Hz)
max. cooling capacity	39,96 kW (70 Hz)	16,41 kW (70 Hz)	8,57 kW (75 Hz)
Frequency range	25 Hz..70 Hz	25 Hz..70 Hz	30 Hz..75 Hz
Enclosure class	IP66	IP66	IP66

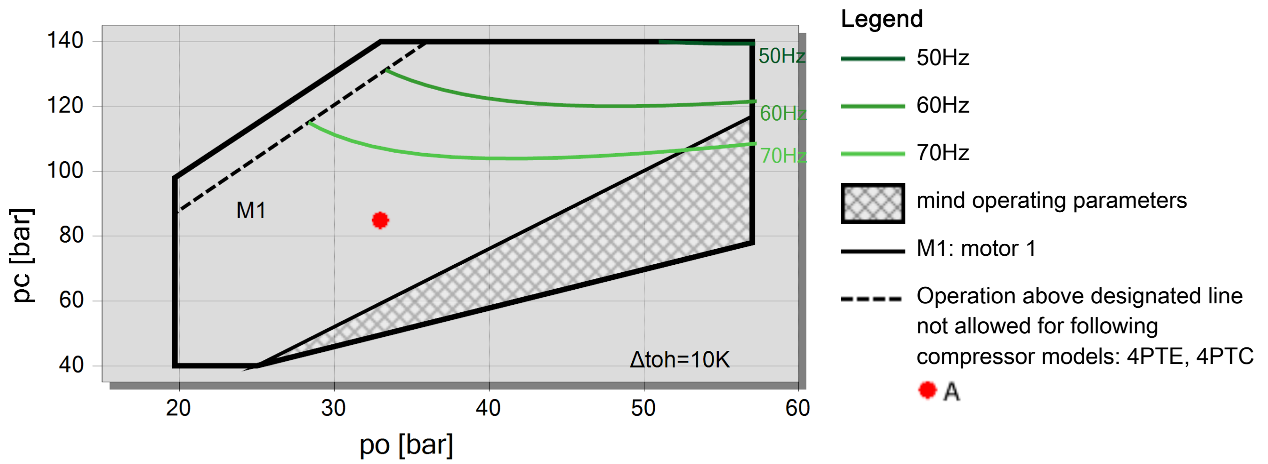
#1: Power consumption at frequency inverter inlet.
#2: Power consumption at frequency inverter inlet.
#3: Power consumption at frequency inverter inlet.

Application Limits 4JTE-15K



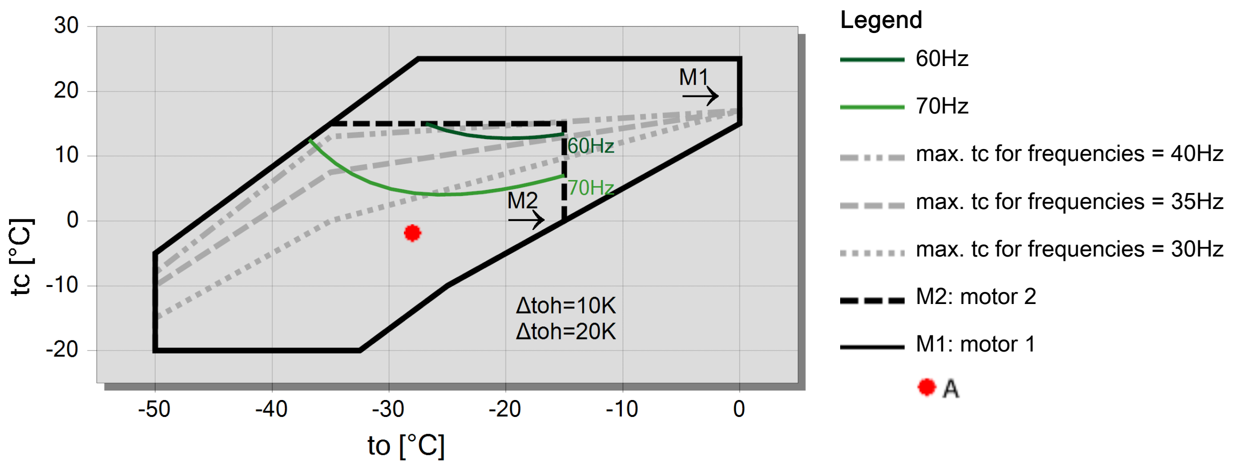
The restriction of the application limit is caused by the frequency inverter. Right or above the frequency lines the maximum output current of the inverter is less than the operating current of the compressor. Select a larger inverter if you need a wider range of the application limit.

Application Limits 4PTE-7K



The restriction of the application limit is caused by the frequency inverter. Right or above the frequency lines the maximum output current of the inverter is less than the operating current of the compressor. Select a larger inverter if you need a wider range of the application limit.

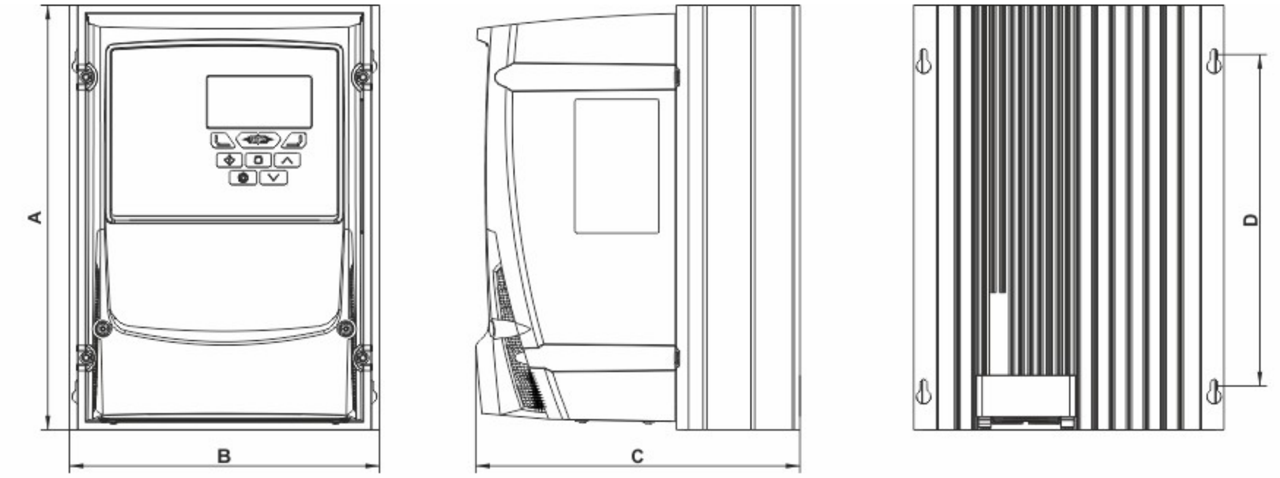
Application Limits 2KME-1K



The restriction of the application limit is caused by the compressor. Right or above the frequency lines the operating current would be greater than the maximum allowed value. Select a larger motor version or a special voltage motor, if you need a wider range of the application limit.

Technical Data: FNY+30-4

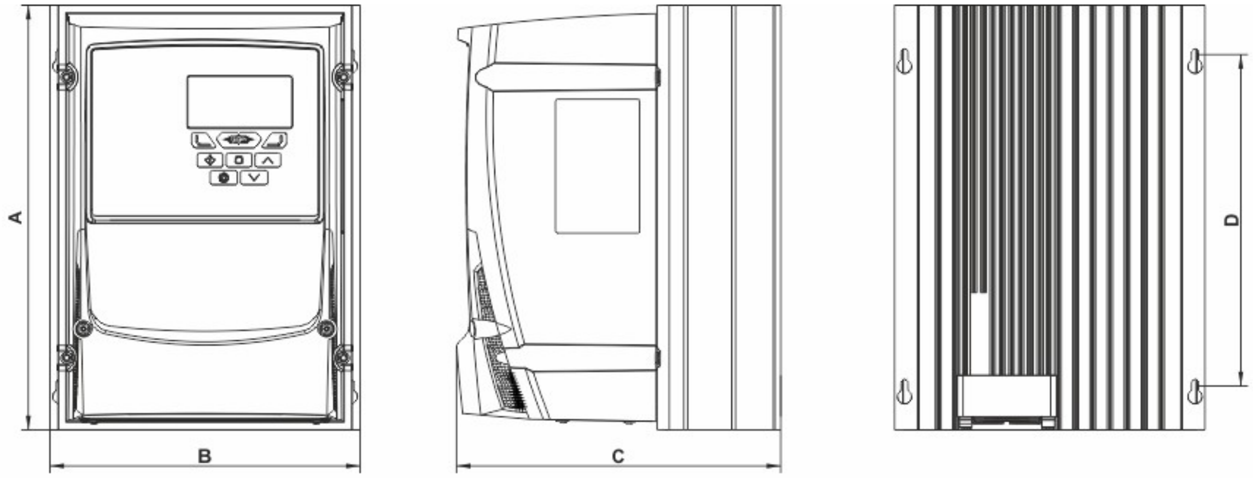
Dimensions and Connections



	A	B	C	D	E
	mm	mm	mm	mm	mm
FMY+6-4, FMY+10-4	257	188	172	200	176
FMY+14-4	257	188	196	200	176
FNY+18-4, FNY+24-4, FNY+30-4	310	211	225	252	198
FOY+39-4, FOY+46-4	360	240	260	300	227

Technical Data: FMY+14-4

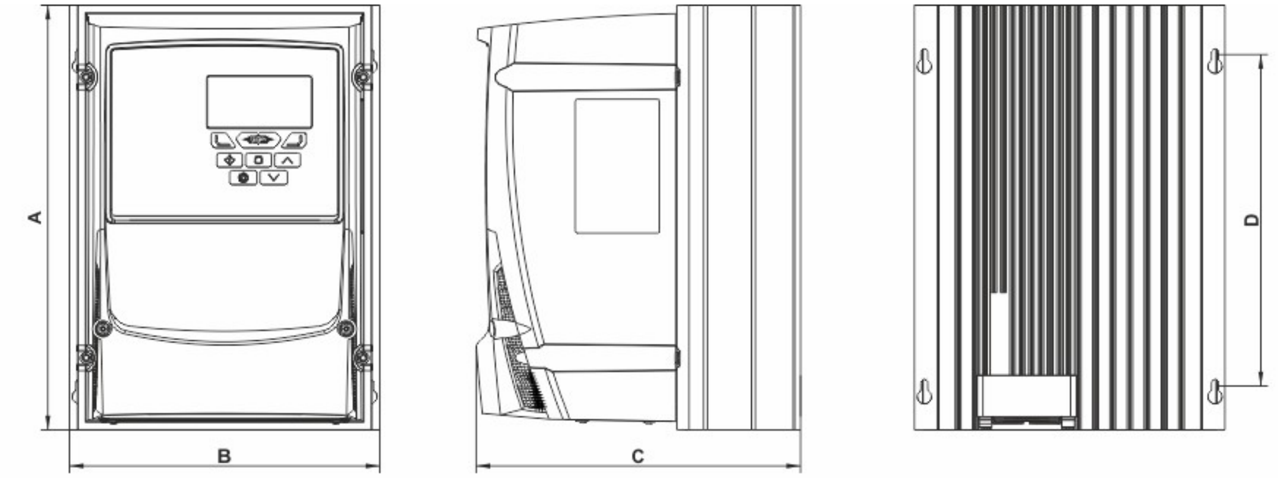
Dimensions and Connections



	A	B	C	D	E
	mm	mm	mm	mm	mm
FMY+6-4, FMY+10-4	257	188	172	200	176
FMY+14-4	257	188	196	200	176
FNY+18-4, FNY+24-4, FNY+30-4	310	211	225	252	198
FOY+39-4, FOY+46-4	360	240	260	300	227

Technical Data: FMY+6-4

Dimensions and Connections

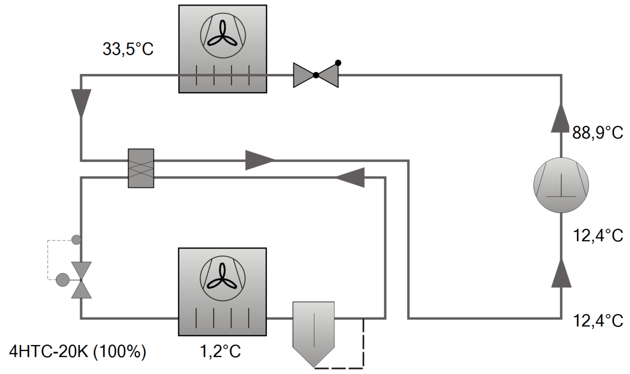


	A	B	C	D	E
	mm	mm	mm	mm	mm
FMY+6-4, FMY+10-4	257	188	172	200	176
FMY+14-4	257	188	196	200	176
FNY+18-4, FNY+24-4, FNY+30-4	310	211	225	252	198
FOY+39-4, FOY+46-4	360	240	260	300	227

Selection: Semi-hermetic Reciprocating Compressors

Input Values

Compressor model	(4HTC-20K)
Mode	Refrigeration and Air conditioning
Refrigerant	R744
Reference temperature	Dew point temp.
Evaporating SST	1,22 °C
High pressure	Auto
Gas cooling outlet	33,5 °C
Suct. gas superheat	11,20 K
Operating mode	Transcritical
Power supply	400V-3-50Hz
Capacity control	100%
Useful superheat	100%



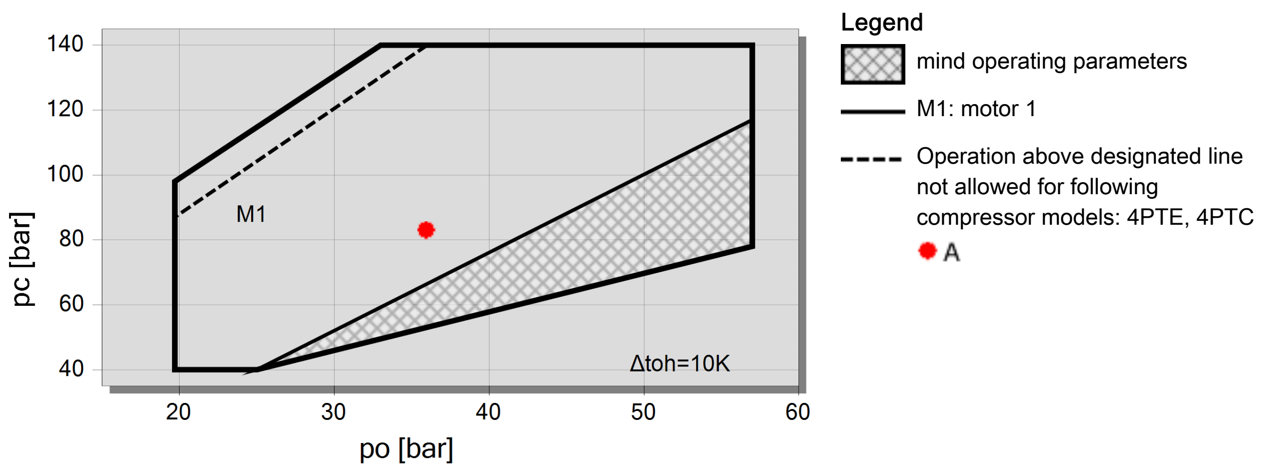
Result

Compressor	4HTC-20K-40P
Capacity steps	100%
Cooling capacity	36,5 kW
Cooling capacity *	36,5 kW
Evaporator capacity	36,5 kW
Power input	13,29 kW
Current (400V)	23,0 A
Voltage range	380-420V
Gas cooler capacity	49,8 kW
COP/EER	2,75
Mass flow	896 kg/h
Discharge gas temp. w/o cooling	88,9 °C
optimal high pressure	82,8 bar(a)

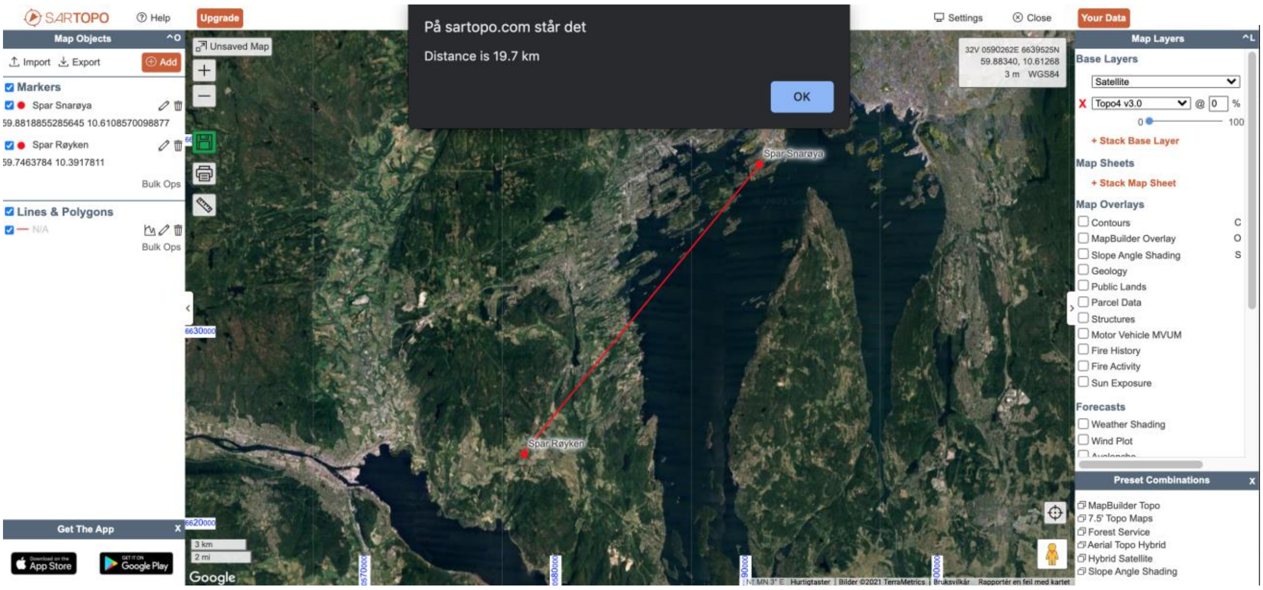
CERTIFIED PRODUCT ASERCOM

Tentative Data.
*Compressor-Performance data certified by ASERCOM (see T.Data/ Notes)
Attention, consider operating parameters. See KP-130 or consult BITZER.
*according to EN12900 (10K suction gas superheat)

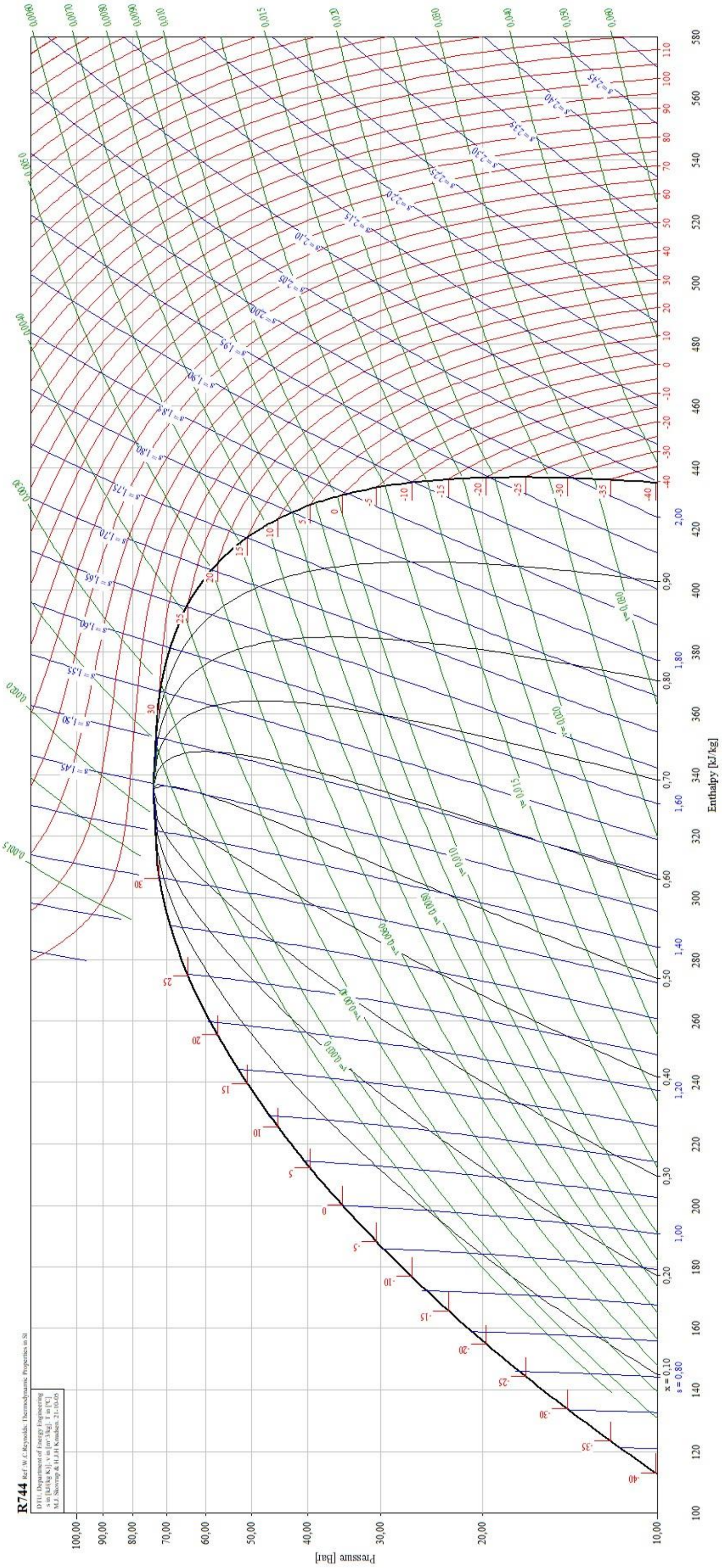
Application Limits 4HTC-20K



Appendix F


På sartopo.com står det
Distance is 19.7 km
OK
Map Objects
Markers
Spar Snarøya
59.8818855285645 10.6108570098877
Spar Røyken
59.7463784 10.3917811
Lines & Polygons
Map Layers
Base Layers
Satellite
Map Sheets
Map Overlays
Forecasts
Preset Combinations
Spar Snarøya
Spar Røyken
Google

Appendix G



Investigation of advanced supermarket refrigeration units for Scandinavian climates

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ABSTRACT

This article presents the highlights of the work done for the master thesis during spring and autum 2021. The main field of research is transcritical refrigeration systems with CO_2 as the refrigerant, but with two different system solutions. One with a low-pressure ejector system and the other one is equipped with a parallel compressor. The main part is directed towards case supermarket Spar Røyken which is the low-pressure ejector system that was implemented in autumn of 2020. This system was analyzed in two phases, as there were some challenges regarding ejector operation. In the first phase which lasted from autumn 2020 to the end of June 2021, there were done no changes from the initial setup, and the analysis showed an average pressure lift of 3.01 bar. As a result of the analysis in phase one there were made a few changes in an effort to improve the operation conditions for the ejector. In phase two of the evaluation of the low-pressure ejector system, one could see the benefits of the changes done after phase one in terms of variable stability. The average pressure lift done by the ejector during this phase were between 4.33 and 4.77 bar. Comparing the ejector operation period with a period without pressure lift from the ejector showed that the period with ejector operation demands less energy with the same run capacity on the MT-level, even though there was an elevation in the low temperature compressors run capacity due to the need to match the pressure lift from the ejector.

Spar Snarøya was also analyzed in terms of energy performance, due to few measuring points in the refrigeration system, the analysis was conducted by use of Bitzer online software. The average coefficient of performance (COP) with operation of the parallel compressor on the hottest day in 2020 was found to be 2.68.

Key words: CO_2, commercial refrigeration, parallel compression, ejector

INTRODUCTION

In a world that's experiencing climate change and global warming, the chase to slow down and prevent depletion of the ozone-layer is on. Among preventative measure for this is phasing out harmful working fluids with high global warming potential. As this is done, natural working fluids are one of the replacement solutions. The refrigerant R744 is one of the natural refrigerants that have been making advances in the market, but the transition toward natural working fluids for all sectors is challenging since each energy concept has its pros and cons. There have been made many developments in the refrigeration systems utilizing CO_2 as the refrigerant in the last years, both regarding components and system configuration. And the suppliers and end-users are continuously interested in increasing the energy efficiency of the systems.

The most widely used CO_2-system in supermarket refrigeration is the transcritical booster system, and the efficiency have been improved by introducing new components and technologies. Two of the newest system solutions are introducing an ejector to perform a pressure lift or inserting a parallel compressor which can compress the flash gas without needing to throttle it down to suction gas level for the MT-compressors, both of these solutions are used to prevent throttling losses as can be big due to the nature of the refrigerant. The parallel compressor have been proven to enhance efficiency in several studies. The ejector solution have been proven efficient, especially in warmer climates where there is a lot of transcritical operation. However, the ejector technology have not been researched a lot in Scandinavian climates where the amount of transcritical operation are significantly lower. The information regarding how well it works is limited, and it is an interesting field to look further into. The potential to reduce throttling losses is important when a more energy efficient system is desirable.

The scope of the work includes a literature review, case supermarket description, developing skills in Excel, estimating the energy demand for the system in different operation scenarios. As well as identifying suitable dates for calculating and evaluating system performance, and discussing the results based on the measured data. This should all be included in a scientific report.

TRANSCRITICAL BOOSTER SYSTEMS

The newest configurations with CO_2 as a working fluid are the transcritical booster systems as shown in the previous section. In these systems R744 covers the entire cooling and freezing need for the supermarket. The booster system gets its name from the separate compressor that elevates the pressure in the low-temperature [LT] level to the pressure at the medium-temperature [MT] level. Downstream of the LT compressor rack, the refrigerant is mixed with the CO_2 from the MT-evaporators and goes through the MT compressor, heat recovery and the gas cooler. The refrigerant is then throttled by a high-pressure control valve to a receiver. This receiver separates the flash gas and the liquid, as well as stores excess refrigerant. The flash gas is produced spontaneously when the liquid is subjected to boiling, and this happens in any refrigeration system due to the pressure drop in the two-phase region. With CO_2 as the refrigerant, the flash gas occurs due to the pressure drop in the high-pressure valve. The higher vapor percentage in transcritical systems results in a larger amount of flash gas in these types of systems compared to subcritical ones (Shecco, 2020).

Measures to increase energy efficiency

Parallel compression

The amount of flash gas in the refrigeration system is larger in transcritical operation, and this increases further with higher external temperatures. The specific cooling capacity is reduced and leads to a larger amount of refrigerant that needs to be compressed by the MT compressor. To take care of this extra amount of flash gas an extra auxiliary compressor can be installed in the system. This compressor will suck parts of the entire volume of flash gas and compress it to the pressure level for the gas cooler. By utilizing a parallel compressor, the losses due to flashing is reduced. Instead of throttling the flash gas by a flash gas by-pass valve down to the pressure level required before the MT compressor, the auxiliary compressor can compress the flash gas directly which will reduce the amount of compressor work needed. The parallel compressor [PC] will only be operating if the amount of flash gas is sufficient, to ensure that the compressor is operating under appropriate conditions. So, if the amount of flash gas is low enough, the flash gas is throttled through the FGBV as in the standard booster system mentioned in the previous section (Kauko, et al., 2016). Below a log p-h diagram for the booster system with parallel compression is shown. The energy efficiency of a transcritical booster system is raised by use of a parallel compressor due to the saved compressor work. The auxiliary compressor operates with a lower pressure lift than the MT-compressor, and thus require less energy. However, the auxiliary compressor should compress the total amount of flash gas to accomplish the best performance. Therefore, a need for a large parallel compressor would increase the total investment cost, and an analysis of cost vs energy trade-off should be executed (Gullo, et al., 2016).

Ejector

When the refrigeration system utilizes R744, one of the major contributors to losses in the system is throttling losses. This is due to the high $\frac{Cp}{\Delta h_f}$-value, and the throttling loss is particularly large in CO_2-systems operating in warm climates because of a higher exiting temperature from the gas cooler which gives higher specific enthalpy. But even in colder climates such as in Norway, the throttling losses are significant. One of the newest technologies to recover this loss is by replacing the high-pressure control valve with ejectors.

The ejector converts expansion energy with a given temperature and pressure into increased suction pressure for the compressor, which results in the ejector doing compressor work without the need for extra power consumption. The refrigerant is then discharged to the receiver. The amount of refrigerant available for the ejector depends on the available expansion work. Using a system with an ejector enables the possibility to operate the evaporators without superheat, flooded mode, which provides higher heat transfer rate and better utilization of the heat exchanger area. As a consequence, the refrigeration system can be operated with higher evaporation temperature which leads to increased overall energy efficiency (Kauko, et al., 2016).

The pressure profile of the mixing process can be seen below. Here, the gas enters from the gas cooler at a high pressure (denoted P_H), and it flows through the throat which accelerates the flow. The gas is now at supersonic speed, which creates the low pressure (P_s). The pressure P_s is lower than the suction fluid pressure (P_L) so the suction fluid now flows into the suction port. As the flows are combined in the mixing section, the pressure gradually increases. At the diffuser, the flow velocity decreases causing the pressure to rise further. When the gas leaves the ejector through the diffuser, the pressure (P_D) is higher than the pressure of the suction fluid (P_L) (Danfoss, 2018).

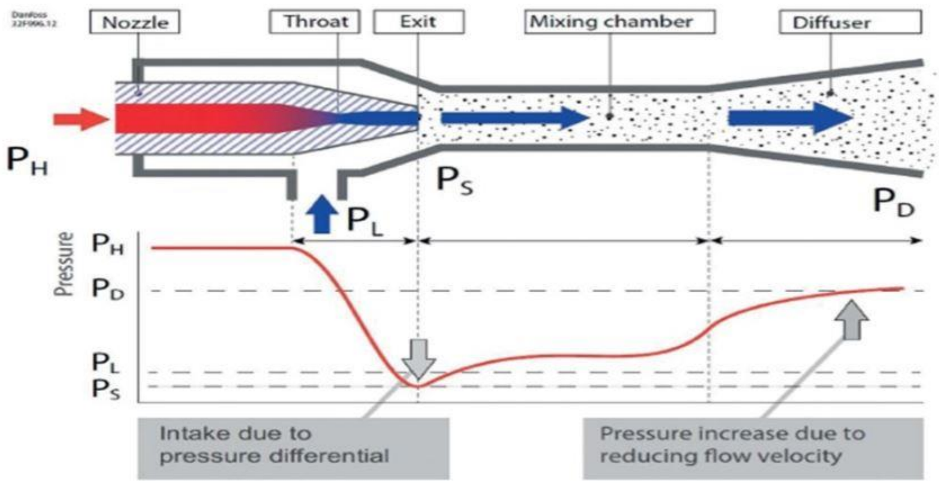


Figure 1:Pressure profile ejector (Danfoss, 2018)

SPAR SNARØYA

The refrigeration system at Spar Snarøya is a transcritcal booster system with R744 as the only refrigerant, which was implemented in 2018. This system utilizes parallel compression, according to the concept explained in the literature review. The system at Spar Snarøya has a capacity of 45kW/-2.5°C and 9.3kW/-27°C, and is charged with 140kg R744. The parallel compressor is activated by the opening degree of the flash gas bypass valve. When the FGBV opening degree is 26% and above for more than 280 seconds, the parallel compressor is turned on. So 26% opening degree of the FGBV correlates to the lowest capacity of the compressor. When the parallel compressor is activated it is internally regulated directly in the frequency inverter, which is suction pressure/receiver pressure regulated. In the low temperature circuit, there are two compressors, where one of them is a VSD-compressor.

Spar Snarøya have two different operation modes, depending on whether the parallel compressor is running or not. In operation mode one the parallel compressor is running.

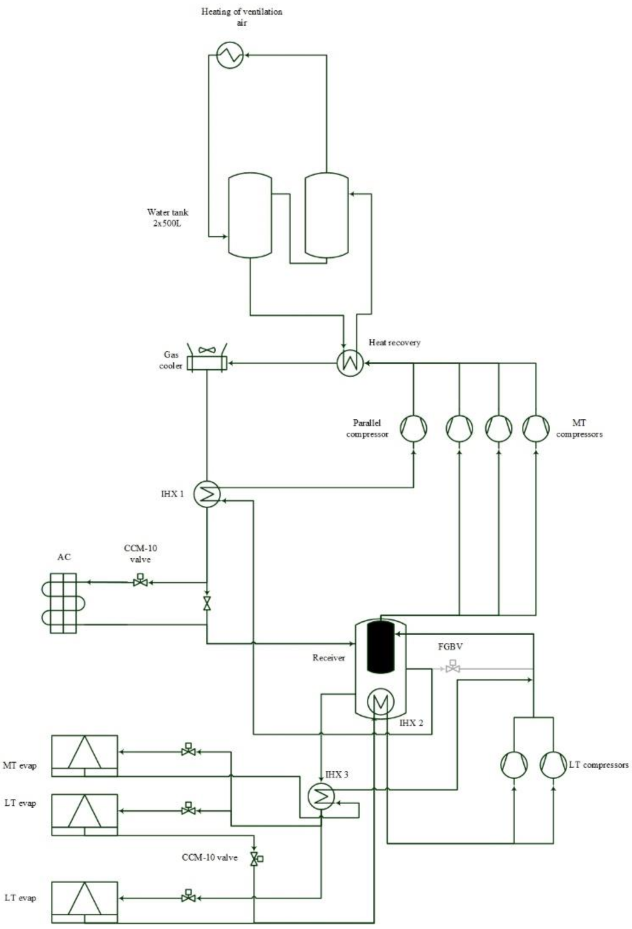


Figure 6: Schematic of Spar Snarøya, operation mode one

The flash gas bypass valve closes, and the gas is supplied to the parallel compressor via IHX 1, which exchanges heat between the flow from the gas cooler and the flash gas from the receiver. The LT-evaporators are supplied refrigerant from the receiver, the refrigerant then travels through IHX 2 in the outer shell of the receiver before it enters the LT-compressors. The refrigerant is then supplied to the inner shell of the receiver. The MT-evaporators are supplied with refrigerant from the outer shell of the receiver through IHX 3, where there is heat exchange between the supply flow to the MT-evaporators and the refrigerant leaving the MT-evaporators. The CO_2 is then expanded to the correct evaporation pressure. The two coolers that have lower evaporation pressure than the rest of the MT-evaporators is throttled to LT-level. The other part of the MT refrigerant exits the evaporators, goes through IHX 2, mixes with the flow from the LT-compressors and is supplied to the inner shell of the receiver. The MT-compressors is supplied from the inner shell of the receiver where the pressure is elevated, the refrigerant then travels through the heat recovery heat exchanger to the gas cooler where the temperature is lowered. Downstream of the gas cooler is IHX 1, where the flow transfers heat to the supply refrigerant to the parallel compressor. Part of the refrigerant travels through the AC loop if this valve is opened before it is supplied to the outer shell of the receiver.

COP from Bitzer simulation during operation mode one and p-h diagram

Figure 2 is graphic displaying the different COP-values from the Bitzer software with the hourly average values as input. The COP ranges from 2.34 to 3.13.

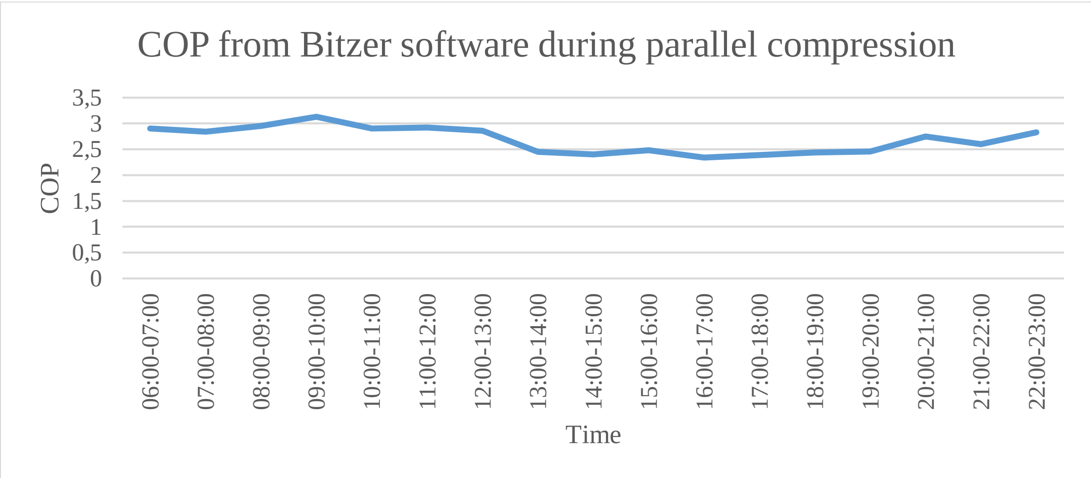


Figure 2: COP from Bitzer software during parallel compression

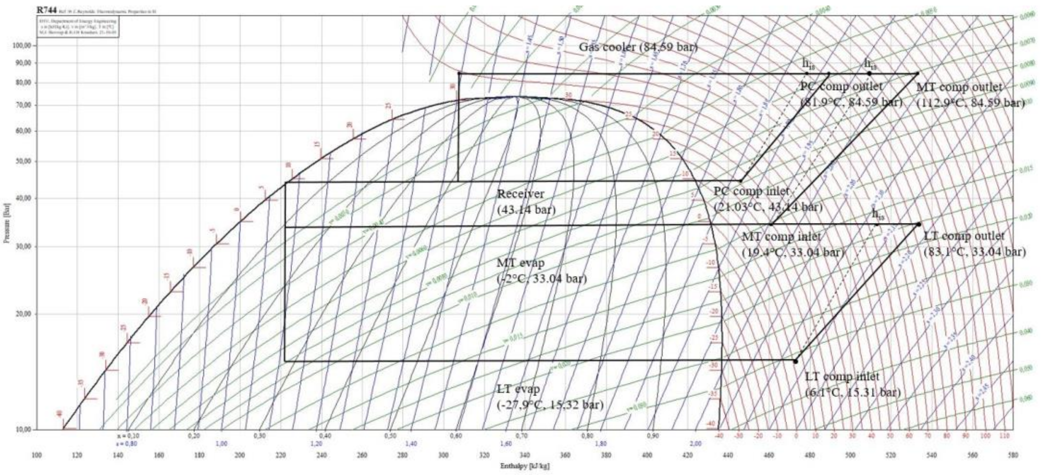
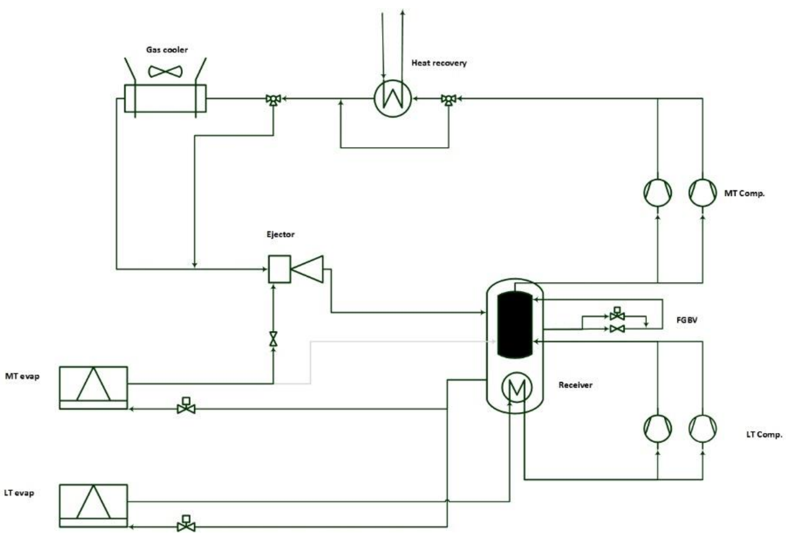


Figure 3: P-h diagram during parallel compression

SPAR RØYKEN

This is a system which had its start-up in the autumn of 2020. The system was refurbished by Kelvin AS, originally it was a subcritical refrigeration system with R744 as the refrigerant from 2008. The rehabilitation turned the system into a transcritical refrigeration system, with a low pressure ejector and a gas cooler. The system at Spar Røyken have a capacity of 56kW for cooling and 13kW for freezing and is charged with 120kg R744.



With ejector operation the mass flow from the LT-evaporators travel from the evaporators through an internal heat exchanger, via the LT-compressors and is deposited in the inner tank in the receiver. The flow exiting the MT-evaporators is now the suction fluid for the ejector, it is mixed with the flow from the gas cooler which is the motive fluid. The flow is then mixed, and travels through the diffuser part of the ejector. It is then fed to the outer shell of the tank. When the ejector function is active, the parallel valve to the flash gas bypass valve is also open as an addition to the FGBV. And the refrigerant passes through these two valves to the inner tank in the receiver.

Results phase one

Figure 4 shows the development of the chosen variables in a 60 minute period. When voltage input is shown as 1, the ejector is running, and 0 when the ejector is off. The ejector was on from 10:25-10:58.

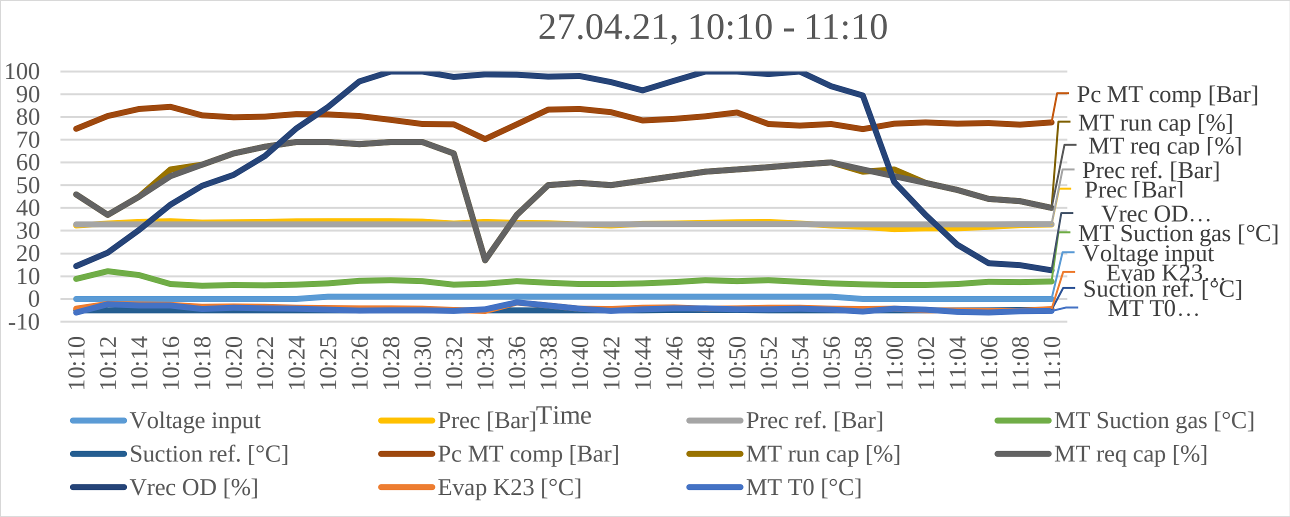


Figure 4: Variables phase one, ejector operation

As a result of the runtime of the ejector, a certain pressure and temperature lift was obtained, the pressure lift ranges from 0.6 bar to 4.6 bar, with an average of 3.01 bar. The temperature lift has its lowest value as 1.3 °C and maximum value as 4.7 °C, with an average of 3.57 °C temperature lift.

Results phase two

In this period the ejector runs from 13:06 to 14:26, a total of 80 minutes. The 90 minute period shown in the graphics below includes the same variables as used in phase one.

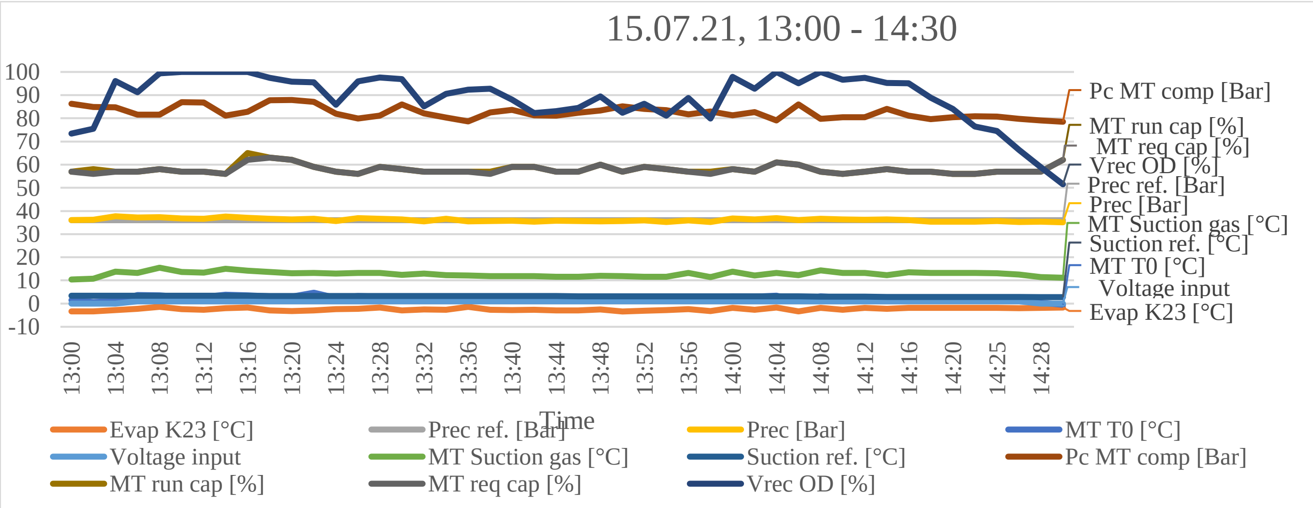


Figure 5: Variables phase two, ejector operation

In this phase, the pressure lift ranges from 5.6 bar as the highest value to 3.1 bar as the lowest, with an average pressure lift of 4.41 bar. The temperature lift has its maximum value as 5.9 °C, and minimum value as 3.5 °C, with an average temperature lift of 4.91 °C. Figure 6 shows the p-h diagram with average values from the period 13:30 to 14:00, during ejector operation.

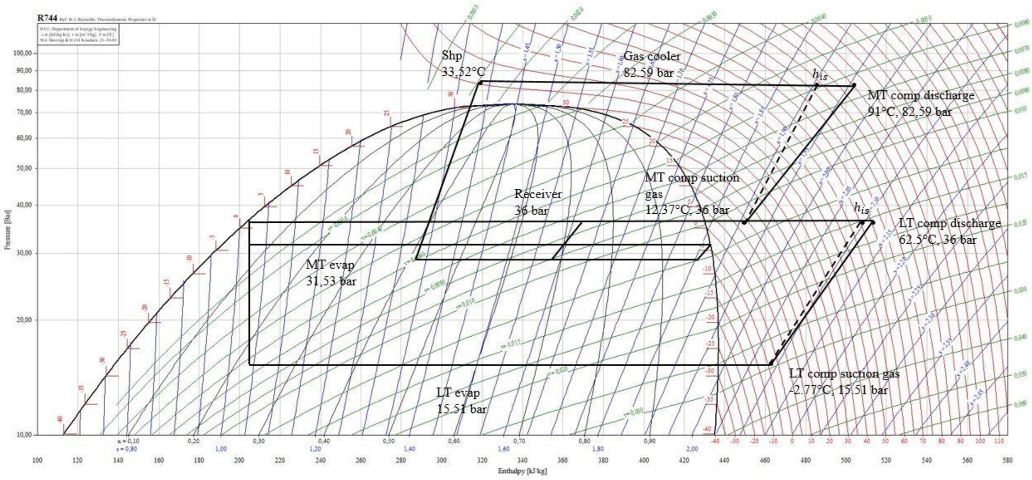


Figure 6: P-h diagram during ejector operation

Comparison ejector on/off

Comparison between two periods the same day, when the ejector is on and off.

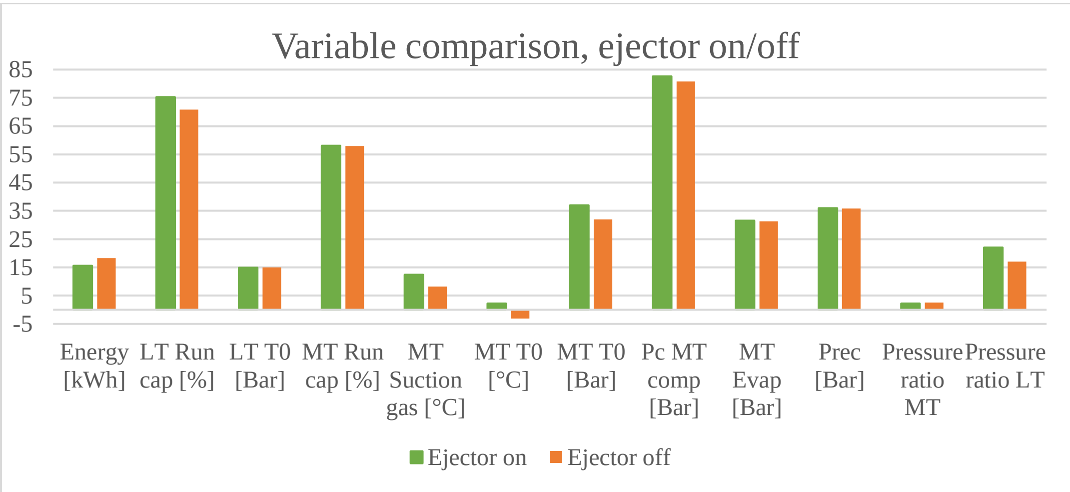


Figure 7: Variable comparison, ejector on/off

As one can see demonstrated, the energy demand is somewhat higher for the period where the ejector is not performing a pressure lift, the energy need is 15.6kW when the ejector is running, compared to 18.3 kW when the ejector is off. The MT suction gas is higher with the ejector running, 12.37°C compared to 8.19 °C when the ejector function is off. One also notes the shift in MT T0 as a result of the change in reference point between the operation modes. The discharge pressure is higher during ejector operation with 82.59 bar compared to 80.75 bar when the ejector is off, which gives a pressure lift of 45.63 bar when the ejector is on and 48.71 bar when it is off. The pressure ratio over MT compressor racks is 2.23 when the ejector is running, and 2.52 when the ejector is in operation mode one. LT compressor run capacity is slightly elevated in ejector operation compared to when the ejector is off, 75.31 % in contrast to 70.8. The LT compressor rack also have a higher pressure ratio when the ejector is in operation, 22.03 bar, compared to 17.06 when the ejector is off.

Discussion and conclusion

Ideally the system COP should have been calculated by values given from sensors in the refrigeration system, but due to the missing values surrounding the parallel compressor this was not possible. The trend in the COP-estimation at Spar Snarøya shows promise, however, it is hard to make a conclusion based on the presented data. The results are estimated in several aspects, and this increases the degree of uncertainty. Further studies have to be conducted to make a definite conclusion regarding the system performance.

One of the major things in phase one for Spar Røyken was that it takes the compressor racks 10 minutes to respond to the ejector being turned on by changing the requested capacity. As a result of this a drop in suction pressure is observed. With the slow response from the MT compressor rack, the receiver pressure also declines without response. The receiver pressure is also set lower to create a more flash gas and in turn a larger opening degree in the flash gas bypass valve. The receiver pressure declines, and the MT compressor rack isn't responding fast enough. In phase two the MT-compressors responds faster than in phase one which causes the system to operate with a higher degree of running stability. The decline in receiver pressure is also absent in this phase as a result of this. One can also note a change in the variable called MT T0 due to reference changes from phase one. In phase one MT T0 refers to evaporator temperature and pressure, but in this phase it is changed during ejector operation. The reference point for MT T0 is now receiver pressure and temperature. The evaporation temperature for unit K23 is also more stable during ejector operation in this phase, there are fewer high spikes in temperature values. A higher pressure lift is also noted during phase two, and the difference between the minimum and maximum pressure lift is also smaller. The average pressure lift is increased by at least 1.32 bar during ejector operation. The decline in MT request capacity is not observed as expected, a slight decline can be seen when the ejector is running but not the amount as expected. However there is an increase in LT run capacity because the LT compressors have to match the pressure lift done by the ejector. So the decline in the requested capacity for the MT compressor rack might be masked by the increased demand on the LT-level. The energy demand is a bit higher when the ejector isn't running, due to the pressure lift done by the ejector. The LT run capacity is slightly higher in the period where the ejector is operation, due to need to match the pressure lift done by the ejector as the refrigerant streams are being fed to the receiver. The change in reference point for MT T0 is also noted in the figure, showing that the reference point shifts from evaporation temperature to receiver temperature during ejector mode. The suction gas temperature to the MT compressors are also elevated as a result of the pressure lift done by the ejector. The discharge pressure is slightly higher when the ejector is in operation, but due to the higher suction pressure the pressure ratio across the compressor is lower when the system is operation in ejector mode. This is contributor to the lower energy demand when the ejector is lifting the pressure. One can also note a higher pressure ratio across the LT compressor, but this doesn't equalize the lower energy demand on MT-level.

As a conclusion from the collected data, the ejector system can be beneficial in Scandinavian climates. The ambient temperatures in the eastern part of Norway are high enough to facilitate transcritical operation of a refrigeration system with CO_2 as the refrigerant. With a well dimensioned system one can utilize the ejector to perform a pressure lift to lower the energy demand. If the ambient temperature is one of the variable that influence whether the ejector solution is appropriate for Nordic climates, there are several days with transcritical operation hours. These are viewed as potential operation hours for the ejector, but this isn't confirmed as the ejector only operated once during the summer period of 2021. However, further studies have to be conducted to determine how beneficial the system solution can be, preferably from a larger data pool.

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Investigation of advanced supermarket refrigeration units

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