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Design and implementation of an indirect refrigeration system for laboratory testing

Master's thesis
Aalto University
School of Engineering
Department of Energy Technology

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Tiivistelmä

Kylmä-alan globaalissa liiketoiminnassa uusia tuotteita ja konsepteja kehitetään jatkuvasti. Nykyajan haasteet ja mahdollisuudet vievät kehitystä kohti ympäristöystävällisempää ja älykästä kylmätekniikkaa, pienemmillä tilavaatimuksilla ja asennuskuluilla. Epäsuorilla kylmäjärjestelmillä on omat etunsa, kuten hyvin pienet kylmäaine lataukset verrattuna suoriin kylmäjärjestelmiin, varsinkin jos putkistoilta vaaditaan pituutta ja kylmäkohteita on monta. Viessmann ESyCool on edistynyt ja kattava energiajärjestelmä ruokakauppoihin. ESyCool järjestelmässä hyödynnetään epäsuoraa kylmätekniikkaa.

Työn tarkoituksena oli suunnitella ja toteuttaa epäsuora jäähdytysjärjestelmä laboratoriotestauksen tarpeisiin. Testattavat ovat erilaisia epäsuoralle jäähdytykselle suunniteltuja tuotteita ja laitteita, joista osa sisältyy myös ESyCool-konseptiin. Työssä esitellään myös aihepiiriin liittyvää kylmä- ja jäähdytystekniikkaa ja teoriaa, mukaan lukien kunnollinen katsaus kompressori-, CO_2- ja epäsuoriin jäähdytysjärjestelmiin.

Kaiken kaikkiaan projekti voidaan nähdä onnistuneena ja toimiva kylmälaitos halutuilla ominaisuuksilla saatiin aikaan. Projektistamme valmistuneella kylmälaitoksella voidaan toteuttaa laadukasta epäsuoraa jäähdytystä vuosikymmeniä, kunhan jäljellä olevat pienet työt, huolto ja laitoksen operointi hoidetaan asiaankuuluvasti. Tämän tutkimuksen perusteella kylmäteholtaan noin 10-50kW:ssa samantyyppisissä epäsuorissa kylmälaitosprojekteissa voidaan päästä melko hyviin tuloksiin liitteiden 4-7 ja 10, ja tässä työssä esiteltyjen tietojen mukaisella suunnittelulla. Toteutuksen osalta voidaan päästä selvästi lyhyempiin asennusaikoihin ja hieman parempaan lopputulokseen soveltamalla tässä työssä mainittuja ja läpi käytyjä keinoja. Tätä projektia ja siitä saatuja oppeja voidaan käyttää jatkossa benchmarkkina samantyyppisissä energiantuotantolaitosprojekteissa, varsinkin kylmä-alan projekteissa.

Avainsanat Kylmäjärjestelmät, Välillinen jäähdytys, Epäsuorat jäähdytysjärjestelmät, Energiaprojektit

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Abstract of master's thesis

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Abstract

New products and concepts are being constantly developed in worldwide refrigeration business. Modern challenges and opportunities drive some development towards more environmentally friendly and smart refrigeration, with smaller demands for space and installation expenses. Indirect refrigeration systems have their own advantages, such as possibility of very small refrigerant charges compared to direct systems, especially with long piping and many cooling targets. Viessmann ESyCool is an advanced and comprehensive energy system for food retail stores. ESyCool system utilizes indirect refrigeration.

Aim of this work is to design and implement an indirect refrigeration system for laboratory testing of various indirect refrigeration products and equipment, including some from ESyCool. Some relevant refrigeration technology and theory are also introduced, including a proper review of vapor compression, CO2 and indirect refrigeration systems.

All in all, this project was successful and a working system with wanted properties was accomplished. Decades of high quality indirect refrigeration can be achieved with the project's refrigeration plant if recommended adjustments, operation and maintenance are carried out. Based on this study, for around 10-50kW similar kind of indirect refrigeration plant projects, considerably good results can be achieved with design according to appendixes 4-7 and 10, and details described in this work. For implementation, significantly shorter installation times and a bit higher quality can be reached with means mentioned and discussed in this work. This project and its learnings can be used as a benchmark for similar custom made energy production plant projects in future, especially in refrigeration branch.

Keywords Refrigeration systems, Secondary cooling, Indirect cooling systems, Energy projects

Acknowledgements

This master's thesis is composed based on a work made for Viessmann Refrigeration Systems Oy between March and November 2019. I want to thank the company for a unique opportunity to carry out my master's thesis in such an interesting design and implementation project. The topic was offered from the company based on their actual needs for a new refrigeration testing capabilities, which was exciting and offered me a lot of valuable know-how and experience.

Viessmann's effort for wellbeing and team spirit of employees was respectable. I got to participate in couple of nice and educational working trips to Europe. The annual Viessmann's Football tournament in Allendorf was a great experience and offered us a good chance to get to know our colleagues in Team Finland and meet some fellow abroad colleagues as well.

I want to thank my advisor D. Sc. Petri Majander and my supervisor Professor Annukka Santasalo-Aarnio for the entire project and the master's thesis. I would also like to thank Jari Virtanen for offering some valuable consultation throughout this project, Maria Karvonen for managing my practical matters in the company smoothly and all the colleagues from Viessmann and co-operating companies I worked with during the project. Thanks to Jyrki Kaarikivi, Ari Salmela, Timo Kontio, Jarno Mäkinen, Kimmo Heinävaara and the entire team that worked on installing and commissioning of the system.

Special thanks to Aalto University, Otaniemi and all the people I spend time with during my studies. We have fantastic education opportunities in Finland, and one of the best and most state supported student lives in the whole World. Working life is welcoming for young professionals in Finland, and thanks belong also to my previous employer, AQVA Finland Oy, times there have proved to be valuable for this project's challenges too.

Last but not least, I would like to thank my Family and Friends. Special thanks to my Dad, Matti Koivisto for all the support and encouragement in the field of science and technology.

Porvoo, 22.9.2019

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Abbreviations

ASHRAE	American Society of Heating, Refrigerating and Air-conditioning Engineers
CO2	Carbon Dioxide
COP	Coefficient Of Performance
EER	Energy Efficiency Ratio
ESyCool	Viessmann's Energy System for Cooling applications. An advanced energy system for food retail business customers, market stores for example.
EU	European Union
GUI	Graphical User Interface
GHG	Greenhouse gases
GWP	Global Warming Potential
HFC	Hydrofluorocarbons (often referring to HFC-based refrigerants)
HFO	Hydrofluoroolefins (often referring to HFO-based refrigerants)
HVAC	Heating, Ventilation and Air Conditioning
IIR	International Institute of Refrigeration
LT	Low temperature (for refrigeration fixtures, temperatures a bit over 0°C)
MT	Mid temperature (for refrigeration fixtures, freezer temperatures)
NAM	North America
ODP	Ozone Depletion Potential
PCO	Plug in Control Operated (for example a refrigeration unit)
PFC	Perfluorocarbons (often referring to PFC-based refrigerants)
RCO	Remote Control Operated (for example a refrigeration unit)
UK	United Kingdom
VCRS	Vapor Compression Refrigeration System. Also referred to as Compressor heat pump-based refrigeration system or shortly Compressor refrigeration system.
VTT	Teknologian Tutkimuskeskus VTT Oy. VTT Technical Research Centre of Finland Ltd.

Symbols

C_p	Specific heat capacity in constant pressure	J/kgK
h	Specific enthalpy	J/kg
L	Latent heat, for vaporization or melting for example	J/kg
$\dot{m}$	Mass Flow	kg/s
M	Molar mass	kg/mol
p	Pressure	Pa, bar
P	Power	W
Q , Φ	Heat power. Used for both heating and cooling	W
RH	Relative humidity	% , -
T	Temperature	K, °C
x	Absolute Air humidity	kg_v/kg_{da}

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1. Introduction

Refrigeration systems are cooling systems capable of lowering temperatures below ambient temperatures, or below temperatures introduced to the cooled material. Refrigeration is defined in a following way by ASHRAE (2019):

“ (1) cooling of a space, substance or system to lower and/or maintain its temperature below the ambient one (removed heat is rejected at a higher temperature)
(2) artificial cooling. ”

For example cooling with cold seawater in heat exchangers is not refrigeration. It works just as long as seawater flow remains colder than the flow being cooled. Arrangement cannot supply temperatures below ambient temperature or introduced material, seawater in this case. This kind of cooling is categorized in free cooling technologies, for example in VTT’s cooling technology-themed report by Laitinen et al. (2016).

By far, the most used refrigeration technology today is vapor compression refrigeration technology, with share of at least 90% of all cases. (Laitinen et al, 2016) (Siemens, 2017). Such refrigeration is made with vapor compression cycle, and system in general is called Vapor Compression Refrigeration System (VCRS). VCRS is practically a compressor heat pump moving heat from refrigeration targets to the heat sink or heat recovery. “Compressor/compression refrigeration system” and “Compressor heat pump-based refrigeration system” are names that can be used for essentially the same system and technology.

1.1 Refrigeration branch, regulation and upcoming trends

Refrigeration is a big, worldwide business. International Institute of Refrigeration (IIR) has published some interesting data from refrigeration branch in 2015 and 2019, giving a good view of refrigeration sectors size, different applications and latest development. Selected statistics collected from aforementioned publications are collected in Table 1. Refrigeration demand is growing globally, especially in many still developing countries. (University of Birmingham, 2017, p.7) Worldwide growth can be seen in table 1 statistics, all selected numbers have grown clearly since 2015. With globally raising cooling demand, energy efficient and more climate friendly technologies and refrigeration products are necessary for bringing World’s CO2-equivalent emissions into an acceptable level.

Table 1. Selected refrigeration-related statistics from IR's Informatory notes on 2015 and 2019.

	Operative refrigeration, air condition and heat pump systems	Employed people in refrigeration sector	Sales of refrigeration, air condition and heat pump systems per year (USD)	Share of refrigeration sector in overall electricity usage (%)
November 2015	3 billion	12 million	300 billion	17
June 2019	5 billion	15 million	500 billion	20

All numbers are Worldwide. 1 billion = 10^9.
1 USD = 0,8917 EUR (Rate 18.7.2019 16.45) (Kauppalehti, 2019)
2015 numbers are from (IRR, 2015) 2019 numbers are from (IRR, 2019)

Current implementation rate of refrigeration varies a lot between different countries. In 2015, less than 4% of India's fresh produced foodstuff was transported under low-temperature preservation. Same number of UK was over 90% (IIR, 2015, p.7). In India's case for example, future economic and technological growth could also implement a massive amount of new installed refrigeration capacity. In addition to India, there are many countries is which the installed capacity may increase in the future. Whether this new capacity is modern, efficient and environmentally friendly can make a notable impact on the average performance and sustainability on refrigeration worldwide.

Refrigeration sector consumes globally around 20% of all electricity these days, number includes air conditioning. Supermarket refrigeration alone consumes about 3% of all electricity in developed countries (University of Birmingham, 2017, p.11). Energy consumption of refrigeration systems plays also a big role in the amount of sectors arising GHG (greenhouse gas) emissions. University of Birmingham (2017, p.3) estimates, that roughly 75% of worldwide cooling's GHG emissions are generally seen to be result of energy consumption. With corresponding share of cooling's 25%, refrigerant leakages are identified as another big GHG source, and these emissions are centered upon relatively small group of actors. From this can be inferred, that most retailers for example have practically no refrigerant leakages from their system, but when such event happens, quite a big GHG emissions follow. How refrigeration system manufacturing was taken into account in statistics is a good question, since accurate calculation of material- and early and late GHG lifecycle effects can be challenging. The whole lifecycle of product has to be taken into account, to prevent for example moving GHG emissions to some phase in product lifecycle that is simply excluded in calculations (VTT, 2017).

Various decisions and regulations have been entering food retail refrigeration business in decent decades all over the World. Some essential decisions governing food retail refrigeration since 1990 are presented in figure 1. Article 5 refers to over 140 still developing countries worldwide (IIR, 2017). NAM stands for North America. One particularly influential event in refrigeration business has been F-gas regulation in EU-area, which is to decrease HFC (hydrofluorocarbons) -based refrigerant supply by almost 80% by 2030 (University of Birmingham, 2017, p.3, p.8). Since 1990, food retail market's refrigeration systems have also experienced clear changes in what it comes to technology. Market shares of global food retail refrigeration equipment types are presented in figure 2. Use of remote operated CO2 refrigeration systems has increased steadfastly since around 2010, replacing HFC-based refrigerant operated remote systems. Propane still has its own grand demand in plugin refrigeration systems, even though some semi-plugin water loop products have been entering the market since later 2010's.

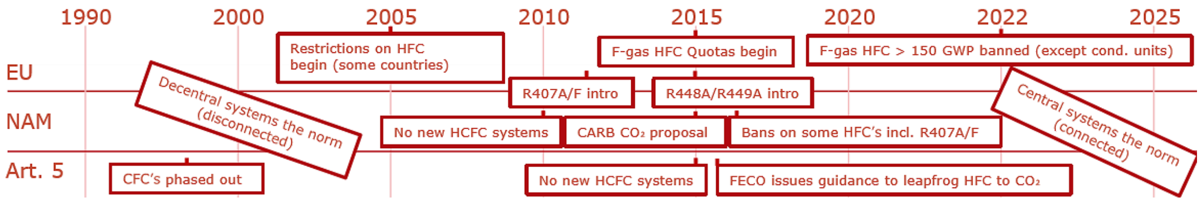


Figure 1. Some essential decisions governing food retail refrigeration since 1990. (Danfoss, 2018A)

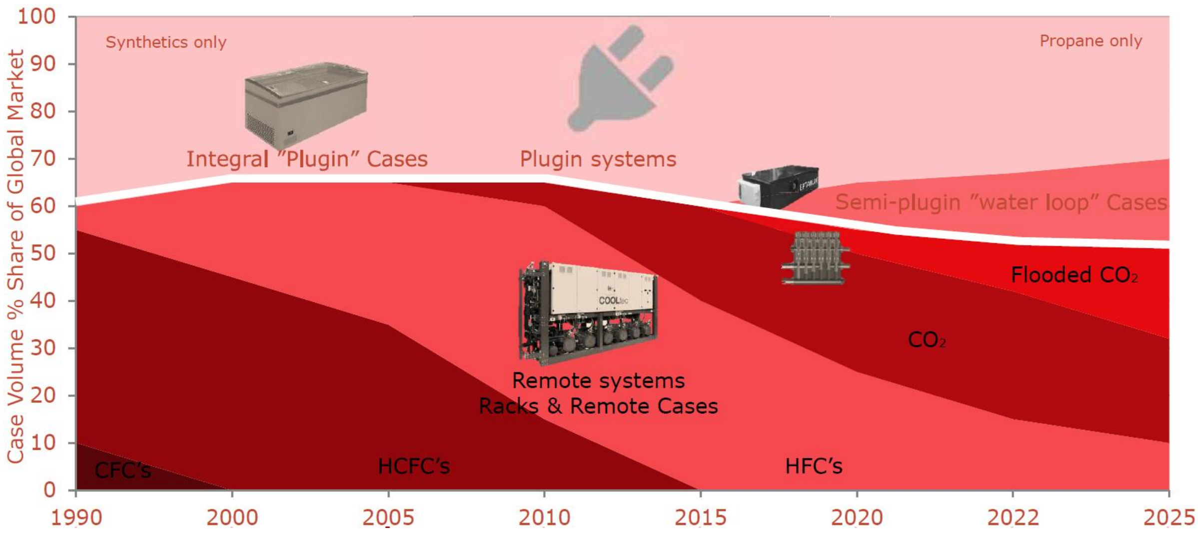


Figure 2. Market shares of global food retail refrigeration equipment types by case volume (%). (Danfoss, 2018A)

Some present and upcoming trends in global food retail business according to Danfoss (2019) are presented in figure 3. Regulations and environmental aspects give natural refrigerants some competitive advances and hinder the use of halogenated refrigerants. New installations with natural refrigerants are entering the food retail markets. At the moment use of CO2 in refrigeration systems is increasing quite rapidly. CO2 can be used in refrigeration systems as refrigerant, or as secondary coolant as well. Marketing material and introduction about advantageous properties of CO2 as a secondary coolant can be found from for example Danfoss (2015). Increasing F-gas prices have an additional effect for decreasing share of HFC-based systems on the markets and turning some new investment choices for favor of CO2 or propane-based refrigeration systems. Two interesting future trends are shortage of trained refrigeration service technicians and increasing expenses spent on services. These trends make some new opportunities in refrigeration business attractive, for example to be able to execute some installation and maintenance jobs with regular plumbing professionals, without having to give up on good quality of work. E-commerce, urbanization and increasing real estate prices, especially in cities and urban areas set new situations and challenges for space use in food retail business. Danfoss (2019) estimates, that this leads to smaller store footprints and therefore demand for more compact food retail refrigeration systems. Connectivity to intelligent control- and operation networks and programs is also identified as a future trend, as technology digitalizes. Wärtsilä, a huge Finnish player in smart technologies and complete lifecycle solutions for the marine and energy markets, defines data and digitalization as one of its main drivers in energy markets in their 2018's annual report (Wärtsilä, 2018, pp.29-30). Plugin and semi-plugin refrigeration systems are estimated to grow slightly in share in future decades.

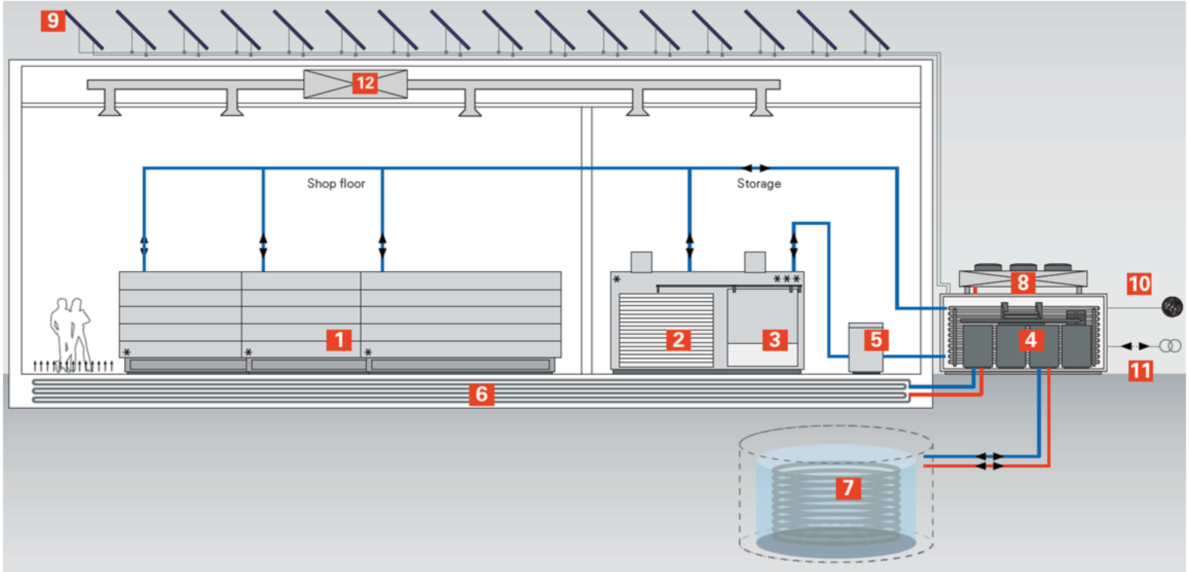
Figure 3. Some present and upcoming trends in global food retail refrigeration business. (Danfoss, 2019) Red area describes trends in demand and business sector, and gray area describes trends in new applications coming to markets.

One technique solving many current and upcoming challenges is to use an indirect refrigeration system. Technology enables a lot smaller refrigerant charges and can limit refrigeration machinery to its own compact technical space, so that cooling displays in the actual sales area can be cooled with low pressure secondary coolant circuits. In Finland, secondary cooling has been used a long time in certain applications, such as artificial ice rinks, air conditioning and foodstuff and process industries. Suomen Kylmäyhdistys Ry (2019, p.2) has freshly estimated that installed capacity of indirect systems will grow in the future, because of tightening regulations for some refrigerants and flammability of some new refrigerants.

1.2 Motivation and ESyCool concept

As environmental friendliness, economical aspects, performance and safety are constantly being developed in the use of technology worldwide, refrigeration technologies are also taking part in the process. One particular solution to refrigeration industry's both modern and traditional challenges are indirect refrigeration systems. IIR (2018, pp.1-3) has introduced some advancements in new refrigeration system technologies in its fresh publication; "Advancements in supermarket refrigeration". In addition to higher energy efficiency, smaller refrigerant charges and refrigerant leakages were found to be key enhancements on achieving lower CO_2 emissions from refrigeration. From publication's numbers can be concluded, that potential for refrigerant charge and leakage reduction is significant: In many existing supermarkets, refrigerant charge can be up to 3000kg, and even 30% of total charge volume can leak out annually. As mentioned in the end of previous chapter, indirect systems can reduce the refrigerant charge into a fraction compared to direct systems. Refrigeration machinery, refrigerant and high pressure levels can therefore be limited to technical room, while in most of the cooling system's area only some low pressure heat transfer fluid is circulated to cool final refrigeration targets. In addition to improved safety, smaller refrigerant charge can help companies' refrigeration systems comply better with new regulations concerning maximum refrigerant charges for some traditional refrigerants. Indirect refrigeration systems would also allow cheaper and more convenient installation and maintenance, while regular plumbing professional could install all devices except the refrigeration machinery room demanding refrigeration technician. Potential segment for such technology is very large and competitive, for example food retail, restaurants, grocery production and storage.

One remarkable concept utilizing indirect refrigeration technology is Viessmann ESyCool (Energy System for Cooling applications), an advanced energy system for food retail business customers. ESyCool system diagram using discount store as an example is presented in figure 4. ESyCool can take care of the food retail store's refrigeration in all temperature levels. ESyCool system cools fixtures in refrigerator temperatures (above 0 °C) with heat transfer fluid, changing sensible heat only. For freezer temperatures there is an additional CO_2 VCRS freezer unit. Therefore in ESyCool concept, indirect cooling carried out entirely with heat transfer fluid cooled by Vitocal heat pumps is applied only for refrigerator temperature fixtures. Cooling to the system is produced with ESyCool heat pump unit (And CO_2 freezer unit). There are 2 different versions of ESyCool heat pump units available. ESyCool Classic with synthetic R410A refrigerant, and ESyCool Green with natural R290 refrigerant (propane). In ESyCool Classic this heat pump unit contains four Vitocal heat pumps in a rack, put in a weatherproof casing. On proximity of Vitocal heat pump rack, there is a dry cooler rejecting heat into outdoor ambient air. Store can be heated with concrete core activation on a floor plate. Concrete core activation can also be used for cooling of the building on hot weather. Ventilation system is optional. Vitovolt photovoltaic system can be also optionally chosen to produce renewable electricity from the sunlight striking the store roof. ESyCool is connected to public electric power grid for power supply. ESyCool has connections to internet, GUI (graphical user interfaces), smart devices and intelligent local- and remote control panels.



Refrigeration consumers

1. TectoDeck multideck cabinets (shop floor)
2. TectoCell coldroom (storage)
3. TectoCell freezer room (storage)

Energy centre

4. ESyCool unit in weatherproof casing
5. CO_2 freezer unit

Components

6. Concrete core activation
7. Ice energy store
8. Dry cooler
9. Vitovolt photovoltaic system
10. Internet (EDI, Meteolink)
11. Public power grid
12. Ventilation system (optional)

Figure 4. ESyCool system diagram using discount store as an example (Viessmann, 2017)

Ice energy storage comes with some, widely equipped versions of ESyCool system. It can act as heat- or cold storage depending on a season and weather. In winter time, when ambient outdoor air is below -10 °C, water in ice storage tank can freeze in atmospheric pressure (0°C), releasing a lot of heat per mass. Heat pump can use this heat easier than below -10°C outdoor air for heating of store via concrete core activation. In summer time frozen water in the tank can be used as a cold storage supplying some additional cold during hot weather. Ice storage has a certain capacity of heat or cold supply, in vicinity of 1MWh per complete phase change of one standard sized 10m^3 water/ice energy store vessel. After phase change has fully occurred, some water has to be frozen or molten to be able to get some heat or cold supply from storage again. Use of Vitovolt photovoltaic system enhances usability of ice storage even further, because it gives us a chance to use solar electricity during a daytime to freeze the ice storage, and this achieved cold reserve can be used to keep cooling targets cold during the nighttime.

Some key benefits of ESyCool system are presented in figure 5. Notable reserve for customization according to customers' needs is there, for example photovoltaic system, ice energy storage and ventilation are optional equipment, and can be supplied if seen to be beneficial in customer's store facilities. ESyCool system is enhanced by Viessmann's multibusiness experience. Viessmann is a major player in heat pump business, and a big supplier with wide range of products in heating, industrial and refrigeration applications. Vitocal heat pump unit consist of for example four Vitocal 300-G heat pumps in a cascade, enabling 4*12kW of refrigeration capacity. Energy efficiency class of ESyCool racks is A++. This energy efficiency is based on EU regulation number 811/2013 and set under certain use and circumstances. (Viessmann, 2017) (European Commission, 2013) There are many models of Vitocal 300-G produced by Viessmann, which all can be used for both heating- and cooling applications, in for example food retail refrigeration and real estate ground source heating businesses. This means that one product can give more sales- and production volumes, cost effective applications and incentives for concentrated product development.

ESyCool – benefits at a glance

- Industrially manufactured, tested and certified outdoor unit in a container (also available as an indoor unit)
- Tailored to meet the demands of advanced building services in international food retailing (discounters, convenience stores and supermarkets)
- For the integration of food refrigeration and demand-dependent heating technology in buildings, supplemented by an innovative ice energy store
- Integrated hydraulic module for brine based refrigeration (refrigerators and coldrooms) and concrete core activation on the water side for heating the building
- Can be extended with a PV system to make use of renewable solar energy
- Can be extended with a ventilation system customised for the building

Figure 5. Some key benefits of Viessmann ESyCool advanced energy system. (Viessmann, 2017)

ESyCool is a Remote control operated (RCO) system. In contrast a regular household refrigerator for example, is a Plug in control operated (PCO) product. Refrigeration units, cabinets- and other fixtures, such as fresh fish counters, normally represent either of these types. PCO products include the entire refrigeration machinery in a single unit. RCO products have centralized refrigeration machinery (compressor and condenser) in technical premises, and only evaporator and expansion valve are in the product, located in the shop sales floor. Typically RCO-products go to a supermarket sized customers. RCO system is typically a fair large in size and some practicality, scale and efficiency goals are pursued with centralization. PCO products go for example to smaller restaurants, shops and cafés, needing only a few units for each temperature level. My work is related to development of RCO products, which include only a sensible heat exchanger in each refrigeration fixture. This is a transition from direct to indirect cooling system, with some associated benefits. Such products however have to be tested well during development process, and our project's refrigeration system is built precisely for this job.

1.3 Goals of this project and research objectives of the thesis

Goal of this R&D-facility project was to successfully design, install and commission an indirect refrigeration system into the K-building of Viessmann's R&D laboratory facilities in Porvoo. Refrigeration in the system was made with a single Vitocal 300-G compressor heat pump unit. System had four main purposes. Most importantly it was needed to supply cold heat transfer fluid to one laboratory room EN-LAB1 in constant, controllable temperature. This capability was needed in the laboratory facilities to test refrigeration products, which utilize heat transfer fluid only via temperature change, working as secondary cooling products. Secondary, some similar cold supply was needed to K-buildings main hall, to test various products and equipment. Testing in hall can be done without interfering testing in controlled environment of EN-LAB1. Third, some cold supply was wanted for air cooling of laboratory room. Temperature- and humidity requirements are strict during testing, and most accurate conditions can be met with simultaneous and adjusted heating and cooling of laboratory room's ambient air. Finally, laboratory facilities have had some challenges during summertime with a large CO2-refrigeration system, supplying a lot of cold for testing. Some additional cold supply is introduced into the condenser/gas cooler of this CO2 system in order to make it work better during hot weather conditions.

In introduction we familiarized ourselves with refrigeration business branch and got some perspective about indirect refrigeration systems and potential benefits they can offer. In the rest of this work we report and analyze the process of this refrigeration R&D-facility project and explain some refrigeration technology-related theory relevant to this project. Theory is approached mostly via a literature review and universal refrigeration-related knowledge, whereas project report consists mostly from the description of working steps, design principles, system characteristics and results, backed up with information from literature. In the end we present results, findings, have some discussion about them and refrigeration technology, and finally make conclusions of this study and the project.

First research objective of this thesis was to find the best design and layout for the indirect refrigeration system distributing Vitocal 300-G's cooling power to our four cooling targets. Best result would be a reliable and well controllable system, which could supply very high percentages of produced cooling power to final cooling targets. Usability of system is also very important, since it is used for testing of various refrigeration products with a high level of accuracy. Second research objective was to find the best way of implementing the system. Best implementation in this case is cost effective, feasible in our projects' time window, as simple as possible and of course, brings us the good quality system with wanted properties and capabilities. Implementation also includes the definition of a reasonable and effective design process, since time is money and such system was designed for the first time.

2. Refrigeration Theory and Technology

2.1 Refrigerants, refrigeration machine oils and heat transfer fluids

2.1.1 Refrigerants

Refrigeration system's working fluids are called refrigerants. Refrigerant goes through thermodynamic cycles in refrigeration system, supplying cold to cooling targets. Refrigerant is evaporated and again condensed in compression refrigeration cycle, absorbing and releasing heat, respectively. Such phase change moves a lot of heat per refrigerant's mass flow, which awards a change to have refrigeration machinery with relatively small mass flow compared to primary cooling with some sensible heat exchange-based arrangement. This effect can be clearly noticed from figure 10 in chapter 2.2. Refrigerant properties are significantly dependent on temperature and pressure.

Refrigerant choice has a grand impact on refrigeration process and equipment suitable for refrigeration. Very different thermodynamical cycles are achieved with different refrigerants. One way of planning refrigerant selection for some application is viewing pressure-temperature (p-T) curves of different refrigerants. From p-T curves required pressure levels for desired temperature areas in planned refrigeration can be checked conveniently. Critical- and triple points and available saturated state temperature range can be examined. There are charts viewing p-T curves of many different refrigerants in the same chart, which makes narrowing of search into most suitable refrigerants feasible. Figure 6 shows p-T curves for selected quite common refrigerants. We can see from curves that CO2 has some special characteristics compared to ammonia and R134A. This also applies between CO2 and most refrigerants in general, since most refrigerants have properties closer to ammonia and R134A, rather than CO2. CO2 has very narrow temperature range of subcritical operation, and critical point is at very low temperature. For given temperature, CO2 has high operating pressures. Triple point of CO2 is at higher than atmospheric pressure, which might create challenges unless this special feature is not properly considered during design and implementation. (Danfoss, 2013)

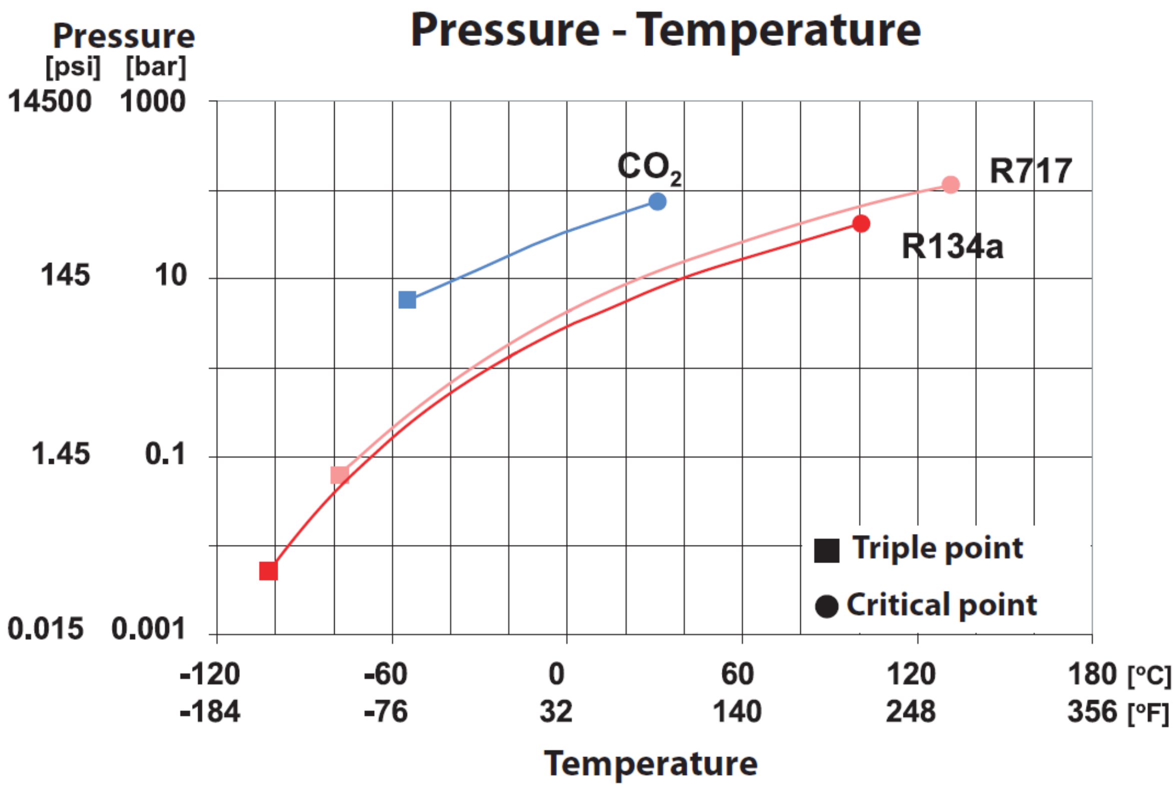


Figure 6. p-T curves of CO2, R717 (Ammonia) and R134A. Temperatures are at saturated fluid state. (Danfoss, 2013)

P-T charts include temperatures in saturated fluid state, and more detailed information about refrigerant's properties can be found for example in its logp-h chart. Logp-h chart is also an effective way of examining possible thermodynamical cycles for planned refrigeration process. Logarithmic pressure is presented of y-axis and specific enthalpy in x-axis. There should be noted, that specific enthalpy is always given in comparison to some reference state, state in which enthalpy is 0 kJ/kg. Different charts may have different specific enthalpy 0-points, which must be taken into account while using them. Logp-h chart of CO2 in a broad minded form, CO2 states and CO2's critical- and triple points are presented in figure 7. More detailed logp-h chart is required for proper thermodynamical refrigeration process design work, which for CO2 is presented in appendix 1. (Hakala et al, 2013, pp.10-22) (Siemens, 2017, pp.36-45)

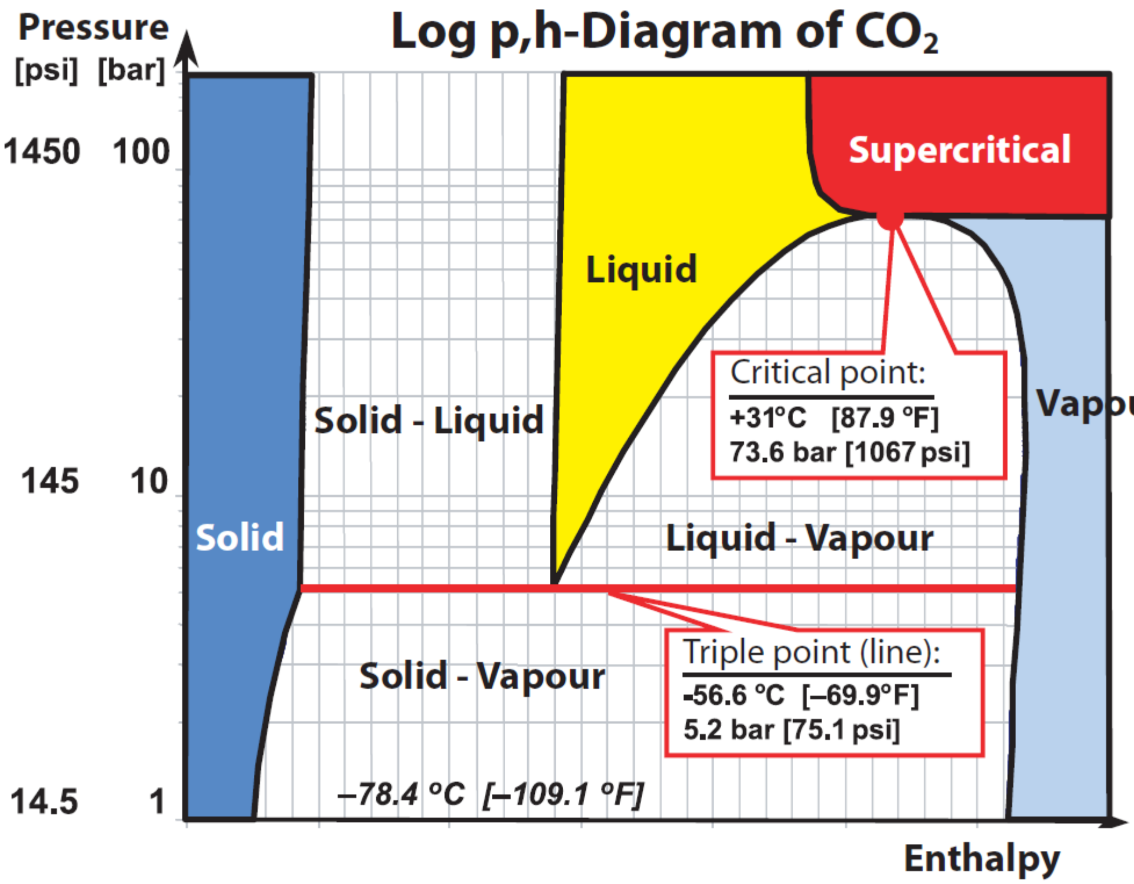


Figure 7. Large scale logp-h diagram of CO2. CO2 states. CO2 critical- and triple points. (Danfoss, 2013)

In addition to information available from p-T- and logp-h charts, there are many other properties effecting refrigerant choice. Table 2 shows some selected properties of CO2, ammonia and R134A. CO2 and ammonia are natural refrigerants, whereas R134 is halogenated hydrocarbon-based refrigerant. ODP and GWP are commonly used numbers to measure environmental friendliness related to a refrigerant. R134A has clearly higher GWP than two others, otherwise all these have small environmental load measured by ODP and GWP. As mentioned, CO2 has very different critical- and triple points compared to ammonia, R134A or most other refrigerants. Ammonia's flammability and toxicity set some challenges for its use, whereas CO2 and R134A are not flammable or toxic. CO2 however is dangerous in excessive concentrations if it is leaked into a space, where it can replace oxygen and cause CO2-poisoning.

Table 2. Some properties of CO2, ammonia and R134A. (Danfoss, 2013)

Refrigerant			**R 134a**		**NH_3**		**CO_2**	
Natural substance			NO		YES		YES	
Ozone Depletion Potential (ODP)*			0		0		0	
Global Warming Potential (GWP)*			1300		-		1	
Critical point	bar	[psi]	40.7	[590]	113	[1640]	73.6	[1067]
	°C	[°F]	101.2	[214]	132.4	[270]	31.1	[87.9]
Triple point	bar	[psi]	0.004	[0.06]	0.06	[0.87]	5.18	[75.1]
	°C	[°F]	–103	[–153]	–77.7	[–108]	–56.6	[–69.9]
Flammable or explosive			NO		(YES)		NO	
Toxic			NO		YES		NO	

Refrigerants can be classified based on their chemical content. Many organic fluids are suitable as refrigerants because of their favorable thermodynamical properties, especially pressure levels on typical temperatures present in refrigeration systems. One common criterion is halogen content, which is also the one widely used in legislation field, because of this approaches useful link to environmental aspects. Figure 8 presents classification of refrigerants, halogen content as main criterion. Halogenated refrigerants are hydrocarbons, in which hydrogen is replaced with halogen-molecules in different ways of processing. Natural refrigerants do not contain halogen-molecules. Natural refrigerants can be further divided to pure hydrocarbons and non-organic refrigerants, which both can be naturally found on Earth and their GWP and ODP are typically 0 or almost 0. For example ammonia, CO2, propane and water are natural refrigerants. (Suomen Kylmäyhdistys Ry, 2017)

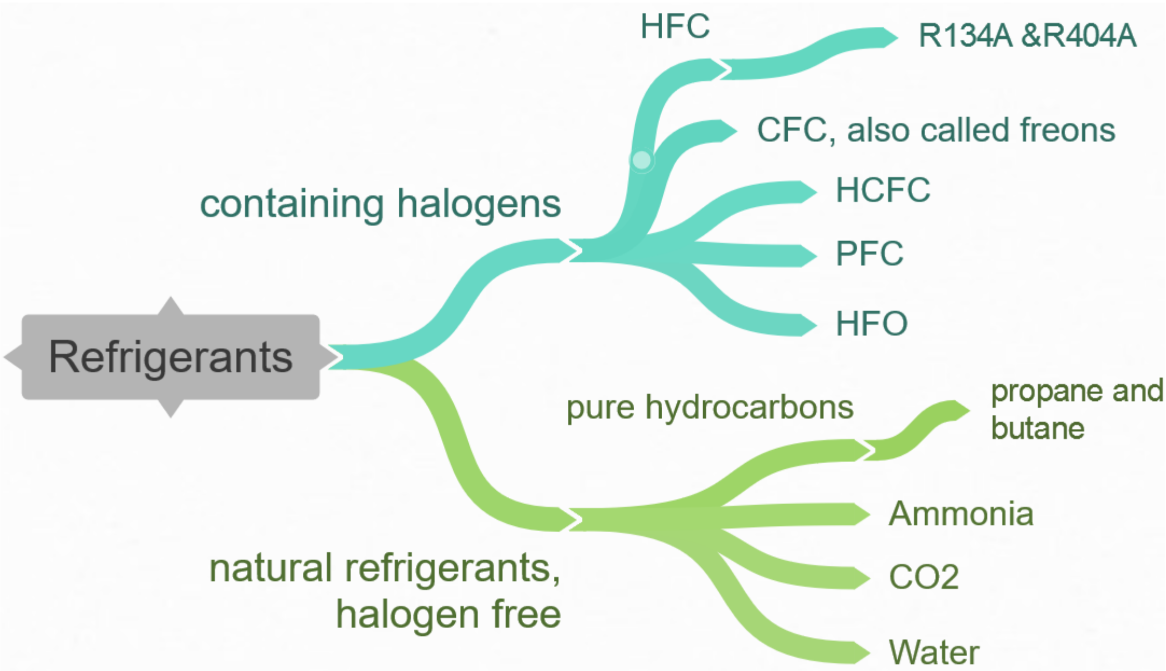


Figure 8. Classification of refrigerant groups and certain refrigerants, halogen-content as main criterion.

Some refrigerants are mixtures consisting of two or more pure refrigerants. For example a quite lot used R404A is a mixture of R125, R143A and R134A. Mixed refrigerants can be azeotropic or non-azeotropic (zeotropic) type. In constant pressure, azeotropic refrigerant mixtures have a constant evaporation temperature, just like pure refrigerants. From practical properties, azeotropic mixture refrigerants can be seen to behave like pure refrigerants, just some mixing has been done to improve some qualities or prevent downside of some otherwise great pure refrigerant. Zeotropic refrigerant mixtures have a sliding evaporation temperature, which sometimes must be considered while designing heat exchanging components and systems. Slide of evaporation temperature depends a lot on a mixture. For example aforementioned R404 has a slide of only about 0,7°C, whereas commonly used R407A, R407C and R407F all have slides of 6°C or more. (Hakala et al, 2013, pp.23-24) (SWEP, 2019B)

Refrigerant manufacturers name their products, and therefore same refrigerant with same chemical content and properties can have multiple trade names. Same applies for heat transfer fluids and refrigeration machine oils. Even though chemical composition would be practically the same (shares of main volatile ingredients), different products can have a difference in for example anti-corrosion agent composition, and the exact composition including additives can be patented.

Range of refrigerants in use has been under clear change due to some legislation in recent and ongoing decades. Halogen-containing refrigerants have been limited because of their unwanted effects to atmosphere. First there were some bans and restrictions concerning refrigerants with high ODP, and a bit later concerning refrigerants with high GWP. F-gas regulation in EU has set regulations or reporting liabilities to HFC, PFC and HFO refrigerants. F-gas regulation was adopted in 2006, and current updated version become valid in 2015. Regulations are expected to increase the share of natural refrigerants and raise F-gas prices in food retail refrigeration business (Danfoss, 2019). Numerous substitutive refrigerants for F-gases have been specified and implemented, for example for R404A there are many alternative choices in food retail refrigeration, such as R454C. R404A was Sweden's most commonly used refrigerant in grocery stores in 2017 (Gustafsson et al. 2017). Substitutive refrigerants can however bring other challenges, for instance aforementioned change means a slight increase in flammability (From class A1 to A2 according to ISO817-standard). (Kapanen, 2017)

2.1.2 Refrigeration machine oils

Oil is commonly used in refrigeration machines to lubricate compressors during operation. Some literature sources such as Hakala et al. (2013) and Siemens (2017) see, that at a time practically all compressor types need some suitable refrigeration machine oil for trouble free operation. Even though oil lubricated compressors perhaps still dominate the market, nowadays there are some applications that make use of VCRS technology without oil. Magnetic bearings can be used for oil-free operation, and technology also enables higher rotation speeds, and therefore higher cooling powers (Laitinen et al. 2016). For HVAC applications, Danfoss Turbocor series can be nominated as an example of compressors with magnetic bearing technology for cooling systems (Danfoss, 2018B). Oil-free compressors are not fully unheard of for household refrigeration appliances either, Embraco Wisemotion can be nominated as an example (Embraco, 2014) (CTCN, 2018).

Oil is in contact with circulating refrigerant, which causes unintentional penetration of lubrication oil into the refrigerant flow. Usually oil content is causing disadvantage in refrigeration process, even though this is not explicit, especially with low concentrations. It has been noticed, that in low concentrations (1-4%) oil might even improve refrigerant's heat transfer properties with some refrigerant-oil combinations. Oil lowers efficiency (kJ/kg) of latent heat exchangers (for example evaporator and condenser), since it does not change phase and therefore carry heat in same temperature as refrigerant. Liquid oil is also a problem in compressor among gaseous or supercritical refrigerant, since its very low compressibility. Oil can accumulate in low flow speed areas of circulation, releasing at once and causing a liquid hammer in compressor (CARLY, 2019). (Hakala et al, 2013)

Many refrigeration systems are equipped with oil separator. Separation capacities in refrigeration machines are typically around 97-99%. Some refrigeration machines are manufactured without oil separator deliberately, for example small factory-made refrigeration machines with below 0°C evaporation and small hermetically sealed refrigeration units. For performance of refrigeration system and separability of oil, it would be favorable, if oil type would either dissolve totally or not at all with refrigerant used. Efficient oil recovery can set some demands for refrigerant flow speeds in the piping, that have to be taken into account while designing refrigeration systems. Oil should not be accumulated in certain locations in the system, and high recovery rates should become achievable in the collecting device. (Hakala et al, 2013)

Just like refrigerants, there is a large variety of different refrigeration machine oils available for different refrigeration systems. Oil choice and arrangement has an effect on successful operation and efficiency of refrigeration machinery. Factors effecting oil choice are for example viscosity in different temperatures, purity of oil product, compatibility with compressor sealing materials, solubility with used refrigerant, moisture- and acid content, foaming tendency and chemical stability. (Hakala et al, 2013)

2.1.3 Heat transfer fluids

Heat transfer fluids are used to carry heat in indirect cooling systems. Unlike refrigerants, heat transfer fluids transfer heat via sensible heat exchange only, without phase change in refrigeration systems. There are many names for refrigeration system's heat transfer fluids used generally, for example secondary fluids, secondary working fluids, secondary refrigerants, and secondary coolants. Salt-water solutions, brines are sometimes used as expression for refrigeration heat transfer fluids too, because they are often used in such purpose. Similarly anti-freeze solution is sometimes used as an expression, perhaps to highlight fluids property of staying in liquid and well flowing form under low temperatures. (Melinder et al, 2015, pp.47-60)

Like refrigerants, there are many different secondary coolants available, and many properties that have to be taken into account while making choices. In refrigeration systems, freezing point must be low enough for secondary coolant to flow well and not freeze, breaking equipment and causing hazards. Many secondary coolants are water based solutions, due to water's good thermodynamical- and various other properties. Mixing with some other substance is usually needed for desirable freezing point below 0°C to be achieved. Freezing points of some water-based solutions are presented in figure 9. Freezing points are given according to additive concentration in weight %. Besides secondary coolant choice, freezing point can be lowered with higher concentration of additive. This will however typically make other fluid properties less favorable for a secondary coolant, since water concentration decreases. This would rationalize the use of additive with higher frost protection in such refrigerant system. Table 3 presents practical operation span for different secondary coolants according to Melinder et al. (2015). Most of the secondary coolants considered are water-based solutions. Also CO2 can be used as a secondary coolant. It has some clear benefits and challenges, which are discussed in more detail in chapter 4.2. One distinctive property of CO2 is a very low viscosity also in low temperatures. (Melinder et al, 2015)

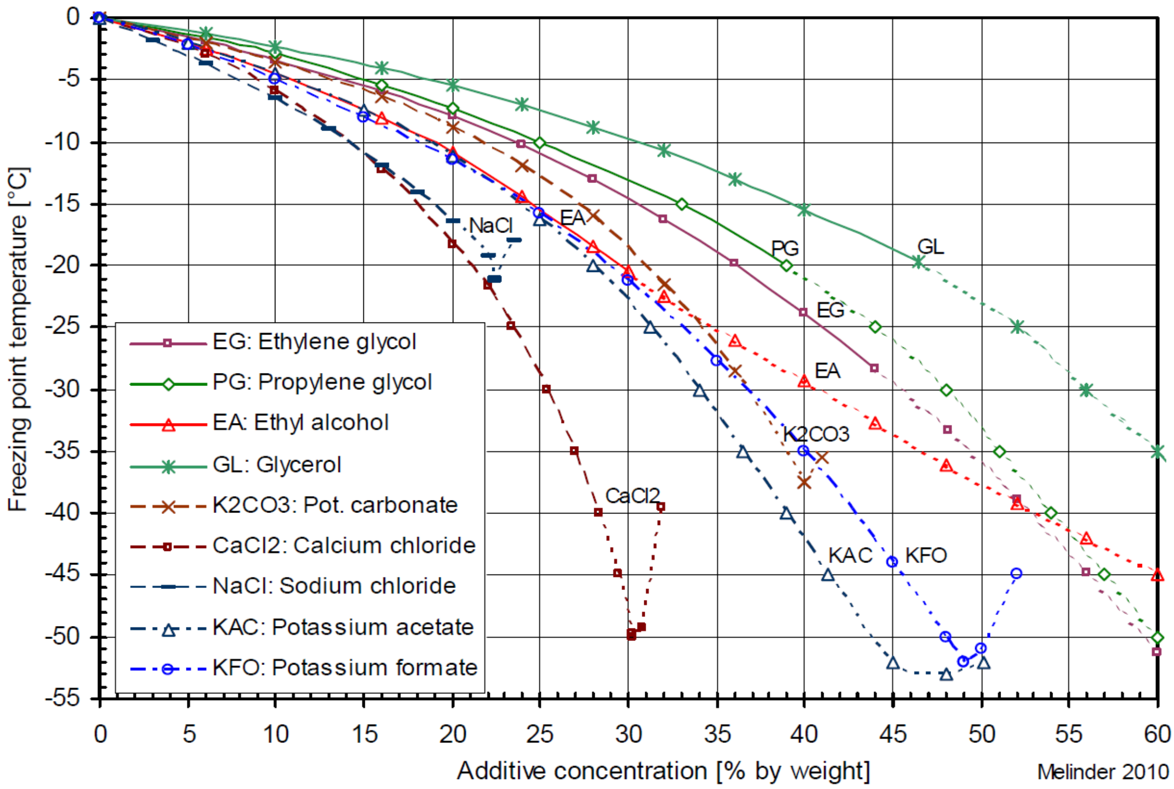


Figure 9. Freezing points of some water solutions according to additive concentration in weight %. (Melinder et al, 2015)

Table 3. Practical operation span for different secondary coolants. (Melinder et al, 2015)

Practical operation temperature

Water solutions	100°C	80°C	60°C	40°C	20°C	0°C	-10	-15	-20	-25	-30	-35	-40	-45	-50°C
Water	+	+	+	+	+	+									
Water solutions															
Ethylene glycol	+	+	+	+	+	+	+	+	+	+	+				
Propylene glycol	+	+	+	+	+	+	+	+	+						
Ethyl alcohol				+	+	+	+	+	+	+					
Glycerol				+	+	+	+	+	+						
Calcium chloride				+	+	+	+	+	+	+	+	+	+		
Sodum chloride				+	+	+	+	+							
Pot. carbonate				+	+	+	+	+	+	+	+				
Pot. acetate		+	+	+	+	+	+	+	+	+	+	+	+	+	
Pot. formate				+	+	+	+	+	+	+	+	+	+	+	
Betain				+	+	+	+	+	+						
Methyl alcohol				+	+	+	+	+	+	+	+	+	+	+	+
Ammonia					+	+	+	+	+	+	+	+	+	+	+
Litium chloride				+	+	+	+	+	+	+	+	+	+	+	+
Two phase media															
Ice slurry						+	+	+	+	+	+	+			
Carbon dioxide								+	+	+	+	+	+	+	+

There are some quite recently published literature available about secondary coolants and their desirable properties, for example from Melinder et al. (2015) and Suomen Kylmäyhdistys Ry (2017). Based on these resources and knowledge gained during design- and implementation project of secondary cooling system during master's thesis work, certain properties important for a secondary coolant can be identified, which are collected in table 4. Table 4 also shows pure water's status with given properties, because many important secondary coolants are water based. Classical act is to mix certain amount of suitable additive to water, achieving lower freezing point and sometimes lesser corrosion. Additive concentration is tried to keep as low as practically possible, for water's good thermodynamical- and other properties to still dominate in mixture characteristics.

Table 4. Important properties for secondary coolants and water's related status.

Property	**Favourable property status**	**Water's status**
freezing point (°C)	lower than coldest operating temperature	too high for most refrigeration applications
specific heat capacity (kJ/kg°C)	high	high
thermal conductivity (W/m°C)	high	high
density (kg/m^3)	high	high
viscosity (kinematic m^2/s)	low in operating temperatures	low in typical operating temperatures
corrosivity	low with system materials	some water-based solutions or water are corrosive to for example metals.
environmental friendliness	high	high
non-toxic	yes	yes
non-flammable	yes	yes
economical and well accessible	yes	yes
current know-how and applications	high level, a lot in operation	yes

Thermodynamical properties are important for a good secondary coolant. In addition to sufficient frost protection, secondary coolant should have high density, specific heat capacity and thermal conductivity. Viscosity of secondary coolant should be low in operating temperatures of a refrigeration system. Secondary coolant with aforementioned thermodynamical properties carries a lot of heat per volumetric flow, supplies cooling efficiently and facilitates compact component sizes for given output. Good heat transfer properties give us smaller achievable pinch points in heat exchangers, leading to better efficiency, carrying more heat and allowing more cooling capacity with same temperature difference. Viscosity and density of secondary coolant effect in pressure drop in refrigeration system's piping and components. In ideal situation pressure drops are low. This leads to lower circulation pump power consumption and is better for system durability. Pressure drop can be adjusted more economically with high density and low viscosity fluid. If fluid is too viscous, for example use in warmer temperature or increasing of pipe sized is needed for smaller pressure drops, which may be hard to achieve without compromising refrigeration performance requirements.

There are also numerous other aspects effecting secondary coolant choice, in which some important ones are low corrosivity with refrigeration system's materials, environmental friendliness, non-toxicity, non-flammability, good economical prospects and good accessibility. A good global or local knowledge about secondary coolant and available installation, consultancy and component supply can also play big role, especially in some applications and projects. For example supply for components and spare parts compatible with water is high in most product categories, whereas components withstanding some more rare and corrosive secondary coolant can be harder and more expensive to find. Environmental friendliness includes a lot of different aspects, for example GWP, ODP, biological degradation rate in nature, ease of recycling and health aspects for humans and diverse wildlife.

We chose propylene glycol-water solution as secondary coolant in our project's refrigeration system. We chose it over ethylene glycol because of its significantly better safety- and health properties, even though frost protection was slightly lowered. Good accessibility from markets was also a criterion. Propylene glycol content was set to 30% or 50%, depending on area in the system. There were many trade names for propylene glycol; we purchased Antifrogen L 100% pure propylene glycol, which was mixed with water in suitable ratios to deliver aforementioned mixtures for our system. Antifrogen is a product from Swiss company, Clariant. Information about Antifrogen L can be found in appendix 2. Technical datasheet, specific heat capacity-chart, thermal conductivity-chart and relative pressure drop-chart are presented. Charts are in function of temperature for each propylene glycol-concentration in water solution. (Clariant International Ltd. 2014) A few remarks can be pointed out: As with most of commonly used secondary coolants, specific heat capacity and thermal conductivity decrease and relative pressure drop increases as we increase propylene glycol concentration in water solution. This is because pure water has better aforementioned characteristics than propylene glycol. Some propylene glycol is mixed with water to get enough frost protection for below 0°C temperatures. There are also some anti-corrosion agents in Antifrogen L to give it some resistance against corrosion. (Clariant International Ltd. 2014)

2.2 Heat exchangers in refrigeration systems - Evaporators, condensers, gas coolers and sensible heat exchangers

Heat exchangers move heat between two media in refrigeration systems. Evaporators and condensers are heat exchangers involving phase change in at least one side of the heat exchanger surfaces, whereas heat exchanger is a general term for components exchanging heat between two media, utilizing temperature difference. In transcritical CO2 refrigeration systems, heat exchangers rejecting heat from supercritical CO2 into ambient air or other heat sink are called gas coolers. This is, even though CO2 is actually in supercritical, not gaseous phase throughout this component. (While gas cooler working in constant pressure)

Heat exchanger as a term is sometimes used referring particularly to a fluid-fluid heat exchanger without phase change. For example in indirect systems, heat exchangers may be named according to type of a possible phase change, and a term heat exchanger is used exclusively for sensible heat exchangers in the system. While talking about a heat exchanger, sometimes specification about more detailed name use is necessary. This is because heat exchangers are often named based on medium on only either one side. For example, in basic household's fridges a component called evaporator, evaporates the refrigerant, fridges technical working fluid in temperature below 0°C. For fridge's air side, where foodstuff is at, the same component works as sensible heat exchanger, taking heat out from the fridge air and foodstuff. A small amount of moisture in fridge's air is condensing and freezing on the heat exchanging back wall in fridge, therefore there is an occasional automatic defrost phase or need for manual defrosting in household fridges. (And usually in refrigeration systems in general) Household refrigerators have typically an automatic defrosting (HELEN, 2017).

Evaporator's purpose is to transfer heat from space being cooled into the refrigerant. Temperature becomes colder or stays cold in the cooled space because of this removed heat. Condenser finally moves this heat away from the refrigerant circulation into the heat sink, outdoor air for example. In indirect systems there might be heat transfer fluids to carry heat from cooled space to evaporator and/or from condenser to heat sink. Some indirect systems may have heat exchangers moving heat from one heat transfer fluid to another, without phase change. Only sensible heat is exchanged. In our project's system, we have two such heat exchangers. In refrigeration systems all these components ultimately aim for moving heat from cooled space to the heat sink or heat recovery.

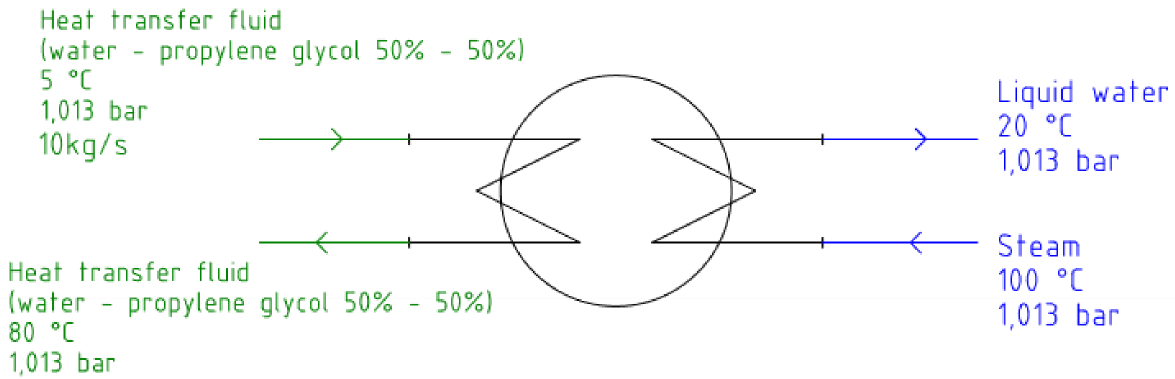
Q (W), heat transferred in heat exchanger can be calculated with equation 1. Applies for all aforementioned heat exchanger types, interpretation depending on more detailed heat exchanger type.

$$Q = \dot{m} * \Delta h = \dot{m} * (h_1 - h_2) = \dot{m} * L_{eva} = \dot{m} * C_p * \Delta T \quad (1)$$

Where $\dot{m}$ (kg/s) is mass flow of refrigerant, heat transfer fluid or some medium taking part in heat exchange. Δh (J/kg) is change of enthalpy in this mass flow. h1 (J/kg) is enthalpy of mass flow before, and h2 (J/kg) after the heat exchanger. L_{eva} (J/kg) is latent heat of vaporization or condensation. L_{liq} (J/kg) can be used instead, when calculating medium liquefying or solidifying in either side of heat exchanger, for example water in ice cold storages. C_p (J/kgK) is specific heat capacity of mass flow's medium in constant pressure. ΔT (K) is a medium's temperature difference over the heat exchanger.

An example of various heat exchanger calculations and using equation 1 are presented in figure 10. Case involves multiphase heat transfer, and also highlights well the large amount of energy transferred via phase change, condensing in this case. (Compared to sensible heat transfer, related to temperature change)

Question: We are heating water-glycol heat transfer fluid with some temporarily excess process steam. With given properties, how much steam is atleast needed for successful process?



Solution: On left side there is sensible heat treansfer only. Heat power needed to heat transfer fluid is

$Q = m*Cp*dT = 10kg/s * 3621\ J/kgK * 75K = 2715750W = 2{,}716MW$

On right side there are both latent- and sensible heat transfer.
Steam condensation to liquid and liquid cooling to 20°C, respectively.
Mass flow needed to supply Q can be calculated from following equation

$Q = m*L\_eva + m*Cp*dT$

by rearranging terms, we get:

$m = Q\ /\ L\_eva+Cp*dT$

$= 2715750W\ /\ (2256800J/kg + 4185{,}4J/kgK * 80K)$

$= 1{,}048\ kg/s$

Protips:

- Calculations can be efficiently done with some computer programs, which also fit well in quick checking and illustrating results. This calculation was made with SWEP SSP heat exchange-program.

- Used specific heat capacities (Cp's) should represent accurately the entire temperature interval of temperature change, if single value is used. (water 100-20°C and pr.gl 5-80°C) Hence, Cp is temperature dependent. In this case we used average values in 60 and 37,5°C, after checking that Cp varies only a little and in approximately linear manner. Cp for water is easily accessible in relevant table books, for pr. gl. we got values from SWEP SSP's Fluid property calculator.

- Q could have been also expressed with shorter equation using specific enthalpies for water. Like this:
$Q = m * dh = m * (h1 - h2)$
In this case we need to get h1 and h2 from a table book, well readable diagram or computer program. Same applies generally for all media, also glycol-water mixtures.

Figure 10. Heating of glycol-water mixture with excess process steam. An example of various heat exchanger calculations and using equation 1.

2.3 Safety of refrigeration systems

Safety and associated risks vary a lot depending on applied technology, refrigeration system size and more detailed characteristics. Some safety risks in refrigeration systems and examples of corresponding security measures are collected in table 5. Careful design of the system is very important for safety. Many risks can be avoided right in the design phase, with for example safe fluid selections. System should be under control and technical knowledge at all times. Well planned refrigeration system could work under its designed operation environment safely on its own, and special safety devices and practices can act as an extra layer of security. As we learned in our R&D refrigeration system project, design choices can have a big effect on safety features, even if system performance and fixed design guidelines were practically the same. One example from our project is heat transfer fluid choice for two piping circuits going to heat rejection and to CO2 heat exchanger. Propylene glycol was chosen over ethylene glycol as 50% additive to water; even that frost protection rate was lowered a bit with this choice. Reason was ethylene glycol's toxicity, whereas propylene glycol is much better for health and therefore a safer choice.

Table 5. Some safety risks in refrigeration systems and examples of corresponding security measures.

Risks	**Security measures for example**
High pressures	Safety valves, pressure vessels, high pressure components
High and low temperatures	Insulation, warning signs, restricted access to hot/cold area
Toxic fluids	Non-toxic fluid choice, leak detection & alarm system
Flammable fluids	Non-flammable fluid choice
Otherwise dangerous fluids	Non-otherwise dangerous fluid choice
Corrosion	Corrosion resistant materials
Trapped liquid	Safety valves, pressure vessels, good concept design
Non-verifiable fluid states in system	Sight glasses, sensors, careful concept design
Fluid leaks	Safe fluid choices, pressure testing, leak proof components and installation techniqués
Component breaks	Good component quality, pressure durability and compatibility. Regular inspections
Cavitation and liquid hammers	Slight overpressure on compressor suction lines, pressure vessel locations
Injuries while installation and O&M	Right workwear, tools and techniques. Work only with safety secured. If questions, work waits until safety is secured.
Varying operating conditions	Careful concept design, proper operation- and control system, wide working areas of components
Foodstuff contamination risk if cold chain breaks	Foodstuff insulation, temperature tracking, toss of any questionnable foodstuff
Compressor short cycling, overworking or lack of oil	Caferul concept design, proper controllability of system
Person knowing system not available all times	Contacts well marked, clear instructions for safe system shutdown, automatic safety devices, proper documentation

Safe operation of refrigeration system should be well noticeable. Design of refrigeration system should allow adequate amount of controllability and inspectability of the system. Remote and well visualized access to operating values and phases of fluids are good ways of ensuring the right state of operation, and handling possible problems or inefficiencies in an early stage. Sight glass can be used in some components to verify the correct phase of refrigerant, verify that air has been successfully removed from the system, or check that entirely gaseous or liquid state is reached in some component. Cold chain of foodstuff is globally important safety- and health aspect. Reliable and good quality food chain has also a big economical impact, since spoilage of foodstuff storage can ruin vital income for actors in food retail branch and disrupt foodstuff availability for consumers.

High pressure levels can bear potential risks in refrigeration systems. For example CO2 compression refrigeration systems have typically high working pressures, transcritical operation requires over 73,6 bar for CO2 to enter the supercritical fluid state. (Emerson, 2016) Pressure levels and pressure differences in system should be high enough for efficient and well performing operation, but in safety perspective not significantly higher than necessary. This is because high pressures have a potential for powerful accidents, if something goes wrong, in form of explosive bursting of equipment for example. High pressures are typically utilized more in refrigerant-, rather than heat transfer fluid circulations, because heat rejection in condenser or gas cooler demands high enough pressure for desirable phase change-/ superheated fluid cooling temperature. Safety valves for overpressure are standard equipment in pressurized and/or closed refrigeration systems. Safety valve is set to open if certain pressure level is exceeded, releasing some fluid and pressure into the atmosphere or some safe space. It is necessary to install discharge of safety valve to a safe location and direction, so that possible high pressure discharge cannot hit any people. Opening pressure and blow-off speed of safety valve must be sized based on a worst case scenario. Safety valve must open well before tolerance of weakest component in the system is surpassed. Blow-off speed must be enough for even the hardest need for discharge. Typically this case is a fire, when refrigerant and other fluids in system heat up rapidly, expand and can build pressure quick. Especially large liquid bodies, buffer tanks for example must be considered in safety valve blow-off speed calculations. (Hakala et al, 2013)

High temperatures can be found in compressor discharge line for example, especially when used heat sink is considerably warm and refrigerant has to be in higher temperature for efficient heat rejection. Also heating of components due to vibrations and work is possible. Cold fluids can cause severe frost damages if spilled on people, since they can conduct heat lot faster than air, and there is always some refrigerant in colder temperature than the coldest cooling target present in operating evaporator of the system (this is a physical and technical requirement for cooling to be possible in compressor refrigeration systems, since heat travels naturally from hotter to colder material).

Finding the best fluid for refrigeration application leads often to a compromise of some aspects, since it is hard to achieve all demanded properties in the same fluid. Toxic, flammable or otherwise dangerous fluid properties are currently found in many common refrigeration systems as refrigerants or heat transfer fluids. For example ammonia and ethylene glycol-water mixtures are toxic, and most hydrocarbons-based refrigerants are highly flammable. CO2 can cause choking if accidentally released by replacing oxygen (Danfoss, 2013, p.101). One guaranteed way of preventing these particular problems, is to choose a fluid that is not toxic, flammable or otherwise dangerous, even though this is a hard task, and performance-related and thermodynamical properties have to be most likely yield on in many applications. Water has many excellent properties needed for a working fluid or a heat transfer fluid. It is non-toxic, non-flammable and does not produce choking hazard in the level we consider refrigeration system fluids. Water however has non-desirable phases and/or temperatures of phase change in pressure levels economically achievable in many refrigeration applications. In our R&D refrigeration system project for example, propylene glycol is mixed with water for additional frost resistance.

Corrosion prevention has to be considered by choosing suitable materials for used refrigerant and heat transfer fluids. Also regular inspections are often helpful. Corroded materials can lose their pressure durability or other vital properties, leading to system failures, pressure bursts or other dangerous events. Flash point of refrigerant is an important fire safety property. Flash point is a lowest temperature, in which vapours of fluid will ignite while exposed to a spark. Flash point of fluid should be well higher than highest temperatures in the system, so that possible vapours do not ignite with a little spark.

Ventilation of spaces should be properly taken care of, for both healthy air quality and for avoiding accumulation of flammable gas mixtures to air. There should be some way of ensuring, that possible leaks will be detected somehow in the system. One way is monitoring fluid pressures. In closed, overpressurized system leaks tend to lower the system pressure. Refrigeration systems may have very many joints and possible locations of leaking, and leaks can happen from very small holes. Therefore a visual inspection is a good, but not all-inclusive way of ensuring that system has not leaked.

3. Vapor Compression Refrigeration Systems

3.1 Basic concept

Vapor Compression Refrigeration Systems (VCRS, also referred to as Compressor heat pump-based refrigeration systems or shortly Compression/Compressor refrigeration systems) are widely used for cooling in different sized spaces, household appliances, industry, food and- beverage sector, cold storages and many other applications having cooling demand. At least 90% of today's refrigeration and air conditioning capacity is fulfilled with compressor heat pump technology (Laitinen et al, 2016) (Siemens, 2017, p.46). System is suitable for refrigeration, without introduction of any originally cold material with the space or product being cooled. Compressor heat pump moves heat from cooler space into warmer space with electricity-made work.

Compressor heat pump works in same principle while refrigeration or heating action. In refrigeration application, goal is to cool cold side and heat is removed to heat sink, often outdoor air. In heating application, goal is to move heat from outdoor air into heated space. Outdoor air will then act as a practically infinite heat source. In supermarket refrigeration system for example, cold space and heat sink are all freezers and refrigerator-cabins in a supermarket and outdoor air, respectively. VCRS technology utilizes a large amount of latent heat energy stored in gaseous refrigerant, when compared to liquid state in the same temperature.

Compressor heat pump-based refrigeration is powered with compressor work, and different compressor types enable different set of strengths and challenges. Household refrigerators are an excellent example of technology's feasibility and popularity for small scale systems, whereas even large scale district cooling can be efficiently achieved with turbo compressor-based heat pump technology (Laitinen et al, 2016, pp.10-15, 51).

Heat pumps are categorized into three groups based on a used heat source in Khartchenko et al (2014, p.541). Air source, ground source, and water source heat pumps. These all can be found in VCRS technology. Our project's Vitocal-300G heat pump transfers heat from one water-glycol heat transfer fluid stream to another. Vitocal 300-G is commonly used in geothermal heat pump applications, but as previously introduced, in our project its job is to cool four cooling targets in our laboratory facilities.

Main components of compressor heat pump-based refrigeration system are presented in figure 11. Figure 11 shows also a rough process path of working fluid in logp-h diagram. Working fluid is also commonly termed as refrigerant, especially in cooling- and refrigeration applications. In refrigeration applications, heat is removed from the cooled space into flowing refrigerant in the evaporator. Compressor raises gaseous refrigerant's pressure and temperature into desired level. Condenser rejects heat into a heat sink, usually outdoor air. Expansion valve lowers pressure and temperature of refrigerant by letting it expand by an applicable volume. Associated working fluid cycle and thermodynamical perspectives are discussed in more detail in next chapter, 3.2.

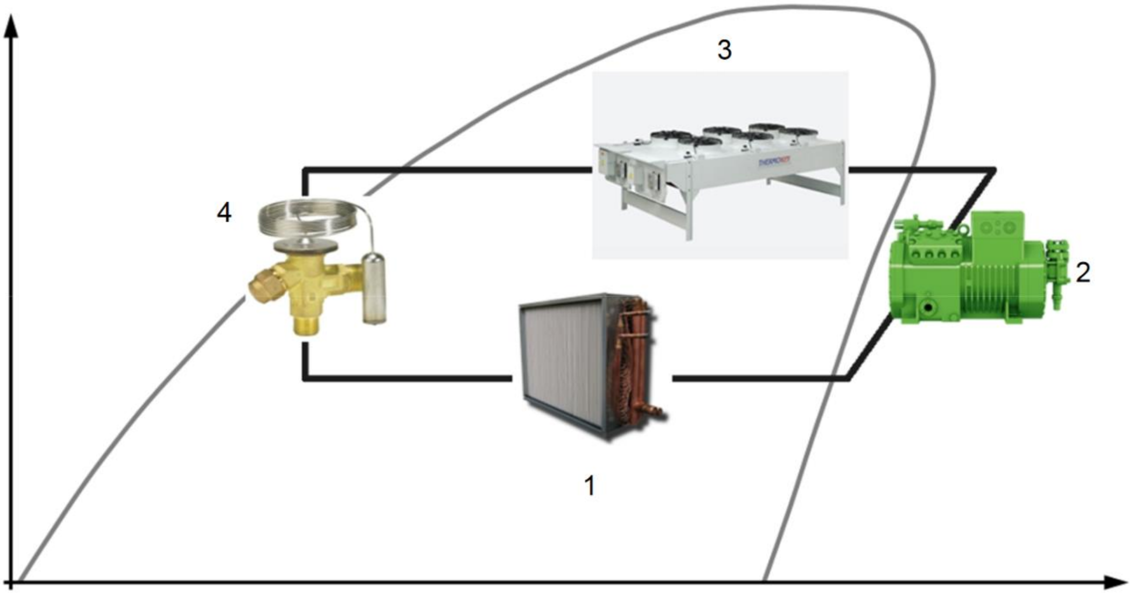


Figure 11. Main components of compressor heat pump-based refrigeration system and a rough process path of working fluid in logp-h-diagram. 1) Evaporator 2) Compressor 3) Condenser 4) Expansion valve. (Siemens, 2017, p.46)

3.2 Vapor compression refrigeration process

Vapor compression refrigeration process composes of four serial stages, evaporation, compression, condensing and expansion. Process is dynamic, cyclic and continuous, there is some refrigerant in every stage at given time. Purpose of the system is to remove heat from the cooled space, lowering its temperature and/or keeping it low. Vapor compression refrigeration process is presented in figure 12. Refrigerant phases, heat- and work flows, some exemplary temperature levels and pressure levels in the system are also shown.

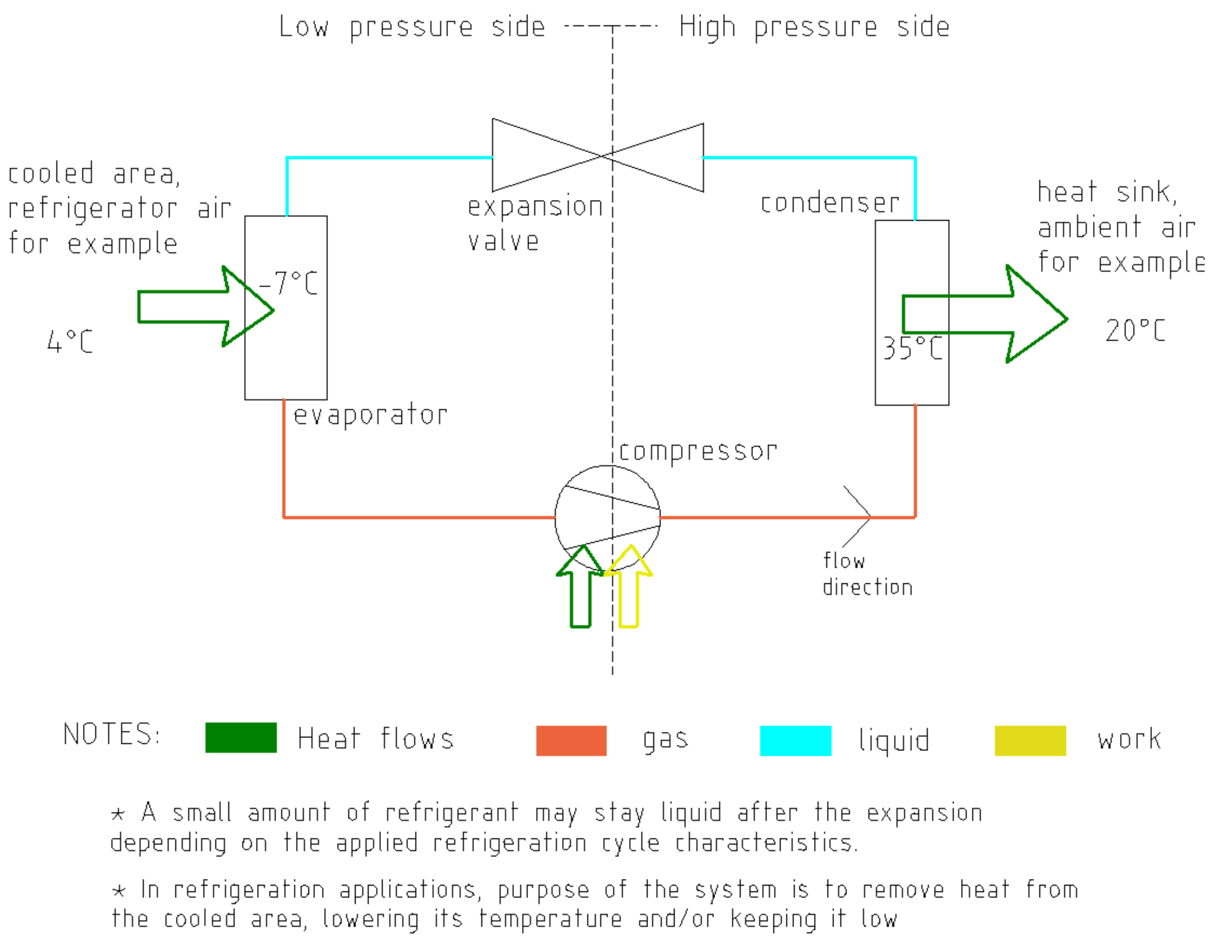


Figure 12. Vapor compression refrigeration process. Refrigerant phases, heat- and work flows, some exemplary temperature levels and pressure levels in the system are shown.

VCRS process and associated refrigerant cycle can be explained component at a time as follows: Cooling is supplied to cooling targets in evaporator. Refrigerant turns from liquid to gas, absorbing heat from the cooled space or medium. For this to happen, refrigerant has to evaporate in lower temperature than the cooling targets temperature, heat to move the right way. This is controlled with suitable pressure, resulting in desired evaporation temperature of the refrigerant. In evaporator, refrigerant's specific enthalpy raises with the latent heat of evaporated refrigerant, plus the superheat. Absorbed heat is off from the cooling target, and corresponds to the accomplished cooling power.

Compressors raise gaseous refrigerant's pressure after evaporation process. This is necessary in order to reject heat out of the refrigeration cycle. Refrigerant temperature also increases, because gas compression naturally raises both temperature and pressure, even with thermodynamically very efficient compressors. This means, that compressing adds heat into refrigeration cycle, that has to be also rejected in condenser. Refrigerant's condensing temperature (=boiling point) depends on its pressure. High enough pressure has to be achieved with compression, for refrigerant's boiling point to settle in desired temperature. Boiling point has to be higher than heat sink (or other media taking the rejected heat) temperature, so heat can move away from the refrigeration cycle. If ambient air temperature is 30°C, compression has to achieve at least around 35-40°C for sufficient temperature difference including margins for heat to transfer from refrigerant to air. This is why heat rejection to air demands higher refrigerant pressure, and therefore also larger compressor power in hot weather.

VCRS extracts heat from the cooling target by absorbing it in the evaporating refrigerant. For cooling to be continuous, this heat and a little bit of heat added by compressor have to be get rid of. Otherwise refrigerant temperature will rise quite a fast, and cooling is no longer possible. Heat is rejected in the condenser. Refrigerant turns from gas to liquid, releasing latent heat of condensing into the heat sink, outdoor air in many cases. Rejected heat can be captured with some heat recovery method, for preheating of water for example. Heat recovery can be a promising option, especially with high cooling power systems with also some heat demand nearby.

After condenser, refrigerant is again in liquid form, and has removed some heat out from the refrigeration cycle. In transcritical refrigeration cycles, refrigerant rejects heat in gas cooler (instead of condenser), in critical fluid state. Either way, refrigerant is still in high pressure and temperature. These need to be lowered into desirable level for evaporation process. This is made by expanding refrigerant in expansion valve. If there is not enough expansion, refrigerants evaporation temperature (boiling point) is settled too high, and heat cannot be transferred from cooler space into refrigerant. If there is more expansion that required with fair margins, refrigeration cycle wastes energy, since compressor has to make more work than necessary, while working over higher pressure difference. After the expansion, given refrigerant volume flows again to evaporator, and starts to travel through a new refrigeration cycle.

3.3 Energy efficiency of Vapor Compression Refrigeration Systems

Figure 13 presents the main energy flows of vapor compression refrigeration cycle. Φ_0 (W), cooling capacity equals heat power absorbed by refrigerant in evaporator. Evaporation removes heat from space being cooled by absorbing it to energize evaporation process, which gives us the desired cooling power. P_{el} (W), compressor power goes for pressurizing refrigerant, but also bringing some heat to it, because of the thermodynamics of gas compression process. Φ_c (W), heat rejection in condenser (or gas cooler) dumps the heat out of the refrigeration cycle. Both heat removed from the cooled space in evaporator and heat brought to process in compression have to be get rid of. The energy balance of VCRS is presented in equation 2.

$$Q_c = Q_0 + P_{el} \tag{2}$$

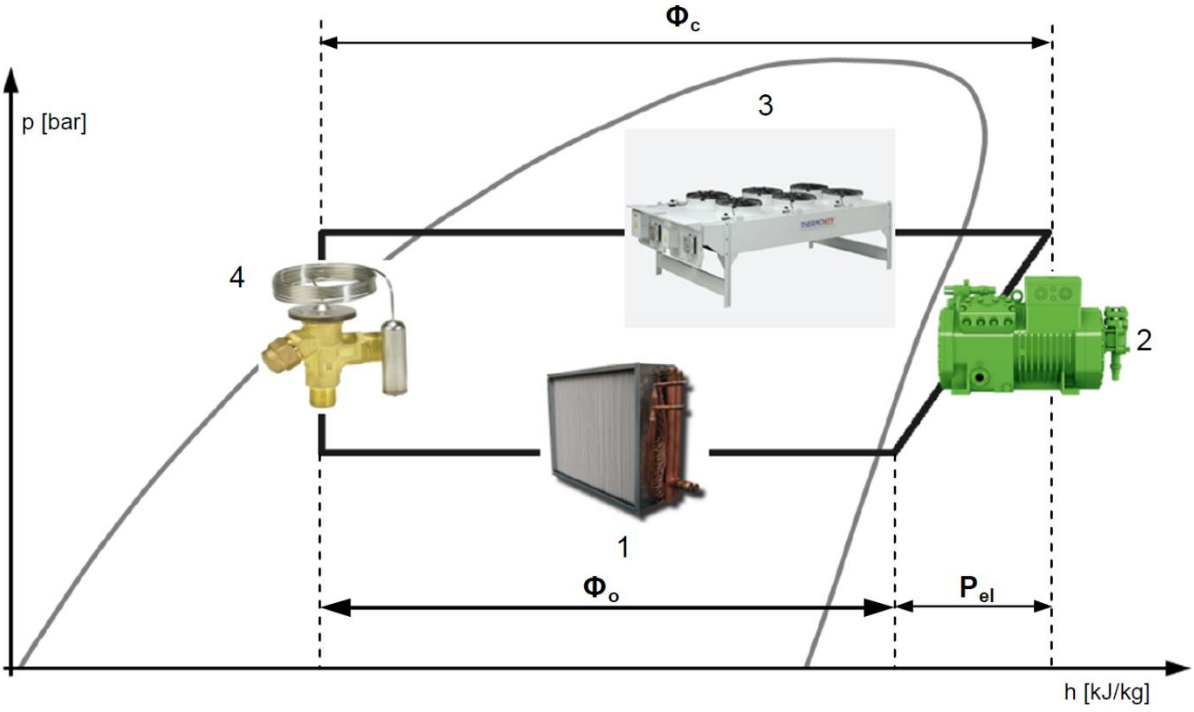


Figure 13. Main energy flows of vapor compression refrigeration cycle. Figure 11 accompanied with cooling capacity Φ_o , heat rejection in condenser Φ_c and compressor power P_{el} . (Siemens, 2017, pp.66-68)

Energy efficiency of vapor compression refrigeration cycle can be measured with Energy Efficiency Ratio (EER). Coefficient of Performance (COP) is another common expression used for the same thing. Some literature recommends use of EER with refrigeration systems, and COP for heat pumps, for example Siemens (2017, pp.81-82). In European heat pump standards COP is used for heating- and EER for cooling-related efficiencies (Laitinen et al, 2016, p.19). EER of vapor compression refrigeration system can be calculated with equation 3.

$$EER = \frac{\Phi_o}{P_{el}} = \frac{\dot{m} * \Delta h_{eva}}{\dot{m} * \Delta h_{cmpr}} = \frac{\Delta h_{eva}}{\Delta h_{cmpr}} = \frac{h_2 - h_1}{h_4 - h_3} \qquad (3)$$

EER is a number comparing benefit and cost (Siemens, 2017, p.81-82). Φ_o (W) is the desired cooling power and P_{el} (W) is electrical power needed to drive the compressor. In equation 3, $\dot{m}$ (kg/s) is mass flow of refrigerant, Δh_{eva} (J/kg) is enthalpy change of refrigerant in evaporator, Δh_{cmpr} (J/kg) is enthalpy change of refrigerant in compressor, h_1 (J/kg) is enthalpy of refrigerant before evaporator, h_2 (J/kg) is enthalpy of refrigerant after evaporator, h_3 (J/kg) is enthalpy of refrigerant before compressor and h_4 (J/kg) is enthalpy of refrigerant after compressor.

P_{el} can be defined in at least two different ways, which's difference is important to be aware of. In figure 13, P_{el} is pointing the power transferred to refrigerant. In real process, electricity consumption of the compressor is a bit higher, since 100% of used electricity cannot be transferred to refrigerant; at least a small part goes to losses. Therefore, in terms of figure 13, only EER of the refrigeration cycle can be calculated with equation 3. If we want to calculate the EER describing ratio of desired cooling output to realized electricity consumption, we have to add power of losses to P_{el} , or straight use the electricity consumption as P_{el} in equation 3. Similar thing can be outlined for Φ_o . Is all heat really removed from the space intentionally being cooled, or is some heat perhaps removed in vain? This speculation can be continued longer than necessary in this work, but aforementioned refers to importance of knowing, which efficiency is actually measured.

Overall energy efficiency of a VCRS often includes more aspects than refrigeration machinery's cycle EER. The whole refrigeration system can have other electricity consumption such as lightning and defrosting. On the other hand, for example heat recovery can give additional benefits that can be taken into account while considering overall efficiency. Our project's refrigeration system is VCRS-based indirect cooling system. In addition to Vitocal 300-G heat pumps electricity consumption, we for example have 5 heat transfer fluid circulation pumps, dry cooler and a temperature control valve using electricity. Also all cooling power applied to heat transfer fluid in Vitocal 300-G will not be usable in some of our four cooling targets due to losses, some of the cooling power is lost to environment. Realized overall efficiency depends on the weather and operation mode of our system. Some kind of efficiencies can be calculated also to separate parts of the refrigeration system. In our project's refrigeration system we could be interested in the overall efficiency, defined as supplied average cooling power per average system electricity consumption. However, accurate performance of our refrigeration system in its duties is perhaps priorized over energy efficiency as long as electricity costs are expectedly somewhat low. Only one of such systems will be produced, and most of benefits come from successful fulfillment of its tasks and know-how accumulating to company, and most of costs from working hours or some other things than systems electricity consumption.

4. CO2 Refrigeration Systems

4.1 Basic concept and market view

As we learned in chapter 1.1, CO2 refrigeration systems are currently increasing in share of Worlds refrigeration capacity. CO2 refrigeration systems are vapor compression refrigeration systems with carbon dioxide (CO2) as refrigerant. CO2 is also suitable for use as a heat transfer fluid in secondary cooling systems. Worldwide amount of stores with transcritical CO2 refrigeration systems in October 2018 is presented in figure 14. As we can see, currently CO2 systems are most common in Europe and Japan. North America has many CO2-equipped stores, even though amount is relatively small taking a huge size of market into account. Many countries and continents have some installed capacity, but a proper generalization has not yet taken place. CO2 systems are a common solution for supermarket- and other food retail locations. CO2 systems can be designed and operated in many ways, which can be optimized for different worldwide climates and conditions. Some basic information about CO2 as a refrigerant and a heat transfer fluid was covered in chapter 2.1.

Figure 14. Worldwide amount of stores with transcritical CO2 refrigeration systems. Numbers are from October 2018. (Danfoss, 2019)

4.2 CO2 System advantages and disadvantages

CO2 systems got some clear advantages. CO2 systems have significantly higher volumetric cooling capacity compared to conventional refrigerants. As an environmentally friendly and cooling capacities compared to systems with conventional refrigerants. As an environmentally friendly and natural refrigerant, CO2 has a competitive advantage to non-environmental friendly refrigerants, which are being targeted by legislations. Refrigeration system safety features are good in some areas, for example because of CO2's non-flammability and low toxicity. Very low viscosity of CO2 allows small pipe sizes and pressure drops in the system. This spares space, lowers component costs and makes economical pumping of CO2 in refrigeration system possible. Many characteristics of CO2 are different from traditional refrigerants and heat transfer fluids, as discussed in chapter 2.1. This bears both advantages and challenges, which also depend a lot on CO2 and compared refrigeration system types, design and details. Some main challenges of CO2 refrigeration systems are high operation- and standstill pressures and typically a complex design compared to traditional systems. High pressures and low viscosity of CO2 increase leak potential, even though leak is less harmful to environment compared to some traditional refrigerants. CO2 systems demand some components suitable for high pressures and good leak proofness. (Emerson, 2016)

4.3 Subcritical and transcritical systems

CO2 refrigeration processes and systems can be divided into subcritical and transcritical types. Subcritical and transcritical refrigeration processes are presented in logp-h diagrams in figures 15 and 16, correspondingly. Subcritical refrigeration process happens entirely below CO2's critical pressure. Consequently CO2 remains in gaseous state after compression, and rejects latent heat of condensing out of the refrigeration cycle via condenser. In transcritical process, CO2 is compressed into supercritical state. Heat rejection is made in a component called gas cooler, cooling supercritical CO2 in standard, supercritical pressure. Evaporation, expansion and compression happen partly or entirely in subcritical area also in transcritical systems. Subcritical cycle is used in lower temperature applications, freezers for example. Transcritical cycle is however practically needed for entire food retail shop solution, since efficient heat rejection often demands supercritical temperatures of CO2 in the heat rejection, especially in hot climates. If ambient air temperature is near or over CO2's critical temperature, 31,1°C, heat rejection demands supercritical operation. Otherwise CO2 in gas cooler can't be in high enough temperature, for heat to leave the refrigeration cycle to surroundings.

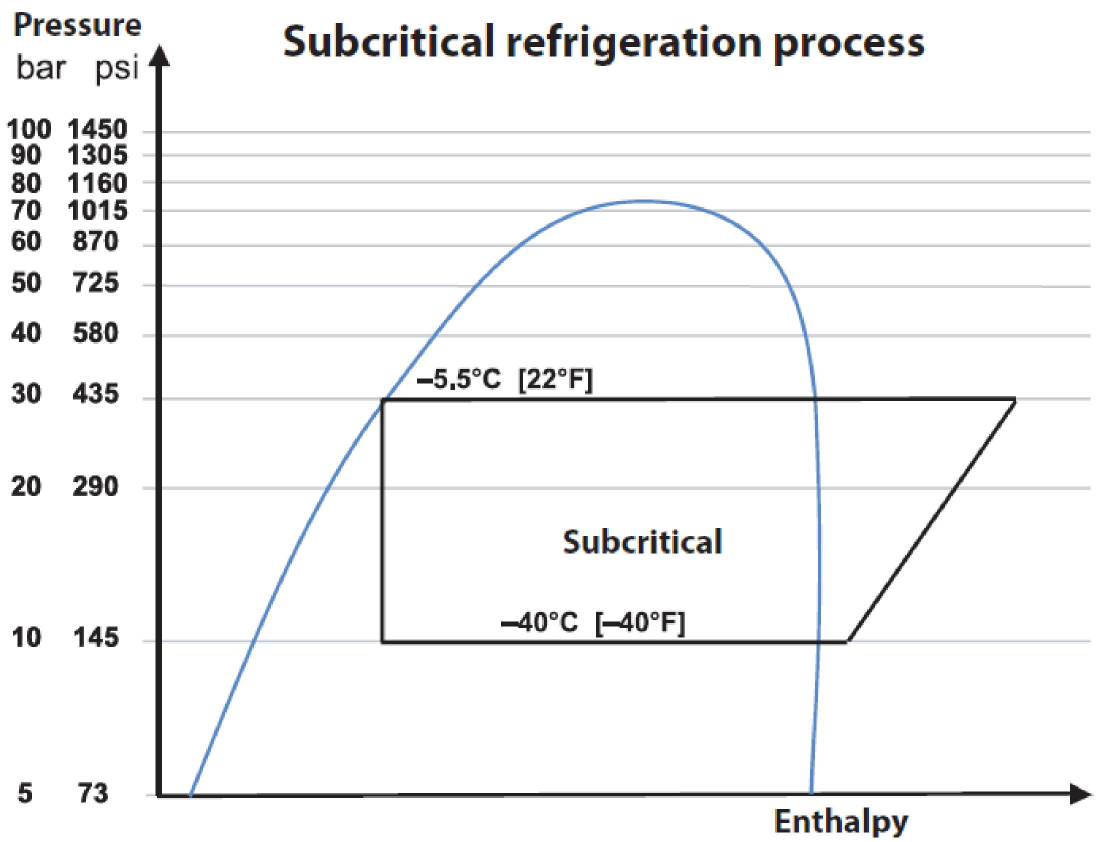


Figure 15. Subcritical CO2 refrigeration process in logp-h diagram. (Danfoss, 2013, pp.96)

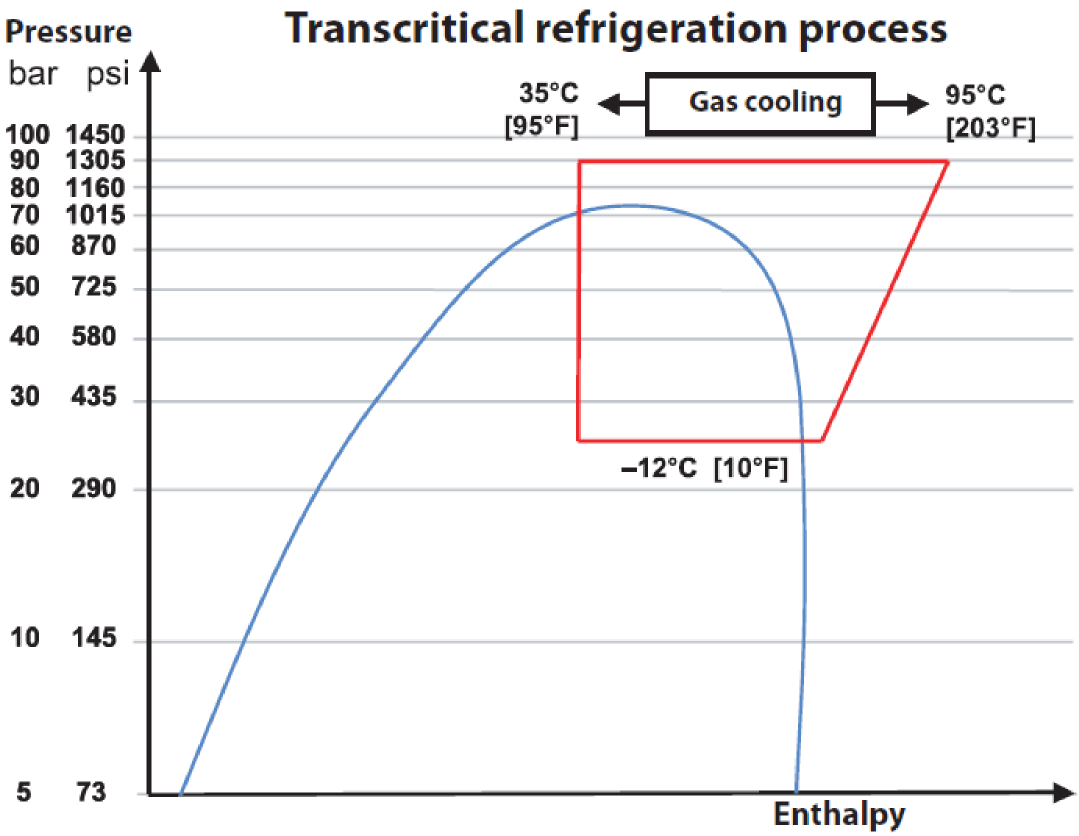


Figure 16. Transcritical CO2 refrigeration process in logp-h diagram. (Danfoss, 2013, pp.97)

Classic solution for refrigeration system capitalizing both subcritical and supercritical operation is a booster system, introduced later in this chapter. Booster system cools low temperature cooling targets (freezers for example) with subcritical cycles, and medium temperature cooling targets (just above 0°C refrigerator temperatures for example) with transcritical cycles. Booster system uses both cycles in the same refrigeration machinery, achieving benefits through integration. In figures 15 and 16 we can see, that subcritical process has a longer line of evaporation in enthalpy's direction, providing more cooling power per kg of refrigerant. Subcritical process cannot however reject heat to very high temperature ambient air or other heat sink. Transcritical process starts expansion from higher temperature, resulting in smaller liquid content available for evaporation. This is because transcritical cycle is usually used in warm climates and weathers, where ambient temperature is high. Ambient air in high temperature cannot cool CO2 cooler than its temperature in gas cooler/condenser. Transcritical cycle is feasible for heat recovery arrangements, because CO2 can be compressed into pretty high temperatures.

4.4 CO2 Refrigeration system types

4.4.1 Cascade system

Two typical CO2 refrigeration system types are a cascade system and a booster system. Cascade system is presented in figure 17. In cascade system, CO2 circulates in the area of cooling targets in two temperature levels. In over 0°C fixtures CO2 is evaporated in some below 0°C temperature and corresponding pressure. CO2 is expanded into lower temperature and pressure for freezer temperature fixtures. Ratio of CO2 going to each temperature level evaporators is determined by cooling demand in each temperature level, and can be adjusted. CO2 goes from both temperature levels to a common condenser, in which heat is removed to a compression refrigeration cycle with traditional refrigerant, R134A for example. This cycle finally removes heat into the environment or heat recovery. One advantage is, that cascade system needs lower pressure level in order to get the heat out of the system, by arranging heat rejection via some traditional refrigerant cycle. With CO2 only this would have demanded transcritical operation. Normal cascade system works with subcritical process in all refrigerant cycles.

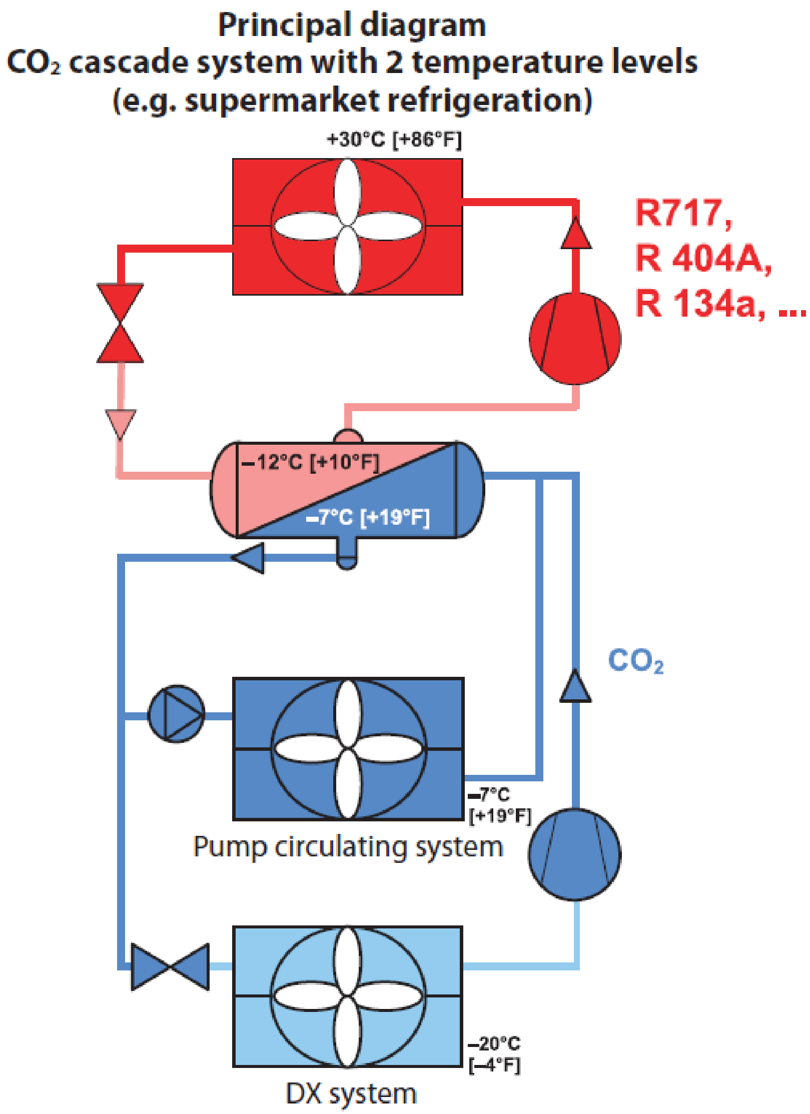


Figure 17. Principal diagram of CO2 cascade refrigeration system with two temperature levels. Common application in supermarket refrigeration. (Danfoss, 2013)

4.4.2 Booster system

CO2 booster refrigeration system is presented in figure 18. Liquid CO2 flows from flash tank to cooling fixtures of both temperature levels, MT (mid temperature, in a bit over 0°C) and LT (low temperature, freezer temperatures). Evaporation pressure and temperature is controlled with electronic expansion valves, ensuring suitable CO2 states in evaporators. For freezer fixtures, CO2 pressure and temperature are naturally dropped to lower level of these two. After evaporation, LT CO2 gas goes to LT compressors, which bring it to the same pressure level with CO2 gas coming from MT evaporation, and these CO2 gases get mixed to one fluid stream. After that, CO2 gases go to transcritical compression, bringing all CO2 into the supercritical state. Next CO2 goes through oil separation phase. Now supercritical CO2 goes to gas cooler, rejecting heat out from the cycle into the ambient air heat sink or heat recovery. Finally CO2 flows through high pressure valve to the flash tank, and process begins again. Booster refrigeration system uses only CO2 as refrigerant, but requires operation on high pressure levels.

CO2 booster process is a combination of sub- and transcritical refrigeration cycles. Both can also be seen in associated process logp-h diagram, but with a common heat rejection process. Refrigerant going to MT fixtures goes through a transcritical refrigeration cycle, and refrigerant going to LT fixtures is compressed two times before flowing back to flash tank and being distributed again to fixtures at either temperature level.

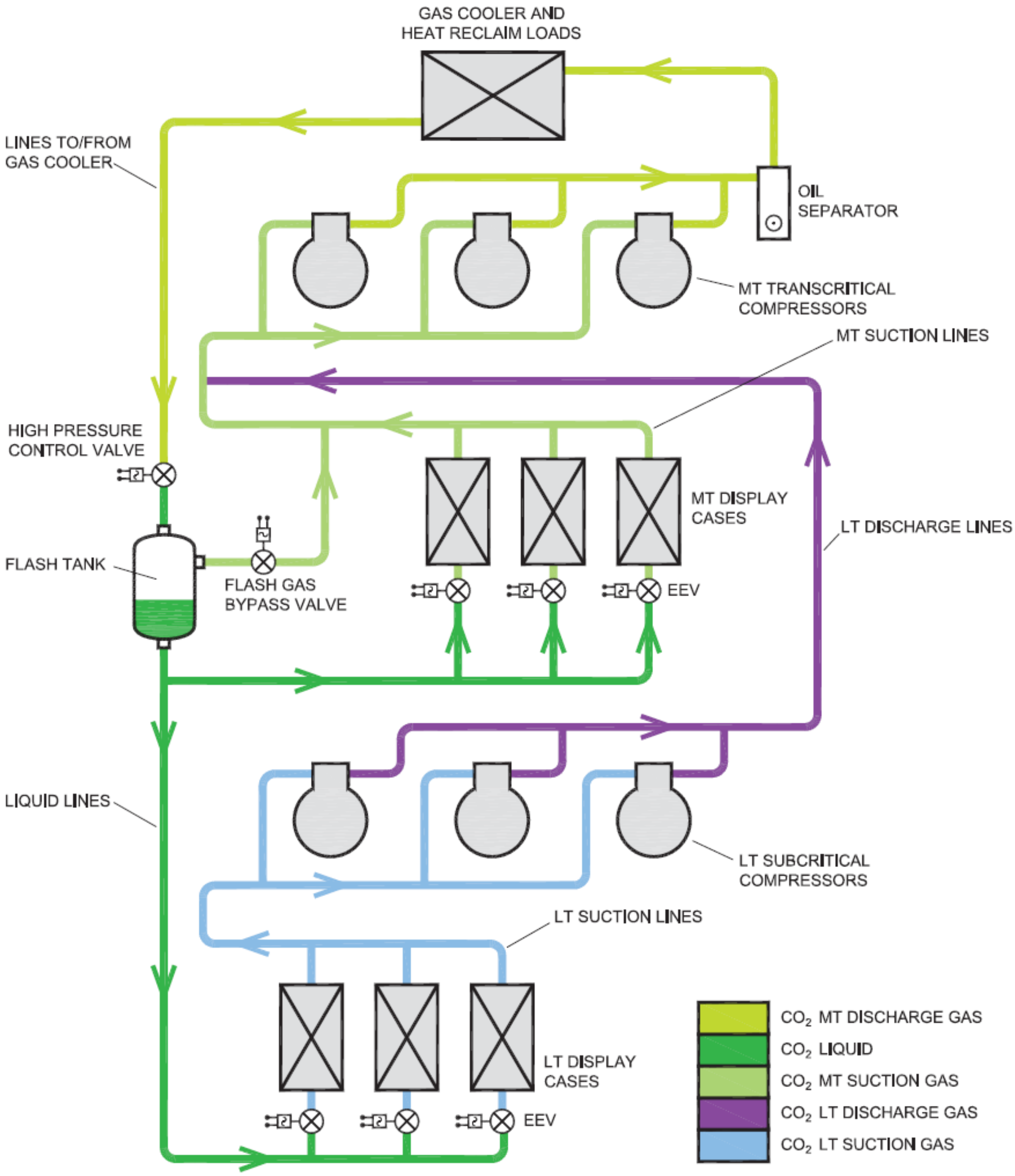


Figure 18. CO2 booster refrigeration system. (Hillphoenix Inc. 2016)

4.4.3 Advantages and disadvantages of CO2 cascade, booster and secondary cooling systems

Some advantages and disadvantages of transcritical booster systems, cascade systems and secondary cooling systems with CO2 are presented in table 6. Achieved efficiencies depend in the climate and weather. Transcritical CO2 booster systems have even better efficiencies than refrigeration systems with HFC refrigerants in mild climates. In warm climates, cascade and HFC systems have usually better efficiencies than booster systems. In booster systems using only CO2 is an advantage, whereas cascade system utilizes two simple systems with different refrigerants. One refrigerant means one set of properties and refrigerant-related cautions for the entire system. In secondary systems, CO2 can be used as heat transfer fluid, so one fluid for all refrigeration and heat transfer is possible with indirect refrigeration systems too. Even though integrated system provides some benefits, possibility for a system fault in one cycle affecting the whole combined system can be seen as disadvantage in booster and cascade systems, even though good quality design and implementation can lower this risk into a very low level. Low viscosity of CO2 makes small pipe diameters and very low pumping power consumptions possible. In Emerson (2016, pp.13-20), need for special, low power pumps is seen as a disadvantage, assumingly for current lower availability and unfamiliarity to refrigeration engineers. While CO2 systems and such pumps are seen to further generalize in refrigeration systems (Danfoss, 2018A), abovementioned problems can be turn over with lower pumping electricity consumption and compact equipment sizes, which also might give an opportunity for economical production, if pump structure can be kept considerably simple.

Table 6. Some advantages and disadvantages of transcritical booster systems, cascade systems and secondary cooling systems with CO2. (Emerson, 2016, pp.13-20)

System	Advantages	Disadvantages
Transcritical Booster	• One refrigerant • One system, lowest system costs • Better efficiency than HFC systems in mild climates	• LT applications require two-stage compression • System faults in coupled systems affect MT and LT • High operation pressures • Lower efficiency as HFC systems in warm climates
Cascade	• Two simple systems • LT with low R-744, the MT with a low GWP HFC refrigerant • Standard HFC components for medium and low temperature cycles • Better efficiency in warm climates	• Two refrigerants although R-744 can be used in the high stage • Temperature difference in the cascade heat exchanger reduce the efficiency slightly for the LT cycle • System faults in coupled systems affect MT and LT
Secondary	• Using R-744 as a secondary fluid using the latent heat, very low pump power required • Simple chiller system for the high -stage with readily available components (separate chiller for LT and MT) • System works at constant pressure without any pressure pulsation • Option to combine LT and MT, pump circulation system for the MT using R-744 combined with a LT booster system • Chiller could use low GWP HFCs or HCs	• Additional heat exchange and temperature difference slightly reduce the efficiency • R-744 pumps required • Pumps in this size are not readily available and are unfamiliar to many refrigeration engineers

5. Secondary cooling and indirect refrigeration systems

5.1 Direct and indirect systems

Compressor heat pump-based refrigeration systems can be of direct or indirect type. In direct systems, evaporator is straight connected or located to the space being cooled, and evaporating refrigerant extracts heat straight out from there. Condenser is correspondingly straight connected or located to the area where heat is rejected, outdoor air or water body for example. Condensing refrigerant therefore rejects heat out from the refrigerant cycle, straight into this heat sink area. Basic idea and main components of direct refrigeration system are presented in figure 19.

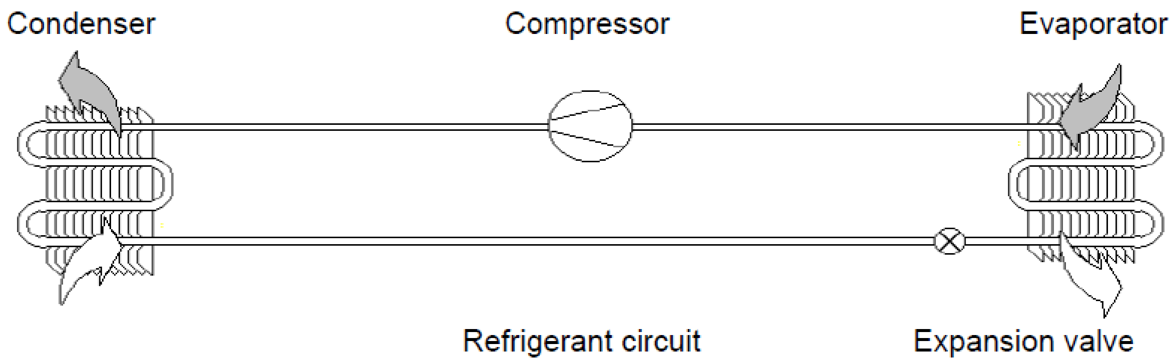


Figure 19. Basic idea and main components of direct refrigeration system. (Melinder et al, 2015)

Indirect systems use secondary fluids to transfer heat between evaporator and cooled space, and/or between condenser and heat rejection. This typically allows more compact arrangement of refrigeration machinery and smaller refrigerant dose. Secondary fluid can be chosen specially based on a good heat transfer properties, without having to worry about phase change properties (except that heat transfer fluid stays in liquid form in applied temperatures). Indirect systems are often used in systems with long piping demand to spaces being cooled or large amount of points where cooling is needed. Good examples of such systems are artificial ice rinks and large foodstuff refineries, respectively. (Melinder et al, 2015, pp.iii,1-7) Basic idea and main components of indirect refrigeration system are presented in figure 20.

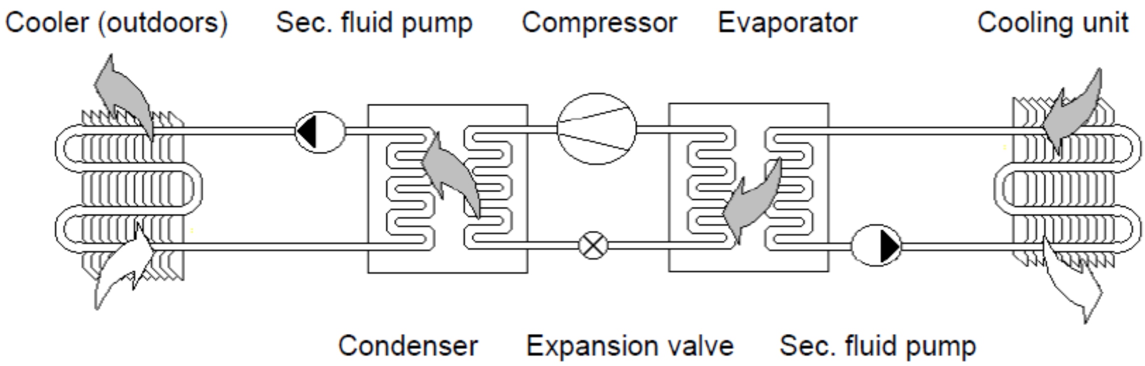


Figure 20. Basic idea and main components of indirect refrigeration system. (Melinder et al, 2015)

5.2 Energy calculation basics

Indirect refrigeration system with some nomenclature and temperature profiles is presented in figure 21. Refrigerant circulates in primary working fluid cycle, moving heat between secondary fluids from lower to higher temperature level with electricity-made work. Cooling is provided and heat is finally rejected in secondary fluid cycles. Secondary fluids can be brine or water-glycol mixtures for example. Refrigerant could be R134A or propane for example. Q_2 (W) is the heat extracted from the cooling target, the desired cooling capacity. Q_1 (W) is the heat rejected from the cycle, in dry cooler for example. E_c (W) is compressor work into the refrigerant, E_{P1} (W) and E_{P2} (W) are circulation pump works in secondary fluid cycles. Q_1 must include Q_2 and heat resulted from compressor and circulation pump works. (Assuming that heat and work is exchanged only in aforementioned points between the system and its surroundings. Piping in secondary fluid 1 side could well be left without insulation in real life indirect refrigeration systems, causing some advantageous extra heat rejection to surroundings, for example) t_1 (°C) and t_2 (°C) are condensing and evaporating temperatures, correspondingly. t_{HSink} (°C) is heat sink temperature and $t_{HSource}$ (°C) is heat source temperature. Heat sink could be outdoor air, and heat source supermarket refrigeration fixtures, for example. Heat sink and source temperatures are shown to change slightly in heat exchange process. This description applies with for example arrangements with counter current heat exchangers holding some dynamic, but limited fluid volume at the time. Fluid enters the heat exchanger, and cools or heats slightly before leaving the heat exchange. If we increase mass flow, temperature change is smaller for same heat exchange power, and vice versa. θ_1 (°C) is the temperature difference between condenser and heat sink temperatures. θ_2 (°C) is the temperature difference between evaporator and heat source temperatures. Δt_{SF1} (°C) is the difference in secondary fluid 1's highest and lowest temperatures during the circulation. Δt_{SF2} (°C) is the difference in secondary fluid 2's highest and lowest temperatures during the circulation. In design's point of view, aforementioned temperature differences should be big enough for achieving sufficient pinch points in all heat exchangers. Larger temperature difference transfers more heat per mass flow. On the other hand, small temperature differences can be compensated with bigger mass flows, if feasible and pumping costs remain manageable. Successful design in this subject can be seen a combination of suitable temperature difference and mass flow, resulting in desired power of heat exchange. This aspect was important in our project's design in terms of determining an efficient, well available and controllable values for all heat exchangers and fluids.

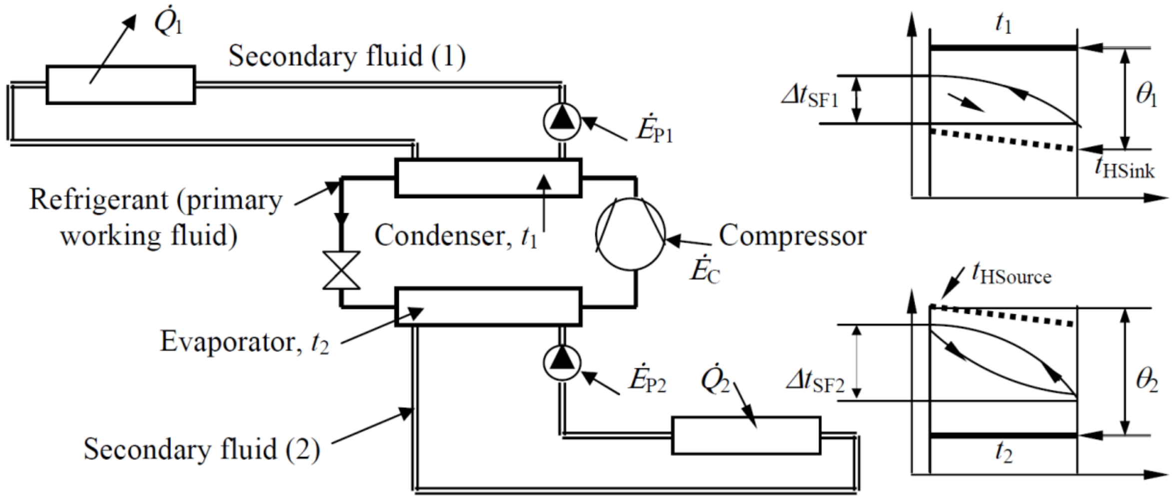


Figure 21. Indirect refrigeration system with some nomenclature and temperature profiles. (Melinder et al. 2015)

5.3 Artificial ice rink

One remarkable indirect cooling system application is an artificial ice rink. It is a good example of using indirect cooling in applications, where large area of heat exchange is needed, and implementation with a vast evaporator solution would not be practical and/or economical. Flow diagram of an artificial ice rink's secondary coolant side is presented in figure 22. Heat transfer fluid is pumped through secondary coolant side of the compression refrigeration cycle's evaporator, where it is cooled. Cooled heat transfer fluid goes then under the ice, absorbing heat from it and therefore cooling the ice. Then heat transfer fluid continues to circulation pump and the cycle starts again. In ice rink application, so called heat exchanger for cooling supply is large and has a lot surface area, the entire ice field. Ice rink temperature should be as even as possible. For good quality ice it is important to get even group of parallel cooling tubes under the ice, covering the whole ice area. (Melinder et al, 2015, pp.1-23)

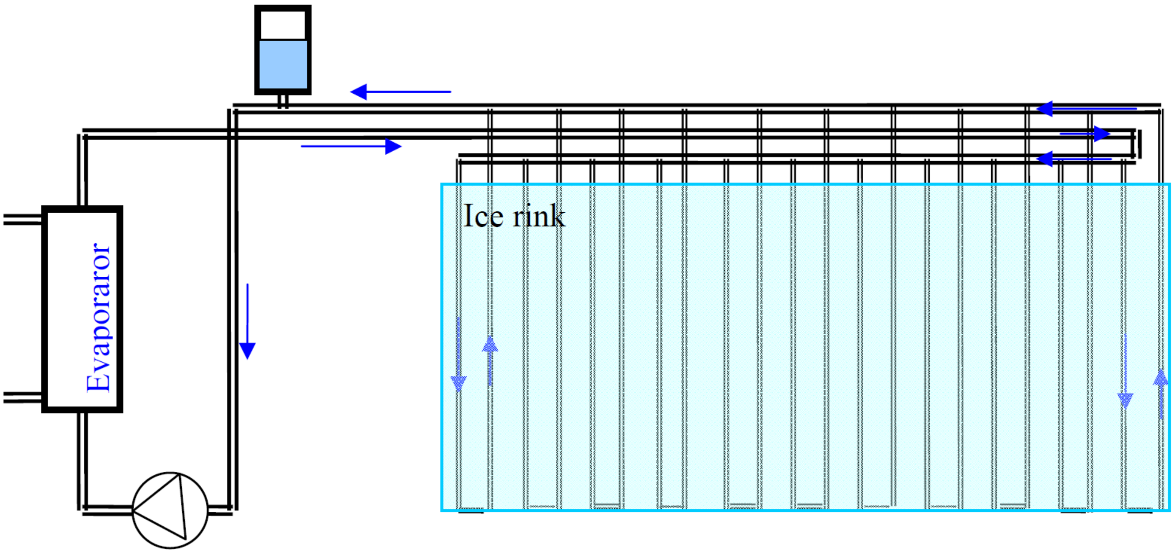


Figure 22. Flow diagram of an artificial ice rink's secondary coolant side. (Melinder et al. 2015, p.7)

5.4 Pressure drops in indirect systems

Suitable pressure drops are important for performance and operation of indirect refrigeration systems. Indirect systems are used in many applications with long piping or numerous cooling targets (Melinder et al, 2015, p.iii). Long piping and many cooling targets may set some challenges for keeping pressure drops in well working and economical level. Pressure drops can be calculated separately for piping and components, and this approach was applied in our project's design, introduced in more detail in chapter 6.10. In closed system, only the flow resistance of heat transfer fluid has to be overcome by the pumping work (Melinder et al, 2015, p.89). Energy needed for lifting fluid in raising pipes is balanced by descending fluid in the system. Since loop is closed, net lift of fluid must be 0. (ITT Water & Wastewater Suomi Oy, n.d.)

Pressure drop of a pipeline, Δp (Pa) can be calculated from equation 4. Equation 4 is introduced in for example Kotiaho et al. (2004, p.56). Pressure drop is proportional to length of pipe L (m), density of fluid ρ (kg/m^3) and flow's friction coefficient λ (λ is "virtauksen kitkakerroin" in Finnish in Kotiaho et al. (2004)). λ is calculated or obtained differently for laminar and turbulent flows. Pressure drop is inversely proportional to pipe diameter D (m). What should be noted, is that pressure drop is proportional to flow velocity's v (m/s) square. With higher flow velocities, pressure drops can increase fast. What is further notable, is that flow velocities can be lowered easily by choosing a larger pipe diameter. In our calculations it was seen, that change to even a one size larger pipe diameter made a great relief in expected pressure drop with given cooling power. Simply decreasing heat transfer fluid mass flows also decreases flow velocities, but this would then have to be compensated by larger temperature changes in all heat exchangers for the same cooling power.

$$\Delta p = \frac{\lambda * L * \rho * v^2}{2 * D} \tag{4}$$

Local pressure losses can be calculated with equation 5. This equation and many examples for local pressure coefficients ξ can be found from Kotiaho et al (2004, pp.57-60). Local pressure losses result for example in pipe curves and inlets and outlets from a pipe to a large fluid body, in buffer tank for example. Also all pipe diameter changes generate some pressure losses, even though typically very small, even neglectible for practical calculations.

$$\Delta p = \frac{1}{2} * \xi * \rho * v^2 \tag{5}$$

Reynolds number is often used dimensionless number in flow calculations. In our project, it was utilized to calculate whether pipe flow is laminar or turbulent, and to calculate λ or read it from the charts. Reynolds number for pipe flow can be calculated with equation 6, where v (m/s) is flow velocity, D (m) is pipe diameter, ρ (kg/m^3) is fluid density, μ (Pa*s) is dynamic viscosity of fluid and ν (m^2/s) is kinematic viscosity of fluid. Reynolds number has some different definitions for different geometries, for example flat plate has different definition than pipe flow. (Cohen et al. 2010)

$$Re = \frac{v * D * \rho}{\mu} = \frac{v * D}{\nu} \tag{6}$$

6. Design of an indirect refrigeration system for laboratory testing

In this chapter we report, how we designed some new laboratory facilities in Viessmann's R&D laboratory testing refrigeration products. System installation and commissioning were a large part of the project, which will be described and commented on chapter 7. In engineer's perspective, associated problem solving offered highly educational experiences. Learning about refrigeration technology and viewing some relevant literature was done throughout the project, especially in the first couple of months. Laboratory was located in the city of Porvoo, Southern Finland. My commission was broad-minded in the beginning, and became more precise during first months of working. Finally after a fair month, we decided the official goals for our project. Priorities of our project, introduced already in chapter 1.3, are listed in table 7. Some minor changes and the entire priority number 2 were booked in as late as August. Our workflow with this project and master's thesis consisted of multiple phases, collected on appendix 3. Working was versatile and classification in appendix 3 is not very precise on single day level. Many days included for example 4h of system- and concept design, and rest of the day was spent on 4 times short 0,5-1,5h tasks of other phases. Classification of tasks is not also exact all times, fetching and learning to use equations for excel calculations for example can be seen as studying refrigeration, especially when some background knowledge and supplementary information is often examined at the same time. Appendix 3 can however be seen as fairly accurate for viewing total time spent on each working phase and approximate periods and order of working phases.

Table 7. Priorities of our laboratory development project.

1	Cooling for LAB1 refrigeration products testing
2	Cooling for various testing in K-buildings hall
3	Cooling CO2-system, preventing heat reject problems in summer
4	Cooling for LAB1 ambient air

Studying refrigeration began soon after my employment began, and was continuous for a couple of months; alongside with other more project focused tasks. After around week 22 focus was more shifted to design tasks and calculations of the project. Studying was done every now and then in later weeks too, since demand for learning about refrigeration theory and technology came up sometimes. Many literature sources were found, mostly on internet. I used Scopus and Google Scholar most of the time, or simply Google search to reach some company's websites in refrigeration branch for example. There were a few main sources which proved themselves valuable, in which I made systematic studying from throughout the project. Most important of those were Danfoss (2013), Hakala et al. (2013), Laitinen et al. (2016), Melinder et al. (2015) and Siemens (2017). In principle we can say, that more detailed questions we got during the project, more important the discussions with experienced coworkers and/or manufacturer or selling party became, since small details were experienced to be hard to find in scientific publications, especially with given, special case.

Design of the refrigeration system can be divided into two categories, system- and concept design and detailed design and component design- and considerations. System- and concept design was mainly done earlier, but had to be reconsidered few times when changes or new information appeared. Main reconsideration on later stage was adding fourth cooling target, cooling for various testing in K-building hall to the system plans. Installation had already started, which ruled some design chances practically out. A good solution was still discovered, to install fourth cooling target piping in parallel with two other, already planned cooling pipelines. Other first planned option would have been to do installation parallel with two heat exchangers, into same main pipeline (see appendix 4). This would however have disturbed testing with products with integrated heat transfer fluid circulation pump, since suction would have happened though already installed pump, or demanded installation of optional pass-by.

Detailed design included a lot of component considerations, applicability research and practical problem solving. System was to be installed, so there was no room for leaving any design into a theoretical level, if some questions came up. Heat transfer fluid selection is a good example of this. In theoretical calculations we would have not needed to change the heat transfer fluid to propylene glycol-water mixture because of ethylene glycol's health hazards. We would have just reminded in the end of calculations, that toxicity causes some restrictions to use in some applications. Here we had to change the heat transfer fluid, check the proper frost protection because it was lowered a bit because of the change and change some values to calculations, taking changes into account. It was noticed to be very important for a good result, that any encountered critical issues were solved comprehensively before continuing the design. All design work was more or less connected, and many detailed issues lead to work with concept design, and vice versa.

Design phase was known to be important for projects success, schedules and costs. After certain procurements and installing works were done, changes to design were expected and seen to be harder and possibly increase costs and demand time. Some useful literature about this phenomenon can be found for example with key word "cost influence curve" with online scientific search tools. Timing of accumulated costs, abilities to influence on costs and use of resources during a project are presented in figure 23. Figure 23 can be seen to represent a typical way that aforementioned aspects are realized in projects involving presented phases, such as design and construction. According to gained experience, figure 23 pretty was valid for our indirect refrigeration project too. Cost of working hours was without a doubt the biggest expense in our project. I would estimate that cost structure was more working hour-concentrated than in an average project modeled by figure 23. Time of high or semi-high resource use was however longer than in figure 23's model, because installing took a long time.

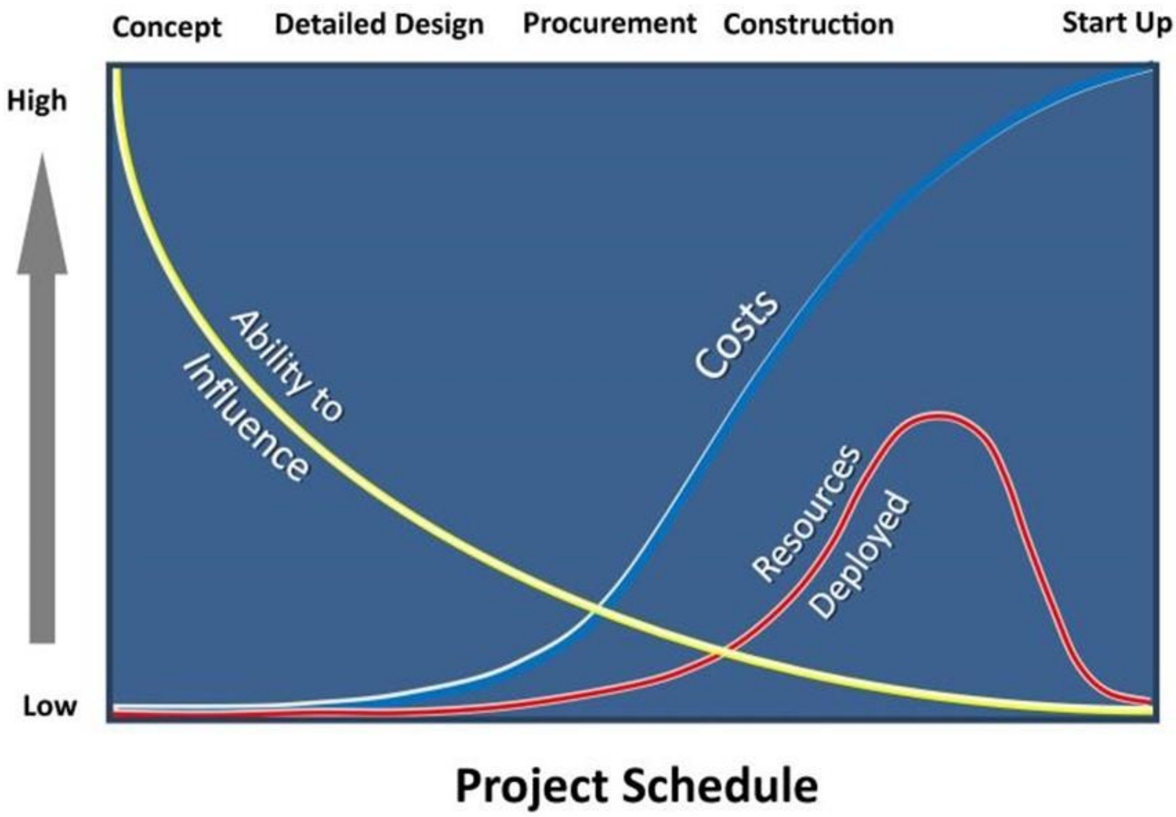


Figure 23. Timing of accumulated costs, abilities to influence on costs and use of resources during a project. (University of Toledo, 2018)

6.1 System- and concept design

System- and concept design started with project introduction. First I got to see some preliminary drawings about the wanted system, and got to see the facilities, where system was going to be installed to. There was some equipment already reserved for the installation, Vitocal 300-G heat pump and Wikora WKS 305 buffer tank for example. I examined their technical manuals, and studied briefly more detailed operation principles and compatibilities, to get a good image on how the design should be.

Most important drawings about our project's refrigeration system are collected in Appendixes 4, 5 and 6. Appendix 4 shows the main drawing and a flow diagram of the system. It was used as the instruction for our projects installers and electricians, and will be the most important updated drawing about the system in the future. Since the document is used for installing, some parts of it are in Finnish, and this feature is maintained in appendix 4-6 to show the newest drawing in its original form. Appendix 5 shows state points used in Excel-calculations. Components are shown in appendix 6. Appendix 4 accompanied with state point and component numbers could have delivered all aforementioned information in a single drawing, but I believe that specialized drawings are more efficient in their specific purposes.

CAD-modeling of system was made in a single LibreCAD drawing utilizing different layers. Layers allowed us to view and print different set of active layers in one picture, producing prints for different purposes. Most important drawings for daily work were the main drawing for planning and installers and the one showing state points for design and calculations.

There were two main design tools I used for the entire project, versatile Excel calculations and modeling of the project's refrigeration system in LibreCAD. Excel was chosen because of its already well known performance capabilities and table-features. LibreCAD was selected among accessible 2D CAD-softwares. LibreCAD is a free open source CAD-program, kept up by a dedicated community. Since program is freely available, program can be accessed anywhere, without company license. Viessmann has Vertex Systems commercial and very versatile programs, but LibreCAD was seen sufficient for simple 2D modeling. Quick search gave a simple and comprehensive image of LibreCAD, which was proven right in this project according to my user experience.

Many other programs were used throughout the project for various shorter tasks. SWEP SSP was used for design of heat exchangers, and picking the right ones for the project from the supplier. SWEP SSP includes also a highly useful fluid property calculator, which was used to fetch properties of different water-glycol mixtures under various circumstances. Unilab Coils was for a special task of heat exchanger design for CO2 air cooling. This task is introduced in more detail in chapter 6.4. Vertex G4 and Vertex Flow were used in the same task for modeling the heat exchanger in detail and giving the evaporator-product factory the demanded documents for production.

Our project's refrigeration system is designed to operate in certain pressure levels. A constant pressurization relative to atmosphere has to be maintained all the time, for example to prevent air entering the system by time. Otherwise pressures should stay somewhat low; under 3 bar-g that will open safety valves in each piping. Pressurization is not needed in secondary cooling circles, but some pressure variation has to be allowed, so that normal temperature changes in the system are allowed. Original Finnish document placed in the refrigeration plant's info wall for heights, pressure levels and operation of the system can be found in appendix 10.

Four cooling targets in a various locations set some restrictions on the suitable system layout. Centralized cold production and storage with Vitocal 300-G heat pump and Wikora buffer tank were a clear choice. Another thing to decide was the distribution of cold heat transfer fluid to cooling targets. Both pumping arrangement and piping layout were to decide. Arrangement with many smaller pumps in main circle was chosen, because of a chance to use only demanded amount of pumping power and electricity, and because this way additional control valves were not needed for flow adjustment. Another choice could have been to use a single pump with good power controllability in main circle and valves to adjust the wanted flow to each branch. For testing of products with an own pump, a separate line could have been introduced. Pump lift height could be then adjusted to match the overall consumption. This choice could have potentially lowered the amount of circulation pumps in the system from current 5 to 3. Overall results can be seen to be good with the selected arrangement. Cold supply could have also been arranged with a main pipe going around the technical room, even though this was not yet investigated with this small amount of cooling targets.

6.2 Operating modes

Our project's refrigeration system was planned to operate on several different modes, which are presented in table 8. The most common mode of use will be mode 1, cooling to EN-LAB1 testing and air. Cooling for various testing in K-hall is an alternative option for cooling to EN-LAB1 cabins. These two modes are designed not to be used at the same time, even though at small cooling loads it is technically fully possible. In mode 1 CO2 circle is passed with closing valves. When there is a hot weather and cooling to CO2 system is needed, two other operation modes are also possible. In these modes main circle to EN-LAB1 testing is closed with valves, and heat transfer fluid is circulated in the double HE-circle. Cooling is supplied to CO2 system, and also to EN-LAB1 air cooling if needed. EN-LAB1 circle can be also closed with valves, if for example weather is really hot and all cooling capacity is reserved for CO2 cooling. One Vitocal 300-G can supply no more than 14kW of cooling power, which rules out the option of simultaneous operation of all four or potentially even three cooling targets, or cold supply to EN-LAB1 testing and CO2 with full designed cooling power. Simultaneous operation of even all four cooling targets is however technically fully possible with moderate and small cooling loads, especially when efficient system operation is well learned.

Table 8. Planned operating modes of our project's refrigeration system.

	Operating modes * cooling to K-hall testing alternative option
1	**Cooling to LAB1 cabins and air.** CO2 circle passed. *
2	**Cooling to LAB1 air and CO2.** Heat transfer fluid circulated only in a double HE-circle. LAB1 cabins-circle
3	**Cooling to CO2 only.** LAB1 cabins-circle closed and LAB1 air circle passed.

All in all the system is multifunctional and its configuration theoretically enables very many different operating modes, even though it is specifically designed for a few purposes. Multifunctional operation with low or moderate cooling loads is expected to have good process controllability. Major limiting features are Vitocal 300-G's cooling power and piping made for certain volumetric flows. System is designed to use with propylene glycol-water mixtures, but the concentration can be changed with a moderate effort.

6.3 Cooling for the CO2 system

Current CO2 system was designed for food retail refrigeration, and had two high pressure compressors, with frequency control in only one of them. This is a standard and working arrangement in such applications. Laboratory testing however, demands considerably even cold supply temperatures for accurate and well verifiable results. In warm weather second compressor goes on, and since frequency control does not allow sufficient partial loads, cold supply temperature varies for a moment. This would not be a problem in terms of refrigeration, but is undesirable during testing for aforementioned reasons.

Various CO2 cooling solutions were examined and compared. Indirect cooling with heat exchangers was selected for implementation. Vitocal 300-G heat pump provides the cooling to propylene glycol-water heat transfer fluid. This cooling is transferred to another propylene glycol-water heat transfer fluid in plate heat exchanger. This fluid finally cools outdoor air going to CO2 system's condenser/gas cooler unit, helping CO2 system to operate better in warm weather conditions. This arrangement supplies cold with many serial heat exchange events, which lowers the overall efficiency. One restriction of this arrangement is also, that cooling capacity will be enough only, when outdoor temperature is just a few °C higher than allowed for successful operation before the arrangement. Reason to this will be shown in more detail in chapter 6.9. However, this case was expected to cover most of the warm days, currently setting challenges for CO2 system. In task handout it was requested, that solution should not demand modifications to current CO2 system, or demand for example opening of CO2 piping. This aspect was met with selected arrangement very well. Other strengths were good compatibility with the whole indirect cooling system and well known technology. Reliable and well known solution with moderate effectiveness was priorized over more effective, but potentially troublesome one.

A couple of other solutions were considered in CO2 cooling concept design phase. Equipping also another compressor with frequency converter, and this way improving the process controllability could have possibly been a precise solution, but it was ruled out for some practical reasons. This solution with its possible effectiveness and challenges was not further studied. I saw pretty good potential in this option, and would like to study compressor controlling techniques closer in the future, but for now efforts were focused in the selected solution and making the most of it.

Potential of cooling with well positioned and shaped ice batch was also calculated. Table 9 shows values of ice cooling calculations and the theoretical maximum cooling power with given cooling time, which corresponds to melting time of one ice batch. A solid ice batch still properly fitting under the CO2 condenser/gas cooler unit, has a theoretical maximum cooling capacity of around 310MJ, allowing constant cooling power of 8,6kW for 10 hours. Latent heat of melting is considered in calculations. Sensible heat from cold water is excluded in calculations, because water is expected to flow away and not be able to significantly cool the air flow going to the aforementioned unit. Cooling time of at least 8 hours should be considered, if one batch is expected to provide cooling for the entire warm summer day.

Table 9. Values of ice cooling calculations and theoretical maximum cooling power with given cooling time.

Ice cooling	
L (m)	1,8
W (m)	0,8
H (m)	0,7
V_ice (m^3)	1,01
ρ_ice (kg/m^3)	920
m_ice (kg)	927
Latent heat of melting (kJ/kg)	334
Q_cool (kJ) (All ice melts)	309738
t (h)	10
t (s)	36000
P_avg (t) (kW)	8,6

Ice would have been placed under the CO2 condenser/gas cooler unit, in restricted space. Ice geometry should enable a large surface area to be in contact with air, and constant air flow would increase the melting pace compared to ice located in still or quite a still air. A solid ice block would most likely melt and release cold to air too slowly, so ice would have to be introduced to the air flow in slices or some other geometry allowing more contact to get the desired cooling power. Some air-regulating plates would have been installed to enhance the contact with ice. Snow could have been used instead of ice due to its perhaps better availability, but this would have decreased the cooling potential available in space under the condenser/gas cooler unit. Good sides of ice cooling solution would have been the possibility of simple installation and an easy switch between active summer mode and the inactive mode during the rest of the year. For example with heat transfer fluid-based cooling, fluid has to be circulated every now and then to keep the circle in a good shape, and some other maintenance must be sometimes done as well.

From table 9 can be seen, that around 1m^3 of ice has just the needed amount of cooling energy in it, but no more than that. In reality, it would be hard to utilize the cooling energy of the ice efficiently enough, so that clearly more than 1m^3 of ice would not be needed. For practical reasons, this option was ruled out pretty soon. Constant need for heavy ice supply and availability of suitable ice would have been challenges, and there was not a known, practical arrangement in which we could control the melting pace of ice and therefore adjust the desired cooling power. All in all, practical implementation would still have needed a lot of work and the possible result was not seen worth investigating further at the time.

6.4 Heat exchanger to CO2 system

Selected solution demanded a suitable heat exchanger to be placed under CO2 systems' condenser/gas cooler unit. Heat exchangers task is to cool air before it enters the unit during warm weather, improving the CO2 process and its controllability. Heat exchanger was decided to be manufactured in Viessmann's local evaporator factory in Porvoo. Heat exchanger was designed by me and my advisor, I did the drawing in Vertex G4 and an experienced designer made the last editions and ordered the manufacturing.

Two different phases can be seen in design and modeling of the heat exchanger. First, suitable heat exchanger thermodynamical performance and geometry were calculated with Excel and Unilab Coils-program. Most equations used in calculations are introduced in theory of heat exchangers in refrigeration systems and humid air calculations, chapters 2.2 and 6.9. Second, suitable documents had to be created in Vertex, for our factory to be able to deliver the heat exchanger. Manufacturing based on PDF printed from Unilab Coils was not enough to allow production.

Heat exchanger was produced successfully and it was turned out to fit its installation area perfectly. Pictures of produced heat exchanger from two directions are presented in figure 24.

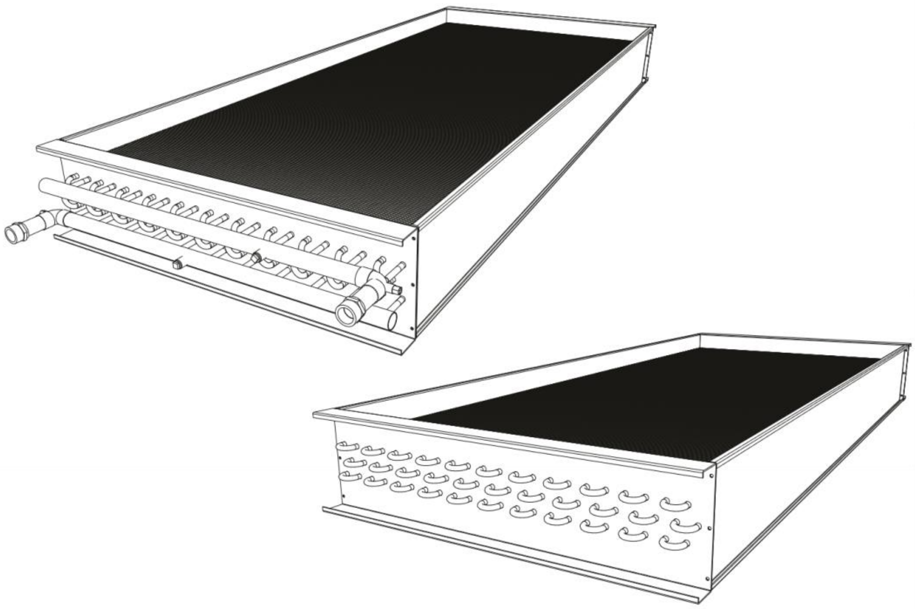


Figure 24. Pictures of produced heat exchanger from two directions. Picture is printed out from Vertex G4 program.

6.5 Cooling powers and temperature levels

First, I determined some key operating values for our project's refrigeration system. Cooling power, temperature and temperature difference values are collected in table 10. Powers are evaluated mostly based on the task handout, what was the requested cooling power for each cooling target. Peak cooling powers were requested roughly as 10kW for EN-LAB1 testing (or alternatively for various testing in K-hall), 5kW for EN-LAB1 air cooling and 10kW for CO_2 summertime cooling. Both largest peak capacity and maximum average capacity for continuous operation were given. Desired temperature levels for EN-LAB1 testing were specified around -5°C, and after discussions 5°C was estimated as an appropriate temperature difference over the EN-LAB1 cooling fixture's sensible heat exchanger. These temperature levels were used for EN-LAB1 air cooling too, since good suitability for an application was noted and this resulted in some design advances, for example simplicity of system. Since CO_2 cooling heat exchanger was put in series after EN-LAB1 air cooling heat exchanger in our arrangement, temperature levels of heat exchange were expected to be a bit higher. Cooling medium was also calculated to give more, about 10°C of its sensible heat away in CO_2 heat exchanger, demanding smaller mass flow for the same cooling power.

Table 10. Cooling power, temperature and temperature difference values of our project's refrigeration system. Values given for each of the four cooling targets. (Same supply to EN-LAB1 and K-hall)

Refrigeration targets	EN-LAB 1 cabins or K-hall	EN-LAB 1 air	CO2 cooling
Peak P_cold (kW)	10	5	10
Avg. P_cold (kW)	7	3	5
T_in fluid (°C)	-5	-5	0
T_out fluid (°C)	0	0	10
ΔT fluid (°C)	5	5	10

Temperature differences and therefore also cooling powers are naturally different depending on the use of the system. If cooling begins with cooling target in ambient temperature, difference between cold heat transfer fluid and fluid being cooled is large, and heat transfer fluid's temperature change will be also larger over the sensible heat exchanger. This leads naturally to a larger cooling power. When desired temperature is achieved in the cooling target, heat losses to environment determine how much cooling power is needed to keep the temperature low. In our project's refrigeration system, Vitocal 300-G heat pump's cooling power decides how quick desired cooling temperatures can be reached, and how big total losses (W) in cooling targets can be compensated by cooling. Temperature differences in table 10 are a lot more than required for almost steady state operation with typically tested products, when cooling target's cold temperature is maintained. Cooling targets are kept in around between -6 and +5°C depending on the tested product. This is a good thing, it helps us achieve test temperatures quickly, gives us some overcapacity to test products with a high cooling demand, we achieve suitable compressor cycles (avoid short cycling) and we can utilize our huge, 300L buffer tank longer before Vitocal has to start charging it.

6.6 Refrigeration machinery performance

After cooling target values, I calculated demanded values for refrigeration machinery performance. Cooling power, hot fluid side heating power and electric power consumption of our project's refrigeration system are presented in table 11. Vitocal 300-G is supplying the cold, so values correspond to its duties. Maximum design cooling powers were calculated as simultaneous cooling of EN-LAB1 testing and EN-LAB1 air, both average and peak. Energy efficiency ratio (EER) for Vitocal 300-G in operation was assumed to be 3. With EER and cooling powers, demanded electric power was calculated. Vitocal's compressor was assumed to add heat to refrigerant with ratio of 0,5 compared to total compressor work. This is a conservative assumption, rather exaggerating needless, but mandatory heat power added to the refrigeration cycle from compression. With this ratio we calculated heating powers to hot side fluid of Vitocal 300-G. More heat has to be removed than the cooling power, because compression unintentionally adds some heat to refrigeration the cycle, which also has to be removed, eventually in outdoor air in our system's case.

Table 11. Cooling power, hot side fluid heating power and electric power consumption of our project's refrigeration system. Values calculated for Vitocal 300-G heat pump based on table 10 information and additional knowledge of our system.

Refrigeration machinery	Vitocal 300-G
Peak P_cold (kW)	15
Avg. P_cold (kW)	10
Peak P_hot side (kW)	17,5
Avg. P_hot side (kW)	11,67
COP (EER in cooling)	3
Peak P_elec (kW)	5
Avg. P_elec (kW)	3,33
Compressor Heat / P_elec	0,5

6.7 Specifying heat transfer fluid properties

Properties of used propylene glycol-water mixtures had to be specified in the beginning of our Excel calculations. Propylene glycol contents of 30-50% depending on piping were chosen in concept design phase due to co-worker's recommendations and properties presented on appendix 2. We used 3 sources in defining and double checking the used values. Our propylene glycol mixtures were made of water and Antifrogen L 100% propylene glycol, and manufacturer's product documents were the first place we fetched values from (Clariant

International Ltd, 2014). All values were checked also with SWEP SSP Fluid property calculator (SWEP, 2019A). Engineering Toolbox was sometimes also used, if additional way of value presentation was wanted or two first mentioned did not offer us a double check possibility (Engineering ToolBox, 2001). Properties of 30% and 50% propylene glycol-water mixtures are presented in table 12.

Table 12. Properties of 30% and 50% propylene glycol-water mixtures.

	30% propylene glycol + 70% water	50% propylene glycol + water
ρ (kg/m^3)	1030	1050
Cp (kJ/kg°C)	4	3,5
v (m^2/s) (-10°C)	2,0E-05	3,6E-05
v (m^2/s) (+10°C)	6,5E-06	1,1E-05

Properties were clearly dependent on temperature and propylene glycol content, which had to be taken into account. Viscosity of propylene glycol-water mixtures raises quick when temperature decreases, and acceptable viscosity had to be specified in the lowest possible operation temperature. We considered it as -10°C, a couple degrees colder than what Vitocal 300-G will supply in coldest in our refrigeration system. Heat transfer fluid temperature can go below -10°C in wintertime with two pipings going outside, if system is not used for a while, because of heat transfer with cold outdoor air. In such case, operation must be started with a small heat transfer fluid mass flow, and raised to intended mass flow after higher temperature – and therefore an acceptable viscosity is achieved. Freezing in 50% mixture will start in around -32°C, but first a slush ice, with no bursting effect due to incompressibility is formed. According to Antifrogen L product sheet, around -50°C is demanded for a bursting effect with 50% mixture (Clariant International Ltd, 2014).

Density of propylene glycol is slightly higher than water's, and a mixture density is raised a bit with increasing propylene glycol content. Specific heat capacity of propylene glycol is lower than water's, so it is lowered in mixture with increasing propylene glycol content. All in all density and specific heat capacity were assumed constant, because of fairly small changes along the designed temperature interval. Viscosity cannot be assumed constant, but was calculated in worst case scenario during refrigeration system operation. Possibility of even colder – and worse conditions according to viscosity was acknowledged, and system's safe and practical return to acceptable operating conditions was noted to be manageable, by above mentioned means.

6.8 Refrigeration system state point calculations

After our project's refrigeration system layout drawings proceeded in LibreCAD, we started to define state points to heat transfer fluids for process calculation. After some time, final state points were specified. Selected properties of state points 1 – 19 during maximum designed cooling loads are shown in appendix 7. State point locations are shown in appendix 5's state point drawing. No state points were specified for various testing piping in K-hall, but wanted maximum cooling powers and flow rates are smaller than what is reserved for the primary alternative, EN-LAB1 cabin cooling. Calculations were ready when K-hall cooling option was ordered, and calculations with state points 3 and 4 were enough to assure that capacity and designed performance are more than enough for this option too.

Mass flows needed for cooling supply were calculated from equation 1. We have a certain cooling power demands, heat transfer fluids thermodynamical properties and estimated available temperature differences, which determine suitable mass flows. Temperatures were taken from table 10 or concluded based on them and suitable temperature levels and differences. For example it was estimated, that in hot summer day heat transfer fluid has to flow through the dry cooler in at least 40 – 47°C, to be able to dump heat to ambient, possibly even 32°C air, with fair margins considered. Pipe inner diameter was selected as 25,6mm in almost all piping, which is speculated more in chapter 6.10. Pipe lengths were evaluated based on our LibreCAD drawing and knowledge of the installation location. Even though state point refers to a specified spot, in piping length all areas of piping were classified to lie on some state point's area, so that all piping was covered. This approach proved to be quite suitable for our calculations. Pipe volumes are calculated based on pipe dimensions and lengths.

Appendix 7 had to be constantly updated as choices were made and details changed. For example, initial plan was to use ethylene glycol-based water solution as heat transfer fluid in two circles going outside. This was changed to propylene glycol-based water solution, because of health and safety aspects. This changed heat transfer fluid's thermodynamical properties, and updates had to be made. Heat transfer fluid pressures are not specified in calculations, since there was no need for pressurization for cooling process purposes. Pressure levels in the system were generally set as low as possible, but high enough to prevent cavitation in circulation pumps. State points 14 and 15 were outside our system, and only crucial thing we needed to calculate was the supplied cooling power in plate heat exchanger.

A sight glass can be installed to some pipeline when system is anyway emptied from liquid, to add a way of examining the system, air venting efficiency and heat transfer fluid condition. This can be updated to the appendix 7, and drawing in the technical room should then also be updated to the newest one. Sight glass can also be used to verify state points, for example installation to cold outlet pipe from Vitocal would allow us to confirm, that even at the coldest point of the main circle slush ice is not formed and heat transfer fluid is kept in a fully liquid state.

State point 9 is the mixing pipe for temperature control system. For laboratory testing requirements, cooling must be supplied to EN-LAB1's tested cabins in constant, controllable temperature. Without temperature control system, heat transfer fluid to testing would be cold, but not in constant temperature. For example variation between -5 – -8°C would show up. Therefore some returning, warmer fluid is let back through state point 9 pipe into cooling supply in small, just right amount to give us steady cooling supply temperature. Supply temperature is measured by temperature sensor included in the system, and controlled by adequate opening status of the mixing valve. In appendix 7, amount of maximum return is specified by maximum mixing ratio. Value is based on an educated guess, performed in conservative manner. Also some values for energy calculations about CO2 refrigeration system are shown. State points 18 and 19 are before and after the condenser/gas cooler unit, correspondingly. 25kW is a typical operating value for our CO2 system, and based on that and state points, approximate mass flow for CO2 in the system with aforementioned load was calculated. Implemented CO2 cooling process was planned and calculated specifically based on air properties introduced in next chapter. Values from the actual CO2 system were examined for early stage of design and to get familiar with the process and its dynamics.

6.9 Humid air calculations for CO2 system cooling

Some moist air state points were calculated while designing the air cooling plate heat exchanger under the CO2 systems condenser/gas cooler. Values for all air state points and specific values for air state points 20 – 22 are shown in appendix 7. Air state points were inspected from Mollier diagram of humid air, which proved to be very handy for examination of air cooling processes (Kotiaho et al. 2004, pp.40-42). In Excel calculations we however needed numerical equations, to be able to check numbers with various conditions quick and accurately. Used equations are introduced in this chapter. Flow area is based on the condensers/gas coolers dimensions. Flow velocity was estimated in a conservative way from manufacturer's charts, discussions with colleagues and quick measurements with air velocity metering device. Based on these, maximum air volumetric flow was calculated.

Temperature and relative humidity of air were taken as design parameters. Absolute humidity, specific enthalpy, vapor pressures and dew point can all be calculated with these values. We used selected equations from Lampinen (2015, pp.1-8). Vapor pressure of saturated moist air in air's temperature, p'_v (T) (Pa) was calculated from equation 7. p_0 (Pa) is the ambient air pressure, 100000Pa was used in our calculations. Vapor pressure of moist air, p_v (Pa) was calculated from equation 8. RH is relative humidity of air, and it is between 0 for absolutely dry and 1 for completely saturated moist air, correspondingly. Vapor pressures were calculated, because they were needed for accurate calculations of air's moisture content, specific enthalpy and dew point temperature. Otherwise we were not specifically interested in these values.

$$p'_v(T) = p_0 * e^{11,78*\left(\frac{T-372,79}{T-43,15}\right)} \tag{7}$$

$$p_v = RH * p'_v(T) \tag{8}$$

Absolute moisture content of air, x (kg_v/kg_{da}) was calculated from equation 9. It is important for us to know, how much water vapor air includes, since possible condensation and evaporation in our systems custom made heat exchanger can impact the effectiveness of CO2's air cooling significantly. M_v (kg/mol) is molar mass of water and M_{da} (kg/mol) is molar mass of dry air. p_{da} (Pa) is partial pressure of dry air in moist air. p (Pa) is the pressure applying to moist air we are interested of, ambient pressure p_0 is naturally used in these calculations.

$$x = \frac{M_v * p_v}{M_{da} * p_{da}} = 0{,}622 * \frac{p_v}{p_{da}} = \frac{0{,}622 * p_v}{p - p_v} \qquad (9)$$

Specific enthalpy of moist air, h (J/kg) was calculated from equation 10. c_{pda} (J/kgK) and c_{pv} (J/kgK) are specific heat capacities of dry air and water vapor, correspondingly. L (J/kg) is the latent heat of water's evaporation.

$$h = c_{pda} * T + x * \left(L + c_{pv} * T\right) \qquad (10)$$

In addition, we wanted to check the dew point temperature of air state points. For this, we derived equation 11 starting from equation 7 as follows: In dew point temperature T (K), our vapor pressure is and equals saturated vapor pressure. Therefore we can substitute p'_v with p_v. Then we arrange terms and take an e-based logarithm. From this new equation we can solve T, which is presented in equation 11. We used dew points to evaluate the cooling power spent to condensing of air's moisture at various weather conditions.

$$T = \frac{43{,}15 * \ln\frac{p}{p_0} - 4391{,}47}{\ln\frac{p}{p_0} - 11{,}78} \qquad (11)$$

Air state points 20 – 22 are presented in appendix 7. State points are shown for a scenario, where air is cooled from 30°C to 27°C, which presents a very hot summer day in Finland. Maximum absolute humidity of air what our system should be designed to encounter was discussed to be around 0,012 kg_v/kg_{da}, which corresponds to 25°C air with 60% relative humidity. Cooling leads inherently to condensing of water vapor into heat exchanger surfaces, because they are colder than air's dew point. Therefore some of the cooling power goes for drying of air, and air's absolute humidity is lowered in the cooling process. Relative humidity will increase even though, because cooler air can contain less humidity. As described in this chapter, enthalpies, absolute moisture contents, dew points and vapor pressures of air state points are determined by temperature and relative humidity. Air maximum mass flow is calculated based on air's maximum volumetric flow and expected density. Mass flow has to be the same as which is currently going through the CO2 condenser/gas cooler unit in a hot summer day, because in our arrangement naturally the same air mass flow has to travel through our custom made air cooler heat exchanger too.

Air's enthalpy change and cooling power demand for 30°C to 27°C scenario is presented in appendix 7. We can see, that air cooling by three °C would demand more cooling power than our system can supply with its Vitocal 300-G heat pump. We ran calculations with many different circumstances in Excel, and got a picture, that air can be reliably cooled by about two °C with full cooling power of our project's refrigeration system. When additional air cooling is needed for big CO2 system to function desirably (see chapter 6.3), CO2 condenser/gas cooler's fan is already blowing with full speed. This means, that cooling has to be enhanced by air cooling, because air flow cannot be increased anymore.

When Air cooling is started with our air cooling plate heat exchanger, it can cool air by abovementioned two °C. If only 1 °C is needed for big CO2 systems desirable operation, we only need cooling supply of around 5,17kW, as can be seen from appendix 7. If ambient air temperature rises further, our air cooling plate heat exchanger can increase power up to around 13-14kW, which is enough when air a bit over 2°C hotter that desirable for big CO2 system. If this is exceeded, cooling would demand more than its capacity from our system, and big CO2 system begins to experience problems even if air cooling was applied with full power. When assessing our cooling solutions effectiveness to initial problem with our big CO2 system, we can say that it will help the situation. Secondly, cooling capacity is enough if ambient air is two °C or less too hot. After that, our solution will lose its effectiveness and with for example over three °C too hot air, it will not help at all. With air over around 31°C, all cooling load falls on our projects secondary cooling system, since air cannot cool subcritical CO2 anymore without additional air cooling. Big CO2 systems second compressor will go on and problems are met. Taking sufficient pinch points into account, in reality this happens a couple of °C earlier than 31°C.

6.10 Pipe sizing, pressure drops and flow calculations

Pipe sizing and flow calculations followed, after state points were calculated. We changed pipe diameters in all state point pipings and checked how it affected pipe flows. We tried to set pipe diameters large enough to give small pressure drops, but not larger than necessary due to economical aspects and few other factors, such as liquid volume stored in piping. Unite pipe dimensions according to possibilities were seen as a good thing due to economical aspects, easy construction and simplicity of system. There was one pipeline in which a little additional pressure drop was even targeted, the temperature control system's returning flow pipe, in state point 9 area. This was because flow needed for temperature control is known to be small. A bit higher pressure drop gives us possibility for more open position of temperature control valve's returning pipe side, which was thought to give us better temperature controllability.

Pipe length has a straightforward effect on pipe's pressure losses, and pipe diameter was considered carefully with long pipelines. While designing the system, attention was paid to make as short pipelines as practically possible. Pipe lengths and importance of as short piping as possible were discussed with installing crew before and during the installation. All in all accurate calculations were noted to be hard and practically impossible, because even this sized system was complex for accurate flow calculations and various operation modes and dynamic nature of the operation were present. With many rounds of enhancing calculations and simply gaining experience on the pressure drop calculations with our system, a good picture of factors effecting the pressure drops and reliable results were achieved. It turned out, that even though accurate prediction of some circles total pressure drop is hard, maximum pressure drop with fair margins can be estimated with given maximum flow. Very important in planning was that pipe diameters were chosen large enough. More mass flow cannot simply be transferred if the pipe diameter is too small.

Our flow calculations began with calculation of Reynolds numbers with equation 6. With finally selected and implemented pipe diameters and other properties, we got laminar flow to all pipelines. Our next task was to read flow friction coefficient λ from Kotiaho et al. (2004, p.56) Selected flow- and tube properties are collected in table 13. To that we needed Reynolds numbers, and pipe roughness value for used material. Pulled copper tube was seen to be close enough representation for used copper pipe. Value k/D and Reynolds number were used to read λ from the chart. For turbulent flow, λ was read straight from the charts visually. Since there was even more accurate formula for λ given for laminar flow, it was used. λ for laminar flows are presented in appendix 7.

Table 13. Selected flow and tube properties.

pipe material	pulled copper tube
roughness k (mm) (max)	0,0015
based on D (mm)	25,6
k/D	0,0000586
vastuskerroin λ , **Re over 2300**, turbulent flow	0,05
vastuskerroin λ , **Re under 2300**, laminar flow	Calculated in State points-charts.

Pressure losses of pipelines were calculated using equation 4. Flow and pressure drop-related properties of piping in area of state points 1-13, 16 and 17 in full cooling load are presented in appendix 7. Calculations were double checked with Pipe Flow Calculations online software in the beginning for added reliability and educational purposes (Pipe Flow Calculations, n.d.). Our Excel calculations were proven to calculate accurately with given values. Differences in values calculated with my Excel and the program were very small, sometimes non-existent. This may be the case, if actually the same equations are used, and then only differences in the result come from rounding accuracy. Now values still needed to be adjusted according to our project's real operational refrigeration system. Pressure drops were calculated separately for laminar and turbulent flows, since λ was calculated with formula for laminar, and visually read from charts for turbulent flows. Only either one can be naturally used at a time, in our project's refrigeration system all flows stayed laminar even in maximum cooling load calculations. From Appendix 7 can be seen, that biggest pressure losses clearly come in state point areas with longest pipe lengths, 12, 13, 16 and 17.

Some local pressure losses were calculated for our system with equation 5. Selected local pressure losses for different curves and pipe inlets and outlets are collected to table 14. Maximum flow velocity was assumed as 1,5m/s, as a conservative value it is above the highest flow velocity estimated to be present in our system with full cooling load. As we can see, local pressure losses in table 14 are very low compared to pressure losses in piping, even with conservative assumptions. Gently sloping curves were recommended for piping, since strict curve can increase pressure losses significantly, and very steep curves can already give significant pressure losses if there are many of them.

Table 14. Selected local pressure losses for different curves and pipe inlets and outlets.

v_loc (m/s)	1,5		
Piping elements	**ξ**	**Local Δp (pa)**	**Local Δp (bar)**
Round 90° curve, R=0,5*D	0,9	1043	0,010
Round 90° curve, R=D	0,33	382	0,004
Round 90° curve, R=5*D	0,1	116	0,001
Circular pipe inlet, submerged in fluid	0,9	1043	0,010
Circular pipe inlet, wall-supported	0,5	579	0,006
Circular pipe outlet, both types	1	1159	0,012

Many pressure drops had to be evaluated without calculations. Almost all pressure drops in our system's most important components were read from manufacturer's technical manuals. We considered the flow rate corresponding to our maximum cooling load. Custom made Air cooling plate to CO2 container's roof was an exception to this, its maximum pressure drop was obtained from the used design program, Unilab Coils. This design phase is introduced briefly in chapter 6.4. Maximum pressure losses of most important components and elements of our project's refrigeration system are collected to table 15. Evaluation was quite difficult and time consuming for some objectives, especially when good functionality with small cooling loads and flows had also to be taken into account.

Table 15. Maximum pressure losses of most important components and elements in our project's refrigeration system.

max Δp of components and pipe elements	
Component	**Δp (bar)**
1-way valve 1 kpl	0,15
3-way valve for T-control	0,1
CO2 cooling plate	0,4
CO2 plate HE	0,3
Güntner	0,5
LAB1 Air Cooling HE	0,2
Manual closing valve 1 kpl	0,05
Vitocal	0,1
Strainer	0,05
Element	
Round 90° curve, R=D	0,0038
Circular pipe inlet, submerged in fluid	0,011
Circular pipe outlet, both types	0,0116

Peak mass flows and piping pressure losses for each of our project's refrigeration systems cooling circles are presented in table 16. Each circle has one heat transfer fluid circulation pump. Peak values are calculated with the highest possible designed cooling power. As a result of calculations, four MAGNA3 25-120 and one MAGNA3 25-80 circulation pumps were purchased, with corresponding lift heights of 12m and 8m. Mass flows in so called heat exchanger circle were calculated as 0,25kg/s in appendix 7, but were doubled here because of some room for capacity increases in the future. Cabin test circle's pressure loss does not include the cabin's pressure loss. Pumping in this circle is normally done with cabin's own circulation pump, but an additional pump was installed to the system, that can be taken into action if needed, but is normally passed by with valve arrangements (See flow diagram in appendix 4). Circle's total pressure losses were calculated by simply adding all circle's pressure losses from piping, single elements and components. Adding was made carefully, since forgotten origin of pressure losses could lead to underestimating the circle pressure losses and pumping powers needed for desired flow rates.

Table 16. Peak mass flows and piping pressure losses for each of our project's refrigeration systems cooling circles.

Pump	Peak ṁ (kg/s)	Peak max piping Δp (bar)
CO2 circle	0,571	1,22
Heat dump	0,714	1,48
Vitocal - Buffer tank	0,75	0,34
HE circle	0,5	1,09
Cabin test circle *	0,5	0,63

* cabin's Δp not included

As can be seen, for heat dump circle, lift height will probably be not enough for hardest circumstances. This is because these are calculated in cold ambient- and fluid temperatures. Warmer heat transfer fluid is circulated in heat dump circle during real operation. What practically results, is that if the system is in standstill for several hours during cold weather, full mass flows and cooling power can be reached only after a while, when fluid has warmed some and viscosity has again reached a normal level in designed operation. Highest peak pressure losses are expected in the heat dump circle. Main reasons for highest pressure losses in this circle are high peak flow rates, relatively long piping and high pressure losses in the Güntner dry cooler. Lowest pressure losses are in the circle connecting Vitocal 300-G to the buffer tank. This is because of very low flow resistance, even though flow rates are expected to be highest of all circles. Low flow resistance is a result of short piping, just a few components causing pressure losses and absence of single components that cause significant losses (which are found in other circles).

6.11 Liquid volume and expansion vessel calculations

Liquid volumes of each three heat transfer fluid circles were calculated, so that enough propylene glycol and expansion vessels can be ordered and fluid fill can be done successfully. First step was to list liquid volumes for all components in the circles, that hold significant amount of fluid. Component's liquid volumes are collected in table 17. Almost all component fluid volumes were found on products technical manuals. Volume for custom made heat exchanger for CO2 air cooling was obtained from the Unilab Coils design program in the technical printout. Cabin's liquid volume is different depending on the tested cabin and a conservative value was used according to discussions with our team. Small amount of liquid has to be refilled after each change of tested cabin. Small refill has to be done even if new cabin was with exact liquid volume as the old cabin, because even minimal differences in liquid volumes make clear differences in standstill and operation pressures of the system, and suitable pressure levels in fluid circles are adjusted with the accurate liquid fill.

Table 17. Liquid volumes of three heat transfer fluid circle's components in our installation.

Single liquid volumes	
Component	**V (litre)**
Vitocal	6
Güntner	8
CO2 HE	1
CO2 cooling plate	8
Wikora buffer tank	300
LAB1 Air cooling HE	1
Cabin, being tested	15

In addition with components, pipes hold some liquid volume. Rough pipe liquid volumes can be seen from appendix 7. Liquid volumes for all three circles without pressure vessel were calculated by adding together all piping and component volumes of the circles. Piping was followed from the LibreCAD drawing, and therefore made sure that all volume-bearing components and pipelines were noticed.

Next we evaluated largest possible expansion percentages of the heat transfer fluid in each circle during normal operation. Flamco's (2014) material was utilized, and corresponding maximum expansion percentages were evaluated from charts based on glycol content and highest possible temperature. Design conditions considered a hot summer day and direct sunlight to some parts of the circle. Fluid heating to 50°C was considered, except 60°C for heat dump circle, because it will naturally work with higher heat transfer fluid temperatures than two other circles, and has to be in higher than ambient temperature during successful operation. Expansion volume V_{exp} (m^3), heat transfer fluid volume after an expansion event V (m^3), and nominal volume of a suitable expansion vessel in our project V_{ev} (m^3) were calculated with equations 12, 13 and 14. V_{ev} includes both liquid- and gas side volumes of the expansion vessel.

$$V_{exp} = x_{exp} * V_0 \quad (12) \qquad V = V_0 + V_{exp} \quad (13) \qquad V_{ev} = 2 * V_{exp} \quad (14)$$

V_0 (m^3) is volume of the heat transfer fluid before expansion. x_{exp} is fraction of expansion volume compared to fluid volume before expansion. If fluid expands for example 2% because of warm weather, x_{exp} is 0,02. V_{ev} was decided to be twice the largest possible expansion volume, because then expansion vessel would still have half of its volume filled with pressurized gas, even in maximum expansion. Highest allowed ratio of liquid to total volume is 0,63, and therefore at least 37% of total volume must be filled by pressurized gas side all the time (for expansion vessels used in our project). Higher fill can stretch the membrane and damage the vessel.

It should be remembered, that heat transfer fluid volume before expansion V_0 is always defined in some particular temperature and pressure. The total mass of the fluid in the circle is practically a constant, but temperature and pressure depend on conditions and operation. Therefore the heat transfer fluid temperature during the fill has an effect on how the fluid volume changes and interacts with expansion vessel. Say for example, we fill liquid exactly to the volume of the circle without expansion vessel, so that that liquid pressure at expansion vessel inlet equals pre-set gas side pressure of the expansion vessel, but the vessel membrane has barely not yet been moved by expanding liquid. Now if fill was made in a warm temperature (which usually is the goal, to get as much air out of the system as possible) and refrigeration systems starts to operate in a lower temperature, liquid volume and therefore system pressure decrease. More liquid has to be filled to be able to get interaction with pressure vessel and keep stable pressures above the atmospheric pressure in the system. A slight overpressure is usually targeted in the system at all times to prevent air entering the system. Circle volumes with and without pressure vessel, expansion in percentages and volumes and suitable pressure vessel volumes are presented in appendix 8. We can see from table 17 and appendix 8, that the cold buffer tank holds the most of the total volume of all circles. Also the cabin's test circle, in which the buffer tank is in, is by far the largest circle by liquid volume. (Flamco, 2014) (Melinder et al, 2015, pp.37-38)

There are many ways of sizing the expansion vessel, and the one described by Flamco (2014) was used in this project. The actual picking of size was made very roughly by doubling the expected maximum expansion volume, as mentioned earlier. This choice was then further examined by making calculations about expansion, presented in appendix 8. As recommended, pre-set pressure in vessels gas side was set according to each circles highest point from vessel level, and resulting hydrostatic pressure. Lowest operation pressure was set into same or a little bit higher than pre-set pressure to have a very slight liquid fill in the vessel when filling the system. Maximum operating pressure was set a few tenths of a bar below safety valve opening pressures. Vessel fill levels without expansion and with maximum expansion were calculated. Efficiency coefficients show the amount of liquid per total volume in maximum expansion. According to calculations shown in appendix 8, selected pressure vessel sizes were noted to be good for our system and its pressure levels, and the mentioned maximum efficiency coefficient, 0,63 was not exceeded. One reason for calculating many scenarios in Excel was to learn about the dynamics of expansion with various fill levels, temperatures and system pressures.

7. Implementation of an indirect refrigeration system for laboratory testing

7.1 Purchasing and acquiring components

As our design advanced, components were picked for building the system. Component list of our project’s refrigeration system is presented in table 18. List is not exact and some piping parts are not mentioned, list is made with suitable accuracy for project’s demands. For example count of manual closing- and air vent valves is realized finally after the installation is completed, and additional valves were added according to noticed needs. Vitocal 300-G heat pump, buffer tank, dry cooler, NIRA hand pump, copper pipes and some basic piping goods were already arranged in our property in the beginning of project. Most of the components were purchased during the project by me. Table 18 shows the seller for main purchases. All in all purchasing and acquiring components for the system took a significant amount of time, but I consider the results successful. Components shown in a flow diagram are collected in appendix 6. Each component is marked in the drawing once.

Table 18. Component list of our project's refrigeration system. Numbers are compatible with LibreCAD drawing showing components in appendix 6. Each component number is marked in the drawing just once.

Number	Component	Amount	Information	Purchased from *
1	Heat pump	1	Vitocal 300-G	Viessmann
2	Buffer tank	1	Wikora WKS 305	Viessmann
3	Plate HE	2	SWEP BX8T	Refair
4	Circulation pump	5	MAGNA3. 4x model 25-120 and 1x model 25-80.	Grundfos
5	Expansion tank	3	Flexcon Top 2, 4 and 18L.	Onninen
6	Dry cooler	1	Güntner GFHC WD	Viessmann
7	HE to CO2-system	1	Design and manufacture in Viessmann, Porvoo.	
8	Temperature control system	1	Includes T-sensor and controllable 3-way valve, self-operating system.	Wexon
9	no-return valve	2		Konwell
10	manual shut-off valve	25		
11	safety valve	4	3 bar-g opening pressure.	
12	air vent valve	10	To highest points and possible air pocket spots	
13	pressure indicator	3		
14	filter, strainer	3	1 for each circuit.	
15	NIRA hand pump	1	For liquid fill	
16	Tanks for safety valve discharge	4	Small, 2x 5L and 1x30L are more than enough	
17	Container for propylene glycol	2	For filling and collecting during deaeration. Must be closed, but allow contact to air.	Cipax
	Propylene glycol content meter	1	Optical metering device	Würth
	Propylene glycol 30% - water 70%	liquid mixture		
	Propylene glycol 50% - water	liquid mixture		
	Propylene glycol 100%	135,4	In Litres.	Ahlsell
	25,6mm copper pipe		Outer diameter 28mm. For practically all piping	
	17,25mm copper pipe		Outer diameter 19,05mm. Short bit for temperature control piping	
	Pipe insulation			

* Viessmann: Purchased from Viessmann Division 1.

I designed plate heat exchangers with SWEP SSP-program myself and purchased them from Refair. Circulation pump properties were calculated in Excel, and suitable pumps were purchased from Grundfos. Grundfos has a good literature for pump selection and dimensioning, which was a great benefit in the process. Pumps are important part of the system, and also the system control and operation is largely carried out by adjustment of pumps. Pumps show lift height and flow rate in real time, which is very good feature for operation and inspection of the system. Expansion tanks were dimensioned in Excel and purchased via

Onninen, manufacturer Flamco's good online material was utilized to help the dimensioning. Temperature control system was ordered from Wexon, and suitable product was found in co-operation with the representative salesman. No-return valves were selected based on manufacturer's charts and bought from Konwell. From my previous working experience in AQVA Finland Oy I knew Cipax Oy, who supplied advanced tanks for propylene glycol-water mixing and filling. Pure propylene glycol was purchased from Ahlsell for mixing the wanted heat transfer fluid to water. Optical meter for water solutions propylene glycol content was bought from Würth. It was estimated, that ability to accurately measure the content was going to be crucial for successful commissioning and operation of the system.

Small components were bought on local HVAC stores during installation, such as most of manual shut off valves, strainers, pipe tees and safety valve parts. Some small components were also taken from the laboratory facilities, if some free stuff was discovered. Stash from the recent fair installation was also available, and some stuff was utilized. Challenge was to verify the condition of used stuff, since no original installer or otherwise familiar person was caught up. Likely good and useful propylene glycol-water mixture was not used, since we did not figure out how to verify quality and exact composition in economical and reasonable way. Similar situation was encountered with few other components, such as fluid circulation pumps from the fair installation. It was learned, that inspection on relatively cheap components can be easily more expensive and time consuming than purchasing the new ones. This can be seen as a shame for technical component's high reusing rates in environments perspective, but I believe it is a common phenomenon in technical business around the World.

Component purchases could have perhaps been made quicker with other kind of arrangement. Now suitable component properties were assessed, and potential suppliers contacted via phone call, online sales enquiry or email. This was a precise way of gathering components, and suppliers were able to tell about their products and suitability for our installation quite well. Another way worth trying would have been the following: When design of system was made and an approximate component list is ready, product enquiry document with the list and brief description of our project would have been sent to potential suppliers. Suppliers would then offer all components they have to offer for our project. This could have probably spared some time. Purchasing strategy was also perhaps encouraged by the fact that this was a first system design I have done of this type. It was natural to purchase a component right after getting to know its requirements before moving to next main component. Also with used purchasing strategy, more attempts to buy components from the same sellers could have been made. Diversity of components was a challenge in this perspective, and it was experienced a bit hard to find suppliers able to contribute with multiple of our main components. From table 18 can be seen, that many suppliers were used and none supplied lot of different main components in this project.

Viessmann's own purchasers were not contacted much during our project. This was probably because our project's refrigeration system was a single system, including more or less different components that which were used in our factory made refrigeration fixture products. Type and characteristics of components would have been given to purchaser, who would have then used his or her knowhow and supply chains to purchase best available component for given purpose. Purchaser also knows which components have to be searched outside current product portfolio, which saves settling time. Theoretically using company's own purchasers can be efficient if product demand and specializations match. Range of available products is typically well known and prices are low, especially when components are purchased to industrial purposes usually in large quantities, and perhaps from well-known suppliers. For example in our project, one HVAC store was noted to be pretty expensive afterwards, even though service and product availability were good. Purchase from another supplier could have spared some costs. Paperwork and purchasing chain can potentially be also handled effortlessly in co-operation with technical purchasers. In a big company there can also be some suitable leftover components in storage, and purchaser or some of his/her contacts could be aware of this. Discussions with a purchaser could have been educational experiences to both parties, even if the outcome would have been the same, to purchase components independently.

7.2 Installation

Installation began properly in the beginning of August. Some preparatory installations were made already in week 26. Installation was thought to start already in late May according to original visions. Refrigeration system's designs essential parts were done early in June, but summer holidays and some practical things delayed beginning of the installations. Installation pace was slow and steady, as planned, even though system was ready and in operation a lot later than what was the original vision. All in all I would estimate, that similar 14kW refrigeration system could be installed in about a week, if another system with reasonably same features was ordered, well planned and installers use their own basic piping parts. Two to three is for example a good size of the installing crew for this kind of task.

Installation had two significant phases. Both were done according to the flow diagram, acting as installation- and electrical drawing in appendix 4. I, system's head designer, was in involved in every phase of the installation helping to interpret the drawing and planning installation steps so that desired system can be reached. I also did a lot of supporting work during installation, such as fetching piping components from local HVAC stores. First all heat transfer fluid circles with full, closed piping and equipment were installed. This work took by far most of the installation time. After this, circles were ready for liquid fill and the commissioning of the system.

Second phase was electricity installation. This phase was completed in about two weeks, and it was ready even before heat transfer fluid cycles and their piping. Circulation pumps, Vitocal 300-G heat pump and other electricity-using equipment were supplied with power. Some minor automation installations were considered, and also carried out, even though system was going to run mainly with self-operating equipment. In our system, there was no connection to some centralized system, which allows control and operation of all system devices and data from one set of computer screens. This selection was ordered while project task handout, to keep the system simple enough. Operation of system is mainly made with circulation pumps and Vitocal 300-G, by setting values from each equipment's own operation panel.

Installation- and electrical drawing in appendix 4 was proven to be good and accurate for installing purposes. Drawing is a certain flow diagram, which shows exactly the way of heat transfer fluid and order in which each component is met. All significant components are market in the drawing 6. Main principles of the drawing 4 and some comments in it must be followed precisely during installing.

There was however some aspects, that give room, and even some need for skilled applying during installation. Such things were for example the closer placing of components in the room, and details of piping routes. As discussed, more compact layout than expressed in flow diagram was a good thing and local decisions can be made, as long as installation is made according to the flow diagram and compulsory requirements are met. Some room for applying was a known decision, because for example short piping and wise pipe routes were known to demand some decisions in the location while installing. (3D-drawings would have been otherwise needed, which would have been too time consuming and not effective in this project) All in all this strategy worked well and was shown practical for the work flow. The system became a bit more compact than indicated in the installation- and electrical drawing, without giving away accurate match in vital requirements or compatibility with flow diagram. Most of circulation pumps and Vitocal 300-G's were installed close to each other, and originally screens to same direction. Orientation of some pump screens had to be changed, which was a quick task, but result in a slightly impractical location of control panels. Placement of liquid fill station was successful. One of the installers was a coming user of the system, and he did some handy tricks during the installation, making the system more practical to use. Total length of all piping in the system became a bit longer than originally planned, partly because of realized pipe routes and partly because of the fourth cooling target ordered at a later times during the project. Therefore also systems' liquid volume and propylene glycol demand were raised a bit.

Our project's refrigeration system was my first time as project's head designer. System was first in its kind for installers too, even though they had experience with works among other type of refrigeration systems, including some technically more challenging than ours. First time affected our schedules a lot, especially because of basic piping components had to be decided and fetched while installing, those were not available in a high volume storage. If similar system was to be designed and installed somewhere else again, next time would take probably only a fraction of time used in this project. Our project's installation was perhaps also not as urgent as some works ordered by a paying customer. Some delays were simply accepted rather than investigating affordable ways of arranging adept installing force, to complete the system sooner and getting it to work.

7.3 Commissioning

Commissioning of the system started in around week 40. Main phases in the process were leak and pressure testing, liquid fill and air purging, component starts and testing, establishing labels to important parts of the system, cold insulation and documentation about the system. Some late installations and fixes to system were carried out at the same time as early commissioning. For example some leaks, non-air tight insulation and components installed in a wrong orientation were discovered and fixed. Preparing documents for the system's users and to some other purposes was done a lot during commissioning. A big magnetic table was placed in the technical room's wall, where all key documents and some tools about the system can be found. Flow diagram, pressure level document, propylene glycol chemical information, list of pending repairs and improvements, instructions to operate Vitocal 300-G heat pump and a set of air vent keys were for example provided.

Leak proof and pressure testing were an important part of our system's commissioning. After circles were installed they were filled with nitrogen gas to detect all possible leaks. Leaks were fixed and testing continued alongside with other works, such as installation of missing air vent and emptying valves. After some time testing was continued with air, since it had better availability, and it was seen to have leak potential possibly closer to water, than what nitrogen had. After system charging with final propylene glycol-water mixtures, leak and pressure testing continued. Leak detection became easier with liquid, and all fixes were completed with withdrawal of only couple of ten liters at maximum. This was achieved, because we had plenty of closing valves to isolate the pipeline in which the repair was done and careful planning of repairing beforehand. After system was leak proof, pressure testing continued alongside with air purging and equipment starts. Official pressure test was carried out and documented in the latest stage of commissioning. Some instructions for pressure test were checked from Melinder et al. (2015) and Suomen Kylmäyhdistys Ry (2019). Pressure test was successful and proved that our refrigeration plant is leak proof and lasts highest pressures that our system might be exposed to. Time under at least 2,9 bar-g test pressure was 3,5 hours.

In system charging stage circles were filled with appropriate propylene glycol-water mixtures. Flushing and pressure testing with water was considered, but fill straight with final mixture was chosen. There are pros and cons in this choice. One of the main pros is that since only final liquid is filled, we do not have to worry about water, that is going to be left in the system even after careful emptying. It was assessed, that for example the main circle would hold a couple ten liters of water after emptying, which would have to be compensated by adding a balancing volume of a stronger mixture to the circle. Another pro was the higher practicality of these two options. Pro of water flush choice would have been a better chance to flush the system before charging, but cleanliness of piping was assessed to be good enough for a straight-away charge.

Air has to be removed as well as possible from the closed secondary fluid system. Air purging took place after and during each liquid fill, and sometimes air vent valves were opened after some running of the system to see if some air has risen to the high location of the valve. Releasing air lowers pressure of the circle, and after air purging, a small amount of liquid was added to compensate the air that left the system. Air purging was done with air vent valves in all highest locations of the piping. Both manual and automatic air vent valves were used, depending on the spot and situation. Manual valves were chosen outside to prevent the problem of freezing automatic valves. Small closing valves were installed between most air vent valves and the system to ease the use, and because a tendency to leak was mentioned by Melinder et al. (2015) and Suomen Kylmäyhdistys Ry (2019). Air purging with low pressure suction was considered, but we assessed that a good result can be achieved with air vent valves only. There are a lot of possible couplings in which low pressure air purging can be done in the future is needed for some reason.

All in all it seems, that air purging can be done quite efficiently from current air vent valves, and at least purging is can be done practically whenever needed. Result of air purging was assessed based on whether any air comes out while opening air vent valves, flow rates in the system (trapped air typically increases pressure losses), sounds and amount of exited air. Pressure gauges were also watched during liquid fill, that was actually a very good way of estimating whether there is still a significant amount of air in the system. Air compresses, and causes oscillations to the pressure gauge if present in the system. Also the speed of pressure rise with one pumping movement with the liquid pump indicates the situation well, air in the system compresses and causes a slow pressure rise. Pure liquid is almost incompressible, and system pressure will rise quickly and more suddenly with even a small amount of added liquid.

All piping and equipment had to be insulated to prevent condensation and/or freezing of air's moisture to the cold surfaces of the system. Insulation has to be air tight, so that air and its moisture can't get to the system's cold surfaces. Only piping that will be not insulated, is the heat rejection pipe going outside from the heat pump, since it carries warm fluid and is therefore not at risk. Return pipe was insulated, because in the cold weather fluid can cool outside and cause condensation as pump is turned on and cold fluid is moved to the technical room. Some of the insulation was already done before, and some were finished after the official pressure test.

All components were started and tested during the commissioning. Most of them started to work well right away and only some basic adjustment of operating values were made. Small works had to be done in the process, for example connection of missing electric wire in the electric cabinet. Temperature control valve started working well right away, even though I was ready to spend some time getting it to right settings, in co-operation with the manufacturer's engineer. Commissioning can be seen to end in the testing of a liquid cooler, the first real task of our project's indirect refrigeration system.

Even though the system is successfully completed and operative, there are still some works that have to be done before full utilization of the system. In the second week of November, some fluid circles were not yet fully cold insulated, and piping from safety valve outlets to downwards position were still missing. For better system controllability, Güntner dry cooler should still be equipped with responsive operation setting or a part-load possibility. Some small fixes and improvements can be recommended to the system in future, such as adding a sight glass, adding second fill station and getting better documentation and experience about operation of Vitocal 300-G heat pump.

8. Completed refrigeration system and its performance

Completed system is a custom made refrigeration plant for specific needs. In addition to moderate, 14kW cooling capacity it has four unique cooling targets and some pretty strict quality requirements, concerning for example cold supply temperatures and operational reliability. Some pictures about our project's refrigeration system are presented in figure 25. In left upper corner we got our system's refrigeration machine, Vitocal 300-G heat pump. It can supply maximum cooling power of around 14kW, which consequently is the maximum output of our system. In the same picture we have our buffer tank, holding 300L of propylene glycol heat transfer fluid as a cold reserve. Up middle picture shows dry cooler for heat rejection, and heat exchanger in CO2 system container's roof, placed under CO2 condenser/gas cooler unit. Up right picture shows heat transfer fluid fill station and the NIRA hand pump used for an accurate fill. Down right picture shows our two SWEP heat exchangers. Heat exchangers are compact and can be found in middle left in the picture. As can be seen, cold insulation was not finished in CO2-circle's area by the time picture was taken. Two circulation pumps, all three safety valves and all three expansion vessels can be spotted in figure 25's pictures. Refrigeration plant's magnetic information board can be seen on left down picture. The biggest content is an A2-sized flow diagram for system operators, installers and engineers. Information board aims to show all the key information about the system at a glance.

Vitocal 300-G heat pump, buffer tank and view to main piping

Dry cooler and heat exchanger for CO2 system

Heat transfer fluid fill station

Refrigeration plant's information board

Heat exchangers and associated piping

Figure 25. Pictures about our project's refrigeration system.

Our project's refrigeration system was used first time for product testing in week 43. Therefore this can be seen as a moment, when the system was successfully installed, commissioned and taken into use, even though it was not completely ready and tuned yet. The first tested product was a small air cooling sensible heat exchanger. We can use these test results to assess our completed refrigeration system's performance. Test results for a certain two-hour testing period are collected in appendix 9. Appendix 9 shows cooling power, heat transfer fluid's supply and return temperatures and heat transfer fluid mass flows through the sensible heat exchanger.

We can see from appendix 9, that our system is capable of delivering stable mass flows. In this particular test the heat transfer fluid circulation was made with tested product's own circulation pump, as designed. In this test, mass flows were maintained very closely in a bit over 0,1 kg/s. Our system is designed to deliver mass flows of at least 0,5 kg/s for product testing. Results show, that mass flows rise very quickly to desired level after circulation pump starts, and goes quickly to zero as it stops, which indicates an excellent controllability of mass flows while testing products with our system. In this test, mass flows were not close to the designed maximum. Future testing will show us, whether larger mass flows can be maintained with as outstanding precision as in this test.

Results from heat transfer fluid cold supply and return temperatures are less certain. What can be seen, is that cold supply temperatures stay above return temperatures at all times, with a rough difference of around 1°C. This is a good thing, and tells us that cold supply works in general level, and that the tested product can deliver cooling to air. If supply and return temperatures would sometimes go to the same value, it would mean that no cooling is done, and heat transfer fluid is only circulated without any cooling effect. Especially important is that supply is colder than return during the cooling periods, when circulation pump works. This means that heat is taken from the cooled space, and cooling occurs.

Cooling action with pump on is seen in data by semi-steady development or lowering of both temperatures, and occurs four times in our two-hour testing period. Beginning of cooling can be spotted in large, but short time spikes in return temperatures. It is the moment in which circulation pump starts pumping. Some reasons for the spikes can be assumed to be starting of the pump and/or warming of heat transfer fluid during standstill in the heat exchanger just before the temperature meter for returning fluid.

Cooling power is quite steady based on appendix 9. After spikes right after pump starts, cooling power stays in the area of 0,3 – 0,6 kW, and fluctuations are calm. Just like mass flows, cooling powers are applied quickly after pump starts, and cooling stops sharp when pump is shut. This again, is a sign of a good controllability of cooling power with our system, and it is achieved with accurate controlling of heat transfer fluid mass flows.

9. Findings and discussion

One of the main goals for our project's refrigeration system was the ability to supply heat transfer fluid to cooling in steady temperatures. Wanted accuracy of cold supply temperatures was not specified in the design phase, but variation less than 1°C was somewhat targeted. In times of steady cooling, cold supply temperature seems to be around -5°C with wanted 1°C accuracy. Based on that we can say that steady cold supply temperature was accomplished, at least most of the time during each cooling period with pump on. Clearly lowering temperatures in around 16:20 are most likely caused by reactions of the temperature control valve. It has made a closing move to the returning fluid mixing pipe, which would result in lowering temperature of cooling supply. Another possibility would be that colder fluid is withdrawn from the buffer tank due to varying temperature profile in the tank. Either way, we expect to reach a bit higher level of cold supply temperature control accuracy in future, that what was achieved in these first recorded tests.

We have to remember, that cooling is ultimately done according to needs of the tested product, its temperature is measured and kept between some desired values. Our project's refrigeration system's goal is to facilitate this and test the performance of various refrigeration products. Therefore even though we introduced important results from our refrigeration system's performance in the previous chapter, its operation depends also on the tested product. Therefore while inspecting our systems capabilities, also air temperatures before and after the sensible heat exchanger were considered. All in all we can say about the results, that system operates well under quite a small cooling loads from one cooling target. Based on the first test and our expectations, good performance will be maintained also with higher loads and with more simultaneous cooling targets.

Temperature differences over the tested product have an effect on the accuracy of cold supply temperatures. Functioning of our temperature control valve is dependent on the tested cooling application. Bigger temperature difference over the tested product is, more accurate heat transfer fluid supply temperatures we get. This is because our temperature control valve is a mixing valve at its functioning type. For example in the test run presented in appendix 9, we got temperature differences of only around 1°C. In our test run, supply temperature was quite a steady either way, even though from this perspective tested cooling product was challenging for our system's supply temperature control. Supply temperature control over a larger temperature difference in future testing will be easier. Future testing will also be more reliable and efficient in future, because small installations and optimizations will be made to the system, and we get some experience operating the system. Examples of such improvements are finishing of the complete insulation, installation of sight glass in the piping, partial load or automatic control possibility to our heat rejection's blower and finishing all temperature sensor installations to the cold fluid buffer tank.

Maintenance of the system is crucial for its high class performance, safety and durability. Heat transfer fluid should be circulated also in CO_2 circle at least once a month to ensure good condition of piping and liquid, even though main use will be only in summertime. Floor and all possible liquid spills should be washed regularly, and the technical room must be kept clinical and clean. Function of safety valves should be checked at least in every half years or more often. Air venting, changing of strainers and checking gas side pressure levels in expansion vessels should be done regularly too. Checking of the entire system performance on different loads is important for seeing the overall condition of the system, but perhaps this is checked naturally without a separate event if system is in frequent use. One important aspect is to see, that advised pressure levels are met with use in varying temperatures and loads of the system. It is advisable to check heat transfer fluid quality and corrosion levels of piping first time after for example two years of operation.

Width of duty was a clear but welcomed challenge in this project; I delivered most of the design, including for example HVAC- and component model determinations. Interdependence of design phases was also a clear challenge. For example final pipe sizes had to wait until mass flows and type of heat transfer fluids are known. On the other hand, available pipe sizes were a minor design parameter themselves, which made simultaneous, converging design tactique somewhat advantageous. For example rough HVAC plan could have been first assigned to a HVAC designer. This saves time, and is an excellent opportunity to train some tricks, so that HVAC design can be next time made partly or entirely with own efforts. While this was my first this kind of project, doing all design myself was on the other hand very educational. Some of efficient HVAC programs are costly and require some practice to be used efficiently, and searching of free or economical program takes time. This can be made quicker by good advisory by someone who knows the scene.

Master's thesis was noted to be challenging, but an educational task for a writer. There are lot of technical details and a big picture to be hold on, and a ready master's thesis is practically always a first edition, compared to classical engineering-related books for example, some of which are 5th edition or even more. In addition of solving problems, writer has to open all key tricks in the written work. If some design is changed, text has to be edited correspondingly. Many times in engineering work, only the outcome and main documents have to be edited and kept in the best possible condition. Some classical challenges with thesis' scope management were noticed. It was occasionally hard to decide what should be mentioned in the work and what not. Because of late finishing of installation, some results were acquired less than a month before the master's thesis deadline, and these details could not be fully written in the work before that. I could have shifted the focus of written thesis towards the earlier phases of the project to solve this problem, but I am happy about my choice to wait for the completing of the system, so that the whole project all the way to the operational refrigeration plant got included in the work. All in all, master's thesis was noted to be too small for all theory studied and work done during the project, and some compaction and prioritization were carried out. Next time if a document like this is prepared or advised, I will pay attention to aforementioned aspect earlier in working process. Hence, the final thesis is good and compact enough in my opinion.

10. Conclusions

Indirect refrigeration systems have been used for a long time in certain cooling applications, especially with long piping and many cooling targets. Wider use in food retail and supermarket field of business is seen promising because of some technical and legislation related advantages, such as small refrigerant charges, low pressure levels outside technical room and possibilities for improved safety and environmental friendliness. Efficient design and implementation of an indirect refrigeration system and whether indirect system type is the best choice in given case, depends on the desired size, application, location and amount of cooling targets, and there can be many equally excellent possible choices.

Best feasible design and layout in our project's case was discovered to be according to information presented in this work. To be precise, appendixes 4-7 and 10, and tables 7, 8, 10, 11, 16 and 18. Four cooling targets in separate locations demanded a large central buffer tank and refrigeration machinery with high capacity to supply cooling, and compact piping was lead to each cooling target. Temperature control system was added to fulfill the need for product testing's especially accurate cold supply temperatures. Pumping was arranged with distributed circulation pumps instead of one big pump accompanied with regulation valves for each cooling targets. System is controlled straight from the equipment's displays because of request for a simple system, and connection to central computer aided system was not done. Propylene glycol was chosen for water's additive to all heat transfer fluid circles over ethylene glycol, which improved refrigeration plant safety and simplified the system and its operation. Fill was arranged from a single station for all heat transfer fluids, for all glycol contents. Numerous other design choices were made, which can be closer examined from chapters 6 and 7, and appendixes 4-8 and 10.

For implementation, some strategy for purchasing and installation work should be decided in an early stage. Main components can be efficiently acquired by the head designer and the planning team, and this can for its part ensure suitable product and good qualities. Tubing, insulation and various installation parts can be included in the deal with the installing party, with a lot of benefits. Installing party can decide which parts to use, as long as given requirements are met, such as pressure durability, material compatibilities and suitability for applied cold temperatures. Schedules and practical things should be discussed and agreed on with the installing party before the choice to ensure reasonable installation times and quality. Installing arrangement should allow us to do something, if schedules or quality of the work does not meet our expectations. If designing team takes care of installation parts too, acquisition in packages from one business partner can spare costs, time and effort, and make authentication of component qualifications easy. Search of a good supplying partner can be started by sending offer requests to all potential suppliers when plans of the refrigeration plant are ready, even though this procedure was not tested in this study. Other chance is to speak with company's purchasers, whether some good known party could meet requirements also for our project's needs. Clear benefits can be achieved if all installation components come from the same, familiar supplier, and prices can be negotiated in a lower level than with random purchasing from a long variety of stores.

Having operating technicians of the coming system on board already in the installation phase is a clear advantage, and improves our ability to use the system and do small repairs and enhancements ourselves in future when necessary. If problems are met during the installation, it is crucial to decide what to do, fix the problem and only then continue installation. Otherwise problems are accumulated and encountered later in even worse extent. Careful design is an effective way of ensuring good results, and major costs and risks typically start to accumulate after the installation begins in energy production plant projects like ours.

Improvements to the similar system in future can be investigated from separating production and distribution of cooling more clearly, and perhaps implementing one circulation pump for all supply in main circle. Pumping would be done according to total demand; flow to final cooling targets would be adjusted with control valves. Connection to centralized control panel would also improve the system and its use, even though added value through this decision is dependent on a clear and good implementation. Fill station should include separate fill pump for different mixtures, or a proper possibility to empty previous fluid and vent the fill piping. Attention should be paid already in a design phase, to keeping pipe layout compact awarding us shorter overall piping and a simpler system. Wanted performance and details should be defined early in the project, so that a simple and robust design can be reached.

All in all, this project was successful and a working system with wanted properties was accomplished. Decades of high quality indirect refrigeration can be achieved with the project's refrigeration plant if recommended adjustments, operation and maintenance are carried out. Based on this study, for around 10-50kW similar kind of indirect refrigeration plant projects, considerably good results can be achieved with design according to appendixes 4-7 and 10, and details described in this work. For implementation, significantly shorter installation times and a bit higher quality can be reached with means mentioned earlier in this chapter. This project and its learnings can be used as a benchmark for similar custom made energy production plant projects in future, especially in refrigeration branch.

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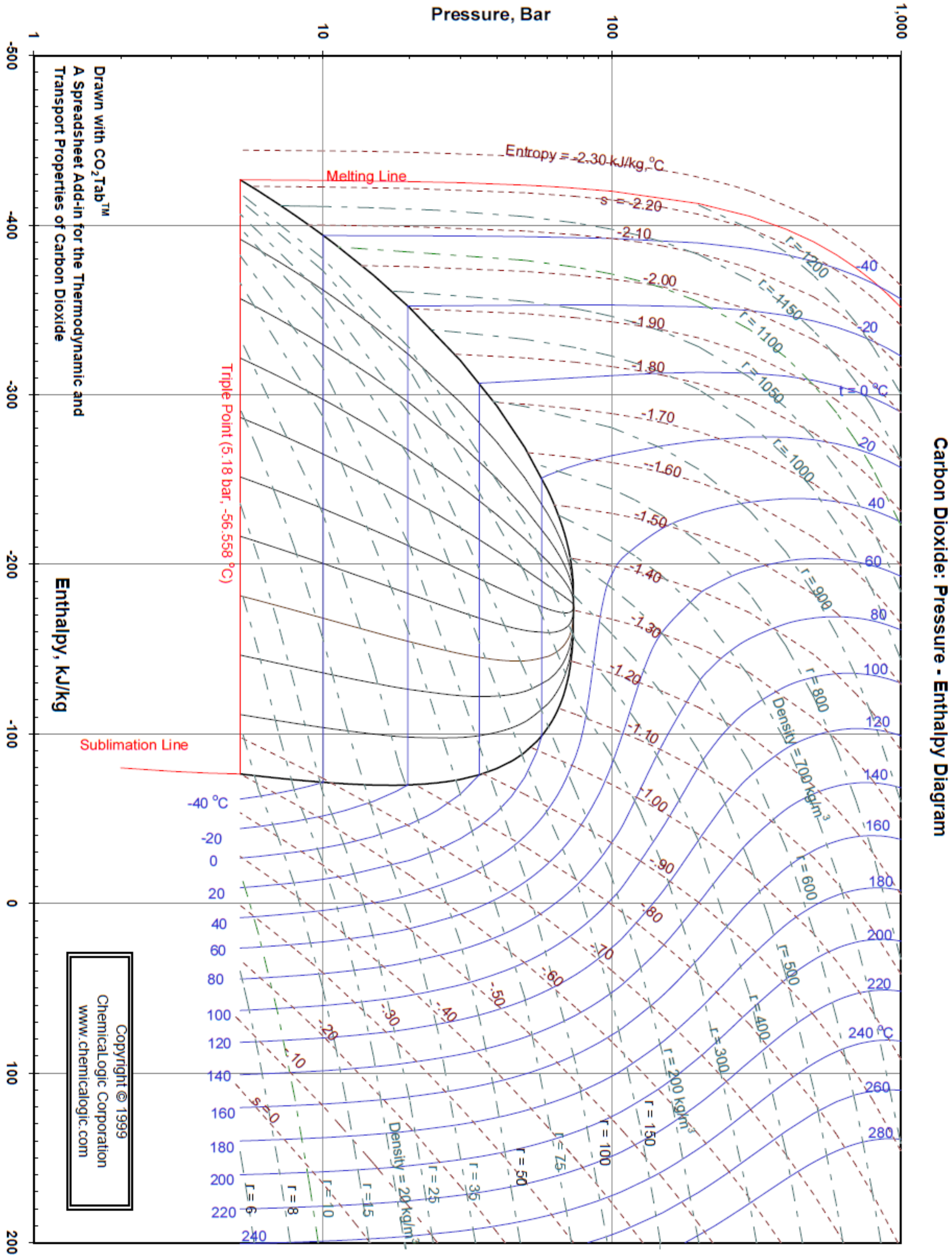
APPENDIXES

APPENDIX 1. Logp-h diagram of CO2, also known as R744. (ChemicaLogic, 1999)



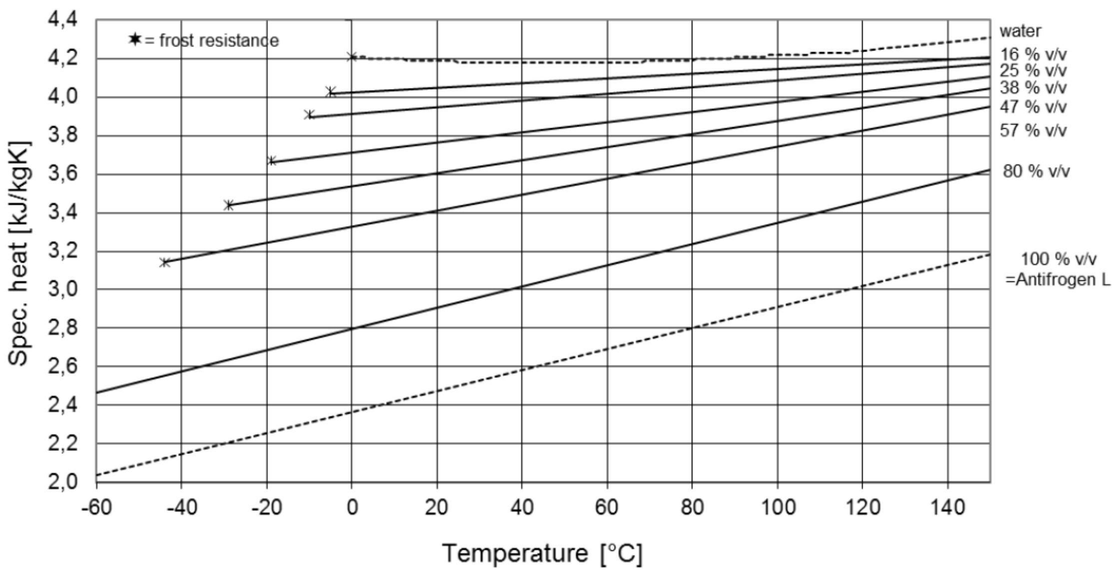
APPENDIX 2. Information about Antifrogen L- heat transfer fluid from Swiss Clariant. Technical datasheet, specific heat capacity-chart, thermal conductivity-chart and relative pressure drop-chart. Charts are in function of temperature for each propylene glycol-concentration in water solution. (Clariant International Ltd. 2014)

Technical data:

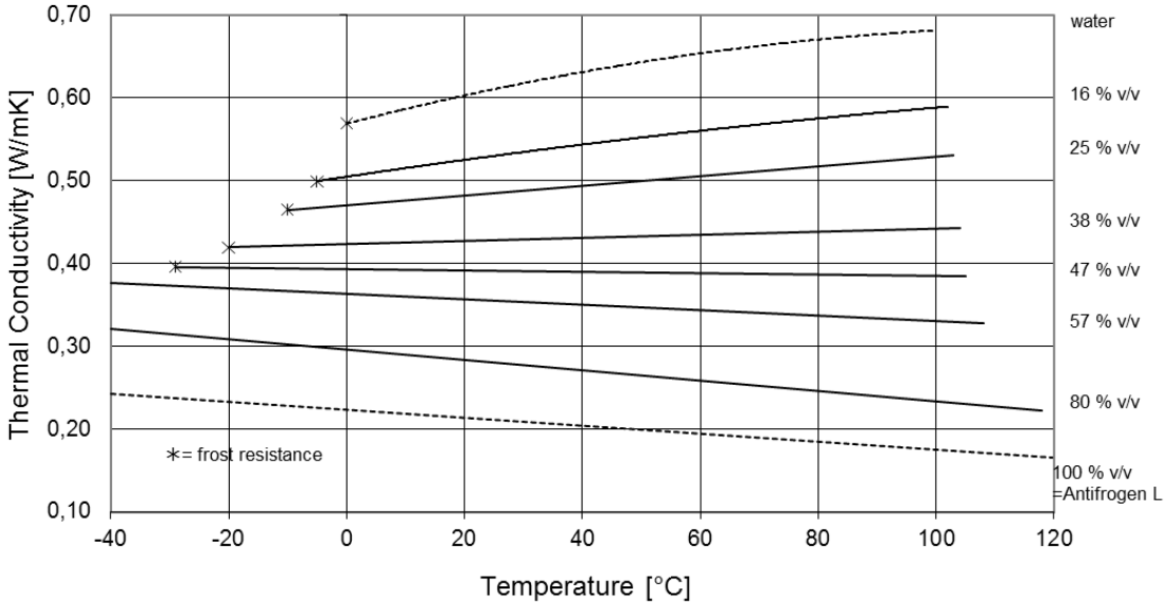
Density at 20 °C (DIN 51757)	g/cm³	approx . 1.043
Refractive index at 20 °C (DIN 51423, Teil 2)		approx. 1.432
pH-value (Antifrogen® N : Wasser = 1:2, DIN 51369)		approx. 8.6
Reserve alcalinity (ASTM D 1121)	ml c (HCI) 0.1 m	min. 4
Boiling point at 1013 mbar (ASTM D 1120)	°C	approx. 155
Pour point (DIN 51583)	°C	approx. -58
Kinematic viscosity at 20 °C (DIN 51562)	mm²/s	approx. 59
Surface tension at 20 °C (Antifrogen® N : water = 1:2, ASTM D 1331)	mN/m	approx. 47
Spec. el. conductivity at 25 °C (Antifrogen® N : water = 1:2)	µS/cm	approx. 2800
Specific heat at 20 °C	kJ/kg·K	approx. 2.5
Thermal conductivity at 20 °C	W/m·K	approx. 0.21

Specific Heat

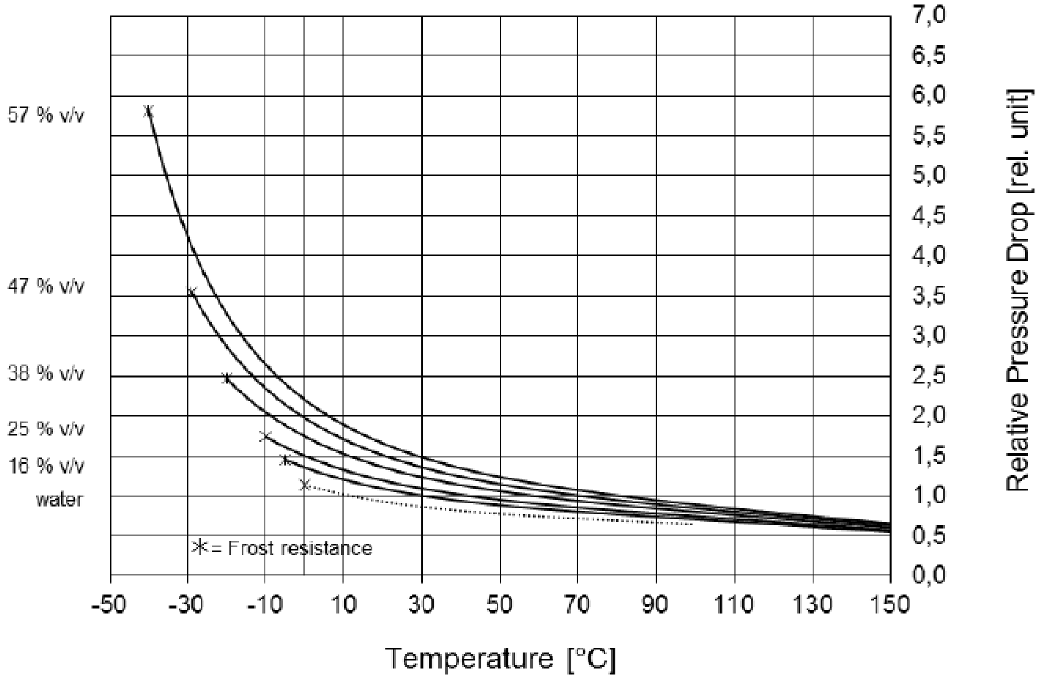
of Antifrogen L-water mixtures of different concentrations



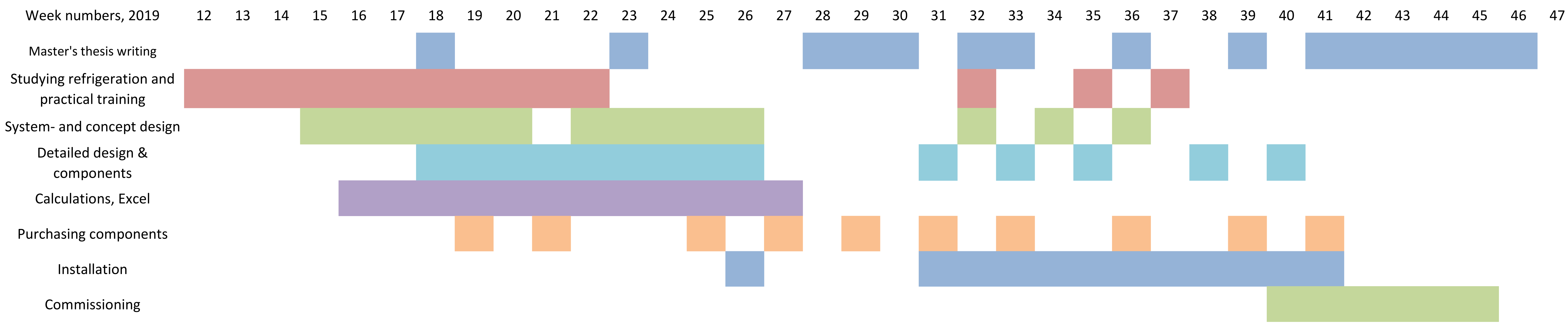
Thermal Conductivity
of Antifrogen L-water mixtures of different concentrations



Relative Pressure Drop
of Antifrogen L-water rmixtures in comparison with water (+10°C) in turbulent flow

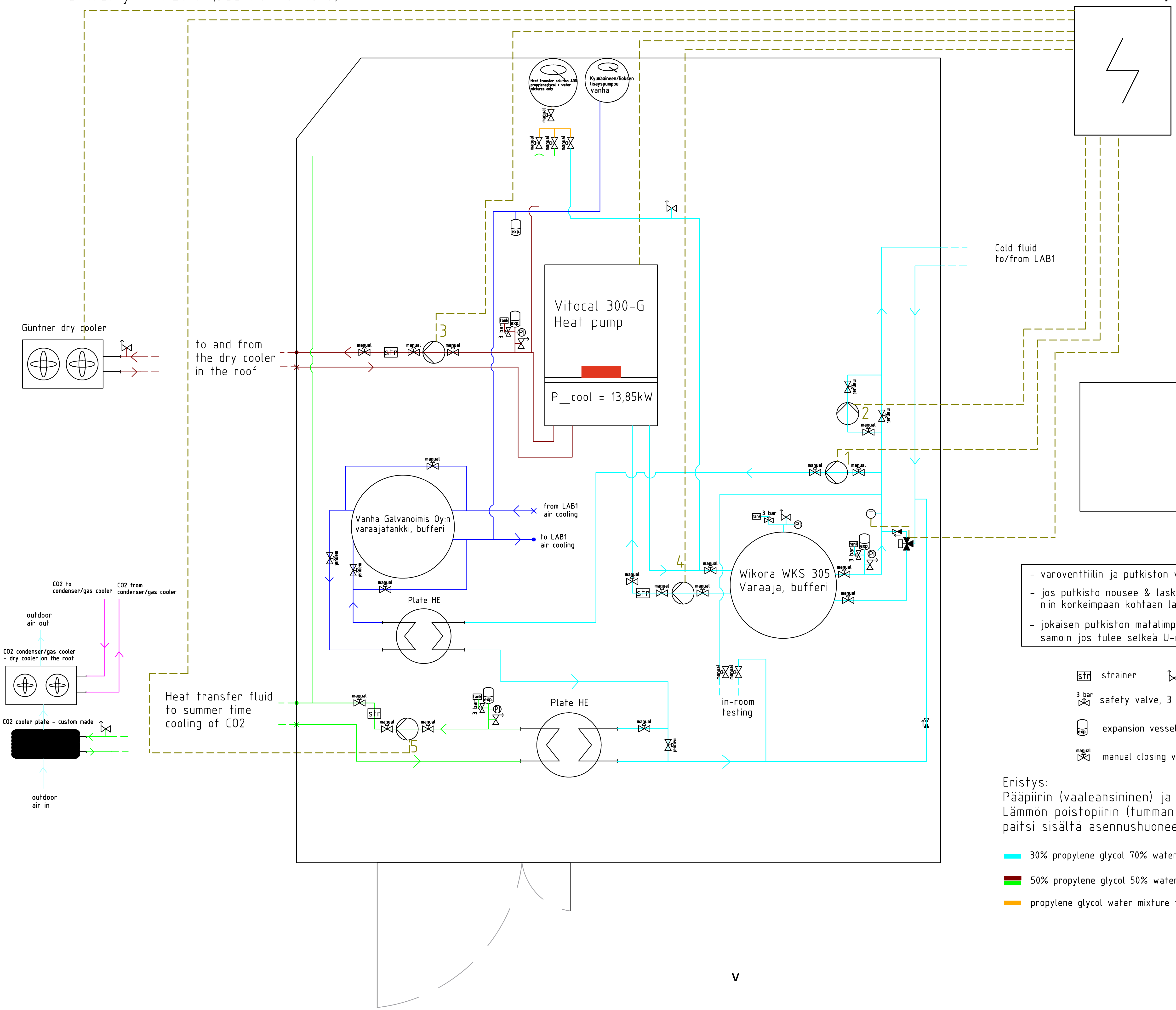


APPENDIX 3. Approximate Workflow - Indirect refrigeration system project and master's thesis
Working contract was 18.3 - 18.11.2019. (Weeks 12 - 47)



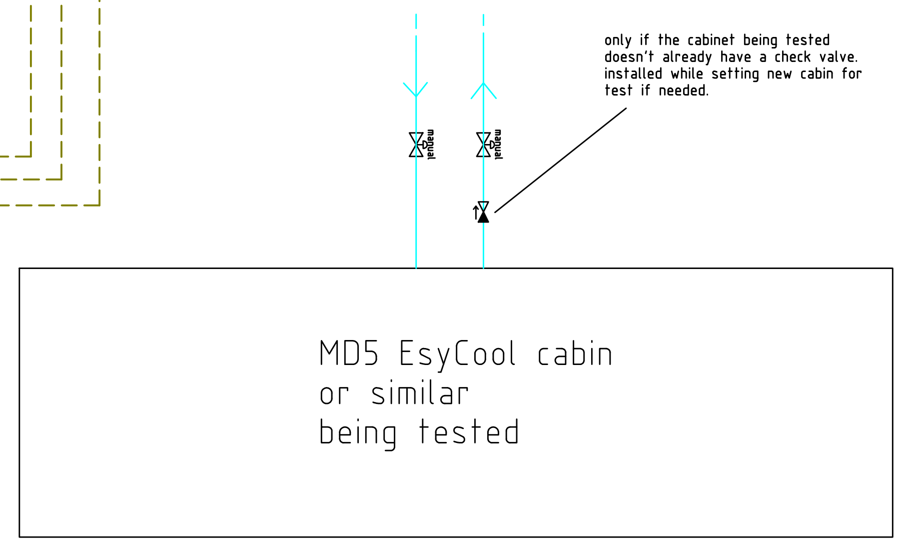
APPENDIX 4 – Flow diagram of our project's refrigeration system

Päivitetty 4.10.2019 (Jaakko Koivisto)



SÄHKÖKOHTEET:
- 5 kylmäliuospumppua
- Vitocal 300-G heat pump
- Lämmönsäädin
- Güntner dry cooler

kaikki varoventtiilit 3 bar-g
paisuntasäiliöt 6 bar
muiden osien pitää kestää min. 10 bar



- varoventtiilin ja putkiston väliin EI saa laittaa sulkuventtiiliä
- pumput lattiatasoon
- jos putkisto nousee & laskee selvästi jossain kohtaa (ylösalaisin oleva U-mutka), niin korkeimpaan kohtaan laitettava ylimääräinen ilmausventtiili
- jokaisen putkiston matalimpaan kohtaan täytyy laittaa tyhjennysventtiili! samoin jos tulee selkeä U-muoto, johon neste jäisi muuten killumaan.

HUOM!

str strainer
air vent valve
pressure indicator
3 bar safety valve, 3 bar-g opening pressure
exp expansion vessel
no-return valve
manual closing valve
tank safety valve discharge tank

Eristys:
Pääpiirin (vaaleansininen) ja CO2-piirin (vihreä) putket eristetään kokonaan.
Lämmön poistopiirin (tumman punainen) putkia ei eristetä,
paitsi sisältä asennushuoneesta paluuputki eristetään!

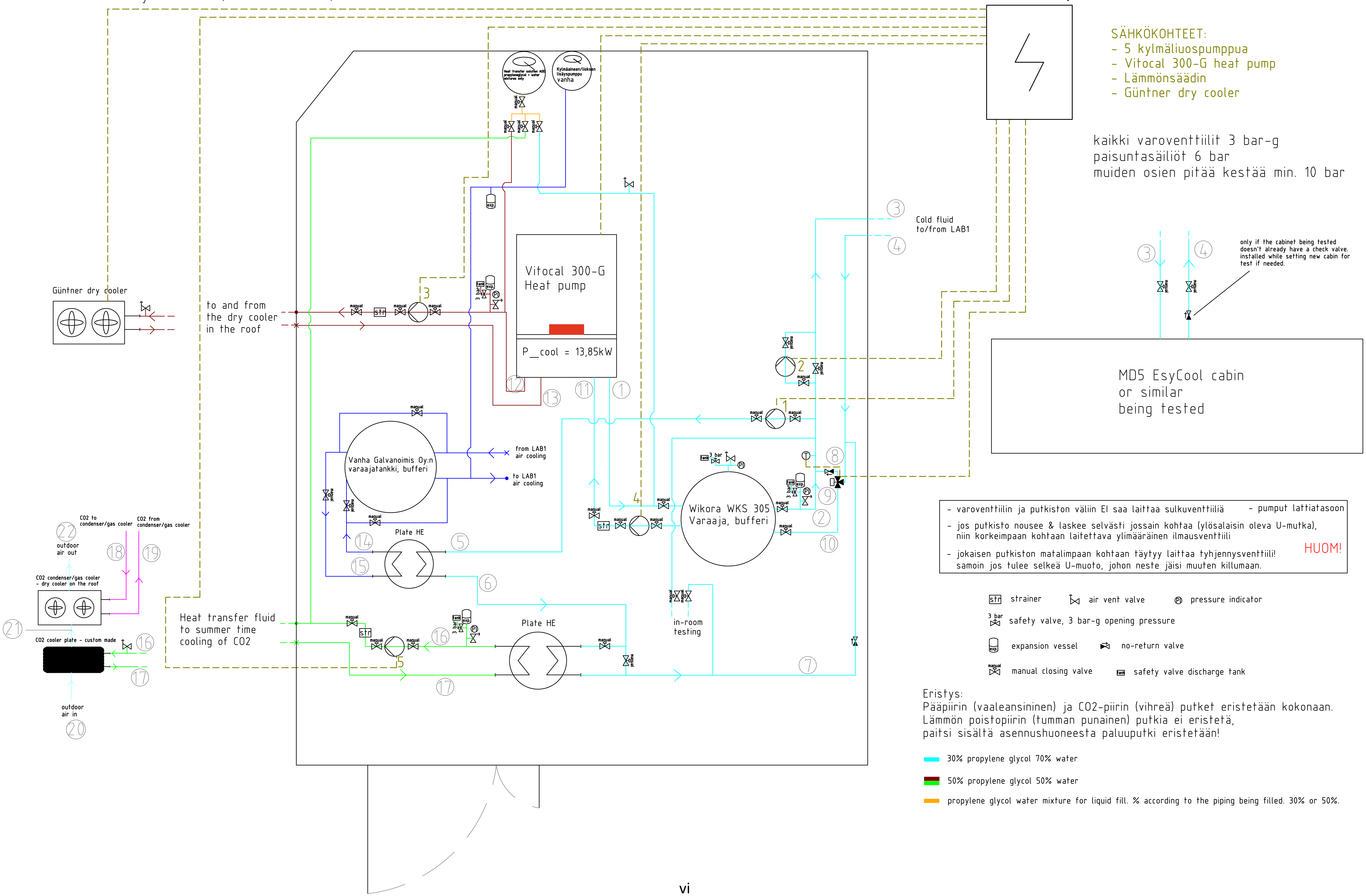
30% propylene glycol 70% water

50% propylene glycol 50% water

propylene glycol water mixture for liquid fill. % according to the piping being filled. 30% or 50%.

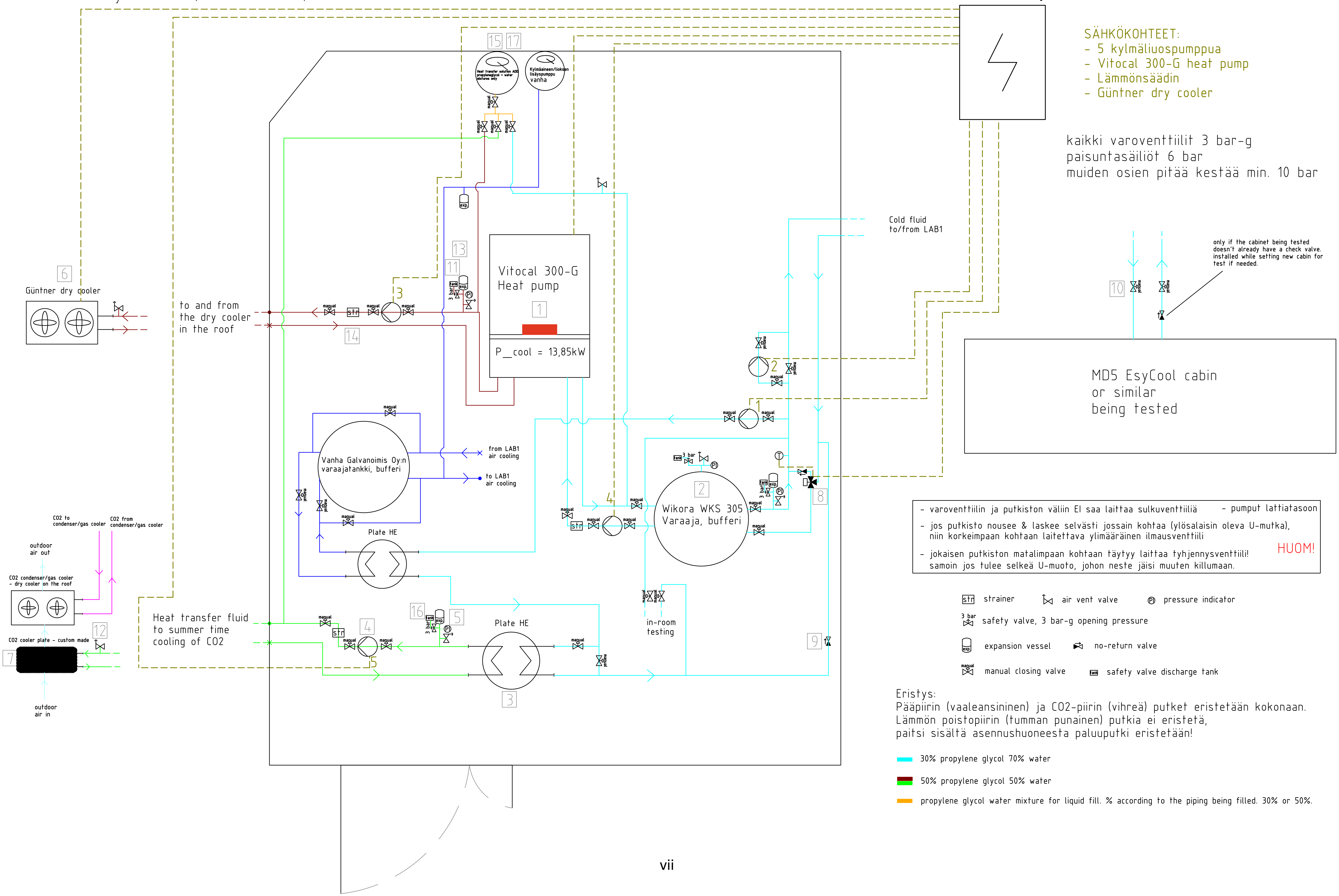
APPENDIX 5 - State points of our project's refrigeration system

Päivitetty 4.10.2019 (Jaakko Koivisto)



APPENDIX 6 – Components of our project's refrigeration system

Päivitetty 4.10.2019 (Jaakko Koivisto)



APPENDIX 7 - STATE POINTS AND SOME OTHER PROCESS VALUES

State points	substance	phase	m (kg/s)	T (°C)	p (bar)	D (mm)	v (m/s)	L (m)	V_pipe (litres)	Re	λ (laminar flow, Re under 2300!)	Δp (bar) laminar	Δp (bar) turbulent
1	pro-gly 30%	liquid, SC	0,75	-7		25,6	1,4	2	1,0	1811	0,035	0,028	0,040
2	pro-gly 30%	liquid, SC	0,75	-5		25,6	1,4	2	1,0	1811	0,035	0,028	0,040
3	pro-gly 30%	liquid, SC	0,5	-5		25,6	0,94	8	4,1	1207	0,053	0,076	0,072
4	pro-gly 30%	liquid, SC	0,5	0		25,6	0,94	8	4,1	1207	0,053	0,076	0,072
5	pro-gly 30%	liquid, SC	0,25	-5		25,6	0,47	3	1,5	604	0,106	0,014	0,007
6	pro-gly 30%	liquid, SC	0,25	0		25,6	0,47	4	2,1	604	0,106	0,019	0,009
7	pro-gly 30%	liquid, SC	0,25	10		25,6	0,47	4	2,1	604	0,106	0,019	0,009
8	pro-gly 30%	liquid, SC	0,75			25,6	1,4	1	0,5	1811	0,035	0,014	0,020
9	pro-gly 30%	liquid, SC	0,19			17	0,80	1	0,2	682	0,094	0,018	0,010
10	pro-gly 30%	liquid, SC	0,75			25,6	1,4	1	0,5	1811	0,035	0,014	0,020
11	pro-gly 30%	liquid, SC	0,75			25,6	1,4	2	1,0	1811	0,035	0,028	0,040
12	pro-gly 50%	liquid, SC	0,71	47		25,6	1,3	14	7,2	947	0,068	0,34	0,25
13	pro-gly 50%	liquid, SC	0,71	40		25,6	1,3	14	7,2	947	0,068	0,34	0,25
14		liquid, SC											
15		liquid, SC											
16	pro-gly 50%	liquid, SC	0,57	15		25,6	1,1	9	4,6	758	0,084	0,17	0,10
17	pro-gly 50%	liquid, SC	0,57	20		25,6	1,1	9	4,6	758	0,084	0,17	0,10
18	CO2 (R744)	gas, SH	0,11	100	70								
19	CO2 (R744)	gas-liq mixture	0,11	29	69,8								

SC subcooled SH superheated R744 is CO2 as refrigerant

All propylene glycol heat transfer fluids are mixed with water.

max mix ratio (m9/m2)	0,25
h_18 (kJ/kg)	526,8
h_19 (kJ/kg)	294,9
Δh (kJ/kg)	231,8
P_CO2 (kW)	25
x_CO2 (Sat. Liq.)	0

Values for All Air state points	
p (bar)	1,013
A (m^2) flow area	1,44
v (m/s)	2
V (m^3/s)	2,88
ρ (kg_da/m^3)	1,15
Cp_da (kJ/kg°C)	1,007
Water Vap. Heat (kJ/kg)	2430
p0 (Pa)	101300
Cp_H2O_sat (kJ/kg°C)	1,883

State points	sub-stance	RH (%)	m (kg_da/s)	T (°C)	x (kg_H2O /kg_da)	h (kJ/ kg_da)	p' (Pa) (if sat)	p (Pa)	Dew Point (°C)
20	Air	45	3,31	30	0,0120	60,92	4263	1918	16,8
21	Air	50	3,31	27	0,0112	55,75	3581	1790	15,7
22	Air	42	3,31	30	0,0112	58,83	4263	1790	15,7

Dew point depends on x.

p and p' are partial water vapor pressures in moist air.

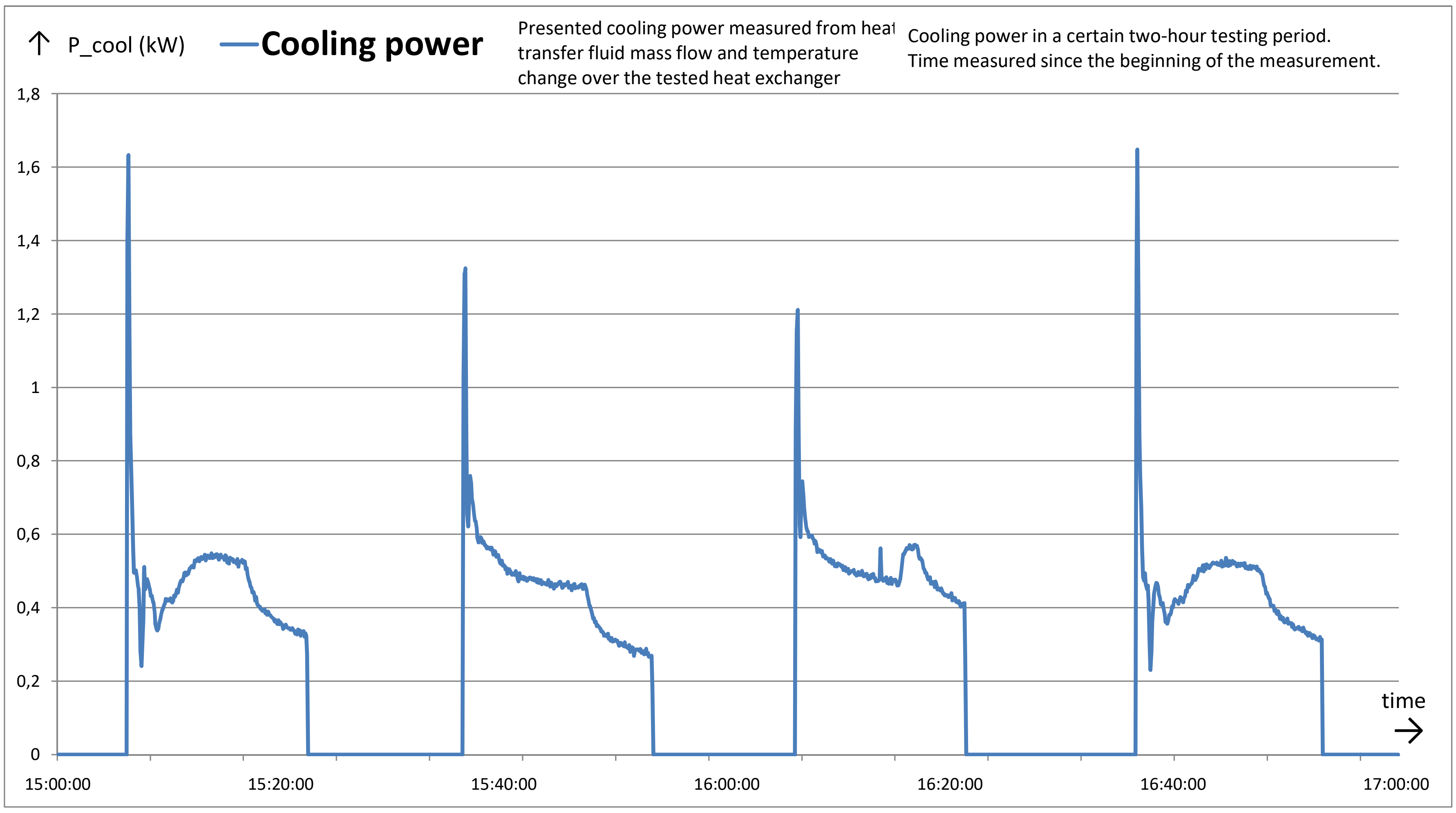
Δh Air over custom made Air-HE (kJ/kg_da)	5,17
Cooling demand for custom made Air-HE (kW)	17,1

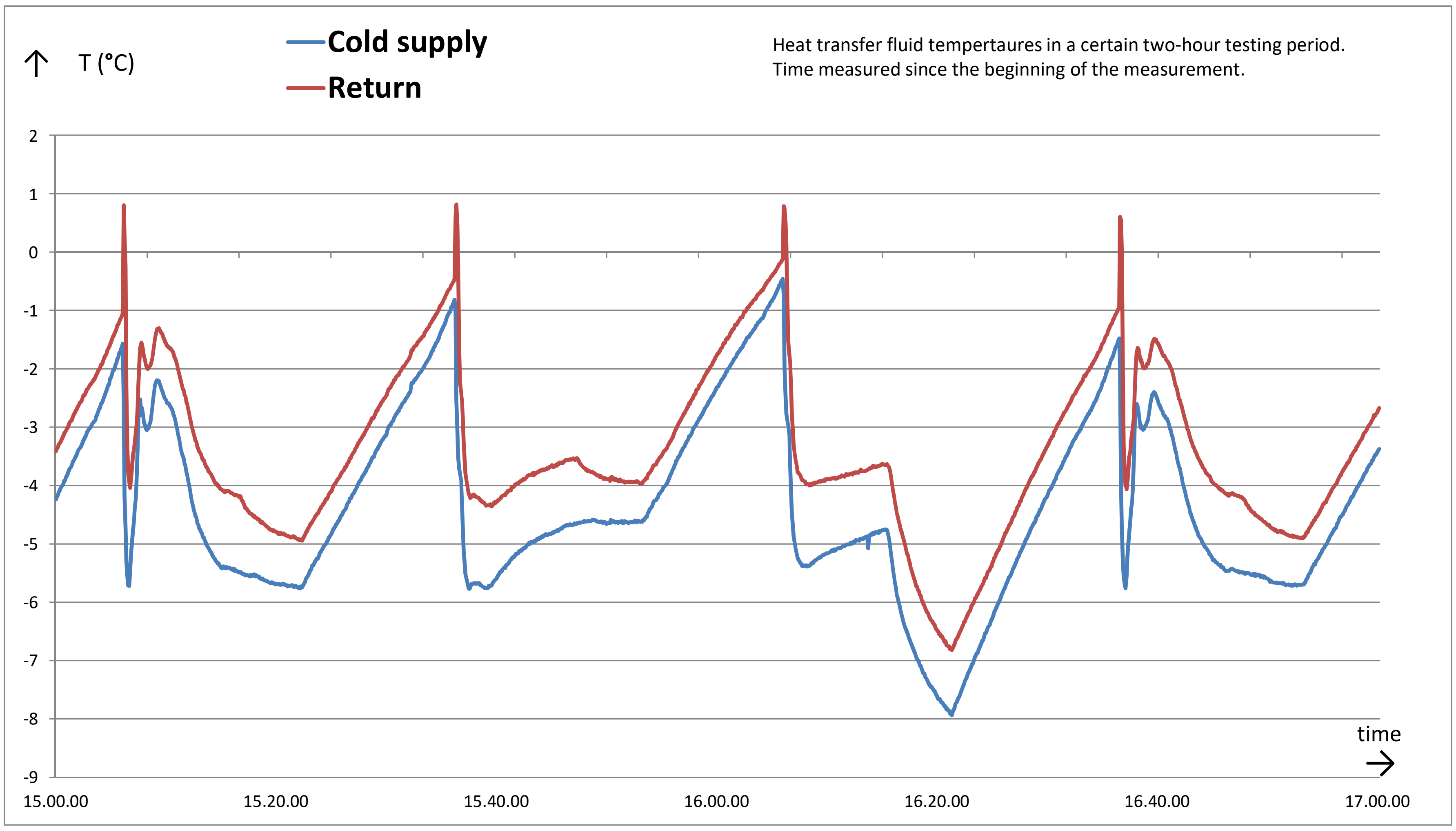
APPENDIX 8. Calculations - Circle and pressure vessel volumes and expansion events

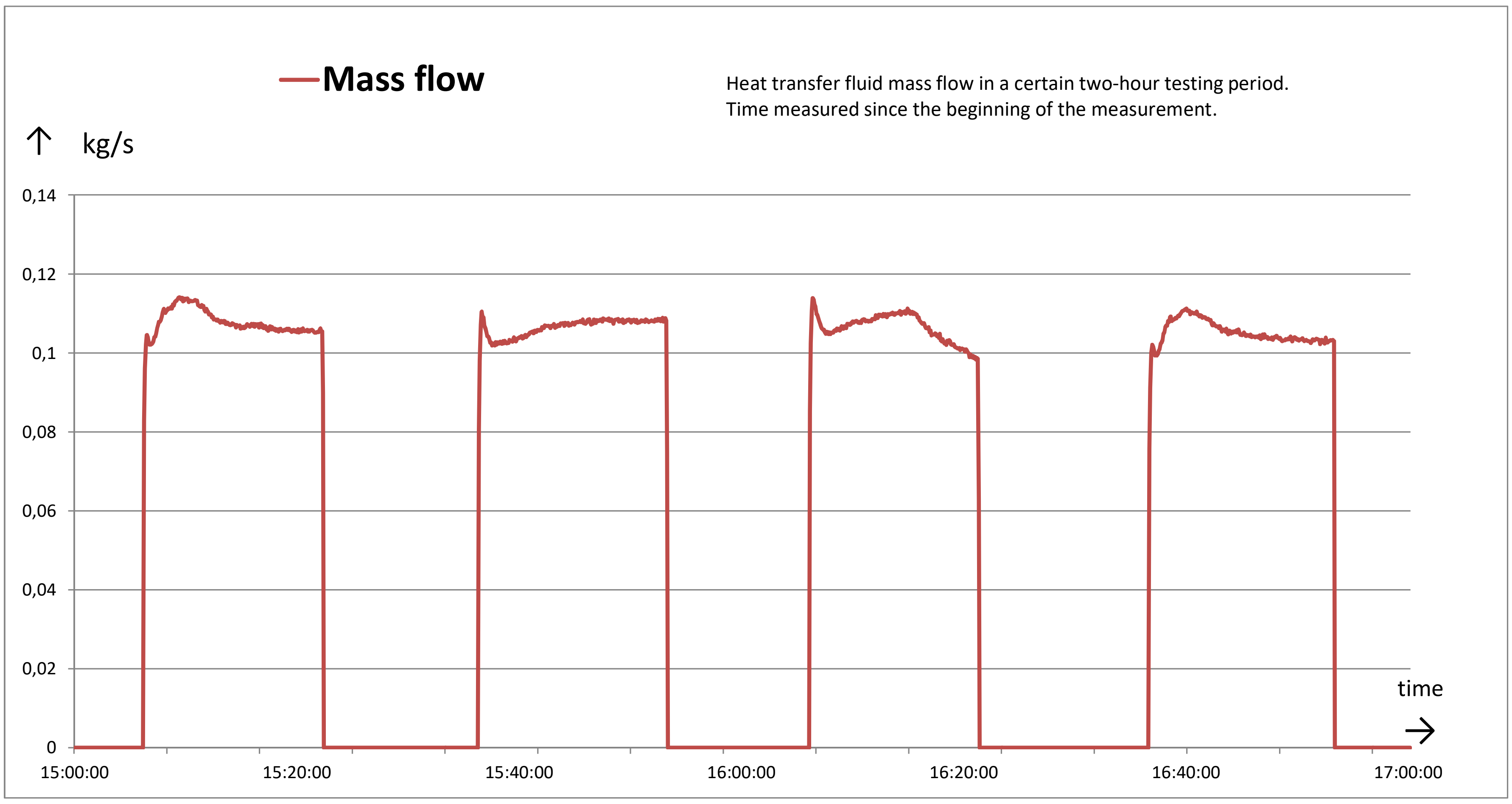
Circle	Circle volume without pressure vessel (L)	Expansion (%)	Expansion (L)	Heat transfer fluid total volume (L)	Suitable pressure vessel total volume (L)	Circle highest point (m)	Pre-pressure (Esipaine) (bar-a)	Lowest opera-tion pressure (bar-a) *	Max vessel pressure (bar-a) *	Vessel fill-level	Remai-ning fill	Efficiency coefficient
Cabin's test circle	349,0	2,5	8,72	359,9	17,45	3	1,5	1,65	4	0,09	0,91	0,53
CO2 circle	21,9	4	0,87	23,0	1,75	4	1,5	1,75	4	0,14	0,86	0,48
Heat dump circle	30,5	5	1,52	32,4	3,05	5	1,5	1,85	4	0,19	0,81	0,44

* Lowest operation pressure is chosen so, that at least 0,5 bar overpressure to atmosphere is in each circle's highest point.
Lowest operation pressure is given at the height of pressure vessel inlet.
* Max operation pressure 2,5 bar-g, but in capacity calculations we consider pressure in which safety valves open.

APPENDIX 9 - Test results - Our project's refrigeration system performance







Appendix 10. Heights and pressure levels of our project's indirect refrigeration system

EPÄSUORAN JÄÄHDYTYSJÄRJESTELMÄN KORKEUS JA PAINEET	Pääpiiri	CO2 piiri	Lämmön poistopiiri	
Putkiston korkein kohta (m)	3	4	5	
Jakotukin korkeus, jossa paine-säiliö ja painemittari (m)	1,5	1,5	1,5	
Putkiston korkeimman kohdan ja jakotukin korkeusero (m)	1,5	2,5	3,5	
Korkeuseroa vastaava paine-ero (bar)	0,15	0,25	0,35	
Painesäiliön esipaine (bar-g) (= Korkeus-eroa vastaava paine-ero pyöristettynä ylöspäin 0,5bar tarkkuudella)	0,5	0,5	0,5	Suositeltu täytekaasu typpi. Ilma vain jos typpeä ei ole saatavilla.
Tavoitepaine putkiston korkeimmassa kohdassa (bar-g)	0,5	0,5	0,5	
Jakotukin korkeudella olevan painemittarin lukema halutussa tavoitepaineessa (bar-g)	0,65	0,75	0,85	

HUOM! Kylmäluoksen lämpötila piireissä voi muuttua kun laitos käynnistyy.
Tyypillisesti: - Pääpiiri kylmenee - CO2 piiri kylmenee - Lämmönpoistopiiri lämpenee

Tämä voi vaatia liuoksen lisäämistä/vähentämistä, jotta pysytään riittävän lähellä tavoitepainetta käytön aikana.

- Laitoksessa oltava koko ajan ylipaine.
- Maksimi paine 2,5 bar-g.

Esim. kylmennyksen aikana liuoksen tilavuus pienenee ja paine pyrkii laskemaan.
--> Liuosta voi joutua lisäämään.
Esim. lämmityksen aikana liuos laajenee ja paine pyrkii nousemaan.
--> Liuosta voi joutua vähentämään, eli laskemaan hieman pois piiristä.
Varoventtiilit 3 bar-g.

Päivitetty ja tarkistettu: 4.10.2019 (Jaakko Koivisto)