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Reversible R744 (CO2) heat pumps applied in public trains in Norway

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Problem Description

1. Opportunities for use of reversible R744 heat pumps in the Norwegian public train sector. Literature survey.

2. Possible system layout and safety requirements for such heat pumping systems, able to achieve and maintain good indoor environment.

3. System simulation applying Csim, which results in seasonal performance data

4. Experimental System performance investigation, perform laboratory measurements on a R744 air conditioning system. Analysis and discussion of the results.

5. Compare Life Cycle Climate Performance (LCCP) of reversible R744 heat pumps with existing heat/cooling systems.

6. Discussion of LCCP results

7. Conclusion

Assignment given: 22. January 2009
Supervisor: Arne Mathias Bredesen, EPT

ABSTRACT

This report presents opportunities for use of CO_2 as refrigerant in the air conditioning system in public trains. The CO_2 system shall provide cooling in the summer and heating in the winter. CO_2 is a natural fluid which means that it exists naturally in the biosphere. Today 75% of the air conditioning systems in trains use R134a as refrigerant. The GWP of R134a is 1410 while CO_2 used as refrigerant is 0. A replacement from R134a to CO_2 gives possibilities of large environmental savings.

Three different technical system solutions of the heat pump are presented, each with its own method of provide cooling and heating. Solution I changes between cooling and heating by change the direction of the refrigerant flow through the system. Solution II changes between cooling and heating by change the configuration of the air streams through the heat exchangers. In Solution III the refrigerant flow direction and the configurations of the air streams is always the same. The whole heat pump is placed on a rotatable unit and the change between cooling and heating is done by rotating the whole heat pump 180°. In all the three technical solutions there are separated heat exchangers for fresh and exhaust air. This gives an energy efficient system which recover heat from the exhaust air. Computer simulation shows that a system solution with one evaporation pressure and one stage compression is problematic for low ambient temperatures; the system must stand temperatures to -40 °C. A system solution with two levels on the evaporation pressure and a two stage compression showed to improves the COP from 1,7 to 3,2 when the ambient temperature at -40 °C.

A railway coach need cooling when the ambient temperature is above 20 °C and heating below 15 °C. Norway is a country with cold climate. Weather statistic show that a train which drives in Oslo every day from 0600 to 1800 throughout a year will need cooling 3% of the time and heating 83% the time. This heating should be done by a heat pump and not with electrical heating as today. Results of the computer simulation shows that the annual energy consumption of heating the train will be reduced by 78 % if the designed CO_2 heat pump is used in stead of electrical heating.

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ABBREVATIONS

CO_2	Carbon Dioxide, R744
COP_c	Coefficient of Performance in cooling mode
COP_h	Coefficient of Performance in heating mode
EPA	Environmental Protection Agency
Eq	Equation
eq.	Equivalent
Evap1 and Evap2	Evaporator 1 and Evaporator 2
GC1 and GC2	Gas cooler 1 and Gas cooler 2
GHG	Greenhouse Gas
GW	Global Warming
GWP	Global Warming Potential
HP	Heat Pump
HXsim	Heat exchanger simulation programme developed by SINTEF
JAMA	Japanese Automobile Manufacturers Association
LCA	Life Cycle Analyse
LCCP	Life Cycle Climate Performance
MAC	Mobile Air Condition
OEM	Original Equipment Manufactures
Pinch	Pinch point temperature. The minimal temperature difference between hot and cold fluid in a heat exchanger
ProII	PRO II/PROVISION simulation software
R&D	Research and Development
SAE	Society of Automotive Engineers
TEWI	Total Equivalent Warming Impact
VDA	German Motor Vehicle Industry Association

PREFACE

In this Master Thesis reversible CO_2 heat pumps in public conveyance in Norway are investigated. The Master Thesis is written the Norwegian University of Science and Technology

Figures which do not have a reference attached to it are graphs and diagrams extracted from the calculation model made in Microsoft Excel 2003. The sketches are made in Microsoft Visio 2003 and Google SketchUp.

I want to give thanks to my supervisor Armin Hafner for always make me feel positive and encouraged when I leaving his office. I also give thanks to my lab partner Pitt Gotze, to the SINTEF employees Trond Andresen and Yves Ladam helping with computer simulation and at last every members of the "8 o'clock Café Sito Coffee Club" giving me a good start of every morning.

A INTRODUCTION

The objective of this Master Thesis is to research opportunities for use of reversible CO_2 heat pumps in the Norwegian public train sector and to find possible system layout for such a system, simulate the system which results in seasonal performance data and to compare the CO_2 system with existing heat- and cooling systems. In the report the HVAC system is referred to as MAC (Mobile Air Condition) or heat pump. However in all cases the system is going to provide cooling in the summer and heating in the winter. CO_2 is interesting as a refrigerant because it is environmentally friendly and a sustainable solution for the future. Old refrigerants such as CFCs and HFCs are, because of their devastating impact on the environment either phased out or in some cases lay taxes on. CO_2 is a natural fluid while CFCs and HFCs are synthetic fluids. Natural means that the fluid is present naturally in the biosphere while synthetic fluids are chemicals made by people. Refrigerants often have long and complicated names; therefore they have "nicknames", which will be used in this report. When CO_2 is used as a refrigerant it is often called R744. R stand for refrigerant, the first number 7 tells that the fluid is natural and the last number 44 is the molar weight. Tetrafluoroethane, which in this report always will be named R134a, is an HFC refrigerant which consist of 2 carbon atoms, 2 hydrogen atoms and 4 flour atoms and a from a "name giving procedure[1]" the fluid is called R134a. CO_2 refrigerant systems can not use the same technology as existing refrigerant systems due to the high pressure and the low critical temperature. The most used refrigerant in MAC systems today are R134a, but that hopefully will change in the future. In the car industry this fluid will be phased out from 2011 and CO_2 is a promising replacement. New technology has to be developed to exploit the properties of CO_2 to make high efficient, environmental friendly and economic competitive refrigerant systems.

A.1 Background

CO_2 is not a new refrigerant. The article *Fundamental process and system design issues in CO2 vapour compression systems* (Kim et al., 2004) gives a historically summary of the refrigerant. The first patent was made by Alexander Twining in 1850 and the first system was built in the late 1860. CO_2 was the dominating refrigerant until 1950-1960 when CO_2 started to be replaced by CFC fluids and ammonia. The Montreal Protocol focused on a phase out of CFC due to the destructive effect on the ozone layer. New fluids had to be found and HFC fluids were a promising replacement. Lorentzen and Pettersen (1993a) predicted that HFC fluids such as R134a were no sustainable solution.

> Post-war experience has demonstrated numerous cases where the introduction of new chemicals, foreign to nature, has led to unexpected and often serious problems…
> The present drive to replace CFCs with a new generation of synthetic chemicals may well result in a similar result in due course. (Lorentzen and Pettersen, 1993a).

1. [1]number: Number of C-atoms minus 1
2. number: Number of H-atoms plus 1
3. number: Number of F-atoms   (2001) Varmepumper Grunnleggende varmepumpeteknikk. SINTEF Energiforskning AS Klima- og kuldeteknikk.

Lorentzen and Pettersen concern about CFC fluid showed out to be well founded and today it is renewed interest of CO_2 as a refrigerant. The focus on natural fluids and development of technology of systems which can use these fluids has also led to a positive development of HFC refrigerant technology due to higher efficiency, less leakage and ratio of charge (Stene, 2008)

A.1.1 Global Warming Potential (GWP)

Definition

> The GWP is a measure of the future radiative effect of an emission of a substance relative to the emission of the same amount of CO_2 integrated over a chosen time horizon (Hafner and Nekså, 2006a).

The current practice of the time horizon is 100-year. The GWP of CO_2 is equal to 1 since CO_2 is the reference value. When CO_2 is used as a refrigerant the value is 0 since the CO_2 are a leftover product from the industry. The most used refrigerant in MAC today is R134a and has a GWP value of 1410 (IPCC, 2005). R134a got under international pressure only seven years after it was introduced into the market. In the year 2000 R134a represented three quarters of the worldwide HFC production (Schwarz, 2000).

A.1.2 Green house gas

Natural GHG include water, Carbon Dioxide, methane and Nitrous oxide (N_2O). Without GHG the average temperature would be -18 °C in stead of ca 15 °C. The manmade GHG emission gives a further warming effect. The GHG emission per capita in Norway is about twice as high as the world's average. Norway has the opportunity to develop new technology which is friendly for the climate and which can be used also in other countries. (www.miljoverndepartementet.no)

A.1.3 EU directive restrictions of use of GWP gasses in air condition systems for vehicles

Norway is today not a member of EU, however the EU directive mentioned below will be implemented in Norway, "Kjøretøyforskriften" § 20-3 (Lovdata, 2007). Directive 2006/40/EC of The European Parliament and of the Council of 17 May 2006 gives restrictions of use of refrigerants used in MAC in cars.

There are possibilities for large environmental savings if the fluid in MAC system in the transport sector changes from R134a to R744. Hafner and Nekså (2006b) estimated that between 38 and 93 x 10^6 metric tons of GHG emission would be saved if China had started using R744 in air condition systems in new cars in 2008 in stead of 2012. If India does the same the environmental savings is between 13 and 31 x 10^6 metric tons GHG.

From January 1, 2011, cars can not be EC type-approved or national type-approved if the air conditioning system use a refrigerant with higher GWP than 150. Cars which have been type-approved after January 1, 2011, is not allowed to get installed a new air conditioning system which use a refrigerant with higher GWP than 150 or refill the air condition system with a refrigerant with GWP higher than 150. From January 1, 2017, it is prohibited to sell or register all new cars with air conditioning system which use refrigerant with GWP higher than 150 and it is not allowed to refill any air conditioning system with refrigerant GWP higher than 150 in any car. The rules mentioned above are valid for cars of category M1 and N1. However the directive could be extended depending among other technological and scientific developments.

(Official Journal of the European Union Directive 2006/40/EC)

Category M1. “Vehicles used for the carriage of passengers and comprising no more than eight seats in addition to the driver's seat.”(Council Directive 70/156/EEC, 1970)

Category N1. “Vehicles used for the carriage of goods and having a maximum weight not exceeding 3,75 metric tons.”(Council Directive 70/156/EEC, 1970)

A.2 Carbon dioxide as a refrigerant

A.2.1 CO_2 Properties

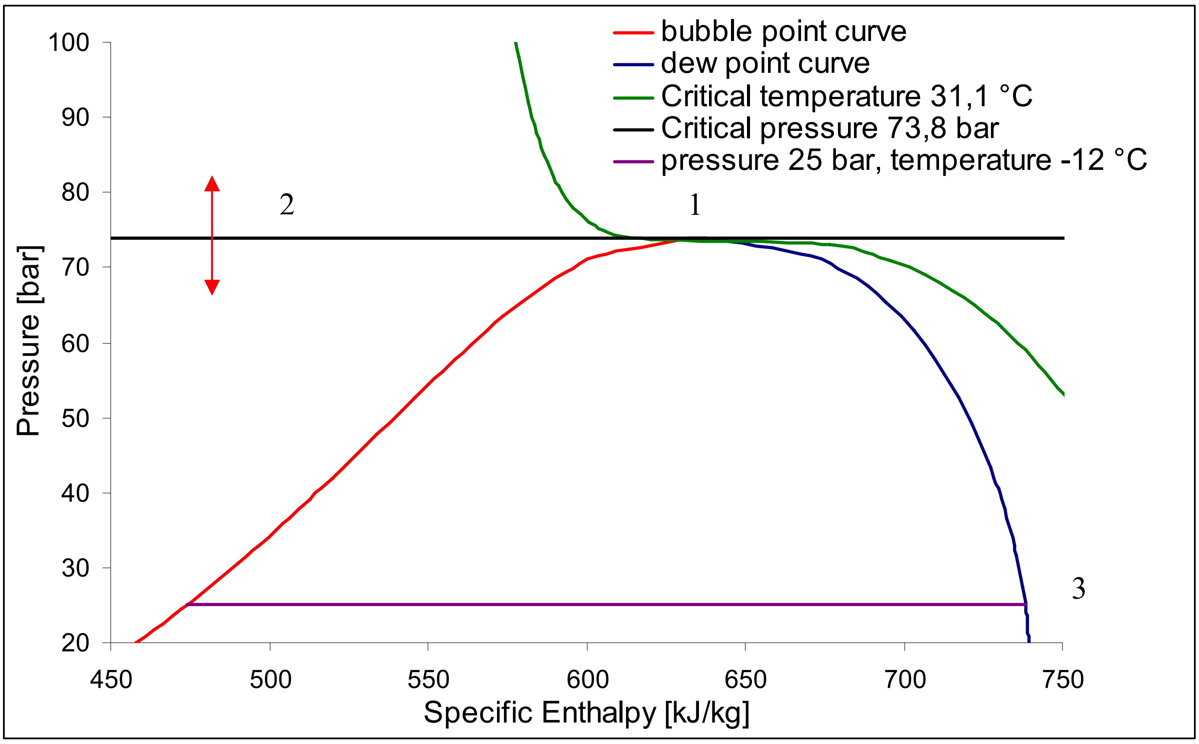


Figure A. 1 Pressure/enthalpy diagram of CO_2 (RnLib, 2007)

Figure A. 1 shows the pressure/enthalpy diagram of CO_2 and there are outlined some important points which makes CO_2 special compared to other refrigerants. Point 1 show the critical point where the pressure is 73,8 bar and temperature 31,1 °C. This mean that at temperatures above 31,1 °C CO_2 can not occur as liquid at any pressures, and at pressures higher than 73,8 Bar CO_2 can not occur as liquid at any temperatures. When CO_2 are used in heat pumps in most cases the high pressure side is higher than the critical pressure. The heat pump than give away its heat from a Gas Cooler in stead of a condenser. Point 2 shows the distinction between the supercritical and the subcritical area. A refrigerant cycle is called subcritical if it is below critical pressure and supercritical if its above critical pressure. If a refrigerant cycle operates both in subcritical and supercritical area the cycle is called transcritical, which often is the case for CO_2 systems. Point 3 shows a line where the pressure is 25 bar. The high pressure side of a R134a system usually have a high pressure side between 25-28 bars (Stene, 2008). At this pressure the temperature of CO_2 is -12°C which indicates that CO_2 requires a higher pressure standard. A transcritical CO_2 cycle has typically a high pressure side between 80 – 120 bar.

A.2.2 CO_2 systems are small and compact

In- a CO_2 cycle the density of the gas into the compressor is high. If the evaporation temperature is -20 °C the density of CO_2 is 14,8 times bigger than for R134a and evaporation enthalpy(h_{fg}) is also bigger for CO_2 (RnLib, 2007). This gives a high volumetric refrigerant capacity[2] (VRC) which again results in a small required volume flow through the compressor. Despite the high pressure level the pressure ratio between high and low pressure side is small for CO_2 cycle which gives compressor with high isentropic efficiency.

A.2.3 Typical design and components of a CO_2 system

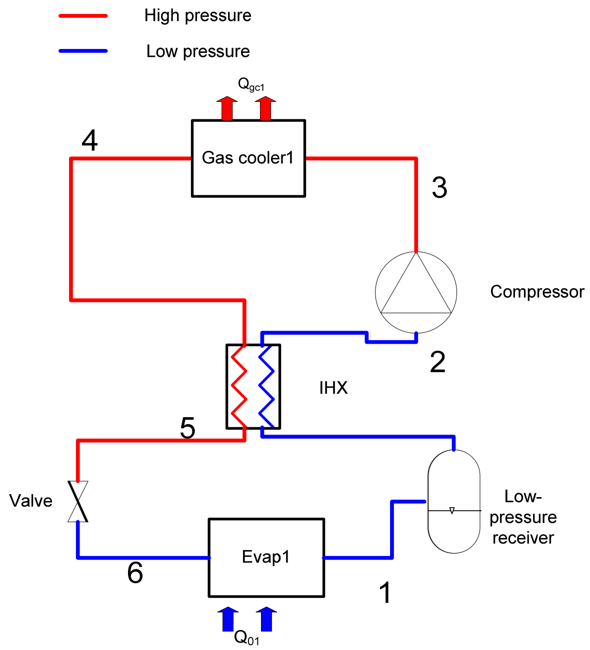


Figure A. 2 Transcritical CO_2 cycle consist of a compressor, gas cooler, interior heat exchanger, expansion valve, evaporator and receiver (Lorentzen and Pettersen, 1993b).

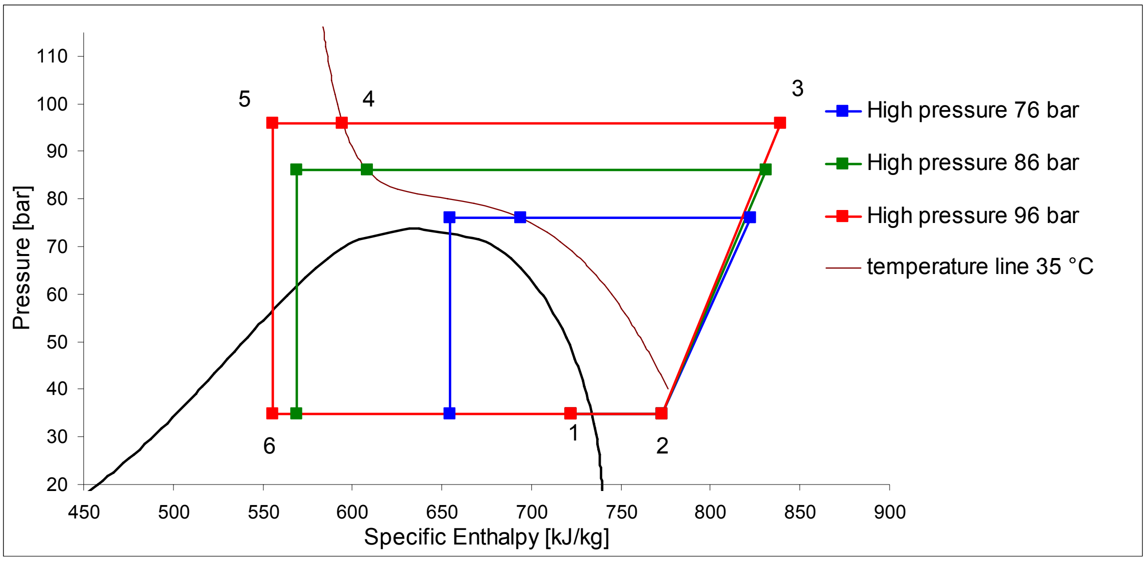


Figure A. 3 Transcritical CO_2 cycle for different high pressures in a Pressure Enthalpy diagram (RnLib, 2007)

[2] $VRC = h_{fg*}\,\rho [kJ/m^3]$

Compressor
The compressor is a volume machine. It receives gas and through the compressor the volume of the gas is reduced. The work needed to reduce the volume is transferred to the internal energy of the gas and the temperature and pressure is increased.
Gas cooler
Hot gas from the compressor exchanges heat with a cold medium for example air, water, glycol etc. The CO_2 is giving away its heat with a gliding temperature. This gives a good temperature profile between hot and cold fluid, if the temperature lift of the cold fluid is big, for example when water are heated from 5 to 80 °C.
IHX
To reduce the throttling loss it is important that the temperature of CO_2 before the expansion valve is as low as possible. Throttling losses is further explained chapter D.1. In the IHX high pressure gas is exchanging heat with low pressure gas. The temperature of the high pressure side is reduced and the temperature of the low pressure side increased. At the same time the IHX reduce the chance of having liquid in the compressor inlet conditions.
Expansion valve
In the expansion valve the refrigerant is going from high to low pressure and the temperature is decreased. The expansion valve is regulated so the amount of liquid into the evaporator is in balance with the suction of CO_2 into the compressor. Than the evaporator temperature is constant.
Evaporator
The temperature in the evaporator is decided by the pressure. The pressure has to be set such that the temperature in the evaporator is lower than the temperature of the medium which will make the refrigerant evaporate.
Low pressure receiver
Different operation conditions for the system required different mass flow of CO_2. The receiver works as a CO_2 buffer and guarantee enough refrigerant for the compressor when the conditions change. If the CO_2 system has a low pressure side which is not dimensioned for high pressures the receiver have another important task. When the system is not running all the CO_2 should flow to the receiver. This is to make sure that the pressure on the low pressure side does not exceed the pressure it is designed for. If there is “trapped” liquid in the evaporator and the system is not running the liquid will after a get the same temperature as the ambient surroundings. If the surrounding temperature is 30 °C the CO_2 pressure can reach 72 bar and that can destroy the evaporator.

A.2.4 Gas cooler pressure

The gas cooler pressure is an important parameter in a CO_2 heat pump. Optimal pressure gives maximum COP. Figure A. 3 illustrates the importance of choosing optimal pressure. The outlet gas cooler temperature is 35 °C for each pressure level. When the pressure is increased from 76 bar to 86 bar the specific evaporation enthalpy get much larger compared to the increased compressor work. This result in a higher COP. When the gas cooler pressure is increased from 86 bar to 96 bar the COP decreases. This is because the increased compressor work is bigger than the increased evaporation enthalpy. As Figure A. 3 shows the constant temperature line get close to vertical when the pressure continues to increase in that case the pressure is too high.

$$\Delta COP = \left(\frac{\dot{Q}_0 + \Delta \dot{Q}_0}{W_{comp} + \Delta W_{comp}} \right) \qquad \text{Eq 1}$$

$$\Delta W_{comp} > \Delta \dot{Q}_0 \qquad \text{Eq 2}$$

A change in the pressure has impact on the evaporation capacity and the compressor work. If the impact of increased high pressure changes the compressor work more than the evaporation capacity the COP will decrease and visa versa.

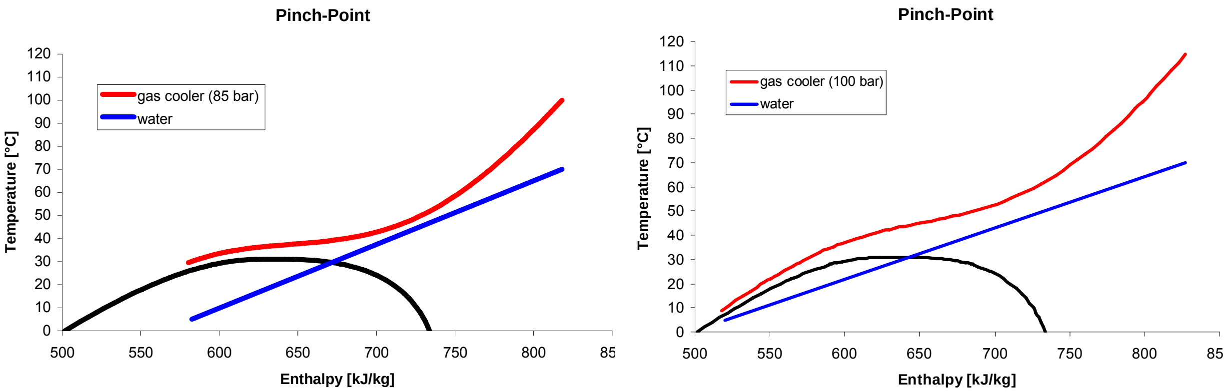
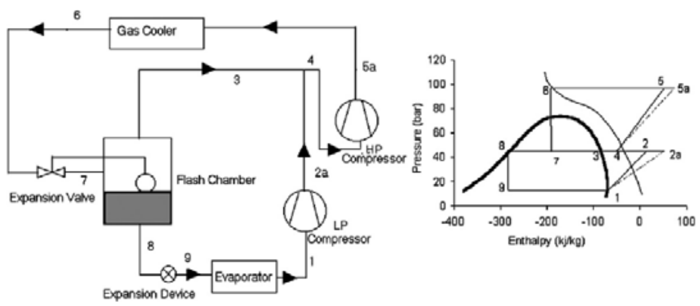


Figure A. 4 The figures shows the importance of the gas cooler pressure when water are heated from 5 °C to 70 °C

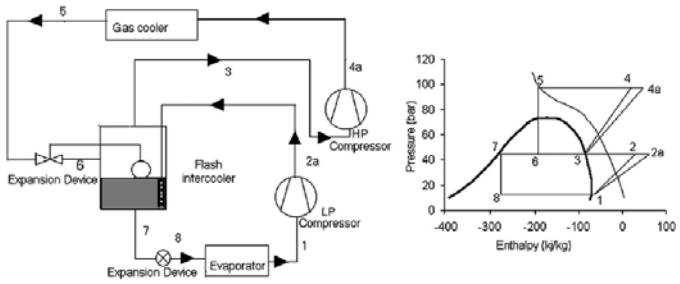
A transcritical cycle reject heat with gliding temperature typically between 30- 100 degrees (Stene, 2008). At low gas cooler pressure the cooling curve has an "S" shape, because of the non linear c_p value, the shape becomes more linear with increasing pressure. To reduce the throttling losses it is important to achieve lowest possible temperature in the gas cooler outlet. Every heat exchanger has a minimum temperature difference between hot and cold fluid below this temperature difference the heat exchanger is not able to exchange heat. The pinch-point is where this temperature limit occurs. As Figure A. 4 shows the pinch-point have to be located in the gas cooler outlet to achieve the lowest possible outlet temperature. The location of the pinch-point is dependent on the gas cooler pressure and the CO_2 mass flow rate. An increase of pressure or mass flow rate moves the pinch-point towards the gas cooler outlet. The water inlet and outlet temperature are constant, but when the pinch are in the gas cooler outlet more water can be heated than if the pinch are inside the gas cooler.

A.2.5 Two stage compression

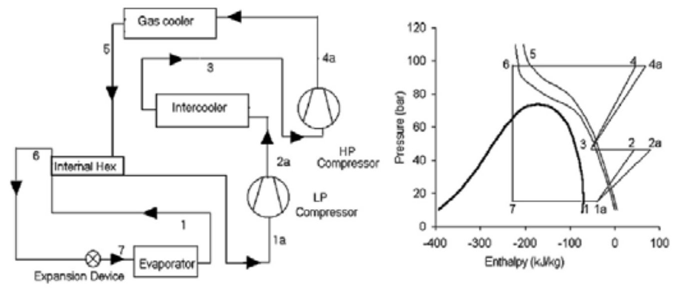
A high compressor outlet temperature is a problem when for low evaporator temperatures and high pressure ratios. It should be considered to use a two-stage heat pump system. The article *Optimization of two-stage transcritical carbon dioxide heat pump cycles* (Agrawal et al., 2007), present three different systems solutions shown in Figure 1. In all the cycles it has been assumed that stage 1 out of the evaporator is saturated vapour.



Two-stage cycle with flash gas bypass (system 1)



Two-stage cycle with flash intercooling (system 2)



Two-stage cycle with compression intercooling (system 3)

Figure 1 Different system solution two-stage system (Agrawal et al., 2007)

In the flash bypass cycle the fluid out of the LP compressor, state 2a, mixes with saturated vapour from the flash chamber, state 3. The supercritical mix at state 4 is compressed in the HP compressor and further cooled in the gas cooler to state 6. The fluid gets expanded into the flash chamber, state 7 and saturated liquid from the flash chamber is further expand to evaporation pressure and enters the evaporator.

In the flash intercooling cycle the fluid from stage 2a enters the flash intercooler, the entering fluid is de-superheated by evaporation of the CO_2 liquid in the flash intercooler. This increases the mass of vapour, stage 3, to the HP compressor. The rest of the cycle is similar to the cycle above.

In the system with intercooling supercritical fluid from the IHX is entering the LP compressor and compressed to 2a. It is cooled down in the intercooler to stage 3. Further the fluid is compressed in the HP compressor and cooled down to state 6, first through the gas cooler than through the IHX. The fluid is then expanded in one step to the evaporator temperature and enters the evaporator.

After optimization and simulation of the three systems with evaporator temperatures between -50°C to -30°C the article found out that at any given evaporation temperature system 1 gives the highest COP and it also gives the reason why it is like that. System 2 has lower COP due to the increased mass flow through the HP compressor which increases the compressor work. System 3 has the lowest COP due to big throttling loss. System 1 has the highest COP because the vapour in state 7 is separated in the separator and therefore bypasses the evaporator, where it would not produce any cooling effect, in stead it enters the HP compressor at a higher pressure and save compressor work.

The discharge temperature (temperature out of HP compressor) is reduced most compared to a one-stage system is system 2, because of the large reduction in temperature in the intercooler. The simulations showed that with a evaporation temperature of -50°C the discharge temperature gets reduced with 30% for system 2 while the reduction is only 10 % for system 3. System 1 is between, but discharge temperature is closer to system 2 than system 3.

The article: *Studies on a two-stage transcritical carbon dioxide heat pump cycle with flash intercooling* describe simulation studies on a identical cycle as system 2. The simulation results led to the conclusion that flash intercooling is not economical with CO_2 refrigerant. Fore some conditions the COP is even lower for the two-stage flash intercooling system than for a single-stage cycle. The reason is, as mentioned above in the system 2 description, that de-superheating of vapour in the intercooler almost doubles the mass flow rate to the HP compressor. The increase depends on the discharge temperature from the LP compressor and the mass flow rate in the evaporator. (Agrawal and Bhattacharyya, 2007). With this in mind it seems like the two-stage cycle with flash gas bypass is the best choice.

A.2.6 Life Cycle Climate Performance (LCCP)

To calculate the impact an MAC have on the environment it is necessary to analyse the whole life cycle of the system. LCCP is: “A measure of the overall GW-impact of equipment over its entire life cycle” (Hafner and Nekså, 2006a).

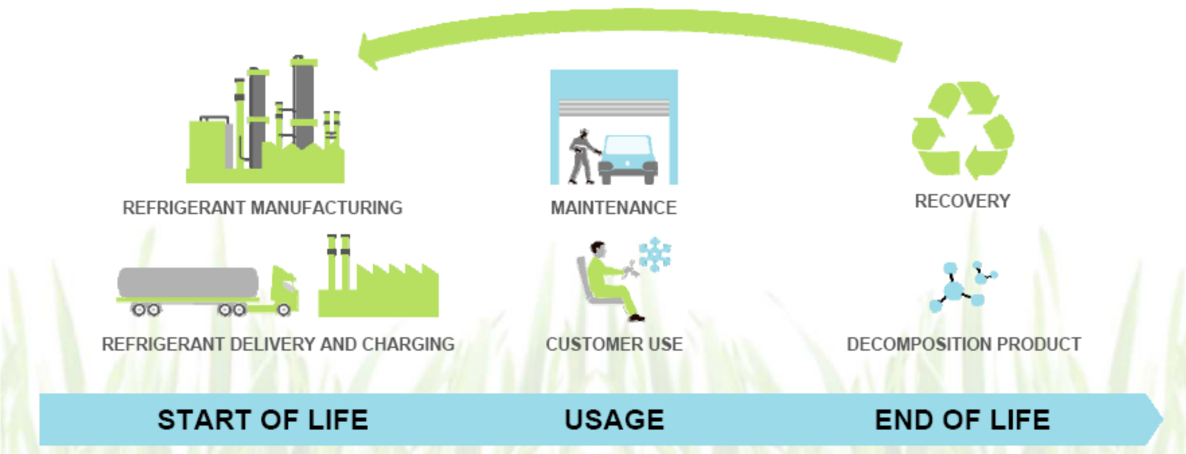


Figure A. 5 Life Cycle of an MAC refrigerant (1234yf OEM Group, 2008)
As Figure A. 5 shows the impact to the environment is dependent on both start of life and end of life which is important to consider when an existing refrigerant is going to be change with another.

The evolution and history of LCCP analysis is described by Papasavva and Andersen (2008). It started with Life Cycle Analyses for alternative refrigerants by Oak Ridge National Laboratory and the study started the term Total Equivalent Warming Impact (TEWI). Dr. Papasavva introduced a more advanced LCA in her 1997 Ph.D. thesis called Life Cycle Warming Impact (LCWI). In the 1999 report of Montreal Protocol Technology and Economic Assessment Panel (TEAP) comprehensive LCA was promoted by Dr. Andersen and the term LCCP was introduced for the first time.

$$TEWI = (GWP_{direct} + GWP_{indirect})_{lifetime\ MAC} \quad [kg\ CO_2\text{-eq.}] \quad \text{Eq 3}$$
$$LCCP = TEWI + (GWP_{direct} + GWP_{indirect})_{start/end\ of\ life} \quad [kg\ CO_2\text{-eq.}] \quad \text{Eq 4}$$

TEWI indicates the GW-impact of a MAC lifetime, but to get a total picture of the GW-impact the start and end of life also have to be evaluated. That is why the LCCP was developed. When new technology in general is developed it is important that the total amount of pollution is reduced and not only moved to another stage of the life cycle. The LCCP gives a more reliable foundation to compare different systems and refrigerants against each others than the TEWI.

The LCCP value of a MAC comes from three segments which is mass, direct and indirect contributions (Hafner and Nekså, 2006a).

1. The mass contribution is emissions of fuel required to transport the MAC
2. The direct contribution is leakages from the MAC, leakages and emissions during production, service, accidents and end-of-life
3. The indirect contribution is emissions related to run the MAC and to run the manufacturing plant

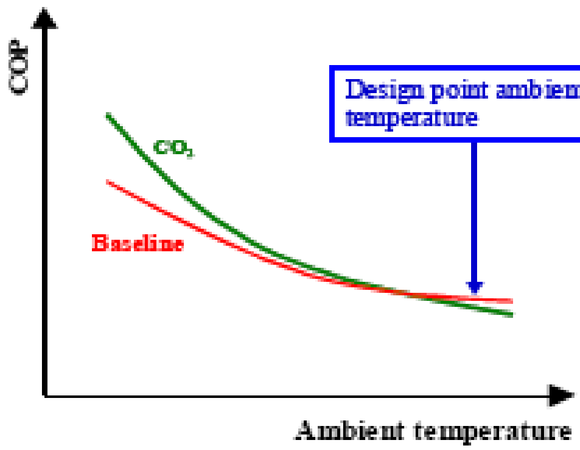


Figure A. 6 Principal COP progress for a CO_2 and conventional cycle with varying ambient temperature (Hafner and Nekså, 2005)

The LCCP is influenced by the COP. The COP_c for R744 is more sensitive for ambient temperatures than conventional refrigerants. When different systems and refrigerants are compared extreme condition is often used and the results will be misleading. In most of the operation time of the MAC the ambient temperature is in the region where R744 have the highest COP. In LCCP calculations mean/average conditions, or conditions based on seasonal climatic variations are used. (Hafner and Nekså, 2005)

Schwarz presented a forecast of the potential emission reduction if all MAC systems in new cars in Germany changes from R134a to R744 gradually from 2007 until 2009. The lifetime of a car was assumed to 12 years. In the year 2010 the savings of green house gas emissions is calculated to 1 million tonnes CO_2-eq. In the year 2021 there are no car left with R134a and the forecast showed that 4,5 million tonnes CO_2-eq greenhouse gas emissions could be saved annually from this year and onwards (Schwarz, 2000)

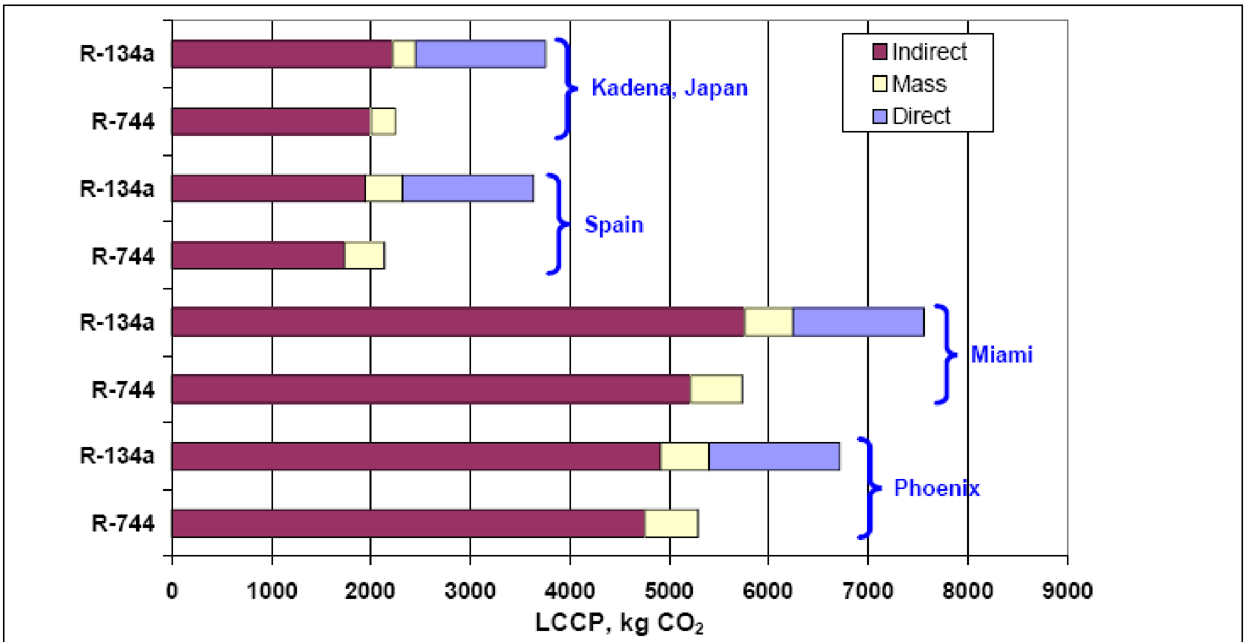


Figure A. 7 LCCP comparison between R744 system and R134a system in different climatic areas. The leakage of the R134a system is assumed to 80 g/year (Pettersen and Nekså, 2003)

Pettersen and Nekså did a comparison between a small 2002 R744 system and an R134a system showed a 20 to 40% reduction in LCCP, Figure A. 7. The comparison also showed a reduction of energy, due to lower need of fuel, to about 3% in Phoenix and 10% in Spain, Japan and Miami. Only at conditions that seldom occurs the COP for an R134a system is better than a R744 system i.e. COP is no reason against R744 (Pettersen and Nekså, 2003)

A.2.7 GREEN-MAC-LCCP©

As discussed above the LCCP is a measure of GW-impact. However the results of the analysis are highly dependent on assumptions and input values of the calculations. The following information is extracted from the presentation "GREEN-MAC-LCCP© The Metric for MAC Environmental Superiority" by Papasavva and Andersen (2008) which are representing General Motors Research & Development Center and U.S. EPA respectively. GREEN-MAC-LCCP© is a standard method to calculate LCCP values developed by General Motors in cooperation with JAMA, SAE, U.S. EPA and 50 world experts. The model can be used in Microsoft Excel and is free to download. The model is under continues developing and new inputs from industry and government are implemented consecutive. The presentation conclude that test results confirm that GREEN-MAC-LCCP© model is transparent, robust and Flexible. The LCCP CO_2-eq emission estimates are accurate and the model is a good tool for engineers and policy makers to use in selecting refrigerants and pursuing energy efficiency.

The most articles and presentation of the GREEN-MAC-LCCP© is written by those who make this model and the presentation is more like advertisement than an objective report.

As a result of improved energy efficiency in the car engine new cars today produce less waste heat. In cold climate there is need for supplementary heating for the passenger comfort. An alternative is to run the MAC as a heat pump. The following information about CO_2 as a heat pump in car is extracted from the article “Carbon dioxide (R-744) as supplementary heating device” by Hammer and Wertenbach (2000). CO_2 is a better suited refrigerant than R134a due to good thermo physical properties both for AC and heat pump. The HP can use ambient air, exhaust air or engine coolant as heat source. In this article Obrist Engineering adopt an R744 HP system to an Audi A4 1,6 ltr fuel engine and the HP using engine coolant as heat source. With ambient temperature of -20 °C and heating of the passenger compartment from -20 to +20 °C test result showed a reduction of 50% heating up time compared to standard heater core. The heat-up time of the engine did not increase because the load of the car engine is increased by the HP compressor.

A.3 Alternative refrigerants

It is no doubt that R134a should be replaced by other refrigerants for the future. The refrigerant that will be the replacement is still not decided. Today it seems like the car industry have three alternative refrigerants, R744 which is a natural fluid or the two synthetic fluids R152a and HFO-1234yf (Ghodbane et al., 2003). The main advantage of the two synthetic fluids is that these refrigerants can use current MAC technologies. The chemical industry will loose their income if CO_2 becomes the preferred choice, therefore this industry makes papers and documents presenting the synthetic refrigerants as the best sustainable solution, while they focuses on the negative sides of CO_2. The refrigerant HFO-1234yf is developed by DuPont and Honeywell and DuPont’s homepage tells that HFO-1234yf “has the potential to deliver the best balance of properties and performance, versus existing candidates, such as CO_2 (R744) and HFC-152a, in MAC systems… Technical and safety issues regarding the use of CO_2 in car air conditioning remain unresolved by the industry, despite years of development.”(www.dupont.com). Barbara Minor from DuPont tells in a presentation that HFO-1234yf has a GWP value of 4 and that the refrigerant is the global solution with good performance in all climates. She also says that it is the transition which requires lowest cost. LCCP results made by JAMA shows that HFO-1234yf have 20%-30% less LCCP values than R134a and CO_2, where CO_2 have the highest LCCP value (Minor, 2008). R152a and HFO-1234yf are both flammable, R152a slightly more than the other. In a front end collision it is chance that these two refrigerants can start to burn (Graz and Wuitz, 2008). A German TV channel, ZDF, demonstrated that HFO-1234yf is flammable and they also informed that the gas from the HFO-1234yf decomposes into toxic gasses which can be deadly for the passengers in the car (Umweltfreundliche Klimaanlage - auf den Inhalt kommt es an, 2008b). Shecco express a concern about the still not clear long term-impact on the nature if HFO-1234yf will be preferred as a global replacement for R143a (Global trends and opportunities for next generation MAC Refrigerants, 2008a).

A.4 Reversible R744 heat pumps in the Norwegian public train sector

Politicians play an important role if CO_2 heat pumps will be used in trains ore not. The Norwegian Pollution Control Authority (SFT) is a directorate under The Ministry of the Environment (Miljøverndepartementet). On SFT's home page they tell that "SFT work for a future without any pollution. We carry out the pollution politics and we are a guide, a guardian and administer for a better environment."(www.sft.no)

SFT was asked if there are or will be laws and regulations which will start a phase out of R134a in Norwegian trains and if Norway has one particular refrigerant they prefer as replacement for R134a

Alice Gaustad leader of the Climate and Energy section in SFT tells that they have no knowledge of previous studies on this field. There are today import taxes on HFCs and there are repayment arrangements for HFC fluids which are handed in for safe destruction. There are no specific plans of a phase out program for R134a in the train sector, but SFT encouraging the use of refrigerants which is less harmful to the environment.
SFT have not decided a particular fluid which they prefer as replacement for R134a. (Gaustad, 2009)

Regarding to trains NSB have decided to buy 50 new trains of the type "Staedler Flirt". The Trains have a total value of 4 billion NOK (€ 460 million). The largest single contract in NSB history (Hanssen, 20.08.2008). Information about the trains is found on NSB's home pages (www.nsb.no). Each of the 50 trains has 5 coaches. The top speed is 200 km/h and maximum amount of passengers 308/259 depending if it is a short or a long distance train.

NSB have started energy saving program for a five year period. The information about the project is extracted from the article *NSB skal spare 60 GWh hvert år* (Johnsen, 2008). The economical bounds of the project are 118 million NOK and the goal is to achieve an energy saving of 60 GWh each year compared to the consumption in 2005. To achieve the goal NSB cooperate with Enova[3] and Entro Energi AS[4] who is hired as branch experts. In the start of the project the power consumption was calculated to be approximately 280 GWh for driving the trains, 80 GWh driving the air conditioning plants and ca 40 GWh electricity heating parked trains. The calculations were made by Bane Energi. The project focuses on three main subjects: 1) Driving the trains in a way that use minimal energy. 2) Reduce the energy demand for parked trains. 3) Reduce the energy demand of ventilation, heating and cooling of the railway coach and at the same time achieve better indoor climate.

Frode O. Gjerstad, representative from Enova tells that he do not know what will be done ore what already have been done regarding CO_2 heat pumps in train. Gjerstad says that for NSB as a Green Company (environmental friendly) it would be a great matter to use a natural refrigerant, but he is worried about the high pressure.

[3] Enovas objective: "Enova is established to achieve an environmental change of the energy consumption and energy production in Norway" WWW.ENOVA.NO.

[4] Entro Energy AS objective: "Entro are going to do business activities more energy efficient and environmental friendly" WWW.ENTRO.NO.

Hans Olav Ness senior counsellor in Entro have the opinion that CO_2 heat pumps may be will be the future solution, but it is not a possible solution in the Staedeler Flirt trains. This means that CO_2 systems not will be considered before next time NSB are going to buy new trains, which at least not will happen before after 2012.

Geir Vadseth is the person in charge for the HVAC systems in NSB and following information in this section is based on personal communication with Vadseth. Totally NSB have over 800 air conditioning installations. R134a is the refrigerant used for the passenger coach while small systems for the driver of the train use R22 or R134a. A few numbers of trains have floor heating in the entrance with purpose to melt snow. In the new NSB trains the HVAC systems will also be able to work as heat pumps, there will be heat recovery from the exhaust air, the amount of fresh air will be regulated regarding to heat/cooling load. There are also put option on use of R744. Floor heating will be a part of the heating system combined with heating elements in the walls and in the ventilation batteries. The heating system must be dimensioned for ambient temperatures as low as -40 °C and speed of 200 km/h. The heat capacity of a coach is typically between 40 and 50 kW.

A.4.1 Mobile MACs in transport sector other than road transport

It is not only the road transport sector which have HFC emissions. The following information and figures about MACs in transport sector other than road transport is extracted from the article “Final Report Maritime, Rail, and Aircraft sector” by Schwarz and Rhiemeier (2007), if not other credits is given.

In 2006 the rolling stock of the Railway, tram and metro operators consisted in the EU of 175 000 units, 65 000 of them equipped with air conditioning systems. 75% of these MACs use R134a as refrigerant and 25% use R407c. Figure A. 8 show that the total charge of refrigerant in the railway sector is 1 605 tonnes CO_2 eq. This is 26% more than in the Maritime sector. However as Figure A. 9 shows the emission from the Maritime sector is much higher than for the Railway. This is because of the high leakage rate in the Maritime sector. It is estimated to 40% per year, even in indirect system it is common with a loss of 20% per year. In the Railway Sector the leakage rate is 5% for the majority of the vehicles (Appendix E).

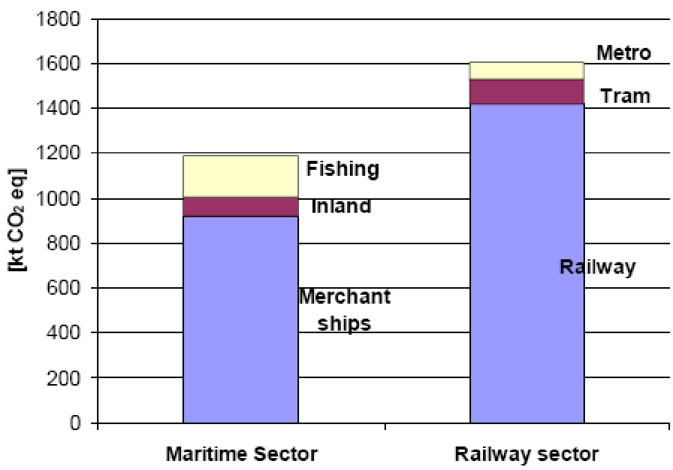


Figure A. 8 HFC charge in CO_2 eq in the Maritime and Railway sector in 2006

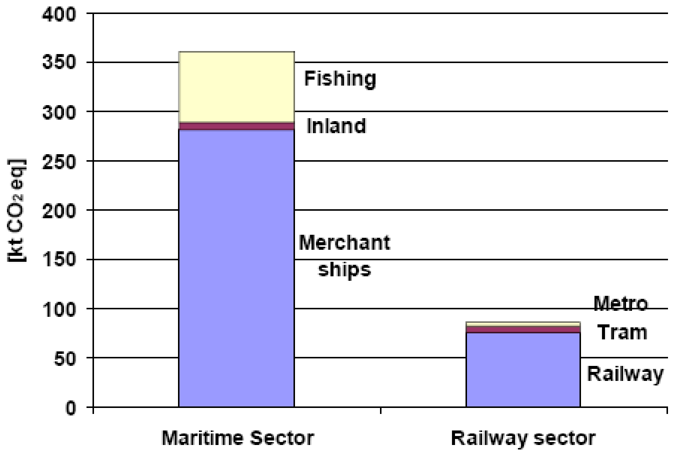


Figure A. 9 HFC emission in CO_2 eq in the Maritime and Railway sector in 2006

Calculations of the emissions in 2020 with a business-as-usual assumption shows that the emission in the railway sector will double to 174 tonnes CO_2 eq while the emissions from the maritime sector will increase to the threefold to 1 141 tonnes CO_2 eq. For comparison emission from the passenger car fleet in 2020 is also estimated in the article. Compared to the total GW emissions from all mobile air conditioners the Maritime sector will contribute to 5% while the Rail sector will contribute to 0,8%

The article concludes that it is more important to do changes in the Maritime sector than in the Railway sector. In the Maritime sector an emission reduction of 30% is relatively easy to achieve by using detectors to find leakages, better maintenance by trained staff and better regulation in the systems. This reduction will cost less than € 100 per tonne CO_2 eq. In the rail sector it is more difficult to achieve emission reductions because of the lower refrigerant charge per system and the low rate of leakages. A change over from HFC to CO_2 as a refrigerant will cost € 1 000 per tonne CO_2 eq (Schwarz and Rhiemeier, 2007).

This report focuses on the Railway sector by the setting that every sector should reduce its emissions as much as possible. New technology than will be developed which also can be important for other application than in the Railway sector. The railway sector also has the advantage that the MAC system can be mass produced. In the Maritime sector the refrigerant system is usually custom-made for each vessel.

Morgenstern and Ebinger have done a research on future railway vehicle air conditioning systems and how the energy consumption can be reduced. CO_2 and HFO-1234yf is two evaluated refrigerants. It is possible to use well known technology and the MAC systems does not have to be changed much with if HFO-1234yf will be used. However the authors show their worry about this new fluid because, as mentioned earlier, this refrigerant is flammable. Neither are all thermodynamic and technical properties of this refrigerant known or published. CO_2 have the disadvantage that the MAC have to stand much higher pressure (high pressure side ca 80-120 bar) and use gas cooler in stead of condenser. New technology has to be applied. This will give a high cost of the MAC. The explosion of CO_2 system may also be a problem, but many experiments show that due to low refrigerant charge and improved system regulation this concern almost disappears. CO_2 have the advantage that it is natural, environmentally friendly and freely available and there are no unknown negative side effects. HFO-1234yf will use known technology and do not have potential for big energy savings in the future, with CO_2 the main advantages in energy savings are still to come. Figure A. 10

Annual energy savings compared with conventional R134a system for air conditioning of passenger coaches(Morgenstern and Ebinger, 2008) Show results of an experiment were annual energy demands for different MAC system in railway passenger coaches are compared to an existing R134a system. In this experiment a prototype CO_2 system had lower energy demands than an optimized R134a system. (Morgenstern and Ebinger, 2008)

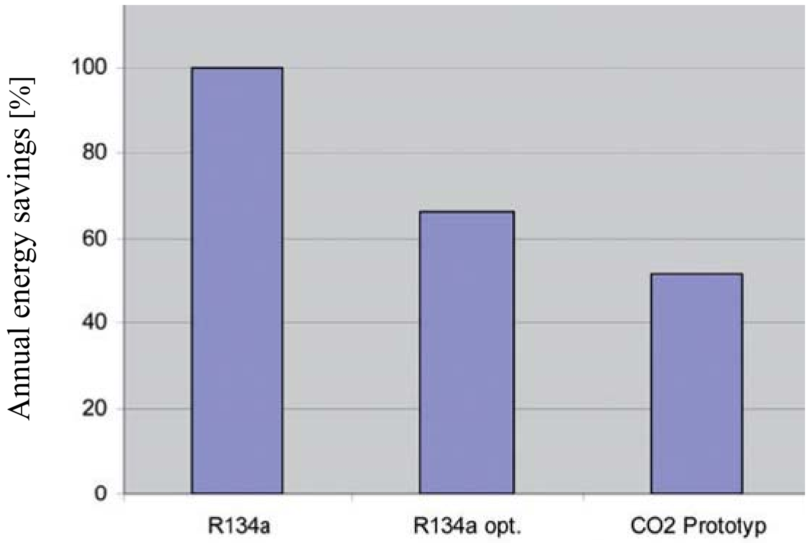


Figure A. 10 Annual energy savings compared with conventional R134a system for air conditioning of passenger coaches(Morgenstern and Ebinger, 2008)

B SYSTEM

This chapter outline some systems specifications and possible system layout for the heat pump system used in trains in the passenger coach.

B.1.1 System specifications

NSB use the international standard NS-EN 13129-1 for their HVAC systems. This standard gives some requirements for the system such as fresh air flow and indoor temperatures. The approximately specification of the rooftop unit and the required cooling and heating capacities for the new NSB trains is given by Lars Hiesche from the company Faiveley Transport Leipzig GmbH.

Table B. 1 Rooftop unit, heat and cooling capacity specifications

	Passengers	locomotive engineer
Length	2,5 m – 4 m	0,7 m – 1m
Width	1,8 m	1,6 m – 1,8 m
Height	0,35 m – 0,55 m	0,35 m – 0,55 m
Cooling capacity	20 kW – 70 kW	2,5 kW – 5 kW
Heating capacity	25 kW – 60 kW	2,5 kW – 7 kW

B.1.2 Fresh air flow

Table B. 2 Fresh air flow as a function of exterior temperature (Te) according to NS-EN 13129-1 (2002)

Exterior temperature (Te)	Minimum fresh air rate equivalent to +20°C and 50% relative humility	Minimum fresh air per coach (NSB usually have 70 seats per coach)
Te ≤ -20 °C	10 m^3/h per seat	700 m^3/h
-20 °C < Te ≤ -5 °C	15 m^3/h per seat	1050 m^3/h
-5 °C < Te	20 m^3/h per seat	1400 m^3/h
Te ≥ 20°C and Tim[5] ≤ 24°C	≥ 20 m^3/h per seat	1400 m^3/h

B.1.3 Temperatures

The indoor temperature has to be between 22 °C and 27 °C.
The temperature of the supply air outlets shall not exceed 65 °C, except during preheating. In case of floor heating maximum temperature of the surface is 27 °C. When the temperature is regulated the change of interior temperature shall be less than 0,1 °C per minute.

Table B. 3 Temperature limits

Area	Maximum temperature difference between mean temperature (Tim) and interior temperature settings (Tic)[6]
Comfort area	± 1 °C
Corridor	Heating mode: maximum 6 °C colder than Tic Cooling mode: maximum 5 °C hotter than Tic
Vestibules	Heating mode: Temperature shall lie between +10 °C and Tic Cooling mode: not more than 9 °C above Tic or never greater than +35°C

[5] Tim: Mean interior temperature. Arithmetic mean of the interior temperatures measured 1,10 m above the floor.
[6] Tic: Interior temperature setting. Theoretical temperature to be achieved by the room air.

B.1.4 Maximum and minimum exterior temperatures

Norway is defined as zone III both in winter conditions and summer conditions. According to this the exterior temperature is minimum -40 °C and maximum +28 °C.

B.1.5 Air ducts

The coach will be cooled and heated with air and it is needed an air duct system to supply the coach with treated air and remove exhaust air.

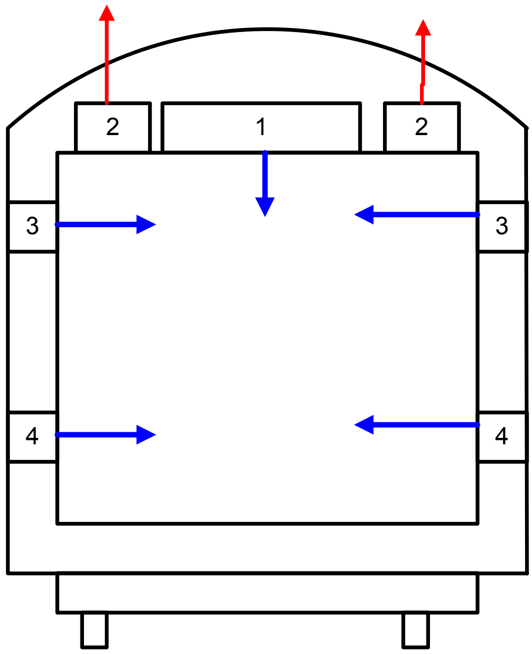


Figure B. 1 Possible solution for the air duct system. This system is used in some of NSB trains today (Vadseth)

The figure has the following ducts:

- Type 1: The duct in the middle of the roof ceiling contains treated air.
- Type 2: Exhaust air duct one on each side of the middle duct. In the different system described below the exhaust air ducts will take the air back to the HVAC unit. The wanted amount will be recycled and the rest will leave the train through a CO_2 heat exchanger.
- Type 3: Air ducts with treated air, one per side, along the baggage rack
- Type 4: Air ducts with treated air, one per side, along the wall under the windows.

There have to be an air duct system which leads air into and out of the rooftop unit.

- The air duct which leads exhaust air into the rooftop unit have to be separated in two flows, one air duct with recycled air and another air duct with exit air. There also have to be some kind of regulation system which sends the correct amount of air to recycling and exiting.
- There have to be an air duct which supply fresh air which exchange heat with gas --cooler 1 in cooling mode and evaporator 2 in heating mode. This air is going out of the system again.
- There have to be an air duct which supply fresh air to evaporator 2 in cooling mode and gas cooler2 in heating mode. This air is treated air is mixed with the treated recycled air and distributed around in the railway coach in the air ducts shown in Figure B. 1.

B.1.6 System safety

The CO_2 heat pump must fulfil the standard in “Norsk kulde- og varmepumpenorm”. The safety specification information is extracted from *kuldehåndbok* (2007) if not other credits are specified. CO_2 is non toxic however if the CO_2 concentration in the air exceed the concentration in our exhalation it get dangerous because the body is unable to get rid of the

CO_2 in the body (Haukås et al., 2007). There is a standard way to describe three levels of dangerousness of a gas:

TLV	Threshold Limit Value; maximum concentration of which do not give injurious to health with daily exposure (8 h/day)
IDLH	Immediate Danger for Life and Health; maximum concentration of which do not give serious health injuries with exposure over 30 minutes.
Deadly	Limit which will cause death

The TLV value of CO_2 is 70 gram/m^3 of air. If the system has a leakage it has to be found out in a worst case scenario if the concentration will exceed TLV. If it does it is necessary with an alarm system. Since CO_2 is heavier than air the CO_2 concentration detectors must be close to the floor and be protected from damage. The danger due to explosion is a function calculated from the pressure multiplied with the refrigerant volume. CO_2 systems have higher pressure, but less volume than R134a systems which is the reason why they are in the same danger classification group.

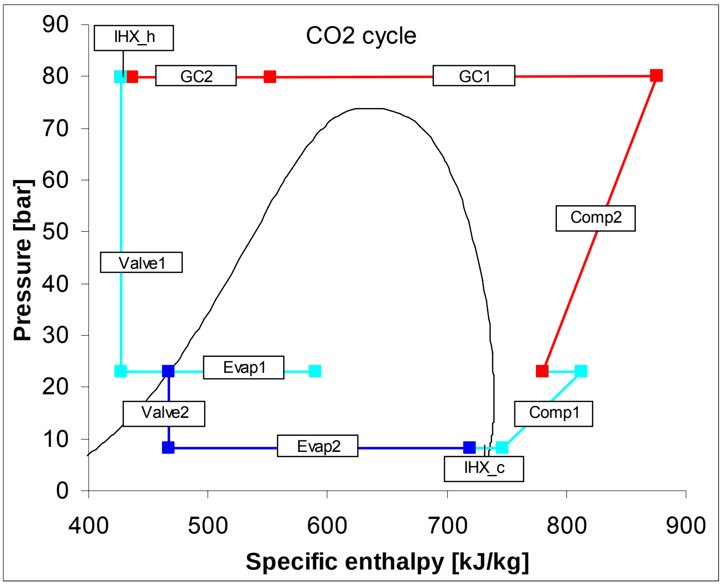
B.2 System layout

There are many ways to design the HVAC system. In this section four different system layouts will be presented and discussed.

Solution I. The heat pump is reversible. The system changes from cooling mode to heating mode by reverse the direction of the refrigerant flow through the system. In this solution the heat exchangers have to be able to operate both as evaporator and gas cooler.

Solution II. The heat pump is not reversible i.e. the CO_2 always flow in the same direction and the air heat exchanger do not change function from gas cooler to evaporator when the system change from cooling to heating mode. The system change from cooling to heating mode by change the direction and configuration of the air streams through the heat exchangers.

Solution III. The heat pump is located on a rotatable unit and change from cooling to heating mode by rotate 180 °. The CO_2 always flow in the same direction and the configuration of the different air streams is also similar in the two modes.



	Cooling mode (T_ambient > T_exhaust)	Heating mode (T_ambient <T_exhaust)
Gas cooler 1	Fresh air (exit)	Exhaust air (recycled)
Gas cooler 2	Exhaust air (exit)	Fresh air (enter)
Evaporator 1	Fresh air (enter)	Exhaust air (exit)
Evaporator 2	Exhaust air (recycled)	Fresh air (exit)

Figure B. 2 Enthalpy Pressure diagram of the CO_2 cycle and configuration of the air streams.

All the system solutions have four heat exchangers which exchange heat with air, one glycol exchanger for floor heating and one internal heat exchanger. There are two gas coolers and two evaporators because the air has different temperature levels. The fresh air have one temperature and volume flow and the exit/recycled air have another temperature and volume flow. The exhaust air is already treated and has a temperature closer to the wanted temperature in the indoor coupe than the ambient temperature i.e. In the summer the exhaust air is already cooled, and in the winter the exhaust air is already heated. To be as energy efficient as possible it is important to exploit the exhaust air. A solution where fresh and exhaust air first is mixed together and then sent to a heat exchanger is not preferred since some of the potential energy in the exhaust air is lost in the air stream mixing. Therefore there is separated evaporators and gas coolers for fresh and exhaust air. The glycol heat exchanger is for floor heating and is only in use in heating mode. In both heating and cooling mode the coldest air crosses gas cooler 2. This is to get a lowest possible temperature out of the gas cooler which give lowest possible throttling loss. The simulation results, which are presented and discussed in Chapter C, show that a design where the two evaporators is in series and having similar evaporation pressure is not a optimal design. It is two reasons for this. The first is that when CO_2 come out of evaporator 1 with a vapour fraction around 0,5 it is hard to get the CO_2 flow distributed equally into evaporator2, some of the pipes in the heat exchanger may be empty and others full. Therefore a receiver is implemented between the evaporators, in this way saturated liquid enters evaporator2. After the receiver it is put a second expansion valve, because of the temperature difference between recycled air and fresh air. In that way the two evaporators have different evaporation temperatures. In heating mode recycled air meets evaporator1 and fresh cold air meets evaporator 2 because recycled air have higher temperature than fresh air. In cooling mode it is visa versa.

B.2.1 Solution I

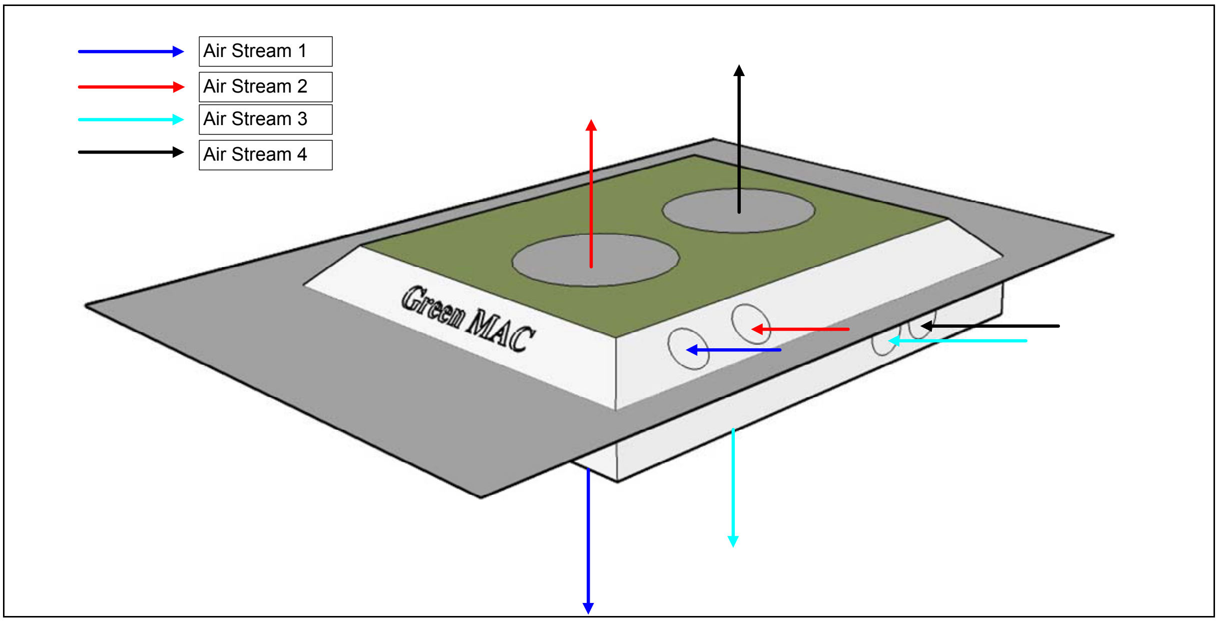


Figure B. 3 Principal sketch of the rooftop unit. (Dimensions according to Table B. 1: Length 2,5 to 4 m, Width 1,8 m, Depth 0,55m)

The two circles on the front side are where fresh air enters the system. One fresh air stream enters the coach and the other exit the system. The two circles on the rear side in the coach rooftop ceiling is the intake for exhaust air. One exhaust air stream is recycled and enters the coach and the other exit the system. The front circle on the rooftop of the unit is the exit of fresh air. The rear circle on the rooftop unit is the exit of exhaust air. The air streams are according to Figure B. 2 and the arrows in Figure B. 3:

Air stream 1.	Fresh air which is going to enter the system as treated air. Exchange heat with Evaporator 1 (summer) / Gas cooler 2 (winter)
Air stream 2.	Fresh air which is going to be exit the system. Exchange heat with Gas cooler 1 (summer) / Evaporator 2 (winter)
Air stream 3.	Exhaust air which is going to be treated and recycled. Exchange heat with Evaporator 2 (summer) / gas cooler 1 (winter)
Air stream 4.	Exhaust air which is going to exit the system. Exchange heat with Gas cooler 2 (summer) / Evaporator 1 (winter)

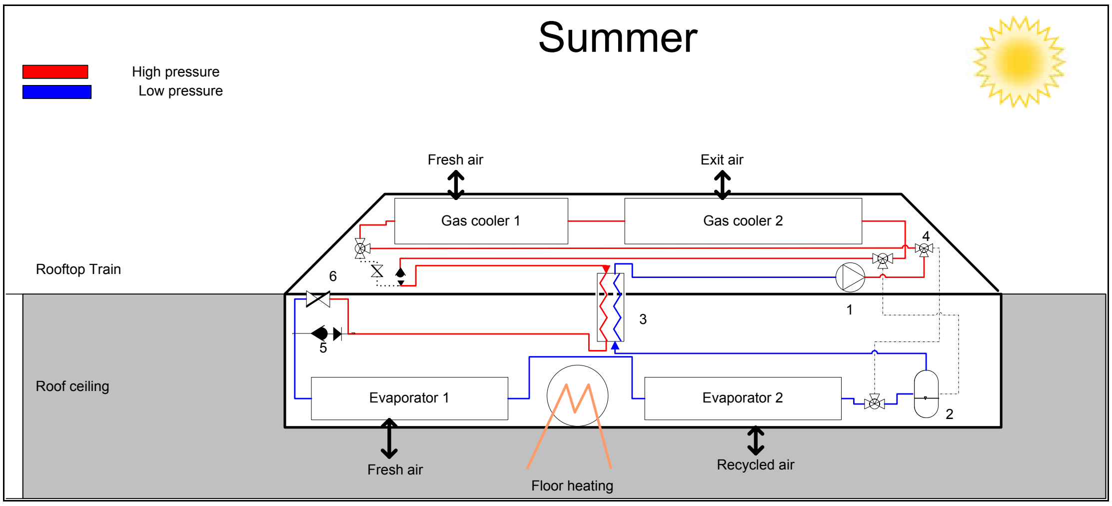


Figure B. 4 System circuit of solution I cooling mode. 1) Compressor. 2) Low pressure receiver. 3) Internal heat exchanger. 4) Three way valve that control the direction of the refrigerant flow. 5) Bypass valve that is open from one side and closed on the other side. 6) Expansion valve

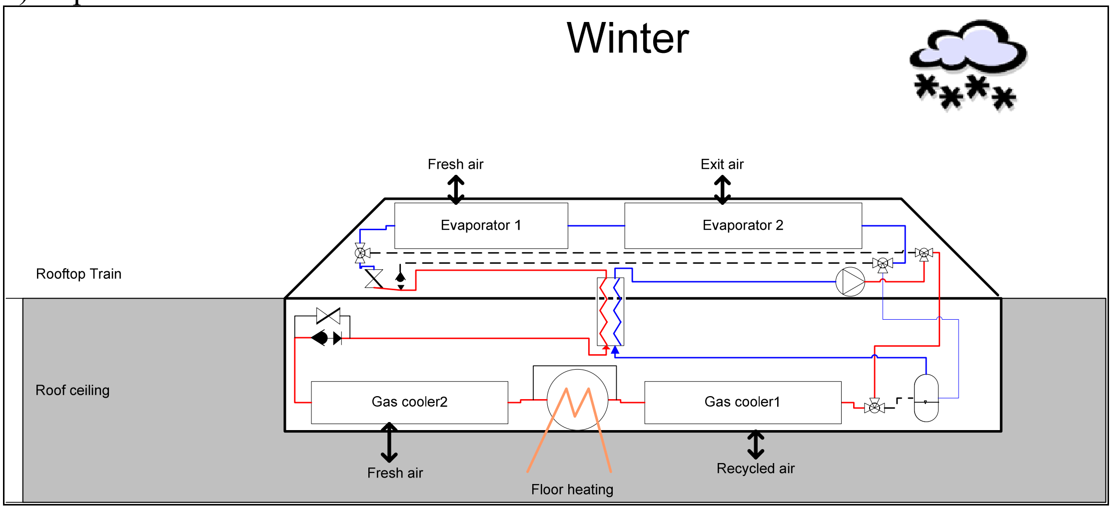


Figure B. 5 System circuit of solution I heating mode

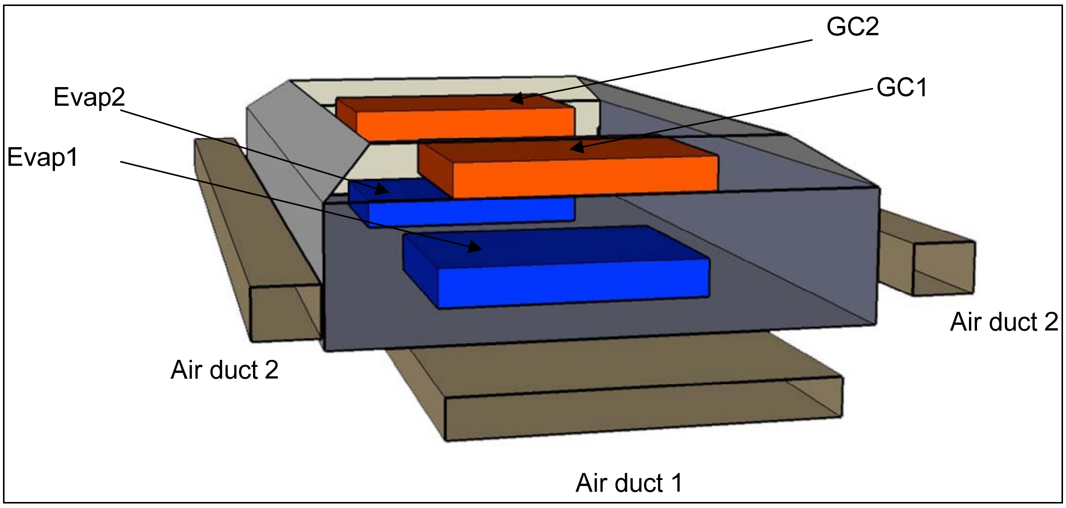


Figure B. 6 Airflow through the heat exchangers in solution I

As Figures of Solution I show the heat pump is reversible and the heat exchangers have to be able to work both as gas cooler and evaporator. Three way valves are used to control the direction of the refrigerant. To provide that the air streams are on the same place on the unit in both heating and cooling the reversion of refrigerant must be done in the way the figures shows. In the summer (Figure B. 4) it looks like it is easier to let the refrigerant flow from the compressor to the closest heat exchanger, but then the air streams on the top of the unit also have to change when the system change from heating to cooling. In the winter (Figure B. 5) the internal heat exchanger is not counter current. However, in heating mode it is not that important with an efficient IHX. The aim of the IHX is, as explained in chapter A.2.3, to reduce the temperature before the expansion valve which again leads to a reduced throttling loss. In the summer the CO_2 temperature out of GC2 is high because there is no cooling source with low temperature. GC2 exchange heat with exhaust air at ca 25 °C and in this case an efficient IHX is important. In the winter there is a cooling source with low temperature, the fresh air flow which is going to enter the coach. The temperature out of GC2 is than much lover than in the summer and an efficient IHX is not that important since the throttling loss already is reduced.

Figure B. 6 is a principal sketch showing the location of the heat exchanger in cooling mode and the air duct system in the rooftop ceiling as in Figure B. 1. Air duct 2 contains exhaust air and brings the exhaust air to the two exhaust air intake in the rooftop unit as showed in Figure B. 3. There have to be a regulation system which divide the exhaust air to correct volume flow to the exit and recycled air flow. The fresh air enters the system in the front of the unit. The treated air flows from the two bottom heat exchangers to air duct 1 and is distributed around in the railway coach. Figure B. 6 is similar for the heating mode regarding air duct system and air streams, but the gas coolers will be at the bottom of the unit and the evaporators at the top of the unit.

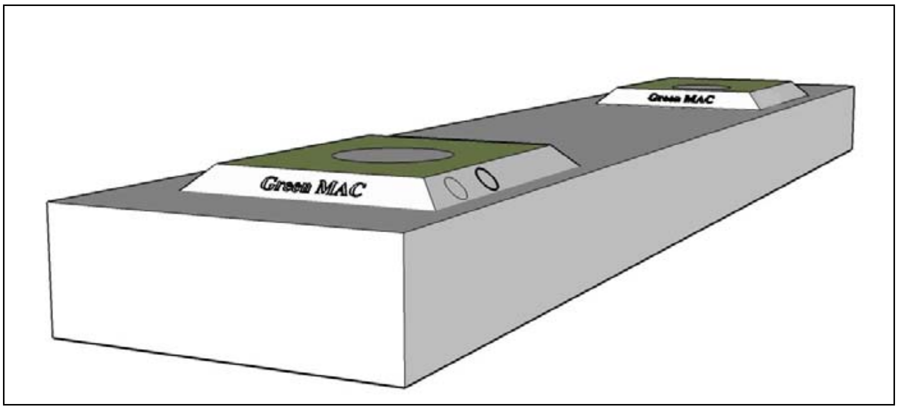


Figure B. 7 Principal sketch of Solution I with two rooftop units.

Since solution I have the two fresh air heat exchangers in the front of the unit and the two exhaust air heat exchanges in the back of the unit it is also possible to split the heat exchangers into two units. The front unit deals with fresh air and rear unit exhaust air. With two rooftop units the size of the heat exchanges can increase. The volume flow of the air streams is fixed by standard mentioned in chapter B.1.2. When the face area of the heat exchanger is small the air face velocity is high which will cause noise.

B.2.2 Solution II

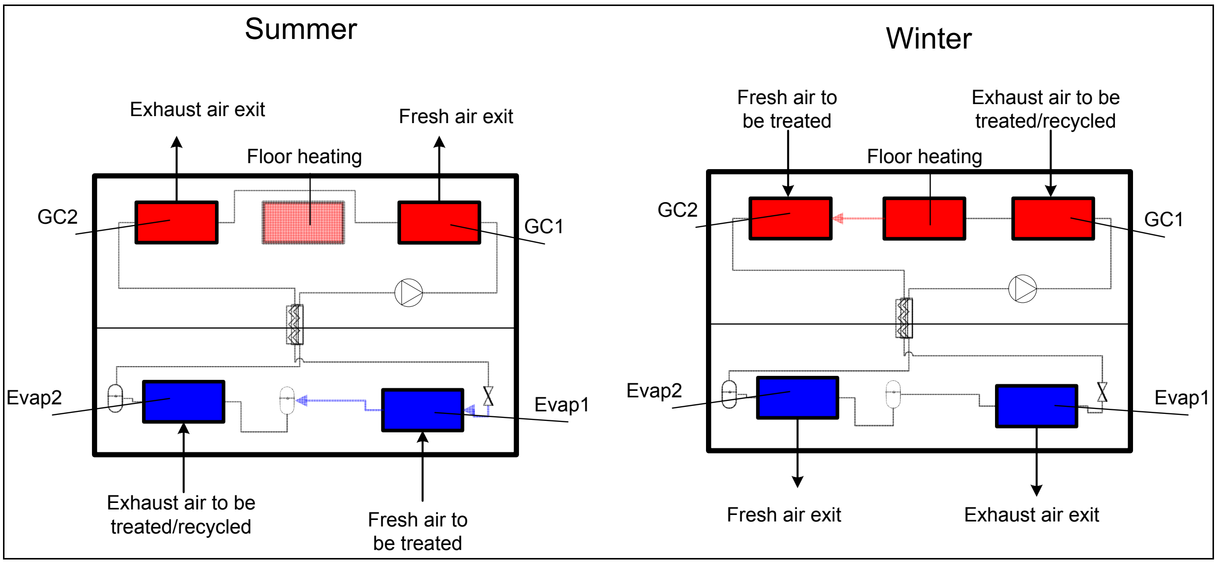


Figure B. 8 System circuit of Solution II. (Seen from above)

Solution II is a non reversible heat pump. As the figure show the circuit and location of the heat exchanger is similar summer and winter. The air flows changes and they change diagonal as the figure shows. This system requires a air duct and fan system that controls the airstreams and change them after summer and winter modus. The evaporators do not have to be dimensioned to stand high pressure side since they always operates as evaporators. The system circuit is simple and do not have any three way valves to control direction of CO_2 flow. Neither bypasses valves to control where the throttling will happens as in Solution I. This will reduce the chance of failure and leakages in the system circuit.

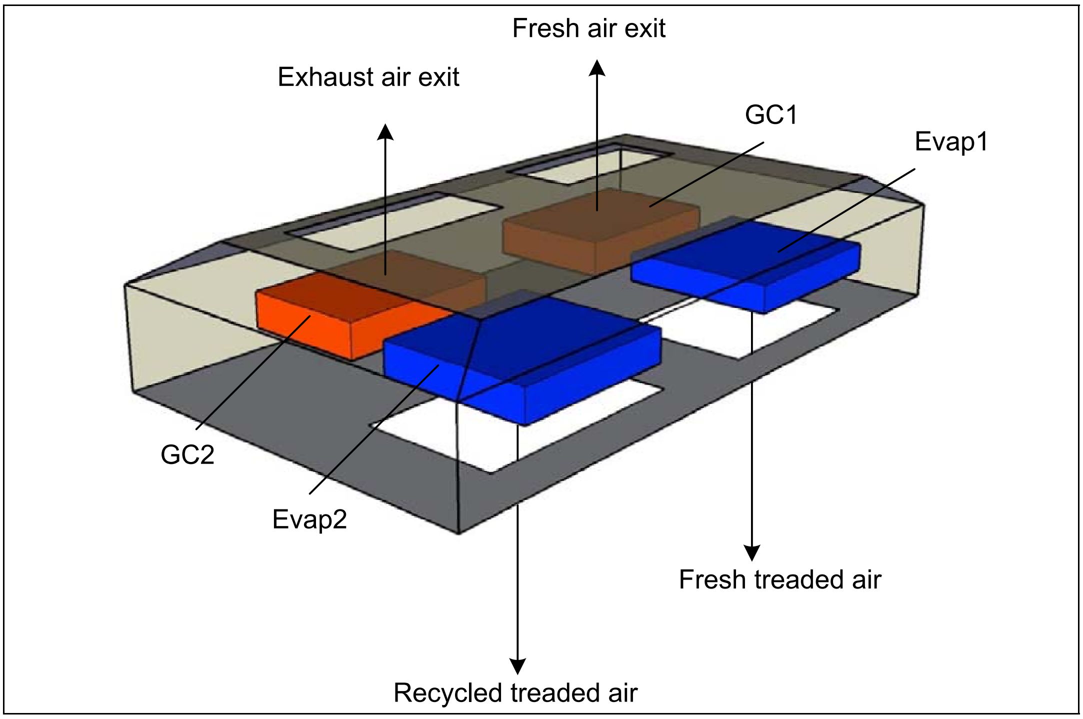


Figure B. 9 Principal sketch of rooftop unit, location of heat exchangers and air flows for Solution II cooling mode. The unit will also have a wall to separate hot and cold side.

The exhaust air which is going to be recycled flows from air duct 2 (Figure B. 1) and have to flow to the top face area of Evap2 and than through the heat exchanger and into the railway coach. Fresh treated air enters the fresh air intake and flows through Evap1 and into the railway coach. As the figure shows there is a hole under each evaporator to let the treated air enter the coach. On the gas cooler side there are two holes in the unit roof to let the air exit the system. Fresh air on the gas cooler side first has to flow down to meet the face area of GC1 bottoms side and than through the heat exchanger to exit the system. Exhaust air enters the unit and exchange heat with gc2 then exit the system.

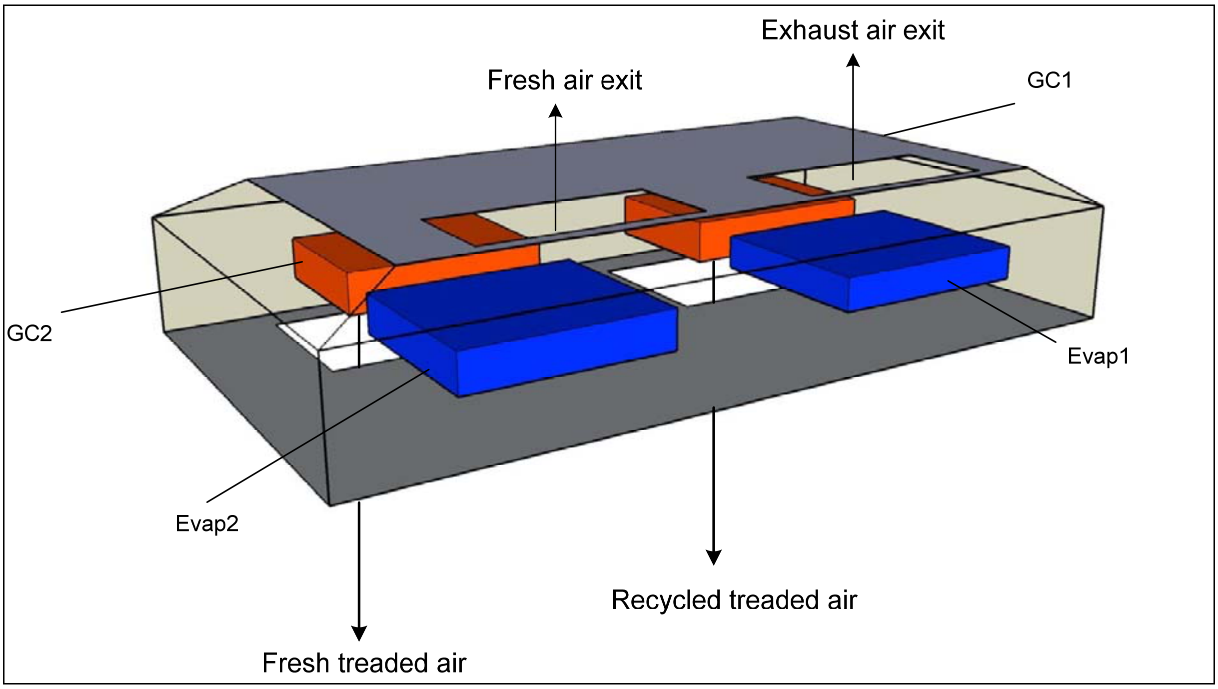


Figure B. 10 Principal sketch of rooftop unit, location of heat exchangers and air flows for Solution II heating mode.

As the figure shows the heat exchanger is the same as in cooling mode but all the air streams have changed diagonal. The holes in the floor have also moved from under the evaporators to under the gas cooler, since in heating it is the warm air that will enter the coach. The holes in the roof are moved from above the gas coolers to above the evaporator, since it is the cooled air which will exit the system. It is necessary with a system to close and open the holes in the rooftop unit.

B.2.3 Solution III

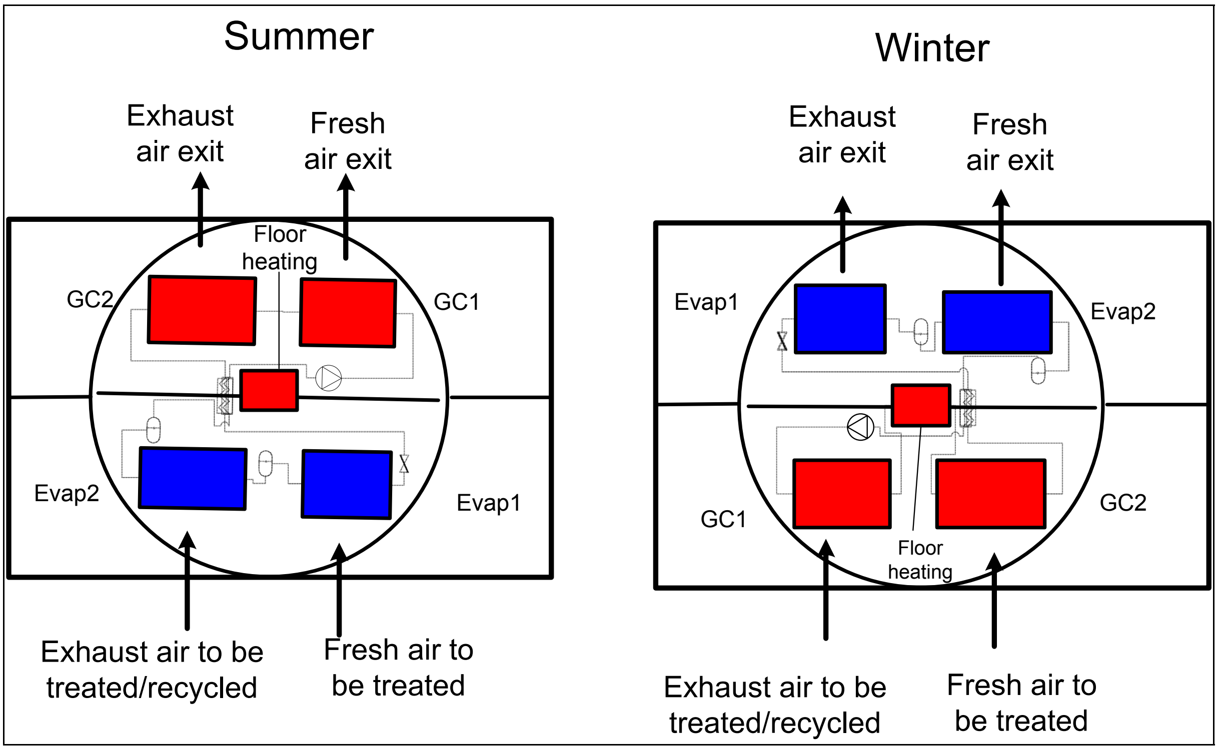


Figure B. 11 System circuit of Solution III.

In this solution the heat pump is located on a rotatable plate. In Solution II all the air streams changes diagonal when to mode change from cooling to heating. When the heat pump rotates 180° the heat exchangers position changes diagonally. Solution III have a circuit where the CO_2 flows in the same direction as in Solution II, and the air streams is similar in cooling and heating mode as the figure shows.

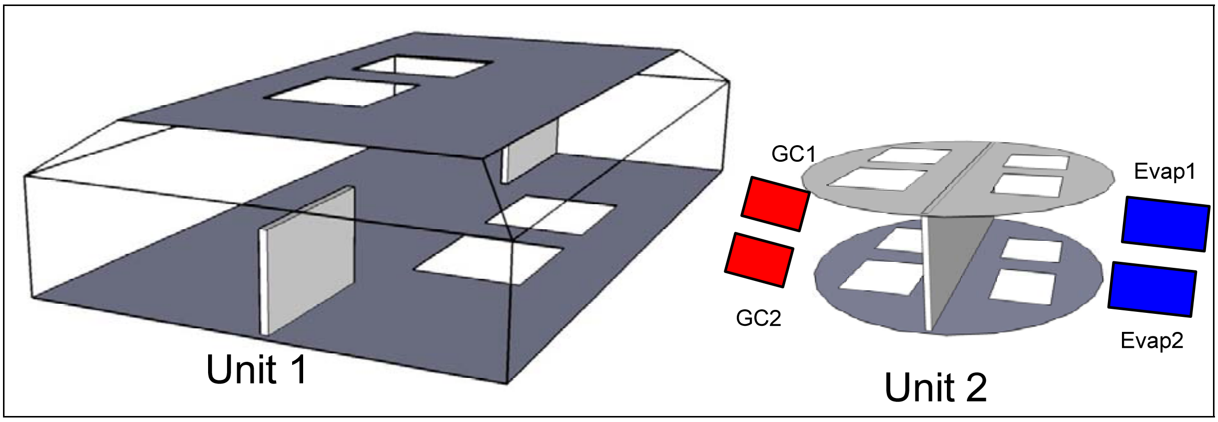


Figure B. 12 Principal sketch of solution III

The rooftop unit (in this explanation called unit1) is on the left of the figure and the rotatable unit (in this explanation called unit2) is to the right. The heat pump is located on unit2. The reason why unit2 is taken out unit1 is to simplify the explanation of solution III. Unit1 have a wall at each side and unit2 have a wall in the centre, together these walls separate the hot and cold side.

Unit1 have to holes in the floor on the right side. This is where the treated air leaves unit1 and is distributed around in the railway coach. Unit1 also have two holes in the roof on the left side. This is where exit air leaves the system. Unit2 have two holes in the floor in both right and left side, and two holes in the roof in both right and left side. In the summer the treated air leave the evaporator side and enters the treated air duct through the holes in unit2 and unit1. The holes fit together. The holes in the roof above the evaporators on unit2 right side are closed by the roof on unit1. In the same way the holes in the floor of unit2 left side is closed by the unit1 floor. In the summer exhaust and fresh air exchange heat with the gas cooler side flow through the holes in the roof of unit2 and unit1 and than exit the system. When the system changes from cooling to heating unit2 is rotated 180 °. Than the holes in the floor fits for the gas cooler side to enter the railway coach and the holes in the roof fits for the evaporator side to exit the system, while the holes in the floor on the evaporator side is closed and the holes in the roof on the gas cooler side is closed.

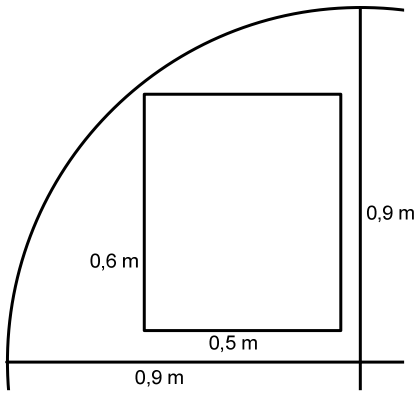


Figure B. 13 Dimension of the heat exchangers for Solution III

The heat exchangers in Solution III have to be small enough that the whole heat pump can be located on the rotatable unit. If not it is impossible to rotate the unit. If all the four air heat exchangers have the same size and the diameter of the rotatable unit is 1,8m (Table B. 1 and Figure B. 3) the dimension of the heat exchanger is maximum ca: Width 0,5 m, Length 0,6 m and Depth 0,4 m. This dimension is used as guide for the decision of the heat exchanger layout in Chapter C.2.1.

B.3 Discussion of system layout

Solution I has a complicated system circuit. All the three way valves and the bypass valves can result in a higher risk of leakage and fatigue failure. The positive with Solution I is that the air streams have the same configuration in heating- and cooling mode. It is also positive that the heat exchangers which exchange heat with air streams going to exit the system is located on the top of the unit and heat exchangers for the treated air streams are located at the bottom of the unit. The air streams flows in a simpler path than in Solution II and Solution III where some of the air streams first have to flow over/under the heat exchanger before it can flow through it. In heating mode the IHX is co current which will reduce the efficiency of the heat exchanger. A counter flow heat exchanger is more efficient than a co current because the temperature fit between hot and cold fluid is better. The outlet temperature of the cold fluid may exceed the outlet temperature of the hot fluid, which is not possible in a co current heat exchanger (Incropera and DeWitt, 2007, s 676). However, as explained in the presentation of Solution I it is in cooling mode an efficient IHX is most important. In Solution I the whole system has to be dimensioned for high the high pressure side which will increase the cost of the heat exchangers.

Solution I with to rooftop units may in total have greater heat transfer surface area because of the two units there is more space for the heat exchangers. The solutions with one rooftop units the location of it is in the middle of the coach. The treated air streams must be divided and distributed to the front and to the rear of the coach. With two rooftop units all the exhaust and treated air in the ducts flows from the units or towards the units in the whole length of the coach. Two rooftop unites requires a longer piping system and a higher CO_2 mass charge. The pipes between the units have to be isolated. In heating mode it is not wanted to loose heat before the CO_2 reaches the gas cooler. The cold pipeline also has to be isolated because the evaporation temperature is lower than the surrounding temperature. If the pipe is not isolated the fluid may start to condense on the way to the receiver which results in a lover vapour fraction in the receiver inlet.

Solution II and III have simple system circuits and do not have to reverse the refrigerant flow. Since it is not reversible the solutions have one high pressure side and one low pressure side. In Solution II all the components in the heat pump are fixed, which is good for the safety while solution III is located on a rotatable unit.

The air distribution system is more complex for Solution II than III. Solution II needs four outlets in the rooftop unit with belonging air fans. It is also necessary with a devise that block the outlets not in use and opens the outlets in use. Solution III only needs one outlet for the treated air and one outlet for the exit air. There is no need for an outlet closing and opening devise like in solution II.

The Computer simulation in next chapter calculates on a non reversible heat pump and the dimensions of the heat exchanger fits to solution III. The final system circuit is modified to have two levels of evaporation pressure and a two stage compression. The system circuit of solution I will in this case be even more complex. Solution II and III seems like the best option.

C COMPUTER SIMULATION

The computer simulation is done by the programmes RnLib, HXsim and ProII.

- RnLib is a programme developed by NTNU-SINTEF. It calculates thermodynamic data and transport properties of refrigerants and refrigerant mixtures. The programme is installed and used together with Microsoft Excel.
- HXsim is a simulation programme developed by SINTEF. It is for dimensioning and calculations of CO_2 heat exchangers where air is the cooling/heating source.
- PROII/PROVISION is a simulation programmes which simulates a whole system circuits with heat exchangers, compressors, receivers, expansion valves and so on.

In the description of this report the simulation should be done applying Csim, which also is a SINTEF developed simulation programme. It is not possible to have more than one evaporator in Csim, therefore the programmes above is used in stead.

C.1 Cooling mode

C.1.1 System description

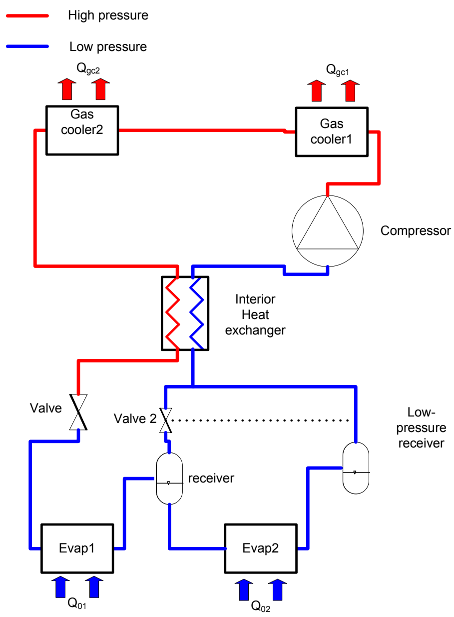


Figure C. 1 System circuit for the simulation in cooling mode.

The reason of the receiver between the two evaporators is to send saturated liquid into Evap2. It is problematic to have two evaporators in series without a receiver between them, because the vapour fraction in Evap2 inlet will be high which make the distribution of the CO_2 into Evap2 problematic. Some pipes may be empty and others full. Because of the receiver the mass flow in Evap1 and Evap2 is not similar.

$$\dot{m}\_evap2 = \dot{m}\_evap1 * vapour\_fraction \quad \text{[Eq 5]}$$

Valve 2 is regulated such as the pressure loss over the valve is similar to pressure loss in Evap2. This is to secure that the pressure in the connection between stream out of valve2 and out of low pressure receiver is similar. The efficiency of the compressor is given by the pressure ratio in the compressor. The compressor efficiency is assumed to be similar to Hrnjak (2006) efficiency curve of compressors (Appendix B, Appendix Figure 7).

As the figure above shows the simulation is done without two evaporation pressures and without two stage compression as in heating mode. If this is implemented in the model also for cooling the simulation results will improve. Than the temperature difference between fresh (hot) air and evaporation temperature of evap1 can be optimized. Discharge temperature out of second stage compression will also be decreased compared to one stage compression and less energy to cool down the CO_2 is needed.

C.1.2 Simulation using Excel and RnLib

It has been made a calculation model of the CO_2 cycle using Excel and RnLib. Assumptions and method of this model is explained in Appendix A some key results of the calculation model are presented.

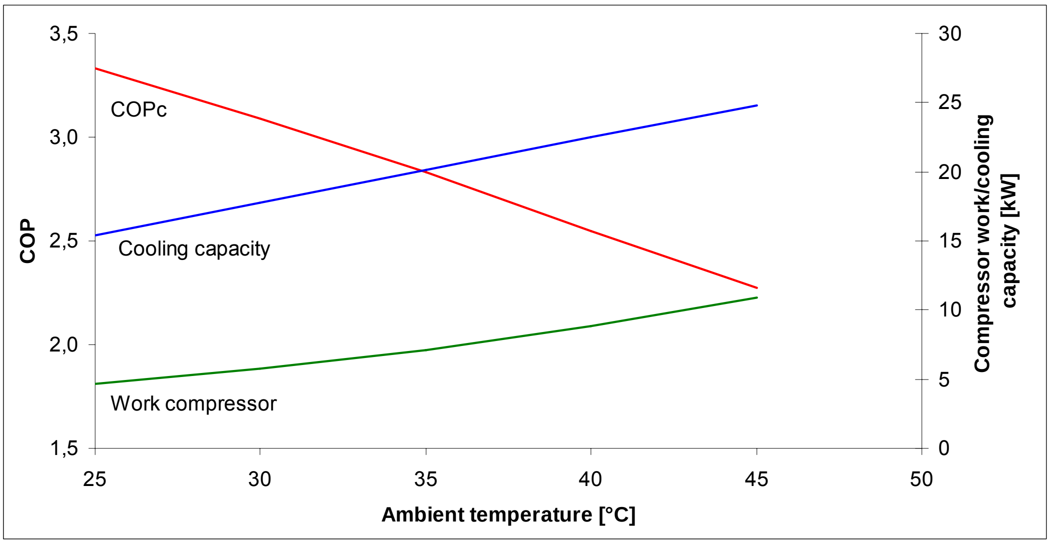


Figure C. 2 Cooling capacity, COP_c and compressor work due to ambient temperatures in cooling mode.

In cooling mode the ratio between fresh and recycled air is constant at all temperatures. The specific cooling capacity (Cp) for air is also assumed constant. This is why the cooling capacity increases linear to the ambient temperature. The reason why the compressor work ad COP is not totally linear is the efficiency of the compressor. The pressure ratio increases with increased ambient temperature. Increased pressure ratio leads to decreased efficiency of the compressor decrease.

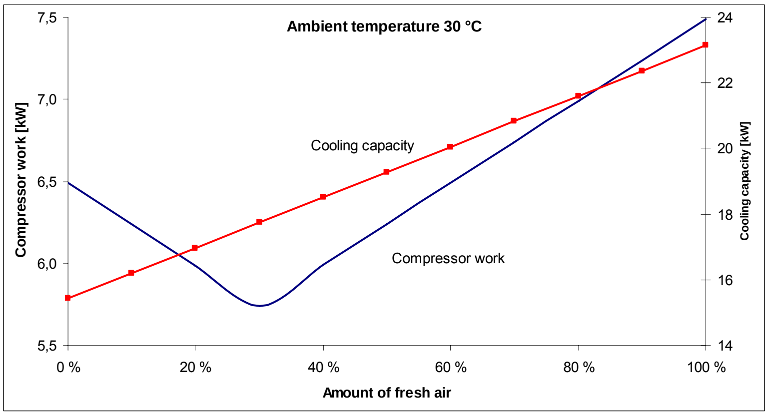


Figure C. 3 Compressor work influenced by the amount of fresh air at 30 °C.

This figure illustrates the importance of a having the CO_2 temperature before throttling as low as possible and to recover energy from the exhaust air. The compressor work is not at minimum when the amount of fresh air is zero. This is the operation conditions which needs the less cooling capacity, since only exhaust air at 25 °C have to be cooled. When no fresh air enters the coach no exhaust air can exit. GC2 is unused and the ambient air is the only cooling source. When the amount of fresh air increase, exhaust air also has to leave the system and GC2 get a colder cooling source than the ambient temperature. The minimum compressor work occurs when the volume flow of exhaust air which exit the system have enough cooling capacity to cool down the CO_2 outlet temperature of GC2 to its minimum (T_gc2_out_min = T_exhaust_air + 2 °C). This exhaust air volume flow occurs when the amount of fresh air is around 30 %.

The optimal amount of fresh air is influenced by the ambient temperature. When ambient temperature decrease the CO_2 inlet temperature of GC2 also decrease, since temperature of the cooling source for GC1 is reduced. This effect reduces the optimal amount of fresh air. However, the temperature difference between exhaust air and fresh air decreases the amount of fresh air does not have that great impact on the compressor work. When ambient temperature is 25 °C the cooling capacity and the compressor work will be constant at all amount of fresh air.

When ambient temperature increase the CO_2 inlet temperature of GC2 also increases. The volume flow of exhaust air than have to increase to keep the CO_2 outlet temperature of GC2 at its minimum. This effect increases the optimal amount of fresh air. The gradient of the cooling capacity will be steeper when the ambient temperature increases, because the temperature difference between recycled and fresh air have increased. This effect will reduce the optimal amount of fresh air. The effect of reduced throttling loss of a low GC2 outlet temperature may have less influence on the compressor work than the increased cooling capacity needed when the amount of fresh air increase. It is not sure that optimal amount of fresh air occur at the point where exhaust air have enough capacity to cool down CO_2 temperature in GC2 to minimum temperature. Simulations of different ambient temperatures are necessary to find out if the optimal amount of fresh air increases of decreases with higher ambient temperatures.

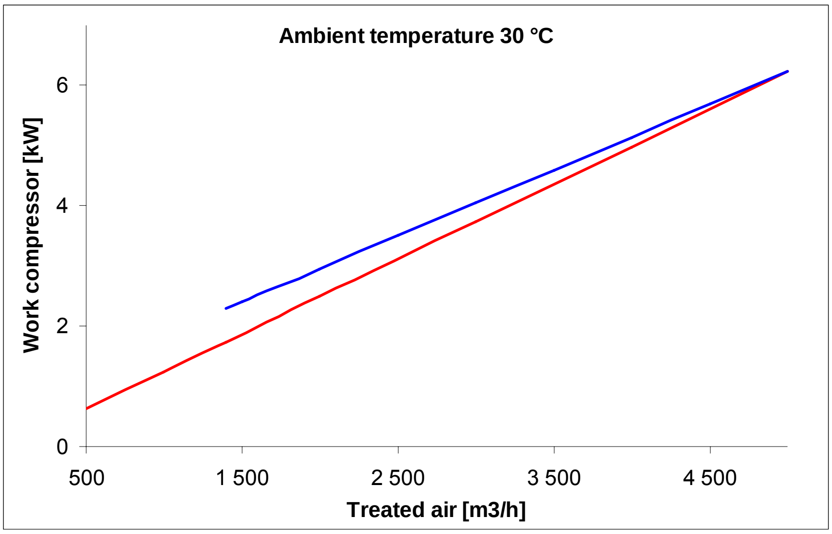


Figure C. 4 Compressor work influenced by the amount of treated air.

The blue curve always keep the standard of 1400 m^3/h fresh air (Table B. 2) while the red curve always have 30 % amount of fresh air the amount which gives minimal compressor work at ambient temperature 30 °C. Today the treated air stream always is 4600 m^3/h, because it is assumed that every seat is occupied. A regulation system which regulates the amount of treated air to the actual number of passengers will save energy. When the coach is half full, 35 seats occupied, 700 m^3/h of fresh air is needed to cover the requirement of 20 m^3/h of fresh per seat. This gives 2300 m^3/h of treated air if the amount of fresh air is 30 %. The compressor energy decreases from 5,52 kW to 2,76 kW. This shows that the potential of savings with passenger regulator. The amount of passengers vary widely due to time of the day, in the morning and evening many travel to and from work while in the middle of the day few people travel.

C.2 Simulation using HXsim and ProII

HXsim is a heat exchanger simulation programme. It simulates only air cooled/heated heat exchangers and only one heat exchanger at the time. In the simulated system there are four CO_2/air heat exchangers. ProII is used to simulate all other components than the four CO_2/air heat exchangers. To find cycle data four HXsim programmes must be opened and the ProII model must be opened at the same time and it is necessary to iterate between the programmes to find the correct value. In cooling mode the volume flow and temperatures of the air is known. Evap1 meet ambient air while Evap2 meet recycled air at 25 °C. The evaporators are dimensioned such as treated air is around 15 °C. In cooling mode it is the evaporators that decide the mass flow of CO_2. However, it is not possible to fix the mass flow in the simulation. The mass flow is an output value of the simulation as a result of input values. Input values of the simulation are air temperature, relative humidity and volume flow of air, evaporation temperature and inlet evaporator conditions (the temperature and pressure before the expansion valve is the input to find inlet evaporator conditions). For gas cooler pressure the optimal pressure found in the Excel/RnLib model is used. The values found in the HXsim/ProII model are documented in a excel sheet (file name: hxsim_pro2_excel_sheet.xls. The excel sheet also contains procedures to be used in the different iteration processes to give a good value of "next guess"). A more detailed explanation of method of simulation of all the iterations between the four HXsim programmes and ProII programme is given in Appendix B.

C.2.1 Deciding heat exchanger layout in HXsim

Design of heat exchanges must be done before the total cycle can be simulated. When different design of heat exchangers are tried out it is important that the exchanger must be able to full fill the need of heating the coldest day and the need of cooling the warmest day. The size of the heat exchanger can not exceed the limits 0,6 m wide, 0,6 m ling and 0,4 m deep as found in chapter B.2.3. To find the best solution of the design different ambient temperatures are used and with different humilities. The final design is given in the table below.

Table C. 1 Design values of the four air heat exchangers

	EVAP1	Evap2	GC1	GC2
Fin data				
Vertical tube picth	0,06	0,038	0,03	0,038
Fin density [FPI]	6	2	6	10
Fin thickness [mm]	0,75	0,75	0,75	0,75
Heat exchanger design				
Pipe diameter	0,009	0,007	0,007	0,007
duplications	5	8	10	10
vertical tubes	2	2	2	2
horizontal tubes	4	10	10	10
vertical tube pich	0,06	0,038	0,03	0,03
horizontal tube pich	0,09	0,04	0,04	0,04
depth	0,04	0,4	0,4	0,4
height	0,6	0,608	0,6	0,6
core width	0,42	0,42	0,42	0,42
spece between	0,01	0,01	0,04	0,04
length	0,36	0,38	0,39	0,39

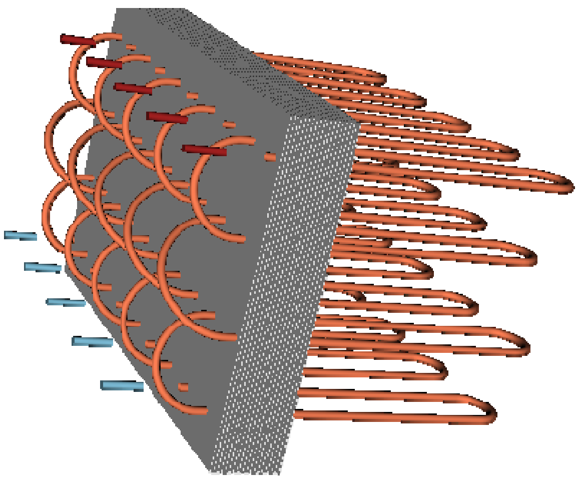


Figure C. 5 Illustration of Evaporator 1.

It is possible while testing different design in HXsim to at all time get a visualization of the heat exchanger rotate the heat exchanger to get a view and understanding of it. The figure above is taken from HXsim. The understanding of the figure and main design values from the table above is the following: 5 duplications means that the CO_2 is distributed to five pipes before it enters the heat exchangers. 2 vertical tubes, each pipe take a tour retour trip before it goes one level down. 4 horizontal tubes, the heat exchanger has 4 levels from the hot to the cold outlet.

C.2.2 ProII model cooling mode

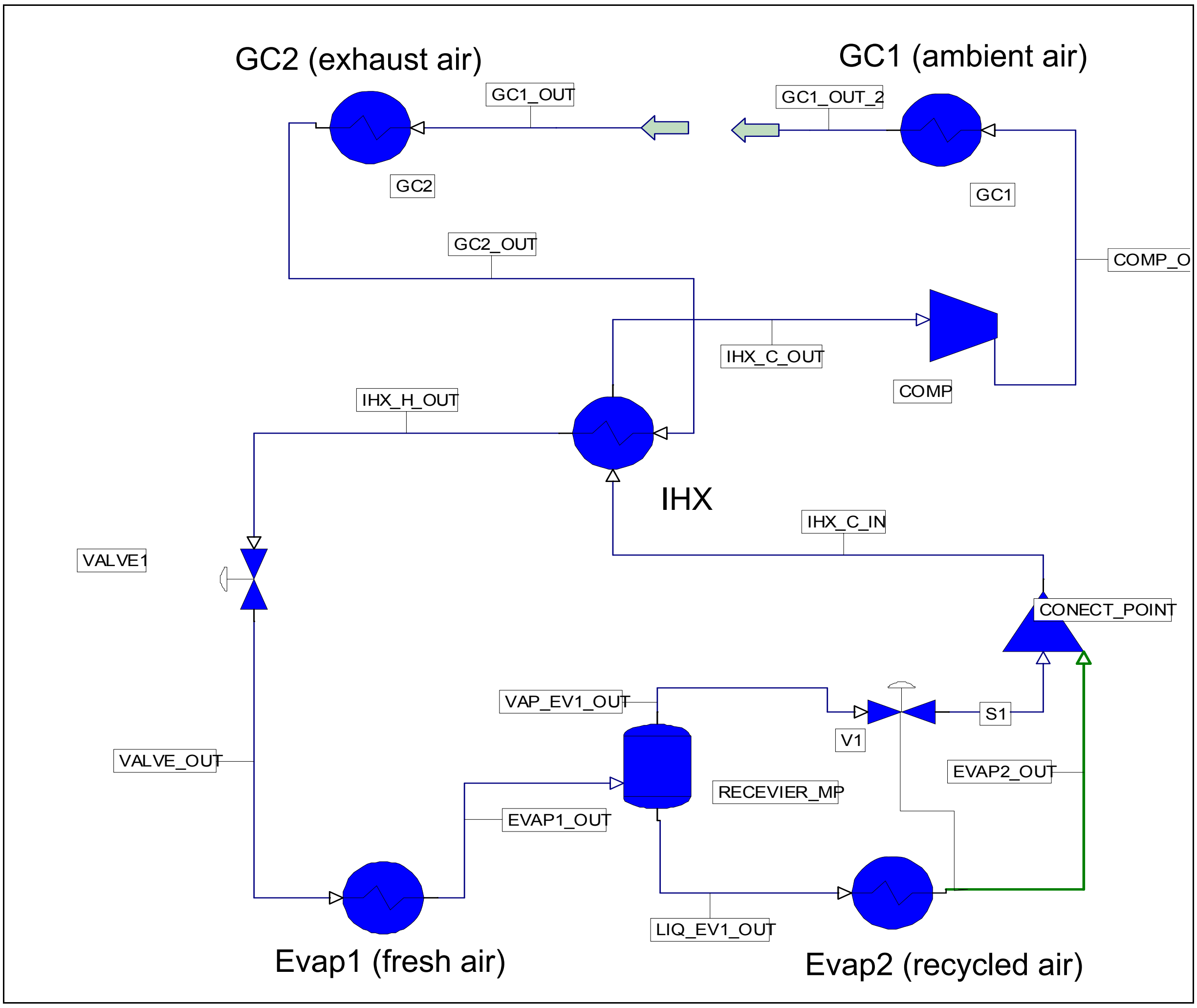


Figure C. 6 System drawing of proII cooling model

In ProII it is not possible to close a circuit. The staring point of the simulation is selected to be GC2 inlet. This is because output of temperature of GC1 can be controlled by the fresh air. The fresh air in GC1do not enters the system and the mass flow can be set such as the CO_2 outlet temperature out of GC1 reach its wanted temperature. Optimal high pressure found in the Excel/RnLib model (

Appendix A, Appendix Figure 5) is used in HXsim/ProII model. The information about the heat exchangers is taken from the simulation in HXsim and inserted in the ProII model. As mentioned above it is necessary to iterate between the HXsim files and the ProII model. If conditions in the GC1 out is not the same as the starting values of the cycle a new iteration must be done. The IHX can not be simulated with HXsim. In ProII the IHX is assumed to have a minimum pinch of 5 °C. Outlet temperature and pressure of the hot side of the IHX is used to define the inlet conditions of Evap1 in HXsim together with the evaporation pressure. In the first simulation in the IHX_h temperature must be guessed. After calculating the hole cycle the guessed temperature must be controlled in ProII and a new simulation process must be done if the temperature is wrong. The middle pressure receiver, the efficiency of the compressor and the expansion valve before the connection point is explained in Figure C. 1.

C.2.3 Presentation and discussion of HXsim/ProII results cooling model

Table C. 2. Results cooling mode relative humility of air 0 %

Ambient temperature °C		**25**	**30**	**35**	**40**	**45**
Total cooling capacity	[kW]	11,8	13,4	14,5	15,7	16,9
Work compressor	[kW]	2,6	3,3	4,0	5,0	6,1
COP	[-]	4,5	4,1	3,6	3,2	2,8

Table C. 3 Results cooling mode relative humility of air 30 %

Ambient temperature °C		**25**	**30**	**35**	**40**	**45**
Total cooling capacity	[kW]	11,7	14,4	17,5	21,0	24,6
Work compressor	[kW]	2,5	3,6	5,3	7,8	10,6
COP	[-]	4,6	4,0	3,3	2,7	2,3

Table C. 4 Results cooling mode relative humility of air 60 %

Ambient temperature °C		**25**	**30**	**35**	**40**	**45**
Total cooling capacity	[kW]	18,8	23,5	28,1	34,2	39,9
Work compressor	[kW]	4,6	7,2	10,4	16,4	21,0
COP	[-]	4,1	3,3	2,7	2,1	1,9

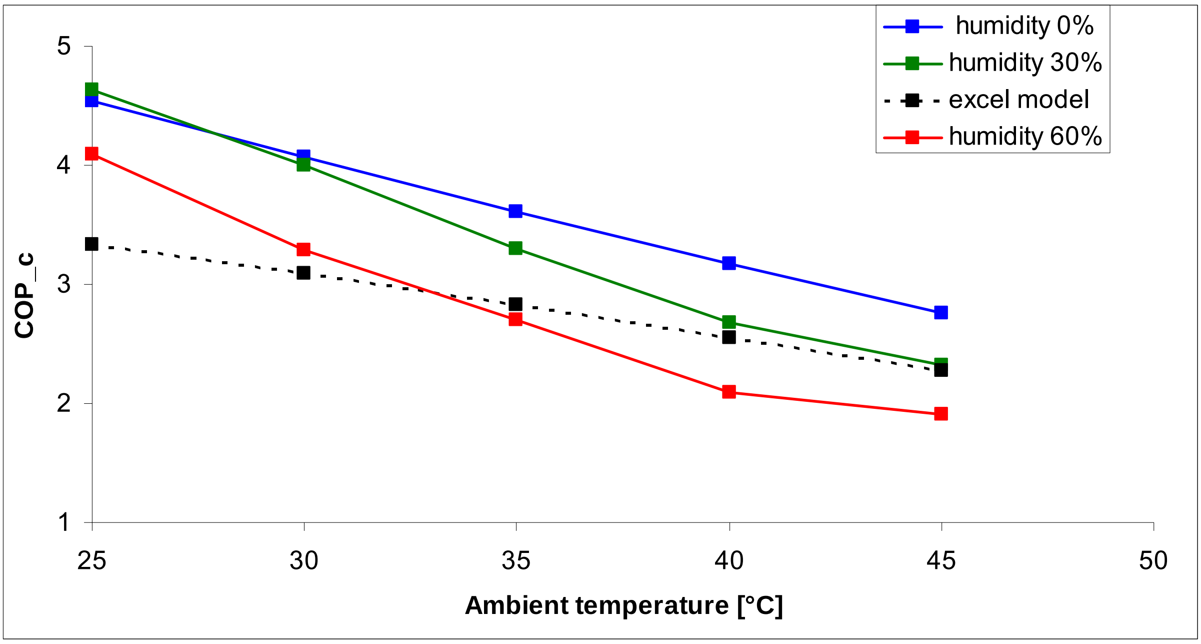


Figure C. 7 COP_c due to ambient temperature. Comparison of calculations in the Excel model and the HXsim /ProII model for three different air humidity levels

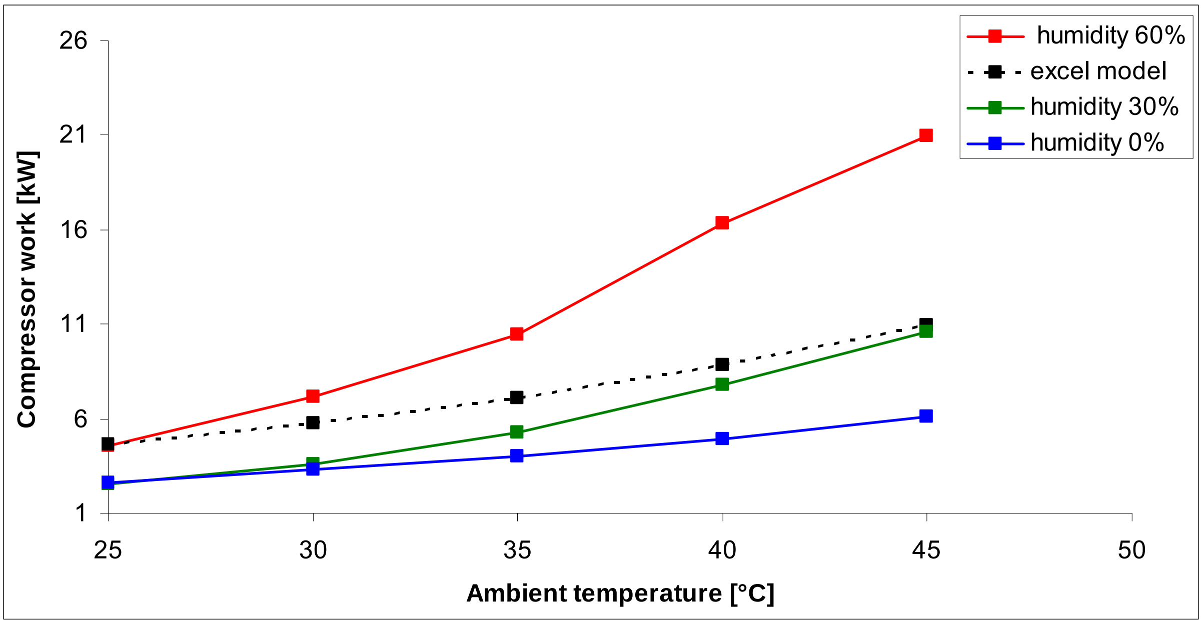


Figure C. 8 Compressor work due to ambient temperature. Comparison of calculations in the Excel model and the HXsim /ProII model for three different air humidity levels

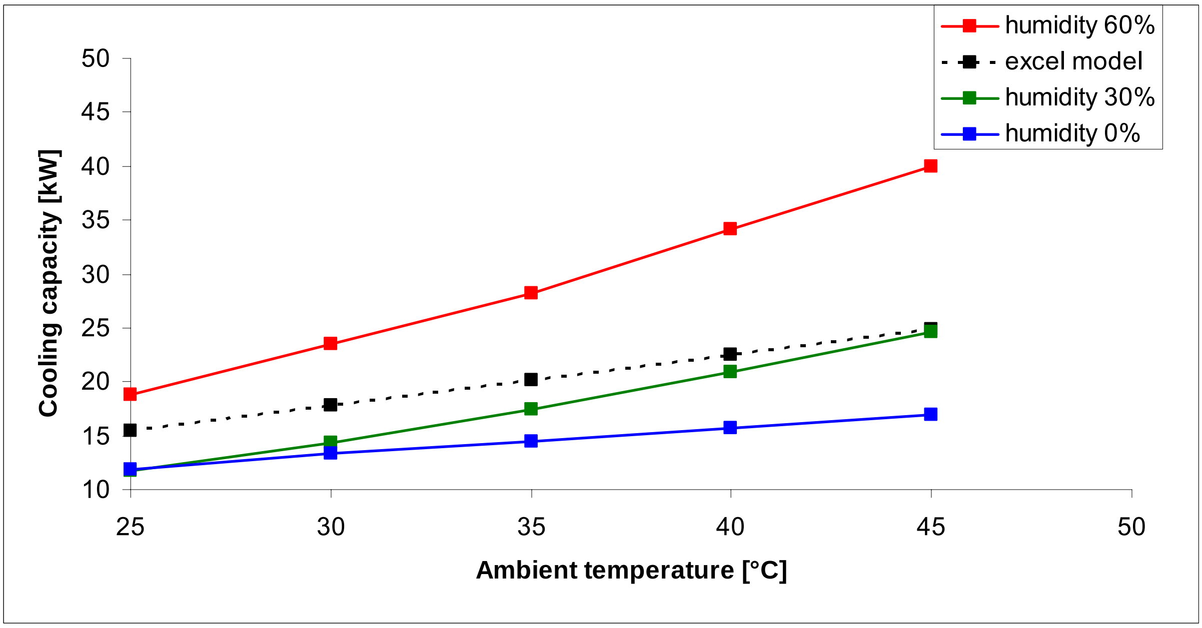


Figure C. 9 Cooling capacity due to ambient temperature. Comparison of calculations in the Excel model and the HXsim /ProII model for three different air humidity levels

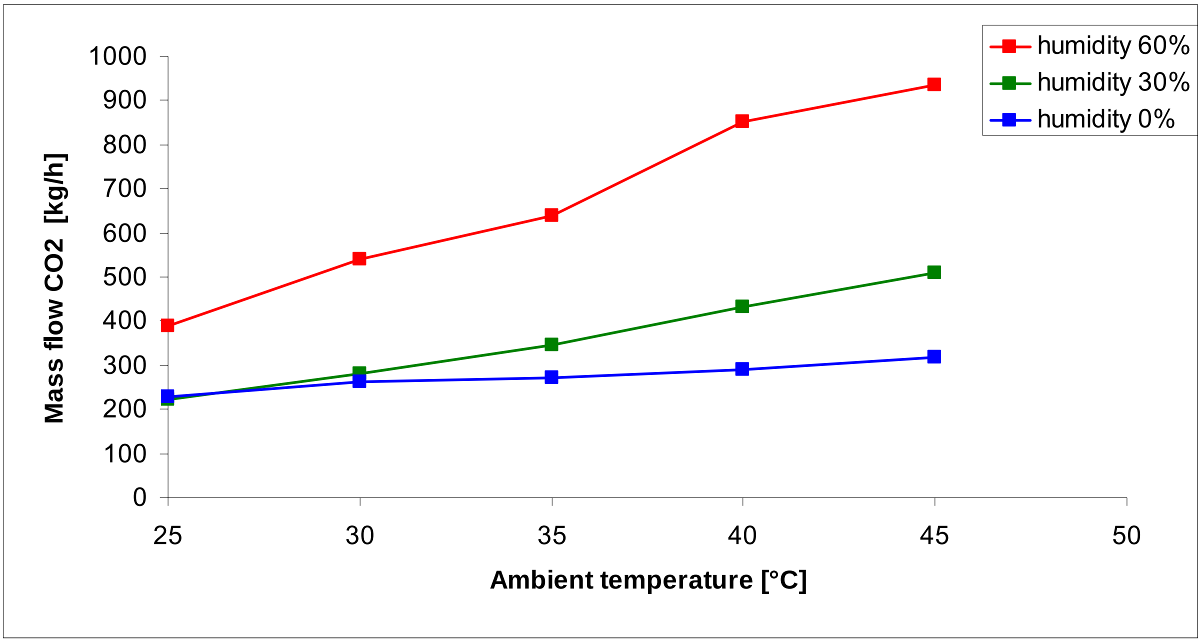


Figure C. 10 Mass flow of CO_2 due to ambient temperature. Comparison of calculations in the Excel model and the HXsim /ProII model for three different air humidity levels

The COP of the cycle decrease as the ambient temperature increase, this is because more cooling is needed to keep treated air at 15 °C. The evaporation temperature is constant and a higher CO_2 mass flow is required to increase the cooling capacity. The optimal high pressure also increases. This is because the CO_2 temperature before throttling gets higher and a higher pressure keep the cycle to the left in the pressure/enthalpy diagram (chapter A.2.4).

The figures also show that the relative humidity has influence on the CO_2 cycle. Increasing humidity has a negative effect of the cycle due to higher compressor work and decreased COP. Values for the simulation series of ambient temperature 40 °C and 60 % relative humidity is used to explain the negative effect. The fresh air evaporator cools air at 40 °C and 60 % relative humidity to 18 °C and 94% relative humidity. The recycled air evaporator cools air from 25 °C and 60% relative humidity to 15 °C and 90 %. The relative humidity increases throughout the evaporators and the specific heat capacity of the air increases. The required cooling increases for each temperature step towards the outlet temperature.

For the gas cooler the air temperature increases throughout the heat exchanger and the relative humidity decreases. Fresh air in gas cooler 1 is heated from 40 °C and 60 % relative humidity to 73 °C and 17% relative humidity. The exhaust air in Gas cooler 2 is heated from 25 °C and 60% relative humidity to 48 °C and 17 % relative humidity. The specific heat capacity of the air decreases throughout the gas cooler. The two effects on the relative humidity of air in the evaporators and the gas coolers both have a negative effect on the COP.

C.3 Heating mode

C.3.1 Simulation using Excel and RnLib

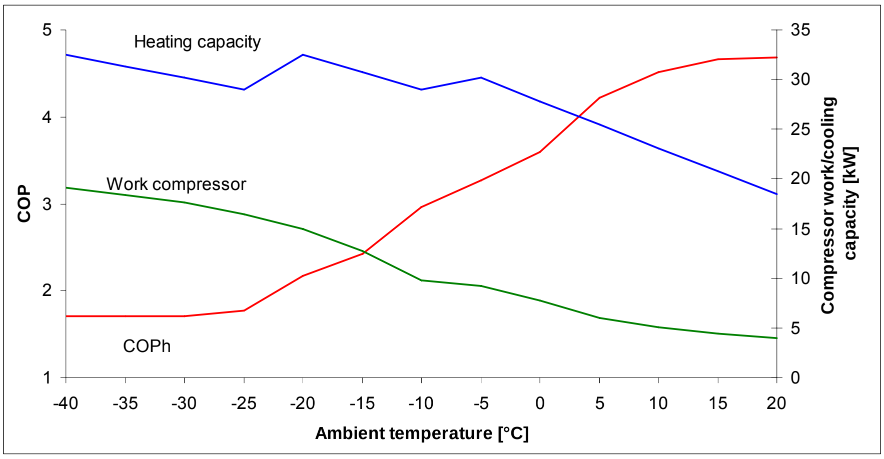


Figure C. 11 Heating capacity, COP_h and compressor work due to ambient temperatures in heating mode.

Based on the simulation in the Excel/RnLib model the system solution is modified. The reason of this is the problematic cold ambient temperatures. As the figure above shows the COP for temperatures below -25 °C is low.

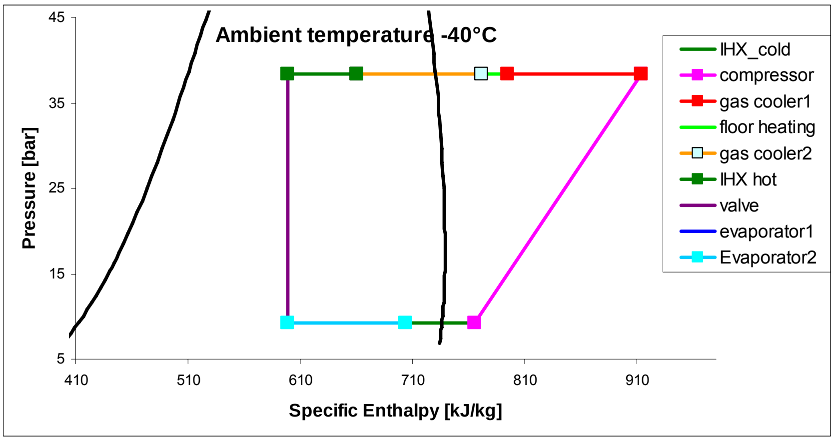


Figure C. 12 Pressure Enthalpy diagram of the CO_2 cycle at -40 °C Excel/RnLib model.

The figure shows that the cycle operates badly. The evaporation temperature is low and the high pressure side is controlled by the discharge temperature which should be kept below 160 °C. A higher temperature will damage the oil in the system. The specific evaporation temperature of the evaporators is low and a high mass flow of CO_2 is needed. This is because the CO_2 is throttled before it is on the saturated liquid line. Based on the knowledge of the Excel/RnLib model a new system circuit was made.

C.3.2 New system circuit

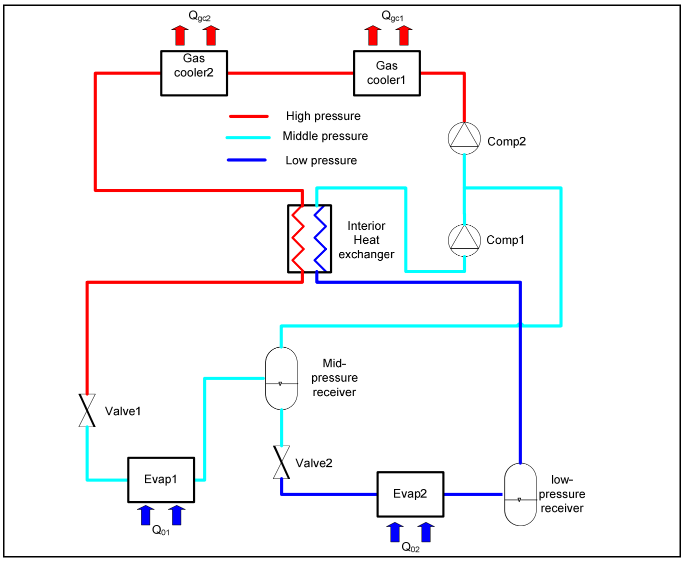


Figure C. 13 New design of CO_2 circuit.

The new design has two evaporator pressures. With this design the evap1 pressure is not decided directly by the ambient temperature. Since Evap1 exchange heat with exhaust air at 25 °C the evaporation temperature do not have to be as cold as the evap2 temperature which exchanging heat with cold ambient air. In the simulation different pressures for Evap1 is simulated to find the optimal pressure. Valve 2 set the evaporation pressure of Evap2 and the corresponding evaporation temperature to 15 °C lower than the ambient temperature. The two stage compression may be done in a two stage compressor or in two separated compressors.

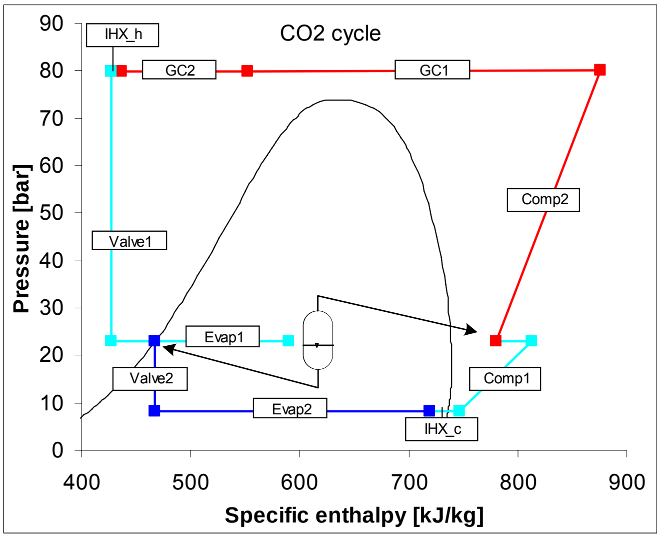


Figure C. 14 Pressure/enthalpy diagram of the new system scheme.

This plot is from the HXsim /ProII result for ambient temperature at -30 °C and the relative humidity of air 0%. As the figure show the exhaust air make Evap1 evaporate to 45 % vapour fraction. Saturated liquid from the receiver is throttled to into Evap2. The saturated vapour from the receiver is entering the second stage compression and cools down the discharge temperature of the first compression stage.

C.3.3 Simulation using HXsim and ProII

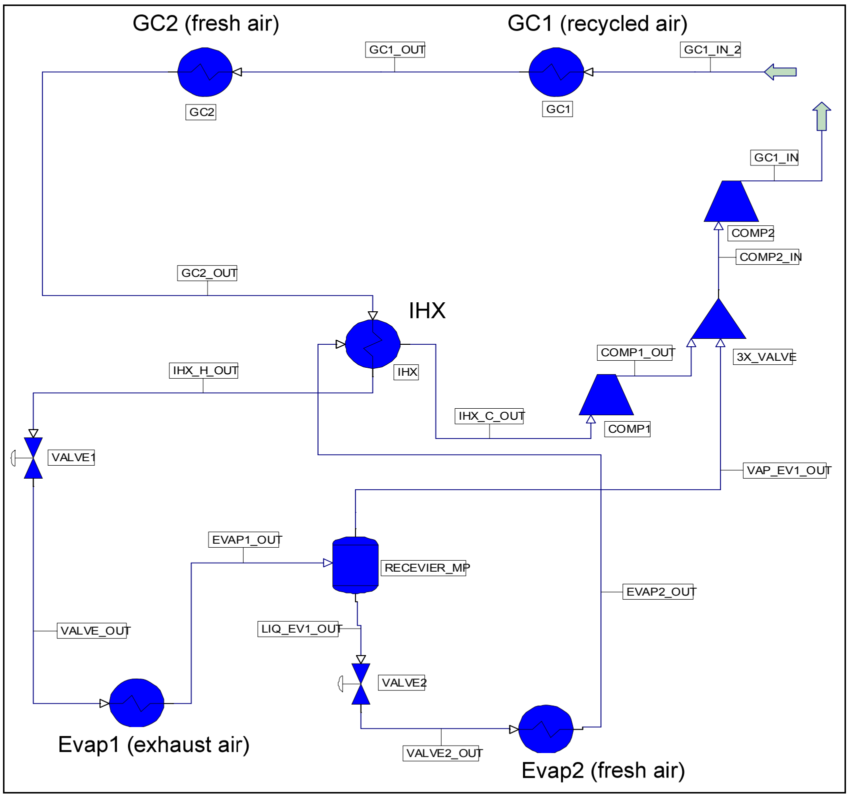


Figure C. 15 ProII circuit of heating mode.

In heating mode as in cooling mode the simulation in the HXsim/ProII model need many iteration processes between the four HXsim programmes and the ProII model. Optimal evaporation temperature of Evap1 is found by doing a whole simulation process with different Evap1 temperatures. Explanation of the simulation method and results of the Evap1 optimal temperature research is found in Appendix B. As the figure above show the cycle is opened between GC1 and the second stage compression. Iteration process must go on until the input values of GC1 are similar as the simulated output values of the second stage compression. As in cooling mode the compressor efficiency is assumed to be similar to Hrnjak (2006) efficiency curve of compressors (Appendix B, Appendix Figure 7) and the IHX is calculated by the requirement of minimum pinch temperature 5 °C. In the HXsim/ProII model the floor heating is excluded. It is not possible to simulate a CO_2/glycol heat exchanger in HXsim, but it is still possible to include the floor heating in the ProII model, but it will requires some more iterations for the GC2 inlet conditions. The floor heating in trains do not a main source for air heating. The maximum temperature in the floor is 27 °C according to NS- EN 13129-1 (2002). The positive having floor heating is to dry up the floor for snow and water. It is also comfortable for the passengers, since it is usual to be cold on the feet in the winter.

C.3.4 Presentation and discussion of HXsim/ProII results heating model

Table C. 5 Heating relative humility of air 0 %

Ambient temperature °C		**15**	**10**	**0**	**-10**	**-20**	**-30**	**-40**
Total heating capacity	[kW]	12,5	11,9	14,2	14,1	18,3	14,0	13,8
Work compressor	[kW]	2,1	2,2	3,2	3,4	5,1	4,3	4,3
COP	[-]	5,8	5,3	4,5	4,1	3,6	3,3	3,2

Table C. 6 Heating relative humility of air 30 %

Ambient temperature °C		**15**	**10**	**0**	**-10**	**-20**	**-30**	**-40**
Total heating capacity	[kW]	12,5	13,6	15,5	16,7	17,7	15,7	15,3
Work compressor	[kW]	2,1	2,6	3,5	4,2	4,8	4,7	4,5
COP	[-]	5,9	5,3	4,5	4,0	3,7	3,3	3,4

Table C. 7 Heating relative humility of air 60%

Ambient temperature °C		**15**	**10**	**0**	**-10**	**-20**	**-30**	**-40**
Total heating capacity	[kW]	12,5	11,6	14,6	14,9	19,3	15,7	16,8
Work compressor	[kW]	2,1	2,2	3,1	3,5	5,0	4,4	4,8
COP	[-]	5,9	5,2	4,7	4,3	3,9	3,5	3,5

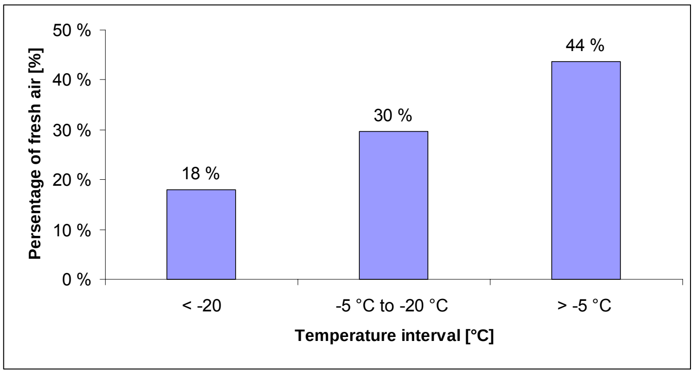


Figure C. 16 The standard of required amount of fresh air is dependent on the ambient temperature (NS-EN 13129-1, 2002)

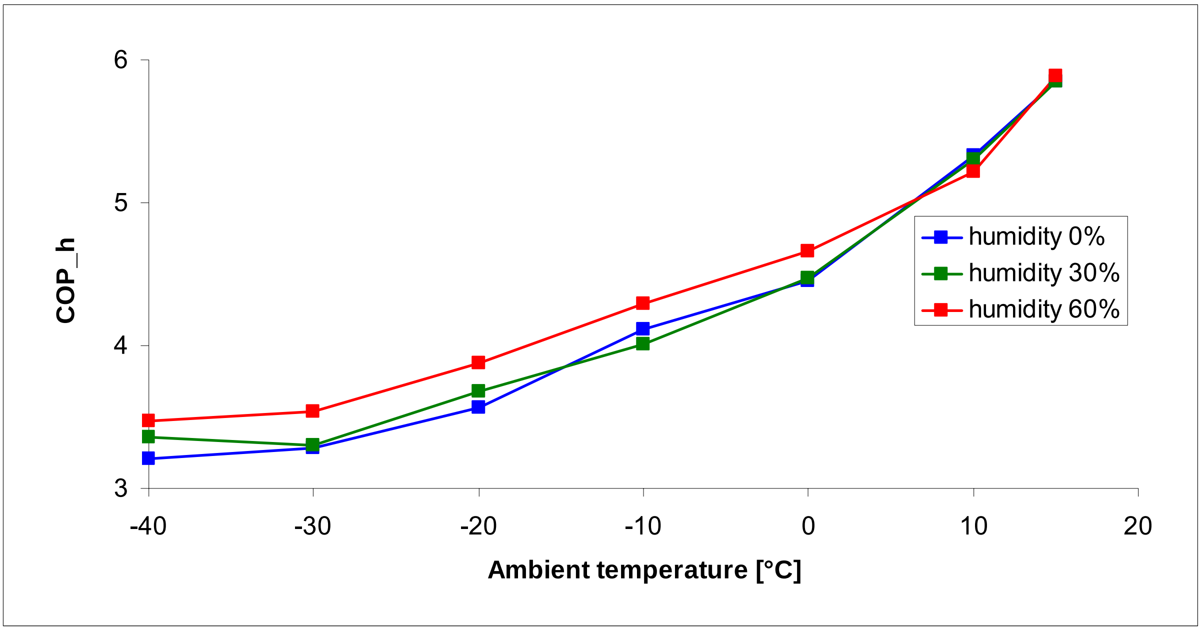


Figure C. 17 COP_h due to ambient temperature. Comparison of calculations HXsim /ProII model for three different air humidity levels

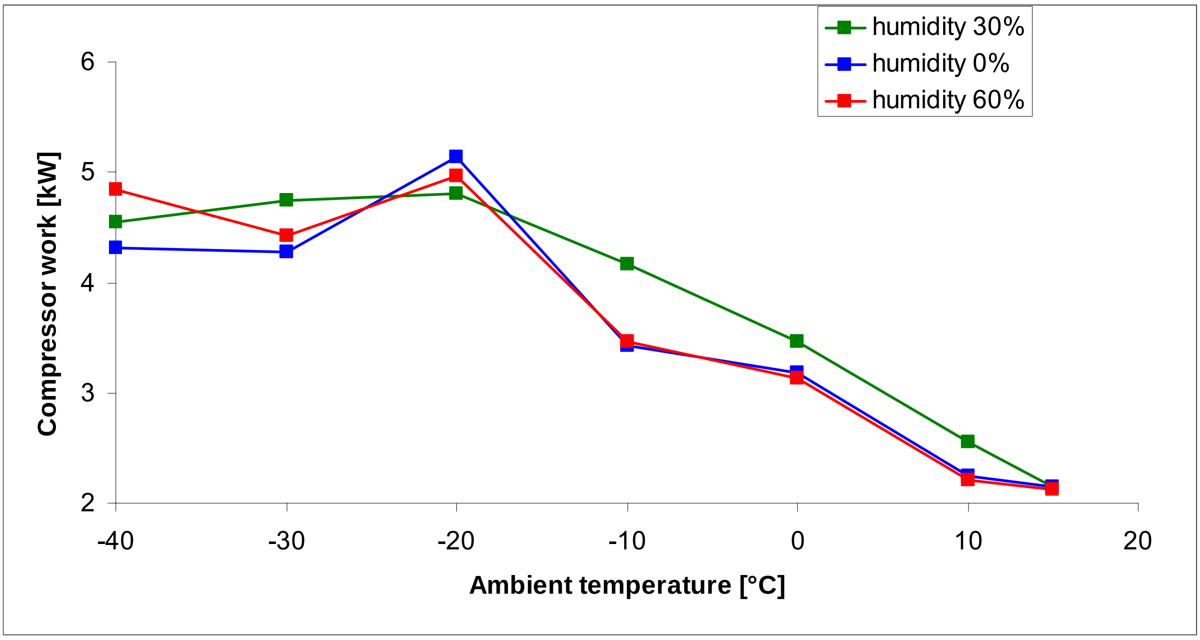


Figure C. 18 Compressor work due to ambient temperature. Comparison of calculations in the Excel model and the HXsim /ProII model for three different air humidity levels

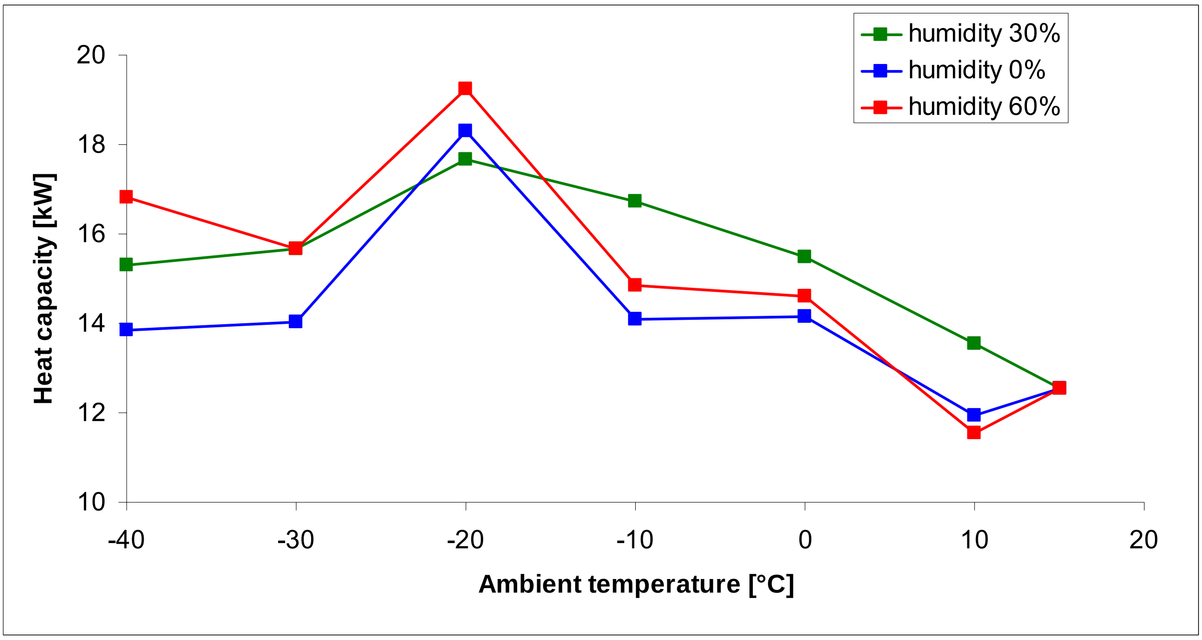


Figure C. 19 Heat capacity due to ambient temperature. Comparison of calculations HXsim /ProII model for three different air humidity levels

The heating mode results are not either increasing or decreasing curves as in cooling mode. In heating mode the required amount of fresh air changes with the temperature. Minus 30 °C needs 18 % of fresh air while -20 °C requires 30% of fresh air. The needed heat capacity is larger for ambient temperature at -20 °C than -30 °C which also leads to maximum compressor work at -20 °C. The same happens at -10 °C and 10 °C where the amount of fresh air changes.

The COP of the HXsim/ProII has increased compared with the Excel/RnLib model, due to the improved system circuit. When the ambient temperature is -40 °C and 0 % humidity the COP value is increased from 1,7 to 3,2

Figure C. 17 show the COP for different ambient temperatures. The COP decreases with decreasing temperatures due the increased pressure ratio both in the first and second stage compression and the increased temperature lift from ambient temperature to wanted treated air temperature.

Table C. 8. Impact relative humidity of air has on the heat exchangers. Ambient temperature is -30 °C.

	Q_Evap1	Q_evap2	Q_GC1	Q_GC2
0 %	5 146	4 525	10 335	3 691
60 %	7 335	3 898	11 365	4 290
	43 %	-14 %	10 %	16 %

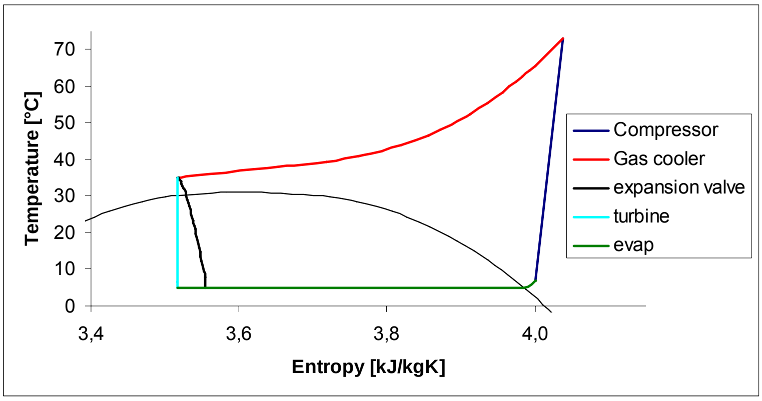
The humidity of air has positive impact of the COP and not negative as in cooling mode. The reason for this is the same as why the relative humidity has a negative impact of the cooling mode. When the relative humidity of air increases the potential “free” recovery heat in evap1 from the exhaust air increases. At ambient temperature at -30 °C the heat capacity of Evap1 at 0 % relative humidity is 5146 [kW]and at 60 % relative humidity 7 335 [kW] Increase of 43 % of “free” reheating. The increased relative humidity result in increased need of heating capacity, but the increase is not as big because the relative humidity in the air decreases throughout the gas cooler. As explained in the explanation of the relative humidity in cooling mode. The wanted vapour fraction out of Evap2 is 0,95. When the humidity of air increase to 60% the evaporation capacity increase that much that Evaporator2 need less heat to evaporate to 0,95 vapour fraction. The needed fresh air flow for Evap2 is reduced, which also reduce the fan work to get the fresh air flow through Evap2.

D EXPERIMENTAL WORK

D.1 Throttling losses

A challenge with CO_2 is the big throttling loss and this loss is kept to a minimum when the temperature before throttling is as low as possible. That is why an IHX is used in CO_2 systems. The IHX decrease the temperature before throttling and secure that no liquid enters the compressor inlet. When CO_2 systems use air as cooling source the throttling loss will increase with high ambient temperatures, this loss is possible to recover by using a two-phase flow turbine, in this report referred to as expander, in stead of an expansion valve. The maximum efficiency for CO_2 expanders is around 50% (Tøndell, 2006, page vi).

Compared to other refrigerant such as R134a and R152a the throttling loss for CO_2 is big, these refrigerants have a vaporization enthalpy in the same region, but CO_2 have higher heat capacity. A fluid with high heat capacity will have a high entropy formation. (Tøndell, 2006)



Compressor efficiency	0,7	[-]
Expander efficiency	1	[-]
Cooling capacity	30	[kW]
Gas cooler pressure	85	[bar]
Gas cooler outlet temperature	35	[°C]

Figure D. 1 Temperature Entropy diagram illustrating the throttling loss calculated in Rn-lib

The figure shows a temperature/entropy-diagram of a CO_2 cycle without internal heat exchanger. The area in the diagram between the isentropic expander and the expansion valve illustrate the throttling loss. The throttling loss is calculated to 2,8 kW 27% of the compressor work is lost in the throttling. With these conditions an expander with 50% efficiency will increase the COP with 19%. The example shows the potential of CO_2 expander especially when the temperature before throttling is high. In reports where different refrigerants are compared R744 is the best solutions when the ambient temperatures are low and R134a with high temperatures, regarding to COP because of the high throttling loss for CO_2 in high ambient temperatures.

Nekså et al.(2007) compare an R134a system with two different CO_2 systems. System A is a conventional R134a system and System B is the most common R744 system with internal heat exchanger. System C is a CO_2 system for the future, consisting of an expander (efficiency 55%) and two compressors with a gas cooler between them. The second stage compressor is driven by the expander.

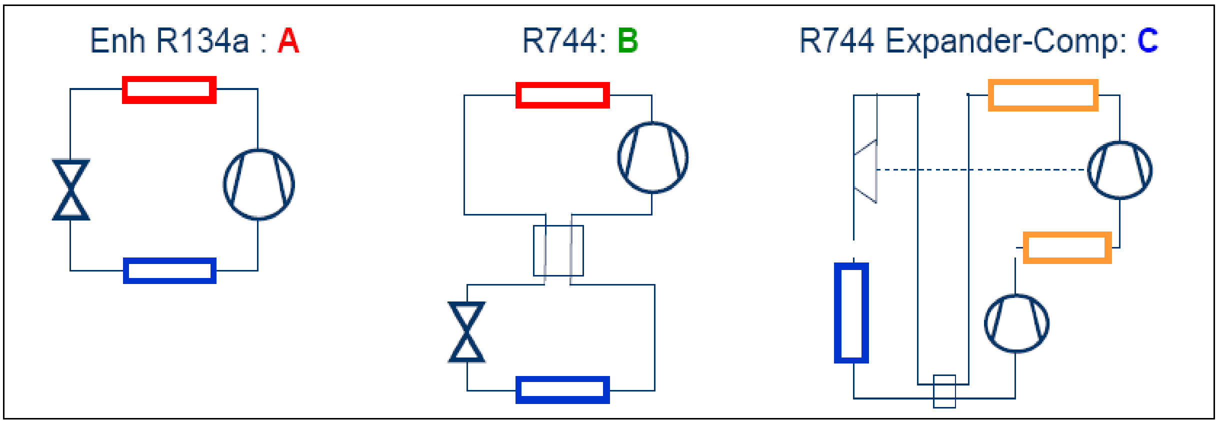


Figure D. 2 Sketch of three different refrigerant systems (Nekså et al., 2007).

Test results are used for all efficiencies in the different systems components. For ambient temperatures below 38°C System B have a higher COP than the R134a system. In Norway temperatures above 38 °C occur seldom and system B will be better than system A most of the time. At ambient temperatures at 38 °C where system A and B have equal COP System C have 27% higher COP than the two other systems. For ambient temperature 45 °C System C still have 8% higher COP than System A. The report shows the good opportunities for CO_2 and concludes that the potential improvements for CO_2 systems are much higher than for R134a systems.

D.2 Case study expansion turbine

In this case study it has been calculated on three CO_2 systems operating in cooling mode. The system design is equal to the systems in Figure D. 2. System C has been calculated on two different ways. In calculations of System C1 the expander is driving the second stage compressor and the middle pressure between the first and second compressor is decided by the expander work. Calculations of System C2 the middle pressure between the first and second compressor is the optimal middle pressure. This means that the second stage compressor is partly driven by the expander and partly from another power supply.
The potential energy savings with an expander is compared for different conditions.

Figure D. 1 shows that the specific evaporation enthalpy is increased when an expander is used. Increased evaporation enthalpy reduces the mass flow needed which again gives a reduced compressor work. COP is defined as cooling capacity divided by the total work needed to run the system; therefore the expander output power is subtracted from the compressor work in the COP calculation.

D.2.1 General assumptions

• Compressor efficiency	80	[%]
• Evaporator temperature	5	[°C]
• Evaporator outlet	Saturated vapour	[°C]
• Efficiency IHX	80	[%]
• Gas cooler outlet conditions	Ambient temperature + 5 °C	[°C]
• Pressure drop heat exchangers	0	[bar]

D.2.2 Equations

Evaporation heat	$\dot{Q}_0 = \dot{m}_{co_2}(\Delta h_{evap})$	[W]	Eq 6
Gas cooler heat	$\dot{Q}_{gc} = \dot{m}_{co_2}(\Delta h_{gc})$	[W]	Eq 7
Compressor work	$W_{compressor} = \dot{m}_{co_2}(\Delta h_{compressor})$	[W]	Eq 8
Throttling loss (T_gc_out [Kelvin])	$W_{throttling\_loss} = \dot{m}_{co_2} T\_gc\_out(\Delta s_{valve})$	[W]	Eq 9
Power expander	$P_{\exp} = \dot{m}_{co_2}(\Delta h_{\exp}) * \eta_{is.\exp} * \eta_{\text{mechanical.exp}}$	[W]	Eq 10
Coefficient of Performance	$COP = \left(\frac{\dot{Q}_0}{W_{compressor}}\right)$	[-]	Eq 11
Coefficient of Performance with turbine	$COP = \left(\frac{\dot{Q}_0}{W_{compressor} - P_{\exp}}\right)$	[-]	Eq 12
Calculate work by revolution and torque	$W = 2\pi * torqe[Nm] * \frac{revolution[1/\min]}{60[s/\min]}$	[W]	Eq 13
IHX	$\Delta q_{ihx} = \min(\Delta h_{ihx\_cold} : \Delta h_{ihx\_hot}) * \eta_{ihx}$	[kJ/kg]	Eq 14
	$\Delta q_{ihx\_cold} = h_{c\_out} - h_{c\_in}$ $\Delta q_{ihx\_hot} = h_{h\_in} - h_{h\_out}$	[kJ/kg] [kJ/kg]	Eq 14.1 Eq 14.2
h_{c_in} is the evaporator outlet conditions while h_{c_out} is the maximum specific enthalpy i.e. temperature is as high as similar to the hot refrigerant inlet temperature while the pressure is similar to the evaporator temperature. h_{h_in} is the gas cooler outlet conditions while h_{h_out} is the minimum specific enthalpy i.e. temperature is as low as possible similar to cold refrigerant inlet temperature and pressure similar to the gas cooler pressure. Δq_{ihx} is the smallest the values Δq_{ihx_cold} and Δq_{ihx_hot} multiplied with the IHX efficiency			
Enthalpy in the compressor outlet	$h_{comp\_out\_actual} = h_{comp\_in} + \frac{(h_{comp\_out\_is} - h_{comp\_in})}{\eta_{is\_comp}}$	[kJ/kg]	Eq 15
Enthalpy expander	$h_{\exp\_out} = h_{\exp\_in} - \frac{P_{\exp}}{\dot{m}_{co_2}}$	[kJ/kg]	Eq 16

D.2.3 Calculation method used in the case study

Temperature, pressure, specific enthalpy and specific entropy are found before and after each unit in the system as explained below.

Compressor

The evaporator outlet is the start of the calculation. It has been assumed an evaporation temperature of 5 °C and that the outlet conditions are on the saturated vapour line. By this information it is possible to find pressure (p), specific enthalpy (h) and specific entropy (s) using RnLib. These values are the input values of the compressor if there are no IHX. In the system with IHX the specific enthalpy is h_comp_in = h_evap_out + q_ihx. The pressure drop is assumed zero so the temperature, h and s can be found with RnLib.

To find the outlet conditions of the compressor it is first assumed that the compressor is isentropic. In an isentropic process the entropy is constant. The compressor outlet pressure is known (explanation further down) and with known p and s the other properties can be found using RnLib. The real compressor outlet is found by Eq 15

Gas cooler

The gas cooler is set to have a pinch of 5 °C and the pinch occurs in the gas cooler outlet. The gas cooler outlet temperature then is ambient temperature + 5 °C. The pressure is known and the other values can be found using RnLib. This method is also used for system C where there are two gas coolers. The outlet and the pressure are known for both of the gas coolers.

IHX

The inlet conditions of the hot and cold fluid in the IHX is known and high pressure side is similar to compressor outlet and low pressure side is similar to evaporation pressure. Eq 14 is used to find the exchanged heat between the hot and cold refrigerant.

Valve

The refrigerant is going from high pressure through the valve and is throttled down to evaporator pressure. This process the enthalpy is constant. Then the enthalpy before the evaporator is known and the pressure is known so the rest of the properties can be found using RnLib.

Expander

To find the outlet conditions of the expander it is first assumed that the expander is isentropic, i.e. the entropy is constant. The pressure outlet pressure is the evaporation pressure. To find the real specific enthalpy Eq 16 is used. The other properties can be found using RnLib.

Mass flow CO_2

If the needed demand of cooling [kW] is known the CO_2 mass flow can be calculated by Eq 6. The specific enthalpy values of the inlet and outlet of the evaporator is found in the calculations above. If the system operates as a heat pump the heating demand is deciding the mass flow and Eq 7 can be used.

Optimal high pressure

The optimal high pressure is the pressure giving the highest COP. This value have to be found be iteration. Initially a high side pressure is selected and then different pressures have to be tried to find the optimal. In Figure D. 3 the ambient temperature is 35 °C and the initial value of the gas cooler pressure is 75 bar for system A. When the pressure rises the specific compressor work increase linearly, because of the assumption of constant efficiency of the compressor. With high pressures between 75 and 95 bar the specific evaporation enthalpy increase rapidly and therefore the COP also increase rapidly. When the optimal pressure is reached a further increase of the gas cooler pressure will have negative impact of the COP because the evaporation enthalpy increasing to slow compared to the increasing compressor work. The figure also show that if the regulation of the high pressure side is inaccurate it is better to have a higher than a lower high pressure than the optimal pressure. For high ambient temperature the curve up to the optimal pressure is rounded, but for ambient temperatures where the optimal pressure is around the critical pressure, the curve up to the optimal pressure is very steep. In that case a few bars to low high side pressure have very negative impact of the COP. Therefore the high pressure side is usually set to minimum 80 bars if the cycle is transcritical (Stene, 2008).

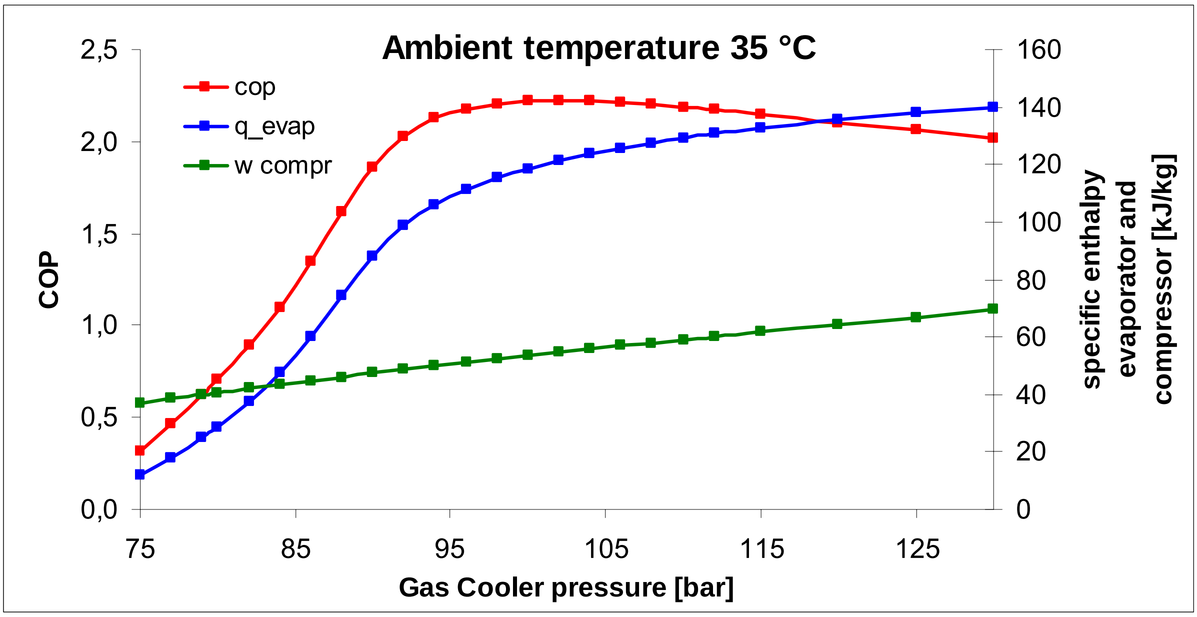


Figure D. 3 COP, Specific evaporator and compressor enthalpy for different gas cooler pressures calculated on system A.

Middle pressure for system C1 and C2

For system C1 the output power from the expander is used to drive the second stage compressor. The method which is used to find the optimal high pressure and the middle pressure is to guess a starting value of the high side pressure and how many bars the pressure increase in the second stage compressor (Δp_comp2). The correct value of Δp_comp2 is where the expander output work and the work of the second stage compressor is equal, since the expander is driving the compressor. The optimal pressure and correct Δp_comp2 is found by iteration or to use the solver function in excel. The high side pressure of System C2 is the same as System C1. The system has two compressors and in System C2 the optimal middle pressure is found. The optimal middle pressure can be found by find the Δp_comp2 which gives the highest COP for the system (using the solver function).

Table D. 1 Optimal high side pressures and middle pressures for the three systems. In system C1 the expander efficiency is 50%

	A	B	C1		C2	
Ambient temperature	high pressure [bar]	high pressure [bar]	high pressure [bar]	mid pressure [bar] (exp ef. 50%)	high pressure [bar]	optimal mid pressure [bar]
25	75	75	75	69	75	56
26	77	76	75	68	75	56
27	80	79	77	69	77	57
27,5	81	80	78	70	78	57
30	88	86	84	74	84	60
32,5	95	92	89	78	89	63
35	102	98	95	82	95	65
40	117	111	105	89	105	70
45	133	123	115	96	115	74
50	151	136	125	103	125	78

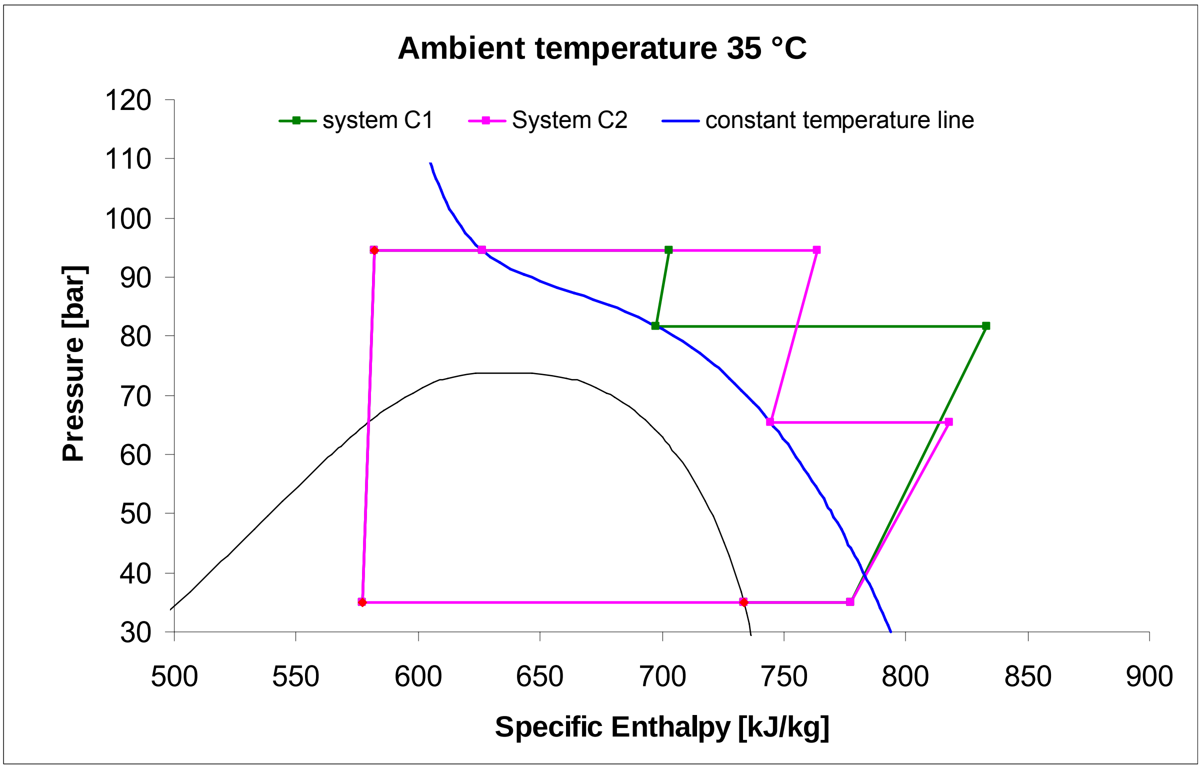


Figure D. 4 Pressure specific enthalpy diagram of system C1 and system C2.
The expander efficiency is 50 %. The figure shows that all the gas cooler outlet temperature is the same and that the power of the expander does not have enough power to have optimal middle pressure. The COP of system C1 is 2,81and 2,92 for system C2.

D.2.4 Results of expander calculations

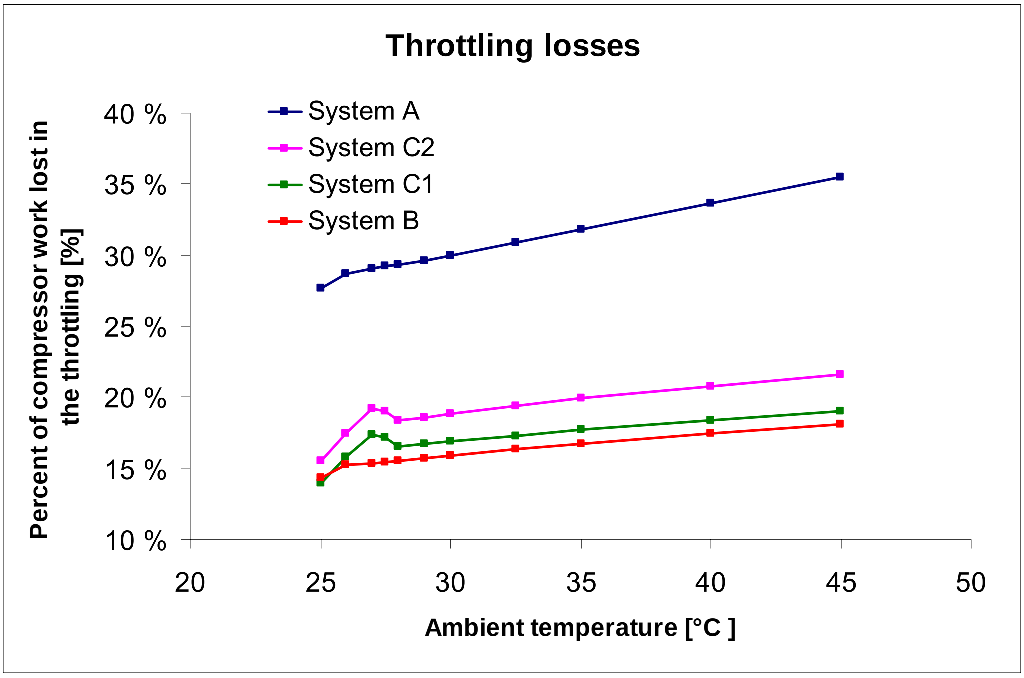


Figure D. 5 Percent of compressor work lost in throttling

System A has the highest potential for savings if an expander is used. That is because there are no IHX and the temperature before throttling than is high. However system A is a bad design when CO_2 is the refrigerant.

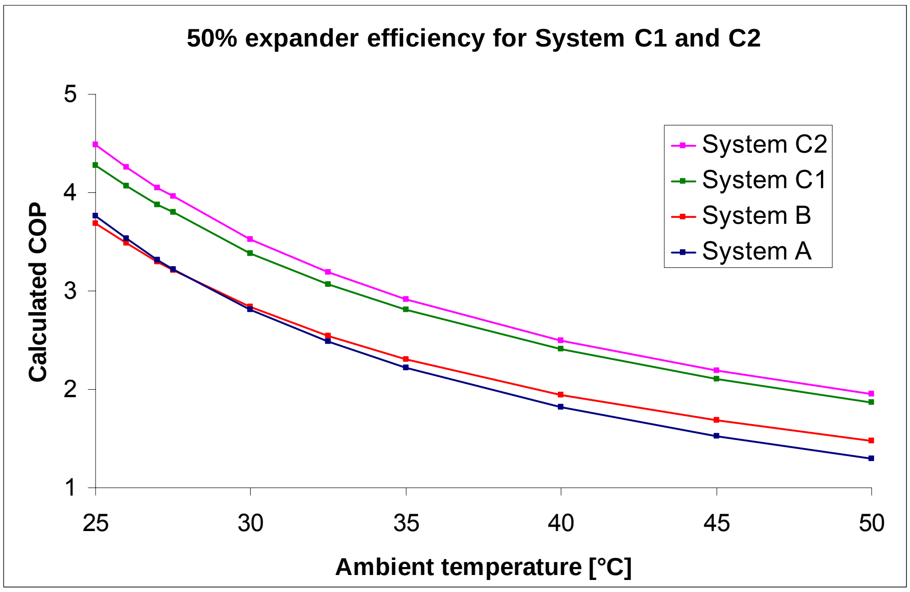


Figure D. 6 Calculated COP for the systems when the efficiency of the expander is 50% in System C1 and C2, while System A and System B have expansion valves.

The potential savings for system C looks promising, but the cost of this system will be high. In Figure D. 6 it is only System C1 and C2 having an expander. If the two other systems are modified to have an expander in stead of a valve, System A will have a big improvement of the COP because the potential savings are the largest of the systems. System B does not have that big potential of saving and the COP does not improve so much with an expander.

Table D. 2 COP Improvement for System A, B, C1 and C2 when expansion valve is replaced by an expander with efficiency at 50%. (The impact of an expander having 100%, 75 % and 25 % efficiency is found in Appendix D)

Ambient temperature [°C]	A		B		C1		C2	
	COP	COP improvement	COP	COP improvement	COP	COP improvement	COP	COP improvement
25	4,44	17,99 %	4,01	8,73 %	4,27	9,19 %	4,49	9,56 %
26	4,20	18,96 %	3,82	9,41 %	4,07	10,52 %	4,26	10,92 %
27	3,96	19,40 %	3,61	9,62 %	3,88	11,81 %	4,05	12,22 %
27,5	3,85	19,63 %	3,52	9,72 %	3,80	11,78 %	3,96	12,18 %
30	3,40	20,74 %	3,13	10,25 %	3,38	11,90 %	3,52	12,29 %
32,5	3,03	21,99 %	2,82	10,75 %	3,07	12,58 %	3,19	12,96 %
35	2,74	23,30 %	2,56	11,25 %	2,81	13,23 %	2,92	13,60 %
40	2,29	26,09 %	2,18	12,24 %	2,41	14,41 %	2,50	14,80 %
45	1,97	29,13 %	1,90	13,20 %	2,11	15,61 %	2,19	16,03 %
50	1,71	32,50 %	1,69	14,19 %	1,87	16,81 %	1,95	17,27 %

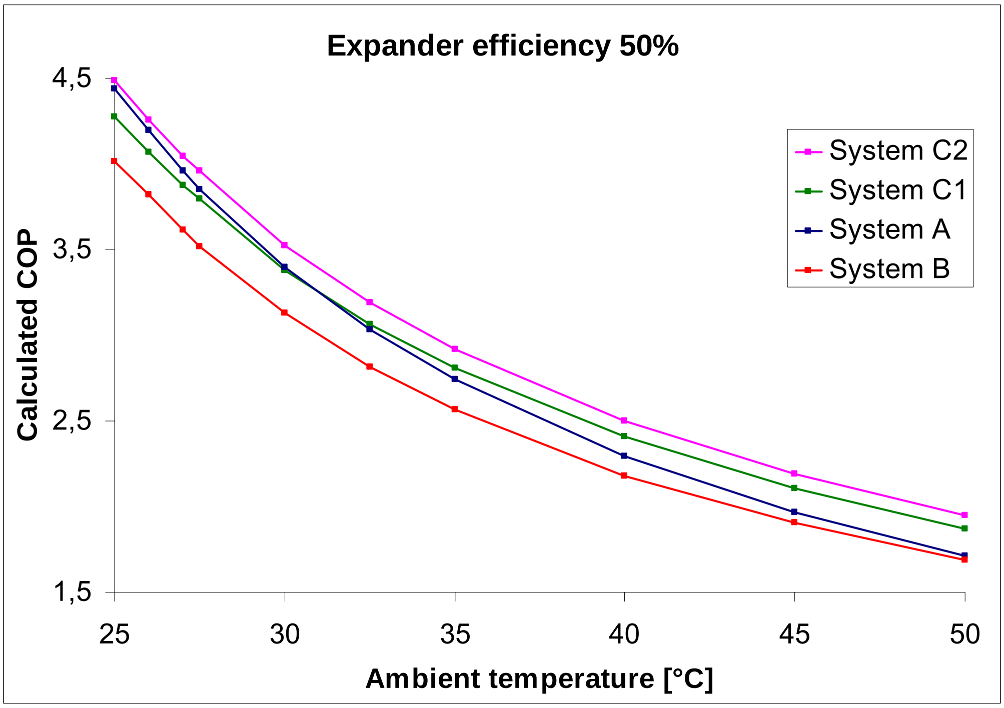


Figure D. 7 Calculated COP if all the system has an expander with efficiency 50 %

The difference between the systems due to COP is now not that great. System A is better than System C1 for ambient temperatures less than 30 °C. When the expander efficiency increase System A getting better and better compared to the other systems. This shows that system with a high efficient expander the system design can be simplified and still have high COP. This applies particularly for climatic areas where temperatures above 30 °C occur seldom.

D.3 Expander turbine experimental work

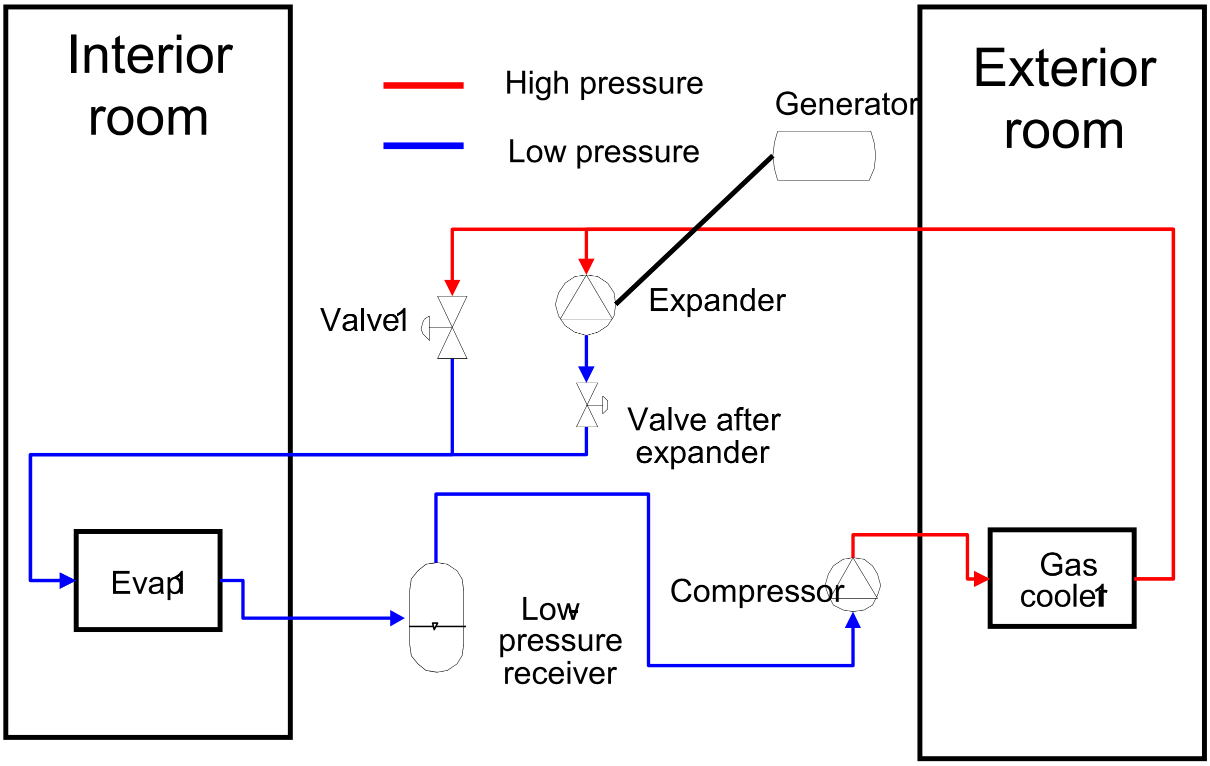


Figure D. 8 Sketch of the main components on the test facility

The aim of the work on the laboratory was to test an expander. Every test was done with constant conditions in the interior and exterior room. The temperature in these rooms is fixed in a computer programme and is regulated to keep the room temperatures constant. It is also possible to control the fans in these rooms to control the airstream through the heat exchangers. In all tests of the expander valve 1 was closed so the whole CO_2 mass flow passed through the expander. The expander is made by OBRIST Engineering and they keep all information about it confidential.

In general it was a problem for all tests to achieve a high pressure before the expander. With a low revolution speed in the expander the pressure was at the highest but then the output power from the expander gets low. With a high revolution speed in the expander the high pressure side decrease, this is because the CO_2 flow meets a smaller resistance going through the expander. To control the revolution of the expander an electrical generator was used. The generator controlled the torque of the expander and by this control the revolution.

The valve after the expander is implemented to the system to increase the opportunities to control the high pressure side and by this move the cycle to the left in temperature/entropy diagram. Further in the report when information about the valve position is given, it is always information about the expansion valve after the expander and not expansion valve 1. Expansion valve 1 is always totally closed when the expander is tested.

Figure D. 8 shows there are no IHX in the system. The IHX is bypassed to minimize the pressure drop between the compressor outlet and the expander inlet. The temperature before the expander gets higher without the IHX and as showed in the report introduction it is important that this temperature is as cold as possible, but as Figure D. 7 shows the potential work recovery by the expander increases when the expander inlet temperature increases.

In the pressure enthalpy diagrams presented below (Figure D. 11, Figure D. 12 and Figure D. 13) each point has a measured value for the pressure and the temperature. The enthalpy is calculated from CO_2-lib except for the enthalpy value out of the expander. The power of the expander is used to find the enthalpy of the expander outlet.

The power of the expander and the compressor can be calculated by the torque and the revolution speed found on a measuring devise, Eq 16. The compressor work can also be found by using the pressure enthalpy diagram, Eq 8. Comparison shows that the difference between the compressor work calculated from the pressure enthalpy diagram and the measured values of compressor torque and revolution differ maximum 3,9%.

D.4 Results of test sequence 3

In this sequence the compressor revolution is kept constant at 30 Hz (1800 rpm) and the mass flow constant at 3,9 kg/min. The sequence consists of three test series each with a constant expander revolution 1800, 3000 and 4000. In each test series different positions of the expansion valve is tested. The total pressure drop between the high and low pressure side is divided in two parts one over the expander and the rest over the valve. The position of the valve decides how big part of the total pressure drop occurs in the expander. When the valve is open the whole pressure drop are done in the expander and the efficiency than gets low. The output power from the expander is however high because the potential of energy recovery increases when the pressure drop over the expander increases.

The CO_2 mass flow was manually kept constant by reduce of increase the total amount of CO_2 in the system. When the expansion valve was opened the amount of CO_2 had to be increased, and when the expansion valve was closed CO_2 have to be filled into the system to keep the mass flow constant.

D.4.1 Expander efficiency and power output

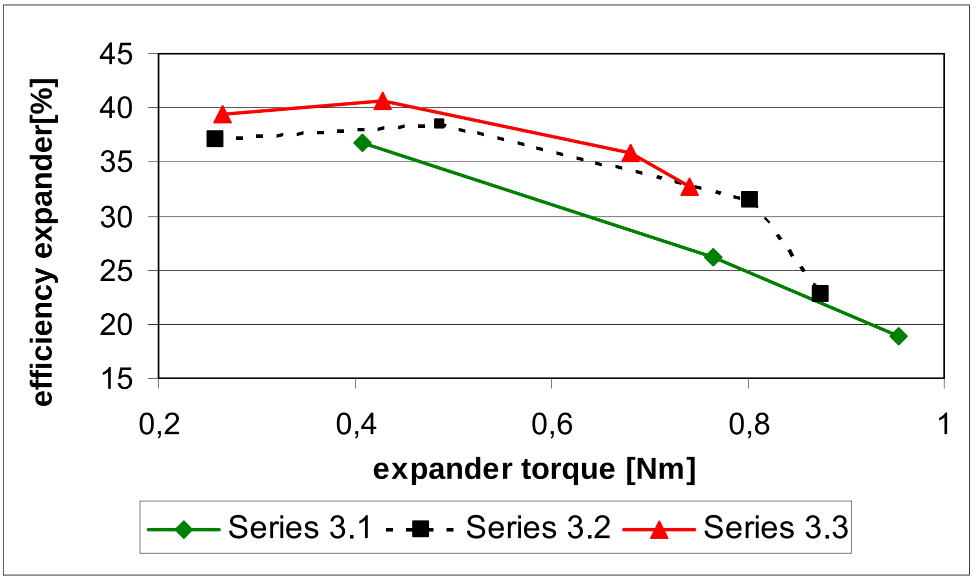


Figure D. 9 Efficiency for the expander due to expander torque

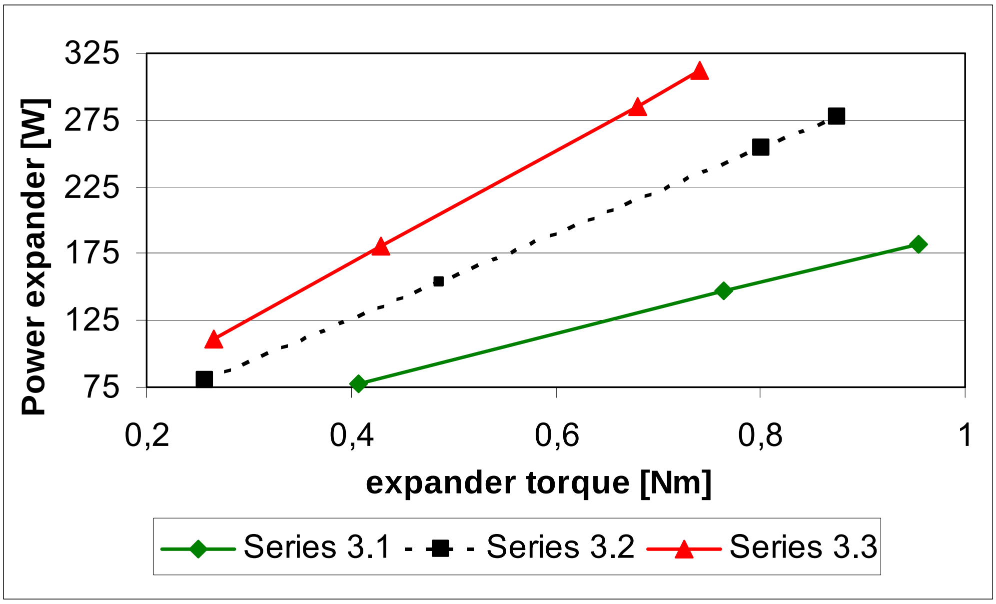


Figure D. 10 Power output from the expander due to expander torque

D.4.2 Series 3.1

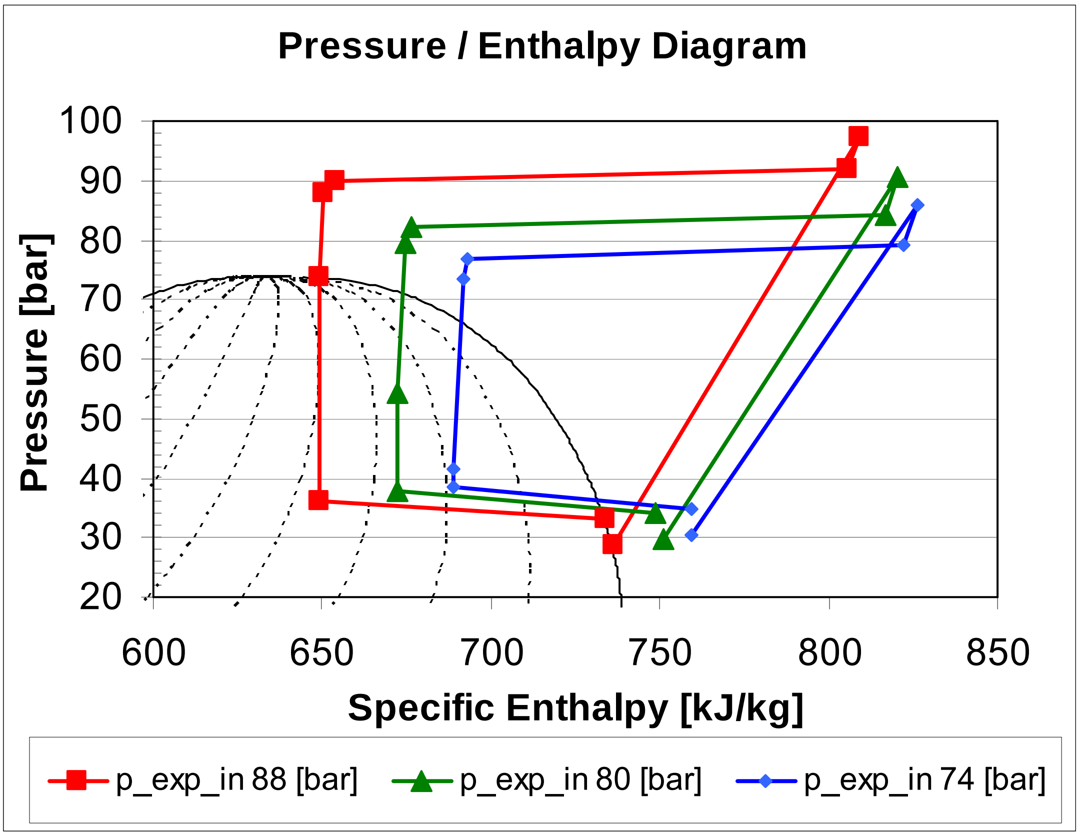


Figure D. 11 Pressure/Enthalpy Diagram series 3.1

Table D. 3 Data values of series 3.1

Torque	**Nm**	0,41	0,76	0,95
rpm	**min^{-1}**	1821	1839	1825
power [W]	**W**	77,8	147,1	182,2
massflow $_{CO2}$	**kg/min**	3,92	3,87	3,87
p$_{_exp_in}$ [bar]	**bar**	87,9	79,6	73,6
p_exp_out [bar]	**bar**	73,6	54,2	41,5
Δp$_{_Expander}$ [bar]	**bar**	14,3	25,4	32,1
t$_{_exp_in}$	**C**	39,3	35,8	32,8
t$_{_exp_out}$	**C**	30,9	17,7	6,8
exp efficiency [%]	**%**	36,7	26,2	18,9
Q0	**W**	5523	4942	4531
Work Compressor	**W**	4741	4501	4322
COP	**[-]**	1,18	1,13	1,09

D.4.3 Series 3.2

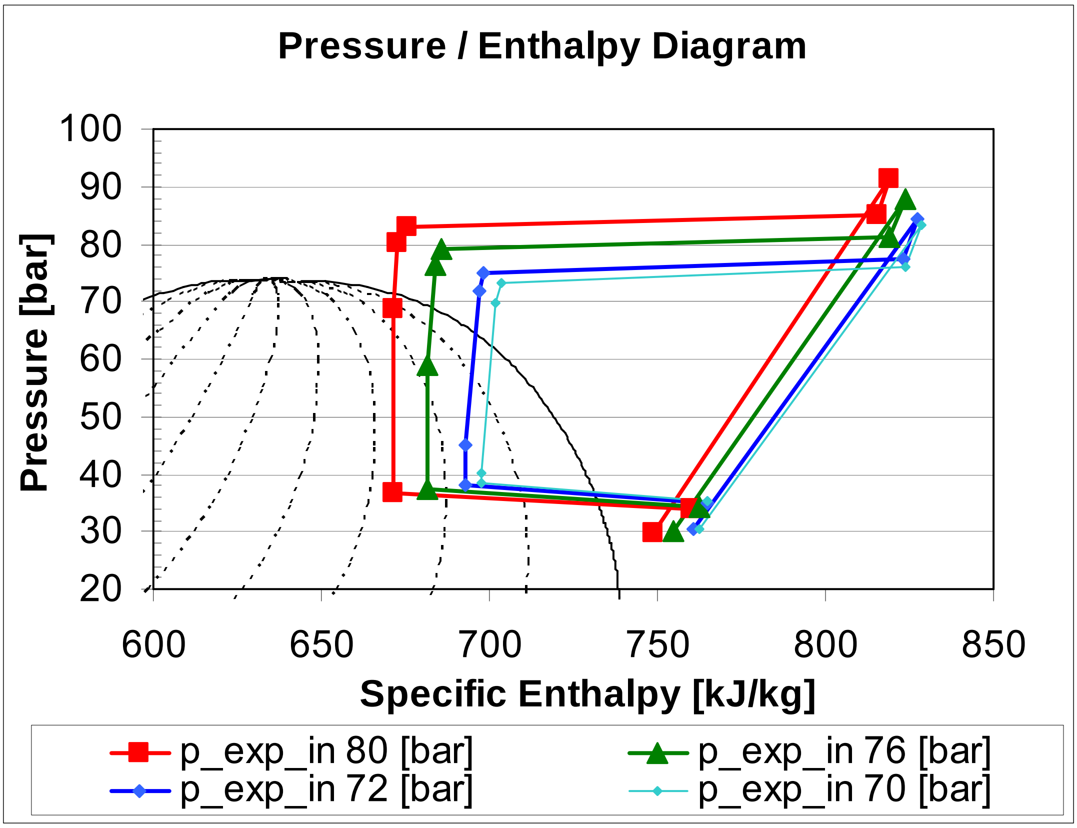


Figure D. 12 Pressure/Enthalpy Diagram series 3.2

Table D. 4 Data values of series 3.2

Torque	**Nm**	0,26	0,49	0,80	0,88
rpm	**min^{-1}**	2991	3008	3031	3025
power [W]	**W**	80,7	153,1	254,4	277,3
massflow $_{CO2}$	**kg/min**	3,89	3,87	3,87	3,90
$p_{\_exp\_in\ [bar]}$	**bar**	80,3	76,3	71,7	69,8
p_exp_out [bar]	**bar**	68,8	59,1	45,0	40,2
$\Delta p_{\_Expander\ [bar]}$	**bar**	11,5	17,2	26,7	29,6
$t_{\_exp\_in}$	**C**	36,1	34,1	31,8	31
$t_{\_exp\_out}$	**C**	27,9	21,3	10	5,5
exp efficiency [%]	**%**	37,0	38,5	31,5	22,7
Q0	**W**	5762	5223	4616	4359
Work Compressor	**W**	4553	4426	4296	4279
COP	**[-]**	1,29	1,22	1,14	1,09

D.4.4 Series 3.3

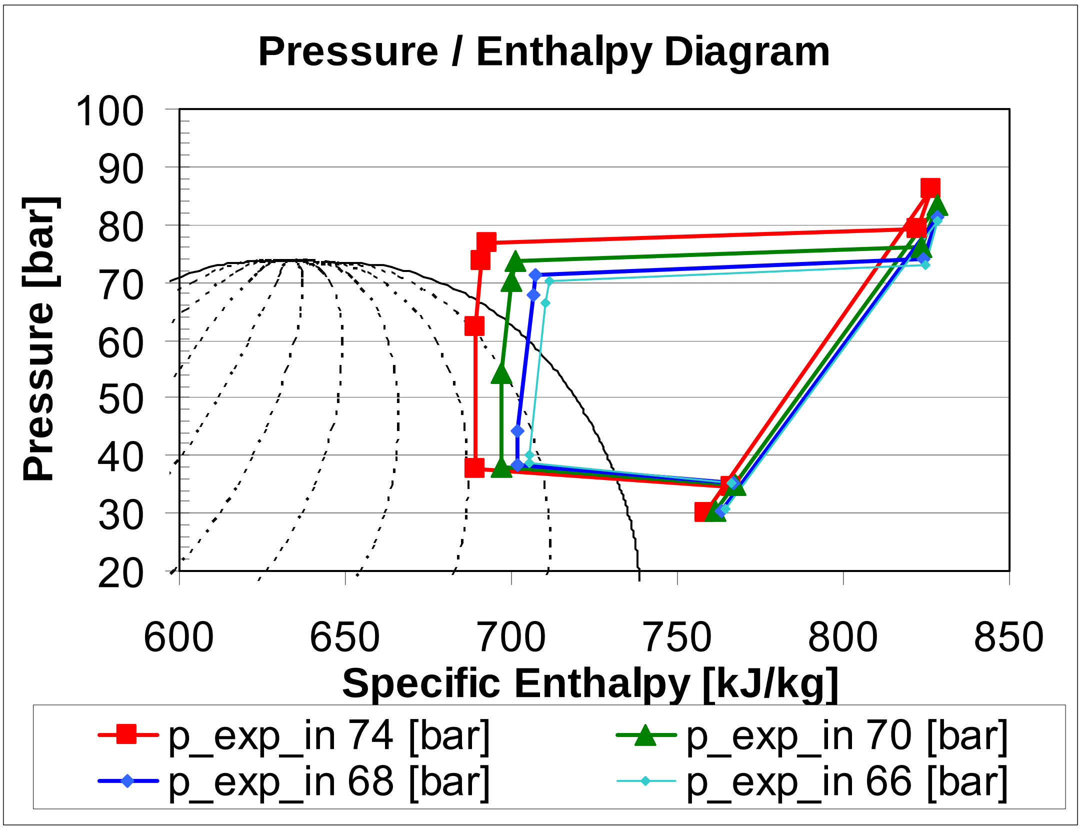


Figure D. 13 Pressure/Enthalpy Diagram series 3.3

Table D. 5 Data values of series 3.3

Torque	**Nm**	0,27	0,43	0,68	0,74
rpm	**min^{-1}**	3980	4014	4007	4015
power [W]	**W**	111,0	180,2	285,2	311,7
massflow $_{CO2}$	**kg/min**	3,86	3,85	3,88	3,89
$p_{\_exp\_in\ [bar]}$	**bar**	73,8	70,3	67,7	66,3
p_exp_out [bar]	**bar**	62,2	54,4	44,0	40,2
$\Delta p_{\_Expander\ [bar]}$	**bar**	11,6	15,9	23,7	26,1
$t_{\_exp\_in}$	**C**	32,4	31	29,9	29,4
$t_{\_exp\_out}$	**C**	23,5	17,8	9,1	5,5
exp efficiency [%]	**%**	39,4	40,6	35,9	32,7
Q0	**W**	4938	4496	4176	3957
Work Compressor	**W**	4378	4252	4176	4165
COP	**[-]**	1,16	1,10	1,07	1,03

D.4.5 Discussion of experimental expander work

The pressure/enthalpy diagrams show that no of the cycles operates in good conditions. The compressor inlet condition is too much superheated. The evaporator outlet is in the superheated area and that is the reason why all the series have no liquid in the receiver. Increased CO_2 mass flow will move the whole cycle to the left in the pressure/enthalpy diagram. This could not be done since the expander has a mass flow limitation for about 4 kg/min.

The test sequence also show that COP decreases when the valve is opened and more of the pressure drop is taken by the expander. The highest COP in the test series occur when the expander is not in use. This is because the cycle conditions are better when an expansion valve is used than with an expander taking the whole pressure drop. The influence of the bad cycle conditions has a more negative effect on the COP than the positive effect from the output expander work. The highest expander power occurs when the whole pressure drop is in the expander, but the expander efficiency is bad in these conditions (Figure D. 9 and Figure D. 10).

The test sequence shows that expander has a negative impact on the COP, but as mentioned the reason is the bad operation conditions. In different experiments with different conditions the expander efficiency usually is between 30% and 40%. When better operation condition for the cycle are reached and the expander efficiency are in the same region the expander can improve the cycle COP between 8% and 20% depending on the expander inlet temperature (Figure D. 7).

E LCCP COMPARATION OF REVERSIVLE CO_2 HEAT PUMP WITH EXISTING R134a SYSTEM

The LCCP chapter does not include calculation of "start of-" and "end of life". It is assumed that this is similar for the different refrigerants. However, for HFC's it is important to have a good system for delivering of the refrigerant when the Air Conditioning system is taken out of service.

E.1 Global warming impact for different refrigerants in the train sector

The calculations below are to show a scenario of the Global Warming impact different refrigerants used in Trains will have on the environment. Today 75% of the air conditioning units in train use R134a as refrigerant.

Data extracted from the an article prepared for the European Commission with the title "*Analysis of the emissions of fluorinated greenhouse gases from refrigeration and air conditioning equipment used in the transport sector other then road transport and options for reducing these emission"*(Schwarz and Rhiemeier, 2007). All the data from the report is approximately which results in inaccurate calculations. However the main picture of the Global Warming impact of different refrigerants and the importance of keep a low annual leakage rate is correct.

Totally the EU railway-, tram- and metro operators consist of 175 000 units and about 65 000 of them is equipped with air conditioning systems. The total refrigerant charge of these vehicles is 1 180 metric tons. The average annual leakage rate is 5% in the railway sector. Based on this data each unit have approximately 18 kg refrigerant per unit. NSB trains consist of 5 coaches which give each train a refrigerant charge of 90 kg. The calculations show that leakage from a train with the average annual leakage of 5% contributes to a annual Green House Gas emission of 63 tonnes CO_2 eq.

Table C. 9 Calculated GW-impact of different refrigerant used in railway air conditioning. Different annual leakage rates are assumed to get a view of the importance of having a low leakage rate.

Refrigerant	GWP (CO2 eq. per kg refrigerant)	tonnes CO2 eq			
		5 %	10 %	20 %	30 %
R134a	1 410	63	127	254	381
R407c	1 600	72	144	288	432
R22	1 700	77	153	306	459
R410a	1725	78	155	311	466

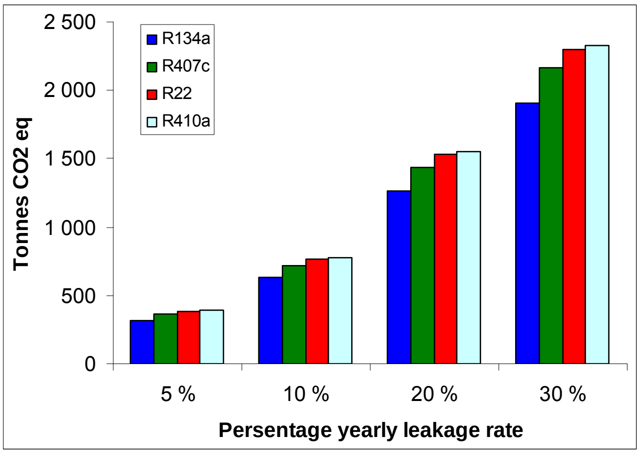


Figure C. 20 Calculated GW-impact of different refrigerant used in railway air conditioning

For HFC refrigerant system the leakage charge should be kept as low as possible as the figure show. CO_2 is not included in this calculation, because the GWP value of CO_2 is 1, since CO_2 is the reference value. When CO_2 is used as refrigerant the value of GWP is zero. This is because the CO_2 refrigerant is a waste product and can be taken from the industry. This is one of the main advantages with CO_2. A leakage will have a negative impact of the CO_2 circuit if the refrigerant charge get to low, but a leakage does not have any negative impact of the environment. When the system is discharged the CO_2 is let out to the air. This is also positive if an emergency situation in the CO_2 system occurs, safety valves discharges the system to the ambient without environmental damages.

E.2 Case study on driving the CO2 system one year in three cities

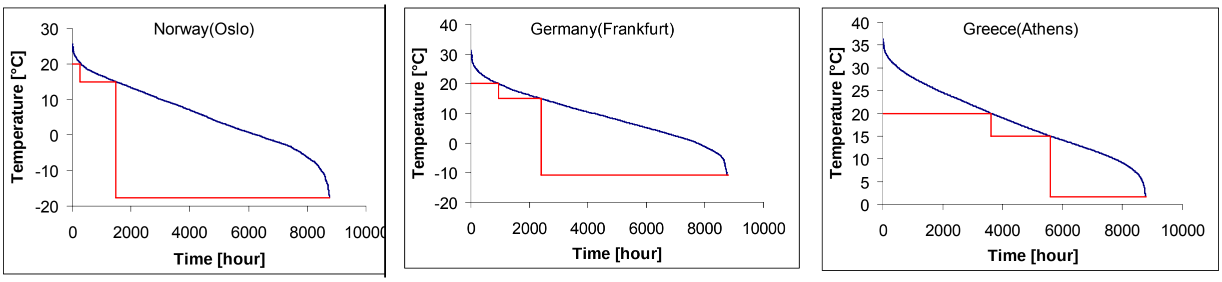


Figure E. 1 Temperature/time diagram

The figure is based on data from Meteonorm (Edition 5.1) which give temperatures for each hour around the year for different cities. In the diagram the temperatures is sorted from high to low temperature. Temperature above 20 °C occurs in 264 hours in Oslo, 958 hours in Frankfurt and 3600 hours in Athens. The temperature above 20 °C is the interval the Air conditioning is in cooling mode. Temperatures below 15 °C occur in 7272 hours in Oslo, 6390 in Frankfurt and 3198 hours in Athens. Based on the climate static it is clear that a heat pump for Norwegian train will save much energy because of the long winter.

A case study is done where a train is used from 0600 to 1800 every day throughout a year in Oslo, Frankfurt and Athens. Data from the HXsim/ProII simulation model is used to calculate energy consumption and required heating/cooling capacity. It is assumed that relative humidity of air is 30 °C. Data from Meteonorm is used for temperatures in the cities. Solar radiation is not included in the calculations. Implementing of solar radiation would lead to increased total hours of air conditioning and increased total need of cooling. This is because weather statistics is taken in the shadow. The calculation method is explained in detail in Appendix C

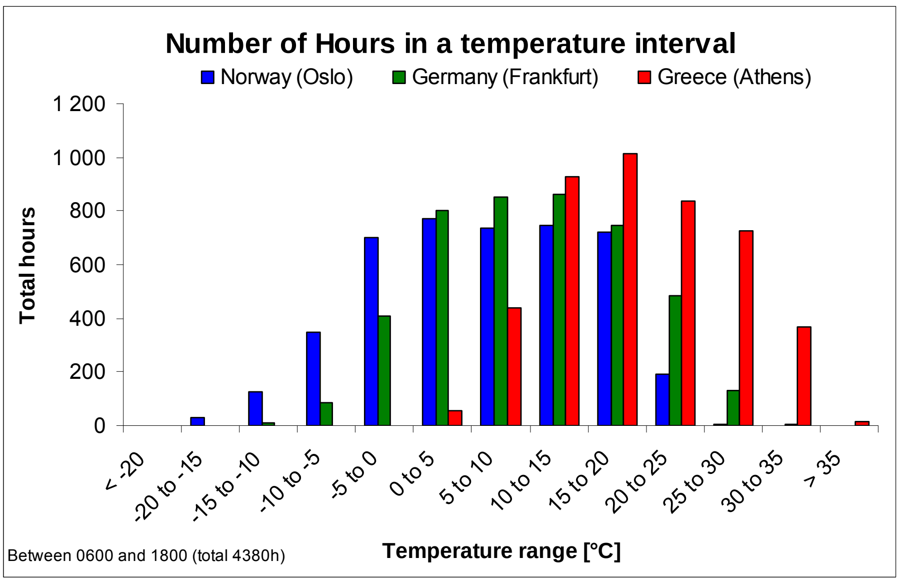


Figure E. 2 Diagram which shows how many total hours a temperature occurs in a temperature interval for the cities Oslo, Frankfurt and Athens. Norway is the country with the coldest climate and therefore have the greatest potential energy saving using a heat pump in stead of electrical heating.

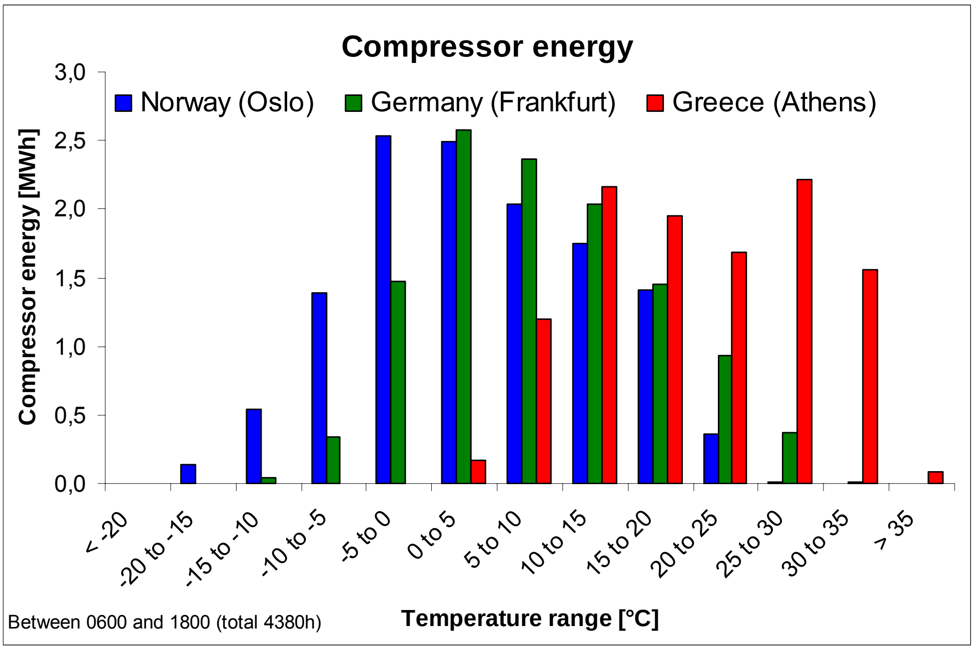


Figure E. 3 Compressor energy [MWh] needed to provide sufficient cooling/heating for the train. The train is in service between 0600 and 1800 each day for one year.

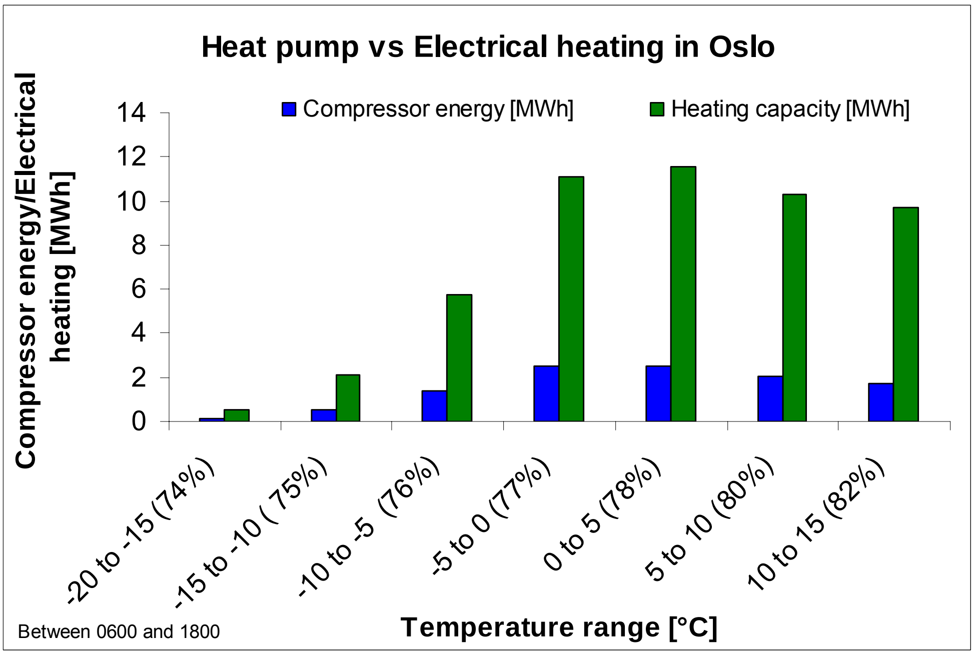


Figure E. 4 Potential savings of using CO_2 heat pump in stead of electrical heating in Oslo. (The percentage values on the x-axis are the percentage reduction of energy if the train is heated with a CO_2 heat pump in stead of using electrical heating). In Norway today train use electrical heating. This Figure shows a great saving potential of using a CO_2 heat pump.

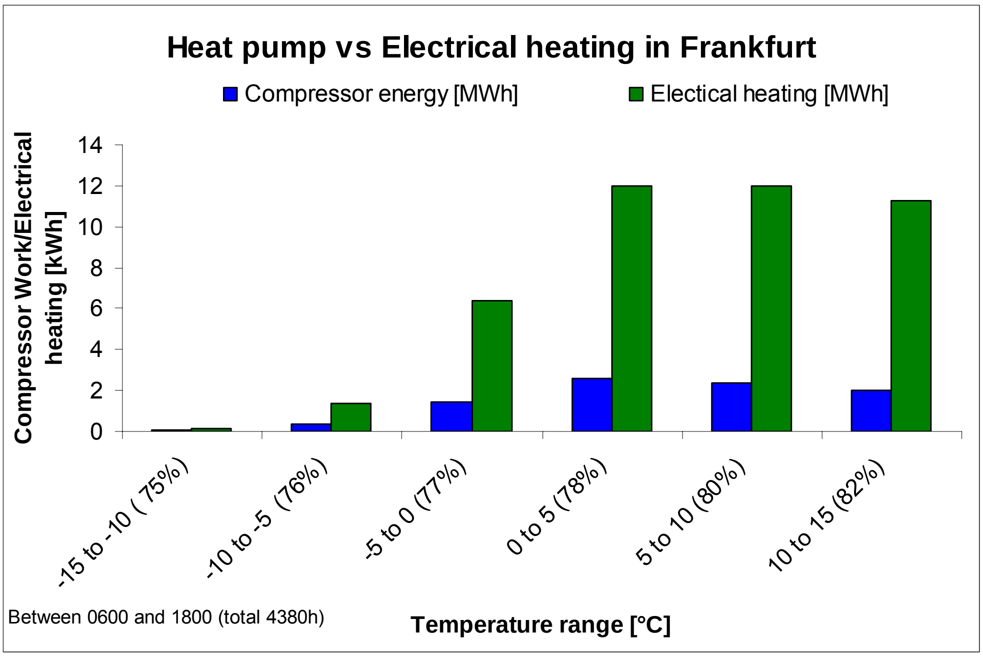


Figure E. 5 Potential savings of using CO_2 heat pump in stead of electrical heating in Frankfurt.

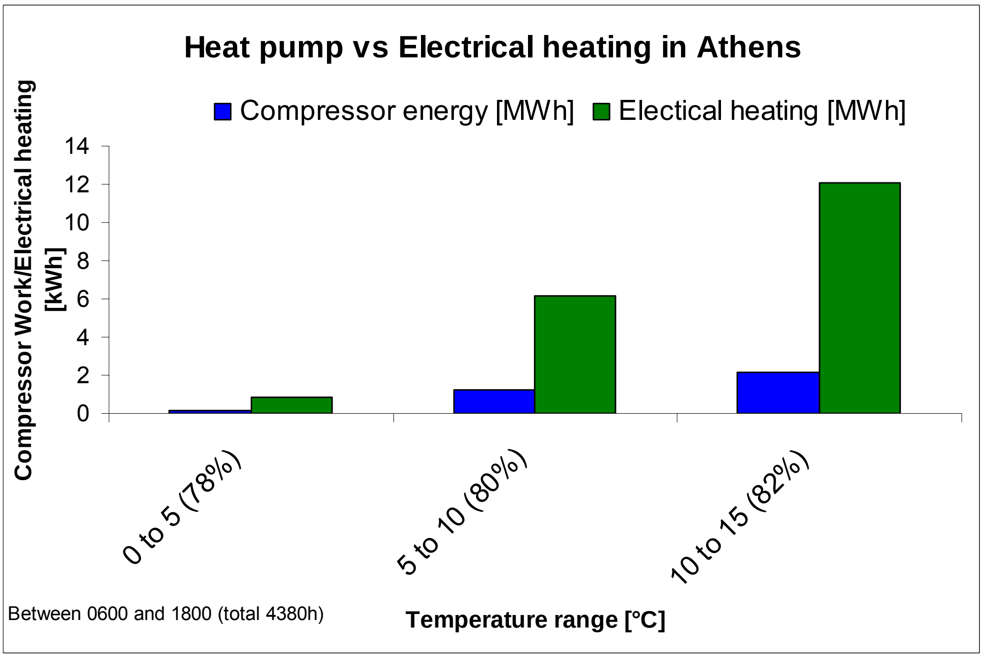


Figure E. 6 Potential savings of using CO_2 heat pump in stead of electrical heating in Athens.

Athens is in a warmer climate than Frankfurt and Oslo. Low ambient temperatures occur seldom and never below zero degrees. The potential heat pump savings is therefore not that big. However, when heating is needed the heat pump does not have to deal with extreme temperature lifts so the COP of the heat pump will be high.

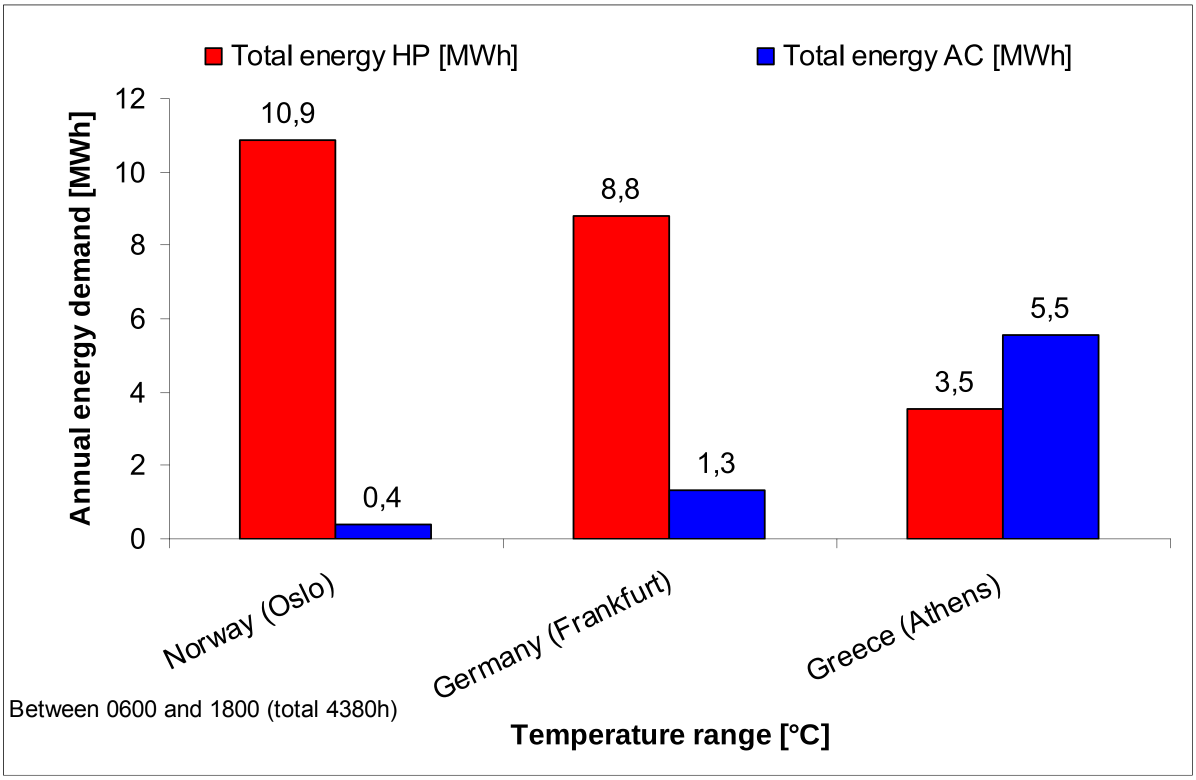


Figure E. 7 Total energy demand of heating and cooling for the CO_2 heat pump according to HXsim/ProII cooling mode and heating mode simulations.

The weather statistics for Oslo for the calculated time of the day (0600 to 1800) show that the train will need cooling 3% of the time and heating 83% the time. This heating should be done by a heat pump and not with electrical heating as today. Results of the computer simulation shows that the annual energy consumption of heating the train will be reduced by 78 % if the designed CO_2 heat pump is used in stead of electrical heating. NSB have a goal to reduce its annual energy consumption by 60 GWh (Johnsen, 2008) to use CO_2 heat pump would contribute to reach the goal.

F CONCLUTION

R134a is today the most common fluid in mobile air conditioning, also in the railway sector where 75 % of the total refrigerant is R134a. All refrigerants system has leakages. The average annual leakage from the railway sector is today 5%. This approximately corresponds to a green house gas emission at 63 tonnes CO_2 eq. annually for a train with five coaches. CO_2 is a natural refrigerant when applied as refrigerant the GWP value is zero. Leakages from CO_2 system would not have any negative impact of the environment.

The system design of the heat pump does not have to be reversible although the system has to provide both cooling and heating. The deciding matter if the system operates in cooling or heating mode is which heat exchangers the treated air and which heat exchanger the exit air exchange heat with. The final design has two evaporation pressures and a two stage compression. With this design compared to a design with one evaporation pressure and a one stage compression the COP improved from 1,7 to 3,2 at ambient temperature of -40 °C.

Experimental results of a CO_2 expander showed an efficiency of 40 %. In the experimental work the expander did not improve the system. The reason for this was the bad operation conditions for the system. Calculations of different system design show that an expander with efficiency of 40% improves the COP from 8 % to 20 %, depending on the system design.

A railway coach need cooling when the ambient temperature is above 20 °C and heating below 15 °C. Norway is a country with cold climate and a scenario where a train is used from 0600 to 1800 every day throughout a year in Oslo, weather statistics shows that the train will need cooling 3% of the time and heating 83% the time. This heating should be done by a heat pump and not with electrical heating as today. Results of the computer simulation shows that the annual energy consumption of heating the train will be reduced by 78 % if the designed CO_2 heat pump is used in stead of electrical heating. Using the designed CO_2 heat pump in Norwegian trains will save the environment due to reduced green house gas emissions and due to reduced energy consumption for heating. NSB want to reduce their annual energy consumption by 60 GWh, applying a CO_2 heat pump system would contribute to reach the goal.

Further work

Further work is to convince NSB and politicians that Norway, as one of the wealthiest countries in the world, should be a number one country regarding to developing and use of new environmental friendly technologies. There is large saving potential using CO_2 heat pumps in trains and this can be one important step to reduce pollution and Green House Gas emissions. If Norway develop the technology and show that it can be applied in real life other countries will follow.

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Appendix A Excel/RnLib calculation model

To calculate on the system it have been made a model in Excel with use of RnLib a thermodynamic and transport properties of refrigerants and refrigerant mixtures computer Programme developed at NTNU-SINTEF.

General assumptions

- The work of the fans to distribute the air is not implemented in the model
- Pressure drop thorough the heat exchangers have been neglected

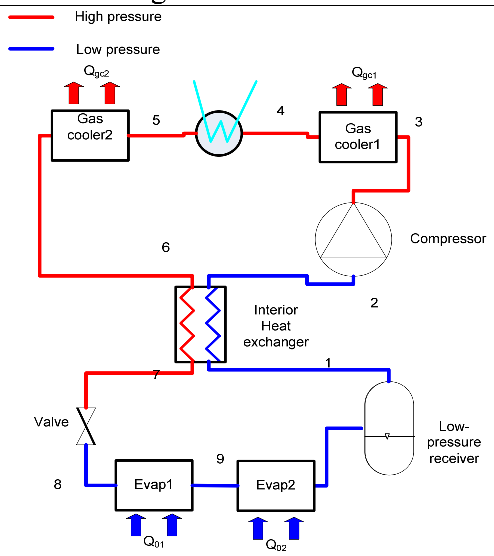
- The system circuit as figure to the left.
- The IHX is always counter flow

Air flow cooling mode

GC1	Fresh air
GC2	Exhaust air to exit
Evap1	Exhaust air to recycling
Evap2	Fresh air

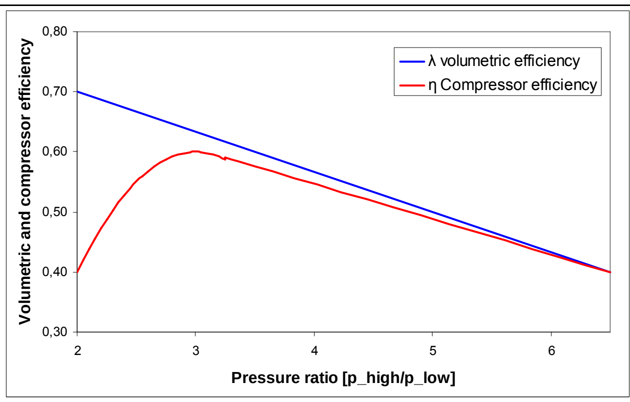
Air flow heating mode

GC1	Exhaust air to recycling
GC2	Fresh air
Evap1	Fresh air
Evap2	Exhaust air to exit



Appendix Figure 1 System circuit

- The total efficiency and the volumetric efficiency of the compressor are given in the figure to the left.



Appendix Figure 2 Compressor and volumetric efficiency

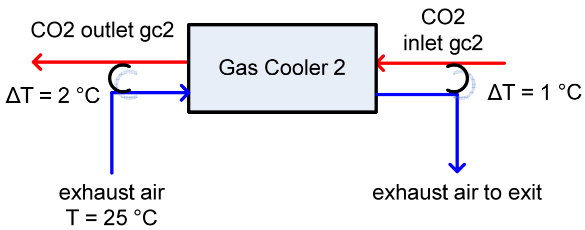
- Terminal efficiency of the internal heat exchanger 0,8
- Temperature inside the coach 22°C
- Vapour fraction out of the receiver 0,9 [kg_{vapour}/kg_{liquid}]
- Seats per coupe 70
- Amount of treated air (fresh air + recycled air) [7] 4600 m^3/h

- Fresh air streams according to descriptions in system specifications

Amtient temperature T	Fresh air rate
T< -20 °C	700 m^3/h
-20°C< T < -5 °C	1050 m^3/h
T > -5 °C	1400 m^3/h

Cooling mode

Exhaust air	25 °C
Treated air	15°C
Evaporators	Temperature difference between the ambient air and the evaporator temperature ΔT=10°C
Gas cooler 1	Fresh air cools down gas cooler 1 T_out_gc2 = t_ambient +5 [°C]
Gas Cooler 2	



Heating mode

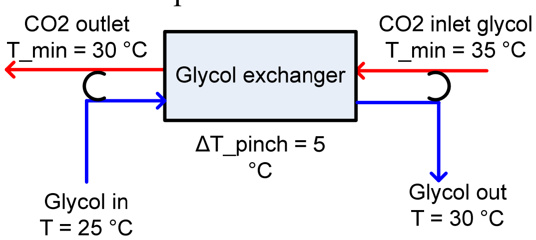
Exhaust air	20 °C
Treated air	30°C(average temperature of treated fresh- and recycled air)
Evaporation temperature	T_ambient – 10°C

Floor heating

$Glycol_{cold}$	25 °C
$Glycol_{hot}$	30°C
Q glycol[8]	3 kW

Gas cooler

High side pressure	Opperates under and over critical area. Minimum pressure when the cycle is transcritical is 80 bar
Gas cooler1	Maximum temperature out of compressor 160 °C. To avoid problems for the compressor lubricant
Glycol exchanger	



Gas cooler 2	Pinch temperature is set to 5 °C. When the cycle is transcritical the pinch point occur in the outlet. When the cycle is subcritical the pinch occur inside the exchanger.

Explanation cooling mode

The main task in the cooling mode is to find the optimum pressure for a given ambient temperature which gives the highest COP_c.

The procedure of a simulation is shown in the flow chart to the left. It is necessary to verify that the calculation model fulfil the specifications and assumptions.

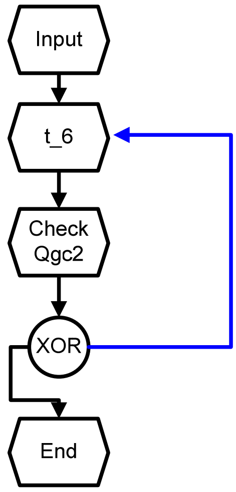
Ambient temperature and gas cooler pressure is input values.

CO_2 temperature out of gas cooler2 (t_6) have to be checked.

The maximum heating capacity of GC2 is defined by the exhaust air, equation 8, i.e. the value calculated from the CO_2 cycle, equation 7 can not be exceed the maximum capacity of the exhaust air.

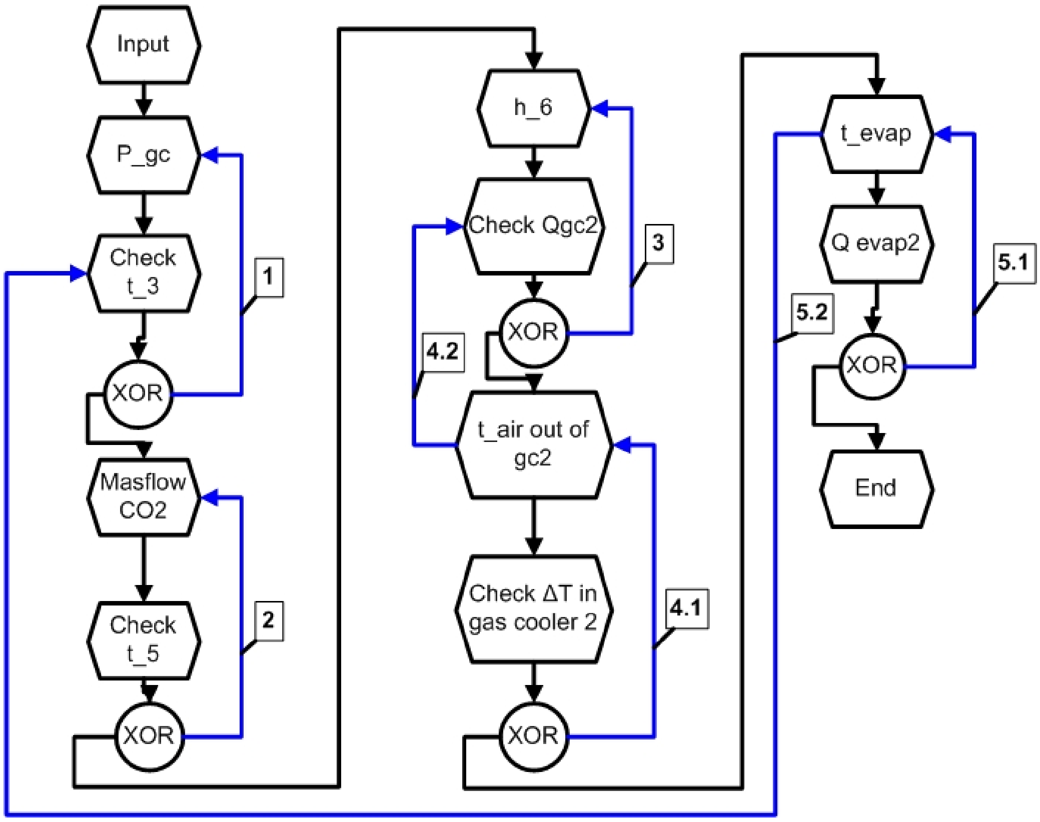
IF Q_gc2_CO_2 > Q_gc2_air the temperature out of the gas cooler is to low and t_6 have to be raised. T_6 have influence on the IHX and the CO_2 mass flow rate. Finding the correct t_6 is therefore an iteration process.

When correct t_6 is found the results simulation is finished. To find optimal pressure different pressures have to be tried.



Appendix Figure 3 Flow chart cooling mode

Explanation heating mode



Appendix Figure 4 Flow chart heating mode

The procedure of a simulation in heating mode is more complex than in cooling mode. The flow chart is valid when the cycle is subcritical. When the cycle is transcritical the pressure is set to minimum 80 bar. If the ΔT between fresh air and CO_2 in Gas cooler 2 is less than 5°C the pressure have to be increased such as ΔT = 5°C

Explanation of the flow chart:

1. Ambient temperature is input value. If the temperature out of the compressor (t_3) exceed 160°C the pressure are set to a value such as t_3 = 160 °C
2. If temperature out of the glycol exchanger (t_5) is lower than 30°C the mass flow of CO_2 have to be increased such as t_5 = 30 °C
3. Q_gc2_co2 - Q_gc2_air = ΔQgc2 If this value is not zero then a new value of h_6 are inserted such as ΔQgc2 = 0. Enthalpy is used in stead of temperature because gas cooler2 operates as a condenser when the cycle is subcritical.

4.1 Temperature of fresh air out of gas cooler2 (t_air_out_gc2) are given a value such as ΔT = 5 °C. This is valid as long as t_air_out_gc2 < 30°C

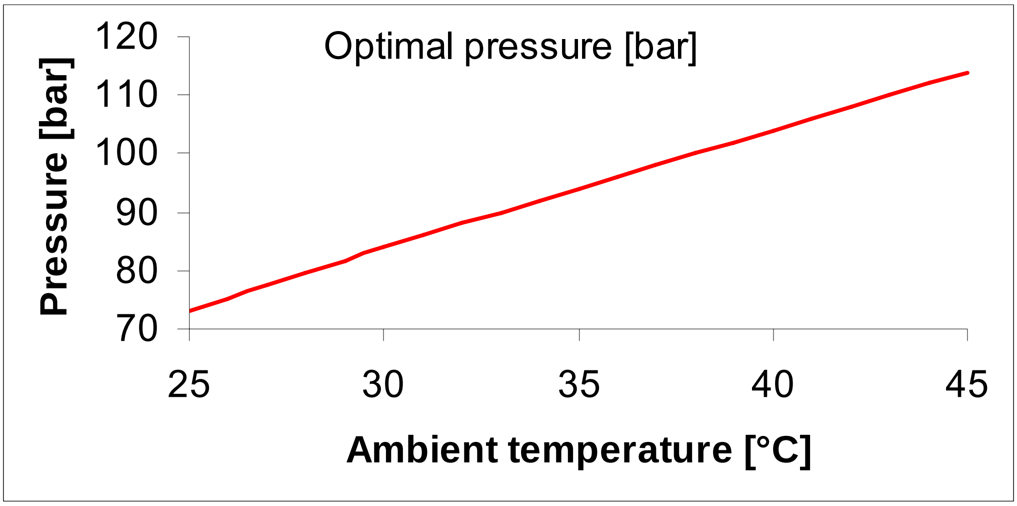
4.2 If t_air_out_gc2 is changed Point 3 must be repeated.

5.1 When the ambient temperature is decreasing there are a limit when cooling capacity of evaporator2 cover more than the total need of cooling. Evaporator1 then will be bypassed and the evaporation temperature set higher to a temperature such as Qevaporation = Qevap2

5.2 If the evaporation temperature is changed temperature out of compressor (t_3) changes and the iteration prosess move back to point 1.

Appendix Table 1 Compressor work influenced by the amount of fresh air

Persent fresh air	Evaporator 1	Evaporator 2	Compressor	
	Qevap [kW]	Qevap [kW]	m_co2 [kg/s]	Work [kW]
0 %	15,43	0,00	0,10	6,49
10 %	13,88	2,31	0,10	6,24
20 %	12,34	4,63	0,10	5,99
30 %	10,80	6,94	0,10	5,74
40 %	9,26	9,26	0,10	5,99
50 %	7,71	11,57	0,10	6,24
60 %	6,17	13,88	0,11	6,49
70 %	4,63	16,20	0,11	6,74
80 %	3,09	18,51	0,12	6,99
90 %	1,54	20,82	0,12	7,24
100 %	0,00	23,14	0,13	7,49



Appendix Figure 5 Optimal pressure for different ambient temperatures for air flows according to standard NS-EN 13129-1 (2002)

Excel/RnLib calculations with formulas in cooling mode

Input		
Refrigerant	R744	
t_ambient	30	[°C]
t_cupe	22	[°C]
Vapor fraction out of receiver	0,9	[kg/kg]
efficiency_IHX	0,8	
p_gc	75	[bar]
Air		
cp	1,006	[kJ/kgK]
density	1,2	[kg/m^3]
t_treated_air	15	[°C]
Fresh air flow	1400	[m^3/h]
Recirculated air flow	3200	[m^3/h]

Output		
t_evap	=t_treated_air-10	[°C]
p_evap	=r_psatliq_t(fluid;t_1)/100000	[bar]
t_inlet_gc2	=t_ambient+5	[°C]
t_out_gc2	=t_exh_air+2	[°C]
Isentropic efficiency compressor	=IF(p_ratio<=2,75;n_1;n_2)	
Volumetric efficiency compressor	=lamnda	
Air		
t_exh_air	=t_cupe+3	[°C]
Mas flow fresh	=fresh_air*density_air/3600	[kg/s]
Mas flow rec	=recirculated_air*density_air/3600	[kg/s]
Refrigerant cycle		
Cooling capacity	=SUM(B32:B33)	[kJ/kgCO2]
Cooling capacity evap1 (rec air)	=cooling_capacity1/masflow_co2	[kJ/kgCO2]
Cooling capacity evap2 (fresh air)	=cooling_capacity2/masflow_co2	[kJ/kgCO2]
Q_gc	=(h_3-h_6)	[kJ/kgCO2]
Q_gc1	=(h_3-h_4)	[kJ/kgCO2]
Q_gc2	=(h_4-h_6)	[kJ/kgCO2]
Work compressor	=h_3-h_2	[kJ/kgCO2]
mas flow CO2	=cooling_capacity/(h_10-h_8)	[kg/s]
COP	=(h_10-h_8)/(h_3-h_2)	
pressure ratio	=p_gc/p_evap	
Compressor volume flow	=masflow_co2*v_2/λ*3600	[m^3/h]

Cooling capacity	=SUM(D32:D33)	[kW]
Cooling capacity evap1 (rec air)	=cp_air*density_air*(recirculated_air*(t_exh_air-t_treated_air)/3600)	[kW]
Cooling capacity evap2 (fresh air)	=cp_air*density_air*(fresh_air*(t_ambient-t_treated_air)/3600)	[kW]
Q_gc	=masflow_co2*(h_3-h_6)	[kW]
Q_gc1	=masflow_co2*(h_3-h_4)	[kW]
Q_gc2	=masflow_co2*(h_4-h_6)	[kW]
Work compressor	=B39*masfow_co2	[kW]

IHX heat exchange		SI
h_7 when t_7=t_evap	=r_hgas_tp(fluid;t_evap;p_gc*100000)/1000	[kJ/kg]
h_2 when t_2=t_out_gc2	=r_hgas_tp(fluid;t_6;p_1*100000)/1000	[kJ/kg]
q_max_IHX_hot	=h_6-h_7min	[kJ/kg]
q_max_IHX_cold	=h_2max-h_1	[kJ/kg]
q_IHX	=MIN(q_max_IHX_hot;q_max_IHX_cold)*efficiency_IHX	[kJ/kg]

IHX (cold side)	In (1)	Out (2)	SI
Temperature	=t_evap	=r_tgas_hp(fluid;h_2*1000;p_2*100000)	[°C]
pressure	=p_evap	=p_1	[bar]
enthalpy	=r_hx_tpx(fluid;t_1;p_1*100000;x_1)/1000	=h_1+q_IHX	[kJ/kg]
entropy	=r_sx_tpx(fluid;t_1;p_1*100000;x_1)/1000	=r_sgas_tp(fluid;t_2;p_2*100000)/1000	[kJ/kgK]
vapor fraction	=x_1		[kg/kg]

Compressor	In(2)	Out(3)	SI
Isentropic temperature		=r_tgas_hp(fluid;h_3s*1000;p_3*100000)	[°C]
Temperature	=t_2	=r_tgas_hp(fluid;h_3*1000;p_3*100000)	[°C]
pressure	=p_2	=p_gc	[bar]
Isentropic enthalpy		=r_hgas_sp(fluid;s_2*1000;p_3*100000)/1000	[kJ/kg]
enthalpy	=h_2	=h_2+(h_3s-h_2)/n_is	[kJ/kg]
entropy	=s_2	=r_sgas_tp(fluid;t_3;p_3*100000)/1000	[kJ/kgK]
spesific volume	=r_vgas_tp(fluid;t_2;p_2*100000)		[m^3/kg]

Gas cooler 1	In(3)	Out(4)	SI
temperature	=t_3	=t_inlet_gc2	[°C]
pressure	=p_3	=p_3	[bar]
enthalpy	=h_3	=r_hgas_tp(fluid;t_4;p_4*100000)/1000	[kJ/kg]
entropy	=s_3	=r_sgas_tp(fluid;t_4;p_4*100000)/1000	[kJ/kgK]

Glycol exhanger	In(4)	Out(5)	SI

Nothing happens

Gas cooler 2	In(5)	Out(6)	SI
Temperature	=t_4	=t_6	[°C]
pressure	=p_4	=p_6	[bar]
enthalpy	=h_4	=h_6	[kJ/kg]
entropy	=s_4	=s_6	[kJ/kgK]

IHX (hot side)	In (6)	Out (7)	SI
Temperature	=t_out_gc2	=r_tgas_hp(fluid;h_7*1000;p_7*100000)	[°C]
pressure	=p_gc	=p_6	[bar]
enthalpy	=r_hgas_tp(fluid;t_6;p_6*100000)/1000	=h_6-q_IHX	[kJ/kg]
entropy	=r_sgas_tp(fluid;t_6;p_6*100000)/1000	=r_sgas_tp(fluid;t_7;p_7*100000)/1000	[kJ/kgK]

Exspansion valve	In (7)	Out (8)	SI
Temperature	=t_7	=t_evap	[°C]
pressure	=p_7	=p_evap	[bar]
enthalpy	=h_7	=h_7	[kJ/kg]
entropy	=s_7	=r_sx_th(fluid;t_8;h_8*1000)/1000	[kJ/kgK]

Evaporator1	In (8)	Out (9)	SI
Temperature	=t_8	=t_evap	[°C]
pressure	=p_8	=p_evap	[bar]
enthalpy	=h_8	=h_8+cooling_capacity1/masfow_co2	[kJ/kg]
entropy	=s_8	=r_sx_th(fluid;t_9;h_9*1000)/1000	[kJ/kgK]

Evaporator2	In (9)	Out (10)	SI
Temperature	=t_9	=t_1	[°C]
pressure	=p_9	=p_1	[bar]
enthalpy	=h_9	=h_1	[kJ/kg]
entropy	=s_9	=r_sx_th(fluid;t_10;h_10*1000)/1000	[kJ/kgK]

Reciever	In (10)	Out (1)	SI
Temperature	=t_10	=t_1	[°C]
pressure	=p_10	=p_1	[bar]
enthalpy	=h_10	=h_1	[kJ/kg]
entropy	=s_10	=s_1	[kJ/kgK]

Excel calculations with formulas in heating mode

Input		
Refrigerant	R744	
t_ambient	5	[°C]
t_cupe	22	[°C]
Vapor fraction out of receiver	0,9	[kg/kg]
efficiency_IHX	0,8	
p_gc	80	[bar]
t_glycol_in	25	[°C]
t_glycol_out	30	[°C]
capasity glycol exchanger	3	[kW]
Air		
cp	1,006	[kJ/kgK]
density	1,2	[kg/m^3]
t_treated_air	30	[°C]
Treated air flow	4600	[m^3/h]
Amount of fresh air	=fresh_air/recirculated_air	
Fresh air flow	=IF(t_ambient<-20;700;IF(t_ambient<-5;1050;1400))	[m^3/h]
Recirculated air flow	=treated_air-fresh_air	[m^3/h]

Output		
p_critical	=r_pcrit(fluid)/100000	[bar]
t_critical	=r_tcrit(fluid)	[°C]
Condensing temperature	=IF(p_gc>p_critical;"over critic";r_tsatgas_p(fluid;p_gc*100000))	
t_evap	=t_ambient-10	[°C]
p_evap	=r_psatgas_t(fluid;t_evap)/100000	[bar]
t_out_gc2	=IF(p_gc<p_critical;t_condensing;t_ambient+5)	[°C]
Isentropic efficiency compressor	=IF(p_ratio<=2,75;n_1;n_2)	
Volumetric efficiency compressor	=lamnda	
Air		
t_exh_air	=t_cupe-2	[°C]
t_air_out_gc2	30	
t_air_out_gc1	=((m_fresh_air+m_recirculated_air)*t_treated_air-m_fresh_air*t_air_out_gc2)/m_recirculate	
Mas flow fresh	=fresh_air*density_air/3600	[kg/s]
Mas flow rec	=recirculated_air*density_air/3600	[kg/s]
Refrigerant cycle		
Q_gc	=(h_3-h_6)	[kJ/kgCO2]
Q_gc1	=heating_capacity_gc1/masflow_co2	[kJ/kgCO2]
Q_glycol	=heating_capacity_glycol/masflow_co2	
Q_gc2	=heating_capacity_gc2/masflow_co2	[kJ/kgCO2]
Cooling capacity	=h_1-h_8	[kJ/kgCO2]
Cooling capacity evap1 (fresh air)	=cooling_capacity1/masflow_co2	[kJ/kgCO2]
Cooling capacity evap2 (exhaust air)	=cooling_capacity2/masflow_co2	[kJ/kgCO2]
Work compressor	=h_3-h_2	[kJ/kgCO2]
mas flow CO2	=masflow_coe_calculated	[kg/s]
COP	=(h_3-h_6)/(h_3-h_2)	
pressure ratio	=p_gc/p_evap	
Compressor volume flow	=masflow_co2*v_2/λ*3600	[m^3/h]

Cooling and heating capacaties in kW

Q_gc	=capasity_glycol+(m_fresh_air+m_recirculated_air)*cp_air*t_treated_air-m_fresh_air*cp_ai	[kW]
Q_gc1	=m_recirculated_air*cp_air*(t_air_out_gc1-t_exh_air)	[kW]
Q_glycol	=capasity_glycol	[kW]
Q_gc2	=m_fresh_air*cp_air*(t_air_out_gc2-t_ambient)	[kW]
Cooling capacity	=(h_1-h_8)*masflow_co2	[kW]
Cooling capacity evap1 (fresh a	=cooling_capacity-cooling_capacity2	[kW]
Cooling capacity evap2 (exhau	=m_fresh_air*cp_air*(t_exh_air-(t_evap+5))	[kW]
Work compressor	=B51*masfow_co2	[kW]

IHX (cold side)	In (1)	Out (2)	SI
Temperature	=t_evap	=r_tgas_hp(fluid;h_2*1000;p_2*100000)	[°C]
pressure	=p_evap	=p_evap	[bar]
enthalpy	=r_hx_tpx(fluid;t_1;p_1*100000;x_1)/1000	=h_1+q_IHX	[kJ/kg]
entropy	=r_sx_th(fluid;t_1;h_1*1000)/1000	=r_sgas_tp(fluid;t_2;p_2*100000)/1000	[kJ/kgK]
vapor fraction	0,9		[kg/kg]

Compressor	In(2)	Out(3)	SI
Isentropic temperature		=r_tgas_sp(fluid;s_2*1000;p_gc*100000)	[°C]
Temperature	=t_2	=r_tgas_hp(fluid;h_3*1000;p_3*100000)	[°C]
pressure	=p_2	=p_gc	[bar]
Isentropic enthalpy		=r_hgas_tp(fluid;t_3s;p_3*100000)/1000	[kJ/kg]
enthalpy	=h_2	=(h_3s-h_2)/n_is+h_2	[kJ/kg]
entropy	=s_2	=r_sgas_tp(fluid;t_3;p_3*100000)/1000	[kJ/kgK]
spesific volume	=r_vgas_tp(fluid;t_2;p_2*100000)		[m^3/kg]

Gas cooler 1	In(3)	Out(4)	SI
temperature	=t_3	=r_tgas_hp(fluid;h_4*1000;p_4*100000)	[°C]
pressure	=p_3	=p_gc	[bar]
enthalpy	=h_3	=h_3-q_gc1	[kJ/kg]
entropy	=s_3	=r_sgas_tp(fluid;t_4;p_4*100000)/1000	[kJ/kgK]

Glycol exhanger	In(4)	Out(5)	SI
temperature	=t_4	=r_tgas_hp(fluid;h_5*1000;p_5*100000)	[°C]
pressure	=p_4	=p_gc	[bar]
enthalpy	=h_4	=h_4-q_glycol	[kJ/kg]
entropy	=s_4		[kJ/kgK]

Gas cooler 2	In(5)	Out(6)	SI
Temperature	=t_5	=t_6	[°C]
pressure	=p_5	=p_6	[bar]
enthalpy	=h_5	=h_6	[kJ/kg]
entropy	=s_5	=s_6	[kJ/kgK]

IHX (hot side)	In (6)	Out (7)	SI
Temperature	=t_out_gc2	=r_tgas_hp(fluid;h_7*1000;p_7*100000)	[°C]
pressure	=p_gc	=p_gc	[bar]
enthalpy	=r_hgas_tp(fluid;t_6;p_6*100000)/1000	=h_6-q_IHX	[kJ/kg]
entropy	=r_sgas_tp(fluid;t_6;p_6*100000)/1000	=r_sgas_tp(fluid;t_7;p_7*100000)/1000	[kJ/kgK]

Exspansion valve	In (7)	Out (8)	SI
Temperature	=t_7	=t_evap	[°C]
pressure	=p_7	=p_evap	[bar]
enthalpy	=h_7	=h_7	[kJ/kg]
entropy	=s_7	=r_sx_th(fluid;t_8;h_8*1000)/1000	[kJ/kgK]

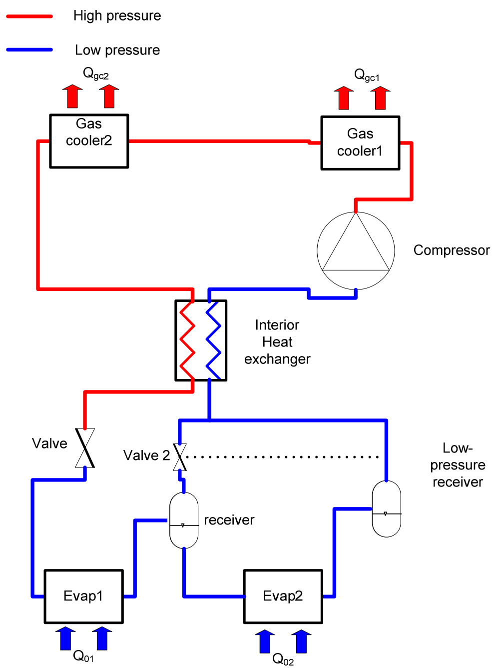
Evaporator1	In (8)	Out (9)	SI
Temperature	=t_8	=t_evap	[°C]
pressure	=p_8	=p_evap	[bar]
enthalpy	=h_8	=h_8+q_evap1	[kJ/kg]
entropy	=s_8	=r_sx_th(fluid;t_9;h_9*1000)/1000	[kJ/kgK]

Evaporator2	In (9)	Out (10)	SI
Temperature	=t_9	=t_evap	[°C]
pressure	=p_9	=p_evap	[bar]
enthalpy	=h_9	=h_9+q_evap2	[kJ/kg]
entropy	=s_9	=r_sx_th(fluid;t_10;h_10*1000)/1000	[kJ/kgK]

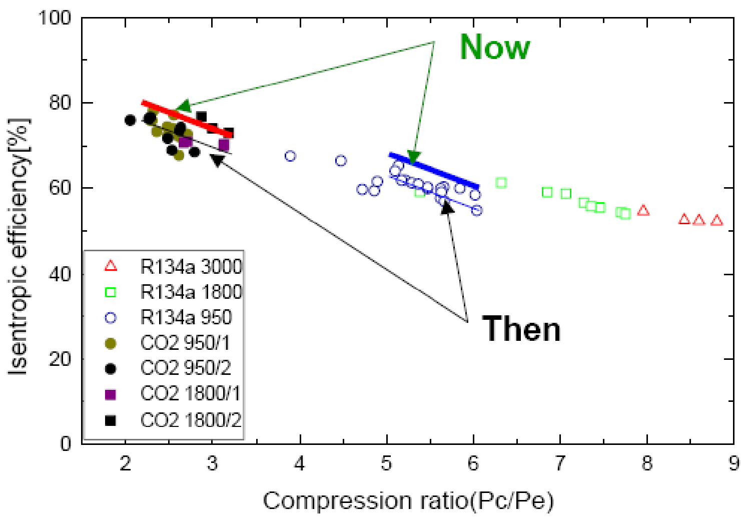
Reciever	In (10)	Out (1)	SI
Temperature	=t_10	=t_1	[°C]
pressure	=p_10	=p_1	[bar]
enthalpy	=h_10	=h_1	[kJ/kg]
entropy	=s_10	=s_1	[kJ/kgK]

Appendix B HXsim/ProII calculation model

Explanation of calculations of the HXsim/ProII model in cooling mode



Appendix Figure 6 System sketch of cooling mode



Appendix Figure 7 Isentropic efficiency of the CO_2 compressors are calculated from Hrnak researches on compressors. As the figure show CO_2 compressor have higher efficiencies than R134a compressors since the pressure ratio is lower.

Method of iteration process in cooling mode

Evaporator 2
The reason why Evap2 is the first exchanger which is simulated is that the inlet conditions are known. The CO_2 inlet is saturated liquid since it comes from the receiver. The pressure is evaporation pressure evap1 minus a assumed pressure drop on 0,5 bar in evap1 (This evaporation pressure is corrected later in the process).
Input values for the simulation
- Set ingoing air temperature to ambient temperature
- Set volume flow of air
- Set relative humility
- Set evaporator temperature to 4,5 °C (this value correspond to an assumption of a Δp = 0,5 bar in evap1)
- Set outlet vapour fraction to 0,95
- Set the pressure before throttling (found from Excel/RnLib model)
- Set temperature before throttling (use RnLib to find the temperature which gives saturated liquid at the evap2 inlet. This is done by first find the enthalpy at the Evap2 inlet. RnLib formula: r_hsatliq_p("CO_2", pressure evap2). Enthalpy in the expansion valve is constant. Temperature before throttling is found by formula: r_tgas_hp("CO_2", h_evap2_inlet;p_before throttling))
- Run simulation
- The most important output of this simulation CO_2 mass flow in Evap2. The CO_2 mass flow in Evap1 and Evap2 is not similar since the receiver is taking away the vapour of the Evap1 outlet.
- Mass flow evap2 = Mass flow evap1 * vapour fraction evap1 outlet

Evap1 (find operation point of evap1)
- Set temperature of air to recycled air temperature, 25 °C
- Set volume flow of air
- Set relative humidity of air
- Set evaporation to 5 °C
- Set temperature before throttling. (This is the hot side product temperature from the IHX first iteration value is value found in Excel/RnLib model)
- Set the pressure before throttling (Found in Excel/RnLib model)
- Run simulation. Iterate with changing evaporation outlet vapour fraction until the correct CO_2 mass flow is found in Evap1. (To reduce CO_2 mass flow increase value of the vapour fraction outlet and visa versa to increase CO_2 mass flow)
- Insert simulation values in excel sheet.

Gas cooler 1
- Set ingoing air temperature to ambient temperature
- Set volume flow of air
- Set relative humidity of air
- Set CO_2 mass flow (found in the simulation of Evap1)
- set inlet temperature of CO_2 (found in the Excel/RnLib model)
- Set the high pressure (found in the Excel/RnLib model)
- Insert simulation values in excel sheet.

Gas cooler 2

- Set temperature of air to recycled air temperature, 25 °C
- Set volume flow of air
- Set relative humidity of air
- Set CO_2 mass (found in the simulation of Evap1)
- Set inlet temperature of CO_2 (found in simulation of GC1)
- Set inlet pressure (Inlet pressure GC1- pressure drop GC1)
- Insert simulation values in excel sheet.

ProII

Insert values found from the HX simulation in the ProII model. Run simulation.

- Check if temperature into Gas cooler 1 calculated used in HXsim and calculated in ProII is simulilar, if not run new simulations in HXsim
- Check if temperature of hot side product temperature of IHX assumed in HXsim is similar as calculated in ProII model, if not run new simulation in HXsim

Ambient temperature 35 °C 0 humility

Evap2			
to	x	Massflow	p evap
	0,95	0,031	39,195

1)First simulation to find mass flow evap2

Evap1			
x	0,630	0,550	0,588
massflow	0,0667	0,0862	0,0756
massflow liquid	0,025	0,039	0,031
next gusee x	0,588		

t before throttling
11,27

2) Iteration prosess to find find evap1 massflow and vapour fraction corresponding to mass flow evap2

5) Check t before throttling is the same as calculated in pro2 (in this case start iteration was 11,27 and after iteration iteration with pro2 ending up with 17,2 °C

Massflow evap1	0,0756	må oppdateres

	EVAP1	Evap2	
vapour fraction in	**0,11**	**0,000**	
to	**5**	**4,95**	
p0	**39,69**	**39,64**	
p before throtling	**94**	**94**	
h_in	**537,9**	**513,97**	
t before throtling	17,2	0,00	throtling which give sat liq
Q	7924	**6583**	
massflow	0,0756	0,031	
pressure drop ref [bar]	0,0520	0,0620	
pressure drop ref [kPa]	5,20	6,2	
pressure drop air	310	13	
air face velocity	4,36	1,8	
air outlet temperature	14,87	15,3	
vap fraction out	0,587	0,979	
Massflow evap2	0,031		
next guess	0,979	0,974	

	Evap2				
x evap2	0,99	0,95	0,979		
m evap2	0,0308	0,032	0,031		

3) Insert correct value of p_evap2 (found p_evap1 - pressure drop evap1) Iterate to find vapour fraction such as the mass flow in evap2 again is 0,031 kg/s

Gas cooler 1 and 2

Mass flow gc1 and gc2	0,0756	
temperature gc1 in		
	GC1	**GC2**
tin	100	41,36
p_in	94	93,8173
t_out	41,36	27,78
Mass flow	0,0756	0,0756
Q	12566	5423
pressure drop ref [bar]	0,1827	0,0989
pressure drop ref [kPa]	18,27	9,89
dp air	312	281
air velosity	4	1,8

4) Check if t in gc1 is the same as calculated in pro2

Appendix Figure 8 Example from “hxsim_pro2_excel_sheet.xls” cooling mode

Result cooling

Appendix Table 2 Cooling relative humility of air 0 %

Ambient temperature °C		**25**	**30**	**35**	**40**	**45**
				Evaporator 1		
Evaporation capacity	[W]	7,583	7,885	7,924	8,015	8,201
Evaporation capacity proII	[W]	7,721	7,835	7,830	8,207	8,542
Mass Flow CO2	[kg/s]	0,064	0,073	0,076	0,081	0,088
Treated air temperature	[°C]	15,32	14,93	14,87	14,75	14,51
Vapour fraction in	[-]	0,11	0,13	0,11	0,07	0,07
Vapour fraction out	[-]	0,68	0,63	0,59	0,54	0,50
Evaporation pressure	[bar]	39,69	39,69	39,69	39,69	39,69
pressure drop CO2	[bar]	0,04	0,05	0,05	0,05	0,06
				Evaporator 2		
Evaporation capacity	[W]	4,235	5,510	6,583	7,675	8,713
Evaporation capacity proII	[W]	4,153	5,484	6,579	7,439	8,753
Mass Flow CO2	[kg/s]	0,020	0,027	0,031	0,037	0,044
Treated air temperature	[°C]	12,5	13,7	15,3	16,5	17,8
Vapour fraction out	[-]	0,95	0,95	0,98	0,95	0,92
Evaporation temperature	[°C]	4,96	4,95	4,95	4,95	4,94
pressure drop CO2	[bar]	0,03	0,05	0,06	0,08	0,08
				Gas Cooler 1		
Heating capacity	[W]	11,552	10,789	12,566	13,854	14,997
Heating capacity proII	[W]	11,726	12,181	12,954	14,051	15,559
CO2 inlet temperature	[°C]	83	88	100	112	120
CO2 inlet pressure	[bar]	80	84	94	104	114
CO2 outlet temperature	[°C]	33,5	36,4	41,4	46,1	51,0
Pressure drop CO2	[bar]	0,14	0,16	0,18	0,20	0,22
Pressure drop air	[Pa]	307	314	312	309	307
				Gas Cooler 2		
Heating capacity	[W]	2,722	4,125	5,423	6,481	7,802
Heating capacity proII	[W]	2,754	4,429	5,486	6,541	7,863
CO2 inlet temperature	[°C]	33,5	36,4	41,4	46,1	51,0
CO2 outlet temperature	[°C]	26,4	27,6	27,8	27,8	28,4
CO2 inlet pressure	[bar]	79,86	83,84	93,82	103,80	113,78
Pressure drop CO2	[bar]	0,07	0,08	0,10	0,11	0,13
				Cycle data		
Work compressor	[W]	2,600	3,288	4,020	4,950	6,120
Compressor efficiency	[%]	76,4	76	74,88	73,8	72,7
COP Pro2	[-]	4,56	4,05	3,58	3,16	2,82
COP Hxsim	[-]	4,55	4,07	3,61	3,17	2,76

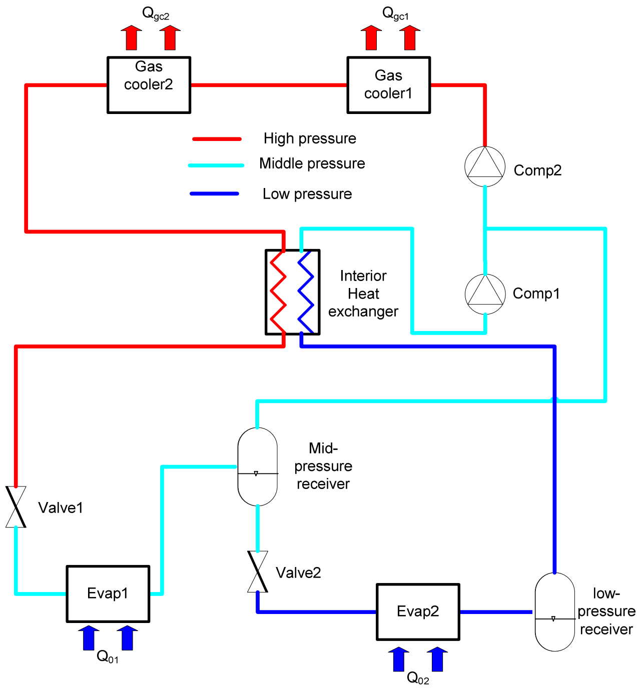
Appendix Table 3 Cooling relative humility of air 30 %

Ambient temperature °C		**25**	**30**	**35**	**40**	**45**
				Evaporator 1		
Evaporation capacity	[W]	7,485	8,053	8,490	8,941	9,292
Evaporation capacity proII	[W]	7,441	7,848	5,933	9,320	9,850
Mass Flow CO2	[kg/s]	0,062	0,078	0,096	0,120	0,141
Treated air temperature	[°C]	15,45	14,72	14,17	13,6	13,18
Vapour fraction in	[-]	0,11	0,13	0,13	0,14	0,14
Vapour fraction out	[-]	0,67	0,61	0,54	0,49	0,45
Evaporation pressure	[bar]	39,69	39,69	39,69	39,69	39,69
pressure drop CO2	[bar]	0,04	0,06	0,08	0,11	0,14
				Evaporator 2		
Evaporation capacity	[W]	4,238	6,301	8,972	12,013	15,322
Evaporation capacity proII	[W]	4,143	6,244	11,516	12,332	15,462
Mass Flow CO2	[kg/s]	0,020	0,031	0,044	0,061	0,078
Treated air temperature	[°C]	12,5	13,9	15,5	17,1	18,7
Vapour fraction out	[-]	0,95	0,95	0,95	0,93	0,92
Evaporation temperature	[°C]	4,96	4,94	4,92	4,89	4,86
pressure drop CO2	[bar]	0,03	0,06	0,11	0,19	0,27
				Gas Cooler 1		
Heating capacity	[W]	11,385	12,101	15,341	19,234	22,812
Heating capacity proII	[W]	11,506	13,062	16,099	20,373	24,559
CO2 inlet temperature	[°C]	83	91	106	120	133
CO2 inlet pressure	[bar]	80	84	94	104	114
CO2 outlet temperature	[°C]	33,3	36,6	42,4	48,5	54,6
Pressure drop CO2	[bar]	0,13	0,22	0,31	0,46	0,65
Pressure drop air	[Pa]	316	314	312	309	307
				Gas Cooler 2		
Heating capacity	[W]	2,584	4,529	6,537	8,957	7,396
Heating capacity proII	[W]	2,612	4,626	6,651	9,086	11,362
CO2 inlet temperature	[°C]	33,3	36,6	42,4	48,5	54,6
CO2 outlet temperature	[°C]	26,2	30,0	31,8	34,8	37,8
CO2 inlet pressure	[bar]	79,87	83,78	93,69	103,54	113,35
Pressure drop CO2	[bar]	0,07	0,12	0,17	0,26	0,35
				Cycle data		
Work compressor	[W]	2,530	3,590	5,300	7,810	10,600
Compressor efficiency	[%]	76,4	76	74,86	73,75	72,63
COP Pro2	[-]	4,58	3,92	3,29	2,77	2,38
COP Hxsim	[-]	4,63	4,00	3,29	2,68	2,32

Appendix Table 4 Cooling relative humility of air 30 %

Ambient temperature °C		**25**	**30**	**35**	**40**	**45**
				Evaporator 1		
Evaporation capacity	[W]	11,026	12,297	12,942	13,936	14,276
Evaporation capacity proII	[W]	11,246	12,925	12,548	16,802	14,913
Mass Flow CO2	[kg/s]	0,108	0,150	0,178	0,237	0,260
Treated air temperature	[°C]	15,54	14,89	14,6	14,28	14,07
Vapour fraction in	[-]	0,16	0,22	0,24	0,23	0,22
Vapour fraction out	[-]	0,64	0,62	0,59	0,55	0,51
Evaporation pressure	[bar]	39,69	39,69	39,69	39,69	39,69
pressure drop CO2	[bar]	0,11	0,21	0,27	0,66	0,49
				Evaporator 2		
Evaporation capacity	[W]	7,787	11,242	15,207	20,219	25,637
Evaporation capacity proII	[W]	7,715	11,670	15,425	20,515	25,986
Mass Flow CO2	[kg/s]	0,039	0,057	0,073	0,107	0,128
Treated air temperature	[°C]	12,8	14,4	16,3	17,7	20,3
Vapour fraction out	[-]	0,91	0,95	0,98	0,88	0,94
Evaporation temperature	[°C]	4,89	4,80	4,73	4,35	4,51
pressure drop CO2	[bar]	0,09	0,17	0,26	0,44	0,61
				Gas Cooler 1		
Heating capacity	[W]	15,629	18,780	23,861	31,108	35,842
Heating capacity proII	[W]	19,862	25,457	29,468	40,322	45,348
CO2 inlet temperature	[°C]	87	97	113	129	144
CO2 inlet pressure	[bar]	80	84	94	104	114
CO2 outlet temperature	[°C]	33,8	36,5	43,3	49,5	56,7
Pressure drop CO2	[bar]	0,43	0,88	1,17	2,04	2,37
Pressure drop air	[Pa]	316	314	312	309	307
				Gas Cooler 2		
Heating capacity	[W]	3,467	4,879	7,668	9,912	8,361
Heating capacity proII	[W]	3,697	6,320	8,936	14,044	16,515
CO2 inlet temperature	[°C]	33,8	36,5	43,3	49,5	56,7
CO2 outlet temperature	[°C]	30,1	33,5	38,0	40,9	44,7
CO2 inlet pressure	[bar]	79,57	83,12	92,83	101,96	111,63
Pressure drop CO2	[bar]	0,20	0,42	0,64	1,09	1,32
				Cycle data		
Work compressor	[W]	4,600	7,170	10,430	16,360	20,960
Compressor efficiency	[%]	76,4	76	74,77	73,63	72,41
COP Pro2	[-]	4,11	3,42	2,68	2,28	1,95
COP Hxsim	[-]	4,09	3,28	2,70	2,09	1,90

Explanation of calculations of the HXsim/ProII model in heating mode



Appendix Figure 9 System sketch of heating mode

Assumptions

- High pressure 80 bar
- Minimum pinch value in IHX 5 °C
- Efficiency compressor values according to Hrnjak (2006)
- Evap2 evaporation temperature= ambient temperature – 15 °C

Important

1. When ambient temperature changes check volume flow of recalculated and fresh air that they are correct according to standard NS-EN 13129-1 (2002)
2. In heating mode the gas coolers decides CO_2 mass flow

Gas cooler

1. Set a inlet temperature and a mass flow in gc1
2. Set the same flow in gc2 and use values from simulation of gc1 as input in gc2
3. Write down air temperatures out of the gas coolers and the mass flow in excel sheet
4. Iterate to find the CO_2 mass flow which give treated air temperature air around 30 °C (the temperature when treated air from GC1 and GC2 are mixed)
5. When mass flow is found insert values from the gas coolers in the excel sheet

proII

1. To find approximately inlet of evap1 run a simulation in the proII model where CO_2 mass flow and simulation results of GC1 and GC2 are inserted in the ProII model

Optimal evap1 pressure

1. Evap2 pressure is fixed while there are a optimal evaporation pressure for evap1 which give max COP. To find the optimal pressure for evap1, different temperatures for evap1 must be simulated.

Evap1

1. Use temperature and pressure out of IHX_h simulated in proII as input values in Evap1

2. iterate in excel sheet to find correct vapour fraction in evap1 which give correct CO_2 mass flow.

Evap2

1. Set pressure and temperature before throttling in HXsim
2. Iterate with different air face velocity and find value which gives vapour fraction 0,95 out of evap2.

proII

1. Insert values from simulation of evap1 and evap2 into the ProII model and run a simulation
2. Check that GC1 temperature is correct, if not a new simulation in HXsim with corrected GC1 temperature must be simulated.

Ambient temperature minus 30 °C 0 humility

Gas cooler 1 and 2

Mass flow gc1 and gc2	0,0320
temperature gc1 in	

1) CO2 value which give treated air temperature to 30 °C

	GC1	GC2
tin	143	21,14
p_in	80	79,96
t_out	21,14	-29,75
Mass flow	0,0320	0,032
Q	10335	3691
pressure drop ref [bar]	0,0351	0,0209
pressure drop ref [kPa]	3,51	2,09
t_out_air	37,16	-7,04
dp air	450	98
air velocity	4,88	0,88

2) Use output values from GC1 simulation as input to GC2

5) Check that GC1 inlet tempreature is correct according to Pro2

rec air temp	25
vanted treated air temp	30
next guess	#DIV/0!

Evap1 wanted mass flow

to	Massflow	p evap
	0,032	

3) Iteration process to find vapour fraction which give correct co2 mass flow in Evap1

	Evap1						
x	0,500	0,400	0,450				
massflow	0,028	0,036	0,032				
massflow liquid	0,014	0,021	0,018	0,000	0,000		
next gusee x	0,450						

Massflow evap1	0,0320	må oppdateres

	EVAP1	Evap2
vapour fraction in	**0**	**0,170**
to	**-15**	**-45,00**
p0	**22,92**	**8,33**
p before throtling	**79,8**	**22,9**
h_in	**467,5**	**407,68**
t before throtling	-35,0	-15,00
Q	5146	**4525**
massflow	0,032	0,018
pressure drop ref [bar]	0,0070	0,1085
pressure drop ref [kPa]	0,70	10,85
pressure drop air	31	25
air face velocity	0,95	2,21
air outlet temperature	-9,49	-38,62
vap fraction out	0,450	0,95

-15,00 throtling which give sat liq

liq fraction 0,550

6) Check that temperature before throttling evap1 is correct according to Pro2.

Husk å forandre på t og p before throttling evap2

Evap2			
phase vel	2	4	2,215
m evap2	0,0163	0,028	0,018

4) Iteration process to find which air face velocity wich give correct CO2 mass flow in Evap2, when the vapour fraction is fixed to 0,95

Massflow evap2	0,018
next guess	2,215

Appendix Figure 10 Example from “hxsim_pro2_excel_sheet.xls” heating mode

Result heating

Appendix Table 5 Optimal Evap1 temperature at ambient temperature - 30 °C

Evaporation temperature Evap1	CO2 temperature GC2 out	COP
-10	148	3,15
-15	**143**	**3,29**
-20	146	3,2

Appendix Table 6 Optimal Evap1 temperature at ambient temperature -10 °C

Evaporation temperature Evap1	CO2 temperature GC2 out	COP
-10	106	4,08
-5	**106**	**4,14**
0	109	3,99

Appendix Table 7. Optimal Evap1 temperature at ambient temperature 10 °C

Evaporation temperature Evap1	CO2 temperature GC2 out	COP
0	**80**	**5,32**
-5	81	4,93

Appendix Table 8 Optimal Evap1 temperature at ambient temperature 15 °C

Evaporation temperature Evap1	CO2 temperature GC2 out	COP
5	**75**	**5,84**
0	75	5,51

Appendix Table 9 Heating relative humility of air 0%

Ambient temperature °C		**15**	**10**	**0**	**-10**	**-20**	**-30**	**-40**
					Evaporator 1			
Evaporation capacity	[kW]	3,345	4,451	7,026	4,740	6,924	5,146	6,575
Evaporation capacity proII	[kW]	3,439	4,267	6,759	4,934	6,928	5,370	6,454
Mass Flow CO2	[kg/s]	0,050	0,044	0,045	0,040	0,046	0,032	0,030
Vapour fraction in	[-]	0,02	0,01	0,00	0,00	0,00	0,00	0,00
Vapour fraction out	[-]	0,33	0,44	0,62	0,38	0,46	0,45	0,60
Evaporation temperature	[°C]	5	0	-10	-5	-15	-15	-25
Evaporation pressure	[bar]	39,7	34,9	26,5	30,5	22,9	22,9	16,8
Pressure drop CO2	[bar]	0,01	0,02	0,03	0,01	0,09	0,01	0,01
Air temperature out	[°C]	10,45	7,33	0,06	1,87	-6	-9,49	-16,91
Air face velocity	[m/s]	1,91	1,91	1,91	1,43	1,43	0,95	0,95
Pressure drop air	[Pa]	87	87	87	57	57	31	31
					Evaporator 2			
Evaporation capacity	[kW]	7,046	5,274	4,217	5,783	6,351	4,525	3,288
Evaporation capacity proII	[kW]	6,944	5,431	4,211	5,784	6,309	4,437	3,268
Mass Flow CO2	[kg/s]	0,034	0,025	0,017	0,025	0,025	0,018	0,012
Vapour fraction in	[-]	0,03	0,03	0,02	0,15	0,13	0,17	0,18
Vapour fraction out	[-]	0,95	0,95	0,95	0,95	0,95	0,95	0,95
Evaporation temperature	[°C]	0	-5	-15	-25	-35	-45	-55
Evaporation pressure	[bar]	34,86	30,47	26,47	30,46	22,83	8,33	5,55
Pressure drop CO2	[bar]	0,09	0,06	0,05	0,12	0,16	0,11	0,08
Air temperature out	[°C]	7,3	1,91	-8,47	-17,98	-27,82	-38,62	-47,64
Air face velocity	[m/s]	4,55	3,23	2,37	3,33	3,59	2,21	1,44
Pressure drop air	[Pa]	89	45	25	89	63	25	11
					Gas Cooler 1			
Heating capacity	[kW]	11,273	10,428	11,438	10,991	13,516	10,335	9,844
Heating capacity proII	[kW]	11,272	10,429	11,450	11,005	13,540	10,343	9,909
CO2 inlet temperature	[°C]	75	80	94	106	126	143	147
CO2 inlet pressure	[bar]	80	80	80	80	80	80	80
Air temperature out	[°C]	36,43	35,88	37,88	36,89	40,71	37,16	36,86
Pressure drop CO2	[bar]	0,074	0,058	0,034	0,051	0,072	0,035	0,031
Pressure drop air	[Pa]	319	319	319	377	377	450	450
Air velosity	[m/s]	4	4	4	4,44	4,44	4,88	4,88
					Gas Cooler 2			
Heating capacity	[kW]	1,265	1,510	2,719	3,098	4,777	3,691	4,004
Heating capacity proII	[kW]	1,269	1,511	2,722	3,103	4,791	3,714	4,160
CO2 inlet temperature	[°C]	23,69	22,5	23,32	21,87	23,73	21,14	20,93
CO2 outlet temperature	[°C]	15,12	10,1	0,15	-9,8	-19,68	-29,75	-39,78
CO2 inlet pressure	[bar]	79,93	79,94	79,97	79,95	79,93	79,96	79,97
Air temperature out	[°C]	19,29	15,32	9,44	3,76	0,03	-7,04	-15,43
Pressure drop CO2	[bar]	0,04	0,03	0,03	0,03	0,04	0,02	0,02
Pressure drop air	[Pa]	283	285	262	185	185	98	98
Air velosity	[m/s]	1,75	1,75	1,75	1,31	1,31	0,88	0,88
					Cycle data			
Work compressor	[kW]	2,145	2,241	3,180	3,423	5,134	4,277	4,315
Compressor 1 efficiency	[%]	80,1	80,0	80,0	77,2	76,7	72,8	71,5
Compressor 1 efficiency	[%]	76,4	75,2	72,1	73,8	70,1	70,1	67,2
COP Pro2	[-]	5,84	5,32	4,46	4,12	3,57	3,29	3,26
COP HX sim	[-]	5,85	5,33	4,45	4,12	3,56	3,28	3,21

Appendix Table 10 Heating relative humility of air 30 %

Ambient temperature °C		**15**	**10**	**0**	**-10**	**-20**	**-30**	**-40**
					Evaporator 1			
Evaporation capacity	[kW]	3,360	4,622	7,459	5,044	7,746	6,080	8,020
Evaporation capacity proII	[kW]	3,476	4,761	7,389	5,243	7,551	6,222	8,070
Mass Flow CO2	[kg/s]	0,050	0,050	0,049	0,047	0,045	0,036	0,034
Vapour fraction in	[-]	0,02	0,04	0,00	0,00	0,00	0,00	0,00
Vapour fraction out	[-]	0,33	0,43	0,62	0,33	0,53	0,47	0,68
Evaporation temperature	[°C]	5	0	-10	-5	-15	-15	-25
Evaporation pressure	[bar]	39,7	34,9	26,5	30,5	22,9	22,9	16,8
Pressure drop CO2	[bar]	0,01	0,02	0,04	0,01	0,02	0,01	0,02
Air temperature out	[°C]	10,42	6,86	0,75	1,74	-4,24	-8,2	-15,88
Air face velocity	[m/s]	1,91	1,91	1,91	1,43	1,43	0,95	0,95
Pressure drop air	[Pa]	87	87	87	57	57	31	31
					Evaporator 2			
Evaporation capacity	[kW]	7,021	6,012	4,554	7,311	5,418	4,817	3,022
Evaporation capacity proII	[kW]	6,902	6,261	4,550	7,390	5,363	4,801	2,934
Mass Flow CO2	[kg/s]	0,033	0,028	0,018	0,032	0,021	0,019	0,011
Vapour fraction in	[-]	0,05	0,06	0,02	0,15	0,13	0,18	0,18
Vapour fraction out	[-]	0,95	0,95	0,95	0,95	0,95	0,95	0,95
Evaporation temperature	[°C]	0	-5	-15	-25	-35	-45	-55
Evaporation pressure	[bar]	34,86	30,47	26,46	30,46	22,90	8,33	5,55
Pressure drop CO2	[bar]	0,09	0,07	0,05	0,17	0,12	0,12	0,07
Air temperature out	[°C]	7,3	2,19	-8,35	-17,64	-28	-38,42	-49,33
Air face velocity	[m/s]	4,53	3,81	2,60	4,40	3,00	2,37	1,30
Pressure drop air	[Pa]	88	63	30	91	43	28	9
					Gas Cooler 1			
Heating capacity	[kW]	11,273	11,604	12,324	12,862	13,056	11,490	10,720
Heating capacity proII	[kW]	11,272	11,608	12,338	12,882	13,078	11,516	10,798
CO2 inlet temperature	[°C]	75	80	95	109	122	141	137
CO2 inlet pressure	[bar]	80	80	80	80	80	80	80
Air temperature out	[°C]	36,42	37,12	38,94	38,99	40,02	38,35	37,36
Pressure drop CO2	[bar]	0,074	0,075	0,077	0,071	0,068	0,044	0,039
Pressure drop air	[Pa]	319	319	319	377	377	450	450
Air velosity	[m/s]	4	4	4	4,44	4,44	4,88	4,88
					Gas Cooler 2			
Heating capacity	[kW]	1,265	1,952	3,152	3,861	4,622	4,182	4,568
Heating capacity proII	[kW]	1,270	1,955	3,461	3,868	4,634	4,208	4,749
CO2 inlet temperature	[°C]	23,69	23,98	24,5	23,4	23,37	21,45	21,26
CO2 outlet temperature	[°C]	15,12	10,16	0,2	-9,7	-19,7	-29,67	-39,7
CO2 inlet pressure	[bar]	79,93	79,92	79,92	79,93	79,93	79,96	79,96
Air temperature out	[°C]	19,29	16,63	10,68	6,42	-0,5	-4,75	-12,74
Pressure drop CO2	[bar]	0,04	0,04	0,04	0,04	0,03	0,02	0,02
Pressure drop air	[Pa]	283	285	287	181	185	98	98
Air velosity	[m/s]	1,75	1,75	1,75	1,31	1,31	0,88	0,88
					Cycle data			
Work compressor	[kW]	2,143	2,555	3,461	4,171	4,805	4,740	4,547
Compressor 1 efficiency	[%]	80,1	80,0	80,1	77,1	76,7	72,8	71,5
Compressor 1 efficiency	[%]	76,4	75,2	72,1	73,8	70,1	70,1	67,2
COP Pro2	[-]	5,85	5,31	4,47	4,01	3,68	3,31	3,41
COP HX sim	[-]	5,85	5,31	4,47	4,01	3,68	3,31	3,36

Appendix Table 11 Heating relative humility of air 60%

Ambient temperature °C		**15**	**10**	**0**	**-10**	**-20**	**-30**	**-40**
				Evaporator 1				
Evaporation capacity	[kW]	3,704	5,465	9,662	6,203	9,785	7,335	9,605
Evaporation capacity proII	[kW]	3,672	5,454	9,564	6,209	9,783	7,438	9,549
Mass Flow CO2	[kg/s]	0,050	0,045	0,047	0,043	0,050	0,037	0,038
Vapour fraction in	[-]	0,01	0,04	0,00	0,00	0,00	0,00	0,00
Vapour fraction out	[-]	0,35	0,56	0,85	0,48	0,65	0,59	0,73
Evaporation temperature	[°C]	5	0	-10	-5	-15	-15	-25
Evaporation pressure	[bar]	39,7	34,9	26,5	30,5	22,9	22,9	16,8
Pressure drop CO2	[bar]	0,02	0,02	0,04	0,01	0,03	0,01	0,03
Air temperature out	[°C]	11,29	9,19	3,19	5,09	-2,43	-5,83	-14,49
Air face velocity	[m/s]	1,91	1,91	1,91	1,43	1,43	0,95	0,95
Pressure drop air	[Pa]	87	87	87	57	57	31	31
				Evaporator 2				
Evaporation capacity	[kW]	6,714	4,413	1,726	5,300	4,583	3,898	2,779
Evaporation capacity proII	[kW]	6,684	4,370	1,736	5,250	4,492	3,853	2,736
Mass Flow CO2	[kg/s]	0,032	0,020	0,007	0,022	0,018	0,015	0,010
Vapour fraction in	[-]	0,04	0,07	0,01	0,15	0,12	0,17	0,18
Vapour fraction out	[-]	0,95	0,95	0,95	0,95	0,95	0,95	0,95
Evaporation temperature	[°C]	0	-5	-15	-25	-35	-45	-55
Evaporation pressure	[bar]	34,86	30,47	26,46	30,45	22,89	8,33	5,55
Pressure drop CO2	[bar]	0,08	0,05	0,01	0,10	0,09	0,08	0,07
Air temperature out	[°C]	6,92	1,05	-9,61	-18,33	-28,54	-38,94	-49,55
Air face velocity	[m/s]	3,00	1,80	0,70	2,65	2,27	1,81	1,16
Pressure drop air	[Pa]	38	15	2,57	33	25	16	7
				Gas Cooler 1				
Heating capacity	[kW]	11,273	10,488	11,630	11,478	13,941	11,365	11,656
Heating capacity proII	[kW]	11,272	10,448	11,641	11,492	14,436	11,395	11,742
CO2 inlet temperature	[°C]	75	77	90	101	116	131	131
CO2 inlet pressure	[bar]	80	80	80	80	80	80	80
Air temperature out	[°C]	36,41	35,69	37,82	37,1	40,61	37,74	38,02
Pressure drop CO2	[bar]	0,074	0,060	0,069	0,058	0,083	0,046	0,048
Pressure drop air	[Pa]	319	319	319	377	377	450	450
Air velosity	[m/s]	4	4	4	4,44	4,44	4,88	4,88
				Gas Cooler 2				
Heating capacity	[kW]	1,265	1,063	2,969	3,383	5,315	4,290	5,152
Heating capacity proII	[kW]	1,270	1,559	2,898	3,391	4,858	4,316	5,361
CO2 inlet temperature	[°C]	23,69	22,59	23,68	22,28	24,49	21,4	21,71
CO2 outlet temperature	[°C]	15,12	10,1	0,17	-9,8	-19,6	-29,64	-39,61
CO2 inlet pressure	[bar]	79,93	79,94	79,93	79,94	79,92	79,95	79,95
Air temperature out	[°C]	19,29	15,62	10,17	4,74	1,79	-4,28	-10,11
Pressure drop CO2	[bar]	0,04	0,03	0,04	0,03	0,04	0,03	0,03
Pressure drop air	[Pa]	283	285	287	185	185	98	98
Air velosity	[m/s]	1,75	1,75	1,75	1,31	1,31	0,88	0,88
				Cycle data				
Work compressor	[kW]	2,129	2,214	3,132	3,460	4,970	4,421	4,840
Compressor 1 efficiency	[%]	80,1	80,0	80,0	77,1	76,7	72,9	71,5
Compressor 1 efficiency	[%]	74,3	75,2	72,1	73,8	70,1	70,1	67,2
COP Pro2	[-]	5,89	5,42	4,64	4,3	3,88	3,55	3,53
COP HX sim	[-]	5,89	5,22	4,66	4,30	3,87	3,54	3,47

Appendix C Results and explanation of calculations for the CO_2system in operation over a year

	Temperature interval		Norway (Oslo) Hours in intervall	Compressor energy [MWh]	Electical heating [MWh]	Energy Saved [%]
Heating mode	below -20	< -20	0	0	0	0 %
	between -20 & -15	-20 to -15	29	0,13	0,50	74 %
	between -15 & -10	-15 to -10	126	0,54	2,13	75 %
	between -10 & -5	-10 to -5	350	1,39	5,72	76 %
	between -5 & 0	-5 to 0	703	2,54	11,06	77 %
	between 0 & 5	0 to 5	771	2,49	11,56	78 %
	between 5 & 10	5 to 10	735	2,04	10,30	80 %
	between 10 & 15	10 to 15	748	1,75	9,74	82 %
	between 15 & 20	15 to 20	721	1,41	8,59	84 %
Cooling mode	between 20 & 25	20 to 25	192	0,36		
	between 25 & 30	25 to 30	5	0,01		
	between 30 & 35	30 to 35	0	0,00		
	above 35	> 35	0	0,00		

correction (Nr. of night hours)	4380
Nr. of hours within the 'time-window'	4380

Total energy AC [MWh]	0,4
Total energy HP [MWh]	12
Total energy electrical heating [MWh]	60
Saved energy [M'Wh]	47

Part of the day, only	start at	end at
hour of the day	6	18

Appendix Figure 11 Temperatures are divided into intervals. The calculation is taken over a year and "start at" and "end at" is the time of the day the calculation is going to consider.

Qalculation of compressorwork, heating and cooling capacity and COP
Equations are found from the Hxsim/pro2 model

	Temperature interval	Compressor work [kW]	Heating/Cooling capacity [kW]
Heating	below -30	0,0193x + 5,319	0,0384x + 16,824
	between -30 & -20	0,0065x + 4,935	0,2006x + 21,69
	between -20 & -10	-0,0634x + 3,537	-0,0955x + 15,768
	between -10 & 0	-0,071x + 3,461	-0,1247x + 15,476
	between 0 & 10	-0,0906x + 3,461	-0,192x + 15,476
	between 10 & 15	-0,0824x + 3,379	-0,2036x + 15,592
	between 15 & 20	**-0,0824x + 3,379**	**-0,2036x + 15,592**
Cooling	between 20 & 25	**0,212x - 2,77**	**0,5262x - 1,432**
	between 25 & 30	0,212x - 2,77	0,5262x - 1,432
	between 30 & 35	0,342x - 6,67	0,6216x - 4,294
	between 35 & 40	0,502x - 12,27	0,6984x - 6,982
	above 40	0,558x - 14,51	0,732x - 8,326

Appendix Figure 12 From the HXsim/ProII simulation model equations for the compressor work and heat/cooling capacity is found for each temperature interval. The reason of the two red lines is that it is not sure that heating is needed in the temperature interval between 15 &

20 °C. The equation in this also assumed to be similar as the temperature interval above. It is the same reason why 20 & 25 is red. Is not sure that whole this interval needs cooling, and equation is assumed to be similar as the temperature interval 25 to 30 °C.

Temp	helping column	Compressor work [kWh]	helping column	heating/cooling capacity	COP	Need of electricity whitout heat pump [kWh]	Tempearture if whitin time intervall
3,5	H	3,14	H	14,80	4,71	14,80	100
3,5	H	3,14	H	14,80	4,71	14,80	100
3,3	H	3,16	H	14,84	4,69	14,84	100
3	H	3,19	H	14,90	4,67	14,90	100
2,8	H	3,21	H	14,94	4,66	14,94	100
2,5	H	3,23	H	15,00	4,64	15,00	2,5
2,3	H	3,25	H	15,03	4,62	15,03	2,3
2	H	3,28	H	15,09	4,60	15,09	2
1,8	H	3,30	H	15,13	4,59	15,13	1,8

Appendix Figure 13. Temperatures for each hour throughout the year are given from Meteonorm database version 5.1. (2005). The column to the right, "temperature if within time interval", is to sort out what temperatures which are going to be included in the calculation or not according to the wanted time interval (Explanation 1), the helping column is there to reduce the amount of "if" sentences in the next column. For heating and cooling capacity formulas from the HXsim/ProII model is used (Explanation 2). If electrical heating, the electrical energy needed is similar to heating capacity (Explanation 3).

Explanation 1.
"Temperature if whitin time interval"
If the time is before desided start value is 100. If the time is after desided stop of mac the value is 200. If not he value of the cell is temperature for this hour.

Kvasi formula

```
IF "start at" and "end at" is the same number
   MAC is operating 24 hrs a day and result cell gives temperature for corresponding hour

ELSE

IF  hour is less than the time desided that the MAC will start running
      100
ELSE
     IF hour is more than the time desided that the MAC will stopp running
         200
      ELSE
          result cell gives temperature for corresponding hour
       END
END
```

Explanation 2.
Compressor work [kWh] (every cell have 1 hour time step therefore kWh)
The compressor work for a given temperature (assumed humidity 30%) is found by the simulation in the Hxsim/pro2 model. The equations for the compressor work is found in summary sheet.

Formula finds compressor work for every temperature

IF (ambient temperature for this hour is whithin [-40:-30]
compressor work formula for this temperatureinterval
Else (ambient temperature for this hou is whitin [-30:-20]
compressor work formula for tis temperatureinterval
|
|
|
END

The "helping collum" is there becaus it's not possible to have enough if sentenses in one cell to cover every temperature intervall.
If the temperature is in heating mode the "helping collum" cell value is H. if not formula for compressor work for this temperature intervall is used, and "Compressor work" cell than reffere to "helping collum" cell

Explanation 3.
Need of electicity if the train have to be heated by electicity in heating mode.
Electicity = heating capasity

If the temperature is in cooling mode value of the cell is 0

Appendix D Tables from expander calculations

Appendix Table 12 Percent of compressor work lost in throttling

	System A			System B		
ambient temperature [°C]	throttling loss [kJ/kg]	compressor work [kJ/kg]	[%] of compressor work lost in throttling	throttling loss [kJ/kg]	compressor work [kJ/kg]	[%] of compressor work lost in throttling
25	10,3	37,1	27,7 %	6,7	47,2	14,3 %
26	11,0	38,5	28,7 %	7,4	48,4	15,2 %
27	11,7	40,3	29,0 %	7,8	50,9	15,4 %
27,5	12,1	41,3	29,2 %	8,1	52,1	15,5 %
28	12,3	42,2	29,3 %	8,3	53,3	15,5 %
29	13,0	44,0	29,6 %	8,7	55,7	15,7 %
30	13,7	45,8	30,0 %	9,2	58,1	15,9 %
32,5	15,5	50,2	30,9 %	10,4	63,8	16,3 %
35	17,3	54,4	31,8 %	11,6	69,4	16,7 %
40	21,1	62,7	33,6 %	14,0	80,1	17,4 %
45	25,1	70,9	35,4 %	16,4	90,4	18,1 %
50	29,5	79,0	37,3 %	18,9	100,4	18,8 %

	System C1			C2		
ambient temperature [°C]	throttling loss [kJ/kg]	compressor work [kJ/kg]	[%] of compressor work lost in throttling	throttling loss [kJ/kg]	compressor work [kJ/kg]	[%] of compressor work lost in throttling
25	6,6	47,2	14,0 %	6,6	42,5	15,5 %
26	7,5	47,4	15,8 %	7,5	42,9	17,5 %
27	8,5	49,3	17,3 %	8,5	44,5	19,2 %
27,5	8,7	50,7	17,2 %	8,7	45,6	19,1 %
28	8,5	51,7	16,5 %	8,5	46,6	18,3 %
29	9,0	54,0	16,7 %	9,0	48,5	18,6 %
30	9,5	56,1	16,9 %	9,5	50,3	18,8 %
32,5	10,6	61,4	17,3 %	10,6	54,7	19,4 %
35	11,8	66,5	17,7 %	11,8	59,0	19,9 %
40	14,0	76,1	18,3 %	14,0	67,2	20,8 %
45	16,2	85,2	19,0 %	16,2	74,9	21,6 %
50	18,4	93,7	19,6 %	18,4	82,3	22,4 %

Appendix Table 13 COP improvement for System A, B, C1 and C2 when expansion valve is replaced by a expander with efficiency at 100%

Ambient temperature [°C]	A		B		C1		C2	
	COP	COP improvement	COP	COP improvement	COP	COP improvement	COP	COP improvement
25	5,34	41,94 %	4,38	18,76 %	4,82	20,28 %	4,94	20,71 %
26	5,10	44,49 %	4,20	20,31 %	4,66	23,49 %	4,76	23,91 %
27	4,83	45,61 %	3,98	20,77 %	4,50	26,64 %	4,58	27,03 %
27,5	4,71	46,18 %	3,88	21,00 %	4,41	26,54 %	4,48	26,92 %
30	4,19	48,97 %	3,47	22,18 %	3,92	26,74 %	3,99	27,10 %
32,5	3,78	52,15 %	3,14	23,30 %	3,58	28,33 %	3,63	28,66 %
35	3,46	55,51 %	2,87	24,42 %	3,30	29,86 %	3,34	30,16 %
40	2,96	62,73 %	2,46	26,63 %	2,86	32,67 %	2,89	32,95 %
45	2,60	70,70 %	2,17	28,79 %	2,54	35,54 %	2,56	35,81 %
50	2,32	79,59 %	1,94	31,00 %	2,28	38,43 %	2,30	38,71 %

Appendix Table 14 COP improvement for System A, B, C1 and C2 when expansion valve is replaced by a expander with efficiency at 75%

Ambient temperature [°C]	A		B		C1		C2	
	COP	COP improvement	COP	COP improvement	COP	COP improvement	COP	COP improvement
25	4,85	29,05 %	4,19	13,57 %	4,54	14,48 %	4,71	14,91 %
26	4,61	30,71 %	4,00	14,65 %	4,36	16,68 %	4,50	17,12 %
27	4,36	31,45 %	3,79	14,98 %	4,18	18,82 %	4,30	19,25 %
27,5	4,25	31,83 %	3,69	15,14 %	4,09	18,76 %	4,21	19,19 %
30	3,76	33,68 %	3,29	15,98 %	3,64	18,92 %	3,74	19,33 %
32,5	3,38	35,79 %	2,97	16,78 %	3,32	20,02 %	3,40	20,42 %
35	3,07	37,99 %	2,71	17,57 %	3,05	21,08 %	3,12	21,46 %
40	2,60	42,72 %	2,32	19,13 %	2,63	23,02 %	2,69	23,39 %
45	2,25	47,92 %	2,03	20,66 %	2,32	24,99 %	2,37	25,37 %
50	1,99	53,67 %	1,81	22,23 %	2,07	26,97 %	2,12	27,38 %

Appendix Table 15 COP improvement for System A, B, C1 and C2 when expansion valve is replaced by a expander with efficiency at 50%

Ambient temperature [°C]	A		B		C1		C2	
	COP	COP improvement	COP	COP improvement	COP	COP improvement	COP	COP improvement
25	4,44	17,99 %	4,01	8,73 %	4,27	9,19 %	4,49	9,56 %
26	4,20	18,96 %	3,82	9,41 %	4,07	10,52 %	4,26	10,92 %
27	3,96	19,40 %	3,61	9,62 %	3,88	11,81 %	4,05	12,22 %
27,5	3,85	19,63 %	3,52	9,72 %	3,80	11,78 %	3,96	12,18 %
30	3,40	20,74 %	3,13	10,25 %	3,38	11,90 %	3,52	12,29 %
32,5	3,03	21,99 %	2,82	10,75 %	3,07	12,58 %	3,19	12,96 %
35	2,74	23,30 %	2,56	11,25 %	2,81	13,23 %	2,92	13,60 %
40	2,29	26,09 %	2,18	12,24 %	2,41	14,41 %	2,50	14,80 %
45	1,97	29,13 %	1,90	13,20 %	2,11	15,61 %	2,19	16,03 %
50	1,71	32,50 %	1,69	14,19 %	1,87	16,81 %	1,95	17,27 %

Appendix Table 16 COP improvement for System A, B, C1 and C2 when expansion valve is replaced by a expander with efficiency at 25%

Ambient temperature [°C]	A		B		C1		C2	
	COP	COP improvement	COP	COP improvement	COP	COP improvement	COP	COP improvement
25	4,08	8,40 %	3,85	4,22 %	4,01	4,37 %	4,28	4,60 %
26	3,84	8,83 %	3,65	4,54 %	3,79	4,97 %	4,04	5,23 %
27	3,62	9,03 %	3,45	4,64 %	3,58	5,55 %	3,82	5,83 %
27,5	3,52	9,13 %	3,36	4,69 %	3,51	5,54 %	3,74	5,82 %
30	3,08	9,63 %	2,98	4,94 %	3,12	5,60 %	3,32	5,87 %
32,5	2,74	10,19 %	2,67	5,18 %	2,82	5,91 %	3,00	6,18 %
35	2,46	10,78 %	2,43	5,41 %	2,58	6,21 %	2,74	6,48 %
40	2,04	12,03 %	2,06	5,88 %	2,19	6,74 %	2,33	7,04 %
45	1,73	13,39 %	1,79	6,34 %	1,90	7,28 %	2,03	7,62 %
50	1,49	14,89 %	1,58	6,81 %	1,67	7,82 %	1,80	8,19 %

Appendix E Leakage rates of Railway Air Conditioning

Appendix Table 17. The Table is found in Final Report Maritime, Rail, and Aircraft Sector (Schwarz and Rhiemeier, 2007)
http://www.oekorecherche.de/english/berichte/volltext/Maritime-Rail-Aircraft.pdf

HFC emissions from the maritime, rail and aircraft sector 54

ANNEX V Railway Vehicles Survey at DB

Survey on Leakage Rates of Railroad Vehicle Air Conditioners at Deutsche Bahn, Plants for Technical Revision Wittenberge, Neumünster, Krefeld, and Kassel. Calender Year 2006.

Wittenberge *Doubledeck Coaches*	
refill kg	700
interval, years	4
cars	200
charge/car, kg	22
refrigerants, kg	4400
refrig x interval, kg	17600
annual leakrate	**4,0%**

Neumünster *IC Coaches*	
refill kg	1953
interval, years	4
IC/EC pass coaches	
cars	730
charge/car, kg	15
refrigerants, kg	10950
Dining cars	
cars	41
charge/car, kg	30
refrigerants, kg	1230
Sleeping cars	
cars	18
charge/car, kg	30
refrigerants, kg	540
Bistro cars	
cars	31
charge/car, kg	15
refrigerants, kg	465
all refrigerants, kg	13185
refrig x interval, kg	52740
annual leakrate	**3,7%**

Krefeld *Electrical Multiple Units*	
refill kg	810
ET 423,424,425	49
interval, years	6
cars	196
charge/car, kg	11
cabins	98
charge/cabin, kg	2
refrigerants, kg	2352
refrig x interval, kg	14112
ICE-2 (EMU-7)	6
interval, years	3,5
cars	42
charge/car, kg	15
cabins	12
charge/cabin, kg	5
refrigerants, kg	690
refrig x interval, kg	2415
ICE-T	10
interval, years	3,5
centre cars	48
charge/car, kg	15
end cars	20
charge/car, kg	7,5
refrigerants, kg	870
refrig x interval, kg	3045
all refrigerants, kg	3912
refrig x interval, kg	19572
annual leakrate	**4,1%**

Kassel I *DMU compact AC*	
refill kg	103
VT 612 (DMU-2)	
interval, years	4
cars	92
charge/car, kg	10
cabins	92
charge/cabin, kg	2
refrigerants, kg	1104
refrig x interval, kg	4416
annual leakrate	**2,3%**

Kassel II *DMU open compressor*	
refill kg	579,5
interval, years	7
VT 643/VT644	
cars	56
charge/car, kg	14
refrigerants, kg	784
VT 642	
cars	2
charge/car, kg	12
refrigerants, kg	24
VT 650	
cars	20
charge/car, kg	14
refrigerants, kg	280
all refrigerants, kg	1088
refrig x interval, kg	7616
annual leakrate	**7,6%**

Explanation: The plant's refill in the calendar year 2006 is related to the refrigerant quantity in all vehicles coming to inspection after a certain time interval (in years) since last revision. The annual leak rate is refill in kg divided by "refrigerants x interval, kg" (all vehicle refrigerants multiplied by the interval years).

The data were provided in March/April 2006, by DB Fahrzeuginstandhaltung GmbH: Jürgen Wicklein and Robert Gründl (Werk Kassel), Helmut Schröers (Werk Krefeld), Ralf Meinsen (Werk Neumünster), Klaus Letto (Wittenberge).