Norwegian University of
Science and Technology

Air reversing CO2 heat pumps

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Master of Science in Energy and Environment
Submission date: July 2010
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Problem Description

1.Litterature survey of the opportunities for use of reversible R744 heat pumps in the transport sector
2. Test out other simulation tools that PRO/II, e.g. Modelica for further investigations of R744 heat pumping circuits applying an ejector for expansion work recovery
3. Investigate possibilities of applying the system layout of mobile HVAC systems for other applications than train; e.g. buses and also heating and cooling of buildings
4. Look into possibilities to integrate additional heating systems in existing designs
5. Find the most appropriate type of heat exchangers for the air reversing R744 system based on performance requirements, space restrictions and economy.
6. Develop a tool for calculation of annual energy consumption for the air reversing R744 system for different climate (heating and cooling mode)
7. Make a draft for a journal paper
8. Suggest improvements for further work

Assignment given: 16. February 2010
Supervisor: Trygve Magne Eikevik, EPT

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NTNU

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and Process Engineering

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MASTER THESIS

for

Stud.techn. **Hanne Elisabeth Andreassen**

Spring 2010

Air reversing CO_2 heat pumps

CO_2 varmepumper med reversibel luftstrøm

Background and objective

Existing Air Conditioning systems use mainly R134a as refrigerant. The global warming potential for this fluid is 1410 times bigger than for carbon dioxide (CO_2) a sustainable alternative refrigerant called R744. There are posibilities for large green house gas (GHG) emission savings if new air condition systems are developed and R744 is used as working fluid. The objective for the master thesis is to evaluate the potential of applying R744 in air reversing heat pumping systems mainly for public conveyance sector and to investigate other potential areas of application.

The following questions should be considered in the master thesis:

1. Literature survey of the opportunities for use of reversible R744 heat pumps in the transport sector.
2. Test out other simulation tools than PRO/II, e.g. Modelica for further investigations of R744 heat pumping circuits applying an ejector for expansion work recovery.
3. Investigate possibilities of applying the system layout of mobile HVAC systems for other applications than train, e.g. busses and also heating and cooling of buildings.
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Within 14 days of receiving the written text on the diploma thesis, the candidate shall submit a research plan for his project to the department.

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

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

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

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

One – 1 complete original of the thesis shall be submitted to the authority that handed out the set subject. (A short summary including the author's name and the title of the thesis should also be submitted, for use as reference in journals (max. 1 page with double spacing)).
Two – 2 – copies of the thesis shall be submitted to the Department. Upon request, additional copies shall be submitted directly to research advisors/companies. A CD-ROM (Word format or corresponding) containing the thesis, and including the short summary, must also be submitted to the Department of Energy and Process Engineering

Department of Energy and Process Engineering, January 12th 2010

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Summary

CO_2 is an environmentally friendly refrigerant that has a no global warming potential when used as refrigerant. The current refrigerants used for air conditioning in public transport are chemical components, and have a high global warming impact. The possibility of replacing the conventional refrigerants by CO_2 is investigated for various parts of the transport sector.

A possible CO_2system for heating and cooling for public transport has been modeled and simulated. This system is a turntable prototype which is reversing the airflows to provide either cooling or heating. It has two gascoolers and two evaporators for separate treatment of ambient and recycled air. The plate is rotated 180̊ to switch from heating to cooling mode.

CO_2 has large potential for expansion work, due to the normally large throttling losses for high ambient temperatures. An ejector has therefore been implemented in the heat pump circuit. The turntable prototype is modeled by the simulation tool Modelica, and it is investigated how this ejector system adjusts to varying ambient conditions and power demand.

Weather data from the climate database Meteonorm was used as a basis for calculation of heating and cooling demand for a train compartment in five different cities, covering a variety of climates. A case study was performed based on an occupancy rate profile and operative hours of the heat pump for the compartment.

Simulations were performed of the air reversing heat pump based on the heating -and cooling demand calculations for the five cities. The COP values obtained are very positive, and they are in general higher for heating than cooling mode. The COP is depending on the load, and decreases with reduced occupancy rate. For cooling mode the COP ranged from 3.1 to 6. For heating mode it ranged from 8.2 to 2.8. With the occupancy rate chosen, the annual energy savings is about 80% for all the 5 cities of the study.

The fan work of the heat pumps was also included for 4 different operating modes. This reduced the total COP by between 10 to 40%, depending on heating and cooling power requirement and ambient conditions.

The fin and tube gas coolers that were used in the Modelica model were compared to a set of MPE gas coolers. The total mass of the heat exchangers was reduced by 50%. One would still have to weigh the reduced mass and increased LCCP performance against the increased investment cost of the MPE heat exchangers.

Sammendrag

CO_2 er et miljøvennlig kuldemedium som ikke har noen global oppvarmingsfaktor når den blir brukt som kuldemedium. Kuldemediene som blir brukt for offentlig transport i dag er kjemiske sammensetninger som har høy global oppvarmingsfaktor. Muligheten for å erstatte de konvensjonelle kuldemediene med CO_2 er undersøkt for flere områder innen offentlig transport.

En prototyp CO_2 varmepumpe for varming og kjøling i offentlig transport har blitt modellert og simulert. Dette systemet reverserer luftstrømmen for å bytte mellom varme og kjøle modus. Den har to gasskjølere og to fordampere for separat behandling av uteluft og resirkulert luft. Varmepumpen er montert på en plate som snurrer rundt 180 grader for å bytte mellom varme- og kjøle modus.

CO_2 har stort potensial for å gjenvinne ekspansjonsarbeid, grunnet det normalt store strupningstapet ved høye omgivelsestemperaturer. En ejektor har derfor blitt montert i systemet. Varmepumpen er modellert ved hjelp av simuleringsverktøyet Modelica, og det er undersøkt hvordan ejektorsystemet tilpasser seg varierende betingelser og last.

Statistiske klimadata fra klimadatabasen Meteonorm er blitt brukt som basis for å beregne varme- og kjølebehov for en togkupé i fem forskjellige byer. Beregningen av behovet ble utført ut i fra en antagelse om antall passasjerer time for time, samt antall operative timer for varmepumpen i togkupeen.

Simuleringer ble gjort ut i fra varme- og kjølebehovet for de fem byene. Resultatene for de oppnådde COP-verdiene er veldig positive. Og er generelt høyere for varmemodus enn kjølemodus. COP avhenger av lasten, og synker ved synkende antall passasjerer for det samme lastintervallet. I kjølemodus varierte COP fra 3.1 for maks last til 6 for minste last. For varmemodus varierte den fra 2.8 for maks last til 8.2 for minste last. Med det antatte antall passasjerer er energibesparelsen rundt 80 % for alle de fem byene.

Viftearbeidet ble også inkludert for fire situasjoner med forskjellig omgivelsestemperatur og last. Dette redusert oppnådd COP med mellom 10-40 % avhengig av kulde- eller varmebehov og omgivelsesbetingelser.

Tube-i-finne gasskjølerene som ble brukt i simuleringene i Modelica ble sammenliknet med MPE-gasskjølere. Den totale massen for varmevekslerene ble redusert med 50 %, men den reduserte vekten, og medfølgende redusert energiforbruk til frakt av systemet, må veies mot den økte innkjøpskostnaden for MPE-gasskjølere.

Preface

The possibilities of applying a CO_2 system for other areas of the transport sector are discussed in theory. Since there is more investigation being done for HVAC systems for cars, a large part of the literature study is based on experiences from that area of the transport sector.

Heat load calculations and simulations of performance of a reversible CO_2 heat pump are performed for a train compartment. Emphasis is put on moist air treatment and enthalpy differences, as this is important for the condition of the compartment air.

Several simulation tools have been used. Thus a major share of the work has consisted of learning how to use the simulation tools, their structure and limitations, in addition to modifying the system components.

Due to the simulation process and the development of the energy calculation tool being very time consuming, it was decided to only perform one case study, namely for a train compartment. The results generated are nevertheless also applicable for buses, as they have more or less the same size and operating conditions.

Parts of the theory are taken from the project work performed during autumn 2009, as the base of the topic is the same for both projects.

I want to give thanks to my academic supervisor Trygve M.Eikevik for constructive advice, and to my research supervisor Armin Hafner for his enthusiasm and for introducing me to the concept of ejectors. I would also give credits to Harald T. Walnum for help with the Modelica simulations. At last cannot be omitted the invaluable help from the 3rd floor coffee machine and the adjacent Anja's Snack bar.

Trondheim, 13.07.2010

Hanne Elisabeth Bø Andreassen

Contents

Abbreviations

CO_2: Carbon dioxide, R744

COPc/COPh: coefficient of performance for cooling and heating mode

Evap1: evaporator 1

Evap2: evaporator2

EPA: Environmental Protection Agency

EU European Union

GC1: gascooler1

GC2: gascooler 2

GWP: Global warming potential

HFC's : group of refrigerants containing hydrofluorocarbons

HVAC: heating ventilation and air conditioning

M-HVAC: mobile HVAC

IHX: internal heat exchanger

LCCP: Life Cycle Climate Performance

MAC: mobile air conditioning (only provides cooling)

VHC/VRC: volumetric heating capacity/volumetric refrigerating capacity

EV: Electric Vehicle

HEV: Hybrid Electric Vehicle

OEM: Original Equipment Manufacturer

MPE: Multiport Extruded Tubes

FT: fin-and-tube

1 Introduction

The purpose of this master thesis is to investigate the opportunities for replacing the traditional heating and air conditioning systems used for the transport sector with a reversible heat pump using CO_2 as refrigerant.

The performance of a prototype air reversing CO_2 heat pump for mobile purposes will be investigated for various climate conditions. The simulation software called Modelica will be used for these simulations. The prototype model has an ejector implemented in the circuit for expansion work recovery.

Based on hourly statistical values of climate data, a tool for calculating heating and cooling power demand can be developed. To investigate the performance of the reversible CO_2 HVAC system, a case study will be performed of a train compartment to find heating- and cooling power demand and annual energy consumption of the HVAC system in different climates.

Two different heat exchanger types available for the HVAC system will also be compared and evaluated in terms of economy, mass and LCCP.

The mobile air conditioning system is referred to as MAC, and the mobile system for providing of both heating and cooling is referred to as M-HVAC.

1.1 Background

Today the refrigerants used for mobile air conditioning (MAC) are chemical compositions called hydrofluorocarbons, or HFC's. They are green house gases, which means that they have a high global warming potential. The global warming potential, GWP, is defined as a measure of the future radiative effect of a substance, relative to the emission of the same amount of CO_2, and integrated over a chosen time horizon. The GWP is measured in kg CO_2 equivalents, and the time horizon is often chosen to a 100 years. (Hafner & Nekså, 2006)

CO_2 is a natural refrigerant, and is often titled R744 when used as refrigerant. CO_2 forms part of the natural greenhouse gases, and the existence of a certain amount of these gases in the atmosphere is vital for the climate on earth. When using CO_2 as refrigerant, the GWP is equal to 0, since it is taken from leftovers from the industry, and no new CO_2 is formed in the process.

The HFC R134a, which is the most used refrigerant for MAC's, has a GWP of 1410. The Kyoto Protocol from 1997 is an agreement between 187 countries to fight global warming, and it states a deep concern about the global warming potential of HFC's.

These concerns led in 2006 to the F-gas directive for the European Union, which contains regulations on safety and leakage control of HFC systems. In combination with this the MAC directive was created. (Morgenstern, 2008) This prohibits the installation of air conditioning systems in cars with a GWP higher than 150, starting from 2011. With this development it is likely that a similar regulation will be passed on HVAC for public transport as well in the near future. Consequently there is a need to find more environmentally friendly refrigerants that do not exceed the GWP limit. Among the possible options is CO_2, with a GWP of 1.

1.2 CO_2 as refrigerant - Thermodynamic properties

CO_2 has some thermodynamic properties that distinguish it from other refrigerants. First of all, by taking a look at the pressure-enthalpy diagram below, the most apparent feature is the critical temperature of only 31.1 C. For comparison R134a has critical temperature of 101.1C.

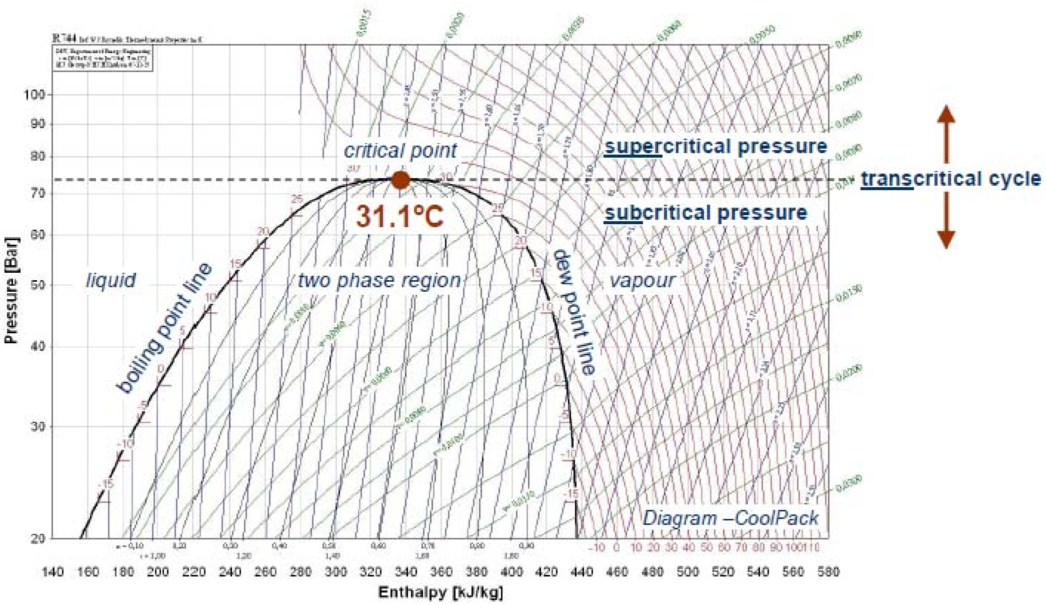


Figure 1-1: Pressure-enthalpy diagram for CO_2 (Stene, 2009)

This means that for air conditioning purposes, and often also heating purposes, the heat rejection will take place in the supercritical area, above the phase envelope. In the supercritical area the gas does not condensate. Instead it is cooled by gliding temperature in a gascooler. The operating pressures will be a lot higher than for other refrigerants. This gives smaller pressure ratio, which again leads to better efficiency of the compressor.

CO_2 has high vapour density in the gas phase, ρ_v. The necessary compressor volume is determined by the volumetric refrigerating capacity or the volumetric heating capacity for cooling and heating respectively. This is calculated by the formula:

$$VHC / VRC = \rho_v \cdot q_{HX} \quad (1.1)$$

With a high ρ_v, the VHC or VRC is high, and the following necessary compressor volume of CO_2 compressors will therefore be typically 15-20% the size of other refrigerants, which is very convenient for M-HVAC systems where space is often limited.

Because of the very low critical temperature of CO_2 compared to other refrigerants, the throttling losses are higher for CO_2 than for other refrigerants. This means that the throttling takes place further to the right in the phase envelope, and the vapour fraction after throttling is thus higher than desired. This problem increases when working with high ambient temperatures, and because of this it is important to get the temperature before throttling as low as possible in order to avoid high throttling loss. Therefore an internal heat exchanger, IHX, is standard equipment in CO_2 circuits. The hot CO_2 gas from the gascooler is then cooled by the cold gas from the suction line going into the compressor. A typical CO_2 circuit will then look like this:

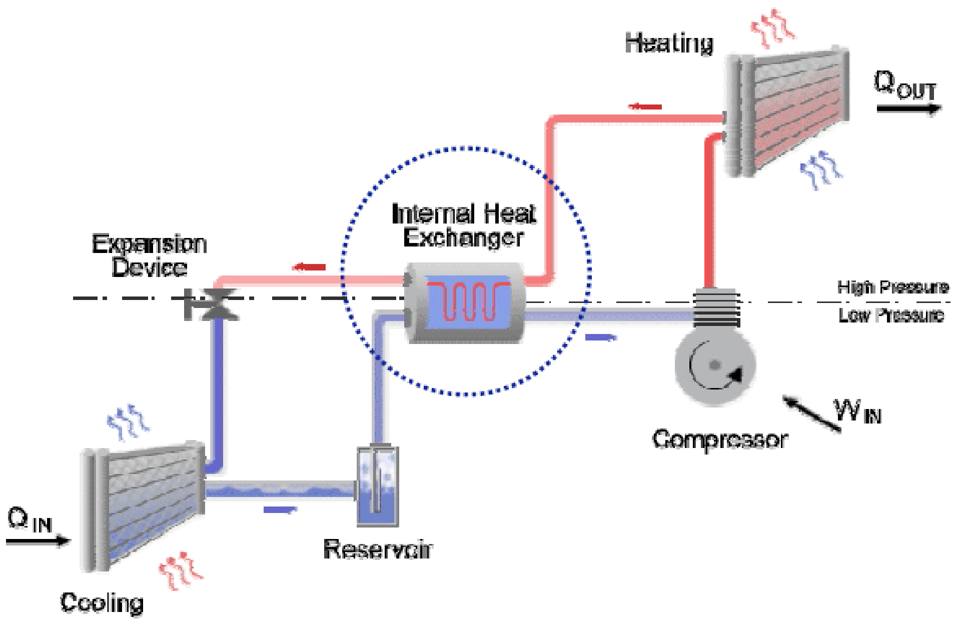


Figure 1-2: System layout of a simple CO_2 circuit with internal heat exchanger. (Stene, 2009)

The duty of the IHX influences the necessary area of the gascooler. A bigger IHX reduces necessary area of the gas cooler without need of increased high pressure. The IHX also ensures pure gas phase into the compressor.

In order to recover some of the throttling loss, an expander or ejector can be used instead of the regular expansion valve. This reduces the vapour fraction of the CO_2 entering the evaporator, and it is proved that CO_2 has more potential for expansion work recovery than e.g. R134a. During an experiment for a car air condition system with an ejector implemented, a COP improvement of 44% was achieved for CO_2, versus only 13% for R134a. (Hrnjak, 2006)

Optimal gas cooler pressure

The coefficient of performance for a heat pump cycle is determined by the formula:

$$COP = Q / W \tag{1.2}$$

Q equals the specific duty of the gascooler or the evaporator, and W is the compression work performed by the compressor.

The increase in compressor work is relatively linear for a certain increase in pressure. On the contrary, the increase in duty is not linear since the constant temperature lines increase their verticality as the pressure increases. This results in a maximum point for the COP at a certain gascooler pressure. Below is shown a P-h diagram for a typical transcritical cycle. Here the duty equals a-b or f-d for the gascooler and evaporator duty respectively.

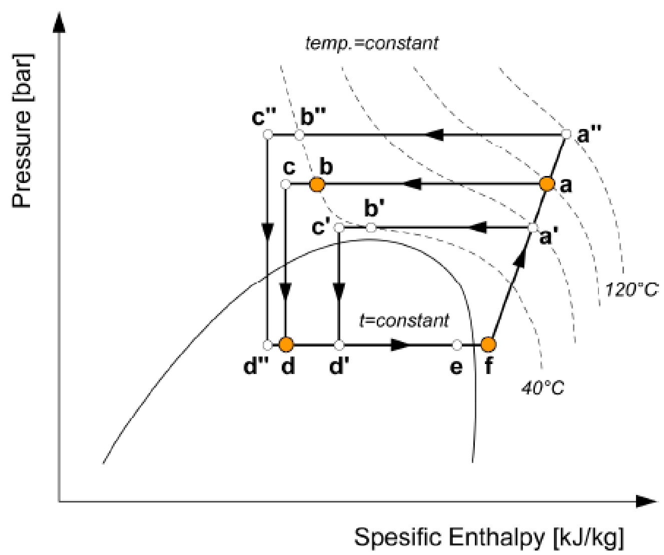


Figure 1-3: A typical transcritical cycle with varying gas cooler pressure. (Stene, 2009)

This feature is of great importance when deciding the operating pressures of a CO_2 heat pump in transcritical operation.

System COP versus cycle COP

The above mentioned COP is based on the compressor work with only the heat losses during compression included. To get the total power input to the compressor, the mechanical and process losses in the compression process has to be included, as well as the motor losses of the compressor engine.

To get the total power input to the cycle, the power to the fans of the heat exchangers also has to be included. The fan power is often considerable compared to the compression work, and increases with increased air volume ratio, or air face velocity.

1.3 Life Cycle Climate Performance for CO_2

Life cycle climate performance, or LCCP, is a measure of the overall global warming impact of equipment over its entire lifecycle. This includes equipment production and end of life. (Hafner & Nekså, 2006) This means that all greenhouse gas emissions from all stages of the lifetime of the equipment are included. This way it is not possible to "hide" greenhouse gas emissions by simply moving these to another part of the life cycle and keep them outside the boundaries of analysis.

For MAC's, the LCCP is divided into three parts representing the equivalent GHG emissions:

- Mass: related to energy used for transportation of the MAC.
- The direct impact: related to the refrigerant. This is direct leaks of refrigerant into the atmosphere, and is evaluated based on the GWP of each refrigerant and the quantity of refrigerant emitted.
- Indirect impact: related to the energy consumption due to MAC manufacturing, operation and end-of-life. It considers the carbon content of the fuel utilized in each process and during operation of the vehicle that uses the MAC.

Since the leakage rates from the MAC's are a lot higher in the automotive sector than in the railway sector, there has been a lot more research on LCCP for the automotive sector. An LCCP has yet to be performed for MAC's in public transport. Nevertheless, parallels can be drawn to that sector as well; for trains, that are run by electricity, the emission due to fuel consumption would be replaced by the emissions due to production of the electricity.

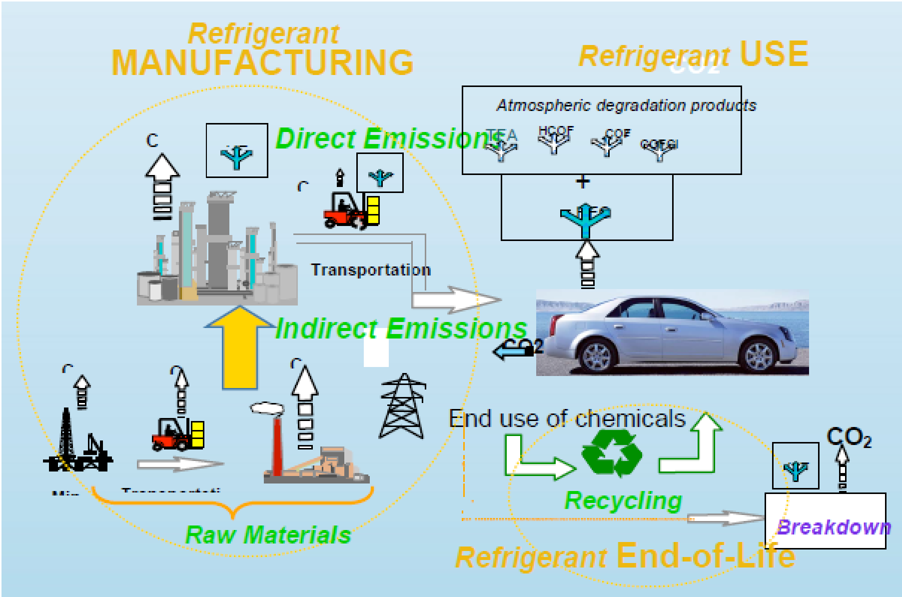


Figure 1-4: Model of LCCP for a Car (EPA, 2007)

As can be seen from the description of LCCP, an energy efficient MAC is important not only from a cost perspective but is also playing an important role in the indirect impact part of LCCP. As J.S.Brown et.al stated in their article comparing CO_2 and R134a:"*The refrigerant's environmental impact on climate change is determined not only by the refrigerant's trapping of infrared radiation, which is the direct effect indicated by the GWP. It also impacts by trapping of infrared radiation by CO_2 that is released upon burning of fossil fuels needed to power the air conditioning system. This is the indirect effect influenced by the AC system's efficiency. It is therefore important to prove that a CO_2 MAC can be as energy efficient as the conventional R134a."* (Brown et.al., 2002)

The SAE Alternate Refrigerant Cooperative Research Project was investigating the performance and efficiency of various alternate refrigerants compared to R134a. They concluded that CO_2 has better energy efficiency than R134a at most relevant temperatures. Only in extreme operating conditions CO_2 has a slightly lower COP, and this will not affect the annual energy use. If the design takes into account the special characteristics of CO_2, these systems can be made equally or more efficient than R134a. (SAE, 2004)

At higher temperatures capacity is the most important factor, and the CO_2 system can achieve a faster pulldown of temperature, due to higher maximum capacity. Below is a graphical illustration of the COP at varying inlet temperature.

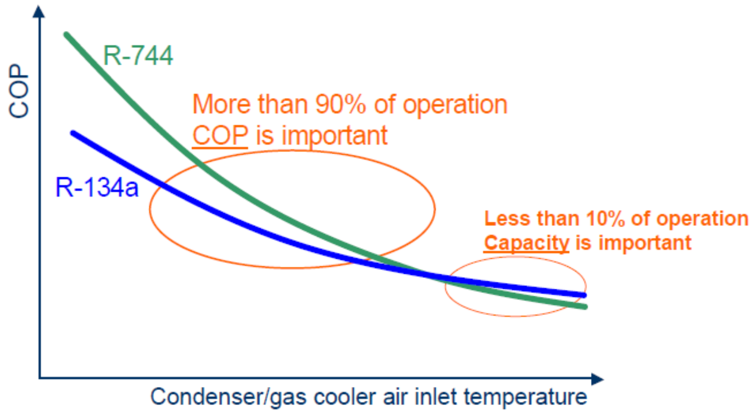


Figure 1-5: Typical energy efficiency at varying gas cooler air inlet temperature. (Hafner, Jacobsen, Nekså, & Pettersen, 2004)

It is clear that over the main part of the temperature range, and therefore also the main share of operating time, CO_2 has higher COP than R134a.

An LCCP analysis was performed based on performance and efficiency data for CO_2 and R134a automotive MAC. This shows 20-40 % reduced LCCP values for CO_2 compared to R134a. The least reduction was obtained for the hottest climate zones.

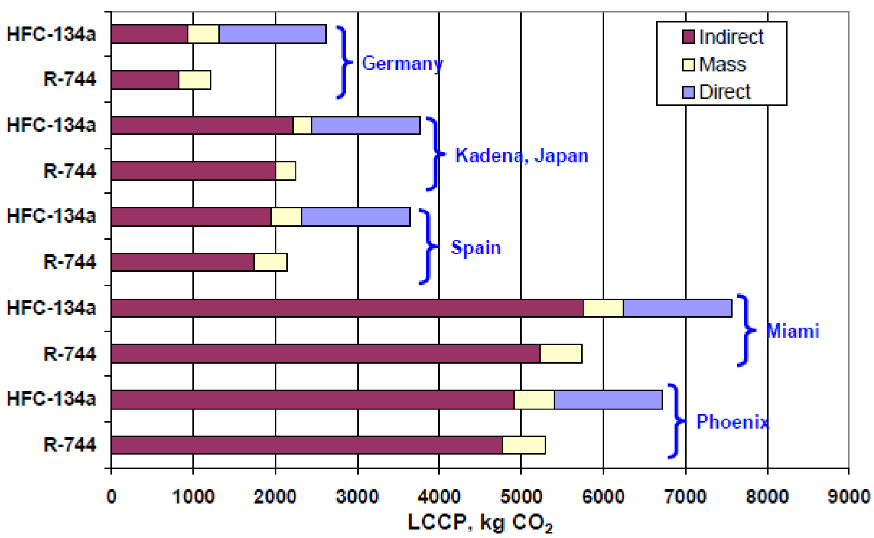


Figure 1-6: LCCP comparison between R134a and CO_2 for warm climates (Hafner, Jacobsen, Nekså, & Pettersen, 2004)

As can be seen from the figure, since CO_2 has a GWP of 0, the direct impact part is non-existing for the CO_2 system.

1.4 Reduction of greenhouse gas emissions by MAC

There are three ways to reduce greenhouse gas emissions from M-HVAC's in the transport sector:

1 .Reduction in energy consumption: Air conditioning in vehicles have a significant share of the total energy use. (Ebinger I. , 2008) The energy consumption of a vehicle can be reduced both by improving the energy efficiency and reducing the mass of the MAC (indirect impact of LCCP). Measures can also be taken on the vehicle itself to lower the amount of energy required for air-conditioning. (Ebinger & Morgenstern, 2008)

2. Reduction in refrigerant emission: HFC-emissions from MAC's are the largest source of HFC-emissions, and the resulting greenhouse gas emissions are the second largest within the refrigeration sector.

3. Research and development of new refrigerants with low or no global warming potential: In addition to CO_2, there are two chemical refrigerants that are being promoted by the industry. These are HFO 1234yf (GWP=100), and HFC 152a (GWP=140).

Both 1234yf and R152a are toxic and flammable, but they can use the equipment from previous refrigerants like R134a. R152a is accepted by several car producers as the replacement for R134a. (EPA, 2007) 1234yf is still under development, but the producer, DuPont Honeywell, allege it to have higher energy efficiency than CO_2 (Spatz & Minor, 2008) It has not yet been tested independently so meanwhile it only remains claims from the producer. (See appendix 11.3.6 for detailed information of the environmental properties of the refrigerants.)

CO_2 will need all new equipment due to higher pressure class, and will thus lead to higher investment costs. A benefit for CO_2 is that since it does not have any global warming potential, no refrigerant recovery is needed, and the service costs will be lower. Since CO_2 is freely available in the atmosphere, it costs a lot less than the chemically produced refrigerants. CO_2 refrigeration systems also weigh less due to the smaller components, and thus reduce the power requirement due to transporting the mass of the module.

Even though CO_2 is not flammable or toxic, it can be dangerous to humans in higher concentrations as it replaces air. CO_2 detectors will therefore be required inside the train compartments/buses. (Haukås, 2007)

1.5 HFC emissions from the transport sector

1.5.1 Railway sector

Today the main refrigerant for European railway air conditioning is R134a, in addition to some use of R407. In 2006 the total emissions from the air conditioning systems in the EU railway sector amounted to 56.7 metric tons, or 76.4 kilotons CO_2-equivalents. Because of the strong increase in air conditioning in Eastern Europe, this number is estimated to increase to the double by 2020 with the use of conventional refrigerants. (Schwarz & Rhiemeier, 2007)

Leakage rates of air-conditioning systems of rail vehicles are still relatively low compared to other transport sectors, and amounts to 5% per year for the majority of the vehicles (numbers from Deutsche Bahn). If all conventional MAC's were replaced by CO_2, the emissions would be reduced by almost 170 000 tonnes CO_2 -equivalents by 2020. Nevertheless it is only about 0.8% of the total estimated global warming emissions of the EU in 2020, considered a business as usual model. (Schwarz & Rhiemeier, 2007)

National railway operators are regulated by public law, and are obliged to follow the environmental protection laws. If a ban were put on the use of HFC's in MAC systems for the railway sector, like what has been done for cars, natural refrigerants like CO_2 could successfully be introduced for the railway sector as well. (Schwarz & Rhiemeier, 2007)

1.5.2 Automotive sector

The main refrigerant for the automotive sector is R134a, and the HFC-emissions from the automotive sector are the dominating source of refrigerant emissions to the atmosphere. A study performed for the German Environment Agency showed that average annual emission rate of R134a for cars were 10.2% of the refrigerant charge, including irregular and disposal emissions. (Schwarz, 2002)

The annual leakage rate for bus air condition is up to 50%of the charge, due to longer pipe distances and higher service frequency. (Repice & Schulz, 2004) Even though this seems drastic, 95% of all MAC's are passenger car air conditioners, so the total HFC- emissions from cars exceed by far the emissions from buses. (Schwarz, 2002)

As a consequence of the F-gas directive, the German Association of Automotive Industry (VDA) agreed to use CO_2 as refrigerant in future air conditioning systems. They were the first group of automotive companies to start this initiative, and the HFC emissions from cars in the European Union will thus gradually decrease as the old cars with conventional refrigerants are being scrapped. (Morgenstern, 2008) This directive is only for passenger cars, so buses, trucks etc. are not restricted by the regulation.

Nevertheless, for the global picture, the newly industrialized countries in e.g. Asia still have a rapid increase in instalment of R134 MAC's. Passing a law similar to the European F-gas directive could save several hundred metric tonnes of GHG emissions in China and India. (Hafner & Nekså, 2006)

1.6 Research and development for CO_2 M-HVAC

1.6.1 Railway

Traditionally, the heating of electrically driven trains is done by direct electrical heating or diesel if it is a diesel train. The air conditioning is provided by using conventional refrigerants, and mostly R134a and R407c. The frequency of air conditioning and heating installation is depending on climate zone.

As of 2010, there is no commercially installed CO_2 MAC- or M-HVAC systems in trains in Europe. Some prototype systems have been made, and since 2006 tests have been performed for the driver's cabin of a train. Compared to a conventional R134 system, the energy consumption seemed to be approximately 10% higher. The developers are also concerned about safety issues, like the displacement of atmospheric oxygen in case of leaks. (Schwarz & Rhiemeier, 2007)

Merak is one of the world leaders in the market of air conditioning for railroad vehicles, and they are working on new alternative refrigerants in order to help reach the aims defined in the Kyoto Protocol. In this respect, Merak has developed a HVAC prototype using the natural refrigerant R744. The system has been tested successfully at Merak's thermodynamic laboratory facilities, and validated with satisfactory performances. Nevertheless it is not installed in any operating public trains as of 2010. (Merak, 2010)

Because of the need for all special designed components for CO_2 systems for trains, the investment costs are considered to be up to 30% higher than with conventional refrigerants. Combined with the comparatively leak tight systems and small refrigerant charges, this gives high reduction costs per tonne reduced CO_2 -equivalent, and is per 2007 not found cost efficient. (Schwarz & Rhiemeier, 2007)

This cost will be much lower with the development of more efficient CO_2 technology. With the current progress of development for the commercial refrigeration sector, it is likely that the soon-to-be commercialized components from that sector can be modified and applied for M-HVAC systems in the future, thus reducing costs. (Nekså, 2010)

Different solutions for providing heating and cooling in public trains by using a reversible CO_2 heat pump have been developed at NTNU/Sintef. One of the solutions is a heat pump which is located on a rotatable plate, and it changes from cooling to heating mode by rotating the unit 180 degrees. The direction of the CO_2 flow is thus the same all the time. Simulations of this system configuration were done using the simulation tools RnLib, HxSim and PROII, and for both cooling and heating mode the results were promising. The annual energy savings were also calculated for three different climates, where using the CO_2 M-HVAC instead of electrical heating. For all three cities considered there was considerable energy saving potential. (Christensen 2009)

This prototype turntable system is investigated further in this work, and is explained in detail later on in this report, starting from chapter 4.

1.6.2 Bus

The heating of buses is normally performed by using engine waste heat in combination with an auxiliary fuel burner. The cooling is done by a conventional R134a air conditioning system. The MAC is often using the compressor situated in the motor compartment. (Leideck, 2009) With the MAC module situated on the bus roof, this leads to long tube stretches, and higher possibilities of refrigerant leaks.

Konvekta, a German thermo system supplier for public transport, has been the world's first to successfully install a refrigeration system with CO_2. They have installed a reversible CO_2 M-HVAC system in city buses various places in Germany in addition to Singapore. Even with the extremely hot and humid climate in Singapore, the CO_2 M-HVAC proved to be more efficient than a standard R134 system. After these custom made installations, Konvekta is now ready to do a series production of CO_2 M-HVAC's. (Fõrsterling, Tegethoff, & Sonnenkalb, 2009)

In addition to these physical tests, a simulation model was made to evaluate the thermal behaviour of the bus. It was proved that for heating the CO_2 heat pump system is much more efficient than the auxiliary fuel burner that is normally being used for buses. The following graph shows the comparison of performance of the CO_2 heat pump and the fuel burner:

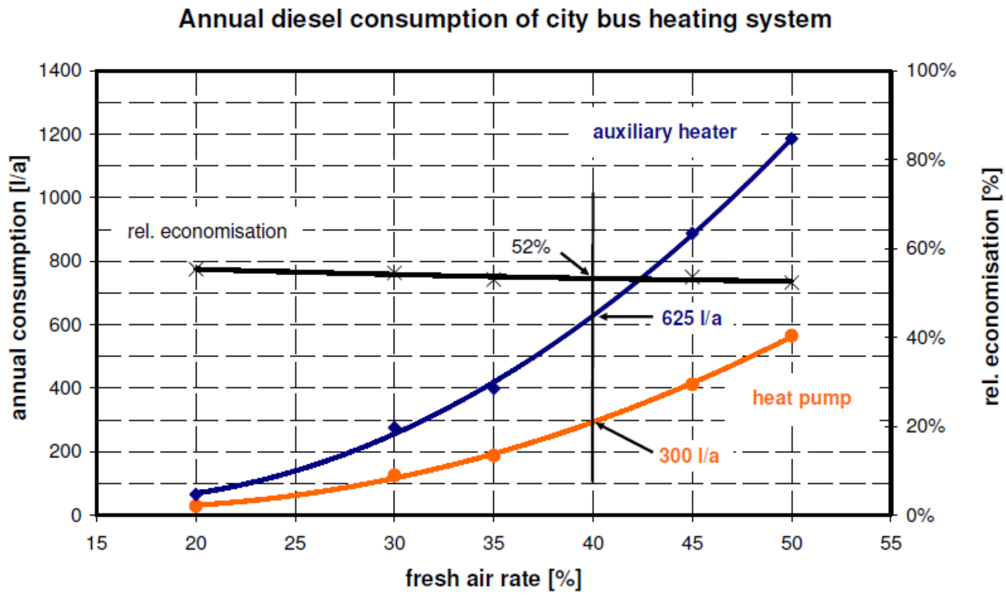


Figure 1-7: Annual fuel consumption and relative economization for CO_2 heating system compared to auxiliary heater (fuel burner). (Fõrsterling, Tegethoff, & Sonnenkalb, 2009)

Naturally the fuel consumption increases with increased ambient air intake, but the increase is about doubled by the use of auxiliary heater.

1.6.3 Automobiles

Combustion engine cars

Traditionally the heating of conventional combustion cars is done by using waste heat from the car's engine. The cooling is done by a MAC, and the most common refrigerant is R134a. (Hafner & Nekså, 2005)

There is a problem of insufficient waste heat for heating of passenger compartment during winter. With the use of engine heat, the heat up-period is long and the defrost-function is too slow on the coldest days. As a mean of supplementary heating, the solution can be to use the air condition as heat pump instead of the electrical heating that is the most common, namely an M-HVAC, since a CO_2 heat pump has the benefits of high COP and capacity also at low ambient temperature. (Hammer & Wertenbach, 2000)

As the graph below shows, the heat pump reaches the desired foot outlet temperature after 5 minutes, while the engine heat system uses more than 20 minutes.

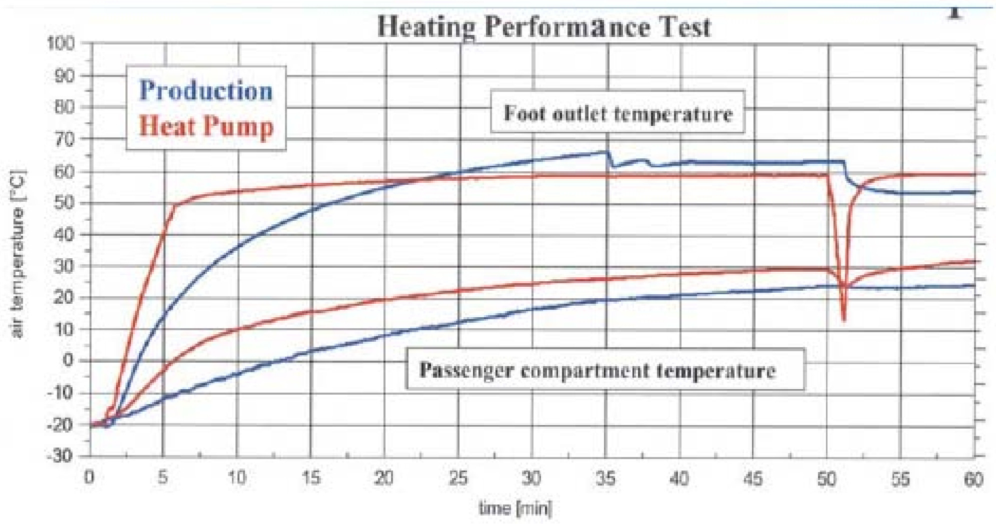


Figure 1-8: Measured air temperatures in during start-up of an Audi A4 test vehicle, and the same car with CO2 heat pump. (Hammer & Wertenbach, 2000)

There is also a 50% lower heat up-time from -20 to +20. This heat pump version used the engine coolant oil as heat source. Exhaust air or ambient air can also be used. When ambient air is used, there is a risk of frosting, but the system will be simpler and less costly. (Hafner & Nekså, 2005)

A CO_2 M-HVAC system has been developed which is constructed such that the waste heat from the heat pump cycle during dehumidification of the incoming air is recovered and used as an auxiliary heat source instead of an electric heater. The results showed that the cooling performance was equal to that of the conventional R134a, whereas the heating mode was 30% more efficient than a similar R134a system. (Tamura, 2005)

Flammability is an important issue in combustion engine cars, and the other possible low-GWP refrigerant, R152a, is highly flammable. Thus a secondary loop system is needed in order to be suitable for automotive M-HVAC. When using CO_2 this is not necessary, so a simpler and lighter system can be used. As a consequence, the German OEM's has decided to not consider flammable refrigerants any more. (Hafner & Nekså, 2005)

Over the last years, the German Motor Vehicle Industry Association (VDA) has coordinated development and testing of CO_2 systems, and several car manufacturers have had test vehicles on the road since the late nineties. (Hafner & Nekså, 2005) Presentations made by BMW, Audi and DaimlerChrysler at an industry meeting in 2002 showed the following consistent MAC related results from independent studies by the three companies:

- higher performance in cool-down mode for CO_2 than for R134a
- lower compartment temperature and faster temperature pull-down with CO_2
- reduced fuel consumption for the CO_2 system

The problem with the CO_2 system for mobile air conditioning is the high pressure levels that lead to more complex components. Meanwhile, Obrist engineering has successfully equipped more than 25 different car models with CO_2 air condition technology to many of the leading car manufacturers. (Wolf, 2004)

Visteon is another leading company for automotive parts, and they have also made a R744 clime system for various car manufacturers. (Visteon, 2010) There is nevertheless no long term or wide base experience, and as of 2008 there were no mass production experience of R744 M-HVAC for cars. (Morgenstern, 2008)

Electric Vehicles-EV

Where combustion cars have insufficient waste heat for compartment heating, the electrically driven cars have almost no waste heat available. Thus the heating needs to be provided by other means.

The EV producer Think has applied an auxiliary electric battery for heating (Think, 2010), and Reva is offering either a diesel burner or an additional electric battery for heating. (Reva, 2009)They both use conventional air conditioning, so they get a weight penalty with the extra equipment for heating.

The bigger car manufacturers like GM and Toyota have produced models with reversed air conditioning for both heating and cooling. This way there is no need for extra equipment for heating, and it is also more energy efficient and environmentally friendly than using oil burner or direct electricity. Nevertheless they are both using conventional refrigerants. (Barreto, 2008)

Investigations have been done for a reversible AC-system for EV's using CO_2.
The prototype investigated has a third heat-exchanger added, thermally connected with the battery-core, to the conventional MAC system used in previous EV-models like the GM EV1 and Toyota RAV4 EV. The concept has proven to bring several benefits such as prolonging the life-span of the electric battery, as well as improving the performance and overall energy-efficiency of the EV. (Barreto, 2008)

Hybrid electric cars-HEV

A hybrid electric vehicle combines the conventional combustion engine with an electric battery, which has the intention of increasing the fuel economy. Hybrids, just like combustion engine cars, rely on waste heat from the engine for compartment heating. Obviously the engine needs to run more frequently to keep up with the heating demand on cold days. The longer the combustion engine operates instead of the electric battery, the fuel economy decreases. (Hybrid Electric Vehicles, 2010)

Running the air MAC also reduces the fuel economy. Several cars were tested for mixed driving and found that the cars tested got 15 to 27% less fuel economy with the air condition system turned on. (hybridCARS, 2006) This indicates that it is of vital importance to have a highly efficient AC system.

2 Heat exchangers for CO_2 heat pumps

2.1 Evaporators:

CO_2 has very preferable heat transfer properties, of which influence the design of the evaporator.

With conventional refrigerants, the first part of the evaporator has low efficiency due to low liquid velocity. Because CO_2 has low surface tension, bubble boiling will start occurring at an earlier stage and the heat transfer is less dependent on high liquid velocity. This gives a good heat transfer throughout the whole length of the evaporator.

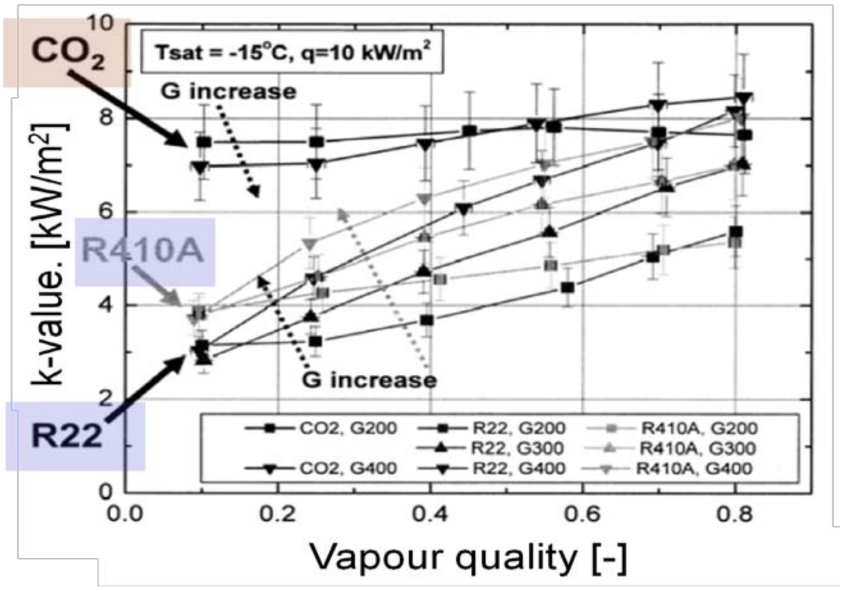


Figure 2-1: heat transfer coefficient as function of vapor fraction for different common refrigerants for air conditioning. (Stene, 2009)

Optimally designed evaporators for CO_2 will typically have 15-20% higher k-value (heat transfer coefficient) than other refrigerants. (Haukås, 2007)

Since CO_2 operates at higher pressure levels, the decrease in temperature for certain decrease in pressure is a lot smaller for CO_2 than for other refrigerants. From the figure below can be seen that the temperature loss is a lot smaller for CO_2 for the whole range of ambient temperatures.

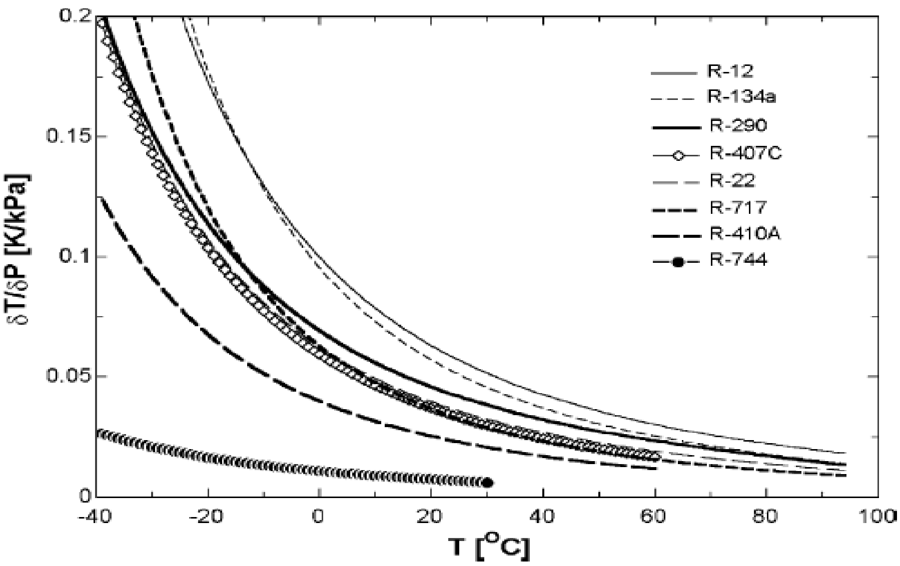


Figure 2-2: Slope of saturation pressure curve dp/dt for different refrigerants. (Haukås, 2007)

Due to the low temperature loss compared to other refrigerants, the optimal length per tube circuit in the evaporator will be considerably longer. This means less parallel circuits, and fewer problems with distribution of refrigerant on the different circuits. As this is often a problem, eliminating this issue can in fact be an even bigger plus than the improved heat transfer.

As a conclusion one can say that correctly constructed evaporators for CO_2 will in general be more efficient than for traditional refrigerants. This can be exploited in several ways:

- Reduced evaporating area. -Gives equal evaporating temperature but reduced costs.
- Reduced LMTD- Leads to higher evaporating temperature and thereby higher energy efficiency, but equal costs.
- A combination of reduced area and reduced temperature difference.

For an M-HVAC the goal is to make the components as small and light as possible, and by using CO_2 the heat exchangers can be designed smaller than for all other refrigerants.

2.2 Gascoolers:

The duty of a gascooler is dependent on the temperature approach between the incoming air and the outgoing CO_2. This temperature difference should optimally be around 2-4 K, and it is regulated by the high side pressure. In a temperature enthalpy diagram it is easy to see how the duty increases by increased gascooler pressure:

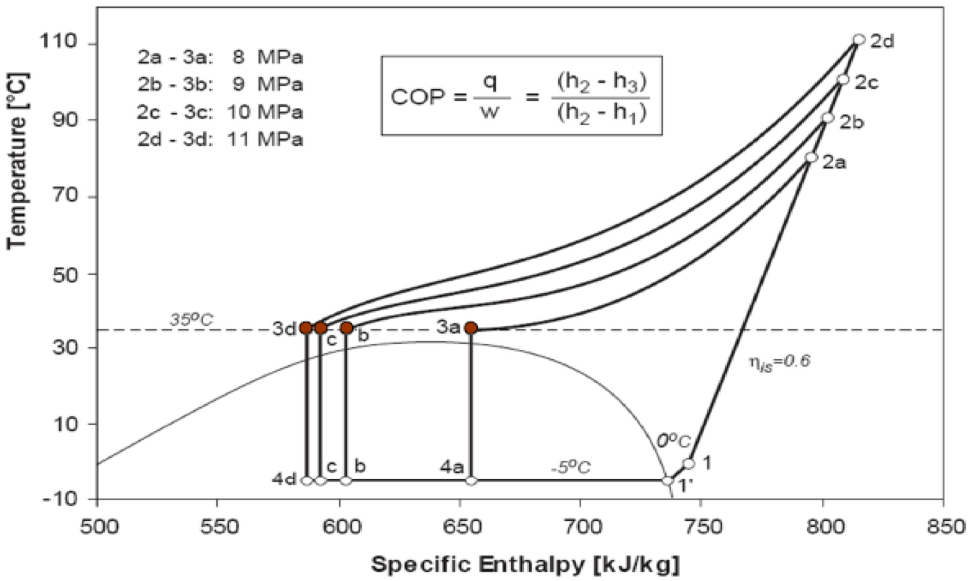


Figure 2-3: Temperature-enthalpy diagram showing gascooler duty with varying gascooler pressure. (Stene, 2009)

The optimal pressure is dependent on both the ambient temperature and the necessary heat exchanger duty. The heat transfer in the supercritical area is very high. This allows for a more compact design than what is possible for conventional condensers. This is preferable in mobile HVAC modules where space is a restriction. Alternatively, if the volume is held constant, the air side surface can be enhanced.

2.3 Type of heat exchangers for CO_2 M-HVAC

The most common heat exchanger for HVAC purpose is the fin-and-tube. This consists of round copper tubes and aluminum fins. The refrigerant runs through the tubes, and is cooled/heated by the air passing through on the outside of the tubes. The purpose of the fins is to enhance the heat transfer between the refrigerant and the air.

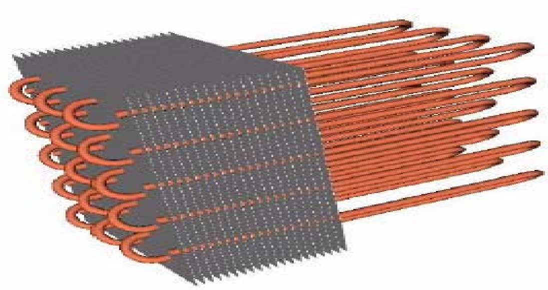


Figure 2-4: Fin-and-tube heat exchanger (Stene, 2009)

In the picture a part of the fins are removed to visualize the tubes. The fins are actually covering the whole length of the tube.

An alternative to the fin-and-tube heat exchanger is the multiport extruded heat exchanger, or MPE. The tubes are replaced by extruded aluminum channels with folded fins in between:

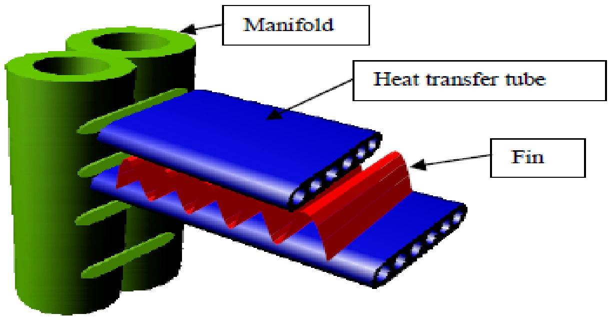


Figure 2-5: Principle of the MPE heat exchanger; Multiport extruded tubes with microchannels, folded fins and a compact "double barrel" manifold. (Pettersen, Hafner, & Skaugen, 1998)

MPE heat exchangers are well apt for CO_2 since the channels have a very small diameter, typically around 1mm. This way they can be made very efficient and compact.

There is nevertheless a problem with distribution of two-phase flow in the microchannels. The potential drop in heat exchanger capacity could be more than 30% due to maldistribution of two-phase flow in the inlet manifold. (Hafner & Nekså, 2005) MPE heat exchangers are thus normally only an option for the gas coolers.

3 Moist air considerations

3.1 Enthalpy of moist air

The energy contained by the air entering a heat exchanger can be measured in enthalpy. This is a function of both temperature and humidity, so by using enthalpy differences instead of temperature differences for energy flow calculations, the humidity of the air is automatically accounted for. (Monforte, 2007) From the chart below it is observed that at 50^{o}C ambient temperature and 20% humidity, the enthalpy of the air is the same as for 35^{o}C and 80% humidity.

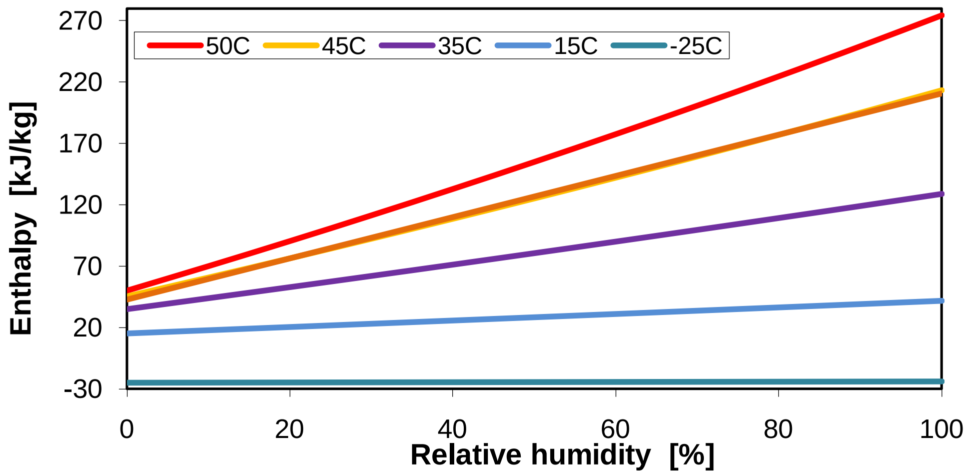


Figure 3-1: Enthalpy at varying relative humidity

The humidity has a greater influence on enthalpy as the temperature rises. At -25^{o}C, there is almost no change in enthalpy by increasing relative humidity.

3.2 Comfort levels for transport applications

The American Society of Heating, Ventilation and Air-conditioning, ASHRAE, has made a standard for indoor air conditions, called "Thermal environmental conditions for human occupancy." (Monforte, 2007). This is originally intended for buildings, but can also be applied for transport purposes. The comfort conditions for indoor environment are stated in the following Mollier-diagram, where the blue frame is the summer comfort conditions, and the red frame is the winter comfort conditions:

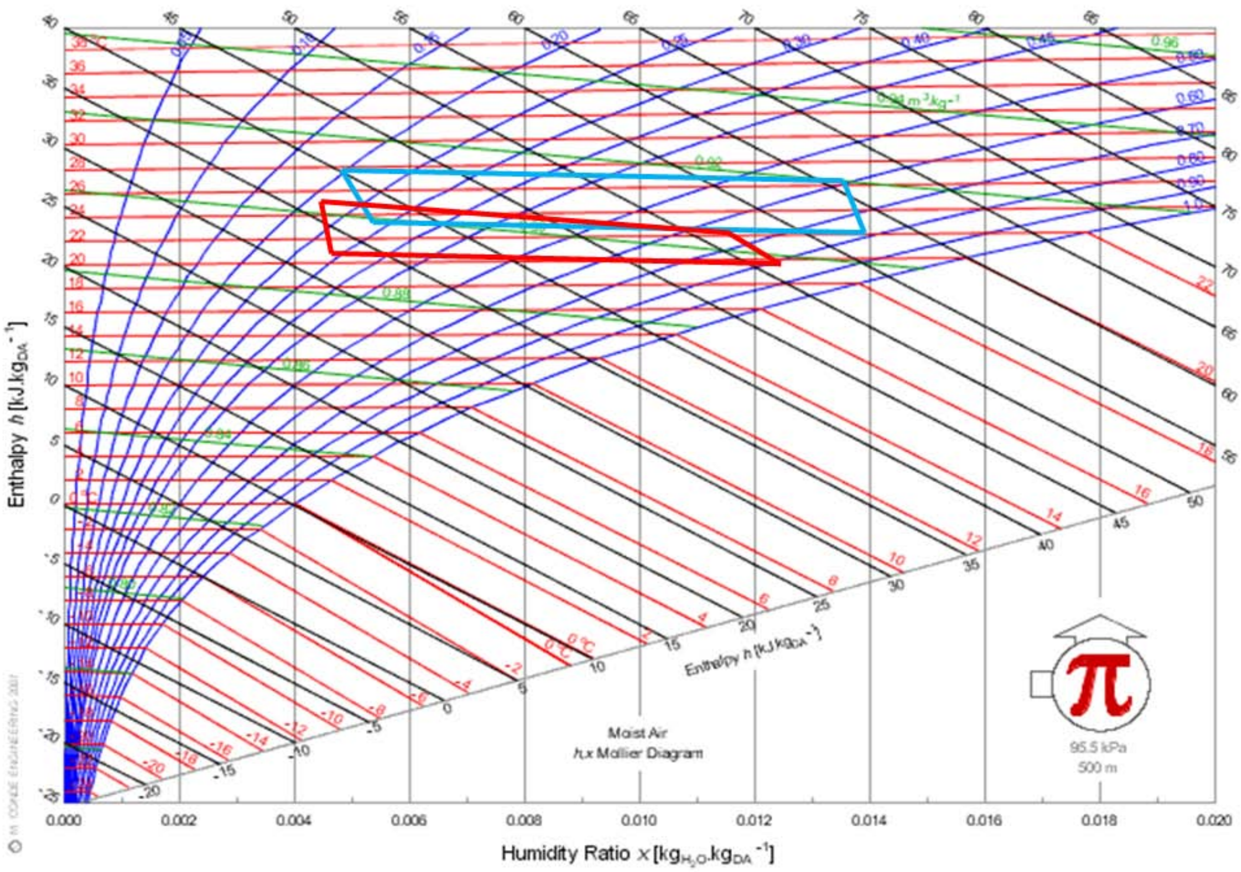


Figure 3-2: Comfort conditions for temperature and humidity according to ANSI/ASHRAE 55-2004.

As visible in the chart, the winter comfort conditions for temperature and humidity are in general a bit lower than the summer comfort conditions. There is also an upper limit for allowable absolute humidity; it should not exceed 0,0015kg/kg. Meanwhile, there is no recommended lower limit for the absolute humidity.

It is also not necessary to maintain the same indoor conditions at all ambient temperatures, and it is more energy efficient to adjust the indoor conditions to the ambient conditions. The ASHRAE limits for comfort conditions are stated in the following table (Monforte, 2007):

Table 3-1: Compartment comfort conditions:

	Enthalpy [kJ/kg]	Temperature [˚C]	Relative humidity [%]
Minimum	32	20,6	30
Mean	49	23	45
Max	57	25,6	60

4 Case study of a train compartment in 5 cities

To design an M-HVAC for a train compartment, a model of the actual compartment needs to be established, as well as the operating conditions. The M-HVAC is thought to provide both heating during winter and cooling during summer. Therefore it is necessary to identify the heat flow balances for both operational modes in order to find the necessary power requirement and energy consumption for the M-HVAC.

An energy requirement calculation tool was developed in order to find the heating and cooling demand for the train compartment, as well as the resulting energy consumption by the M-HVAC system. This tool makes use of HxLib, which is a moist air properties library developed at Sintef/NTNU. The calculation tool is constructed such that by implementing the input data, which is measures of the compartment, the operational hours of the train and the occupancy rate during the operational hours, the resulting power demand is generated hour by hour throughout the year for the locations chosen. This is explained in detail in appendix, chapter 11.2.

4.1 The train compartment

The train chosen is the high speed train set Velaro from Siemens, which has a top speed of 350 km/h. It currently operates in various contries all over the world. The picture below is a Velaro train set operated by the Spanish railway operator Renfe.

Figure 4-1: Siemens Velaro train set operated by Renfe (SiemensAG)

The Velaro train compartment has the following measures:

Table 4-1: Measures of the compartment

Compartment length [m]	24,175
Compartment height [m]	3,89
Compartment height [m]	2,95
Window area [%]	25

The M-HVAC will be turned off when the train is not in operation, which for the case study is between 10pm and 6am. The occupancy rate during the day is chosen according to the following table:

Table 4-2: Occupancy rate profile

Hour of Day	Occupancy
6	One third
7	One third
8	Full
9	Full
10	Two thirds
11	One third
12	One third
13	One third
14	One third
15	Two thirds
16	Two thirds
17	Full
18	Full
19	Half full
20	One third
21	One third

See appendix, chapter 11.2.1 for complete input data sheet for calculation.

4.2 Climate

For this case study 5 different cities with different climates are used: Trondheim, Frankfurt, Madrid, Moscow and Athens. The train model chosen is, among other countries, currently being operated in Russia, Spain and Germany. Trondheim is included as a local reference, and Athens is included to represent the extreme point of cooling demand in Europe. These 5 cities represent a great variation in climate, and most other cities in Europe can thus be related to one of the cities explored in this case study.

Climate data from the climate database Meteonorm is used to calculate the ambient load for a train compartment for the 5 cities. This is a global meteorological database for solar energy and applied meteorology, and from this database the statistical hourly values of ambient air temperature, humidity and the solar radiation are found. (Meteonorm)

A good indication of the different climates is the ambient air enthalpy profile, which is shown below for the 5 cities:

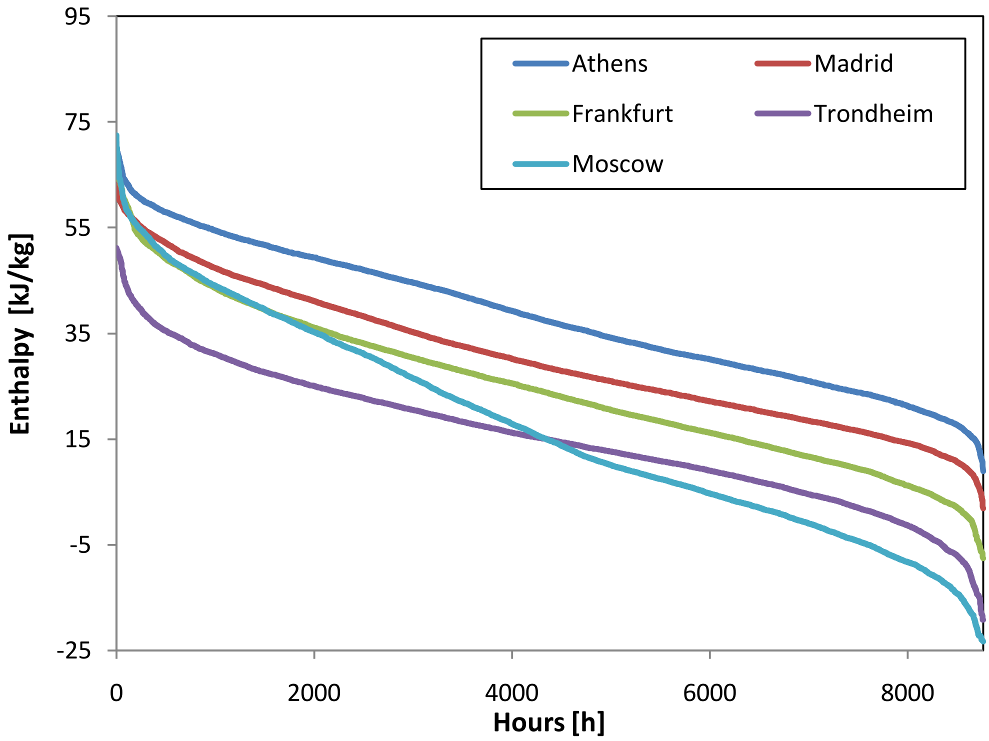


Figure 4-2: Ambient air enthalpy profiles the five different cities.

It is evident that Athens has the most demanding climate for cooling conditions, while Moscow has the most varying climate, stretching from almost maximum enthalpy, signifying high cooling power demand, to the minimum air enthalpy, signifying high heat power demand.

The solar radiation through the compartment windows is relatively high in the northern cities during winter. This is due to that the sun's position is closer to the horizon, and the direct radiation on a normal plane is therefore higher here than in e.g. Athens. This can be seen from the comparison of annual radiation through the compartment windows in Trondheim and Athens below:

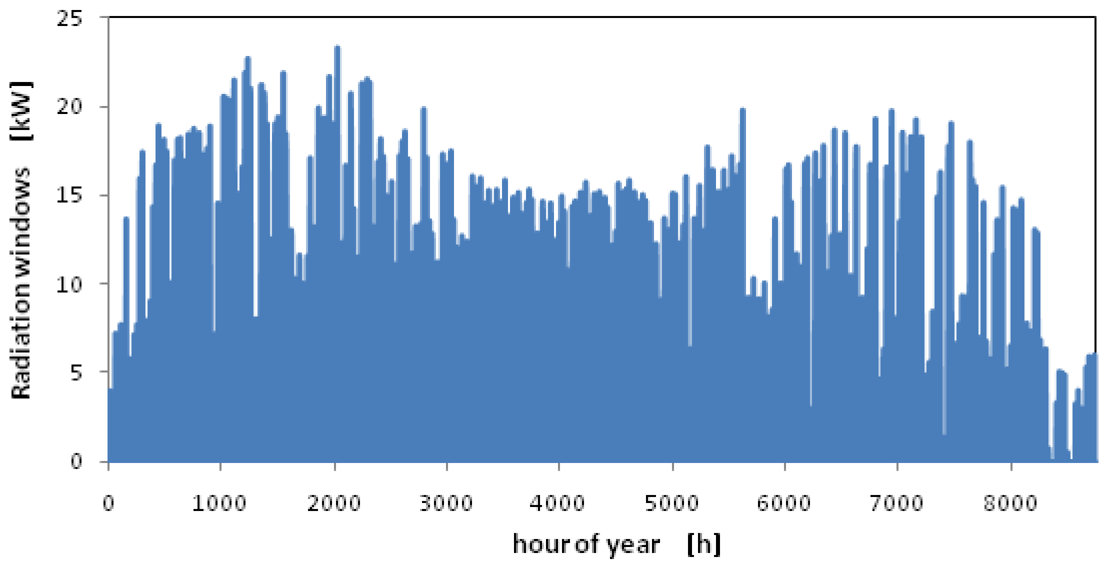


Figure 4-3: Variation of solar radiation during the year in Trondheim

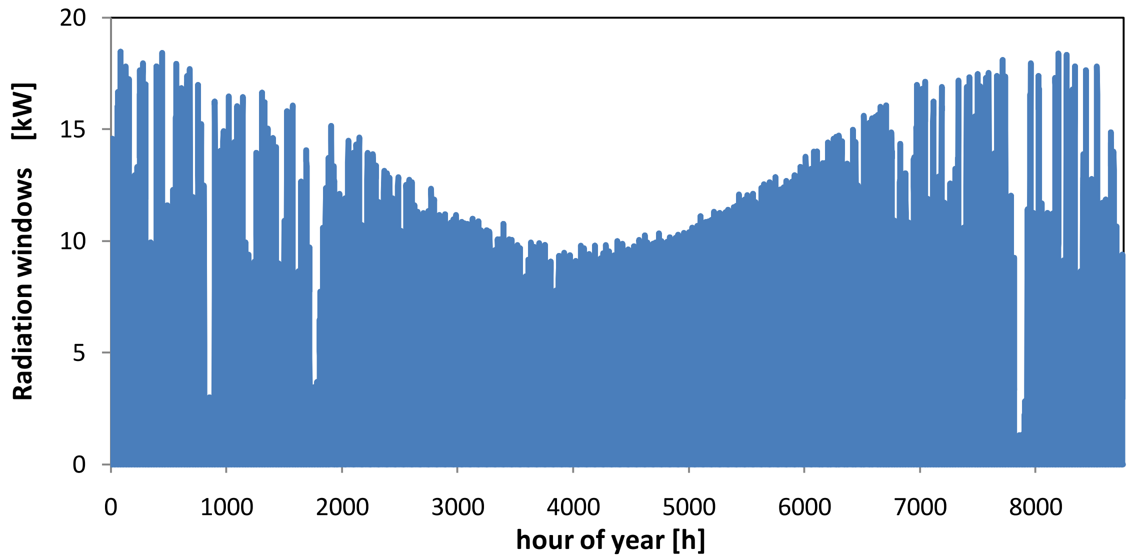


Figure 4-4: Variation of solar radiation during the year in Athens

On a sunny day in Trondheim, the radiation is actually larger than in Athens. For both places the radiation is larger in Spring/Autumn than during summer, due to the sun's position in the sky. It is thus clear that the solar radiation through the windows is independent of ambient temperature, and needs to be taken into account in power demand calculations.

4.3 The air reversing heat pump

The CO_2 MAC system that will be applied for this case study is a prototype reversible heat pump developed at Sintef Research Centre. The M-HVAC is reversible in the sense that it is placed on a rotatable plate. To switch from heating to cooling mode the turntable is turned 180 degrees so that the heat exchangers switch place diagonally. The layout is sketched below:

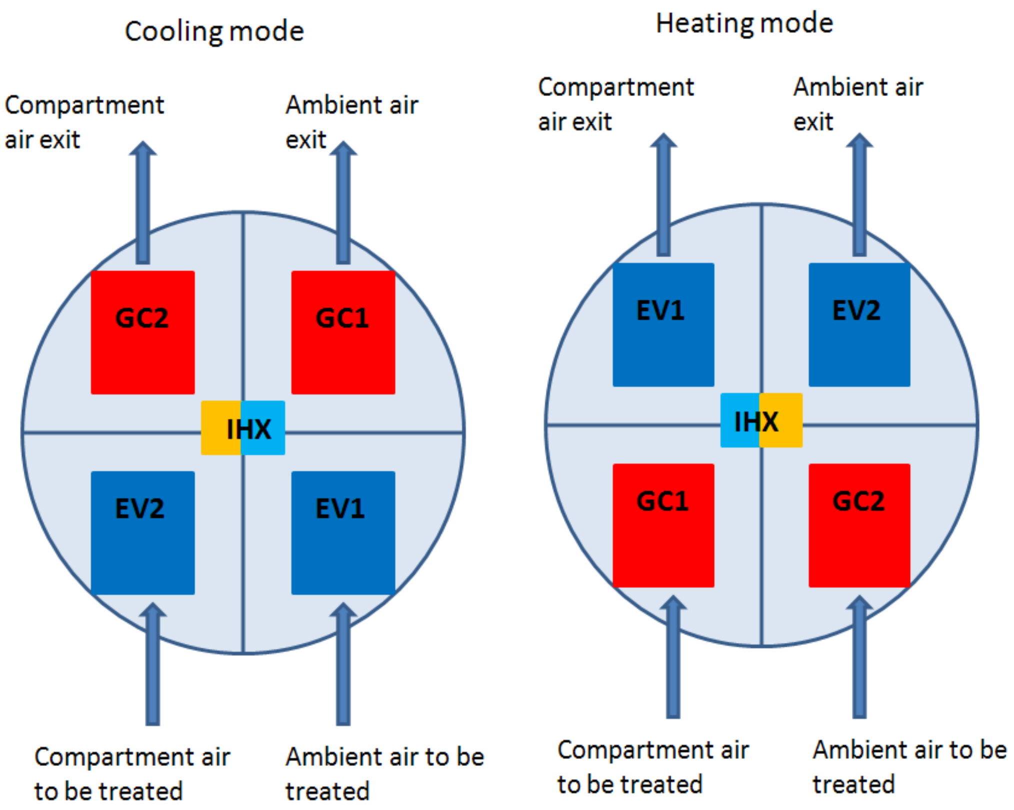


Figure 4-1: Layout of the turntable prototype in cooling and heating mode

In this system the air always flows in the same direction through the M-HVAC. An equal quantity of air is leaving and entering the system, so the exhaust air exiting the system always equals the admixed ambient air. With this layout the heat exchangers can be optimized for one purpose, since the gascooler always functions as a gascooler, and the evaporator always functions as an evaporator.

As can be seen from the layout, the system has 4 heat exchangers, in addition to an internal heat exchanger in the middle. This way one gascooler and one evaporator will treat entering/exiting air, and one gascooler and one evaporator will treat recycled air and bypassing air.

The air flow directions for each heat exchanger are stated in the table below to easily compare the function of each heat exchanger in heating and cooling mode:

Table 4-1: Heat exchanger function in cooling and heating mode

Heat exchanger	Heating mode	Cooling mode
Evaporator 1	Exhaust air exit	Fresh air to be treated and sent into compartment
Evaporator 2	Ambient air enter and exit HX (bypass air)	Exhaust air is recycled
Gascooler 1	Exhaust air is recycled	Fresh air enter and exit HX(bypass air)
Gascooler 2	Ambient air to be treated and sent into compartment	Exhaust air exit

The heat exchangers of the turntable cannot exceed the space limitations of the train roof. The radius of the turntable is 0.9m, so the resulting maximum size of the heat exchangers is: width: 0.5m, length: 0,6m. Besides the depth should not exceed 0.4 m. These limitations are used as a guide when designing the heat exchanger for the M-HVAC. (Section 6.2)

4.4 The process of the airflows in the M-HVAC

To get a picture of what is happening to the air properties when passing through the heat exchanger of the M-HVAC is chosen two operating modes, maximum cooling load and maximum heating load. They are shown in a Mollier diagram to better understand the process.

4.4.1 The air conditioning in cooling mode and full compartment

The following chart shows how the air is treated in cooling mode, with ambient air temperature of 35.7^{o}C and 35% relative humidity.

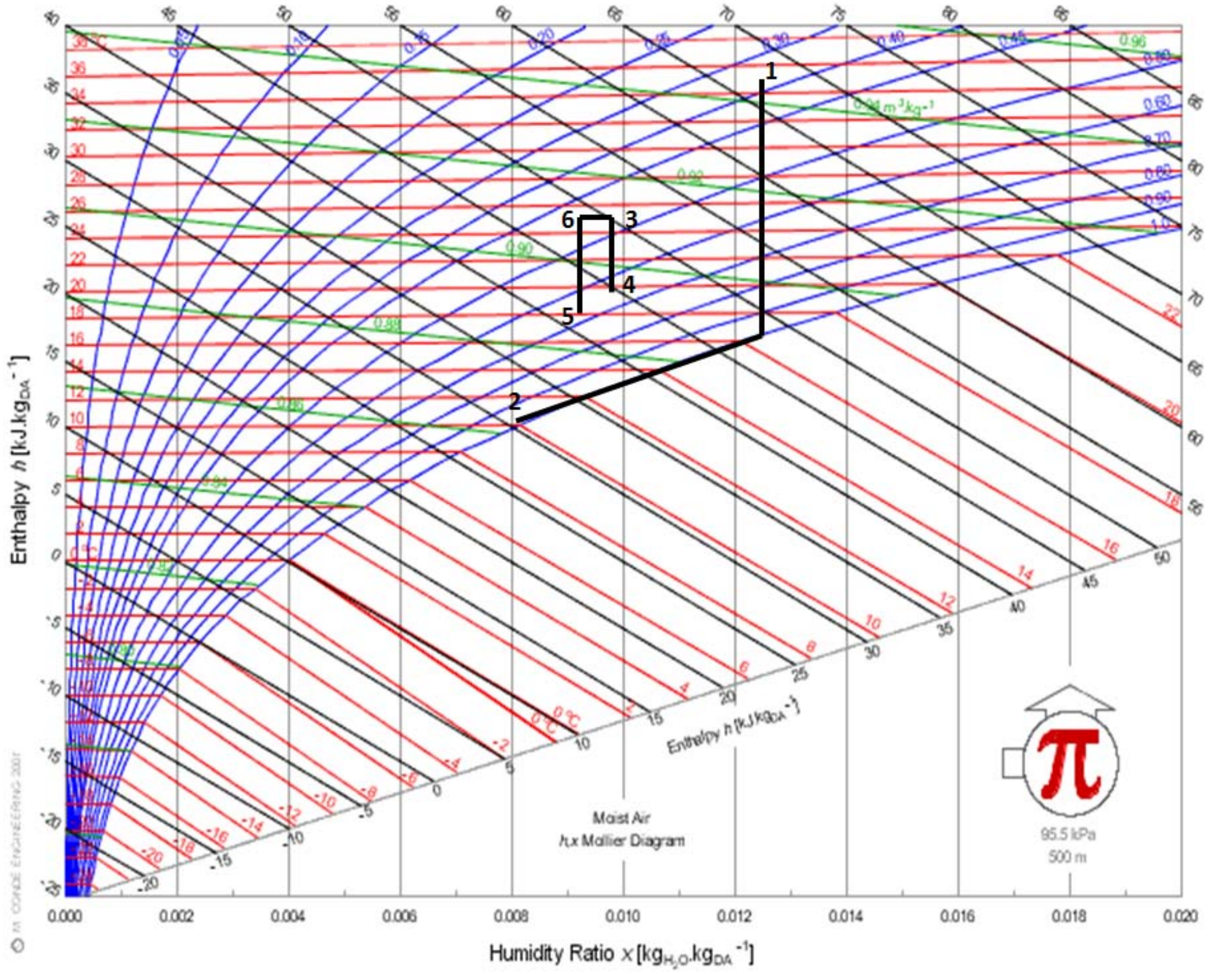


Figure 4-5: Process of the airflows in cooling mode.

Steps of air treatment according to the numbers in the chart:

1-2: The ambient air is cooled with constant absolute humidity until it reaches the saturation line. It is then further cooled along the saturation line until it reaches the wanted enthalpy of the ingoing ambient air.

3-4: The recycled air is cooled down from compartment temperature with constant absolute humidity until it reaches desired ingoing air temperature.

5-6: The ambient airflow and the recycled airflow are mixed in point 5 and are being heated up to point 6 due to the heat loads of the compartment.

6-3: The absolute humidity of the compartment air increases due to respiration from occupants.

4.4.2 The air conditioning in heating mode and full compartment

The following picture shows the treatment of air in heating mode. The ambient temperature is -20˚C and the relative humidity is 88%.

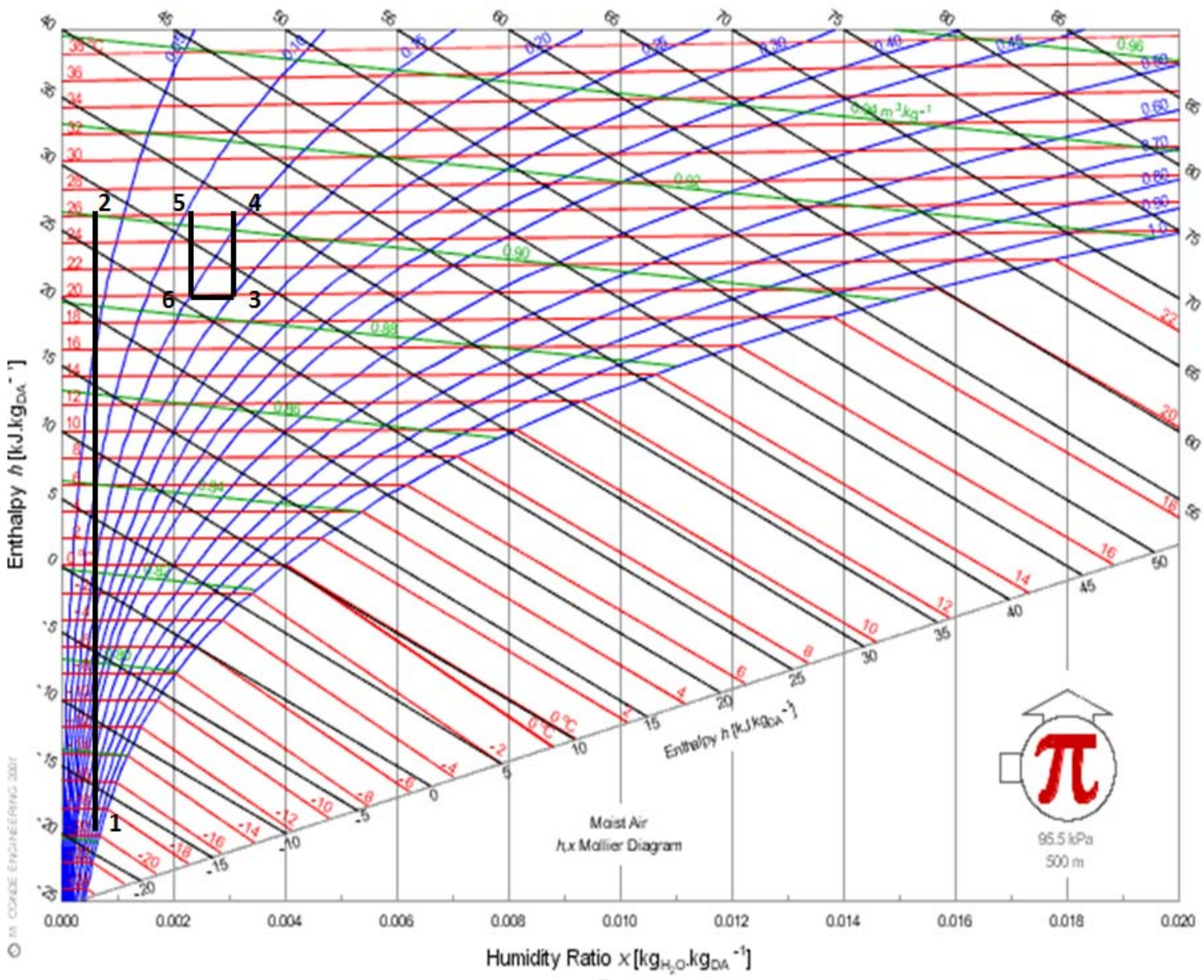


Figure 4-6: Process of the air flows in heating mode

Steps of air treatment according to the numbers in the chart:

1-2: The ambient air is heated with constant absolute enthalpy from ambient temperature to the desired temperature of ingoing ambient air.

3-4: The recycled air is heated with constant absolute humidity until it reaches desired temperature.

5-6: The mix of the ambient and recycled airflows is cooled with constant absolute humidity due to heat loss from the compartment.

6-3: The absolute humidity of the compartment increases due to respiration from passengers.

See appendix, chapter 11.1 for model of the compartment air flows.

Taking a look at the comfort zone of Figure 4-6 it is clear that in heating mode the humidity will for the coldest ambient temperatures be lower than the comfort conditions, due to that there is no moisturizing function in the M-HVAC. Moisturizer is very energy demanding and is therefore not a wanted feature.

4.5 The energy balances for the compartment

To get a picture of the differences in energy flows for cooling and heating mode, they are shown graphically in the following picture. The red arrow means heat flow entering or leaving, and blue arrow signifies cold flow entering, which requires heating.

Ambient temperature > compartment temperature	*Ambient temperature <compartment temperature*

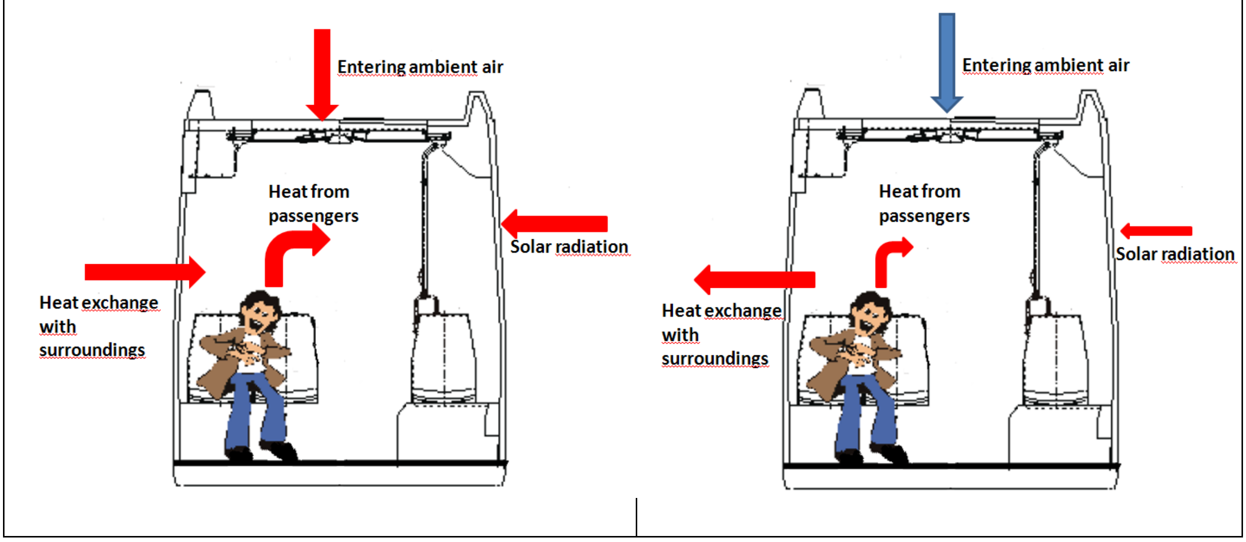


Figure 4-7: The energy flows in a train compartment for $t_{comp}<t_{amb}$ and $t_{comp}>t_{amb}$

As shown in the pictures above, the heat flows that determine the capacity demand for the MAC module are as follows:

1. Heat exchange with surrounding area caused by heat transfer from the ambient through walls, windows and roof.
2. Heat emission from passengers, both from heat emission and respiration
3. Admixing of ambient air
4. Solar radiation

From this, the energy balances around the train compartment can be established for both heating and cooling mode:

$$\dot{Q}_{HVAC} = \dot{Q}_{passengers} + \dot{Q}_{solar} + \dot{Q}_{ambientair} + \dot{Q}_{heatexchange} \tag{4.1}$$

If the resulting Q_{HVAC} is positive, cooling is required, and if the Q_{HVAC} is negative, heating is required. Due to the impact of sun radiation and heat from occupants, it may result in cooling mode even though the ambient air temperature is below compartment temperature. Depending on the ambient air temperature and the total load, the cooling can for those situations be provided simply by letting ambient air in. This fact is evaluated further in section 4.6.

The different heat flows are calculated as follows:

<u>Heat flow 1: Heat exchange with surrounding area:</u>

$$\dot{Q}_{heatexchange} = \dot{Q}_{walls} + \dot{Q}_{windows} \tag{4.2}$$

$$\dot{Q}_{wall/window} = U_{wall/window} \bullet A_{wall/window} \bullet \Delta T \tag{4.3}$$

$$\frac{1}{U_{wall}} = \left(\frac{1}{h_{amb}} + \frac{\delta}{\lambda} + \frac{1}{h_{in}} \right)^{-1} \tag{4.4}$$

Assumptions:

-Wall thickness	*0.05m*
-Heat transfer coefficient, h_{am}	*20W/m^{2}K*
-Heat transfer coefficient indoor, h_{in}	*7W/m^{2}K*
-Heat conductivity, λ	*0.03W/mK*
-Heat transmission coefficient window, U_{win}	*3W/m^{2}K*

According to the ASHRAE regulations, (Lynch, 2004), the set point for the compartment temperature is a function of the ambient temperature, and for the calculations the following is used:

Table 4-3: ASHRAE limits for compartment temperature:

Ambient temperature T_a [˚C]	Compartment temperature [˚C]
$T_a>30$	25,6
$-10<T_a<30$	23
$T_a<-10$	20,6

<u>Heat flow 2: Heat emission from passengers:</u>

The occupants cause the air temperature to rise, and the air humidity to rise due to perspiration. The energy contribution can be divided into 2 parameters:

Q sensible- the heat that one can feel. This is often set to a standard value of 100W per person.

Q latent- evaporation from skin and respiration. The latent heat is the only heat load on the compartment that leads to an increase in the absolute humidity. The amount is about 40g/h, and is often estimated to 30W. (NTNU-Sintef, 2007). This is nevertheless depending on the airflow going through the room, so this heat load will be calculated every hour of the year.

To find the contribution from respiration, it is first necessary to find the increase in absolute humidity.

$$\Delta x = \frac{\dot{m}_{water} \bullet \#_{passengers}}{\dot{m}_{air}} \tag{4.5}$$

The m_{water} is assumed to be 40 g/h per person.

The heat contribution from respiration is then

$$\dot{Q}_{respiration} = \dot{m}_{tot,air}(h_{comp,\Delta xresp} - h_{comp,xmix}) \tag{4.6}$$

Where:

$h_{comp,xmix}$ is the enthalpy at compartment temperature, and at the absolute humidity of the air of the mixed air entering the compartment from the M-HVAC.

$h_{comp,\Delta x}$ is the enthalpy at compartment temperature, with the respiration from passengers included.

The total heat load from passengers is then:

$$\dot{Q}_{passengers} = \#_{passengers} \cdot \dot{Q}_{person,dry} + \dot{Q}_{respiration} \qquad (4.7)$$

Where $Q_{person,dry}$ is 100W/person.

Heat flow 3: Admixing of ambient air:

The work needed to heat or cool ambient air is found from the difference between the wanted enthalpy of the ambient air entering the compartment and the enthalpy of the ambient air:

$$\dot{Q}_{ambientair} = \dot{m}_{ambientair} \cdot (h_{ambient} - h_{amb\_inl}) \qquad (4.8)$$

There is a European standard that limits the minimum percentage of the admixed ambient air per person, which will be applied in the heat load calculations:

Table 4-4: Minimum ambient air entrainment ratio:

Ambient temperature T_a [˚C]	Minimum ambient air flow rate per passenger [m^3/h]
$T_a \leq -20$˚C	10
-20˚C<$T_a \leq$-5˚C	15
-5˚C< T_a	20
T_a>20˚C	≥20

Heat flow 4: Solar radiation

The total solar heat gain is obtained by adding the direct normal radiation, diffuse horizontal radiation and radiation reflected from the ground. (Malic & Bullard, 2004) The different fluxes are obtained for every hour of the year, and for all cities, from the Meteonorm climate database. The solar load on the compartment is depending on the amount of windows. It is for the case study chosen to 25% of the surface of the train compartment. The following equations are used to calculate the solar radiation through the compartment windows:

$$\dot{Q}_{solar} = \dot{Q}_{solar\_direct} + \dot{Q}_{solar\_diffuse} + \dot{Q}_{solar\_ground} \tag{4.9}$$

$$\dot{Q}_{solar\_direct} = \tau_{glass} \bullet q_{direct} \bullet A_{glass} \bullet F \tag{4.10}$$

$$\dot{Q}_{solar\_diffuse} = \frac{\tau_{glass} \bullet q_{diffuse} \bullet A_{glass}}{2} \tag{4.11}$$

$$\dot{Q}_{solar\_diffuse} = \frac{\rho_{ground} \bullet \tau_{glass} \bullet (q_{direct} + q_{difuse}) \bullet A_{glass}}{2} \tag{4.12}$$

F is an exposure factor for direct solar radiation. Since the train is moving and is partly in the shadow and in the sun, the exposure factor is set to 0.25. The transmittance factor is set to 0.8, and the ground reflectance is set to 0.2.

See appendix 11.1.2 for detailed calculation method of the energy flows and load of the HVAC, and appendix 11.2.2 for excel-sheet with an example of energy flow calculations for one hour in Frankfurt.

4.6 Results from energy flow calculations:

When energy flows for every hour of the year is identified and added together, the total annual energy demand is as follows for the 5 cities of the case study:

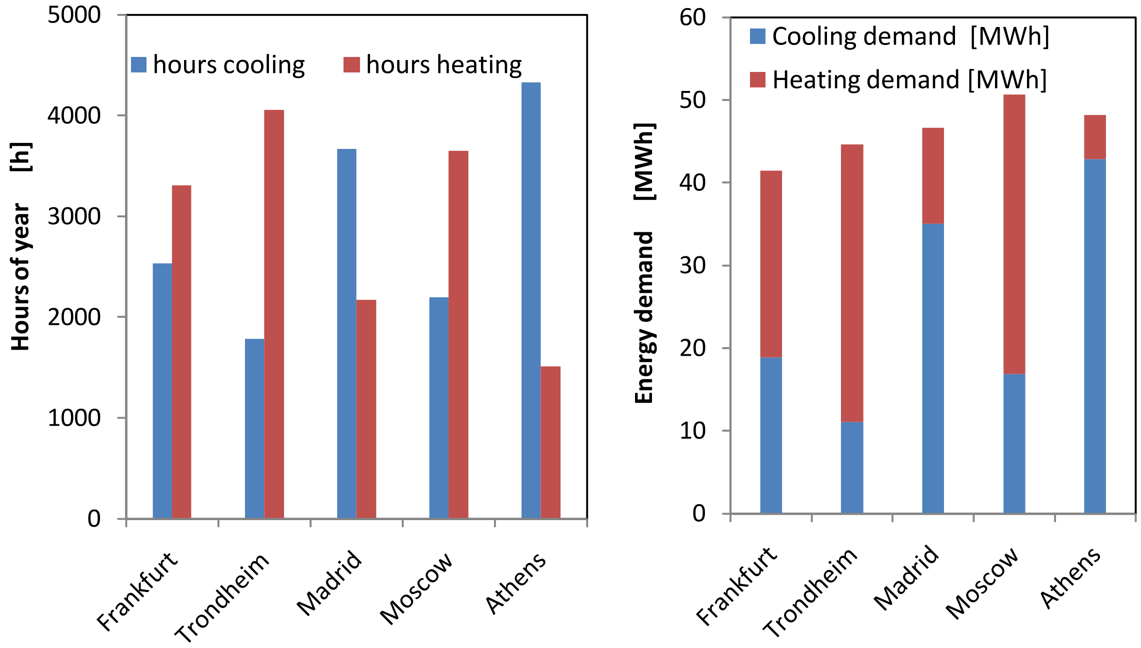


Figure 4-8: Hours of cool/heat load and total annual energy demand

Athens has the most hours of cooling, while Trondheim has the highest amount of heating hours. Moscow also has the highest total energy demand. This corresponds to the enthalpy profile in Figure 4-2, where Moscow reaches both the highest and the lowest levels of enthalpies.

When analyzing the demand hour by hour, it is discovered that for low cooling loads, there is sometimes needed heating of the incoming ambient air, and at the same time cooling of the recycled compartment air. This is not energy efficient, and the smartest solution for those situations will then be to turn off the M-HVAC and only let ambient air in without treatment. Since the ambient air in those cases is below compartment temperature, cooling is then efficiently covered by letting fresh air in. Cooling loads up to 10kW is thus thought to be covered only by fresh untreated air.

For heat loads up to 5kW the M-HVAC is also thought to be turned off, so that only the required ambient air will be entering. The M-HVAC system will for such low loads be very unstable, and the ambient air temperature is so close to the set point for compartment temperature that this would only lead to a slight temperature decrease.

Below is a chart of the share of hours of each power demand interval. Negative power demand indicates heating demand.

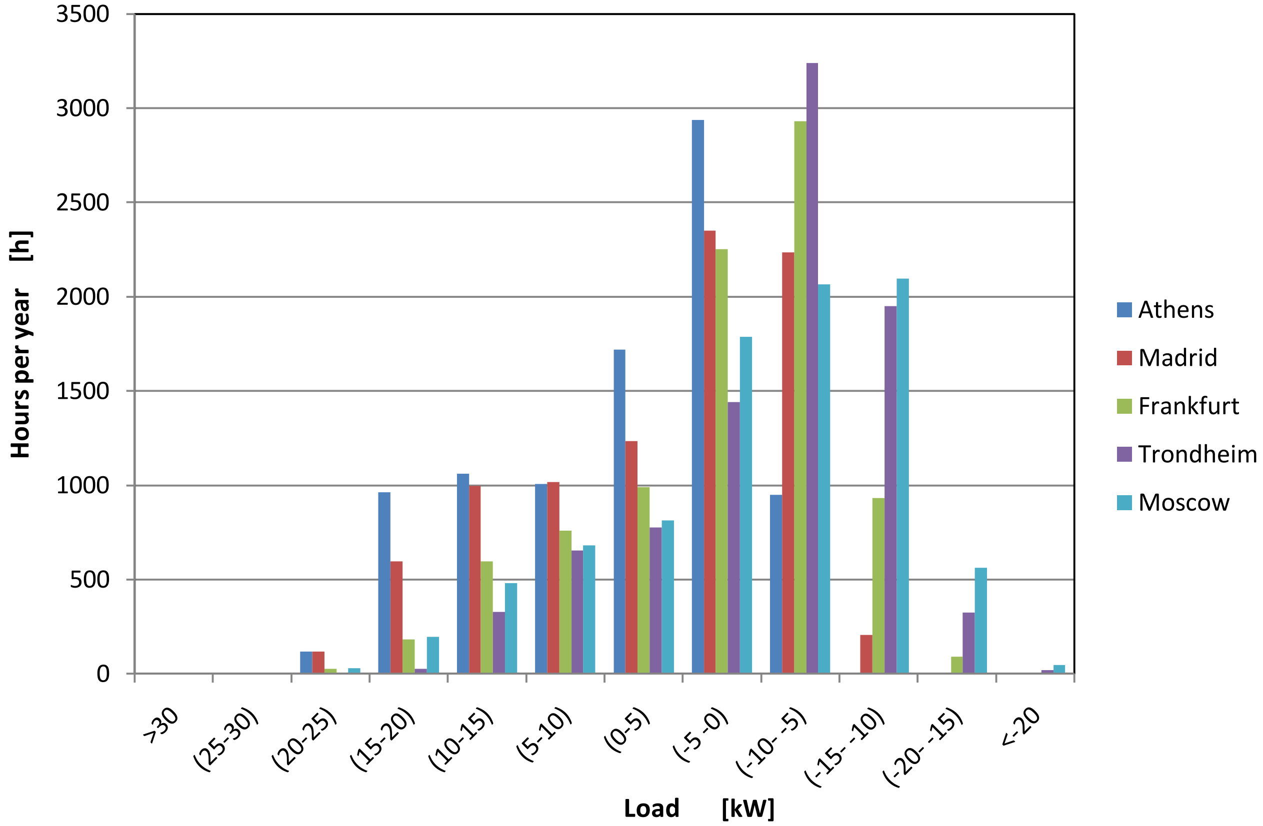


Figure 4-9: Amount of hours on the different load intervals for the case study.

By eliminating the heat loads between 10kW cooling and 5kW heating, the total amount of operational hours for the M-HVAC is reduced substantially. The new annual amount will then be as follows:

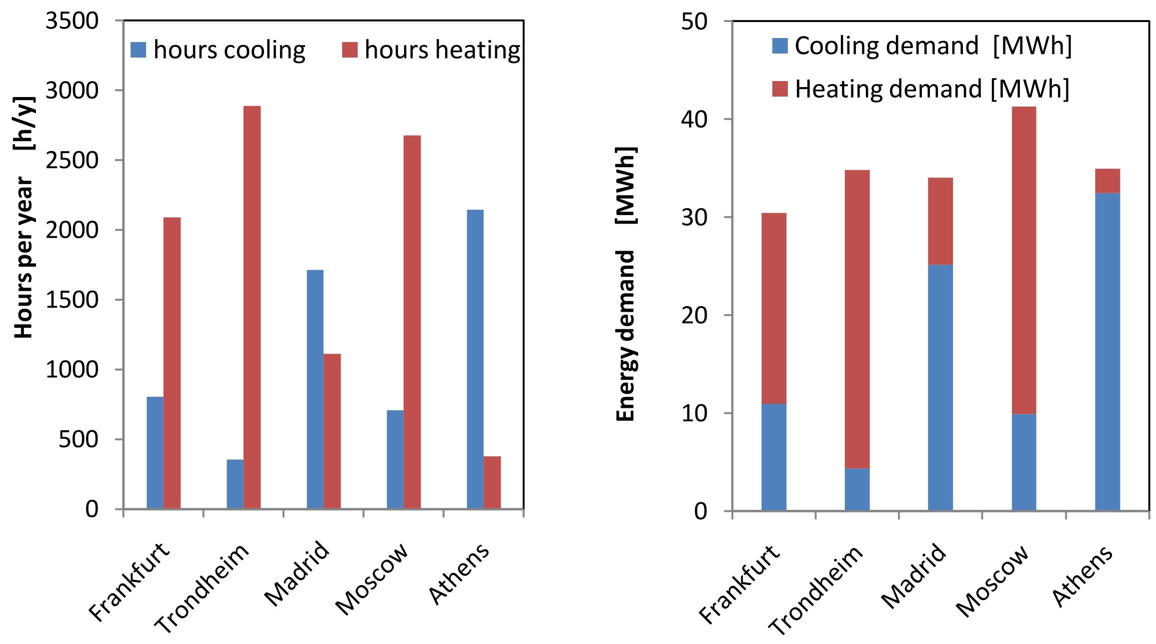


Figure 4-10: Hours of cool/heat load and total annual energy requirement that have to be provided.

The total operational hours of the M-HVAC are now reduced by about 1000 hours for all cities. The cooling demand is being more reduced than the heating demand, since all the hours with cooling loads lower than 10kW are eliminated from the total, and only the hours with heat load less than 5kW is eliminated.

Even though the energy demand for heating is higher than the total cooling demand, the maximum cooling power demand is higher than the maximum heat power demand (32kW vs. 23kW). This is partly due to that the air flow rate of entering ambient air is higher in cooling mode, and it is the cooling of this airflow that requires most power. The radiation is also an important contribution to the cooling load, which on the contrary reduces the heating load. The maximum cooling load will therefore be the design condition for the design of the M-HVAC.

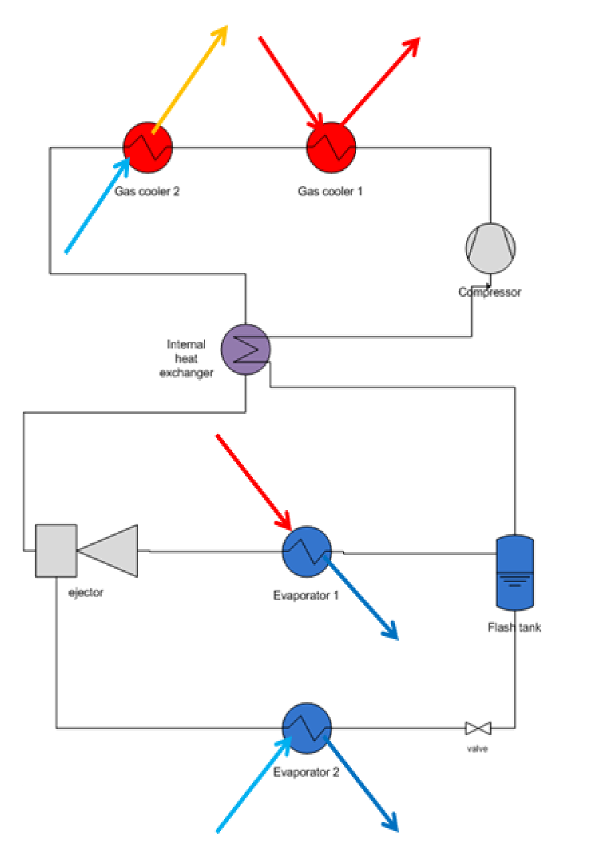
5 The M-HVAC system layout

When the heat flows and following airflows have been identified for the train compartment, the M-HVAC is ready to be simulated in order to find the operating conditions for the system.

As mentioned, due to the low critical temperature, CO_2 has higher throttling losses than other refrigerants used for air conditioning. This can be reduced by implementing an ejector instead of the regular expansion valve in the circuit. An ejector is therefore included in the turntable M-HVAC for the train compartment.

The principal cycle with airflows in heating and cooling mode is shown below. Arrow pointing upwards means air entering the heat exchanger from the compartment, and arrow pointing downwards indicates air entering the compartment.

Cooling mode



Heating mode

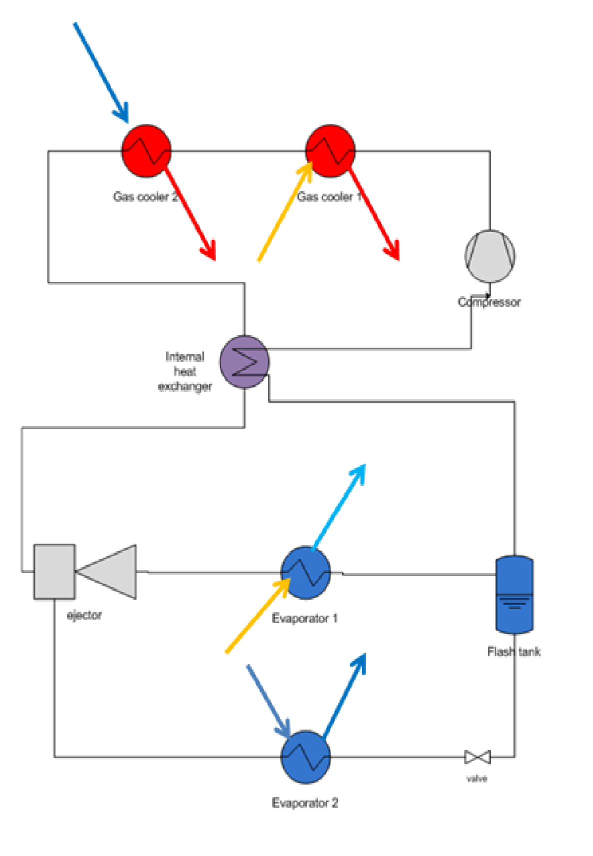


Figure 5-1: The MAC layout with ejector in heating and cooling mode and with corresponding air flows

For the prototype M-HVAC, there are two evaporators, namely one that treats ambient air, and one that treats recycled air. The evaporator that treats ambient air will be at the intermediate pressure, or ejector pressure. The evaporating pressure of evaporator 2 will be lower, and the pressure levels are regulated by the work recovery from the expansion process of the ejector. The ejector process is explained further in the following section.

5.1 Implementation of ejector to reduce throttling losses

An ejector works basically like a pump, but without moving parts. The aim is to use the energy that is contained in the high pressure gas from the gas cooler to compress the gas coming from the evaporator up to a middle pressure. Below is shown the principle of an ejector:

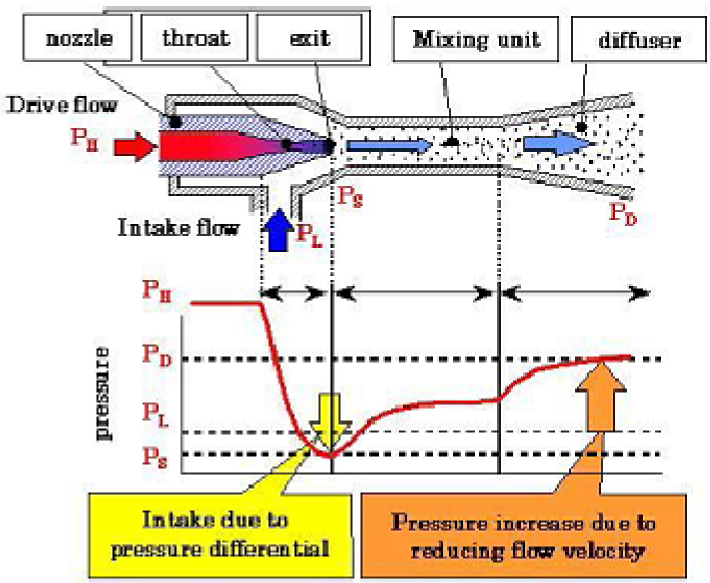


Figure 5-2: The ejector process (Stene, 2009)

Step-by-step the ejector process goes as follows: (Hrnjak & Elbel, 2007)

1. High pressure gas from the gas coolers is accelerated in the high pressure nozzle (red area). The static pressure is low and the kinetic energy is very high.

2. The low pressure stream coming from the evaporator is pre-accelerated in the suction nozzle to reduce mixing losses caused by shearing forces.

3. The lower-energetic stream from the evaporator is entrained (mixed) and accelerated by momentum transfer from the high pressure to the low pressure stream. Pressure decreases to below the low pressure stream pressure, before the mixing causes the two velocities to equalize, and pressure raises.

4. A subsonic diffuser converts the remaining kinetic energy into static pressure.

Instead of defining ejector efficiency, the performance of the ejector is often given in pairs of:

- Mass entrainment ratio: $\phi_m = \frac{\dot{m}_{evap}}{\dot{m}_{GC}}$

  (5.1)
- Suction pressure ratio: $\Pi_{sp} = \frac{P_{diff,out}}{P_{evap,out}}$

  (5.2)

The ejector performance is relatively independent of ambient temperature. This is a very important aspect for CO_2, where the efficiency normally decreases significantly at increasing ambient temperature

6 Simulations of the turntable M-HVAC using Modelica

The simulations of the M-HVAC system are performed in the computer language Modelica, which is an open, object-oriented language for modeling of large, complex and heterogeneous physical systems. To simulate the Modelica system, various other simulation tools are used:

Dymola is used as a platform, and it is the interface that translates Modelica language to C through a C compiler. It makes use of modeling methodology based on object orientation and equations. (Dymola, 2009)

TIL is a Modelica library for steady-state and transient simulation of thermodynamic systems. It uses TILMedia, which is an interface library to provide fluid and solid properties from various existing property databases to different applications. (TLK-Thermo, 2010)

StateViewer is a tool used for graphical presentation of transient thermodynamic measurements or simulation data. The simulation files from Modelica can thus be transported to StateViewer to get a better understanding of the process performance. (TLK-Thermo, 2010)

6.1 Model of the M-HVAC in Modelica

The layout of the M-HVAC system in Modelica is shown in the following picture:

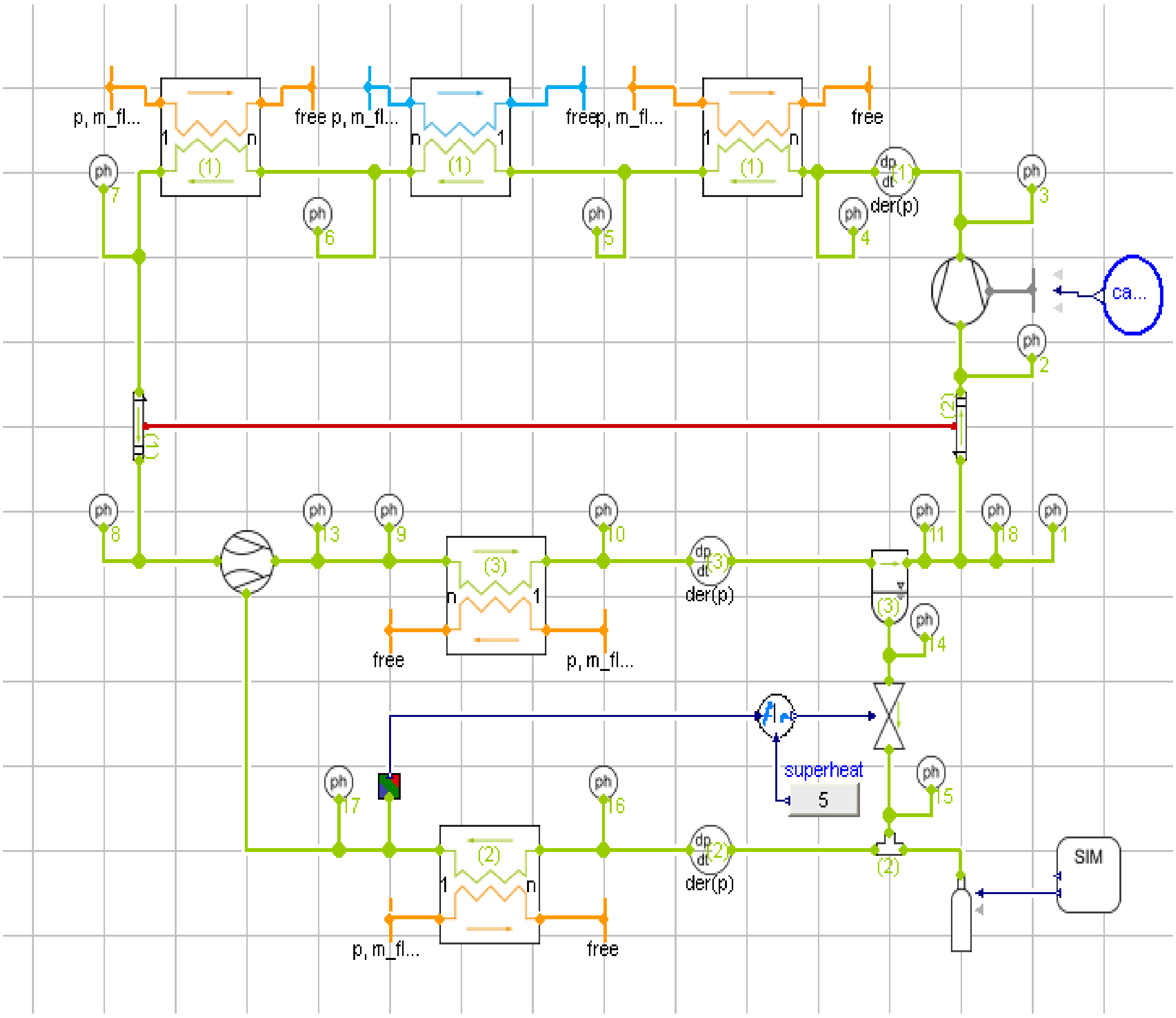


Figure 6-1: System layout of the M-HVAC in Modelica.

The red horizontal line indicates the internal heat exchanger, and the orange flows are the air flows entering and leaving the compartment. The main difference from the principal system chart (Figure 5-1) , is that in addition to the two air cooled gas coolers, a glycol heat exchanger is included to provide for floor heating. The floor heating is not investigated in the case study, as the air cooled gas coolers are designed to provide enough heat to cover the maximum heat load alone.

The principal difference of Modelica compared to earlier used simulation tools is that it makes transient simulations. This provides for a more realistic simulation process and gives a more thorough understanding of how such a system in real life would be adjusting to varying operating conditions.

The part load regulation of the compressor is done by decreasing the rotational speed. A regulator is connected to the compressor, identified as the blue circle to the right of the compressor. It regulates the frequency according to the heating or cooling requirement.

See appendix 11.3.4 for details on the design of the compressor

6.2 Configuration of the heat exchangers for the M-HVAC

In Modelica, the only choices for moist air heat exchangers are Fin-and-tube and MPE. For the simulations, parallel flow fin and tube is chosen for all four heat exchangers. Nevertheless, the important for the simulations is that heat exchangers are able to cover the power demand, and that the space restrictions are fulfilled.

HxSim was used to find the configurations for the different heat exchangers. This is a simulation tool by Sintef, and it is especially made for CO_2 heat exchangers that are cooled or heated by air.

The hour with the highest cooling load was identified in the energy calculations, and the corresponding operating conditions were used to model the heat exchangers. As an example, the gas cooler 1 designed for the M-HVAC is shown below:

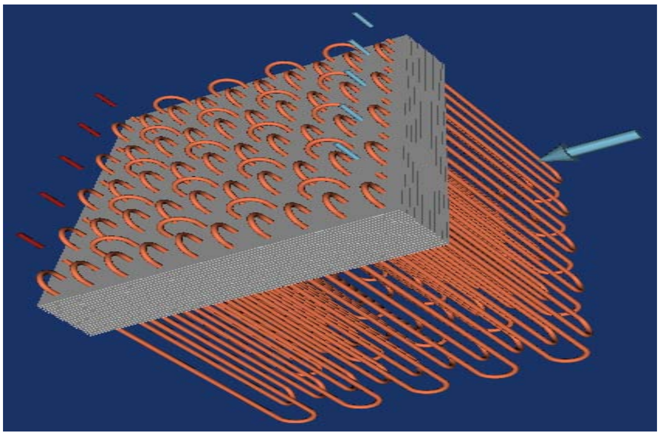


Figure 6-2: Gascooler 1 designed in HxSim for the M-HVAC.

See appendix, chapter11.3.1 for a detailed explanation of the configuration process of the heat exchangers and appendix 11.3.2 for the heat transfer models used for simulation.

The design of all the four heat exchangers is stated in the following table:

Table 6-1: **Heat exchanger configurations**

	Evap 1	Evap 2	Gascooler1	Gascooler2
Tube configuration	Tube-in-fin	Tube-in-fin	Tube-in-fin	Tube-in-fin
# Duplications	8	7	5	5
#Vertical tubes	3	3	3	3
#Horizontal tubes	9	8	10	12
Vertical tube pitch [m]	0,025	0,025	0,04	0,04
Horizontal tube pitch [m]	0,04	0,05	0,04	0,03
Core Depth [m]	0,36	0,4	0,4	0,36
Height [m]	0,6	0,525	0,6	0,6
Core width [m]	0,42	0,42	0,42	0,42
Fin density [FPI]	6	9	6	9
Fin thickness [m]	9,00E-04	1,00E-04	7,50E-04	1,00E-03
Pipe outer diameter [m]	0,011	0,007	0,007	0,007
Weight HX [kg]	137,1	40	106	171

6.3 System process

System process in cooling mode

Below is a pressure-enthalpy diagram of the process in cooling mode. The example used is for the maximum cooling load, which occurs at the ambient temperature of 35.7 ˚C and 100% occupancy rate.

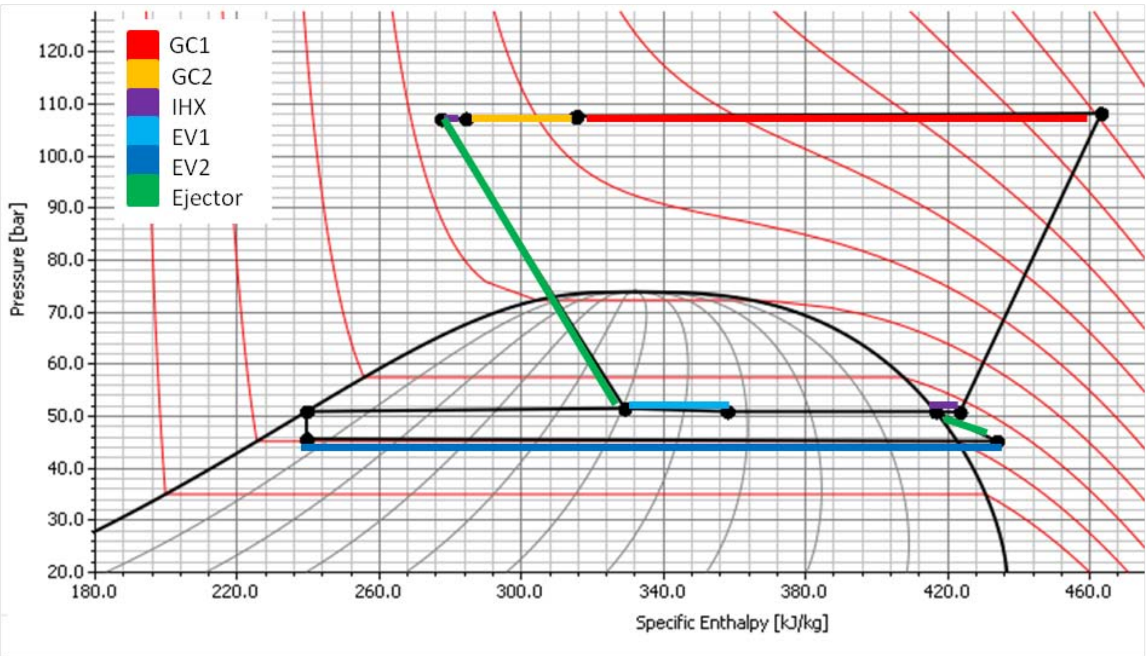


Figure 6-3: P-h diagram of the process in cooling mode

System process in heating mode

The following pressure-enthalpy diagram is of the maximum heating load. The ambient air temperature is -20˚C, and the occupancy rate is 100%.

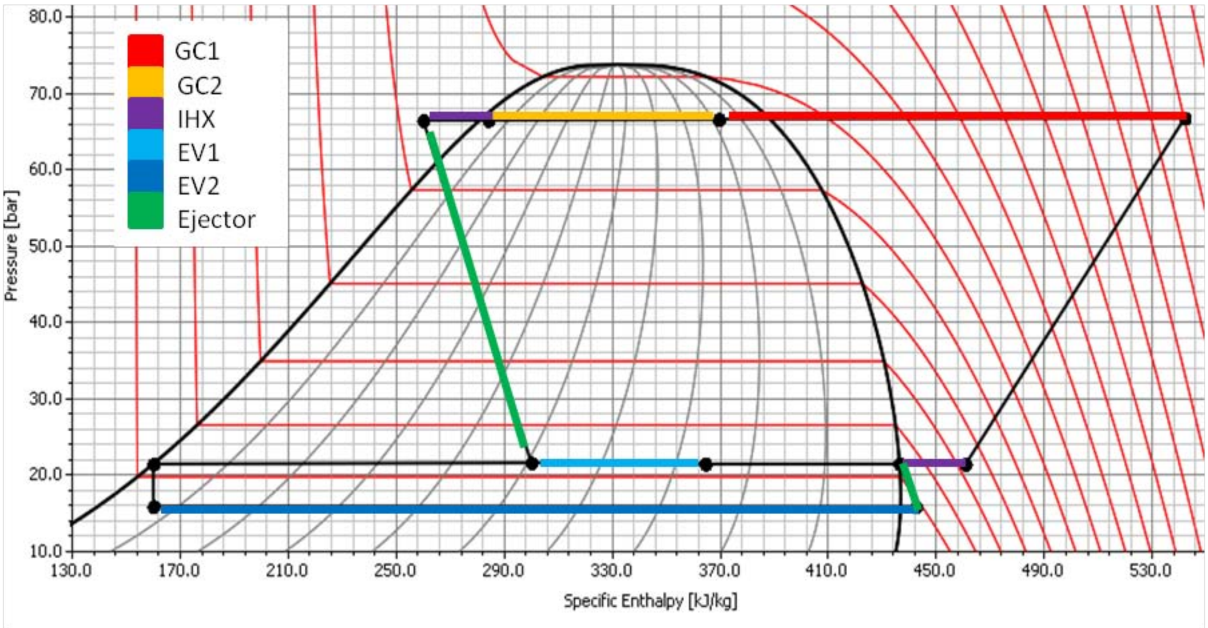


Figure 6-4: P-h diagram of the process in heating mode

Comments on the processes

In cooling mode, the process is transcritical; i.e. that the heat rejection takes place in supercritical area, above the phase envelope. The pressure of evaporator 1, light blue line, is around 5K below the ambient air temperature. The pressure difference between the two evaporators is about 5 bars and is depending on the cooling power requirement and the work recovery of the ejector.

In transcritical operation, the gas cooler pressure is highly depending on the ambient temperature, in addition to the power requirement. The optimal pressure will vary between 120 to 80 bars, and in this case it is 107 bars.

In heating mode the process is subcritical. In the extreme case of -20C ambient temperature, a great part of the heat rejection in gascooler 1 takes place in the superheated area. This is due to that the evaporator pressure is very low and the compressor process thus terminates far to the right of the phase envelope.

In heating mode, the gain of the internal heat exchanger is higher, due to that the temperature between the high pressure flow from gas cooler 2 and the suction gas to the compressor is bigger. This can be seen as there being a lot more constant temperature lines between the gascooler pressure and suction pressure in heating mode. This gain will be reduced by increasing ambient temperatures.

6.4 Dynamics of the system

To get a better understanding of how the Modelica system adjusts to varying operating conditions, two different cases are chosen:

Case 1

First is investigated a scenarios where the temperature is constant at 35^{o}C, while the occupancy rate varies. The system then operates according to the following chart:

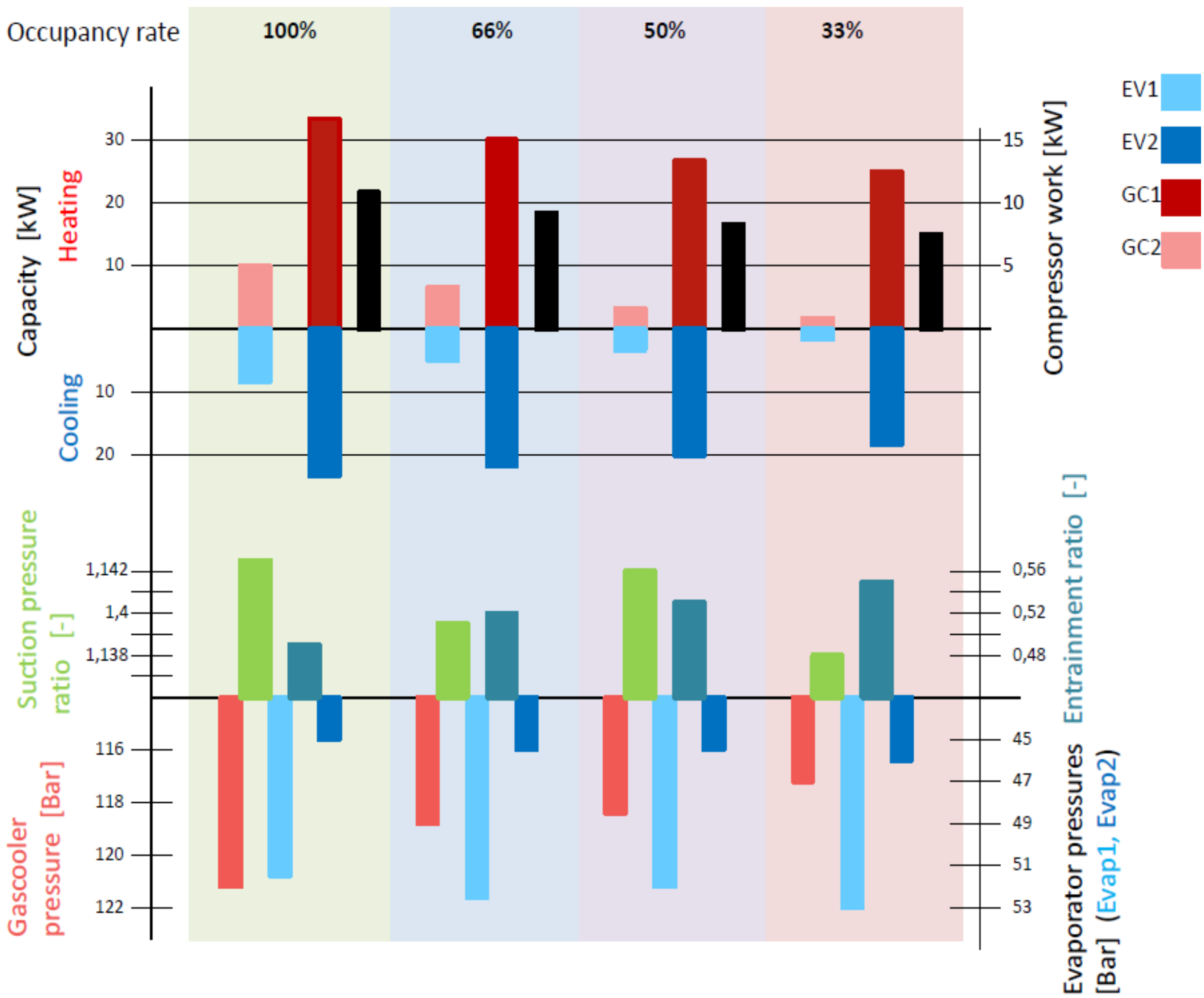


Figure 6-5: System operation at constant ambient temperature and varying occupancy rate.

See Table 11-4 for exact values of the simulation parameters.

The uppermost part of the chart shows the capacities of the different heat exchangers. It is clear that the total cooling demand of capacity decreases as the occupancy rate decreases, due to less ambient air needing treatment. The decrease in capacity is relatively equal for all the 4 heat exchangers. The main share of the cooling demand is covered by evaporator 2, which cools the recycled compartment air. Accordingly, the main share of the heat rejection is done in gas cooler 1, which is cooled by bypass ambient air. The reason for this is that the air flow rates passing through these heat exchangers are far greater than for gas cooler 2 and evaporator 1. The compressor work is also decreasing with decreased occupancy, but not as much as the decrease in heat exchanger capacity. This leads to a slight decrease in COP.

The lowermost part of the chart shows the pressures, the entrainment ratio Φ, and the suction pressure ratio ∏. Studying the scale of the axis of the suction pressure ratio is discovered that this is almost unchanged for all the occupancy rates. Even though the evaporator pressures decrease or increase for the different occupancy rates, the difference between the pressures is almost unchanged, leading to almost constant pressure ratio. This is due to that the ambient temperature is constant for the 4 situations.

The mass entrainment ratio increases with decreased occupancy rate. Both the suction flow rate and the motive flow rate are being reduced with the decreasing load, but the motive flow rate is being a bit more reduced than the suction flow rate, thus leading to increased mass entrainment ratio. The optimal gas cooler pressure is also decreasing with the decreased capacity demand.

Case 2

The second scenario is the opposite of the first, namely that the occupancy rate is constant, but the ambient temperature is decreasing. The occupancy rate is here 33%, and the temperature ranging from maximum to minimum temperature. For the two first cases, 35C and 20C, the system operates in cooling mode, while for the two last cases, 5C and -17C the system operates in heating mode. The system then operates according to the chart below:

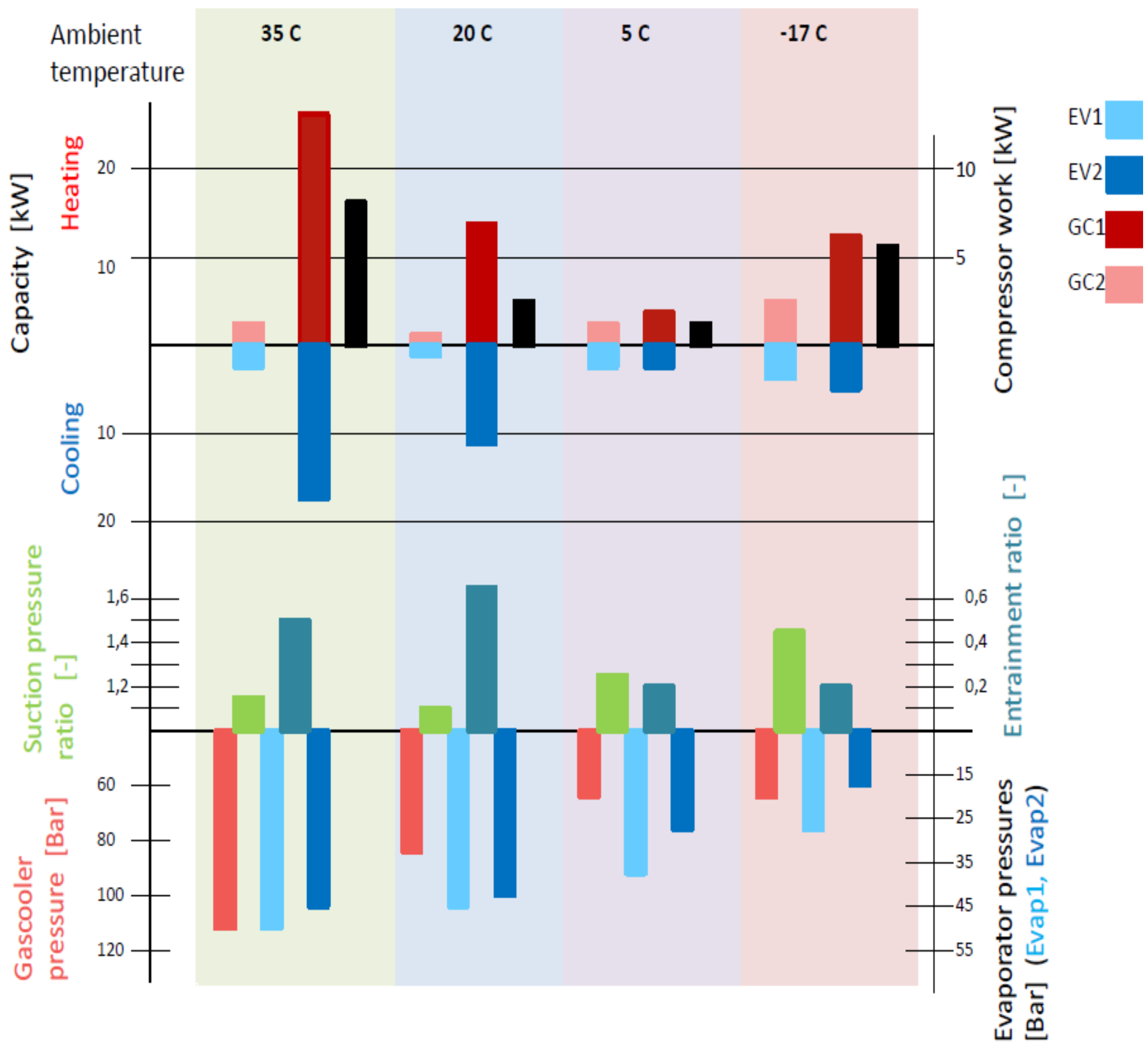


Figure 6-6: System operation at constant occupancy rate of 33% and varying ambient temperature.

See Table 11-5 for exact values of the simulation parameters.

As for the first case, the evaporator 2 and gas cooler 1 have the largest cooling and heating capacity respectively. In cooling mode when the capacity demand decreases, the necessary compressor work decreases more than the capacity, leading to higher COP (2.92 vs. 4.98). This is equal for heating mode, where the capacity with falling temperatures increases less than the compressor work, leading to reduction in COP (4.95 vs. 3.92).

The suction pressure ratio is decreasing with decreasing load, and is in general higher for heating than cooling mode. This can be explained by the pressures levels, which are in general lower for heating mode, so a certain increase of the evaporator pressure difference will give a relatively bigger increase in the pressure ratio than what would be the case for cooling mode.

The evaporator pressures levels are decreasing with decreasing ambient temperature. In cooling mode, the difference between the evaporator pressures are decreasing with decreased capacity, but in heating mode the pressure difference is slightly increasing with decreased capacity, and the difference is in general larger than for cooling mode.

In cooling operation, the mass entrainment ratio is increasing as the ambient temperatures and capacity demand decreases. This is, as for case 1, due to the general decrease in flow rate with decreasing capacity demand. The same is happening to the mass entrainment ratio in heating mode, where the mass entrainment ratio increases with increased ambient temperature, due to decreased capacity demand. The difference is not as big as for cooling mode, due to that the difference between the gas cooler pressure and evaporator pressure is in general much lower for heating mode, and the mass entrainment ratios thus also in general lower.

The optimal gascooler pressure is for cooling mode decreasing a lot according to capacity demand and ambient temperature. In heating mode this is only decreasing slightly with increased demand. This is also due to that the operation is subcritical in heating mode and the variation of gas cooler pressure is thus a lot smaller.

6.5 Input for simulations of annual performance in Modelica

The purpose of the simulations is to find the COP at varying operational conditions, and from this be able to calculate the annual compressor power consumption of the M-HVAC for the train compartment.

In order to find the operational conditions for the simulations, an analysis was performed of the energy flow calculations in order to find design conditions that would give a good approximation of the ambient conditions through the year. The parameters needed for the simulations in Modelica that are not given by regulations are the following:

- Ambient air temperature
- Relative humidity of ambient air
- Relative humidity of compartment air
- Mass flow of recycled compartment air
- Mass flow of bypass ambient air

The range of power demand found in the heat load calculations was divided into intervals of 5kW. For each city and occupancy rate, an average value of the above listed input parameters was calculated within each load interval. Below is a graphical display of the average values of the input parameters for each load interval for the five cities with a 100% occupancy rate. Positive value of load is cooling load, and negative value indicates heat load.

See appendix 11.2.3 for example of design points for Frankfurt with a 100% occupancy rate.

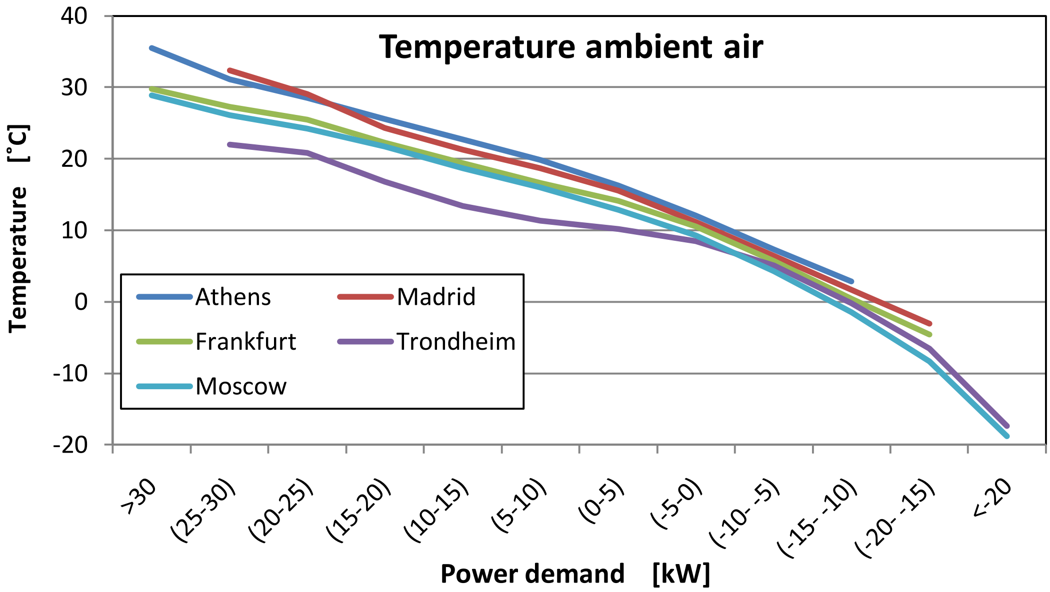


Figure 6-7: Average ambient temperature at varying demand for full compartment

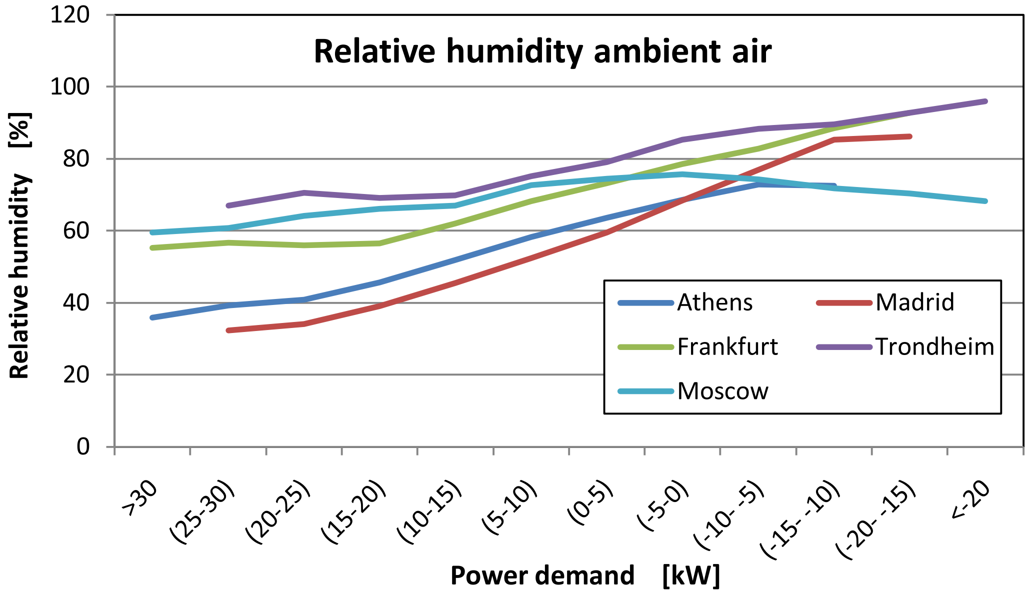


Figure 6-8: Relative humidity of the entering ambient air at varying demand for full compartment

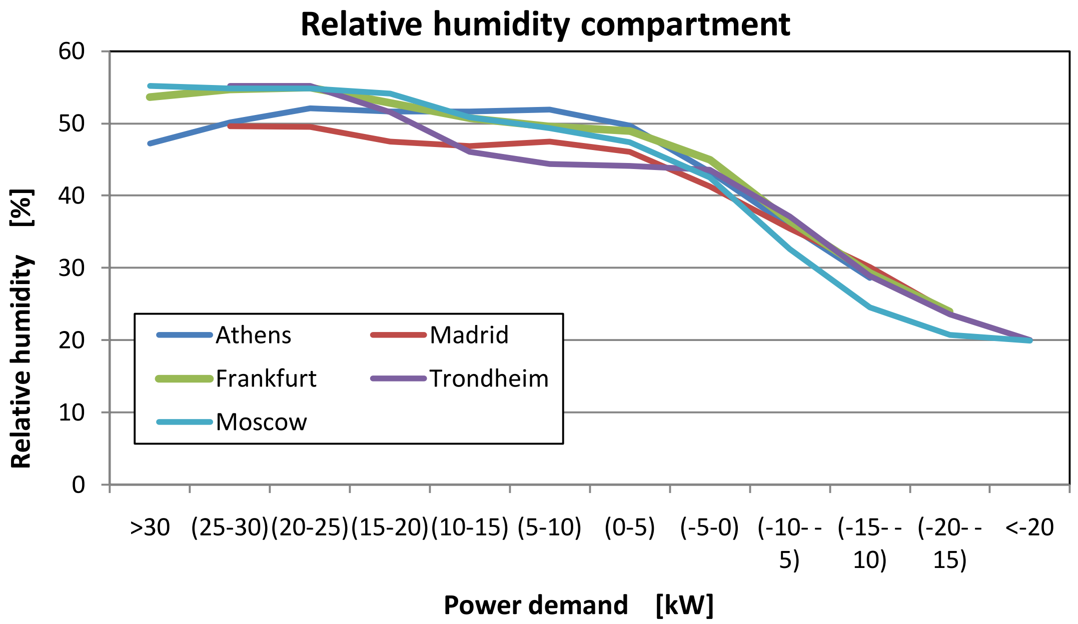


Figure 6-9: Relative humidity of compartment air at varying demand for full compartment.

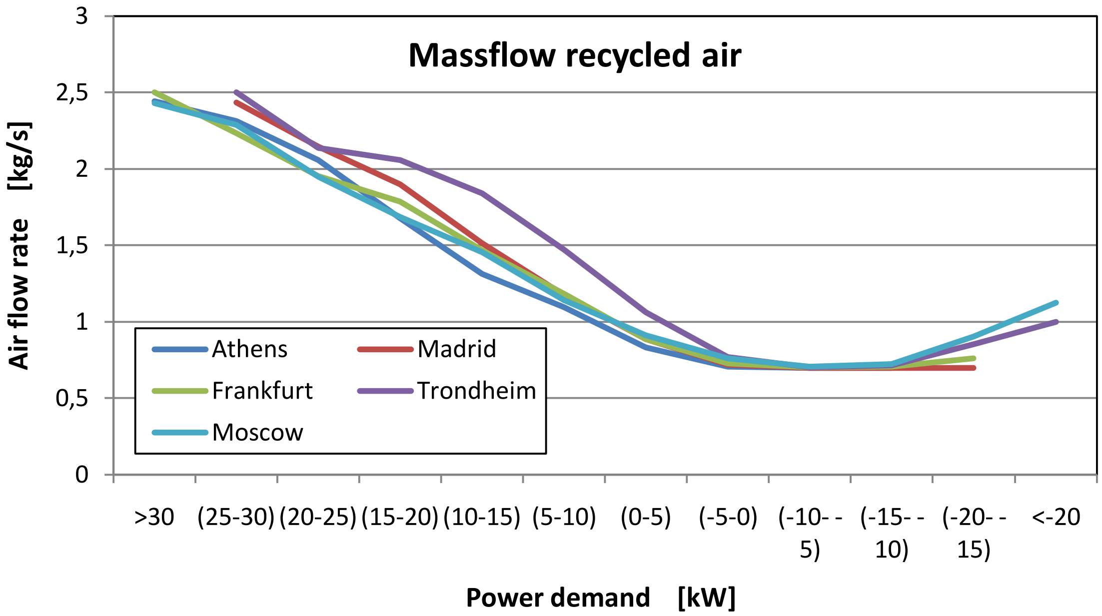


Figure 6-10: Mass flow rate of recycled compartment air at varying demand for full compartment.

Due to climatic variations, the highest heating- and cooling demands are not represented by all the cities. Athens, Frankfurt and Madrid do not have heating demand above 20kW, while Trondheim and Madrid do not have heating demand exceeding than 30KW.

As visible in the charts, there is a clear pattern for the 5 cities for all the input parameters. Frankfurt represents the mean value for all the parameters for almost all the power demand intervals, and can be used as a good representation for the operational conditions. Values from Moscow are used for the maximum heat power demand, since Frankfurt in not represented in that interval.

The last unknown input variable, the mass flow of bypass ambient air, is adjusted according to a balance between obtained COP and necessary fan power.

See appendix, 11.3.3 for further explications on finding the air flow rate of bypass air.

6.6 Results from simulations

Using the input parameters of the previous section, the following performance results were obtained for the different climate conditions and occupancy rates:

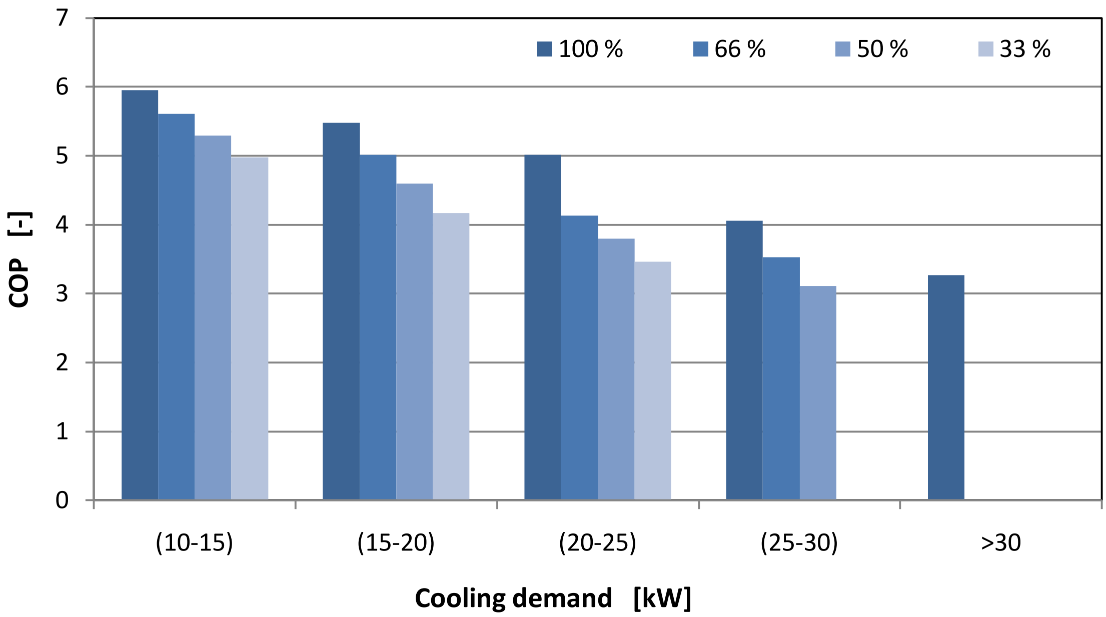


Figure 6-11: COP obtained for varying cooling load and occupancy rate

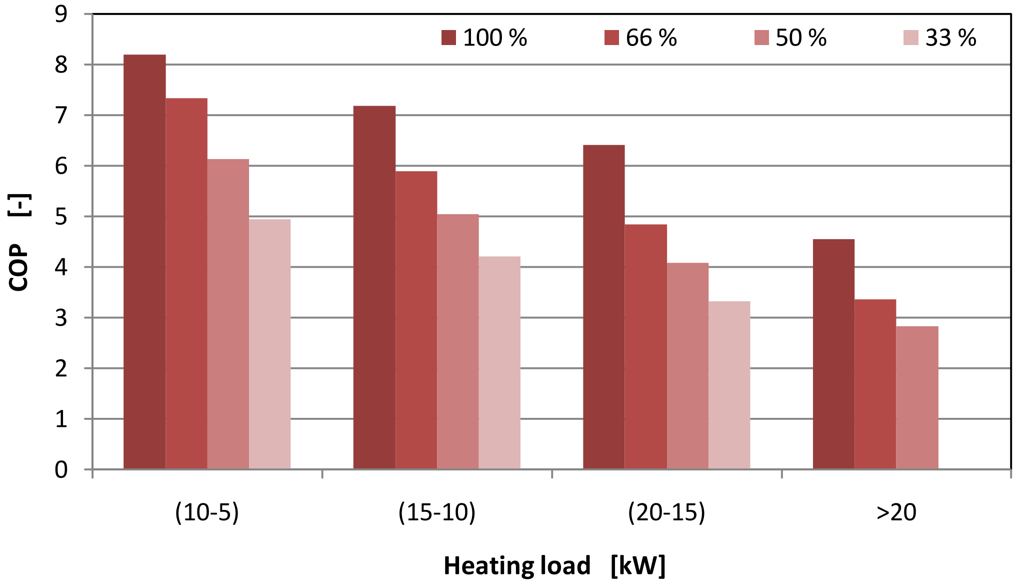


Figure 6-12: COP obtained for varying heating load and occupancy rate

The exact values can be found in Table 11-2.

The COP obtained for both heating and cooling are in general very high. Nevertheless, the increase in COP as the load decreases is realistic. One reason may be the heat transfer correlations of the heat exchangers. For the lowest loads in both cooling and heating mode, Modelica did not allow for the heat exchanger correlations models used in the design cases (Max cool/heat.) so constant heat transfer coefficients were used. Even though the heat transfer coefficients for those cases are found in HxSim, the COP is still very high.

For the heat load interval of 5-10 kW, the compressor frequency is very low, and the system is oscillating. This may also lead to unrealistic values of COP.

The COP decreases as the occupancy rate decreases. The reason for this can partly be seen from the graph below, where the ambient temperature is varying according to load interval and occupancy rate:

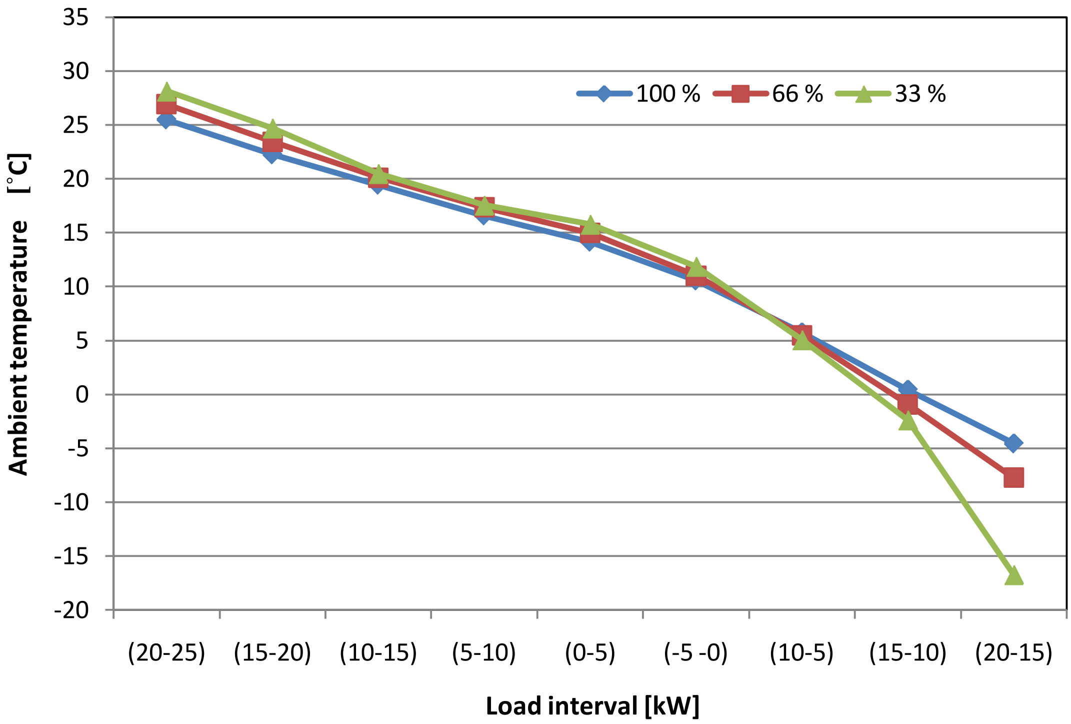


Figure 6-13: Average ambient temperature for the different load intervals and occupancy rates.

In cooling mode the average temperature of one load interval is increasing with decreasing occupancy rate. At the same time less ambient air is coming in, and thus less air exiting the compartment to cool gas cooler 2. The CO_2 is therefore being less cooled after the gas coolers, leading to a decrease in performance.

This is similar for heating mode, where the temperatures are in general lower for the same load interval at decreasing occupancy rate. The decrease in average temperature for each load interval is even higher in heating mode than the increase of temperature in cooling mode. With less heated compartment air exiting through evaporator one, the CO_2 mass flow has to be increased to cover the demand.

Combining the results from Figure 6-13,Figure 6-12 and Figure 6-11, the COP is obtained as a function of ambient temperature instead of heat load:

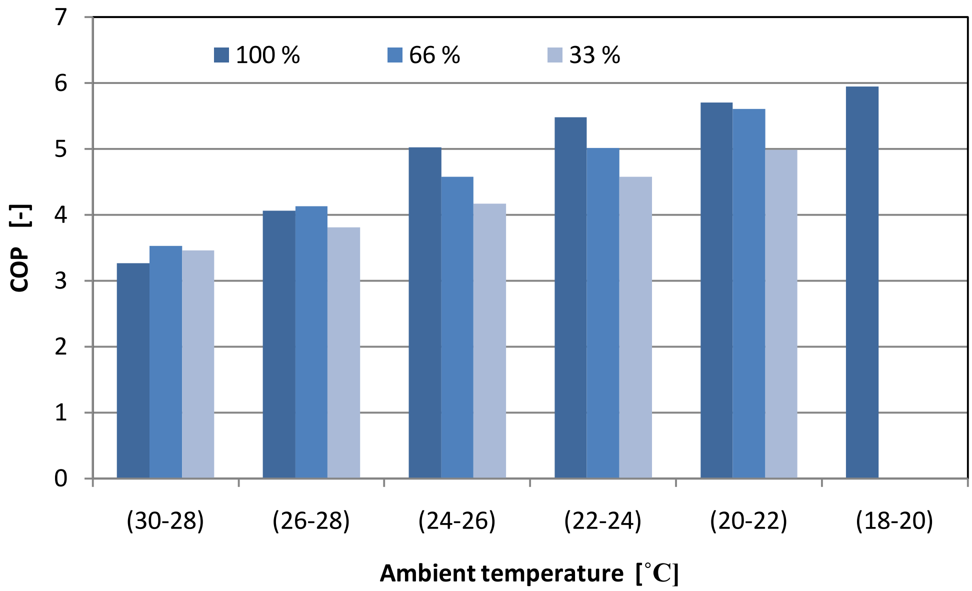


Figure 6-14: COP obtained for varying ambient temperature and occupancy rate

To check how applicable the simulation results are for the other cities in the case study, a sample is done in cooling mode for Athens for the highest load intervals. The following graph shows that the difference is about 10% for the highest cooling load, and the difference is decreasing with decreasing cooling loads.

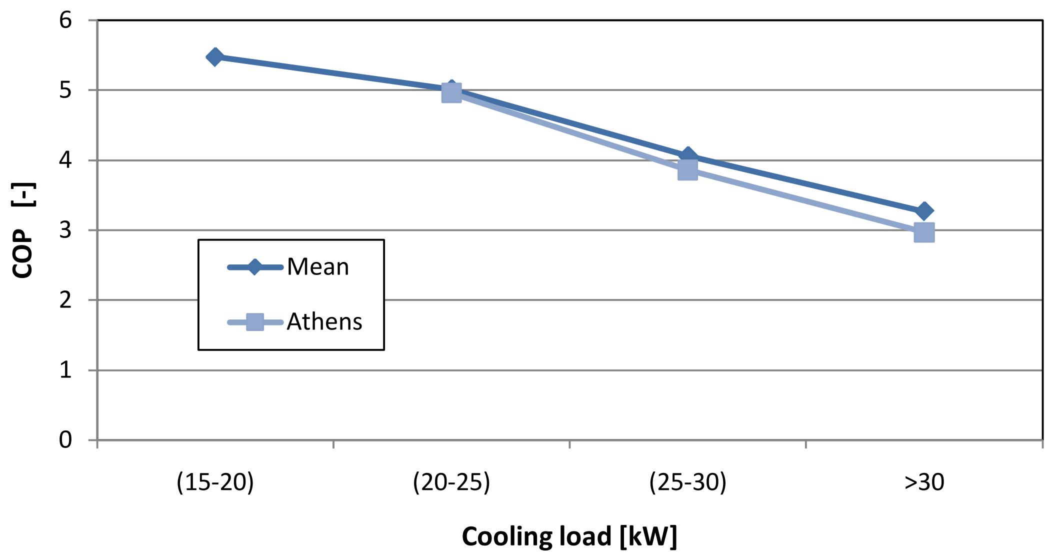


Figure 6-15: Difference in COP for Athens versus the mean. (Frankfurt)

Looking at the COP for heating load below, the difference between the mean values (Frankfurt), and the extreme values (Moscow) is also about 10 %. Since Moscow also represents the design point for max load, there is no difference between mean and extreme case for the maximum heat load interval.

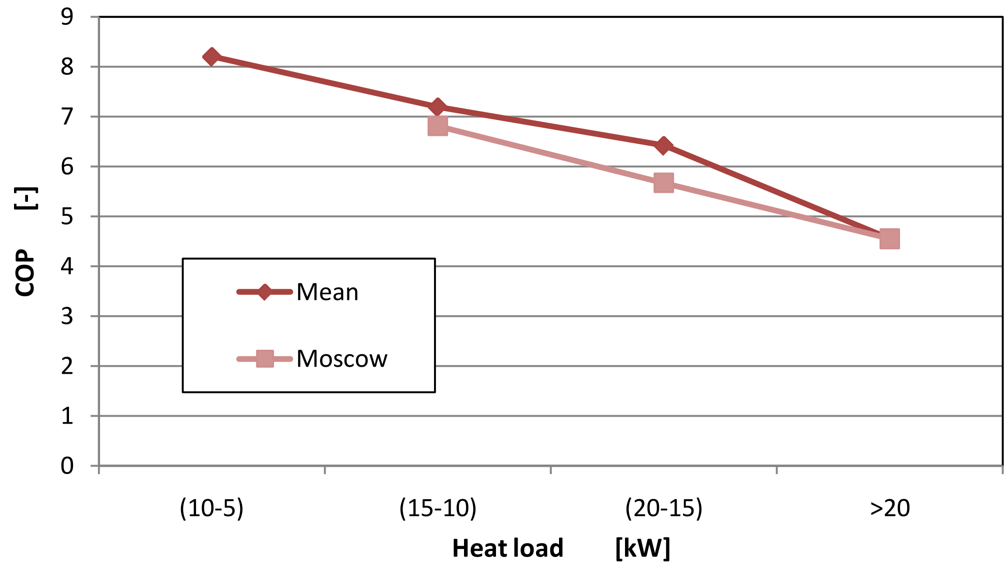


Figure 6-16: Difference in COP for Moscow versus the "mean climate", Frankfurt.

6.7 Total results of energy calculations

By linking the COP results and the heat load calculations hour by hour, the total annual energy consumption by the compressor is obtained, and the result is as stated by the following table and chart:

Table 6-2: Results annual demand vs. consumption

	Frankfurt	Trondheim	Madrid	Moscow	Athens
Cooling demand [MWh]	10,94	4,38	25,13	9,92	32,45
Heating demand [MWh]	19,48	30,45	8,94	31,40	2,47
Total energy demand [MWh]	30,42	34,83	34,06	41,32	34,93
Energy consumed by compressor [MWh]	5,66	6,57	6,62	8,16	7,06

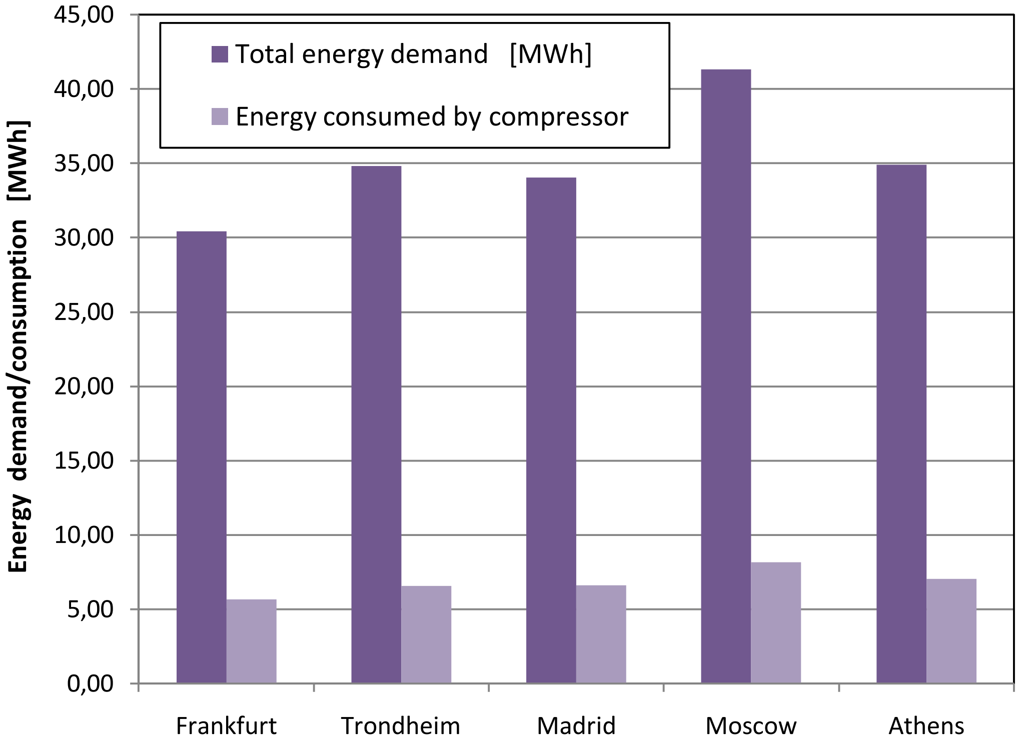


Figure 6-17: Annual energy demand provided by the M-HVAC vs. the energy delivered to the energy consumption by the compressor.

It is evident that with the in general high COP values obtained for all ambient conditions, the annual energy saving is very high. For all 5 cities the energy saving is about 25MWh, or about 80%. The exact numbers are listed in the table below:

Table 6-3: Annual energy consumption and savings

	Frankfurt	Trondheim	Madrid	Moscow	Athens
Total energy demand [MWh]	30,42	34,83	34,06	41,32	34,93
Energy consumption compressor [MWh]	5,66	6,57	6,62	8,16	7,06
Annual energy savings [%]	81,4	81,1	80,6	80,3	79,8

The COP found in simulations is only the "local" COP and not the system COP. This means that the fan work and compressor losses are not included. The fan work is often quite substantial, and highly depending on the air flow rate. (See Figure 11-2 and Figure 11-3 for fan power consumption for gas cooler 1 and evaporator 2.) The airflows of entering and leaving ambient/compartment air are so small that the fan power needed has no effect on the total fan power consumption.

To evaluate the increase in total power consumption by the fans, the fan power demand is investigated for the 4 scenarios of case 2, chapter 6.4. Results for fan power consumption are found in HxSim for the different heat exchangers and air flow rates, so a new COP could be calculated:

Table 6-4: Modified COP with fan power demand included

T ambient [^oC]	35	20	5	-16
Power demand cooling [kW]	21,4	12,2	7,6	17,7
Compressor power [kW]	7,55	2,44	1,536	5,367
COP original [-]	**2,84**	**4,98**	**4,95**	**3,32**
Fan power GC1	1,322	0,1638	0,122	0,566
Fan power EV2	2,92	1,291	0,067	1,29
Total power consumption [kW]	11,792	3,8948	1,725	7,223
COP new [-]	**1,81**	**3,13**	**4,4**	**2,45**
Decrease [%]	**0,36**	**0,37**	**0,11**	**0,26**

For all four operating temperatures, the COP decreases considerably when the fan power is included. The biggest decrease in COP is for moderate temperatures in cooling mode. Since the compressor work is small, the relative increase in power consumption is high, but the percentage decrease in COP is almost equal to the scenario of 35^oC.

In heating mode, the necessary air flows of bypass and recycled air are smaller, so the fan power consumption is in general a lot lower than for cooling mode. The relative decrease in COP with fan power included is increasing with lower ambient temperatures.

It is evident that this will increase the annual power consumption drastically, but at the same time, this fan power would have come in addition no matter what kind of heating or cooling system that were to be applied. The fan power calculated is also only the theoretical fan power, so the motor losses and efficiencies etc. of the fans are not included.

6.8 Comments and discussion of the simulation process and results

When both heating and cooling is needed at the same time, the mode is chosen according to the net total requirement. E.g. if the ambient air needs to be heated and the recycled air needs to be cooled, the mode will be chosen according to the largest load (in absolute value). The model also does not adjust the loads on the heat exchangers exactly according to the energy flow calculations. This is due to that the ejector itself adjusts the evaporator pressures and mass flows according to the ambient temperatures and the power demand. One would thus have to modify the heat exchangers for each ambient temperature in order to get the individual heat exchanger loads according to the energy balance calculations. But as long as the total power demand is covered, it is of less importance that the share of duty is as calculated.

In heating mode, the heat transfer coefficients had to be constant for the model to run. Even though they are according to values generated in HxSim for every case, Modelica handles heat transfer in a different manner than HxSim, as there are various heat transfer correlations possible. This may lead to a higher heat exchanger performance than what is realistic.

The system is struggling to operate when approaching critical point. This is due to that the heat exchanger correlations are not working well since the heat capacity is reaching towards infinity around critical point. It is therefore not possible to decrease the pressure from transcritical to subcritical process in one simulation.

For the lowest heating loads, the result of heat exchanger capacity is oscillating as the simulation progresses. This makes the simulations advance very slowly, and it is clear that it is not a preferred operating condition. A solution to this could be to divide the M-HVAC into different modules to reduce the maximum required capacity of each module. This would lead to less part load and thus more steady operating conditions.

The optimal mode of operation in heating mode is found to be subcritical. If the glycol heat exchanger is to be taken into use (Ref Figure 6-1), a transcritical operation would be necessary in order to provide sufficient heat for floor heating, since a higher gascooler temperature is required for the glycol circuit than for heating of air.

Regarding the profile of the occupancy rate chosen for the case study, it is probably not representative for all cities, as this would be affected by the size of the population and the general diurnal rhythm of the specific location. This would nevertheless only affect the annual power consumption of the M-HVAC and not the COP results. The assumptions made for the heat flow balances would nevertheless affect the necessary power demand as well as the COP results, since the ambient conditions would change for the same load interval and thus change the input values for simulations.

What is not taken into consideration is that there would still be a power demand if the compartment is empty. It is assumed that the compartment will always be populated during operation, and that the M-HVAC will be turned off while not in operation. This would thus lead to an increased power demand in order to reach desired compartment conditions when tuning on the M-HVAC. Nevertheless, the current compressor is designed to handle this extra load at startup.

7 Comparison of MPE and FT gas coolers

The important feature when designing an M-HVAC is to reduce weight. Less weight leads to higher LCCP, due to that the reduced mass requires less energy to transporting the module. (Ref chapter 1.3)

The fin-and-tube gas coolers used in simulations were compared to a set of MPE gas coolers. The reason why this is not investigated for the evaporators is the earlier mentioned difficulties with the distribution of liquid CO_2 between the different channels.

To make a comparison possible, the MPE gas coolers were designed to give the same performance, at equal temperature drop of CO_2 through the heat exchanger, and with equal or less fan power demand. The size and weight was then compared, and is listed in the following table ()

Table 7-1: Comparison of size and weight for MPE and fin-and-tube,(FT), gas coolers.

	GC1 FT	GC2 FT	GC1 MPE	GC2 MPE
Core Depth [m]	0,4	0,36	0,28	0,385
Height [m]	0,6	0,6	0,36	0,36
Core width [m]	0,42	0,42	0,4	0,4
Weight [kg]	**106**	**171**	**25,9**	**22,33**

See appendix 11.3.5 for detailed specification of the MPE gas coolers

In addition to the MPE gas coolers being a bit smaller, the most apparent and important difference is the reduced weight. The weight of the two gas coolers together is 83% lower for MPE, and the total weight of the heat exchangers is reduced by 50%. This would reduce the traction power especially at startup of driving the train, and in general the mass part of LCCP is reduced.

Even though this weight reduction is significant, the increased costs of MPE heat exchangers have to be considered. The M-HVAC systems in use today with HFC refrigerants often have fin-and-tube heat exchangers installed. Thus it would only be necessary to modify the fin-and-tube heat exchangers already in use. Meanwhile with MPE, a total replacement would be necessary, and the investment cost would be a lot higher.

For many M-HVAC applications, the size and weight restriction would be the deciding factor. For the module of the train compartment, the fin-and-tube heat exchangers were within the size limitations, and there were no upper weight limit. Depending on the purchase costs, the reduced power consumption and following increased mass part of the LCCP is not likely to make up for the increased price. Since the performance is equal for the two types, the annual energy used to operate the M-HVAC would also be equal for both types. Thus the indirect part of LCCP, which is linked to energy use due to operation of the M-HVAC, would remain the same.

Meanwhile for automobiles, the weight of the M-HVAC would account for a larger amount of the total weight of the car, so it is a greater possibility of the decrease in fuel consumption needed to transport the system outweighing the increased purchasing cost.

8 Conclusion

There have been concerns about the performance of CO_2 in high ambient temperatures, but it is proven that it is well adapted for use in M-HVAC's also for higher temperatures. Due to some preferable thermodynamic properties, a CO_2 M-HVAC can be constructed more compact than when using other refrigerants, while obtaining the same capacity. This is convenient both for the space limitations in a vehicle and for the mass impact of the LCCP due to the energy required for transportation of the module. Not only from a cost perspective is it of importance that the M-HVAC is energy efficient. The indirect LCCP of the system will be significantly improved by the reduced energy consumption needed to operate the system.

Both in cooling and heating mode the performance of the CO_2 turntable prototype heat pump was very positive, The COP is in general higher for heating mode than for cooling mode, and highest for the most frequently occurring ambient conditions.

The COP is depending on the load, and decreases with reduced occupancy rate. For cooling mode the COP ranged from 3.1 to 6. For heating mode it ranged from 8.2 to 2.8. With the occupancy rate profile chosen for the case study, the annual energy savings is about 80% for all the 5 cities of the study.

It was found that the fan power requirement is highly depending on the air flow, and it reduced the total COP by 10 to 40%, depending on ambient temperature and load. Still the performance is positive, and leads to significant reduction in annual energy consumption.

The total weight of the heat exchangers was reduced by 50% by using MPE gas coolers instead of the regular fin-and-tube heat exchangers. For the case study, the fin-and-tube heat exchangers are found apt, and the increased purchasing costs of the MPE gas coolers are not likely to be justified by the decreased electricity consumption needed to transport the system.

Based on the simulation results obtained from Modelica, a reversible heat pump with CO_2 seems like a very good alternative to the conventional refrigerants used in M-HVAC's today. Since there are currently not installed any CO_2 heat pumps in European trains, proving that a CO_2 M-HVAC system for train is more energy efficient and has lower LCCP would be a valuable incentive for future replacements of the conventional refrigerants.

9 Further work

- Account for the increased work for startup of the M-HVAC, e.g. in the morning when it has been turned off all night. It requires extra power to obtain the desired compartment conditions after the HVAC system has been turned off for a long time.

- It should be looked into what would be the most energy efficient way of maintaining the air conditions when the train is not in operation, whether it is partly reduction of capacity throughout the night, or turning the HVAC system off totally at night and turning it on with extra capacity for a while before startup of the train.

- Include the glycol heat exchanger in energy calculations and simulations. The load for the two air cooled gascoolers is then lowered, and the calculation model would have to be modified. The process would then be transcritical for all ambient temperatures.

- Evaluate the part load regulation system for the compressor. A combination of regulation of rotational speed and compressor volume could give more steady operational conditions for low loads.

- Build a prototype of the turntable heat pump with the right dimensions and capacity.

- Convince NSB and other railway operators that switching to CO_2 M-HVAC systems is the only sustainable way to go.

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10 Draft for a Journal Paper:

Air Reversing CO_2 HVAC system for Use in Public Trains

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Summary

A possible system for heating and cooling of a train compartment has been modeled and simulated. This system is a turntable prototype developed at Sintef, which is reversing the airflows to provide either cooling or heating. It has two gascoolers and two evaporators for separate treatment of ambient and recycled air. The plate is rotated 180˚ to switch from heating to cooling mode.

CO_2 has large potential for expansion work, due to the normally large throttling losses for high ambient temperatures. An ejector has therefore been implemented in the heat pump circuit. The turntable prototype is modeled by the simulation tool Modelica, and it is investigated how this ejector system adjusts the performance to varying ambient conditions and power demand.

Simulations were performed for the air reversing heat pump based on the heating and cooling demand of 5 cities, covering a wide climatic diversity. The COP values obtained are very positive. The COP for cooling mode ranged from 3.1 to 6, depending on load and occupancy rate. The COP in heating mode ranged from 8.2 to 2.8. The COP decreases with reduced occupancy rate. With the occupancy rate chosen for the train compartment, the annual energy savings is about 80% for all the 5 cities of the study.

Background

CO_2 is an environmentally friendly refrigerant that has a no global warming potential when used as refrigerant. The current refrigerants used for air conditioning in public transport are HFC's with high global warming impact. The heating is provided by direct electrical heating.

Compared to the total emissions from automobiles, the HFC emissions from public sector are quite small, but the emissions per system are significant, as the total charge per system is a lot higher for trains than for cars. (Schwarz & Rhiemeier, 2007) Also very important in public trains is the ability to do heat pumping in the winter where you can reduce the power consumption compared to using electricity. As long as the air conditioning part is also done efficiently this could be very promising. (Nekså, 2010)

Because of the need for all special designed components for CO_2 systems for trains, the investment costs are considered to be up to 30% higher than with conventional refrigerants. Combined with the comparatively leak tight systems and small refrigerant charges, this gives high reduction costs per tonne reduced CO_2 -equivalent, and is per 2007 not found cost efficient. (Schwarz & Rhiemeier, 2007)

This cost will be much lower with the development of more efficient CO_2 technology. There is as of today no commercialized CO2 heat pumps for transport sector. (Morgenstern, 2008) With the current progress of development for the commercial refrigeration sector, it is likely that the soon-to-be commercialized components from that sector can be modified and applied for M-HVAC systems in the future, thus reducing costs. (Nekså, 2010)

A prototype air reversing CO_2 heat pump for public trains has been developed by Sintef. This system is reversing the airflows to provide either cooling or heating. It has two gascoolers and two evaporators for separate treatment of ambient and recycled air. The plate is rotated 180̊ to switch from heating to cooling mode.

A possible refrigeration circuit for the turntable heat pump has been explored, using a throttling valve to create two different evaporating pressures. Depending on the humidity of the ambient air, heating COP of between 3 and 6, and cooling COP of between 4.5 and 2 was obtained. It also showed an 82% reduction in the annual energy required for heating. (Hafner & Christensen, 2010)

CO2 has the problem of high throttling losses when working with high ambient temperatures, as is the case for air conditioning. During an experiment for a car air condition system with an ejector implemented, a COP improvement of 44% was achieved for CO_2, versus only 13% for R134a. (Hrnjak, 2006) An ejector is thus implemented in the circuit of the turntable prototype.

New cycle layout of the air reversing M-HVAC including ejector

Due to the low critical temperature, CO_2 has higher throttling losses than other refrigerants used for air conditioning. This can be reduced by implementing an ejector instead of the regular expansion valve in the circuit. An ejector is therefore included in the turntable M-HVAC for the train compartment.

The principal cycle with airflows in heating and cooling mode is shown below. Arrow pointing upwards means air entering the heat exchanger from the compartment, and arrow pointing downwards indicates air entering the compartment

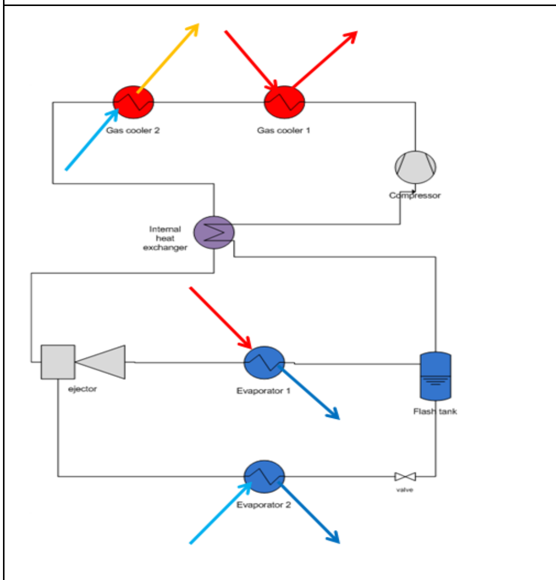
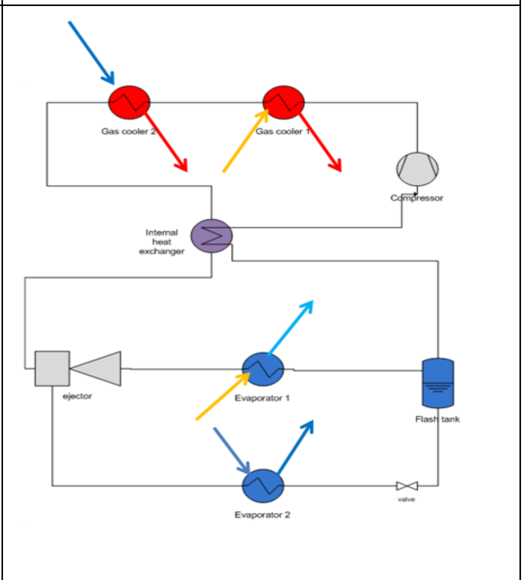
Cooling mode	Heating mode
	

Figure 1: The MAC layout with ejector in heating and cooling mode and with corresponding air flows

For the prototype M-HVAC, there are two evaporators, namely one that treats ambient air, and one that treats recycled air. The evaporator that treats ambient air will be at the intermediate pressure, or ejector pressure. The evaporating pressure of evaporator 2 will be lower, and is regulated automatically by the work recovery from the expansion process of the ejector.

Simulation of the M-HVAC

The simulation of the turntable M-HVAC is performed by using the tools Modelica, Til, Dymola , and HxSim.

-Dymola is used as a platform, and it is the interface that translates Modelica language to C through a C compiler. It makes use of modeling methodology based on object orientation and equations.

-TIL is a Modelica library for steady-state and transient simulation of thermodynamic systems. It uses TILMedia, which is an interface library to provide fluid and solid properties from various existing property databases to different applications.

-StateViewer is a tool used for graphical presentation of transient thermodynamic measurements or simulation data. The simulation files from Modelica can be transported to StateViewer to get a better understanding of the process performance.

-HxSim was used to find the configurations for the different heat exchangers. This is a simulation tool made by Sintef, and it is especially made for CO_2 heat exchangers that are cooled or heated by air.

The layout of the M-HVAC system in Modelica is shown in the following picture.

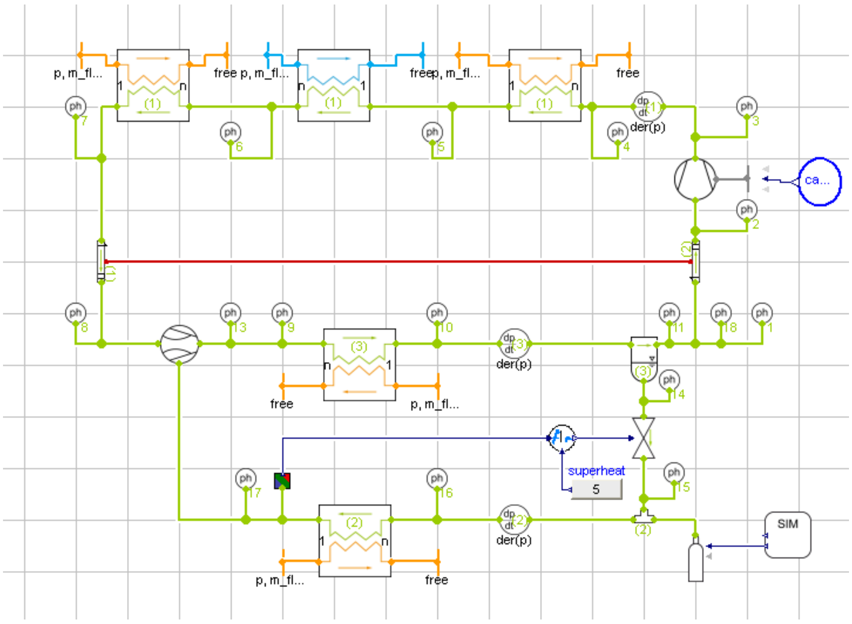


Figure 2: System layout of the M-HVAC in Modelica

The red horizontal line indicates the internal heat exchanger, and the orange flows are the air flows entering and leaving the compartment. In addition to the two air cooled gas coolers, a glycol heat exchanger is included to provide for floor heating. The floor heating possibility is not investigated further in simulations, as the air cooled gas coolers are designed to provide enough heat to cover the maximum heat load alone.

The principal difference of Modelica compared to previously used simulation tools is that it makes transient simulations. This provides for a more realistic simulation process and gives a more thorough understanding of how such a heat pump system in real life would be adjusting to varying operating conditions.

System process

Below is a pressure-enthalpy diagram of the heat pump process in cooling and heating mode:

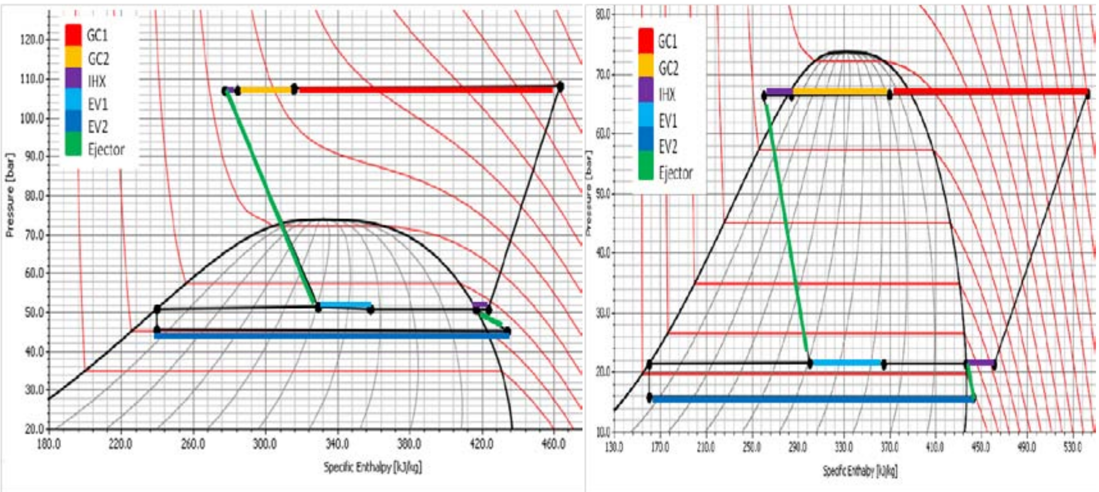


Figure 3: P-h diagram of the process in cooling and heating mode respectively

In cooling mode, the process is transcritical; i.e. that the heat rejection takes place in supercritical area, above the phase envelope. In heating mode, the optimal conditions are found for subcritical operation, as the glycol heat exchanger from the Modelica model is not included. The pressure difference between the two evaporators is depending on the ambient temperatures, power requirement and the work recovery of the ejector. In transcritical operation, the optimal gas cooler pressure is highly depending on the ambient temperature and power requirement.

Dynamics of the system

To get a better understanding of how the Modelica system adjusts to varying operating conditions, two different cases are chosen:

Case 1

First is investigated a scenarios where the temperature is constant at 35.7°C, while the occupancy rate varies. The system then operates according to the following chart:

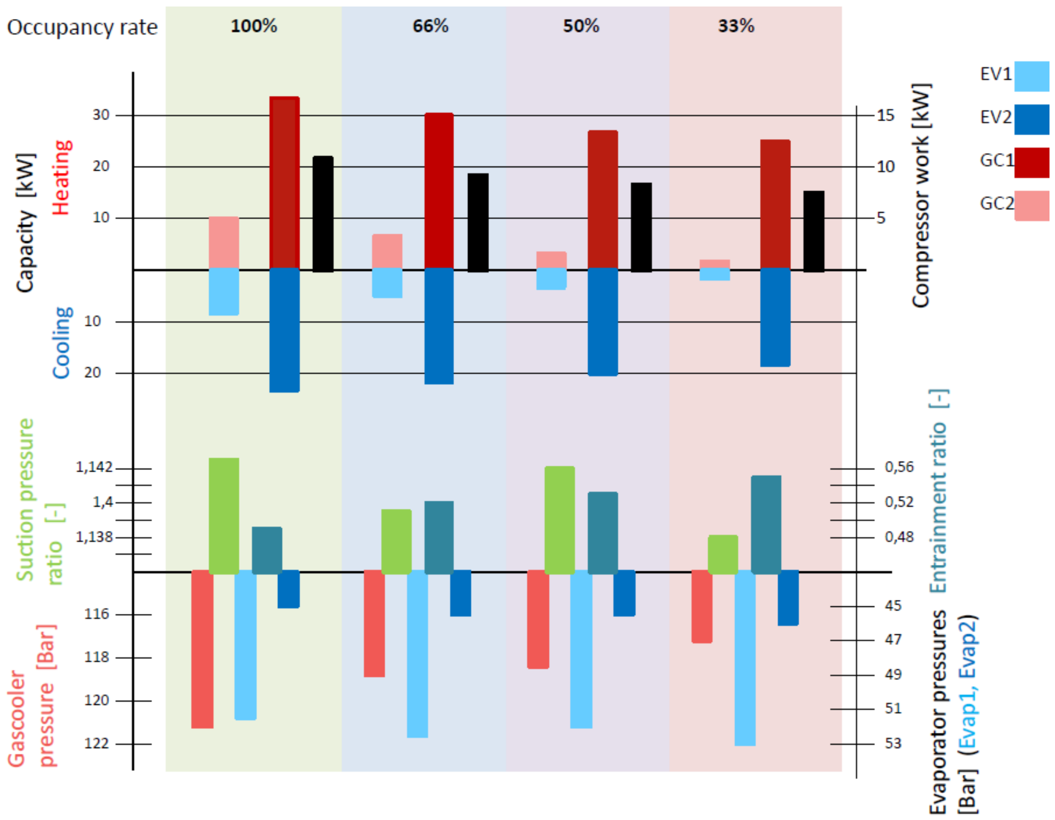


Figure 4: System operation at constant ambient temperature and varying occupancy rate.

The uppermost part of the chart shows the capacities of the different heat exchangers. It is clear that the total cooling demand of capacity decreases as the occupancy rate decreases, due to less ambient air needing treatment. The decrease in capacity is relatively equal for all the 4 heat exchangers. The main share of the cooling demand is covered by evaporator 2, which cools the recycled compartment air. Accordingly, the main share of the heat rejection is done in gas cooler 1, which is cooled by bypass ambient air. The reason for this is that the air flow rates passing through these heat exchangers are far greater than for gas cooler 2 and evaporator 1. The compressor work is also decreasing with decreased occupancy, but not as much as the decrease in heat exchanger capacity. This leads to a slight decrease in COP.

The lowermost part of the chart shows the pressures, the entrainment ratio Φ, and the suction pressure ratio ∏. Studying the scale of the axis of the suction pressure ratio is discovered that this is almost unchanged for all the occupancy rates. Even though the evaporator pressures decrease or increase for the different occupancy rates, the difference between the pressures is almost unchanged, leading to almost constant pressure ratio. This is due to that the ambient temperature is constant for the 4 situations.

The mass entrainment ratio increases with decreased occupancy rate. Both the suction flow rate and the motive flow rate are being reduced with the decreasing load, but the motive flow rate is being a bit more reduced than the suction flow rate, thus leading to increased mass entrainment ratio. The optimal gas cooler pressure is also decreasing with the decreased capacity demand.

Case 2

The second scenario is the opposite of the first, namely that the occupancy rate is constant, but the ambient temperature is decreasing. The occupancy rate is here 33%, and the temperature ranging from maximum to minimum temperature. For the two first cases, 35C and 20C, the system operates in cooling mode, while for the two last cases, 5C and -17C the system operates in heating mode. The system then operates according to the chart below:

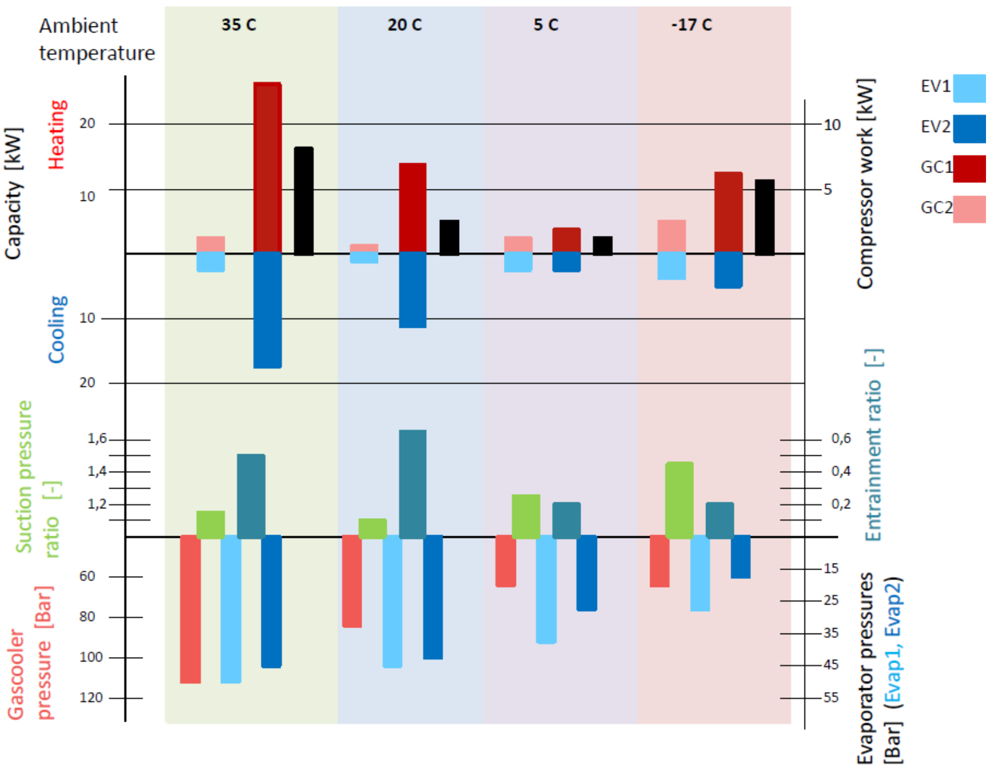


Figure 5: System operation at constant occupancy rate of 33% and varying ambient temperature.

As for the first case, the evaporator 2 and gas cooler 1 have the largest cooling and heating capacity respectively. In cooling mode when the capacity demand decreases, the necessary compressor work decreases more than the capacity, leading to higher COP (2.92 vs. 4.98). This is equal for heating mode where the capacity with falling temperatures increases less than the compressor work, leading to reduction in COP (4.95 vs. 3.92).

The suction pressure ratio is decreasing with decreasing load, and is in general higher for heating than cooling mode. This can be explained by the pressures levels, which are in general lower for heating mode, so a certain increase of the evaporator pressure difference will give a relatively bigger increase in the pressure ratio than what would be the case for cooling mode.

The evaporator pressures levels are decreasing with decreasing ambient temperature. In cooling mode, the difference between the evaporator pressures are decreasing with decreased capacity, but in heating mode the pressure difference is slightly increasing with decreased capacity, and the difference is in general larger than for cooling mode.

In cooling operation, the mass entrainment ratio is increasing as the ambient temperatures and capacity demand decreases. This is, as for case 1, due to the general decrease in flow rate with decreasing capacity demand. The same is happening to the mass entrainment ratio in heating mode, where the mass entrainment ratio increases with increased ambient temperature, due to decreased capacity demand. The difference is not as big as for cooling mode, due to that the difference between the gascooler pressure and evaporator pressure is in general much lower for heating mode, and the mass entrainment ratios thus also in general lower.

The optimal gas cooler pressure is for cooling mode falling a lot according to capacity demand and ambient temperature. In heating mode this is only decreasing slightly with increased demand. This is also due to that the operation is subcritical in heating mode and the variation of gas cooler pressure is thus a lot smaller.

Case study of a train compartment in 5 cities

The annual heating and cooling demand has been calculated for a Siemens Velaro train compartment and for 5 cities with a wide climatic diversity. These cities are Athens, Madrid, Frankfurt, Moscow and Trondheim. The M-HVAC will be turned off when the train is not in operation, which in this case is between 10pm and 6am. The assumption for occupancy rate, as well as the resulting energy demand that will have to be provided by the M-HVAC is shown below:

Hour of Day	Occupancy
6	One third
7	One third
8	Full
9	Full
10	Two thirds
11	One third
12	One third
13	One third
14	One third
15	Two thirds
16	Two thirds
17	Full
18	Full
19	Half full
20	One third
21	One third

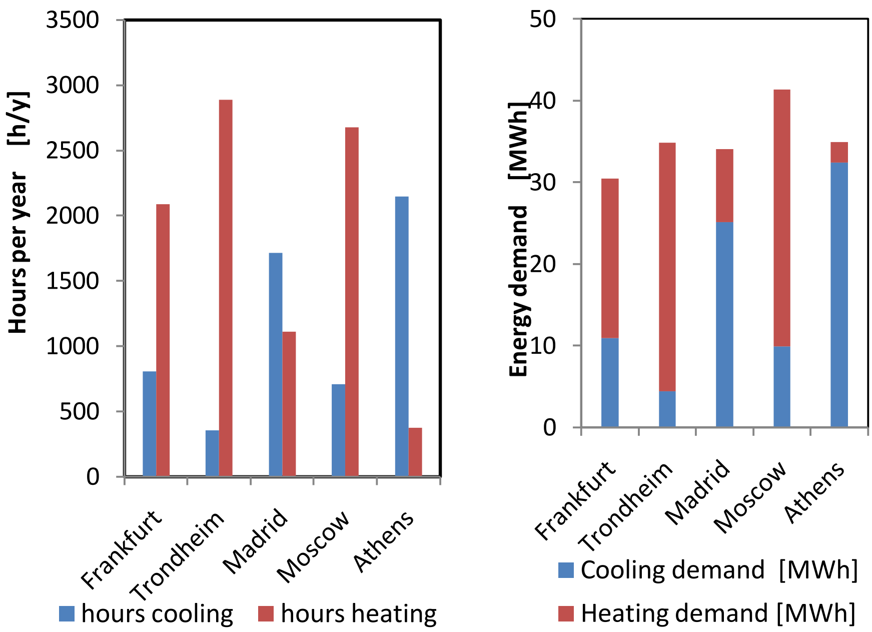


Figure 6: Occupancy Rate and following energy demand for the train compartment in the 5 cities.

Based on the results for heating and cooling demand, simulations were performed in Modelica in order to find necessary compressor work for the M-HVAC. Frankfurt was proven to have the mean values for climate conditions, so input parameters were chosen according to Frankfurt climate. For simulations of maximum cooling demand, Moscow climate is used.

Results from simulations

The following performance results were obtained for the different climate conditions and occupancy rates:

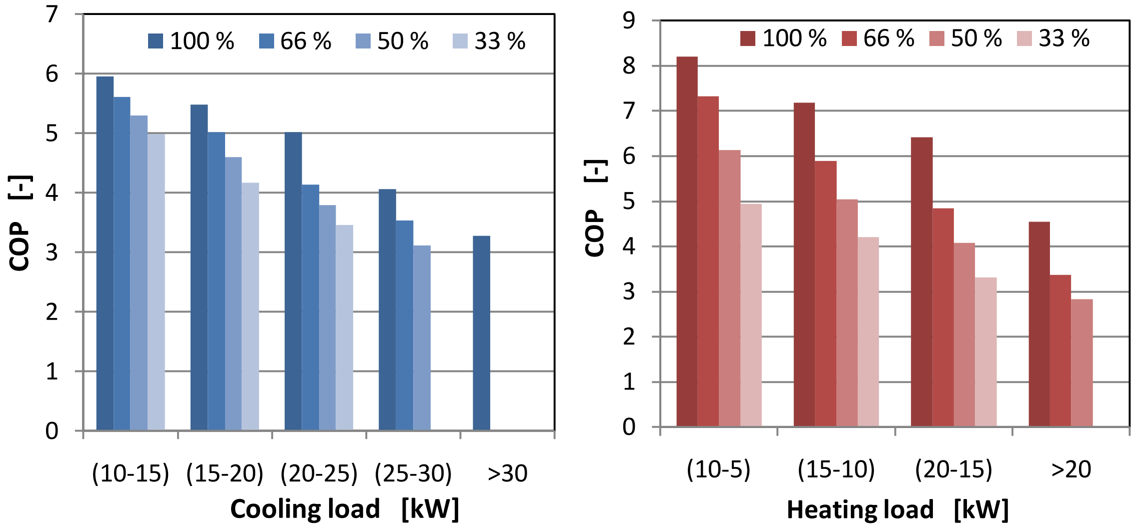


Figure 7: COP obtained for cooling and heating load for varying occupancy rate.

The COP obtained for both heating and cooling are in general very high. Nevertheless, the increase in COP as the load decreases is realistic. The COP decreases with decreased occupancy rate. This is due to that the air flow ratio is dependent on the occupancy rate. When less air exits through gas cooler2, the refrigerant is being less cooled down before entering the ejector, leading to reduced efficiency.

To check how applicable the simulation results are for the other cities of the case study, a sample is done in cooling mode for Athens for the highest load intervals. The following graph shows that the difference is about 10% for the highest cooling load, and the difference is decreasing with decreasing cooling loads.

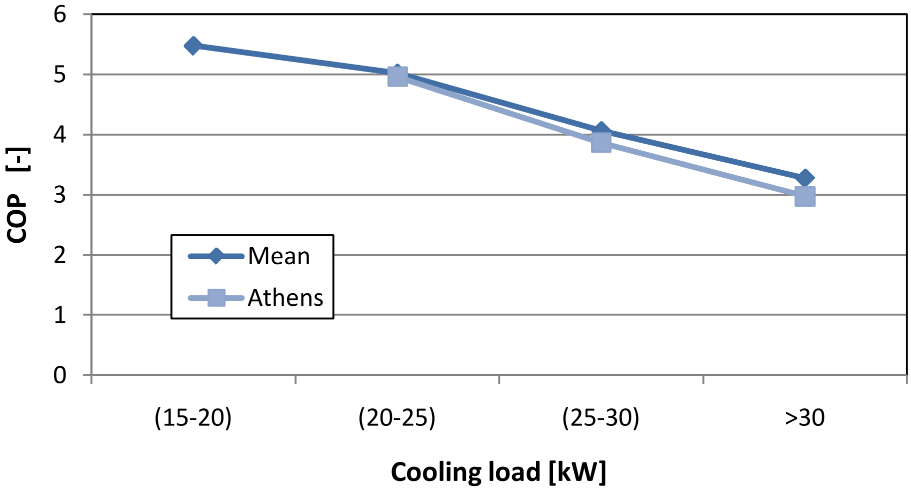


Figure 8: Difference in COP for Athens versus the mean. (Frankfurt)

Looking at the COP for heating load below, the difference between the mean values (Frankfurt), and the extreme values (Moscow) is also about 10 %. As Moscow also represents the design point for max load, there is no difference between mean and extreme case for the maximum heat load.

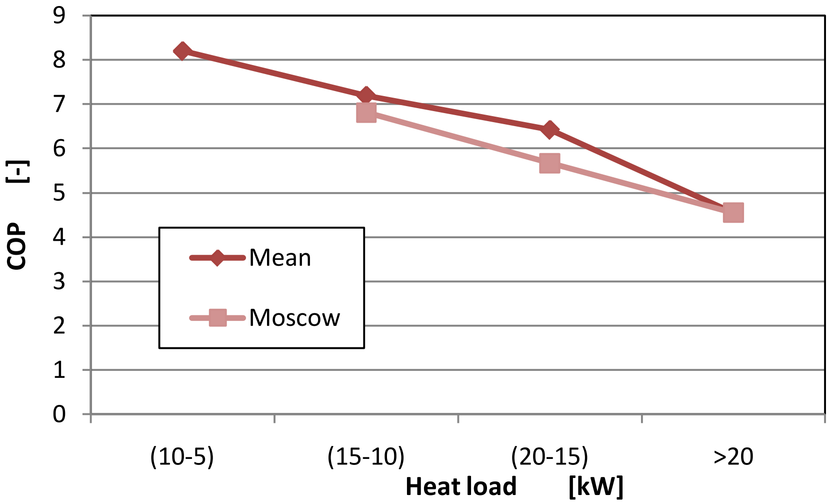


Figure 9: Difference in COP for Moscow versus the "mean climate", Frankfurt.

By linking the COP results and the heat load calculations, the total annual energy consumption by the compressor is obtained, and the results are according to the following chart:

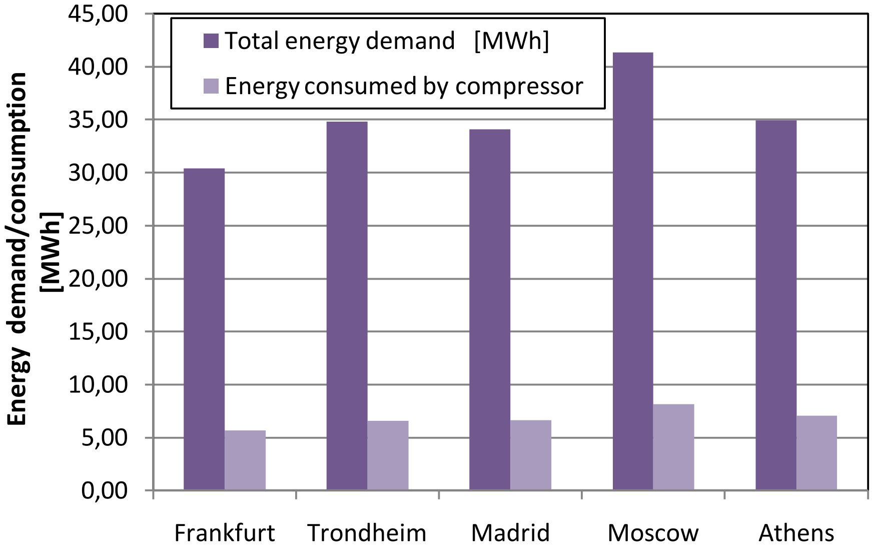


Figure 10: Annual energy demand provided by the M-HVAC vs. the energy delivered to the energy consumption by the compressor.

It is evident that with the in general high COP for all load intervals, the energy saving is very high. For all 5 cities the energy saving is about 25MWh, or about 80%. The exact numbers are listed in following table:

Table 1: Annual energy consumption and savings.

	Frankfurt	Trondheim	Madrid	Moscow	Athens
Total energy demand [MWh]	30,42	34,83	34,06	41,32	34,93
Energy consumption compressor [MWh]	5,66	6,57	6,62	8,16	7,06
Annual energy savings [%]	81,4	81,1	80,6	80,3	79,8

Conclusion

The COP values obtained are very positive. The COP for cooling mode ranges from 3.1 for maximum load to 6 for the minimum cooling load that the system needs to handle. The COP in heating mode is ranging from 8.2 to 2.8, and the variation is thus greater than for cooling mode. The COP is highest for the most frequently occurring ambient conditions, which leads to high energy savings. The COP also decreases with reduced occupancy rate. With the occupancy rate chosen, the annual energy savings is about 80% for all the 5 cities of the study.

Based on the simulation results obtained from Modelica, a reversible heat pump with CO_2 seems like a very good alternative to the conventional refrigerants used in M-HVAC's today. Since there are currently not installed any CO_2 heat pumps in European trains, proving that a CO_2 M-HVAC system for train is more energy efficient and has lower LCCP would be a valuable incentive for future replacements of the conventional refrigerants.

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11 Appendix

11.1 Model and procedure for energy flow calculations

Nomenclature for this section:

Massflows:

m_{amb}: ambient air flow

m_{rec}: recycled compartment air flow

m_{tot}: total air flow entering compartment

Absolute humidities:

X_{comp}: absolute humidity of the compartment air flow, and also the recycled air flow

X_{amb}: absolute humidity of the ambient air

X_{amb_inl}: absolute humidity of the ambient air entering the compartment

X_{mix}: absolute humidity of the mix of recycled and ambient air flows entering the compartment

Δx_{resp}: contribution to the absolute humidity from respiration

Enthalpies:

h_{comp} : compartment air enthalpy

h_{rec_inl}: enthalpy of the recycled air flow entering the compartment

h_{amb}: ambient air enthalpy

h_{amb_inl}: enthalpy of the ambient air flow entering the compartment

h_{mix}: enthalpy of the mix of the ambient air flow and the recycled air flow entering the compartment

Temperatures:

T_{comp}: compartment air temperature

T_{mix}: temperature of the mixed ambient and recycled air entering the compartment

T_{amb}: ambient air temperature

Heat flows:

Q_{load}: transmission, solar radiation, dry heat emission from occupants and respiration

$Q_{load-resp}$: transmission, solar radiation, dry heat emission from occupants

Q_{amb}: duty needed to heat or cool ambient air

Q_{amb_contr}: share of the total load that is covered by the entering treated ambient air.

Q_{rec}: share of Q_{load} being covered by the recycled air.

Q_{HVAC}: the total duty provided by the M-HVAC.

11.1.1 Model of the M-HVAC system for calculations of energy flows and air flows

Below is a sketch of the process of the airflows in the system:

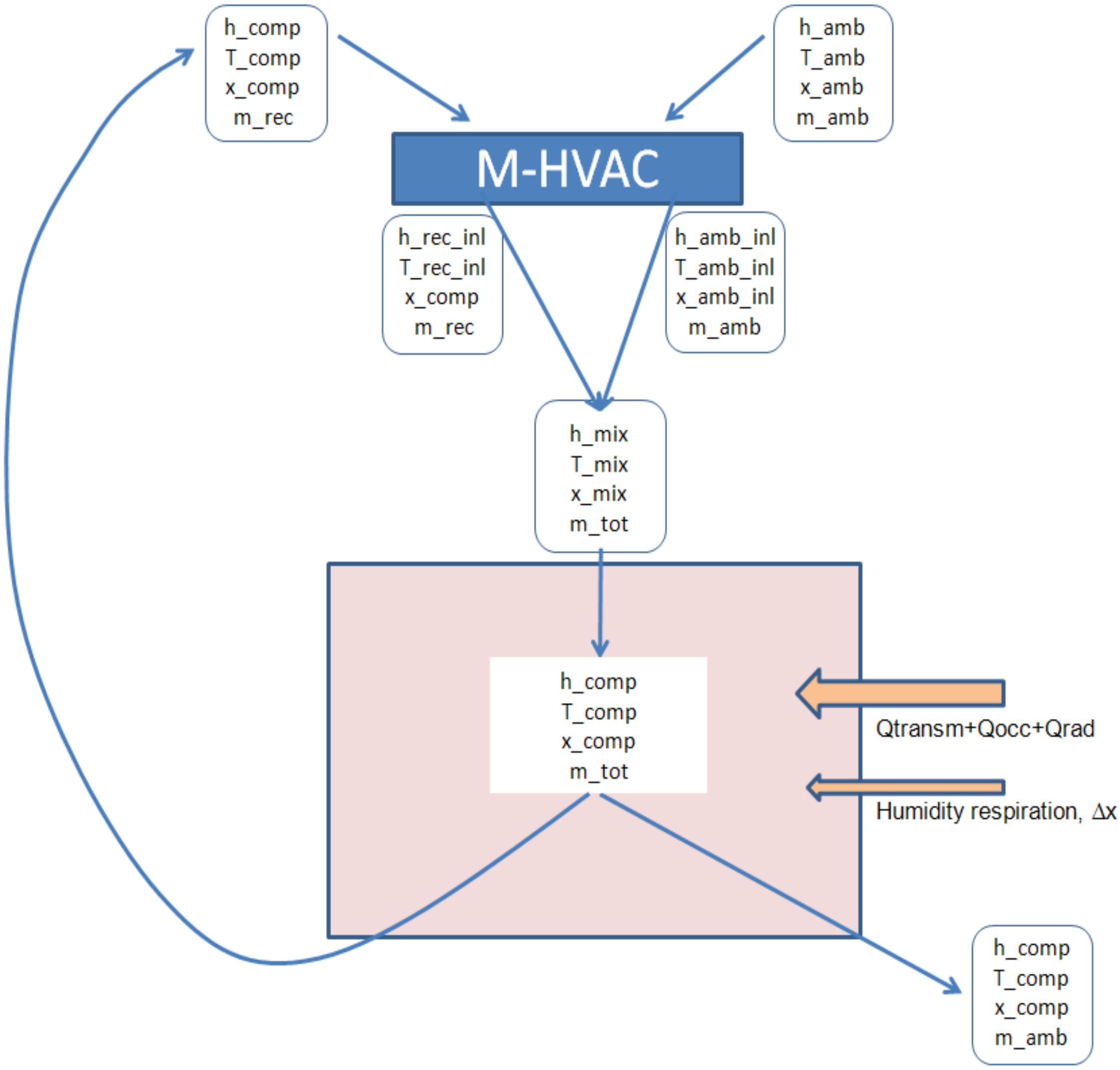

Figure 11-1: Flow pattern for the air flows of the M-HVAC, including the different air properties.

The ambient and recycled airflows are treated separately in two different heat exchangers. They are mixed afterwards before entering the compartment. The necessary enthalpy of the air entering the compartment is given by the total energy balance of the compartment. This is described further in the following section (11.1.2)

11.1.2 Calculation procedure for energy flows:

The total load on the compartment consists of transmission, radiation through windows, dry heat emission from occupants and respiration. This load will be covered by the total airflow entering the compartment, and gives the equation:

$$\dot{Q}_{load} = \dot{m}_{tot}(h_{comp} - h_{mix}) \qquad (11.1)$$

This will be handled by the two airflows, ambient and recycled air, and the equation is thus transformed to:

$$\dot{Q}_{load} = \dot{m}_{amb}(h_{comp} - h_{amb\_inl}) + \dot{m}_{rec}(h_{comp} - h_{rec\_inl}) \qquad (11.2)$$

The total flow of air is given by the amount of recycled and entering ambient air. The ambient air volume is decided by regulations in tableTable 4-4.The amount of recirculated air is decided according to the total load on the compartment, excluding respiration, $Q_{load-resp}$.

To solve this equation, the absolute humidity needs to be identified for both the mixed air and the compartment air.

$$x_{mix} = \frac{\dot{m}_{amb} \bullet x_{amb} + \dot{m}_{rec} \bullet x_{comp}}{\dot{m}_{amb} + \dot{m}_{rec}} \qquad (11.3)$$

The absolute humidity of the entering recycled air is the same as the humidity of the compartment air, and this is equal to the humidity of the mixed air plus the humidity contribution from respiration:

$$x_{comp} = x_{mix} + \Delta x_{resp} \qquad (11.4)$$

Combining eq. (1.3) and (1.4) yields:

$$x_{mix} = \frac{\dot{m}_{amb} \bullet x_{amb\_inl} + \dot{m}_{rec} \bullet \Delta x_{resp}}{\dot{m}_{amb}} \qquad (11.5)$$

If x_{amb} is less than 0,008kg/kg, x_{amb_inl} will be equal to x_{amb}. If the ambient humidity is higher than 0,008, the air will be dried in the evaporator until the absolute humidity is equal to 0,008. The relative humidity will in those cases be 100%.

With the absolute humidity known for each point of the process, both h_{comp} and h_{mix} can be found using HxLib functions.

With the enthalpy and the absolute humidity given for the mixed air, the temperature is also automatically given. The temperature of the incoming ambient air is set to the same as the mixed air. The exception is when the ambient air needs to be dried, and the temperature is given by the relative humidity at 100%. Thus the enthalpy of the entering ambient air can be identified as a function of temperature and absolute humidity, or relative humidity and absolute humidity. Namely:

If x_{amb}<0,008:

$H_{amb\_inl}=f(t_{amb\_inl}, x_{amb})$

If x_amb>0,008:

H_{amb_inl}=f(RH=100%,x=0,008)

To find how much of the total heat load that is covered by the ambient air, the following equation is used:

$$\dot{Q}_{amb\_contr} = \dot{m}_{amb}(h_{comp} - h_{amb\_inl}) \tag{11.6}$$

The rest of the heat load needs to be covered by the recycled air is thus found by combining eq.11.2 and 11.6:

$$\dot{Q}_{rec} = \dot{Q}_{load} - \dot{Q}_{amb\_contr} = \dot{m}_{rec}(h_{comp} - h_{rec\_inl}) \tag{11.7}$$

The work needed to heat or cool the ambient air is then given by

$$\dot{Q}_{amb} = \dot{m}_{amb}(h_{amb} - h_{amb\_inl}) \tag{11.8}$$

The total work that needs to be provided by the MAC is then given by:

$$\dot{Q}_{HVAC} = \dot{Q}_{amb} + \dot{Q}_{rec} \tag{11.9}$$

In cooling mode Q_{amb} and Q_{rec} is to be provided by evaporator 1 and evaporator 2 respectively. In heating mode, Q_{amb} is to be provided by gas cooler 2, and Q_{rec} is covered by gas cooler 1.

11.2 : Case study

11.2.1 : Input table for calculations

The purple fields of the table are the input chosen by the user. As stated in previous section, the flow rate of recycled air is decided according to the load on the compartment, and such that the resulting enthalpy difference over the heat exchanger is reasonable.

Table 11-1: Input for calculations of annual energy demand:

Operational hours	Start MAC	End MAC
hour of day	6	22
Max # passengers	66	
Occupancyrate		
Hour of day	***Occupancy***	
1	Empty	
2	Empty	
3	Empty	
4	Empty	
5	Empty	
6	One third	
7	One third	
8	Full	
9	Full	
10	Two thirds	
11	One third	
12	One third	
13	One third	
14	One third	
15	Two thirds	
16	Two thirds	
17	Full	
18	Full	
19	Half full	
20	One third	
21	One third	
22	Empty	
23	Empty	
24	Empty	

Measures train		
Length [m]	24,175	[m]
Width	2,95	[m]
Height	3,89	[m]
%window area	0,25	[%]
Heat transfer coeff ambient, hamb	20	[W/m2K]
Heat transfer coefficient indoor, hin	7	[W/m2K]
heat conductivity coeff, kwall	0,07	[W/m2K]
Heat transmission coefficient wall,U	1,10	[W/m2K]
Heat transmission coefficient window,U	3	[W/m2K]
Total surface area	259,39775	[m2]
Wall and roof area, excl windows	194,5	[m2]
Window area	64,8	[m2]
Wall thickness	0,05	[m]

Qload (excl respiration) [kW]	**mass flow air [kg/s]**
<5	0,7
>5<10	1
>10<15	1,5
>15<20	2
>20<25	2,5
>25<30	2,8
>30	3

11.2.2 Example of calculation procedure for power demand in Excel

The example chosen is a heat flow calculation for Frankfurt, from the 4^{th} of January at 12am. The first rows down to the purple line are the weather data taken from Meteonorm. The following rows are the resulting energy calculation, using the method of the previous chapter. (11.1)All the 8760 hours of the year have one column each in the spreadsheet. For reference for the formulas, this particular hour is column CG in the spreadsheet.

1	**m**	1	1
2	**dy**	4	4
3	**h**	12	12
4	**hy**	84	84
5	**Ta**	4,9	4,9
6	**RH[%]**	66	66
7	**Global horiz rad. [W/m2]**	238	238
8	**Global tilted rad. [W/m2]**	838	838
9	**Global reflected rad. [W/m2]**	92	92
10	**Diffuse tilted rad. [W/m2]**	316	316
11	**Diffuse horizontal rad. [W/m2]**	84	84
12	**Direct horizontal rad. [W/m2]**	154	154
13	**Direct tilted rad.**	522	=$CG$8-$CG$10
14			
15	**G_direct [kW]**	6,770281	=0,8*CG13*Input_output!$F$16/4000
16	**G_diffuse [kW]**	8,196969	=(0,8*CG10*Input_output!$F$16)/(2*1000)
17	**G_refl [kW]**	2,99345	=(0,8*0,2*((CG13)/2+CG10)*Input_output!$F$16)/2000
18	**G-Windows total [kW]**	**17,9607**	**=CG15+CG16+CG17**
19	**h_amb [kJ/kg]**	13,63287	=hx_h_trh(103000;CG5;CG6)/1000

20	**x ambient**	0,003472	=hx_x_th(101325;CG5;CG19*1000)
21	**x amb_inl**	0,003472	=IF(CG20<0,008;CG20;0,008)
22	**Occupancy**	One third	=Input_output!$B$22
23	**#Passengers**	21,78	=IF(CG22="Full";Input_output!$B$7;IF(CG22="Two thirds"; Input_output!$B$7*0,66;IF(CG22="Half full"; Input_output!$B$7*0,5;IF(CG22="One third";Input_output! $B$7*0,33;0))))
24	**Tot pax contribution,dry [kW]**	**2,178**	**=(CG23*100)/1000**
25	**Indoor temperature [C]**	23	=IF(CG5>30;25,6;IF(CG5>=-10;23;IF(CG5<-10;20,6)))
26	**delta T**	-18,1	=CG5-CG25
27	**delta T abs**	18,1	=ABS(CG26)
28	**Heat transfer wall/window [kW]**	-7,4031	=(Input_output!$F$15*Input_output!$F$12*CG26+ Input_output!$F$13*Input_output!$F$16*CG26)/1000
29	**Vol flow amb air/pax [m3/h]**	20	=IF(CG5<-20;10;IF(CG5<-5;15;20))
30	**Total volume amb air [m3/h]**	435,6	=CG29*CG23
31	**M ambient air [kg/s]**	**0,1452**	**=1,2*CG30/3600**
32	**M recycled air [kg/s]**	1,5	=IF(CG35>30;Input_output!$F$27;IF(CG35>25;Input_output!$F$26; IF(CG35>20;Input_output!$F$25;IF(CG35>15;Input_output!$F$24;IF(CG35>10; Input_output!$F$23;IF(CG35>5;Input_output!$F$22;Input_output!$F$21))))))
33	**Mamb/Mrecycl [-]**	0,0968	=CG31/CG32
34	**Cooling/heating recycled [kW]**	**12,7356**	**=CG18+CG24+CG28**
35	**ABS recycled**	**12,7356**	**=ABS(CG34)**
36	**h_PLUSocc-h_recINL**	7,743162	=(CG44-CG56)/CG32

37	**hum contr [kg]**	0,8712	=0,04*CG23
38	**delta x occ**	0,000147	=CG37/(CG31*3600+CG32*3600)
39	**x_MIX [kg/kg]**	0,004992	=IF(CG31=0;CF46;(CG31*CG21+CG38*CG32)/CG31)
40	**x_comp plusocc [kg/kg]**	0,005139	=IF(CG38=0;CG39;CG39+CG38)
41	**h comp MINocc [kJ/kg]**	35,78996	=hx_h_tx(101325;CG25;CG39)/1000
42	**h comp PLUSocc [kJ/kg]**	36,16416	=IF(CG31="0";CG41;hx_h_tx(101325;CG25;CG40)/1000)
43	**Q resp**	0,615634	=(CG31+CG32)*(CG42-CG41)
44	**Q_totload**	**13,35124**	**=CG34+CG43**
45	**RH plusocc [%]**	29,54724	=hx_rh_tx(101325;CG25;CG40)
46	**x_rec_inl**	0,005139	=CG40
47	**h_recINL [kJ/kg]**	28,421	=CG42-CG36
48	**T rec_INL**	15,36335	=hx_t_hx(101325;CG47*1000;CG40)
49	**h_compOCC-h_mix**	8,115266	=CG44/(CG31+CG32)
50	**hmix**	28,04889	=CG42-CG49
50	**t MIX**	15,36335	=hx_t_hx(101325;CG50*1000;CG39)
52	**RH MIX**	46,21389	=hx_rh_tx(101325;CG51;CG39)
53	**h_ambINL**	24,20484	=IF(CG31=0;"N";IF(CG20<0,008;hx_h_tx(101325;CG54;CG21)/1000; hx_h_trh(101325;CG54;99)/1000))
54	**t_ambINL**	15,36335	=IF(CG31=0;"N";IF(CG20>0,008;hx_t_xrh(101325;0,008;100);CG51))
55	**RH_ambINL**	32,22302	=IF(CG31=0;0;IF(CG21=0,008;100;hx_rh_tx(101325;CG54;CG21)))
56	**Ambient contribution**	1,736493	=IF(CG31=0;0;CG31*(CG42-CG53))
57	**Q left to cover**	**11,61474**	**=CG44-CG56**
58	**Q amb [kW]**	**-1,53505**	**=IF(CG31=0;0;CG31*(CG19-CG53))**
60	**Tot Q MAC [kW]**	**10,07969**	**=CG57+CG58**
61	**ABS Tot Q [kW]**	10,07969	=ABS(CG59)

62	**Q MAC on [kW]**	10,07969	=IF(Input_output!$B$5=Input_output!$C$5;CG59;IF(CG3<Input_output!$B$5;"N";IF(CG3>=Input_output!$C$5;"N";CG59)))
63			
64	**ABS QMAC**	10,07969	=IF(CG61="N";0;ABS(CG61))
65	**Cooling/heating?**	C	=IF(CG61="N";"N";IF(CG61>0;"C";"H"))
66	**Only cooling**	10,07969	=IF(CG63="N";0;IF(CG63="C";CG61;0))
67	**Only heating**	0	=IF(CG$63="N";0;IF(CG$63="H";CG$61;0))
68			
69	**COP**	1	=IF(CG$61="N";0;IF(CG$22="Full";IF(CG$61>30;3,27;IF(CG$61>25;4,06;IF(CG$61>20;5,08;IF(CG$61>15;5,48;IF(CG$61>10;5,95;IF(CG$61>-5;0;IF(CG$61>-10;8,2;IF(CG$61>-15;7,2;IF(CG$61>-20;6,42;IF(CG$61<-20;4,55;))))))))));1))
70	**COP**	1	=IF(CG$22="Two thirds";IF(CG$61>25;3,53;IF(CG$61>20;4,13;IF(CG$61>15;5,014;IF(CG$61>10;5,61;IF(CG$61>-5;0;IF(CG$61>-10;7,33;IF(CG$61>-15;5,89;IF(CG$61>-20;4,85;IF(CG$61<-20;3,37)))))))));CG67)
71	**COP**	1	=IF(CG$22="Half full";IF(CG$61>25;3,11;IF(CG$61>20;3,8;IF(CG$61>15;4,59;IF(CG$61>10;5,29;IF(CG$61>-5;0;IF(CG$61>-10;6,14;IF(CG$61>-15;5,05;IF(CG$61>-20;4,08;IF(CG$61<-20;2,83)))))))));CG68)
72	**COP**	4,98	=IF(CG$22="One third";IF(CG$61>20;3,46;IF(CG$61>15;4,17;IF(CG$61>10;4,98;IF(CG$61>-5;0;IF(CG$61>-10;4,95;IF(CG$61>-15;4,21;IF(CG$61>-20;3,32)))))));CG69)
73	**COP**	4,98	=IF(CG22="Empty";0;CG70)
74			
75	**Energy compressor**	2,024034	=IF(CG71=0;0;CG62/CG71)

11.2.3 Average values from energy calculations

To find the parameters to implement in the Modelica model to find the COP (Ref Chapter 6.5), the energy flow calculation is done separately for each occupancy rate. Thus to find the operational mode for the M-HVAC for 100% occupancy rate, the input parameter sheet, Table 11-1, would be set to full for all the 24 hours of the day. The same is done for the other occupancy rates. The outlined parameters are the ones that will be used in the Modelica in order to find COP. Similar average values are made for 66% and 33% occupancy rate in order to find the COP for all load intervals and occupancy rates.

Tot Q	#hours	Sum h [kJ/kg]	h avg [kJ/kg]	Sum G [kW]	G average [kW]	sum T [C]	T avg [C]	Sum m [kg/s]	m rec avg [kg/s]	Sum %RH [%]	%RH avg [%]	Sum Q [kW]	Q avg [kW]	Sum %RH amb	%RH amb avg [%]
>30	5	332,13	66,43	59,55	11,91	148,9	29,78	12,5	2,50	267,965	53,59	154,7451	30,95	276	55,20
(25-30)	30	1791,37	59,71	345,55	11,52	818,4	27,28	67	2,23	1639,638	54,65	806,3543	26,88	1700	56,67
(20-25)	122	6620,89	54,27	1172,69	9,61	3109,5	25,49	238,5	1,95	6703,23	54,94	2677,582	21,95	6825	55,94
(15-20)	356	16536,05	46,45	3254,33	9,14	7921,3	22,25	635,5	1,79	18802,38	52,82	6133,255	17,23	20139	56,57
(10-15)	613	25675,73	41,89	4307,88	7,03	11914,5	19,44	900,5	1,47	31105,42	50,74	7580,187	12,37	37971	61,94
(5-10)	895	33424,82	37,35	4411,13	4,93	14847,8	16,59	1058,4	1,18	44358,18	49,56	6567,785	7,34	61087	68,25
(0-5)	1230	40494,59	32,92	3497,25	2,84	17405,8	14,15	1090,3	0,89	60209,67	48,95	3000,312	2,44	89952	73,13
(-5-0)	1749	45896,49	26,24	2293,52	1,31	18464,2	10,56	1272,4	0,73	78604,76	44,94	-4733,09	-2,71	137305	78,50
(-10--5)	1890	33106,97	17,52	1220,00	0,65	10853,5	5,74	1324,5	0,70	68821,44	36,41	-14165,8	-7,50	156621	82,87
(-15--10)	1458	13032,77	8,94	311,95	0,21	631,3	0,43	1029	0,71	42680,12	29,27	-17932,8	-12,30	129091	88,54
(-20--15)	412	559,44	1,36	23,56	0,06	-1876,7	-4,56	313	0,76	9864,963	23,94	-6645,64	-16,13	38217	92,76
<-20	0	0,00	0,00	0,00	0,00	0	0,00	0	0	0	0,00	0	0,00	0	0,00

11.3 : Modification of system components

The components of the HVAC model had to be modified in order to fit the desired operating conditions. The heat exchangers and the compressor were adjusted according to the results from Excel calculations for the train compartment.

Nomenclature:

GC1: gascooler 1

GC2: gascooler 2

Evap1: evaporator 1

Evap2: evaporator 2

HX: heat exchanger

11.3.1 Heat exchanger configuration

The heat exchangers had to be designed so that the M-HVAC could cover the maximum cooling power demand, since this is the most demanding operating condition.

The built in heat exchangers that were already in the Modelica model had to be minimized a great deal to fit the size restrictions. The model was giving enough duty to cover the maximum cooling load, so the existing process parameters such as temperatures and pressure levels could be used as initial values for designing the new heat exchangers.

To find the appropriate heat exchanger configurations, an iteration process was necessary between the Modelica model and HxSim. After running the system in Modelica with the correct ambient conditions, the heat exchanger performances were analyzed at maximum COP, and the process parameters such as temperatures, pressures and mass flows were implemented in HxSim to find the optimal design of each heat exchanger.

After changing one heat exchanger configuration, the performance of the other heat exchangers, as well as the other process parameters, are automatically changing. By analyzing the temperatures in the circuit, it was possible to find which HX that needed further modifications in order for the HVAC to cover the load.

Design of evaporators

The evaporators had to be configured such that the conditions of the air leaving the heat exchanger is according to energy flow calculations in excel.

In HxSim, the CO_2 pressures and mass flows were first assumed according to the values at maximum COP of the original model in Modelica. When the heat exchangers are reconfigured, the optimal pressures will change, since the ejector automatically balances the two evaporator pressures according to ambient temperature and mass flow of CO_2. The new pressure and mass flow were then implemented in Hxsim, and the configuration adjusted until the duty and temperatures of air were according to the energy calculations in Excel.

Since Modelica automatically adjusts the evaporator pressure, it is difficult to get the duties of the evaporators exactly as calculated in excel. One would have to adjust the configuration for every part load and change of ambient conditions, which is not possible. The important is nevertheless that the two evaporators together cover the cooling demand.

Design of gas coolers

The GC's were also designed for the maximum cooling load, since this is more demanding than the maximum heat load. The important issue for good cycle performance is to get the outgoing temperature of GC2 as low as possible. The GC's were thus designed in order to get the highest possible cool down of CO_2, and such that the temperature difference out of GC2 was about 3K. An iteration process was done for both GC's between Modelica and HxSim until the temperature profile was acceptable.

11.3.2 Heat transfer model for the heat exchangers

The following factors affect the heat transfer coefficient, k, of the heat exchanger:

- Temperature/pressure
- Vapor quality (flow pattern)
- Mass flux [kg/m2s] – convective boiling
- Heat flux [W/m2] – nucleate boiling
- Tube diameter / geometry

It is thus clear that the k-value will not be constant throughout the heat exchanger, and it is also highly dependent on the ambient conditions. There are various heat transfer correlations that can be applied to account for this variation of the k-value during the heat transfer process.

The heat transfer calculation method is of major importance for the HX performance. As an initial value, the heat transfer coefficient (W/m^2K) was found from HxSim for the different heat exchangers. This is nevertheless only valid for the maximum cooling load, and will be less for the other cases. When using the heat transfer correlations in Modelica this is accounted for, and the initial value is only important for getting the simulation started.

The heat transfer correlations chosen:

Cooling mode:

Evaporators: 2-Phase: Shah Chen/1-Phase: Gnielinski Dittus Boelter

Gas coolers: 2-phase: constant alpha/ 1-Phase: Gnielinski Dittus Boelter.

The reason why the Shah-Shen correlation is not used for the gas coolers is that it only works for two-phase flow. Even though it is supposed to apply Gnielinski for 1-phase flow, Dymola is calculating both correlations at the same time, and applying the correct one afterwards.

Heating mode:

All heat exchangers using: 2-phase: constant alpha/ 1-Phase: Gnielinski Dittus Boelter.

11.3.3 Finding optimal amount of bypass ambient air

The mass flow of air through the heat exchanger is deciding the temperature difference between the incoming air and CO_2. By studying the temperature profile in the heat exchanger, the mass flow of air can be optimized. The fan power demand is also varying according to the air flow rate. The fan power demand increases exponentially with the air flow rate, so the wanted head exchanger capacity has to be weighed against the resulting fan power demand.

The airside pressure drop correlations and fan power consumption is not included in the Modelica simulations. The fan power demand for the varying airflow is thus found separately in HxSim. The fan power demand is only depending on the air flow rate, and is independent of other operating conditions. The fan power demand will thus be valid for all operating conditions for the same heat exchanger.

In cooling mode it is the airflow in GC1 that has to be optimized. Here the optimal temperature difference between incoming air and exiting CO_2 is about 5-10K, but is found by reducing the bypass air flow until the decrease in HX capacity is less than the decrease in fan power demand. Below is a graph of the theoretical fan power demand as a function of air flow:

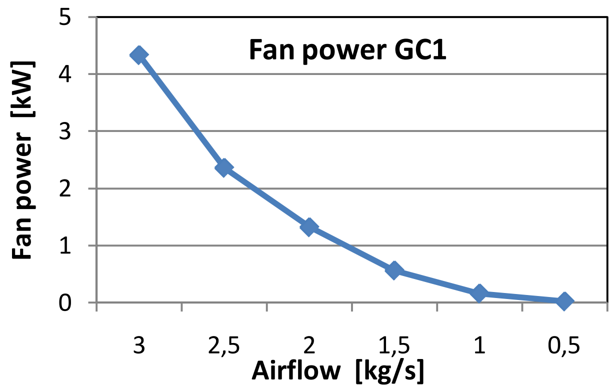

Figure 11-2: Fan power demand for gas cooler 1

In heating mode it is the airflow of evap2 that has to be optimized. The temperature difference between the incoming air and the incoming CO_2 should be about 5K. Less air is necessary to get the wanted temperature difference for evap 2 for heating mode than for GC2 in cooling mode. The wanted temperature difference is found at a relatively low airflow rate, so the risk of getting extremely high fan power demand is not as high as in cooling mode. This can be seen from the graph below. At maximum heat power demand, only 1.5kg/s of ambient air is necessary, so an optimization process is not needed.

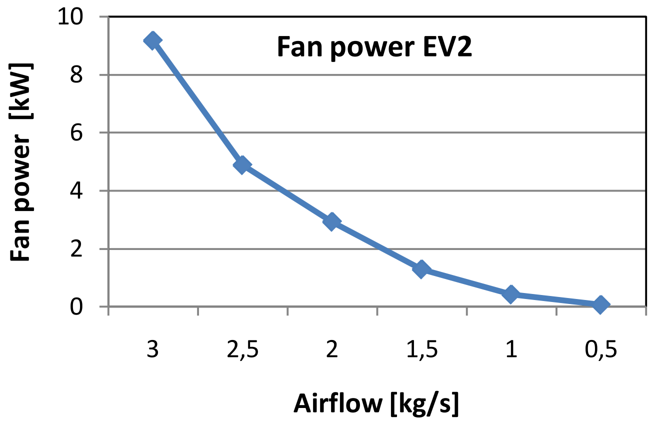

Figure 11-3: Fan power demand for evaporator 2

11.3.4 : Fan power demand for EV1 & GC2

Below is shown the fan power demand of evaporator 1 and gascooler 2, which treat entering/exiting air. These airflows are relatively small compared to the air flows of the other two heat exchangers, so the resulting power demand is not worth considering. Note that the y-axis in this case is [W] and not [kW]:

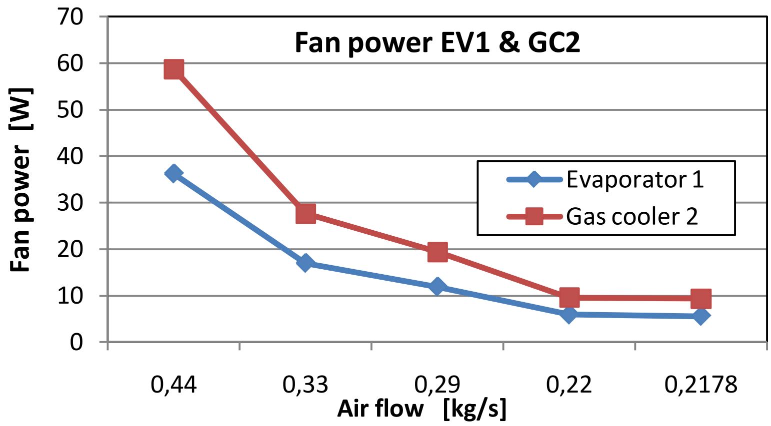

Figure 11-4: Fan power requirement for gascooler2 and evaporator1

11.3.5 Deciding the compressor volume:

The compressor has to be designed so that it can deliver enough capacity to cover the refrigeration demand at maximum cooling load. At the same time, it should not be overdimensioned, so that it has to run on low part load much the time. The design parameter is the displacement volume:

$$V_C = \frac{\dot{m}_s}{\rho_s \cdot \lambda \cdot n} \qquad (11.10)$$

Where:

V_c is the necessary displacement volume

M_s is the refrigerant mass flow into the compressor

P_s is the density of the refrigerant at the compressor inlet

Λ is the volumetric efficiency of the compressor, which is set to 0.7.

n is the frequency or rotational speed of the compressor

The density of CO_2 at compressor inlet is known from the necessary suction pressure of the case of maximum cooling demand.

In Modelica the volumetric and isentropic compressor efficiencies are set to a constant value of 0.7. It is thus of no importance for the performance whether to choose a small volume, leading to a high necessary rotational speed, or a large volume, leading to a low rotational speed, to cover the same demand. The performance for the Modelica model would be the same, but this is not the case in reality.

The compressor volume is thus chosen such that the difference of rotational speed should not be too large for the maximum load and part load. This is found for a compressor volume of 55cm^3. Then the rotational speed is at 45Hz for maximum cooling load of the case study and about 10 for the minimum load. If a higher power demand should be required, the rotational speed could for a real system easily increase to cover the demand.

11.3.6 Results from Simulations

The following tables are the results obtained for the COP for the different power demands and occupancy rates. The rows of 0 indicates that the load is so low that the heat pump will be turned off, and only the necessary amount of ambient air will enter the compartment untreated:

Table 11-2: COP at varying occupancy rates

TotQ/occ.rate	100 %	66 %	50 %	33 %
>30	**3,27**	**0**	**0**	**0**
(25-30)	**4,06**	**3,53**	**3,11**	**0**
(20-25)	**5,0174**	**4,13**	**3,795**	**3,46**
(15-20)	**5,476**	**5,014**	**4,592**	**4,17**
(10-15)	**5,95**	**5,61**	**5,295**	**4,98**
(5-10)	0	0	0	0
(0-5)	0	0	0	0
(-5 -0)	0	0	0	0
(-10- -5)	**8,2**	**7,33**	**6,14**	4,95
(-15- -10)	**7,19**	**5,89**	**5,05**	**4,21**
(-20- -15)	**6,418**	**4,85**	**4,085**	**3,32**
<-20	**4,55**	**3,37**	**2,83**	0

The following table shows the average temperature of each power demand interval for each occupancy rate:

Table 11-3: Temperature at varying occupancy rate

Tot Q/occ.rate	100 %	66 %	33 %
>30	29,78	0	0
(25-30)	27,28	29,3	0
(20-25)	25,48	26,9	28
(15-20)	22,2	23,4	24
(10-15)	19,43	20,1	20,48
(5-10)	16,58	17,35	17,55
(0-5)	14,15	14,97	15,77
(-5 -0)	10,5	11	11,87
(-10- -5)	5,74	5,4	5
(-15- -10)	0,43	-0,9	-2,35
(-20- -15)	-4,5	-7,7	-16,7
<-20	-20	-20	0

The following tables show the results of the dynamics of the system for varying operating conditions. (Section 6.4) The first table is for case 1, and the second is for case 2.

Table 11-4: Operational mode for the ejector cycle at varying occupancy rate and constant temperature.

	Constant Temp			
Temperature	35,7	35,7	35,7	35,7
Occrate	100 %	66 %	50 %	33 %
COP	2,9	2,88	2,85	2,837
Q gc1	**33737,4**	**29631,5**	**27856,6**	**25846,9**
Qgc2	**10031,3**	**6276**	**4607,5**	**2920,37**
Q ev1	**8543,92**	**5888,16**	**4788,2**	**3141,5**
Q ev2	**23908,4**	**20922**	**19397,8**	**18275,3**
P ev1	51,7972	52,28	51,96	52,36
P ev2	45,34	45,85	45,5	46
P high	121,2	119,3	118,73	117,4
suction pressure ratio ∏	**1,142417292**	**1,1402399**	**1,141978**	**1,13826087**
m suction	0,253	0,211	0,19	0,1744
m motive	0,125	0,11	0,1015	0,096
mass entrainment ratio Φ	**0,494071146**	**0,521327**	**0,5342105**	**0,55045872**

Table 11-5: Operational mode for the ejector cycle at varying ambient temperatures and constant occupancy rate.

	Constant Occupancy rate			
	Cool high	Cool low	Heat low	Heat high
Temperature	35,7	20,5	5	-16,7
Occrate	33 %	33 %	33 %	33 %
COP	2,837	4,98	4,95	3,32
Q gc1	**25846,9**	**13560,7**	**4429,7**	**12456,4**
Qgc2	**2920,37**	**987,85**	**3142,28**	**5218,5**
Q ev1	**3141,5**	**1515,52**	**2340,5**	**4142,9**
Q ev2	**18275,3**	**10667,5**	**2733,3**	**5025,2**
P ev1	52,36	45,44	39,8	26,23
P ev2	46	43	30,9	17,79
P high	117,4	83,3	66,84	68,17
suction pressure ratio ∏	**1,138261**	**1,056744**	**1,288026**	**1,474424**
m suction	0,1744	0,073	0,05	0,084
m motive	0,096	0,051	0,012	0,019
mass entrainment ratio Φ	**0,550459**	**0,69863**	**0,24**	**0,22619**

Below is listed the specifications for the MPE gascoolers used to compare between fin-and-tube and MPE:

Table 11-6: Design of the MPE heat exchangers used for simulations

	Gas cooler1	Gas cooler 2
Tube configuration	MPE	MPE
# Duplications	7	11
#Vertical tubes	8	8
#Horizontal tubes	1	1
Vertical tube pitch [m]	0,04	0,04
Horizontal tube pitch [m]	0,04	0,035
Core Depth [m]	0,28	0,385
Height [m]	0,36	0,36
Core width [m]	0,4	0,4
Fin density [FPI]	25,4	12,7
Fin thickness [m]	1,50E-04	1,00E-04
Micro tube height [m]	0,0014	0,0014
Micro tube width [m]	0,0263	0,0263
# Microtube channels [-]	20	20
Weight	25,9	22,33

11.3.7 Environmental properties of common refrigerants

The following table shows the environmental properties of some common refrigerants. R152a and R744 are the two options considered for replacement of R134a for mobile air condition applications.

Table 11-7: Environmental properties for some common refrigerants.

Refrigerant	R134a	R152a	R290	Blend H	R744
GWP* (-)	1300	140	11	<150	1
ODP* (-)	0	0	0	0	0
ATEL (ppm)	50000	50000	50000	tbd	40000
TWA (ppm)	1000	1000	1000	tbd	5000
Maximal Operating Pressure (MPa)	2,5	2.5	2.5	2.5	14

*(*References: GWP=1 for R744, ODP=1 for R12)*

(Antonijevic, 2007)