HEATING AND COOLING AN ELECTRIC VEHICLE

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HEATING AND COOLING AN ELECTRIC VEHICLE

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KTH Industrial Engineering and Management	**Master of Science Thesis EGI 2008/ETT:442** **HEATING AND COOLING AN ELECTRIC VEHICLE** Ricardo Arboix Barreto

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Abstract

Global warming and climate change is forcing humanity to take action. The electric vehicle has during the past year shown an increasing potential, implementing new Li-Ion batteries. These provide a performance similar to today's fossil fueled vehicles. The drawback of highly efficient cars is however the lack of waste heat, used today for cabin heating.

An interesting alternative to the presently used oil burners is the vapor compression cycle, used for air conditioning cars. The future of refrigerant choice for the automobile sector is today uncertain due to the demand for substitution of R134a.

This MSc thesis discusses the possibility of implementing a transcritical vapor compression cycle based on the natural refrigerant carbon dioxide, also called R744, in future electric vehicles. It would be used for both cooling and heating.

Although R744 was one of the first refrigerants to be used, very little is known about its transcritical performance. This study describes the special characteristics of this cycle. An improved cycle is proposed and compared with the "standard" transcritical cycle using R744 and a subcritical R134a cycle. The improved cycle requires an 8 port 2 way valve, enabling alterations in the refrigerant flow and an expander recovering work. A flash-gas by pass concept, decreasing the pressure drop in the evaporator and improving heat transfer, is also introduced. Optimal correlations and equations of state provided by EES are used in the simulation providing ideal and thus comparative results.

The improved automobile R744 TVCC HP/AC cycle showed an average of 23 % improvement in COP compared to the standard R744 cycle when simulated in heating mode. The nature of the transcritical cycle implies a need to keep the passengers safe from the high-pressure and temperature of R744 entering the gas cooler. At an ambient

temperature of -2 °C the refrigerant was estimated to have a pressure of 120 bar and corresponding gas cooler inlet temperature of 120 °C in order to keep the Cabin warm.

R134a can however be proven to outperform both analyzed R744 cycles. An average of 20 % in reduced power consumption can be expected if operating the standard cycle with R134a compared to the improved R744 cycle. In spite of this, the R134a refrigerant can not be recommended due to its facing out, starting 2011. Therefore it is recommended to continue developing improved R744 alternatives, as well as studying other refrigerants in direct and indirect systems.

Finally this thesis describes further improvements on the proposed cycle. By integrating a second micro-channel tube bundle evaporator **in the EV-battery core**, it is possible to recover heat generated in the battery. The design presents an additional 12- port- 5 way valve enabling constant refrigerant flow direction during the cycles multiple operational modes. This should improve the COP even more and reduce the temperature drop in the cabin during de-frosting. The battery can further on be temperature regulated, heated and cooled from the inside, leading to a more energy efficient use of the EV as well as possibly prolonging the lifespan of the battery.

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Thank you family and friends for being understanding and supportive.

Abbreviations

AC	Air Condition
EES	Engineering Equation Solver
EV	Electric Vehicle
GWP	Global Warming Potential
HP	Heat Pump
ICE	Internal Combustion Engine
ODP	Ozone Depleting Potential
R744	Carbon Dioxide
R134a	Tetraflourethane
SVCC	Subcritical Vapor Compression Cycle
TVCC	Transcritical Vapor Compression Cycle
VCC	Vapor Compression Cycle

Nomenclature

A	area	[m^2]
c_p	specific heat - constant pressure	[kJ/kg, ^oC]
COP	coefficient of performance	[-]
h	specific enthalpy	[kJ/kg]
m	mass	[kg]
$\dot{m}$	mass flow	[kg/s]
P	pressure	[kPa]
Q	energy	[kJ]
$\dot{Q}$	heating/cooling-power	[kW]
R_H	relative humidity	[%]
s	specific entropy	[kJ/kg, ^oC]
T	temperature	[^oC]
U	overall heat-transfer coefficient	[kW/m^2, °C]
V	volume	[m^3]
$\dot{V}$	volume flow	[m^3/s]
W	energy	[kJ]
$\dot{W}$	mechanical-power	[kW]
x	refrigerant quality	[-]

Greek letters

ρ	density	[kg/m^3]
η	efficiency	[-]
τ	time	[s]

Subscripts

a-d	refrigerant state points
amb	ambient
ave	average
battery	electric vehicle battery
cab	cabin
comp	compressor
cons	constant
cool	cooling
el	electric
ev	evaporator/evaporation
exp	expander – work recovery machine
HX1	heat exchanger – inside cabin
HX2	heat exchanger – outside cabin
IHX1	internal heat exchanger - hot side
IHX2	internal heat exchanger - cold side
improved	improved automobile R744 TVCC HP/AC
is	isentropic
l	liquid
lim	limits
m	mechanical
opt	optimal
out	out from heat exchanger – air side
pers	person
ref	refrigerant
rswf	real swept volume flow
s	superheat
standard	standard automobile R744 TVCC AC
sub	subcritical
sw	swept volume
tot	total
trans	transcritical
tswf	theoretical swept volume flow
v	vapor
vol	volumetric
1-6	refrigerant state points
12/5	improved 12/5-EV-HP/AC

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1. Introduction and background

1.1 Introduction

During the recent past years, a grown concern regarding global climate change has led and is still leading the technological development towards alternatives that have the least possible negative impact on the global climate. It is widely known that petrol based fuels have a negative impact on the environment, with special emphasis on the global warming concern.

It has in recent years also been discussed if ethanol fueled vehicles really are a god option for the actions taken in the struggle to reverse the global warming process. Critics mean that the devastation of the forests around the world in order to make room for cultivation of the ethanol producing crops cannot be seen as a sustainable solution to the present global warming situation.

The Electric Vehicle - EV has a great potential in becoming the future option in terms of environmental traveling. Recent development within the EV industry indicates that the implementation of efficient Li-Ion batteries has a great potential in order to meet present and future car owners needs and demands. Environmental critics of the EV mean that there has to be a consensus requiring that the power generation sector also must provide low CO_2 energy for this idea to work. Alternatives such as renewable energy sources and nuclear power generation seem to be the most sustainable way to supply EV's with low CO_2 emitting electricity.

One draw back of the EV is the lack of waste heat for cabin heating. In most climates around the world, this requirement is a necessity for the comfort.

The use of EV's has in recent years increased steadily, see fig. 1. This trend can be expected to continue in the coming years. Today the Li-Ion technology in EV exists in "standard" prototype vehicles, as well as in exclusive EV sports cars that are not in reach for the "common" EV consumer. Several articles found on the web show that consumers that can afford these cars are buying them. EV car manufacturers are also showing a big interest in future mass production of the Li-Ion EV.

A trend in the refrigerating legislative sector is to promote a change from synthetic and fluorine containing refrigerants, to "natural" environmentally friendly ones. The phasing out, in new cars, of the commonly used refrigerant R134a in the vehicle sector is expected happen 2011 in Europe.

This study suggests, presents, discusses and compares an improved AC-HP cycle based on R744 (Carbon dioxide), which could be used with other refrigerants as well, with a standard R744 AC **t**ranscritical **v**apor **c**ompression **c**ycle (TVCC). It should provide both cabin heating and cooling. More over, the integration of the improved AC-HP cycle in the EV-battery is proposed and discussed.

1.2 Background of the EV

The electric car, or yet more commonly known as the electrical vehicle (EV) is defined as - to move with chemical energy stored in rechargeable battery packs.
This technology was already present in the beginning of the mechanically based transportation era. Professor Sibrandus Stratingh of Groningen, the Netherlands, designed a small-scale electric car, which then was built by his assistant Christopher Becker in 1835 [1]. This was the starting point - after Prof. Sibrandus achievements other engineers and inventors lined up to write further history.

The result of a growing interest during the past 30 years, due to environmental and economical concerns, has lead to a fast development of EV. The timeline is filled with ups and downs in general interest from consumers and producers. It can be briefly mentioned that the EV's highlight era came during the 1990's to meet new demands of California's "Zero emission vehicle mandate". This era was abruptly disrupted. See e.g. the movie "Who killed the electric Car?"

The main problem facing the EV has traditionally been the incapacity to meet the car user's demand for speed, power, battery durability and design. A lot of effort has and is been put into resolving these matters. Both small companies financed by private investors, and larger car manufacturing companies have taken part.

1.2.1 Present and future of the EV

All the way from New Zealand, where a state owned company takes initiative towards testing new EV's [2] to the US, were famous people now stand in line to receive their first EV sports car, the obvious recovery from the past 10 years is no longer a saying but instead a becoming. Among others, Mitsubishis own President uses a Mitsubishi i-MIEV sport [3] for transportation. Besides politics and global warming concerns it seems that the idea of cities growing larger and people demanding cleaner air and quiet surroundings is pushing the automobile industry into this direction. The near future can certainly take the form of growing cities with private and public transportations based on electricity. Power generation plants, outside the cities will provide quiet and clean energy. The present urban oil usage, dependence and price development will most likely also open the doors for alternative fuels that can be found locally.

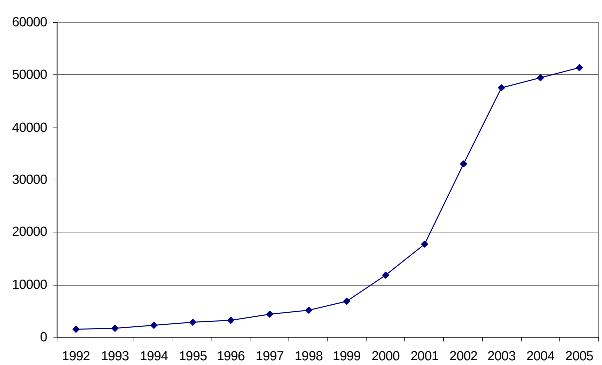


Figure 1. Estimated number of EV's in the US. Data from EIA [4]

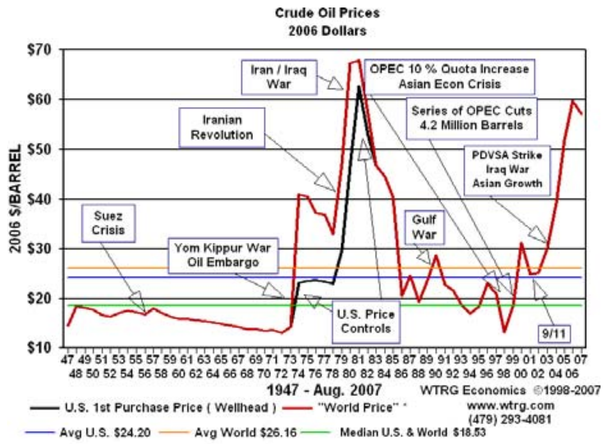


Figure 2. Oil price 1947-2007. Fig. from [5]

Li-Ion batteries are at the moment (2008) performing best and are reported to provide driving ranges of 400-500 km per charge.

Among the EV manufacturers there are varieties of models that span all the way from, small relatively inexpensive EV's to more sophisticated prototypes. One of the most impressive upcoming Li-Ion EV-model is a car made by the American based company Tesla Motors - called the Roadster [6]. The manufacturers promise an acceleration of 0-100 km/h under 4 seconds, a top speed of 200 km/h and a driving range of 392 km before needing to recharge the battery. The European equivalent of this model is the French Venturi Fetish with a similar performance.

Several articles are available in terms of bigger car manufactures announcing further development and interest of plain EV's based on the "new" Li-Ion battery technology. These batteries are under continuous development and progressing fast in EV applications. Further and deeper analysis in respect to the world lithium situation can be found in "The trouble with Lithium" [7].

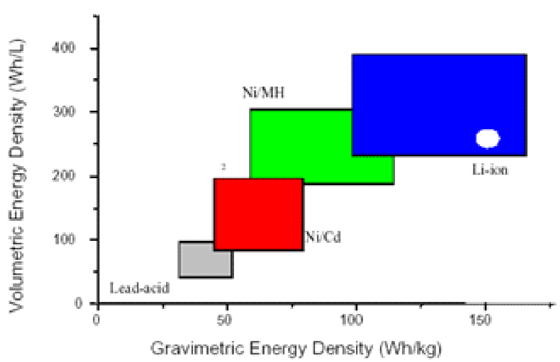


Figure 3. Energy Potential of Li-Ion Batteries. Fig. from [8]

1.3 Present cabin heating of EV

Approaching the present established EV production industry was found to be hard. This can be explained by that the larger car manufacturing companies are still in the development face. They do not generally want to reveal their new ideas concerning the next generation of EV.

The smaller EV car manufacturing companies on the other hand, do not seem to prioritize the heating problem and are instead relying on smaller types of oil based heaters, like Eberspächer burners [9] shown if fig. 4. It seems to be three ways in which the EV is heated today:

- Heat produced from electrical resistance [10]. This method can be proven beneficial for the EV due to the low-weight and small space characteristics enhanced. More over it is commercially available and could thus be an economical feasible choice. More over it requires only one fuel to fully operate the EV.
  However, the drawback cannot only be explained by the necessity of a separate AC-system (a requirement in this thesis), but also from the relatively large electricity consumption.

- Diesel/paraffin burner [11-12]. This method can be proven beneficial for the EV due to its reliability and simplicity. More over it is commercially available and commonly used in today's EV's.

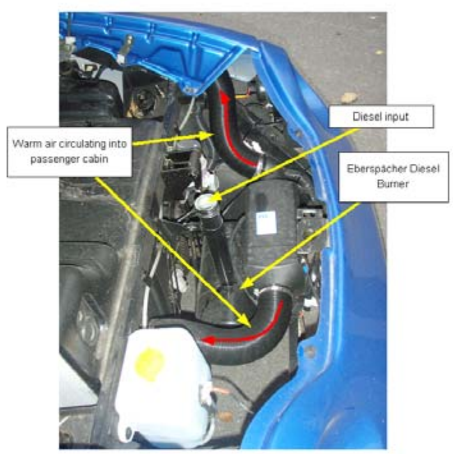


Figure 4. Diesel Burner in a Reva EV

  However, the drawback cannot only be explained by the necessity of a separate AC-system (a requirement in this thesis), but also from an environmental point of view. Two different fuels are needed to fully operate the EV. This can be unpractical for the EV owner. Thus the recharging of the EV and purchasing of oil normally takes place at different locations.

Figure 5. Heat Flow in a Reva EV

- HP (Past EV models – GM EV1, Toyota RAV4 EV [13-14]. Today in efficient diesel vehicles)

☺ Heating and Cooling
☺ Only one fuel necessary
☺ "Environmentally friendly"

😐 Replacement for 134a (GWP=1300)?
😐 Price tag will be?
😐 Consumes electricity – How much?

The cooling of the passenger cabin is done in the same way as in the common Internal Combustion Engine (ICE)-vehicle, thus the air conditioning (AC) system is basically a HP, pumping heat from the interior of the passenger cabin out to the ambient using conventional refrigerants such as R134a.

1.4 Possibilities for the HP in future vehicles

Traditionally the automobile **s**ubcritical **v**apor **c**ompression **c**ycle (SVCC) has been designed mainly for cooling the passenger cabin and in a few cases also for supplementary heating of vehicles, when lacking sufficient engine waste heat. That happens e.g. in some highly efficient diesel cars. The ICE vehicles are today using heat pumps almost only for cooling purposes, thus the heat needed to heat the cabin is mainly taken from approximately 30 % of the waste heat, produced in the internal combustion engine.

As the technology advances in the engine development field for ICE vehicles there will be less waste heat available and the necessity of additional cabin heat can be expected to grow in this sector as well. Thus the HP can be expected to have a promising future as provider of cabin heat in vehicles.

A quick background check in patent databases (esp@cenet and PatentStorm) indicate that Asian countries like Japan and Korea are presently leading this development by delivering and patenting a wide range of HP based solutions for cabin heating purpose of the EV.

In conclusion, the following can be expected to promote future applications of HP's in EV's, and other future vehicles as well;

- Increased electric energy available for traction in the EV.
- Elimination of fuel dependence for heating purpose in the EV.
- Rising fuel prices for oil based cabin heaters as well as consumer demands for "clean" heat.
- Expected growing demand due to demands for higher fuel efficiency in ICE's.
- Demand for a shorter response time for heating the cabin, compared to the engine waste heat method currently being used in ICE vehicles.
- Vehicle demisting operation by cooling the air on the evaporator for water condensation and then re-heating it to provide comfortable dehumidified air.
- A faster de-frosting of the windshield due to the working principle of the HP in ICE vehicles.
- A possible way of decreasing the re-charging-time of the EV-battery, by means of combining a network of micro-channel evaporating tubes for intermediate temperature control between the stacked Li-Ion batteries in future EV's. This concept will be introduced and briefly discussed in this thesis.
- Improved overall energy efficiency of future EV/-HP, based on the above.

These reasons are believed to change the present purpose and design of today's HP-systems as encountered in AC-only systems. The utilization time per year of the AC/HP system (compared to cooling only) can be prolonged, especially in more temperate regions were the heating mode will be more significant. That could justify the necessary investment increase in such a system.

1.5 Goals and expectations

As a consequence of the upcoming transition and phasing out of the commonly used R134a refrigerant in the automobile sector, a large part of this project analyses the R744 TVCC, which is expected to become the next generation refrigerant in future vehicle AC-systems. R744 TVCC can be the future in EV as well.

Due to the present debate whether R744 should be chosen as the replacement refrigerant in future vehicles, this project mainly aims for a deeper understanding and theoretical evaluation of what the TVCC would require when heating an EV. Two different R744 cycles are compared theoretically.

Further on, the relative performance of R744 is compared to R134a. This is expected to give an indication of which refrigerant to be suggested for heating an EV.

The possibility of improving the existing standard automobile R744 TVCC believed to be the next generation of automobile AC-systems is also studied. Finding adaptable alternatives to increase this system is highly interesting in order to reduce the electric power needed to run the system.

Thus, complying with the existing standards given for the R744 TVCC AC-system for vehicle applications it is expected to provide an alternative design for such a system with the **main purpose** of **heating the cabin**.

1.6 Delimitations

Due to the early stage of development of the standard automobile R744 TVCC AC, very little information regarding specifications of components, cycles and their particular performance is available. Due to this, an **economical** evaluation is excluded for the suggested system presented in this report.

The “new” Li-Ion EV’s are as well under a present development phase, thus with the practical absents of such a vehicle and the necessary information of thermodynamic properties concerning it, this study is limited in its accuracy of determine a qualitative thermodynamic analysis of its function.

Unsteady state conditions and transient behavior of the cabin thermal analysis is not analyzed due to the complexity and absence of a real test vehicle as well as a real test HP cycle. The analysis is thus based on estimated heat demand in order to keep a certain temperature difference between the cabin and the ambient temperature under steady state conditions. The heat demand has largely been based on recommendations given by Eberspächer [15].

The thermodynamical analysis of compared cycles, assumes only the ideal case were perfect heat transfer occurs between the refrigerant and its surroundings. Pressure drops have as well been excluded from all calculations, thus also being dependent on the component. For the same reason, the required fan power inside the cabin as well as on the outside is excluded in this study.

Competing synthetic refrigerants with R744 are in the development phase. Thus, the absence of thermodynamic properties of such refrigerants makes a further comparison difficult.

1.7 Method and procedure

To gather past and present achievements of the R744 TVCC, Science Direct database has been used extensively. A review of past papers published in this database has been the primary tool for understanding the behavior of the cycle.

Contacts with the industry were initially established but were difficult as they were still in the development phase concerning R744.

Engineering **e**quation **s**olver (EES) [16] software was used for the analysis of the presented cycles. EES provides high accuracy thermodynamic properties for carbon dioxide based on the equation of state developed by R. Span and W. Wagner [16]. Properties for R134a are provided based on the fundamental equation of state developed by R. Tillner-Roth and H.D. Baehr [16].

By assuming a required heat capacity output from the cycle and defining the degree of super heat, evaporator outlet quality and a temperature difference between the refrigerant gas cooler/condenser outlet and the heated air in the cabin it is possible to achieve a theoretical evaluation of each cycle and refrigerant performance.
The results obtained from 100 runs of sets of the calculated parameters, using EES, are then plotted.

Cool Pack software [17] developed by the department of mechanical engineering at the technical university of Denmark, was used for the estimation of the de-frosting case. This software is semi integrated with EES thus e.g. providing the same equations of state for moist air as can be found in EES.

2. R134a phase out

2.1 Observations on the ongoing refrigerant replacement debate

As mentioned earlier, the automotive industry is facing an upcoming challenge were at the present refrigerant R134a used in the AC systems is to be replaced and faced out starting in the year 2011 in Europe.

At this point the most promising refrigerant seems to be carbon dioxide, also called R744. Likely it will be chosen as the upcoming replacement in the automobile sector. The implementation of this refrigerant will need to address practical differences concerning design and performance, compared to normally used refrigerants until today. Refrigerants like HFO-1234yf, a “slow burning” and low GWP refrigerant could though be an alternative – promoted by the fluorocarbon industry [18].

One of the main reasons for the choice of R744 is that it complies with the demands requested for refrigerants in automobiles for being non-flammable and non-toxic in small amounts. The low **g**lobal **w**arming **p**otential (GWP=1) strengthen its case further.
The biggest practical challenge for the implementation of this refrigerant is the nature of the cycle, operating at relatively high-pressures. The development of high-pressure

components seems to have developed greatly though during recent years. Still very little is known about the resurrected R744 TVCC in terms of function, both theoretically and practically. The German carmakers have recently decided to choose R744 as the replacement refrigerant [19] for future AC systems.

Behind the replacement of the R134a refrigerant lays the Kyoto and Montreal Protocols initiating actions towards a more environmental and sustainable development for the future of global climate.

Further on, several interesting arguments see ref. [20], in favor of the introduction of R744, like the fact that R744 is already available in abundant amounts and therefore a future production and distribution system will not be a problem because of this. Another interesting aspect is the future possibility to end the refrigerant-producer monopolies, thus resulting in cheaper refrigerant for future consumers.

Within the automobile industry the present stage of development now is to make prototypes and carefully test them. Ixetic announced [21] in December 2007, to be the first company to start a serial production of a R744 car compressor model LA25K. Japan, with Sanyo, has already commercialized stationary R744 HP systems for household applications, and has been reported to sell large amounts of units.

Further noticeable optimism is seen by the fact that the Society of Automotive Engineers SAE, is presently working on standardizations for R744 TVCC. At this moment (2007), SAE J639 exists, while SAE J2772, SAE J2769, and SAE J2774 etc are still under development. A complete set of standards for automobile R744 TVCC, not only for cooling the passenger cabin but also for heating, from behave of SAE is thus still necessary for the implementation of such a system on the market.

By reviewing the SAE Automotive Alternate Refrigerant Systems Symposium presentations, from Events of 2002, it is clearly noticeable that R744 dominates that years happening. By looking at this year's events (2007), it can be seen that the chemical compounds are catching up for a place in the spotlight as a future option.
Noticeable is thus that bigger chemical companies like DuPont offering the refrigerant DP-1, INEOS-Flour offering the refrigerant AC-1 and Honeywell offering the refrigerant Fluid-H followed by Solvay, Sinochem and Arkema are participating in the race of future opportunity in the automotive sector.
Also noticeable is that ACEA, European Automobile Manufacturers Association, is at this moment not excluding other refrigerants, such as R152a, sometimes referred to as GAR refrigerant, as an alternative for the approaching critical refrigerant transition year 2011 [22]. HFO 1234 yf, as previously mentioned, is also participating in this race.

The positive perception of R744 is thus not shared by everybody. This can be explained not only from the quote below, but also from [23]. The opposition of the possible transition to R744 in the automotive sector is discussed and summarized with the conclusion that R744 should not be chosen as the replacement refrigerant for different reasons.

It can be concluded that there are mainly two different approaches noticeable in the R134a automobile replacement debate. They can be classified in the following way:

- Adapting new refrigerant to present technology:

  Replacement of R134a should be based on finding a proper refrigerant that **meets** the **current** AC-systems **refrigerant properties** and comfort demands, in terms of thermodynamical cycle similarity etc, consequently leading to a much easier transition. Sticking with the subcritical cycle means proper adaptation for existing AC-components in current systems;

  > *"IF there is a consensus that at least one chemical refrigerant will be acceptable, then Renault/PSA/Fiat suggest to give up R-744 now to focus on chemical refrigerants in order to relieve OEM's and Tier's resources" …… [24]*

- Adapting new technology to natural refrigerant:

  Replacement of R134a should be based on finding a proper **environmental friendly** refrigerant that meets the current AC-systems comfort demands. The R744 TVCC is thus a god option, differing in thermodynamical cycle similarity etc, consequently leading to harder transition and means of development and introduction of AC-components in existing refrigerating systems;

  > *"….In reality, according to the second law, the theoretical efficiency depends exclusively on the thermodynamic process that is realized, and is independent of the properties of the refrigerant. Rather than trying to develop a chemical compound to satisfy the requirements of a particular cycle, the cycle should be adapted to the properties of the substance that we want to use…."…… [20]*

2.2 Choosing a refrigerant

The present SAE international Safety Standards for Motor Vehicle Refrigerant Vapor Compressions systems – SAE J639 [25], implies **shall be of low toxicity, non-flammable, and non-explosive.** That rule concerns refrigerants directly exposed to the passenger compartment or air handling system, including blends of refrigerants, both in their original composition and in compositions created because of normal refrigerant system operation conditions. The definitions can be found in the rules from American Society of Heating, refrigeration, and Air Conditioning Engineers, Inc. and Standard ANSI/ASHRAE 34.

Based on the above the most suitable option is presently believed to be R744:

- GWP = 1
- ODP = 0
- Non-flammable
- Non-explosive
- Low toxicity in small amounts

3. Flash fogging and safety issues for automobile HP/AC

3.1 Flash fogging

Flash fogging is discussed briefly in [26]. Flash fogging is referred to as the phenomenon that occurs when the outside of the cabin gas cooler/condenser (the condenser is replaced with a “gas cooler” in a R744 unit) contains an amount of water that rapidly evaporates when the system is turning on. If the windshield is cold, the water vapor will condense on the windshield making it opaque. This is dangerous. As further stated in [26], using the same heat exchanger for both dehumidification/cooling and heating, during seasons when the ambient temperature lays between 0-10 °C, will require means of controlling mist formation. In other words, it can be concluded that it is not possible to switch between heating and cooling mode to rapidly. Moisture control on the reversible evaporator-gas cooler/condenser must be applied.

Two possible solutions are presented:

- Two separate heat exchangers, one for heating and one for dehumidifying/cooling. This option indicates possibilities for a non-EV, in such a way that the HP could be used for heating up the engine coolant which in turn would dissipate its heat in a separate heat exchanger into the cabin. This would accelerate the heating process of the cabin compartment. Due to the absence of engine heat in an EV (which this thesis is primarily about), and for practical/economical reasons this option is not valid here.

- Controlling the heat dissipation of the reversible heat exchanger by reducing its heating capacity during the initial period of heating when the liquid content on the gas cooler surface is large.

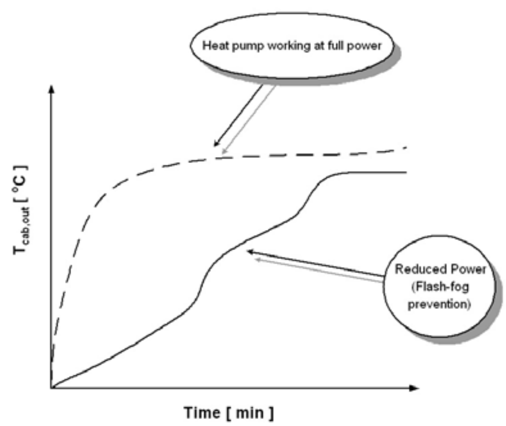


Figure 6. Air Temperatures downstream of Cabin HX vs. Operating Time

The method is based on trials from [26]. Fig. 6 shows how this approach would look like in terms of the hot air leaving the gas cooler/condenser inside the cabin corresponding to a stepwise increase of the heating capacity during start up of a wet evaporator in order to avoid flash fogging.

3.1.1 Flash fogging control method

If applying the same strategy as in [26] flash-fog control could be achieved.
As will be explained in section 6.2, the TVCC requires means of optimal COP control which practically is done by active control of the expansion device opening (mass flow). The method described below (fig. 7) corresponds to this case.
The same control method could of course be applied to a SVCC, with the difference that it would then not require the COP control regulation step, see fig. 7. Of course, it must be said that in order to conclude its function real tests must be conducted as the possibility of success strongly depends on equipment and weather conditions.

By actively measuring the temperature and humidity downstream of the gas cooler/condenser the dew point can be calculated. By actively measuring the corresponding windscreen temperature, the maximum allowable temperature rise (δT) on the windscreen in order to avoid moisture condensation can be calculated. The fan power control can here be assumed to be included in the heat pump controller.

The heat pump controller controls the desired heating/cooling capacity of the HP system by power adjusting the compressor (and displacement – in the case of R744). More over as mentioned earlier it is necessary to actively control the expansion device, e.g. by varying the displacement of an expander as in the case of the improved R744 cycle presented in section 5.4, for optimal COP. Thus, a simultaneous possibility of controlling the optimal COP under flash-fog-prevention-start-up is desired and should be integrated in the control system.

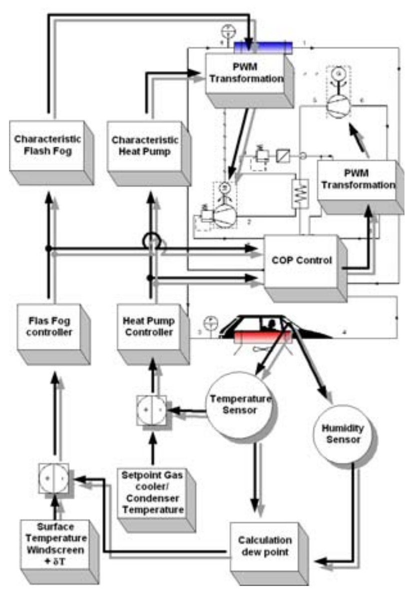


Figure 7. Flash fog prevention technique. Based on [26]

The scheme in fig. 7 shows the process described earlier – Registration of temperature/humidity – electronically processing data – integration of data to optimal COP data (unnecessary for SVCC) – conversion of electrical impulses (Pulse Width Modulation) to mechanical ones in the compressor and expansion device (optional for

SVCC). The cycle in fig. 7 corresponds to a cycle which is presented in this thesis and described in section 5.4. For further detailed information regarding the different components necessary in the method discussed above it is suggested to contact the authors in [26].

This concept could be improved in terms of passenger comfort (thus the initial time in a vehicle demands the highest heating capacity) by adding e.g. electric/- electric seat heaters to the vehicle or including a heat buffer which could be used as additional heating. More over if sufficient electrical power is available; the heat pump could be programmed or activated with a remote control to increase the cabin temperature to a "safe" temperature before entering the cabin. A remote control is e.g. already commercialized for the Eberspächer oil burner and goes under the name "EasyStart R+ remote control" [15]. Another method of attack could consist of window-integrated electric resistance heater with non-visible wires for the driver. This is today relatively common in non-EV.

An alternative HP , fig. 39-41, with the possibility of running the HP under conditions such as the ones discussed here, during recharging of the battery which could consequently keep the cabin/windscreen sufficiently hot, as well as the gas cooler/condenser sufficiently dry, and thus possibly avoiding the flash fog problematic, if not entirely partly. This method could thus be well suited in general, for all vehicle HP applications. This approach would consequently need part of the recharging energy to run the compressor, but at the same time reduce the temperature elevation in the battery, meaning that the battery-recharging process could become more effective and at the same time keeping the cabin at a temperature safe from flash-fogging risk during start up.

3.3 Safety regulations for the R744 TVCC AC-system

The presently existing safety measures given for the automobile R744 TVCC AC systems are composed by SAE J639 [25]. In [25] the high-pressure side is defined as, the refrigerant system from the compressor discharge chamber to the expansion device inlet chamber. The low pressure-side consequently consists of the remaining parts of the VCC.

Among other things, it is said that the maximum high-side pressure should not exceed 17,6 MPa. The maximum low-pressure side should not exceed 13,2 MPa.
The system should have a pressure relief device on both pressure sides. The purpose of the pressure relief device is thus to prevent the system from turning-on, as well as hindering operation when "unsafe" pressures are encountered. More over the VCC-system is recommended to have the appropriate number of pressure relief devices so that all components are protected from possible component burst. E.g. it is specified in [27] that a pressure blow-out disk should be chosen on both sides as a pressure relief device. For more information regarding these disks, please contact the authors of [27].
Further recommendations regarding fittings etc. are given in [25], although not much is said about temperature limits.

The refrigerant temperature limits, due to material stability and safety, are set to be a maximum of 165 °C in permanent running mode and 180 °C for short term operation according to J. Werthenbach at DC et al. [28]. More over Dr. C. Reibinger at Audi AG [29] sets the same limits to a maximum of 175 °C in permanent running mode and 190 °C for short term operation.

From Table 1, it can be seen that the compressor temperature limit has been increased from [28] to [29] during the timeline from 2002 – 2004. Consequently it can be assumed that the increase in temperature (+ 10 °C) for the compressor presented in [29] will thus lead to the following table:

Components	Temperatures [^{o}C]		
	Min	Max	Periods < 5 min
Evaporator, Expansion device	-10	80	80
Compressor	-10	175	190
Flexible Hose Connector to Gas-Cooler	-10	165	180
Gas Cooler	-10	155	170
Other Components	-10	80	80

Table 1. Temperature Limits- Automobile R744 TVCC AC-Components. [28] and [29]

It should of course be mentioned that all limits in terms of safety regulation for the mobile R744 TVCC presented in this section are dependent on the specific components used in the system. These figures can thus bee seen as guidance based on presently existing recommendations.

More over [29] recommends adopting so called safety barriers in the R744 TVCC. This is done by keeping the pressure and temperature below certain limits during its operation. According to [29] the system pressure should be checked before start-up so that it lies within an upper limit of 9,5 MPa and a lower limit of 2,5 MPa to prevent start-up difficulties. The upper limit for the non-operational R744 TVCC is however set to an absolute maximum of 13 MPa, thus corresponding approximately to the limit set in [25]. A tolerance of 2 MPa is suggested in [29] for the activation of the low-pressure relief device of the system during non-operation, consequently activating it at 11 MPa preventing the system to be turned on.

During operation of the TVCC the maximum high-side pressure is recommended to 17 MPa, thus corresponding approximately to the limit set in [25]. A tolerance of 2 MPa is suggested in [29] for the activation of the high-pressure relief device of the system during non-operation, consequently activating it at 15 MPa and thus turning off the system. However for “stable” compressor control, the maximum working pressure is set to 13,3 MPa in [29]. Reaching this limit means that a reduction of the compressor displacement (reduced capacity) should occur without turning off the system entirely.

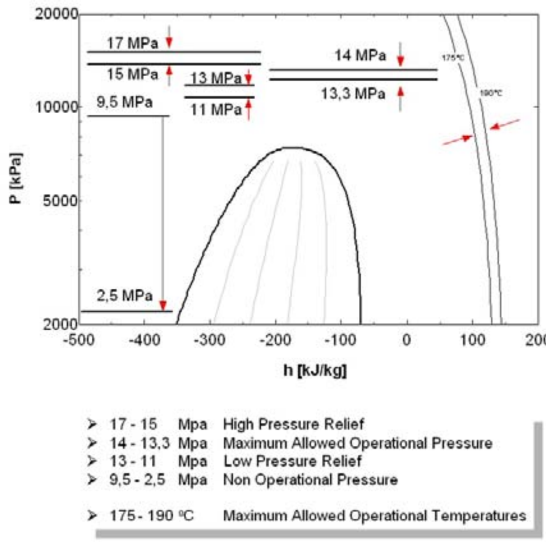


Figure 8. Pressure and Temperature limits- Automobile R744 TVCC AC-System. [28] and [29]

3.3.1 Purpose and function of safety regulations

The safety system works in the following way, the low-pressure relief device checks that the initial pressure in the system is in order, the compressor starts and is supervised by its own control system (could e.g. be included in the HP controller, see fig. 7). The system will not start unless the initial conditions are satisfactory. If the control system should “break down” then the overlooking high-pressure relief device acts like a backup safety system. If these 3 safety measures are supposedly out of function then the systems minimum burst pressure is encountered being twice that of the maximum high- and low-pressure relief device consequently 34 respectively 26 MPa [28]. Note that SAE J639 [25] names this upper limit as the “ultimate burst pressure”.

For further protection of the passenger cabin besides those stated in [29] it is recommended that all connections not being of welded or brazed type, such as e.g. screwed connections, have to be put **outside the passenger cabin**. Meaning that the evaporator (and hot gas cooler when heating) and its connections inside the cabin have to be brazed. The protection of the driver and all passenger is further achieved by a non-bursting and braking of the evaporator (and condenser/gas cooler when heating) inside the passenger cabin. Careful tests based on pressure-, impulse-, temperature-, cycling, vibration- and corrosion should therefore be properly carried out.

In [29] recommendations regarding R744 leakage are presented for the evaporator (cooling mode) inside the cabin. Although the first safety measure is the so called “safe evaporator” the following can be said concerning “worst” operational air cooling conditions based on a starting pressure of 6 MPa and superheated R744 at 50 °C, the diameter of the circular leaking point should not in any case exceed 0,13 mm. This diameter corresponds to the leakage rate of CO_2/ R744 equivalent to the maximum human respiration, or meaning 3 g/min [29]. A reasonable limit of allowed CO_2/ R744 volume concentration, lays in the limit of 5 % of occupied space [27].

Factors that will influence this limit are:

- Rate of R744 outflow into passenger cabin.
- Formation of dry ice inside system at 0,52 MPa.
- Amount of R744 dissolved in lubricant.
- Cabin ventilation rate.
- Stratification due to R744 being heavier than air.

4. Standard automobile R744 TVCC AC

4.1 Introduction to the standard automobile R744 TVCC AC

As basis for the safety regulations implemented earlier lays the standard R744-A/C-system which was presented by German OEMs in 2003, consisting of Audi, BMW, DaimlerChrysler, Porsche and VW. It appears at the SAE, Alternate Refrigerant Systems Symposium – in Phoenix 2003 [28].

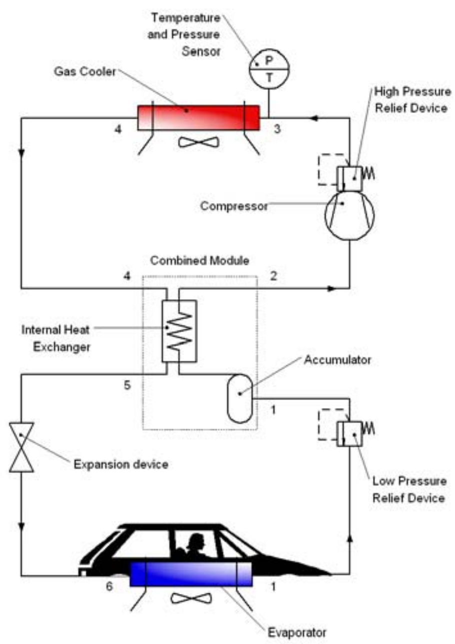


Figure 9. Standard Automobile R744 TVCC AC-System

This cycle will be simulated (with the gas cooler placed inside the cabin) in the comparative case study done in chapter 8-9. It will as well be evaluated for R134a. Note that when using R134a the cycle is referred to as "the standard R134a/- SVCC". This does however not mean that a "standard automobile R134a SVCC AC-system" corresponds to fig. 9.

4.1.2 Purpose and function of the standard automobile R744 TVCC

As stated in [29] the high-pressure relief device has to be positioned at the outlet of the compressor. The pressure and temperature sensor must be positioned between the compressor and gas cooler. *The un-evaporated refrigerant coming out from the evaporator is accumulated in the combined module for evaporation and superheating before going in to the compressor*. The low-pressure relief device has a protection purpose for when the R744 TVCC is turned of, in other words to protect the low-pressure side from reaching higher pressures. It can be placed optionally between the evaporator outlet and compressor inlet.

As further stated in [29] the first safety measure for the introduction of the R744 TVCC is defined by the suggested Standard R744-A/C-system. Secondly, it aims for complete implementation in all kinds and sizes of existing and future vehicles. Thirdly, it aims for implementation into fully controlled as well as manually controlled AC-system.
Safety targets are presented as following:

- Protection of driver.
- Protection of all passengers from any negative effects in regards to health.
- Protection of pedestrians/technicians during service.
- Avoid interference between the R744 safety system and other safety systems existing in the vehicle.

If applying the safety measures described earlier this should be full field.

4.2 Special properties of the transcritical fluid

The term “transcritical” is explained by the cycle working between a supercritical and a subcritical state. In other words when the “condensation” takes place above the critical point it is called gas cooling. The critical point is defined by - the critical temperature, which is the minimum temperature for liquefaction of a gas, and the critical pressure, which refers to the pressure required for liquefaction of a gas.
In the traditional SVCC using R134a, the refrigerant condensates below the critical point meaning that the refrigerant never reaches the supercritical state. The single stage TVCC has its **low** temperature- and pressure-side in the subcritical region and the **high** temperature- and pressure-side in the supercritical region.

Due to the low critical temperature of R744 (31,1 °C) the need for “condensation” is normally above this temperature. This means that it is not possible to transfer heat to the ambient temperature if the ambient temperature is higher than 31,1 °C. Instead we have to see it as cooling a supercritical refrigerant from a high temperature to a low temperature. Because of the high critical pressure, the system has to be treated differently than in traditional heat pumps.

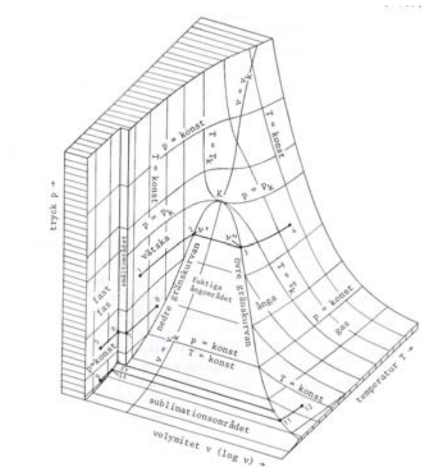


Figure 10. P,v,T-diagram – for a pure substance [30]

The supercritical fluid has a varying specific heat (c_p) while keeping the pressure constant. Around the critical point, this is most pronounced. The energy given off by the refrigerant in the gas cooling process can be expressed by the following equation (in and out stand for the refrigerant state condition):

$$\dot{Q} = \dot{m} \cdot c_p \cdot (T_{in} - T_{out}) \tag{4.1}$$

As can be seen in fig. 11 the c_p first increases and then decreases during the gas cooling process (going from right to left). From equation 4.1 it can be seen that a high c_p will give a low temperature change if keeping the other parameters constant. Pinch points will occur limiting the heat exchange. This indicates that the gas cooler requires extra attention in its design.

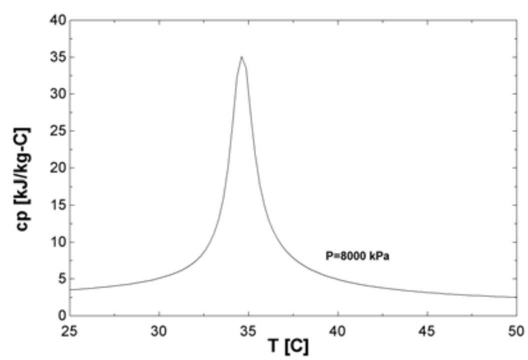


Figure 11. Specific heat variation vs. Temperature near critical point – R744

The pseudo-critical point is used to determine at what temperature the specific heat becomes a maximum for a constant supercritical pressure. This temperature will indicate the location, for a given heat transfer process, were the heat transfer potential reaches its minimum [27].

$$T_{pseudo} = -122{,}6 + 6{,}124 \cdot P - 0{,}1657 \cdot P^2 + \\ + 01773 \cdot P^{2,5} - 0{,}0005608 \cdot P^3 \qquad (4.2)$$

Because the ε-NTU and LMTD methods require a constant specific heat during their analysis they can not be applied to the analysis of a supercritical refrigerant near the critical point. The heat transfer analysis must instead be done, dividing the process piece by piece.

5. Improving the Standard automobile R744 TVCC AC

5.1 Flash-gas bypass concept

The flash gas bypass concept for R744 micro-channel evaporators was experimentally proven beneficial for the performance of the standard R744 TVCC by Stefan Elbel and Pega Hrnjak [31]. The benefit of this adaptation lies in:

- **Pressure drop reduction** in the refrigerant side off the evaporator.
- **Increased heat transfer coefficient**, especially when the expanded refrigerant vapor quality is high.
- An improved refrigerant distribution in the evaporator header and consequently through all the evaporator channels can be expected.

The consequence of feeding the evaporator with a two-phase refrigerant is that the vapor part will reduce the overall cooling effect. This means that only the liquid part will evaporate and thus leaving the vapor part relatively unaffected lowering the refrigerant cooling capacity. This is the case for both the isenthalpic and isentropic expansion process when placing the accumulator downstream of the evaporator, as in the standard R744 TVCC. Due to different densities of the vapor and liquid part of the expanded refrigerant the flash-gas tank serves as both as an accumulator and separator in the cycle.

A flash-gas bypass valve is needed because of the pressure drop that will emerge in reality in the evaporator. The valve then lowers the bypassed vapor-part pressure (isenthalpic) to equal the evaporator exit pressure enabling continuity in the direction of the flow. This valve can suggestively be a thermostatically actuated or a quality censored valve for optimal function.

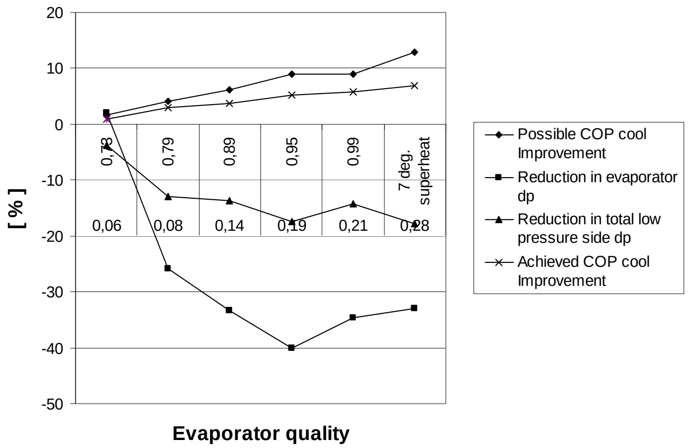


Figure 12. Flash-Gas Bypass Improvement vs. Evaporator Inlet-Outlet Quality [31]

The chart above is correlated from experimental results presented in [31] for a given set of conditions. It shows achieved improvements of COP_{cool} and refrigerant pressure loss reduction in the low-pressure side (consisting of an evaporator and IHX) of a tested R744 TVCC. The comparison is done in respect to the case for when the accumulator is placed after/downstream of the evaporator, as is the case e.g. in the standard automobile R744 TVCC AC. The refrigerant inlet and outlet qualities resulted from the experiments carried out in [31].

By using a variable speed compressor (this was however not the case in [31]) and the flash-gas bypass concept the possible COP_{cool} improvement is believed [31] to become even grater. Simultaneous control of the variable speed compressor and expansion device is explained and shown in [32]. Active control of these parameters is reported to have a 7-9 % COP_{cool} improvement over the case for when the accumulator/refrigerant buffer is placed after/downstream of the evaporator. By improving the COP_{cool} of the system it can be expected to improve the COP_{heat} as well, as:

$$COP_{heat} = COP_{cool} + 1 \qquad (5.1)$$

The authors in [31] varied the cooling capacity in the evaporator thus resulting in different sets of entry and exit qualities of the evaporator as shown in fig. 12. The refrigerant quality, also called vapor quality, is defined by the vapor fraction of the refrigerant, thus within the saturation line (two-phase region) the fluid consist partly of liquid and partly of vapor. As can be seen from the experimental data the evaporator exit quality of 0,95 and corresponding inlet quality of 0,19 showed highest pressure drop reduction. Both the “standard” cycle and flash-gas cycle showed a maximum COP_{cool} at an evaporator exit quality of around 0,9 (90 % vapor and 10 % liquid). Although by looking at fig. 12, it can be seen that the relative improvement of the flash-gas bypass cycle to the “standard” continues to increase in spite of the mutual reduction in COP_{cool} at higher exit qualities. As explained in [27] the evaporator exit is recommended to be kept slightly wet were as the 7 degree super heated exit quality can be disregarded in the real case for the conditions presented in [31], this is also discussed in [31]. For more information regarding the recommended slightly wet R744 evaporator exit, please contact [27] or [31].

It has to be pointed out that, the comparison done in [31] compares the improvement of this concept in respect to an isenthalpic expansion (expansion valve) and not an isentropic expansion (expansion machine). The same argument yields for [32]. In an isentropic expansion process the vapor fraction of refrigerant mixture leaving the expansion device will be slightly lower than in an isenthalpic expansion, see fig. 15. The isentropic line brings the expanded mixture more to the left in P-h diagram meaning that the vapor part of the mixture decreases. Based on this it can be assumed that the flash-gas bypass concept will improve the performance even of an expander cycle, although slightly less in comparison to the possible improvement on the isenthalpic expansion cycle.

5.2 Work recovery trough an expander-generator unit

As a way to improve the R744 TVCC COP, implementation of an expander is presently being widespread investigated. The expander is meant to recover work from the expansion. Presently the research done in this area focuses on combining an expander and a compressor in one unit, thus aiming for as low mechanical losses as possible. Often the work is used in a second stage compressor driven by the expander.
This method requires a controllable intermediate pressure control for optimal COP performance of the cycle. Different approaches have been presented, by varying the expander-compressor unit position from the first stage compression to the second stage compression with inter-cooling in between etc.

Jun Lan Yang et al. [33] analyzed three different approaches for the implementation of an expander in the R744 TVCC, or as they call it single-stage compression cycle with throttle valve (SCV). The SCV is the standard R744 TVCC (Isenthalpic expansion) without an IHX. The cycles are classified in the following order:

- TCOP, Two-stage compression using an optimal intermediate pressure
- TCDH, Two-stage compression with an expander driving the high-pressure stag
- TCDL, Two-stage compression with an expander driving the low-pressure stage

In the TCOP the expander and low stage compressor are operated independently without being coupled to the same shaft. This approach gives high transmission losses compared to the other cycles, but enables a flexible control between the expansion and compression. This approach lets the expander work independently from the compressor meaning that the desired pressure at the exit of the gas cooler can be regulated as needed for optimal cycle performance, and at the same time letting the low stage compressor reach its optimal intermediate pressure without any flow restriction from the expander.

The TCDH has the high stage compressor and the expander on the same shaft, forcing the expansion and compression to work at the same rotational speed. The advantages of this design are primarily the reduced transmission and leakage losses between the expansion and compression processes. The same argument yields for the TCDL.

The results for a given set of conditions by the analysis done in [33] implies that the TCDH outperforms the relative performance of the SCV in terms of COP_{cool} by 45,62 % and the TCOP is improved by 44,58 %.
More over the analysis done in [33] compares the above-mentioned cycles with the so called SCE, single stage compression with expander cycle. This is basically the SCV equipped with an expander instead of a throttle valve.
In respect to the SCE, the larger improvement of the TCDH over the TCOP is only 0,72 %. It is also mentioned that the ratio of the density at the main compressor inlet and the expander inlet should be fixed for the TCDH and TCDL, thus requiring some kind of operational control to avoid deviation from the design point. This is not the case for the TCOP.

5.3 Suggested improvements to the standard R744 TVCC

By the implementation of a combined expander-generator module, the standard R744 TVCC can be assumed to improve its performance by work-recovery, which is otherwise lost in an isenthalpic expansion. This can be explained by the reduced entropy generation achieved by the isentropic expansion, bringing the cycle closer to the ideal Carnot cycle described later. This should be the case even for a SVCC.

Further on it is shown that by changing the position of the accumulator (referring to the flash tank in the flash-gas bypass concept described earlier) placing it upstream of the evaporator instead of behind as in the standard cycle it is possible to increase the performance even more. This can as well be explained by the reduced entropy generation achieved by the reduction pressure drop in the evaporator, bringing the cycle closer to the ideal Carnot cycle described later. This should be the case even for a SVCC.

Finally, it is necessary to make the cycle reversible in order to provide the required de-frosting possibility and AC-function when needed. By implementing the suggested 8/2-valve [31] (8 ports and 2 positions/directions) this necessity is met. More over this particular valve maintains the flow direction in the system without causing any sudden changes in flow momentum due to the change in direction, thus avoiding any further unnecessary losses due to this.

5.4 The improved automobile R744 TVCC HP/AC

By implementing the required safety measures stated earlier, and adapting the standard cycle with the previously presented improvements the cycle takes the following form:

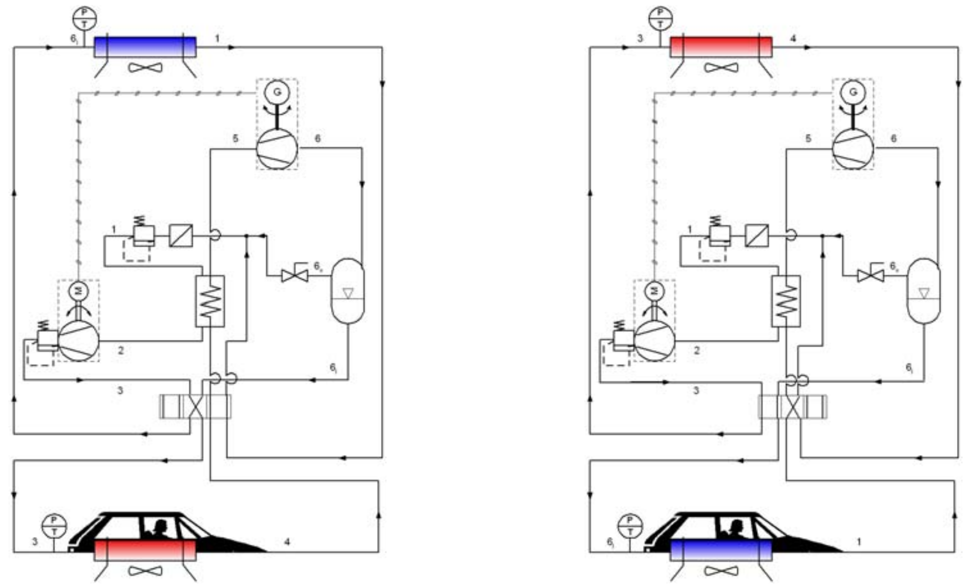


Figure 13. Improved automobile R744 TVCC HP/AC. Left- HP-Mode, Right- AC-Mode

Note that the cycle presented above, as well as the improved 12/5-EV-HP/AC cycle, see fig. 37-43, could of course work with other refrigerants other than R744. The denomination (R744) of the cycle shown above is due to the nature of the comparative work carried out in this thesis. More over the recommendable dryer/filter is located ahead of the low pressure relief device. This location is chosen due to solvent properties of R744 found at high pressures.

6. Basic thermodynamics and cycle analysis

6.1 The ideal Carnot cycle

The Carnot cycle is used in thermodynamical analysis because it is the most efficient cycle. A comparison of another designed cycle's efficiency, or coefficient of performance (COP), is commonly done using its equivalent Carnot-COP as reference. The person who described this cycle was called Nicolas L.Sadi Carnot, whom described it in a paper written in the year 1824 called "Réflexions sur la puissance motrice du feu, et sur les machines proper a déveloper cette puissance." [30] Carnot confirmed the second law of thermodynamics before being officially introduced as an axiom. He postulated that the efficiency of a heat engine depends on the highest and lowest temperatures reached in the cycle. By cycle we define the following, thermodynamic processes that after numerous stages return a system to its initial state.
The theoretical considerations supporting the idealized reversible Carnot cycle are the following:

- An isentropic compression
- An isothermal heating
- An isentropic expansion
- An isothermal cooling

And translating this to a more comprehensive form e.g. imagining a piston compressing and expanding inside a cylindrical tube this can be described as it having a:

- A perfect sealing, in a way that no atoms can escape from the working fluid as the piston moves.
- A perfect lubricant, in a way that no friction emerges.
- A working fluid that does not exist.
- A perfect thermal contact to one or to none of two reservoirs and with a perfect thermal insulation isolating it from all other heat transfers than the ones intended in the cycle.

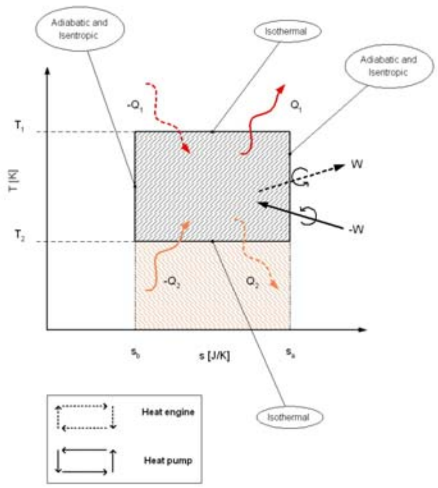


Figure 14. T-s diagram, Ideal Carnot cycle

6.2 COP behavior of the basic TVCC

6.2.1 Varying high side pressure and temperature

In comparison to the basic R134a SVCC it is possible to see a big difference in the COP behavior while applying an elevation of pressure and temperature at the high-pressure and temperature side of the cycle.
In a general form, the cooling COP is:

$$COP_{cool} = \frac{Q_2}{W} \tag{6.1}$$

The influence of the higher pressure and temperature of the cycle will require additional added compressor work and a reduction of refrigerating capacity:

$$COP_{\underset{-}{cool}} = \frac{Q_2 - \delta Q}{W + \delta W} = \frac{Q_{\underset{-}{2}}}{\underset{-}{W}} \tag{6.2}$$

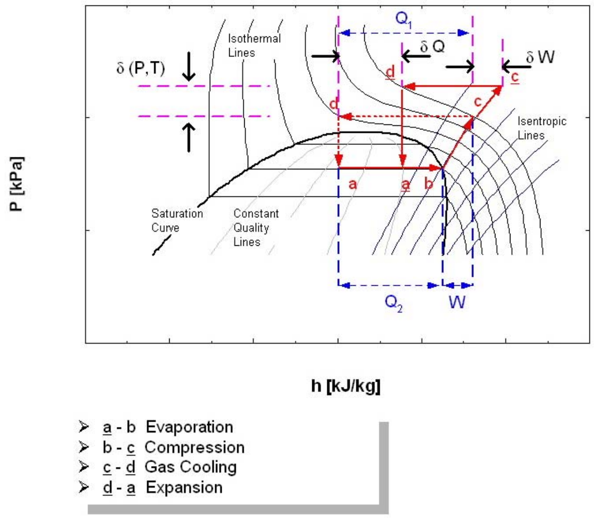


Figure 15. P-h diagram - TVCC – High-side pressure and temperature variation

As Q_2 obviously decreases and W increases this leads to a decrease of the cycles COP_{cool}. It can be seen from fig.15 that there is even a possibility for the COP_{cool} to become smaller than one, meaning that the cycle then would need more power input from the compressor than the actual refrigerating capacity. As for the **heating COP** the same argument yields, as:

$$COP_{heat} = COP_{cool} + 1 \tag{5.1}$$

A conclusion for the basic **TVCC** COP is thus generally – to keep the temperatures down on the high side.

6.2.2 Varying high side pressure

The influence of the higher pressure while maintaining the gas cooler exit temperature constant and evaporator temperature constant as well, now leads to the need of an increment of compressor work as well as an increment of the compressor exit temperature. Note that the cycle will now require additional compressor work and an increment of refrigerating capacity.

$$COP_{\underline{cool}} = \frac{Q_2 + \delta Q}{W + \delta W} = \frac{\underline{Q_2}}{\underline{W}} \tag{6.3}$$

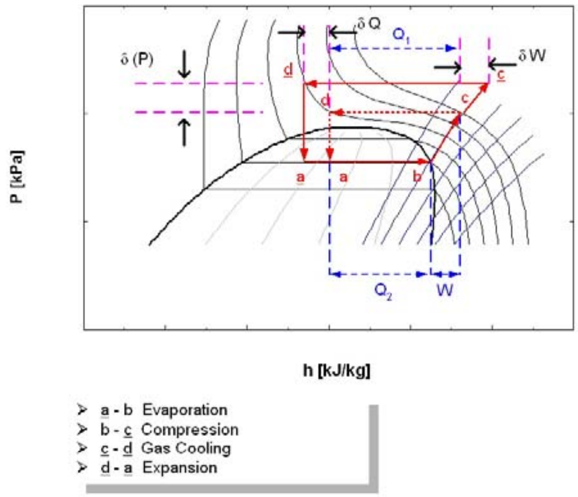


Figure 16. P-h diagram - TVCC – High-side pressure variation

It seems that in some pressure interval the $\mathrm{COP}_{\mathrm{cool}}$ can be kept relatively constant while changing the gas cooler pressure and keeping the same gas cooler exit temperature. It can also be noticed that in some point this relation will change, thus following the isotherm upwards (increasing pressure) from the critical region indicates a nearly vertical slope meaning that the additional compressor work will start decreasing the $\mathrm{COP}_{\mathrm{cool}}$ due to the stop in increment of the refrigerating capacity. At a higher pressure, following the same isotherm, the refrigerating capacity will actually start decreasing if assuming that the equations of state used in ESS [16] for Carbon dioxide are valid, as done in this thesis.

The heating capacity will follow the compressor work increment during its vertical up going path on the isotherm. For an appropriate defined interval, the following can be estimated:

$$\delta W \approx \delta Q \Rightarrow \mathrm{COP}_{\underset{-}{\mathrm{cool}}} \approx \mathrm{COP}_{\mathrm{cool}} \tag{6.4}$$

And thus:

$$\left\{P_{c-d}\uparrow, T_c\uparrow, T_{\text{d-cons}}, P_{\text{a-b-cons}}, T_{\text{a-b-cons}}\right\} \Rightarrow$$
$$\Rightarrow \mathrm{COP}_{\text{cool-cons}} \Rightarrow \lim$$

As for the heating COP it can bee seen that denominator will increase slower, equation 6.5, than the nominator, meaning that the $\mathrm{COP}_{\mathrm{heat}}$ will increase with increasing gas cooler pressure while keeping the gas cooler exit temperature constant within a certain region.

As well as for $\mathrm{COP}_{\mathrm{cool}}$ there seems to be a limit for this trend as well, that is at some point the heating capacity will start increasing almost linearly with added compressor work while maintaining the $\mathrm{COP}_{\mathrm{heat}}$ almost constant.

$$COP_{heat} = \frac{Q_1}{W} \tag{6.5}$$

$$COP_{\underset{-}{heat}} = \frac{Q_1 + \delta Q + \delta W}{W + \delta W} = \frac{Q_{\underset{-}{1}}}{\underset{-}{W}} \tag{6.6}$$

Conclusion for the basic TVCC COP analysis is thus:

$$\begin{aligned}&\left\{P_{c-d} \uparrow, T_c \uparrow, T_{d\text{-cons}}, P_{a-b-cons}, T_{a-b-cons}\right\} \Rightarrow \\ &\Rightarrow COP_{heat} \uparrow \Rightarrow \lim\end{aligned}$$

This concludes a comprehensive regulation of the COP_{cool} and thus leading to a proper regulation of the cycles COP by varying the high side pressure.

Given the above, it can be concluded that there exists an optimum gas cooler pressure for each gas cooler exit temperature following the cycle as described above with a corresponding given evaporation temperature and pressure to achieve an optimum COP for the TVCC. J.Sarkar et al. [34] has defined a correlation for this optimal pressure (**in bar**) as a function of the gas cooler exit temperature and evaporation temperature. The equation is valid for gas cooler exit temperatures ranging from 30 °C to 50 °C, and evaporation temperature from -10 °C to 10 °C [34].

$$P_{opt} = 4{,}9 + 2{,}256 \cdot T_d - 0{,}17 \cdot T_{ev} + 0{,}002 \cdot T_d^{\,2} \tag{6.7}$$

Note [34] states that the above correlation is valid for gas cooler exit temperatures below the critical temperature, 31,1 °C. Such temperatures will however not be addressed to in the analysis done in this study, due to the different nature of the isotherms below that temperature, see fig. 10 and 15. This correlation is thus hereby not recommended to be used below the critical point.

6.2.2 COP control by high side mass regulation

In a system with mass regulation control at the high-pressure side, the basic TVCC indicates that there must bee means for controlling momentary mass of the refrigerant located between the compressor outlet and the expansion device inlet for optimal function. As the refrigerant mass is defined as constant within the whole system, there should consequently bee a refrigerant buffer located somewhere in order to prevent a flooding or drying in the evaporator, e.g. hurting the compressor. This buffer/receiver/accumulator can be placed at various points in the cycle.

- Systems with a low-pressure receiver at the evaporator outlet, as in the case of the standard R744 TVCC, can be explained in the following way:

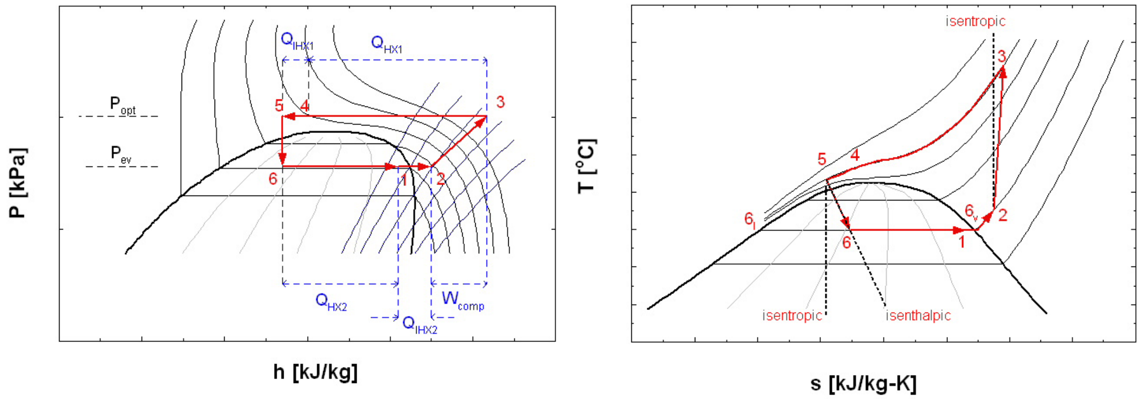
  Controlling the high side pressure by adjusting the expansion device and thus temporarily changing its and the compressor mass flow. Meanwhile there will be a refrigerant accumulation and pressure elevation in the high-pressure side. The vapor fraction at the evaporator exit may temporarily rise and the necessary high-pressure side charge will be taken from the low-pressure buffer until a new balance is found. In practice there will be a need of liquid bleed from the receiver to the compressor for lubricating purposes. As mentioned in [27] the evaporator exit is recommended to be kept slightly wet, whereas a liquid bleed from the receiver/buffer to the evaporator could be suitable.

Consequently, to effectively control the TVCC COP there is a need to adjust the high side pressure. This can easily be done by adjusting the expansion devise. That leads to an accumulation or reduction of R744 mass in the gas cooler/high-pressure side leading to the need of an accumulator at the low-pressure side. An accumulator at the low-pressure side in such a system would therefore work as a supplier of R744 to or from the low-pressure side making sure that there is enough refrigerant for the rest of the system to work properly. Additionally, when intending to operate the TVCC for heating in a reversed mode, it is also beneficial to have a refrigerant buffer for controlling the different refrigerant mass charges needed. This is briefly discussed for a specific system in [35].

- In the improved cycle, the buffer is placed **ahead** of the evaporator. In order to fulfill the reasoning explained in this section there must be a way of controlling the refrigerating power during the heating of the cabin. This can be done by active airside control, thus requiring some kind of airflow control over the outside heat exchanger. A Venetian blind type airflow restrictor can be recommended. This function would then of course also play an important role while de-frosting the evaporator, hindering the ambient air flow from cooling it down

7. Thermodynamic model and correlations

7.1 Energy balance – Standard R744 TVCC – HP-mode



6 - 1	Isobaric	Evaporation
1 - 2	Isobaric	Heat Exchange
2 - 3	Non-Isentropic	Compression
3 - 4	Isobaric	Gas Cooling
4 - 5	Isobaric	Heat Exchange
5 - 6	Isenthalpic	Expansion

6 - 1	Isobaric	Evaporation
1 - 2	Isobaric	Heat Exchange
2 - 3	Non-Isentropic	Compression
3 - 4	Isobaric	Gas Cooling
4 - 5	Isobaric	Heat Exchange
5 - 6	Isenthalpic	Expansion

Figure 17. P-h and T-s diagram, Standard R744 TVCC

Heat/Cooling-Power:

$$\dot{Q} = \frac{\delta Q}{\delta \tau} \tag{7.1}$$

Mechanical-Power:

$$\dot{W} = \frac{\delta W}{\delta \tau} \tag{7.2}$$

Heat balance for the gas cooler (located inside the EV):

$$\dot{Q}_{HX1} + \dot{Q}_{cab} = 0 \tag{7.3}$$

Heat dissipation from refrigerant:

$$\dot{Q}_{HX1} = \dot{m}_{ref} \cdot (h_3 - h_4) \tag{7.4}$$

Refrigerant mass flow:

$$\dot{m}_{ref} = \frac{\dot{V}_{rswf}}{v_2} \tag{7.5}$$

Real swept volume flow:

$$\dot{V}_{rswf} = \eta_v \cdot \dot{V}_{tswf} \tag{7.6}$$

Theoretical swept volume flow:

$$\dot{V}_{tswf} = V_{sw} \cdot \frac{N}{60} \tag{7.7}$$

Isentropic relation, compressor:

$$h_3 = h_2 + \frac{(h_{3is} - h_2)}{\eta_{is}} \tag{7.8}$$

Isenthalpic relation, expansion device:

$$h_5 = h_6 \tag{7.9}$$

Heat emitted from the hot refrigerant in the IHX to the cold refrigerant:

$$\dot{Q}_{IHX1} = \dot{m}_{ref} \cdot (h_4 - h_5) \tag{7.10}$$

Heat absorbed from the cold refrigerant in the IHX:

$$\dot{Q}_{IHX2} = \dot{m}_{ref} \cdot (h_1 - h_2) \tag{7.11}$$

Energy balance over the IHX:

$$h_4 - h_5 = h_2 - h_1 \tag{7.12}$$

Required compressor shaft power:

$$\dot{W}_{comp} = \dot{m}_{ref} \cdot (h_2 - h_3) \tag{7.13}$$

Required compressor electric power:

$$\dot{W}_{el,comp} = \frac{\dot{W}_{comp}}{\eta_{el,comp}} \tag{7.14}$$

Absorbed heat in the evaporator:

$$\dot{Q}_{HX2} = \dot{m}_{ref} \cdot (h_6 - h_1) \tag{7.15}$$

Energy balance over cycle:

$$\dot{Q}_{HX1} + \dot{Q}_{HX2} + \dot{Q}_{IHX1} + \dot{Q}_{IHX2} + \dot{W}_{comp} = 0 \tag{7.16}$$

Definition of COP for the standard cycle:

$$COP_{heat\text{-}standard} = \frac{\dot{Q}_{HX1}}{\left|\dot{W}_{el,comp}\right|} \tag{7.17}$$

Note that the energy balance for the standard R134a SVCC is done in the same way as above. The difference is however that the state point 4 (condenser outlet) is defined on the saturation (quality = 0) curve, corresponding to the approach temperature of 5 °C above condenser air out-let temperature. The pressure does not follow equation 7.27.

7.2 Energy balance – Improved R744 TVCC – HP-mode

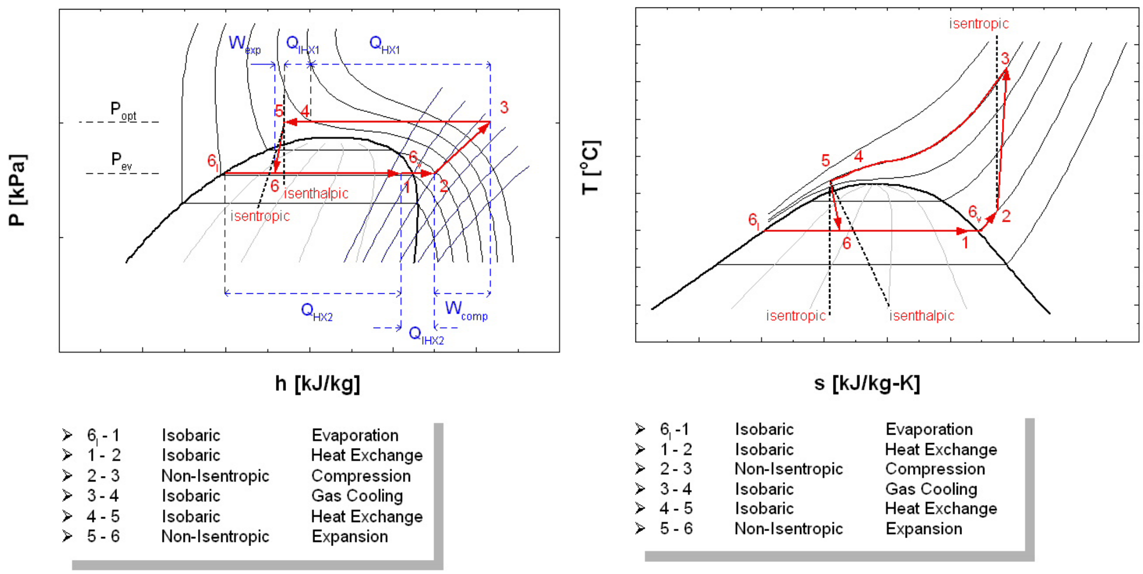


Figure 18. P-h and T-s diagram, Improved R744 TVCC

Absorbed heat in the evaporator:

$$\dot{Q}_{HX2} = \dot{m}_{ref,l} \cdot (h_{6,l} - h_1) \tag{7.18}$$

Liquid part mass flow of the expanded mixture that enters the evaporator:

$$\dot{m}_{ref,l} = (1 - x_6) \cdot \dot{m}_{ref} \tag{7.19}$$

Vapor part of the mass flow that is by-passed the evaporator:

$$\dot{m}_{ref,v} = x_6 \cdot \dot{m}_{ref} \tag{7.20}$$

Isentropic relation, expander:

$$h_6 = h_5 + \eta_{is,exp} \cdot (h_{6is} - h_5) \tag{7.21}$$

Shaft power delivered from the expander:

$$\dot{W}_{exp} = \dot{m}_{ref} \cdot (h_5 - h_6) \tag{7.22}$$

Delivered electric power from the expander:

$$\dot{W}_{el,exp} = \eta_{el,exp} \cdot \dot{W}_{exp} \tag{7.23}$$

Total electricity consumption of the cycle:

$$\dot{W}_{el,tot} = \dot{W}_{el,exp} + \dot{W}_{el,comp} \tag{7.24}$$

Energy balance over cycle:

$$\dot{Q}_{HX1} + \dot{Q}_{HX2} + \dot{Q}_{IHX1} + \dot{Q}_{IHX2} + \dot{W}_{comp} + \dot{W}_{exp} = 0 \tag{7.25}$$

Definition of COP for the improved cycle:

$$COP_{heat\text{-}improved} = \frac{\dot{Q}_{HX1}}{\left|\dot{W}_{el,comp} + \dot{W}_{el,exp}\right|} \tag{7.26}$$

7.3 Adopted correlations

7.3.1 Optimal pressure correlation

As described earlier there exists a high-optimal pressure to get an optimal COP for the R744 TVCC. The correlation described earlier is thus used to achieve optimal COP for the standard R744 TVCC as well as for the improved R744 TVCC. The standard cycle shown in fig. 9 assumes a cabin-heating-case with the gas cooler placed inside the cabin, instead of the ambient as shown. Following this, Sarkar's equation 6.7 [34] becomes the following:

$$\begin{aligned} P_{opt} &= 100 \cdot (4{,}9 + 2{,}256 \cdot T_4 - \\ &- 0{,}17 \cdot T_{ev} + 0{,}002 \cdot T_4^{\,2}) \end{aligned} \tag{7.27}$$

Sarkar's formula is multiplied with a factor of 100 in order to coincide with the other units defined in this analysis. More over the lower gas cooler exit temperature is set to 31,7 °C in this thesis, as explained in section 6.2.2.

7.3.2 Compressor isentropic and volumetric efficiency

The compressor isentropic efficiency is calculated from Douglas et al. [36]:

$$\eta_{is,comp} = 0{,}815 + 0{,}022 \cdot \left(\frac{P_{opt}}{P_{ev}} \right) - \\ - 0{,}0041 \cdot \left(\frac{P_{opt}}{P_{ev}} \right)^2 + 0{,}0001 \cdot \left(\frac{P_{opt}}{P_{ev}} \right)^3 \tag{7.28}$$

Note that this correlation was used in Sarkar's paper [34] providing equation 6.7.
The compressor volumetric efficiency is calculated from the correlation presented by Sarkar et al. [37]:

$$\eta_{vol,comp} = 0{,}9207 - 0{,}0756 \cdot \left(\frac{P_{opt}}{P_{ev}} \right) + \\ + 0{,}0018 \cdot \left(\frac{P_{opt}}{P_{ev}} \right)^2 \tag{7.29}$$

Note that the same correlations are used for R134a in the comparative analysis done in this study. The "optimal" pressure for R134a is in this case the condensation pressure.

8. Comparative case study

8.1 Energy balance – Cabin – HP-mode

This analysis considers the case were the cabin is supplied with sufficient heat to keep a certain over-temperature to the surrounding. No transient effects are taken into account.

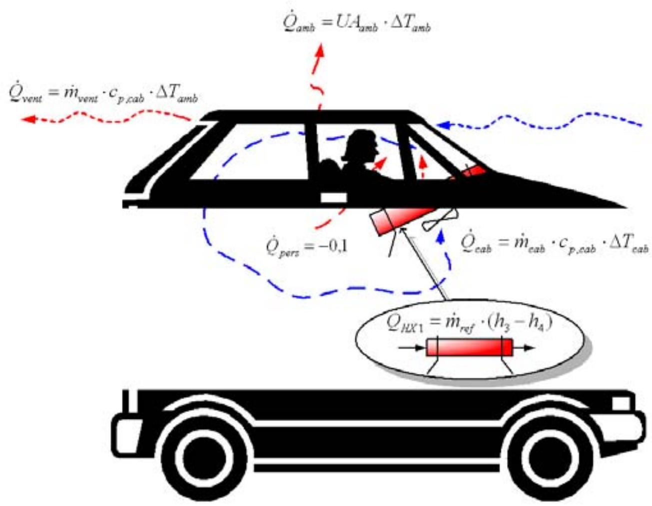


Figure 19. Heat Balance- Cabin

In reality there are numerous factors to take into account when dimensioning a heating system for a vehicle. The conditions inside the cabin as well as the outside vary continuously under normal driving conditions. For a more accurate thermal analysis of the vehicle compartment, it is recommended to contact e.g. the authors of [38].

8.1.2 Estimated UA-value for the cabin

In [39] it is recommended to have at least 1,8 air changes per hour (ACH) to achieve an adequate ventilation rate for a vehicle.

This gives, for the same car as in [15] given that the temperature difference between the ambient and cabin is 30 °C (given that the ambient is -10 °C) and assuming that the same air mass flow rate enters and exits the cabin, the minimum required ventilation losses:

$$\rho_{air} = \frac{348}{(T_{cab} + 273)} = \frac{348}{(15 + 273)} = 1{,}21 \qquad (8.1)$$

$$m_{cab} = \rho_{air} \cdot V_{cab} = 1{,}21 \cdot 4{,}5 = 5{,}4 \qquad (8.2)$$

$$\dot{m}_{vent} = \frac{ACH \cdot m_{cab}}{\tau} = \frac{1{,}8 \cdot 5{,}4}{60 \cdot 60} = 0{,}0027 \qquad (8.3)$$

$$c_{p_{cab}} \approx 1 \qquad (8.4)$$

$$\dot{Q}_{vent} = \dot{m}_{vent} \cdot c_{p_{cab}} \cdot (T_{cab} - T_{amb}) = \\ = 0{,}0027 \cdot 1 \cdot 30 = 0{,}081 \tag{8.5}$$

The heating demand for the simplified models compared in EES is done by assuming that the general recommendations in the Eberspächer [15] brochure are valid, thus for the case of a badly isolated vehicle with a cabin volume of 4,5 m^3 and a temperature difference of 30 °C, the following can be estimated for the heating demand of the cabin:

$$\dot{Q}_{cab} = \dot{m}_{cab} \cdot c_{p_{cab}} \cdot (T_{cab} - T_{cabout}) = -1{,}8 \tag{8.6}$$

Moore over the vehicle is assumed to carry one passenger with a heat dissipation equivalent to 0,1 kW, thus:

$$\dot{Q}_{pers} = -0{,}1 \tag{8.9}$$

Following that the cabin has an overall heat transfer coefficient, the heat dissipation from the cabin to the ambient can be expressed in the following way:

$$\dot{Q}_{amb} = UA_{cab} \cdot (T_{cab} - T_{amb}) = UA_{cab} \cdot 30 \tag{8.10}$$

And the heat balance for cabin becomes:

$$\dot{Q}_{amb} + \dot{Q}_{vent} + \dot{Q}_{pers} + \dot{Q}_{cab} = 0 \tag{8.11}$$

The UA-value for the cabin becomes:

$$UA_{cab} \cdot 30 + 0{,}081 - 0{,}1 - 1{,}8 = 0 \tag{8.12}$$

$$UA_{cab} = 0{,}061 \tag{8.13}$$

Note that this value corresponds approximately to the one measured for a Volvo truck cab in [38] which measured 0,065 kW/m^2.

8.2 Other assumptions

8.2.1 Assumed efficiencies

$$\eta_{el,exp} = 0{,}9 \tag{8.14}$$

$$\eta_{el,comp} = 0{,}9 \tag{8.15}$$

$$\eta_{is,exp} = 0{,}7 \tag{8.16}$$

$$\eta_{vol,exp} = 1 \tag{8.17}$$

$$\eta_{exp,m} = 1 \tag{8.18}$$

$$\eta_{comp,m} = 1 \tag{8.19}$$

8.2.2 Assumed temperature relations

Assumed temperature difference for the air leaving the gas cooler and the refrigerant leaving the gas cooler:

$$T_4 = T_{cabout} + 5 \tag{8.20}$$

Assumed temperature difference for the ambient air and refrigerant evaporation temperature:

$$T_{ev} = T_{amb} - 8 \tag{8.21}$$

Temperature of refrigerant at compressor inlet:

$$T_2 = T_{ev} + T_s \tag{8.22}$$

Assumed super heat:

$$T_s = 3 \tag{8.23}$$

8.2.3 Quality, specific heats, densities and latent loads

The evaporator exit quality is assumed constant and fixed for all cases:

$$x_1 = 0{,}9 \tag{8.24}$$

The specific heat of the ambient air as well as the cabin air is assumed to be constant and fixed for all cases:

$$c_{p_{amb}} \approx c_{p_{cab}} = 1 \tag{8.25}$$

The air is assumed to be an ideal gas, thus for the average ambient air density through the evaporator (placed outside the cabin):

$$\rho_{amb} = \frac{348}{(T_{ave} + 273)} \tag{8.26}$$

Were:

$$T_{ave} = \frac{T_{amb} + T_{HX2,out}}{2} \tag{8.27}$$

Latent heat of evaporation of the humid air in the cabin is not considered. Latent loads of condensation, frost and the cooling of ice formation on the evaporator are not accounted for.

The latent loads can have a significant impact in HP systems efficiency, depending on operating conditions. This goes for both cycles and is thus not accounted for in the comparative analysis done in this study. More over such kind of theoretical analysis should be combined with a real case in order to achieve god results.

8.3 Case

- The cabin temperature is set to 15 °C. The driver is assumed to be dressed warm, as usually is the real case under ambient condition requiring cabin heat.
- $\dot{m}_{vent} = 0{,}0027$: Consistent with equation (8.3)
- R134a is assumed to exit the condensers as saturated liquid (quality = 0)

- Compressor speed is set to 1000 rpm, complying with a variable displacement control for the different cycles and refrigerants operations. This compressor speed is chosen for practical reasons. This strategy is applied because R744 and R134a have very different specific volumes at the compressor inlet. It is therefore easier to compare the two media with the same speed for both compressors than with the same swept volume.

- The mass flow of air through the gas cooler is set to 0,035 kg/s [28].

- The ambient temperature is varied from -2 °C to 7 °C. These limits are restricted in this analysis due to the validity of the correlation [34] used for optimal pressure. When the ambient temperature is set to -2 °C the evaporator temperature will be -10 °C (assumed temperature difference in this study 8 °C) corresponding to the lower limit set by Sarkar et al. [34].
  When the ambient temperature is set to 7 °C the required air temperature out from the gas cooler is calculated to 26,7 °C thus giving a refrigerant temperature at the gas cooler outlet of 31,7 °C (assumed temperature difference in this study 5 °C) corresponding to the lower limit as explained in section 6.2.2.

The cycles are run 100 times each in EES. The equations are set according to chapter 7-8 for the different loads. All parameters are then calculated 100 times from an ambient temperature of -2 °C to 7 °C. The heating demand is calculated with equation 8.11 and 8.13. The compressor displacement volume is set to change to provide the necessary volume flow for the required heating capacity. The performance (COP_{heat}) and power consumption ($\dot{W}_{el,tot}$ and $\dot{W}_{el,comp}$) for each cycle using R744, is then compared against the standard SVCC using R134a as reference (100%). More over the relative performance of the improved R744 TVCC is compared against the standard R744 TVCC as reference (100%).

9. Results and Analysis

9.1 Results - COP_{heat}

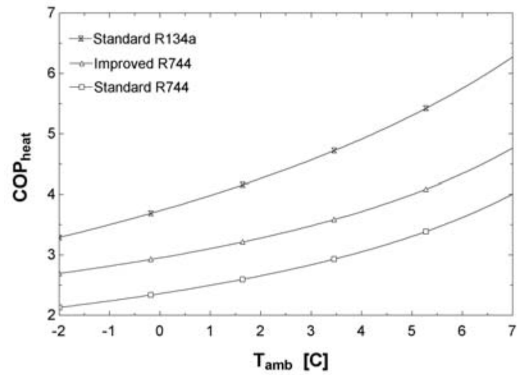


Figure 20. COP_{heat} vs. Ambient Temperature

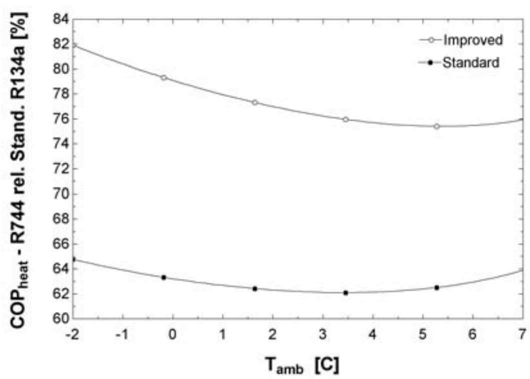


Figure 21. COP_{heat} - R744 vs. R134a

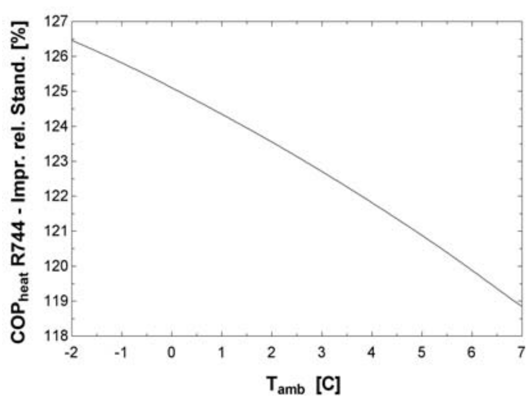


Figure 22. COP_{heat} - Improved R744 vs. Standard R744

9.1.1 Analysis - COP_{heat}

From the results in fig. 20, it is noticeable that the standard SVCC using R134a refrigerant outperforms R744 in terms of heating performance. This trend is continues throughout the tested temperature range.

At an ambient temperature of -2 °C the standard SVCC with R134a shows a COP_{heat} of around 3,3, the improved R744 TVCC around 2,7 and the standard R744 TVCC around 2,1.

At an ambient temperature of 7 °C the standard SVCC with R134a shows a COP_{heat} of around 6,3, the improved R744 TVCC around 4,8 and the standard R744 TVCC around 4,0.

From the results in fig. 21 it can be seen that the R744 cycles have a minimum in their relative merit compared to R134a. For the standard R744 TVCC this minimum seems to be at around 3,5 °C ambient temperature, and for the improved R744 TVCC at around 5,5 °C.

In **average**, see fig. 21, the standard **R134a** SVCC **outperforms** the standard **R744** TVCC with **37 %** and the improved R744 TVCC with **20 %** in terms of COP_{heat}.

$$\text{average}[\%] \approx 100 \pm \left(\min[\%] + \frac{\max[\%] - \min[\%]}{2} \right) \tag{9.1}$$

It can be seen in fig. 22 that the improved R744 TVCC cycle has its best relative merits at low ambient temperatures. In **average** the **improved R744** TVCC **outperforms** the **standard R744** TVCC with **23 %**.

9.2 Results – Electricity consumption

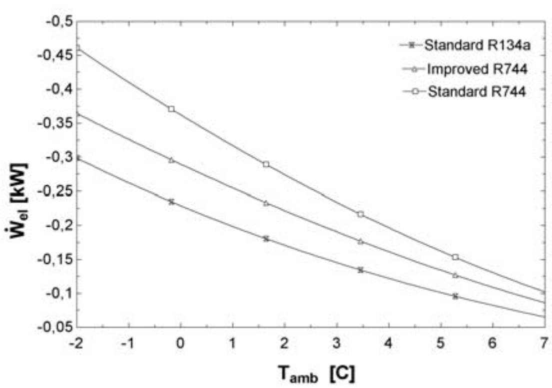


Figure 23. Electricity Consumption vs. Ambient Temperature

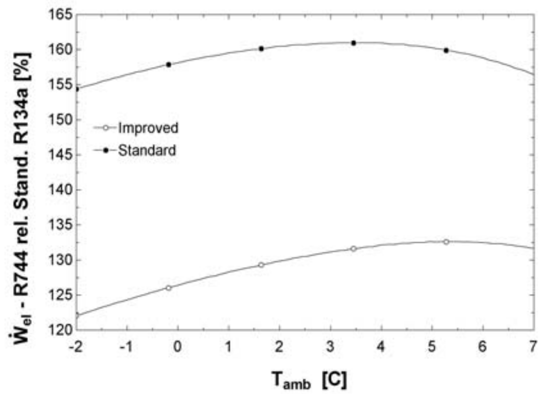


Figure 24. Electricity Consumption - R744 vs. R134a

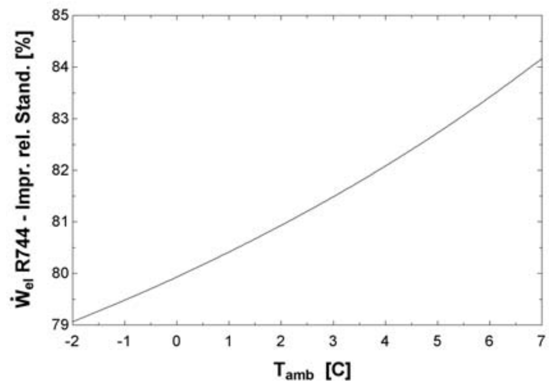


Figure 25. Electricity Consumption - Improved R744 vs. Standard R744

9.2.1 Analysis – Electricity consumption

From the results in fig. 23 it is noticeable that the standard SVCC using R134a refrigerant outperforms R744 with less electricity consumption. This trend is continues throughout the all conditions.

At an ambient temperature of -2 °C the standard SVCC with R134a shows an electricity consumption of around 0,30 kW, the improved R744 TVCC around 0,36 kW and the standard R744 TVCC around 0,46 kW.

At an ambient temperature of 7 °C the standard SVCC with R134a shows an electricity consumption of around 0,065 kW, the improved R744 TVCC around 0,086 kW and the standard R744 TVCC around 0,10 kW.

9.3 Results – Cabin related

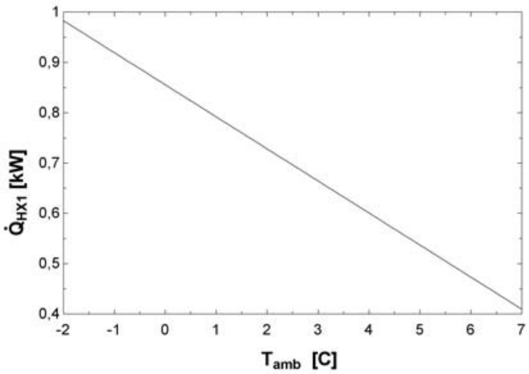


Figure 26. Heat Capacity vs. Ambient Temperature

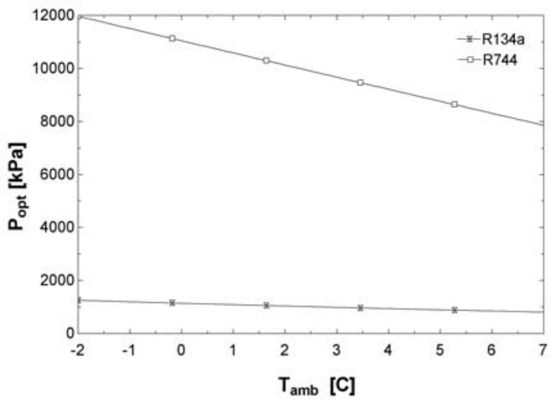


Figure 27. High-side Pressure vs. Ambient Temperature

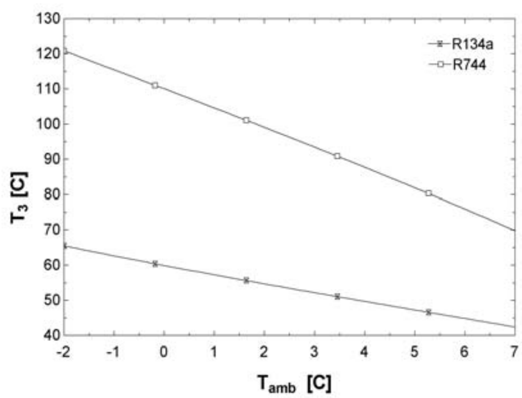


Figure 28. Gas Cooler/Condenser Inlet Temperature vs. Ambient Temperature

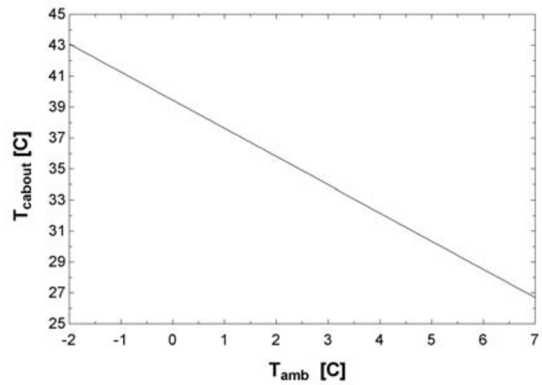


Figure 29. Gas Cooler/Condenser outlet air-flow Temperature vs. Ambient Temperature

9.3.1 Analysis – Heating capacity, pressure and temperatures

In fig. 26, it can be seen that the heat demand at an ambient temperature of -2 °C is approx. 1 kW and at 7 °C around 0,4 kW.

In fig. 27 the required refrigerant pressure of the R744 entering, the gas cooler inside the cabin is 12000 kPa (120 bars) and 7800 kPa (78 bars) at an ambient temperature of -2 °C and at 7 °C respectively. The corresponding refrigerant temperatures at the gas cooler inlet can be seen in fig. 28 and are approx.120 °C and 70 °C.

The required refrigerant pressure of R134a entering the condenser under the same conditions, is approx. 1250 kPa (12,5 bar) and 800 kPa (8 bar) at an ambient temperature of -2 °C respectively 7 °C. The corresponding refrigerant temperatures can be seen in fig. 28 and are approx. 65 °C and 42 °C.

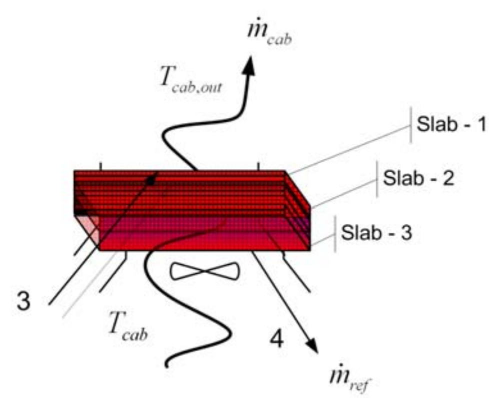


Figure 30. Gas Cooler

The air temperature leaving the gas cooler is approx. 43 °C at ambient temperature of -2 °C, and 27 °C at ambient temperature of 7 °C in order to keep the cabin at 15 °C. This gives an average air input temperature of 35 °C.

9.4 Results – Compressor

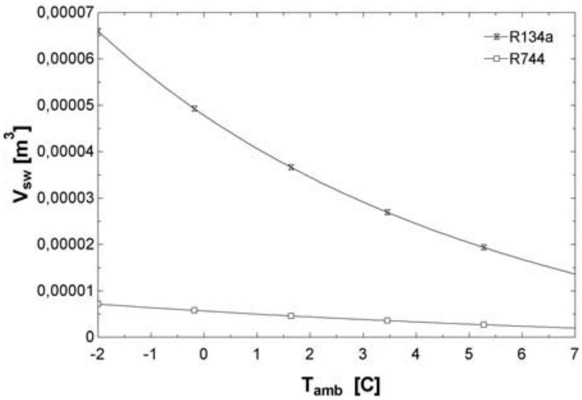


Figure 31. Swept Volume vs. Ambient Temperature (at 1000 rpm)

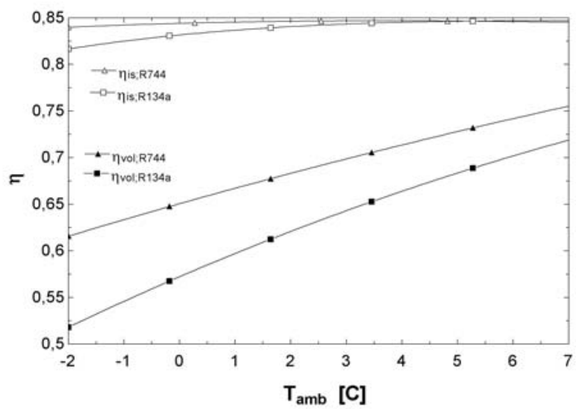


Figure 32. Compressor Efficiencies vs. Ambient Temperature (at 1000 rpm)

9.4.1 Analysis – Swept volume and efficiencies

As seen in fig. 31 R744 requires a much smaller swept volume in comparison to R134a. This is explained by the much lower specific volume of R744 at the compressor inlet.

At an ambient temperature of -2 °C and 1000 rpm the standard R134a compressor requires a swept volume of approx. 0,000066 m^3 (66 cm^3) and R744 needs only 0,000007 m^3 (7 cm^3) for the same process.

At an ambient temperature of 7 °C and 1000 rpm the standard R134a compressor requires a swept volume of approx. 0,000014 m^3 (14 cm^3) and R744 needs 0,000002 m^3 (2 cm^3) for the same process.

As seen in fig. 32 R744 shows overall higher volumetric and isentropic efficiencies. This is by the equations used due to the differences in required pressure ratio for each refrigerant. R134a gives overall a higher pressure ratio compared to R744.
The isentropic efficiency reaches its maximum at a pressure ration around 3. This corresponds to an ambient temperature of approx. 3,5 °C for R744 and 6,5 °C for R134a.

At an ambient temperature of -2 °C and 1000 rpm the standard R134a compressor shows a volumetric efficiency of around 0,52 and R744 0,62.

At an ambient temperature of 7 °C and 1000 rpm the standard R134a compressor shows a volumetric efficiency of around 0,72 and R744 0,75.

9.5 A De-frosting case for the improved R744 TVCC HP/AC

9.5.1 Conditions

- $R_H \approx 1{,}27$ (Corresponding with [28])
- $T_{ev} = -5$ (Assumed evaporation temperature)
- $T_{amb} = 3$ (Assumed ambient temperature)
- $V_{cab} = 4{,}5$ (Consistent with [15])
- $\dot{m}_{vent} = 0$ (Turned of during de-frosting, e.g. a switch)
- $\dot{m}_{air} = 0$ (Turned of during de-frosting, e.g. a Venetian blind)
- $\dot{Q}_{HX2} \approx -0{,}50$ (Calculated in EES – ideal case)
- $T_{HX2,out} = -2{,}8$ (Calculated in EES – ideal case)
- $\rho_{amb} \approx 1{,}27$ (Calculated in EES – ideal case)
- The mass flow of air through the evaporator (during heat pump mode) is calculated based on the effective dimension presented in table 3 and the recommended minimum lower air velocity through the outside heat exchanger of 1,5 m/s discussed in [28].

EXAMPLE - EVAPORATOR [40]		
Effective dimensions	Evaporator	Unit
Core effective	250 X 181 X 89	mm
A_{face}	0,04525	m^2
A_{air}	3,277	m^2

Table 2. Example - Evaporator Characteristics

$$\dot{m}_{air} = \rho_{amb} \cdot vel_{amb} \cdot A_{face} = \\ = 1{,}27 \cdot 1{,}5 \cdot 0{,}04525 \approx 0{,}09 \tag{9.2}$$

9.5.2 Approach

Heat balance for the evaporator:

$$\dot{Q}_{HX2} + \dot{Q}_{air} = 0 \tag{9.3}$$

$$\Rightarrow \dot{Q}_{air} = 0{,}50 \tag{9.4}$$

Emitted heat from the ambient air to the refrigerant in the evaporator:

$$\dot{Q}_{air} = \dot{m}_{air} \cdot c_{p_{amb}} \cdot (T_{amb} - T_{HX2,out}) \tag{9.5}$$

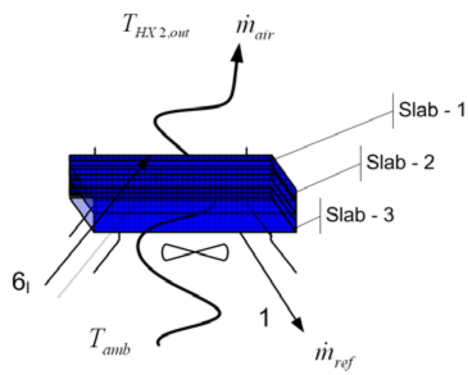


Figure 33. Evaporator

Assumed constant specific heat of ambient air through the evaporator:

$$c_{p_{amb}} \approx 1 \tag{9.6}$$

$$\Rightarrow T_{HX2,out} \approx -2{,}8 \tag{9.7}$$

By calculating the volume flow **in m^3/h** it is possible to estimate the amount of latent load due to frost formation using the program Coolpack [17]:

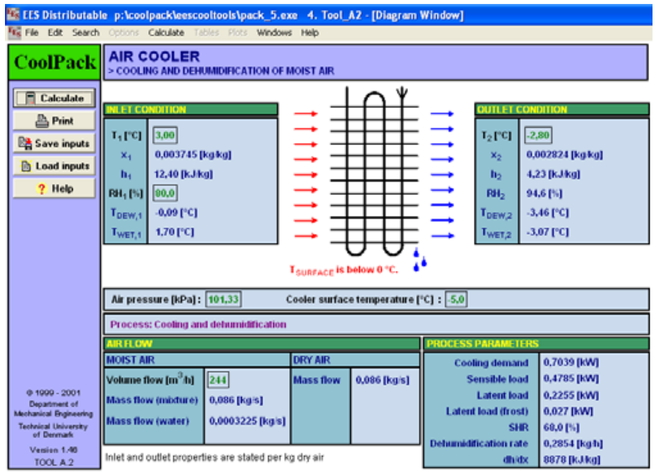


Figure 34. CoolPack Work sheet

"This model can be used to calculate the cooling and dehumidification rates for humid air in air-coolers.
The calculations are based on the assumption that the cooling and dehumidification process can be described by dh/dx = constant (h = enthalpy, x = humidity ratio). In a h,x-diagram this corresponds to the process being a drawn as a straight line from the inlet condition towards the surface condition of the air-cooler (saturated condition).
If the surface temperature is below 0 C and dehumidification occur, frost will form on the surface. The freezing of the water condensate and subsequent cooling of the frost formed creates an extra latent load on the evaporator. The valued displayed for latent load contains this part, but for information the part of the latent load caused by freezing and cooling of the frost formed is displayed as "Latent load (frost)"."…….[17]

$$\dot{V}_{air} = vel_{air} \cdot A_{face} \cdot 60 \cdot 60 \approx 244 \qquad (9.8)$$

$$\Rightarrow \dot{Q}_{latent\text{-}frost} = 0{,}027 \qquad (9.9)$$

If assuming a 10 minute operation under this condition:

$$Q_{latent\text{-}frost} = \int_0^{\tau} \dot{Q}_{latent\text{-}frost} \cdot \delta\tau = 0{,}027 \cdot 60 \cdot 10 \approx 16{,}2 \qquad (9.10)$$

This gives an approx. amount 16,2 kJ consumed by the frost formation and subsequent cooling of the frost. This frost will deteriorate the heat transfer in the evaporator. By reversing the HP process, shutting off the ventilation and neglecting any heat transfer losses from the cabin to the ambient thus assuming that the de-frosting process takes place under a short period of time, then the following can be assumed for the temperature drop in the cabin.

$$Q_{\text{latent-frost}} = Q_{\text{de-frost}} \tag{9.11}$$

$$\begin{aligned} &Q_{\text{de-frost}} = \\ &= V_{\text{cab}} \cdot \rho_{\text{air}\,cab} \cdot c_{\text{p}\,\text{cab}} \cdot (T_{\text{cab}} - T_{\text{cab-after-defrost-improved}}) \end{aligned} \tag{9.12}$$

$$\rho_{\text{air}\,cab} \approx 1{,}2 \tag{9.13}$$

$$\Rightarrow T_{\text{cab-after-de-frost-improved}} \approx 12 \tag{9.14}$$

9.5.3 Comments

The latent heat of frost formation/melting can be assumed to be 334 kJ/kg, Ice [41]. This means that 16,2 kJ corresponds to melting approx. 49 grams of ice corresponds to a temperature drop of approx. 3 °C in the cabin. This indicates that it is recommendable to de-frost continuously with short time intervals in order to avoid large temperature drops in the cabin. If including the actual energy required to heat a real heat exchanger and other equipment in the HP-system this temperature drop will become even higher. Most probably it would be wise to de-frost within shorter time intervals than 10 minutes in the real case. This must however be practically experimented/ tested in order to give any god recommendations. An alternative concept design, which among other things would give a reduced temperature drop during de-frosting, is described in chapter 11.

10. Conclusion and remarks

10.1 Comparative case study

As can be seen from the temperature profiles, R744 requires a very high temperature and pressure due to the characteristics of the R744 properties. Under the given conditions, the refrigerant is required to enter the cabin gas cooler with a maximum temperature of around 120 °C and corresponding approx. pressure of 120 bars. This indicates that the gas cooler should be placed well protected from the passenger of the heated vehicle. These extreme conditions can be crucial in future discussion concerning the suitability of R744 gas coolers operating inside the cabin.

A protection system is needed to keep the passengers fully protected from the system. In terms of safety, a secondary refrigerant loop could be used, thereby keeping the R744 system outside the cabin. This would however mean a further degradation of the performance.

Approx. - Values	COP_{heat}		Electricity Consumption (kW)		Heating Capacity (kW)	
	max	min	max	min	max	min
R134a cycle	6,2	3,3	0,30	0,065	1,0	0,40
Improved R744 cycle	4,7	2,8	0,36	0,086	1,0	0,40
Standard R744 cycle	4,0	2,2	0,46	0,10	1,0	0,40

Table 3. Results of Case Study

Disregarding the high pressures and temperatures as a potential safety issue, the analysis indicates that R744 could be well suited for heating over ambient temperatures around 5 °C with increasing potential as the temperature goes up.

Under the given tested conditions, the result of this analysis shows that R134a has the overall best qualities both in terms of cooling (not discussed in this study) and heating performance as well as in lower electricity consumption in comparison to R744.
The average electricity consumption for the standard R134a SVCC was found to be approx. 20 % lower in comparison to the improved R744 TVCC. More over, the results indicate that R134a would require an average of 37 % less power compared to R744 when operated in the same cycle under same conditions. At the same time, the results of the volumetric behavior of R744 indicate that there is a great potential in making the compressor small in comparison to R134a due to the lower specific volume. If the potential in size reduction of the R744 TVCC should result in a lighter system, the negative effects of the R744 low cycle performance could be alleviated. However, this does not seem very likely to happen in the improved R744 TVCC, thus requiring additional work-recovering components in its design.

10.2 Remarks

The efficiency is of great importance, as the electricity in an EV is needed elsewhere in the car. The improved R744 TVCC showed an average improvement of approx. 23 % in COP_{heat} compared to the standard R744 TVCC. The feasibility of the improved R744 TVCC compared to the standard R744 TVCC was found to be the highest at lower ambient temperature below -2 °C. This suggests that a possible investment in a work recovering expansion device would be most feasible if the cycle operates at lower ambient temperatures.

J. Steven Brown et al. [42] carried out a comparative analysis between R134a and R744 in an automotive air conditioning system. At a compressor speed, equal to the on tested in this study (1000 rpm), R744 showed a 21 - 34 % lower COP compared to R134a. More over it was proven that R744 has a closer approach temperature (3-5,8 °C) in the gas cooler compared to R134a (8,5-11,2 °C) in an equal sized condenser. [42] States that the closer approach temperature did however not over-come the "thermodynamic penalty" associated with the high entropy generation found in the gas cooler.

Tomoichiro Tamura et al. [43] report the following:

> "…..We constructed a CO2 cooling and heating air conditioning prototype system (for medium-sized cars) with performance equal to or exceeding that of a system using HFC134a refrigerant……. The relative heating/dehumidifying COP=1.31 (in comparison with current HFC134a)."…… [43]

More over it is reported in [43] that the so called "COP ratio to HFC134a system" resulted to be 1.31. This could easily be interpreted as a 31 % COP improvement in favor of R744.

The results achieved in this study, in respect to the comparative performance of R134a and R744 does not correspond with the ones achieved in [43] and [42]. It seems thus very reasonable to address the reader to be extra careful when revising existing R744 COP literature, as the results seem to disagree. More over, it is difficult to compare results from one study to another as they strongly depend on the theoretical models used, assumptions made, systems compared, conditions tested and so on. This can consequently, in part explain the different results achieved in this study, [42] and [43].

It has to be pointed out that, as basis for the R744 results achieved in this study lays Sarkar et al. [34] correlation for optimal COP performance of the R744 TVCC and the equations of state (thermodynamic properties) provided in EES [16]. More over the equations of state used in this analysis for carbon dioxide (R744) are based on the work of R. Span et al. [44], and provides high accuracy thermodynamic properties for carbon dioxide assuming real fluid behavior. The corresponding results and thermodynamic properties for R134a are based on the fundamental equation of state developed by Tillner-Roth et al. [45], also given in EES [16].

Due to the nature of the simulation, presented here the pressure drops were not accounted for. As mentioned previously the main advantage of the flash-gas concept is the reduction in refrigerant pressure drop [31]. Due to this, the relative performance as well as the approximation to its Carnot equivalent, the improved R744 TVCC compared to the standard R744 TVCC and SVCC R134a can be expected to be better in the real case.

11. Improved 12/5-EV-HP/AC

11.1 Further possible Improvements

By combining the improved automobile R744 TVCC HP/AC cycle with a proposed 12-port - 5 direction valve and an integrated heat exchanger in the battery core, the cycle can be improved even more in terms of efficiency. See fig. 37-43 in appendix.

11.1.1 Improved de-frosting

By partly/entirely evaporating the refrigerant in the integrated EV-battery micro-channel tube bundle evaporator a reduced temperature drop in the cabin can be expected during de-frosting, see fig. 42. This could be a very important detail in its implementation, thus as shown in section 9.5 the temperature drop was found to be large. This can simply be explained in the following way:

$$Q_{\text{de-frost}} = V_{\text{cab}} \cdot \rho_{\text{air}} \cdot c_{p_{\text{cab}}} \cdot (T_{\text{cab}} - T_{\text{cab-after-de-frost-12/5}}) + Q_{\text{battery}} \quad (11.1)$$

$$T_{\text{cab-after-de-frost-12/5}} = T_{\text{cab}} - \left(\frac{(Q_{\text{de-frost}} - Q_{\text{battery}}) \downarrow}{V_{\text{cab}} \cdot \rho_{\text{air}} \cdot c_{p_{\text{cab}}}} \right) \downarrow \quad (11.2)$$

$$\Rightarrow T_{\text{cab-after-de-frost-12/5}} > T_{\text{cab-after-de-frost-improved}} \quad (11.3)$$

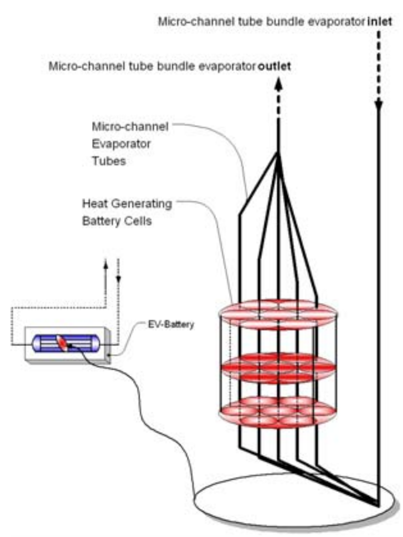


Figure 35. Integrated micro-channel tube bundle concept

Were, Q_{battery} represents the available waste heat in the battery core. There is of course a recommended optimal battery temperature limit for each battery, which should not be trespassed, thus implying that this additional heat capacity is limited.

More over it can be well recommended to isolate the battery pack properly against any possible heat losses to the ambient. This would consequently require a way to cool the battery effectively. By adding another valve on the improved 12/5-EV-HP/AC it could be possible to cool the battery by pumping the heat to the outside gas cooler/condenser. The battery may not heat the cabin during e.g. hot summer days.

In reality, the process of de-frosting an evaporator is very complex. Depending on the case and design, the process can be done quickly or slowly, step-wise or entirely etc. The method, which would be applied in the real case, is very dependent on the components of the system as well as the dimension and thermodynamic properties of the cabin for acceptable thermal comfort. The temperature drop should also be compared to the limited and acceptable time of reheating the cabin to the desired stable cabin temperature. In the case discussed previously the temperature drops down to 12 °C, which can be considered uncomfortable for the driver. If the time of reheating the cabin is short, meaning that the compressor in the system and the other components in the system are capable of operating in the desired way, then it could be assumed that this method could be acceptable. How ever, most probably this is not the real case for a cabin.

As stated by Man-Hoe Kim et al. [27] very little is known about frosting and condensate drainage from ultra-compact micro-channel heat exchangers. In [27] it is further on pointed out that the commonly used method of de-frosting by reversing the HP used in residential heat pumps is not a feasible option for automobiles due to the small volume of air in the passenger compartment. This gives an indication of the importance to look further into detail regarding this complex matter.

The necessary de-frosting demand is strongly dependent on the evaporator design, ambient conditions, operational conditions of the HP, available electricity power from the EV-battery, available compressor power etc. To make an appropriate estimation off the effects of frost formation on a R744 micro-channel evaporator during HP operation in an EV goes beyond the limitations of this study.

11.1.2 Improved COP

The improved performance can be explained in the following way for the expansion-machine cycle:

$$\dot{Q}_{HX1} + \dot{Q}_{HX2} + \dot{Q}_{IHX1} + \dot{Q}_{IHX2} + \dot{Q}_{battery} + \dot{W}_{comp} + \dot{W}_{exp} = 0 \qquad (11.4)$$

As defined throughout the work in this thesis, the energy supplied to the system is defined as negative, and assuming the same ideal case as previous. Then the following can be said:

$$COP_{cool\text{-}12/5} = \left(\frac{(\dot{Q}_{HX2} + \dot{Q}_{battery}) \uparrow}{\left| \dot{W}_{el,comp} + \dot{W}_{el,exp} \right|} \right) \uparrow \qquad (11.5)$$

$$COP_{heat} = COP_{cool} + 1 \tag{5.1}$$

$$\Rightarrow COP_{heat\text{-}12/5} > COP_{heat\text{-}improved} \tag{11.6}$$

And for the case of the expansion valve cycle (fig. 43) the same argument is applied, although without the additional work recovered from the expansion process thus:

$$COP_{heat\text{-}12/5} > COP_{heat\text{-}improved} > COP_{heat\text{-}standard} \tag{11.7}$$

In fig. 41 the possibility of operating the cycle in HP-mode, only by using the heat generated in the EV-battery pack is shown.

11.2 By-passing the IHX and improving COP

The improved 12/5-EV-HP/AC cycle should be designed in a way that the IHX can be bypassed, meaning that the refrigerant evaporates/superheats only in the micro-channel tube bundle evaporator and exterior heat exchanger, and expanding directly after the gas cooler (condenser in SVCC). This function would be beneficial **when the battery generates sufficient waste** heat for the cycle to work properly.

The benefit is explained by the **reduced pressure drop** that is otherwise encountered in the IHX1 (hot side). This should be beneficial for both the isenthalpic (expansion valve) and isentropic (expansion-machine-work-recovery) cycle shown in fig 37-43.
However this approach can be assumed to be more beneficial to the work recovery expansion cycle, this is explained by the double effect the pressure drop reduction has on the cycle.

The pressure drop reduction will not only **reduce the compressor work** (dW_{comp}) but also **increase the expansion-machine power-output** (dW_{exp}), see fig. 36, due to the increased pressure ratio between the expander inlet and outlet. As the flash-tank separates the vapor and liquid after the expansion this concept should not affect the cycle cooling capacity. The above mentioned does of course also apply to the SVCC (when including the flash-gas bypass concept to it).

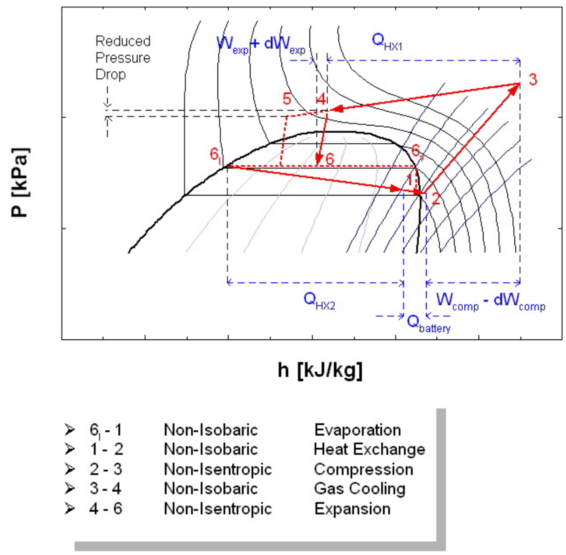


Figure 36. P-h diagram. By-passed IHX in the improved 12/5-EV-HP/AC

For the TVCC this can be explained by the figure above. The COP_{heat} improvement can thus be explained in the following way:

$$COP_{heat\text{-}12/5\text{-}by\text{-}passed\text{-}IHX} = \tag{11.8}$$
$$= \left(\frac{\dot{Q}_{HX1}}{\left| \left(\dot{W}_{comp} - \delta\dot{W}_{comp}\right) - \left(\dot{W}_{exp} + \delta\dot{W}_{exp}\right) \right| \downarrow} \right) \uparrow$$

$$\Rightarrow COP_{heat\text{-}12/5\text{-}by\text{-}passed\text{-}IHX} > COP_{heat\text{-}12/5} \tag{11.9}$$

Conclusively this means that for optimal function of the 12/5-EV-HP/AC, see fig. 37-43, the following should be added to the cycle in its design for optimal function:

- An additional way to entirely by-pass the IHX, so that the evaporation only takes place in the EV-battery and/-or exterior cabin Heat Exchanger.

- An additional way to entirely by-pass the interior cabin Heat Exchanger, so that the gas cooling/condensation can only take place in the exterior cabin Heat Exchanger.

11.3 Summarized

A well insulated battery pack would require further development of the improved 12/5-EV-HP/AC; this can simply be done by adding an extra circuit and valve thus bypassing the interior heat exchanger. It should be done due to the temperature rise encountered in summer time if the battery is well insulated. This would consequently require a way to cool the battery effectively and not heating the cabin at the same time. By adding another valve on the improved 12/5-EV-HP/AC it can thus be possible to cool (or reversibly heat) the battery by pumping the heat to the outside heat exchanger (gas cooler/condenser), thus it is not recommended to heat the cabin during e.g. hot summer days.

If R744 should not become the future refrigerant in EV-HP applications, and instead a refrigerant more similar in its thermodynamical performance to R134a, then it could be suggested to work more on the cycle proposed in fig. 43. This should become a more economical choice, thus no need of an additional expander-generator module.

Improvements presented in this study over the present "standard automobile R744 TVCC AC" can be summarized component-wise:

- 8 port 3 way valve
- 12 port 5 way valve
- Flash tank/accumulator
- Flash gas bypass valve
- Hermetic (generator included) variable displacement expander. Entirely, partly or not integrated with the compressor.
- Integrated micro-channel tube bundle evaporator in the EV-battery.
- An additional way to by-pass the IHX- as discussed previously (not included in fig. 37-43)
- An additional way to by-pass the interior cabin heat exchanger- as discussed previously (not included in fig. 37-43)

12. Future work and final words

12.1 Future work

It is not known if components such as e.g. the expander-generator-hermetic-compressor exist. It could be recommendable to develop components like that, to make R744 a more competitive alternative. If such a component should be found to be to expensive within what is found to be reasonable an alternative proposed cycle, see fig. 43, is recommended for further development.

The flash-gas bypass concept for R744 has been tested [31] for a stationary R744 unit, but little information was found about the necessary components, thus suggesting further work in its development and behavior in a mobile R744 TVCC.

Keeping the passengers protected from the gas cooler is important, however a secondary loop inside the cabin, containing in example brine, would cause a further drop in performance of the already relatively low performing R744 cycle.

An emphasis on the control system for optimal operation of the TVCC is necessary. The control system should have the means of actively controlling the expander flow, flash gas by-pass valve, compressor displacement and compressor velocity. It should also control the secondary systems including the “Venetian blinds” and necessary fans.

Due to the relocation of the refrigerant buffer (accumulator) in the proposed improved system it is necessary to make further studies of the systems vapor feeding to the compressor. Especially mass regulation for optimal COP must be studied in depth.

More over, it is not known if a reversible EV-battery integrated micro-channel heat exchanger capable of working as a gas cooler/condenser and evaporator exists. It could be recommendable to develop components like that, to make the heat pump a more competitive alternative for the heating of an EV.

12.2 Final words

Due to the facing out of the superior R134a refrigerant as a possible working fluid in a HP system for an EV it is suggested to continue the work on improved R744 cycles (for instance the ones proposed here). It is presently (2008) not sure, if future safety regulations will continue to permit the given operational conditions of the improved automobile R744 TVCC HP/AC. Complying with existing regulations, the presented improved cycle can be used in present AC vehicle operations with an improved functionality.

Possible improvements on the R744 TVCC have been suggested by several authors by dividing the expansion and/or the compressor process in multiple steps. This approach has not been further discussed due to the increased complexity that such a system would represent, not only technically but possibly economically as well.

If the necessary operating conditions inside the cabin compartment of the TVCC are found to be unacceptable using R744, then it is suggested to use a secondary working medium with good thermodynamical properties in a secondary loop providing heat to the cabin. This option also increases the possibility of using other refrigerants like e.g. propane, with good thermo dynamical properties. That could otherwise be unacceptable if placed in direct contact with the cabin. If, on the other hand, such an auxiliary system was used – why not use that system for both heating and cooling.

An integrated micro-channel system, see fig 35, placed in between the stacked EV-battery cells/shells working as an evaporator, thus absorbing the relatively large amount of heat generated in the re-charging process (as well as during normal operation) and pumping it into the cabin, see fig. 39-41, can consequently be expected to reduce the re-charging time of the EV-battery while heating the cabin. This would be the case when

sufficient waste heat is generated in the battery core. If this should not be the case, the two evaporators could work together as shown in fig. 39-40.
This method could of course be used reversibly, thus heating the EV-battery when necessary for optimal function of the EV recharging and operational process.

The Li-Ion batteries, e.g. in the Tesla Roadster mentioned earlier, consist of cylindrical Li-Ion cells packed inside the battery package. The integration of micro-channel tubes in the battery core cavities is thus not space consuming and could thus be recommended to be studied into more detail.
However the additional weight of the micro-channel tube bundle has to be studied in more detail, thus this could be decisive.

By regulating the battery core temperature, it can be assumed that the large battery pack in the EV will last longer. It is however not known if such evaporator exists. It could be interesting to develop components like that, to make the HP cycle a more competitive alternative in the EV (even for SVCC). More over it can be recommended to develop a way to use the proposed micro-channel evaporator as a gas cooler/condenser as well. This could be beneficial if using the outside heat exchanger as gas cooler/condenser, thus during hot days the battery needs to be cooled as well as the cabin.

It is presently (2008) not sure, if future safety regulations will continue to permit the given operational conditions of the improved 12/5-EV-HP/AC using R744 as refrigerant. Complying with existing regulations, the presented 12/5-EV-HP/AC can be used in present EV-HP operations with an improved functionality.

If the temperature drop in the cabin is found to be unacceptable during de-frosting a heat buffer is recommended. This can for example consist of a well insulated accumulator containing glycol/brine or a molten salt or paraffin, which is intermediately heated by the gas cooler. This heat can then be used during de-frosting.

The variation of the R744 properties in the gas cooling process indicates that the heat exchanger, if intended to be used both as a gas cooler and as an evaporator, has to be carefully designed. Presently a three slab gas cooler is suggested for the gas cooling process, like the one tested in [46]. A combination of a micro-channel evaporator and gas cooler is probably the best choice for the presented system.

It is problematic to provide sufficient amounts of lithium for future EV-mass-production. This influences the market potential for the HP’s discussed in this study.

This project has not studied the very important initial heating up time of an initially cold cabin. Hopefully the interior of the vehicle has some additional heating facility, beside the air, that can mitigate this demand – like seat heating. Not to over- dimension the HP-system, it is recommended to look into more such additional electric heat sources for the peak loads.

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14. Appendix

14.1 Improved 12/5-EV-HP/AC

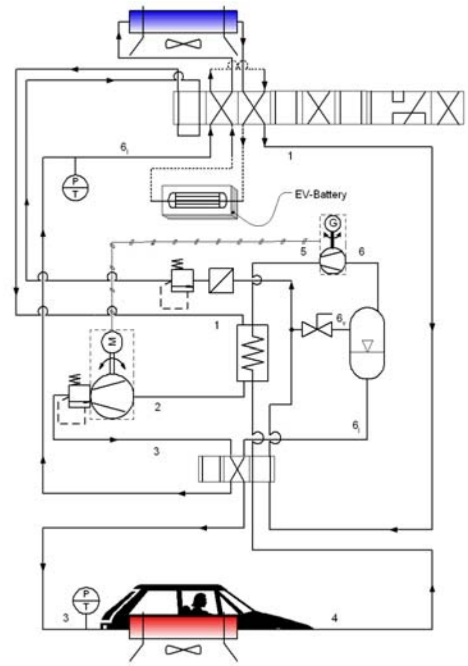

Figure 37. Improved 12/5-EV-HP/AC. HP-mode I

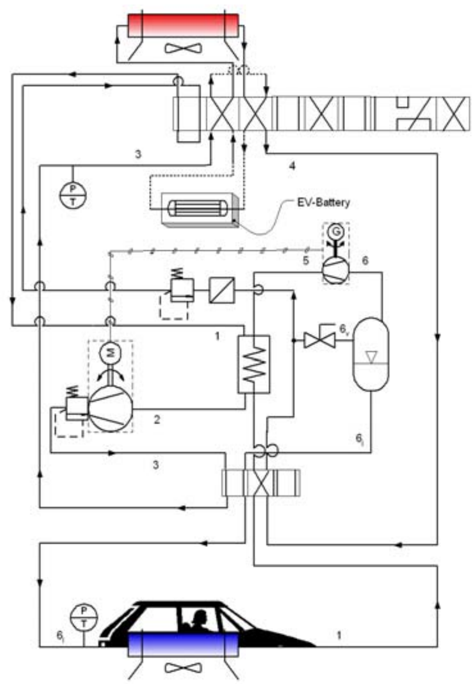

Figure 38. Improved 12/5-EV-HP/AC. AC-mode

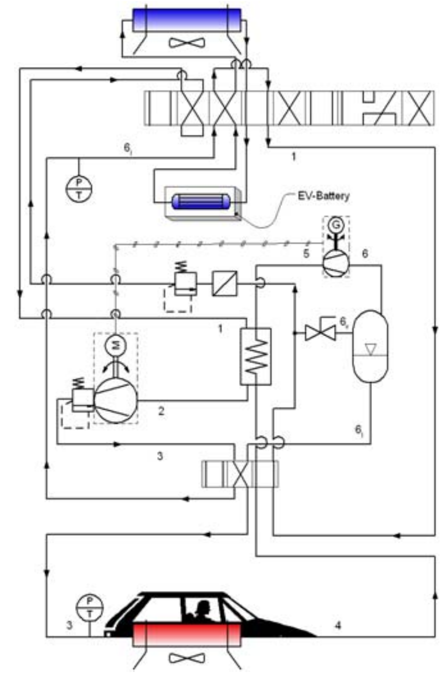

Figure 39. Improved 12/5-EV-HP/AC. Efficient HP-mode I

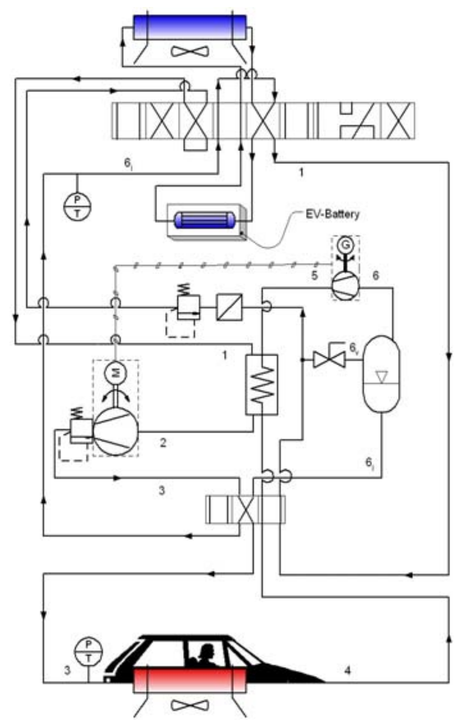

Figure 40. Improved 12/5-EV-HP/AC. Efficient HP-mode II

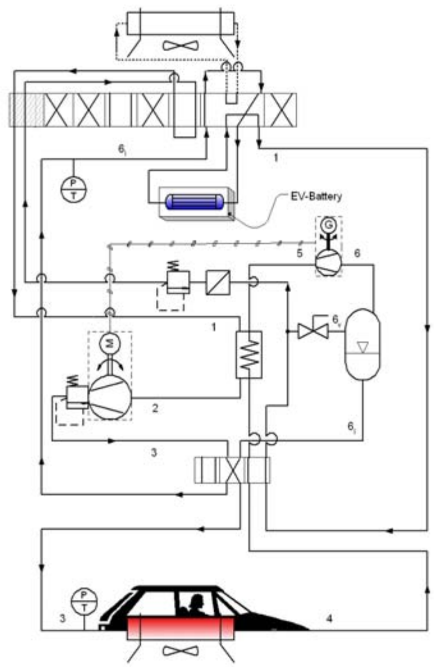

Figure 41. Improved 12/5-EV-HP/AC. HP-mode II

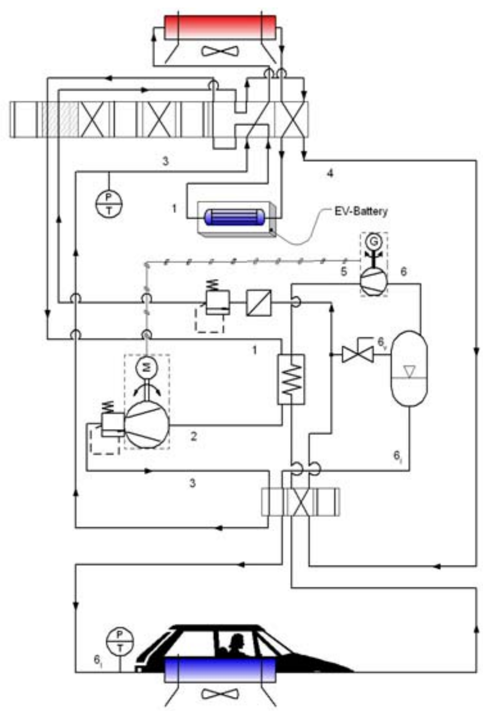

Figure 42. Improved 12/5-EV-HP/AC. Efficient de-frosting/ battery cooling

14.2 Improved - Expansion valve - 12/5-EV-HP/AC

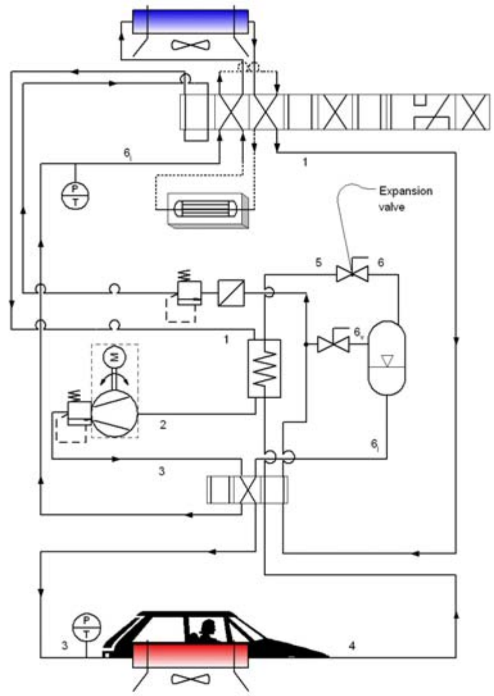

Figure 43. Improved - Expansion valve - 12/5-EV-HP/AC