Direct cooling for fast charging of electric vehicles

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***Abstract*—Fast charging electric vehicles will pose new challenges for battery thermal management systems. The waste heat produced by the battery cell exceeds the maximum cooling capacities of state of the art cooling systems. This paper introduces an alternative approach for battery cooling and heating. Decoupling the battery thermal management from the HVAC (Heating, Ventilation and Air Conditioning) will have positive effects on the HVAC complexity and on the performance of the thermal management. Certainly the fast charging of electric scooters and motorcycles can also benefit from this approach. A qualitative analysis of the advantages and disadvantages is presented in the following paper. In addition a detailed comparison of 7 refrigerants is conducted especially for vehicles which currently feature no HVAC. As a result Propane was found to be an efficient, environmental-friendly and inexpensive refrigerant for the battery thermal management. At last the issue of dissipating large heat flows in a standing vehicle without larger condensers or high fan noise levels is solved by increasing the condenser temperature. This inhibits lower energy efficiency but will have no effect on the driving range as the energy is provided by the charging station.**

Keywords—battery thermal management, fast charging, refrigerant, two-wheeler

I. Introduction

Charging times of 15 minutes are aspired by car manufacturers [1] for future vehicles to shorten the waiting time for customers. Especially in electric two-wheelers fast charging is key to increase the attractiveness as range and thus battery capacities are limited due to volume and mass constraints. Both vehicle classes will be confronted with large waste heat losses during fast charging. The heat generation depending on the internal resistance of the cells and resistances of the electric contacts is manifested as Joule heat [2]. Mahle Behr [3] estimates required cooling capacities of 12 kW for battery cooling during a 15 minute charge of an automobile with 100 kWh battery capacity. This would outperform the available cooling capacity of current HVAC (Heating, Ventilation and Air Conditioning) systems, which can supply up to 8 kW during cabin cool-down [3].

Keeping in mind that steady-state cabin cooling will require an additional power of 3 kW [3],[4] and that battery cell aging may double the battery heat losses, it is not possible to use current automotive battery thermal management systems for fast charging with more than 2C. Cylindrical cells used in electric scooters and motorcycles show higher internal resistances than automotive cells resulting in even larger relative waste heat.

This paper presents an alternative approach and new solutions for battery thermal management to enable fast charging of electric vehicles.

II. State of the Art

Current battery thermal management systems are coupled with the HVAC of the cabin. This allows the cooling of the battery even in extreme environmental conditions. But as the temperature level and thus the pressure are adjusted to the HVAC these systems are not utilizing the full potential of the refrigerant circuit. The two state of the art types of battery thermal management systems are presented in this chapter.

A. Chiller system

The Mahle Behr chiller system [5] is used in plug-in hybrids and electric vehicles. The battery is cooled and heated with a water/glycol circuit. To cool the circuit it is coupled with the HVAC via a plate heat exchanger with an integrated expansion valve (called chiller) and a radiator integrated in the circuit. The heating of the circuit is usually applied by PTC-heaters. At suitable ambient temperatures the battery can dissipate the heat via the radiator. For ambient temperatures above the temperature limit of the battery, its heat is dissipated via the chiller. The cooling circuit allows low fluid pressures and easy control of the battery temperatures. Disadvantages are the numerous heat transmissions (battery cell to cooling circuit, cooling circuit to HVAC, HVAC to ambient) and the resulting temperature levels as well as a specific cell temperature inhomogeneity due to the temperature increase of the cooling fluid.

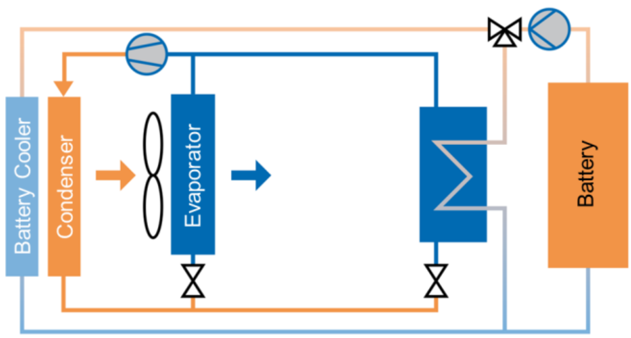

Fig. 1. Mahle-Behr Chiller system

B. *Direct evaporation system*

The direct evaporation system [6] depicted in Fig. 2. cools the battery with a refrigerant evaporator which is often integrated in a cooling plate. This cooling system was applied in the battery of the BMW i3 [7]. The phase change in the cooling plate offers high heat transfer coefficients and homogeneous cell temperatures as the temperature is constant during phase change. However the cooling plate and tubing must withstand the higher system pressure (current refrigerants: up to 20 bar) and water condensation can occur on the cooling plate at high humidity conditions.

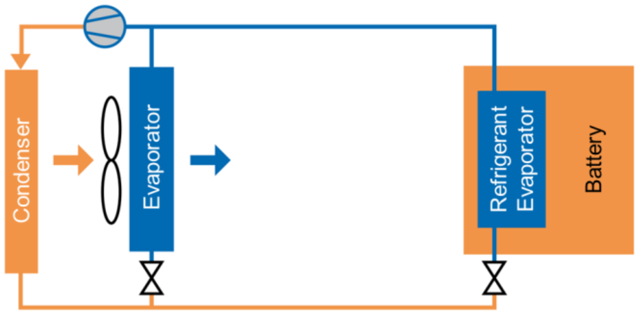

Fig. 2. Direct evaporation system

C. *Impact of fast charging on the state of the art*

Increasing the cooling capacities of HVAC-coupled systems to meet the requirements for fast charging will increase the size of the HVAC refrigerant circuit. This would result in the following changes:

a) Large or fast rotating compressors to increase refrigerant mass flow.
b) Large condensers or large fans which either results in an increase in drag coefficient and installation space or in high noise dissipation.
c) High refrigerant charge in the HVAC-circuit due to enlarged components and thus higher cost.
d) For chiller systems: High volume flows of water/glycol and thus larger pumps to reach homogenous cell temperature distribution.

III. Alternative Approach

The approach the authors propose is the decoupling of the HVAC and the battery thermal management into two separate refrigerant circuits. A decoupled system with integrated heat pump functionality is depicted in Fig. 3. The decoupling offers several advantages, but also disadvantages which are listed in this chapter. The order is not representing the importance as it depends heavily on the goals of the

A. *Simplification of HVAC operation*

The HVAC can be switched to heat pump mode regardless of the battery cooling requirements. Operation modes such as heat cabin, cool battery are easily accomplished. HVAC-coupled systems would require additional valves and interconnections to accomplish that. The refrigerant circuit of the battery can be switched to a heat pump with an additional 4/2-way valve and a bidirectional electric expansion valve. With these two components an additional PTC heater for the battery is not required anymore.

B. *Integration of battery thermal management in drivetrain cooling circuits*

During charging the drivetrain cooling circuit is not required for the electric motor or power electronics cooling. The battery and onboard charger cooling systems can utilize it to avoid additional radiators. By integrating the battery thermal management, the drivetrain cooling radiator only needs to be enlarged slightly as battery waste heat is less in driving operation compared to fast charging. Coupling the circuits as shown in Fig. 3. also offers the possibility to use drivetrain waste heat and thermal mass for battery heating.

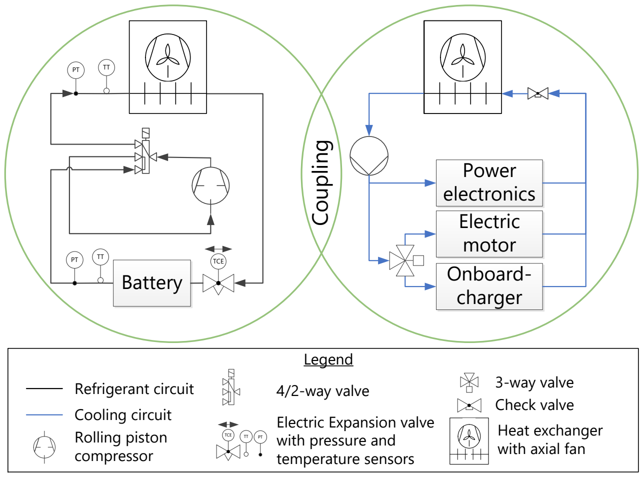

Fig. 3. HVAC-decoupled battery cooling

C. *Increasing the performance and efficiency of the battery thermal management and usage of alternative refrigerants*

Using a separate refrigerant circuit for the battery thermal management allows to adjust the evaporation temperature to the required battery temperature of 25 °C [8]. By increasing the evaporation temperature, the cooling capacity as well as the energy efficiency is increased. The enhancement can be quantified independently of the system size by calculating a one-stage refrigerant cycle. The results for several refrigerants applicable for automotive applications can be found in chapter IV.

D. *Noise reduction*

Dissipating large heat flows in a standing vehicle is only possible by either increasing the radiator size, the fan performance or the temperature level. While the first measure will increase the required space and the drag coefficient and the second measure will increase the noise level. The temperature level only depends on the operation conditions of the refrigerant circuit. Increasing the condensation temperature will mainly reduce the energy efficiency and will have minor effects on the cooling capacity of a decoupled system. An estimation isgiven in chapter V.

E. Reduction of refrigerant charge

The adaption of the evaporation temperature and the shorter distance between evaporator and condenser of the battery thermal management lead to shorter tubing and hose lines. This results in smaller refrigerant charges.

F. Alternative refrigerants

Efficient, eco-friendly and cost-efficient natural refrigerants can be applied when HVAC and battery thermal management are decoupled. These refrigerants are flammable and thus prohibited in HVAC systems due to the increased risk of emission of refrigerant into the cabin during a crash.

G. Additional components

The big drawback of the decoupled battery thermal management is the installation space, mass and costs of the additional components such as compressor, condenser and (expansion) valves. The components can be downsized due to the cooling performance increase of the adapted evaporation temperature and the use of alternative refrigerants. Refrigerant costs can be cut down by using natural refrigerants as the required refrigerant quantity and costs are less than R1234yf.

H. Flammability of alternative refrigerants

Another issue poses the flammability of natural refrigerants such as Propane. Flammable fluids are not allowed in direct HVAC systems as the refrigerant can contaminate the cabin in case of a crash. Quantities below 150 g require no additional safety features [9] and should be aspired for decoupled battery thermal management systems.

IV. Refrigerant Comparison

To access the ideal refrigerant for a decoupled battery thermal management a comparison of several refrigerants was conducted using the cost–utility analysis defined in VDI guideline 2225 [10]. This guideline suggests the weighting of independent criteria, followed by scores depending on the fulfilment of each criterion. Multiplying the scores with the weighting and adding up the products results in a quality rating. This proceeding is depicted exemplary with two refrigerants in chapter IV.E. The overall results are presented in chapter IV.F. The comparison was conducted for three temperature settings separately.

The thermodynamic comparison is based on a one-stage refrigerant cycle. As Carbon Dioxide is hypercritical over 30.98 °C and 73.8 bar the condensation pressure was calculated according to the correlation of Liao et al [11]. For all other refrigerants the condensation pressure is coupled to the condensation temperature in the diphase region. The calculation is also conducted for the current state of the art of a direct evaporation system as reference.

This chapter will present the boundary conditions as well as the criteria and the results for the battery cooling of a small two-wheeled vehicle.

A. Boundary Conditions

The most important boundary condition is the global warming potential (GWP) of the refrigerant. By 2017 all refrigerants with a GWP over 150 are forbidden in automobile applications [12]. Furthermore toxic refrigerants holding the safety class B are ignored. Only refrigerants fulfilling these requirements were chosen for the comparison. The refrigerants as well as the GWP and the safety group can be found in TABLE I.

TABLE I. Compared refrigerants

Name	Short mark acc. to DIN 8960	Chemical composition	GWP	Safety group
Carbon dioxide	R744	CO_2	1	A1
Propane	R290	C_3H_8	3	A3
Propene	R1270	C_3H_6	3	A3
Isobutane	R600a	C_4H_{10}	3	A3
2,3,3,3-Tetrafluor-propene	R1234yf	$C_3H_2F_4$	4	A2L
1,3,3,3-Tetrafluor-propene	R1234ze		6	A2L
Difluorethane	R152a	$C_2H_4F_2$	124	A2

To compare these refrigerants three temperature settings were defined based on automotive standard specifications and average temperatures in mega cities: (1) hot climate with an ambient temperature of 40 °C according to DIN 1946-3 [4] also being the maximum average summer temperature in Abu Dhabi. (2) Cold climate illustrating the average minimal temperature in Moscow of -10 °C. (3) Standard conditions of 20 °C. More boundary conditions are presented in TABLE II.

The evaporation temperature in hot and standard conditions was defined 10 K lower than the optimum battery temperature. The direct evaporation system with R1234yf is defined as reference with an evaporation temperature of current HVACs of -5 °C [13].

The condensation temperature was defined 15 K above the ambient temperature. For the cold climate the evaporation temperature is 15 K lower than the ambient as heat needs to be transferred from the ambient to the battery.

TABLE II. Boundary conditions

Name	Symbol	Hot climate	Standard conditions	Cold climate
Ambient temperature [°C]	t_{cL1}	40	20	-10
Condensation temperature [°C]	t_c	55	35	25
Evaporation temperature [°C]	t_o	15	15	-25
Superheat [K]	Δt_{o2h}	5		
Subcooling [K]	Δt_{c2u}	3		
Isentropic efficiency [-]	η_{is}	0.83		
Mechanical efficiency [-]	η_m	0.87		

The isentropic and mechanical efficiency were defined as constants for all refrigerants with typical literature values [14].

The cooling requirements are based on the fast charging of a small battery system (2 kWh, 48 V) used in a small electric scooter. However the thermo-physical properties are independent of the system size and also apply for automotive applications.

B. Criteria and weighting

The criteria for the evaluation are based on the ideal refrigerant properties in Pohlmann [14]. 13 independent properties were chosen for the comparison. They contain thermo-physical, as well as chemical, electrical, ecological, physiological and economical properties. TABLE III shows the rating as well as the weighting of the properties.

TABLE III. CRITERA

<table>
<tr><th>#</th><th>Weighting</th><th colspan="6">Properties</th></tr>
<tr><td>1</td><td>100%</td><td colspan="6">Volumetric cooling/heating capacity</td></tr>
<tr><td>2</td><td>91%</td><td colspan="6">Compression ratio</td></tr>
<tr><td>3</td><td>83%</td><td colspan="6">Condensation pressure</td></tr>
<tr><td>4</td><td>65%</td><td colspan="2">Hot gas temperature</td><td colspan="2">EER/COP</td><td colspan="2">Evaporation pressure</td></tr>
<tr><td>5</td><td>39%</td><td colspan="2">Toxicity</td><td colspan="2">GWP</td><td colspan="2">Safety group</td></tr>
<tr><td>6</td><td>22%</td><td colspan="6">Refrigerant cost</td></tr>
<tr><td>7</td><td>9%</td><td colspan="3">Electrical conductivity</td><td colspan="3">Water solubility</td></tr>
</table>

The two most important properties directly affect the cooling $\dot{Q}_o$ and the heating $\dot{Q}_c$ capacity of the refrigerant cycle. It can be estimated with equation (1), where $\dot{V}_g$ is the geometric flow capacity of the compressor, λ is the volumetric efficiency of the compressor, Π is the compression ratio and q_{ov} is the volumetric cooling capacity.

$$\dot{Q}_o = \dot{V}_g \, \lambda(\Pi) \, q_{ov} \tag{1}$$

The heating capacity $\dot{Q}_c$ is defined by exchanging the volumetric cooling capacity q_{ov} with the volumetric heating q_{cv} capacity in Eq.1.

The volumetric cooling capacity is defined in Eq. 2. It is the product of the specific enthalpy increase during evaporation Δh_o and the density at the intake of the compressor ρ_1.

$$q_{ov} = \Delta h_o \, \rho_1 \tag{2}$$

For the calculation of the volumetric heating capacity q_{cv} the specific enthalpy decrease Δh_c at the condenser is used instead.

$$q_{cv} = \Delta h_c \, \rho_1 \tag{3}$$

Increasing the volumetric cooling/heating capacity affects the performance of the refrigerant circuit as the geometric flow capacity $\dot{V}_g$ and thus compressor size and tubing diameters can be minimized. Alternatively the same compressor can obtain higher cooling and heating capacities if a refrigerant with high volumetric cooling/ heating capacity is used. The most important property for small vehicles is therefore the volumetric cooling and heating capacity as volume and mass constraints are high.

The second most important property is the compression ratio Π. The volumetric efficiency λ used in Eq. 1 mainly depends on the compression ratio. Increasing the compression ratio will result in a decrease of volumetric efficiency and thus cooling/heating capacity.

The condensation pressure is ranked third as it directly affects the construction of the parts in the refrigerant circuit. All components need to withstand the pressures of the refrigerant circuit in all temperature conditions. High system pressures often imply higher costs for the components. For a small vehicle the costs need to be minimized.

The fourth rank is seized by three thermodynamic properties: (1) The hot gas temperature which is measured after the compressor. Low hot gas temperatures result in an efficient use of the condenser resulting in lower condensation pressures and temperatures. Contrary to the boundary condition in TABLE II the condensation temperature t_c is not fixed but depends on several properties. Among them are the isentropic efficiency of the compressor, the condenser size, the air flow rate or rather fan performance, the refrigerant charge and others. For the comparison the condenser temperature was defined constant to enable a fast comparison of several refrigerants. (2) The Energy Efficiency Ratio (EER) is defined in Eq.4. It is the quotient of the cooling capacity $\dot{Q}_o$ and the compressor power P.

$$\text{EER} = \frac{\dot{Q}_o}{P} = \frac{\Delta h_o}{h_2 - h_1} \, \eta_m \tag{4}$$

In steady-state operation it can be calculated with the evaporation enthalpy increase Δh_o, the enthalpy increase during the compression $h_2 - h_1$ and the mechanical efficiency of the compressor η_m.

For the heat pump operation the Coefficient of Performance (COP) is defined in Eq. 5 with the heating capacity $\dot{Q}_c$ and the enthalpy decrease Δh_c.

$$\text{COP} = \frac{\dot{Q}_c}{P} = \frac{\Delta h_c}{h_2 - h_1} \, \eta_m \tag{5}$$

The EER/COP was classified less important than the volumetric cooling/heating capacity as the thermal management is optimized for fast charging. During driving the boundary conditions and thus the EER/COP improve as the waste heat is much lower and the airstream improves the condensation temperature/pressure. (3) The evaporation pressure must be higher than ambient pressure in all conditions to avoid the refrigerant cycle contamination with air and moisture. This is particularly critical in heat pump operations.

The properties on the fifth and following ranks are not thermo-physical. No direct toxicity is given for the refrigerants. However the decomposition during the incineration of R1234yf, R1234ze and R152a can produce toxic hydrogen fluoride. The GWP was mentioned in TABLE I and was also taken into account.

The safety group defines the flammability of the refrigerant. Carbon Dioxide is the only non-flammable refrigerant followed by R1234yf/ze in the A2L and R152a in the A2 group which is less flammable. Propane, Propene and Isobutane are highly flammable (A3) and thus have restrictions in refrigerant charge and safety measurements.

The refrigerant costs span from 7.6 €/kg (Carbon Dioxide) to 116 €/kg (R1234yf) and were also taken into account.

The electrical conductivity should be low to avoid battery short-circuits. The values for the comparison are based on the measurements of Hegewald and Feja [15]. The natural refrigerants Propane, Propene and Isobutane feature the lowest electric conductivity as all have unipolar polarity. R1234yf and R1234ze exhibit the highest polarity of the considered refrigerants.

C. Thermodynamic properties

The results of the three most important properties are presented in this chapter. The calculation was conducted for 7 refrigerants for the scenarios hot climate (transverse stripes), standard conditions (filled) and cold climate (dotted). An additional calculation was conducted for R1234yf with the evaporation temperature of current HVAC system, representing the state of the art.

1) Volumetric cooling capacity

The volumetric cooling capacity is depicted in Fig. 3. The volumetric heating capacity is depicted for the cold climate.

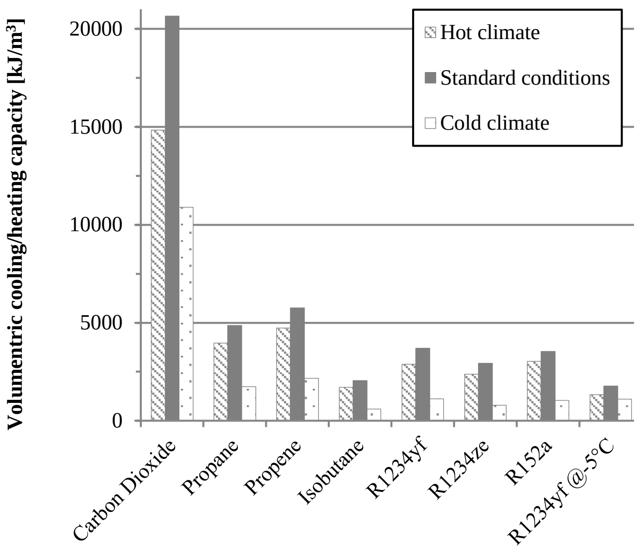

Fig. 5. Volumetric cooling/heating capacity comparison

For a better understanding the comparison will only feature the values for the standard conditions.

At all ambient temperatures Carbon Dioxide offers the largest volumetric cooling capacities. For the standard conditions the value is 20,649 kJ/m^3. The second and third largest values for all scenarios can be obtained with Propene (5,768 kJ/m^3) and Propane (4,856 kJ/m^3). The rating of the other refrigerants is changing depending on the ambient temperature. Their values are lower generally than 3700 kJ/m^3. The lowest values are obtained by the sample case of R1234yf at -5 °C it shows the lowest values except for the cold climate where Isobutane is even lower.

2) Compression ratio

The compression ratio for the three scenarios is depicted in Fig. 4.

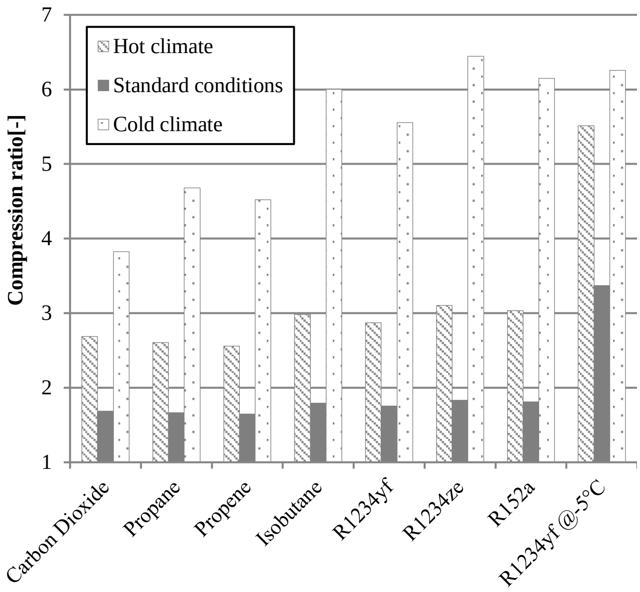

Fig. 4. Compression ratio comparison

The compression ratio for standard conditions is in a limited range between 1.6 and 1.8. Only the sample case exhibits a ratio over 3. The same characteristics apply for the hot climate. The compression ratios are distributed between 2.6 and 3, except for the sample case which has a value of 5.5. In cold climate conditions Carbon Dioxide offers the smallest compression ratio of 3.8. All other refrigerants exhibit compression ratios between 4.5 and 6.4.

3) Condensation pressure

The condensation pressure is depicted in Fig. 5.

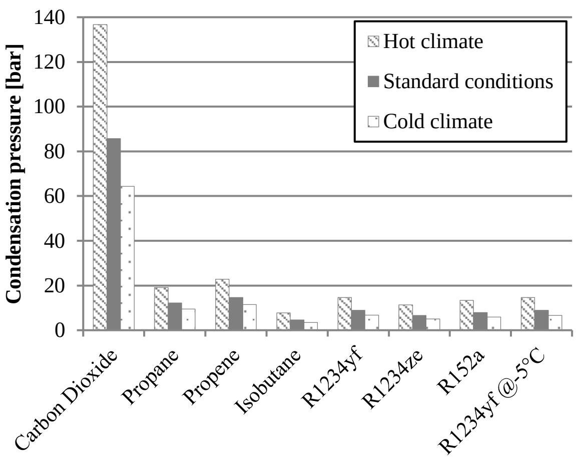

Fig. 6. Condensation pressure comparison

The biggest drawback of Carbon Dioxide as refrigerant is the high system pressure. Especially for hot climate conditions this poses a problem as values can reach up to 137 bar. Moreover, in standard conditions the pressure of a Carbon Dioxide refrigerant cycle is over nine times higher than in an R1234yf cycle.

The second highest condensation pressures in all conditions can be found in a Propene cycle. Pressures up to 22.8 bar can appear in hot climate conditions. The lowest condensation pressures occur with Isobutane. This explains the usage in refrigerators as low pressures enable inexpensive components and the required cooling capacities are low.

D. Scale of Values

The scale of values depicted in TABLE IV. assigns a score from 0-4 depending on the fulfilment of the criteria which were defined in TABLE III. The values for the scores of the thermos-physical properties are calculated with the percentage increase or decrease of the property in comparison to the state of the art.

TABLE IV. SCALE OF VALUES

Criteria \ Score	0	1	2	3	4 (best)
Volumetric cooling/ heating capacity	<+25%	<+75%	between	>+175%	>+225%
Compression ratio	> -46%	> -47%		< -49%	< -50%
Condensation pressure	> +40%	> +20%		< -20%	< -40%
Hot gas temperature	>+17 %	>+12 %		<+2%	<-3%
EER/ COP	<+112%	<+115%		>+121%	>+124%
Evaporation pressure	< 1 bar	-		-	>1 bar
Toxicity	--		-		0
GWP	>6	<=6	<=4	<=2	0
Safety group	A3	A2L	A2	-	A1
Refrigerant cost	>50€/kg	<50€/kg	<40€/kg	<30€/kg	<20€/kg
Electrical conductivity	--	-	0	+	++
Water solubility	>400 mg/l	>300 mg/l	>200 mg/l	>100 mg/l	<100 mg/l

The limits were calculated to assign the highest/ lowest scores to the highest/ lowest values giving the values in the range of the average a score of 2 points. The scale of values was defined for the values during standard conditions and is executed for all three climate conditions.

E. Exemplary execution for two refrigerants

An insight into the comparison using the cost–utility analysis is given for Carbon Dioxide and Propane in this chapter. The scores for the criteria and the calculation of the quality rating for the standard condition is depicted in TABLE V. The quality rating is the ratio of the sum of the scores and weighted scores of each refrigerant to the maximum possible value.

TABLE V. QUALITY RATING FOR STANDARD CONDITIONS

Criteria \ Refrigerant	Carbon Dioxide		Propane	
	value	**weighted**	**value**	**weighted**
Volumetric cooling/ heating capacity	4	4.00	3	3.00
Compression ratio	4	3.64	4	3.64
Condensation pressure	0	0.00	1	0.83
Hot gas temperature	0	0.00	2	1.3
EER	0	0.00	4	2.6
Evaporation pressure	4	2.6	4	2.6
Toxicity	4	1.57	4	1.57
GWP	3	1.17	2	0.78
Safety group	4	1.57	0	0.00
Refrigerant cost	4	0.87	3	0.65
Electrical conductivity	3	0.26	4	0.35
Water solubility	0	0.00	4	0.35
Sum	30.0	15.7	35.0	17.7
Average points	2.50	2.30	2.92	2.59
Quality rating	62.5%	57.4%	72.9%	64.7%

Having a high volumetric efficiency Carbon Dioxide receives the highest score. Concerning the condensation pressure, the hot gas temperature and the EER, Carbon Dioxide is below the average of the compared refrigerants. Both natural refrigerants exhibit evaporation pressures above 1 bar, pose no direct toxicity and have low GWPs. Being in the safety group A3 Propane inherits zero points in that category whereas Carbon Dioxide is the least dangerous refrigerant of the considered refrigerants.

F. Results and refrigerant selection for a small two-wheeled vehicle

The overall results for the refrigerants in the three defined climate conditions are shown in TABLE VI.

TABLE VI. QUALITY RATING

Name	**Short mark acc. to DIN 8960**	**Quality rating**		
		Hot climate	Standard conditions	Cold climate
Carbon dioxide	R744	**57,4**	57,4	**44,1**
Propane	R290	55,2	**64,7**	29,8
Propene	R1270	50,6	62,5	29,9
Isobutane	R600a	48,8	55,2	30,5
2,3,3,3-Tetrafluor-propene	R1234yf	45,5	55	31,5
1,3,3,3-Tetrafluor-propene	R1234ze	42,7	48,5	23,4
Difluorethane	R152a	33,8	46,6	16,9

For hot conditions Carbon Dioxide followed by Propane should be preferred. The quality rating in this condition for the remaining refrigerants is significantly lower. For the standard conditions Propane is the favored refrigerant scoring the highest percentage of 64.7 %. Second is Propene with a 2.2 percentage points lower quality rating. Especially in heat pump operation in a cold climate carbon dioxide is outranking the remaining refrigerants. Eventually the authors consider Propane as the ideal refrigerant for small battery systems in electric two-wheeled vehicles. Carbon Dioxide sets too high requirements onto the components regarding pressure stability and thus will have higher component costs than Propane.

However using Carbon Dioxide in automotive applications already containing a CO_2-HVAC could benefit from the already existing components. The integration of the battery thermal management in the medium pressure level of a CO_2 two-stage cycle should then be considered.

V. INCREASING THE CONDENSATION TEMPERATURE

Dissipating large heat flows in a standing vehicle is only possible by either increasing the radiator size, the fan performance or the temperature level. Increasing the condensation temperature will primarily reduce the energy efficiency while drag coefficient and noise levels will be unaffected. The cooling performance $\dot{Q}_o$ will only decrease slightly as the volumetric efficiency λ decreases due to the higher compression ratio Π and the volumetric cooling capacity q_{ov} will be reduced by the lower specific evaporation enthalpy Δh_o. However the density ρ_1 at the intake of the compressor is unchanged.

TABLE VII. COMPARISON OF CONDENSATION TEMPERATURES

Name	**R1234yf-state of the art**	**R290 (Propane)**	
Condensation temperature [°C]	55		70
Energy Efficiency Ratio (-)	2.1	4.1 (+90%)	2.5 (+19%)
Compression ratio [-]	5.5	2.6 (-54%)	3.5 (-36%)
Volumetric cooling capacity [kJ/m3]	1335	3956 (+197%)	3220 (+141%)

The impact of a condensation temperature of 70°C on a decoupled refrigerant cycle with Propane is depicted in TABLE VII. The values are compared to the state of the art which is also shown in the figures 5-7. The biggest impact of the high condensation temperature is seen in the EER. This will drop from an increase of 90 % to only 19 %. However, this will have no effect on the driving range as the energy is provided by the charging station. The improvements in compression ratio and volumetric cooling capacity are slightly reduced. The condensation temperature increase can provide low-noise fast charging by decreasing the energy efficiency of the refrigerant circuit.

VI. SUMMARY AND OUTLOOK

The idea of decoupling the battery thermal management from the HVAC was presented in this paper. Starting with the state of art several qualitative statements were given concerning the advantages and disadvantages of this approach. This was followed by a comparison of seven refrigerants comparing 13 properties in three temperature scenarios. While Carbon Dioxide was considered appropriate for automotive applications, Propane was chosen to be ideal for small electric vehicles which currently feature no HVAC. To avoid a large condenser as well as a high fan noise level the impact of increased condensation temperatures was calculated. In case of Propane this would reduce the energy efficiency significantly and would decrease the volumetric cooling capacity only slightly. Consequentially being an appropriate procedure for fast charging of electric vehicles in noise sensitive areas.

In the near future simulations will be conducted for several levels of drivetrain cooling integration to assess the performance of the refrigerant circuit. For this purpose component behavior needs to be measured for existing components especially for Propane e.g. isentropic and volumetric efficiency of the compressor. New thermal interfaces to the battery cell as a consequence of the approach will be examined in simulations and prototypes.

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