

Simulative Analysis of Secondary Loop Automotive Refrigeration Systems Operated with an HFC and Carbon Dioxide

Jan Christoph Menken, Martin Ricke, and Thomas A. Weustenfeld
AUDI AG

Juergen Koehler
University of Braunschweig

ABSTRACT

Recent attempts to find energy-efficient thermal management systems for electric and plug-in hybrid electric vehicles have led to secondary loop systems as an alternative approach to meet dynamic heating and cooling demands and to reduce refrigerant charge. The choice of refrigerant for the primary refrigeration cycle is an important issue regarding the overall system performance. In this work, an HFC refrigerant (R-134a) and a natural refrigerant (R-744) are evaluated regarding a potential use in secondary loop systems. To meet the demands of R-744 cycles such as higher system pressure, most components have to be redeveloped. Nonetheless the use of the environmentally friendly refrigerant has advantages such as better applicability and performance in heat pump systems under cold ambient conditions. This work presents Modelica-based design and simulations of secondary loop automotive refrigeration systems with the HFC refrigerant in a subcritical and R-744 in a transcritical primary cycle. Under defined use cases and environmental conditions, both primary cycles are compared resulting in a better performance for HFC in a summer scenario and for R-744 under winter conditions.

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INTRODUCTION

As secondary loop systems become a viable alternative approach for automotive thermal management systems, fundamental design requirements concerning the primary vapor compression cycle need to be specified. An important influencing factor is the choice of refrigerant. This simulative work compares two different primary refrigerant cycles of a secondary loop system: The first system is equipped with R-134a which is widely used throughout the automotive industry. The second system uses R-744 (CO₂) as refrigerant. Due to the fact that both systems exhibit fundamental differences in terms of design and operation, the focus of this paper lies in providing comparability if procurable. This is achieved by presuming equal heating or coolant capacities as well as justifiable assumptions for modification based on state-of-the-art technologies for design. For the system evaluation, real-world driving cycles and representative weather data are taken into account.

The general concept of a secondary loop refrigeration system can be found in various fields of application. Wang et al. [1] offer an extensive review on secondary loop refrigeration systems including performance and risk assessment of flammable refrigerants. Applications belonging to the fields of commercial refrigeration, residential air conditioning/heat pumping and mobile air conditioning are presented. Recently, much research has been performed regarding automotive applications in this field. Ghodbane et al. [2] present a

secondary loop system with the HFC refrigerant R-152a. Kowsky et al. [3] demonstrate a hermetic encapsulated central thermal management unit for the use in electric and hybrid vehicles using the battery waste heat as energy source during heat pump mode. The occurring energy storage effects in such systems can even be increased by the addition of phase change materials [4] or ice-based latent energy storages [5] in the secondary loops.

Due to the high global warming potential (GWP) of R-134a, the European Union's directive for emissions from air-conditioning systems in motor vehicles constrain the use of such fluorinated greenhouse gases [6]. It is specified that new vehicle models have to be fitted with a refrigerant having a GWP less than 150.

A climate-friendly alternative is the refrigerant R-744 (CO₂) with a GWP of 1. Due to its low critical temperature of $T_{crit} = 31.1\text{ °C}$ the R-744 refrigerant cycle discharges heat at a transcritical level. Because of the fact that the cycle is operated at a higher pressure level, R-744 cannot be used as a drop in refrigerant. This is why all refrigerant components have to be engineered from scratch. As shown in Table 1 the volumetric refrigeration capacity of R-744 is 8 times higher compared to R-134a. Therefore a smaller mass flow rate is needed to achieve a certain cooling load which results in a reduced stroke volume of the compressor. Another advantage is based on the low evaporation temperature. Thus, a R-744 equipped heat pump

system can be operated on lower ambient temperatures than a R-134a one. As the density declines with pressure, a higher mass flow rate can be obtained in the CO₂-system.

Table 1. Comparison of selected parameters for refrigerants R-134a and R-744 [20, 21, 22, 23].

	R-134a	R-744
GWP	1300	1
Autoignition Temperature	743 °C	none
Cost	20-30 €/kg	10 €/kg
Critical Pressure	40.7 bar	73.8 bar
Critical Temperature	101.1 °C	31.1 °C
Boiling Temperature at 1 bar	-26.4 °C	-78.4 °C
Volumetric Refrigeration Capacity at 0 °C	2868 kJ/m ³	22545 kJ/m ³

METHODS

Boundary Conditions

As the requirements of the vehicle thermal management system change over the year depending on the environmental conditions, the system evaluation is not a trivial task. Additionally, the driving behavior plays an important role. In order to use representative data describing operating conditions of the evaluated system, the boundary conditions have to be specified. As this work uses the waste heat of electric drivetrain components (battery, electric machine, power electronics), real world driving cycles have to be taken into account. Additionally, the environmental conditions influence the vehicle's transient behavior. Therefore, the ambient temperature $T_{air,amb}$, relative humidity $\phi_{air,amb}$ and solar radiation $\dot{q}_{sol}$ need to be specified. Describing the customer's driving behavior, the Artemis road driving cycle is used. Figure 1 shows the velocity profiles over time [7] as well as the calculated drivetrain losses [8].

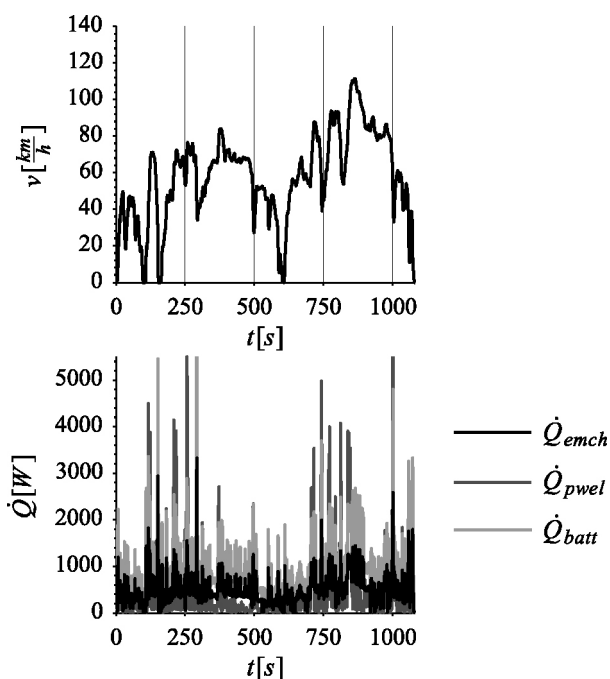


Figure 1. Velocity profile and drivetrain waste heat of electric machine, power electronics and battery for Artemis urban, road and motorway.

Regarding the ambient conditions, Table 2 displays the values for $T_{air,amb}$, $\phi_{air,amb}$ and $\dot{q}_{sol}$ for the use cases 1, 2 and 3 which represent the three typical heating or cooling loads of the passenger compartment that lead to the dynamic cabin heating and cooling loads described in the following section.

Table 2. Ambient conditions for the three use cases heat-up, re-heat and cool-down.

	Heat-up	Re-heat	Cool-down
$T_{air,amb}$ [°C]	5	15	30
$\phi_{air,amb}$ [%]	70	70	42
$\dot{q}_{sol}$ [W/m ²]	0	0	800
$\dot{m}_{col,jc,in}$ [kg/s]	0.15	0.15	0.15
$\dot{m}_{col,chi,in}$ [kg/s]	0.15	0.15	0.15

Dynamic Heating and Cooling Loads

For the evaluation of the system configurations, a battery powered electric vehicle is assumed. Because of the lack of an internal combustion engine, the missing heat load during winter and reheat operation mode is provided by a primary refrigerant cycle operated in a heat pump mode using the waste heat of the electric drivetrain components and an electric heater if necessary as heat source. The reheat operation mode is distinguished by dehumidification of the instream air and subsequent re-heating before entering the passenger compartment. Based on experiments and preliminary simulation, the cooling and/or heating load at coolant-to-air heat exchanger is predefined as shown in Figure 2. Positive heat flow rates represent a heat-up of the instream air, whereas a negative sign indicates the air side cooling demand. It has to be mentioned that during re-heat operation, only the cooling demand at the first heat exchanger is controlled. It is assumed that sufficient heating load for the second cabin heat exchanger is provided by a heat surplus at the condenser.

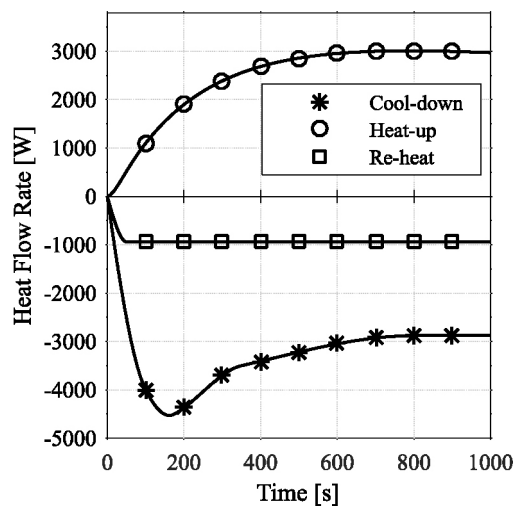


Figure 2. Air side heat flow rates at coolant-to-air heat exchanger within the air handling unit for heat-up, re-heat and pulldown.

Modeling and Simulation

Refrigeration Cycle

All system setups presented in this work are modeled with the TIL Library in Dymola/Modelica. They consist of an electrical scroll compressor, a condenser, an electrical expansion valve, an evaporator and a refrigerant reservoir on the low pressure side (accumulator). This setup is described as optimal from an operating point of view as subcooling can be set while superheating is close to zero [9]. The simulation of the R-134a model is validated in a previous publication [10]. The R-744 cycle additionally contains an internal heat exchanger. Figure 3 shows the setup of the two primary loop refrigerant cycles modeled in Dymola.

Compressor

The compressor is a crucial component regarding the overall energy consumption of the system. Both refrigerant cycles are driven with an electric loss based scroll compressor described in [11]. The parameters based on measurements of an electric R-134a scroll compressor and then extrapolated for the desired maximum cooling load are given in Table 3. As R-134a and R-744 have different specific volumetric heat capacities, the suction volumes have to be chosen respectively (based on the ratio of suction volumes of mechanical swash-plate-piston compressors) [12].

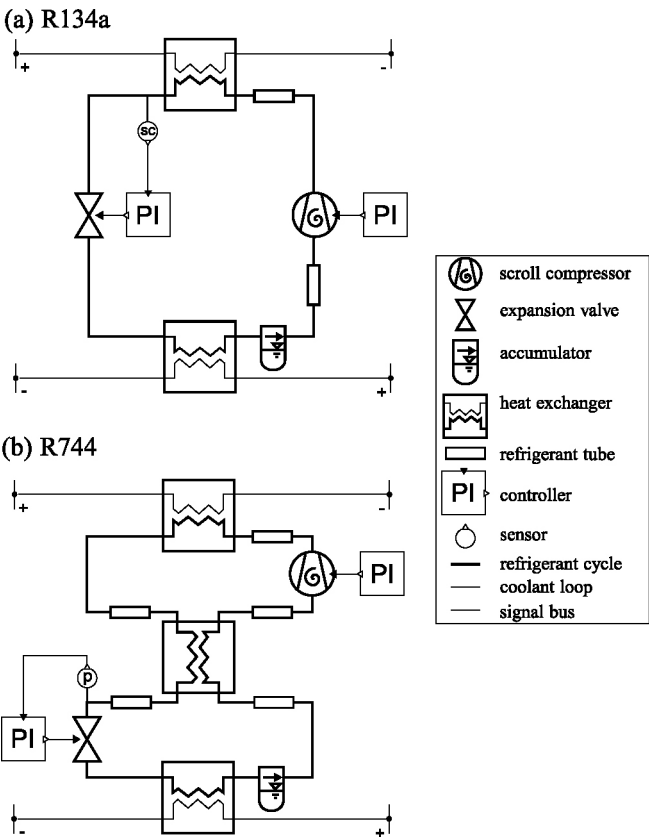


Figure 3. Primary refrigerant cycle for R134a (a) and R744 (b).

Table 3. Selected compressor parameters for R-134a and R-744.

	R-134a	R-744
Suction Volume [cm³]	34	5.49
Effective Leakage Area [mm²]	0.028	0.0035
Electric Efficiency [-]	0.764	0.88

Plate Heat Exchanger

Plate heat exchangers with their compact dimensions and typically high heat flux densities are used as refrigerant-to-coolant heat exchangers. Figure 4 shows the general flow distribution of both mediums. Due to the unavailability of high pressure CO₂ plate heat exchangers for automotive applications, the dimensions, pressure losses and heat transfer coefficients are estimated. To maintain comparability the R-134a heat exchangers are also modeled with constant coefficients instead testbed tuned correlations for pressure drop and heat transfer.

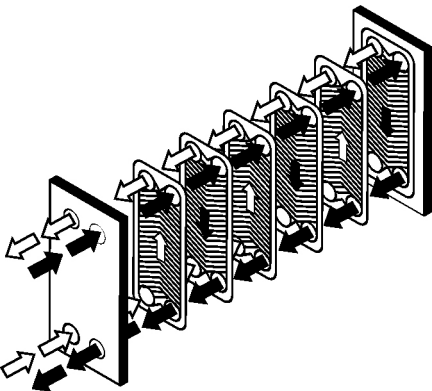


Figure 4. Flow distribution in a plate heat exchanger [24].

Both refrigerant cycles are sized for a cooling capacity of 7.5 kW at ambient conditions of T = 40 °C and φ = 40 % (Table 4).

Table 4. Plate Heat Exchanger geometry data of R-134a and R-744 condenser/ gas cooler and evaporators.

	Condenser	Evaporator
R-134a		
Number of parallel flows per medium [-]	20	20
Total Heat Transfer Area [m²]	0.84954	0.799901
Refrigerant-Side Heat Transfer Coefficient [W/m²K]	2000	2000
Coolant-Side Heat Transfer Coefficient [W/m²K]	4000	4000
Refrigerant-Side Pressure Drop [bar]	0.5	0.6
Coolant-Side Pressure Drop [bar]	0	0
R-744		
Number of parallel flows per medium [-]	20	20
Total Heat Transfer Area [m²]	0.753358	0.428084
Refrigerant-Side Heat Transfer Coefficient [W/m²K]	4000	4000
Coolant-Side Heat Transfer Coefficient [W/m²K]	4000	4000
Refrigerant-Side Pressure Drop [bar]	2	1
Coolant-Side Pressure Drop [bar]	0	0