ABSTRACT

The EU funded project B-COOL deals with the development of a low cost CO₂ (R744) mobile air conditioning system for lower segment cars. The design process is supported by computer simulations using the object oriented MODELICA library AirConditioning in combination with the modelling tool Dymola. Within this paper the main component models of the MAC systems of a Ford Ka and a Fiat Panda are presented. The accumulator will be discussed in detail as it has an important impact on the system behaviour. Test rig measurements of different accumulators are presented and compared with simulation results. Furthermore, steady state simulations of the R744 MAC system using different component and cycle designs are discussed. Transient simulations based on the New European Driving Cycle (NEDC) are presented.

1. INTRODUCTION – THE B-COOL PROJECT

On 31 January 2006 the EU agreed to phase out HFC-134a from air conditioning systems in new vehicle models from 1 January 2011 [1]. About 17 million cars are sold every year in the EU [2]. The lower priced vehicles constitute up to 70% of the present EU car market. This number will rise up to 80% with the EU enlargement.

The B-COOL project objective is the development of a low cost and high efficiency air-conditioning system for lower priced cars based on a vapour compression cycle using CO₂ as a refrigerant.

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2. MODELICA AND AIRCONDITIONING

Within the B-COOL project Modelica/Dymola in combination with the AirConditioning library and own models are used for cycle and component simulation.

Modelica is an object-oriented modelling language used to model large, complex and heterogeneous physical systems. The language is designed for convenient, component-oriented modelling of physical multi-domain systems. Different commercial and non commercial model libraries are available to carry out simulations in different application areas e.g. of mechanical, thermodynamic, magnetic and electric systems.

The German automotive manufacturers have jointly decided to use Dymola and the AirConditioning library as a tool for model exchange and simulation of automotive air conditioning systems [3]. Dymola and the AirConditioning library provide sustainable possibilities for simulating
automotive air conditioning systems and allow for optimizing and verifying the design of an air conditioning system from the early design phases through control design and implementation [4-6]. The library provides detailed models for different types of heat exchangers, compressors, valves, receivers and further components. The user can connect components thoroughly, which makes it possible to realize non-standard configurations such as cycles with parallel evaporators. Both transient simulations and steady state simulations e.g. pull down and driving cycle experiments can be carried out. Further advantages of using the Modelica/Dymola in combination with the AirConditioning library compared to other programs are the open source code and the possibility to exchange models or encrypted models with other AirConditioning users. The models in Modelica are mathematically described by differential, algebraic and discrete equations.

3. COMPONENT MODELS USED FOR THE B-COOL SIMULATION

3.1. Cycle Setup

Part of the B-COOL project is the analysis of different refrigerant cycle setups. For the simulations presented in this publication a cycle as shown in Figure 1 was used.

![Figure 1. Cycle setup and representation using Modelica/Dymola and AirConditioning](image)

The cycle consists of a compressor, gas cooler, internal heat exchanger, orifice tube, evaporator and a low pressure accumulator. The main components are discussed in the following chapters.

3.2. Heat Exchangers

Condensers/gas coolers and evaporators in automotive refrigeration cycles are mostly of cross flow or cross-counter flow type and consist of louvered fins and extruded micro channel flat tubes, both made of aluminium. The models in the AirConditioning library are composed of refrigerant and air cross flow elements with walls between the two media. Heat conduction in the solid material in fluid flow direction is neglected. The dynamic behaviour of the component is influenced by the amount and distribution of the solid wall material and associated heat capacity. On both sides of the wall, several parallel flow channels are lumped into one homogeneous flow for efficiency reasons. The refrigerant path through the component is treated as one pipe flow with variable cross section and one air element associated with each flow segment. Each air element is further discretized or symbolically integrated along its flow. Automatic coupling of air elements is made according to the parameter-specified and component type dependent 3D orientation, e.g. as the evaporator and to the
user defined segmentation of the refrigerant flow. Both parameters are merged into a 3D-matrix, which defines the position of each refrigerant segment with respect to a fixed coordinate system. This approach allows for a wide variety of flow paths and a 2D-interface for inhomogeneous air inlet. However, the interface resolution is directly coupled to the number of refrigerant passes through the heat exchanger and their segmentation [7].

For systems using the refrigerant R744 as the cycle fluid, it is quite common to have an internal heat exchanger between the high pressure side, after the gas cooler, and the low pressure side, between the accumulator and the compressor. The base classes for the internal heat exchanger are identical to those for the refrigerant side of flat tube heat exchangers. Currently used internal heat exchangers come in a wide variety of geometries. Tube-in-tube type internal heat exchangers can be parameterized directly from the geometrical data. For other types of internal heat exchangers, the user has to compute parameters like the hydraulic diameter and the heat transfer areas by hand.

3.3. Compressor Simulation

The compressor is mapped using a quasi-stationary model based on measurement data for full load or partial load operating conditions [6]. Mass flow rate, shaft power and discharge temperature are calculated as a function of compressor speed, relative displacement, pressure ratio for a given set of suction and discharge pressure and suction temperature. Measurements are used to adapt the free parameters of the efficiency functions that are chosen to have physically reasonable asymptotes for high pressure ratios and low displacement. The form of the functions varies slightly for different compressor types. The compressor model uses three functions to characterize the compressor efficiencies, the volumetric efficiency \( \lambda_{\text{eff}} \), the effective isentropic efficiency \( \eta_{\text{eff}} \) and the isentropic compressor efficiency \( \eta_{\text{isen}} \). The efficiencies are defined as

\[
\lambda_{\text{eff}} = \frac{m_{\text{eff}}}{V \cdot n \cdot \rho (p_s, T_s)} \tag{1}
\]

\[
\eta_{\text{eff}} = \frac{P_{\text{isen}}}{P_{\text{eff}}} = \frac{h_{d,\text{isen}} - h_s}{2 \cdot \pi \cdot |M| \cdot n} \cdot \frac{\dot{m}_{\text{eff}}}{V} \tag{2}
\]

\[
\eta_{\text{isen}} = \frac{h_{d,\text{isen}} - h_s}{h_d - h_s} \tag{3}
\]

In the definition above, \( p \) is the pressure, \( T \) temperature, \( V \) displacement volume, \( \rho \) density, \( h \) specific enthalpy, \( P \) power, \( \dot{m} \) mass flow rate and \( M \) the torque of the compressor. The subscripts \( d \) refer to the discharge side, \( s \) to the suction side, \( isen \) to isentropic conditions and \( eff \) to effective values.

The mathematical characteristic functions for the compressor model are factored into two parts: one that captures the influence of the pressure ratio \( \pi \) and the rotational speed \( n \), \( f(\pi, n) \) and another one that takes into account the control of the swash plate angle \( x \) and rotational speed, \( g(x, n) \).

\[
\lambda, \eta(\pi, n, x) = f(\pi, n) \cdot g(x, n), \quad \text{with } g(x = 1, n) = 1. \tag{4}
\]

Measurements of the influence of the swash plate angle are not always available, and due to this separation it is still possible to derive efficiencies for the full load case. A more detailed description of the used approach is described in [6].
3.4. Accumulator

The section starts with the description of the experiments conducted to demonstrate the accumulator behaviour as a function of mass flow rate, vapour quality and diameter of the oil bleed hole. Subsequent sections introduce a new characteristic diagram accumulator model with the relevant model equations. The principal physical phenomena of the separation process are described, comparing measuring results of measurements made with a glass receiver.

3.4.1. Test Rig Measurements

In order to describe the steady state behaviour of the accumulator as a function of charge, measurements were carried out on a special test bench. The following section presents the test bench and the results.

Measurement results from three different oil bleed hole diameters are presented and compared with measurements using the same separator design implemented in a glass cylinder.

![Figure 2. Test rig setup for accumulator measurement](image)

The purpose of the measurements was to acquire data for an accumulator model that is based on a mathematical characteristic function. Consequently, measurements were made to characterize the relationship between the charge in the accumulator and the inlet vapour quality at different refrigerant mass flow rates. A measurement configuration discussed by Raiser [8] for the purpose of determining the liquid level in low-pressure accumulators with carbon dioxide as refrigerant was used (Figure 2). In the experimental setup the charge is determined indirectly by weighing the accumulator. Therefore the accumulator is installed in the cycle in such a way that its weight is not affected by the flexible refrigerant tubes of the refrigerant in- and outlet. To prevent any heat exchange with the environment, the accumulator, refrigerant hoses and tubes are all fitted with insulation material from the measurement points upstream of the expansion valve to the measurement points at the refrigerant outlet, so that all these components can be treated as adiabatic.
The CO₂ mass in the accumulator was determined for a constant suction and discharge pressure of 40 bar and 100 bar respectively, a range of refrigerant mass flow rates (10g/s, 20g/s, 30g/s and 40g/s) as well as an oil bleed hole diameter variation (0.8 mm, 1.0 mm and 1.2 mm). For the experiments a deflector type separator was used in the accumulator, which is shown in Figure 3.

The measurements were performed varying the gas cooler outlet temperature yielding in a variation of the accumulator outlet quality. For the tested accumulator, as seen in Figure 3, the separation characteristic for mass flow rates bigger than 20 g/s the separation efficiency is high (x > 0.93).

Therefore the outlet vapour changes only slightly within a charge range of 50 g to 300 g. Only for small refrigerant mass flow rates, the separation efficiency deteriorates, which is assumed to be due to a bigger fraction of the total mass flow being siphoned through the oil bleed hole. Notwithstanding the results presented by Raiser [8] the separation efficiency of the accumulator is not a smooth function of the mass flow rate. To investigate the flow conditions inside the accumulator in detail, the internal construction of the accumulator was placed into a glass cylinder which was then attached between two blank flanges.

Figure 4 shows the experimental setup and two sample photos taken at mass flow rates of 10 g/s and 30 g/s.

The photos suggest that the decrease in separation efficiency at higher mass flow rates is due to an increasing turbulence of the phase interface, causing droplets to be dragged out with the flow through the outlet pipe. This effect seems to be responsible for the fact that a higher maximum amount of refrigerant can be stored in the accumulator with decreasing mass flow rate. Due to less droplet fog at lower mass flow rates the separation mechanism is flooded only at a higher accumulator mass.
3.4.2. Accumulator Model

The accumulator model presented in this paper is based on the standard separator model implemented in the AirConditioning library. Deviant from the AirConditioning model the accumulator outlet quality is defined by a characteristic diagram correlating the accumulator mass with the outlet vapour fraction.

The characteristic diagram is derived from the measurements described above. The following function was found to be most suitable to fit the accumulator mass as a function of the outlet vapour fraction over the whole mass flow range

\[
m_{\text{accumulator}} = -P_1 \cdot \arctan(P_2 \cdot (x_{\text{outlet}} - P_3)) + (P_1 + 100)
\]  

(5)

whereas parameter \( P_1 \) represents the maximum accumulator mass, \( P_2 \) the separation variance and \( P_3 \) the separation efficiency. The fit of the measuring data with the described function is plotted in figure 3. To implement the mass flow rate dependency, the three parameters are fitted as a function of the mass flow rate.

4. CYCLE SIMULATIONS

4.1. Steady State Simulation

An accumulator sizing was performed using Modelica/AirConditioning. The overall size of an accumulator is typically dictated by four effects: 1. the amount of refrigerant that needs to be stored under different operating conditions 2. the space required to provide adequate separation of liquid and gas phase 3. the space for the volume taken by the internal components and 4. a reserve to adjust for leakage.

The overall dimensions of the cycle components are designed to fit in a FORD KA. The models for the heat exchangers are standard models from the AirConditioning library, the compressor was modelled with a characteristic diagram fitted to measuring results. The used accumulator model is the one presented above. Following Finlayson [9] the accumulator was sized by comparing the refrigerant level in the accumulator at low evaporator heat load conditions and high evaporator heat load conditions.
Figure 5 shows the simulation results of these extreme conditions. The difference in stored refrigerant mass in the accumulator equals 160 g (312 cm$^3$). To account for losses due to leakage (assumed to be 25 g/a) and guarantee an operating period of 5 years without service, an extra of 125 g (63 cm$^3$) has to be added. This yields an accumulator with an internal volume of 450 cm$^3$.

Figure 5. Simulation results: full and empty accumulator

4.2. Transient Simulation

Figure 6 shows an example of a transient cycle simulation. The compressor speed was varied similar to the New European Driving Cycle (NEDC).

The calculated cooling capacity is plotted using the secondary ordinate (right). As a result of an increase of the compressor speed the cooling capacity rises (e.g. $t=140...150$ s). The adjacent interval of constant compressor speed ($t=150...175$ s) shows a decreasing cooling capacity mainly due to the heat capacity of the gas cooler.

Figure 6. Transient Simulation of the cooling capacity
5. CONCLUSIONS

Modelica/Dymola and AirConditioning are appropriate tools for the steady state and transient simulation of mobile air conditioning systems. Examples for both applications have been shown. The modelling of the main components was discussed in the paper. As the accumulator has an important impact on the system behaviour the used model was described in detail. Comprehensive measurements as well as pictures of a glass accumulator in operation were presented.

REFERENCES

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