ANALYSIS OF SUPPLEMENTARY HEATING SYSTEMS FOR AUTOMOBILES BASED ON A R744 AIR-CONDITIONING SYSTEM

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ABSTRACT

A fast warm-up of the passenger compartment is a substantial comfort and safety criterion for automobiles. However, modern high-efficient engines often do not provide enough waste heat at low outside temperatures for a fast heating of the interior. Thus the importance of supplementary heating devices rises. In this context an air conditioning system with the refrigerant R744 offers an efficient possibility. The supplementary heating system can be realized as a simple triangle process or a more complex heat pump. As heat source for the heat pump both the ambient air and the engine cooling cycle are applicable.

The numerical simulation of steady state and transient operating conditions can be a helpful tool for the design and optimization of the thermal management of an automobile. For the transient behaviour the thermal masses of the material and thus the warm-up of the components play an elementary role. In this case the use of simulation models that consider the thermal capacities is necessary.

This paper describes the analysis of different set-ups of a R744 circuit for supplementary heating. Experimental investigations built the framework for the verification of the simulation results and a comparison of an air/air heat pump, a coolant/air heat pump and a hot gas cycle. The systems have been investigated with respect to the system performance, air temperatures and heating capacity. Special attention is drawn to the simulation of the refrigerant cycle at transient operating conditions.

1 INTRODUCTION

In the last years great progresses have been achieved in the design of efficient engines for automobiles. However, the better the engines become with respect to fuel consumption, the less waste heat is available for the warm-up of the passenger compartment. According to Renner (2002) a car with a fuel consumption of 3 litre with the model year of 2000 has only 22% of the available energy for heating compared to a car of the year 1991. Nevertheless a fast warm-up of the passenger cabin is desirable for the comfort of the passenger. Thus the use of supplementary heaters is getting more widespread. For this application R744 (CO_2) heat pump systems have been studied by several authors, e.g. Feuerecker *et al.* (2005), Vetter and Memory (2003), Hünemörder and Kakehashi (2003), Heckt (2004). It seems that the refrigerant R744 is well suited for heating purpose.

The analysis of steady state conditions is of special interest for the design and optimization of automotive air conditioning systems. Within this field the numerical simulation can make a contribution to the reduction of the development times (Schneider *et al.*, 2005). Cullimore and Hendricks (2001) state that in the area of simulation the consideration of the dynamic (transient) behaviour gains in importance. However, up to now only a few publications on transient simulations of the refrigerant cycle of an automotive air conditioning system can be found in the open literature, e.g. Cullimore and Hendricks (2001), Rasmussen *et al.* (2002), Pfafferott (2005) and Adiprasito (1998), but all these publications focus on the cooling mode.

To obtain reliable results the simulation models have to be verified by means of experimental data. Therefore a test facility has been constructed at Graz University of Technology. This allows steady state and dynamic measurements of the refrigerating cycle and its components. A detailed description of the test rig, including the design of the components and further information on the development of components models at the research company "The Virtual Vehicle" can be found in Martin *et al*. (2005).

2 BASICS

A system with limited complexity for a heating set-up is the triangle process or often called hot gas cycle (HGC) (see fig. 1-left). The compressor sucks vapour and compresses the refrigerant to a high pressure (1 → 2). After heat losses (2 → 3) and expansion to a low pressure (3 → 4), the refrigerant is cooled down by the air in the interior heat exchanger (IHX) (4 → 1) and enters the compressor again.

The process of a heat pump is displayed in figure 1-right in the temperature/enthalpy-diagram. There is no major difference in the process whether the heat source is the ambient air or the engine coolant. It proceeds in the following way: the compressor sucks vapour and compresses it to the high pressure (1 → 2). After heat losses (2 → 3) and the heat rejection in the interior heat exchanger (IHX) (3 → 5), the refrigerant is expanded to the low pressure and enters the two-phase region (5 → 6). In the exterior heat exchanger (EHX, either the AC-gascooler or an additional coolant/CO_2-heat exchanger) the refrigerant absorbs heat and evaporates (6 → 1). Afterwards it is fed to the compressor again.

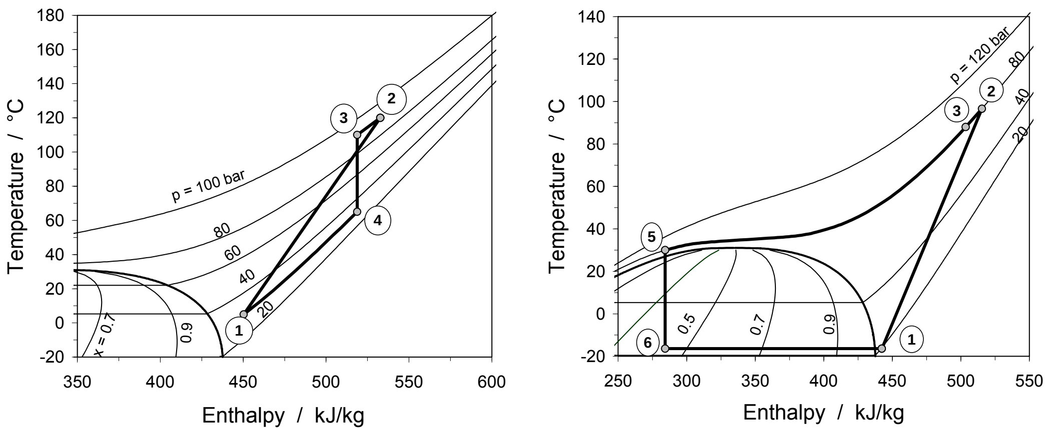

Figure 1: Schematic process of R744 hot gas cycle (left) and R744 heat pump cycle (right)

For the simulation of refrigerant cycles various tools with different modelling depth are available. The calculations presented in this publication, were conducted with KULI (2005), a software tool of the project partner Magna Powertrain. This 1D simulation program allows not only the calculation of refrigerant cycles, but also the simulation of other thermal cycles in vehicles like coolant, oil and charge air cycle. Figure 2 shows a simplified KULI model of the refrigerant cycle (air/air heat pump) using basic components as well as enhanced component models which are linked to the model as external components. For the presented simulations the air/refrigerant heat exchangers were modelled according to geometry data, the models for the compressor and the refrigerant/coolant heat exchanger are based on characteristic curves since no detailed models are available up to now.

In order to be able to simulate the transient behaviour of the system, the thermal inertia of the components must be considered. Figure 3 shows how the refrigerant temperature at the outlet of a component is influenced by its mass. After a temperature change at the inlet of the component, the outlet temperature follows with some delay because of the slow heating up of the component. The time until the outlet temperature reaches a steady state condition is mainly influenced by the thermal inertia of the component and the heat transfer coefficient between the refrigerant and the wall. In the figure, mass1 is about 0.1 kg and mass2 is three times higher and thus the heat capacity of the component is also three times higher.

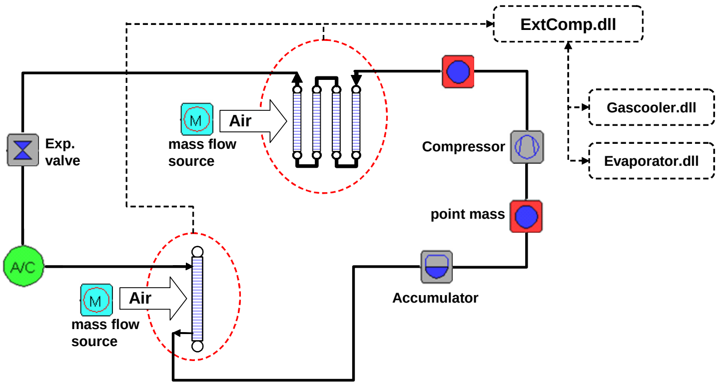

Figure 2: Schematic of a KULI model with implementation of external components

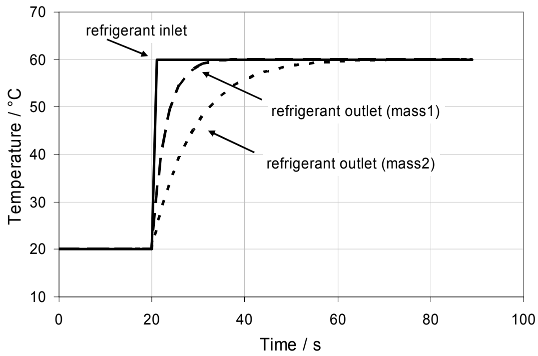

Figure 3: Influence of the components mass on refrigerant outlet temperature

3 STEADY STATE INVESTIGATIONS

In this chapter an air/air heat pump, a coolant/air heat pump and a hot gas cycle are being compared by means of experiments and simulation. The systems are investigated with respect to capacity, air temperatures and COP for steady state operating conditions. The simulation of the transient behaviour is discussed in detail in the following chapter.

All experiments were conducted with a compressor speed of 900 rpm at an ambient temperature of -5°C and a relative humidity of 90 % (about 2 g_W/kg_{dA}). The test series were carried out with different air volume flow rates over the interior heat exchanger. The compressor was operated with maximum stroke (33 cm^3) during all tests. Furthermore the cross section of the expansion valve was kept constant throughout the entire experimental investigations, even though the heating capacity is mainly influenced by the control of the expansion valve (Feuerecker *et al.*, 2005). Some analysis of the influence of the expansion valve cross section for the heat pump mode can be found in Martin *et al.* (2006). For the investigations of the air/air heat pump the air volume flow rate over the exterior heat exchanger was kept constant at a value of 1500 m^3/h; the coolant/air heat pump operated with a coolant mass flow of 730 kg/h and a constant coolant inlet temperature of -5°C.

Figure 4-left shows the obtained heating capacities and a comparison of measurement and simulation. As expected the maximum capacity is achieved at the maximum air volume flow rate.

The highest heating capacity of 4.2 kW can be reached with the coolant/air heat pump (HPCA). However, the operating point with the highest heating capacity will not be optimal in the vehicle, although the COP is high (see fig. 5), because the obtainable air temperature at the interior heat exchanger outlet and thus at the inlet into the passenger compartment is rather low (see fig. 4-right), e. g. at a flow rate of 500 m³/h the air outlet temperature is lower than 20°C. Substantially higher temperatures can be obtained with lower air flow rates. At 100 m³/h an air outlet temperature of 60°C can be achieved with the heat pump systems. Under this condition a heating capacity of approximately 2.5 kW can be obtained.

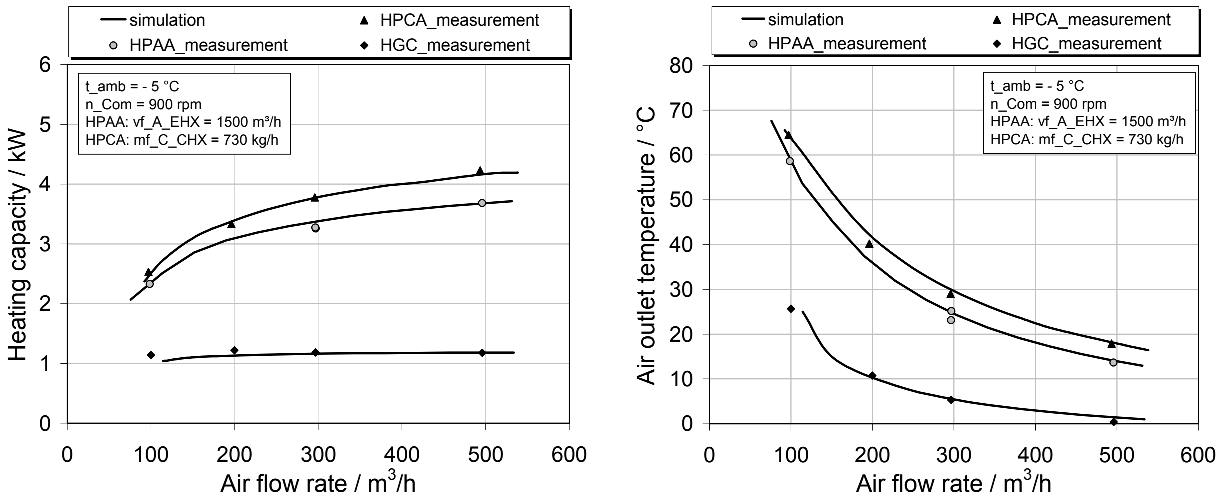

Figure 4: Heating capacity (left) and air outlet temperature (right) vs. IHX air volume flow rate (measurement vs. simulation)

The heating capacity of the hot gas cycle (HGC) depends strongly on the driving power of the compressor. With the used compressor at 900 rpm a heating capacity of 1.1 kW was obtained (see fig. 4-left). The necessary driving power was 1.3 kW, this leads to a COP of 0.88 (see fig. 5). It shows up that the achievable heating capacity is nearly independent of the air volume flow rate over the interior heat exchanger in a wide range. The compressor driving power and thus the COP is also nearly constant since the suction and discharge pressure are on an almost constant level at the investigated air flow rates.

The obtained COPs for the different systems are shown in figure 5. For all systems the maximum COP appears with the highest air flow rates. For all investigated operating conditions the highest COP could be reached with the coolant/air heat pump. The maximum is almost 3.5 at an air flow rate of 500 m³/h, nevertheless the COP of the air/air heat pump is just somewhat lower. However Schäfer *et al.* (2003) have concluded that for a system with a 1.9 l TDI engine the COP of a heat pump with coolant as heat source should be in the range of 1.5 to 1.8 to ensure that the engine coolant does warm-up within the same period of time as the basis system without a heat pump, where no heat is rejected from the coolant cycle. This COP can be reached in the region between 100 and 150 m³/h where also convenient air temperatures were achieved.

It should be noticed here, that the heat losses on the high pressure side, which are on a high temperature level, can not be neglected and therefore have to be considered within the simulations.

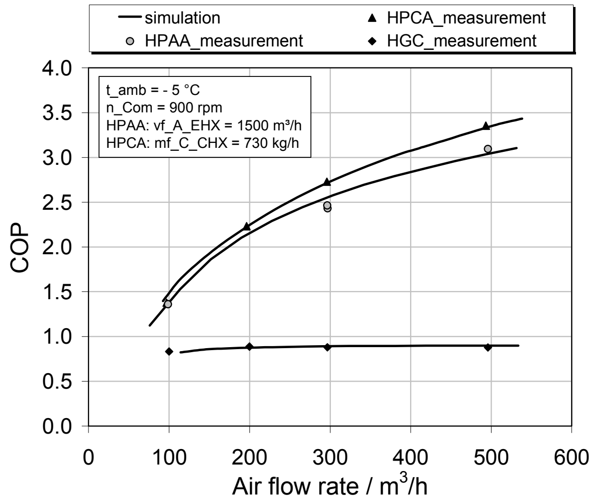

Figure 5: COP vs. IHX air volume flow rate (measurement vs. simulation)

A comparison of measurements and simulation results shows that the model provides reliable results for a wide range of operating conditions (compare fig. 4 and 5). The heating capacity was computed for most points with an error smaller than 5 %. Therefore the air temperature at the outlet of the interior heat exchanger could be calculated with high accuracy as well. Hence the model can be used for further steady state calculations.

Although freezing of the exterior heat exchanger (evaporator) in the air/air heat pump mode is not the main issue of the current work, it should be mentioned here that frosting was not a big issue for operating points with rather low air flow rates over the exterior heat exchanger and low heating capacities. But it can be significant when the air flow rate is enlarged and the heating capacity rises, or at other ambient conditions.

4 TRANSIENT INVESTIGATIONS

In this chapter the transient behaviour of the refrigerant cycle will be investigated in detail. In this context quick changes of the compressor speed, as they occur during the "New European Driving Cycle" (NEDC), are a matter of particular interest.

AIR/AIR HEAT PUMP

In the following the transient behaviour of an air/air heat pump is analysed. During these investigations the compressor speed was varied within short periods of time according to the first minutes of the NEDC (see fig. 6). The air volume flow rates over the heat exchangers were kept constant at 300 m³/h (IHX) and 1500 m³/h (EHX) at a temperature of -5°C and a relative humidity of 90%. The investigations were conducted with a cold start.

First of all a comparison of measurement and a quasi steady state simulation (i.e. for each time step a new steady state calculation is performed) should be discussed for the obtained heating capacity. The thermal inertia of the masses plays such a crucial role that the quasi steady state simulation does not deliver satisfying results compared to the measurements (see fig. 7). The damping, which is caused by the thermal masses of the components, is of essential relevance not only during the warm-up of the system within the first minutes, but also after a longer period of time.

Figure 7 also shows the results of a transient simulation with consideration of the thermal capacities by means of point masses. Thereby the computed air outlet temperature downstream of the interior heat exchanger deviates by less than 2 K from the measured values during the whole test. Even the fast load changes, i.e. the rising and falling flanks, are accurately considered by the simulation.

The air temperature downstream the interior heat exchanger is not shown here, but they are directly coupled to the heating capacity since the air flow rate was kept constant during the test. As can be seen in figure 7 the heating capacity varies between 2 and 5 kW which corresponds to a variation of about 30 K of the air temperature for the investigated conditions. Of course this will not be very suitable for the application in the car since this variation in temperature will be noticed in the passenger compartment. Therefore adequate control strategies have to implemented, e.g. a variable displacement of the compressor.

Again freezing of the exterior heat exchanger was not a big issue within these investigations since the air flow rate over the exterior heat exchanger was rather low and the period of time was not that long. However freezing could be a problem of the air/air heat pump that will be avoided with the coolant heat pump.

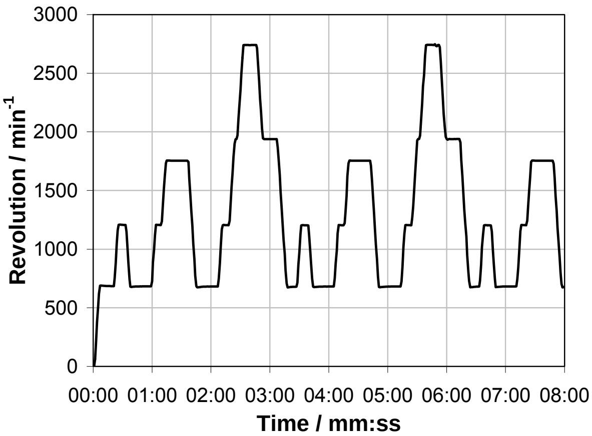

Figure 6: Variation of compressor speed depending on time

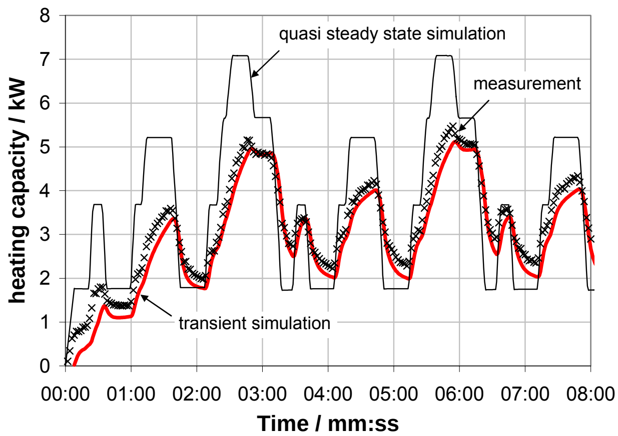

Figure 7: Comparison of quasi steady state and transient simulation with measurement

HOT GAS CYCLE

Further transient simulations were conducted for the hot gas cycle. Some investigations concerning the warm-up behaviour from a cold start and the verification of the simulation model can be found in Martin *et al*. (2006).

Within this paper a simulation of a load change at an ambient temperature of -5°C is shown (see figure 8). For these investigations the initial point was a steady state condition with a compressor speed of 600 rpm and an air mass flow rate over the interior heat exchanger of 500 kg/h (operating point "a"). At the time step 01:00 min the compressor speed was set to 1200 rpm, the other boundary conditions were kept constant. Figure 8 shows the refrigerant inlet and air outlet temperature at the interior heat exchanger as well as the corresponding refrigerant cycles in the t/h-diagram for three different operating points.

For this load change an interesting behaviour shows up, which is caused by the thermal inertia of the components: The air is cooled down in the first seconds (operating point "b") and it takes about 15 seconds before the air is heated up. This behaviour can be explained by the fact that during this phase the refrigerant is compressed to a high pressure quite fast, but the components on the high pressure side store much energy and the inlet temperature of the refrigerant into the expansion valve is therefore rather low. Thus the outlet temperature of the expansion valve and as a result the inlet temperature into the interior heat exchanger is even lower than the ambient temperature and therefore the refrigerant absorbs heat from the supply air. The effect becomes enforced by heat losses on the high pressure side. The systems needs about 5 minutes until it reaches a steady state condition again (operating point "c"). It should be stated here, that this behaviour is not only of theoretical interest, but has also been observed within experimental tests.

For the operation of the hot gas cycle in a car it has to be avoided that the air cools down for a while since this is not very comfortable for the passenger, of course. In order to avoid this, the masses on the high pressure side may not be too large and their thermal inertias have to be considered within the design of the system.

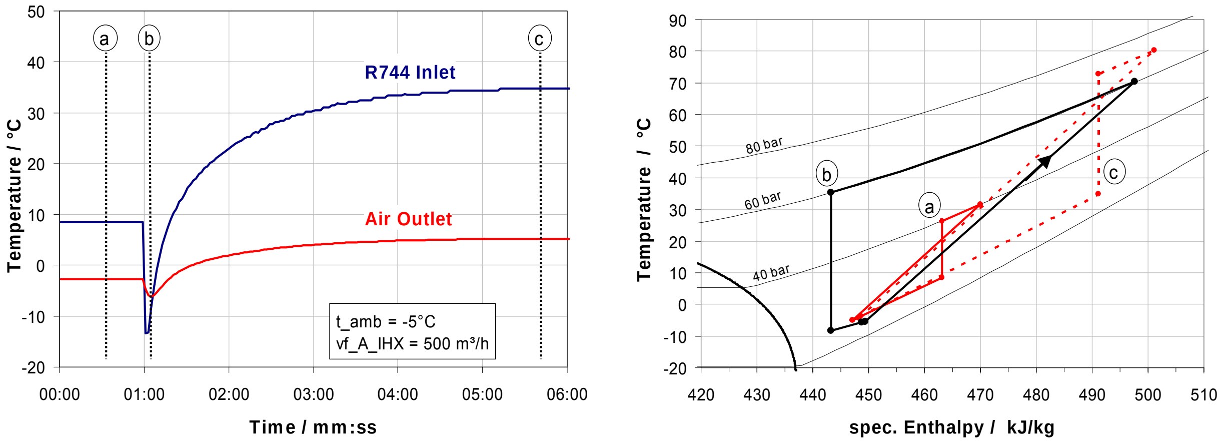

Figure 8: Simulation of a load change for the HGC (n_Com from 600 to 1200 rpm)
Temperatures at Interior HX (left) and corresponding process in t/h-diagram (right)

5 SUMMARY AND OUTLOOK

The on-going development of high-efficient engines leads to a heating capacity deficit of the vehicles. Thus the use of supplementary heaters gains in interest. A promising solution is the use of the air conditioning system with the refrigerant R744 either as a hot gas cycle or as a heat pump. A comparison of an air/air heat pump, a coolant/air heat pump and a hot gas cycle at a compressor speed of 900 rpm shows the fact that the heat pump systems are far superior to the hot gas cycle since the COP of the hot gas cycle is always lower than one. With the heat pump systems COPs higher than 3 can be reached at high air flow rates over the interior heat exchanger. However these conditions seem not to be favourable for the application in the car, since the air inlet temperature into the passenger compartment will be rather low. To exceed an air temperature of 40°C the air flow rate must be lower than 200 m³/h for the heat pump systems investigated. With the hot gas cycle such temperatures can not be obtained at a compressor speed of 900 rpm.

An emphasis of the investigations is set on the simulation of the transient behaviour of the refrigerant cycle. It is shown that the thermal masses of the components play a considerable role and therefore a quasi steady state simulation does not provide satisfying results for the investigated driving cycle. The developed methods have been verified on the basis of measurements and deliver reliable results in the comparison with the measured data. Even at fast load changes within a few seconds, as they occur e.g. during a driving cycle, the air temperatures can be predicted with an uncertainty less than 2 K.

An interesting effect at the operation of the hot gas cycle was shown by means of a transient simulation. During a load change, e.g. variation of the compressor speed, the air can be cooled down below the ambient temperature for a while, if the thermal masses on the high pressure side are too large.

For the future the entire thermal management of the vehicle should be investigated; the effect of the cooling/heating system on the vehicle will be analyzed both experimentally and by means of simulation.

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