

REVERSIBLE MOBILE A/C SYSTEM

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Abstract: 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 a quick heat-up of the passenger compartment. This makes supplementary heaters necessary for a fast warm-up at low ambient temperatures. For this purpose the utilization of the R744 refrigerant cycle seems to be a promising solution. This can be realized either by reversing the refrigerant process to a heat pump cycle with the coolant or the ambient air as the heat source, or by a quite simple hot gas cycle.

This paper describes the analysis of a refrigerant cycle that can be used for cooling and heating by switching the conventional refrigerant cycle to an air/air heat pump, a coolant/air heat pump or a hot gas cycle. The different systems have been investigated by means of experiments and numerical simulations. All systems are based on the R744 refrigerant circuit of the air conditioning system and they are therefore suitable for heating and cooling of the passenger cabin.

It has shown that rather large heating capacities of more than 2.4 kW can be achieved with the heat pump systems even at unfavourable operating conditions (low compressor speed and low air flow rate). For the hot gas cycle a maximum heating capacity of 1.1 kW was obtained at the investigated conditions.

Key Words: *mobile air conditioning, R744, hot gas cycle, heat pump*

1 INTRODUCTION

The motivation of this study is based on two challenges:

- The need of supplementary heating for automobiles:
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 litres with the model year of 2000 has only 22% of the available energy for heating compared to an average 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.
- The potential use of the refrigerant R744 in mobile air conditioning systems:
According to the Kyoto protocol and consequent regulations in the EU, the automotive industry is now in the process of replacing the presently used refrigerant

R134a. The German automotive OEMs have announced to use the natural refrigerant R744 as the substitute (VDA, 2007).

Compared to the refrigerant R134a, R744 is characterized by the following fluid properties:

- high saturation pressure, density and volumetric refrigeration capacity even at low temperature (-20°C or lower)
- temperature glide of refrigerant at supercritical conditions

These properties are especially favourable for heat pump systems and therefore R744 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) which all show a great potential for R744 heating systems.

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. 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

This chapter describes the basics of this study, i.e. the design of the different systems that were investigated (for cooling and heating) and the fundamentals of the used simulation tool.

2.1 System Design

Cooling mode (AC)

The investigated system for the cooling mode was a "conventional" R744 system as it is described by several authors, e. g. Kampf et al. 2002. The flow scheme of the refrigerant can be derived from Figure 1-left: The compressor sucks superheated vapor from the internal heat exchanger, compresses it to the high pressure ($1 \rightarrow 2$). Afterwards the refrigerant is directed to the gas cooler by means of a switching valve. After heat rejection at a gliding temperature in the gas cooler ($2 \rightarrow 6$) the refrigerant is further cooled down in the suction line heat exchanger ($6 \rightarrow 7$) in counter flow with the low-pressure refrigerant from the evaporator. When the high pressure refrigerant passes the throttling device into the low pressure part of the circuit ($7 \rightarrow 5$), it enters the two-phase region. In the evaporator ($5 \rightarrow 3$) the refrigerant absorbs heat from the ambient air and evaporates. After passing the switching valve and the accumulator, the refrigerant is superheated in the low-pressure side of the internal heat exchanger and fed to the compressor again ($6 \rightarrow 1$).

Heating mode

The process of a heat pump is displayed in Figure 1. 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 ($1 \rightarrow 2$). The switching valve then leads the refrigerant directly to the indoor heat exchanger (IHx). After heat losses ($2 \rightarrow 3$) the heat is rejected from the refrigerant in the indoor heat exchanger ($3 \rightarrow 5$) and thereby heats up the air flow. Subsequently the refrigerant is

expanded to the low pressure and enters the two-phase region (5 → 6). In the exterior heat exchanger (EHX, either the A/C-gascooler or an additional coolant/R744-heat exchanger) the refrigerant absorbs heat and evaporates (6 → 1). Afterwards it is fed to the compressor again.

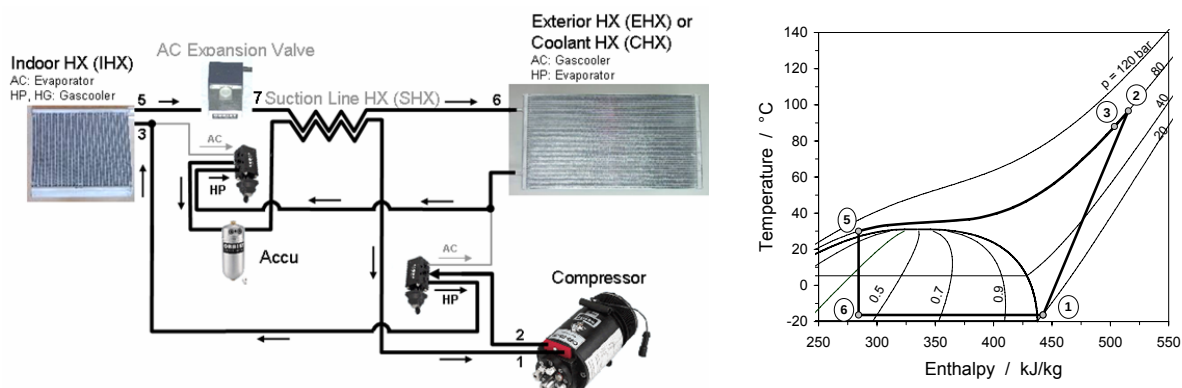


Figure 1: Flow sheet of reversible R774 cooling/heat pumping system (left) and process of R744 heat pump cycle in t/h-diagram (right)

A system with limited complexity for a heating set-up is the triangle process or often called hot gas cycle (HGC) (see Figure 2). 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 indoor heat exchanger (IHx, this is the evaporator in A/C mode) (4 → 1) and enters the compressor again.

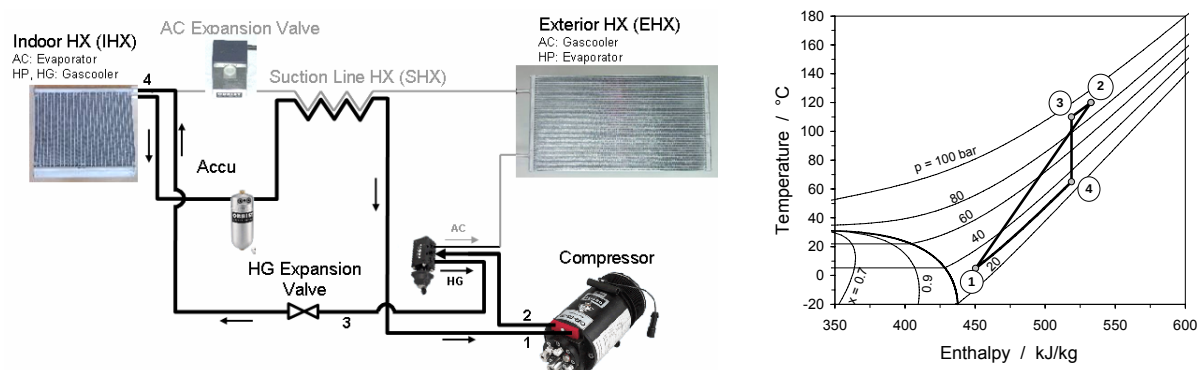


Figure 2: Flow sheet of reversible R774 cooling/hot gas cycle system (left) and process of R744 hot gas cycle in t/h-diagram (right)

For all the described investigations a variable displacement compressor with a maximum swept volume of 33 cm³ was used. It is equipped with a control unit for air conditioning and heat pump applications to control the temperature after the indoor heat exchanger, i. e. the evaporator in cooling mode. However the compressor has been operated with the maximum swept volume during the entire tests presented in this paper.

The expansion valve has been a bidirectional needle valve and is therefore suitable for both A/C and heat pump application. It can be used to control the high pressure by varying the cross section. The accumulator is also applicable for heat pump systems and includes an oil management system based on an u-tube design.

2.2 Simulation Tool

The differences between R744 and R134a and the new challenge of designing a system for heating and cooling demands the development of new models for simulation tools. In the following paragraphs simulations for steady state conditions of an A/C-System with R744 as refrigerant are shown. The results are compared to experiments conducted with the described systems. The simulations have been carried out with the program KULI (2005). This software offers the possibility to model refrigerant circuits for several working fluids, including R744 and is based on the one dimensional heat transfer between refrigerant and single-phase fluids like humid air or coolants.

Figure 3 shows the principal simulation models that were used for the investigations in this paper. They mainly consist of the same components as the test rig. These are compressor, exterior heat exchanger, suction line heat exchanger (SHX-LP and SHX-HP), expansion valve, indoor heat exchanger and accumulator. The tubes and bends are not considered, as long as their influence on the entire cycle is not of notable relevance for this simulation.

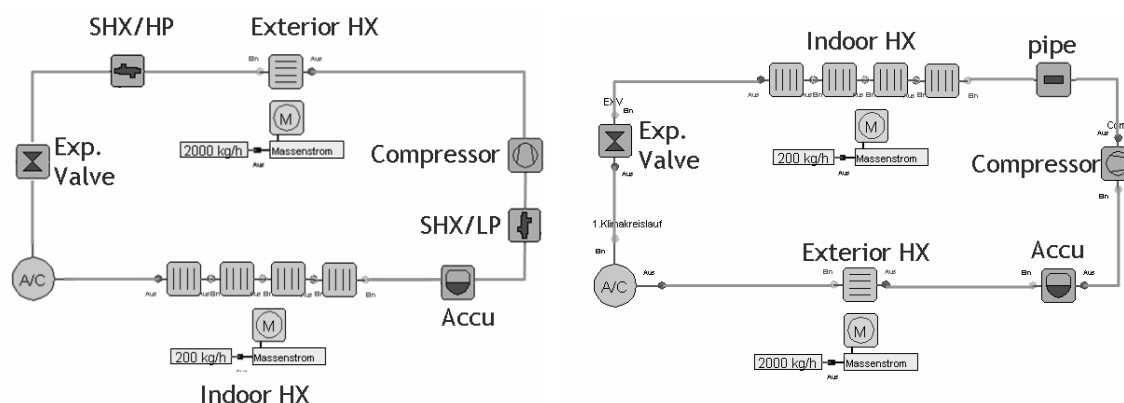


Figure 3: Simulation-Model for a/c mode (left) and air/air heat pump (right)

The model of the compressor is essentially described by experimentally determined characteristic curves. For the expansion valve an orifice model with a fixed throttle cross section was used. The cross section was varied to adjust a certain high pressure.

The accumulator model enables to predict the behavior of a refrigerant accumulator which is placed downstream of the evaporator. Usually the discharge u-tube has a hole at the bottom in the u-turn section. The diameter of this hole determines a certain value of refrigerant vapor quality at the accumulator outlet, which is needed for the oil return to the compressor. One can enter the geometric data and then the outlet vapor quality is predicted by calculating the pressure differences and drops in the various sections and tubes.

The heat exchanger models are based on commonly used equations, mostly from the VDI-Wärmeatlas (1994). Therefore the inputs provide a detailed description of the heat exchangers geometry and their layout. In this case the heat exchanger is modeled according to the heat exchangers geometry installed in the test rig, i.e. the indoor heat exchanger consists of four layers and the exterior heat exchanger of one. Furthermore other geometric data, like number of tubes and passes as well as the inner geometry on the refrigerant side are considered by the model.

3 RESULTS IN COOLING MODE

This chapter shows steady state simulation results conducted with the simulations models described before in comparison to measurement results.

Throughout the experiments and calculations shown in Figure 4 the cross section of the orifice was varied to control the compressor outlet pressure. For these tests the ambient temperature was 35°C, the compressor speed 900 rpm (representing idle conditions), and the air volume flow rates were 500 m³/h for the evaporator and 3000 m³/h for the gascooler, respectively.

While the driving power increases almost linearly with rising high-side pressure the cooling capacity increases on a diminishing scale. However, the COP shows a maximum at a pressure of about 95 bar, but this optimum could not be predicted exactly by the simulation.

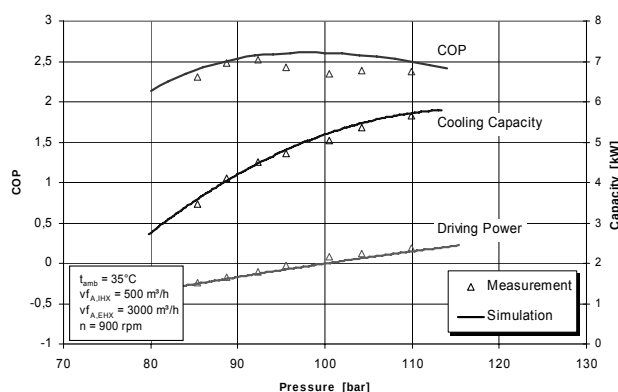


Figure 4: Cooling capacity, COP and driving power against high pressure

However, it can be concluded that the simulation results show a good agreement with the experimental data, although deviations exist in some operation points. Nevertheless the model is suited for steady state simulations of the investigated R744 system, but of course there is potential for optimization. For instance experimental data for some models is required and the capabilities for simulating transient phenomena are limited. Therefore more advanced component models are under development.

4 RESULTS IN HEATING MODE

In this chapter different heating systems are being compared by means of experiments and simulation. These are an air/air heat pump, a coolant/air heat pump and a hot gas cycle. The systems are investigated with respect to capacity, air temperatures and COP for steady state operating conditions.

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 indoor heat exchanger. The compressor was operated with maximum stroke (33 cm³) 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 m³/h; the coolant/air heat pump was operated with a coolant mass flow rate of 730 kg/h and a constant coolant inlet temperature of -5°C.

Figure 5-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 Figure 6) because the obtainable air temperature at the indoor heat exchanger outlet and thus at the inlet into the passenger compartment is rather low (see Figure 5-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.

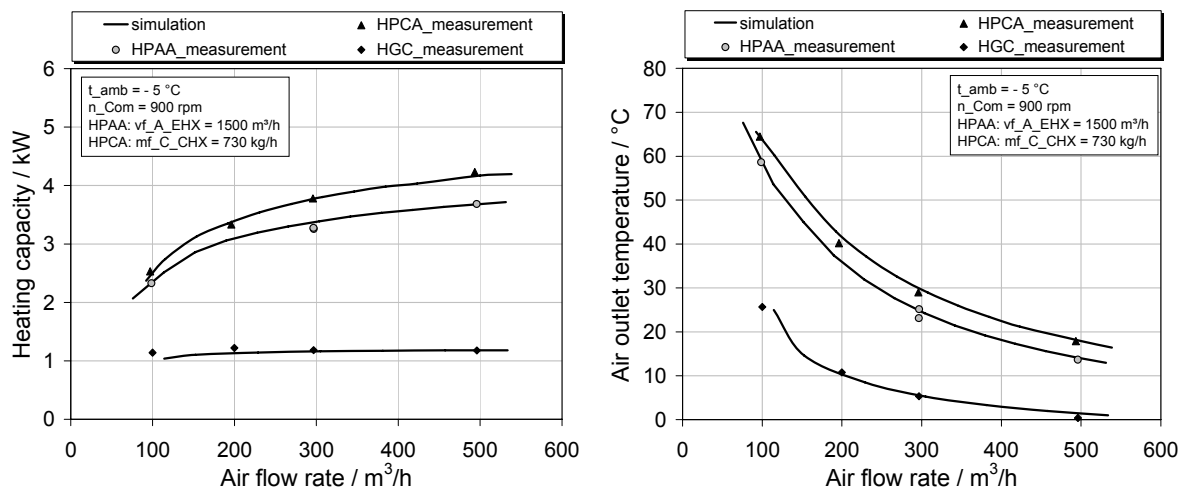


Figure 5: Heating capacity vs. IHX air volume flow rate (left) and Air outlet temperature vs. IHX air volume flow rate (right)

The heating capacity of the hot gas cycle (HGC) strongly depends on the driving power of the compressor. With the used compressor at 900 rpm a heating capacity of 1.1 kW was obtained (see Figure 5). The necessary driving power was 1.3 kW, this leads to a COP of 0.88 (see Figure 6). It shows up that the achievable heating capacity is nearly independent of the air volume flow rate over the indoor 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 within the investigated air flow rate.

The obtained COPs for the different systems are shown Figure 6. 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 Schaefer 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 are achieved (compare Figure 5).

A comparison of measurements and simulation results shows that the model provides reliable results for a wide range of operating conditions (compare Figure 5 and Figure 6). The heating capacity was computed for most points with an error smaller than 5 %. Therefore the air temperature at the outlet of the indoor heat exchanger could be calculated with high accuracy as well. Hence the model can be used for further steady state calculations. Additional information on the investigation of the heating systems, especially the simulation of the transient behaviour can be found in Martin et al. (2006).

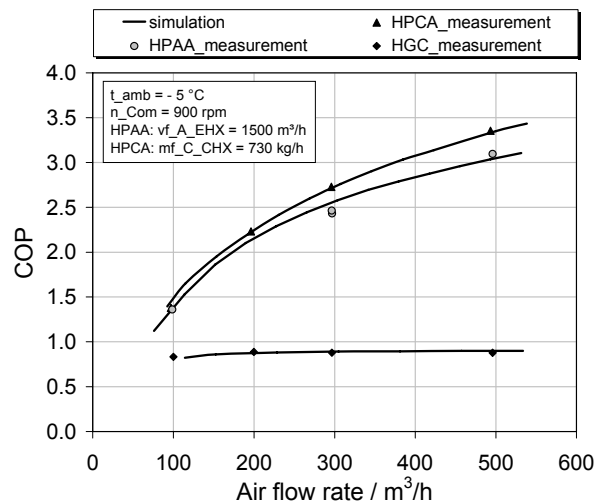


Figure 6: COP vs. IHX air volume flow rate (measurement vs. simulation)

Regarding the air/air heat pump it should be mentioned here that frosting of the exterior heat exchanger (evaporator) was not a big issue for the presented operating points. However, it can be significant when the air flow rate over the exterior heat exchanger is enlarged, or at other ambient conditions.

5 CONCLUSION

R744 seems to be the future refrigerant for automobile A/C systems. Due to the differences compared to R134a a lot of engineering work is necessary. Therefore a test rig for a R744 system analysis was constructed and simulation models for the R744 refrigerant cycle have been tested extensively. It has been shown that the influence of several system parameters like air flow rates and high-side pressure can be predicted with acceptable accuracy over a wide range of operating conditions by means of steady state simulations for both the cooling and the heating mode.

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 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 and heating capacities higher than 3.5 kW can be reached at high air flow rates over the indoor 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 investigated heat pump systems. With the hot gas cycle such temperatures can not be obtained at a compressor speed of 900 rpm.

For future investigations a detailed consideration of the transient behaviour of the refrigerant cycle seems to be of special interest. Furthermore the heating systems should be investigated within the context of the entire thermal management of the vehicle; i. e. the effect of the heating system on the coolant cycle and the engine which is especially caused by the additional load of the compressor. The presented verified simulation models can be used as part of such investigations.

6 ACKNOWLEDGMENTS

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