Controller Development for an Automotive Ac-system using R744 as Refrigerant

Sanaz Karim Hubertus Tummescheit
Modelon AB
Ideon Science Park, SE-22370 Lund, Sweden
Hubertus.Tummescheit@modelon.se, Sanaz.Karim@gmail.com

Abstract

Due to recent regulatory changes in Europe, CO_2 or R744 is considered a serious alternative to be the successor of R134a for the AC-system of cars for the European market. Research into R744 as a working fluid for automotive AC started in the early nineties and continues even today. There are still open issues in both design and control of R744 systems, e.g. the choice of an expansion device that satisfies both cost and performance constraints, control in the subcritical region and controling transcritical transients. In a Masters thesis project organized in cooperation with Daimler AG in Sindelfingen, these issues were investigated using a well validated model of an R744 prototype system modeled using the AirConditioning Library by Modelon AB and Dymola from Dynasim AB. The preferred choice for the expansion device from a cost point of view is a two-stage orifice with a pressure-activated bypass for high load conditions. The solution with the two-stage valve is compared to a reference system that uses an electronically controlled valve that is controlled to the COP-optimal high side pressure. Unfortunately, the two-stage valve can exhibit both limit cycling behaviour and multiple steady states depending on the plant operation history, both undesirable properties. For the investigated system the drawbacks could be eliminated by proper control design. Another problem that was investigated was the load distribution between a front- and a back seat evaporator for a two-evaporator version of the same system. Again for cost reasons, the refrigerant side of the second evaporator is not controlled, instead flow is split between the two evaporators using a fixed expansion device for the rear evaporator.

Keywords: air conditioning; compression cycle; simulation; CO2; R744; control design, COP optimization

1 Introduction

Under the Kyoto protocol agreement, by the year 2012, industrialized countries have to reduce their collective emissions of greenhouse gas 5% below their 1990 levels. Since the current refrigerant used in vehicles, R134a, has a GWP (Global Warming Potential) of 1410, R744 (CO_2) technology has been proposed as a natural alternative to current R134a-based systems. The main benefits of R744 as a refrigerant are:

- Energy-efficient
- Non-toxic
- Non-flammable
- No ozone depletion potential (ODP=0)
- Low global warming potential (GWP=1)

Apart from the environmental benefits listed above, using R744 as a refrigerant for air-conditioning (A/C) systems can decrease the fuel consumption under some climate conditions.
Daimler AG and some of its suppliers have developed and validated specific component and system models for R744-cycles based on the AirConditioning Library by Modelon. These models were used to investigate control strategies for both the single evaporator and the dual evaporator system prototype for an S-class Mercedes.

2 A/C Systems Optimization and Control

The role of the HVAC-unit in the A/C system is to provide maximum cooling power in order to cool down the air and dehumidify it before re-heating and ventilation. To increase cooling power at very high ambient temperature, traditionally a lower COP (more fuel consumption) is accepted. The current practice is to control the air temperature after the evaporator to a constant, low temperature (slightly above 0 Celsius to avoid frost) and control the actual cabin temperature by mixing in warm outside air in the HVAC box to obtain the desired temperature in the cabin. However, most of the operating times the optimization of the COP is the more reasonable control target from the point of view that fuel consumption should be minimized. These two control targets can be fulfilled by inserting two decoupled SISO control loops, one of them controling the high pressure and the other one controling the evaporator outlet temperature by considering the strong crosscoupling between these two variables.

2.1 Optimum High-Pressure Control

To achieve maximum COP in R744 systems, a simple SISO control strategy with two control loops has been proposed by [1]. They consider the high- pressure as the main variable that affects the COP and cooling power. Since the heat rejection process of the R744 refrigeration cycle takes place in the supercritical region, where the pressure is independent of the temperature, the system efficiency is a nonlinear function of the working pressure and the ambient temperature.
For each ambient (gas cooler air inlet) temperature, there is an optimum high-pressure, which results in the maximum COP. With the increase of the ambient temperature, the optimum pressure increases.
The other boundary conditions (evaporator temperature, air humidity and flow rate) have negligible effect on the optimum high-pressure.

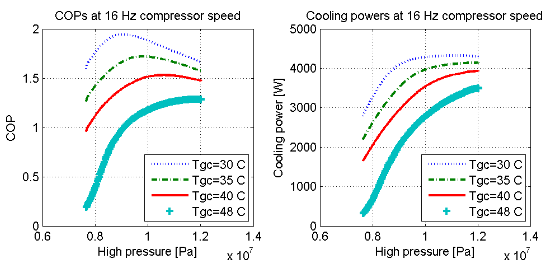

Figure 1: Comparison of COP and cooling power with the change of high-pressure

A variable swash plate controller is used as low-pressure (evaporator air outlet temperature) controller and since any change in a angle of the swash plate will affect the pressure ratio as well as the compressor power; it is expected that it changes the optimum high-pressure as well. Therefore a high-pressure regulator which controls the refrigerant flow, based on the ambient temperature and compressor speed is suggested by [1]. For the purpose of simplification the effect of speed is neglected and the controller is reduced to a controller which works just based on the ambient or gas cooler temperature and is designed at a low speed, since at higher speeds of the compressor the role of the optimum high-pressure is less significant.
An electronic expansion valve like a PWM-valve can be used as an actuator to change the flow rate to achieve desired the high-pressure, but to get rid of the costs of the high-pressure controller and gas cooler temperature measurement device, a two-stage orifice expansion valve has been developed whose internal control mechanism is described in section 3.1.

2.2 Evaporator Temperature Control

Under low load conditions, it is necessary to control the compressor power to reduce the cooling power to the desired range for the A/C system and not let it reach its maximum possible capacity. These conditions are low cooling load and/or high engine speed. Since the compressor of the automotive A/C unit draws its driving force from the engine, its power is a function of the engine speed, which is a highly fluctuating variable. Control of the compressor capacity is necessary to compensate engine speed disturbances, to satisfy the comfort requirements and to avoid temperature variations. Control is particularly important at higher speeds, which cause an undesirable power of the compressor and too low temperature at the evaporator.
Among the various methods proposed to control the compressor capacity, using a variable displacement compressor is the most attractive one. The most popular variable displacement compressor for automotive use today is the swash-plate compessor[1]. Changing the inclination of the swash plate changes the displacement of each of the many pistons of the compressor. This causes a change of the pressure ratio, both high-pressure as well as low-pressure are affected, but the effect of the expansion valve on the high pressure is dominant. The control of the swash plate angle and thus the relative volume is used as a low pressure controller in spite of its influence on the high pressure.
In the sub-critical region, where the heat rejection takes place isothermally, evaporator refrigerant and air outlet temperature are functions of the low-pressure,

[1]For hybrid cars with sufficient electrical power, other options would be advantageous, because they open up the new possibility of using a speed control of the compressor.

thus the swash plate control makes it possible to control the evaporator temperature and via the temperature also the power.
Concerning the previous section, at a constant speed, it is acceptable to neglect the cross coupling between the first SISO loop which tries to maximize COP by high-pressure control and the second one which aims to control the low-pressure (evaporator air outlet temperature), but it is not satisfactory to decouple these loops in the case of speed changes.
Assuming constant speed, control of the evaporator air outlet temperature in the case of low cooling load can improve the COP significantly due to a smaller pressure ratio and consequently smaller power uptake of the compressor.

3 AirConditioning Library

The AirConditioning Library and the simulation tool Dymola, both based on the standardized, freely available modelling language Modelica, have been selected by the German automotive OEM as the preferred tool for model development and exchange for the A/C system in passenger cars. The library contains a complete range of component models and templates of typically used and proposed A/C system architectures and all currently used as well as new and proposed refrigerants for automotive applications. The modeling detail is appropriate for component selection, system architecture design, system integration for overall vehicle thermal management and climate control design. Prototype systems for future technologies often contain components that differ from those needed for conventional designs, but due to the open code and the given modeling infrastructure, it is straightforward to add unusual components to the Library. In this case a two-stage orifice model with a pressure operated bypass had to be added.

3.1 Two-Stage Orifice Model

This valve has an internal mechanism to drastically change its Kv value based on the pressure difference between the low- and high-pressure side [6]. It consists of a standard orifice and a bypass which is closed for small pressure differences. As shown in Figure 2, the refrigerant flows only through the orifice at pressure differences below a pressure difference Δp, in this case set to 73 bar. The bypass starts to open at a rising pressure difference of Δp with a very steep gradient, and for higher pressures, the Kv-value rises almost linearly with the pressure difference. This results in a very non-linear pressure – mass flow characteristic which is prone to limit-cycling behaviour. The cycle is caused by interaction between the dynamics if the mass storage at the high- and low pressure levels in combination with the differences between the mass flow characteristics of the compressor (almost no change for pressure difference above and below Δp) and the valve (almost a step function at Δp). When the rising pressure opens the valve for a pressure difference higher than Δp, the opening bypass will increase the mass flow from the high pressure side so rapidly that the pressure difference falls below the bypass opening limit, because the compressor mass flow does not increase in the same degree and the cycle starts again.

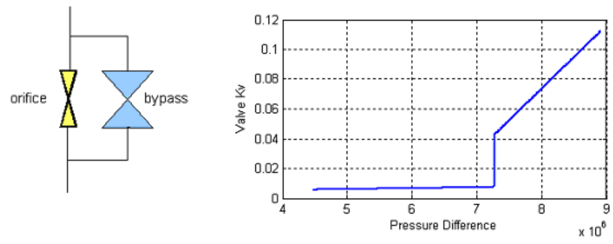

Figure 2: Two stage orifice valve

The highly nonlinear behaviour of the valve's Kv-value can under some situations give rise to limit cycling around the steep part of the characteristic where the valve opens, and it may even lead to two steady states with different COP's, one at a pressure difference above the opening pressure, the other one at a pressure difference below the opening pressure, at identical boundary conditions.

4 Single Evaporator, Two-stage Orifice Valve System Control Design

While no direct control of the high-pressure is possible anymore when using the two-stage orifice valve, it is still desired to keep the COP as close as possible to its optimal value in order to reduce fuel consumption. As previously mentioned, the first control target remains to regulate the evaporator temperature by means of the compressor relative volume control, the COP control is of secondary importance. To achieve these goals, a simplified control structure proposed by [4] was used as a starting point for the control design. That structure was developed for the same type of two-stage orifice valve and used a complex feed-forward map with three

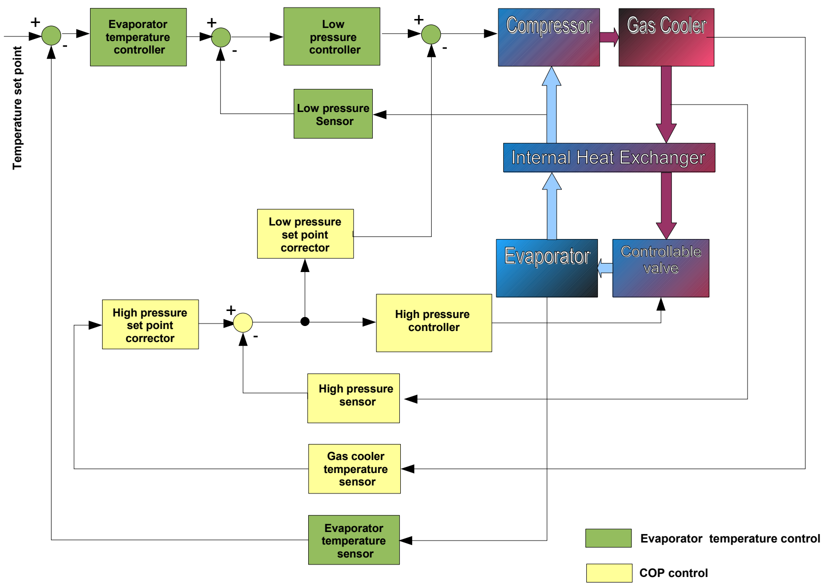

Figure 3: Control structure for R744 AC-cycle with electronically controllable expansion device, assuming COP-optimal control via the valve to control the high pressure side and temperature/power control via the compressor to control the low pressure side.

inputs (engine speed, air mass flow and inlet air temperature) to mimic the optimal high pressure control with a fully controllable valve. There are a number of reasons why the control structure suggested in [4] uses a high-pressure controller in place of the low-pressure one for controlling the evaporator temperature. The refrigerant high-pressure sensor required for controlling the high-pressure is already present for monitoring and protection functions in todays R134a circuits, so no additional sensors are needed and this suggestion gets rid of the cost for a low-pressure sensor. There are a number of reasons why the control structure proposed by [4] was dropped in favour of a simpler one. On the low pressure side the existing evaporator outlet temperature sensor can be used due to the simple temperature-pressure relationship of the saturation curve:

- The feed forward is not robust to changes in the environment conditions, in particular not to changes in humidity, which today is not measured due to too costly sensors. The feed forward only works well in a limited range of operating conditions and actually decreases control performance in other situations. A feed forward based design that includes humidity measurements would most likely avoid the robsutness drawback.

- Using the components chosen in the given prototype R744 system with the two-stage orifice valve, undesirable limit cycling behaviour occurs at some operating points. It is not possible to remove the limit-cycling behaviour with the given control structure.

- For engine speed disturbances, the feed forward scheme for controling evaporator outlet temperature from with a feedback on the high side pressure did not work reliably.

- The occurence of multiple steady states, see section 4.3.

The current investigation was not done with a fully realistic sensor model for the evaporator temperature. If a cost-effective temperature sensor would be too slow to control engine speed variations, a low pressure controller would still be preferrable to the high pressure one with feedforward due to the list of drawbacks above.

4.1 Performance of the Valves

To compare the operation of the controllable valve in an optimized cycle and a two-stage valve without control of the high pressure, all boundary conditions and the compressor speed are kept constant and simulation were performed for three different load cases and both valves.

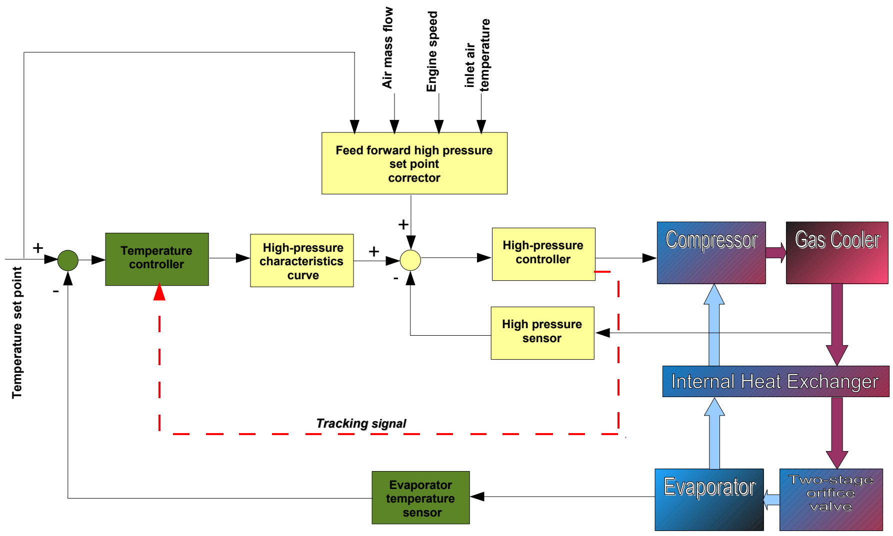

Figure 4: Proposed Control Structure by [4], simplified compared to the control structure in 3. In this case it is also assumed that the temperature set-point for the evaporator is adapted at low load to improve the COP.

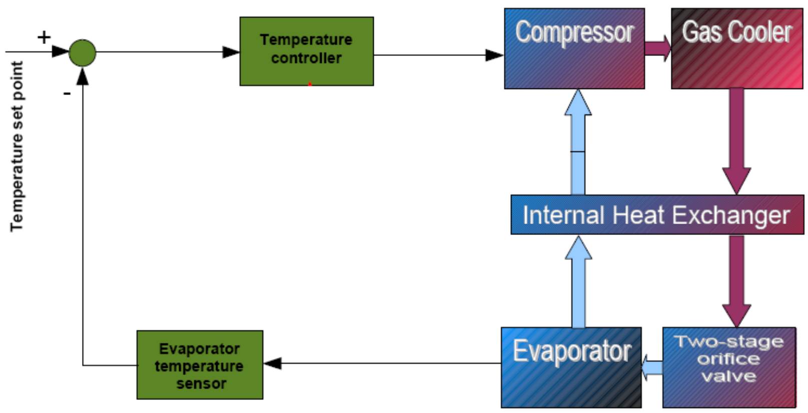

Figure 5: Alternative control structure for control of the low pressure side. The evaporator temperature set point is used to improve COP, which means that a higher complexity is needed in the supervisory part of the HVAC control that needs to determine the proper temperature set point.

1. Low cooling load and no control on evaporator outlet air temperature (Fixed relative volume of the compressor)

2. High cooling load and no control on evaporator outlet air temperature (Fixed relative volume of the compressor)

3. Evaporator temperature controlled (low cooling load)

4.1.1 Case 1

The system with two-stage valve has lower COP and higher cooling power than the optimized cycle, even for the lower ambient temperature. At higher temperature losses decrease, see Figure 6.

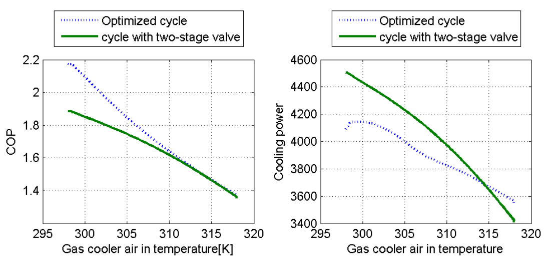

Figure 6: Comparison of the two valves, case 1

The high-pressure with two-stage orifice valve is kept fixed around 110 bars, while the variable Kv valve allows the pressure to change in a wider range. The reasons is behind the internal mechanism of the two-stage orifice valve which does not result in a Kv-value close to the controlled Kv for most of this range (Figure 7).

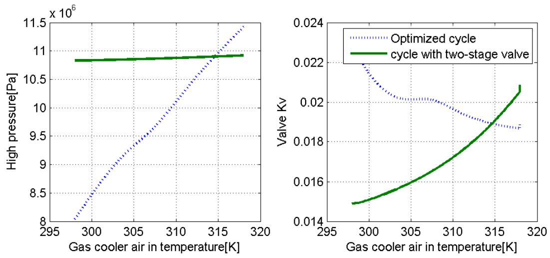

Figure 7: Comparison of the two valves, case 1

4.1.2 Case 2

In comparison with the previous case, at higher loads, the cycle with two-stage orifice valve has a COP near to the optimum value but at higher ambient temperatures it does not achieve equally high cooling power as the optimized cycle.

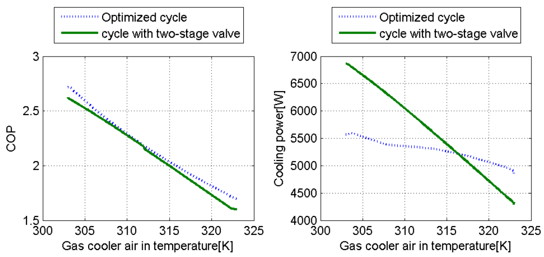

Figure 8: Comparison of the two valves, case 2

For this cooling load, the Kv shows a smaller deviation from the optimized one.

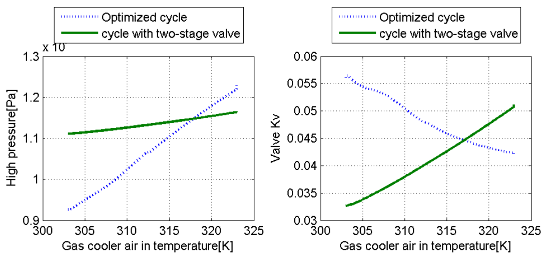

Figure 9: Comparison of the two valves, case 2

4.1.3 Case 3

When the low-pressure is controlled via the relative displacement of the compressor, the COP is improved for both cycles.

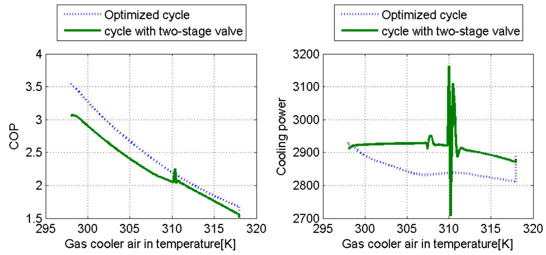

Figure 10: Comparison of the two valves, case 3

Since the pressure difference is low, at the lower ambient temperatures, the refrigerant passes through the fixed orifice of the two-stage orifice valve and provides the high-pressure that is needed for better COP. Both the valve-Kv values and correspondingly the resulting high pressures are closer to one another for this load case and control scheme than for the previous two ones.

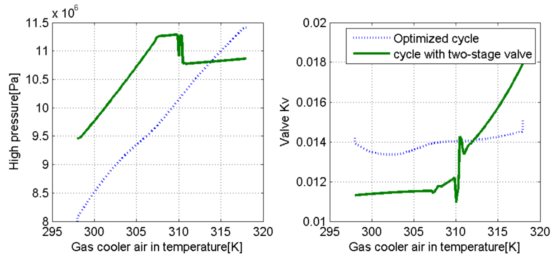

Figure 11: Comparison of the two valves, case 3

As has been demonstrated in this section, for lower ambient temperatures, the COP of the cycle with two-stage orifice is up to 40% less than ideal cycle, therefore it is suggested that in this range of ambient temperatures, the evaporator temperature is controlled to the highest possible value to improve the COP. Assuming the evaporator temperature is controlled with the two-stage valve cycle, the differences between the solution are not as dramatic as a first look suggests. The worst case scenario is, however, handled better with the optimized cycle that provides the highest cooling power at the highest load case.

4.2 Limit-Cycling Behaviour

In some operating points, which result in a higher pressure-difference than 73 bars, as a consequence of the rising pressure, the bypass starts to open and decreases the high-pressure, the decrease in high-pressure causes the closing of the bypass and this limit cycle continues until one of the inputs alters the pressure difference and mass flow rate. To observe the role of flow rate and pressure change in the limit cycle phenomenon directly, all the boundary conditions are kept constant and the relative volume of the compressor is changed manually to provide the appropriate pressure difference and flow rate. Figure 12 illustrates above explanations.

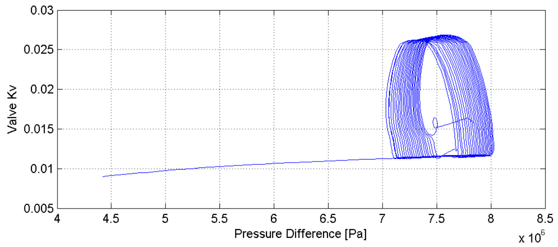

Figure 12: Limit cycle

Other output parameters, which are correlated with the high-pressure, will also show this limit cycle. The effect on the evaporator outlet air temperature is negligible (less than 1^{o}C in this case) and it is seen in Figure 13 that the low-pressure controller can remove the fluctuations. Therefore passengers do not sense the oscillations of the temperature.

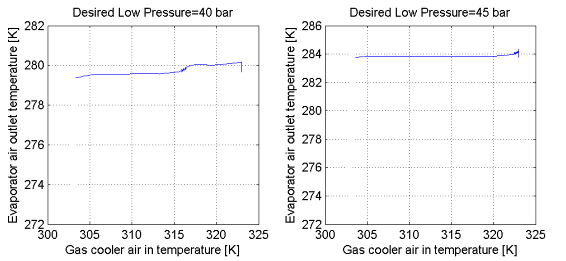

Figure 13: Temperature and limit cycle

But the effect on the cooling capacity and COP is quite considerable. In the temperature interval where this phenomenon happens, the highest deviation of the COP is about 50% less than the expected average value (Figure 14).

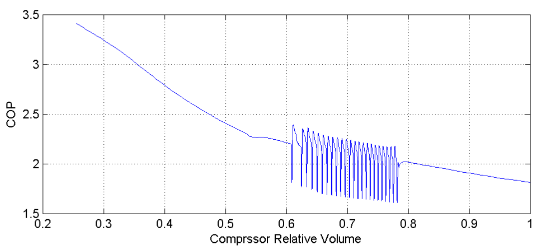

Figure 14: COP and Limit cycle

However, this limit cycle does only occur at few operating points and its characteristic differs in different circumstances. The following observations demonstrate this statement: Assuming a low-pressure controlled cycle, the ambient temperature varies in the range from 30^{o}C to 45^{o}C, and other operating conditions are kept constant. Figure 15 shows the phase portrait plot of two different cases when the limit cycle takes place. One of them happens when the desired low-pressure is 40 bar and the other one at 45 bar.

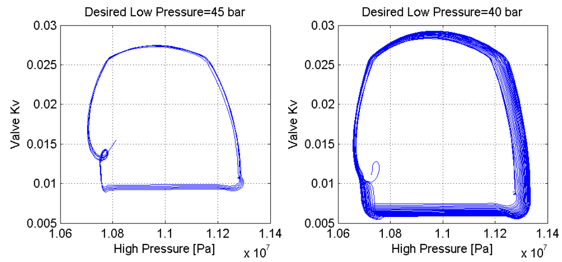

Figure 15: Portrait plot of the valve Kv against the pressure-difference

Under normal driving conditions, boundary conditions will rarely ever be constant for a sufficiently long time such that these limit cycling conditions will be noticeable, but they are nonetheless an undesired side effect of the valve construction.

4.3 Multiple Steady-States

In the case of high-pressure control and in the vicinity of 73 bar pressure-difference, when the two-stage orifice valve changes its flow configuration, a *bistability* phenomenon takes place. In this case, any disturbances which leads to small variance in the pressure-difference, causes the valve to jump to the alternate path while the high-pressure is kept constant by the controller. Therefore the system is able to exist in either of two steady states, while the high-pressure is fixed. Figure 16 shows that a small disturbances of the pressure, pushes the system to another steady state and causes a significant change in the cooling power. Although this will be compensated by the outer loop later on, it is another situation where the high-pressure loop in combination with the two-stage valve acts against the main purpose of control.

5 Dual Evaporators

Today luxury cars allow passengers to control a different climate in up to four climate zones. This requires the presence of two or even three evaporators to generate the cooling capacity for front and rear passengers. The Electronic Control Unit (ECU) controls the position of the different temperature blend doors

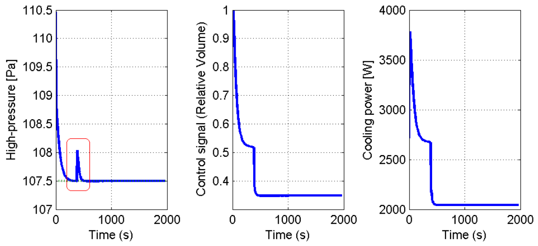

Figure 16: Bi-stable behaviour for high pressure control

to provide the passengers with their desired temperature in different zones. In the cooler unit, the high-pressure refrigerant splits and flows from two different expansion devices to the front and rear evaporator. The cooling capacity is divided accordingly between both evaporators. But the amount of the division depends on the operating conditions and structure of the valves. If a variable displacement compressor is used to control the front evaporator outlet air temperature, and a two-stage orifice valve to improve the COP, then a fixed orifice can be used to pass the refrigerant to the rear evaporator. In this case, there is no direct control on the outlet air temperature of the rear evaporator. To have full control authority on both evaporator temperatures, a controllable second expansion device would be needed. Alternatively, a model-based controller could be designed to control the compressor relative volume based on the measured value of the outlet air temperature of both evaporators. The easier way to control the cooling capacity of the rear evaporator is to use a variable speed fan and change the air flow around the evaporator, while the temperature of the front evaporator is controlled with the compressor relative volume variation using the same SISO approach as for the one-evaporator system. This will change the balance point of the rear evaporator low-pressure and this in turn changes the front evaporator low-pressure. The behaviour of the latter control system is investigated in [2], where in the modeling of the dual evaporator system, it is supposed that the front evaporator uses fresh air for ventilation and the rear compartment has just one zone. The outlet air of the front evaporator enters the car cabin, it is mixed with recirculation air of the rear compartment and then enters the rear evaporator for the second phase of cooling.

5.1 Cooling Power Distribution

To compare the cooling power of the one-evaporator system with the two-evaporator one, both systems are simulated under the same cooling load and at the same operating conditions. Note that all other components are the same, which means in particular that the heat rejection capacity via the gascooler is identical for both systems. Figure 17 illustrates that the summation of the capacity of the front and rear evaporator is equal to the capacity of one-evaporator system in this condition. It also shows that the outlet air temperature of the front evaporator is same for both cases. With a perfect model which includes the corresponding effects of the rear compartment on the front one, this distribution scheme may change a little and more compressor work will be needed to keep the front evaporator temperature constant.

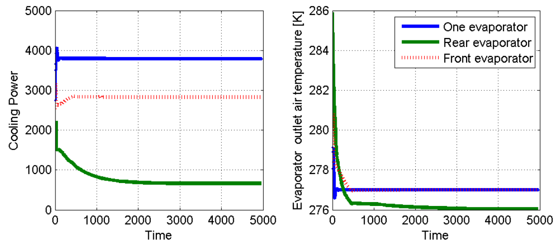

Figure 17: Cooling power distribution between two evaporators in comparison with one-evaporator system

Figure 18 shows the cooling power distribution against the ambient temperature. At higher temperatures, the pattern of distribution will change but acceptable cooling power is still provided for both evaporators.

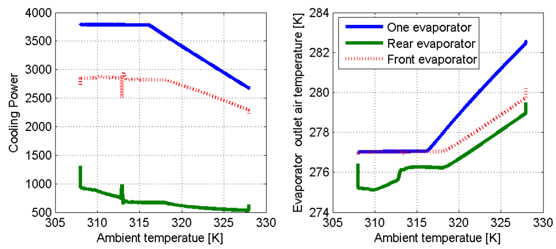

Figure 18: Cooling power distribution between two evaporators in comparison with one-evaporator system

5.2 Rear AirFlow Effect

In order to order to manipulate the cooling power of the rear evaporator, it is possible to change the air mass flow through it. The simulation was run in a limited range of airflow variations under two different cooling loads. Figure 19 shows the change of the rear evaporator cooling power when the air mass flow is changed at 5000 second. Figure 20 shows the cooling power variation against the air mass flow variation under a high and a low cooling load.

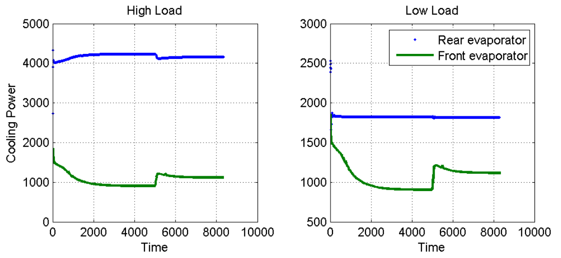

Figure 19: Rear evaporator air flow change.

It is seen that the rear cooling power is changed while the front one is almost kept constant. At lower cooling loads, the rear evaporator capacity is more sensitive to the air mass flow change.

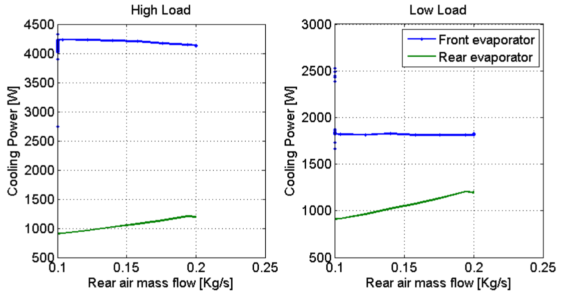

Figure 20: Cooling power over air mass flow.

Therefore, at these conditions, using a two-stage valve besides the front evaporator temperature control is possible, while the capacity of the rear evaporator is controlled by means of adjusting the air mass flow.

6 Conclusions

Various aspects of system and control design for a prototype of a R744 automotive A/C system for the Mercedes S-class were investigated by simulation using the AirConditioning Library and Dymola. Different system designs with a controllable expansion valve and a two-stage bypass orifice were compared and show that the controllable valve gives up to 15 % better COP than the two-stage valve. Several control designs were compared and the result was that the simplest control structure proved to be most robust and had better performance than the more complex versions. Furthermore it is demonstrated that the system with the highly non-linear two-stage valve exhibits limit-cycling behaviour and bistability around the part of the valve characteristic that looks almost like a step function in the valve coefficient Kv. For the two-evaporator system which uses a two-stage orifice valve to regulate the pressure of the front evaporator, simulation results suggest that the same approach of control for the one-evaporator system is also applicable for the dual evaporator system. With the given limited control authority, pressure and temperature of the rear evaporator will always be defined by the controlled conditions for the front evaporator and the boundary conditions. Instead of temperature control for the rear compartment, the capacity of the rear evaporator can be controlled using a variable speed fan, but only within certain limits.

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