

Open Issues in Control of Automotive R744 Air-Conditioning Systems

Sanaz Karim

2007

Master Thesis
Electrical Engineering
Nr: E3492E



DEGREE PROJECT

Electrical Engineering

Programme	Reg number	Extent
Master of Science in Electrical Engineering	E3492E	30 ECTS
Name of student	Year-Month-Day	
Sanaz Karim	2007-03-29	
Supervisor	Examiner	
Dr. Pär Samuelsson	Dr. Björn Sohlberg	
Company/Department	Supervisor at the Company/Department	
Modelon AB	Dr. Hubertus Tummescheit	
Title		
Open Issues in Control of Automotive R744 A/C Systems		
Keywords		
R744, Air-Conditioning, COP, Cooling power, optimization, two-stage orifice valve, Two-evaporator		

Abstract

In this thesis, one of the current control algorithms for the R744 cycle, which tries to optimize the performance of the system by two SISO control loops, is compared to a cost-effective system with just one actuator. The operation of a key component of this system, a *two stage orifice expansion valve* is examined in a range of typical climate conditions. One alternative control loop for this system, which has been proposed by Behr group, is also scrutinized.

The simulation results affirm the preference of using two control-loops instead of one loop, but refute advantages of the Behr alternate control approach against one-loop control. As far as the economic considerations of the A/C unit are concerned, using a two-stage orifice expansion valve is desired by the automotive industry, thus based on the experiment results, an improved logic for control of this system is proposed.

In the second part, it is investigated whether the one-actuator control approach is applicable to a system consisting of two parallel evaporators to allow passengers to control different climate zones. The simulation results show that in the case of using a two-stage orifice valve for the front evaporator and a fixed expansion valve for the rear one, a proper distribution of the cooling power between the front and rear compartment is possible for a broad range of climate conditions.

Table of Contents

Table of Contents	iv
Acknowledgements	vi
1 Introduction	1
2 Overview of Automotive Air-Conditioning	4
2.1 Air-Conditioning Unit	4
2.2 Refrigeration Cycle	6
2.2.1 Subcritical Cycle	8
2.2.2 Transcritical Cycle	8
2.3 Refrigeration Capacity and Power	10
2.3.1 Balance of Refrigeration Cycle	11
2.4 Control System	14
2.4.1 Controllable Components	16
3 System Modeling and Simulation Tools	18
3.1 Modelica Language	18
3.2 Dymola Environment	23
3.3 Modelon Air-Conditioning Library	25
4 Optimization and Control	28
4.1 Optimum High-Pressure Control	28
4.2 Evaporator Temperature Control	33
4.3 Two-stage Orifice Valve	35
4.3.1 Performance of the Valves	36
4.3.2 Limit Cycle of Two-stage Orifice	38
4.4 Behr Alternative Controller	43

5	Four-Zone Air-Conditioning	49
5.1	Two-Evaporator System Model	51
5.2	Simulations	52
6	Summary and Conclusions	55
	References	57

Acknowledgements

This thesis was carried out at Modelon AB in Lund, Sweden. Models of the prototype components have been provided by DaimlerChrysler, Germany.

I should hereby thank the Modelon team for making my stay in Lund very pleasant, most of all I appreciate the great supervision and support of Dr. Hubertus Tummescheit. Besides my thesis, he provided me with an opportunity to learn about object-oriented modeling.

I would also like to thank Dr. Pär Samuelsson, my local supervisor at Department of Electrical Engineering at Dalarna University, and Dr. Björn Sohlberg for their kind guidances during my studies in Sweden.

Chapter 1

Introduction

Under the Kyoto protocol agreement, by the year 2012, industrialized countries must 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 (CO₂) technology has been proposed as a natural alternative to current R134a-based systems.

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)

Besides the environmental benefits mentioned above, at some climate conditions, using R744 as a refrigerant for Air-Conditioning (A/C) systems decreases the fuel consumption through less fuel needed to run the A/C system.

Using R744 creates a need for major changes in the current A/C system components. DaimlerChrysler and its A/C system supplier, Behr GmbH, have designed and developed the prototypes to be used in the future cars. These prototypes have been modeled and validated by means of the Dymola Air-Conditioning library. This library is a Modelica package for dynamic simulation of A/C systems, which has been developed at Modelon AB.

Several issues with the R744 components design and system control remain unsolved or undecided [Wiessler, 2006]. In this thesis some of these control issues will be studied.

From the control viewpoint, two features of the A/C systems are important to be considered:

1. Comfort maintenance: To provide passengers with a nice temperature, an acceptable amount of air flow and low humidity in the cabin.
2. Energy efficiency : To reduce the fuel consumptions of the A/C unit which is one of the major fuel consumers in the car.

The A/C system consists of several different parts which cooperate with each other to satisfy the above goals, but the task of the heat absorption from the car cabin is mostly carried out by one evaporator for ordinary vehicles or two evaporators for luxury or very big vehicles. The absorbed heat of the evaporator is rejected to the outside environment from a condenser by aid of a compressor.

Since the A/C system is a multi-input multi-output process, for a multiple performance control, the model-based multivariable control strategies needs to be applied. For R134a systems, such strategies have already been developed by [Shah et al., 2003] and [He and Liu, 1998].

However such sophisticated controllers are not popular, cost-effective solutions in the opinion of the automotive industry. Therefore decoupled Single-Input Single-Output (SISO) feedback loop techniques must be developed by considering the strong cross-coupling among the various actuating inputs and outputs [Shah et al., 2003].

Variable displacement compressors, electric expansion valves, and variable speed fans for air flow over the heat exchangers are the available controllable components which can be used for the purpose of multiple SISO control. Apparently using several SISO control loops results in increase of control authority but again, cost reduction considerations require as few as possible controllers, sensors and controllable components.

One of the current multiple SISO control algorithms for R744 cycle tries to optimize the performance of the system and satisfy comfort requirements by the help of two controllable devices: the variable displacement compressor and the electronic expansion valve. It would be much more agreeable, if one of these controllable device can be replaced by a simpler and cheaper one. Therefore a mechanical expansion device, the *two stage orifice expansion valve*, has been built to replace with the electronic expansion valve. However the operation of this device in various climate conditions has to be studied more carefully for concerns of unit efficiency and comfort.

One of the major A/C component suppliers, Behr GmbH & Co., has designed its second generation of the R744 A/C system by accepting this valve and claimed “at minimum additional cost, their new system achieves the same level of steady-state cooling performance as present R134a systems” [Lochmahr et al., 2005]. This claim is also going to be examined precisely.

The problem of the decoupling is more complicated for two-evaporator systems because of their numerous coupled inputs. It should be verified whether the same simplified control approach of one-evaporator systems is applicable for two-evaporator systems.

According to this concise introduction, the main goals of this thesis can be classified as follows:

Goals

- Verification of the current multiple SISO loop control algorithm for different cooling loads and in different climate conditions.
- Comparison between the operation of a system with two controllable devices and system with a two-stage orifice valve for different cooling loads and in different climate conditions.
- Examination of the Behr alternative controllers.
- Simulation of the two-evaporator system in different climate conditions and cooling loads, in order to investigate the problems of the SISO control for this system.

Thesis Outline

Chapter 1 gives an overview of different parts of A/C systems. The thermodynamical principle of the process can also be found in this chapter. In order to figure out the control operation, it is necessary to understand these fundamentals. As mentioned earlier, modeling of the system was carried out with the Dymola A/C library. The modeling principles of this language and properties of the library are explained in chapter 2. In chapter 3, control algorithms are discussed and examined through several simulations. Chapter 4 talks about two-evaporator structure in the four-zone A/C system and cooling power distribution between them. In the last part a summary of the report and conclusions can be found.

Chapter 2

Overview of Automotive Air-Conditioning

2.1 Air-Conditioning Unit

To provide the passengers in a car with a comfortable cabin, the automotive air-conditioning (A/C) unit should handle unpleasant effects of temperature, humidity, airflow and heat radiation. To perform this task, the A/C unit needs to cool and heat air, dehumidify and distribute the air flow properly in different parts of the car cabin, by means of four major parts; an air intake mode selector, a blower, cooler and heater. The location of these parts differs in vehicles, but they generally are installed in the instrument panel of the vehicle, with the configuration of Figure 2.1.

Air which is introduced to the blower might be fresh or recirculation air. Depending on the driving condition and the cooling load, the air intake mode selector switches between recirculation and fresh air. In the recirculation mode, inside air of the cabin is re-circulated and in the fresh air mode, fresh air from the outside is used for air conditioning. In a place where it is full of exhaust gas or dirty ambient air or when high cooling power is needed, the air mode selector is switched to re-circulation air, but normally fresh air is preferred. The cooler is located after the blower. In the cooler unit, air is dehumidified and cooled and then by the means of a heater, it is re-heated to reach the desired temperature. The air distribution unit controls the volume of air to be re-heated in the heater part.

The cooler unit stores the evaporator and expansion valve. Heat, which is absorbed

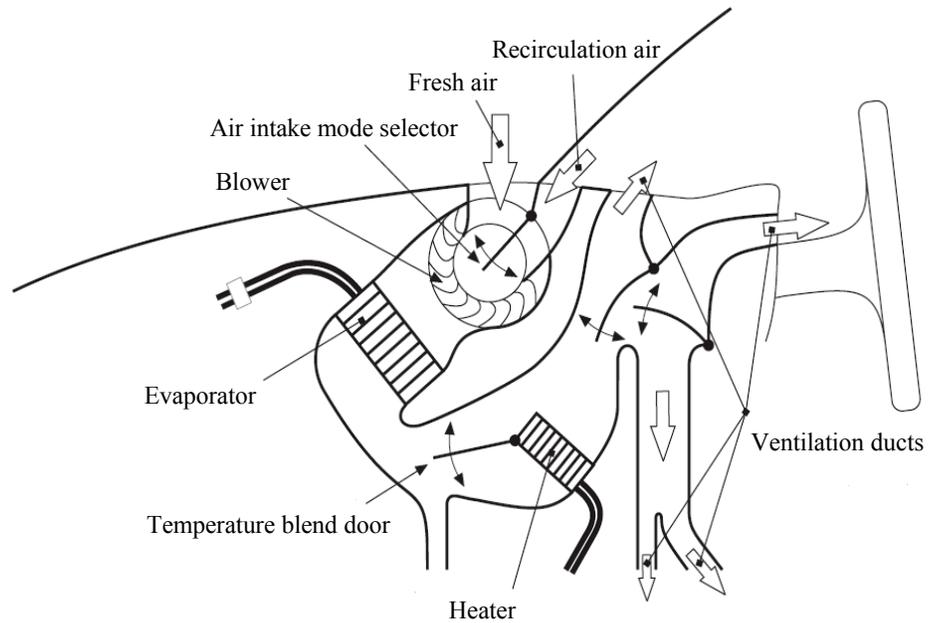


Figure 2.1: Automotive A/C archetype

by this unit, is rejected to outside environment by a condenser which is located in the front of the car. A compressor provides work needed for the heat rejection. In Figure 2.2, the location of these parts in Mercedes-Benz S-class is shown. This A/C unit has been designed by Behr GmbH and BHTC.

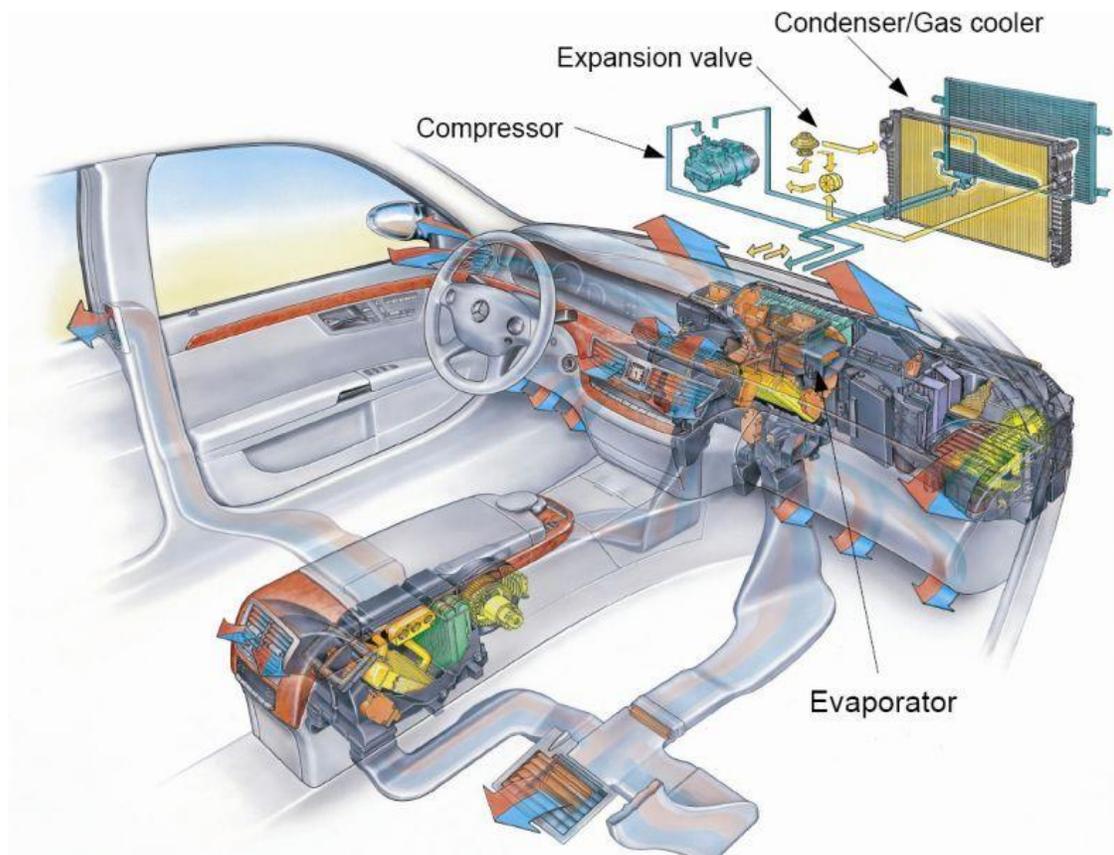


Figure 2.2: Air conditioning of Mercedes-Benz S-class [Behr GmbH & Co. KG]

2.2 Refrigeration Cycle

In this section, the cooling process will be explained by the help of pressure-enthalpy (PH) diagram. This diagram shows the phase changes of a fluid through different enthalpies and pressures. Enthalpy is a measure of the useable energy content of a fluid. Figure 2.3 shows the PH diagram of R134a which is one of the most in-use refrigerant in vapor compression cycles. In this figure, point “a” which is located in the subcooled or liquid region shows the phase of the refrigerant when its temperature is below its boiling point. If heat is added to this refrigerant at the constant pressure, its enthalpy will increase and make saturated liquid at point “b”, with more heat, it enters into the two-phase or mixed vapor and liquid region. Afterwards by adding more heat, the vapor portion of refrigerant will increase and at point “c” it becomes saturated vapor. The vapor fraction of the refrigerant is called *quality* with a value between 0 and 1, so at point “b” the quality of the refrigerant is 0 and at

point “c” it equals to 1. The heat that is absorbed in the two-phase region is called latent heat of evaporation because this heat does not increase the temperature and just increase internal energy or enthalpy. Therefore the evaporation of the refrigerant occurs *isothermally* in the two-phase region. With further heat increase, the refrigerant state moves into the “superheated vapor” region, and temperature will increase in this region. On the PH diagram, it is seen that the saturated liquid and vapor curves unite at a point are called “critical point”. At this point the refrigerant temperature and pressure called critical temperature and critical pressure respectively. The “Supercritical region” is located above the critical pressure, where the refrigerant state does not undergo distinct phase transitions by heat addition or reduction.

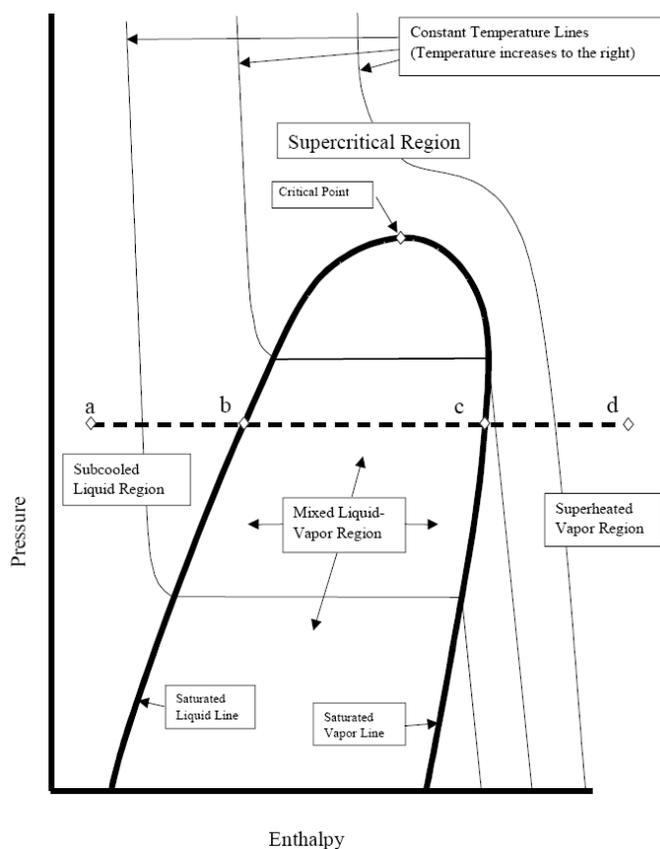


Figure 2.3: Pressure-Enthalpy Diagram

To absorb heat from the cold space (car cabin) and discharge it into the hot space (outside environment), a specific cycle of phase transitions needs to take place, which will be explained in the next section.

2.2.1 Subcritical Cycle

Figure 2.4 illustrates a subcritical refrigeration cycle with its related cooling system. The refrigerant commonly used in this system is R134a. At point 1 the refrigerant is sucked into a compressor at a low pressure and compressed adiabatically (no heat is removed), at a higher pressure and temperature, higher than the critical temperature of fluid, it enters a condenser at point 2. Since it has higher temperature than the hot space temperature, the refrigerant is transferring its heat to the air and condensing. It leaves the condenser as a subcooled liquid at point 3. Through an expansion device, it expands and enters to the low-pressure part of the system. This expansion process moves the refrigerant into the two-phase region at a low pressure and temperature. Then in the evaporator, the refrigerant with a low pressure and a temperature lower than the ambient temperature absorbs heat from the cold space to vaporize and then reach the point 1 as superheated vapor. To be sure that just pure vapor enters the compressor and no liquid, a component called *receiver* is fixed before the compressor, which separates the liquid fraction of the refrigerant. Another option is to place a receiver (accumulator) after the condenser to subcool the refrigerant, thus the refrigerant entering the expansion device is always a saturated liquid.

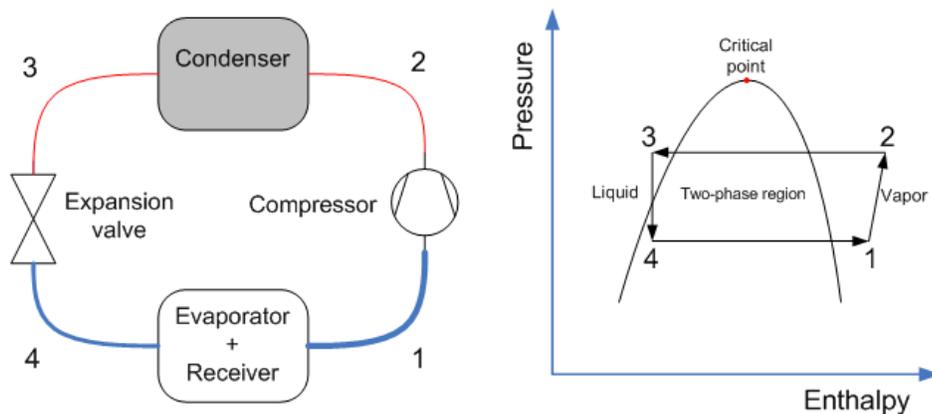


Figure 2.4: Subcritical refrigeration cycle

2.2.2 Transcritical Cycle

Apparently subcritical heat rejection is just possible for refrigerants with the critical temperature much higher than the hot space temperature. But the critical temperature of R744 is 31°C , which for many the cases is less than the ambient temperature.

In such caess the heat rejection process does not happen in the subcritical region anymore and the refringent will enter the supercritical region to reach higher temperatures. So the subcritical cooling system needs some modifications to be able to work for R744. Figure 2.5 illustrates this modified system. In this system, the condenser is replaced by a gas cooler. Another component named *internal heat exchanger* is also added to the structure. Refrigeration cycle starts at the point 1 when the vapor refrigerant enters the compressor, then the compressor raises the refrigerant pressure, temperature and enthalpy. The refrigerant with high pressure leaves the compressor for the gas cooler. Unlike the condensing process in subcritical systems, the refrigerant remains in the gas phase and the heat rejection in the supercritical region does not occur isothermally.

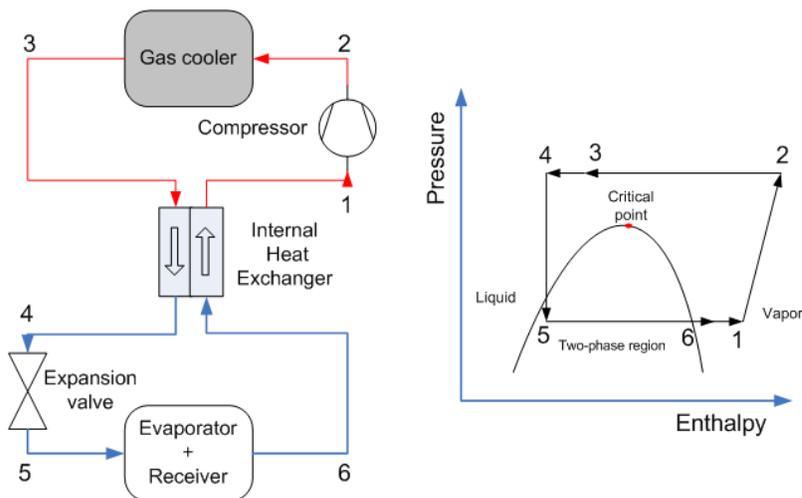


Figure 2.5: Subcritical refrigeration cycle

To increase the system efficiency and reject more heat, an additional heat exchanger is needed. The inner part of this internal heat exchanger removes excess heat from gas leaving the gas cooler and the outer part, which contains the refrigerant coming from the evaporator, will absorb this excess heat. After the internal heat exchanger, the refrigerant leaves the high-pressure section by passing through an expansion device and at a low pressure and temperature enters the evaporator to absorb heat from the cooled space. Finally, in the inner part of the internal heat exchanger, it absorb more heat then the superheated refrigerant is ready to become compressed again.

2.3 Refrigeration Capacity and Power

The PH diagram is depicted for refrigerant per unit weight, to obtain actual capacities and powers, the refrigerant charge circulating in the cycle must be known.

The mass flow rate of a fluid G_r [Kg/h] that is circulating in a cycle by power of a compressor is given as:

$$G_r = \frac{V_c}{v_s} \quad (2.3.1)$$

Where, V_c [m³/h] is the compressor suction volume and v_s [m³/kg] is the specific volume of the fluid and the compressor suction volume is obtained as below [Watanabe, 2002]:

$$V_c = \frac{V_1 \times N \times 60}{10^6 \times \eta_v} \quad (2.3.2)$$

V_1 : Compressor cylinder volume [cc]

N : Compressor speed [rpm]

η_v : Volumetric efficiency

The cooling power can be represented as the change of enthalpy of the refrigerant, when the evaporator absorbed heat from the cooled space (car cabin). Hence for the subcritical system it is computed by:

$$Q_{er} = (h_4 - h_1)G_r \quad (2.3.3)$$

Where, h_4 is the enthalpy at the evaporator inlet and h_1 is the enthalpy at the outlet. Since the adiabatic compression increases the enthalpy, the compressor work performed on the fluid can be represented as:

$$P_C = (h_2 - h_1)G_r \quad (2.3.4)$$

Where, h_2 is the discharge enthalpy . The efficiency of the cycle is defined by a quantity named the Coefficient Of Performance (COP); It is the ratio of the absorbed heat from the cooled space to the amount of compressor work required for this absorption:

$$COP = \frac{Q_{ev}}{P_C} \quad (2.3.5)$$

Higher COP values indicate that more heat is removed for a given amount of work. The COP is not only a function of the system features but also a function of the operating conditions, such as the car cabin and environment temperature.

2.3.1 Balance of Refrigeration Cycle

According to the heat transfer principle, the amount of the heat which is removed from air by the evaporator Q_{ea} should be equal to the evaporator heat absorption capacity on the refrigerant side Q_{er} (2.3.3). The absorbed heat on the air side Q_{ea} is proportional to the difference between the refrigerant temperature T_{er} and the air temperature T_{ea} . By air here, we mean warm air which is entering the evaporator (inlet air), not cooled air after the evaporator (outlet air).

$$Q_{ea} = \phi_e \cdot C_a \cdot G_{ea} (T_{ea} - T_{er}) \quad (2.3.6)$$

Substituting G_r from the 2.3.1 in the equation 2.3.3, the evaporator capacity on the refrigerant side is:

$$Q_{er} = \frac{V_c}{v_s} (h_4 - h_1) \quad (2.3.7)$$

ϕ_e : Evaporator temperature efficacy

V_c : Compressor suction volume

C_a : Specific heat of air

G_{ea} : Air mass flow rate

h_4 : Enthalpy of the refrigerant at the evaporator inlet

h_1 : Enthalpy of the refrigerant at the evaporator outlet

T_{er} : Temperature of inlet air

T_{ea} : Temperature of the refrigerant in the evaporator

From the (2.3.6), it is seen that the lower the refrigerant temperature T_{er} , the higher capacity on the air side Q_{ea} can be obtained. In other words, the refrigerant temperature, changes the intercept of the Q_{ea} line in Figure 2.6. On the other hand, the lower refrigerant temperature T_{er} results in the lower capacity on the refrigerant side Q_{er} and changes the slope of the Q_{er} in Figure 2.7, because the lower refrigerant temperature means the lower low-pressure (as it is seen from the PH diagram in Figure 2.3) and due to the fact that the specific volume of gas is less at the lower pressures, so according to equation 2.3.7 this decreases the capacity on the refrigerant side. Thus the refrigerant flows at a low-pressure which satisfies $Q_{er}=Q_{ea}$. We call it balance point for the low-pressure. Similar to the evaporator and low-pressure, the high-pressure balance point is defined by the air and refrigerant side properties of the condenser as indicated in Figure 2.6a. The only difference is that the high-pressure variation does not change the refrigerant flow rate considerably.

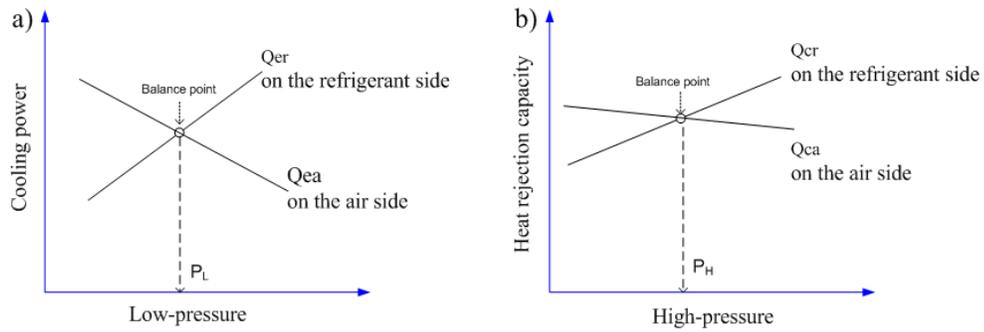


Figure 2.6: Balance of low-pressure and high-pressure

The intercept of the Q_{er} line in Figure 2.6 is changing by the inlet enthalpy, which is a function of the high-pressure. The higher the high-pressure the lower the refrigeration capacity on the refrigerant side Q_{er} becomes. Therefore, depending on the high-pressure balance point location, the low-pressure balance point location varies (Figure 2.7). Therefore the pressure and capacity at the balance point in the refrigeration cycle are obtained when the high-pressure, low-pressure and cooling power are balanced.

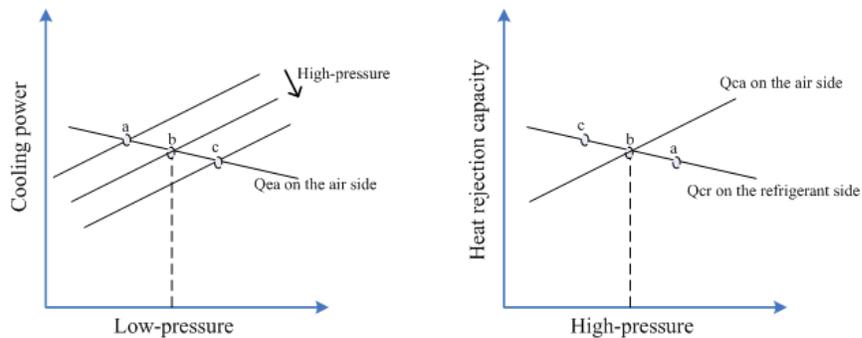


Figure 2.7: Balance of refrigeration cycle

As indicated by the equation 2.3.6, air properties around the evaporator also affect the position of Q_{ea} line in Figure 2.6 and balance point. The effect of the airflow rate on low and high-pressure balance is explained briefly here.

When the airflow rate around the evaporator increases, the refrigeration capacity on the air side increases and this leads to higher low-pressure on the refrigerant side and changes the balance point location as indicated in Figure 2.8. Therefore, the refrigerant flow rate increases and the condenser can reject more heat. Thus based on the balance principle for the condenser, the high-pressure corresponding to the refrigerant temperature increases.

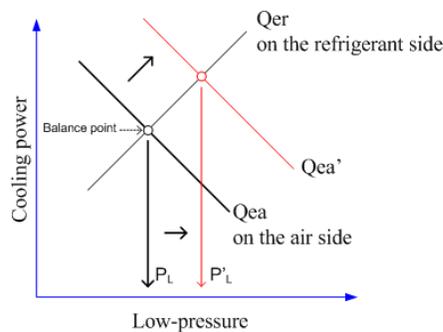


Figure 2.8: Balance change by evaporator airflow rate

When the airflow rate around the condenser decreases, the heat rejection capacity on the air side decreases. To compensate for this, the high-pressure increases to balance with the refrigeration capacity on the refrigerant side (Figure 2.9).

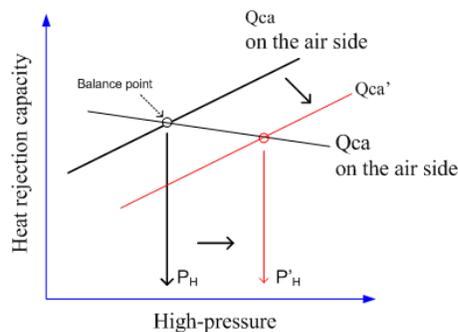


Figure 2.9: Balance change by condenser airflow rate

Another parameter which influences the balance of the cycle is the compressor rotation speed; When the speed increases, the refrigerant flow rate also increases. Since this increases the refrigeration capacity on the refrigerant side, the balance point location is changed because the low-pressure decreases to achieve the heat absorption capacity on the air side, and the high-pressure increases to achieve the heat rejection capacity of the condenser [Watanabe, 2002].

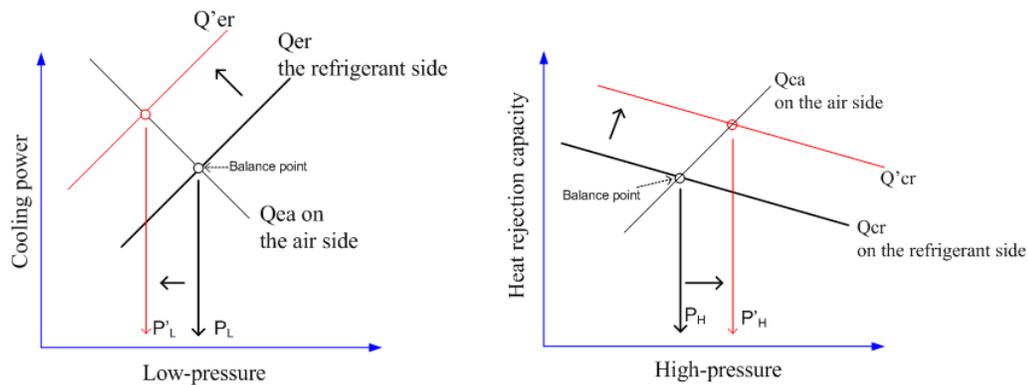


Figure 2.10: Balance change by the evaporator refrigerant flow rate

2.4 Control System

The air-conditioning control system consists of four basic parts:

- Sensors that detect the ambient air temperature, the cabin condition and the A/C system operating condition.
- A control panel that indicates the temperature, the operating condition, etc.
- The air-conditioning Electronic Control Unit (ECU) which is responsible to run the logic of A/C system control. Air-conditioning ECU calculates signals in order to control the outlet air temperature and the outlet airflow volume based on the various sensor signals and A/C panel signals, etc.
- The air-condition unit that operates according to the signal from the ECU.

Control goals of an A/C system can be classified as below:

- Temperature Control

- To provide the desired temperature, the angle of the temperature blend door in Figure 2.1 is controlled by the ECU based on the outside environment and car cabin condition. By this control, the outlet air temperature varies between the outlet temperature just after the evaporator and the heater core temperature.
- Airflow volume control
 - With the airflow volume control, ECU changes the speed of blower motor of A/C unit (See Figure 2.1), to increase the airflow volume when high cooling/heating capacity is required and decrease it when the temperature becomes closer to the desired temperature and a large capacity is not required anymore.
- Airflow distribution mode control
 - Besides the temperature degree, passengers comfort is also dependent on the direction of the airflow. In some cars A/C unit there is a air distribution door which directs the airflow either from face or head ventilation duct (Figure 2.1).
- Air intake mode control
 - This action is done to switch between fresh or recirculation air mode, when it is required. (See Section 1.1)
- Humidity control
 - The evaporator is responsible to absorb the humidity of air cabin by providing enough cooling power.
- Capacity Control
 - In section 2.3.1, we have seen that the cooling power and efficacy of the A/C unit are variable quantities relating to operating conditions. To have control on these quantities, two components of the refrigeration cycle are made as controllable components: expansion valve and compressor. Therefore control system can change the refrigerant flow rate by means of these components to achieve desired capacity.

2.4.1 Controllable Components

Expansion valve: The expansion valve could be a linear proportional solenoid valve. Usually the K_v of this valve, which defines its flow area, can be changed by pulse width modulation (PWM) techniques.

Compressor: The compressor is a belt driven pump that most often is fastened to the car engine via a clutch. There are primarily four types of compressors used in the automotive A/C: reciprocating, scroll, screw and centrifugal. The most preferred one in R744 systems is the variable displacement swash-plate compressor which belongs to the reciprocating category. In reciprocating compressors, the refrigerant vapor is compressed by a piston located inside a cylinder. The piston is connected to the crankshaft by a rod. As the crankshaft rotates, it causes the piston to travel back and forth inside the cylinder. A suction valve and a discharge valve, are used to trap the refrigerant vapor within the cylinder during this process. The piston travels away from the discharge valve and creates a vacuum effect. Reduction in the pressure within the cylinder to below suction pressure forces the suction valve to open and the refrigerant vapor is drawn into the cylinder. The piston reverses its direction and travels toward the discharge valve, compressing the refrigerant vapor, the suction valve is then closed and traps the refrigerant vapor inside the cylinder. As the piston continues to travel toward the discharge valve, the refrigerant vapor is compressed. The discharge valve is forced to open and the compressed refrigerant vapor leaves the cylinder. To have control on the capacity of the compressor, variable displacement

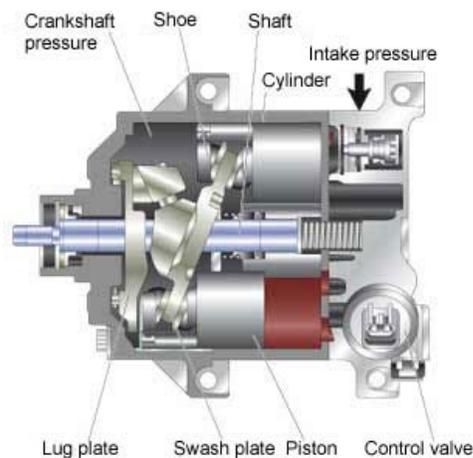


Figure 2.11: Variable displacement compressor (*by Toyota Industry*)

compressors have been developed in to types: swash and wobble plate. In these compressors, the required amount of refrigerant gas is sucked in and compressed by changing the pressure balance within the compressor. The angle of the swash or wobble plate determines the length of the piston stroke. In a variable displacement compressor, that angle can be changed, which changes the length of the pistons' stroke and, therefore the amount of the refrigerant displaced in each stroke [Gordon, 2005].

Chapter 3

System Modeling and Simulation Tools

The model of the air-conditioning system which is used in our simulations is explained in this chapter. This model is built based on the prototype components of Daimler-Chrysler R744 A/C system, by means of the Air-Conditioning Library. This library has been designed and developed at Modelon AB as a commercial Modelica library for the steady-state and transient simulation of automotive air conditioning systems.

3.1 Modelica Language

Modelica is a declarative, object-oriented modeling language which allows convenient, component-based modeling of complex systems which could contain mechanical, electrical, electronic, hydraulic, thermal, control, and electric components as well as multi-domain structures. It supports continuous, discrete event and hybrid modeling approaches. The key features of this language are described briefly here:

Object-orientated Modeling: Although it has many features in common with the traditional object-oriented programming language like C++ or Java, Modelica concept of object-orientation is different from them in some aspects. Modelica uses *declarative* mathematical descriptions for its models. A declarative representation of system behavior does not determine *how* something is calculated, instead, it defines *what* it is [Tummescheit, 2001], so when you look at the Modelica code of a

system you will find similarities with the explicit mathematical description of that system. Therefore the task of procedural description is derived by the compiler not by the programmer.

Classes: Like any other object-oriented language, Modelica provides the notions of classes and instances. Modelica classes can be classified in six major categories:

- **Type:** All data objects in Modelica are instantiated either from basic data types (*Real*, *String*, *Boolean*, *Integer*) or from enumeration types. It is possible to define different attributes of a variable other than its value. For example, the attributes of *Real* type are predefined as value, quantity, unit, display unit, min, max, etc. Therefore physical types (e.g. pressure, temperature, mass) can be derived from basic types but with their own special features. All the well-known **SIunits** are already instantiated from the basic types and exist in the Modelica standard library to ease declaring physical quantities and reduce the risk of errors in programming.
- **Model:** It is a general term for a complete Modelica object which usually is a structured description of the physical systems. A component is an instance of a model.
- **Connector:** Interaction between objects (components) of well-structured classes (models) is usually done through the connectors. For example each electrical component should contain connectors named **Pin** which defines voltage and current in connection with other electrical components based on Kirchhoff's laws.
- **Block:** Blocks have the well-known internal semantics with known inputs, attached to input connector, from which the outputs are computed.
- **Function:** Similar to the block definition, it defines the relation between its inputs and outputs but these are just variables not connectors. Functions can be called inside an equation.
- **Package:** A package refers to a collection of Modelica models which are meant to be used together [Tiller, 2001].

Inheritance: Similar to other object-oriented languages, the original class or the *base class*, is extended to create a more specialized version of that class known as *child class*. This derived class inherits the behavior and properties such as variable declarations, equations and other contents of the original classes. Modelica also supports the concepts of abstract classes under the name of *partial models*. These are not complete models and can not be instantiated, only used in inheritance.

Equations: Modelica is primarily an equation-based language in contrast to ordinary programming languages, where assignment statements build the body of the code. Assignment fix the computational causality but equations have no pre-defined causality. For example in a assignment statement like: $z := x + y$, the left-hand side of the statement is assigned a value calculated from the expression on the right-hand side and z has no effect on the value of x and y , but an equation may have expressions on both its right and left-hand sides, for example: $x + y = z$ just describes an equality. The translator and analyzer of the simulation engine have to manipulate and sort the equations according to data-flow dependencies to determine their order of execution and which components in the equation are inputs and which are outputs.

Acausal Physical Modeling: Modelica supports both two common approaches to modeling in engineering: causal (or block-oriented) and acausal (component-based) modeling formalism. This differs from other general purpose modeling packages such as Simulink that just use causal modeling methods by means of block diagrams. By the help of component and connector definition in Modelica, it is possible to built system models without considering the order in which the variables need to be calculated. This enables models to be defined in a more general way. The key advantage of acausal modeling is that it speeds up the model development process as it simplifies what the user must do. Component models are simply dragged into the model and connected in an recognizable manner without considering the calculation order. It also simplifies model maintenance as there is only one definition of each component model that needs to be maintained [Tiller, 2001].

Algorithms: In some cases where nondeclarative constructs are needed, algorithmic statement can be written within the *algorithm* section of Modelica. In the algorithm section, assignments, if-then-else statements, for-loops and while loops are available to use.

Functions are reusable algorithms with an fixed relation between its inputs and outputs but when a function is called inside an equation, it can be used to calculate an input from a given output.

Example: In Figure 3.1, Modelica code of a simple electrical circuit is shown. In this model, the components **R1** and **R2** have been instantiated from the **Resistor** class and so on about the other components. Then in equation section, the order of connections between different components has been defined.

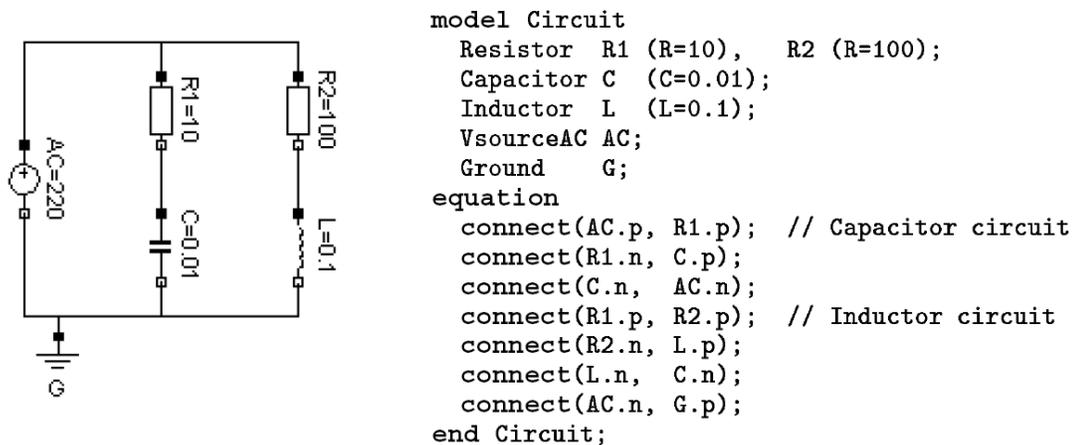


Figure 3.1: Modelica model of an electrical circuit

Figure 3.2 illustrates the block-oriented modeling of the same circuit. Here the physical topology is lost.

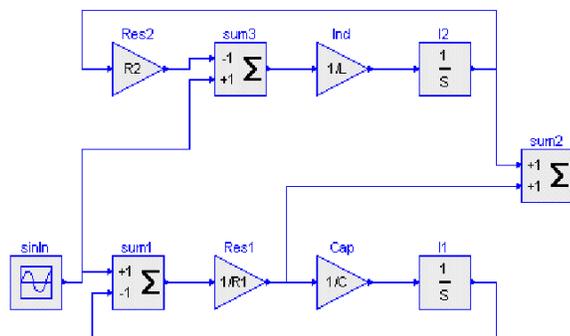


Figure 3.2: Equivalent Simulink model

In the equation section, the command `connect(Pin1, Pin2)` tells the compiler to compute two equations: $\text{Pin1.v} = \text{Pin2.v}$ and $\text{Pin1.i} + \text{Pin2.i} = 0$. The **Pin** connector definition is shown below. Similar laws apply to flow rates in a piping network and to forces and torques in a mechanical system. The sum-to-zero equations are generated when the prefix flow is used in the connector declarations.

```

connector Pin
  Voltage v;
  flow Current i;
end Pin;

partial model TwoPin
  Pin p, n;
  Voltage v;
equation
  v = p.v - n.v;  p.i + n.i = 0;
end TwoPin;

```

To model a resistor or any other simple electrical components, it is useful to define a *shell* class: **TwoPin**. This class has two pins, p and n, and a quantity, v, that defines the voltage drop across the component. See the code below. The equations define common relations between quantities of a simple electrical component. In order to be useful, another complementary equation must be added. The keyword *partial* indicates that this model class is incomplete. To define a model for a resistor, start from TwoPin and add a parameter for the resistance and Ohms law to define the behavior. For the capacitor it just needs little change in the equation section. Derivation is possible in Modelica and the expression `der(v)` means the time derivative of v.

```

model Resistor "Ideal resistor"
  extends TwoPin;
  parameter Resistance R;
equation
  R*p.i = v;
end Resistor;

model Capacitor "Ideal capacitor"
  extends TwoPin;
  parameter Capacitance C;
equation
  C*der(v) = p.i;
end Capacitor;

```

3.2 Dymola Environment

Dymola (Dynamic Modeling Laboratory) is a front-end for Modelica which contains a *symbolic translator* to pre-process the Modelica equations and then generate C-code for simulation. Symbolic manipulation can solve or simplify differential algebraic equations of any order at translation time, in order to make less numerical calculations when running the simulation. [Fritzson, 2003]

Dymola has a graphic editor for composing Modelica models. It can also import other data and graphics files. The generated C-code can be exported to Simulink and hardware-in-the-loop platforms. Scripts can be used to manage experiments and to perform calculations. Figure 3.3 from *Dymola 6.0 User Manual*, shows Dymola links with other environments.

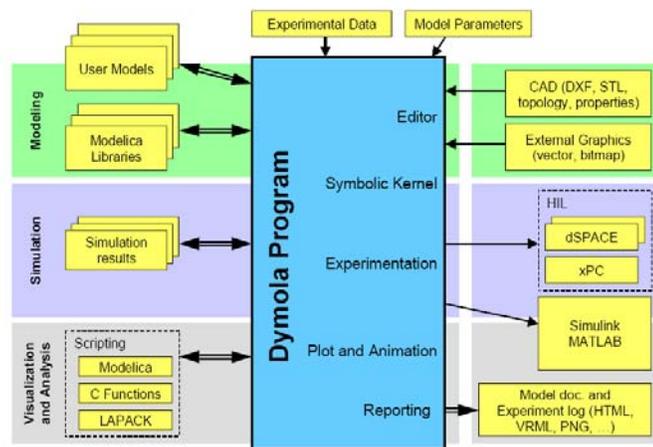


Figure 3.3: Architecture of Dymola

Dymola has two kinds of windows: Main window and Library window. The Main window operates in one of two modes: Modeling or Simulation. The Modeling mode of the Main window is used to compose models and model components. The Simulation mode is used to make experiment on the model, plot results and animate the behavior. The Simulation mode also have a scripting subwindow for automation of experimentation and performing calculations.

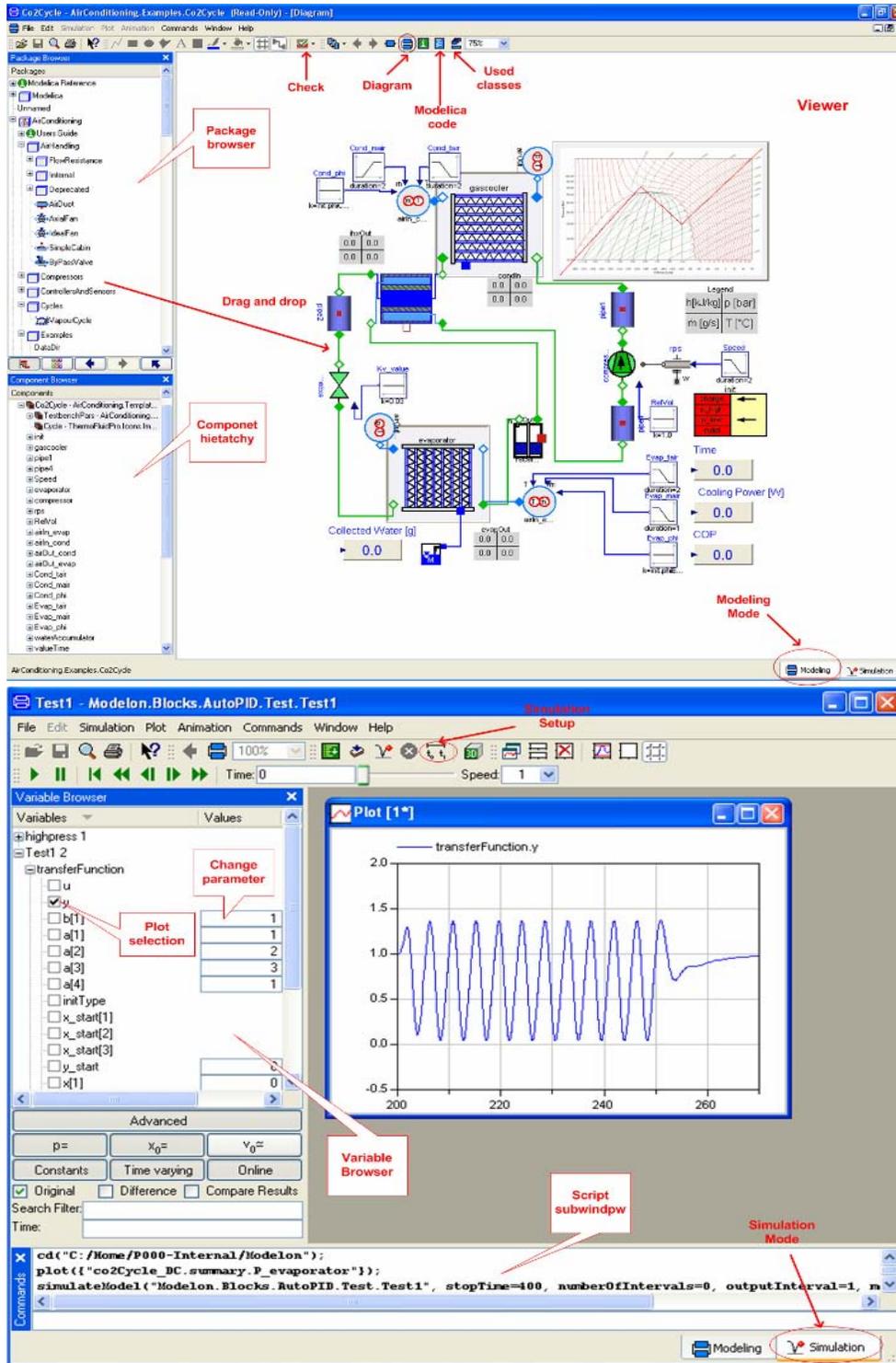


Figure 3.4: Modeling and Simulation mode of Dymola

3.3 Modelon Air-Conditioning Library

The aim of the modeling is to create a library with physical based models of the A/C system components. Such a library with models of these components and of additional components for testing, like air sinks and sources, can be used for investigations of both, single components and complete refrigeration cycles. Furthermore it is of great interest to make dynamic simulation as well as steady state simulation of refrigeration cycle and single components, especially heat exchanger [Pfafferott and Schmitz, 2002]. The Air-Conditioning Library provides component models for automotive A/C systems. This library has been derived from the Modelica library ThermoFluid and the ACLib library. Traditionally, A/C system level models are only used as steady-state models, with the exception of very simplistic, often linear models for control design. ThermoFluid provided accurate dynamic models, but could not be used for steady-state tasks. Air-Conditioning bridges that gap and is suited both for dynamic and steady-state design computations, eliminating the need for multiple platforms and models [Tummescheit et al., 2005].

Three levels of models are available in the A/C library:

- Ready-to-run templates for components and cycles for example template for standard R744 refrigeration cycle .
- Component models for drag- and drop
- Base classes for fully user-defined models

Users can connect component as they desire, which makes it easy to build also non-standard configuration such as two-evaporator, internal heat exchange and losses at any place in the cycle.

Among the others, the standard model of these main components are available in A/C the library:

Heat exchangers: The heat exchanger model is composed from two fluid objects (air and refrigerant) and one wall element. The wall mass is determined from detailed geometry input data and therefore reflects distributed capacities. Heat conduction in the solid material is modeled one-dimensional and perpendicular to both fluids. Heat is transferred between wall and fluid using a heat connector class (Figure 3.3).

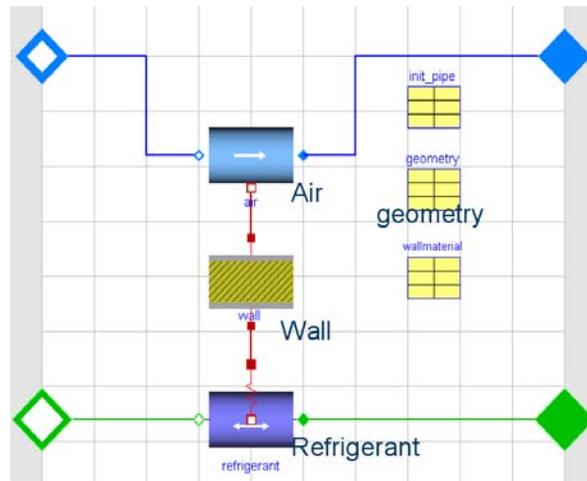


Figure 3.5: Heat exchanger modeling

Compressors: To model different types of A/C compressors, mass flow and change of enthalpy are calculated by algebraic equations. Efficiencies can be provided by non-linear approximation functions or tabulated data. Approximation functions are verified with data sets for many compressor types. Displacement volume can be controlled by an external signal.

Besides these, models of expansion devices, pipes and volumes, homogenous and inhomogeneous air sources are also available.

The other important factor in the complete transient simulation of a vapor cycle is the refrigerant mass. During different boundary conditions (e.g. changing air temperature, air massflow through heat exchanger) the refrigerant mass is moving to different parts of the system and must be observed and then the optimal charge of the system and the changing process behavior during any variation of the compressor speed, air massflow and temperature can be found out. [Limperich et al., 2005]

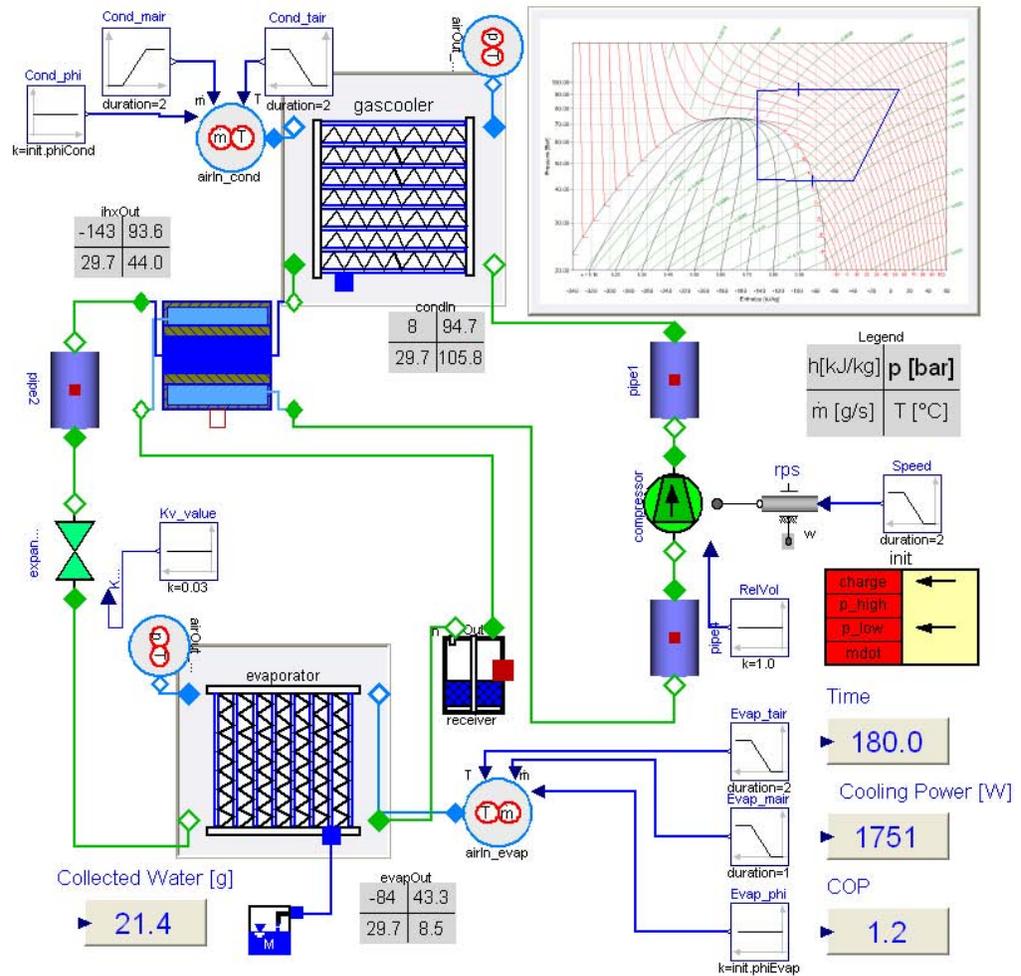


Figure 3.6: A/C library template for R744 refrigeration cycle

Figure 3.6 shows a generic R744 refrigeration cycle whose template can be found in the A/C library.

Chapter 4

Optimization and Control

As mentioned in (Chapter 1.4), a variety of control actions should be carried out in the A/C unit to keep the passenger cabin in a comfortable condition and to maintain system efficiency.

This chapter deals with the control of COP and cooling power in the cooler unit. The role of the cooler 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. Its second target is to maximize COP in order to reduce the fuel consumption. To increase cooling power at very high ambient temperature, usually a lower COP is accepted. However most of the operating times the optimization of COP is the dominant control target. The control actions of the cooler unit are performed under the general logic of the A/C control unit.

4.1 Optimum High-Pressure Control

To achieve maximum COP in R744 systems, a simple SISO control strategy has been proposed by [Yang et al., 2005].

Its performance for the R744 system with the specifications stated in Chapter 1, is studied in this section. 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 ambient temperature. In the supercritical region, the CO₂

working pressure typically ranges from 7500 to 13000 kPa. By changing the valve opening to achieve this desired high-pressure, the simulation is conducted over this pressure range, in order to study the effect of high-pressure on the COP and cooling power in some typical European climate conditions.

First, to study the effect of high-pressure on COP of different *gas cooler air inlet temperature*¹, six other boundary condition are kept constant with the values of Table 4.1.

Table 4.1: Boundary Conditions

Parameter	Value
Gas cooler relative humidity	60%
Gas cooler air flow rate	0.6 kg/s
Evaporator air inlet temperature	25°C
Evaporator relative humidity	40%
Evaporator air flow rate	0.1 kg/s
Compressor speed	16Hz

Figure 4.1 shows the variations of the COP and cooling power in the R744 system, with respect to the high-pressure variation, at four different gas cooler air inlet temperatures: 30°C, 35°C, 40°C and 48°C.

Figure 4.1 indicates that:

- Higher ambient (gas cooler air inlet) temperature gives lower COP and cooling power. In lower ambient temperatures, the sensitivity of the COP to the optimum high-pressure is more pronounced.
- 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 exact optimum values are shown in Table 3.2.
- The cooling power increases with high-pressure but there exists a maximum capacity.

¹Since the condenser (gas cooler in R744 system) is located in the front of the car and exchanging the heat directly with the environment, the gas cooler air inlet temperature is assumed to be equal to the ambient temperature. However in some particular conditions such as an idle car, which is placed in exposure of sun radiation, these two temperatures might differ a lot.

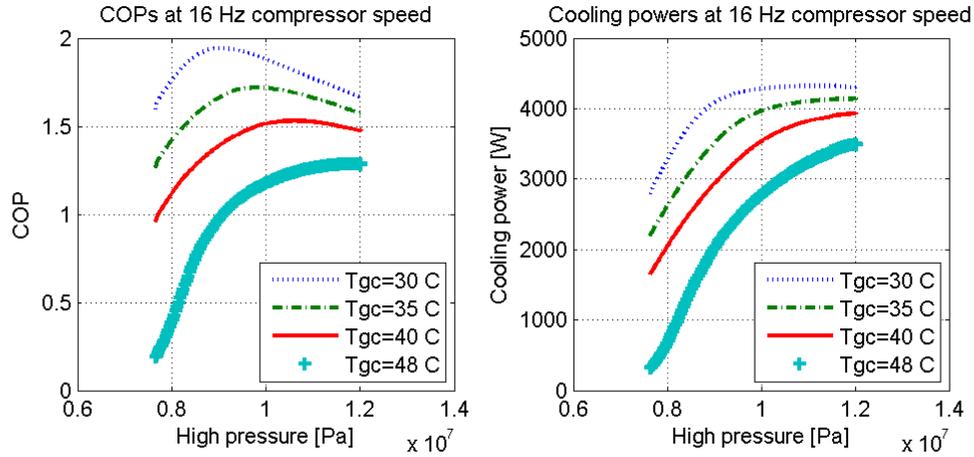


Figure 4.1: Comparison of COP and cooling power with the change of high-pressure

- The optimum high-pressure which brings about the maximum COP can not provide the maximum cooling power but the cooling power at this pressure is close to its maximum value especially at higher ambient (gas cooler air inlet) temperatures.

Table 4.2: Optimum high-pressure

Gas Cooler temperature[°C]	Optimum high-pressure at 16 Hz [bar]
30	90
35	97
40	106
48	118

Another parameter, which is expected to influence the optimum high-pressure, is the compressor speed because the power of the compressor and the low-pressure hence the COP and cooling power are nonlinear functions of the speed (Figure 4.2).

Figure 4.3 illustrates significant effect of the speed on the optimum high-pressure.

Based on Figure 4.3, it can be concluded that:

- Both the COP and cooling power are strong function of the compressor speed.
- Higher compressor speed gives lower COP but higher cooling power.

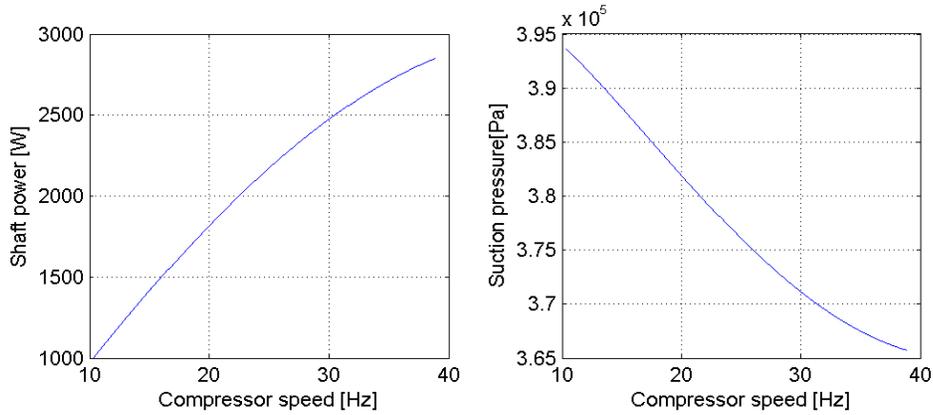


Figure 4.2: Effect of compressor speed on shaft power and low-pressure

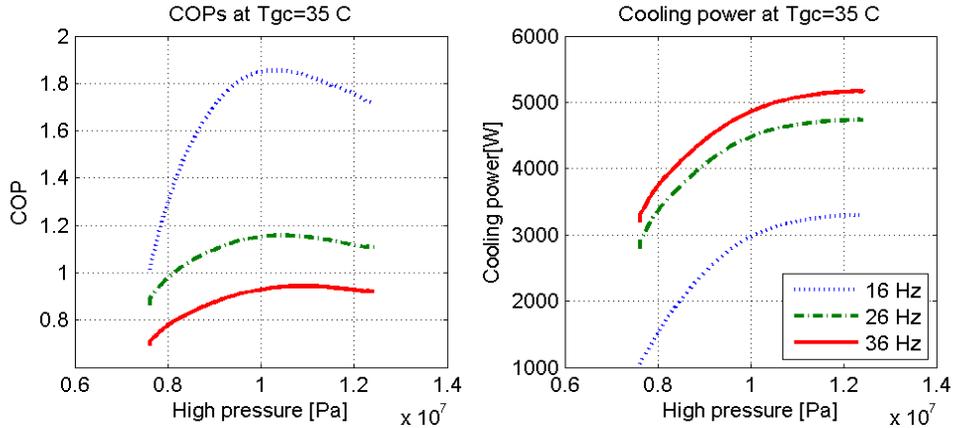


Figure 4.3: Comparison of COP and cooling power with the change of speed

- The sensitivity of the COP to the high-pressure variations is larger at low speeds and there exists a different optimum pressure for each speed. Table 3.3 gives the exact value of this optimum pressure for our system.

In [Yang et al., 2005] it is also shown that other boundary conditions (evaporator temperature, air humidity and flow rate) have negligible effect on the optimum high-pressure.

As it stated in the first chapter, a variable swash plate controller is going to be 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. Figure 4.4 shows that, although COP and cooling power reacts to it considerably, the value of

Table 4.3: Optimum high- pressure and speed

Compressor speed[Hz]	Optimum high-pressure at Tgc=35 °C
16	97
26	108
36	111

the optimum high-pressure does not vary too much.

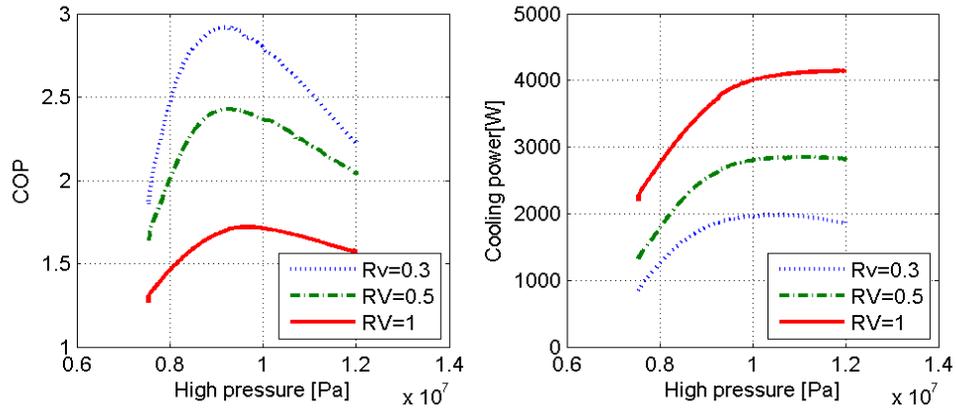


Figure 4.4: Comparison of COP and cooling power with the change of relative volume

Table 4.4: Optimum high-pressure and relative volume

Relative volume	Optimum high-pressure at 16 Hz [bar]
1	97
0.5	93
0.3	92

Therefore a high-pressure regulator which controls the refrigerant flow, based on the ambient temperature and compressor speed is suggested. 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. The model of a typical variable-Kv valve has been described in section 3.3. Figure 4.5 shows the effect of variation of Kv value of this valve on both pressures at a constant speed and temperature.

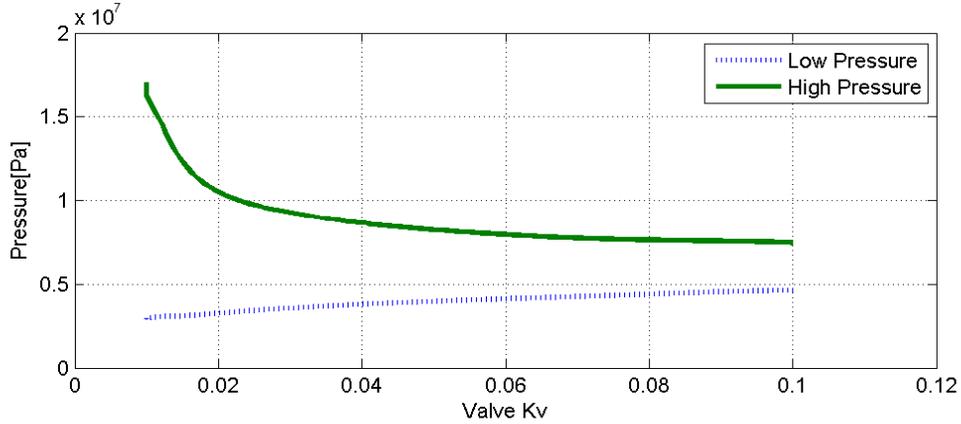


Figure 4.5: Pressures against the valve Kv

4.2 Evaporator Temperature Control

So far, besides COP considerations, getting maximum cooling power has been the main purpose of control of the cooler unit of the A/C system, especially in extreme conditions such as very high ambient temperature or high cooling loads; however in some other conditions, it is necessary to control the compressor power to keep the cooling power in the acceptable 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 acquires its driving force from the engine, its power is a function of the engine speed, which is a very fluctuating variable, thus some kind of control on compressor capacity is needed to compensate engine speed disturbances to satisfy the comfort requirements and avoid temperature variations. Furthermore, it is needed especially at higher speeds, which bring about an undesirable power of the compressor and a very cold evaporator.

Among the other methods proposed to control the compressor capacity, using a variable displacement compressor ([McEnaney and Hrnjak, 2005]) is the most attractive one. As indicated by the PH diagram and the discussion of section 2.3.1, cooling power is a function of suction pressure. Due to the fact that in the sub critical region,

where the heat rejection takes place isothermally, evaporator refrigerant and air temperature are linear functions of the low-pressure, so the swash plate control makes it possible to control cooling power and the evaporator temperature.

Changing the inclination of the swash plate (relative displacement or relative volume) of the compressor causes a change of the pressure ratio; namely high-pressure as well as low-pressure (Figure 4.6) but the effect of the expansion valve on the high-pressure is dominant (Figure 4.5), thus the relative volume is regarded as a low-pressure controller. Concerning the previous section, at a constant speed, it is acceptable to

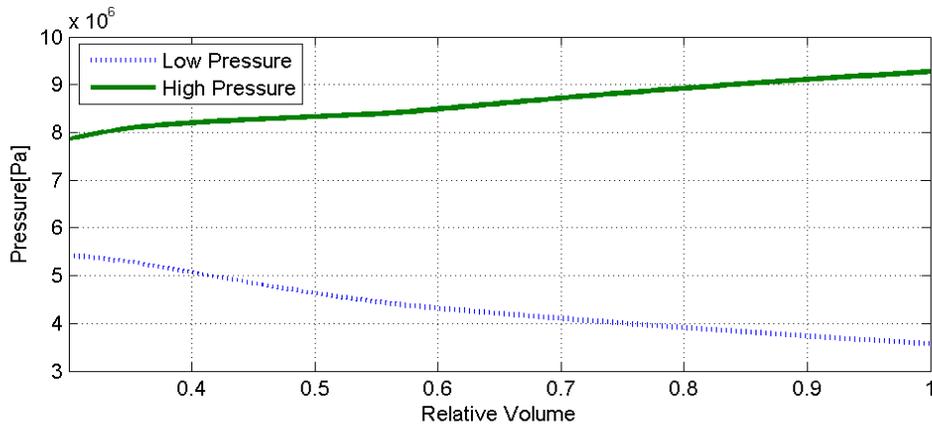


Figure 4.6: Effect of Compressor relative volume on pressures

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. Figure 4.7 shows the advantage of this control at a low speed and low cooling load.

Figure 4.8 shows that in these operating conditions, without any control on evaporator air outlet temperature control, it goes a couple of degrees below zero which is not desired for low cooling loads, but by the low-pressure control, it remains in an acceptable range. As mentioned earlier, in the case of speed changes, to control the evaporator air outlet temperature and optimize the cycle at the same time, a control strategy is needed which considers the cross coupling between these two loops.

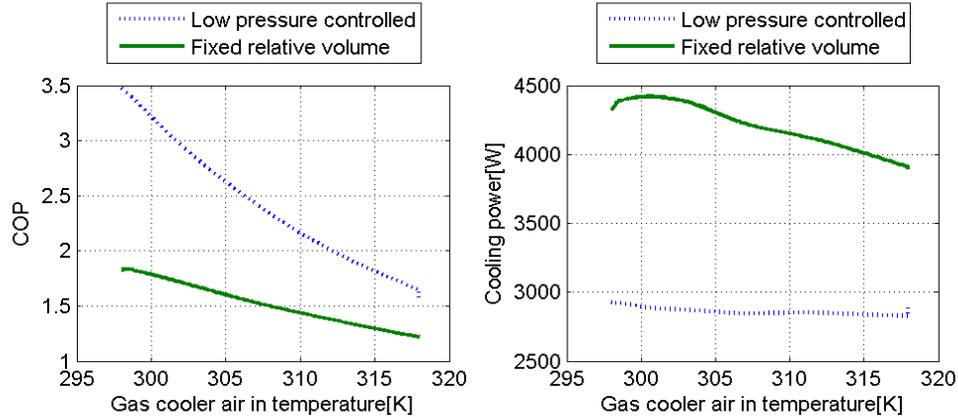


Figure 4.7: COP and cooling power in the low-pressure controlled cycle

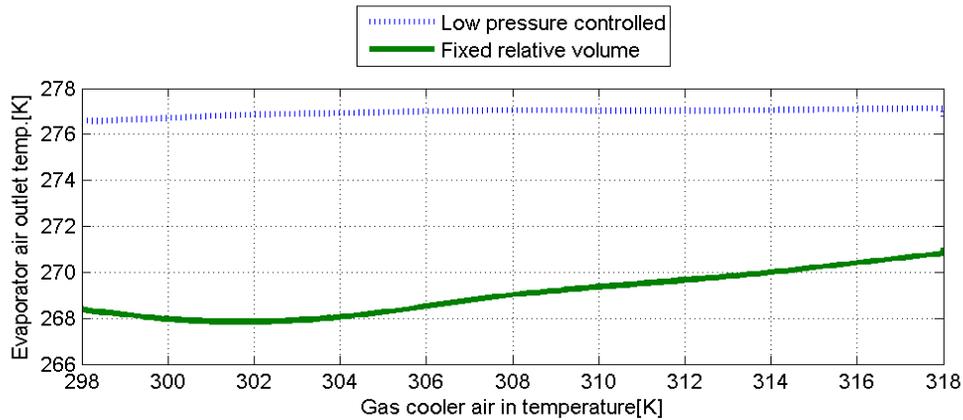


Figure 4.8: Evaporator air outlet temperature

4.3 Two-stage Orifice Valve

To remove the costs of the high-pressure controller and gas cooler temperature measurement device, a two-stage orifice expansion valve has been produced by Egelhof GmbH and modeled by DaimlerChrysler. This model has been modified by Modelon AB and is used in our simulations. This valve has an internal mechanism to change the Kv value of the valve based on the pressure difference between the low- and high-pressure side [Lemke et al., 2005]. It consists of a standard orifice and a bypass; as it is shown in the Figure 4.9 at pressure differences below 23 bar, the refrigerant flows only through the orifice. The bypass starts to open at 73 bar with a jump, and for

higher pressures, the Kv-value rises linearly. If this increase results in pressure difference higher than 73 bar, the bypass will start to open again and the Kv increases linearly with the pressure difference.

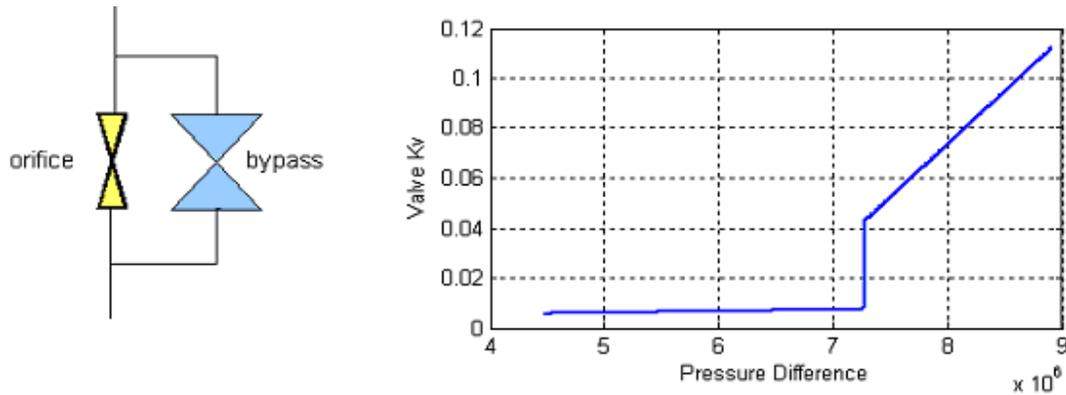


Figure 4.9: Two stage orifice valve

4.3.1 Performance of the Valves

To compare the operation of the controllable valve in an optimized cycle and a two-stage valve, all boundary conditions and the compressor speed are kept constant and the simulation runs three cases for both of them.

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

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 (Figure 4.10). 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 provide a very close Kv to the controlled Kv for most of this range(Figure 4.11).

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

In comparison with the previous case, at the higher loads, the cycle with two-stage orifice valve has a COP near to the optimum value but in the higher

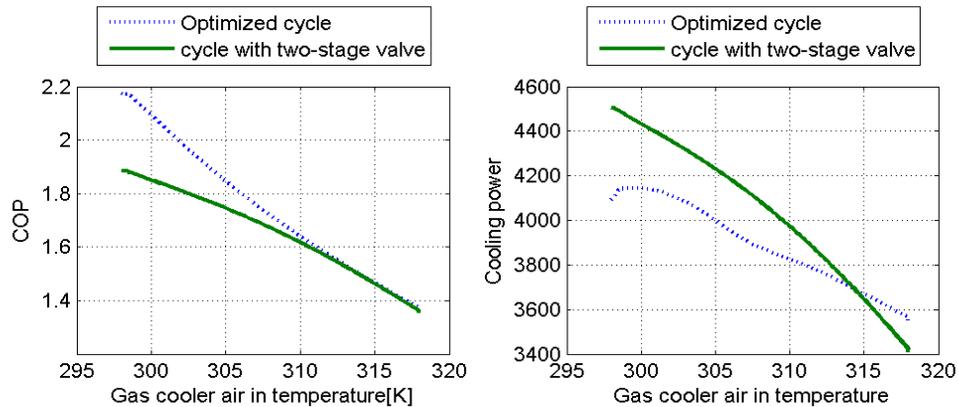


Figure 4.10: Comparison of the two valves, case 1

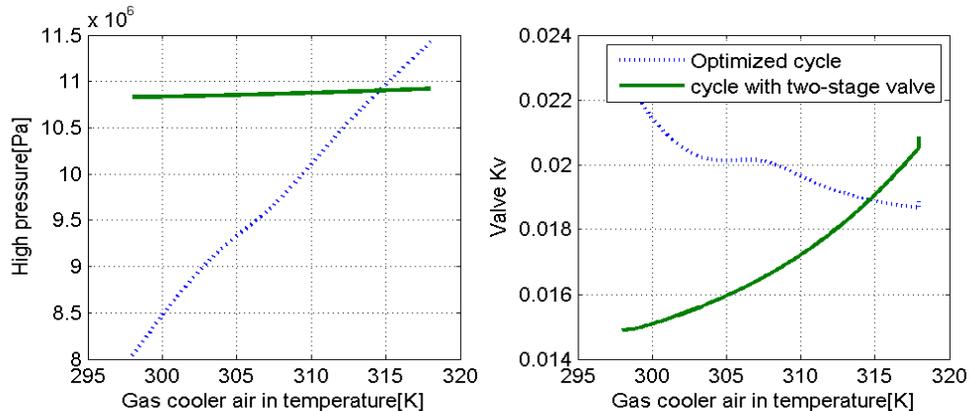


Figure 4.11: Comparison of the two valves, case 1

ambient temperature it cannot maximize the cooling power opposed to the optimized cycle.

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

3. Evaporator temperature controlled (low cooling load)

When the low-pressure is controlled via the relative displacement of the compressor, the COP is improved for both cycles. 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. The slope of the high-pressure and valve Kv are closer to the optimized cycle.

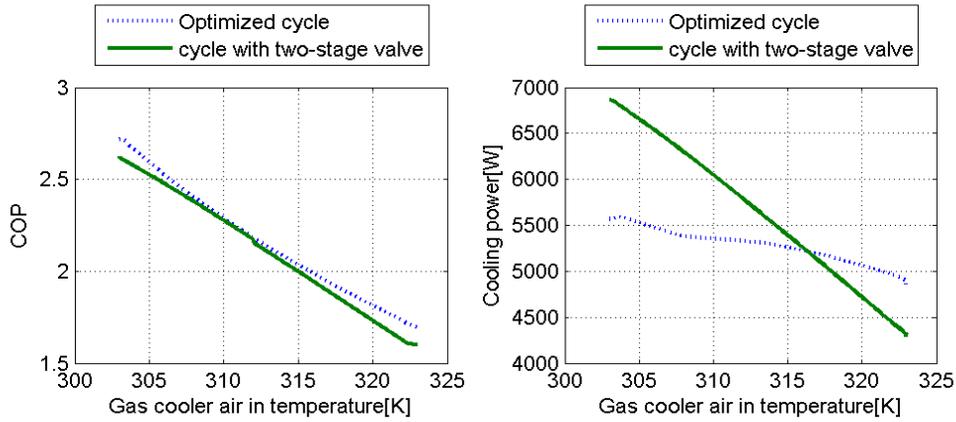


Figure 4.12: Comparison of the two valves, case 2

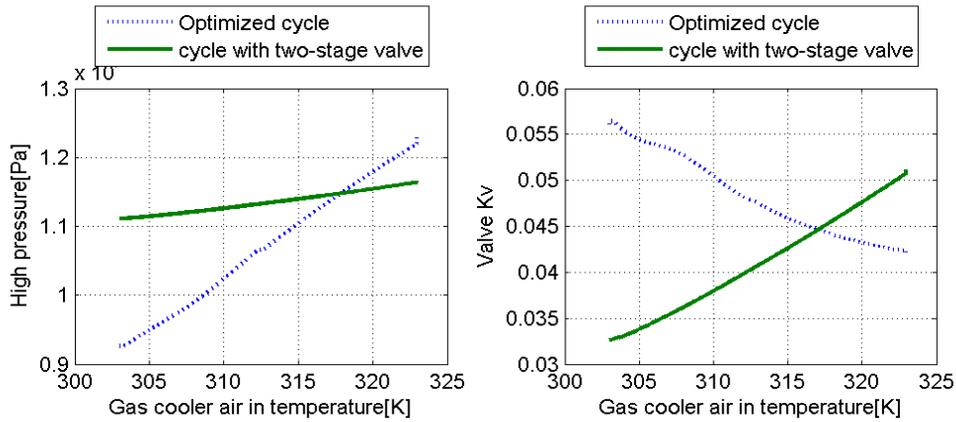


Figure 4.13: Comparison of the two valves, case 2

As has been seen in this section, for lower ambient temperatures, 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 kept as high as possible to improve the COP.

4.3.2 Limit Cycle of Two-stage Orifice

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 again, the decrease in high-pressure causes the closing of the bypass and this limit cycle continue until one of the inputs alter the pressure difference and

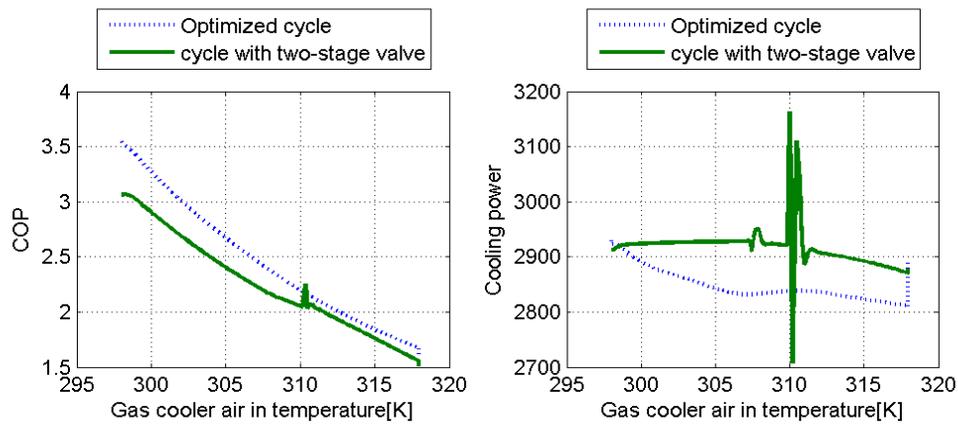


Figure 4.14: Comparison of the two valves, case 3

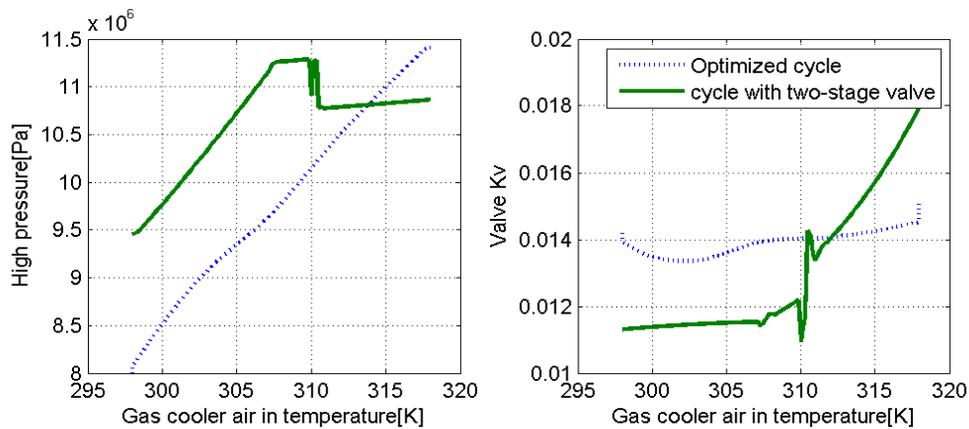


Figure 4.15: Comparison of the two valves, case 3

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 4.16 illustrates above explanations.

Figure 4.17 shows the portrait plot of the valve Kv against the pressure-difference.

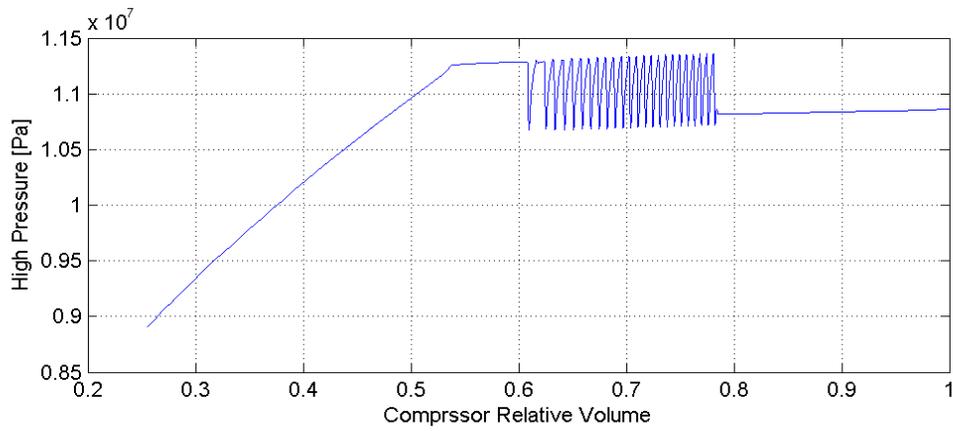


Figure 4.16: Limit cycle

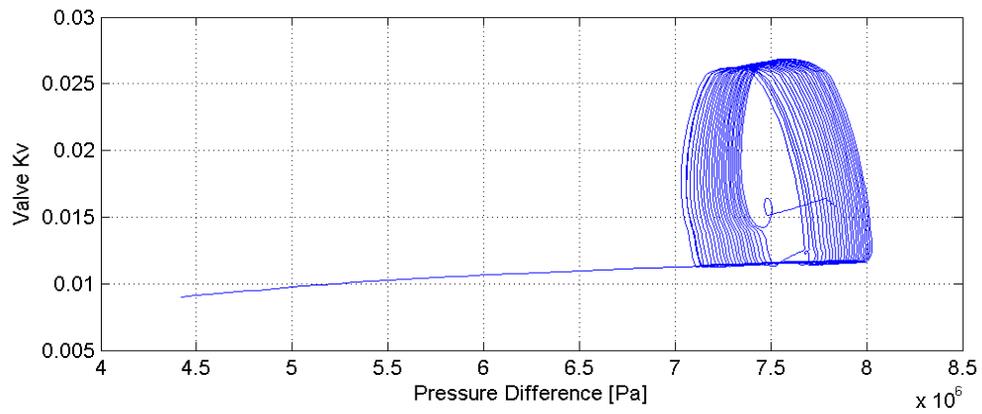


Figure 4.17: Limit cycle

Because of this phenomenon, the COP is subject to fluctuations, its highest deviation is about 20% less than its average value in this range. Therefore the limit cycle occurrence will reduce the expected average COP.

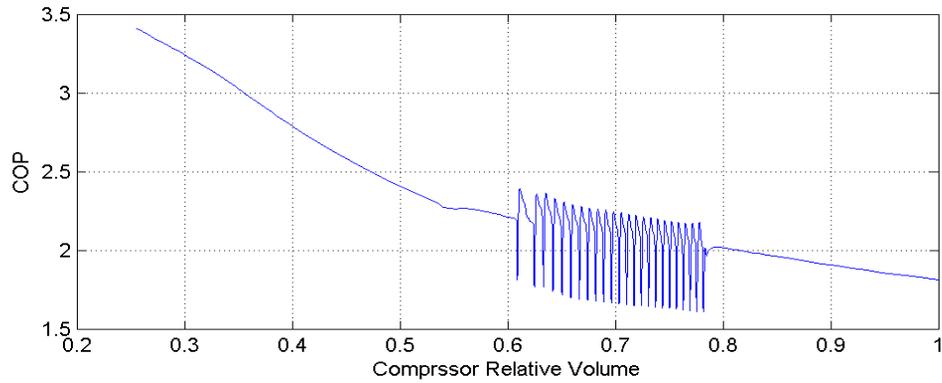


Figure 4.18: Limit cycle

However, this limit cycle is not a consistent behavior and its characteristic differ in different circumstances. The following observations demonstrate this statement: Assuming a low-pressure controlled cycle, the ambient temperature varies in the range of 30°C to 45°C and other operating conditions are kept constant. Figure 4.19 shows the 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.

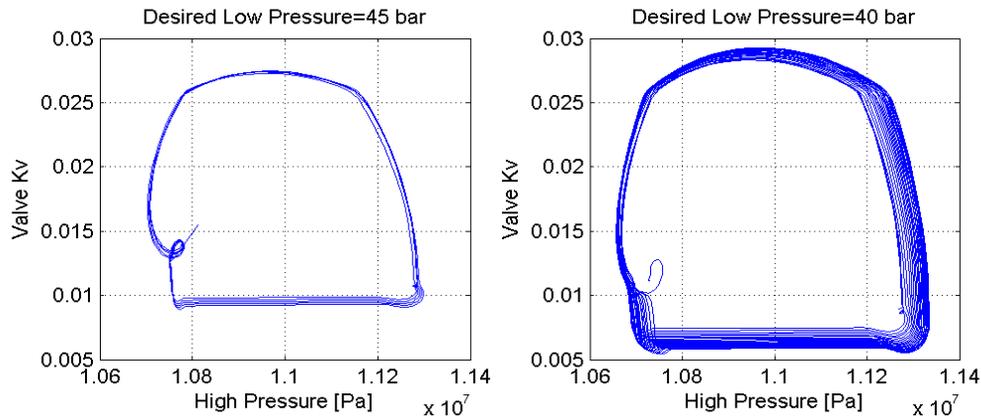


Figure 4.19: Portrait plot of the valves Kv against the pressure-difference

As depicted in Figure 4.19, the number of limit cycle orbits and the amplitude of the oscillations differ in these two different desired evaporator air outlet temperatures.

Other output parameters, which are correlated with the high-pressure, will also show this limit cycle. The effect on the outlet air temperature is negligible (less than 1°C in this case) and it is seen in Figure 4.20 that the low-pressure controller can remove the fluctuations. Therefore passengers do not sense the oscillations of the temperature. But the effect on the cooling capacity and COP is more considerable.

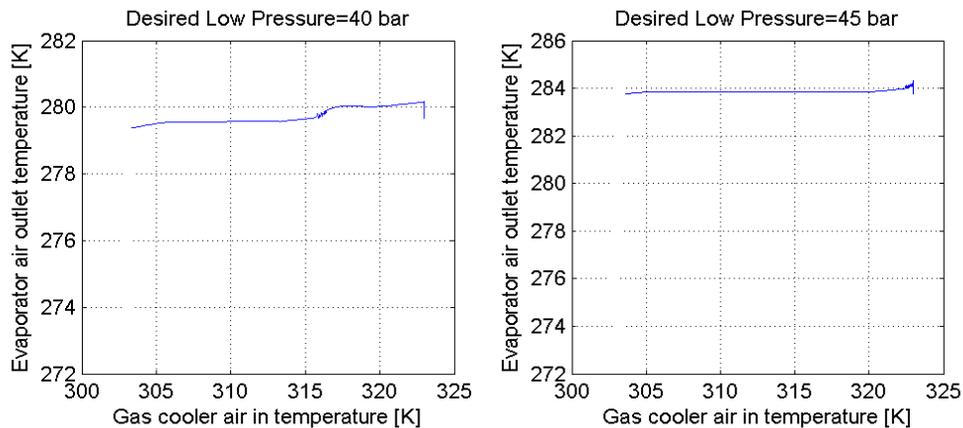


Figure 4.20: Temperature and limit cycle

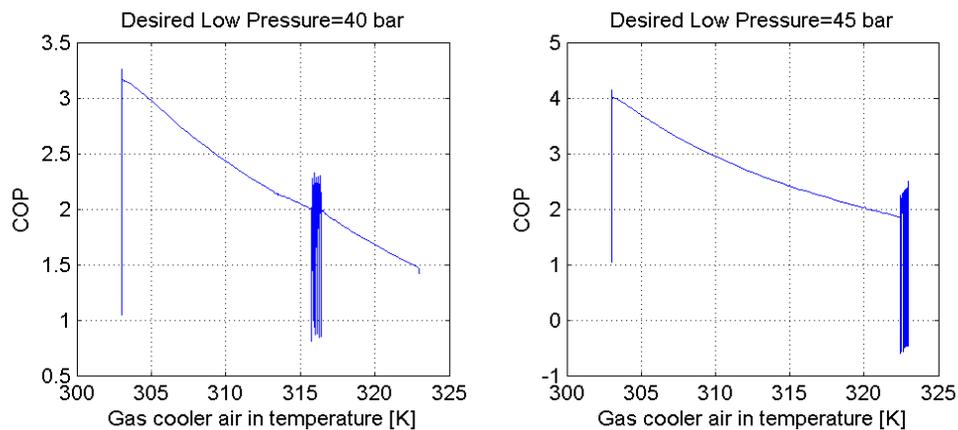


Figure 4.21: COP and limit cycle

In the temperature interval where this phenomenon happens, the highest deviation of the COP is about 50% less than expected average value.

4.4 Behr Alternative Controller

Behr GmbH & Co. KG, Stuttgart, is a systems partner for the international automotive industry who has developed different components as well as new control methods for R744 systems.

Lochmahr, Barushke and Britsch-Laudwein in Behr GmbH have introduced two types of improved controllers for the R744 system ([Lochmahr et al., 2005]). The first controller has been designed for the optimized system with a controllable valve and the second one for the system with a two-stage orifice valve. The advantages of these modified version, in comparison with simple controllers that we have seen in the previous sections, are not mentioned in the article. Therefor according to cascade control methodology recommended by [Åström and Hägglund, 2006] , the probable advantages of having these controllers will be verified in this section.

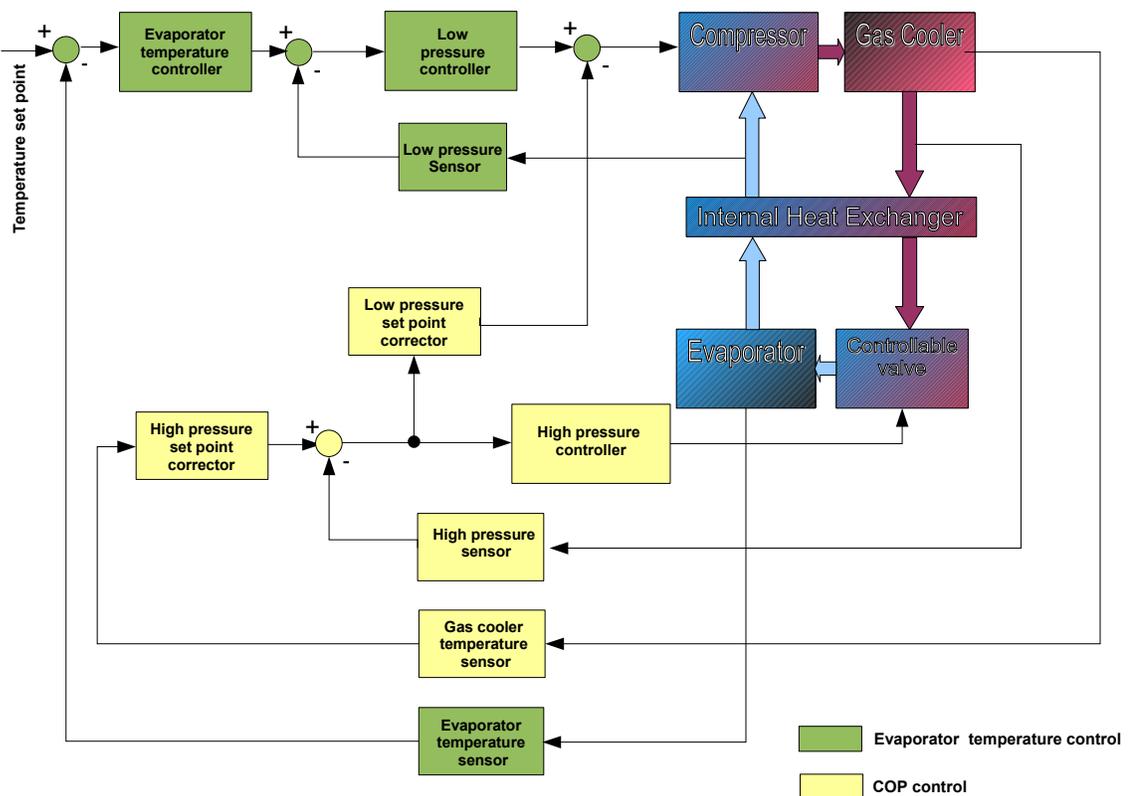


Figure 4.22: Control System for R744 cycle

In the first method, they use the same COP optimization approach mentioned in section 4.1, but instead of the low-pressure or temperature control, they propose a tighter control by nesting both the low-pressure and evaporator outlet air temperature control loops in a cascade loop. Figure 4.22 shows structure of this controller. Basic rules to have a reasonable cascade control, suggested by Åström and Hägglund, are examined for this controller:

- **Rule 1: There should be a well-defined relation between the primary and secondary measured variables.**

In the subcritical, two-phase region, the low-pressure and refrigerant temperature have such a relation. Since the evaporator outlet air temperature sensor is inserted into the evaporator, the relation between evaporator outlet air and refrigerant temperature is also linear. Therefore such a well-defined relation exists between the low-pressure and evaporator outlet air temperature.

- **Rule 2: Essential disturbances should act in the inner loop.**

Since the low-pressure and temperature are correlated according to balance principle in section 2.3.1, it is not possible to distinguish between the effect of disturbances on the inner and outer loop, except for engine speed disturbances which acts directly on the compressor power and changes the low-pressure firstly. Therefore this rule just holds true on the effect of engine speed disturbances.

- **Rule 3: The inner loop should be faster than the outer loop. The typical rule of thumb is that the average residence times should have a ratio of at least five.**

The thermal conductivity of R744 is very high and the design of the evaporator facilitates the system with a high heat transfer rate in the subcritical region; as a result of this fact, the air temperature dynamic characteristics are close enough to the low-pressure dynamics. Figure 4.23 shows this fact by illustration the relation between the low-pressure and evaporator air outlet temperature in an uncontrolled system, when the compressor speed is changing very fast.

Therefore for our system, using cascade controller gives no significant advantage and one of these control loops can be eliminated from the cascade loop.

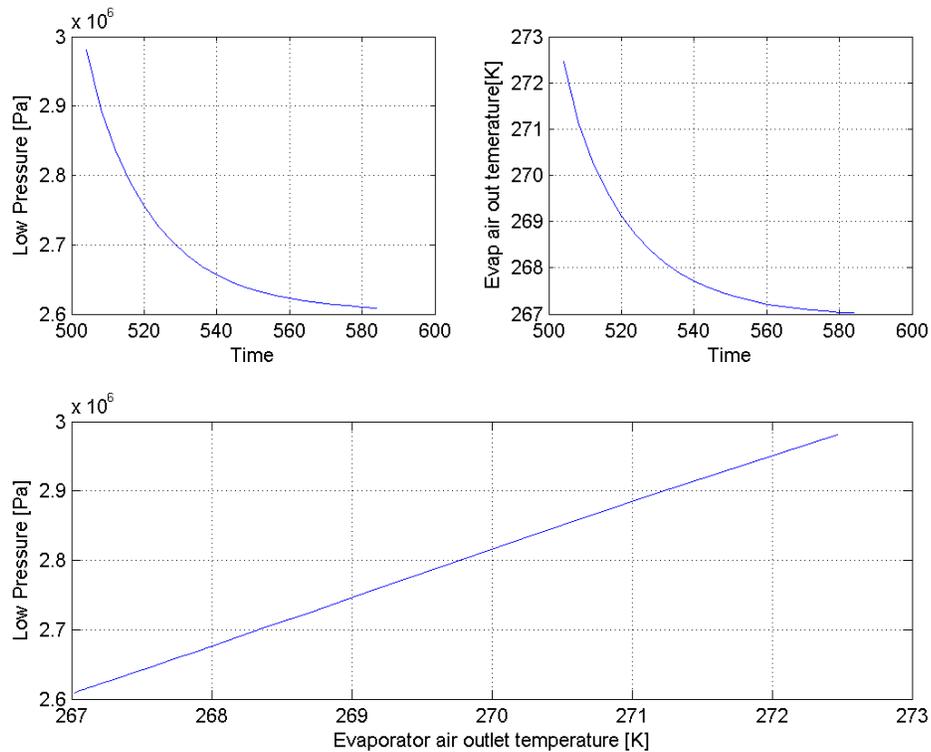


Figure 4.23: Evaporator temperature and low-pressure

In the second proposed controller, instead of the controlled valve, the two-stage orifice valve is used; therefore there is no direct control of the COP. They also suggest to enter a high-pressure controller in place of the low-pressure one in the inner loop, because the refrigerant high-pressure sensor required for controlling the high-pressure is already present for monitoring and protection functions in today's R134a circuits, so no additional sensors are needed and this plan remove the cost of low-pressure sensor.

The configuration of the system with this controller is shown in Figure 4.24.

The first rule of the cascade control is no more true here. Due to the fact that the evaporator outlet air temperature is not a direct function of the high-pressure, the idea of measuring high-pressure instead of low-pressure, causes some problems in the controller design that should be solved. A feedforward compensator has been proposed in [Lochmahr et al., 2005] to solve these problems. In the following argument

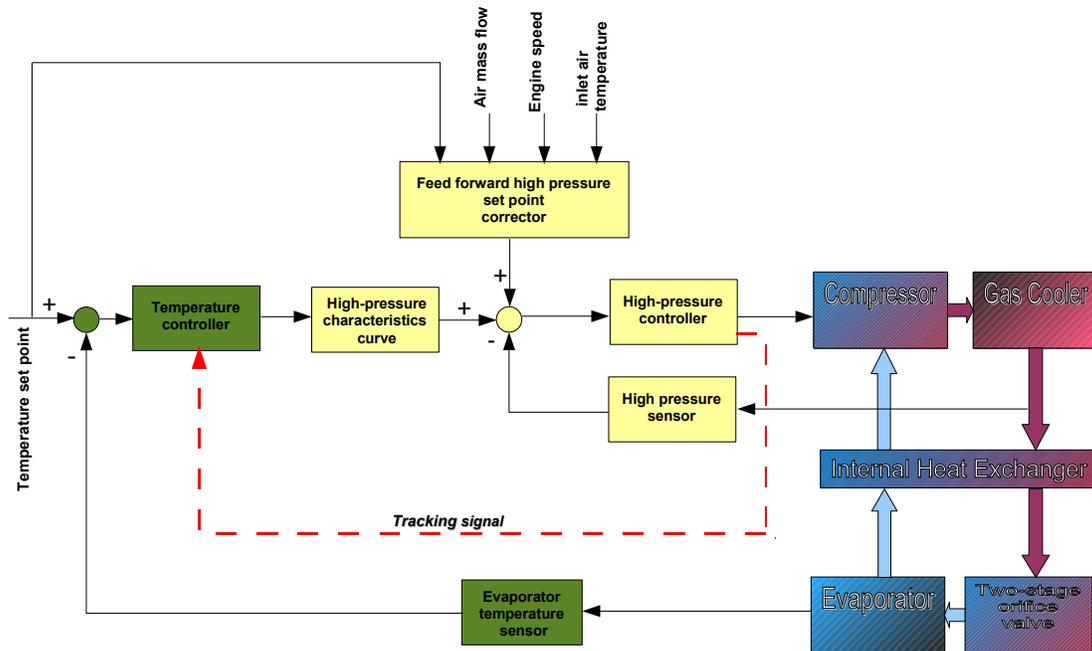


Figure 4.24: Alternative control system for R744 cycle with two-stage valve

we will show that the idea of adding the feedforward compensator to the system besides the temperature outer control loop will handle a part of these problems to some degree but it needs very accurate design and can not provide good robustness.

To investigate the relation between the high-pressure and evaporator outlet air temperature, the system input and disturbances can be divided in two groups:

1. Engine speed and any disturbances on the side of compressor and expansion valve.
2. Boundary conditions and any disturbances on the air side of the gas cooler and evaporator.

If any items of the first group changes, the system input power will change and the pressure lines on the PH diagram move in the opposite direction of each other. In this case, the high-pressure is just a function of the input power and not a function of the low-pressure. For example, any increase of the compressor speed, causes higher high-pressure and lower low-pressure which in turn decreases the evaporator temperature. Therefore by the high-pressure control, the input power of the cycle is maintained and

the effects of the first group of disturbances on the temperature can be suppressed, so the two first rules of cascade control are satisfied. But if any items of the second group changes, according to the balance principle of the refrigeration cycle in section 2.3.1, both the low and high-pressure will vary. In this case, the high-pressure variation is not a result of the input power variation, so when the controller tries to maintain the input power based on the high-pressure measuring, the evaporator temperature will vary in a wrong direction. Hence, the high-pressure controller set point needs to be corrected to give the proper control action. The role of the feedforward compensator in this configuration is to correct the high-pressure set point in proportion of the second group of parameters variations. In the [Lochmahr et al., 2005] proposed controller, four parameters are entered to feedforward block: engine speed, temperature set point, inlet air temperature and mass flow rate.

It is investigated that this design has these problems:

- Since the high-pressure controller (inner loop) is fast enough to compensate the effect of speed disturbances, insertion of the speed measured value in the feedforward compensator is not required. Furthermore, according to the multi variable modeling approach in [Rasmussen, 2002], high-pressure is a function of speed and four other states, besides this if the nonlinear behavior of the valve in this system is considered, the relation between speed and high-pressure is highly nonlinear, therefore adding a simplified feedforward transfer function for the speed may tune the high-pressure in a limited range of operating conditions but increases the risk of overcompensating in other conditions.
- The gas cooler boundary conditions are not measured in this scheme and not entered in the feedforward compensator, consequently before the outer loop reacts to the resultant temperature variation and compensate the effect of these disturbances, because of the fixed set point, the high-pressure controller acts falsely. For example, if the ambient temperature increases, then as a result of the balance principle for the gas cooler, the high-pressure will increase therefore the high-pressure controller decreases the compressor power and the cooling power while in this case, higher power is needed to maintain cabin temperature.
- Humidity of the cabin is also ignored in this design. Due to the fact that the outlet temperature is a nonlinear function of the air temperature and humidity, such a feedforward compensator can not cover a wide range of different humidities.

- The output signal of the secondary controller, i.e. the compressor relative volume, is limited and the integral action should be used in both the inner and outer control loops, therefore it is necessary to have a scheme to avoid *integral windup* [Åström and Hägglund, 2006]. This requires being able to inject a tracking signal into outer loop. The connection place of this tracking signal is shown in Figure 4.24 which should be added to the original configuration.
- 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 path, *bi-stability* phenomenon takes place. In this case, any disturbances which leads to small variance in the pressure-difference, cause 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 4.25 shows that a small disturbances of the pressure, pushes the system to another steady state and causes 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 acts against the main purpose of control.

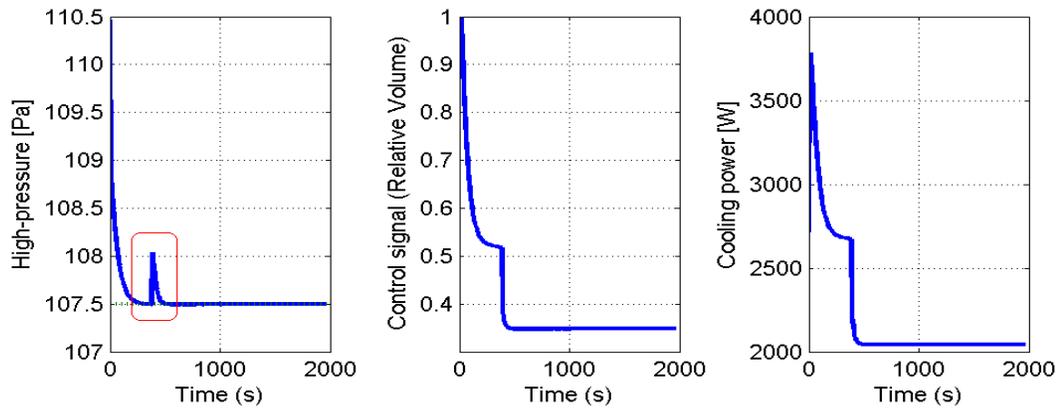


Figure 4.25: Cycle bi-stability

Chapter 5

Four-Zone Air-Conditioning

Modern luxury cars allow passengers to control a different climate in up to four climate zones. This requires the presence of two 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 to provide the passengers with their desired temperature in different zones. In the cooler unit, the high-pressure

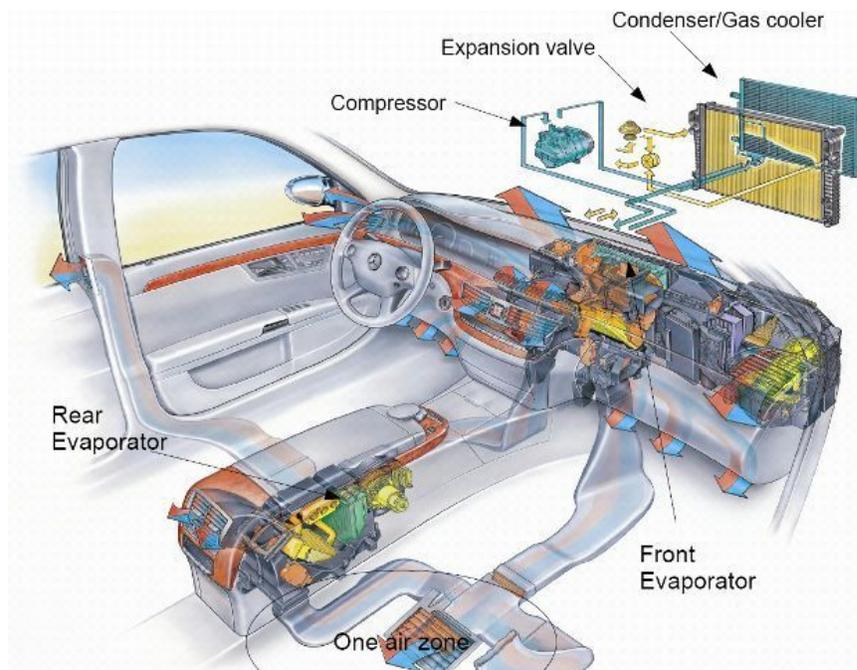


Figure 5.1: Four-Zone Air-conditioning of Mercedes-Benz S-class [*Behr GmbH & Co.*]

refrigerant splits and flows from two different expansion devices to the front and rear evaporator. Then the cooling capacity is divided between these 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 temperature, a model-based controller is needed to be designed to control the compressor relative volume based on the measured value of the outlet air temperature of both evaporators and two desired values (Figure 5.2).

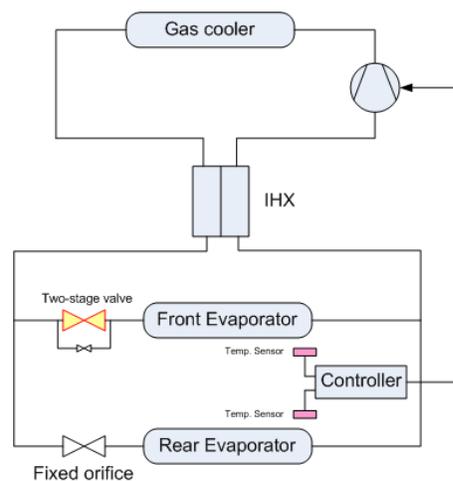


Figure 5.2: Two-evaporator temperature control

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 (The same SISO approach as for the one-evaporator system). Based on the balance principle in section 2.3.1, this will change the balance point of the rear evaporator low-pressure and this in turn, changes the front evaporator low-pressure. Since the temperature controller tries to fix the front evaporator low-pressure, it is expected that under certain cooling loads, this control plan develops limit cycle behavior. In this section some simulations will be done to investigate whether this phenomenon happens or not.

5.1 Two-Evaporator System Model

To model the four-zone system perfectly, a car cabin model is needed which includes the air interchanging among the different zones. Since we do not have such a complete model yet, the simple car cabin model of the A/C library is used to model the air mixing of the front and rear evaporator. In this model 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. Figure 5.3 shows the structure of this model.

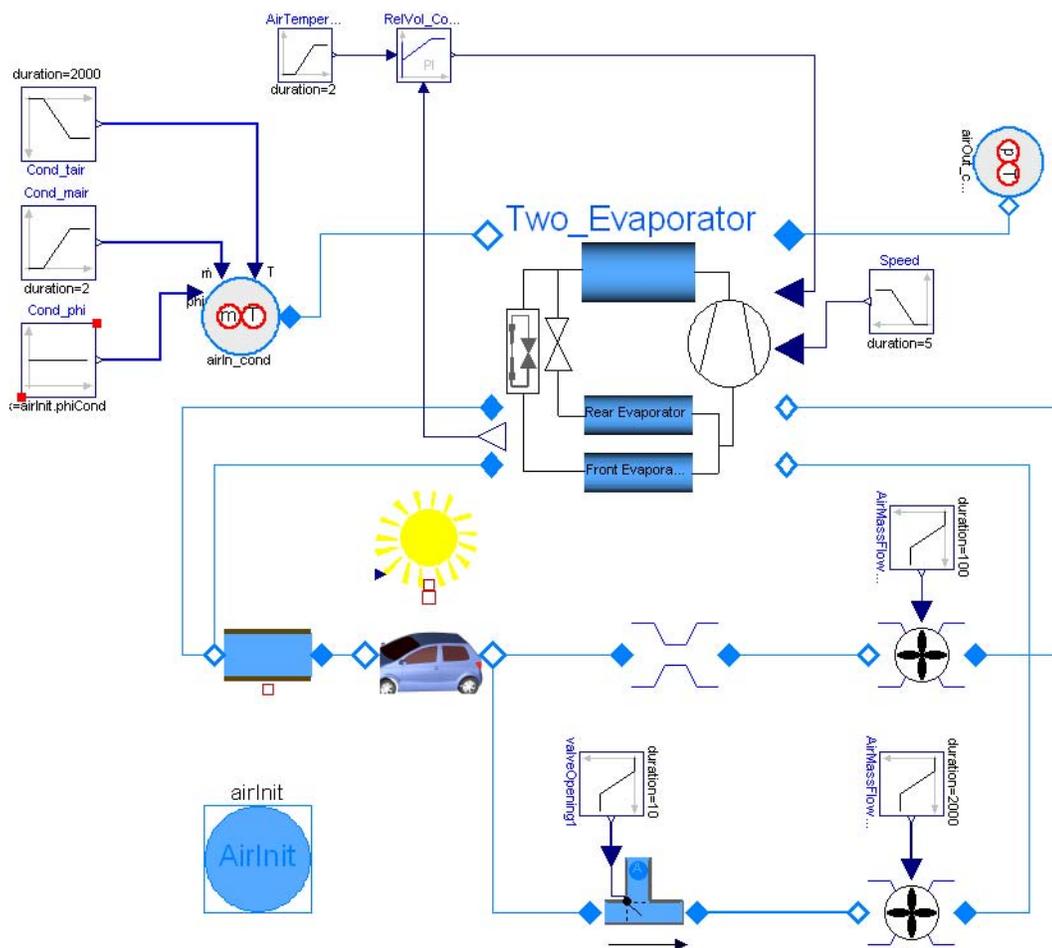


Figure 5.3: Model of the two-evaporator system

5.2 Simulations

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 (typical European climate conditions of Table 4.1). Figure 5.4 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.

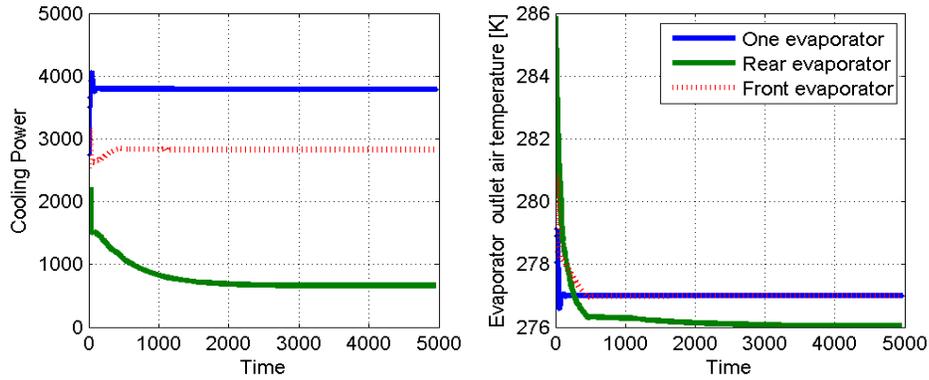


Figure 5.4: Cooling power distribution between two evaporators in comparison with one-evaporator system

Figure 5.5 shows the cooling power distribution against the ambient temperature. At higher temperatures, the pattern of distribution will change but it still provides acceptable cooling power for both evaporators.

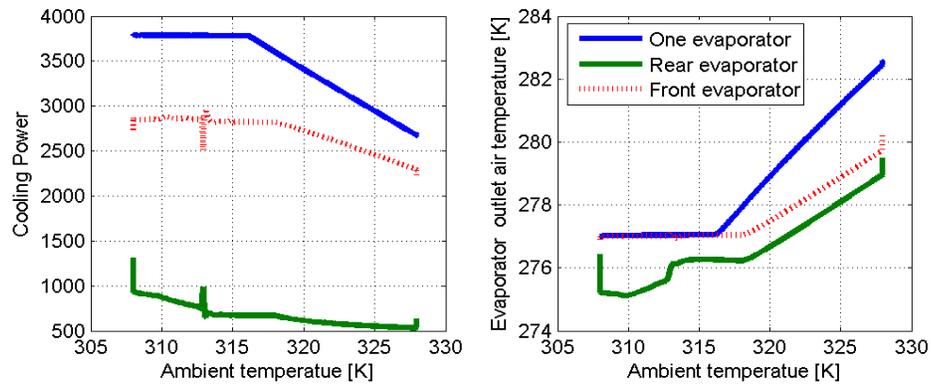


Figure 5.5: Cooling power distribution between two evaporators in comparison with one-evaporator system

Rear AirFlow Effect As we said earlier, it is possible to change the air mass flow around the evaporator in order to change the cooling power. The simulation has run in a limited range of airflow variation under two different cooling loads. Figure 5.6 shows the change of the rear evaporator cooling power when the air mass flow is changed at 5000 second. Figure 5.7 shows the cooling power variation against the air mass flow variation under a high and a low cooling load.

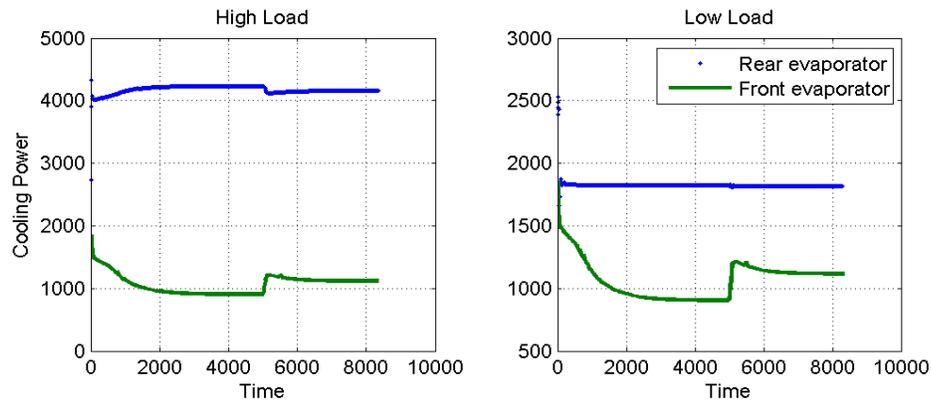


Figure 5.6: Rear evaporator air flow change

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 can change.

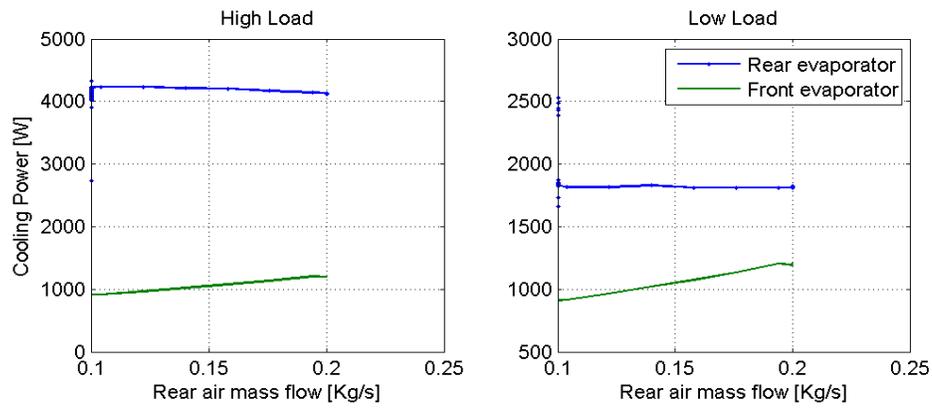


Figure 5.7: Rear evaporator air flow change

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 air mass flow changing.

Chapter 6

Summary and Conclusions

In this thesis some control issues of R744 A/C system have been studied. First of all, the problem of decoupling among different inputs to fulfil two control tasks, optimization and temperature control, have been verified. Decoupling between the COP optimization loop and the evaporator outlet air temperature control loop is possible for a wide range of different climate conditions and cooling loads by ignoring the effect of speed disturbances. Since at higher compressor speeds, the role of the high-pressure is less significant on the COP optimization, those two control loops should be designed at a low speed to provide better efficiency.

Secondly, the operation of the two-stage orifice valve has been studied and compared with a controllable valve. If a high COP is desired by using this valve, it is necessary to increase the setpoint of the temperature control loop, especially at lower cooling loads. If we do not need to use the full cooling power, the behavior of the cycle with two-stage valve is close to the optimized cycle. However, it can never provide the same efficiency. One other drawback of using this valve is the risk of trapping in a limit cycle at some operating conditions. This limit cycle has no significant effect on the sensed temperature by passengers but can reduce the average COP.

In the next step, the performance of the first and second generation controllers proposed by Behr GmbH have been studied. It has been shown that, in a first alternative,

one of the unnecessary cascade control loops can be removed. Design problems of the high-pressure control loop in the second alternative have also been detected, especially the drawbacks of the feedforward compensator design in the controller structure.

At last, a brief description of the two-evaporator system has been given, then through some simulations it has been concluded that the same approach of SISO control for the one-evaporator system is also possible to apply for this system, and in order to change the capacity of the rear evaporator, a variable speed fan can be used.

Future works

Since the A/C system with a two-stage orifice valve is highly nonlinear, proper tuning of the PID controller which is used for the capacity control is another issue which can be studied. It is expected that a systematic PID parameter optimization method increases the average COP for the whole driving cycle.

The structure and internal mechanism of the two-stage orifice valve can be modified to achieve better performance. For example, to make the COP closer to the optimized cycle COP, it is required that the valve opens at different pressures depending on different climate conditions. A tunable valve is suggested to make this possible. The problem of the limit cycle event can be also solved by changing the bypass opening mechanism.

For the two-evaporator system, the possibility of the COP optimization by using controllable valve has not been studied in this thesis. The drawbacks of the limit cycle problem of the two-stage orifice valve are not investigated either for this system. Therefore it is not proved that using a two-stage orifice valve is the best solution for this type of systems, and future investigation should be considered.

Bibliography

- [Åström and Hägglund, 2006] Åström, K. and Hägglund, T. (2006). *Advanced PID Control*. ISA-The Instrumentation, Systems, and Automation Society.
- [Fritzson, 2003] Fritzson, P. (2003). *Principles of Object-Oriented Modeling and Simulation with Modelica 2.1*. A John Wiley & Sons, Inc., Publication.
- [Gordon, 2005] Gordon, J. (2005). Variable displacement A/C compressor. *Motor Age*, 4:54–55.
- [He and Liu, 1998] He, X.-D. and Liu, S. (1998). Multivariable control of vapor compression systems. *HVAC R Research*.
- [Lemke et al., 2005] Lemke, N., Tegethoff, W., Kohler, J., and Horstmann, P. (2005). Expansion devices for R744 MAC units. *Vehicle Thermal Management Systems 7 Conference and Exhibition, SAE International*, 2041.
- [Limperich et al., 2005] Limperich, D., Braun, M., Prölß, K., and Schmitz, G. (2005). System simulation of automotive refrigeration cycles. *Proceedings of 4th International Modelica Conference*.
- [Lochmahr et al., 2005] Lochmahr, K., Baruschke, W., and Britsch-Laudwein, A. (2005). Control system for R744 refrigerant circuits. *ATZ worldwide*, 9:796–799.

- [McEnaney and Hrnjak, 2005] McEnaney, R. and Hrnjak, P. (2005). Clutch cycling mode of compressor capacity control of transcritical R744 systems compared to R134 systems. *Vehicle Thermal Management Systems 7 Conference and Exhibition, SAE International, 2023*.
- [Pfafferott and Schmitz, 2002] Pfafferott, T. and Schmitz, G. (2002). *Modeling and Simulation of Refrigeration Systems with the natural refrigerant CO₂*.
- [Rasmussen, 2002] Rasmussen, B. (2002). Control-oriented modeling of transcritical vapor compression systems. Master's thesis, Illinois University, Urbana.
- [Shah et al., 2003] Shah, R., Alleyne, A., and Ramussen, B. (2003). Application of multiple adaptive control to automotive air conditioning systems. *ASME International*.
- [Tiller, 2001] Tiller, M. (2001). *Introduction to Physical Modeling with Modelica*. Kluwer Academic Publishers.
- [Tummescheit, 2001] Tummescheit, H. (2001). *Design and Implementation of Object-Oriented Model Libraries using Modelica*. PhD thesis, Lund Institute of Technology.
- [Tummescheit et al., 2005] Tummescheit, H., Ebron, J., and Prölß, K. (2005). Air-conditioning a modelica library for dynamic simulation of ac systems. *Proceedings of 4th International Modelica Conference*.
- [Watanabe, 2002] Watanabe, S. (2002). *Automotive Air-Conditioning*. DENSO corp.
- [Wiessler, 2006] Wiessler, P. (2006). Air conditioning and global warming. *Automotive Engineering International, SAE International, 114:52–54*.
- [Yang et al., 2005] Yang, W., Fartaj, A., and Ting, D. (2005). CO₂ automotive A/C system optimum high pressure control. *Vehicle Thermal Management Systems 7 Conference and Exhibition, SAE International, 2022*.