TRANSIENT SIMULATION OF A DIRECT-EVAPORATING CO2 COOLING SYSTEM FOR AN AIRCRAFT

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Abstract

Integrating a direct-evaporating system in an aircraft requires an exact analysis of the operating conditions. The boundary conditions can vary significantly in temperature and humidity inside and outside the aircraft. The heat that is removed from the galleys by the refrigerant is rejected to the ambient air at the gas cooler. The presented paper shows the advantages of transient numerical simulations. Conditions which cannot be adjusted at the existing test rig can be investigated having the opportunity to see even processes like the mass movement in the system or the quality change in the heat exchangers. Furthermore it is possible to investigate operating conditions which could lead to the damaging of the test rig.

1 Introduction

Two main cooling systems are installed in a modern passenger aircraft. An air cycle system controls the temperature in the fuselage taking care of the heat loads generated by passengers, electronic equipment and solar radiation. A second cooling system is present to remove heat from the installed galleys to meet the required temperature for storage of food and beverages.

The development of this system towards weight reduction and efficiency optimisation has become more relevant in the last decade caused by rising health, comfort and reliability demands. This results in increased complexity and performance requirements. In this paper a direct-evaporating CO_2 system for galley cooling is discussed using a non-linear model written in the language Modelica.

1.1 State-of-the-Art On-board Cooling Systems

A simple way of cooling beverages and food in a galley is using dry ice. The lack of flexibility of this method led to active cooling devices. In the first step a so called "Air Chiller" (ACS) was introduced using R22 and later R134a as a refrigerant in a simple refrigeration cycle. The air circulating through the galley is cooled in the evaporator of the Air Chiller. Therefore they are positioned directly at the galley to avoid extensive air ducting, which would lead to an increased fan power demand, installation space and weight increase. The waste heat from the condenser is discharged against exhaust air from the cabin. This system is still a widespread technology for on-board cooling systems.

A more integrated system is the "Remote Chiller System" (RCS). This system consists of a refrigeration unit for the cold production and a secondary loop for the cold distribution using the non-toxic and non-flammable liquid Perfluoropolyether. The coolant is lead through 2 to 4 refrigeration units and supplies 1 to 4 galleys in parallel. An "Air Chill Unit" (ACU) is located at the galley including a coolant control valve, a fan and a heat exchanger. This system offers more flexibility in terms of integration space and temperature levels compared to the Air Chiller System, since the ACS offers only one constant air outlet temperature.

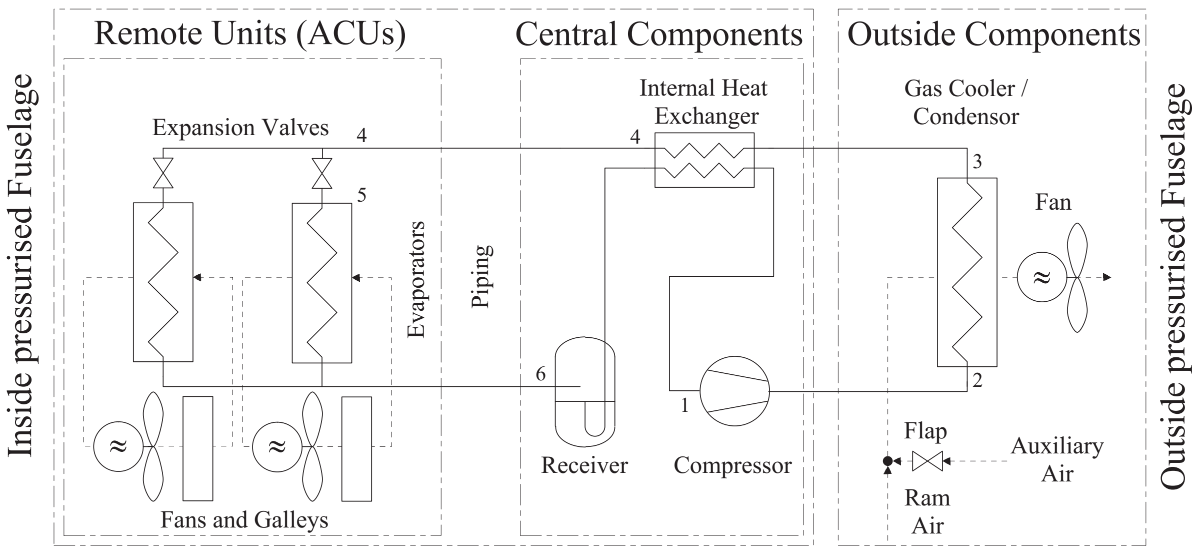

Fig. 1 CO2 direct evaporating system

The next generation and even more integrated on-board cooling system is the "Supplemental Cooling System" (SCS). A centralised R134a vapour cycle located in centre section of the aircraft generates the cooling capacity for two redundant distribution lines which supply the complete aircraft using the same coolant as the RCS. Contrary to the previously described cooling systems the condenser is connected to the ambient air outside the pressurised fuselage. The ACU is similar to that of the RCS system. In addition to galleys also electronic equipment can be cooled by the SCS.

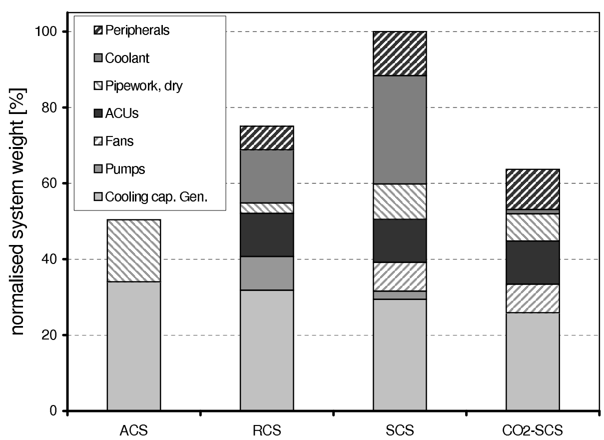

Fig. 2 Weight estimation for on-board cooling systems

1.2 Direct evaporating CO_2 cooling system

A slightly modified approach is realised by the centralised CO_2 system in Figure 1. It is defined as a single loop direct evaporating system with CO_2 as refrigerant/coolant following the Lorentzen Process. The heat transfer at the galleys is realised similar to the Remote Chiller and Supplemental Cooling System. For heat discharge at the gas cooler, ambient air is used.

The potential advantages using CO_2 as a refrigerant are its high specific cooling capacity and the low Global Warming Potential (GWP=1). The refrigerant weight can be reduced and smaller pipe diameter can be used. Although this effect is partially leveled off by the wall thickness due to the high operating pressure the comparison of the system weight in Figure 2 shows, that the CO_2 system is second behind the Air Chiller System. In this Figure the normalised overall weight for all mentioned technologies are shown designed for a long range reference aircraft. The mass of the CO_2 system is based on several assumptions and has an uncertainty of approximately ± 10 %. As there is no secondary loop with an inefficient coolant, the system offers more functionality, ef-

ficiency and dynamic capabilities. To get more knowledge of the functionality and dynamic behaviour of this system a test rig and numerical simulations are used.
The main disadvantage in regard to using this system in an aircraft is the high operating pressure of up to 130 bar in ground conditions. Due to the fact, that the piping of the cold distribution is installed close to the passengers or essential aircraft systems, effort has to be put into protection against critical failure conditions. Current investigations are dealing with the minimisation of such risks.

2 Model of a three consumer direct evaporating CO2 system

The complete model presented in this paper is put together using sub-models which represent the components used in a 3 consumer test rig erected at the department of Aircraft System Engineering [4]. These components are mainly used in automotive applications.
At the Applied Thermodynamics Group of the TUHH a modeling library named ACLib based on the ThermoFluid library, see [5] [1], was developed in a joint research project by DaimlerChrysler AG, Airbus Deutschland GmbH and the Hamburg University of Technology. The aim of this project was to investigate the transient behavior of complex air conditioning systems [3].
This library is written in the describing language Modelica. Physical models can be easily created due to the fact that the equations can be written non-causal and the language supports object-oriented constructs, which enables the reuse of models.
Partly based on the ACLib a more user friendly library named AirConditioning library is being developed by the company Modelon AB. This library is mainly used for the buildup of the presented three consumer system (Figure 3). The three evaporators a named Galley2, Galley7 and Galley1c. In the next subsections the sub-models are explained in brief.

2.1 Heat exchangers

The heat exchanger model has to capture the transient and steady-state behavior of the used compact heat exchanger which consists of flat tubes with micro channels in combination with louvered fins on the air side. The approach used in the AirConditioning library requires no experimental input data for a wide range of applications. This is achieved by using physical parameters and heat transfer correlations on the refrigerant and air side. The geometry of the heat exchanger is collected via a dialog window and a single pipe approach combines all parallel refrigerant flows through the component in a single flow with variable cross section, resulting in an array of cross flow elements. This pipe is discretized using a finite volume method. The heat conduction in the wall is modeled one-dimensional and perpendicular to both fluids, longitudinal conduction is neglected. The air side heat transfer can be calculated using a analytical or a discretized method. The second method enables a calculation of condensing water from the humid air. The dynamic and steady state results of the test rig have been used to validate the heat exchangers models used in this paper.

2.2 Compressor and Expansion valve

A quasi-stationary model of an Obrist C99 compressor is used in the model. Three functions describe the compressor characteristic: the effective volumetric efficiency, the effective isentropic efficiency and the isentropic compressor efficiency [2]. The displacement is fixed to 33.5 cm^2.
The mass flow through the expansion valve is determined using the equations from the DIN EN 60534-1. A characteristic line given by the manufacturer of the used expansion valves shows the interrelationship of the valve lift and the flow coefficient *kv*. These equations characterize the valve quasi-stationary.

2.3 Receiver

The receiver is situated in the low pressure part of the system and has the task of storing refrig-

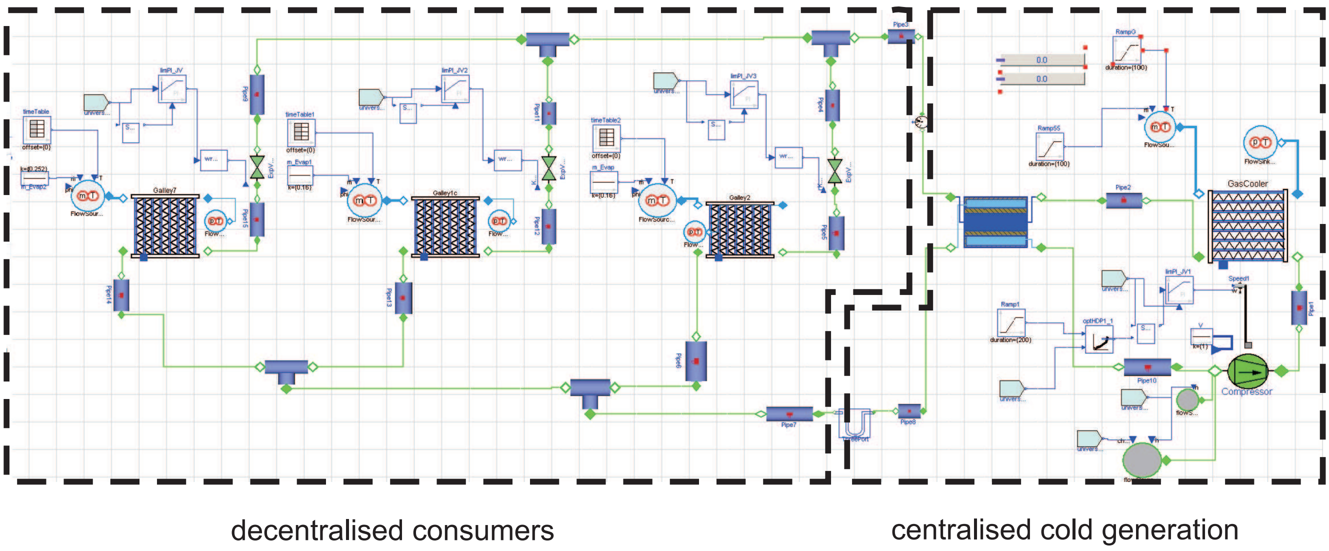

Fig. 3 Dymola/Modelica model of a three consumer direct evaporating CO2 system

erant in order to run the system under various conditions. The receiver is modeled as an upright standing cylinder with a refrigerant inlet at the top and an outlet through an u-bend. The u-bend has a small hole at the lowest point which is in most operation conditions in the liquid phase so that collected lubricant can be drawn out of the receiver (the influence of the lubricant is not taken into consideration). It is assumed that the two phases in the receiver are in equilibrium. The outlet properties depend on the liquid level, the height of the outlet and the flow resistance ratio of the u-bend and the hole in the bend.

3 Simulation of Ground and Flight Case

The aircraft is operating in a wide range of climatic conditions. To be able to design a system which meets the requirements three design cases have been set up.

- Design Case 1: Tropical Ground Case
  34°C, 76%rH, 0.1013 MPa → outside
  29°C, 70%rH, 0.1013 MPa → inside

- Design Case 2: Desert Ground Case
  40°C, 13%rH, 0.1013 MPa → outside
  34°C, 13%rH, 0.1013 MPa → inside

- Design Case 3: Flight Case
  ISA+16°C, cruise altitude 35kft, velocity 0.82 Ma → outside
  23°C, 13%rH, 0.076 → inside

To achieve the required air temperature in the galleys the air temperature at the evaporator outlet is controlled to -1°C using the expansion valves as regulating devices.
The humidity of the cabin air cannot be neglected. Condensing humidity can be modelled correctly, if the air side is discretized. The disadvantage of doing this is the enlargement of the equation system and in result the increasing of the simulation time. Therefore dry air is used. The effect of the condensing water is taken into consideration by an increased required cooling capacity. The cooling requirements can be seen in Table 1.

The air inlet temperatures at the evaporators are adjusted to the cooling requirements and kept constant during the simulations. The air mass flow at the evaporators is also kept constant.
The control strategy of the system depends on the ambient temperature at the gas cooler. In the Ground Cases ($T_{amb} > 28°$) the refrigerant process is transcritical. Contrary to subcritical processes, this enables a flexible adjustment of the high pressure, leading to an optimisation

Required cooling power				
	G1c	G2	G7	
design case 1	1,21	0,66	2,18	kW
design case 2	0,97	0,515	1,79	kW
design case 3	0,805	0,435	1,505	kW

Table 1 Required cooling capacity at the galley interface

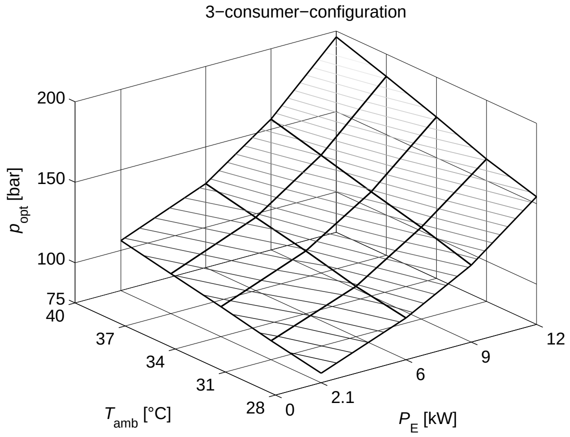

Fig. 4 Optimal pressure for the 3 consumer configuration

problem. Maxima of coefficient of performance (COP) exist for each set of varying ambient conditions.

Using the simulation model a characteristic diagram, shown in Figure 4, for the optimal pressure with maximum COP was generated. This was achieved by simulating the 3 consumer configuration with different ambient conditions and heat loads at the evaporators. Controlling the compressor speed according to the optimal high pressure and the expansion valves according to the evaporator air outlet temperature the processes shown in Figure 5 are obtained for Design Case 1 and 2. The COP in Design Case 1 is 1.2 and 1.5 in Design Case 2 at a specific charge of 300 kg/m^3.

The control of the system in the Flight Case condition ($T_{amb} << 28°$) is different. In the simulations for the Design Case 3 the high

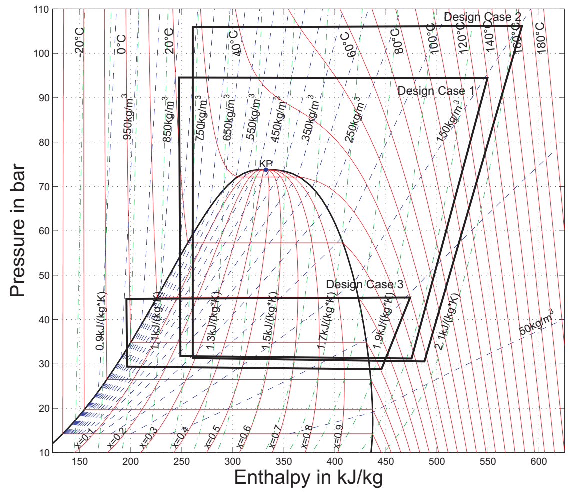

Fig. 5 Design casel 1,2 and 3

pressure has been fixed to 45 bar, controlled by the compressor speed. The sub cooling in the gas cooler/condenser is controlled by the air mass flow at the gas cooler/condenser. The resulting process can be seen in Figure 5. The COP achieved in Design Case 3 is 9.8 at a specific charge of 300 kg/m^3.

3.1 Ambient temperature change at the gas cooler

The air temperature at the gas cooler inlet is changing significantly between the ground cases and the flight case. It is therefore interesting to see the changing of the parameters during climb. The process is getting subcritical when the air temperature is getting under the critical temperature of CO_2.

As mentioned before the high pressure is controlled by the compressor speed. To get a first impression the characteristic diagram is extrapolated to lower temperatures. The high pressure is decreasing with the temperature until the lower limit of 45 bar is reached. The change of the gas cooler air inlet temperature and the high pressure set point can be seen in Figure 6.

During the pressure and temperature change the refrigerant mass is first transported into the receiver until the high pressure and air temperature are both constant again. The refrigerant mass is then moving back into the gas cooler. This can be seen in Figure 7 and 8.

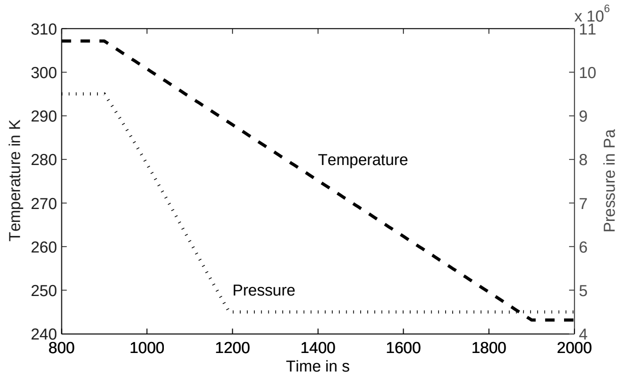

Fig. 6 Air temperature at the gas cooler and high pressure change

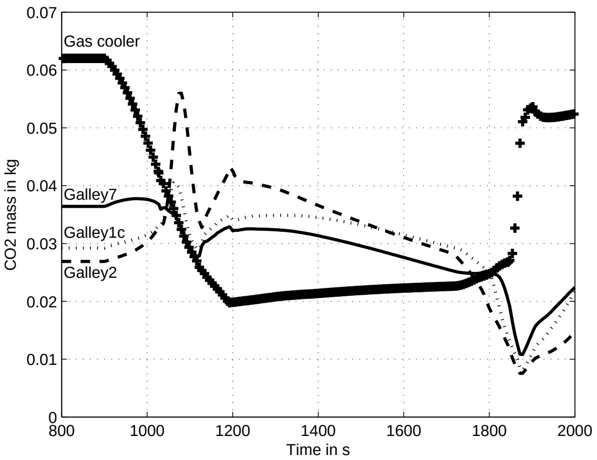

Fig. 7 CO_2 mass in the main components (ambient temperature change at the gas cooler)

The temperature controls at the evaporators cannot handle this fast change, see Figure 9. Because of the rapid change of the high pressure set point, the heat rejection at the gas cooler is first decreasing leading to an increase of the air temperature at the evaporator. In second part of the simulation the heat rejection at the gas cooler increases leading to an decrease of the air temperature. The heat fluxes can be seen in Figure 10.

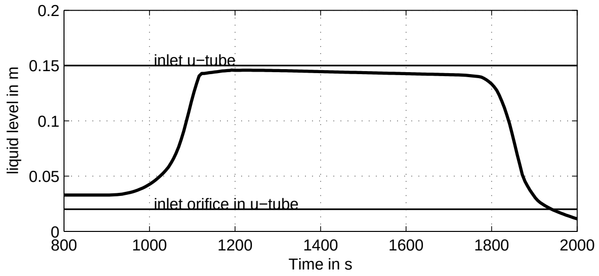

Fig. 8 Liquid level in the receiver (ambient temperature change at the gas cooler)

At constant high pressure and air temperature the controller for the galley temperature reaches the set point again.

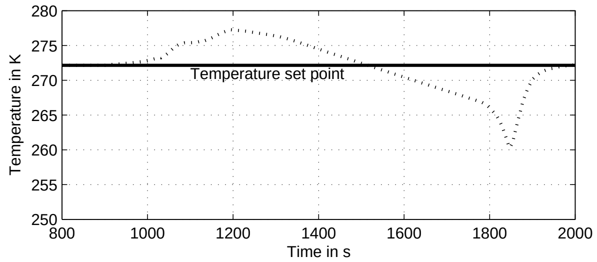

Fig. 9 Air outlet temperature at a evaporator (ambient temperature change at the gas cooler)

Resulting from the changes of the heat fluxes at the evaporators and the power of the compressor the COP, see Figure 11, is decreasing to a minimum at simulation time 1200s and then increases to a maximum of about 9.8 at the end of the simulation. The sub cooling controller reduced the mass flow at the gas cooler from $\dot{m}$ = 0.25 to 0.067 kg/s. This has a direct and positive effect on the aircraft drag.

To get the required cooling performance also in climb conditions the coupling of the air temperature at the gas cooler inlet and the high pressure set point has to be investigated in more detail. Due to fact that the properties of CO_2 changes rapidly at the critical point numerical problems occur making it impossible to change the high pressure decrease freely. Solving this problem

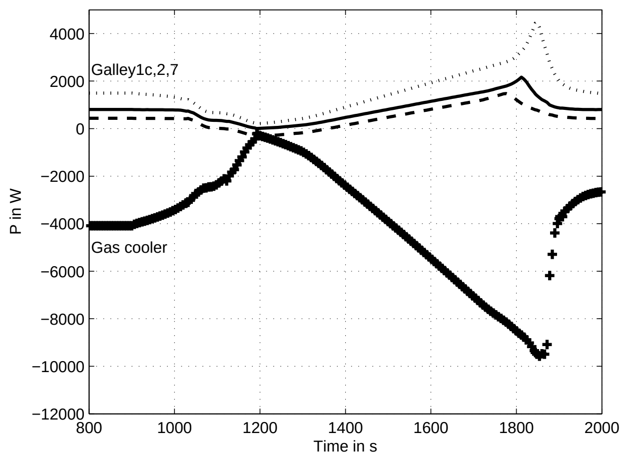

Fig. 10 Heat flow rate at the evaporator and gas cooler (ambient temperature change at the gas cooler)

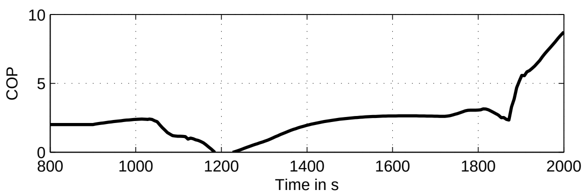

Fig. 11 COP (ambient temperature change at the gas cooler)

should make it possible to get a control for the climb condition.

3.2 Charge level study of Design Case 1

The charge level influence is important because of two main reasons. The first reason is the design of the receiver volume. The system should only contain the minimum amount of refrigerant mass to achieve all requirements. The second reason is a maintenance issue. Small leakage should not lead to a failure of the system. The correct choice of the receiver volume in addition with the charge level has influence of the COP and maintenance rates. Changing the total mass of the system as shown in Figure 12 has no effect of the evaporator air outlet temperature in the simulation interval.

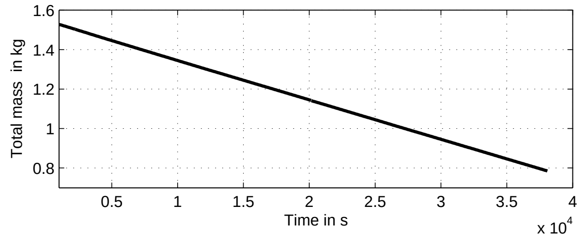

Fig. 12 Total mass in the system

The COP increases until there is no liquid in the receiver any more, see Figure 13 and 14. The high pressure control has to increase the compressor speed to keep the high pressure set point. This leads to an increase of the CO_2 discharge temperature, see Figure 15, at the compressor which would lead to a shut off in a real system.
The pressure/temperature of the CO_2 in the evaporator is decreasing leading to a high temperature difference between air and CO_2 and therefore results in a inefficient heat exchanger, see Figure 16.

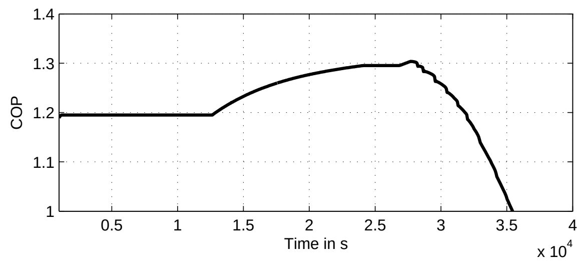

Fig. 13 COP, emptying process

4 Conclusion

A transient numerical model offers the opportunity to investigate the direct CO_2 evaporating system in conditions which cannot be adjusted at the test rig. As shown in this paper low temperatures and temperature changes can be investigated having the possibility to see all parameters also in the components like quality distributions and mass. Moreover intensive

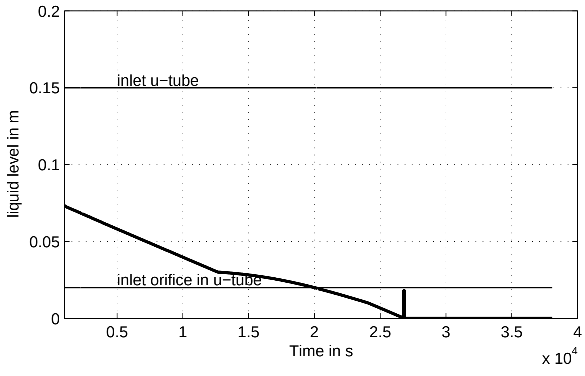

Fig. 14 Liquid level in the receiver, emptying process

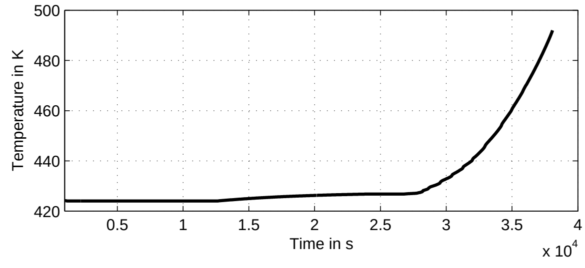

Fig. 15 Compressor discharge temperature, emptying process

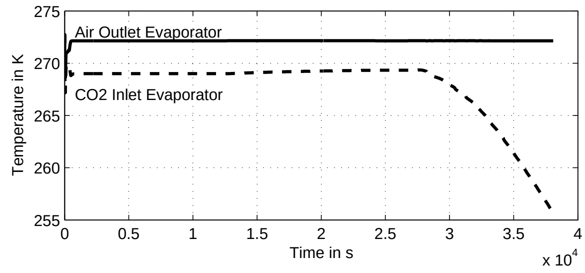

Fig. 16 Temperature at the evaporator, emptying process

parameter studies to get for example a characteristic diagram for the COP can be performed.
The charge level study has the great advantage that the change of the discharge temperature at the compressor can be investigated before experimental studies take place.
Effort has to be put in some numerical issues. The simulation of the presented 3 consumer model has required some time to find for example the correct discretisation to get correct results on the one hand and a stable model on the other hand. The simulation around the critical point of CO_2 is still a problem.

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