COP-OPTIMISED PRESSURE CONTROL FOR A CENTRALISED CO_2 COOLING SYSTEM IN AIRCRAFT APPLICATIONS

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ABSTRACT

The existence of optimal pressures in transcritical refrigeration processes with varying ambient conditions can be used for process control to optimise the coefficient of performance (COP). Dominant factors are the gas cooler inlet temperature and the evaporator heat load. This is experimentally investigated on a direct evaporating CO_2 cooling cycle test rig, designed for the verification of a centralised aircraft on-board cooling system, where optimal pressures calculated from process states are passed to an integrated pressure control. In addition to positive effects regarding the process efficiency, the dynamics of the system concerning compensation of changing boundary conditions is improved. The process control is structured into two principal closed loops, meaning the pressure and the temperature control. This enables flexible system layouts with multiple parallel cooling consumers that can be operated and controlled separately within the integrated cooling system.

1. INTRODUCTION

Several cooling tasks are present inside the pressurised fuselage of modern passenger aircraft. The cabin air conditioning system is a mobile high-capacity cooling system with a nominal output of up to 50 kW. Solar radiation and the passengers are the two main heat sources for this system. Additional cooling positions are galleys, where food and beverages are cooled for health and comfort reasons, and electric consumers, such as avionic or in-flight entertainment computers. In this paper, these additional on-board cooling systems are discussed.
Developments towards weight reduction and efficiency optimisation have become more relevant for these systems throughout the last 10 years, because the complexity and performance requirements have increased. This is mainly the result of rising health, comfort, and reliability demands.

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

Motivated by the insufficiencies linked to using dry ice for the cooling of food and beverages in aircraft galleys, the need for active cooling devices emerged. In the first place, small units with a standard compression cycle using R22 and later R134a, as refrigerant, were used. These units consist of a scroll compressor, condenser, constant thermal expansion valve, evaporator, and two fans. They are referred to as "Air Chillers", which are connected to a ducting for cold air distribution and waste heat removal. The latter is discharged against exhaust air from the cabin, while cooled air exits the evaporator and is cycled through the galley compartments. This requires an Air Chiller location close to the cooling positions, in order to avoid extensive air ducting, which would lead to an increased fan power demand, installation space and insulation effort. Each galley to be cooled is connected to one independently operated Air Chiller, commonly installed in the lateral areas underneath the cabin floor.
The Air Chiller is still the most widespread technology for on-board cooling systems. A development towards integration is the "Remote Chiller System" (RCS), resulting from a limited installation space for certain aircraft. The refrigeration units are similar to the Air Chiller units, while a secondary cycle with a liquid coolant, instead of direct air cooling, distributes the cooling capacity to the consumers. It consists of the piping and an additional pump. The coolant is lead through 2 to 4 serial refrigeration units and supplies 1 to 4 galleys, each in parallel.

A non-toxic and non-flammable liquid Perfluoropolyether is used as coolant. Remote Chiller and pump unit are located underfloor, analogue to the Air Chillers, but in variable distance from the galleys. To transfer heat from the cooling positions to the coolant loop, an additional heat exchanger including coolant control valve and fan is installed on site. This system architecture is more flexible regarding installation location and temperature levels, since the Air Chiller System offers only one constant air outlet temperature.

A more integrated on-board cooling system was developed by expanding the coolant loop to supply the complete aircraft with two redundant distribution lines. It is referred to as "Supplemental Cooling System" (SCS). This system is based on a centralised R134a vapour cycle generating the cooling capacity for both coolant loops. All cold generation components are located in the centre section of the aircraft. Contrary to the previously described cooling systems, the heat sink for the condenser of the vapour cycle is ambient air from outside the pressurised fuselage. The cooling position heat exchange is realised identical to the Remote Chiller System, since the same coolant and supply temperature is used.

For this type of cooling system, the consumers are represented by galleys as well as equipment located in the Electric/Electronic Compartment, which require active cooling under on-ground conditions in warmer regions.

The centralised CO_2 cooling system follows this development towards integration. It is defined as a single loop direct evaporating system with CO_2 as refrigerant/coolant. The heat transfer at the cooling positions is realised similar to the Remote Chiller and Supplemental Cooling System, but without secondary coolant cycle. For heat discharge at the gas cooler, ambient air is used. Currently, the emphasis lies on investigations considering transcritical processes, meaning moderate or warm ambient temperature levels at the gas cooler on ground.

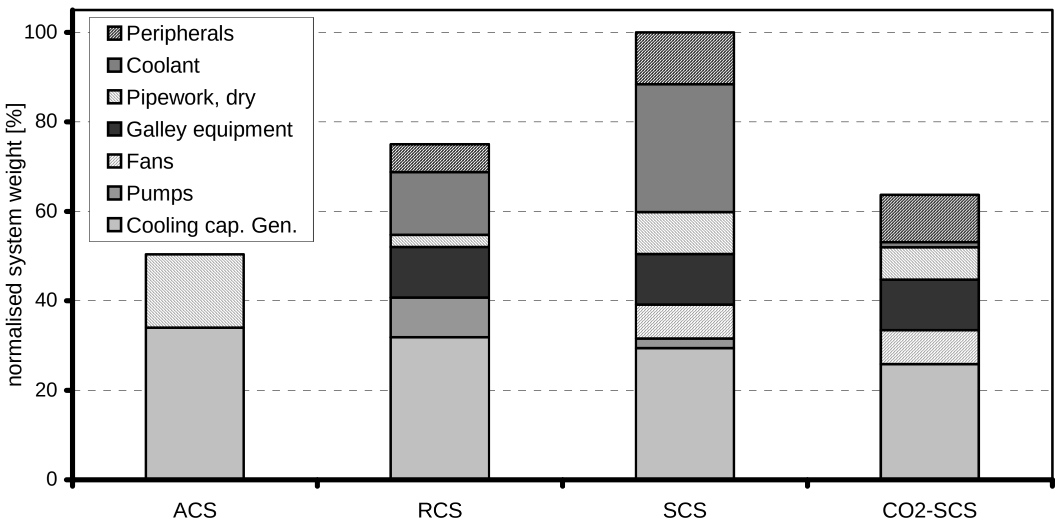


Figure 1: Weight estimation for on-board cooling systems

1.2 Advantages of CO_2

Potential advantages concerning the use of CO_2 as refrigerant for aircraft on-board cooling systems lie in the high specific cooling capacity. Compared to currently used refrigerants (R134a) and coolants (Perfluoropolyether), lower fluid charges are required, which results in a significant coolant weight reduction. Additionally, the required mass flows are lower due to the heat transfer coefficient for two phase CO_2 compared to liquid synthetics. This leads to smaller pipe diameters which again reduce the system weight and installation space. This effect is partially levelled off by the wall thickness of the CO_2 piping due to high operating pressures.

Comparing the system weight, the diagram in Figure 1 shows the normalised system mass for all introduced technologies dimensioned to a long range reference aircraft. The CO_2 cooling system is second in weight behind the Air Chiller System (ACS). At the same time, the CO_2 system structur-

ally offers more functionality, efficiency and dynamic capabilities. Its mass is based on several assumptions and has an uncertainty bandwidth of approximately ±10 %.
High pressure pneumatic systems in aircraft, as which the centralised CO_2 cooling system is considered, underlie stringent safety requirements. Due to the fact, that distribution piping with high pressures up to 130 bar is installed close to 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. Investigations in this article focus on functionality and dynamic behaviour of the refrigeration cycle.

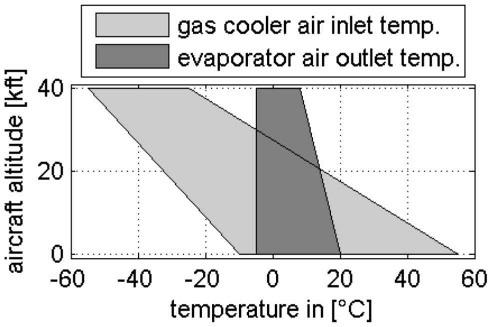


Figure 2: Simplified operation envelope for aircraft cooling systems wiht outboard air supply

Passenger aircraft are exposed to a wide range of ambient conditions. Figure 2 illustrates this fact. It can be obtained, that the gas cooler inlet temperature T_{amb} can drop significantly below the demanded evaporator air outlet temperature T_{dem}, especially at cruise altitude. Extreme supercooling can be avoided by lowering the gas cooler air mass flow, which positively affects the aircraft drag, caused by a ram air inlet. Solely ground conditions of T_{amb} >28 °C with a transcritical cooling cycle are covered in this paper, since they represent the design case with maximum heat loads for the cooling system.

2. PROCESS INVESTIGATION ON CO_2 CYCLE

Experimental and theoretical analysis of a direct evaporating CO_2 vapour cycle system was performed. In this section, a summary of the experimental facilities and applied methods is given.

2.1 Experimental System

For experimental investigations of a centralised CO_2 cooling system for aircraft applications, a full scale test rig, adapted to specific aircraft dimensions, has been set up. A scheme of this system is displayed in Figure 3. The two main subsystems are the centralised cold generation unit and the parallel consumer stations (2 illustrated in Figure 3). Compressor, gas cooler, internal heat exchanger, and low pressure reservoir are integrated in the centralised system part. The consumer stations consist of expansion valve, evaporator, closed air cycle, and an electric heater to simulate variable heat loads. Another heater is installed at the air inlet channel of the gas cooler to realise higher temperature levels for waste heat removal. The distribution piping connecting the two subsystems has a total length of approximately 110 m. A subset of system characteristics is listed in Table 1.

Table 1: Characteristics of CO_2 Test Rig

transferable heat/heat load		Unit	Compressor data		Unit
Gas Cooler	12	[kW]	max. speed	4000 (1900)	[1/min]
Consumer 1	0.6..2	[kW]	mass flow	40..300	[kg/s]
Consumer 2	1..4	[kW]	max. pressure	130	[bar]
Consumer 3	1..8	[kW]	max. temperature	140	[°C]

A swash plate variable displacement piston compressor as used in mobile applications is used for pressure generation. Its displacement is set to full stroke, while the operating pressure p_{HP} is controlled by the compressor speed n_C. Thermocouples, pressure sensors, mass flow and speed sensors are used to measure or determine refrigerant states at essential system components for analysis and process control, respectively.

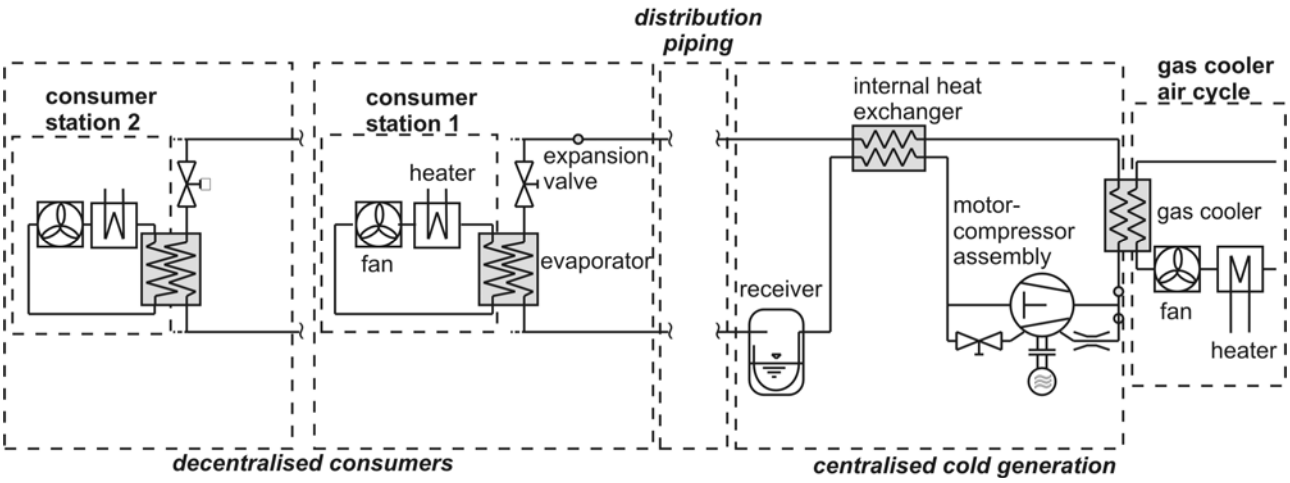


Figure 3: Scheme of CO_2 Test Rig

2.2 System Modelling and Simulation

The simulations are based on a non-linear model of the CO_2 cycle. It was implemented in MODELICA with use of the modelling environment DYMOLA. The library for this object-oriented model is the *ACLib* (Pfafferot and Schmitz 2004) which is based upon the THERMOFLUID library, and was validated by measurements on the described experimental system. All refrigerant states can be accessed from the simulation results, supporting experimental analysis.

From the complex, non-linear model in DYMOLA/MODELICA, a linear state space model was derived. The focus for this model is to reflect the dynamic behaviour of the real system and allow analytical controller design. The first step was a separate modelling and examination of elementary system parts, which can then be connected to achieve a state space model for the complete system. All parameters and boundary conditions for the process under consideration and operating points are derived from simulation data from the DYMOLA/MODELICA model, providing a basis for controller design.

2.3 Controller Concept

Two main controller tasks have to be accomplished for a safe and efficient process, according to Figure 4. The primary control variable is the air outlet temperature of the evaporator $T_{E,air}$. It is held at a constant demanded temperature T_{dem} using the adjustment travel of the expansion valve s_{EV} as actuating signal. The parameter T_{dem} is the output of a prefilter with the manually adjusted target temperature as input signal. At two phase refrigerant inlet and outlet flow at the evaporator, the expansion valve controls the refrigerant charge of the heat exchanger. A direct control of the evaporation pressure is not applied.

To adjust defined high pressure levels, the secondary controller is linked to the constant displacement compressor. It controls the demanded pressure p_{HP} by varying the motor/compressor speed n_{C} and thus the refrigerant mass flow $\dot{m}_{CO2}$.

These two controller types are designed regarding the according plants as SISO systems. Hence, single controllers for the compressor and the three expansion valves are implemented. A MIMO controller could optimise the system behaviour towards changing boundary conditions, taking the thermodynamic couplings of each plant into account. The dominant coupling results from the equal high and low pressure levels for all consumers.

A SISO solution was chosen to obtain a more flexible system with a controller concept working with each configuration of one, two, and three parallel consumers. The applicability of such a controller concept was previously validated by Schade and Carl, 2002, 2003.

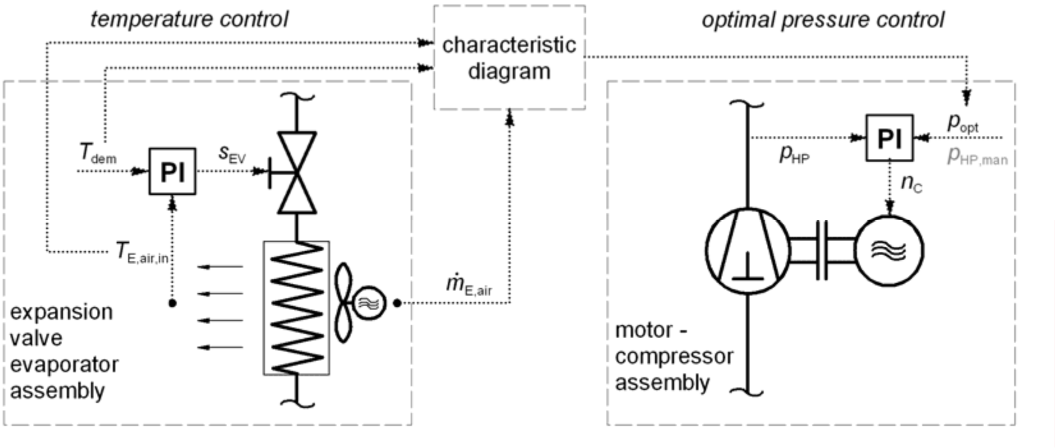


Figure 4: Scheme of temperature and optimal pressure controller setup

3. OPTIMAL PRESSURES FOR CO_2 REFRIGERATION CYCLES

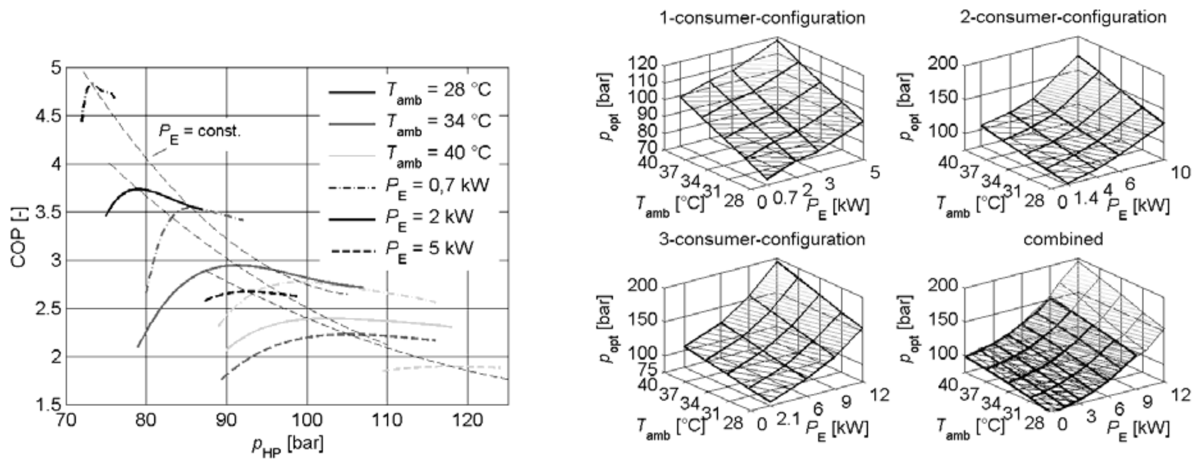
Regarding ambient temperatures of T_{amb} > 28 °C, the refrigeration process at the experimental system is transcritical. Contrary to subcritical processes, this enables a flexible adjustment of the high pressure p_{HP}, leading to an optimisation problem. Maxima of the coefficient of performance (COP) exist for each set of varying ambient conditions (Kauf, 1999).
Using the non-linear model in DYMOLA/MODELICA, a characteristic diagram for the optimal pressure with maximum COP was generated. This was performed by simulating 3 consumer configurations with the particular ambient conditions heat load (P_E = 0.7/2/3/5 kW per consumer) and ambient temperature (T_{amb} = 28/31/34/37/40 °C). Set values for the evaporator outlet temperature T_{dem} were varied for a 1 consumer configuration (T_{dem} = 3/5/8/15 °C) only, because the effect of the temperature level on optimal pressures is negligible. Figure 5a shows a summary of the achieved results. The COP is plotted over p_{HP} for certain subsets of boundary conditions showing lower COP at higher optimal pressures p_{opt}. In Figure 5b, all simulation data is used to calculate a 3 dimensional characteristic diagram for p_{opt} dependent on T_{amb} and P_E. Calculation of the COP is done using the evaporator heat load P_E in:

$$\mathrm{COP} = \frac{P_E}{\dot{m}_{CO2} \cdot (h_{C,o} - h_{C,i})} \quad , \tag{1}$$

where the compressor inlet and outlet enthalpies $h_{C,i}$ and $h_{C,o}$, respectively, are calculated from measured pressures and temperatures. To integrate the target values for p_{HP} from the characteristic diagram into the controller, T_{amb} and $P_{E,1\text{-}3}$ must be assigned. Air temperature $T_{E,air,in}$ and air mass flows $\dot{m}_{E,air}$ allow calculation of the actual evaporator heat load:

$$P_E = \dot{m}_{E,air} c_{p,air} \cdot (T_{E,air,in} - T_{dem}) \tag{2}$$

Using the specific heat capacity $c_{p,air}$, this equation yields the effective evaporator heat load P_E including heat dissipation from the fans. The sum of all heat loads $P_{E,1\text{-}3}$ and T_{amb} are used in two third order polynomials to calculate the optimal pressure, which then can be directly passed to the pressure control.



a) b)

Figure 5: COP for varying ambient temperature and heat load a), optimal pressures for different system configurations b)

4. EXPERIMENTAL INVESTIGATIONS

Three cases are considered for investigation of the system behaviour with and without optimal pressure control. A target temperature step from T_{dem} = 5→3→5 °C was performed. A rising p_{HP} occurs in this case, due to the temporarily increased P_E (compare Equation 2). In the second step, the ambient temperature was increased and reset to T_{amb} = 34→36→34 °C, and least the net heat load from the electric heaters was set to $P_{H,tot}$= 2.1→3.3→2.1 kW. Figure 6 illustrates the measurement results from these steps. The relevant time periods are highlighted. The changing pressures p_{opt} and p_{HP} indicate the effect of the optimal pressure control. A maximum overshoot at the $\pm\Delta T_{dem}$ step can be seen. This points out the need for the applied prefilter to avoid excessive pressure oscillation.

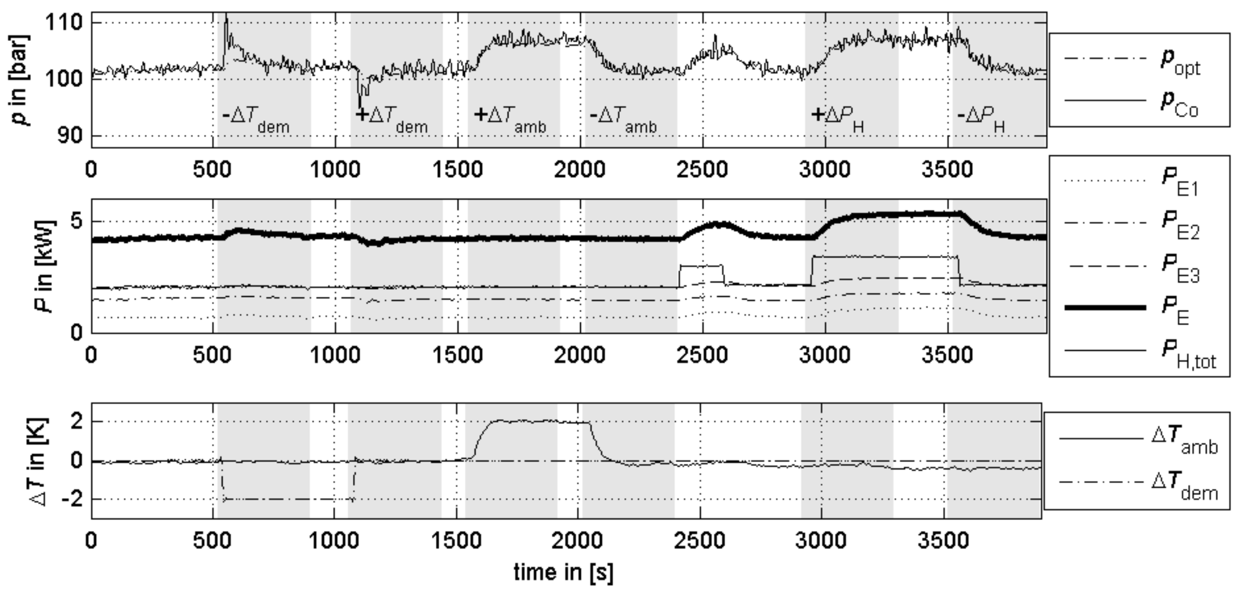
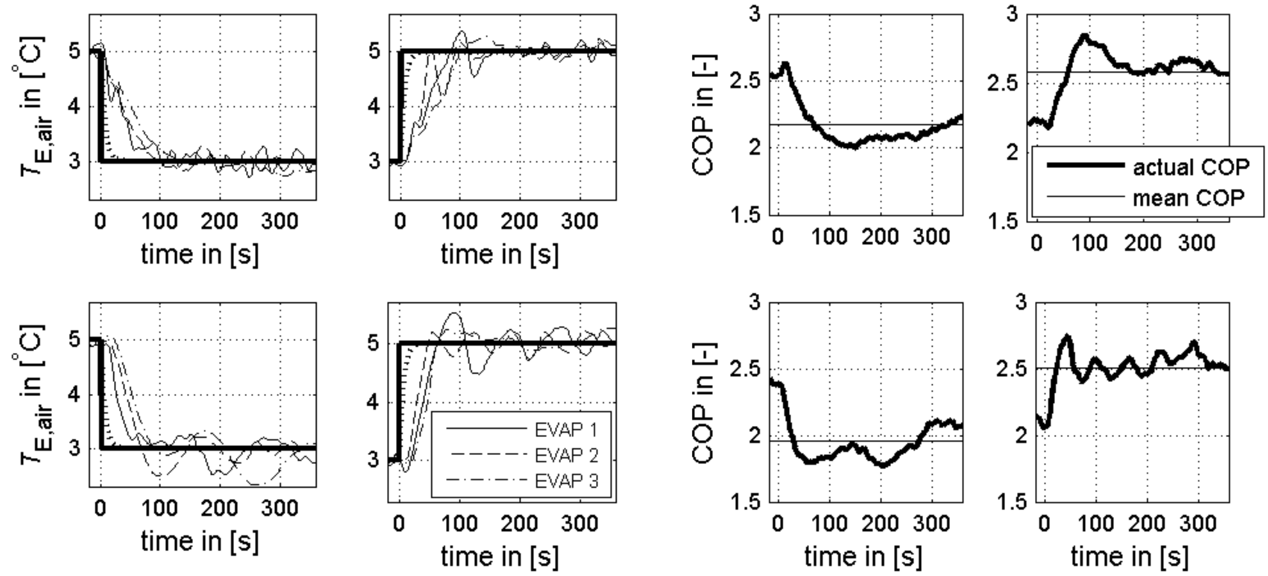


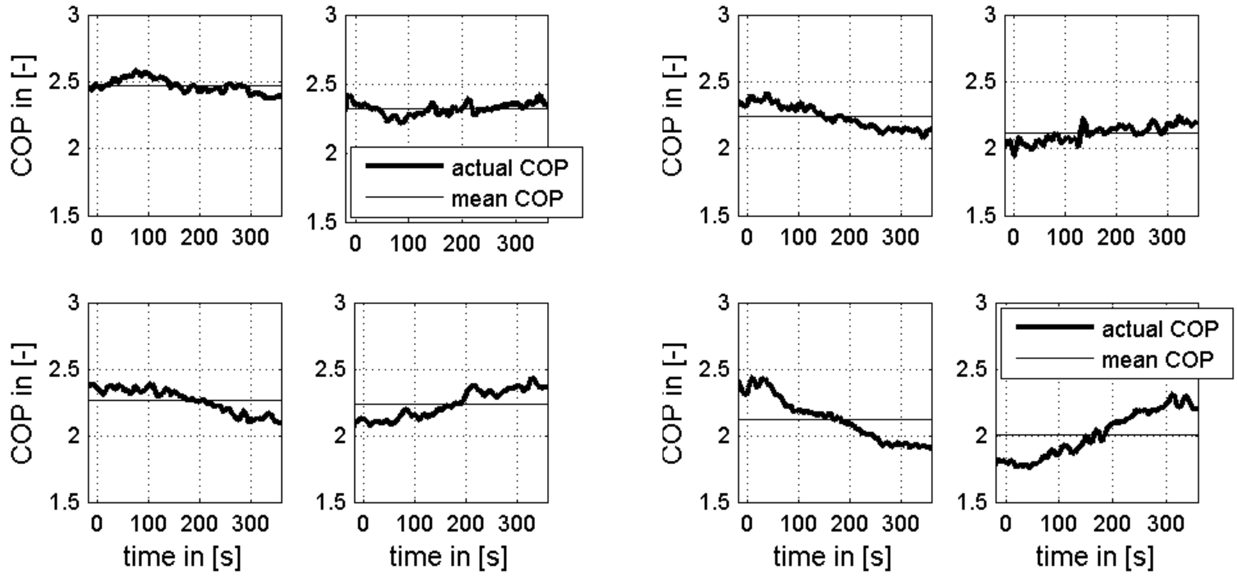
Figure 6: Pressures, heat loads and temperatures with optimal pressure control

In Figure 7a, the step response of the evaporator temperature $T_{E,air}$ for T_{dem} = 5→3→5 °C is displayed. The prefilter output signal is represented by the dotted lines. All upper diagrams in Figure 7



a) Evaporator temperatures b) COP

Figure 7: Step response for T_{dem} step; upper diagrams: at p_{opt}; lower diagrams: at constant p_{HP}



a) ambient termperature T_{amb} step b) heat load P_H step

Figure 8: Coefficient of performance; upper diagrams: at p_{opt}; lower diagrams: at constant p_{HP}

and 8 refer to the optimal pressure control, while the lower diagrams show results at a constant pressure p_{HP} = 101 bar, being the optimal pressure for the starting conditions, for the same steps.
Figure 7a illustrates, that the overshoot and settling time of $T_{E,air}$ are both higher at constant high pressure. As expected in this case, the mean COP displayed in Figure 7b is lower over the regarded time period. Start and final COP values regarding each time period for both controller operations are approximately equal, verifying the negligible effect of the evaporator temperature level on the COP and p_{opt}. As it can be seen in Figure 6, the pressure level with optimal pressure control returns to the initial value. Adapting p_{HP} to the optimum p_{opt} allows faster compensation of the temporarily rising heat loads and offers, as can be seen from these results, a better COP, also for this transient operation.

Regarding Figure 8, the influence on the COP is more obvious regarding ambient temperature T_{amb} and heat load P_E steps. Here, the COP is illustrated for the transition time periods. With optimal pressure control, the COP is noticeably higher (≈ 0.2-0.3) after each first step. For moderately changing heat loads, it can almost be held constant, while the resulting pressure changes for p_{opt}, displayed in Figure 6 are equal. Since the average COP is higher with activated optimal pressure control, the benefit is already noticeable for small time periods.

5. CONCLUSION

Process control of a centralised, direct evaporating CO_2 cooling system for aircraft on-board cooling can be optimised concerning both dynamic behaviour towards disturbances, and efficiency, by implementing an optimal pressure control. Thus a characteristic diagram is required, which was generated based on simulation results from a non-linear model in DYMOLA/MODELICA, validated by test results. Experimental investigations show an optimised COP during transient operation, in combination with smaller settling times for the primary control signal $T_{E,air}$. For steady state operation, the system can operate at the maximum coefficient of performance with a limited set of measurement signals. With this controller concept, some of the advantages of a real MIMO controller can be utilised, while being flexible regarding varying consumer configurations with multiple SISO controllers.

5. ACKNOWLEDGEMENTS

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NOMENCLATURE

COP	coefficient of performance	(-)	Subscripts / Abbreviations	
c_p	specific heat	(kJ/kg/K)	amb	ambient
h	specific enthalpy	(kJ/kg)	C	compressor
$\dot{m}$	mass flow	(kg/s)	dem	demanded/target
n	compressor speed	(1/s)	E	evaporator
p	pressure	(bar)	H	heater
P	heat load	(kW)	ACS	Air Chiller System
s_{EV}	expansion valve position	(m)	RCS	Remote Chiller System
T	temperature	(bar)	SCS	Supplemental Cooling System

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