Numerical Simulation of a New Cooling System for Commercial Aircrafts

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ABSTRACT

This paper presents the results of the modeling of a complex cooling system of a future commercial aircraft. The results from transient calculations will be presented, for instance the system behavior during the climb phase of the aircraft. Hereby the feasibility of a first control strategy will be investigated. Moreover the experience solving this modeling task coupling two different simulation environments will be discussed.

1. INTRODUCTION

Recent considerations when developing aircrafts – like for example the omission of engine bleed – result in additional cooling tasks which require an elaborated cooling architecture. The system under consideration consists of two R134a - vapour cycles which are linked to the consumers by secondary cooling loops. Consumers which can run at higher temperatures, like power electronics, are cooled directly by cooling loops not connected to any vapour cycle.

The challenge of such a system lies in the fact that the system has to work very reliable under a wide range of ambient conditions. Not only do the ambient conditions change over one flight phase but also the heat loads vary significantly during flight. In this context, system control is an important issue, also because different minimum and maximum temperatures have to be assured for different consumers.

For pre-design and feasibility studies as well as for the set up and testing of different control strategies and monitoring concepts it is important to have a powerful model of the cooling architecture. In a previous study the system was sized using a steady-state model to meet the cooling requirements. In the second step presented here a one dimensional transient simulation model of the entire system is set up.

In order to obtain a numerical efficient model, two different simulation tools are used instead of one. Therefore the model is divided into two parts: the cold distribution via a one-phase coolant is modeled using a commercial pipe network flow program. The vapour cycle as well as the cold plates are modeled using the Modelica language. The co-simulation is then performed by coupling the two programmes using a coupling software. The devices to be cooled (like power electronics) itself are not modelled in detail, but their time-dependent heat losses are input values to the simulation model.

2. SYSTEM DESCRIPTION

2.1 General

The investigated cooling system (figure 1) shall remove heat from several cold consumers of an aircraft. These consumers can be divided into mainly two groups: consumers which require a cooling temperature below normal fuselage temperature (like for instance galleys which require a temperature of 4°C) and consumers whose temperatures can be above normal fuselage temperature, mostly power electronics whose case temperatures can be around 80°C.

The first category requires vapour cycle cooling, whereas for the second category a simple cooling loop is sufficient. Therefore the system is divided into two main parts: one part containing two R134a vapour cycles which are linked to the consumers by secondary water-glycole cooling loops and a second part containing two water-glycole cooling loops. The heat is rejected to ambience in ram air channels where the necessary air flow on ground is ensured by ram air fans. In this example architecture the temperature level of the power electronics (H1 – H6) as well as of some of the other cold consumers (C1 – C5) must not fall below a certain temperature. This minimum temperature is assured by mixing the coolant from the supply line with coolant from the return line.

Depending on the required reliability of the system, different cross couplings of the cooling loops could be realized. In this example both hot loops pass a ram air heat exchanger in each ram air channel, so all heat from the power electronics could be removed even if one ram air flow is lost. The two cold loops are not linked, even though each vapour cycle already has two evaporators (daisy-chained with respect to the refrigerant flow), so a linking could be easily implemented.

2.2 Control Strategy of the System

An exemplary control strategy is presented here which will be tested on the the developed system model (figure 2). As usual the superheating of the refrigerant is controlled by a thermostatic expansion valve and the coolant temperature at the evaporator outlet by the compressor speed. Since the outside air temperature varies significantly during a flight period a head pressure control has to be implemented in the vapour cycle. This can be done for instance with refrigerant partly bypassing the condenser or by reducing the air mass flow rate. In this example the minimum head pressure is ensured by adjusting the air mass flow rate, which is done on ground by controlling the fan speed and during flight by an actuator at the ram air channel inlet. On the side of the fluid network the only devices to be controlled are the valves and the pumps. The valves directly upstream the cold consumers ensure a maximum outlet temperature. The pump speeds are controlled by the highest opening ratio of those valves, so the pump speed decreases if all opening ratios are below a certain limit. In the hot loops a bypass ensures a minimum supply temperature for all power electronic devices. In the cold loops each consumer requests a different minimum temperature so there is a control valve for each branch of the return line which adjusts the flow which is injected from the return line into the supply line of each cold consumer.

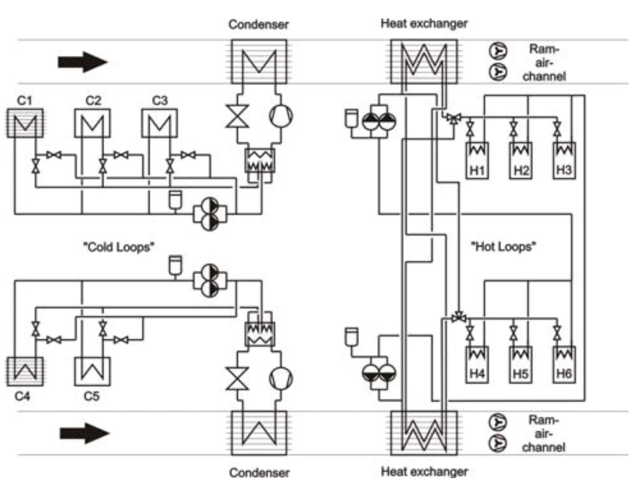

Figure 1: Schematic description of the investigated cooling system

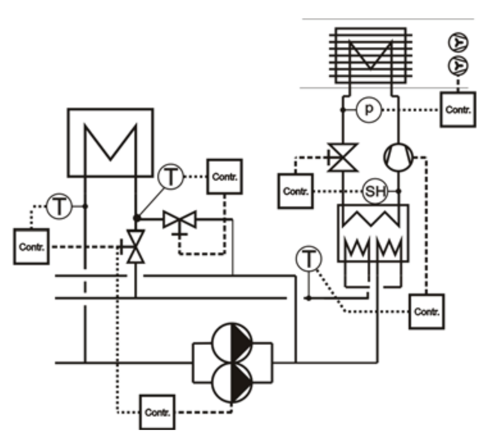

Figure 2: Implemented system control

3. MODELLING OF THE SYSTEM

From the modelling point of view, the system is divided into four fluid networks including pipes, valves and pumps on the one hand and into the vapour cycles and the heat exchangers of the cold consumers on the other hand. The cold distribution networks can be conveniently modelled using a commercial pipe network flow program where the solver is optimized for such a system. However such a program does not allow for detailed modelling of the heat exchangers and for the modelling of the vapour cycle. The heat exchangers and the vapour cycles are therefore modelled seperately. The different models are then coupled using a coupling tool (Kossel *et al.*, 2006), which exchanges the data using the tcp/ip protocol, so subsystems can be allocated on different machines. The advantages of object oriented distributed modelling are for instance described by Krus (2001). So is for example the vapour cycle of much faster dynamics than the large fluid network, and the use of distributed solvers allows for different time steps for the different systems.

3.1 Modelling of the Vapour Cycle

The vapour cycle and the cold consumers were modelled using the object-oriented modeling language Modelica. Modelica is a tool-independent modelling language which was developed in order to describe hybrid algebraic differential equations. In order to maintain the physical structure of the mapped system, the connector definition was introduced in Modelica. Connectors define the external interfaces of components and allow for mass and energy transfer across the system boundaries of the components. When connections are established using special connect-equations, all flow variables (such as mass flow rate) sum up to zero and all potential variables (such as pressure) are set equal. In fluid systems two different connectors are of interest: thermal connectors, with temperature and heat flow rate as quantities in the connector and flow connectors containing basically specific enthalpy, pressure and mass flow rate.

There exist already several libraries written in Modelica, for example the free Modelica_Fluid library (Casella *et al.*, 2006), which provides zero- and one-dimensional components for the modeling of thermo-fluid systems. Amongst others the Modelica_Fluid contains a base class providing dynamic mass, energy and momentum balances for pipe flows with one-dimensional spatial discretization. This base class is used as a starting point for the modelling of the heat exchangers. A similar approach was used successfully for CO_2 vapour cycles by Pfafferott (2004).

The compressor on the other hand is described by static mass and energy balances and a variable isentropic efficiency. However when setting up the whole vapour cycle the resulting algebraic coupling between the compressor inlet and condenser inlet results in an additional nonlinear system of equations. Even though this system of equation is rather small (only three iteration variables: $p_{Compr,\,in}$, $p_{Compr,\,out}$, $h_{Compr,\,out}$) it requires a large amount of computation time. In order to break this nonlinear system a small volume was introduced downstream the compressor. This measure decreases for this specific set up of the vapour cycle the computation time by about the factor 1.4.

The one-phase medium models used (air and incompressible liquids with table based properties) are already provided by the Modelica Standard Library whereas the refrigerant medium models were implemented using the ExternalMedia. The ExternalMedia is an interface library developed by Casella and Richter (2008) which can be used to include external fluid property code in Modelica. So even if the computation results presented here were obtained using R134a as refrigerant, the vapour cycle can be easily re-used with other refrigerants.

3.2 Modelling of the Condenser

Exemplary for the modelling of a heat exchanger the way the condenser was modelled shall be presented here. Since the Modelica models are used to perform system simulation, an equilibrium has to be found between detailed modeling of the components and fast simulation time. The condenser is supposed to be a cross-flow heat exchanger which is divided into several passes with different number of small tubes (see figure 3a). It is modelled using a single pipe for the refrigerant flow and several parallel pipes for the air flow; where all pipes are discretized in flow direction. Hereby the number of the parallel air pipes equals the number of the discrete elements of the refrigerant pipe. The refrigerant side of the heat exchanger could be modelled by connecting several pipes with different geometries in flow direction. However, since the simulation tool can solve the system of equations faster if the equations for mass and energy transport are implemented directly and not via the connect-equations, a single pipe is used, where the geometry differs along the pipe length.

Each pipe contains *n* heat transfer models, determining the heat transfer coefficient *k* and one pressure drop model.

The thermal coupling between the pipes is performed in a way, that all thermal ports of one air pipe are connected to the thermal port of the corresponding refrigerant discrete element (see Figure 3b).

Since the heat transfer coefficient on the refrigerant side depends significantly on the phase of the refrigerant, the refrigerant pipe often requires a high number of discrete elements. The resulting large number of parallel air pipes increases the nonlinear system of equations and slows down the simulation. However it can be seen that even though the air temperature may vary a lot along its flowpath it does not vary much over the area, certainly not if the subcooling is reasonable. Therefore the condenser can also be replaced by a condenser having only as many parallel air pipes as there are refrigerant passes, which allows for different air temperatures in the case air comes in contact with strongly subcooled refrigerant. The thermal ports of one air pipes are connected to all thermal ports of the discrete elements belonging to the same pass of the refrigerant pipe (see Figure 3c). For a vapour cycle with a condenser with $n_{Ref} = 9$, $n_{pass} = 3$ the simulation time reduces by about the factor 1.6. An even larger simplification can be achieved if only one air pipe is used. The condenser, as all other components, were checked for plausibility, but not compared to measurement data.

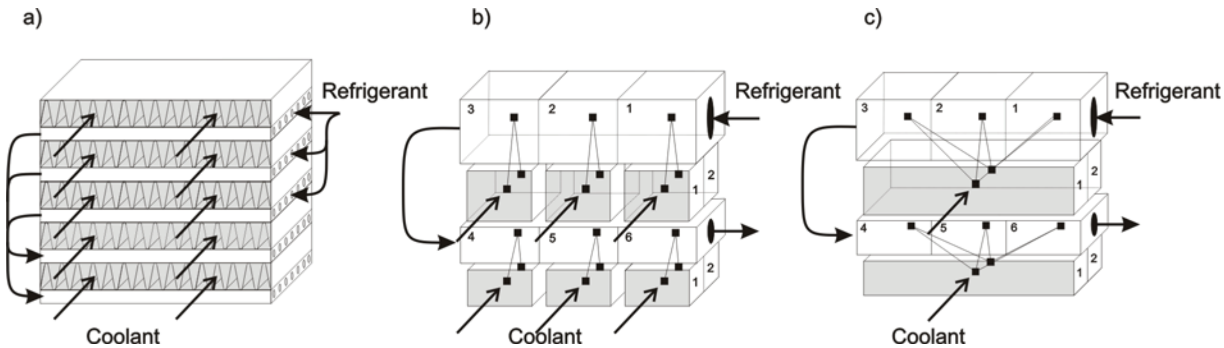

Figure 3: Modelling approach of the condenser: Schematic of the condenser (a), condenser with high number of air pipes (b) and simplified approach (c). The small black squares symbolize the heat ports.

3.3 Co-simulation

In order to perform the co-simulation, first the interfaces between the different subsystems have to be defined. Since the fluid network is modelled as a closed loop the coolant mass flow rate is calculated there, depending on the pressure the pumps generate. The pressure loss in the Modelica heat exchangers has to be submitted to the network models, where they are implemented using a valve model. On the other hand, the calculated mass flow rate is submitted to the different Modelica models. Also the back pressure of the coolant in the Modelica models could be supplied by the network models. However, since the coolants are modelled as incompressible liquids whose fluids properties do not depend on the pressure, a (discrete) change in the back pressure may only slow down the simulations, without affecting the results, so any constant back pressure in the Modelica models can be used. The thermal coupling between the models of the subsystems could be realized by either handing over the heat load or the outlet temperature of the heat exchanger from the Modelica models to the heat exchangers in the network models. However when handing over the heat load to the network models the heat exchanger outlet temperature is there calculated using

$$T_{out} = \frac{\dot{Q}}{\dot{m} \cdot c_p} + T_{in} \tag{1}$$

The mass flow rate used here would be the actual (instantaneous) mass flow rate in the network system which is not equal to the mass flow rate the heat flow rate was calculated with in the Modelica model, simply because the mass flow rate in the network changed since the last time the data was synchronized. If the gradient of the mass flow rate is large, an extremely unrealistic temperature could be the result. Therefore the outlet temperature is handed over by the Modelica model rather than the heat flow rate.

Also the time interval at which the data shall be synchronized has to be fixed. In general simulation time decreases if the time interval increase, since each time data is synchronized all simulation tools are stopped and have to be restarted. Also, since the Modelica model runs with variable step size, the maximum step size is constrained by the time interval of the synchronization. On the other hand too large intervals may result in large discrete changes of the exchanged data, which can cause trouble for instance when re-initializing the Modelica model. This is discussed in the following result section.

4. RESULTS

4.1 Influence of the Time Interval of the Synchronization

As mentioned above the time interval at which data is exchanged plays an important role when co-simulating. To study the influence of the synchronization interval Δt_{sync}, the start-up procedure is studied here: the coolant is initialized with 20°C and then cooled down by the vapour cycle, variing the value for Δt_{sync}. As can be seen in figure 4 and figure 5, the synchronization interval has a significant effect on the result: if the synchronization interval is too large, the coolant temperature at the evaporator inlet is overestimated and so is the coolant temperature at the evaporator outlet. Therefore the compressor controller increases the compressor speed, resulting in a high condenser inlet pressure which results in the end in an overestimation of the electrical power consumption of the compressor. A variation of Δt_{sync} showed that the synchronization interval should be around 2 s in order to give reliable results.

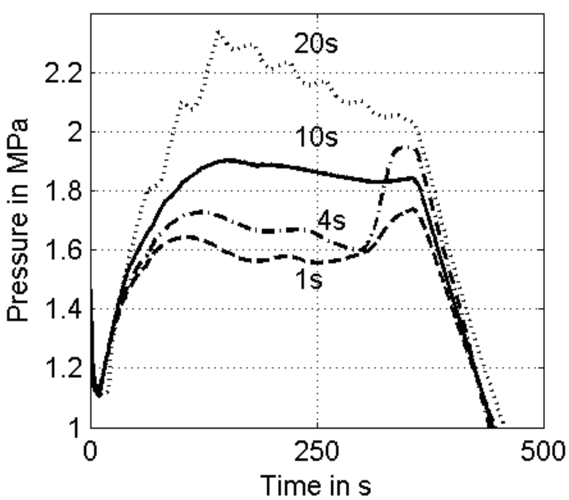

Figure 4: Refrigerant pressure at condenser inlet for different values of $\mathbf{\Delta t_{sync}}$

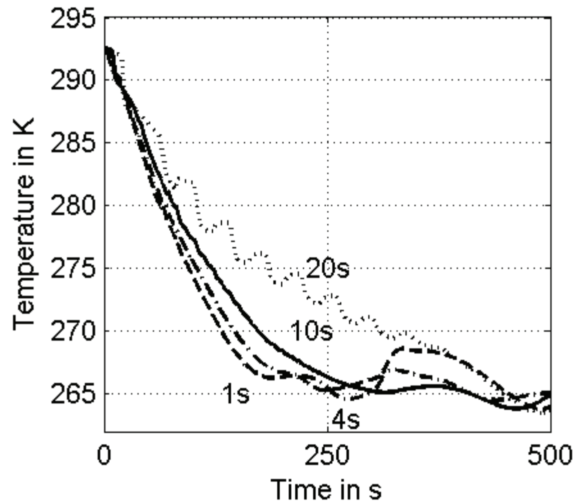

Figure 5: Coolant temperature at evaporator outlet for different values of $\mathbf{\Delta t_{sync}}$

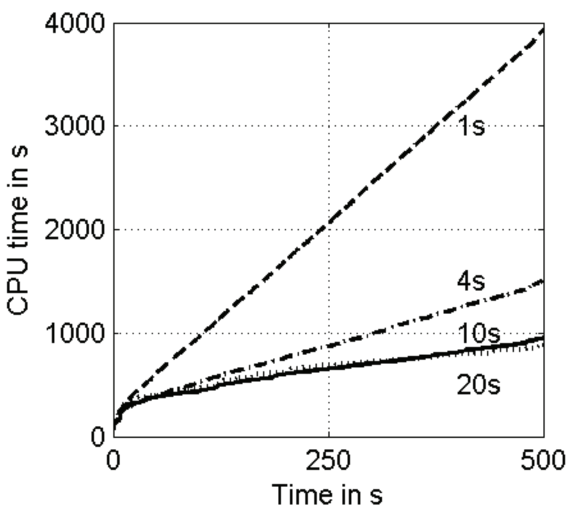

Figure 6: CPU time for different values of $\mathbf{\Delta t_{sync}}$

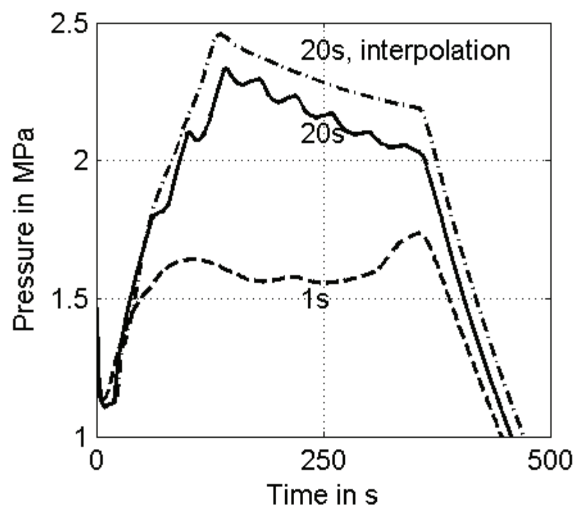

Figure 7: Refrigerant pressure at condenser inlet with and without cubic spline interpolation

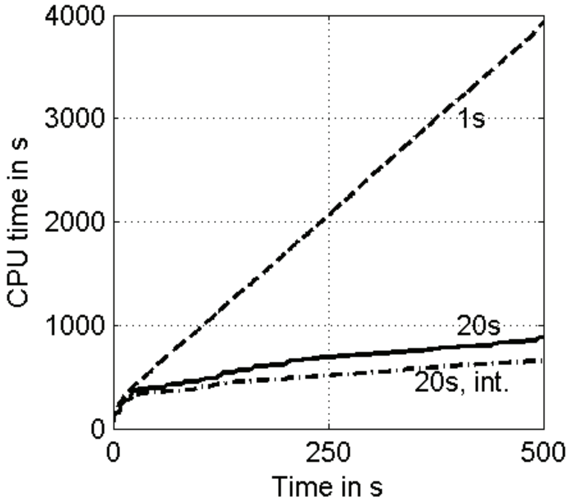

Figure 8: CPU time with and without cubic spline interpolation (int.) and for different values of Δt_{sync}

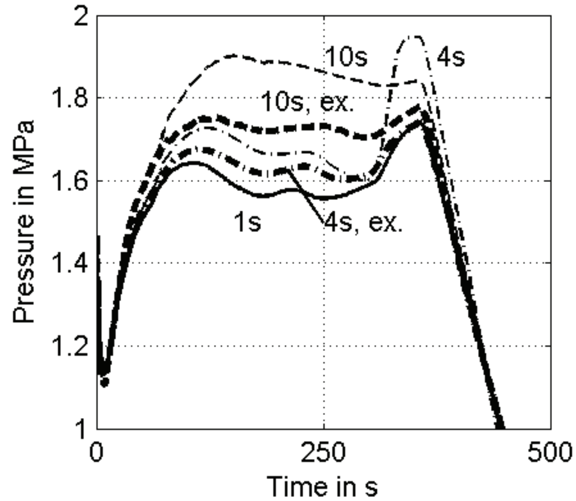

Figure 9: Refrigerant pressure at condenser inlet with and without linear extrapolation (ex.) and for different values of Δt_{sync}

In general the CPU time (figure 6) decreases with increasing synchronization interval, however, there is no further decrease between Δt_{sync} = 10 s and Δt_{sync} = 20 s. This may be due to the fact that for the latter the discrete changes in the variable values are already quite large, such slowing down the re-initialization process of the Modelica model. The problem of too high discrete changes can be avoided when interpolating the exchanged data. Therefore the data was interpolated using a natural cubic spline with three knots. The comparison between spline interpolation and normal synchronization for Δt_{sync} = 20 s can be seen in figure 7 and figure 8. As expected the CPU time decreases due to the now smoother behaviour of the variables. However, the difference to the solution with Δt_{sync} = 1 s is even higher, since the coolant temperature decreases even slower than without interpolation. This observation also holds for the other values of Δt_{sync}.

So even though (spline) interpolation is a very good measure in order to increase computation speed and to avoid numerical problems, it is not applicable in order to get better (i.e. more exact) results with a higher synchronization interval. To increase the accuracy at higher values for Δt_{sync}, the data has to be extrapolated. In general extrapolating data is quite dangerous, since it can lead to unphysical values. However since in any case the synchronization interval has to be chosen so that the discrete changes are not too high, extrapolation can be used here. One could extrapolate the cubic spline used for interpolation, however since for the natural cubic spline the second derivative of the spline polynomials are set equal to zero at the ends of the interpolation interval, the extrapolated value after another $\Delta t = \Delta t_{sync}$ equals exactly the value of a linear extrapolation. Therefore a simple linear extrapolation is used here. The result is significantly improved compared to normal synchronization (figure 9). So a higher synchronization interval (4 s instead of 2 s) can be used, achieving good results with lower computation times.

Nevertheless one has to bear in mind that extrapolation can lead to problems and can only be used if the changes in the exchanged data are not too high.

4.2 Simulation Results for Implemented System Control

The results of the simulation of the investigated system with the corresponding control strategy are shown in this section. In this example calculation the boundary conditions (flight level and heat loads of the cold consumers) are varied over the simulation time as shown in figure 10.

In figure 12 it can be seen that at t = 1500 s the air mass flow rate is decreased rapidly, in order to maintain the desired head pressure (in this case $p_{cond,\ out,\ min}$ = 0.7 MPa, see figure 11). Since the air passes the ram air heat exchanger after it passed the condenser it is already strongly heated up (due to the small air mass flow rate). Therefore the coolant temperature in the hot loops increases significantly (see figure 13) even though the hot loop pumps run at full speed and the control upstream the cold consumers are fully open (see figure 14). In order to maintain a lower coolant temperature the control strategy has therefore to be extended – for example a combination of controlling the air mass flow rate as well as a bypass valve will be investigated.

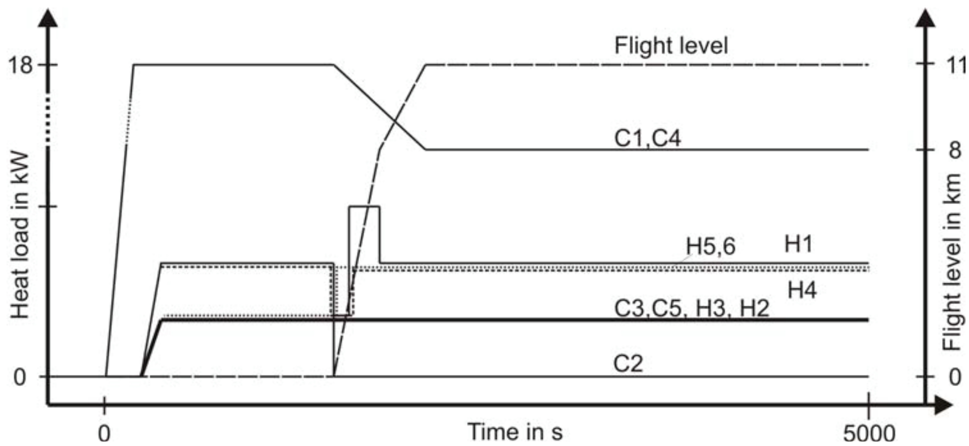

Figure 10: Boundary conditions of simulation (flight level and heat loads)

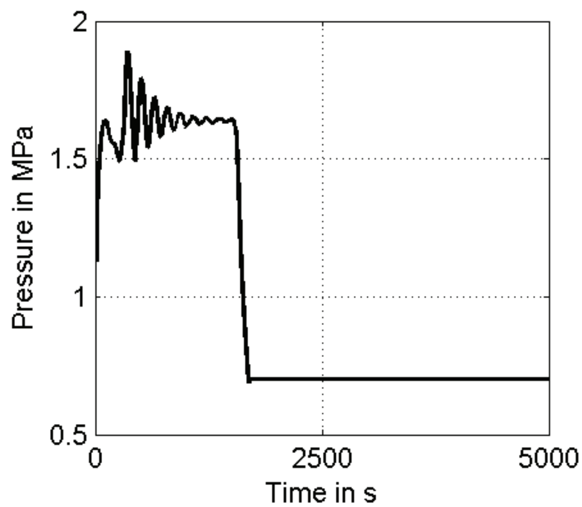

Figure 11: Pressure of refrigerant at condenser inlet

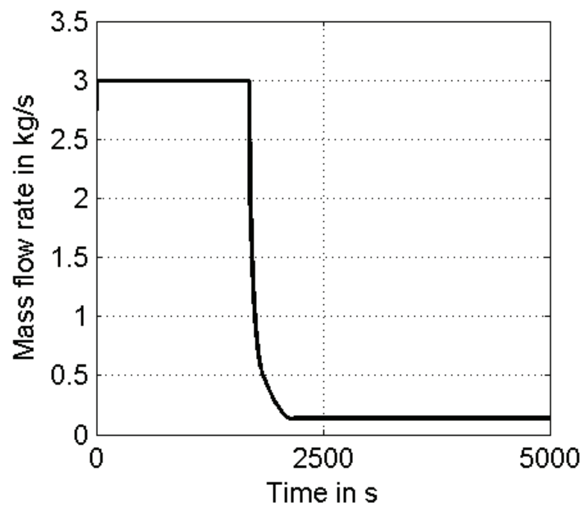

Figure 12: Air mass flow rate in one ram air channels

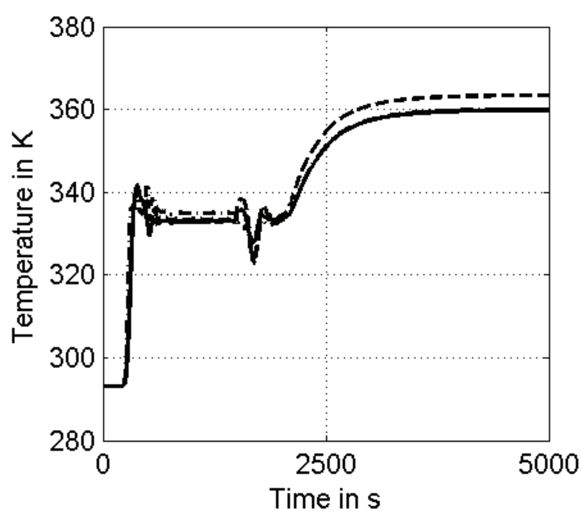

Figure 13: Coolant temperature at outlet of consumers in hot loop 1

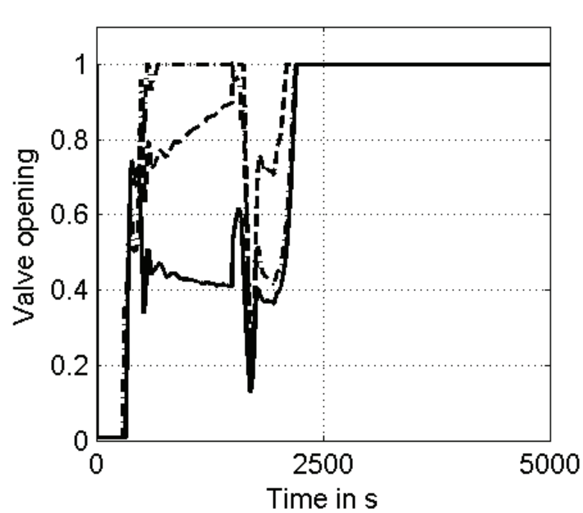

Figure 14: Valve positions of the control valves of each consumer in hot loop 1

6. CONCLUSIONS

The simulation of the investigated cooling system shows:

- The time interval at which data between coupled tools is exchanged has a huge effect on the result.
- Extrapolation of the exchanged data can be performed in order to decrease the computation time without affecting the result.
- Interpolation of the exchanged data improves the numerical behaviour but deteriorates the result.
- To control the high pressure of the vapour cycles the air mass flow rate is used as the controlling variable. A too small air mass flow leads to an increase of the hot loop coolant temperatures. Different control strategies such as a head pressure control valve have to be investigated.

NOMENCLATURE

c_p	specific heat capacity	(kJ/kgK)
$\dot{m}$	mass flow rate	(kg/s)
n	number (of discrete elements)	(-)
p	pressure	(Pa)
$\dot{Q}$	heat flow rate	(W)
Δt	time interval	(s)
t	time	(s)
T	temperature	(K)

Subscripts

compr	compressor
cond	condenser
in	inlet
min	minimum
out	outlet
sync	synchronization

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