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Implementation of a Modelica Library for Simulation of Refrigeration Systems

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Abstract

The physical modelling and transient simulation of refrigeration systems can be useful within the specification, development, integration and optimisation. Therefore, a model library for vapour compression cycles has been implemented. The library is based on the free Modelica library ThermoFluid and contains basic correlations for heat and mass transfer and pressure drop, partial components for control volumes and flow resistances and advanced ready-to-use models for all relevant components of refrigeration systems like pipes, heat exchangers, compressor, expansion devices and accumulator. The library currently enables the use of two refrigerants (CO_2 , R134a), but due to the structure of the library the extension to other refrigerant medium models is quite easy to realise. The modelling approach, the structure of the library and some validation results are presented in this paper.

1 Introduction

The modelling and simulation of refrigeration systems is of interest for several problems:

- Development and testing of control strategies and controller configurations
- Prediction and investigation of cycle dynamics like cool-down performance, start-up behaviour, pressure gradients and torque at the compressor
- Prediction of power consumption and COP (Coefficient Of Performance)
- Design and evaluation of heat exchangers
- Determination of optimal refrigerant charge
- Integration of refrigeration system as a subsystem within other systems like air-conditioning systems of automotives, buildings and aircrafts
- Development of combined heat pump systems for cooling and heating
- ...

This listing is incomplete but it clarifies the need of transient analysis of refrigeration systems by dynamic simulation.
The development of a Modelica library for refrigeration systems is being realised within two joint research projects founded by Airbus Deutschland GmbH, Hamburg, and DaimlerChrysler AG, Stuttgart. The aim of both projects is the development of a tool for transient simulation to support the research and development of new refrigeration technologies and to optimise currently used systems. In general, both projects focus on vapour compression cycles. A schematic of a vapour compression cycle is shown in Figure 1 (left figure). The main components are compressor, condenser, accumulator, expansion valve and evaporator. The working fluid is compressed in the compressor from suction line state to the high pressure line. In the following condenser the heat is rejected isobaric from the working fluid to the ambient or to a secondary coolant. After the condenser the accumulator is placed, where the subcooled fluid is dried by a desiccant. The expansion valve throttles the fluid isenthalpic to the low pressure level. In the evaporator the fluid is evaporated and superheated isobaric by removing a heat flux from the cooling medium.
The working fluid has to fulfil several requirements depending on the area of application. One of these

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requirements is the environmental sustainability of refrigerants. At the moment, this fact is an important driving force for the development of new refrigeration technologies, since the prohibition of currently used refrigerants is discussed. In the beginning of the 1990s the HCFCs based refrigerants were prohibited due to their ozone depletion potential (ODP). The used substitutes, the so called CFCs, have no ODP but the global warming potential of these refrigerants is very high in comparison to other, so called natural refrigerants like water, CO_2 , ammonia or hydrocarbons (propane, butane) [1]. Therefore, the prohibition of CFCs is discussed within the European Union and it seems to be realistic, that these refrigerants will be prohibited until the end of this decade [2].

Carbon dioxide (CO_2 , R 744) as a natural refrigerant was rediscovered and has recently demonstrated a very high potential to substitute currently used refrigerants in the area of mobile/automotive air-conditioning and cooling [3], [4]. This development is caused by the thermodynamic, transport and environmental properties of CO_2 . A schematic of a transcritical CO_2 cycle is shown in Figure 1 (right figure). The cycle consists of the same components as other vapour compression cycles, but it is supplemented by an internal heat exchanger. The internal heat exchanger is an essential component in a CO_2 -cycle to realise an acceptable COP.

For the automotive application the working fluid R134a is state-of-the-art. Therefore one purpose of the developed library is focused on this area. In the automotive application the AC-system is one important key for the passenger's comfort. On the other hand the AC-system influences the fuel consumption and the emission of the vehicle. For future automotives and for the aircraft application new, CO_2 -based refrigeration technologies are investigated [5]. The support of this development is also one purpose of the library. In this paper the basis of the library and an outline of the library content is given. Furthermore validation results of the CO_2 -models are presented and discussed with regard to the measurement uncertainty and the uncertainty of parameters.

2 Library for refrigeration systems

The aim of the modelling is to implement a library with physical based models of components of refrigeration systems. At the moment the library enables investigations with two refrigerants (CO_2 , R134a). But the realised structure allows the extension of the library by other refrigerants. Such a library can be used for investigations of components and complete refrigeration cycles. Furthermore it is of great interest to conduct dynamic simulation as well as steady state simulation of systems and single components, especially heat exchangers. This should be able with one tool and using the same models. The numerical investigation of heat exchanger components is of particular interest to find optimised heat exchangers for limited space. On the other hand, the concept of connectors in Modelica provides the opportunity of using the same heat exchanger models for single component simulation as well as for a complex cycle simulation. Finally, the library can be used for simulation and evaluation of different system designs in various applications.

2.1 ThermoFluid library

The implemented refrigeration library is based on the free Modelica library ThermoFluid [6], [7], [8]. The ThermoFluid library, especially its base classes and partial components, offers a good base for the modelling of refrigeration systems with respect to the implementation of the three balance equations and the method of discretisation. The basic design principles of the library are:

- models are designed for system level simulation,
- one-dimensional one- and two-phase flow is considered,
- one unified library for lumped and distributed parameter models,
- bi- and unidirectional flows are supported,
- conservation laws are implemented separately from the medium models, in order to improve reusability.

The use of distributed parameter models suggests the finite volume method as discretisation method. The finite volume method is very common for system modelling and one-dimensional discretisation [9]. The thermodynamic model holds the equations for total mass and internal energy for a control volume with constant volume:

$$\frac{dM}{dt} = \dot{m}_{in} - \dot{m}_{out} \quad (1)$$

$$\frac{dU}{dt} = \dot{m}_{in} \cdot h_{in} - \dot{m}_{out} \cdot h_{out} + \dot{Q} + \dot{W}_s \quad (2)$$

The fluxes on the border of the control volume are calculated by the half grid staggered flow model, which

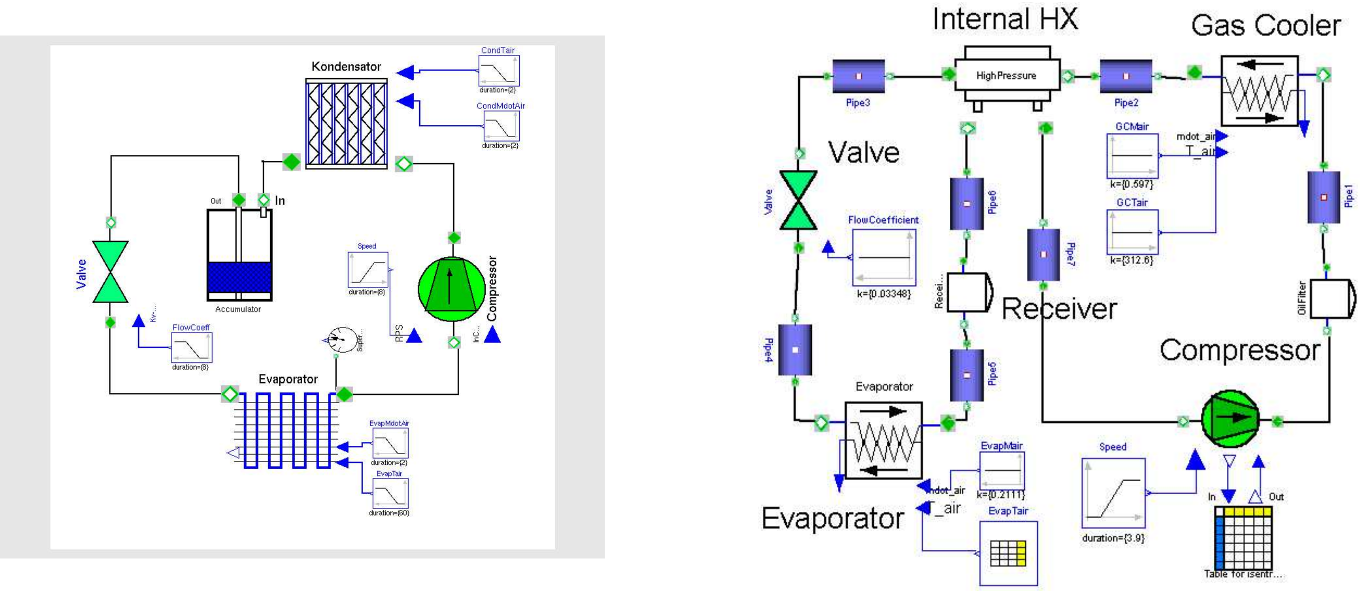

Figure 1: Schematic of vapour compression cycle and of a transcritical CO_2 process

holds either a stationary pressure drop model or the dynamic momentum balance:

$$\Delta z \cdot \frac{d\dot{m}}{dt} = \dot{I}_{in} - \dot{I}_{out} + (p_{in} - p_{out})A - \Delta p_{loss}A - M \cdot g \cdot \sin(\beta). \qquad (3)$$

The state variables of $\{M, U\}$ for the thermodynamic model are numerically not efficient. Therefore, the equations (1) and (2) are transformed into a form with $\{p, h\}$ as state variables. The constitutive equations needed for the calculation of pressure drop and heat flow in the equations (2) and (3) are not implemented in the ThermoFluid library yet.

2.2 Content of the library

So far, the most important models and classes have been implemented in the model library. As already mentioned the structure of the library enables the extension by other refrigerant models. This leads to a separation into a more general part and a specialised part. The general part holds the implementation of base and partial models like heat transfer and pressure loss correlations or flow resistance. All these models are independent of a specific medium model. Nevertheless, most of the models need medium properties for their execution. Therefore, the different medium models have to be implemented in the same way using the same names of variables and records and realising the same structure. In the specialised part of the library holds ready-to-use models.

Some of the models which have been implemented in the library are:

- **Heat transfer and pressure loss relations for the whole fluid region:**
  These constitutive equations are used for the calculation of heat flux and pressure drop due to friction, which are terms within the balance equations of energy and momentum. Most of the implemented correlation are state-of-the-art:
  The heat transfer of *single phase* flow can be calculated with Nusselt-Number based correlations for laminar and turbulent flow and smooth transition between both [10]. The heat transfer coefficient for *condensation* is computed by the assumption of film condensation using the correlation from *Shah* [11]. For the *boiling heat transfer* a superposition model introduced by *Chen* can be used, which takes forced convection and nucleate boiling into account [11]. These correlations are valid for two phase flow of refrigerants in general. A more accurate correlation has been implemented for the evaporation of R134a, which is based on the superposition model and has been adapted by *Gungor and Winterton* [12]. All these correlations calculate local heat transfer coefficients. Therefore, a discretised modelling of flow with heat transfer is required.
  The pressure drop of single phase flow is calculated depending on the Reynolds-Number [13]. To avoid event iterations, a function has been implemented generating a smooth transition be-

tween the different correlations and the areas of validity [16]. The pressure drop of two-phase flow is computed by a correlation of *Friedel*, which is simple but fairly accurate [10]. The implemented correlations have been validated for R134a [12] and CO_2 [14] with experimental data and they show mostly a fair accuracy. The advantage of using the same correlations for both refrigerants is the simplicity of their implementation. The loss of accuracy by not using specialised correlations is acceptable since the most important heat transfer is outside the pipe.

- **Models for the air side of heat exchangers:**
The balance equation of energy is implemented by the finite volume method [9]; heat transfer correlations for the air side have been implemented [15] as well as medium properties of air using polynomial fitting. The condensation of humidity is taken into account in the energy balance but not in the heat transfer coefficient. The approach for modelling the condensation is quite simple: If the air temperature is equal to or below the dew point temperature of the humid air, condensation occurs. By this approach the humid air is saturated at the outlet.

- **Medium models:**
The medium properties are calculated based on the implementation in ThermoFluid. The medium models for R134a and CO_2 have been customised with regard to the implemented constitutive equations; e.g. the transport properties, the phase boundary properties and the surface tension are calculated within the medium model. To avoid event iterations at the phase boundary a crossing function has been implemented generating a smooth transition between the model of the two phase and that of the single phase area.

- **Pipes and heat exchangers:**
Based on the medium model, classes of ThermoFluid, the heat transfer and pressure drop correlations and the air side models, pipes and heat exchangers have been modelled. The pipes are modelled as one-dimensional discretised flow just like the air flow itself. By this assumption the refrigerant flow is treated as homogenous. For the air side a non-homogenous distribution of the air mass flow can be provided. Since the heat exchangers are built up from basic elements for the refrigerant flow, the wall and the air flow, different types of heat exchangers can be modelled easily. Due to the discretisation of the flow, the state of the refrigerant and the heat transfer along the heat exchanger can be predicted with the models. Up to now several types of counter flow, cross flow and cross-counter flow heat exchangers have been modelled and validated successfully. A more detailed description of the modelling of the heat exchangers is given in [16] and [17].

- **Compressor:**
The model is made for a reciprocating compressor. Therefore, the mass flow is calculated by the general equation (4) of a reciprocating compressor and the enthalpy change is calculated according to the isentropic efficiency by equation (5):

$$\dot{m}_{co} = f \cdot \lambda \cdot \rho_{in,co} \cdot V_{dv} \tag{4}$$

$$\Delta h_{co} = h_{out,co} - h_{in,co} = \frac{h_{out,co,is} - h_{in,co}}{\eta} \tag{5}$$

By using these equations the compressor is assumed to have no dynamics. The efficiencies can be provided by measured characteristic fields of a known component or are set as constant parameters if they are unknown and must be estimated.

- **Expansion valve:**
The throttling process is treated as isenthalpic and the pressure drop is calculated according to the flow coefficient of the valve using an algebraic equation [18]:

$$\dot{m}_{ev} = \frac{1}{3600} \cdot K_V \cdot Y \cdot N_6 \cdot \sqrt{x \cdot p_{in,ev} \cdot \rho_{in,ev}} \tag{6}$$

where is:

$$x = \frac{p_{in,ev} - p_{out,ev}}{p_{in,ev}} \tag{7}$$

$$Y = 1 - \frac{x}{3 \cdot F_\gamma \cdot x_T} \tag{8}$$

Since the flow coefficient K_V and the critical differential pressure ratio x_T result from specific valve data and construction, the model has to be parameterised with corresponding data.

- **Accumulator / Receiver:**
In general, the function of an accumulator or a receiver in a refrigeration system is to accumulate refrigerant, since the necessary refrigerant charge depends on the operating mode of the system. Therefore, additional charge has to be stored. In R134a systems the accumulator is placed after the

condenser and contains a desiccant for drying the refrigerant, see Fig 1. Whereas the receiver in CO_2 systems is placed at the suction line after the evaporator, see Fig 1. For modelling, the receiver is separated in a separator, a tube for the gaseous outflow, an orifice for the liquid outflow and a junction mixing the two outflows. This modelling approach is similar to [19]. The incoming two phase flow is separated into its liquid and vapour phase. The outlet condition is calculated by mixing the two mass flows through the tube and the orifice, which are modelled as flow resistances with specific friction factors. The friction factors can be estimated for steady state; then the vapour fraction at the receiver outlet is the same as at the receiver inlet. The receiver is modelled as adiabatic.

- **Flow splits and junctions:**
  For these models, classes of ThermoFluid are used. The pressure drop in the momentum equation uses special correlations for splits and junctions taking the ratio of mass flow into account [20]. The change of mass flow direction is also taken into account in the implementation.

3 Experimental setup

The experiments were carried out at the CO_2-experimental system built at the Department of Aircraft Systems Engineering of the TUHH described in detail by Schade [21]. The test rig was constructed with prototype components of the automotive application. It realises the process of a transcritical cycle introduced by Lorentzen and Pettersen [22], which is extended in the realisation by three, parallel cooling points/evaporators. The main objective of the experimental investigations is control-oriented. Furthermore, steady state and transient data from the test rig should be used for the validation of the simulation models. The gas cooler is a cross-flow heat exchanger with three passes at the refrigerant side. The evaporators are cross-counter flow heat exchangers with eight passes in two layers. The internal heat exchanger is built as a counter-flow heat exchanger with coaxial tubes. The used compressor consists of an axial piston unit with variable or fixed displacement. The gas cooler is installed in an open channel whereas the evaporators are built in closed loop air-cycles. The temperature and mass flow rate of the air at the heat exchanger inlet is conditioned by electrical air heaters and fans. The temperature of CO_2 at inlet and exit of each component is measured with thermocouples attached to the surface. The pressure is also measured at inlet and exit of each component. The CO_2 mass flow rates are measured at different points in the system by using Coriolis type meters. Hot-wire anemometers are used to measure the air mass flow rates through the heat exchangers. The air inlet and exit temperatures are measured by thermocouple grids. The uncertainties for the measurements are listed in Table 1. Especially the uncertainties of the air temperature after the gas cooler is very high due to the inhomogeneous distribution of temperature. Due to error propagation the resulting uncertainty of the calculated capacities can be up to ± 12 % for both gas cooler and evaporator.

Table 1: Absolute, resp. relative error of measurement

Pressure at suction line	± 50 kPa
Pressure at high pressure side	± 100 kPa
CO_2 temperature	± 0.7 K
Air temperature evaporator in/out	± 1 K
Air temperature gas cooler in	± 1 K
Air temperature gas cooler out	± 3 K
CO_2 mass fbw rate	± 0.2 %
Air mass fbw rate	± 4 %

4 Validation of air-CO_2 heat exchanger models

Simulations in a test configuration have been run with the gas cooler model discretised with $n_{CO2} = 9$ for the CO_2 flow and $n_{air} = 4$ for the air-side flow; the evaporator was discretised with $n_{CO2} = 8$ and $n_{air} = 4$. These discretisations were chosen for the simulation due to acceptable execution time for a simulation run of a complete refrigeration cycle.
The test configuration consists of a source providing pressure and enthalpy at the heat exchanger inlet and a mass flow sink generating a defined mass flow at the outlet. The source and sink are used to set the boundary conditions resulting from measured data at the component.
Figure 2 shows the measured and simulated capacity at the gas cooler for a wide set of operating conditions (p_{CO2}:7-11.3 MPa, $\dot{m}_{CO2}$:45-230 kg/h, $T_{CO2,in}$:345-400 K). What can be seen from the comparison, is, that most of the simulated capacities are in within the error of ± 12 %. The deviation becomes higher near

the critical point which can be traced back to the chosen discretisation of the model. A higher discretisation would represent the influence of the pseudo-critical point more accurately. The discretisation also affects the exit CO_2 temperature, which the model predicts for supercritical gas cooling within 1.1 K and 2.6 K higher than the experimental data and outside the error of ± 0.7 K, see Figure 2. For operating conditions below the critical pressure the model predicts the capacity very well. The influence of discretisation with regard to consistence with experimental data is shown by Limperich [17].

In Figure 3 the results of the evaporator model are compared with experimental data. The boundary conditions were p_{CO2} within 3.017-5.01 MPa and $\dot{m}_{CO2}$ within 45-140 kg/h. As Figure 3 shows, the model predicts the capacity within ±7.4 %. The air outlet temperature is predicted within ± 0.8 K which is within the uncertainty of measurement. The humidity of the air was not taken into account since the evaporator is integrated in a closed loop air-cycle. Therefore it can be assumed that the air is dehumidified after a short time of operation.

The validation of the internal heat exchanger is shown in Figure 4. The comparison of the transferred heat fluxes shows a good agreement within an uncertainty of ± 10 %.

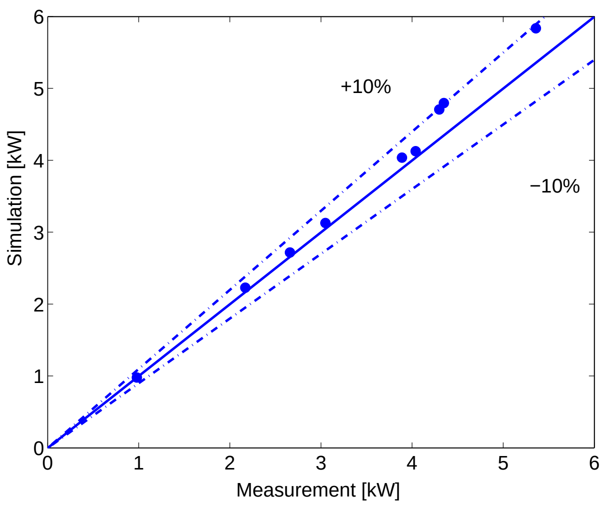

Figure 4: Measured and simulated capacity at the internal heat exchanger within ± 10 %

5 Transient simulation of a CO_2-system

In the following, results of the transient simulation of the above mentioned CO_2-system are presented. The simulated model is shown in an object diagram in Figure 1. This configuration represents the available CO_2-test rig on basic level with one evaporator. The results are compared with data of a start up of the system and following step changes in compressor speed as shown in Figure 5. The air inlet temperature of the evaporator also changed during the experiment, Figure 5. The other boundary conditions stayed constant and are listed together with the initial states in Table 2. All these data were taken from the experiment.

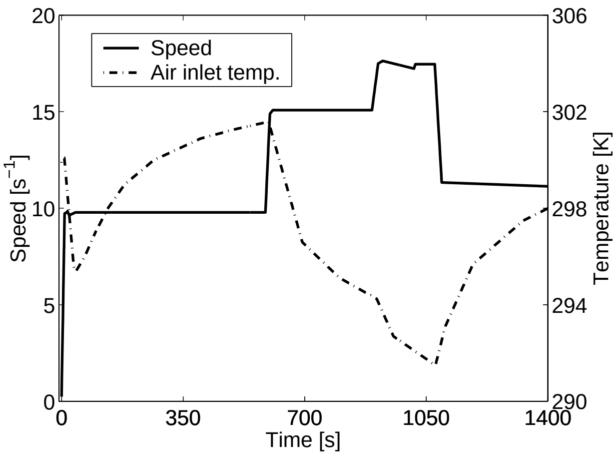

Figure 5: Step changes in compressor speed and run of air inlet temperature at the evaporator in the experiment; set as boundary condition of simulation run

Table 2: Boundary conditions and initial values of the simulation run corresponding to the experiment

Compressor	fixed displacement $V_{dv} = 33.5$ ccm
Expansion valve	100 % open $K_v = 0.0264$ m^3/h
Gas cooler	$\dot{m}_{air} = 2100$ kg/h $T_{air,in} = 312$ K
Evaporator	$\dot{m}_{air} = 760$ kg/h
System volume	$V_{tot} = 3.62$ l
Specific refrigerant charge	267 kg/m^3
Initial value	$p_0 = 5.7$ MPa $h_0 = 425$ kJ/kg
Initial value receiver	$p_0 = 5.7$ MPa $h_0 = 295$ kJ/kg

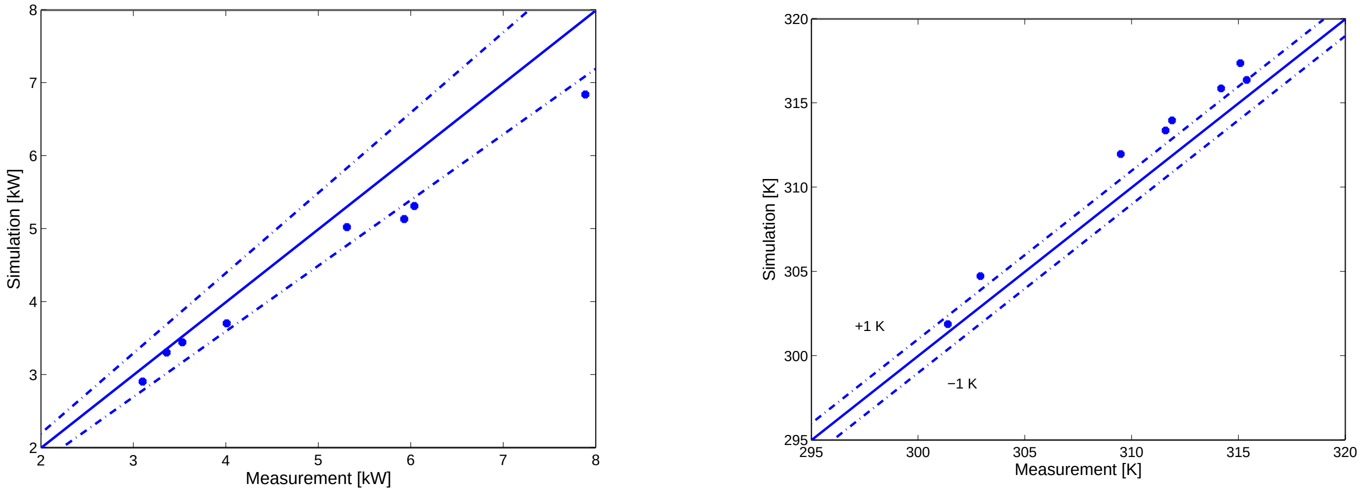

Figure 2: Measured and simulated cooling capacity at the gas cooler within ± 10 % and approach of refrigerant temperature at gas cooler exit (± 1 k)

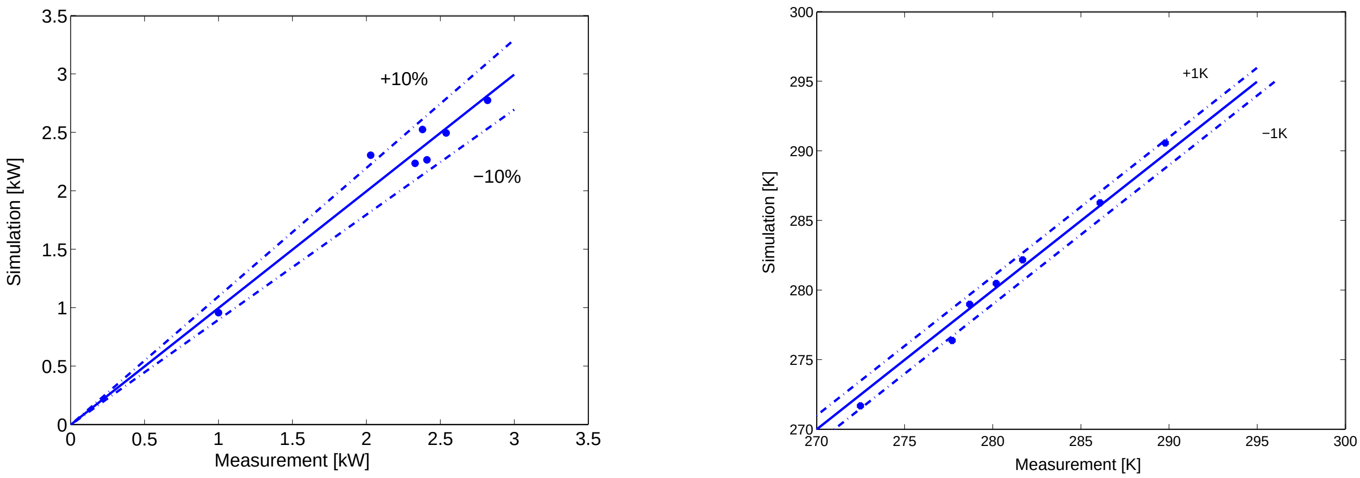

Figure 3: Measured and simulated cooling capacity at the evaporator within ± 10 % and approach of air outlet temperature (± 1 k)

In Figure 6 the simulated and measured pressure at compressor inlet and exit is plotted versus time. The plotted experimental data are filtered due to the very high measurement noise. What can be seen from the comparison is a fair agreement of the absolute values as well as the time response for the pressure at the compressor inlet. At the compressor exit there is only a partial agreement; especially at the beginning there is a clear deviation in absolute values and time response. The model predicts a pronounced undershoot whereas the experimental data show a smaller undershoot. This behaviour can also be seen in the comparison of the mass flow rate at the expansion valve in Figure 7. In general, there is a systematic underestimation of the mass flow rate by the model, which is larger then the tolerance of the used mass flow meter. The run of pressures and mass flow rates are coupled in such systems. Therefore deviation of one value influences the other values and vice versa. Reasons for this behaviour can be seen in the modelling of the compressor using algebraic equations instead of a physical model. This leads to the use of characteristic fields for the efficiencies, which were generated by measurements at steady state. Especially at the start up of the system the used efficiencies in the model are probably different from the real behaviour of the compressor. Furthermore the available values of the flow coefficient of the expansion valve are independent from the inlet state and the pressure difference at the valve. The flow coefficient is only a function of the open ratio. From physical point of view it seems to be obvious, that this simplified characteristic does not represent the complete operating range. So, the uncertainty of component-specific parameters like compressor efficiencies and flow coefficient of the valve influences the simulation results. This known influence can be accepted for the level of system simulation and has to be taken into account for the validation of the models.

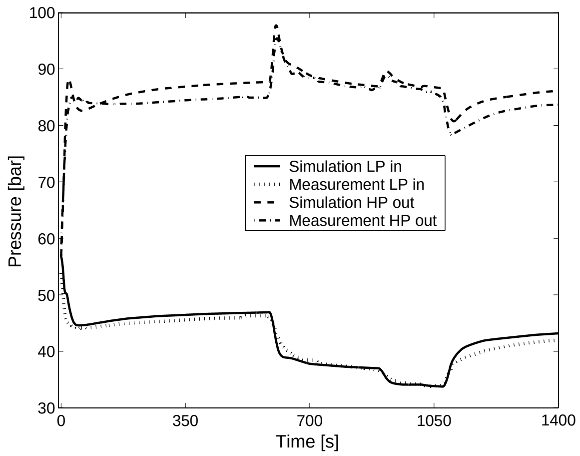

Figure 6: Transient run of the pressure at compressor inlet and exit; comparison between simulation and measurement

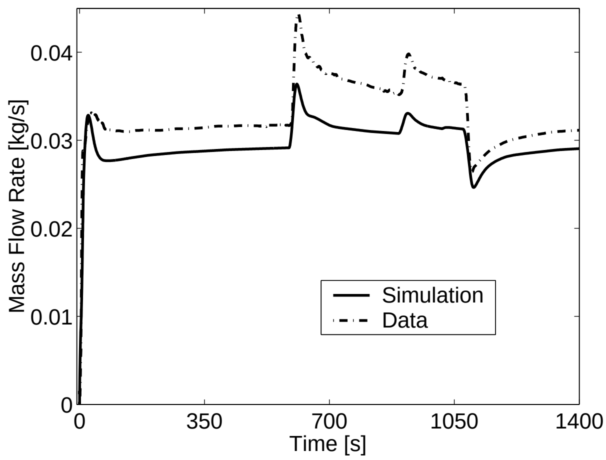

Figure 7: Transient run of the mass flow rate at the expansion valve; comparison between simulation and measurement

The object diagram of a CO_2 -loop with two parallel evaporators is shown in Figure 8. This schematic represents the extension of the above mentioned CO_2 test rig. By this example it can be shown that the transient simulation of such a system, especially the split of the mass flow is predicted correctly by the models. In Figure 9 the simulated and measured mass flow rates are plotted. Their comparison shows a good agreement. The mass flow rates in the branches differ, since the expansion valves have different flow characteristics.

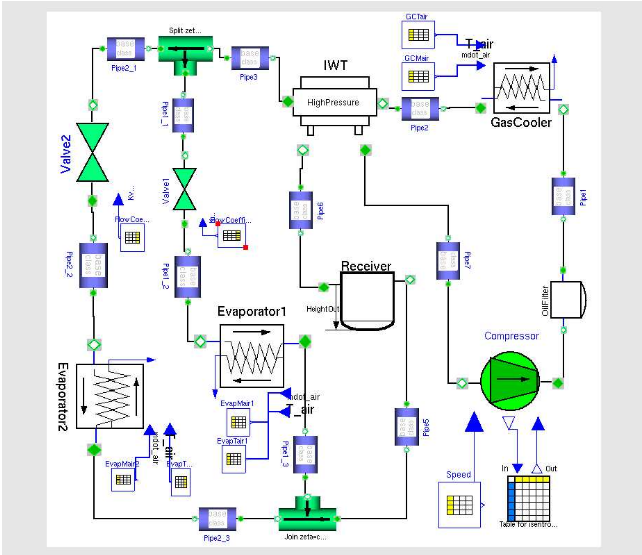

Figure 8: Object diagram of the simulated CO_2 -loop with two parallel evaporators; representing the CO_2 test rig

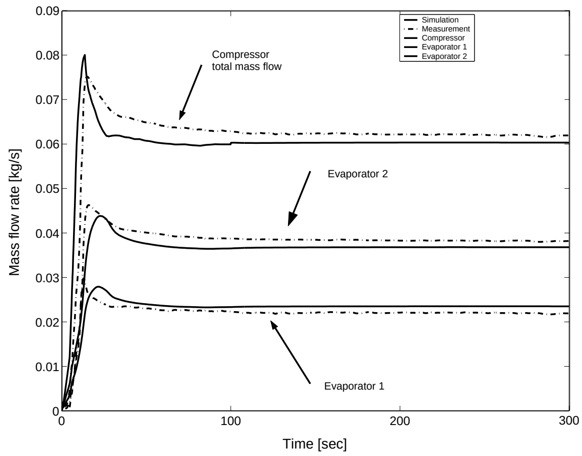

Figure 9: Simulated and measured mass flow rates

6 Conclusion

In this paper the modelling and implementation of a Modelica library for refrigeration systems was presented. The implemented library provides both base models for modelling of components and usable models of components for the automotive and aircraft application. The intention is to create a library for the simulation of single components and complete cycles.

Such a library can be used to make fundamental investigations of refrigeration systems. Furthermore, it can be used for the optimisation of specific heat exchangers, for the evaluating of an optimal system configuration and for the layout and optimisation of the system control. The presented simulation results for the steady state of different types of CO_2-heat exchangers show a fair correspondence with measured data. The results of the transient simulation show a good agreement in comparison with experimental data.

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