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Transient 1D heat exchanger model for the simulation of domestic cooling cycles working with R600a

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ABSTRACT

Generally, domestic refrigerators and freezers are running in non-continuous operation mode most of the time, which is a necessity to match cooling capacity to thermal loads. In currently available domestic appliances this matching is realized either by on/off or variable frequency control of the hermetic compressor, leading to a repetitive and transient change of the system state. In case of longer compressor runtimes when cooling capacity demand is high (e.g. pull down cycles, initial operation) steady state operating conditions might be reached.
The cycling transients cause losses in system efficiency, thus they should be reduced or avoided. To understand the complex transient physical processes and to optimize the cooling system efficiency, the use of numerical methods has turned out to be a promising approach.
For this reason, a 1D heat exchanger model, which has been successfully implemented in a domestic cooling cycle simulation tool, is presented in this work. The heat exchanger model is a further development of the model being presented in Berger *et al*. (2012). The same mathematical framework is used for modeling the evaporator and condenser. In order to compute the void fraction, the pressure drop and the heat transfer in case of evaporation and condensation special empirical models, which are proposed in literature, have been implemented. Finally, the numerical predictions are compared to experimental data gained from a purpose-built test rig.

1. INTRODUCTION & REVIEW

The heat exchangers of refrigeration devices – no matter whether they are integrated in domestic or commercial appliances – are the main components to influence the transient thermal performance of the whole system. The working process within the heat exchangers can be described as a complex 3-dimensional fluid flow with strong non-linear phase transitions of the refrigerant, which are caused by a combined heat transfer including convection, radiation and heat conduction. Furthermore, the working fluid is not a pure refrigerant but actually a refrigerant/lubrication oil mixture, whose composition is unknown and hardly detectable. Regarding on/off controlled systems, the phase changes during start-up and shut-down are highly transient and consequently crucial for all further responses of the system.
The handling of these severe transients is the major challenge in developing a transient heat exchanger model. Generally, heat exchanger models can be classified into single component models (component level analysis) and models which are implemented in cycle simulation tools (system level analysis). The system approach requires a compromise between computation accuracy and computation speed because the results of one component are given as boundary condition for the adjacent components (Berger *et al.*, 2012).

The development of refrigeration cycle simulation tools, and therefore transient heat exchanger models, began in the late 1970s (detailed summaries can be found in Philipp, 2002 or Hermes and Melo, 2008). Nowadays, typically used cycle simulation tools are non-commercial and simplified combinations of 0-dimensional and 1-dimensional modeling approaches to approximate the real 3-dimensional working process with an appropriate accuracy. Even though computer power has increased significantly in recent years, 3-dimensional simulations with commercially available software packages have only been used to investigate the air flow in the refrigerator cabinet but not for the computation of the two phase flow in the heat exchangers – neither for steady state conditions nor for transient conditions (Laguerre *et al.*, 2007, Gupta *et al.*, 2007).

The following literature review is a representative summary of research activities dealing with transient heat exchanger modeling and the experimental validation within the last 30 years.

In 1988, Melo *et al.* presented their numerical and experimental investigation of the dynamic behavior of a vapor compression refrigerator. Condenser and evaporator were modeled by three and two control volumes, respectively. Experiments were carried out using a top-mount refrigerator and a comparison between simulation data and experimental data was shown for the suction and discharge pressure as well as for the air temperature in the freezer and in the cabinet. No transient thermal information about the heat exchangers was presented.

A moving boundary element approach for the heat exchangers can be found in Janssen *et al.* (1988) in their theoretical and experimental investigation of a dynamic model for small refrigerating systems. An upright freezer with evaporator shelves was used for the experimental validation. Regarding the heat exchangers, their measured temperatures differed considerably from the simulated data especially during the highly transient start-up phase. This fact was explained thereby that the model was adjusted with steady-state experimental data.

In his study on energy optimization of refrigeration systems, Jakobsen (1995) chose a lumped parameter approach to describe the physics of the heat exchangers. He compared the simulated results with experiments of an R134a charged refrigerator, which had a built-in hot-wall condenser and a roll-bond evaporator. Because of the simple lumped parameter approach, temperatures, especially the wall temperatures, were not directly comparable with the measured ones. Nevertheless the predicted trends seemed to be quite reasonable.

A general moving boundary formulation was shown by Willatzen, Pettit and Ploug-Sorensen (1998a, 1998b) for the dynamic simulation of evaporators and condensers in refrigeration. The mathematical framework was described in detail – however no experimental validation was carried out. The authors stated that the simulation results matched their experiences with refrigeration systems and experimental work.

A numerical simulation of a variable speed refrigeration system, which was charged with R12 or R134a was carried out by Koury *et al.* (2001). The heat exchangers were modeled using the distributed parameter method. Concerning the experimental validation no data of the transient evolution of temperatures and pressures in the evaporator or in the condenser was shown.

In a comprehensive study, Philipp (2002) used numerical methods to optimize domestic refrigerators. Philipp used the moving boundary approach for modeling the condenser and the evaporator. He compared the simulation data with experimental investigations of an R600a charged refrigerator which was equipped with a roll-bond evaporator and a finned-tube condenser. During compressor shut-down the simulated mean wall temperature of the evaporator deviated by 5K but when the compressor was running, the deviations decreased to 3K.

The transient response of a finned-tube condenser in a household refrigerator was investigated by Porkhial *et al.* (2006). The authors applied the distributed parameter method with an efficient two-level iteration method. For the experimental validation a top-mounted household refrigerator working with R12 was used. The comparison of the measured data and computed data yields that the predicted temperatures deviate about 10% from the measured ones especially at the inlet and the outlet of the condenser.

A first-principles simulation approach was introduced by Hermes and Melo (2008, 2009) to assess the energy performance of household refrigerators including the start-up and the cycling transients of household refrigerators. An R134a charged frost-free top-mount refrigerator with a tube-fin evaporator and a wire-and-tube condenser was used for their experiments. The only presented transient data compare suction and discharge pressure, air temperature in freezer and cabinet and the compressor power. No information about the transient thermal evolution of the heat exchangers was presented.

The dynamic behavior of a vapor compression cycle including the shut-down and start-up was investigated by Li and Alleyne (2010). The authors applied the moving boundary element method with an advanced switching scheme to capture the dynamic transition of the refrigerant state within the heat exchangers. Different validation cases were carried out with a special experimental system which consisted of a tube-and-fin evaporator and a tube-and-fin condenser. The experimental results were agreeing well with the computed ones.

Nan Liang *et al.* (2010) presented a dynamic simulation of variable capacity refrigeration systems under normal and abnormal conditions. The authors developed a moving boundary model for evaporator and condenser. For the validation of the model a specific testing system was built. Calculated values from the mathematical model show reasonable accordance to the experimental data.

To sum up, various modeling approaches, ranging from the simple lumped parameter approach, the moving boundary approach to the distributed parameter approach, have been applied by researchers. It has to be mentioned that many works lack a validation of the thermal behavior of the model because in many cases only the transient evolution of the resulting pressure was presented but no information on the thermal evolution of the heat exchangers was shown. An as accurate as possible prediction of the pressure is absolutely essential but considering cycle simulations also the thermal conditions are important because the states at the inlet and the outlet of the heat exchangers serve as boundary conditions for the adjacent components. In very few papers, a good holistic conformity between the computed and simulated data was presented.
Therefore, the aim of this work was the further development of the transient, 1-dimensional heat exchanger model, which was presented by Berger *et al.* (2012). A distributed parameter approach was chosen and the authors paid peculiar attention to the implementation of recently published works in order to describe the pressure drop, the heat transfer mechanisms and the void fraction mathematically. Experiments were carried out on a purpose-built test rig to validate the predictive capability of the heat exchanger model. In this study the focus was put on the high pressure side of the refrigeration cycle and thus on the thermal behavior of the condenser.

2. THE HEAT EXCHANGER MODEL

2.1 Fundamental model for evaporator and condenser

For the mathematical description of the physical processes, which occur in the heat exchanger, a distributed parameter approach was chosen. Both heat exchangers were divided into equally sized finite volumes and in each of these cells, the first law of thermodynamics, which governs the conservation of energy, is solved separately for the refrigerant and the heat exchanger walls. With the help of the equation of continuity, the mass in the cell and the mass fluxes have been calculated. Because of the complexity of the working process within the heat exchangers, the following simplifying assumptions have been applied in the modeling of the condenser and for the evaporator:

- The refrigerant flow was assumed to be 1-dimensional.
- The refrigerant channels were considered as straight, horizontal tubes with a constant cross-section and without any wires.
- Gravitational effects were neglected.
- The axial heat conduction in the refrigerant was neglected.
- The terms considering the kinetic and potential energy in the energy conservation law were eliminated.
- The liquefied refrigerant and the vaporized refrigerant were assumed to be in thermal equilibrium.
- The presence of moisture was neglected.
- The working media is pure refrigerant R600a (isobutane). The lubricant oil/refrigerant mixture was not considered in the heat exchangers.
- The single phase heat transfer coefficients were computed by implementing the Dittus-Boelter-Kraussold correlation according to Philipp (2002).

Based on these general model simplifications, different empirical models which account for the condensation and evaporation effects in the heat exchangers were implemented. Recently published correlations for the computation of the void fraction and the two phase heat transfer coefficient were chosen. In contrast to the majority of works on this topic, the pressure drop in the heat exchangers was not neglected in this work, as experiments have shown that the pressure drop, especially in the evaporator, can be significantly high. With the negligence of the pressure drop in the evaporator, the simulation would over-predict the suction pressure of the compressor and thus the discharged mass flow rate.
The following semi-empirical correlations were implemented in the condenser model:

- The void fraction was computed using the model proposed by El Hajal *et al.* (2003).
- For the calculation of the heat transfer coefficient in the two phase region, the flow pattern based heat transfer model by Thome *et al.* (2003) was chosen.
- The pressure drop model proposed by Quiben and Thome (2007) was implemented.

- A combined radiative and natural convection heat transfer coefficient approach according to Philipp (2002) was used to model the heat transfer at the condenser surface.

Because of the different physics of condensation and evaporation, other correlations were implemented in the evaporator model:

- The Steiner horizontal tube version of the vertical tube expression of Rouhani-Axelsson void fraction model was integrated (Steiner, 2006).
- The heat transfer coefficient during evaporation was computed by the flow pattern based heat transfer model proposed by Wojtan *et al.* (2005a, 2005b).
- The pressure drop model proposed by Quiben and Thome (2007) was implemented.
- The radiation in the refrigerator compartment was neglected.

2.2 Properties of the working media R600a

All investigations in this study were carried out with the refrigerant R600a (isobutane). The R600a thermophysical properties were computed with the REFPROP software (Lemmon *et al.*, 2012).

2.3 Developed computation algorithm (see Figure 1)

The following computation algorithm is based on the fundamental algorithm, which was presented by Berger *et al.* (2012). Besides the pressure iteration loop, an additional cell iteration to solve the energy conservation equation was included, which leads to a complex and nested heat exchanger model.

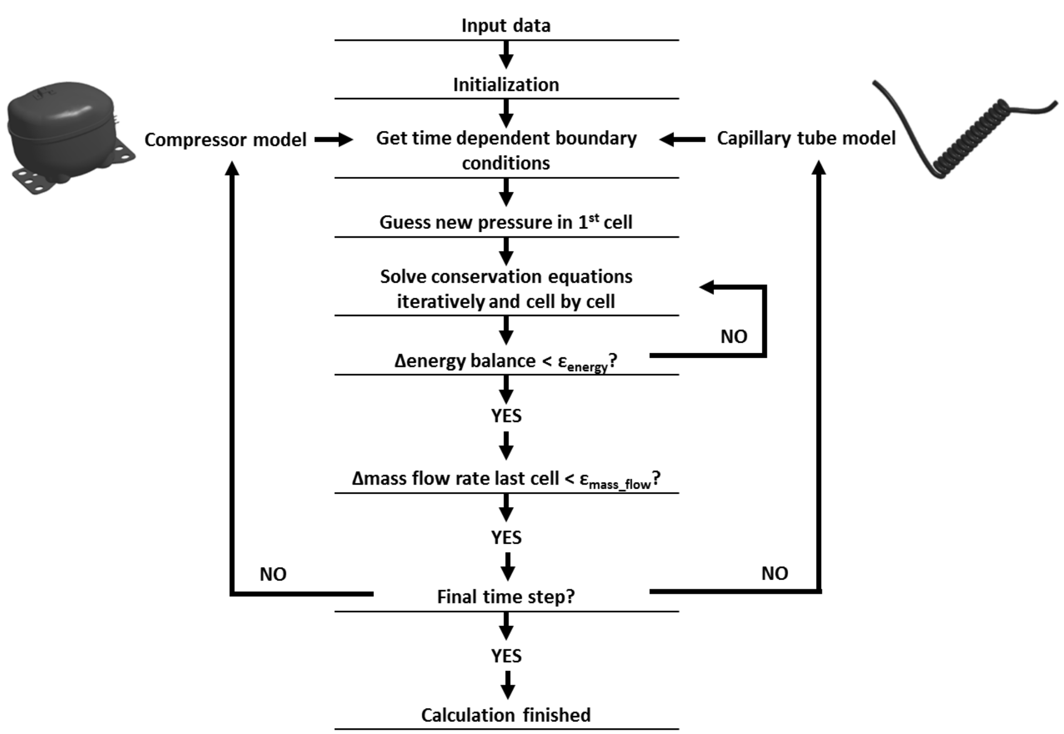

Figure 1: Iterative computation algorithm

The heat exchanger model was developed to be implemented in a cycle simulation tool. Thus different strategies were elaborated and advices mentioned in literature were considered to increase the computation performance:
The properties of the working media R600a at the saturation lines were approximated by cubic splines. All the other property correlations of the working media, which were needed during the computation, were provided in look-up tables.

A hybrid Regula-falsi/Newton's method iterative solver was developed and programmed in order to guarantee high stability as well as a good speed of convergence. To further accelerate the computation, extrapolation algorithms in time and space were implemented to improve the start values of the iteration loops.
Finally, an automatic adaption of the time step, which had a minimum of 0.5 seconds and a maximum of 2.5 seconds, respectively, was integrated.

3. VALIDATION OF THE CONDENSER MODEL

3.1 Experimental setup

The experimental validation of the condenser was carried out on a small cooling capacity refrigeration test rig. The test rig consisted of a hermetic reciprocating compressor for domestic appliances, an evaporator and a condenser. Furthermore, a capillary tube without internal heat exchanger was chosen as expansion device.
This study focused on the condenser. All the condensation correlations for the void fraction, for the pressure drop and for the heat transfer, which were implemented in the heat exchanger model, had been developed for horizontal tubes without any wires. Thus, a coil condenser (see Figure 2) with a small inclination was made from a simple copper tube to approximate the horizontal tube. The total length of the coil condenser was 20m and the inner and outer diameter of the copper tube was 0.004m and 0.006m, respectively. The resulting coil condenser had a height of 1.1m and its diameter was 0.6m.

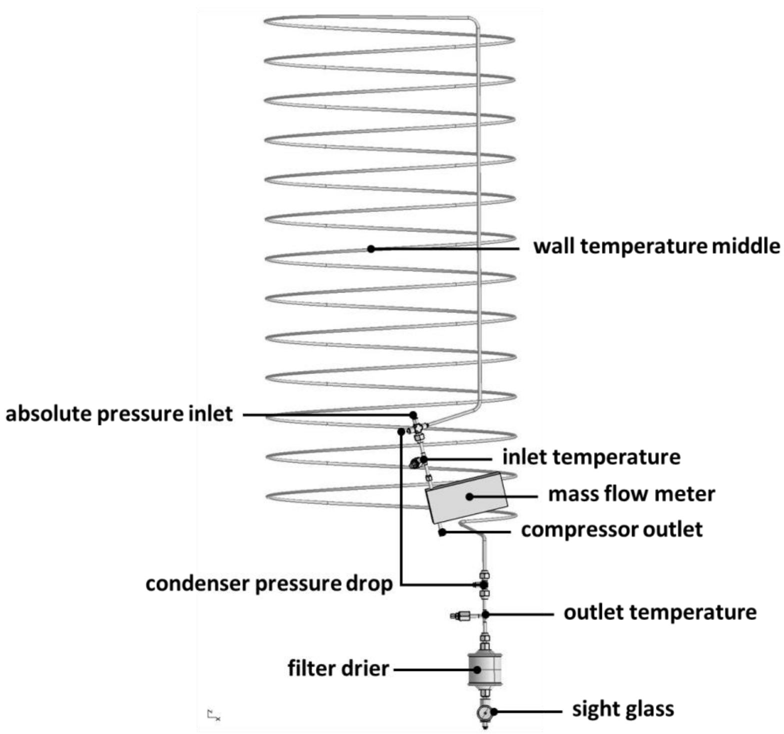

Figure 2: Research coil condenser equipped with measurement instrumentation

The coil condenser was equipped with carefully chosen sensor technology. To get the mass flow rate at the inlet of the condenser, a coriolis mass flow meter was placed there. To compute the enthalpy of the refrigerant at the condenser inlet, an absolute pressure transducer and a thermocouple were mounted at the outlet of the mass flow meter. A differential pressure transducer was installed to determine the pressure drop of the condenser. One thermocouple was added to the condenser surface in its middle to get information about the wall temperature. The pressure at the outlet of the condenser was computed with the absolute inlet pressure and the measured pressure drop value. Another thermocouple was placed in front of the filter drier.
All measurement data was collected via a NI Data Acquisition System (NI cDAQ 9188) and recorded by a LABVIEW routine.

3.2 Validation strategy of the condenser model

The transient validation of the condenser is quite complex. This can be emphasized for example by taking a look at the mass flow rates which enter and leave the condenser: The start-up period is characterized by the fact that the mass flow rate at the inlet is much higher than the mass flow rate at the outlet because the capillary tube chokes the refrigerant flow. Therefore, refrigerant is caught in the condenser and the refrigerant mass in the condenser increases. When the compressor is switched off, an inverse behavior can be observed: The mass flow rate at the inlet decreases to zero, nevertheless refrigerant is leaving the condenser towards the evaporator through the capillary tube because of the pressure gradient as long as the suction pressure and the discharge pressure are equalized.

For this reason, two mass flow meters would be necessary to capture the mass distribution during the cycling transients. The disadvantage of this approach is that the transient response of the whole system changes significantly because of the physical properties (relatively high thermal inertia, pressure drop) of the measuring device.

Hence, a different validation approach, which has no influence on the working process, was chosen: The outlet mass flow rate of the condenser was computed using a validated, adiabatic capillary tube model (Heimel *et al.*, 2014). The boundary conditions for the capillary tube model (pressure and enthalpy at the inlet; pressure at the outlet) were taken from the experimental investigations (see Figure 2). The measured mass flow rate and the enthalpy which was computed with the help of the measured pressure and temperature of the refrigerant after leaving the coriolis mass flow meter were defined as boundary conditions at the condenser inlet.

With this approach, it was possible to validate the start-up transients. During the shut-down period another challenge in transient validation occurs, as the refrigerant changes its state from sub-cooled liquid to a two-phase mixture, thus it is not possible to compute an explicit enthalpy based on pressure and temperature measurements. However, the enthalpy is an essential boundary condition for the capillary tube model. Therefore only start-up cycles were investigated in the validation process.

3.3 Results

Figure 3 shows the mass flow rates at the inlet and the outlet of the condenser, which are defined as boundary conditions. The mass flow rate at the inlet was derived from experiments. The mass flow rate at the outlet was computed by means of the capillary tube model by Heimel *et al.* (2014). This validation approach is justified by the fact that the mass flow rates at the inlet and the outlet become identical after a certain period.

Because of the difference between the two mass flow rates, refrigerant is caught in the condenser. The transient increase of the refrigerant is also depicted in Figure 3. An experimental validation of the accumulated mass is extremely time-consuming (see Björk and Palm, 2006) and therefore it was not carried out. Nevertheless, the pictured trend is quite reasonable and meets the authors’ expectations.

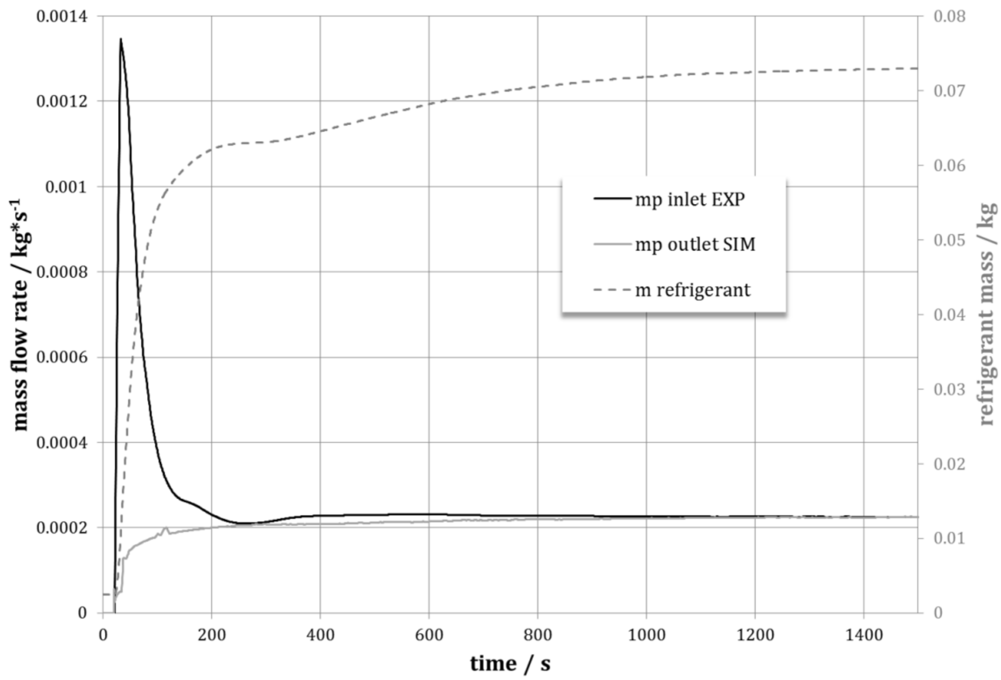

Figure 3: Transient mass flow rates and refrigerant mass in the condenser

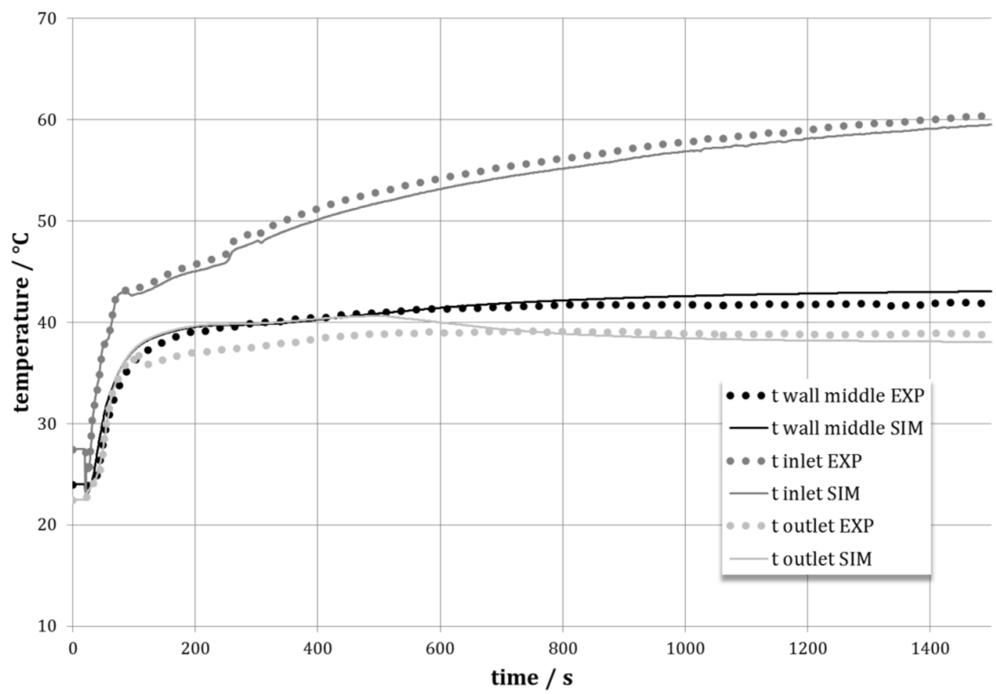

Figure 4: Temperature trends at different locations of the condenser

In Figure 4, the time variation of different temperatures during the start-up is depicted. Two temperatures of the refrigerant and one wall temperature in the middle of the condenser are compared. It can be seen, that the measured data matches the simulated data fairly well. The simulated wall temperature is higher than the measured temperature because the thermocouple is mounted on the surface of the coil condenser. Therefore the thermocouple is partly cooled by the ambient air.

The deviations of the refrigerant temperatures at the inlet and the outlet of the condenser can be explained by errors in the heat transfer predictions because of the uncertainties of the implemented models.
The transient pressure trend is shown in Figure 5. Especially in the first 100 seconds, a remarkable agreement between simulation and experiment can be seen. The deviation after 200 seconds can be explained by errors in the computation of the heat transfer and the mass flow rate at the condenser outlet.

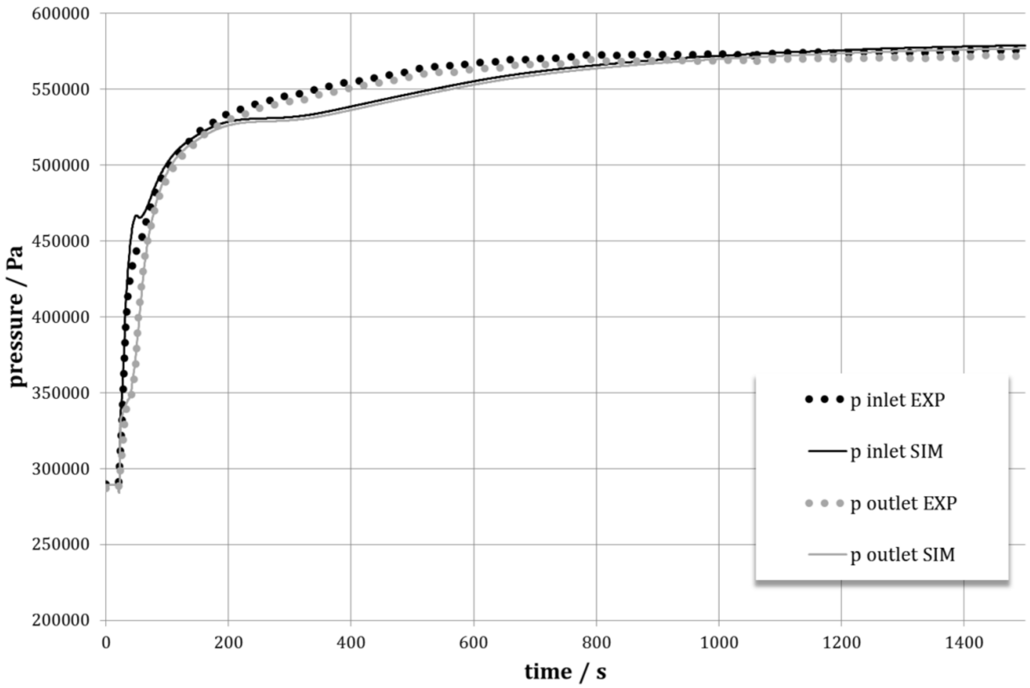

Figure 5: Transient pressure trends at the condenser inlet and at the condenser outlet

4. CONCLUSION

In this work a detailed heat exchanger model based on a distributed parameter approach was presented. The experimental validation has shown truly satisfying agreement with the predicted data. The minor deviations between experiment and simulation can be explained on the one hand by the simplifying assumptions, which were listed in chapter 2 and on the other hand by the major uncertainty factor in the computation of thermodynamic systems which is the approximation of the real heat transfer.
An essential part of this work was the extensive literature review on modelling the evaporation and condensation processes. Recently published correlations for refrigerants were chosen and implemented in the model to account for the different physical effects of condensation and evaporation. The high complexity of these models causes a high programming effort and it is extremely time-consuming to get them executable, especially in the case of a cycle simulation. As the comparison of the experimental data with the simulated data has shown already a truly satisfying agreement with the chosen correlations, no analysis of the sensitivity of the model concerning other correlations for condensation and evaporation was carried out up to now.
Regarding numerical issues it has to be emphasized that the convergence of the simulation is mainly influenced by the highly transient state changes during the star-up and the shut-down of the compressor. For this reason an adaption of the time step which depends on the mass flow rate of the compressor and of the capillary tube was integrated.
In future, this model is going to be implemented in a refrigeration cycle simulation tool and experimental validations with different heat exchanger designs (wire-and-tube condensers, forced convection condensers, finned-tube condenser) are going to be carried out.

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