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Investigation of the Dynamic Heat Transfer Coefficient of R-134a in a Horizontal Pipe

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ABSTRACT

This paper presents a test rig to measure the dynamic heat transfer coefficient of the refrigerant R-134a in a small diameter horizontal pipe. The experiment layout is basically a cooling cycle. The basic components are a receiver tank filled with R-134a, a pump, an evaporator pipe and a condenser. The evaporator pipe made out of copper is the main element. Thermocouples, pressure sensors and a mass flow sensor provide the information necessary to determine the heat transfer coefficient of the horizontal pipe. In addition to the experimental investigations, a numerical model is created. The results are compared with the test rig results to validate the numerical model.

Keywords: Dynamic Heat Transfer Coefficient, Two-Phase Flow, Refrigerants.

1. INTRODUCTION

The German automotive industry is trying to achieve full electric mobility in Germany. Therefore, the research focus is on the battery as the new primary energy storage. Shorter range and higher costs of this technology are not the only problem. In addition, the battery is a highly dynamic and demanding consumer from a thermodynamic point of view. The working temperature is limited to the range between 20 and 40 °C. Below 0 °C and above 60 °C the life expectancy and the capacity would suffer. Thus the air conditioning cycle of the automobile needs to be adapted to the new challenge.

Since the industry wants to simulate every state of the dynamic air conditioning system in the car, numerically robust models are required. In general, it can be said that there is a growing need for more dynamic simulations. Therefore more dynamic experiments have to be examined (Wang 2013). Experience shows that in particular low heat loads and mass flow rates close to zero cause problems to the solvability of such dynamic cooling cycles. One aspect of these numerical problems is the dynamic heat transfer coefficient at these marginal loads. Hence the aim of the study at the Institute of Thermo-Fluid Dynamics is to investigate the dynamic behavior of the heat transfer coefficient under these conditions for the refrigerant R-134a.

2. TEST RIG

A test rig at the Institute of Thermo-Fluid Dynamics is used to investigate the dynamical heat transfer behavior of R-134a. It is the most common refrigerant in the automotive industry worldwide.

2.1 Setup

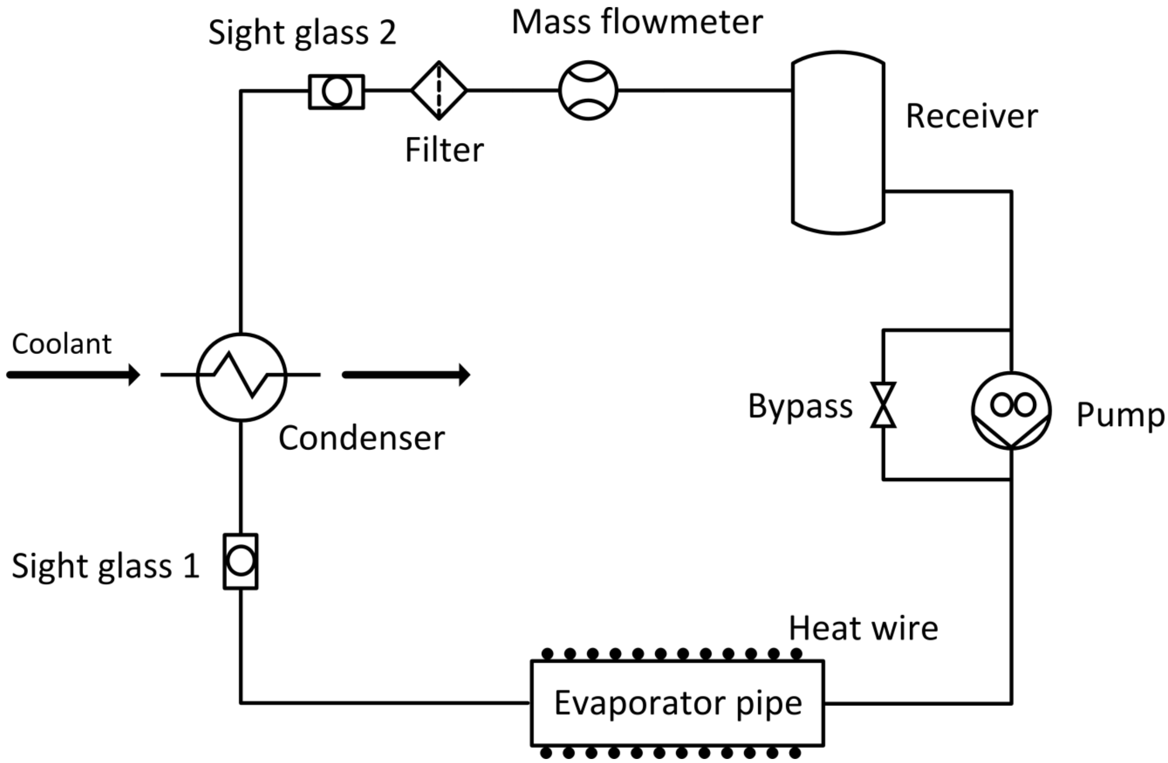


Figure 1: Schematic of test rig cooling cycle

The setup of the test rig is like a cooling cycle. The main parts are a pump, an evaporator pipe, a condenser and a receiver. The schematic is shown in figure 1. The test rig is filled with R-134a in its liquid phase at 20 °C and 6 bar. The gear pump can be used to establish a maximal, steady mass flow of 1 g/s or it can be bypassed to examine natural circulation processes.

The evaporator pipe is on a greater height than the pump. It is made out of copper and the length is 10 cm. The inner diameter is 6 mm and a wall thickness of 1 mm. It is wrapped with heat wire and heat-conductive paste to get a constant and equally heat load on the outside wall of the pipe. Thermocouples are glued between the wires to the pipe wall. There are 11 Thermocouples on the top and 11 on the bottom. They start at 0 cm of the measured section and end at 10 cm in a distance of 1 cm to each other. Thermocouple type T is used to measure the pipe wall temperature. Under the assumption of a small Biot number, the outer wall temperature is similar to the inner wall temperature T_{W}. Inside the evaporator pipe are also two thermocouples, one at the inlet T_{In} and one at the outlet T_{Out}. These two temperatures are used to determine the fluid temperature.

The condenser is located on an even greater height than the evaporator pipe. Thus, when the R-134a evaporates, the vapor can only flow in one direction. The condenser is oversized so the outlet temperature of R-134a can stay constant. The heat sink is cooled down with a Water-Glycol coolant flow. On the way up to the condenser there is a sight glass (sight glass 1) in which the vapor flow can be observed visually.

A second sight glass (sight glass 2) is installed after the condenser to ensure that all refrigerant is in liquid phase again. Because the mass flowmeter works only for completely liquid mass flows. The mass flowmeter is a coriolis flowmeter suitable for very low flow ranges, 0.08-4 g/s with an accuracy of 0.2% (manufacturer's data).

The pressure is measured at the inlet and at the outlet of the evaporator pipe and after the condenser. The material properties of R-134a and Copper can be seen in table 1.

Table 1: Material properties of R-134a and Copper

Property	R-134a (6 bar)	Copper
Temperature T in °C	20	20
Density ρ in $\mathrm{kg/m^3}$	1225,5	8920
Thermal conductivity k in $\mathrm{W/mK}$	0,083	400
Specific heat capacity c in $\mathrm{J/kgK}$	1405	385

The enthalpy of evaporation for R-134a ΔH_V is at about 180 KJ/kg.

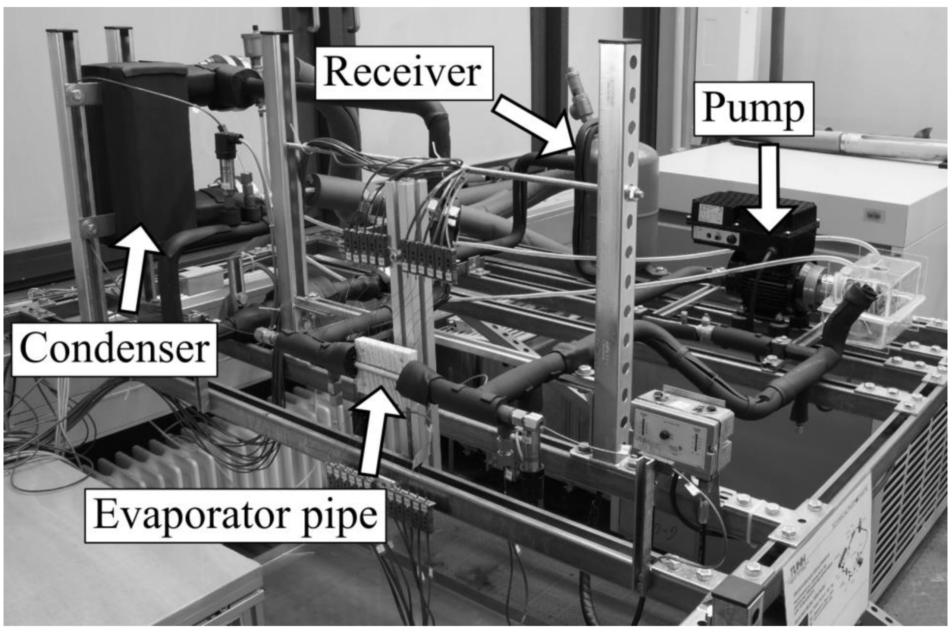


Figure 2: Picture of test rig

Figure 2 shows a picture taken from the test rig. Condenser, evaporator pipe with thermocouples, receiver and pump can be seen. The insulation (black) around the pipes prevents too much influence of the environment.

3. SIMULATION

In addition to the experimental studies a numerical model is built to simulate the dynamic behavior of the test rig. This system simulation is performed with Modelica. Modelica is the most common programming language in the German automotive industry to describe air conditioning cycles in vehicles (Limperich, 2003). With this system simulation tool the dynamic refrigerant flow and the heat transfer are simulated with a one-dimensional model from the Modelica-library AirConditioning®. A precursor version of this library (AClib) was developed by Pfafferott and Schmitz (Pfafferott, 2000). The heat transfer model uses fixed values for specified temperature differences. A reason for these simplifications of the flow and heat transfer model is to save computing capacity and to make the initialization of the system model easier.

But this library is not specialized on simulating small heat loads at marginal mass flows. Particularly the transient heat transfer with evaporating in the test pipe is very complex. Even the simulation of a natural convection process by coupling the Finite Element - software COMSOL Multiphysics® and the equation based modelling language Modelica was not successful. Therefore the Modelica model should be calibrated with measurement data of the experimental test rig.

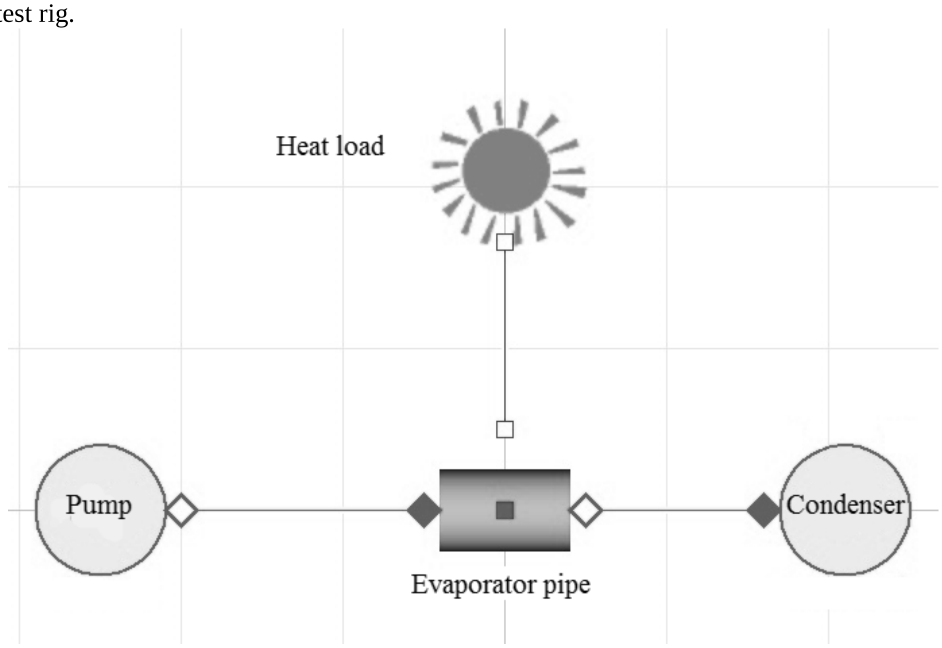


Figure 3: Modelica-model

In figure 3, a graphical representation of the Modelica-model can be seen. The main part is the evaporator pipe, which is connected to a pump model and a condenser model. The heat load is represented by the sun-symbol. The pipe is discretized in 11 finite volumes according to the number of thermocouples. So the model and experiments can be compared more easily.

Boundary conditions to the simulation are the mean mass flow and the system pressure of the test rig. The simulation does not allow zero mass flows, so an average mass flow of the measured value had to be used.

4. RESULTS OF EXPERIMENT AND SIMULATION

To compare the results of the experiments with the simulations the same heat load was used for both as a boundary condition. The time dependent characteristics of the heat load $\dot{Q}$ can be seen in figure 4.

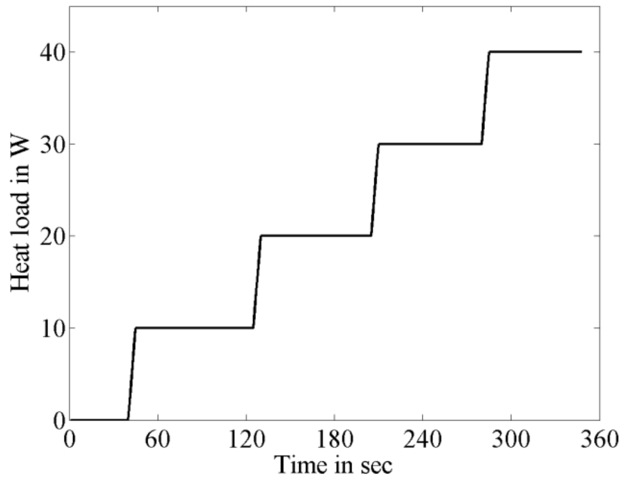


Figure 4: Applied heat load at evaporator pipe over time

For a dynamic investigation of the test rig and the simulations, a step function was used. The increase of the heat load takes 5 s and then the value stays constant for a little more than a minute. The heat load is increased up to the maximum of 40 Watt. Higher heat loads should be tested in the future.

A difference between the simulation and experiments is the mass flow of the refrigerant. The simulation only allows small changes in the altitude. This is not the case in the experiment. When the heating starts the mass flow becomes highly transient.

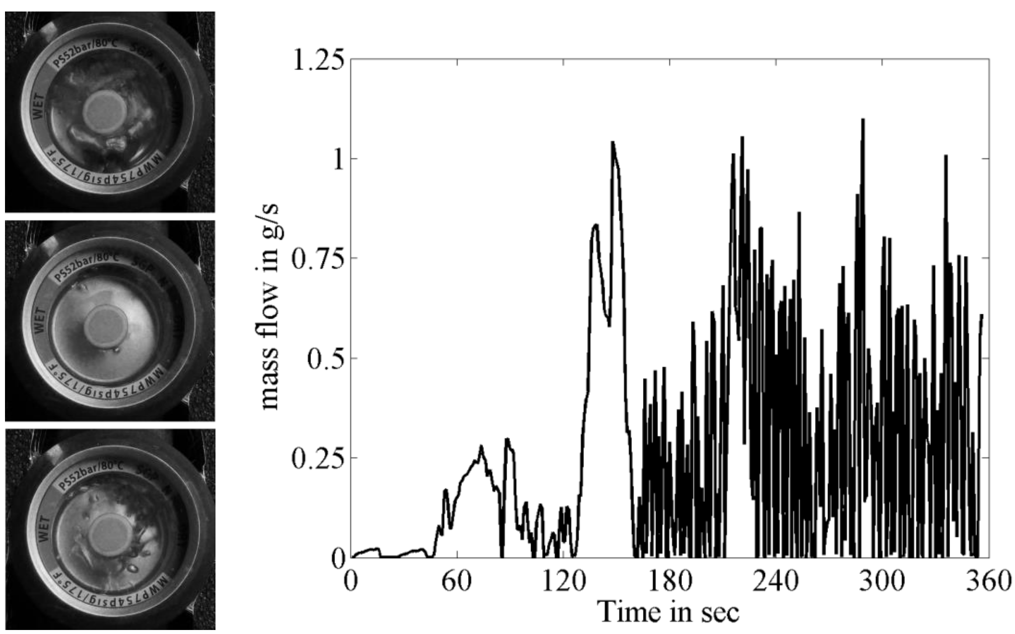
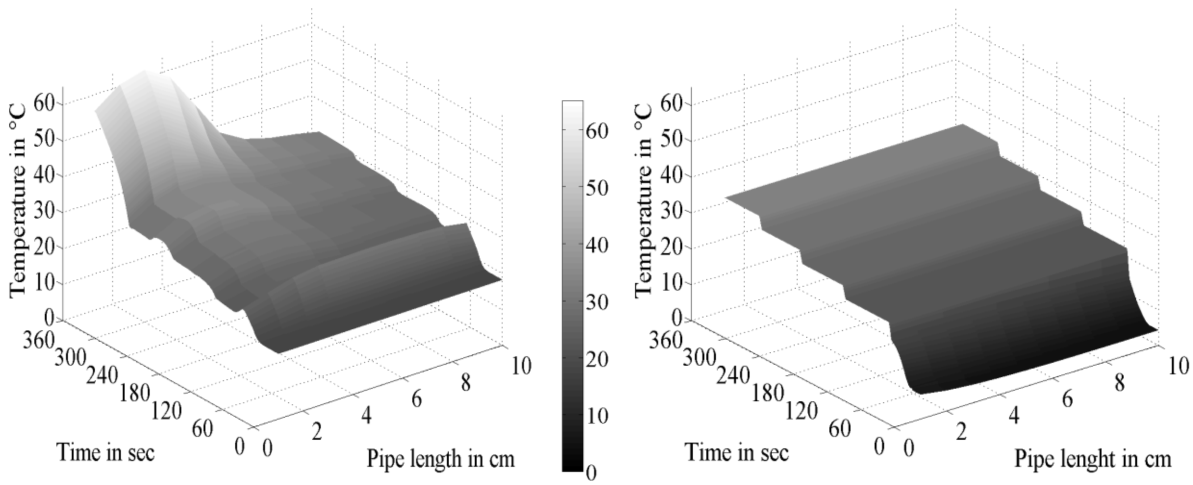


Figure 5: Mass flow of experiment

The right side of figure 5 shows a diagram of the mass flow of the experiment over time. The mass flow starts with the heating of the evaporator pipe. The experiment was carried out without the pump and with an open bypass. After developing a large bubble, the evaporated refrigerant reaches the pipe that leads up to the condenser and some bubbles rise up. This unsteady process can also be observed through the sight glass 1. The pictures on the left side in figure 5 show the moments of the uprising and short stops. The mass flow oscillates with a constant heat flow and an average value of 0.2-0.5 g/s. It reaches maximum values of up to 1 g/s.



Experiment **Simulation**

Figure 6: Temperature of the pipe wall over length and time

Figure 6 shows the results for the 11 temperatures of the pipe wall. Shown on the left side are the results of the experiment and on the right side the results of the simulation. It is the average of the temperature at the top and the temperature at the bottom. The maximum error for this mean value is at 2 K for the experiment. The temperatures at the simulation are the same, due to the one-dimensional modeling.

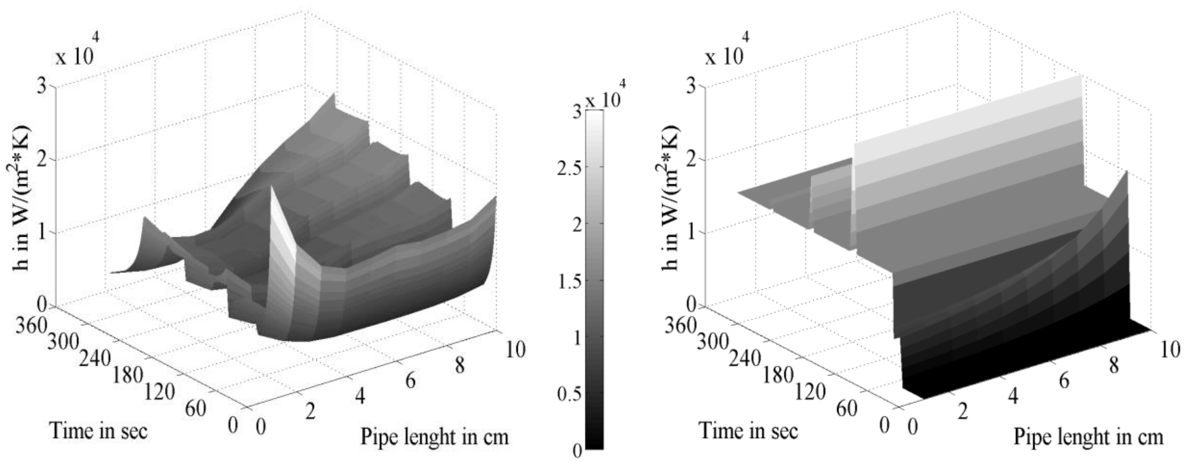
The experimental result on the left side shows a temperature rise over time. The overshoot at the beginning is a result of the appearance of a huge bubble at the beginning of the experiment. The heating is switched on and the refrigerant starts to evaporate until the size is big enough thus it reaches the pipe that leads to the condenser. The unsteady streaming process starts, see also figure 5. After the initial overshoot, the unsteady streaming process ensures a source of liquid refrigerant for the evaporator pipe, so the temperatures stay constant. The temperature corresponds to the level of the heat load. The temperatures are at 20-30 °C. The higher the heat load the higher is the wall temperature. At 40 Watt a strong increase in the wall temperature at the inlet can be observed. The temperature rises up to 63 °C at the first half of the evaporator pipe. This is an indicator for reaching the boiling crisis of the 1st order in the pipe. The refrigerant builds a film of gas in the pipe. That limits the heat transfer strongly and leads to higher wall temperatures. The battery would take damage at this point and would have a reduced lifetime.

The right side of figure 6 shows the results of the simulation for the wall temperatures. The temperature at time=0 is very low, because of initialization problems. The temperature rises according to each step of the heat load and stays on this steady level, because of the continuous mass flow. Their value is also at 20 to 30 °C. There is no model for the boiling crisis of the 1st order implemented thus the temperature rise to 60 °C of the experimental results is unnoticed.

Out of the temperatures and the heat flux, the heat transfer coefficient h can be calculated as:

$$h = \frac{\dot{q}}{T_W - (T_{Out} + T_{In})/2}. \tag{1}$$

The heat load $\dot{Q}$ is divided by the temperature difference of the wall temperature T_W and the average value of the outlet and inlet temperatures T_{Out} and T_{In}. The temperature difference between in- and outlet temperature in the experiment add up to a maximum of 0.5 K. They are the same in the simulation, since there is no superheating.



Experiment **Simulation**

Figure 7: Heat transfer coefficient in the pipe over length and time

Figure 7 shows the heat transfer coefficient h in the pipe over the length of the pipe and over time. On the left hand are again the experimental results and on the right hand are the results of the simulation.

The experimental results show a large overshoot of the heat transfer coefficient at the beginning. It reaches up to 30,000 W/m²K. The reason for this overshoot can be found in the calculation of the heat transfer, as listed in equation (1). The heat load rises very quickly and the temperature difference between the wall and the fluid is very low at the beginning, because the experiment starts from a thermal equilibrium. Thus, the heat transfer is implausible high in the beginning. After the overshoot, it drops to 12,000 W/m²K. With each step the heat load does, the heat transfer coefficient shows a small oscillation. The reason is the quick rise of the head load. The capacities of the heat wire, heat-conductive paste, pipe wall, insolation, air included in the insolation influence the heat transfer. The heat transfer rises until it reaches a maximum 20,000 W/m²K. Only in the first half of the pipe, where the beginning of boiling crisis of the 1st order is located, the heat transfer drops down to 5,000 W/m²K. This is a result of the gas film and the high wall temperatures.

The simulation on the right side of figure 7 shows a much more steady heat transfer characteristic. The heat transfer coefficient is oscillating at the beginning due to the initializing process. After that, it stays constant for the constant heat load. At each step of the heat load, a peak in the heat transfer coefficient can be observed. This is a result of the modeling process, where the rise of the heat load leads to a discontinuity. Except the discontinuities, the heat transfer coefficient stays constant over time and length of the pipe at a value of 15,000 W/m²K.

The dynamic modeling of transient heat transfer behavior is challenging. The constant value of the simulation approximates the measured heat transfer very accurate. A numerical problem appears in the changing of heat loads. A greater problem is that the beginning of the boiling crisis of the 1st order is not registered by the simulation. The temperatures reach a battery-damaging level and the dynamic modeling does not notice.

On the other hand, the experiment has no ideal behavior either. The heat losses to the surrounding and heat capacities of heat-conduction paste or heat wire cause uncertainties in the accuracy of the test rig. In addition, there is high heat conduction in the pipe wall, which heats up the refrigerant before it enters the measurement and evaporating section.

5. OUTLOOK

In the short-term future, experiments and simulations with different heat load profiles like slower raises of the heat load over time are planned. In addition, the maximal heat load is going to be increased.

In the long term, investigations with CO_2 as a refrigerant are planned. In addition, there is interest of the investigation of pipes with smaller diameters. In this case, there might be problems with the measurement instruments, because the influence of thermocouples in the flow will increase. Furthermore, the behavior of a refrigerant, which is flooded in an empty but already heated pipe, should be investigated, to see the effects at switching between different evaporator channels.

In the future different FEM-simulation tools should be used to validate the Modelica models and to get a better understanding of the evaporation process.

6. CONCLUSION

This paper presented the numerical and experimental investigations of the dynamic heat transfer behavior of R-134a in a test pipe. The test rig enables measurements of wall temperatures of a heated copper pipe and by this the determination of the heat transfer coefficient in the pipe. The results are compared with the outcome of a Modelica model of the test rig. The differences and similarities have been discussed. The boiling crisis of the 1st order is unnoticed by the simulation and can lead to false predictions of a cooling cycle.

NOMENCLATURE

c	specific heat capacity	$(\mathrm{J/(kgK)})$
h	heat transfer coefficient	$(\mathrm{W/m^2K})$
k	thermal conductivity	$(\mathrm{W/(mK)})$
$\dot{Q}$	heat load	(W)
$\dot{q}$	heat flux	$(\mathrm{W/\ m^2})$
ρ	density	$(\mathrm{kg/m^3})$
T	temperature	(K, °C)
t	time	(s)
H	enthalpy	(kJ/kg)

Subscripts

In	inlet
Out	outlet
W	wall
V	vapor

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