DETERMINING THE HEAT TRANSFER CHARACTERISTICS OF R125 AT SUPERCRITICAL PRESSURES

Marija Lazova[1]*, Steven Lecompte[1], Michel De Paepe[1]

[1]Ghent University, Department of Flow, Heat and Combustion Mechanics,
Ghent, Belgium
Marija.Lazova@ugent.be

* Corresponding Author

ABSTRACT

Heat exchangers are components of vital importance for the proper operation of the organic Rankine cycles (ORC's). Furthermore, the efficiency of the transcritical ORC's can be improved by acquiring supercritical heat transfer in the heat exchanger at the hot side. However, heat transfer correlations for designing heat exchangers suitable to operate under these conditions are lacking.

Therefore, heat transfer experiments at supercritical state of the working fluid R125 were conducted on the novel test set-up. The test section is horizontally positioned and is a tube-in-tube heat exchanger with an outer tube diameter of 28.6mm. Experiments were conducted at various mass fluxes and the pressure was in the range of (1.05-1.15)p_{cr} of the working fluid.

The experimental results show the effect of the inlet pressure to the heat transfer coefficient and the local maximum of the pseudocritical temperature is determined experimentaly under certain conditions.

1. INTRODUCTION

The total primary energy use has increased significantly in the last decade. In 2014 the industry accounted for about 42.5% of the total primary energy use [1]. According to the statistical investigations up to 50% of the energy used in the industry sector is accounted as a waste heat [2]. Hence, this energy in the form of low-grade waste heat has a great potential to be utilized and recovered. The benefit is multiple, and leads to considerable reducing the energy demand in the industry sector. Additionally, if the waste heat is not discarded to the atmosphere the negative impact to the environment is mitigated.

The organic Rankine cycle (ORC) is a well-developed technology that can exploit the low-grade temperature heat for electricity generation. There is a lot of interest to improve the efficiency of the ORC. One possible way is by providing supercritical heat transfer to the working fluid in the heat exchanger at the hot side. The benefit lies in the better thermal match between the temperature glides of the heating fluid and the working fluid. Hence, the component is so-called a vapour generator (instead of an evaporator) and the cycle the transcritical ORC. The advantage of using transcritical cycles was investigated by Wang et al., 2010 [3], and Schuster et al., 2010 [4]. An improvement of 8% compared to a subcritical ORC was reported by Schuster et al., 2010 [4]. Additional requisite for efficient ORC is the proper selection of a working fluid. However, there is no unique solution and the choice depends on the temperature profile of the heat source Liu et al., 2004 [5]. Furthermore, favorable working fluids are the environmentally friendly fluids with low GWP (global warming potential) and ODP (ozone depletion potential).

R125 is considered as potential working fluid for transcritical ORC's because it has beneficial thermophysical properties and low critical pressure (36.2bar) and critical temperature (66.02°C). In many comparative (numerical) studies for low-grade heat sources Gu and Sato, 2002 [6], Shengjun et al., 2011 [7], Baik et al., 2011 [8], R125 is considered as favorable among the other tested working fluids (CO2, propane, R134a, R123, R41, etc.). It was considered as a cost effective solution for heat

source temperature of 100°C, with higher net power output due to the lower pumping power. Regarding the environmental aspects, R125 has an ODP of 0 but the GWP of 3500 which makes room for a suitable substitute in the near future.
Fig. 1 depicts a simple layout of a transcritical ORC cycle and Fig. 2 shows a T-s diagram of the respected cycle.

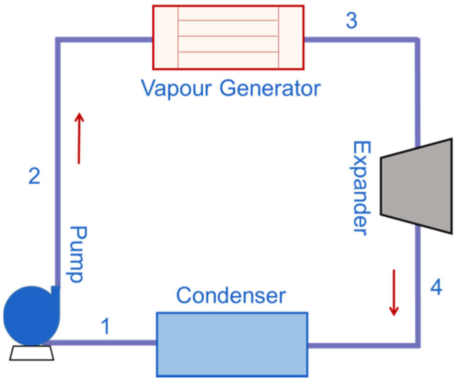

Figure 1: Basic layout of a transcritical ORC

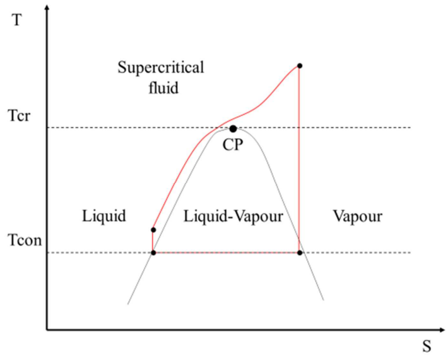

Figure 2: T-s diagram of the transcritical ORC

Moreover, an experimental investigation related to heat transfer and pressure drop characteristics of the organic fluids (refrigerants[1]) which are potential working media for transcritical ORC is scarce. However, in the literature there is valuable data linked to supercritical heat transfer for the fluids water, CO_2 and helium. These fluids were tested for applications (nuclear reactors, steam cycles, refrigeration) different than ORC. Moreover, the results from the previous work do not show a good agreement with the organic fluids because they are characterized with unique heat transfer and pressure drop features. Additional difference is the layout of the test sections, made of small tube diameters tested mainly in vertical position. All these differences limit their use in practical application.
Therefore, determining the heat transfer phenomena of organic fluids at supercritical state in a horizontal layout and large tube diameter applicable for ORC conditions is significant. An experimental investigation of R125 at supercritical pressure in a test section of 4m length, large tube diameter (d_o= 28.6mm; d_i=24.78mm) positioned horizontally is presented in this article.

2. HEAT TRANSFER AT SUPERCRITICAL STATE

A fluid at supercritical state is above its critical pressure and temperature. Near the pseudo-critical temperature T_{pc} there is a rapid variation of the thermophysical (density, viscosity, thermal conductivity, specific heat capacity) properties. This is the area where the specific heat capacity reaches a peak which yields to a maximum local heat transfer coefficient in the vicinity of that region. Additionally, the density, viscosity and the thermal conductivity vary significantly within a small temperature range near the T_{pc}. All these rapid changes have a significant effect on the heat transfer mechanisms. Fig. 3 shows the variations of these properties as a function of the temperature for R125. Furthermore, the heat flux, the mass flux, the tube diameter, the flow direction and the buoyancy have an influence on the heat transfer to a fluid at supercritical state.
Research activities about heat transfer at supercritical state started in the early 50s in the 20th century with the work of Schmidt et. al [9]. As already mentioned, mostly water, CO_2 and helium were tested under supercritical conditions and in vertical flow. One of the first studies regarding heat transfer to water at supercritical state in horizontal flow was performed by Vikrev and Lokshin [10]. The buoyancy effect has a significant influence on the supercritical heat transfer in horizontal flow because there is a temperature difference at the top and bottom of the tube. This temperature variation leads to a reduction in the heat transfer coefficient.

[1] In this article all working fluids used in ORC's are regarded as organic fluids.

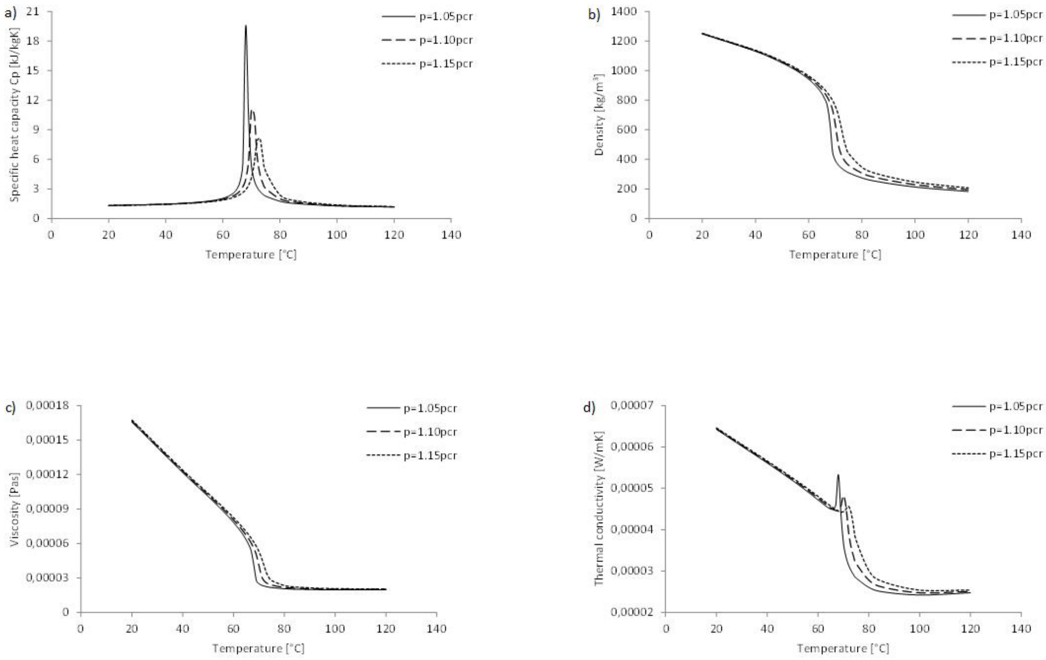

Figure 3: Property changes of R125

In the work of Miropolsky and Pikus [11] it was reported that there is a reduction in the heat transfer coefficient with a factor of 4 between the upper and the lower surface. Measurements in horizontal flow in a large tube diameter of 22.14mm and wide range of mass and heat fluxes was done by Adebiyi and Hall [12]. In their study buoyancy free and buoyancy dependent cases were distinguished. Jackson and Hall [12] highlighted that the buoyancy effect are present in horizontal flow due to the flow stratification of the working fluid. The hotter and less dense fluid is close to the top of the tube and due to dumping effect of the stabilizing density gradient turbulence might occur near the upper surface. Common for all these studies is the possibility of heat transfer deterioration appearing at the top surface and heat transfer enhancement at the bottom surface. This heat transfer phenomena occurs at high heat fluxes and a detailed investigation is required. However, this is not the scope of this paper.

Recently, several organic fluids (R134a, R22, R404A, R410A) have been tested at supercritical pressures in either cooling or heating application. Zhao and Jiang [13], tested the fluid R134a at supercritical pressures (1.10-1.35) p_{cr} in small tube diameters. The results from the measurements show that the mass flux and the pressure of the working fluid, as well as the heat flux have a strong influence on the heat transfer coefficients. Tian et al. (Tian *et al.*, 2019) experimentally investigated the heat transfer characteristics of R134a in a horizontally positioned tube with a diameter of 10.3mm. Furthermore, a heat transfer correlations was also proposed. In 2012, Jiang et. al. [14] conducted test by using R22 and ethanol. The conclusion was that the flow acceleration and the buoyancy have an influence on the heat transfer coefficients. Regarding the frictional pressure drop, it increases significantly with the heat flux compared to ethanol. The local heat transfer coefficients increase with the enthalpy rise for both fluids.

3. EXPERIMENTAL INVESTIGATION

3.1 Experimental Facility

In Fig. 4 the basic layout of the experimental facility iSCORe is presented. It consists of the heating, the cooling and the experimental loop. A thermal heater with a power of 20kW is used to provide heating fluid (low synthetic thermal oil) to the experimental loop. A compact independent thermal unit electrically heats and provides the heating fluid (Terminol ADX10) to the test section. The cooling loop consists of a 37kW chiller and is used to control the temperature of the cold source. The cooling fluid is a mixture of water/glycol (70%/30% by volume) that is provided to the condenser to sub-cool the R125.

The experimental loop represents a basic ORC cycle, consisting of a pump, a condenser, a heat exchanger (the test section) and an expansion valve (instead of an expander). In this loop there are additionally two tube-in-tube preheaters and one in-line electrical preheater with a capacity of 10kW.

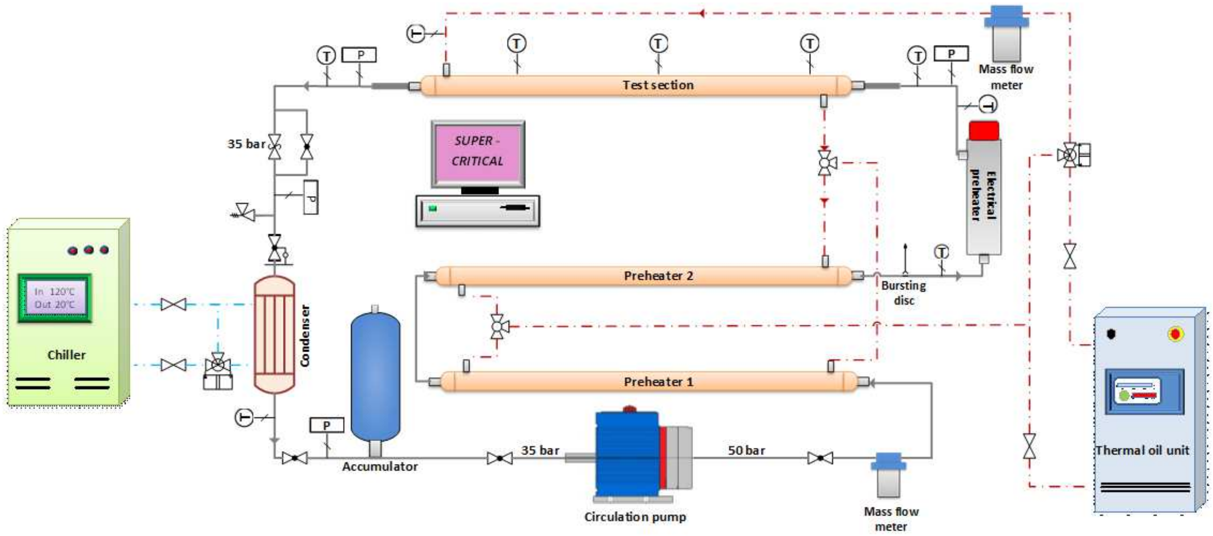

Figure 4: Experimental facility 'iSCORe'

3.1.1 Test Section

The test section is constructed as a horizontal counter-flow, tube-in-tube heat exchanger. In the annulus circulates the heating fluid and provides the heat flux to the working fluid R125 that runs in the inner tube. Fig. 5 depicts the layout of the test section.

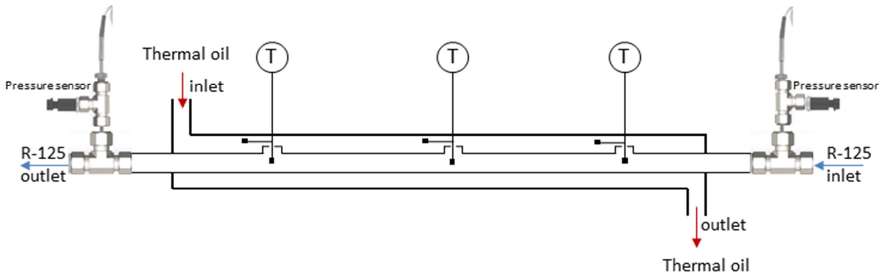

Figure 5: Layout of the test section

The inner tube is made out of copper alloy tube with the following dimensions: d_o= 28.6mm; d_i=24.78mm and heating length of L=4m (L/Di=162). Moreover, for fully-developed flow, the test section is connected with unheated adiabatic test section with a length of 1m. Measurements in the test section are performed with a large number of sensors. Four thermocouples Pt100 are mounted at the inlet/outlet of the test section to measure the temperatures of both fluids. There are 11 T-type thermocouples positioned on equal distance of 0.0.33m measuring the bulk temperature of the working fluid. Additionally, 3 K-type thermocouples placed along the test section on a distance of 1m measure the bulk temperature of the thermal oil. To reduce the heat losses to the environment the tubing of the system is insulated by an insulation material with thickness of 0.055mm and thermal conductivity of 0.04 W/mK.

3.2 Data Reduction and Error Analysis

The heat transfer coefficients htc's of the 11 subsections are determined with Eq. (1):

$$htc = \frac{q}{T_{w,i} - T_b} \tag{1}$$

where T_b is the measured bulk temperature of R125, and $T_{w,i}$ is the inner wall temperature of the copper tube calculated with Eq. (2):

$$T_{w,i} = T_{w,o} - \frac{q}{2k_{cu}L} ln\left(\frac{d_o}{d_i}\right) \tag{2}$$

d_i and d_o are the inner and outer tube diameters, L is the length of the test section and $T_{w,o}$ the outer wall temperature. The outer wall temperature of the copper tube was calculated from the thermal resistance depicted in Eq. (3):

$$R_{tot} = \frac{1}{h_{R125}\pi d_i L} + \frac{ln\left(\frac{d_o}{d_i}\right)}{2k_{cu}L} + \frac{1}{h_{oil}\pi d_o L} + \frac{1}{h_{oil}\pi D_i L} + \frac{ln\left(\frac{D_o}{D_i}\right)}{2k_{steel}L} + \frac{1}{h_{air}\pi D_o L} \tag{3}$$

Taking into account the enthalpy of the oil at the outer diameter of the copper tube (R_3, the third section) the $T_{w,o}$ can be determined and is presented with Eq. (4):

$$Q = \frac{1}{R_3}(T_{w,o} - T_{hf,b}) \tag{4}$$

where $T_{hf,b}$ is the bulk temperature of the heating fluid.
The heat flux q was determined from the enthalpy change of the heating fluid (thermal oil) along the test section and is presented with Eq. (5):

$$q = \frac{\dot{m}_{hf} c_p (T_{hf,in} - T_{hf,out})}{\pi d_o L} \tag{5}$$

By comparing the enthalpy changes of the heating fluid and working fluid Eq. (5) the energy balance was determined. The deviation is less than 3%. However, near the pseudo-critical region the error is larger and it can reach even 15%.

$$\dot{Q}_{wf} = \dot{m}_{wf}(h_{wf,out} - h_{wf,in}) \tag{6}$$

3.2.1 Error Analysis

Pressure, temperature measurements were performed with a large number of sensors. The parameters of the heating fluid for this set of measurements were kept constant. The pressure was in the range of 1.5bar, the temperature 125°C and the mass flow rate was 2kg/s. However, the pressure of the working fluid was above the critical point of R125 and was in the range (1.05-1.15)p_{cr}. The inlet temperature and the mass flow rate of the working fluid were kept stable for this set of measurements and were in the range of 60°C and 0.26kg/s.
The pressure sensors have an absolute range of 0-60bar and the accuracy is 0.1% FS BFT (Full Scale Best Fitting Line). The pressure drop during the measurements was very small and it was in the range of 5-10kPa with a relative uncertainty of 1.5%. All temperature sensors were recalibrated in-house and Pt100 and T-type have an accuracy of 0.07°C while the K-type thermocouples have an accuracy of 0.2°C. In Table 1 an overview of the uncertainties is presented. As already mentioned the error of the energy balance was 3% far from the near-critical region. The mean uncertainty of the heat transfer coefficients is 11.9%. However, it has to be mentioned that the thermo-physical properties of R125 have uncertainties. The uncertainty of the density near the critical region is 0.5% and the isobaric specific heat capacity has an uncertainty of 1% in the critical region (Bell 2014). Heat loss to the environment was less than 3%.

Table 1: Page margins for manuscripts

Parameter	Range	Relative error %
Heat input	5-10 kW	2.62
Pressure	38-55 bar	1.5
Temperature	60-100 °C	3.71
Mass flow rate	0.26 kg/s	2

4. RESULTS AND DISCUSSION

4.1 Influence of the inlet pressure to the local heat transfer coefficients

Each set of measurements was done at steady-state conditions by assuring that the measured values (temperature, pressure, mass flow rate) varied ±1% during the process. The inlet temperature of R125 was 60°C, and the inlet temperature of the heating fluid was 125°C. Furthermore, the mass flow rate of the heating and working fluid were stable in the range of 2kg/s and 0.26kg/s respectively.

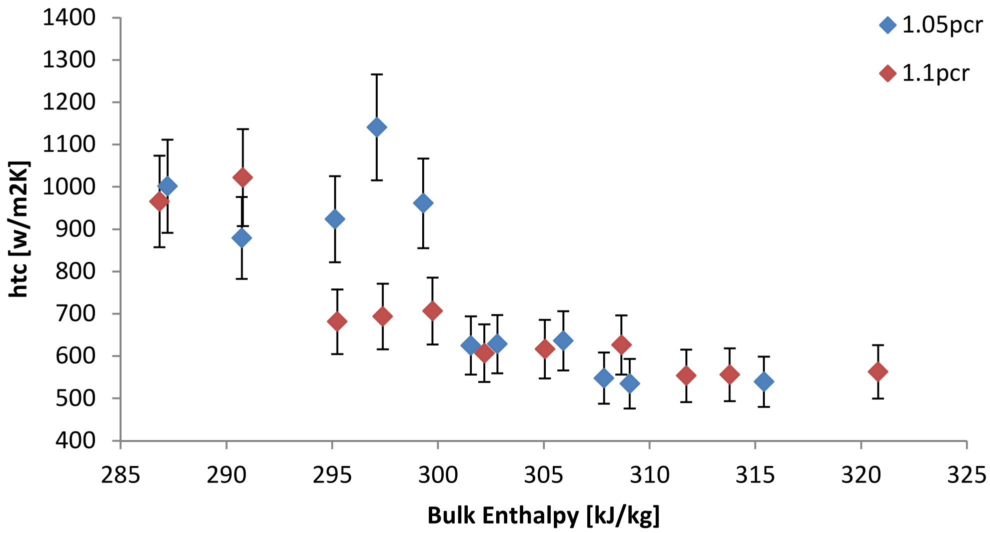

Figure 6: Variation of the local heat transfer coefficients

In Figure 6 the variation of the local heat transfer coefficients at different inlet pressure of R125 is presented. The inlet pressure was in the range of (1.05-1.10) p_{cr}. The heat transfer coefficients along the test section gradually rise up to a local peak. At these test conditions higher values for the local htc's are present at the inlet pressure of 38 bar or 5% higher than the critical pressure of R125, where the local enthalpy was 300kJ/kg. In comparison with the results obtained at 40bar ($p=1.1p_{cr}$), the local peak of the htc is lower. The inlet pressure has a significant effect on the local htc's. The experimental results show that the changes of the thermophysical properties of R125 have a significant effect on the heat transfer coefficients. Near the critical region, the specific heat capacity rises abruptly and the htc's increase greatly in this range.

4.2 Comparison of the pseudo-critical temperature at different inlet pressures

Determining the pseudo-critical temperature is important, since this is the region where the specific heat capacity has its maximum value. The set of measurements presented in this analyse is the same like in the previous subchapter. At different inlet pressure the specific heat has its local maximum. Hence this has an influence on the htc's, and it is expected to be the highest. Far from the critical pressure of R125 the specific heat capacity follows the trend of having a peak value but is signififcantly lower, when compared to lower pressures. This can be seen on Figutre 7 and a representative case is the pressure of $p=1.15p_{cr}$. From the experiments, the pseudo-critical temperature has a maximum value at inlet pressure of $p=1.05p_{cr}$, where it reaches 19kJ/kgK at a temperature of 68°C.

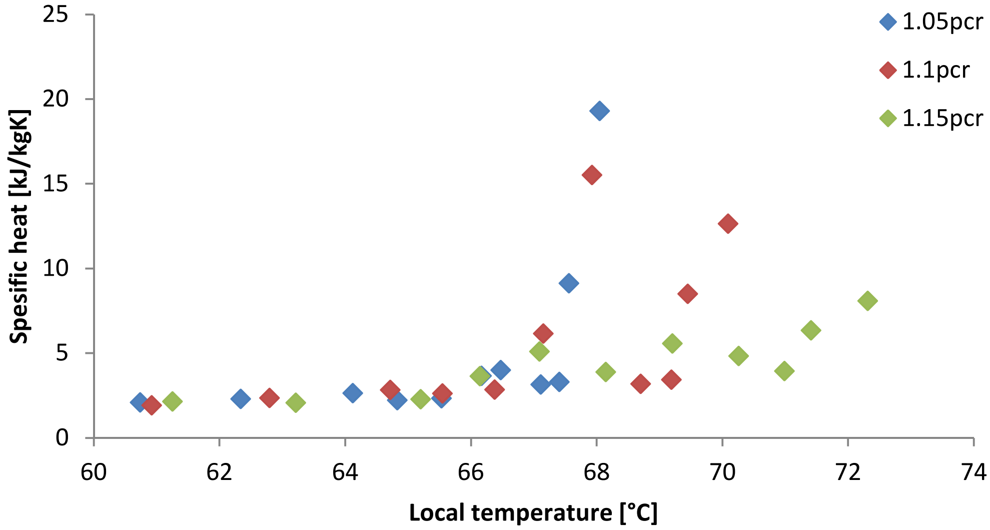

Figure 7: Comparison of the pseudo-critical temperature at different inlet pressures

5. CONCLUSIONS

In this article, results from the forced convection heat transfer measurements at supercritical state, done on the new test set-up are reported. Utilization of a heating fluid temperature of 125°C and ensuring supercritical heat transfer in the test section was achieved. At supercritical pressure and bulk fluid temperatures close to the pseudo-critical temperature heat transfer coefficient has a peak value. Determining the pseudo-critical temperature has a strong impact on the heat transfer rate because this is the region where the specific heat capacity has its maximum value. A next step of this research is to obtain wide data from local temperature measurements and to determine the local heat transfer coefficients. Furthermore, using the results from the measurements a new heat transfer correlation for designing supercritical heat exchangers suitable to operate under (transcritical) organic Rankine cycle (ORC) conditions is going to be derived.

NOMENCLATURE

A	heat transfer area	(m2)	p	pressure	(bar)
cp	specific heat capacity	(J/kgK)	ṁ	mass flow rate	(kg/s)
d	diameter	(m)	T	temperature	(°C)
h	enthalpy	(kJ/kg)	R	resistance	(W/K)
htc	heat transfer coefficient	(W/m2K)	Q	heat transfer rate	(kW)
λ	thermal conductivity	(W/mK)	q	heat flux	(kW/m2)

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