Experimental investigation at sub- and supercritical conditions of a helical coil heat exchanger particularly designed for a small-scale solar ORC

M Lazova[1], A Kaya[1], H Huisseune[1] and M De Paepe[1]

[1] Ghent University –UGent
Sint-Pietersnieuwsstraat 41, 9000 Ghent, Belgium

E-mail: marija.lazova@ugent.be

Abstract. A detailed experimental investigation on the heat transfer, effects of the system pressure and thermal match of a helical coil heat exchanger installed in a solar Organic Rankine Cycle was carried out. The heat exchanger was specially designed for supercritical working conditions in a small-scale installation with a net cycle capacity of 3 kW. There are many parameters such as the heat source temperature, mass flow rate and operational pressure that influence on the performance of the heat exchanger. The tests are performed at sub- and supercritical operating conditions, for low-grade temperature heat source of 95 °C. From the experimental evaluation of the heat exchanger it was found that a better performance is achieved when operating at supercritical conditions.

1. Introduction

The wide usage of the conventional energy resources (fossil fuels) through the decades has led to their depletion on one hand and environmental problems on the other. This situation necessitates using renewable energy sources and developing new technologies for electricity production. The Organic Rankine Cycle is one of the technologies that has been studied intensively by many researchers in the past years. This technology is a promising process for conversion low and medium heat source temperature into electricity. As low grade heat is considered a heat source with a temperature below 400 °C due to it can't be efficiently utilized by conventional thermal processes. ORC can exploit the low grade heat from several renewable energy sources such as biomass, geothermal, solar, waste heat from various processes etc.

Solar (power) is a continual and important source of renewable energy. There are many technologies that have been developed for deployment of the solar energy such as photovoltaic systems concentrating solar power, solar receivers etc. The produced heat from the solar thermal collectors can be transferred to a power cycle such ORCs for electricity generation. Many state-of-the art small-scale ORCs applications that can be combined with solar desalination reverse osmosis system, concentrated photovoltaics/thermal collectors or solar receiver have been investigated. Small-scale ORC is a promising technology for decreasing the investment cost due to the installed power can be reduced to kW scale and the operation is at relatively low temperature. The first ORC prototype for solar application appeared in early 70s when many experimental and theoretical investigations about different working (organic) fluids and configurations took place [1].

In the literature there is lack of experimental data for (solar) ORCs operating at supercritical conditions. However, many of the work published elaborates the benefits of operating with supercritical CO_2 in solar power cycles. Haskins *et al.* (1981) [2] were among the first that had research activities of supercritical ORC application. More particularly a development of a solar receiver coupled to an ORC engine that uses toluene as working fluid was investigated.

The performance objective of the solar receiver design was to maximize the thermal efficiency and heat capacity of the core. A solar receiver concept is based on waste heat utilization of directly-heated, mono-tube normally operating at supercritical pressure. In 2010 Schuster *et al.* [3] performed simulation of the ORCs and optimization potential of the process when using supercritical parameters of various working (organic) fluids for different applications and heat sources. Experimental investigation of transcritical solar ORCs was restarted 10 years ago.

Analyze about a combined concentrating solar power system and a geothermal binary plant based on a supercritical heat transfer in an organic Rankine cycle (ORC) was performed in 2011. Astolfi *et al.* [4] designed the optimal utilization of an intermediate enthalpy geothermal source. Moreover, in the plant a solar parabolic trough field was included, in order to increase the power production. The performance analyze of the power cycle have been performed by carrying out the estimation of the yearly power output by using a detailed solar field model. Finally, a differential economic analyze have been conducted in order to determine the cost of the electricity generated by the solar source.

In 2012 a new combined power and desalination system was investigated. Li *et al.* [5] proposed a system that can utilize low grade heat source from solar energy, geothermal or waste heat. The system itself is a combination of transcritical (supercritical) Rankine cycle, an ejector and a multi-effect distillation (MED) system which could be used for sea water or concentrated brine. In order to quantify the performance of the combined power and water system a parametric sensitivity of the model was carried out. This combined system showed good results for desalination with no additional energy input except heat supply to the power cycle.

Research activities regarding desalination process by means of transcritical (supercritical) Rankine cycle driven sea water reverse osmosis system continued in 2013 by Li *et al.* [6]. In this work a comparison between SCORC-RO and ORC-RO using two types of low grade heat sources with a maximum temperature of 150°C was performed. The obtained results show that SCORC-RO system provides stable performance while using different heat sources. Moreover, a comprehensive list of suitable working fluids for SCORC-RO is proposed. A co-generation system that produces electricity and fresh water by a solar driven transcritical (supercritical) ORC coupled with a desalination unit was examined by Li et al. [7]. The system was coupled with parabolic trough solar collectors that could produce 700kW thermal energy with temperature of 400 °C at peak condition. Cycle efficiency close to 21% could be achieved thanks to the use of hexamethyldisiloxane as a working fluid in the transcritical ORC. Based on variable incident solar radiation, the proposed system can generate electricity only or water-electricity co-generation. This system could decrease the negative influence of intermittent solar energy without thermal energy storage by converting solar energy to desalinated water and is ideal for small/medium applications.

Combined system of concentrated PV/thermal coupled with ORC engine that is designed to operate at supercritical conditions is investigated experimentally. In this work the performance of all components is evaluated while the main focus was on assessing the performance of the scroll expander Kosmadakis *et al.* [8] and the supercritical heat exchanger Lazova *et al.* [9]. The net power output of this system was 3 kW. In this article an experimental study of this heat exchanger designed for supercritical operation in ORC is presented. Further, the focus of this study is to examine and compare the performance of the heat exchanger operating at supercritical and subcritical condition in solar ORCs.

Today, the number of commercial transcritical ORC power plants worldwide is 3 with 28 MWe power and are installed for solar, geothermal and waste heat applications. The first solar transcritical ORC pilot plant for power and industrial heat with net power output of 150 kW was built in 2014 in Wallsend, NSW, Australia [10]. However, there is an interest for new transcritical (supercritical) ORCs power plants worldwide.Transcritical Organic Rankine Cycle

2. Sub- and Supercritical heat transfer in the heat exchanger in the solar Organic Rankine Cycle

In the recent years a lot of attention has been paid on improving the efficiency of ORCs. However, there are many parameters that influence on the cycle efficiency such as a proper selection of the working fluids, adequate selection/design of the components, the operating conditions etc. Further, according to the theoretical studies [11] promising results are obtained by ensuring supercritical heat transfer in the heat exchanger (vapour generator). Namely, a better thermal match between the heat source and organic fluid temperature glide yields to improved heat transfer in the heat exchanger. The difference between the sub- and transcritical cycle lays in the heat addition process in the heat exchanger (evaporator, vapour generator, boiler). A T-s diagram representing the sub- and transcritical cycle is depicted in Figure 1.

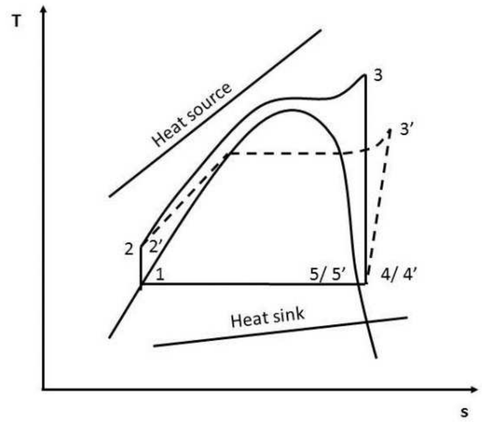

Figure 1. T-s diagram of sub- and transcritical ORC.

Due to the organic fluids have lower critical temperature and pressure, compared to water that is used in classical Rankine cycle thermal process, they can be pressurized above its critical pressure and heated above its critical temperature while avoiding the two phase region. By bypassing the isothermal boiling process, a better thermal match is found between the temperature curve of the heat source and the working fluid, reducing entropy generation and thus raising cycle efficiency.

2.1. A helical coil heat exchanger

A helical coil heat exchanger, presented in Figure 2 was particularly designed and built for the new CPV/T-Rankine test set-up. Details about the design procedure of the heat exchanger can be found in Lazova *et al.* [12]. This heat exchanger integrates the ORC engine with the concentrating PV/thermal collectors in one system (Figure 3). Further, the decision to work with a helical coil heat exchanger is due to several advantages such as compact design, easy integration in the system, enhanced heat transfer, cost-effectiveness etc. This component was designed with a capacity of 41 kWth and can operate properly at relatively high pressure and temperature. Several set of measurements were conducted at sub- and supercritical working condition.

The shell side of the heat exchanger is fabricated out by two concentric cylinders in which a metal coil tube with length of 66 m and coil diameter of 0.6 m is fitted. Moreover, the heating fluid - water is flowing downwards in the shell and the working fluid - R404a circulates in upward direction in the coil resulting in a counter flow heat exchanger.

Figure 2. Helical coil heat exchanger installed in the laboratory in an ORC installation.

The heat transfer of both fluids takes place across the coil wall with a total heat exchange area of ~7m^2. Furthermore, the heat exchanger is well insulated in order to reduce the heat loss to the environment.

2.2. Description of the transcritical CPV/T- Rankine set-up

Small-scale solar ORC with a net capacity of 3 kW was built in Athens, Greece [8], [9]. The new CPV/T-Rankine set-up integrates two technologies such as the concentrated photovoltaic/thermal collectors and transcritical Organic Rankine cycle in one system. In Figure 3 a basic layout of a CPV/T-Rankine set-up is presented.

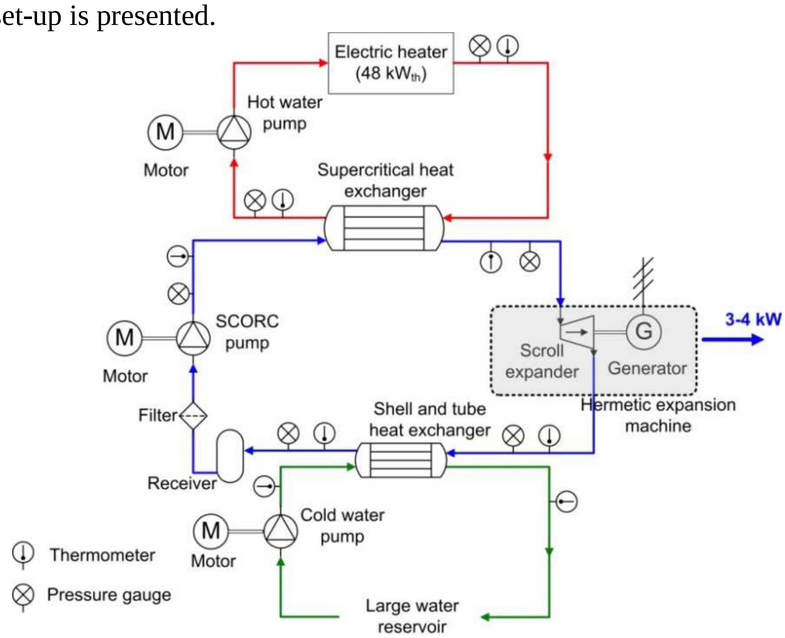

Figure 3. Layout of the experimental CPV/T- Rankine set-up.

On the layout, with the red line the heating loop is denoted and represents the solar PV/thermal collectors that simultaneously generate heat and electricity. The excess thermal heat from the collectors is utilized by the helical coil heat exchanger that couples the CPV/T field and the supercritical ORC engine in one system. The transcritical ORC loop is denoted with the blue line and consists of a pump, condenser, expander and evaporator/vapour generator. The cooling of the system is accomplished by an air driven condenser that is part of the cooling loop and denoted with green line on the layout.

2.3. Test procedure and uncertainty analysis

The first tests were conducted in the laboratory where an electrical heater with capacity of 48 kW_{th} was used instead of the solar collectors. All the measurements reported in this paper were performed at sub- and supercritical operating conditions and inlet temperature of the heating fluid of 95 °C. During the measurements the pressure of the heating fluid was constant at 3 bar and the pressure of the working fluid R404a was varied between 18 bar to 41 bar. The mass flow rate of the heating fluid was kept stable at 2.7 kg/s while the mass flow rate of the organic fluid varied between 0.10 kg/s to 0.30 kg/s. All set of measurements were conducted at steady state while keeping the temperature, the pressure and the mass flow rate at the inlet of the heat exchanger at hot and cold side stable.

Once all the measurements were done and the values of the pressure and temperature were recorded the heat transferred to the organic fluid was calculated from the enthalpy changes at the inlet and at the outlet of the heat exchanger with the following equation (1):

$$\dot{Q} = \dot{m}_{wf}(h_{wf\_out} - h_{wf\_in}) \tag{1}$$

To calculate the overall heat transfer coefficient, equation (2) was used:

$$U = \frac{\dot{Q}}{A.\Delta T_{log}} \tag{2}$$

where Q is the heat transferred, U is the overall heat transfer coefficient, A is the total heat transfer area, $\dot{m}_{wf}$ is the mass flow rate of the organic fluid, h_{wf} is the enthalpy of the working fluid, ΔT_{log} is the logarithmic temperature difference or LMTD.

2.3.1. Uncertainty analysis. Temperature and pressure measurements were performed with temperature sensors Pt100 and pressure transducers type 21Y (manufactured by Keller) placed at the inlet and at the outlet of the hot and cold side respectively. All sensors have high accuracy ($\pm$0.2 % temperature error - 1 % full scale pressure error) and their positioning in the system is indicated in Figure 3. Mass flow meters are not used because all the measurements were performed at steady-state conditions. The parameters at the hot side are calculated from the conditions from the ORC side, because a positive displacement pump that has a linear correlation of the flow rate with the speed and has a constant coefficient of 0.0205 l/min/RPM that provides reliable calculation of the volumetric flow rate is used. An estimated accuracy of this method is 2%. From the measured temperature and pressure of the fluid at the pump outlet using EES/Refprop database for R-404a the mass flow rate can be calculated [8]. The control of the rotation of the pumps was done with a frequency inverter. All thermodynamic properties were calculated with an accuracy of about 1.2%.

In Table 1 the mean values of the relative measurement error for each parameter are included. The thermal efficiency has the highest error, since many parameters are included in its calculation formula. However, this error is still low and does not influence the relative differences of the results.

However, it is important to be noted that local temperature measurements on the coil and hence determining the local heat transfer coefficients is not possible. The performance evaluation of the heat exchanger is rather done as a black box modelling, taking into account the temperature and pressure measurements at the inlet and at the outlet of the heat exchanger.

Table 1. Accuracy of calculated parameters

Parameter	Range	Relative error (%)
Heat input to ORC	12 - 48 kWth	2.62
Pressure ratio	1.7 – 2.6	1.4
Thermal efficiency	0 – 4.2 %	3.71
Volume flow rate	1 – 30 l/min	2
Expander power production	0.5 – 3 kWe	2.62
Expansion efficiency	20 – 85 %	2.66

3. Performance evaluation of the heat exchanger at sub- and supercritical operating conditions

3.1. Heat transfer at variable mass flow rate of the organic fluid at constant heat source temperature of 95 °C

Evaluation in terms of the heat transfer of the heat exchanger at sub- and supercritical operational condition was performed at constant inlet properties of the heating fluid such as temperature at 95 °C, pressure at 3 bar and mass flow rate at 2.7 kg/s. Several set of measurements were performed, where the mass flow rate and the inlet pressure of the organic fluid R404a were varied in the range between 0.10 kg/s to 0.30 kg/s and 18 bar to 41 bar respectively. The inlet temperature at the cold side is dependent from the mass flow rate of the organic fluid.

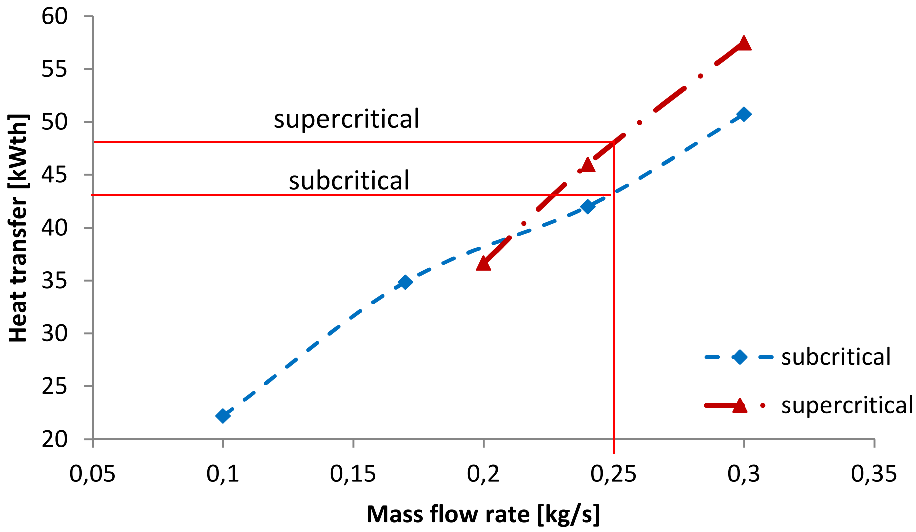

Figure 4. Heat transfer on organic fluid side at different pressure and mass flow rates at 95 °C.

A comparison of the nominal designed values of the heat exchanger such as the heat transfer of 41 kW, heating fluid inlet temperature of 95 °C and a mass flow rate of the organic fluid of 0.25 kg/s with the measurements at sub- and supercritical state was performed. At these inlet conditions at hot and cold side of the heat exchanger a heat transfer of 43 kWth at subcritical and 48 kWth at supercritical state was achieved. This yields a better performance in terms of heat transfer of ~15 % in the evaporator. Figure 4 presents the heat transfer at sub- and supercritical state of the organic fluid. It can be concluded that the correlations used for designing the heat exchanger under-predict the heat transfer and new correlations for heat transfer at sub- and supercritical state in ORC's conditions need to be developed.

3.2. Effect of the system pressure on the heat transfer

Investigation of the system pressure on the heat transfer of the organic fluid that circulates in a helical coil heat exchanger is evaluated next. The heat transfer is increasing as the system pressure is near to the critical pressure of R404A. At the nominal designed pressure of 38.4 bar a heat transfer of 58 kWth is achieved. However, at higher system pressure of 40 bar and 41 bar the heat transfer is lower compared to the inlet pressure of 26 bar and 34 bar.

Figure 5 shows the effects of the system pressure on the performance of the evaporator. At relatively high inlet temperatures of the heating fluid, in this case of 95 °C the heat transfer in the heat exchanger at sub- and supercritical conditions is comparable. This indicates that an optimal heat transfer in the heat exchanger can be achieved at relatively lower system pressure and operating at high system pressure is no longer required.

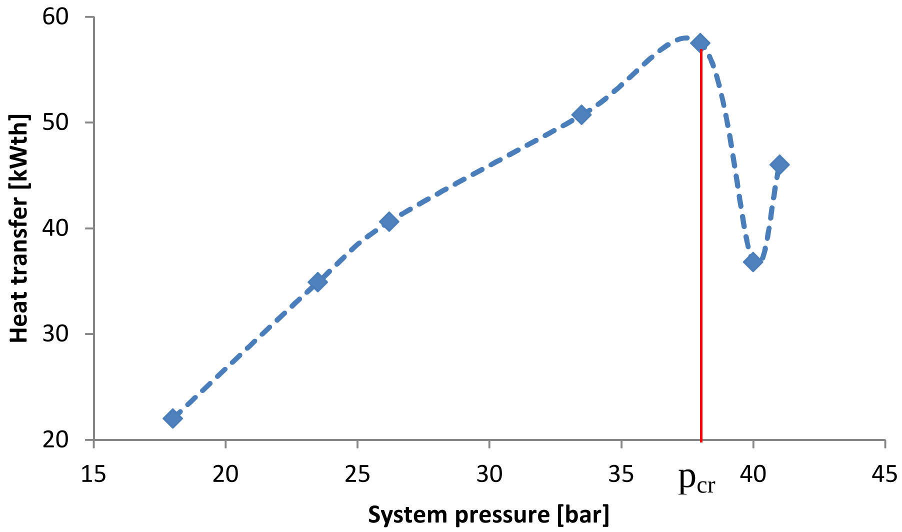

Figure 5. Effects of the system pressure to the hat transfer.

However, the best performance is reached at near critical region or ~5% higher than the critical pressure of the working fluid of the organic fluid R404a.

3.3. Thermal match analysis at sub- and supercritical operating conditions at 95 °C

Analyses of the thermal match in the helical coil heat exchanger were conducted at sub- and supercritical operating conditions. During the measurements, the inlet temperature of the heat source was kept constant at 95 °C, while the inlet temperature of the organic fluid depends from the mass flow rate and the pressure. Further, the pressure of both fluids was also kept steady at the inlet of the heat exchanger. The pinch point temperature difference is determined by the flow rates and the inlet temperatures of the heating and organic fluids. Best performance in terms of heat transfer was achieved at mass flow rate of the heating and working fluid of 2.7 kg/ and 0.3 kg/s respectively, taking into account when the fluid is superheated at sub- and supercritical operating conditions.

Due to local heat transfer measurements in the particular prototype are not possible the values from the developed EES model were used to make the plot in Figure 6.

Line 1 illustrates the heat source that flows downwards in the annulus of the heat exchanger. A temperature drop of 3 °C occurs between the inlet and the outlet of the hot side in the heat exchanger. At the cold side, the working fluid circulates in the coil in upward direction and is evaporated/ vaporized at constant pressure.

Line 2 shows the heat transfer to the organic fluid R404a at supercritical state. The organic fluid is pressurized above its critical pressure and the temperature increases during the heat transfer process as illustrated with line 2. At supercritical heat transfer process a good thermal match is obtained at the outlet of the heat exchanger of only 2 °C. The heat transfer to the supercritical fluid is higher and the irreversibility and exergy destruction are considerably lower in this case.

When the organic fluid is evaporated at constant subcritical pressure as presented with line 3, the average temperature increase of R404a, during the heat transfer is significantly lower than the temperature of the heat source. In this case a significant exergy destruction and irreversibility occurs.

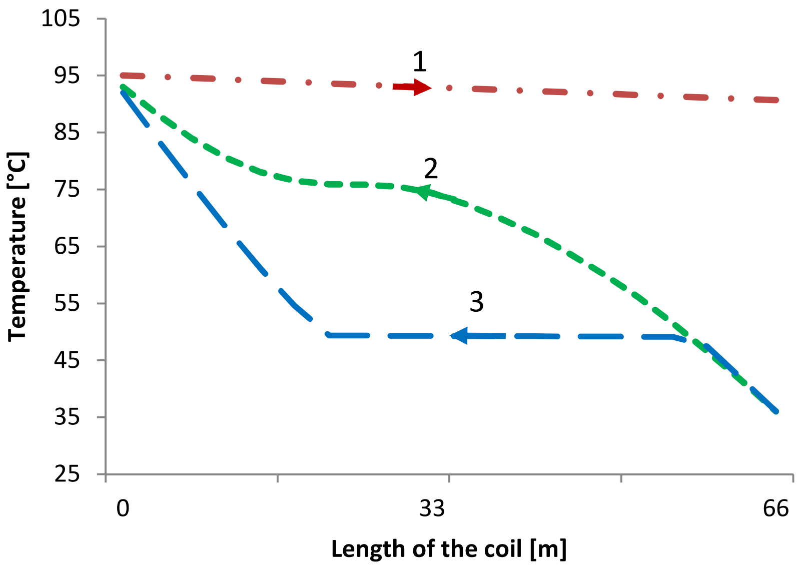

Figure 6. Thermal match in the heat exchanger for sub- and supercritical heat addition.

From the analysis described in the text above it can be concluded that improved thermal match in the heat exchanger is reached for supercritical fluid.

4. Conclusion

An experimental study was conducted to evaluate the performance of the helical coil heat exchanger at sub- and supercritical operating conditions for ORC applications. For all set of these measurements the inlet temperature of the heating fluid was kept constant at 95 °C.

From the measurements and the analysis it can be concluded that in terms of heat transfer, better performance is achieved at supercritical conditions. However, compared to the nominal designed values such as mass flow rate of 0.25 kg/s of the working fluid and inlet temperature of the heating fluid of 95 °C, the heat exchanger outperforms ~ 15% at both operating conditions.

Evaluating the effects of the system pressure on the heat transferred in the heat exchanger yields a conclusion that best performance is achieved at near critical region. While operating at higher system pressure doesn't bring any improvement of the heat transfer compared to lower system pressure.

However, a better thermal match is achieved at supercritical state where the irreversibility and exergy destruction are considerably lower compared to the subcritical heat transfer in the heat exchanger.

In order to have optimal design and operation and cost-effective heat exchanger from the investigation results from this study it is clear that new and more accurate correlations need to be developed for designing a heat exchanger suitable to operate in ORC conditions.

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