# DESIGN OF A SUPERCRITICAL HEAT EXCHANGER FOR AN INTEGRATED CPV/T-RANKINE CYCLE

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## ABSTRACT

The worldwide interest for low grade heat utilization by using Organic Rankine Cycle (ORC) technologies has increased significantly. An Organic Rankine Cycle can be combined with several renewable sources, such as solar energy. Concentrating solar power is a well proven technology and it can be efficiently combined with ORC technology for electricity generation. The goal was achieved by utilizing the excess heat source from PV collectors through a low temperature supercritical heat exchanger in the Organic Rankine Cycle. The motivation for working at supercritical state in the heat exchanger is the better thermal match between the heat source and the working fluid, leading to better overall cycle efficiency.

A designed prototype is a helical coil heat exchanger with R404a used as working fluid flowing in the coil and the heat source fluid in the shell. The design of this heat exchanger was done using heat transfer and pressure drop correlations available from literature. There is a large uncertainty on these correlations for the considered application because they were derived for working fluids water, $CO_2$, R410a and R404a more than ten years ago. In order to have adequate performance and heat transfer rate the heat exchanger was oversized by 20%. Next, the prototype was built and installed in a test set-up. In this paper measurements on the supercritical heat exchanger prototype are reported. The measurements on the prototype show that the heat exchanger is indeed oversized. Based on the measurements, a new heat transfer correlation is suggested. In future design this correlation can be used as less oversized (and thus cheaper) heat exchanger.

## 1. INTRODUCTION

An increased demand for energy and environmental issues on a worldwide level have stimulated a number of researchers to work on improving the efficiency of thermodynamic cycles and look for ways of utilizing renewable energy sources. Concentrated solar power is one of the renewable energy sources that have great potential for electric power generation. A new system integrating two technologies the Concentrated Photovoltaics/Thermal (CPV/T) and Supercritical Organic Rankine Cycle (SCORC) was developed and implemented in Athens, Greece. One solar collector presents a combination of photovoltaic panel and a solar thermal collector that simultaneously generates heat and electricity. The SCORC engine includes the supercritical heat exchanger that is used for recovery of the low grade excess heat from the CPV/T loop. In the SCORC, R404a is used as working medium where the critical pressure and temperature is 37.29 bar and 75.02 °C respectively.

The reason of introducing this innovative concept CPV/T in combination with the SCORC was because the Organic Rankine Cycle is considered a suitable technology for converting low-grade heat sources (e.g. from process industry, solar energy, geothermal, etc.) to electric power generation. This is because the organic (working) fluid has relatively lower critical pressure and temperature compared to water/steam used in classical Rankine cycle. In order to have a good performance of the cycle, not only a good selection of the working fluid is important but also a proper design and selection of the cycle components is essential. A way of enhancing the overall cycle efficiency of an ORC is introduced with supercritical heat transfer in the heat exchanger.

A main challenge to work with supercritical ORCs is a better thermal match of the temperature profiles of the heat source and the working (organic) fluid in the heat exchanger. Moreover, at supercritical state there are strong variations of the thermophysical properties of the fluid. As the value of the heat transfer coefficient depends on these variations, it is important to study and understand the behaviour of the fluid properties when going from subcritical to supercritical state. In order to have a proper design of heat exchanger suitable to work at supercritical conditions it is important to determine the local heat transfer coefficients and correlations.

Other important parameters that influence on the heat transfer are the working fluid flow direction, tube diameter, heat and mass flux, buoyancy and selection of proper organic fluid.

A lot of research activities regarding supercritical heat transfer started in the second half of the 20th century. Many studies were related to heat transfer at critical and near-critical region for a variety of working fluids such as water, carbon dioxide and helium. Back in 1957, Bringer and Smith (1957a) [1] were the pioneers on experimental research for heat transfer to supercritical fluids. Because of the rapid variations of thermal conductivity, viscosity and density they have found out that the existing empirical and semi-theoretical correlations did not give accurate results. The rapid variations of thermal conductivity, viscosity and density were identified as the main reasons for the deviation between experimental results and expectations from the correlation. Dickinson and Weich (1958) [2] are one of the first researchers who did investigation about heat transfer to supercritical water. The work was followed by Shitsman (1959) [3] who did heat transfer research at the near critical region, not only on water but on oxygen and $CO_2$ as well. Krasnoschekov and Protopopov (1959) [4] published a work related to the heat transfer at the supercritical region in tubes for the fluids such as water and $CO_2$. In 1963 Shitsman (1963) [5] published a work related to impairments on heat transmission at supercritical pressure. Bishop *et al.* (1964) [6] investigated the forced convection heat transfer to water at near critical temperatures and supercritical pressure. In 1970, Ackerman (1970) [7] investigated the parameters that influence on the pseudo-boiling heat transfer to supercritical water in smooth and ribbed tubes. Yamagata *et al.* (1972) [8] conducted research related to forced convection heat transfer to supercritical water flowing in vertical tubes. Jackson and Fewster (1975) [9] did work on forced convection to supercritical fluids. Most of the experimental investigations had been mainly done in vertical positioning of the tube. In 1964, Vikrev and Lokshin [10] performed one of the earliest studies about supercritical heat transfer to water at horizontal flow. This study is of great importance because it was first attempt to quantitatively formulate deterioration of heat transfer coefficients in supercritical conditions. The buoyancy effect in a turbulent and vertical flow was first studied by Jackson and Hall (1979) [11].

In order to provide accurate correlations for design of a heat exchanger the heat transfer process to the working fluids at supercritical conditions has to be studied. Even though the research activities regarding supercritical heat transfer started long time ago the first published paper found in the literature regarding supercritical ORC dates from 1981. Haskins (1981) [12] performed research activities of solar receiver coupled to a supercritical ORC engine in order to maximize the thermal efficiency by using toluene as working fluid. Furthermore, ten years later a first paper regarding numerical investigations of the flow pattern and forced convective heat transfer in supercritical flows, such as those encountered in compact heat exchangers had been published [13]. While the first work related to ORC and the influence of the parameters on plate heat exchanger design was presented by Schuster *et al*. (2012) [14].

Many heat transfer and pressure drop correlations have been proposed from the authors mentioned above. It is important to notice that the correlations are derived mainly for fluids such as water, $CO_2$ and helium. However, it is important to be pointed out that this does not marginalize the importance and the scientific value of previously performed research.

From the literature review by Lazova *et al.* (2014) [15] can be concluded that there is very limited experimental and heat transfer data at supercritical state for organic fluids used in transcritical ORCs. The main reason is the difference of the working conditions of an ORC plant such as relatively higher

temperature and pressure. This lack of knowledge necessitates the development of new heat transfer correlations suitable for working fluids used under the supercritical conditions in ORC. Inaccurate correlations lead to an over-sizing of heat exchangers, thus resulting in a lower economic feasibility of such cycles. The heat exchanger is a key component in every ORC engine. This component dictates the efficiency of the cycle and the total cost of an ORC plant. It is estimated that the cost of the heat exchangers is usually up to 30% of the total cost of an ORC where evaporator, (regenerator) and condenser are taken into account Zhang *et al.* (2010) [16]. The cost of the heat exchanger in the CPV/T-Rankine prototype system is only 12% because the major part belongs to the solar collectors. Because the ratio of the total heat exchanger area to net power output in an ORC is considerably high it presents an important issue of consideration.
Hence, from the arguments mentioned above it can be concluded that more accurate design of the heat exchanger with appropriate correlations leads to improved cycle efficiency and lowering the cost of such installation. In this work a supercritical heat exchanger is designed and constructed using literature correlations. Next, the heat exchanger is tested and the measurements are compared to the design specs.

## 2. DESCRIPTION OF THE CPV/T – RANKINE TEST SET-UP

A new test set-up that integrates two technologies such the Concentrated Photovoltaics/Thermal (CPV/T) and Supercritical Organic Rankine Cycle (SCORC) was developed and built in Athens, Greece. The supercritical heat exchanger is the component that couples the CPV/T field from one side with the SCORC engine. For the first measurements the supercritical heat exchanger was tested in the laboratory where an electrical heater was used instead of solar collectors. The final engine is coupled with solar collectors. Heat sources that are of interest in this research work are with temperature range between 70°C to 100°C. Figure 3 illustrates simplified layout of the experimental test set-up. During the experimental campaigns temperature and pressure measurements were conducted, while the mass flow rate of the organic fluid was determined from correlations of the positive displacement pump (SCORC feed pump). The positioning of the pressure and temperature sensors is indicated in Figure 3. In order to evaluate the performance of the heat exchanger, one temperature sensor is placed at the inlet and one at the outlet of the heat exchanger and the heat source respectively. It is important to be mentioned that the system is well insulated, which means that the heat loss to the environment is reduced

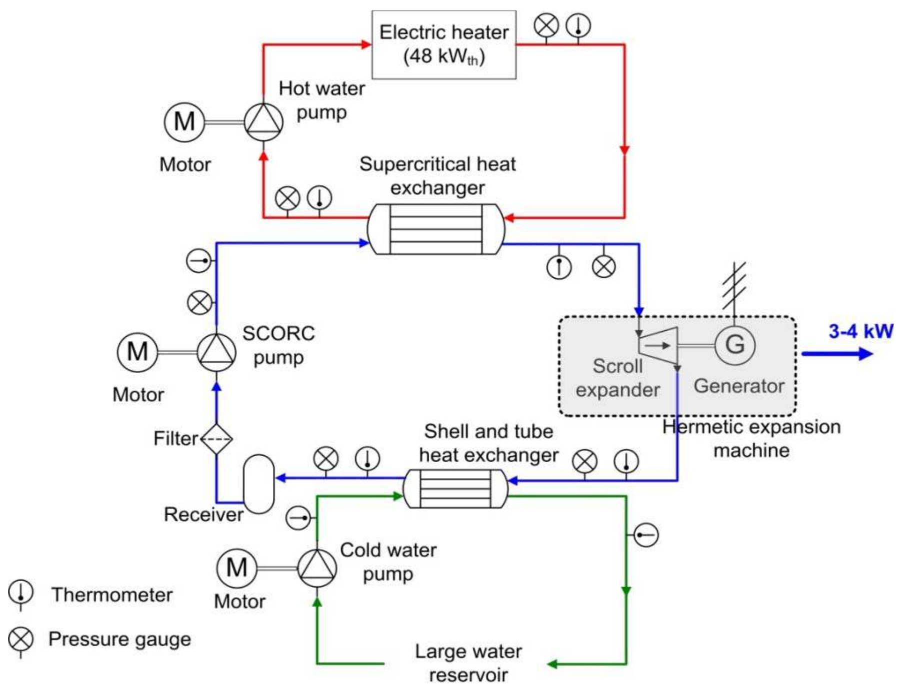


**Figure 3:** Simplified layout of the experimental test set-up

Several measurements campaigns were done, where the supercritical state was achieved under the following conditions presented and compared with the designing condition in Table 1; While running the measurements these values were held constant. As presented in Table 1, there is variation between the design and measurements conditions which gives lower heat transfer capacity.

**Table 1:** Summary of the design and measurement conditions

| | MEASUREMENTS | | DESIGN | |
|---|---|---|---|---|
| **m_wf** | 2,7 | [kg/s] | 2,5 | [kg/s] |
| **T_hf_in** | 101 | [°C] | 95 | [°C] |
| **m_wf** | 0,226 | [kg/s] | 0,2539 | [kg/s] |
| **T_wf** | 36,3 | [°C] | 27,37 | [°C] |
| **p_crit** | 1,026 | | 1,034 | |
| **Q** | 36 | kW | 41 | kW |

The difference between the initially designed model and the built component in the terms of heat transferred is 41 kW and 36 kW respectively.
The aim of these measurements was to evaluate the performance of the heat exchanger working at supercritical state. This component is first of its kind specially designed and build for an ORC installation, suitable to operate at relatively higher pressure and temperature.

## 3. DESIGN OF A SUPERCRITICAL HEAT EXCHANGER

The heat exchanger is part of the SCORC engine that is coupled to the CPV/T circuit in one system. This component was designed by using correlations (Pethukov, Garimella, Mokry) available from literature. The heat capacity of this supercritical heat exchanger is 41 $kW_{th}$. According to the results of the simulations helical coil type heat exchanger was selected based on its ability to withstand high pressure, compact size, performance, easy integration to the system, relatively simple manufacturing and cost. This component is first of this kind specially designed and built for ORC application suitable to operate at supercritical conditions.

### 3.1 Design characteristics

A counter flow heat exchanger of helical coil type Patil *et al.* (1982) [17] is fabricated out of metal coil - tube that is fitted in annular portion of two concentric cylinders. The dimensions of both cylinders are determined by the velocity needed to meet heat transfer requirements. At the annulus side in downward direction the heat source (water-glycol) flows while the working fluid R404a runs in upward direction in the helical coil. The heat transfer takes place across the coil wall. Figure 4 presents the configuration of the helical-coil heat exchanger.

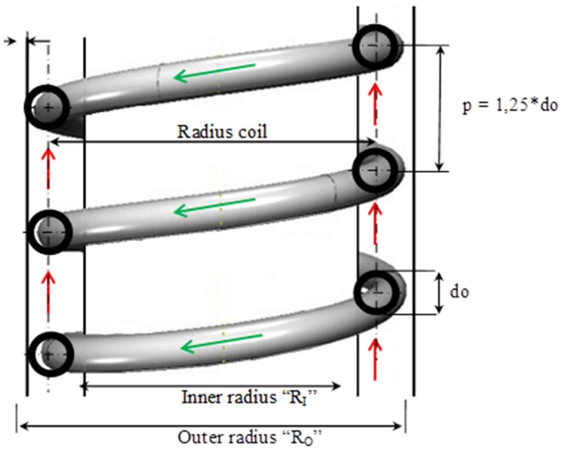


**Figure 4:** Schematic representation of the helical coil heat exchanger

A representative supercritical heating process is presented in Figure 5, showing the temperatures of the heat transfer fluid and an organic fluid R404a, with a pinch point temperature difference of 10 K, which exists at the organic fluid's outlet.

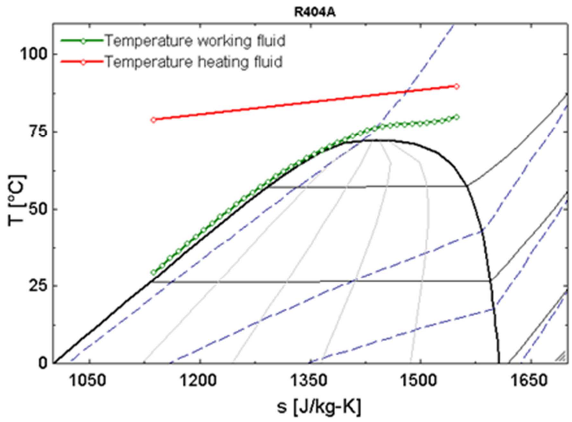


**Figure 5:** T,s-diagram of the heating process in the supercritical heat exchanger

The design of the heat exchanger is accomplished taking into account that the velocity and pressure drop in the coil-tube and annulus are within the allowable ranges. In the calculation procedure, the velocity ranges of the working fluid R404a were fixed at minimum 0.5 m/s and maximum 2.17 m/s, while the overall pressure drop was neglected. It was calculated afterwards and should be lower than 40 kPa. The heating fluid is flowing relatively slow (Re = 4200 - 5900).

### 3.2 Methodology for designing the heat exchanger

A widely used method of calculating the heat transfer capacity (UA) and eventually sizing the heat exchanger is the logarithmic mean temperature difference (LMTD) method, applied between the inlet and outlet of the heat exchanger Cayer *et. al* (2010) [18], Roy *et. al* (2010) [19], Claesson (2005) [20] and given by Eq. (1).

$$\dot{Q} = U.A.\Delta T_{log} = U.A.\frac{\Delta T_1 - \Delta T_2}{ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (1)$$

where $Q$ is the heat transferred, $U$ the overall heat transfer coefficient, $A$ the total heat exchanging surface and $\Delta T_{log}$ is the logarithmic temperature difference or LMTD, and '1' and '2' is the ends of the heat exchanger at which the hot and cold streams enter or exit on either side.
However, the LMTD-method is based on constant fluid properties. When working with fluids at supercritical state this assumption leads to incorrect results. An alternative solution is to discretise the heat exchangers to a large number of control volumes so that the properties variation in each step is small and an average constant value can be assigned within each volume. The discretization is performed in EES (Engineering Equation Solver) by dividing the overall enthalpy change for one of the streams in $N$ (here $N$ = 20) equal differences $dh$. Discretization is advisable to be in the range between 20 and 40 equal distances. Lower than 20 leads to inaccurate results, while above 40 is not suggested, due to the large increase of required computational time (the accuracy is more or less the same).

3.2.1 Heat transfer coefficient at the shell side (annulus): In a helical-coil heat exchanger, the heating fluid is circulated in the annulus. As the flow rate of the heating fluid is rather low, the following Nusselt-correlation, valid for Reynolds number (Re) between 50 and 10000 can be used Eq. (2) Coates (1959) [21]

$$Nu = 0{,}6Re^{0{,}5}Pr^{0{,}31} \quad (2)$$

where $Pr$ is the Prandtl number.

For higher Reynolds number ($Re$ > 10000), Eq. (3) is used Kern (1950) [22]

$$Nu = 0{,}36Re^{0{,}55}Pr^{\frac{1}{3}}(\mu/\mu_w)^{0{,}14} \quad (3)$$

where $\mu$ is the fluid's bulk viscosity and $\mu_w$ is the fluid's viscosity at the wall temperature.

3.2.2 Heat transfer coefficient at the helical coil side: At the helical coil side supercritical fluid is circulated in upward flow. Several correlations can be found in the literature for the calculation of the heat transfer coefficient at supercritical conditions. In this work three correlations for sizing of the heat exchanger are identified and compared. The conventional heat transfer correlations for single phase flow (calculation of the Nusselt number) cannot be used in the current case, due to the variations of the fluid properties around the critical point. For the calculations of the helical coil heat exchanger, three heat transfer correlations are compared.

Petukhov *et al.* (1961) [23] developed correlations for supercritical fluid parameters. The correlations have a correction factor, which neutralizes the effect of the variations of the thermo-physical properties around the pseudo-critical point and provides more stable and accurate results. The proposed Nusselt-correlation originally developed for carbon dioxide in the supercritical range at high temperature drops takes into account the difference in properties between the wall and the bulk, and is given below.

$$Nu_b = Nu_{0,b}\left(\frac{\bar{c}_p}{c_{p,b}}\right)^{0.35}\left(\frac{\lambda_b}{\lambda_w}\right)^{-0.33}\left(\frac{\mu_b}{\mu_w}\right)^{-0.11} \quad (4)$$

where $b$ refers to the bulk fluid temperature and $w$ to the wall temperature.

The heat-transfer coefficient of the organic fluid flowing inside the coil is calculated using the correlations for supercritical heat transfer in a straight pipe. This coefficient is then corrected for a coiled tube by multiplying it by a factor: $F_{helical}$ given by Schmidt's correlation, which has a large application range

$$F_{helical} = 1 + 3{,}6\left[1 - \frac{d_i}{D_H}\right]\left(\frac{d_i}{D_H}\right)^{0{,}8} \quad (5)$$

This expression is applicable for 2 x $10^4$ < $Re$ < 1.5 x $10^5$ and for 5 < $R/a$ < 84, with $R$ the radius of the coil [m] and $a$ the radius of the tube [m].

The term $Nu_{0,b}$ is calculated using the following Petukhov-Kirillov (1958) correlation [24] and the bulk temperature of the fluid.

$$Nu_{0,b} = \left(\frac{\frac{f_b}{8}\, Re_b\,\overline{Pr}}{12.7\left(\frac{f_b}{8}\right)^{0.5}\left(\overline{Pr}^{\frac{2}{3}} - 1\right) + 1.07}\right) \quad (6)$$

where the Darcy friction factor ($f$) is expressed as:

$$f = (1.82log_{10}(Re_b) - 1.64)^{-2} \quad (7)$$

The average specific heat $\bar{c}_p$ is defined as:

$$\bar{c}_p = \frac{h_b - h_w}{T_b - T_w} \quad (8)$$

Garimella (2005) [25] developed correlations for supercritical heat transfer based on measurement data from refrigerants bends R410a and R404a. Three regions of heat transfer were identified based on the state of the heat transferring fluid: Liquid-like region, Pseudo-critical transition and Gas-like region. For each region a separate correlation for Nusselt number and friction factor was identified. However, these correlations were developed for smaller diameter (9,4 and 6,2 mm) and for supercritical heat transfer cooling applications. As already mentioned, the tube diameter influences the heat transfer rate. The designed supercritical heat exchanger has relatively higher inner diameter of 26 mm and the working conditions are different from one for ORC application. Therefore, these correlations were taken into account and compared to other without completely relaying during the design process. The average uncertainties in these heat transfer coefficients were ±10%. These correlations are listed in continuation;

Liquid-like region:

$$Nu = 1{,}421Nu_{churchil-modfied}\left(C_{p,w}/C_{p,b}\right)^{0{,}444}(d_{actual}/d_{baseline})^{-0{,}183} \quad (9)$$

Pseudo-critical transition:

$$Nu = 1{,}350 Nu_{churchil-modfied}\left(C_{p,w}/C_{p,b}\right)^{0{,}249}(d_{actual}/d_{baseline})^{-0{,}066} \quad (10)$$

Gas-like region:

$$Nu = 1{,}556 Nu_{churchil-modfied}\left(C_{p,w}/C_{p,b}\right)^{-0{,}212}(d_{actual}/d_{baseline})^{-0{,}308} \quad (11)$$

These correlations are valid for the following working range: 200 kg/m²s < G < 800 kg/m²s and 1.0 < $P/P_{cr}$ < 1.2. Also there are correction factors developed for all flow regimes boundaries.

The majority of empirical correlations were proposed in the 1960s – 1970s, when experimental techniques were not at the same advanced level as they are today. Also, thermo-physical properties of water have been updated since that time (for example, a peak in thermal conductivity in critical and pseudo-critical points within a range of pressures from 22,1 to 25 MPa was not officially recognized until the 1990s). Therefore, a new or an updated correlation, based on a new set of heat-transfer data and the latest thermo-physical properties was recently developed and evaluated by Mokry et al. (2011) [26]

$$Nu = 0{,}0061 Re^{0{,}904} Pr^{0{,}684}(\rho_w/\rho_b)^{0{,}564} \quad (12)$$

This correlation is valid for the following working range: 200 kg/m²s < G < 1500 kg/m²s.

### 3.3 Dimensions of the heat exchanger

As summary, the final design of the heat exchanger leads to coil length of 66 m and inner diameter of the coil of 26 mm. To account for heat transfer correlation uncertainty the heat exchanger is oversized by about 20%. Table 1 presents summary of the heat exchanger design.

**Table 2:** Summary of the heat exchanger design

| Helical coil heat exchanger | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $d_o$ [mm] | t [mm] | L [m] | $D_i$ [m] | $D_o$ [m] | $D_c$ [m] | H [m] | A [m²] | Q [kW] | h_hf_avg [W/m²K] | h_wf_avg [W/m²K] |
| **33.7** | **4** | **66** | **0.526** | **0.674** | **0.6** | **1.508** | **6.988** | **41** | **403** | **2200** |

where $d_o$ is the tube outer diameter, t the tube thickness, $D_i$ the inner shell diameter, $D_o$ the outer shell diameter, $D_c$ the coil diameter, H the height of the HX and A the total heat exchanger surface. Q is the heat transfer capacity of the heat exchanger coefficient. h_avg is the average heat transfer coefficient.

## 4. COMPARISON OF THE CALCULATED WITH THE MEASURED DATA OF THE SUPERCRITICAL HEAT EXCHANGER

### 4.1 Analysis of the constraints of the designed and built model

To check the performance of the helical coil heat exchanger, the influence of changing mass flow rate to the heat transferred and the outlet temperatures at cold and hot side is investigated. The constraints are presented in Table 2. The outlet temperatures at the cold and hot side and the pinch point temperature difference are determined by the flow rates and inlet temperatures. In figures 6 and 7the pinch point temperature difference (design and measurements) is presented.

Changing the flow rate of the heating fluid without changing the flow rate of the working fluid will result in a decrease of the outlet temperature of the heating fluid, a decrease of the outlet temperature of the working fluid and an increase of the pinch point temperature difference.

The designed pinch point temperature difference is 10 K. From the sets of measurements covering supercritical operation it is shown that the main advantage is the very low temperature differences between the heating fluid and the organic fluid R404a (in the range of 1.5-2.5 °C). Moreover, the pressure drop of the heating fluid is very small and equal to 0.1-0.2 bar, while that of the organic fluid R404a is higher and in the range of 0.6-1 bar. This value is low, which should be however considered during the design stage, in order to select the correct size of this component.

In figure 6 and 7 the pinch point temperature difference (design and measurements) between the heating fluid and the working fluid R404a is presented.

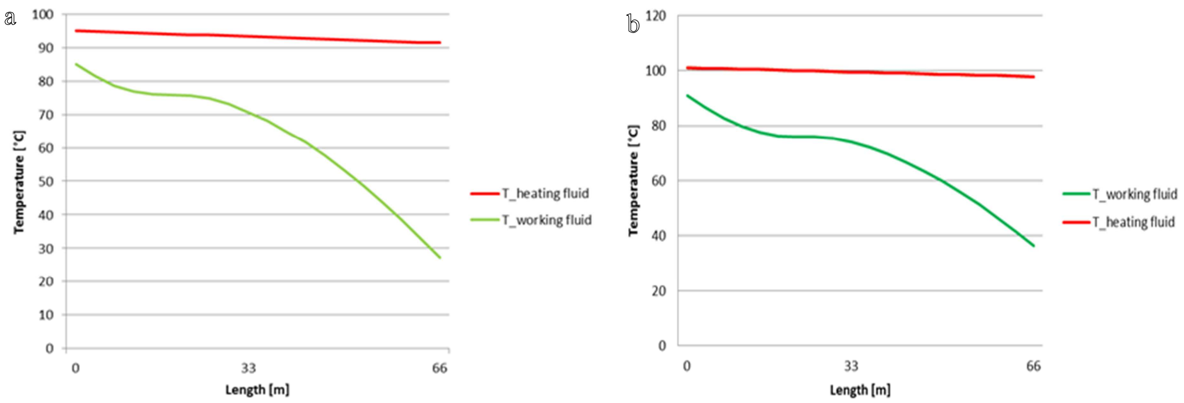


**Figure 6:** Pinch point temperature differences: a. designed model; b. built model

**4.2 Development of new correlation suitable for helical coil heat exchanger**

Using a modified Wilson method introduced by Shah (1990) [27], Jose *et. al* (2005) [28] the mean value of the convection coefficient outside the tubes and the convection coefficient inside the tubes as a function of the working fluid mass flow (or velocity) are obtained. Figure 8 is logarithmic graph that describes the Nu number as a function of the Re and Pr numbers. This graph presents a comparison of the experimental data and the correlations used from the literature. Moreover, the coefficient C and the exponent of the Reynolds number m of the general dimensionless correlation $Nu = CRe^{m}Pr^{n}$ are also obtained, thus the general correlation is determined assuming only the value of the exponent of the Prantdl number, n.

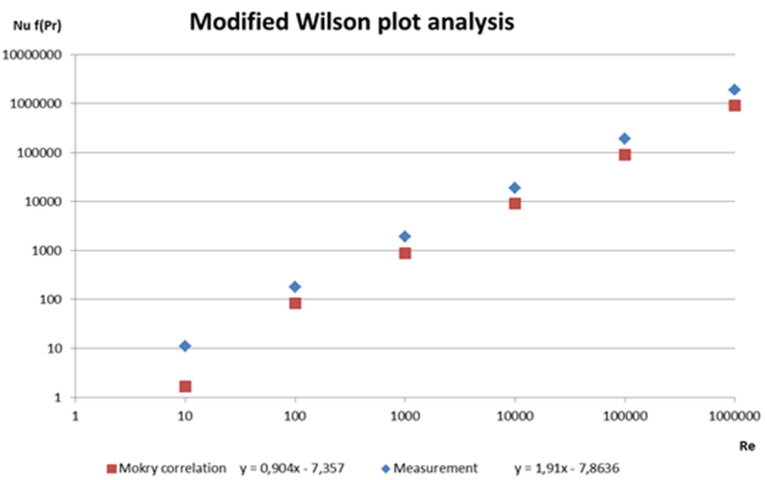


**Figure 8:** Logarithmic graph describing the Nu number as a function of the Re and Pr numbers (comparison of calculated with measured data)

$$Nu = 0{,}0044Re^{1{,}91}Pr^{0{,}4} \quad (13)$$

In the design process of this supercritical heat exchanger, three calculation methods for supercritical heat transfer were implemented and compared (Petukhov, Garimella, and Mokry). These heat transfer correlations were developed independently. A safety factor was implemented to account for heat transfer correlation uncertainty. The measurements and the new correlation derived from this experiment indicate (Figure 8) that the measured heat transfer is about 10% higher than the used correlations. Hence, the size of the heat exchanger can be reduced by 10% and still keeping the good performance (heat transfer rate). On the other hand, by reducing the size of this component, the cost will be decreased and an economic benefit on the whole plant will be achieved. The benefit would be more accurate design and use of less material, which leads to lower costs and lower pressure drop for both fluid circuits.

## 6. CONCLUSIONS

In this work a supercritical heat exchanger suitable for ORC applications is investigated. Even though ORC is not new technology, there is still room for improvement by working at supercritical state of the organic fluid. However, there is still lack of experimental (accurate) data, especially suitable for ORC installations. A helical coil heat exchanger was designed and built. Correlations available from literature were used. These correlations were developed for water, $CO_2$ and refrigerants like R404a and R410a. The correlations were derived for smaller diameter and different working conditions than ORCs. In order to check the performance of the designed and built heat exchanger at supercritical working conditions, measurements have been conducted. The inlet temperatures of the working and heating fluid were held constant 36,3°C and 101°C respectively. For this measurement the mass flow rate was 0,226 kg/s and the heat exchanger showed good performance. The pinch point temperature difference at these working conditions is lower than 10K. From the mentioned arguments in this work, it can be concluded that more accurate design of the heat exchanger with appropriate correlations leads to improved cycle efficiency and lowering the cost of such installation.

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