Experimental investigations of forced convective heat transfer to fluids operating at supercritical state in Organic Rankine Cycles

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Supervisor: Prof. dr. ir. Michel De Paepe
Counsellor: Marija Lazova

Master's dissertation submitted in order to obtain the academic degree of Master of Science in Electromechanical Engineering

Department of Flow, Heat and Combustion Mechanics
Chair: Prof. dr. ir. Jan Vierendeels
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Preface

This thesis is the final work of the Master in Electromechanical Engineering. I decided to investigate supercritical heat transfer to organic fluids. I have always been curious about the subject and my interest has grown even more during the year.

I would like to thank several persons that have had an impact or have contributed to the writing of this thesis.

First of all, I thank my promoter Prof. Dr. ir. De Paepe to give me the opportunity to reflect about this interesting subject. He gave me feedback during several meetings throughout the year.

Secondly, I would like to thank ir. Marja Lazova for her input and her guidance.

Thirdly, my mother is always there to motivate me and to inspire me every day. She always believes in me, no matter what I do, she is there for me.

Lastly, my girlfriend deserves a special thank you for her support and for the long hours we worked together, both writing our thesis to finish our studies simultaneously. Her critical reading helped review the final thesis.

Thibault Stepman, June 2016
Admission to Loan

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Thibault Stepman, June 2016
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by

Thibault STEPMAN

Master’s Dissertation submitted to obtain the academic degree of Master of Science in Electromechanical Engineering - Energy Engineering

Academic year 2015–2016

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Summary

This thesis investigates the supercritical heat transfer phenomenon with organic working fluids in large diameter tubes. There is not a lot of academic research to be found, as the main working fluids are either water, CO₂ and helium. This is partly because they have well documented properties, but also because they are readily available.

The main goal of this thesis is to develop a new correlation for supercritical heat transfer to organic working fluids in large diameter tubes. Therefore, a literature study is conducted in order to gain more insight in the subject. Two important researches are being treated in depth, namely the work of Garimella and Zhao et al. Chapter 3 describes the test set-up that is used to conduct the measurements. Built at Ghent University, the iSCORe test set-up simulates a transcritical Organic Rankine Cycle with a test section of 4 m long. It is equipped with multiple temperature and pressure sensors to have overall and local heat transfer measurements.

The next chapter discusses the control of the test set-up. This is necessary as several variables have to be controlled. A program is written in LabVIEW, with PID controls for each component that needs to be controlled.

As we come to chapter 5, things get interesting. This chapter explains how all collected data are transformed to Nusselt numbers and heat transfer coefficients.

Chapter 6 explains the uncertainty propagation method and applies it to the equations used in Chapter 5. It is a necessary step to usefully interpret the collected data.

Arriving at chapter 7, results are discussed and compared to other correlations from the literature. Lastly chapter 8 concludes this thesis.
Keywords

ORC, Organic Rankine Cycle, Transcritical, iSCORe, R125, experimental, supercritical heat transfer
EXPERIMENTAL INVESTIGATION OF FORCED CONVECTIVE HEAT TRANSFER TO FLUIDS OPERATING AT SUPERCRITICAL STATE IN ORGANIC RANKINE CYCLES

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ABSTRACT

The Organic Rankine Cycle (ORC) allows the conversion of heat coming from low-grade waste heat sources (60 °C – 250 °C) into electrical energy. Transcritical ORCs, in which the heat transfer from the heat source to the organic working fluid occurs at supercritical conditions, are relatively simple and very efficient cycles that show great potential for industrial applications. In order to design these transcritical cycles, accurate correlations for supercritical heat transfer in large diameter tubes need to be available.

This paper reports on the initial measurements done on a transcritical ORC set-up, iSCORE. The results obtained provide a meaningful insight in the working of the set-up, which will be applied for further research.

INTRODUCTION

The total primary energy consumption across the globe has increased with 99 % in the last 40 years and is expected to keep increasing in the years to come. In 2013 the industry accounting for about 29 % of the total primary energy consumption [1]. Each process involves losses and statistical investigations show that up to 20 - 50 % of the energy input is lost as waste heat [2, 3] with low-grade waste heat accounting for up to 50 % of the heat loss [4]. If this energy in the form of low grade waste heat is recovered, it would decrease the primary energy demand in industry significantly.

Several cycles are available that utilize low grade heat (T ≤ 180 °C) such as the Kalina Cycle, Trilateral Flash Cycle and the Organic Rankine Cycle (ORC). Compared to the other two cycles, the Organic Rankine Cycle is simpler and more straightforward in design which makes it a very promising technology and a the focus of many researches these days. ([5, 6, 7, 8, 9]).

The ORC is very similar to the well known Rankine Cycle but uses an organic working fluid rather than water. Because of the lower boiling point of the organic working fluid, energy recuperation from low grade heat sources is possible at relative high efficiencies. A division within the ORCs is made based upon the operational temperatures and pressures of the organic working fluid, where the subdivision is based on the critical point of the fluid. At the critical point, the liquid- and gaseous phase are no longer indistinguishable. A subcritical cycle is when the temperature and pressure remain below the critical point of the fluid. At the critical point, the liquid- and gaseous phase are no longer indistinguishable. A subcritical cycle is when the temperature and pressure remain below the critical point of the fluid. In other words heat addition and rejection to and from the organic working fluid is accompanied with phase change and two-phase fluids. A supercritical cycle is when the temperature and pressure remain above the critical point for the entire cycle while a transcritical cycle has heat addition to the organic working fluid at supercritical pressures and temperatures, but heat rejection at subcritical conditions. There is no phase change during the heat addition because the two-phase region is omitted.

The focus of this study is heat transfer measurement to supercritical fluids. Therefore, a new test set-up with a test section length of 4 m was built. The purpose of this study is to have and determine local heat transfer measurements and local heat transfer coefficients respectively. The test set-up is a transcritical cycle with the main focus on the heat exchanger and instead of an expander, an expansion valve is used.

In Figure 1 a schematic is shown of an transcritical ORC. A pump compresses the organic working fluid to a pressure equal or above the critical pressure (1-2)
Figure 1: Schematic of a transcritical ORC

and leads it to the evaporator where the low grade heat source is used to heat up the organic working fluid (2-3) to a supercritical state. The supercritical organic working fluid is then expanded over an expansion machine (3-4) to generate electricity. In the condenser (4-1) the remaining heat can be used for heating purposes, depending on the temperature, or cooled down using a cooling circuit. Leaving the condenser as a (sub-cooled) liquid, it flows back to the pump to finish the cycle.

Suitable organic working fluids for transcritical ORCs with low grade heat sources include, but are not limited to: R125, R134a, R245fa,... The most suitable working fluid for a transcritical cycle depends on the temperature of the heat source as the critical temperature of the organic working fluid must be lower than the heat source operating temperature. A general selection criteria is that the critical temperature should be around 0.8 times the temperature of the heat source [10].

For applications in real cycles one has to look at the Global Warming Potential (GWP) and Ozone depletion potential (ODP) of the fluid. The ODP is the relative amount of damage a chemical substance could do to the ozone if it is released in the atmosphere compared to CO\textsubscript{2}, while GWP is the relative amount of heat the chemical substance captures when released in the atmosphere compared to R11.

**SUPERCRITICAL HEAT TRANSFER**

At supercritical pressures an unusual phenomenon occurs with the thermophysical properties, they vary very rapidly and is the focus of some studies such as from [11], [12] or [13]. Figure 2 shows the rapid variation of the density and specific heat capacity at supercritical pressures of 40 and 45 bar. The specific heat capacity shows a clear peak at a temperature called the pseudo-critical temperature, T\textsubscript{pc}. By increasing the pressure, the peak decreases in magnitude but also shifts to higher temperatures. When the organic working fluid is at temperatures below T\textsubscript{pc} it behaves more as a liquid with more liquid like properties, but when it is near to the pseudo-critical temperature it enters a transition to liquid-gas like. With temperatures above T\textsubscript{pc} the organic working fluid behaves as a gas with gas-like properties. This can cause so called pseudo-boiling, a phenomenon similar to subcritical film boiling but at supercritical pressures. The bulk of the fluid, with temperatures below T\textsubscript{pc}, behaves as a liquid while the outer layer, which is at temperatures above T\textsubscript{pc}, behaves as a gas and prevent the colder liquid to contact the heating surfaces by forming a protective film. This leads to deteriorated heat transfer [14].

The first investigations on supercritical heat transfer were fluids such as water, CO\textsubscript{2} and helium. Oh and Son [15] investigated the cooling of supercritical CO\textsubscript{2} in a smooth horizontal tube-in-tube heat exchanger with and inner diameter of 4.55 mm and 7.75 mm and with length of 4000 mm and 6000 mm respectively. Oh and Son [15] created a new correlation based on the Dittus-Boelter type but with addition of two terms that account for the density variations and the specific heat capacity variations between the bulk and the wall of the fluid.

Gupta et al. [16] conducted experiments on heating of the supercritical CO\textsubscript{2} in a vertical tube with an inner diameter of 8.06 mm and a length of 2280 mm. A Dittus-Boelter correlation was proposed with a thermal conductivity ratio, dynamic viscosity ratio and the density ratio to cope with the property variations between the bulk and wall of the working fluid.

The density variation has a stronger influence in the correlation of Oh, as with a horizontal tube the buoyancy plays a major role in the heat transfer process.

Another study on horizontal tubes was performed by Garimella [17] on cooling of R-410A, with diameters of 0.76 mm, 1.52 mm, 3.05 mm at various supercritical pressures. He defined three separate flow regimes by using a dimensionless number E\textsubscript{0}. This dimensionless number is the ratio of the flow work to the heat convected out during the cooling process. Three regions are defined and form the basis of the pressure drop correlation: the liquid-like, the pseudo-critical and the gas-like. Garimella’s heat transfer correlation is less pronounced with density variations, because in small diameter tubes the buoyancy effects are reduced compared to large diameter tubes.

Studying supercritical heat transfer and pressure drop of R-134a was done by Zhao and Jiang [18], using an inner diameter of 4.01 mm. The resulting equation, based upon the Gnielinski equation, takes into account the density variations from bulk to wall and the specific heat capacity variations along the axial direction. It is no surprise to see the density variations being strongly present in the correlation as he uses a horizontal set-up.

Many other studies that have been done are either numerical, ([19], [20]) or on smaller diameter tubes. The focus of this study is heat transfer and pressure drop in large diameter tubes with an inner diameter of 24.7 mm using the organic fluid R125. To the knowledge of the author no studies have been done with supercritical heat transfer in large diameter tubes using hydro-fluorocarbons. This study is an effort to fill this void in literature.

**TEST SET-UP**

A test set-up has been constructed specifically for supercritical heat transfer measurements in a transcritical Organic Cycle, iSCoRe. Built at Ghent University it’s aim is to develop new correlations for supercritical heat transfer with organic fluids in horizontal large diameter tubes.
tubes. Consisting of three loops (heating, cooling and an experimental) it was designed for high accuracy measurements over a wide operating range. Figure 3 show the test set-up with all it’s components and measuring equipment. For more detailed information about the parts in the test set-up see Daelman [21] and Bauwens [22].

Table 1: Overview of the organic working fluid properties

<table>
<thead>
<tr>
<th>Property</th>
<th>HFC-125</th>
<th>HFC-134a</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chemical formula</td>
<td>C₂HF₅</td>
<td>CF₃CH₂F</td>
</tr>
<tr>
<td>Tₑcrit</td>
<td>66.02 °C</td>
<td>101.03 °C</td>
</tr>
<tr>
<td>Pₑcrit</td>
<td>3.618 MPa</td>
<td>4.060 MPa</td>
</tr>
<tr>
<td>ODP</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>GWP (100y)</td>
<td>3500</td>
<td>1430</td>
</tr>
</tbody>
</table>

Table 1 shows the properties of the two working fluids that will be tested in the first phase on this test set-up. These fluids are very commonly used as working medium in refrigerating applications but will be replaced by other fluids, with lower GWP, in the near future. However as a first study on the new set-up it can provide meaningful insight in the supercritical heat transfer because of their low critical temperature and pressure.

**Heating loop, cooling loop and experimental loop**

The heating loop consists of a thermal heater with two 10 kW heating elements with a built in circulation pump. It provides the heat source for the organic working fluid and the heating lengths can be changed from 0 m to 8 m in steps of 4 m by bypassing none, one or two preheaters respectively. Flow rate is electronically controlled by a three way valve placed in diverting position.

The cold source for the organic working fluid is provided by the condenser, a plate heat exchanger, that uses a 70%/30% volumetric ratio of water/glycol as a coolant. This coolant is cooled by a 37 kW chiller in a 900 l buffer vessel and the flow rate is also electronically controlled by a three way valve.

The experimental loop is composed of a pump, two tube-in-tube preheaters and an electrical preheater, a test section, and expansion valve, a condenser and an accumulator. This expansion valve is very important as without it the pump would not be able to function because of the inlet pressure limitation of the pump. By definition there is now a high pressure side and a low pressure side that have to be controlled and for this several sensors are used. A total of 5 Pt100 temperature sensors, 5 pressure transducers and 42 thermocouples are used to monitor not only the test section but the entire set-up.

**Test section**

A tube-in-tube counterflow heat exchanger with a length of 4 m is used as the test section. The test section is insulated with an insulating ring and wrapped in an insulating blanket. More details about the heat losses are included in the section Data reduction. The benefits of a tube-in-tube heat exchanger are the uniform heating along the test section with wall temperatures representative for practical applications which is not the case with Joule heating. The inner tube consists of a copper tube with inner diameter of 24.77 mm and outer diameter of 28.57 mm. The outer shell of the tube-in-tube heat exchanger is made from galvanized steel with an
Figure 4 is a schematic representation of the test section and shows also the placement of the thermocouples. In total 33 thermocouples are placed upon the copper wall divided over eleven equally spaced measuring points. In each of these points a thermocouple is placed at the bottom, side and top measuring the wall temperatures of the copper tube. These measurements will be used to determine the flow variation in the organic working fluid as well as developing a new heat transfer correlation. The heating fluid bulk temperature is measured using 9 thermocouples at three equally spaced measuring points. Again the bottom, side and top of the fluid temperature is measured. All these thermocouples were calibrated at Ghent University with a temperature controlled oven for an increased accuracy. More details can be found later in the section Error analysis.

Measuring the pressure drop across the test section is achieved with two pressure transducers at the in- and outlet of the test section allowing the operational pressure to be monitored at all times.

The mass flow rate through the test section of the organic working fluid is measured by a coriolis mass flow meter and varies between 0.19 and 0.31 kg/s. The flow rate of the heating fluid can be controlled by a three way valve as previously said, but is not varied in this study.

The operating conditions are listed in Table 2. Most of the measurements are done at supercritical pressures and temperatures with R125. In order to test the set-up and know how it behaves, some experiments have been done at near-critical pressures and temperatures. The organic working fluids selected for this study (R125, R134a), are compatible with the installed components and materials used. One of the prerequisites of these working fluids is that it is non-flammable, non-harmful to people, has an ODP of 0, are colourless and have a relatively low critical point. The latter is an advantage because supercritical conditions can easily be reached.

### Table 2: Working conditions

<table>
<thead>
<tr>
<th>Property</th>
<th>Operating range</th>
</tr>
</thead>
<tbody>
<tr>
<td>$p_{wf}$ [bar]</td>
<td>35 - 50</td>
</tr>
<tr>
<td>$G_{wf}$ [kg/m²s]</td>
<td>395 - 645</td>
</tr>
<tr>
<td>$T_{wf}$ [°C]</td>
<td>40 - 90</td>
</tr>
<tr>
<td>$T_{hf}$ [°C]</td>
<td>70 - 120</td>
</tr>
</tbody>
</table>

### DATA REDUCTION

#### Heat loss

To minimize the losses to the environment the test set-up is insulated with high-performance insulation of a low thermal conductivity. The test section and preheaters are both protected with extra insulation because of the higher temperatures that occur in the outer shell due to the heating fluid. It is wrapped in an insulating ring around which is covered by an insulating blanket with a total thickness of 10 mm for the ring and blanket together.

Calculating the heat losses is easily done using the heat resistance network in Figure 5 and the following equation:

$$ Q_{loss} = \frac{T_{b,hf} - T_{amb}}{R_{wall} + R_{ins,r} + R_{ins,b} + \frac{R_{conv} R_{rad}}{R_{conv} + R_{rad}}} \quad (1) $$

Iteration is necessary to calculate the radiation and convective resistances because the temperature at the outer surface of the insulation, $T_{ins}$, is unknown. Low heat losses to the environment of 70 to 100 W were achieved depending on the ambient and operating temperature.

### Heat transfer coefficient

The overall heat transfer coefficient is determined by subdividing the test section in N cells with equal enthalpy difference. If the number of cells is chosen high enough, the assumption of constant thermophysical properties in each cell is valid. This allows the
use of the Logarithmic Mean Temperature Difference (LMTD) method as, see equation 2. The LMTD method assumes constant properties at the inlet and outlet of the heating fluid and the organic working fluid, which is not the case if few cells are chosen. The number of cells chosen in this study is 80, a balance between computation time and precision. The constant enthalpy difference, together with the assumption of linear pressure drop along the test section and known inlet temperature, determines the temperature profile at the in- and outlet of each computational cell.

\[
LMTD = \frac{(T_{h,f,in} - T_{w,f,out}) - (T_{h,f,out} - T_{w,f,in})}{\ln\left(\frac{T_{h,f,in} - T_{w,f,out}}{T_{h,f,out} - T_{w,f,in}}\right)}
\]  

(2)

When the temperature profile is known, a set of \( N+2 \) equations need to be solved to find the overall heat transfer coefficient \( U \). The heat transferred to the organic working fluid, \( Q_{w,f} \), is divided in \( N \) equal parts \( Q' \). The heat transferred to the working fluid in each cell also equals the product of the overall heat transfer coefficient \( U \), the surface heat transfer area \( A \) in each cell \( A_{HT,i} \) and the LMTD in each cell. To complete the set of equations, the sum of the heat transfer surface areas in each cell must equal the overall heat transfer surface area \( A_{HT,wt} \). With \( N+2 \) equations and \( N+2 \) unknowns, this set can be solved to find the overall heat transfer coefficient \( U \).

\[
\begin{aligned}
NQ' &= Q_{w,f} \\
\sum_{k=1}^{N} A_{HT,k} &= A_{HT,wt} \\
Q' &= U \cdot A_{HT,k} \cdot LMTD_k 
\end{aligned}
\]  

(3)

The calculation of the overall heat transfer coefficients allows for the calculation of the convection heat transfer coefficient at the working fluid side \( h_{tc,w} \). This is determined using equation 4. In this equation the fouling resistance at the inside and outside of the tube are neglected as the set-up is new. Furthermore, the heating fluid and the working fluid are compatible with the material for the tube and no corrosion is to be expected. The convection heat transfer coefficient at the heating fluid side, \( h_{tc,o} \), is unknown and should be determined with a heat transfer correlation suitable for fully developed, turbulent flow with low viscosity fluids.

\[
\frac{1}{U \cdot A_{HT,i}} = \frac{1}{h_{tc,i} \cdot A_{HT,i}} + \frac{1}{R_w} + \frac{1}{h_{tc,o} \cdot A_{HT,o}}
\]  

(4)

To calculate the convection heat transfer coefficient at the heating fluid side, equation 5 is used. It is the standard Dittus-Boelter equation with a viscosity term to account for the changes in viscosity.

\[
\begin{aligned}
N_{u,tf} &= 0.023 \cdot \frac{Re_{h,f}}{8} \cdot Pr_{h,f}^{1/3} \cdot \left(\frac{b_{o,h,f}}{b_{w,h,f}}\right)^{0.14} \\
N_{u,bf} &= \frac{htc_{w,Do,h,f}}{b_{o,h,f}} \\
N_{u,fb} &= \frac{htc_{o,Do,w,f}}{b_{w,h,f}}
\end{aligned}
\]  

(5)

All the data necessary to calculate the Nusselt number of the heating fluid is available in the documentation provided by the manufacturer such as the density, the viscosity, etc.

The calculation of the local heat transfer coefficient is also possible by the wall mounted thermocouples along the test section. The test section can be subdivided in 4 sections of 1 m parts where the heat flux can be taken as a constant. The heat flux is also considered constant in between each measuring point and is calculated as the ratio of the heat flux to the heat transfer area at the inner surface of the tube (equation 6).

\[
\begin{aligned}
htc_{loc} &= \frac{Q_{w,f}}{A_{HT,wt} \cdot (T_{w,in} - T_{w,out})} \\
q_{w,f} &= Q_{w,f} \cdot \frac{T_{w,in} - T_{w,out}}{4k_{tub}} \\
T_{w,in} &= T_{w,out} - \frac{Q_{w,f} \cdot ln\left(\frac{T_{w,out}}{T_{w,in}}\right)}{2 \pi L \cdot \lambda_{tub}}
\end{aligned}
\]  

(6)

The wall temperature at the inner surface of the tube is determined assuming one-dimensional heat conduction using the measured external wall temperature. The external wall temperature can be either measured the top, side of bottom wall temperature, which gives us a local heat transfer coefficient for the top, side and bottom of the organic working fluid.

\[
T_{w,out} = T_{top} \quad T_{w,out} = T_{side} \quad T_{w,out} = T_{bottom}
\]  

(7)

**Pressure drop**

The pressure drop across the test section is measured using two high accuracy absolute pressure sensors and is can be seen as a combination of three phenomenon: a frictional pressure drop, an adiabatic pressure drop and a pressure drop due to acceleration of the organic working fluid. We could consider the static pressure drop as well, but since this depends on the flow angle relative to the horizontal plane, it is zero in our case. The adiabatic pressure drop depends on the lengths of the adiabatic section, the density and velocity of the working fluid and the through flow diameter [18]

\[
\Delta p_{ad} = f \cdot \frac{L_{ad}}{d_{ad}} \cdot \rho \cdot u^2
\]  

(8)

\[
f = [1.82 \cdot \log (Re_{b}) - 1.64]^2
\]  

(9)

The pressure drop due to acceleration is a result from density variations across the test section. As the mass flow rate has to remain constant and the density decreases along the test section, the organic working fluid accelerates. This is calculated using equation 10 [23]:

\[
\Delta p_{ac} = G_{wf} \cdot \left(\frac{1}{\rho_{out}} - \frac{1}{\rho_{in}}\right)
\]  

(10)

With these we can calculate the frictional pressure drop through the test section as follows:

\[
\Delta p_f = \Delta p_{meas} - \Delta p_{ac} - \Delta p_{ad}
\]  

(11)

**Error analysis**

In this study three parameters are measured: the temperature, the pressure, and the mass flow rate. Table 3 is an overview of the measurement accuracies. Uncertainties are not only in the measured quantities, but also in tabulated data. CoolProp [24] is used to calculate the thermophysical properties, and the accuracies of the equation of state used for calculating the density, specific heat capacity, etc., are relatively high around the critical region. This is because the equation
of state was developed with the scarce available data around the critical region. This experimental data is hard to achieve and has inherently higher errors.

Table 3: Overview of the uncertainties in the measurements

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{w}$</td>
<td>± 0.10 % - 0.14 %</td>
</tr>
<tr>
<td>$T_{wf,in}$</td>
<td>± 0.021 %</td>
</tr>
<tr>
<td>$T_{wf,out}$</td>
<td>± 0.021 %</td>
</tr>
<tr>
<td>$T_{hf,in}$</td>
<td>± 0.018 %</td>
</tr>
<tr>
<td>$T_{hf,out}$</td>
<td>± 0.018 %</td>
</tr>
<tr>
<td>$\dot{m}_{wf}$</td>
<td>± 0.07 %</td>
</tr>
<tr>
<td>$\dot{m}_{hf}$</td>
<td>± 0.08 %</td>
</tr>
<tr>
<td>$p_{wf,in}$</td>
<td>± 0.005 %</td>
</tr>
<tr>
<td>$p_{wf,out}$</td>
<td>± 0.008 %</td>
</tr>
</tbody>
</table>

When all absolute uncertainties of all sensors and tabulated data are known, equation 12 is used to see how these inaccuracies propagate through the data reduction. Equation 12 represents the absolute error of $q$ that is a dependant of parameters $x,...,z$ which have the independent and random absolute errors $\delta x,...,\delta z$ ([25]).

$$\delta q = \sqrt{\left( \frac{\partial q}{\partial x} \delta x \right)^2 + \cdots + \left( \frac{\partial q}{\partial z} \delta z \right)^2} \tag{12}$$

RESULTS

The control of the set-up for these measurements was done with a LabVIEW program, but the program didn’t run autonomously but was monitored by the users. This was done in order to be familiar with the set-up and how it would react to various stimuli. With these initial measurements, the positioning of the K-type thermocouples and Pt100 temperature sensors, as well as the correct functioning of the pressure transducers and mass flow meters could be verified. Unfortunately, the initial measurements prove that the K-type thermocouples are not attached well and the wall measurements could not be obtained. Achieving steady state is another difficulty that was not completely resolved. Experiments at near steady state are performed as the set-up was manually controlled and not completely PID-controlled which lead to inaccuracies in the measurements.

Heat transfer coefficient

Figure 6 shows the obtained Nusselt numbers for the tube-in-tube heating of R125 at supercritical pressures in function of the bulk temperature of the working fluid. With increasing mass flow rates, an increase in the Nusselt number and thus the heat transfer coefficient is obtained. This is mainly due to the increased heat transferred at higher mass flow rates. With increasing bulk temperatures of the working fluid, the Nusselt numbers at mass flow rates of 0.19 kg/s and 0.25 kg/s show an increasing trend. This is not the case for the highest mass flow rate. As all the bulk temperatures of the organic working fluid remain the pseudo-critical temperature ($T_{pc} = 67.04 \, ^\circ C$ at 37 bar, $T_{pc} = 76.45 \, ^\circ C$ at 45 bar), no conclusion can be made relating the Nusselt number to the pseudo-critical temperature. More measurements at bulk temperatures above the pseudo-critical temperature are required.

This increase in Nusselt number at higher mass flow rate is also shown in Figure 7. It shows the Nusselt number in function of the Reynolds number. The Nusselt numbers ranged between 530 and 1061 with corresponding heat transfer coefficients of 969.82 J/(kg.K) and 1931.6 J/(kg.K). The measured overall heat transfer coefficient has a mean accuracy of 11.95 %. The corresponding mean accuracies of the convection heat transfer coefficient and the Nusselt number are respectively 12.01 % and 12.14 %.

Pressure drop

Figure 8 shows the frictional pressure drop in function of the working fluid inlet temperature at constant mass flow rates. At a constant mass flow rate, higher heating fluid temperatures are the dominant factor with respect to the pressure drop. Higher mass flow rates do not correlate to higher frictional pressure drops as seen in Figure 8. The inlet temperature of the working fluid also influences the frictional pressure drop. For heating fluid temperatures of 80 °C and 90 °C, an increase in inlet temperature of the working fluid leads to a decrease in the frictional pressure drop. This could be attributed to the lower viscosity at higher working fluid inlet temperatures but is not consistent for all the measurements, namely at a heating fluid of 100 °C.

CONCLUSION

Experimental data was obtained on the test set-up and the initial measurements are presented in the text. As the set-up is new, automatic control of the parameters was not yet possible because of the unknown dynamic behaviour. This dynamic behaviour is now documented and a suitable control is being developed. During the first measurements, it was soon discovered that the wall mounted thermocouples, that measure the wall temperature at eleven different points, are not attached.
well to the wall of the inner tube. These thermocouples are heavily influenced by the heating fluid and a new way of attaching has to be considered in further work. The temperature sensors at the in- and outlet of the test section measure the bulk temperature of the heating and working fluid respectively. However, due to the configuration they are placed in, reaching steady-state conditions takes longer. For measurements at steady state this is not a problem, but as the set-up was not yet automatically controlled, steady state was not completely achieved. This influenced on the accuracy of the measurements.

Now that a more thorough insight in the dynamic behaviour of the set-up is achieved, accurate control systems can be implemented for full automated control used for further research.

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List of Abbreviations

CO₂  Carbon Dioxide
DHT  Deteriorated Heat Transfer
EES  Engineering Equation Solver
EHT  Enhanced Heat Transfer
GWP  Global Warming Potential
HFC  Hydrofluorocarbon
HTC  Heat Transfer Coefficient
LMTD Logarithmic Mean Temperature Difference
N₂  Nitrogen
NHT  Normal Heat Transfer
ODP  Ozon Depletion Potential
ORC  Organic Rankine Cycle
PID  Proportional-Integral-Derivative
RMS  Root Mean Square

Roman characters

\[ A_{HT} \quad [m^2] \quad \text{heat transfer surface area} \]
\[ c_p \quad [kJ/(kg.K)] \quad \text{specific heat capacity} \]
\[ \bar{c}_p \quad [kJ/(kg.K)] \quad \text{mean specific heat capacity} \]
\[ d_{wf} \quad [mm] \quad \text{diameter of the working fluid installation} \]
\[ D_{hf} \quad [mm] \quad \text{diameter of the heating fluid installation} \]
\[ G \quad [kg/(m^2.s)] \quad \text{mass flux} \]
\[ h \quad [kJ/kg] \quad \text{enthalpy} \]
\[ L \quad [m] \quad \text{length} \]
\[ \dot{m} \quad [kg/s] \quad \text{mass flow rate} \]
\[ p \quad [bar] \quad \text{pressure} \]
\[ \Delta p \quad [bar] \quad \text{pressure drop} \]
\[ R \quad [K/m^2] \quad \text{heat transfer resistance} \]
<table>
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<th>Unit</th>
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<td>°C</td>
<td>temperature</td>
</tr>
<tr>
<td>ΔT</td>
<td>°C</td>
<td>temperature difference</td>
</tr>
<tr>
<td>q</td>
<td>W/m²</td>
<td>heat flux</td>
</tr>
<tr>
<td>Q</td>
<td>W</td>
<td>heat flow rate</td>
</tr>
<tr>
<td>U</td>
<td>W/(m².K)</td>
<td>overall heat transfer coefficient</td>
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**Greek characters**

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<tr>
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<td>[-]</td>
<td>emissivity</td>
</tr>
<tr>
<td>λ</td>
<td>W/(m.K)</td>
<td>thermal conductivity</td>
</tr>
<tr>
<td>μ</td>
<td>mPa.s</td>
<td>dynamic viscosity</td>
</tr>
<tr>
<td>ρ</td>
<td>kg/m³</td>
<td>density</td>
</tr>
<tr>
<td>σ</td>
<td>W/(m².K⁴)</td>
<td>Stefan-Boltzmann constant</td>
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**Subscripts**

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<td>adiabatic</td>
</tr>
<tr>
<td>air</td>
<td>air</td>
</tr>
<tr>
<td>b</td>
<td>bulk</td>
</tr>
<tr>
<td>conv</td>
<td>convective</td>
</tr>
<tr>
<td>crit</td>
<td>critical</td>
</tr>
<tr>
<td>f</td>
<td>frictional</td>
</tr>
<tr>
<td>hf</td>
<td>heating fluid</td>
</tr>
<tr>
<td>i</td>
<td>inner</td>
</tr>
<tr>
<td>in</td>
<td>inlet</td>
</tr>
<tr>
<td>ins</td>
<td>insulation</td>
</tr>
<tr>
<td>ins,r</td>
<td>insulating ring</td>
</tr>
<tr>
<td>ins,b</td>
<td>insulating blanket</td>
</tr>
<tr>
<td>loc</td>
<td>local</td>
</tr>
<tr>
<td>max</td>
<td>max</td>
</tr>
<tr>
<td>o</td>
<td>outer</td>
</tr>
<tr>
<td>out</td>
<td>outlet</td>
</tr>
<tr>
<td>pc</td>
<td>pseudo-critical</td>
</tr>
<tr>
<td>rad</td>
<td>radiative</td>
</tr>
<tr>
<td>ts</td>
<td>test section</td>
</tr>
<tr>
<td>w</td>
<td>wall</td>
</tr>
<tr>
<td>wf</td>
<td>working fluid</td>
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**Dimensionless numbers**

- $E_0$ [-] dimensionless criterion
- $Nu$ [-] Nusselt number
- $Ra$ [-] Rayleigh number
- $Re$ [-] Reynolds number
- $Pr$ [-] Prandtl number
Chapter 1

Introduction

The total primary energy consumption across the globe has increased with 99% in the last 40 years and is expected to keep increasing in the years to come. In 2013 the industry accounting for about 29% of the total primary energy consumption [12]. Each process involves losses and statistical investigations show that up to 20 - 50% of the energy input is lost as waste heat [13, 14] with low-grade waste heat accounting for up to 50% of the heat loss [15]. If this energy in the form of low grade waste heat is recovered, it would decrease the primary energy demand in industry significantly. Several cycles have been developed to recover the energy in the low grade waste heat, with the transcritical ORC as the most promising.

This thesis investigates the supercritical heat transfer phenomenon with organic working fluids in large diameter tubes. There is not a lot of academic research to be found, as the main working fluids are either water, CO$_2$ and helium. This is partly because they have well documented properties, but also because they are readily available.

The main goal of this thesis is to develop a new correlation for supercritical heat transfer to organic working fluids in large diameter tubes. Therefore, a literature study is conducted in order to gain more insight in the subject. Two important researches are being treated in depth, namely the work of Garimella and Zhao et al. To be able to conduct measurements at supercritical conditions, the iSCORe test set-up is built at Ghent University. The iSCORe test set-up simulates a transcritical Organic Rankine Cycle with a test section of 4 m long. It is equipped with multiple temperature and pressure sensors to have overall and local heat transfer measurements.

After that, the control of the test set-up is discussed. This is necessary as several variables have to be controlled simultaneously. A program is written in LabVIEW, with PID controls for each component that needs to be controlled.

As we come to chapter 5, things get interesting. This chapter explains how all collected data are transformed to Nusselt numbers and heat transfer coefficients.
A measurement is only as good as the accuracy with which it has been measured. Chapter 6 explains the uncertainty propagation method and applies it to the equations used in Chapter 5. This step is done to be able to interpret the collected data with a critical point of view.

Arriving at chapter 7, results are discussed of the initial measurements and compared to other correlations from the literature. The Wilson plot method is used and explained, together with the modified Wilson plot method that is needed to develop a new correlation. Lastly, chapter 8 concludes this masters thesis.
Chapter 2

Literature study

The world’s demand for energy increases every year and will not likely stop in the near future. While the need for more energy increases, the principal energy sources, such as fossil fuels, are finite and diminishing in size. This leaves three options for the future. The first option is increasing the efficiency of the energy use on global level. Secondly, another possibility is decreasing the energy use while the third option is to investigate new potential primary resources and developing new technologies. Therefore, in the recent years many researchers focus on finding alternative technologies to utilize renewable energy sources to provide power in the future. Solar and wind energy have emerged as environmental friendly energy sources, but also biomass and geothermal are another form of (clean) energy sources. Therefore, multiple ways to harness this energy have been investigated. The list consists of but is not limited to Stirling engines, Organic Rankine Cycle (ORC), Kalina cycle and Trilateral Flash Cycle (Ziviani et al. [16]). The ORC, which is the focus of many researches today (Le et al. [17], Yu et al. [18], Melorose et al. [19], Kaya et al. [20]), is a very promising technology due to it’s relative simple and straightforward design compared to other thermodynamic cycles.

2.1 Organic Rankine Cycle

An ORC is a simplified thermodynamic cycle suitable to convert low grade temperature heat into usable work. In this cycle an organic fluid is used as working medium instead of water or steam like in a classical Rankine cycle. Organic working fluids typically have a much lower boiling point than water. Therefore low and medium temperature heat sources, as found in biomass, solar, geothermal and waste heat from various processes are considered as potential heat sources for this cycle. The industry accounts for about a third of the worldwide energy consumption and 20 - 50 % of the industrial energy input is lost as waste heat. This makes low grade heat from the industry an abundant heat
source that can be recovered in an ORC (Le et al. [17]).

The simplified layout for an ORC is seen in Figure 2.1. The working fluid is compressed to a higher pressure, heat is added to the working fluid in the heat exchanger called the evaporator. After that the working fluid is expanded to a lower pressure using an expander (turbine, inverted compressor), and it is (sub-)cooled in the condenser. The expansion work is recuperated by attaching an electric generator to the shaft of the expander. In many experimental set-ups and applications, such as heat pumps and refrigerators, an expansion valve is used instead of an expansion machine. In the new iSCORe set-up built at Ghent university [21], an expansion valve is used to simulate the Organic Rankine Cycle because the aim is not to recover the energy in the expander, but to better understand and develop a new correlation for supercritical heat transfer in large diameter tubes. The main focus of this study is low grade heat recovery within temperature below 180 °C.

2.1.1 Organic Rankine Cycle Classification

The ORC can be subdivided in three categories based upon where the heat addition and heat rejection is situated in the T-S diagram. These three categories are the sub-critical, supercritical and transcritical ORCs and they are shown in Figures 2.2, 2.3 and 2.4. The subcritical ORC heats up and cools down working fluids through the two phase region at subcritical pressures.
2.1. ORGANIC RANKINE CYCLE

While a subcritical ORC has two phase flow, the supercritical ORC adds and rejects heat at supercritical pressures, thus avoiding this multiphase region (Feher [22]).

Similar to the supercritical cycle, in a transcritical cycle heat is added at supercritical pressure while the condensation takes place in the two-phase region as in the subcritical cycle. This has the advantage of a better thermal match between the temperature glide of the heat source and the working fluid in the heat exchanger (evaporator, vapour generator, boiler, etc.). In transcritical cycles the operational pressure and temperature of the working fluid in the heating process must be above the critical point, which results in higher requirements for the materials used compared to subcritical cycles.
2.2. WORKING FLUIDS USED IN TRANSCRITICAL ORCS

There are four processes in a Rankine cycle, identified by numbers in Figures 2.2, 2.3, and 2.4. The focus for this thesis is performing heat transfer measurements to supercritical fluids in the heat exchanger (test section) in the new built test set-up. The iSCORe test set-up actually represents a transcritical ORC. Therefore, in the text below only the details for a transcritical cycle are explained.

Process 1-2': Isentropic compression of the working fluid. This is done by the pump and the temperature of the working fluid increases slightly.

Process 2'-3: Isobaric heat addition to the working fluid. This process is performed in the evaporator and the amount of the heat added to the working fluid depends on the heat transfer area and the temperature difference between the heat source and the working fluid.

Process 3-4': Isentropic expansion of the working fluid. In the expander (turbine, inverted compressor), the high pressure fluid (vapor) is expanded and is converted into useful work.

Process 4'-1: Isobaric heat rejection of the working fluid. The amount of heat rejected by the working fluid in the condenser depends on the heat transfer area and the temperature difference between the cold source and the working fluid.

In reality these processes are not ideal and have losses such as pump losses that result in shift from 2' to 2, a small pressure drop in the heat exchangers due to irreversibilities (not shown) and a non-ideal expander that result in a shift from 4' to 4 (due to irreversibilities).

2.2 Working fluids used in transcritical ORCs

Multiple working fluids are available for use in transcritical ORC. To be eligible for use in a transcritical ORC only a few conditions apply. The critical temperature must be
lower than the temperature of the heat source and the condensation temperature must be higher than the cold source. This implies that CO$_2$ is not likely to be used in real cycles because of its low critical temperature, $T_{cr} = 30.98 \, ^\circ C$. This temperature can be achieved during summer days which means that sub-cooling the fluid becomes very difficult. For operational safety reasons the pressure of the working fluid should be as low as possible. Thus working fluids with a low critical pressure are desired. The aim of this study is to focus on low grade heat with temperatures between 100 °C and 180 °C. However, the critical pressure and temperature of the fluid are not the only criteria for choosing the working medium. Economic and environmental factors also play a major role in the selection process. A working fluid may have a very desirable characteristic, nevertheless if that new fluid is very costly, it might be more interesting to purchase a cheaper alternative. There is a big interest from companies producing organic fluids to make new fluids and replacements for the existing ones with more desirable characteristics. Many researches focus on environmental friendly Hydrofluorocarbons (HFC) with no Ozon Depletion Potential (ODP) and a low Global Warming Potential (GWP) for ORCs. The ODP is the relative amount of damage a chemical substance could do to the ozon if it is released in the atmosphere, while GWP is the relative amount of heat the chemical substance captures when released in the atmosphere. Preferably a non flammable fluid is chosen with good thermophysical characteristics that is compatible with the heat source operating conditions.

An overview of frequently used working fluids in ORCs can be found in Table 2.1. The fluids can be used in sub- and transcritical cycles, but require different temperatures for the heat source and cold source. For determining the thermophysical properties of the fluids, CoolProp [23] was used.

<table>
<thead>
<tr>
<th>Working fluid</th>
<th>$T_c$ [K]</th>
<th>$p_c$ [MPa]</th>
<th>Cycle</th>
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<tr>
<td>R125</td>
<td>339.17</td>
<td>3.62</td>
<td>Trans [24]</td>
</tr>
<tr>
<td>R22</td>
<td>369.30</td>
<td>4.99</td>
<td>Trans [25]</td>
</tr>
<tr>
<td>R134a</td>
<td>374.21</td>
<td>4.06</td>
<td>Sub/Trans [25]</td>
</tr>
<tr>
<td>R227ea</td>
<td>374.90</td>
<td>2.93</td>
<td>Sub/Trans [25]</td>
</tr>
<tr>
<td>R152a</td>
<td>386.41</td>
<td>4.52</td>
<td>Sub/Trans [25]</td>
</tr>
<tr>
<td>R236fa</td>
<td>398.07</td>
<td>3.20</td>
<td>Sub/Trans [25]</td>
</tr>
<tr>
<td>R245fa</td>
<td>427.01</td>
<td>3.65</td>
<td>Sub/Trans [25]</td>
</tr>
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</table>

Different conditions and cycles require various working fluids for optimum efficiency. Vivian et al. [25] concludes that the temperature difference between the inlet of the heat source and the critical temperature of the working fluid, $\Delta T$, influences the choice of the working fluid for optimum efficiency. For a subcritical ORC three categories are defined
based upon the temperature difference $\Delta T$. Fluids with high temperature difference ($<55 \, ^\circ C$) should be excluded, fluids with a negative temperature difference are sub-optimal and fluids with temperature difference close to 35 $^\circ C$, are the optimal choice.

The working fluid selection for transcritical ORCs is also heavily influenced by the temperature difference $\Delta T$, but the optimum efficiency can be achieved in a broader region of pressures and temperatures. When the temperature difference is between 35 $^\circ C$ and 55 $^\circ C$ the optimum is achieved, while having $\Delta T$ smaller than 35 $^\circ C$ does not improve significantly over the subcritical ORC [25]. A general selection criteria for the working fluid is that the critical temperature should be around 0.8 times the temperature of the heat source [26].

### 2.3 Thermophysical characteristics of supercritical fluids

A supercritical fluid is a fluid at a temperature and pressure above it’s critical point. At supercritical state the fluid has a combination of liquid-like and gas-like properties that depend on the pressure and temperature. The different phases of a working fluid can be seen in Figure 2.5. A working fluid that is heated at a constant supercritical pressure from a sub-cooled liquid to a supercritical fluid, undergoes no real phase change as there is no real distinction between the phases in the supercritical region (see Figure 2.4).

![Figure 2.5: Phase diagram of CO$_2$](image)

Supercritical fluids have always occurred in nature like in underwater volcanoes and some planetary atmospheres, such as Venus, Jupiter and Saturn. Only in the mid 1800’s sci-
2.4. HEAT TRANSFER TO FLUIDS AT SUPERCRITICAL PRESSURES

Scientists started to process minerals in water at or above the critical point. This process, called hydrothermal processing, is still used today to produce high quality crystals [27]. Later on in the 1950’s supercritical water was considered to be used in nuclear reactors but was abandoned with the emergence of so called Light Water Reactors. The idea of using supercritical water is coming into focus again due to high thermal efficiency and simpler design [28]. Another working fluid that is commonly used is supercritical \( \text{CO}_2 \). This is used in enhanced oil recovery in mature oil fields and supercritical fluid extraction [29]. More recently organic working fluids are studied to be used at supercritical conditions, e.g. R134a as a working fluid in a transcritical ORC. To optimize performance of these processes a good understanding of the conditions at supercritical state is required.

The thermophysical property variations of R134a (\( p_{\text{crit}} = 4.06 \text{ MPa}, T_{\text{crit}} = 374.1 \text{ K} \)) at supercritical pressures are depicted in Figure 2.6. The specific heat capacity shows a sharp peak around a temperature that is defined as the pseudo-critical temperature, \( T_{\text{pc}} \). Other properties such as thermal conductivity, density and viscosity also show a large change in the vicinity around \( T_{\text{pc}} \). This pseudo-critical temperature increases and the change in thermophysical properties is less pronounced with increasing pressures [28]. Modelling of the transport properties of refrigerants R125, R134a, R32 and R125+R32 mixtures has been done by Kiselev et al. [30]. The models take into account the rapidly changing properties around the critical point and the smoothing out at regions away from this point. They achieved relative deviations of 10 % when comparing the thermal conductivity to the experimental results. The model for the thermal conductivity has been integrated into the Engineering Equation Solver \( \text{EES-program} \) [31].

2.4 Heat transfer to fluids at supercritical pressures

When discussing heat transfer to supercritical fluids, specific terms and expressions are used for these conditions. These terms include, but are not limited to: Enhance Heat Transfer (EHT), Deteriorated Heat Transfer (DHT), Normal Heat Transfer (NHT), pseudo-boiling and pseudo-film boiling. Figure 2.7 illustrates these various terms.

**NHT** has wall heat transfer coefficients comparable with those of subcritical convective heat transfer, calculated with the single phase Dittus-Boelter-type correlations (Eq. 2.3). **EHT** has wall heat transfer coefficients higher than the NHT and **DHT** has wall heat transfer coefficients lower than the NHT. Pseudo-boiling is a phenomenon similar to subcritical nucleate boiling but at supercritical pressures. Some layers of the fluid may attain temperatures higher than \( T_{\text{pc}} \), while the bulk temperature of the fluid is still below \( T_{\text{pc}} \). Pseudo-film boiling is similar to subcritical film-boiling at supercritical pressures. A fluid with temperatures above \( T_{\text{pc}} \), which is a low-density gas, prevents a fluid with temperatures below the \( T_{\text{pc}} \), which represents a high-density liquid, to contact the heating surface [28].
In the literature, there are many heat transfer correlations derived for supercritical fluids. Mainly, these heat transfer correlations are of Gnielinski (2.1, 2.2) and Dittus-Boelter (2.3) type and were derived for working fluids such as water, CO$_2$, helium, etc. Furthermore, these heat transfer correlations have been derived for different working conditions than those of ORCs but nevertheless, they are of high importance. In order to derive heat transfer correlations that can be applied for designing a heat exchanger suitable to operate at supercritical conditions in an ORC cycle, further research is required.

$$Nu_D = \frac{(f/8)(Re_D - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)}$$ \hspace{1cm} (2.1)

$$f = (0.79 \ln(Re_D) - 1.64)^{-2}$$ \hspace{1cm} (2.2)

$$Nu_D = C \cdot Re_D^m Pr^n$$ \hspace{1cm} (2.3)

As already mentioned in the text, the first investigations of supercritical heat transfer were performed to working fluids such as water, CO$_2$ and helium (Durham [32], Bourke et al. [33], Section [34]). Oh and Son [7] investigated the cooling of supercritical CO$_2$ in a smooth horizontal tube-in-tube heat exchanger with an inner diameter of 4.55 mm and 7.75 mm and with lengths of 4000 mm and 6000 mm respectively. This research was conducted to find the heat
2.4. HEAT TRANSFER TO FLUIDS AT SUPERCRITICAL Pressures

Figure 2.7: Temperature and heat transfer coefficient profiles along heated length of vertical circular tube (Kirillov et al., 2003): Water, D=10 mm and Lh=4 m

The operating pressures are between 7.5 MPa and 10 MPa ($T_{crit} = 304.15$ K, $p_{crit} = 7.38$ MPa) and inlet temperatures of the fluid vary between 90 °C and 100 °C. Mass flux ranges from 200 - 600 kg/m²s for the smaller diameter tubes, $d_i = 4.55$ mm, and between 200 - 400 kg/m²s for the larger diameter tubes. This study makes a comparison of the experimental data with the work of others such as Yoon et al. [35] and X. et al. [36]. This heat transfer correlations are found in Table 2.5 and were chosen for the similar conditions as the experiments conducted by Oh and Son. When compared, these equations both under predicted the experimental data and had a mean deviation of 29.4 % and 26.8 % respectively. These variations are explained by the lack of data fitting for Fang, since the heat transfer correlation was based upon numerical simulations while Yoon’s correlation did not take into account the critical enhancement of the specific heat. The proposed correlation is based on the Dittus-Boelter (2.3) correlation with the addition of two terms: the density ratio $\rho_b/\rho_w$ and the specific heat ratio...
2.4. HEAT TRANSFER TO FLUIDS AT SUPERCRITICAL PRESSURES

This density ratio takes into account the density gradient between the wall and the bulk of the fluid and the buoyancy effect that accompanies this density gradient. The heat transfer correlation that is proposed is subdivided for bulk temperatures above and below the pseudo-critical temperature and can be found in Table 2.5. Small differences exist for the two regions and are given in Table 2.2.

Table 2.2: Coefficients for Oh and Son heat transfer correlation

<table>
<thead>
<tr>
<th>C</th>
<th>n1</th>
<th>n2</th>
<th>n3</th>
<th>n4</th>
</tr>
</thead>
<tbody>
<tr>
<td>T_b/T_{pc} &gt; 1</td>
<td>0.023</td>
<td>0.7</td>
<td>2.5</td>
<td>0</td>
</tr>
<tr>
<td>T_b/T_{pc} ≤ 1</td>
<td>0.023</td>
<td>0.6</td>
<td>3.2</td>
<td>3.7</td>
</tr>
</tbody>
</table>

Gupta et al. [8] conducted experiments on heating of supercritical CO\textsubscript{2} in a vertical tube with an inner diameter of 8.06 mm and with a length of 2400 mm. Only 2280 mm was heated by an electrical current that flows through the tube. The pressure varied between 7.57 MPa and 8.8 MPa and mass fluxes varied between 900 kg/m\textsuperscript{2}s and 3300 kg/m\textsuperscript{2}s. With an electrical heater the amount of heat added to the test section could be changed easily and varied between 9.3 kW/m\textsuperscript{2} and 616.6 kW/m\textsuperscript{2}. The proposed correlation was produced using the Buckingham Π-Theorem and is similar to the Dittus-Boelter correlation with the addition of three terms: the thermal conductivity ratio \( \lambda_w/\lambda_b \), the viscosity ratio \( \mu_w/\mu_b \) and the density ratio \( \rho_w/\rho_b \). These terms are added to account for the property variations that can occur due to the differences in temperature on the wall and bulk of the working fluid. The heat transfer correlation can be found in Table 2.5. The coefficients for the correlation are different depending on what characteristic temperature is used, which is the temperature to evaluate the majority of the thermophysical properties. Three different approaches are used, the classical bulk fluid temperature \( T_b \), the wall temperature \( T_w \) and the combination of both, the film temperature \( T_f = (T_w + T_b)/2 \). The reasoning behind this is that at higher heat fluxes the wall and bulk fluid temperature may differ. Table 2.3 shows the different coefficients. The biggest difference is the absence of the viscosity ratio when calculating the Nusselt's number with the film temperature approach. The most accurate approach is the wall temperature approach with the lowest root mean square (rms) errors for the Heat Transfer Coefficient (HTC). The mean errors for the HTC are 0.9 %, 0.8 % and 0.2 % for the bulk temperature, wall temperature and film temperature approach respectively, when compared to the experimental results. The RMS errors are 22.4 %, 20.3 % and 21.7 % respectively. The correlations of Oh and Son and Gupta et al. are devised for cooling and heating respectively. Both of them are based on the Dittus-Boelter correlations and show similarities to each other. The same working fluid was used and the pressures are in the same region. Oh and Son used a specific heat correction factor while Gupta et al. used a thermal conductivity and viscosity correction.
Table 2.3: Coefficients for Gupta et al. with different approaches

<table>
<thead>
<tr>
<th>Approach</th>
<th>C</th>
<th>n₁</th>
<th>n₂</th>
<th>n₃</th>
<th>n₄</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bulk fluid temperature</td>
<td>0.01</td>
<td>0.89</td>
<td>-0.14</td>
<td>0.93</td>
<td>0.22</td>
</tr>
<tr>
<td>Wall temperature</td>
<td>0.0038</td>
<td>0.96</td>
<td>-0.14</td>
<td>0.84</td>
<td>-0.75</td>
</tr>
<tr>
<td>Film temperature</td>
<td>0.0043</td>
<td>0.94</td>
<td>0</td>
<td>0.57</td>
<td>-0.52</td>
</tr>
</tbody>
</table>

factor. Gupta et al. calculated the correlation using the Buckingham Π-Theorem that produces the Nusselt number in function of several dimensionless terms. Oh and Son started from the basic Dittus-Boelter equation and corrected it for use in the supercritical region. The density ratio is also more pronounced in the correlation by Oh and Son due to the buoyancy effect that has a bigger impact in horizontal tubes than in vertical tubes.

2.4.1 Supercritical heat transfer to organic fluids

R-410A

The literature is far less extensive for supercritical heat transfer to organic fluids than for water, CO₂ or helium. Garimella [9] investigated the cooling of sub- and supercritical R-410A, an azeotropic mixture of 50 % R32 and 50 % R125, \((T_{\text{crit}} = 345.95 \, \text{K}, p_{\text{crit}} = 4.86 \, \text{MPa})\) in smooth horizontal tubes. Three different tube diameters are investigated, where the two smallest are multiport test sections. The diameters are 0.76 mm, 1.52 mm and 3.05 mm and the heat transfer lengths are 3048 mm, 3048 mm and 1524 mm respectively. The 0.76 mm diameter tube has 17 ports while the 1.52 mm tube has 10 ports. Three different pressures are chosen, \(1.0 \times p_{\text{crit}}, 1.1 \times p_{\text{crit}}\) and \(1.2 \times p_{\text{crit}}\). The flow in the supercritical region is subdivided in separate flow regimes, defined by a dimensionless criterion \(E_0\), which represents the ratio of the flow work to the heat convected out during the cooling process. Three regions are defined: the liquid-like region, the pseudo-critical region and the gas-like region. The division in flow regimes is used as a base to develop the pressure drop. The friction factor is a variation to the Churchill friction factor, which accounts for the changes in bulk-to-wall properties, and has different coefficients defined per flow regime. They are found in Table 2.4. The heat transfer is a variation on the Churchill’s Nusselt number and is divided in two regimes depending on the temperature with different coefficients (see Table 2.4). The heat transfer correlation is multiplied with a \(R_{\text{bulk}}\) term to account for the increased Nusselt number and lower Reynolds number at lower diameters. The dependence of the experimental data to the \(R_{\text{bulk}}\) can be seen in Figure 2.8. In a smaller diameter the dependence of the Nusselt number to the Reynolds number changes. Gravitational effect is less significant in smaller tubes leading to a flow similar to annular flow. The correlations are found in Table 2.5.
2.4. HEAT TRANSFER TO FLUIDS AT SUPERCRITICAL PRESSURES

### Table 2.4: Coefficients for Garimella

| T < \( T_{pc} \) | 0.56 | 0.022 | 0.010 |
| T > \( T_{pc} \) | 0.19 | 0.118 | 0.011 |

(a) Coefficients for Nusselt’s number

| \( C_2 \) | 1.16 | 1.31 |
| \( n_3 \) | 0.91 | 0.25 |

(b) Coefficients for friction factor

The mean deviation for the friction factor varies from 16 %, in the liquid-like and gas-like region, to 22 % in the pseudo-critical region. The mean deviation for the heat transfer coefficients are 22 % and 23 % for temperatures lower and above than \( T_{pc} \) respectively.

### R134a

The cooling of supercritical R134a in horizontal tubes is investigated by Zhao and Jiang [2]. A smooth horizontal tube of 1120 mm long with an inner diameter of 4.01 mm is used. The heat transfer section is 1000 mm long with adiabatic sections before and after of 60 mm each. The cooling is done by a low viscosity silicon oil flowing counterflow. The total pressure drop is calculated by the sum of the frictional pressure drop, the adiabatic pressure drop and the pressure drop from the fluid contractions due to the density variations during cooling.

\[
\Delta p_{\text{total}} = \Delta p_{\text{ad,in}} + \Delta p_{\text{ad,out}} + \Delta p_f + \Delta p_{\text{con}}
\]  

The density variations play a major role in the frictional pressure drop. Density decreases with increasing bulk temperature and the average velocity increases to maintain a continuous flow. Increasing velocities with rising temperatures mean an increase in frictional pressure drop with elevating temperature. Four different heat transfer correlations are compared to the experimental results of the heat transfer coefficients: Gnielinski, Pitla et al., Dang and Hihara and Yoon.

Figure 2.9 shows the experimental results and the various correlations at two different pressures. For temperatures below the pseudocritical temperature Gnielinski, Pitla, and Dang and Hihara’s heat transfer correlations predict the experimental values well. But Yoon’s heat transfer correlation overestimates all the experimental values, while Gnielinski underestimates the values slightly at temperatures \( T_b > T_{pc} \). Even though Pitla and Dang and Hihara’s heat transfer correlation are derived for a working fluid such as CO\(_2\), they have a relatively good approximation for the experimental results with R134a. Nevertheless these heat transfer correlations have a correction factor that over-corrects the property variations of R134a.
Figure 2.8: Experimental Nu versus Re$_{\text{bulk}}$ (Garimella 2013)
The proposed correlation for the heat transfer coefficient is designed with the axial and radial property variations in mind. Based upon Gnielinski’s correlation it has extra terms to compensate the axial and radial property variations. The two extra components are the density ratio, \( \rho_w/\rho_b \), representing the radial variation and the specific heat capacity ratio \( \bar{c}_p/c_{p,b} \), which expresses the axial variation. The mean specific heat capacity is defined as the fraction of the enthalpy difference to the temperature difference at the in- and outlet:

\[
\bar{c}_p = \frac{h_{in} - h_{out}}{T_{wf,in} - T_{wf,out}}
\]  

Comparing the equation to the experimental dataset shows that 90 % of this data can be predicted with an accuracy of 15 %, but no mean deviation was given for the results. The correlations can be found in Table 2.5.

Figure 2.9: Comparison of experimental HTC and various correlations at different pressures (Zhao 2011)
R125

The organic working fluid R125 is often compared to other organic fluids in theoretical studies regarding transcritical cycles. The relatively low critical point, perfect for usage in combination with low temperature heat sources, together with the commercial availability, makes R125 an interesting fluid to be used in transcritical cycles.

Shengjun et al. [37] compared several organic working fluids, including R125, in subcritical and transcritical ORC power cycles for low temperature geothermal power generation. He concluded that R125 is preferable over the other organic working fluids because it showed excellent economical and environmental performance while maximizing the usage of the geothermal power. Even though R123 in a subcritical ORC has a better thermal efficiency and exergy efficiency, R125 provides 20.7 % larger recovery efficiency.

Baik et al. [38] made a comparative study of power optimization between a R125 transcritical cycle and other HFC organic Rankine cycles with low-temperature geothermal heat source of about 100 °C. He found that R125 is the best choice when the total overall conductance, \((UA)_v + (UA)_c\), is larger than 35 kW/K compared to subcritical cycles. This was because the power output in the transcritical cycle was improved by a better thermal match between the heat source and working fluid during heating. Below 35 kW/K, subcritical cycles are favoured because the higher condensing temperature and larger pumping power, negated the increase in power in transcritical cycles.

In a previous article, Baik et al. [24], Baik et al. investigated the performances of CO2 and R125 transcritical cycles with a low grade heat source of 100 °C. He came to the conclusion that even though the carbon dioxide cycle has better heat transfer and pressure drop characteristics, the increased pumping power was detrimental to the power output of the carbon dioxide cycle. This lead to R125 producing 14 % more power and being the preferred working fluid when working with a temperature heat source of 100 °C.

All these researches are theoretical and need to be experimentally validated. To this date, this has not been done and neither is there any experimental work on supercritical heat transfer of R125.
### 2.4. HEAT TRANSFER TO FLUIDS AT SUPERCRITICAL Pressures

<table>
<thead>
<tr>
<th>Diam.</th>
<th>Correlation</th>
<th>Conditions</th>
<th>Ref.</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.55 - 7.75 mm</td>
<td>$Nu = CRe^a \frac{Pr^a}{Pr_{w}^{a/2}} \left( \frac{Diam}{c_p \nu} \right)^{a_4}$</td>
<td>Cooling : CO$_2$ G = 200-600 kg/m$^2$s</td>
<td>Oh and Son</td>
</tr>
<tr>
<td>4 mm</td>
<td>$Nu = 1.82 \log Re_w - 1.64^{-2}$</td>
<td>$p = 7.5-10$ MPa</td>
<td></td>
</tr>
<tr>
<td>7.73 mm</td>
<td>$Nu = CRe^a \frac{Pr^a}{Pr_{w}^{a/2}} \left( \frac{Diam}{c_p \nu} \right)^{a_4}$</td>
<td>Cooling : CO$_2$ G = 225-450 kg/m$^2$s</td>
<td>Yoon et al.</td>
</tr>
<tr>
<td>C = 0.14, $n_1 = 0.69$, $n_2 = 0.66$, $n_3 = 0$ for $T_b &gt; T_{pc}$ C = 0.013, $n_1 = 1.0$, $n_2 = -0.05$, $n_3 = 1.6$ for $T_b \leq T_{pc}$</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>N/A</td>
<td>$Nu = \left( \frac{f_{w}}{f}\left[R_{w} - \sqrt{R_{w}^{2} - 1}\right] \right) \left( 1 - 0.002 \left( \frac{\nu_{p}}{\nu} \right) \right)$</td>
<td>Cooling : CO$_2$ $Re_w = 3500-2.5 \times 10^4$</td>
<td>Fang</td>
</tr>
<tr>
<td>$f_w = 1.82 \log Re_w - 1.64^{-2}$</td>
<td>$\dot{q}_w/G = -115 - 3$ J/kg</td>
<td>J/kg</td>
<td></td>
</tr>
<tr>
<td>$A = 1 + 7.10^{-8}Re_w$ for $Re_w &lt; 10^{-6}$</td>
<td>$T_f = (T_b + T_w)/2$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$A = 1.07$ for $Re_w \geq 10^{-6}$</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1 - 6 mm</td>
<td>$Nu = \frac{(f_{w}/f)(R_{w} - 1000)^{0.1}}{1.07 + 12.7 (f_{w}/f) (Re_{w}^{2/3} - 1)}$</td>
<td>Cooling : CO$_2$ $Re_w = 7.5 - 10$ MPa</td>
<td>Dang and Hiara</td>
</tr>
<tr>
<td>$f_w = 1.82 \log Re_w - 1.64^{-2}$</td>
<td>$\dot{q}_w/G = -115 - 3$ J/kg</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$Pr = \frac{c_p \mu_b / \lambda_b}{c_p \mu_l / \lambda_l}$ for $c_p \mu_b &gt; c_p \mu_l$</td>
<td>$T_f = (T_b + T_w)/2$</td>
<td>Configuration of Pipes</td>
<td></td>
</tr>
<tr>
<td>$Pr = \frac{c_p \mu_b / \lambda_b}{c_p \mu_l / \lambda_l}$ for $c_p \mu_b &gt; c_p \mu_l$ and $c_p \mu_b &lt; c_p \mu_l$ and $\mu_b / \lambda_b \geq \mu_f / \lambda_f$</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$Pr = \frac{c_p \mu_b / \lambda_b}{c_p \mu_l / \lambda_l}$ for $c_p \mu_b &lt; c_p \mu_l$ and $\mu_b / \lambda_b &lt; \mu_f / \lambda_f$</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8.06 mm</td>
<td>$Nu = CRe^a \frac{Pr^a}{Pr_{w}^{a/2}} \left( \frac{Diam}{c_p \nu} \right)^{a_4}$</td>
<td>Heating : CO$_2$ G = 900-3300 kg/m$^2$s</td>
<td>Gupta et al.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.76 - 3.05 mm</td>
<td>$Nu_{mod} = \left[ 4.364^{10} + \left( \frac{exp(2200 - Re_{bulk})}{4.364^{10}} \right) \right] + \left( 6.3 + \frac{0.070 \left( Re_{mod} + Pr_{mod} \right)^{1/3} \left( Re_{mod} \right)^{-2}}{1 + \left( 1.07 \left( Re_{mod} \right)^{1/3} \right)^{10}} \right)^{1/10}$</td>
<td>Cooling : R410a : R404a G = 792.4-801.9 kg/m$^2$s</td>
<td>Garimella</td>
</tr>
<tr>
<td>$f_{mod} = C_2 \frac{\mu_{wall}^{0.0004}}{(Pr_{mod})^{1/3}}$</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$Nu = \left( \frac{f_{w}}{f}\left[R_{w} - \sqrt{R_{w}^{2} - 1}\right] \right) \left( 1 - 0.002 \left( \frac{\nu_{p}}{\nu} \right) \right)$</td>
<td>Cooling : R134a</td>
<td>Zhao and Jiang</td>
<td></td>
</tr>
<tr>
<td>4 mm</td>
<td>$f = 1.82 \log (Re_w - 1.64)^{-2}$</td>
<td>$p = 4.5-5.5$ MPa</td>
<td></td>
</tr>
<tr>
<td>$C_{Pr} = 0.93 \left( \frac{Pr_{w}}{Pr} \right)^{-0.111} \left( \frac{Re_{w}}{Re_{b}} \right)^{-0.094} \left( \frac{\nu_{p}}{\nu} \right)^{0.06}$ for $T_b \leq T_{pc}$</td>
<td>$T_{bc} = 80-140^\circ$ C</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$C_{Pr} = 1.07 \left( \frac{Pr_{w}}{Pr} \right)^{-0.45} \left( \frac{Re_{w}}{Re_{b}} \right)^{-0.28} \left( \frac{\nu_{p}}{\nu} \right)^{-0.18}$ for $T_b &gt; T_{pc}$</td>
<td>$Re_w = 4.10^{3}-8.10^{4}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Pr$_{bc} = 1.2-8.8$</td>
</tr>
</tbody>
</table>

The heat transfer correlations presented in the Table 2.5 are recently developed and are for organic fluids and CO$_2$. However, the list is not finite and can be extended with more literature correlations that were derived in the past as well.
2.5 Optimization of heat transfer and pressure drop

Heat transfer optimization in ORC is not straightforward since multiple parameters work together to influence the heat transfer. The focus of this study is the supercritical heat transfer in larger diameter tubes for actual ORC application. In this study an expansion valve is included in the test set-up and the optimization of the power output is not investigated. (More information about power optimization can be found in: Le et al. [17], Zabek et al. [29], Shengjun et al. [37], Ayachi et al. [41], Ran et al. [42]).

The diameter of the tube plays a significant role in the heat transfer process. In this study a subdivision of the diameters in three categories is made: micro (0 - 2 mm), mini (2 - 20 mm) and macro (20 - 40 mm). Several studies show that a decreasing tube diameter increases the HTC [7, 9], Figure 2.10a shows this effect for three different diameters at pressures 10 % higher than the critical pressure, $p_{\text{crit}}$. Due to the increase in heat transfer surface to flow area, smaller diameters can achieve a higher HTC. Figure 2.10b indicates that the pressure drop across the tube is inversely proportional with the diameter. At a given mass flux and decreasing diameter, the flow has to accelerate to maintain the mass balance. This increase in velocity causes a bigger frictional pressure drop.
2.5. OPTIMIZATION OF HEAT TRANSFER AND PRESSURE DROP

(a) Effects of D on h for different mass fluxes (Garimella 2013)  
(b) Effects of D on pressure drop for different mass fluxes (Garimella 2013)

Figure 2.10: Influence of diameter on HTC and pressure drop

2.5.1 Influence of mass flux on the heat transfer

An important parameter that influences the HTC is the mass flux. Figure 2.10a and Figure 2.11a show the change in HTC at different mass fluxes in mini tubes. The HTC varies proportional to the mass flux with all diameters, but the effect is the strongest in micro/mini tubes. A phenomenon similar to the two-phase annular flow may develop at high enough mass fluxes. In this case the colder, high-density fluid sits at the outer edge, while the warmer, low-density fluid is surrounded by the high-density fluid. This means an increase in the HTC and is only visible at small diameter where gravity does not effect the flow that much. HTCs in macro tubes are also proportional to the mass flux, but this relationship is not as strong as in mini/micro tubes as a result of buoyancy effects as a limiting factor. The pressure drop increases with increasing mass fluxes, as seen in Figure 2.11b. The conservation of mass ensures that with higher mass fluxes, higher velocities
occur and thus a larger pressure drop. This trend is the same for all diameters.

2.5.2 Influence of pressure on the heat transfer

The pressure also influences the HTC indirectly and locally around $T_{pc}$ due to the changes in thermophysical properties at varying pressures. Figure 2.12a displays that at higher pressures the peak of HTC is lower and shifted to higher temperatures, as expected. At temperatures far away from the $T_{pc}$ the pressure does not effect the HTC in a significant way. The frictional pressure drop decreases with increasing pressure if the temperature is around or above the pseudo-critical temperature. Below $T_{pc}$ no significant changes occur when varying the pressure, this can be seen in Figure 2.12b.
2.5. OPTIMIZATION OF HEAT TRANSFER AND PRESSURE DROP

2.5.3 Influence of heat flux on the heat transfer

The change in HTC is also affected by the heat flux. Wang et al. [43] conducted experiments on vertical tubes with water as a working fluid. A smooth tube with an inner diameter of 19.8 mm and an internally ribbed tube with an inner diameter of 17.63 mm are used. The structure is electrically heated along the length of the test section that is 2000 mm long and pressures between 22.5 MPa and 29.0 MPa are used. The mass flux varies from 650 kW/m$^2$ to 1200 kg/m$^2$s and the heat flux ranges from 200 kW/m$^2$ to 600 kW/m$^2$. Figure 2.13a demonstrates that higher heat fluxes correlate to higher inside wall temperatures. The only change in the HTC occurs around the pseudo-critical point. At lower heat fluxes and high enough mass fluxes, higher HTCs are found. This could be attributed to the energy input not being high enough to overcome the large values of specific heat of the fluid around the pseudo-critical temperature. If the heat flux is very high and the mass flux is low enough, DHT will occur, see Figure 2.13b.
2.5. OPTIMIZATION OF HEAT TRANSFER AND PRESSURE DROP

2.5.4 Influence of buoyancy on the heat transfer

When a supercritical fluid flows in a horizontal tube, buoyancy makes the lighter and hotter fluid rise up to the upper part of the tube, while the heavier and colder fluid will settle on the bottom part of the tube. The hotter and lighter fluid at the top will increase the heat resistance, while at the same time the fluid at the bottom will decrease the heat resistance. This translates into decreased HTC at the top and increased HTC at the bottom. The inside wall temperature will be different at the top and at the bottom of the tube \[3\]. This is indicated in Figure 2.14. The peak in temperature difference of the top and bottom coincides with slightly lower HTCs for the top surface. When the heat flux is high enough the fluid in contact with the top surface reaches the pseudo-critical point first, which is a possible explanation for the latter. Around this point a large reduction in thermal conductivity and density occurs, which decreases the heat transfer coefficient. This is counteracted by the change in specific heat capacity, which increases the amount...
of heat that can be stored in the fluid around this pseudo-critical point. In micro tubes the buoyancy has no noticeable effect on the flow and the HTC, as other parameters dominate the influence on the HTC. Mini tubes are lightly effected by buoyancy at high mass fluxes and low heat fluxes. If the ratio of heat flux to mass flux exceeds 0.4 kJ/kg, the top part of the tube experiences DHT \[3\]. For macro tubes this effect will most likely be the same as for mini tubes, but at different threshold values.

![Figure 2.14: Effect of buoyancy on the inside wall temperature and HTC](image)

In this study we have selected R125 as a working fluid because in a transcritical power cycle, R125 shows excellent economic and environmental performance and can maximize utilization of the low-grade temperature heat \[37\]. Another favourable attribute is the relatively low critical pressure and temperature. However, due to the high GWP it leaves room to be replaced by a more suitable working fluid with even better environmental properties.
### Table 2.6: Overview of influences on HTC

<table>
<thead>
<tr>
<th>mass flux</th>
<th>heat flux</th>
<th>buoyancy</th>
<th>pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>micro</td>
<td>HTC $\propto G$; annular flow at high mass fluxes</td>
<td>HTC $\propto 1/q$ for high mass fluxes; DHT at $q/G$ above a certain value</td>
<td>peak around $T_{pc}$ is lower at higher pressures</td>
</tr>
<tr>
<td></td>
<td></td>
<td>not a noticeable effect</td>
<td></td>
</tr>
<tr>
<td>mini</td>
<td>HTC $\propto G$; annular flow at high mass fluxes</td>
<td>HTC $\propto 1/q$ for high mass fluxes; DHT at $q/G$ above a certain value</td>
<td>peak around $T_{pc}$ is lower at higher pressures</td>
</tr>
<tr>
<td></td>
<td></td>
<td>a small effect is noticeable, mainly with high heat fluxes</td>
<td></td>
</tr>
<tr>
<td>macro</td>
<td>$G \propto HTC$; effect is smaller than in mini/micro tubes as buoyancy has a big effect on heat transfer</td>
<td>Buoyancy effects will cause DHT at $q/G$ above certain values in top part of the tube but not in bottom part</td>
<td>(large) differences in wall temperature in top and bottom part of the tube; results in different HTC for top and bottom part of the tube</td>
</tr>
</tbody>
</table>
2.5. OPTIMIZATION OF HEAT TRANSFER AND PRESSURE DROP
Chapter 3

Test set-up iSCORe

3.1 Test set-up

The test set-up on which experiments are conducted is iSCORe, built at Ghent University [21], with the aim of developing new heat transfer correlation for large diameter tubes. This heat transfer correlation is going to be used for designing a heat exchanger in a transcritical ORC. Figure 3.1 shows a schematic representation of the test set-up, which consists of three loops: an experimental loop, a heating loop and a cooling loop (see also Figure 3.2). Data acquisition hardware, such as CompactRIO from National Instruments and Keithley from Tektronix, are used to monitor and control different parameters on the set-up (see Chapter 5). The test set-up is designed for high accuracy measurements over a wide operating range and is considered to be modular as the inner tube of the test section can be replaced to allow measurements with different conditions (e.g. different tube diameters, different tube materials, corrugated tubes, ...).

The design conditions of the test set-up/test section are the following:

- Horizontal heat transfer
- Working pressure up to 50 bar
- Maximum temperature of the thermal heating fluid up to 180 °C
- Organic working fluids (see Table 3.1 for properties)
  - HFC-125
  - HFC-134a
- Materials used for the test section tube-in-tube
3.1. TEST SET-UP

Figure 3.1: Schematic of test set-up

Figure 3.2: Simplified schematic of the fluid loops in the test setup [4, 5]
3.1. TEST SET-UP

- Copper alloy tube for the organic working fluid flow ($p_{\text{max}} = 120$ bar, $T_{\text{max}} = 200$ °C)
- Galvanized steel tube for thermal oil flow

- **Diameters**
  - Copper alloy tube inner diameter: 24.77 mm
  - Copper alloy tube outer diameter: 28.57 mm
  - Galvanized tube steel inner diameter: 56.65 mm
  - Galvanized tube steel outer diameter: 60.3 mm

- Maximum power of the pump: 5.5 kW
- Maximum power of the electrical preheater: 10 kW
- Maximum power of the thermal oil heater: 20 kW
- Length of the test section: 4 m
- Length of the first tube-in-tube preheater: 4 m
- Length of the second tube-in-tube preheater: 4 m

<table>
<thead>
<tr>
<th>Property</th>
<th>HFC125</th>
<th>HFC134a</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chemical formula</td>
<td>C$_2$HF$_5$</td>
<td>CF$_3$CH$_2$F</td>
</tr>
<tr>
<td>$T_{\text{crit}}$</td>
<td>66.02 °C</td>
<td>101.03 °C</td>
</tr>
<tr>
<td>$p_{\text{crit}}$</td>
<td>3.618 MPa</td>
<td>4.060 MPa</td>
</tr>
<tr>
<td>ODP</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>GWP (100y)</td>
<td>3500</td>
<td>1430</td>
</tr>
</tbody>
</table>

**Cooling loop**

The cooling loop provides the cold source for the organic working fluid flowing through the condenser. A 70%/30% volumetric ratio of a water/glycol mixture is used with a 900 l buffer vessel. This vessel is cooled by a 37 kW chiller and the water that leaves the chiller has a temperature of 5 °C. The flow rate through the condenser is electronically controlled by a 3-way mixing valve and implementation of this control is done by a PID in LabVIEW (see 4.1).
3.1. TEST SET-UP

Heating loop

The heating loop provides the (sensible) heat source for the organic working fluid flowing through the preheaters and the test section. The preheaters can be bypassed, either one or both at the same time, so that the preheating length can be varied from 0 m to 8 m in steps of 4 m. Thermal oil is pumped through the heating circuit using a built in circulation pump and is heated inside the thermal oil unit with two 10 kW heating elements (a total capacity of 20 kW). The flow rate of the thermal oil is electronically controlled by a 3-way valve and is also implemented by a PID in LabVIEW (see 4.1).

Experimental loop

The experimental loop, where the organic working fluid circulates, consists of a pump, three preheaters, a test section, a condenser and an accumulator. The pump is a diaphragm pump with 5 integrated diaphragms and a maximum inlet pressure of 35 bar.
Driven by a frequency controlled induction motor of 5.5 kW, it is pressure limited. A premium quality pressure sensor is installed downstream of the pump that is connected to a panel meter, which can shut down the pump if the pressure exceeds a certain limit. Mass flow rate is a dependent on the pump characteristics. Two tube-in-tube heat exchangers and an electrical preheater are present. The tube-in-tube preheaters are fed with thermal oil from the heating circuit and can be bypassed as previously mentioned. An electrical preheater is included because this allows a more accurate control of the test section inlet temperature. The test section will be discussed in more detail in 3.2. The condenser, a plate heat exchanger, subcools the organic working fluid before it enters the pump to prevent gas bubbles at the pump inlet. An accumulator is needed to account for the volumetric expansion of the organic working fluid from start-up of the test set-up to the test conditions. It contains two fluids, pressurised nitrogen (N\textsubscript{2}) and the organic working fluid, that are separated by a diaphragm.

More information about the design of the test set-up can be found in the thesis of Daelman \cite{5} and of Bauwens \cite{4}.

### 3.2 Test section

The test section is the key component in this set-up. In the design process heat transfer correlations from literature such as Petukhov \cite{44}, Swenson \cite{45} and Shitsman \cite{46} were used in EES environment. These heat transfer correlations are listed below (3.1a - 3.3a). In order to have as large as possible temperature difference over the test section, whilst keeping the dimensions of the set-up within reasonable range, a length of 4 m is chosen from the calculations \cite{5}. A counterflow tube-in-tube test section is chosen over a joule (electrically) heated test section. This allows for more accurate measurements as the heating of the working fluid is more uniform over the test section with a tube-in-tube heat exchanger. It also approaches the heat transfer that would take place in the actual heat exchangers of an ORC in a better way. The wall temperature might be higher than practical applications with an electrical heater influencing the convective heat transfer of the organic working fluid side. Joule heating could also influence the thermocouples attached to the tube, even though they are electrically insulated. For all these reasons a tube-in-tube test section is chosen and is depicted in Figure 3.5.

The heat transfer correlation of Petukhov, Krasnochekov and Protopopov \cite{44} is applicable
for supercritical water and CO$_2$:

$$Nu_b = Nu_{0,b} \cdot \left( \frac{c_p}{c_{p,b}} \right)^{0.35} \cdot \left( \frac{\lambda_b}{\lambda_w} \right)^{-0.33} \cdot \left( \frac{\mu_b}{\mu_w} \right)^{0.11}$$  \hspace{1cm} (3.1a)

$$Nu_{0,b} = \left( \frac{h_b}{8} \cdot Re_b \cdot Pr \right) \left( 12.7 \cdot \left( \frac{h_b}{8} \right)^{0.5} \cdot \left( Pr^{2/3} - 1 \right) + 1.07 \right)$$  \hspace{1cm} (3.1b)

$$f = (1.82 \cdot \log_{10}(Re_b) - 1.64)^{-2}$$  \hspace{1cm} (3.1c)

Swenson, Carver and Kakaralas heat transfer correlation \[45\] is suitable for water in horizontal tubes:

$$Nu = 0.00459 \cdot \left( \frac{D \cdot G}{\mu_w} \right)^{0.923} \cdot \left( \frac{h_w - h_b}{T_w - T_b} \cdot \frac{\mu_w}{\lambda_w} \right)^{0.613} \cdot \left( \frac{\rho_w}{\rho_b} \right)^{0.231}$$  \hspace{1cm} (3.2)

Shitsman heat transfer correlation \[46\] is relevant in horizontal tubes with helium, water or CO$_2$ as the working fluid:

$$Nu_b = 0.023 \cdot Re_b^{0.8} \cdot Pr_{min}^{0.8}$$  \hspace{1cm} (3.3a)

$$Pr_{min} = \min[Pr_b, Pr_w]$$  \hspace{1cm} (3.3b)

The temperature of the thermal oil and the wall temperature of the inner tube are measured at several places along the length of the test section. Two types of temperature sensors are used: Pt100 resistive temperature sensors and K-type thermocouples. The Pt100 sensors are used to measure the inlet and outlet temperatures of the thermal oil and the organic working fluid while the K-type thermocouples are measuring the bulk temperature of the thermal oil and the wall temperature of the central tube.

There are three places where thermocouples are used to measure the bulk temperature of the thermal oil, each placed in a one meter interval. Figure 3.5 shows the radial configuration of the thermocouples and for each of these points, three thermocouples measure the bulk temperature: one at the top, one at the bottom and one on the side of the tube. Measuring the local heat transfer coefficient and the variation of the local heat transfer

Figure 3.5: Detailed schematic of the test section, thermal oil indicated in red and the organic working fluid indicated in blue
coefficient is done on eleven locations along the test section with three thermocouples at each point. These thermocouples are attached to the outer wall of the inner tube with an interval of 0.333 m. At each location three thermocouples are employed: one at the top, one at the bottom and one on the side of the tube. This is to measure the buoyancy effect, which is an important factor in the supercritical heat transfer in horizontal tubes, that creates temperature differences along the circumference of the inner tube.

To measure the pressure drop across the test section, pressure transducers are placed upstream and downstream of the test section and this also allows to monitor the operational pressure.

**Operational conditions**

The operating conditions are listed in Table 3.2. Most of the measurements are done at supercritical pressures and temperatures with R125. In order to test the set-up and know how it behaves, some experiments have been done at near-critical pressures and temperatures.

The organic working fluids selected for this study (R125, R134a), are compatible with the installed components and materials used. One of the prerequisites of these working fluids is that it is non-flammable, non-harmful to people, has an ODP of 0, are colourless and have a relatively low critical point. This is an advantage because supercritical conditions can be easily reached.

<table>
<thead>
<tr>
<th>Property</th>
<th>Operating range</th>
</tr>
</thead>
<tbody>
<tr>
<td>$p_{wf}$ [bar]</td>
<td>35 - 50</td>
</tr>
<tr>
<td>$\dot{G}_{wf}$ [kg/m²s]</td>
<td>395 - 645</td>
</tr>
<tr>
<td>$T_{wf}$ [$^\circ$C]</td>
<td>40 - 90</td>
</tr>
<tr>
<td>$T_{hf}$ [$^\circ$C]</td>
<td>70 - 120</td>
</tr>
</tbody>
</table>
3.2. TEST SECTION
Chapter 4

Control strategies and test procedures

4.1 Control strategy

Several components have to be controlled to be able to reach certain stable and steady state conditions (e.g. pressure or temperature) and this is done by a Proportional-Integral-Derivative (PID) controller. A PID control is a control loop feedback mechanism that calculates the error between the process value and a desired setpoint. The PID controller tries to minimize the error by adjusting the control variable, in our case for example the position of the mixing valve, to a new value determined by a weighted sum:

\[ u(t) = K_p \cdot e(t) + K_i \cdot \int_0^t e(\tau) \, d\tau + K_d \cdot \frac{de(t)}{dt} \]  

Here \( K_p, K_i, K_d \) represent the coefficients of the proportional (P), integral (I) and derivative (D) terms respectively. The P-action accounts for the present values of the error, the I-action takes care of the previous values of the error from the start until the present, while the D-action tries to control the possible future values of the error. Figure 4.1 shows an example of one PID control.

**PID tuning**

The components that are PID controlled are the pump, the electrical preheater, the thermal heater, the expansion valve, the mixing valves of the cooling and the heating loop. All these components will be fine-tuned experimentally but in order to speed up this process a basic understanding of the PID control is necessary. The electrical preheater and the thermal oil unit have built-in PID controllers, which don’t need to be reimplemented.
4.1. CONTROL STRATEGY

The proportional term, P-action, produces a value proportional to the current error value. If the proportional term is high, a large change in the output is to be expected, but if the value is low, a small change in the output is projected and the control might be less responsive. As the proportional term is driven by the current error value a steady state error will occur if no other action is taken.

To be able to remove the steady state error the integral term, I-action, is needed. It takes into account the magnitude and the duration of the error and can be seen as the accumulated error over time that should have been corrected previously. With high values for the integral term overshot is increasingly present, but with too low values the system might react too slow.

To improve the settling time of the system the derivative term, D-action, is used as it calculates the rate of change of the error and tries to predict the future value of the error.

The tuning method that will be used is the ZieglerNichols method [10] by setting the I-action and D-action to zero, while increasing the P-action until oscillation occurs in the output, denoted $P_{osc}$. This oscillation has a certain oscillation period, denoted $T_{osc}$, that is used for controlling the I-action and/or D-action depending on what type of control is desired. This is summarised in table 4.1.

Table 4.1: Ziegler-Nichols tuning rules for PID Ziegler and Nichols [10]

<table>
<thead>
<tr>
<th>Control type</th>
<th>$K_p$</th>
<th>$K_i$</th>
<th>$K_d$</th>
</tr>
</thead>
<tbody>
<tr>
<td>P</td>
<td>$0.5 \cdot P_{osc}$</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>PI</td>
<td>$0.45 \cdot P_{osc}$</td>
<td>$\frac{1}{2} \cdot T_{osc}$</td>
<td>-</td>
</tr>
<tr>
<td>PID</td>
<td>$0.60 \cdot P_{osc}$</td>
<td>$\frac{2}{T_{osc}}$</td>
<td>$\frac{T_{osc}}{8}$</td>
</tr>
</tbody>
</table>
Another possibility for more accurate control is to find the transfer function of each component and fine tune the step response for optimal control. This was not done in this study, but can be included in future work.

### 4.2 Data acquisition

All the data from the sensors is collected using two different data acquisition software, the CompactRIO from National Instruments and the Keithley 2700 from Tektronix. All the K-type thermocouples are collected using the Keithley, while all the Pt100 temperature sensors, pressure transducers, valves and the other components are connected to the CompactRIO data acquisition. The data from the CompactRIO is read in using a LabVIEW program and this is described in more detail in Bauwens [4]. The Keithley is connected to the computer and the data is collected in Excel using the ExceLINX program provided with the Keithley.

### 4.3 Measurement Procedures

#### 4.3.1 Start Up

Before starting up the set-up, certain simple checks have to be done. All the cables need to be plugged in and the pressure that is displayed on the panel meter needs to be between 12.5 bar and 14.5 bar, depending on the room temperature. Afterwards the following procedure to start the program and run the set-up is advised:

**Step 1:** Open the shut-off valve.

**Step 2:** Fully open the mixing valves of the cooling loop and the heating loop.

**Step 3:** Fully open the expansion valve.

**Step 4:** Set the frequency of the pump to 10 Hz and start the circulation. Listen carefully if irregular noise coming from the pump. An unusual noise might indicate damage to one of the diaphragms. Check the side glass for any bubbles at the inlet of the pump, which should be avoided at all times.

**Step 5:** Set the pump to the desired measuring frequency and turn on the electrical heater set at the desired inlet temperature of the test section.

**Step 6:** When the desired temperature of the organic working fluid is reached, at the inlet of the test section, turn on the thermal heater set at the desired temperature.

**Step 7:** Close the expansion valve in order to avoid a pressure higher than 35 bar at the inlet of the pump and wait for steady state to settle in.

The desired temperature at the inlet of the test section can be reached also by using one or two preheaters, but this requires the manual handling of the position of the valves.
and is not accurately reproducible. Nevertheless it can be used if the electrical preheater is unable to reach the required temperature. This can appear in case if there is a large temperature difference at the outlet of the pump and the desired temperature for the measurements. Partially or fully opening one or two valves from the preheaters will reduce the temperature increase the electrical heater has to provide, allowing the system to reach steady state.

4.3.2 Measurements strategy

The mass flow rate of the working fluid depends on the frequency of the pump. In Table 4.2 the measurement strategy that was followed is presented. All the measurements included in the table are at supercritical state.

<table>
<thead>
<tr>
<th>T_{hf} [°C]</th>
<th>T_{wf} [°C]</th>
<th>f_{pump} [Hz]</th>
<th>T_{hf} [°C]</th>
<th>T_{wf} [°C]</th>
<th>f_{pump} [Hz]</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>50</td>
<td>15-20-25</td>
<td>80</td>
<td>50</td>
<td>15-20-25</td>
</tr>
<tr>
<td>70</td>
<td>55</td>
<td>15-20-25</td>
<td>80</td>
<td>55</td>
<td>15-20-25</td>
</tr>
<tr>
<td>70</td>
<td>60</td>
<td>15-20-25</td>
<td>80</td>
<td>60</td>
<td>15-20-25</td>
</tr>
<tr>
<td>90</td>
<td>45</td>
<td>15-20-25</td>
<td>100</td>
<td>45</td>
<td>15-20-25</td>
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<tr>
<td>90</td>
<td>50</td>
<td>15-20-25</td>
<td>100</td>
<td>50</td>
<td>15-20-25</td>
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<tr>
<td>90</td>
<td>55</td>
<td>15-20-25</td>
<td>100</td>
<td>55</td>
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<tr>
<td>90</td>
<td>60</td>
<td>15-20-25</td>
<td>100</td>
<td>60</td>
<td>15-20-25</td>
</tr>
</tbody>
</table>

4.3.3 Shut Down

When shutting down the set-up, a number of steps need to be followed in order to ensure a safe shut down. The temperature of the heating fluid and working fluid need to be lowered as well as the pressure of the working fluid. First step is to turn off the electrical heater but keep the circulation going in the experimental loop. It is very important to leave the circulation on for at least 10 minutes to remove any electrical charges left on the electrical heating rods inside the electrical heater. Otherwise it could damage the electrical heater. Secondly, the thermal heating unit is put in circulation mode instead of heating mode and the mixing valve of the cooling loop is turned fully open. This allows both the heating fluid and working fluid to cool down, while gradually decreasing the pressure. When the temperature of the heating fluid reaches around 35 °C, the thermal heating unit can be shut down. The working fluid needs to be cooled down to temperatures between 25 °C and 30 °C with a pressure between 16 and 18 bar. If this condition is fulfilled, the pump
can be slowed down gradually to 10 Hz. After two minutes of pumping at 10 Hz, the pump can be shut down and the shut-off valve closed.
Chapter 5

Data reduction

In this chapter the derivation of the heat transfer coefficients and the pressure drop with their respective uncertainties are described. Calculation of the thermophysical properties of the organic working fluid is done with CoolProp [47] in combination with MATLAB [48] and an extract of the code can be found in Appendix A.

5.1 Heat losses

Minimization of the heat losses to the environment is achieved by wrapping the test section with an insulating ring of low thermal conductivity ($\lambda_{ins,r} = 0.043 \text{W/(m.K)}$ at 100 °C) and a thickness of 30 mm. An insulating blanket ($\lambda_{ins,b} = 0.046 \text{W/(m.K)}$ at 50 °C) with a width of 25 mm is placed round this insulating ring. Convection and radiation at the outside of the insulation are the reason of the heat losses in the test section and the calculation of the heat losses is shown in detail below.

The resistance network is shown in Figure 5.1 and it is assumed that the wall temperature at the annulus is the average of the bulk temperature of the thermal heating fluid in the test section. The heat loss to the environment is calculated with equation (5.1):

$$Q_{\text{loss}} = \frac{T_{hf,b} - T_{amb}}{R_{\text{wall}} + R_{\text{ins},r} + R_{\text{ins},b} + \frac{R_{\text{conv}}R_{\text{rad}}}{R_{\text{conv}} + R_{\text{rad}}}}$$ (5.1)

The conductive resistances of the galvanized tube, insulating ring and insulating blanket are given by equation 5.2, 5.3 and 5.4 respectively. In these equations the inner and outer diameter of the galvanized tube, $D_{i,hf}$ respectively $D_{o,hf}$, as well as the inner and outer diameter of the insulating ring and blanket, $D_{\text{ins},r}$ resp. $D_{\text{ins},b}$, are needed. The thermal conductivities for the insulating ring and blanket are found from the datasheet supplied by the manufacturer and depend on the temperature. This can be seen in Table 5.1.
5.1. HEAT LOSSES

Table 5.1: Thermal conductivities insulation (provided by manufacturer)

(a) Thermal conductivities insulating ring

<table>
<thead>
<tr>
<th>T [°C]</th>
<th>λ [W/m.K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>0.035</td>
</tr>
<tr>
<td>50</td>
<td>0.037</td>
</tr>
<tr>
<td>100</td>
<td>0.043</td>
</tr>
<tr>
<td>150</td>
<td>0.052</td>
</tr>
<tr>
<td>200</td>
<td>0.062</td>
</tr>
</tbody>
</table>

(b) Thermal conductivities insulating blanket

<table>
<thead>
<tr>
<th>T [°C]</th>
<th>λ [W/m.K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>0.037</td>
</tr>
<tr>
<td>30</td>
<td>0.041</td>
</tr>
<tr>
<td>50</td>
<td>0.046</td>
</tr>
</tbody>
</table>

Figure 5.1: Resistance network of test section
5.1. HEAT LOSSES

\[ R_{wall} = \frac{\ln \left( \frac{D_{o,hf}}{D_{i,hf}} \right)}{2 \cdot \pi \cdot L_{ts} \cdot \lambda_{galv,tube}} \]  
(5.2)

\[ R_{ins,r} = \frac{\ln \left( \frac{D_{ins,r}}{D_{o,hf}} \right)}{2 \cdot \pi \cdot L_{ts} \cdot \lambda_{ins,r}} \]  
(5.3)

\[ R_{ins,b} = \frac{\ln \left( \frac{D_{ins,b}}{D_{ins,r}} \right)}{2 \cdot \pi \cdot L_{ts} \cdot \lambda_{ins,b}} \]  
(5.4)

To be able to calculate the radiation and convection resistance equations 5.5 and 5.8 are used, but the temperature at the outer shell of the insulation, \( T_{ins} \), is unknown. This temperature is calculated by iteration using equations 5.1 through 5.12. Results of this iteration for different ambient temperatures can be found in Table 5.2.

The radiation resistance is inversely proportional to the heat transfer surface area at the outer shell of the insulation, \( A_{HT,ins} \), and the radiative heat transfer coefficient, equation 5.7. The outer layer of the insulation consists of a thin, slightly wrinkled aluminium foil and thus an emissivity of \( \epsilon_{ins} = 0.07 \) is assumed [49]. The Stefan-Boltzmann constant is \( \sigma = 5.670 \times 10^{-8} \text{ W/(m}^2\cdot\text{K}^4) \) and the ambient temperature is measured using a thermocouple. In Table 5.2 three different ambient temperatures are used with an average annulus temperatures of 100 °C in order to give a good indication of the magnitude of the losses in the test section.

\[ R_{rad} = \frac{1}{A_{HT,ins} \cdot h_{rad}} \]  
(5.5)

\[ A_{HT,ins} = \pi \cdot D_{ins,o} \cdot L_{ts} \]  
(5.6)

\[ h_{rad} = \epsilon_{ins} \cdot \sigma \left( T_{ins}^2 + T_{amb}^2 \right) \cdot \left( T_{ins} + T_{amb} \right) \]  
(5.7)

To calculate the convective resistance the heat transfer surface area, equation 5.6, and the convective heat transfer coefficient need to be computed. This last one can be found using the Nusselt number for natural convection around a horizontal cylinder and is given by Churchill [50], equation 5.10. For this the Rayleigh number, equation 5.11 has to be calculated and the thermophysical properties are evaluated at the average air temperature \( T_{air} = \frac{T_{ins} + T_{amb}}{2} \).

\[ R_{conv} = \frac{1}{A_{HT,ins} \cdot h_{conv}} \]  
(5.8)

\[ h_{conv} = \frac{N_{u,conv} \cdot \lambda_{air}}{D_{ins,out}} \]  
(5.9)
\[ Nu_{\text{conv}} = \left( 0.60 + \frac{0.387 \cdot Ra^{1/6}}{1 + \left( \frac{0.559}{Pr} \right)^{9/16}} \right)^2 \] (5.10)

\[ Ra = \frac{g \cdot \beta_{\text{air}} \cdot (T_{\text{ins}} - T_{\text{amb}}) \cdot D_{\text{ins,o}}^3}{\nu_{\text{air}} \cdot \alpha_{\text{air}}} \] (5.11)

In order to complete the iterative process of finding the insulation wall temperature another equation has to be introduced. With this equation an iterative program was written in MATLAB [48] that determines the insulation temperature with a residual of \(1 \cdot 10^{-8}\).

\[ Q_{\text{loss}} = \frac{T_{\text{ins}} - T_{\text{amb}}}{R_{\text{conv}} + R_{\text{rad}}} \] (5.12)

<table>
<thead>
<tr>
<th>( T_{\text{amb}} ) [°C]</th>
<th>( T_{\text{ins}} ) [°C]</th>
<th>( R_{\text{rad}} ) [K/W]</th>
<th>( R_{\text{conv}} ) [K/W]</th>
<th>( Q_{\text{loss}} ) [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>16.00</td>
<td>25.71</td>
<td>1.2134</td>
<td>0.1422</td>
<td>76.324</td>
</tr>
<tr>
<td>20.00</td>
<td>29.41</td>
<td>1.1670</td>
<td>0.1448</td>
<td>73.060</td>
</tr>
<tr>
<td>24.00</td>
<td>33.09</td>
<td>1.1231</td>
<td>0.1473</td>
<td>69.763</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>( R_{\text{wall}} ) [K/W]</th>
<th>( R_{\text{ins, r}} ) [K/W]</th>
<th>( R_{\text{ins, b}} ) [K/W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>( 1.0269 \cdot 10^{-4} )</td>
<td>0.7194</td>
<td>0.2467</td>
</tr>
</tbody>
</table>

### 5.2 Data reduction

To calculate the heat transfer coefficients the test section is divided in a number of cells with equal enthalpy change and in each of these cells the equations [5.17] to [5.19] are to be applied. Furthermore, the heat transfer coefficient will be calculated in different places, as shown in Figure [5.2] and are discussed in more detail below in the text. The pressure drop is also measured at the inlet and the outlet of the test section of the organic working fluid.

#### 5.2.1 Heat transfer coefficient

Both, the overall heat transfer coefficient and the local heat transfer coefficient can be calculated from the temperature measurements. This is possible because there are temperature measurements of the inner tube wall positioned over the entire length of the
5.2. DATA REDUCTION

Figure 5.2: Subdivision of the test section
test section in 11 discrete points. In each of these measuring points, the temperature is measured at the top, bottom and side, see Figure 3.5.

**Overall heat transfer coefficient**

The calculation of the overall heat transfer coefficient $U$ is possible in two ways that are very similar. The first way to calculate it, is by using equations 5.13 to 5.16 and has one inherent problem that lies with the LMTD-method. The LMTD-method assumes (near) constant thermophysical properties between the in- and outlet for the fluid, which is incorrect in our case.

**First method**

$$U = \frac{Q_{wf}}{A_{HT,wf} \cdot LMTD}$$  \hspace{1cm} (5.13)

Determining the heat flux is possible by using the pressure transducers and Pt100 temperature sensors at the inlet and outlet of the test section, together with the Coriolis mass flow meter. With these measurements, the enthalpy at the inlet and outlet of the test section can be calculated and so the heat flux from the thermal heating fluid to the organic working fluid is calculated (with linear approximation of pressure drop in test section). This could also have been calculated using the heat flux from the thermal heating fluid to the working fluid and the environment subtracted with the heat loss to the
5.2. DATA REDUCTION

Figure 5.3: Division of test section in cells with equal enthalpy change

environment in the test section. This is not done here as it would lead to a higher error in further calculations, as some assumptions were made in calculating the heat loss to the environment, see [5.1]

\[ Q_{wf} = \dot{m}_{wf} \cdot (h_{wf,ts\ out} - h_{wf,ts\ in}) \]  

(5.14)

The Logarithmic Mean Temperature Difference (LMTD) is used to determine the temperature driving force in the test section and is defined as follows:

\[ LMTD = \left[ \frac{(T_{hf,ts\ in} - T_{wf,ts\ out}) - (T_{hf,ts\ out} - T_{wf,ts\ in})}{\ln \left( \frac{T_{hf,ts\ in} - T_{wf,ts\ out}}{T_{hf,ts\ out} - T_{wf,ts\ in}} \right)} \right] \]  

(5.15)

Using equation [5.16] we can calculate the heat transfer surface and use the inner diameter of the copper tube to calculate the circumference.

\[ A_{HT,\ wf} = \pi \cdot d_{i,wf} \cdot L_{ts} \]  

(5.16)

**Second method** To be able to use the LMTD-method we subdivide the test section in a number of cells with equal heat flux from the thermal heating fluid to the organic working fluid. Figure [5.3] illustrates this. If the number of cells is chosen high enough we can assume that the thermophysical properties are constant in each cell. Using equations [5.17] to [5.19] we have a N+2 unknowns and N+2 equations, with N the number of cells. Solving this set of N+2 equations results in the correct overall heat transfer coefficient U.

\[ Q' = \frac{Q_{wf}}{N} \]  

(5.17)

\[ \sum_{i=1}^{N} A_{i,HT} = A_{HT,\ wf} \]  

(5.18)

\[ Q' = U \cdot A_{i,HT} \cdot LMTD_i, \ i = 1...N \]  

(5.19)
Local heat transfer coefficient

At eleven measuring points, distributed evenly along the test section, the local heat transfer coefficients can be calculated using the temperature measurements defined by equation 5.20. The heat flux at the inner surface of the tube can be considered constant in each of the four sections, see Figure 5.2. The heat flux density is also considered constant in between each measuring point and is calculated as the ratio of the heat flux to the heat transfer area at the inner surface of the tube (equation 5.21).

\[
HTC_{loc} = \frac{q_{wf}}{T_{wall,in} - T_{bulk}} \quad (5.20)
\]

\[
q_{i,wf} = \frac{Q_{k,wf}}{\pi \cdot d_{i,wf} \cdot L_{ts}/4} \quad (5.21)
\]

\[
T_{wall,in} = T_{wall,out} - \frac{Q_{k,wf} \cdot \ln \frac{d_{out}}{d_{i,wf}}}{2 \cdot \pi \cdot L_{ts}/4 \cdot \lambda_{copper \, tube}} \quad (5.22)
\]

The wall temperature at the inner surface of the tube is determined with equation 5.22. To determine this we need the outer wall temperature. The outer wall temperature can be defined as the average of the top, side and bottom wall temperature, which gives us an averaged local heat transfer coefficient. Another option is to take either the top, side or bottom wall temperature giving a detailed view of the flow at the top, side or bottom of the tube (equations 5.23).

\[
T_{wall,out} = \frac{T_{top} + T_{side} + T_{bottom}}{3} \quad (5.23a)
\]

\[
T_{wall,out} = T_{top} \quad T_{wall,out} = T_{side} \quad T_{wall,out} = T_{bottom} \quad (5.23b)
\]

5.2.2 Pressure drop in working fluid

In order to determine the heat transfer coefficient and the friction factor, the measured pressure drop along the test section is taken into consideration. The pressure drop over the test section is measured, by using two absolute pressure transducers. This pressure drop can be subdivided in three components: the frictional pressure drop, the pressure drop due to acceleration and the adiabatic pressure drop. As this is a horizontal set-up the static pressure drop has no influence in the total pressure drop. As we are interested to find a new friction factor for supercritical heat transfer we will determine this experimentally using the following equations:

\[
\Delta p_{meas} = \Delta p_f + \Delta p_{ac} + \Delta p_{ad} \quad (5.24)
\]
The adiabatic pressure drop occurs in the adiabatic section and is calculated as [2]:

\[
\Delta p_{ad} = f \cdot \frac{L_{ad}}{d_{ad}} \cdot \frac{\rho \cdot u^2}{2}
\] (5.25)

\[
f = [1.82 \cdot \log (Re_b) - 1.64]^2
\] (5.26)

The pressure drop due to acceleration is a result from density variations across the test section. As the mass flow rate has to remain constant and the density decreases along the test section, the organic working fluid accelerates. This is calculated using equation 5.27 [51]:

\[
\Delta p_{ac} = G_{wf} \cdot \left( \frac{1}{\rho_{out}} - \frac{1}{\rho_{in}} \right)
\] (5.27)

With these we can calculate the frictional pressure drop through the test section as follows:

\[
\Delta p_f = \Delta p_{meas} - \Delta p_{ac} - \Delta p_{ad}
\] (5.28)

### 5.2.3 Pressure drop in heating fluid (thermal oil)

Designing a test section requires that the designer makes several choices and assumptions. Based on that several components have to be chosen that are compatible with each other and the overall system. After installing all the components at the heating fluid side, the pressure drop was recalculated. This was done in order to evaluate the performance of each component such as the built in circulation pump of the thermal heater, mixing valve, etc. As a heating fluid a low viscosity synthetic organic heat transfer fluid was used.

The largest pressure drop across the test section for the thermal heating fluid will occur when it has to pass both preheaters. This pressure drop consists of the frictional pressure drop, calculated with the Darcy-Weisbach equation 5.29 and the static pressure drop due to the height differences. Usage of equation 5.29 is allowed since the thermal heating fluid remains in one single phase, the liquid phase as it has a boiling point of 293 °C at 1 bar. However, the maximum operating temperature and pressure is 180 °C and 3 bar.

\[
\Delta p = 4 \cdot f \cdot \frac{L}{D_h} \cdot \frac{v^2}{2}
\] (5.29)

\[
f = \begin{cases} 
\frac{Re}{16} & \text{if } Re < 2300 \\
(1.58 \cdot \log Re - 3.18)^{-2} & \text{if } Re \geq 2300
\end{cases}
\] (5.30)

The heating loop consists of series of straights, bends, tees, ball valves, a mixing valve and a Coriolis mass flow meter. The lengths of the straights are measured using a tape measurer and the equivalent length of the bends, tees and ball valves are calculated using Table 5.3. An average temperature for the bulk of the thermal heating fluid is assumed
equal to the average temperature of the annulus of the test section. The pressure drop across the mixing valve is estimated using the sizing sheet provided by the manufacturer and is about 1 bar. The pressure drop for the Coriolis mass flow meter is also provided by the manufacturer in the data sheet and depends on the mass flow rate. The static pressure drop is determined with the straight tubes and their respective angle to the horizontal plane. As it is a horizontal set-up we only have to account for the vertical tubing that has a total vertical length of 0.545 m of downward flow. With a mass flow representing average testing conditions we get a total pressure drop of 1.34 bar which is acceptable for the heating unit. The sized components (heating unit, mixing valves and ball valves) are all in the acceptable range and work as expected.
5.2. DATA REDUCTION
Chapter 6

Uncertainty analysis

Every experiment done anywhere in the world is only as good as the equipment used and more specifically the accuracy of the equipment and the overall system. All measurements inherently have errors and to be able to make relevant conclusions, a good error analysis has to be performed. This error analysis uses the individual errors of the measuring equipment and/or the acquisition system and combines them using equation 6.5. In the following all errors of the measuring equipment will be presented together with an example of the calculations.

6.1 Measurement equipment accuracy

Mass flow meters

In this set-up two Coriolis mass flow meters are used to measure the mass flow rate of the heating fluid and the organic working fluid respectively. Both mass flow meters are calibrated by the manufacturer and are documented in a calibration document that is provided together with the component. The calibration is done in 5 discrete points with a variation from 3 kg/s to 30 kg/s for the mass flow meter of the organic working fluid and from 6.7 kg/s to 138 kg/s for the mass flow meter of the heating fluid. This is summarized in table 6.1. The calculation of the propagation of errors takes the largest error, 0.07 % for organic fluid and 0.08 % for the heating fluid, as we do not have enough calibration points to cover all the testing conditions.
6.1. Measurement equipment accuracy

Table 6.1: Uncertainties mass flow meters

(a) Organic working fluid mass flow meter uncertainty

<table>
<thead>
<tr>
<th>Mass flow rate [kg/min]</th>
<th>( \delta \dot{m}_{\text{wf}} ) [kg/min]</th>
<th>( \delta \dot{m}<em>{\text{wf}} / \dot{m}</em>{\text{wf}} ) [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.0</td>
<td>0.0003</td>
<td>0.01</td>
</tr>
<tr>
<td>7.5</td>
<td>0.0053</td>
<td>0.07</td>
</tr>
<tr>
<td>15.0</td>
<td>0.0075</td>
<td>0.05</td>
</tr>
<tr>
<td>22.5</td>
<td>0.0000</td>
<td>0.00</td>
</tr>
<tr>
<td>30.0</td>
<td>0.0060</td>
<td>0.02</td>
</tr>
</tbody>
</table>

(b) Heating fluid mass flow meter uncertainty

<table>
<thead>
<tr>
<th>Mass flow rate [kg/min]</th>
<th>( \delta \dot{m}_{\text{oil}} ) [kg/min]</th>
<th>( \delta \dot{m}<em>{\text{oil}} / \dot{m}</em>{\text{oil}} ) [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>6.7</td>
<td>0.0053</td>
<td>0.08</td>
</tr>
<tr>
<td>14.0</td>
<td>0.0084</td>
<td>0.06</td>
</tr>
<tr>
<td>35.0</td>
<td>0.0175</td>
<td>0.05</td>
</tr>
<tr>
<td>70.0</td>
<td>0.0350</td>
<td>0.05</td>
</tr>
<tr>
<td>105.0</td>
<td>0.0525</td>
<td>0.05</td>
</tr>
<tr>
<td>138.0</td>
<td>0.276</td>
<td>0.20</td>
</tr>
</tbody>
</table>

Pressure transducers

The pressure sensors are placed at five locations on the test set-up. Pressure sensor 1 and 3 measure the outlet and inlet pressure of the test section respectively. Further, pressure sensor 2 determines the condenser outlet pressure and is used together with a temperature sensor to check whether the organic working fluid at the inlet of the pump is a sub-cooled liquid. Controlling the expansion valve is done with the aid of pressure sensor 5, which is located behind the expansion valve. To measure the outlet pressure of the pump, pressure sensor 4 is used and is connected to a panel meter that can shut down the pump if the pressure exceeds a certain limit.

As we are only interested in the accuracies the pressure sensors at the in- and outlet of the test section, we will only consider these uncertainties further. The reason for not considering the exact accuracy of the other three pressure sensors is that they are used as a safety or control and therefore their uncertainties will not propagate through the data reduction.

The pressure sensors at in- and outlet have an accuracy of 0.1 % FS BSL (Full Scale Best Straight Line, \( \delta \dot{m}_{\text{acc}} \)), a thermal uncertainty of 0.009 % FS/°C (\( \delta \dot{m}_{\text{temp}} \)) and an accuracy of 0.1 % FS/year (\( \delta \dot{m}_{\text{drift}} \)), due to the drift that increases with the age of the sensor. The thermal error needs to be multiplied, not with the temperature at which the pressure sensor operates, but with the temperature relative to the calibration temperature of 20 °C.

\[
\delta p_{\text{tot}} = \sqrt{\delta p_{\text{acc}}^2 + \delta p_{\text{temp}}^2 + \delta p_{\text{drift}}^2} \tag{6.1}
\]

The accuracies of the pressure transducers ranged between 0.5651 % and 0.6872 % at 45 bar and 37 bar respectively.
6.1. MEASUREMENT EQUIPMENT ACCURACY

Temperature sensors

The temperature sensors can be subdivided in two categories: Pt100 sensors and K-type thermocouples. The Pt100 temperature sensors are Resistance Temperature Detectors (RTDs) that measure the temperature by correlating the resistance of the sensor with temperature. In total, five of these sensors are used in the set-up. Pt100 3 and Pt100 1 measure the bulk of the organic working fluid when it enters and leaves the test section respectively. Pt100 2 evaluates the temperature of the organic working fluid after the condenser. Pt100 4 and Pt100 5 determine the temperature of the thermal heating fluid when it exits and enters the test section with respect to the thermal heating fluid flow respectively.

The K-type thermocouples are used to measure the wall temperature of the copper tube and the bulk temperature of the thermal heating fluid. At eleven different places they are placed to measure the wall temperature at the bottom, side and top. At position 3, 6 and 9 the bulk temperature of the thermal heating fluid is measured as well.

All the thermocouples and Pt100 sensors are calibrated at Ghent University, using a temperature controlled oven and a reference probe with an absolute uncertainty of ±0.064 °C (δTref). The Pt100 sensor placed after the condenser is not re-calibrated as it is not used in the data reduction but only for control of the condenser. The Pt100 sensors are accurate and reliable enough to be used as control, but for increased accuracy in the measurements and data reduction calibration is necessary. The remaining four are calibrated by placing them in the controlled oven that has a temperature range between 0 °C and 100 °C with steps of 10 °C. In each measuring point several measurements are taken by both the Pt100 sensor and the reference probe and the average of each is calculated. The behaviour of the Pt100 sensors can then be represented as a linear curve \( y = Ax + B \), where \( y \) is the actual temperature measured by the reference probe and \( x \) is the temperature measured with the thermocouple. This is then used to correct the measurements done by the Pt100 sensors to improve the accuracy. But this also induces an uncertainty of the fitted curve (δTreg) and this uncertainty for linear regression is given by:

\[
\delta T_{\text{reg}} = \sqrt{\frac{\sum_i (y_i - Ax_i - B)^2}{N - 2}}
\]  

The accuracies of the Pt100 thermocouples varied between 0.0661 % and 0.083 % for the heating fluid. For the thermocouple at the inlet of the test section, measuring the temperature of the organic working fluid, the accuracy is between 0.112 % and 0.149 %. This is very similar for the outlet of the test section, accuracies ranging from 0.0986 % and 0.109 % are found.
Table 6.2: Temperature sensors Pt100 uncertainty

<table>
<thead>
<tr>
<th>PT100 sensors</th>
<th>$\delta T_{\text{regr}}$</th>
<th>$\delta T_{\text{tot}}$ [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pt100 1</td>
<td>0.0205</td>
<td>0.0672</td>
</tr>
<tr>
<td>Pt100 2</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Pt100 3</td>
<td>0.0329</td>
<td>0.0720</td>
</tr>
<tr>
<td>Pt100 4</td>
<td>0.0176</td>
<td>0.0664</td>
</tr>
<tr>
<td>Pt100 5</td>
<td>0.0166</td>
<td>0.0661</td>
</tr>
</tbody>
</table>

The calibration of the K-type thermocouples is done similarly, but over a temperature range of 20 °C to 100 °C. In this case a second order regression is used, $y = A \cdot x^2 + B \cdot x + C$, as the accuracy with a first order curve is insufficient. For the second order curve the uncertainty is given by:

$$
\delta T_{\text{regr}} = \sqrt{\sum_i (y_i - A \cdot x_i^2 - B \cdot x_i - C)^2 / (N - 3)}
$$

The total uncertainty of the thermocouples and Pt100 sensors is a combination of the regression curve uncertainty ($\delta T_{\text{regr}}$) and the reference probe uncertainty ($\delta T_{\text{ref}}$):

$$
\delta T_{\text{tot}} = \sqrt{\delta T_{\text{regr}}^2 + \delta T_{\text{ref}}^2}
$$

Tables 6.2, 6.3 and 6.4 give the uncertainties for Pt100 sensor and K-type thermocouple.

**CoolProp uncertainty**

CoolProp [23] uses the equations of state for R125 published by Lemmon and Jacobsen [52] to calculate the density and specific heat capacity. In this work, the density has an overall accuracy of 0.05 % and the specific heat capacity an overall accuracy of 0.5 %. This is with exclusion of the case for the critical region. The scarce data available around the critical region, as it is not effortless to obtain reliable data around this region, makes it more difficult for an accurate equation of state to be proposed. The uncertainty’s in density around the critical region are generally 0.5 %, but when compared to experimental data can exceed 5 %. The isobaric specific heat capacity has an uncertainty of 1 % in critical region.
6.1. MEASUREMENT EQUIPMENT ACCURACY

Table 6.3: Thermocouple uncertainties measuring wall temperature copper tube

<table>
<thead>
<tr>
<th>Thermocouples</th>
<th>$\delta_{\text{regr}}$</th>
<th>$\delta_{\text{tot}}$ [°C]</th>
<th>Thermocouples</th>
<th>$\delta_{\text{regr}}$</th>
<th>$\delta_{\text{tot}}$ [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Top TC 1</td>
<td>0.0829</td>
<td>0.1047</td>
<td>Top TC 7</td>
<td>0.0830</td>
<td>0.1048</td>
</tr>
<tr>
<td>Side TC 1</td>
<td>0.0596</td>
<td>0.0875</td>
<td>Side TC 7</td>
<td>0.0497</td>
<td>0.0810</td>
</tr>
<tr>
<td>Bottom TC 1</td>
<td>0.1217</td>
<td>0.1375</td>
<td>Bottom TC 7</td>
<td>0.0545</td>
<td>0.0841</td>
</tr>
<tr>
<td>Top TC 2</td>
<td>0.0897</td>
<td>0.1102</td>
<td>Top TC 8</td>
<td>0.0815</td>
<td>0.1039</td>
</tr>
<tr>
<td>Side TC 2</td>
<td>0.0906</td>
<td>0.1109</td>
<td>Side TC 8</td>
<td>0.0597</td>
<td>0.0875</td>
</tr>
<tr>
<td>Bottom TC 2</td>
<td>0.1132</td>
<td>0.1301</td>
<td>Bottom TC 8</td>
<td>0.0493</td>
<td>0.0808</td>
</tr>
<tr>
<td>Top TC 3</td>
<td>0.0877</td>
<td>0.1086</td>
<td>Top TC 9</td>
<td>0.0813</td>
<td>0.1035</td>
</tr>
<tr>
<td>Side TC 3</td>
<td>0.0633</td>
<td>0.0900</td>
<td>Side TC 9</td>
<td>0.0580</td>
<td>0.0864</td>
</tr>
<tr>
<td>Bottom TC 3</td>
<td>0.0891</td>
<td>0.1097</td>
<td>Bottom TC 9</td>
<td>0.0565</td>
<td>0.0854</td>
</tr>
<tr>
<td>Top TC 4</td>
<td>0.0852</td>
<td>0.1066</td>
<td>Top TC 10</td>
<td>0.0829</td>
<td>0.1047</td>
</tr>
<tr>
<td>Side TC 4</td>
<td>0.0652</td>
<td>0.0913</td>
<td>Side TC 10</td>
<td>0.0525</td>
<td>0.0828</td>
</tr>
<tr>
<td>Bottom TC 4</td>
<td>0.0775</td>
<td>0.1055</td>
<td>Bottom TC 10</td>
<td>0.0620</td>
<td>0.0891</td>
</tr>
<tr>
<td>Top TC 5</td>
<td>0.0844</td>
<td>0.1059</td>
<td>Top TC 11</td>
<td>0.0816</td>
<td>0.1037</td>
</tr>
<tr>
<td>Side TC 5</td>
<td>0.0641</td>
<td>0.0906</td>
<td>Side TC 11</td>
<td>0.0677</td>
<td>0.0931</td>
</tr>
<tr>
<td>Bottom TC 5</td>
<td>0.0681</td>
<td>0.0934</td>
<td>Bottom TC 11</td>
<td>0.0428</td>
<td>0.0770</td>
</tr>
<tr>
<td>Top TC 6</td>
<td>0.0835</td>
<td>0.1052</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Side TC 6</td>
<td>0.0661</td>
<td>0.0920</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bottom TC 6</td>
<td>0.0579</td>
<td>0.0863</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 6.4: Thermocouple uncertainties measuring bulk temperature thermal heating fluid

<table>
<thead>
<tr>
<th>Bulk thermocouple</th>
<th>$\delta_{\text{regr}}$</th>
<th>$\delta_{\text{tot}}$ [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Top TC$_b$ 3</td>
<td>0.1075</td>
<td>0.1251</td>
</tr>
<tr>
<td>Side TC$_b$ 3</td>
<td>0.0978</td>
<td>0.1168</td>
</tr>
<tr>
<td>Bottom TC$_b$ 3</td>
<td>0.0883</td>
<td>0.1091</td>
</tr>
<tr>
<td>Top TC$_b$ 6</td>
<td>0.0873</td>
<td>0.1082</td>
</tr>
<tr>
<td>Side TC$_b$ 6</td>
<td>0.0644</td>
<td>0.0908</td>
</tr>
<tr>
<td>Bottom TC$_b$ 6</td>
<td>0.0974</td>
<td>0.1166</td>
</tr>
<tr>
<td>Top TC$_b$ 9</td>
<td>0.0867</td>
<td>0.1078</td>
</tr>
<tr>
<td>Side TC$_b$ 9</td>
<td>0.0862</td>
<td>0.1074</td>
</tr>
<tr>
<td>Bottom TC$_b$ 9</td>
<td>0.1020</td>
<td>0.1204</td>
</tr>
</tbody>
</table>
The calculation of the enthalpy is not done by an equation of state, but is a derived quantity. This makes it hard to determine the accuracy of the enthalpy and is not mentioned by CoolProp [23]. This is a problem to which no solution has been found. As the enthalpy is derived from the specific heat capacity, the uncertainty is taken of 1 %.

### 6.2 Uncertainty propagation

When all absolute uncertainties of all sensors and tabulated data are known, equation 6.5 is used to see how these inaccuracies propagate through the data reduction (section 5.2). Equation 6.5 represents the absolute error of \( q \) that is a dependant of parameters \( x, \ldots, z \) which have the independent and random absolute errors \( \delta x, \ldots, \delta z \) (Taylor [53]).

\[
\delta q = \sqrt{\left( \frac{\partial q}{\partial x} \delta x \right)^2 + \cdots + \left( \frac{\partial q}{\partial z} \delta z \right)^2}
\]  

(6.5)

The overall uncertainty of the (calculated) local and overall heat transfer coefficient and pressure drop depend on the accuracies of its parameters. These accuracies vary depending on the measured values which makes it impractical to have a general number that indicates the overall uncertainty of these calculated values. All the measurements are automatically calculated with their respective inaccuracy using a script in MATLAB [48].

### 6.3 Example uncertainty propagation

This is an example of the uncertainty propagation calculations that are done in this work. The mass flux of the organic working fluid is calculated using equation 6.6 and the error propagation is done in two steps. First we calculate the error on the through flow area, equation 6.7 and with that we calculate the error on the mass flux, equation 6.8. The same is done for the mass flux of the heating fluid, equations 6.9 through 6.11.

**Calculating the mass flux working fluid**

\[
G_{wf} = \frac{\dot{m}_{wf}}{A_{ts,wf}} \quad A_{ts,wf} = \frac{\pi}{4} \cdot d_{i,wf}^2
\]  

(6.6)
\[ \delta A_{ts,wf} = \sqrt{\left( \frac{\partial A_{ts,wf}}{\partial d_{i,wf}} \cdot \delta d_{i,wf} \right)^2} \]
\[ = \sqrt{\left( \frac{\pi \cdot d_{i,wf}}{2} \cdot \delta d_{i,wf} \right)^2} \]
\[ = \frac{\pi \cdot d_{i,wf}}{2} \cdot \delta d_{i,wf} \] (6.7)

\[ \delta G_{wf} = \sqrt{\left( \frac{\partial G_{wf}}{\partial m_{wf}} \cdot \delta m_{wf} \right)^2 + \left( \frac{\partial G_{wf}}{\partial A_{ts,wf}} \cdot \delta A_{ts,wf} \right)^2} \]
\[ = \sqrt{\left( \frac{1}{A_{ts,wf}} \cdot \delta m_{wf} \right)^2 + \left( -\frac{m_{wf}}{A_{ts,wf}^2} \cdot \delta A_{ts,wf} \right)^2} \] (6.8)

Calculating the mass flux heating fluid

\[ G_{oil} = \frac{m_{oil}}{A_{ts,oil}} \quad A_{ts,oil} = \frac{\pi}{4} \cdot (D_{i,oil}^2 - d_{o,wf}^2) \] (6.9)

\[ \delta A_{ts,oil} = \sqrt{\left( \frac{\partial A_{ts,oil}}{\partial D_{i,oil}} \cdot \delta D_{i,oil} \right)^2 + \left( \frac{\partial A_{ts,oil}}{\partial d_{o,wf}} \cdot \delta d_{o,wf} \right)^2} \]
\[ = \frac{\pi}{2} \cdot \sqrt{(D_{i,oil} \cdot \delta D_{i,oil})^2 + (d_{o,wf} \cdot \delta d_{o,wf})^2} \] (6.10)

\[ \delta G_{oil} = \sqrt{\left( \frac{\partial G_{oil}}{\partial m_{oil}} \cdot \delta m_{oil} \right)^2 + \left( \frac{\partial G_{oil}}{\partial A_{ts,oil}} \cdot \delta A_{ts,oil} \right)^2} \]
\[ = \sqrt{\left( \frac{1}{A_{ts,oil}} \cdot \delta m_{oil} \right)^2 + \left( -\frac{m_{oil}}{A_{ts,oil}^2} \cdot \delta A_{ts,oil} \right)^2} \] (6.11)
6.3. EXAMPLE UNCERTAINTY PROPAGATION
Chapter 7

Results and model development

7.1 Adaptations to measuring method

When working on the set-up several difficulties arose that eventually led to an adaptation in the measuring method. In the first attempt to have local wall temperature measurements using Keithley, a problem was encountered. At first, not all the thermocouples measured the same temperature in steady state at room temperature. The side wall temperatures and the heating fluid temperatures measured temperatures of 18 °C to 22 °C higher than what was expected. This was because Keithley lacked a reference Pt100 at the measuring pins and took 0 °C as a reference. This was solved by adding a second reference Pt100 temperature sensor at the connection of the measuring pins.

This however, was not the only complication as the thermocouples placed at the wall did not measure the correct wall temperature. Several checks have been done in order to come to this conclusion. Four different cases are considered depending on the circulation in the fluid and it’s temperature compared to the other working fluid.

i. No flow of the organic working fluid and no flow of the heating fluid at environmental temperatures.

ii. Flow of the organic working fluid and no flow of the heating fluid, with $T_{wf} > T_{hf}$.

iii. Flow in both the organic working fluid and heating fluid, with $T_{wf} < T_{hf}$.

iv. Flow in both the organic working fluid and heating fluid, with $T_{wf} > T_{hf}$.

All these different cases are shown in Figure 7.1. The Pt100 temperature sensors measure the in- and outlet of both the organic working fluid and the heating fluid. The heating fluid, marked in red, is measured at three different positions equally distributed over the test section. Position 0 corresponds with the outlet of the test section, with respect to the
7.1. ADAPTATIONS TO MEASURING METHOD

organic working fluid, while position 12 corresponds with the inlet of the organic working fluid. This means that the flow of the organic working fluid is from right to left, and vice versa for the heating fluid.

In the first case (a), at environmental temperatures, everything is measured as expected. The wall mounted thermocouples measure approximately the organic working fluid. This case does not provide much insight as all the temperatures are within 2 °C. The second case (b) shows, upper right in Figure 7.1, that when there is no flow in the heating fluid, the wall mounted thermocouples measure approximately the organic working fluid. However, as shown in lower left, the third case (c) is far from what is expected. Here, the wall mounted thermocouples measure temperatures close to the heating fluid. This deviation from the in- and outlet bulk temperature measurements of the organic working fluid cannot be explained with the accuracy of the measurements. The fourth case (d), displayed in the bottom right, shows that when there is flow in the heating fluid, the wall mounted thermocouples measure a temperature close to the bulk of the heating fluid. It is obvious that the flow of the heating fluid influences the measurements on the wall.

![Figure 7.1: Measurements done to confirm the bad positioning of the thermocouples](image)

In order to check whether the thermocouples are positioned badly or there is a connection problem, all the cables have been visually inspected for flaws and the connection to Keithley has been verified. The positioning of the thermocouples was also checked by confirming that all the materials used to attack the thermocouples to the wall are compatible with the operating conditions. As no flaws are found, it is reasonable to conclude that the positioning of the thermocouples is faulty and therefore these measurements are completely excluded. No local heat transfer measurements could take place because of
Another unwanted effect that was noticed during the measurements, is the long reacting time of the Pt100 1 and Pt100 3 temperature sensors at the in- and outlet of the test section that measured the organic working fluid temperature. This delay was not noticed at the heating fluid side, as the positioning of the Pt100 temperature sensors is different from those at the organic working fluid side. The temperature sensors are placed directly in the flow of the heating fluid, which is not the case for the organic working fluid side, see Figure 7.2. This was a design choice that was made in order to minimize the flow disturbance in the test section at the organic working fluid side. The Pt100 temperature sensor is mounted in a branch of a tee tube fitting, together with a pressure transducer. This section was placed upon another tee tube fitting connecting the test section to the rest of the set-up. Due to this configuration, the temperature measured by the Pt100 is delayed because the rate of fluid change near the sensor is low. Temperature changes can only be conveyed by conduction and very little by convection because of the small diameter in this configuration. Hotter fluid will also tend to stay longer at the top due to the buoyancy effect.

A proposed solution is the use of cross tube configuration in order to get the tip of the Pt100 temperature sensor closer to, but without disrupting the main flow. This configuration would have the advantage of fast measurements because the tip of the Pt100 is close to the main flow. The flow is lightly perturbed due to the two branches in comparison to the previous configuration, where only one branch exists. This influence is minimal and is justified. The inlet of the test section has a 1 m long adiabatic test section, with the intention of having a homogeneous inlet at the test section. In the adiabatic section, the flow perturbation will be smoothed out before the test section is reached. At the outlet of the test section, flow disturbance is of less concern.
7.2 Results

In this thesis book, results obtained from the initial measurements are presented. The control of the set-up for these measurements was done with a LabVIEW program, but the program didn’t run autonomously but was monitored by the users. This was done in order to be familiar with the set-up and how it would react to various stimuli. With these initial measurements, the positioning of the K-type thermocouples and Pt100 temperature sensors, as well as the correct functioning of the pressure transducers and mass flow meters could be verified. Unfortunately, the initial measurements prove that the K-type thermocouples are not attached well and the wall measurements could not be obtained. Achieving steady state is another difficulty that was not completely resolved. Experiments at near steady state are performed as the set-up was manually controlled and not completely PID-controlled. This lead to inaccuracies in the measurements together with the time delay of the Pt100 temperature sensors that is described in the text above.

Only one parameter at a time can be controlled of the pump and a constant mass flow rate is chosen over a constant pressure. As the pressure of the organic working fluid is temperature dependant, maintaining a constant pressure at different temperatures of the heating fluid is impossible. This results in measurements with constant mass flow rate, but not at constant pressures.

7.2.1 Heat balance

A first indicator of steady state and good measurements is a correct heat balance. This was always performed immediately after each measurement series in order to determine whether it was done correctly. If this was not the case, the measurement was repeated. The first set of measurements, at subcritical conditions, indicated an average difference in the heat balance between the heating fluid and the organic working fluid of 90 W. This
is within the acceptable range as the uncertainty of the heat flux is about 1 % - 2 % at
subcritical conditions.

However, the measurements at near-critical and supercritical conditions showed a larger
difference in the heat balance. An average of 1300 W is found, even with repeated
measurements. This large difference can be explained by the large uncertainty in enthalpy
around the critical region. Uncertainties of 8.52 % up to 25.3 % are found with an average
deviation of 12.7 %. If this is the case, new and improved equations of state for the organic
fluids need to be developed for improved accuracy in the critical region.

Another possibility is achieving pseudo steady state caused by the time delay on the Pt100
temperature sensors at the organic working fluid side. This delay is not constant as it
depends on the temperature, pressure, etc.. This delay then causes a mismatch in the
experimental data, which cannot be retrieved due to the nature of the delay.

The results are nonetheless shown as they might provide some insight in the working of
the test set-up and the supercritical heat transfer phenomena.

7.2.2 Heat transfer results

Figure 7.4 shows the obtained Nusselt numbers for the tube-in-tube heating of R125 at
supercritical pressures in function of the bulk temperature of the working fluid. With
increasing mass flow rates, an increase in the Nusselt number and thus the heat transfer
coefficient is obtained. This is mainly due to the increased heat transferred at higher mass
flow rates. With increasing bulk temperatures of the working fluid, the Nusselt numbers
at mass flow rates of 0.19 kg/s and 0.25 kg/s show an increasing trend. This is not the
case for the highest mass flow rate.

As all the bulk temperatures of the organic working fluid remain the pseudo-critical
temperature (\(T_{pc} = 67.04{^\circ}C\) at 37 bar, \(T_{pc} = 76.45{^\circ}C\) at 45 bar), no conclusion can be
made relating the Nusselt number to the pseudo-critical temperature. More measurements
at bulk temperatures above the pseudo-critical temperature are required.

This increase in Nusselt number at higher mass flow rate is also shown in Figure 7.5.
It shows the Nusselt number in function of the Reynolds number. The Nusselt num-
bbers ranged between 530 and 1061 with corresponding heat transfer coefficients of 969.82
J/(kg.K) and 1931.6 J/(kg.K). The measured overall heat transfer coefficient has a mean
accuracy of 11.95 %. The corresponding mean accuracies of the convection heat transfer
coefficient and the Nusselt number are respectively 12.01 % and 12.14 %.

7.2.3 Pressure drop results

The pressure drop measured with these experiments is very low, between 5.2 kPa and
9.4 kPa with a relative uncertainty of ± 0.04 %. This can be seen in Figure 7.6. The
frictional pressure drop, as calculated in Section 5.2.3, is shown in Figure 7.7. The pressure drop is dependant on the temperature of the heating fluid, which increases as the temperature of the heating fluid increases. Unusual for these measurements is that the frictional pressure drop tends to decrease with increasing mass flow rate, which is the opposite of what is expected as seen in Figure 7.7.
To get a better insight if the pressure drop decreases with increasing mass flow rate, the measured pressure drop is shown in function of the working fluid inlet temperature at constant mass flow rate, Figure 7.8. Figure 7.9 shows the frictional pressure drop in function of the working fluid inlet temperature at constant mass flow rates. At a constant mass flow rate, higher heating fluid temperatures are the dominant factor with respect to the pressure drop. Higher mass flow rates do not correlate to higher frictional pressure drops as seen in Figure 7.9. The inlet temperature of the working fluid also influences the frictional pressure drop. For heating fluid temperatures of 80 °C and 90 °C, an increase in inlet temperature of the working fluid leads to a decrease in the frictional pressure drop.
This could be attributed to the lower viscosity at higher working fluid inlet temperatures but is not consistent for all the measurements, namely at a heating fluid of 100 °C.

Figure 7.8: Pressure drop

Figure 7.9: Frictional pressure drop
7.3 Comparison with literature

7.3.1 Heat transfer

Generally, Gnielinskis correlation [54] is used to evaluate the Nusselt number and the heat transfer coefficient for in-tube single-phase heat transfer for bulk Reynolds number of $2300$ to $10^6$. However, this correlation is developed based on experimental data at subcritical pressures. For fluids at supercritical pressures, the thermophysical properties vary significantly. This will lead to the actual heat transfer coefficients different than those predicted by Gnielinski. New correlations have been proposed by previous researchers such as Zhao and Jiang [2] and Garimella [9]. These two correlations are compared together with the correlations of Petukhov [44], Swenson [45], Shitsman [46] as they were used to design the test section and should show a good affinity with the measured Nusselt numbers. The results are shown in Figure 7.10 at a mass flow rate of 0.19 kg/s, Figure 7.11 at a mass flow rate of 0.25 kg/s, and Figure 7.12 at a mass flow rate of 0.31 kg/s.

Figure 7.10: Comparison of Nusselt numbers from correlation in the literature; $m_{wf} = 0.19$ kg/s

The Gnielinski correlation overpredicts the Nusselt numbers at all mass flow rates. This is because the Gnielinski equation was developed at subcritical pressures and does not account for the rapid property variations around the pseudo-critical temperature.

Petukhov’s correlation does predict the Nusselt number relatively well, but always overpredicts it with a rms error of 6.3755 %. At a mass flow rate of 0.25 kg/s Petukhov’s correlation predicts the measured Nusselt numbers with an accuracy of 0.45 % rms.

Swenson’s correlation always under-predicts the Nusselt number with a rms error of 222 %. This value does not change with the mass flow rate as with Petukhov’s correlation.
On the same note, Shitmans correlation is under-predicting with a rms error of 100.82 %. Together with Swensons and Gnielinskis correlation, it is the worst correlation to fit this data.

Better suited on this data is the correlation proposed by Zhao with an rms error of 16.22 % even though it under-predicts the data. This correlation does account for the thermophysical property variations and was developed with R134a as a working fluid.

Garimellas correlation shows an rms error of 63.53 %, and just like the correlation of Zhao it under-predicts the data. It was developed for an organic fluid, R-404A, with the change of thermophysical properties around the pseudo-critical temperature in mind.

---

**Figure 7.11:** Comparison of Nusselt numbers from correlation in the literature; $\dot{m}_{wf} = 0.25$ kg/s

**Figure 7.12:** Comparison of Nusselt numbers from correlation in the literature; $\dot{m}_{wf} = 0.31$ kg/s
7.4 Model development - Correlation

7.4.1 Wilson plot method

The Wilson plot was developed in 1915 as a way to evaluate the convection coefficients in shell and tube heat condensers by Wilson [55]. The general idea is that the overall heat transfer coefficient can be expressed as a sum of the internal convection heat transfer coefficient, the wall heat transfer resistance and the outer convection heat transfer coefficient:

\[
\frac{1}{U \cdot A_i} = \frac{1}{h_i \cdot A_i} + \frac{\ln(d_o/d_i)}{2 \cdot \pi \cdot \lambda_w \cdot L_w} + \frac{1}{h_o \cdot A_o} + R_{f,i} + R_{f,o} \tag{7.1}
\]

\[
= \frac{1}{h_i \cdot A_i} + R_w + \frac{1}{h_o \cdot A_o} + R_{f,i} + R_{f,o}
\]

\[
= \frac{1}{h_i \cdot A_i} + R_w + \frac{1}{h_o \cdot A_o}
\]

The fouling at the inner and outer surface area of the tube is neglected as the set-up is a new installation. Furthermore, both the heating fluid and the working fluid are compatible with the tubes, so no corrosion is to be expected.

The inner heat transfer coefficient can be seen as a function that depends on the velocity and the other heat transfer coefficients can be considered constant, as theorized by Wilson [55]. If the overall heat transfer coefficient can be determined reliably from experimental measurements, then we can estimate the inner convection heat transfer coefficient as a function of the velocity.

To have an idea of how the correlation should look like, a Wilson Plot is made using the method described below.

The general form of the equation is of the Dittus-Boelter type, \( Nu = C \cdot Re^n \cdot Pr^m \), where the coefficient of the Prandtl number for heating is taken \( n = 0.4 \). This value is often used for heating, as well as \( n = 1/3 \) [56]. It is assumed that the inner heat transfer coefficient, \( htc_i \), is a function of mass flow rate, \( htc_i = f(\dot{m}_{wf}) \).

Calculation of the convection heat transfer coefficient at the working fluid side \( HTC_i \) is done by using equation [7.2]. The convection heat transfer coefficient at the heating fluid side, \( HTC_o \), is unknown and should be determined with a heat transfer correlation suitable for fully developed, turbulent flow with low viscosity fluids.

\[
\frac{1}{U \cdot A_{HT,i}} = \frac{1}{HTC_i \cdot A_{HT,i}} + R_w + \frac{1}{HTC_o \cdot A_{HT,o}} \tag{7.2}
\]

To calculate the convection heat transfer coefficient at the heating fluid side, equation [7.3] is used. It is a Dittus-Boelter type correlation with a viscosity term to account for the
changes in viscosity.

$$
\begin{align*}
N_{u_{hf}} &= 0.023 \cdot R_{e_{b, hf}}^{0.8} \cdot \frac{1}{\nu_{b, hf}} \cdot \left( \frac{\nu_{b, hf}}{\nu_{w, hf}} \right)^{0.14} \\
N_{u_{hf}} &= \frac{H T C_{\alpha} \cdot D_{hyd, hf}}{\lambda_{b, hf}} 
\end{align*}
$$

(7.3)

All the data necessary to calculate the Nusselt number of the heating fluid is available in
the documentation provided by the manufacturer such as the density, the viscosity, etc..

Once the internal convection heat transfer coefficients is known, the Nusselt number can
be calculated:

$$
N_{u_{wf}} = \frac{H T C_{i} \cdot D_{hyd, wf}}{\lambda_{b, wf}}
$$

(7.4)

A graphical representation can be made of the logarithm of the Nusselt number and the
Reynolds number using the measured data. The Nusselt and Reynolds are coupled by
equation (7.5) and this is graphically represented in Figure 7.13.

$$
N_{u} = C \cdot R_{e}^{m} \cdot P_{r}^{0.4}
$$

(7.5)

$$
= C_{0} \cdot R_{e}^{m}
$$

(7.6)

$$
\log (N_{u}) = \log (C_{0} \cdot R_{e}^{m})
$$

(7.7)

$$
= \log (C_{0}) + m \cdot \log (R_{e})
$$

(7.8)

A straight line can be fitted to the data that has a certain slope and intercept. The slope

![Figure 7.13: Wilson Plot method](image)

and intercept correspond to the unknowns in equation (7.6).

Implementing this, a result as seen in Figure 7.14 is obtained. The best straight line has
a slope of $m = 0.6447$ and an intercept of $\log (C_{0}) = -0.4559$. This gives us a value for
the coefficient in equation (7.6) and equation (7.5) of $C_{0} = 0.35$ and $C = 0.2164$ respectively.
The obtained correlation is as follows:

$$
N_{u} = 0.2164 \cdot R_{e}^{0.6447} \cdot P_{r}^{0.4}
$$

(7.9)

The root mean square error of the correlation with the data is 88.85 %. The accuracy of
Figure 7.14: Wilson Plot method with the measurements

the Nusselt number of the organic working fluid varies between 6% and 25% with the main source of uncertainty coming from the overall heat transfer coefficient which varied between 4% and 23%. This is calculated using equation 6.5.

7.4.2 Heat transfer

The Dittus-Boelter correlation is a good correlation when there is a small temperature difference from the bulk of the working fluid to the heating surface of the working fluid. In supercritical heat transfer this is often not the case, and this is why correction factors are added to the standard Dittus-Boelter type correlation.

As the Prandtl number is taken with a fixed exponent that does not need iteration, good correction factors are the viscosity ratio, $\mu_b/\mu_w$, and the specific heat capacity ratio $c_{p,b}/c_{p,w}$. The specific heat capacity ratio and the viscosity ratio are to compensate for the large thermophysical property variations in the vicinity of the pseudo-critical temperature. Another correction factor that needs to be included is the density ratio $\rho_b/\rho_w$. This correction factor accounts for the buoyancy effect, which is a very important phenomenon in horizontal tubes that influences the heat transfer along the height of the tube.

One other correction factor that could be added is the thermal conductivity ratio $\lambda_b/\lambda_w$ accounting for the changes in thermal conductivity at the bulk and wall.

An optimum combination of these factors can be found by combining factors and checking
which correlation has the highest prediction accuracy.

\[
Nu_b = C \cdot Re_b^n \cdot Pr_b^{0.4} \cdot \left( \frac{\mu_b}{\mu_w} \right)^m \cdot \left( \frac{P_b}{P_w} \right)^x \cdot \left( \frac{c_{p,b}}{c_{p,w}} \right)^y \cdot \left( \frac{\lambda_b}{\lambda_w} \right)^z \quad (7.10)
\]

To determine the coefficients of the correction factors, a modified Wilson plot method needs to be implemented as the convection heat transfer coefficient at the inner tube is no longer merely a function of the velocity. It will depend on the other thermophysical properties as well.
Chapter 8

Conclusions

The goal of this master thesis was to control and perform overall and local heat transfer measurements on a test set-up built at Ghent University. This test set-up allows the characterization of heat transfer at supercritical pressures to organic working fluids. From the literature study, it was concluded that there are a limited number of correlations for supercritical heat transfer to organic working fluids and that this supercritical heat transfer is heavily influenced by the thermophysical property variations. It was also concluded that no correlation exist for supercritical heat transfer in large diameter tubes with organic fluids suitable for ORCs.

Experimental data was obtained on the test set-up and the initial measurements are presented in the text. As the set-up is new, automatic control of the parameters was not yet possible because of the unknown dynamic behaviour. This dynamic behaviour is now documented and a suitable control is being developed. During the first measurements, it was soon discovered that the wall mounted thermocouples, that measure the wall temperature at eleven different points, are not attached well to the wall of the inner tube. These thermocouples are heavily influenced by the heating fluid and a new way of attaching has to be considered in further work. The temperature sensors at the in- and outlet of the test section measure the bulk temperature of the heating and working fluid respectively. However, due to the configuration they are placed in, reaching steady-state conditions takes longer. For measurements at steady state this is not a problem, but as the set-up was not yet automatically controlled, steady state was not completely achieved. This influenced on the accuracy of the measurements.

Because of time constraints many of these problems were not solved. Fixing these problems is the logical next step together with the automatic control of the set-up. This would then allow for accurate heat transfer measurements along the test section to better understand the supercritical heat transfer phenomena.

Other work that can be done in the future is replacing the smooth tubes with other
diameters, corrugated tubes, coils and bends to check the influence on the supercritical heat transfer. Cooling instead of heating could also be interesting for applications in those such as gas cooler of a transcritical heat pump cycle.
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## Appendix A

Some excerpts of the MATLAB code used for the data reduction.

### CHAIRMENS

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
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<td>( r_i )</td>
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</tr>
<tr>
<td>( r_o )</td>
<td>Test section inner tube outside tube diameter</td>
</tr>
<tr>
<td>( L_{i,d} )</td>
<td>Test section inner tube inside test diameter</td>
</tr>
<tr>
<td>( L_{o,d} )</td>
<td>Test section outer tube outside test diameter</td>
</tr>
<tr>
<td>( D_{i,wall} )</td>
<td>Outer tube inside diameter, pressurization pipe (in)</td>
</tr>
<tr>
<td>( D_{o,wall} )</td>
<td>Outer tube outside diameter, pressurization pipe (in)</td>
</tr>
<tr>
<td>( b_{hyd} )</td>
<td>Hydraulic diameter, test section (inner tube)</td>
</tr>
<tr>
<td>( b_{hyd,ext} )</td>
<td>Hydraulic diameter, test section (outer tube)</td>
</tr>
<tr>
<td>( D_{hyd,ext} )</td>
<td>Hydraulic diameter, pressurizer</td>
</tr>
<tr>
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<td>Hydraulic diameter, insulator</td>
</tr>
<tr>
<td>( L_{ins} )</td>
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</tr>
<tr>
<td>( L_{ext} )</td>
<td>Insulation outside diameter (measured with caliper)</td>
</tr>
</tbody>
</table>

### LENGTHS

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( L_{FS} )</td>
<td>Length first segment (m)</td>
</tr>
<tr>
<td>( L_{SS} )</td>
<td>Length second segment (m)</td>
</tr>
<tr>
<td>( L_{CT} )</td>
<td>Length cold test (m)</td>
</tr>
</tbody>
</table>

### HEAT CAPACITY OIL

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( C_{oil} )</td>
<td>Heat capacity of oil (J/kg°C)</td>
</tr>
</tbody>
</table>

### THERMAL CONDUCTIVITIES

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( k_{Cu} )</td>
<td>Thermal conductivity of copper (W/m°C)</td>
</tr>
<tr>
<td>( k_{Al} )</td>
<td>Thermal conductivity of aluminum (W/m°C)</td>
</tr>
<tr>
<td>( k_{HR} )</td>
<td>Thermal conductivity of reactor (W/m°C)</td>
</tr>
<tr>
<td>( k_{Ins} )</td>
<td>Thermal conductivity of insulation and blanket (W/m°C)</td>
</tr>
<tr>
<td>( k_{Blk} )</td>
<td>Thermal conductivity of blanket (W/m°C)</td>
</tr>
<tr>
<td>( k_{Ring} )</td>
<td>Thermal conductivity of ring (W/m°C)</td>
</tr>
<tr>
<td>( k_{Lam} )</td>
<td>Thermal conductivity of laminate (W/m°C)</td>
</tr>
<tr>
<td>( k_{Hyd} )</td>
<td>Thermal conductivity of hydraulic (W/m°C)</td>
</tr>
<tr>
<td>( k_{Cal} )</td>
<td>Thermal conductivity of caliper (W/m°C)</td>
</tr>
<tr>
<td>( k_{Extr} )</td>
<td>Thermal conductivity of extrusion (W/m°C)</td>
</tr>
</tbody>
</table>

### Notes

- \( T \) ranges from 0°C to 100°C from reactors to thermal oil bath
- \( q_{oil} \) ranges from 350°C to 400°C from reactors to thermal oil bath
- \( P_{oil} \) ranges from 0 to 100 MPa from reactors to thermal oil bath
LOCAL HEAT TRANSFER COEFFICIENT

\[ A2_{ht} = \frac{p_i d_4}{d_{e} d_{f}^2 \Delta t_{es}} \]

\[ q_1 = \frac{(Q_{hf} - Q_{ta}_{in})/\Delta t_{es}}{A2_{ht}}; \ T_{drop_{wall}} = \frac{(Q_{hf} - Q_{ta}_{in})/\Delta t_{es}}{\log(d_{o}/d_{e})/\Delta t_{es} + \log(d_{e}/d_{f})/\Delta t_{es}} \]

\[ q_2 = \frac{(Q_{hf} - Q_{ta}_{in})/\Delta t_{es}}{A2_{ht}}; \ T_{drop_{wall}} = \frac{(Q_{hf} - Q_{ta}_{in})/\Delta t_{es}}{\log(d_{o}/d_{e})/\Delta t_{es} + \log(d_{e}/d_{f})/\Delta t_{es}} \]

\[ q_3 = \frac{(Q_{hf} - Q_{ta}_{in})/\Delta t_{es}}{A2_{ht}}; \ T_{drop_{wall}} = \frac{(Q_{hf} - Q_{ta}_{in})/\Delta t_{es}}{\log(d_{o}/d_{e})/\Delta t_{es} + \log(d_{e}/d_{f})/\Delta t_{es}} \]

\[ q_4 = \frac{(Q_{hf} - Q_{ta}_{in})/\Delta t_{es}}{A2_{ht}}; \ T_{drop_{wall}} = \frac{(Q_{hf} - Q_{ta}_{in})/\Delta t_{es}}{\log(d_{o}/d_{e})/\Delta t_{es} + \log(d_{e}/d_{f})/\Delta t_{es}} \]

\[ T_{hf}(1) = \frac{T_{A_{avg}} - T_{hf}_{in}}{1}; \ T_{hf}(2) = \frac{T_{A_{avg}} - T_{hf}_{in}}{2}; \ T_{hf}(3) = T_{A_{avg}}; \]

\[ T_{hf}(4) = \frac{T_{A_{avg}} - T_{A_{avg}}}{2}; \ T_{hf}(5) = \frac{T_{A_{avg}} - T_{A_{avg}}}{2}; \ T_{hf}(6) = T_{A_{avg}}; \]

\[ T_{hf}(7) = \frac{T_{A_{avg}} - T_{A_{avg}}}{2}; \ T_{hf}(8) = \frac{T_{A_{avg}} - T_{A_{avg}}}{2}; \ T_{hf}(9) = T_{A_{avg}}; \]

\[ T_{hf}(10) = \frac{T_{A_{avg}} - T_{hf}_{in}}{1}; \ T_{hf}(11) = T_{A_{avg}}; \]

\[ U_{loc_{avg}} = \text{zeros}(1,11); \ U_{loc_{bot}} = \text{zeros}(1,11); \ U_{loc_{sid}} = \text{zeros}(1,11); \ U_{loc_{top}} = \text{zeros}(1,11); \]

\[ K; \% \text{Intermediate variable} \]

for \( i = 1:11 \)

- if \( i < 4 \)
  \[ q_{\text{temp}} = q_1; \ T_{\text{drop temp}} = T_{drop_{wall}}; \]
- elseif \( i < 7 \)
  \[ q_{\text{temp}} = q_2; \ T_{\text{drop temp}} = T_{drop_{wall}}; \]
- elseif \( i < 10 \)
  \[ q_{\text{temp}} = q_3; \ T_{\text{drop temp}} = T_{drop_{wall}}; \]
- else
  \[ q_{\text{temp}} = q_4; \ T_{\text{drop temp}} = T_{drop_{wall}}; \]
end

\[ U_{loc_{avg}(i)} = q_{\text{temp}}/(T_{hf}(i) - (T_{hf}(i) - T_{\text{drop temp}})); \% \text{UNNECESSARY} \]

\[ U_{loc_{bot}(i)} = q_{\text{temp}}/(T_{hf}(i) - (T_{hf}(i) - T_{\text{drop temp}})); \]

\[ U_{loc_{sid}(i)} = q_{\text{temp}}/(T_{hf}(i) - (T_{hf}(i) - T_{\text{drop temp}})); \]

\[ U_{loc_{top}(i)} = q_{\text{temp}}/(T_{hf}(i) - (T_{hf}(i) - T_{\text{drop temp}})); \]

\[ K = 5; \]
end

\[ U_{loc} = [U_{loc_{avg}}; U_{loc_{bot}}; U_{loc_{sid}}; U_{loc_{top}}]; \]

Pressure drop working fluid

\[ \Delta p_{w_{f}} = p_{ta_{in}} - p_{ta_{out}}; \]

\[ \rho_{w_{f}} = \text{CoolProp} \cdot \text{PropsSI}('W', 'T', T_{w_{f}_{ta_{in}}}, 'P', p_{ta_{in}}, 'W'); \]

\[ \rho_{w_{f}} = \text{CoolProp} \cdot \text{PropsSI}('W', 'T', T_{w_{f}_{ta_{in}}}, 'P', p_{ta_{in}}, 'W'); \]

\[ \Delta p_{w_{f}} = \rho_{w_{f}} / (\text{cho_{w_{f}}_{in}} - 1) / \text{cho_{w_{f}}_{in}}; \]

\[ \mu_{w_{f}} = \text{CoolProp} \cdot \text{PropsSI}('W', 'T', T_{w_{f}_{ta_{in}}}, 'P', p_{ta_{in}}, 'W'); \]

\[ A_{f} = \text{pi} \cdot d_{4} / 4; \]

\[ K_{f} = 0; \]

\[ f = 0.5 \cdot 0.5; \]

\[ \Delta w_{f} = 1; \]
FUNCTION T = CalcUIN(h_in, h_out, h_wf, h_sil, Q_sil, T_wf_in, T_sil_out, p_in, p_out, A_tr, WF)
    Delta_h = abs(h_out-h_in)/h;
    Delta_wf = delta_h*WF;
    Delta_sil = Q_sil/h;
    p = linspace(p_in,p_out,N);
    T_wf = zeros(1,N+1);
    T_sil = zeros(1,N+1);
    T_wf(i) = T_wf_max; \( i \) is the inlet of the test section for working fluid
    T_sil(i) = T_sil_out; \( i \) is the outlet of the test section for oil
    h_temp = h_sil;
    LMDT = zeros(1,N); \( \) LMDT(i) gives LMDT in each cell i
    for [i=1] (N+1)
        T_temp = h_temp + Delta_h;
        T_wf(i) = CoolProp.PropsSI('T','H',h_temp,'P',p(i),WF);
        T_sil(i) = CalcQsil(T_sil(i-1));
        LMDT(i) = ((T_sil(i)-T_wf(i)-T_sil(i-1)-T_wf(i-1))/log((T_sil(i)-T_wf(i))/T_sil(i-1)));
    end
end

function T = CalcQsil(T_sil)
    T_sil_table = [50.18 51.08 52.18 53.18 54.18 55.18 56.18 57.18 58.18 59.18]; \( T \) ranges from 0\(^\circ\)C to 120\(^\circ\)C from datasheet thermal oil unit
    op_sil_table = [1590 1990 2030 2070 2100 2140 2180 2220 2260 2290];
    op = interp1(T_sil_table, op_sil_table, T_sil);
end