Experimental investigation of flow boiling in horizontal tubes at low-temperature subcritical ORC conditions

Bram Baert

Supervisor: Prof. dr. ir. Michel De Paepe
Counsellors: Alihan Kaya, Dr. ir. Steven Lecompte

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
Faculty of Engineering and Architecture
Academic year 2016-2017
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Ghent, June 2017

The author

Bram Baert
Preface

I want to thank my supervisor, Alihan Kaya, for his guidance throughout the year, for proof reading my text multiple times and for his trust in my ideas. My gratitudes go to Frederik Martens and Patrick De Pue. Without their technical skills, the construction of the test facility would have been impossible. I would like to give special thanks to my promoter, Prof. Michel De Paepe, for making this project possible. I also want to thank Bernd Ameel and Steven Lecompte, for helping solving the problems of the test facility.

Furthermore, I want to thank my girlfriend for her support and grammatical advice. Finally, I want to thank my friends and family for their interest in my work.
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Summary
The aim of this master thesis is to perform two-phase flow experiments in low temperature ORC conditions. This is done using a test facility built for this purpose.

In the first chapter, the situation and reasons for conducting this investigation are introduced. Thereafter, a literature review follows. It consists of the general functioning of an ORC, some details about the used experimental fluid and a section about two-phase flow heat transfer experiments.

The third chapter describes the experimental test facility. An overview is given of its main components, and they are illustrated in detail. The working principle of the measuring equipment is explained and the functioning of the data acquisition is clarified. The final part of this chapter describes the dimensioning of the new components of the test facility.

The data reduction is discussed in chapter four. It contains the equations and methods for calculating all important quantities of this research, and it includes an estimation of the heat loss in the test section.

Subsequently, there is a chapter about the accuracy of the measurements. The uncertainty of the measuring equipment is reviewed and an uncertainty analysis is done to estimate the uncertainties of the calculated physical quantities. In the last part of this chapter, these uncertainties are evaluated and discussed.

Thereafter, the chapter about the design of experiments follows. It consists of the different techniques that can be used, and one of these techniques is applied in ORC conditions. For these points, an existing flow pattern map and heat transfer correlation were implemented.

The seventh chapter explains the different control loops and the control strategy that are used to control the test facility.

Keywords  Organic Rankine Cycle, flow boiling, experimental, Solkatherm® SES36
Extended abstract
EXPERIMENTAL INVESTIGATION OF FLOW BOILING IN HORIZONTAL TUBES AT LOW-TEMPERATURE SUBCRITICAL ORC CONDITIONS

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ABSTRACT

The Organic Rankine Cycle (ORC) is suitable for generating electricity from geothermal energy, solar energy, biomass and waste heat. It is currently employed for recuperating energy from exhaust gases originating from industrial processes. The ORC is preferred above the classical Rankine cycle in low temperature applications. The evaporator is a key component of an ORC and its design has an impact on the cost and efficiency of the system. This research focuses on the optimization of the evaporator design for waste heat recovery applications, which is done using heat transfer and pressure drop correlations. The correlations will be developed for fluids that are frequently used in these systems. A test facility was built for conducting two-phase flow boiling experiments in ORC conditions: relatively low temperatures and large diameters. In the test section, the local heat transfer coefficients and the two-phase pressure drops are measured, and they are used to make the correlations. As a first step, the evaporation of Solkatherm® SES36 is investigated in a horizontal tube with an inner diameter of 21 mm. The heat transfer measurements are done at saturation temperatures of 110, 115 and 120 °C. The mass velocity is varied between 100 and 300 kg/(m²⋅s), and the heat flux between 5 and 20 kW/m².

INTRODUCTION

Organic Rankine Cycles have promising characteristics for use in the following applications: biomass combined heat and power, geothermal energy, solar power plants and waste heat recovery [1]. This investigation focuses on the recovery of low-grade waste heat (~100 °C – 250 °C), which accounts for 50 % or more of the heat generated in the industry [2-3]. By employing an ORC to recover this waste heat, the overall thermal efficiency of the industrial process increases. Cement production is one of the most energy intensive industries in the world, it is responsible for 5 % of the global greenhouse gas emissions [4-5]. In this industry, waste heat is emitted at temperatures between 215 °C and 315 °C, which can be recuperated with ORCs. Other applications of waste heat recuperating ORCs include internal combustion engines, the chemical and glass industry, etc.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Description</th>
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<tbody>
<tr>
<td>BSL</td>
<td>-</td>
<td>Best straight line</td>
</tr>
<tr>
<td>D</td>
<td>m</td>
<td>Diameter of the tube</td>
</tr>
<tr>
<td>EPDM</td>
<td>-</td>
<td>Ethylene propylene diene monomer rubber</td>
</tr>
<tr>
<td>FS</td>
<td>-</td>
<td>Full scale</td>
</tr>
<tr>
<td>G</td>
<td>kg/(m²⋅s)</td>
<td>Mass velocity</td>
</tr>
<tr>
<td>h</td>
<td>J/kg</td>
<td>Enthalpy</td>
</tr>
<tr>
<td>h</td>
<td>W/(m²⋅K)</td>
<td>Heat transfer coefficient</td>
</tr>
<tr>
<td>L</td>
<td>m</td>
<td>Length</td>
</tr>
<tr>
<td>m</td>
<td>kg/s</td>
<td>Mass flow rate</td>
</tr>
<tr>
<td>ORC</td>
<td>-</td>
<td>Organic Rankine Cycle</td>
</tr>
<tr>
<td>p</td>
<td>Pa</td>
<td>Pressure</td>
</tr>
<tr>
<td>Q</td>
<td>W</td>
<td>Heating power</td>
</tr>
<tr>
<td>q</td>
<td>-</td>
<td>General physical quantity</td>
</tr>
<tr>
<td>q̇</td>
<td>W/m²</td>
<td>Heat flux</td>
</tr>
<tr>
<td>q̇''</td>
<td>W/m³</td>
<td>Volumetric heat flux</td>
</tr>
<tr>
<td>T</td>
<td>K</td>
<td>Temperature</td>
</tr>
<tr>
<td>TEB</td>
<td>-</td>
<td>Thermal error band</td>
</tr>
<tr>
<td>x</td>
<td>-</td>
<td>Vapor quality</td>
</tr>
<tr>
<td>X</td>
<td>-</td>
<td>General physical quantity</td>
</tr>
<tr>
<td>y</td>
<td>m</td>
<td>Distance measured from the test section inlet</td>
</tr>
</tbody>
</table>

Greek characters

<table>
<thead>
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<th>Symbol</th>
<th>Description</th>
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<tr>
<td>δ</td>
<td>Absolute error</td>
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<tr>
<td>λ</td>
<td>Thermal conductivity</td>
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Subscripts

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<thead>
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<tr>
<td>acc</td>
<td>Accuracy</td>
</tr>
<tr>
<td>age</td>
<td>Age</td>
</tr>
<tr>
<td>av</td>
<td>Average</td>
</tr>
<tr>
<td>bot</td>
<td>Bottom (of the tube)</td>
</tr>
<tr>
<td>b</td>
<td>Bulk</td>
</tr>
<tr>
<td>g</td>
<td>Vapor</td>
</tr>
<tr>
<td>i</td>
<td>Inner</td>
</tr>
<tr>
<td>in</td>
<td>Inlet</td>
</tr>
<tr>
<td>l</td>
<td>Liquid</td>
</tr>
<tr>
<td>line</td>
<td>Line (pressure)</td>
</tr>
<tr>
<td>o</td>
<td>Outer</td>
</tr>
<tr>
<td>out</td>
<td>Outlet</td>
</tr>
<tr>
<td>ph</td>
<td>Preheater</td>
</tr>
<tr>
<td>side</td>
<td>Side (of the tube)</td>
</tr>
<tr>
<td>sat</td>
<td>Saturated</td>
</tr>
<tr>
<td>top</td>
<td>Top (of the tube)</td>
</tr>
<tr>
<td>temp</td>
<td>Temperature</td>
</tr>
<tr>
<td>tot</td>
<td>Total</td>
</tr>
<tr>
<td>tp</td>
<td>Two-phase</td>
</tr>
<tr>
<td>ts</td>
<td>Test section</td>
</tr>
<tr>
<td>w</td>
<td>Wall</td>
</tr>
</tbody>
</table>
In Diesel engines, an increase in thermal efficiency of more than 6% can be obtained with a bottoming ORC [6].

An ORC is a Rankine cycle that uses an organic fluid instead of water. The advantage is that these fluids have lower boiling points, which makes them better adapted for recuperation of low-grade waste heat [3]. Compared with the classical Rankine cycle, the ORC is more appropriate for moderate power ranges and/or low temperature applications [1]. ORCs with low temperature heat sources perform better than the water/steam cycle with the same heat source, even at low or moderate evaporation pressures [7].

A typical ORC consists of an evaporator, a turbine, a condenser and a pump, as can be seen in Figure 2. Sometimes, a recuperator is added to this system. This can be beneficial, depending on the working fluid [8], because the evaporator has a smaller heat input in that case. In the evaporator, heat is extracted from the hot exhaust gases, which are cooled down, and the organic fluid is evaporated. Thus, the working fluid enters the turbine in a vapor state. There it is expanded to a lower pressure and work is extracted. This work is used to drive a generator, which converts it into electricity. Then, the fluid passes through the recuperator, where heat is extracted to preheat the liquid before it enters the evaporator. Next, the fluid is condensed completely and finally it is brought back to the high-pressure level by the pump.

Current research is focusing on several aspects to increase the efficiency of the ORC and make it economically feasible: the selection of the working fluid, the turbine or expander and the design of the evaporator and condenser [1,3]. An appropriate evaporator design has a large impact on both the efficiency and the cost of the system [9]. If the evaporator is designed too small, then it is possible that the fluid is not evaporated completely, which can lead to turbine or expander damage.

Designing the evaporator too large, leads to a higher cost and too much superheat, which is detrimental for the thermal efficiency [9]. With a precise knowledge of the heat transfer coefficients, the amount of heat transferred to the fluid in the evaporator is known, and the length of the evaporator can be chosen so that the fluid evaporates completely.

Solkatherm® SES36 is a fluid that has promising properties for waste heat recovery applications with flue gas temperatures below 400°C, because of its relatively high vapor density, which results in a smaller system size [10-11]. The currently existing heat transfer correlations were not validated for Solkatherm® SES36. Most of the research focuses on two-phase flow heat transfer experiments for applications in the heat pump, refrigeration or air conditioning industry [12], with saturation temperatures ranging between -20°C and 40°C. At higher saturation temperatures and consequently higher reduced temperatures, the thermodynamic properties shift, which has an influence on the liquid-vapor interactions, the occurring flow patterns and thus also the heat transfer [12]. Making use of the existing correlations would give rise to relatively large error margins in the design of these evaporators. Therefore, it is
necessary that the heat transfer properties of this fluid are investigated experimentally [9]. Apart from an investigation of the dryout of Solkatherm® SES36 in vertical, small diameter tubes, there is no experimental data of the heat transfer of this fluid available in literature [13].

Hence, a test facility was built for conducting two-phase flow experiments in horizontal tubes at subcritical ORC conditions. In the test section, the local heat transfer coefficients and the two-phase pressure drop are observed. The test facility is suited to have test sections with different tube diameters and tube materials. The first experiments are done in a 21 mm internal diameter, horizontal tube. This relatively large diameter is chosen because the evaporation heat of organic fluids is typically ten times smaller than that of water, which results in a higher mass flow rate. The consequence is that a larger feed pump is needed [7]. In addition to that, the pressure drop in the evaporator increases significantly when the tube diameter becomes smaller than a critical value, and the critical value of the diameter increases for increasing mass flow rates. This is because the flow pattern changes to annular flow. The increase of the pressure drop attenuates the temperatures difference that drives the heat transfer, which results in a longer evaporator length [14]. So, the larger diameters are chosen to reduce the pressure drop and consequently to avoid an increase of the length of the evaporator.

**TEST FACILITY DESCRIPTION**

In the test facility, the mass velocity, the inlet and outlet vapor quality, the heat flux and the saturation pressure are controlled. This way, experiments can be done in different thermodynamic conditions. A schematic overview of the test facility is given in Figure 1. A magnetically coupled sliding vane pump ensures the circulation of the experimental fluid and increases the pressure. The pump is driven by an induction motor with a variable frequency. This way, the speed of the pump is easily adapted. After the pump, the fluid passes through a Coriolis mass flow meter. Then it enters the electrical preheater, where the liquid is heated up until it reaches the desired inlet condition. Subsequently, it enters the test section, where it is evaporated (partially). Next, it is expanded to a lower pressure by an electronic expansion valve. After that, it passes through the condenser, where heat is extracted by a water/glycol mixture. The experimental fluid is condensed and ends up in the liquid receiver. This is a liquid-vapor reservoir. At the bottom, a tube is connected to the subcooler, such that the liquid is subcooled a few degrees and cavitation is avoided in the pump. Between the pump and the mass flow meter, a bladder accumulator is placed. Its function is to dampen pressure and mass flow rate fluctuations. All electric components are connected to a data acquisition system (DAQ), which is connected to a personal computer. The computer is used to control and monitor these devices and to read out and log the measurement data.

The mass velocity in the test section is controlled by adapting the rotation speed of the pump. Because it is a positive displacement pump, the mass flow rate is proportional with the rotation speed. By modifying the power of the preheater, the inlet vapor quality changes. The outlet vapor quality follows from the heat flux in the test section.

The test section is heated by the Joule effect; both ends of the tube are connected to a DC power supply of 12 kW. An electrical current flows through the test section, depending on the voltage difference across the test section and the electrical resistance of the tube. The electrical energy is dissipated and heat is produced. The heating power can be adapted by changing the voltage and consequently the current through the tube. At the inlet of the test section, a capacity sensor is installed. It is used to get an estimation of the vapor quality and to detect the flow pattern. At the inlet and at the outlet, sight glasses are present.

![Figure 3: Cross section of the test section](image)

![Figure 4: Side view of the test section without insulation, with the locations of the thermocouples indicated](image)

In Figure 3, the cross section of the test section is shown. Around the circumference of the tube, three thermocouples are placed: one at the top, one at the side and one at the bottom. In total, thermocouples are placed at 24 axial locations along the test section. At every axial location, there are three thermocouples. However, the thermocouple at the side is placed alternately at the left and at the right sides of the tube. This can be seen in a side view of the test section, in Figure 4. Around the tube of the test section, a Kapton® MT polyimide film is wrapped to insulate it electrically. This film is suitable because it is a good thermal conductor and it is very thin. The thermocouples are attached to this tape, and a thermal paste is spread on it to increase the thermal conductivity. Around the thermocouples, another film of Kapton® MT polyimide is wrapped, such that they remain in place. Finally, an EPDM insulation tube with a thickness of 32 mm is placed around this to minimize heat losses. Some heat is lost to the environment via conduction through the insulation and natural convection at the outside of the insulation. The heat loss was estimated by assuming a one-dimensional model with these two thermal resistances in series. For an outer wall temperature of 120 °C, the heat loss is about 45 W.

The thermocouples in the test section measure the temperature of the tube’s outer wall. The temperature at the inside of the tube, which is one of the components to calculate the local heat transfer coefficient, can be calculated from this.

![Diagram](image)
The inlet and outlet of the test section are connected to a differential pressure transducer, such that the two-phase pressure drop is measured.

DATA REDUCTION

At the inlet of the test section, the absolute pressure is measured. With the differential pressure transducer, the pressure at the outlet of the test section is also known. The pressure between these points is obtained using linear interpolation:

\[ p(y) = p_{ts,in} - \Delta p_{test} \cdot \frac{y}{L_{ts}} \]  

(1)

In which \( y \) is the distance measured from the inlet of the test section and \( L_{ts} \) is the total length of the test section. An energy balance of the preheater results in:

\[ \dot{Q}_{ph} = \dot{m} \cdot (h_{ph,out} - h_{ph,in}) \]  

(2)

In most of the cases, the fluid will be partly evaporated at the preheater outlet. Then the vapor quality can be calculated. It is defined as:

\[ x_{ph,out} = \frac{h_{ph,out} - h_{ph,l,sat}}{h_{ph,g,sat} - h_{ph,l,sat}} \]  

(3)

If equation 2 is substituted into equation 3, then the following equation is obtained for the vapor quality at the outlet of the preheater:

\[ x_{ph,out} = \frac{h_{ph,in} + \frac{\dot{Q}_{ph}}{m} - h_{ph,l,sat}}{h_{ph,g,sat} - h_{ph,l,sat}} \]  

(4)

The vapor quality at the inlet of the test section, is approximately equal to the vapor quality at the outlet of the preheater: \( x_{ts,in} \approx x_{ph,out} \). The vapor quality at the inlet of the test section follows from the enthalpy at the inlet of the preheater, the heating power of the preheater, the mass flow rate and the saturated liquid and saturated vapor enthalpies at the inlet of the test section. From the pressure at the inlet of the test section, the saturated liquid and vapor enthalpies can be evaluated, this is done using CoolProp [15]. The enthalpy at the inlet of the preheater can also be evaluated, with the knowledge of the temperature and pressure at that location. The mass flow rate is known, by the measurement of the Coriolis mass flow meter. From equation 4, it can then be understood that the inlet vapor quality \( x_{ts,in} \) can be adapted by changing the power of the preheater, \( \dot{Q}_{ph} \).

The same thing can be done for the test section:

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

(5)

\[ x_{ts,out} = \frac{h_{ts,out} - h_{ts,l,sat}}{h_{ts,g,sat} - h_{ts,l,sat}} \]  

(6)

\[ x_{ts,out} = \frac{h_{ts,in} + \frac{\dot{Q}_{ts}}{m} - h_{ts,l,sat}}{h_{ts,g,sat} - h_{ts,l,sat}} \]  

(7)

In these equations, it is assumed that the fluid is not completely evaporated at the outlet of the test section. In equation 7, \( h_{ts,0,sat} \) and \( h_{ts,l,sat} \) are evaluated at the local pressure, the pressure at the outlet of the test section. The vapor quality at a location \( y \) in the test section is then found as follows:

\[ x_{ts}(y) = \frac{h_{ph,in} + \frac{\dot{Q}_{ph}}{m} + y \cdot \frac{\dot{Q}_{ts}}{m} - h_{ts,l,sat}}{h_{ts,g,sat} - h_{ts,l,sat}} \]  

(8)

The mass velocity in the test section can be evaluated from the mass flow rate and the inner diameter of the test section:

\[ G = \frac{\dot{m}}{\pi \cdot D^2} \]  

(9)

The heat flux is the heating power per surface unit of heat transfer. It is evaluated using the following formula:

\[ \dot{q} = \frac{\dot{Q}_{ts}}{L_{ts} \cdot \pi \cdot (D^2 - D_l^2)} \]  

(10)

The volumetric heat flux is required to calculate the inner wall temperature, it is defined as follows:

\[ \dot{q}^v = \frac{\dot{q}}{L_{ts}} \]  

In the test section, the two-phase heat transfer of fluids is investigated. The heat transfer is typically expressed with a local heat transfer coefficient, defined as follows:

\[ h_{tp} = \frac{\dot{q}}{T_{w,i} - T_b} \]  

(12)

with \( \dot{q} \) the heat flux, \( T_{w,i} \) the temperature at the inner wall of the tube and \( T_b \) the bulk temperature of the fluid. The latter temperature is evaluated directly from the saturation pressure in case of a two-phase flow, because there is a one-to-one relationship between saturation pressure and saturation temperature. The saturation pressure at a distance \( y \) from the test section inlet is obtained from equation 1. The inner wall temperature can be calculated from the outer wall temperature by assuming a one-dimensional, steady-state heat conduction model. Due to the Joule heating, there is a combined effect of heat conduction and heat generation in the tube. By assuming that the tube is perfectly insulated at its outer diameter, the following equation is obtained:

\[ T_{w,i} = T_{w,o} + \frac{\dot{q}^v \cdot (D_o^2 - D_l^2)}{16 \cdot \lambda} - \left( \frac{\dot{q}^v}{8 \cdot \lambda} \right) \cdot D_o \]  

(13)

Since there are three thermocouples placed at every axial location, an average local heat transfer coefficient can be calculated:

\[ h_{tp,av} = \frac{h_{top} + h_{bot} + h_{side}}{3} \]  

(14)
ERROR ANALYSIS

The different components of the test facility have uncertainties that propagate in the measured physical quantities. All thermocouples have been calibrated individually, and their uncertainties vary between 0.07 K and 0.11 K. There are two types of pressure transducers. The first one measures the pressure at the inlet of the test section, and the second one measures the pressure drop in the test section. Both are used to determine the local saturation pressure and the bulk temperature of the fluid. The pressure transducers have an accuracy of 0.04 % FS BSL. In addition to that, there is also a thermal error. In the compensated temperature range, from -40 °C to 125 °C, this is ± 0.24 % FS TEB. The full-scale (FS) uncertainties need to be multiplied by the total range of the sensor, 45 bar and 35 kPa for the pressure sensor at the inlet of the test section and the differential pressure sensor, respectively. For the differential pressure sensor, two additional effects lead to an uncertainty: the line pressure and the age effect. These uncertainties are combined as follows:

\[ \delta p_{\text{tot}} = \delta p_{\text{acc}}^2 + \delta p_{\text{temp}}^2 + \delta p_{\text{line}}^2 + \delta p_{\text{age}}^2 \]  

This leads to a total uncertainty of 10.9 kPa for the pressure sensor at the inlet of the test section and 0.4 kPa for the differential pressure sensor. The mass flow rate sensor was calibrated, and it has an uncertainty of 0.04 %. The heater of the test section has relative uncertainties of 0.2 % and 0.1 % of the voltage and current respectively. This results in a relative uncertainty of the heating power of the test section of 0.5 %. The uncertainty of the heating power of the preheater is ±1.45/-2.9 %. All uncertainties of the measuring equipment are summarized in Table 1. Moreover, also the geometrical uncertainties and the uncertainties of the fluid properties were considered in the error analysis.

Table 1: Maximal uncertainties of measuring equipment

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Device</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>Thermocouple</td>
<td>±0.107 K</td>
</tr>
<tr>
<td>Pressure</td>
<td>Absolute</td>
<td>±10.9 kPa</td>
</tr>
<tr>
<td>Pressure</td>
<td>Differential</td>
<td>±0.4 kPa</td>
</tr>
<tr>
<td>Mass flux</td>
<td>Coriolis</td>
<td>±0.04 %</td>
</tr>
<tr>
<td>Heating power</td>
<td>Voltmeter</td>
<td>±0.2 %</td>
</tr>
<tr>
<td>test section</td>
<td>Ampèremeter</td>
<td>±0.1 %</td>
</tr>
<tr>
<td>Heating power</td>
<td>Wattmeter</td>
<td>+1.5/-2.9 %</td>
</tr>
</tbody>
</table>

Uncertainties propagate in the physical quantities that depend on them. By making use of the following equation, the error propagation can be evaluated [16]:

\[ \delta q = \sqrt{\sum_i \left( \frac{\partial q}{\partial X_i} \delta X_i \right)^2} \]  

where \( q \) is a physical quantity which is determined by other physical quantities \( X_i \), that have uncertainties \( \delta X_i \). This way, the uncertainties of the mass velocity, the heat flux, the vapor quality, the inner wall temperature and the local heat transfer coefficient can be determined. For the local heat transfer coefficient, the uncertainty propagation may be written in the form of the following equation:

\[ \frac{\delta h_{tp}}{h_{tp}} = \sqrt{\left( \frac{\delta q}{q} \right)^2 + \left( \frac{\delta T_{w,i}}{T_{w,i} - T_b} \right)^2 + \left( \frac{\delta T_b}{T_{w,i} - T_b} \right)^2} \]  

(17)

The uncertainty of the vapor quality results from uncertainties in the enthalpy, the heating power of the preheater and test section, the mass flow rate and geometrical uncertainties (see equation 8). The uncertainties of the mass velocity, heat flux, vapor quality and local heat transfer coefficient vary depending on the experimental condition. The uncertainty of the heat transfer coefficient in a certain experimental condition can only be determined after doing the experiments, because the local wall temperatures are unknown initially, as can be seen in equation 17. To have an idea of the uncertainty of the local heat transfer coefficient, the Kandlikar correlation [17] is used to estimate the local inner wall temperatures and the local heat transfer coefficients. The maximum relative uncertainty of the local heat transfer coefficients is 23 %. The uncertainties are shown in Table 2.

Table 2: Maximal uncertainties of calculated quantities

<table>
<thead>
<tr>
<th>Quantity &amp; unit</th>
<th>Value</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>( G ) [kg/(m²·s)]</td>
<td>100</td>
<td>0.2</td>
</tr>
<tr>
<td></td>
<td>300</td>
<td>0.6</td>
</tr>
<tr>
<td>( \dot{q} ) [W/m²]</td>
<td>5·10³</td>
<td>22.9</td>
</tr>
<tr>
<td></td>
<td>20·10³</td>
<td>91.4</td>
</tr>
<tr>
<td>( x ) [/]</td>
<td>0-1</td>
<td>0.01</td>
</tr>
<tr>
<td>( h_{tp} ) [W/(m²·K)]</td>
<td>1484</td>
<td>284</td>
</tr>
<tr>
<td></td>
<td>4837</td>
<td>643</td>
</tr>
</tbody>
</table>

DESIGN OF EXPERIMENTS

In Table 3, an overview is given of the Design of Experiments (DOE). In the DOE, three levels occur for every factor: the mass velocity, the saturation pressure and the heat flux. The saturation temperature corresponding with the saturation pressure is also shown in Table 3. The levels of the saturation temperature correspond with conditions inside a waste heat recovery ORC evaporator. The heat flux is varied between 5 kW/m² and 20 kW/m², because this corresponds with typical heat fluxes in ORCs [18-19]. The levels of these factors will be combined in a full factorial DOE. This means that all combinations of the three factors will be tested [20], leading to in total \( 3^3 = 27 \) samples. The advantage is that the main and interaction effects can clearly be observed from this DOE. From equation 18, it can be understood that when the heat flux, the geometrical parameters (\( D_i \) and \( L_{ts} \), and the mass velocity are fixed, the enthalpy increase in the test section (\( \Delta h \) ) is also fixed. Therefore, the change of vapor quality is fixed in the test section.

\[ \dot{Q}_{ts} = \dot{q} \cdot \pi \cdot D_i \cdot L_{ts} = \dot{m} \cdot \Delta h = G \cdot \pi \cdot D_i^2 \cdot \Delta h \]  

(18)
In combinations of design parameters with low heat fluxes and high mass velocities, the enthalpy increase is relatively small, the fluid is only evaporated partially. Therefore, the combinations will be subdivided in multiple experiments, each starting and ending with different vapor qualities, such that the entire range of vapor qualities between 0 and 1 is covered. For example, the combination of $G = 100 \text{ kg/(m}^2\text{s)}, p_{\text{sat}} = 7.93 \text{ bar}$ and $\dot{q} = 5 \text{ kW/m}^2$, is split up in 5 experiments. The first one starts with an inlet vapor quality of $x_{\text{ts,in}} = 0$, the second one with $x_{\text{ts,in}} = 0.2$ and the last one with $x_{\text{ts,in}} = 0.8$.

<table>
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<tr>
<th>$G$ [kg/(m$^2$s)]</th>
<th>$p_{\text{sat}}$ [bar]</th>
<th>$T_{\text{sat}}$ [°C]</th>
<th>$\dot{q}$ [kW/m$^2$]</th>
</tr>
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<tr>
<td>100</td>
<td>7.93</td>
<td>110</td>
<td>5</td>
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<tr>
<td>200</td>
<td>8.84</td>
<td>115</td>
<td>10</td>
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<tr>
<td>300</td>
<td>9.83</td>
<td>120</td>
<td>20</td>
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**CONCLUSION**

ORCs can be used to recover low-grade waste heat. The fluid Solkatherm® SES36 is already used in these applications, but its flow boiling characteristics are unknown. To improve the efficiency and the cost of these systems, heat transfer and pressure drop correlations will be developed. A test facility was constructed for performing two-phase flow boiling experiments. The experiments will be done in ORC conditions: relatively low saturation temperatures and large diameters. The uncertainties of the experiments were estimated. For the local heat transfer coefficient, the maximal relative uncertainty is 23%. The next steps are doing the experiments and making the correlations.

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## Abbreviations

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<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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</thead>
<tbody>
<tr>
<td>A</td>
<td>annular</td>
</tr>
<tr>
<td>BSL</td>
<td>best-straight-line</td>
</tr>
<tr>
<td>CCD</td>
<td>central composite design</td>
</tr>
<tr>
<td>CS</td>
<td>capacitance sensor</td>
</tr>
<tr>
<td>D</td>
<td>dryout</td>
</tr>
<tr>
<td>DAQ</td>
<td>data acquisition system</td>
</tr>
<tr>
<td>DOE</td>
<td>design of experiments</td>
</tr>
<tr>
<td>EPDM</td>
<td>ethylene propylene diene monomer rubber</td>
</tr>
<tr>
<td>FS</td>
<td>full scale</td>
</tr>
<tr>
<td>HSC</td>
<td>high speed camera</td>
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<td>I</td>
<td>intermittent</td>
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<td>M</td>
<td>mist</td>
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<tr>
<td>NPSH</td>
<td>net positive suction head</td>
</tr>
<tr>
<td>ORC</td>
<td>organic Rankine cycle</td>
</tr>
<tr>
<td>PID</td>
<td>proportional–integral–derivative</td>
</tr>
<tr>
<td>S</td>
<td>stratified</td>
</tr>
<tr>
<td>SG</td>
<td>sight glass</td>
</tr>
<tr>
<td>SW</td>
<td>stratified-wavy</td>
</tr>
<tr>
<td>TEB</td>
<td>thermal error band</td>
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<tr>
<td>TWC</td>
<td>triple point of water cell</td>
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## Greek symbols

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<td>K$^{-1}$</td>
</tr>
<tr>
<td>$\delta$</td>
<td>error</td>
<td>-</td>
</tr>
<tr>
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<td>Unit</td>
</tr>
<tr>
<td>--------</td>
<td>------------------------------</td>
<td>--------------</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>thermal conductivity</td>
<td>W/(mK)</td>
</tr>
<tr>
<td>$\nu$</td>
<td>kinematic viscosity</td>
<td>m²/s</td>
</tr>
<tr>
<td>$\rho$</td>
<td>density</td>
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</tr>
<tr>
<td>$\theta$</td>
<td>angle</td>
<td>rad</td>
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**Mathematical symbols**
## Subscripts

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<tr>
<td>acc</td>
<td>accuracy</td>
</tr>
<tr>
<td>acc</td>
<td>accumulator</td>
</tr>
<tr>
<td>age</td>
<td>age of the pressure sensor</td>
</tr>
<tr>
<td>av</td>
<td>average</td>
</tr>
<tr>
<td>b</td>
<td>bulk</td>
</tr>
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<td>bottom</td>
</tr>
<tr>
<td>cal</td>
<td>calibration</td>
</tr>
<tr>
<td>cb</td>
<td>convective boiling</td>
</tr>
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<td>cr</td>
<td>critical</td>
</tr>
<tr>
<td>cond</td>
<td>condenser</td>
</tr>
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<td>fluid</td>
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<td>friction</td>
</tr>
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<td>vapor</td>
</tr>
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<td>hydraulic</td>
</tr>
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<td>i</td>
<td>inner</td>
</tr>
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<td>inlet</td>
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<td>ins</td>
<td>insulation</td>
</tr>
<tr>
<td>l</td>
<td>liquid</td>
</tr>
<tr>
<td>lg</td>
<td>liquid-vapor</td>
</tr>
<tr>
<td>line</td>
<td>Line pressure</td>
</tr>
<tr>
<td>lm</td>
<td>logarithmic mean</td>
</tr>
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<td>heat loss</td>
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<td>momentum</td>
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<td>nucleate boiling</td>
</tr>
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<td>outer</td>
</tr>
<tr>
<td>out</td>
<td>outlet</td>
</tr>
<tr>
<td>pb</td>
<td>pool boiling</td>
</tr>
<tr>
<td>ph</td>
<td>preheater</td>
</tr>
<tr>
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<td>linear regression</td>
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<tr>
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Chapter 1

Introduction

The world's energy use is increasing. Up to now, a large extent of the energy was provided using fossil fuels. It is generally known that this can lead to severe consequences, for example the rise of the sea level [1] and higher occurrence of extreme weather events [2]. To meet this energy need in the future in a sustainable way, new technologies are being developed. The organic Rankine cycle (ORC) is one of those technologies. ORCs can be used to generate electricity by utilizing biomass combined heat and power, geothermal energy, solar power plants and waste heat recovered from exhaust gases [3]. This work focuses on the latter case. By use of an ORC, some of the energy contained in the exhaust gases is recuperated and converted into electricity, which increases the thermal efficiency of the process. Generally, there are two important sources of waste heat: industrial processes and internal combustion engines. A typical example of a waste heat generating industrial process is the cement industry, which is responsible for 5 % of the global greenhouse gas emissions, where 40 % of the input energy is carried away through flue gases released to the atmosphere [4, 5]. For the internal combustion engines, ORCs are installed on heavy-duty Diesel engines, increasing the thermal efficiency with up to 6.1 % [6].

An ORC is a Rankine cycle in which water is replaced with an organic fluid. It is suitable for cases in which heat is available at relatively low temperatures, because organic fluids have lower boiling points than water [7]. ORCs are better suited for low to moderate power ranges (~ MWe) and/or low temperature applications, than the classical Rankine cycles [3]. ORCs with low temperature heat sources (< 250°C) perform better than the water/steam cycle with the same heat source, even at low or moderate evaporation pressures [8].

Current research is focusing on several aspects of the ORC, to increase its efficiency and to make its use economically feasible. The various aspects include the selection of the working fluid, the turbine or expander and the design of the evaporator and the condenser [3, 7]. The size of the evaporator affects both the efficiency and the cost of the system [9]. If the evaporator is designed too small, then it is possible that the fluid is not evaporated completely, which can lead to turbine of
expander damage. If the evaporator is designed too large, then the cost of the system increases and
the superheat rises, which is detrimental for the thermal efficiency of the cycle [9]. With a precise
knowledge of the local heat transfer coefficients, the amount of heat transferred to the fluid is known
and the length of the evaporator can be chosen appropriately. The local heat transfer coefficients
are customarily obtained from heat transfer correlations that are fitted on experimental data.

Solkatherm® SES36 is a fluid that has promising properties for waste heat recovery applications
with flue gas temperatures below 400 °C, because of its relatively high vapor density which results
in a small system size [10, 11]. The currently existing heat transfer correlations are not validated for
this fluid or for ORC conditions [9], because most researches focus on two-phase flow boiling heat
transfer experiments for applications in the heat pump, refrigeration or air-conditioning industry,
with saturation temperatures ranging between −20 °C and 40 °C [12]. At higher saturation temper-
atures and consequently higher reduced temperatures, the thermodynamic properties shift, which
has an influence on the liquid-vapor interactions, the occurring flow patterns and thus also the heat
transfer [12]. Making use of the existing correlations would give rise to relatively large error margins
in the design of these evaporators. Therefore, it is necessary that the heat transfer properties of
Solkatherm® SES36 are investigated experimentally [9], which is the purpose of this thesis.

Apart from an investigation of the dryout of Solkatherm® SES36 in vertical, small diameter tubes,
there is no experimental data of the heat transfer of this fluid available in literature [13]. Hence, a
test facility was built for conducting two-phase flow experiments in horizontal tubes at subcritical
ORC conditions. In the test section, the local heat transfer coefficients and the two-phase pressure
drop are observed. The test facility is suited to have test sections with different tube diameters and
tube materials. The first experiments are done in a 21 mm internal diameter, horizontal tube. This
relatively large diameter is chosen because the evaporation heat of organic fluids is typically ten
times smaller than that of water, which results in a higher mass flow rate. The consequence is that
a larger feed pump is needed [8]. But next to that, the pressure drop in the evaporator increases
significantly when the tube diameter becomes smaller than a critical value, and the critical value
of the diameter increases for increasing mass flow rates. This is because the flow pattern changes
to annular flow. The increase of the pressure drop destroys the temperatures difference that drives
the heat transfer, which results in a longer evaporator length [14]. So, larger diameters are chosen
to reduce the pressure drop and consequently to avoid an increase of the length of the evaporator.

This thesis comprises a literature study, a detailed description of the flow boiling test facility and its
working principle, the data reduction and the accuracy of the measurements, the design of experi-
ments and the control of the test facility.
Chapter 2

Literature study

In this chapter, first the working principle of an ORC is explained. Then the properties of the working fluid, Solkatherm® SES36, are discussed. The third part of this chapter is a review about two-phase flow boiling experimental research in horizontal tubes.

2.1 Organic Rankine cycles

A typical ORC consists of an evaporator, an expander (or turbine), a condenser and a pump, as can be seen in Figure 2.1a. In the ORC without recuperator, the subcooled liquid enters the evaporator, where the exhaust gases are cooled down. The enthalpy of the primary fluid increases until it is saturated. The fluid starts evaporating and at the end of the evaporator, the fluid has reached the superheated vapor state. Then it enters the expander. The pressure of the fluid decreases and work is extracted, this work is used to drive a generator that converts mechanical energy into electrical energy. In the condenser, the fluid is brought back to the liquid phase by external cooling. Finally, the pump brings the liquid back to the high pressure level and the cycle starts over again. Depending on the working fluid, it can be beneficial to add a recuperator to this system, because

Figure 2.1: A schematic overview of an ORC without a recuperator (left) and with a recuperator (right) [15]
the evaporator then needs a smaller heat input [15]. The cycle with a recuperator is shown in Figure 2.1b. After the liquid leaves the pump, it goes to the recuperator where it is preheated using the fluid that leaves the expander. This configuration has the advantage that a smaller heat input is required in the evaporator to reach the vapor state. The cycle without recuperator is also depicted in a temperature-specific entropy diagram, in Figure 2.2, where 1-2 is the part where the fluid is pumped. 2-3 represents the evaporator, the liquid is boiled. 3-4 is the part in the expander and 4-1 represents the condenser.

2.2 Solkatherm® SES36

Solkatherm® SES36 is an azeotropic mixture with a mass fraction of 65 % R-365mfc and 35 % Galden® HT 55. In total, this mixture contains more than eight different types of perfluopolyethers, all having low boiling temperatures. The mixture is non-flammable and has a low toxicity. It is promising for waste heat recovery purposes with flue gas temperatures below 400 °C, because of its relatively high vapor density compared to other frequently used fluids [10]. The vapor density determines the size of the system, because a lower vapor density leads to a higher volume flow rate and consequently a higher pressure drop. For the same pressure drop, the system with a higher vapor density has smaller tubes. A system that would use steam to recover waste heat, would be significantly larger than the system using Solkatherm® SES36 [11]. The company Turboden already uses Solkatherm® SES36 for waste heat recovery applications [3].

2.3 Two-phase flow boiling in horizontal tubes

In Figure 2.1, the subcooled liquid refrigerant enters the evaporator. As heating power is applied, the refrigerant starts to heat up until it becomes saturated liquid. When more heat is added, the fluid starts evaporating and a two-phase flow is present in the tube: partly liquid and partly vapor. As evaporation completes, all of the liquid evaporates and only vapor remains.

In general, a two-phase flow is a flow in which two phases coexist, for example liquid and vapor. The process of boiling a fluid in the presence of bulk fluid flow is called flow boiling or forced convection boiling [16], and it is distinctly different from pool boiling. In pool boiling, the liquid is in contact with a surface at a temperature above the saturation temperature, and boiling occurs at the solid-liquid interface. There is no bulk fluid flow, the motion of the fluid originates only from the boiling process [17]. Flow boiling is interesting for industrial applications because of the high heat transfer coefficients occurring.

Two-phase flow boiling is a subject that has already been investigated thoroughly. Various applications where it occurs include: coal fired power plants, refrigerators, heat pumps and nuclear reac-
However, the conditions differ for every application: the temperatures at which the boiling occurs, the size of the tubes, the used refrigerants, etc. In the examples mentioned above, the boiling fluid is always inside a tube and the heat is applied from the outside, this will also be the case in the study below. An important aspect for the experimental work is whether the tubes in which the fluid boils are placed horizontally or vertically. It is easy to understand that the influence of gravity will cause a difference between these two types [19]. In the following, the focus will be on horizontal tubes.

First, the different flow patterns occurring in horizontal tubes for evaporating flow are described. Then, the heat transfer mechanism is explained. Next, several existing heat transfer correlations for two-phase flow are discussed. Then, there is a section about the influence of various factors on the heat transfer coefficient. The pressure drop in two-phase flow is explained. The final part gives an overview of experimental studies that have been performed.

2.3.1 Flow patterns during evaporation in tubes

In two-phase flows, different flow regimes can be distinguished. The categorization of these regimes is not uniquely defined, various divisions can be found in the literature. The flow patterns are different for horizontal and vertical tubes. In vertical tubes, there is no net influence of gravity. In horizontal tubes, gravity forces the liquid to the bottom of the tube. Figure 2.3 illustrates different flow patterns for evaporating flow inside horizontal tubes. The different flow patterns occurring in evaporating flow inside horizontal tubes are:

**Bubbly flow** Most of the fluid is liquid, but multiple spherical vapor bubbles are present in the flow. They are randomly distributed, but more of them are located in the upper part of the tube,
Stratified flow  The top part of the tube consists of vapor and the bottom of liquid, with a straight separation between them.

Plug flow  Large elongated vapor bubbles exist in the tube. They are located in the upper part of the tube, but the entire tube circumference remains wet.

Wavy flow  At high vapor phase velocities, waves start to develop at the liquid-vapor interface. This flow regime is also known as stratified-wavy flow.

Slug flow  The waves become so large that they touch the upper part of the tube. The resulting flow pattern is one where the upper part of the tube is alternately wet and dry. Sometimes, plug and slug flow are grouped together and referred to as intermittent flow.

Annular flow  The liquid forms an annulus at the circumference of the tube and the center is occupied by the vapor phase. The bottom part of the liquid annulus is thicker than the top part, because of the influence of gravity. Liquid droplets are present in the vapor phase in the core of the tube.

Mist flow  This pattern occurs when the entire liquid ring of the annular flow pattern is evaporated. There are still some liquid droplets entrained in the vapor phase, but the tube circumference is dry. The flow pattern between annular and mist flow is sometimes categorized as dryout. Mist flow and dryout flow are not depicted in Figure 2.3.
In literature, many flow pattern maps exist in which the mass velocity and the vapor quality are on the y- and the x-axis respectively. An example can be found in Figure 2.4. When the fluid is completely evaporated, the graph is run through from the left side to the right side. Remark that the flow patterns that appear in the tube, depend on the mass velocity. The flow pattern map changes for other refrigerants, tube diameters, saturation temperatures or heat fluxes.

2.3.2 Heat transfer mechanism

It is crucial to have physical insight into the heat transfer mechanism when doing heat transfer experiments. This way, the experimental results and the influences of the parameters can be interpreted and explained.

Generally, it is assumed that two-phase flow boiling heat transfer is the combined effect of nucleate boiling and convective boiling heat transfer.

Nucleate boiling is similar to pool boiling, but the fluid flow influences the growth and departure of bubbles. Like in the process of pool boiling, the heat transfer increases strongly with increasing heat flux [19] (as long as the heat flux is smaller than the critical heat flux). The relative effect of nucleate boiling compared to convective boiling decreases with increasing mass flow rate and increasing vapor quality. At high mass flow rates and high vapor qualities, the effect of nucleate boiling is suppressed due to the thinner thermal boundary layer of forced convection compared to natural convection. A smaller zone has effective superheat and less bubbles are being formed [22].

Convective boiling is similar to single-phase convective heat transfer, the heat transfer by the movement of fluids, but again there is an influence of the two-phase state of the flow. When vapor is formed, the average density of the fluid decreases and the fluid’s velocity increases. Thus the relative importance of convective boiling is increased when compared to single-phase forced convection.

Steiner and Taborek [23] assumed that the local two-phase heat transfer coefficient can be calculated with the combination of the nucleate boiling coefficient $h_{nb}$ and the convective boiling coefficient $h_{cb}$ in a power law:

$$h_{tp} = \left[ (h_{nb})^n + (h_{cb})^n \right]^{\frac{1}{n}} \quad (2.1)$$

In Figure 2.5, the power law of equation 2.1 is depicted for different powers. The extreme values correspond with:

- $n = 1$ corresponds with the sum of $h_{nb}$ and $h_{cb}$
- $n = \infty$ corresponds with the maximum of $h_{nb}$ and $h_{cb}$

In Figure 2.5, convective boiling is independent of the heat flux, which is a common assumption for flow boiling prediction methods [19]. It is assumed that $h_{nb}$ is a function of the heat flux, this
Figure 2.4: Flow pattern map for R-22 at 5°C in a 13.84 mm tube [21], S= stratified, SW=stratified-wavy, I=intermittent, A=annular, D=dryout, M=mist flow

Figure 2.5: Two-phase heat transfer coefficient ratio versus the heat flux ratio for different powers $n$ [23]
follows from the resemblance with pool boiling. The different existing heat transfer correlations that are summarized below, have various values of the power $n$ in the power law.

### 2.3.3 Heat transfer coefficient correlations

A lot of correlations for predicting the heat transfer coefficient already exist. As stated by Kandlikar [24], flow boiling correlations can be classified into two types. The first type is a correlation that is valid for one fluid, fitted on one or more databases. The second type is fitted on many databases of different refrigerants and has a larger range of applicability.

In the following, the basic principles behind some important heat transfer correlations are explained. A summary of the applicability of the different correlations is given in section 2.3.3.6.

#### 2.3.3.1 Chen correlation

The correlation of Chen [25] has power $n = 1$ in equation 2.1:

\[
htp = h_{nb} + h_{cb}
\]

Chen used the correlation of Forster and Zuber [26] to determine the nucleate boiling heat transfer coefficient and the single-phase heat transfer coefficient was determined by using the correlation of Dittus and Boelter [27]. However, the two terms were multiplied with factors $S$ and $F$ respectively:

\[
htp = h_{pb} \cdot S + h_{sp} \cdot F
\]  \hspace{1cm} (2.2)

where $S$ is called the suppression factor and $F$ the Reynolds number factor. $F$ is generally greater than unity, because the effect of convective boiling is more pronounced than single-phase convection. The factor $S$ varies between zero and one, taking into account the relative importance of nucleate boiling, which is smaller than the effect of pool boiling (see section 2.3.2).

Note that this is a correlation that can only be used for vertical flow, but it is mentioned here because of its historical importance and because it gives insight into the heat transfer mechanism. This correlation was tested against more than 600 data points for several fluids, in vertical tubes for annular and annular-mist flow [25].

#### 2.3.3.2 Shah correlation

Shah [28] developed the equations of a new heat transfer correlation in 1982, after having developed a graphical method in 1976. A distinction was made between four regimes of flow boiling:
• pure nucleate boiling
• bubble suppression
• pure convective boiling with surface fully wet
• convective boiling with partly dry surface

In the pure nucleate boiling regime, it is assumed that nucleate boiling is dominant. In the bubble suppression regime, there is a combined effect of both convection and pool boiling. In the convective boiling part, convective boiling is dominant. For horizontal tubes, a distinction is made between a fully and a partly wet surface, this is the case in stratified flow (see Figure 2.3). This distinction can be made based on the liquid Froude number:

\[ F_{\text{fl}} = \frac{G^2}{\rho_l^2 \cdot g \cdot D} \]

As a Froude number is the ratio of inertia forces to gravity forces, it can be understood that the Froude number gives an indication of the wet or dry state of the tube surface.

In each regime, the two-phase heat transfer coefficient is calculated by taking the maximum of the nucleate boiling and the convective boiling heat transfer coefficient, this corresponds to a power \( n = \infty \) in equation 2.1 and Figure 2.5:

\[ h_{tp} = \max(h_{nb}, h_{cb}) \]

The nucleate boiling heat transfer coefficient is calculated with a correlation depending on the boiling number and the convective heat transfer coefficient is obtained from the correlation of Dittus and Boelter [27].

2.3.3.3 **Gungor-Winterton correlation**

Gungor and Winterton [22] developed a heat transfer correlation that is simple to apply, can be used for subcooled and saturated boiling, is valid for horizontal and vertical flow, is tested against a large database and has a good fit. They started from the Chen correlation (equation 2.2) and iterated to calculate a correlation for the suppression and Reynolds number factor. Instead of the Forster and Zuber correlation [26], the Cooper correlation [29] was used to calculate the pool boiling heat transfer coefficient.

2.3.3.4 **Kandlikar correlation**

Kandlikar [24] realized that it was important to develop a correlation that applies to new fluids and has the ability to correctly predict the dependency of heat transfer coefficient on the system pa-
rameters. For example, it was checked whether the correlation predicted the trend of heat transfer coefficient versus vapor quality correctly. It was stated that the mean deviations for this correlation are smaller than those of other correlations. The following correlation was proposed:

\[
\frac{h_{tp}}{h_I} = c_1 \cdot \text{Co}^{c_2} \cdot (25Fr_l)^{c_5} + c_3 \cdot \text{Bo}^{c_4} \cdot Fr_l
\]

where \(Fr_l\) is a fluid dependent parameter. The coefficients \(c_1, \ldots, c_4\) were first fitted for water in vertical tubes (\(Fr_l = 1\) and \(c_5 = 0\)). Each coefficient has a value for the convective boiling region and for the nucleate boiling region. The two correlations obtained by inserting these two values intersect. So when the largest value of the two equations is chosen, there is no discontinuity when going from dominant convective to dominant nucleate boiling. The fluid dependent parameter was then fitted for other fluids. For horizontal tubes where \(Fr_l < 0.04\), an additional factor dependent on \(Fr_l\) has to be included in the first term.

### 2.3.3.5 Local flow pattern evaporation model of Kattan-Thome-Favrat

The heat transfer correlation of Kattan, Thome and Favrat [30] makes use of a flow pattern map to predict the heat transfer coefficient. It is understandable why this was done: the heat transfer is strongly dependent on the flow pattern. In this method, the occurring flow pattern is identified first, then geometrical parameters of this flow pattern are calculated. The two-phase flow structures were simplified, as can be seen in Figure 2.6. In the top left corner, the stratified flow pattern is depicted. This pattern is replaced by a truncated annulus in the top right corner, with the same dry angle and a thickness \(\delta\). The bottom left corner shows the simplification of annular and intermittent flow, it is assumed that the thickness of the annulus is constant. In the middle of the bottom part of the figure, the simplification of annular flow with partial dryout is shown. The bottom right corner shows the simplified pattern of stratified-wavy flow. Some of the parameters on the figure had to be calculated iteratively. Later on, this was modified by Thome and El Hajal [31], so that iteration was no longer required. Then, the heat transfer coefficient, depending on the geometrical parameters, can be calculated:

\[
h_{tp} = \frac{D_i \cdot \theta_{dry} \cdot h_{vap} + D_i \cdot (2\pi - \theta_{dry}) \cdot h_{wet}}{2\pi D_i}
\]

where \(h_{wet}\) is calculated with an asymptotic equation of power three, depending on nucleate and convective boiling influences. \(h_{vap}\) is calculated with the Dittus-Boelter correlation for turbulent flow [27].

Wojtan et al. [21] updated the flow pattern map of Kattan et al. The stratified-wavy zone was divided in three subzones. This improved the predictions at low vapor qualities and low mass velocities (see Figure 2.4). New heat transfer correlations were proposed, and with the modifications, the heat transfer model can be used for the entire vapor quality range, in all flow regimes except bubbly flow.
2.3.3.6 Overview of the discussed correlations

In Table 2.1, an overview is given of the validated ranges of applicability of the correlations discussed above. This means that the correlations were developed using data points in these conditions, or that it was checked whether the correlation applies with the mentioned conditions. As stated before, these correlations were mainly developed for applications with lower saturation temperatures, they might be unreliable in the conditions of ORC evaporators.

2.3.4 Factors influencing the heat transfer coefficient

In this section, the effect of different parameters influencing the heat transfer coefficient will be discussed. The heat transfer in the tube results from the combined effect of nucleate boiling and convective boiling, this principle was explained in section 2.3.2. The factors that have an influence are: the heat flux, the mass velocity, the vapor quality and the saturation temperature or saturation pressure. Note that it is not always possible to mention the effect of a single parameter, because there is interaction between the parameters.

2.3.4.1 Heat flux

When the heat flux is increased, the contribution of nucleate boiling to the heat transfer increases. Since nucleate boiling is dominant in the low vapor quality and low mass velocity region, an increase in heat flux is expected there. This trend is indeed observed by different authors. Da Silva Lima et al. [33] observed an increase of heat transfer mainly at low vapor qualities, when the heat...
Table 2.1: An overview of the range of applicability of the discussed correlations

<table>
<thead>
<tr>
<th>Correlation</th>
<th>Fluid</th>
<th>Flow orientation</th>
<th>Geometry $D_h$</th>
<th>$x$ (%)</th>
<th>datapoints</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>$D_h$ (mm)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Chen [25]</td>
<td>water, methanol, cyclohexane, pentane, heptane, benzene</td>
<td>vertical</td>
<td>circular</td>
<td>1-71</td>
<td>600</td>
</tr>
<tr>
<td>Shah [32]</td>
<td>water, R-11, R-12, R-22, R-113, cyclohexane</td>
<td>vertical, horizontal</td>
<td>circular, annular</td>
<td>6-25.4</td>
<td>0-100 780</td>
</tr>
<tr>
<td>Gungor-Winterton [22]</td>
<td>water, R-11, R-12, R-114, R-152, ethylene, glycol</td>
<td>vertical, horizontal</td>
<td>circular, annular</td>
<td>2.95-32</td>
<td>0-100 3693</td>
</tr>
<tr>
<td>Kandlikar [24]</td>
<td>water, R-11, R-12, R-114, nitrogen, neon</td>
<td>vertical, horizontal</td>
<td>circular</td>
<td>4.6-32</td>
<td>0-99 5426</td>
</tr>
<tr>
<td>Kattan-Thome-Favrat [30]</td>
<td>R-134a, R-502, R-404a, R-407c, ammonia</td>
<td>horizontal</td>
<td>circular</td>
<td>10.9-16</td>
<td>1-100 unknown</td>
</tr>
</tbody>
</table>

flux was increased (see Figure 2.7). However, when the heat flux was increased for relatively high mass velocities, the inception of dryout occurred at lower vapor qualities. Dryout results in a strong decrease of heat transfer. In the test conditions of Da Silva Lima et al. [33], the local heat transfer coefficient increased with vapor quality until dryout (except for very small vapor qualities). Thus the maximum local heat transfer coefficient was reached just before the onset of dryout, and this maximum value decreased with increasing heat flux. This phenomenon is depicted in Figure 2.7. For example, look at the condition of a saturation temperature of 5 °C. For the low heat flux (Figure 2.7a), dryout occurs at a vapor quality of about 0.86 and the corresponding local heat transfer coefficient is almost $13000 \text{ W/(m}^2\text{K)}$. In the high heat flux condition (Figure 2.7b), dryout occurs at a vapor quality of about 0.78 and a local heat transfer coefficient of almost $11000 \text{ W/(m}^2\text{K)}$. Similar trends can be observed for higher saturation temperatures. Bandarra Filho et al. [34] observed an increase of the heat transfer coefficient over the entire range of vapor qualities with increasing heat flux at low mass velocities. At high mass velocities, there was an increase of the heat transfer coefficient as long as the vapor quality was smaller than 0.5. At higher vapor qualities, all the data points converged to the same values. This indicates that convective boiling is dominant at high mass velocities and vapor qualities. Saiz Jabardo [35] drew the same conclusion, but observed that at very high heat fluxes, the heat transfer coefficient increased over the entire range of vapor qualities. Similar trends to those already mentioned were observed by Greco and Vanoli [36], Wang et al. [37], Cho et al. [38] and Oh et al. [39].
Figure 2.7: Heat transfer coefficients of R-134a versus vapor quality for low and high heat fluxes at a high mass velocity and different saturation temperatures [33], circle=5°C, square=15°C, triangle=20°C, S= stratified, I= intermittent, A= annular, D= dryout, M= mist flow

2.3.4.2 Mass velocity

Increasing the mass velocity leads to an increase of the convective boiling contribution to the heat transfer. As mass velocity increases, a strong increase of the heat transfer can be expected in the region where convective boiling is dominant: low heat fluxes and high vapor qualities. This is confirmed by Greco and Vanoli [36]. Da Silva Lima et al. [33] reported an increase of the heat transfer coefficient with the mass velocity in the entire vapor quality range, but this increase was stronger for higher vapor qualities. However, an increase of the mass velocity led to dryout inception and completion at lower vapor qualities. Bandarra Filho et al. [34] related the effect of mass velocity to the occurring flow pattern. At low mass velocities the heat transfer coefficient was more or less constant over the vapor quality range, here stratified-wavy flow occurred. At higher mass velocities, annular flow occurred. As the vapor quality increased, the liquid film thickness decreased and the effect of a higher mass velocity was more pronounced (see Figure 2.8). Wang et al. [37] confirmed this, and stated that the heat transfer coefficients corresponding to different mass velocities converge at low vapor qualities, this indicates that nucleate boiling is dominant in that region. Saiz Jabardo et al. [35] also confirmed the findings of Bandarra Filho et al. [34]. In the range of low mass velocities there was a weak dependence of the heat transfer coefficient on the vapor quality and stratified-wavy flow occurred. In the range of high mass velocities, there was a strong dependence of the local heat transfer coefficient on the vapor quality, and a local minimum of the heat transfer coefficient occurred at relatively low vapor qualities. It was stated that this happened because the flow pattern changed from intermittent slug flow, with bubble suppression, to annular flow. This local minimum can clearly be observed in Figure 2.8 for the highest mass velocity. Cho et al. [38]
and Oh et al. [39] stated that with CO$_2$, the effect of convective boiling is little due to the small difference between the liquid and vapor density. A large difference between liquid and vapor phase densities leads to a pronounced difference between liquid and vapor phase velocities, and a larger contribution to convective boiling. In the case of using CO$_2$ as a refrigerant, the influence of mass velocity on the heat transfer coefficient is thus smaller than usual.

### 2.3.4.3 Saturation temperature

Greco and Vanoli [36] stated that at higher saturation temperatures, less wall superheat is required to form a stable nucleus. Thus, more bubbles are being formed, the effect of nucleate boiling is increased, and its contribution to the heat transfer leads to an increase of the heat transfer coefficient. For low mass fluxes, this increase was noticeable over the entire range of vapor qualities. For higher mass fluxes, the increase could only be observed at low vapor qualities. Da Silva Lima et al. [33] also observed that the heat transfer coefficient at low vapor qualities increased with saturation temperature. However, at higher vapor qualities the opposite trend was observed: the heat transfer coefficient decreased with the saturation temperature (see Figure 2.7). Two reasons were mentioned for this. First, at higher saturation temperatures the vapor density increases, thus the velocity of the vapor phase lowers. This leads to a smaller contribution of convective boiling. Also, the thermal conductivity of the liquid decreases with saturation temperature. This leads to a larger difference between the wall and bulk temperature, and a lower heat transfer coefficient. Saiz-Jabardo et al. [35] drew similar conclusions to those of Da Silva Lima et al. [33]. But at high heat fluxes, an increase of the saturation temperature led to an equal increase of the heat transfer coefficient in the entire vapor quality range. It was concluded that in their test conditions nucleate boiling was dominant in the annular flow regime until vapor qualities of about 0.9. Similar trends to the ones previously mentioned were observed by Cho et al. [38] and Oh et al. [39].

### 2.3.5 Pressure drop

In the design of evaporators, the pressure drop is important. For example in an ORC (see Figure 2.1), the feed pump has to overcome the pressure drops in the evaporator, expander and condenser. A larger pressure drop requires a higher input of electrical energy and may lower the efficiency of the system. Next to that, it also increases the cost of the system. In two-phase flows, the total pressure drop is the result of three influences: the static pressure drop, the momentum pressure drop and the frictional pressure drop:

$$
\Delta p_{tot} = \Delta p_{stat} + \Delta p_{mom} + \Delta p_{fric}
$$

The static pressure drop is a result of the gravitational potential energy, and will have no influence in horizontal tubes. The momentum pressure drop is a consequence of the change in kinetic energy
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Figure 2.8: Heat transfer coefficient versus vapor quality for different mass velocities with R-407c [34]

of the flow [19]. In evaporating flow, the kinetic energy increases and the contribution of this term is positive. The last term is a pressure drop due to the friction with the tube wall.

To calculate the pressure drop, the two-phase flow can be regarded as a homogeneous flow or as a separated flow. In homogeneous flow models, the two-phase flow is considered a single-phase flow with average properties of the two phases. Then, the pressure drop is calculated using these properties. In separated flow models, it is assumed that the phases are perfectly separated. Each phase flows in a tube with cross-section proportional to the void fraction [19].

Many correlations for predicting $\Delta p_{fric}$ exist: the correlation of Friedel [40], Lockhart-Martinelli [41], Müller-Steinhagen and Heck [42], Grönnerud [43], Moreno Quibén and Thome [44], etc.

The correlation of Moreno Quibén and Thome is the most advanced one [44]:

- it has the highest accuracy
- it is good at predicting the trend of the pressure gradient with the vapor quality
- it is applicable for mist and dryout flow
- it gives acceptable predictions at $x = 0$ and $x = 1$
- it includes the effect of a partially dry perimeter
- it predicts the peak of the pressure gradient at high vapor qualities correctly
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The other methods failed at one or more of these points [44]. The correlation makes use of a flow pattern map. For the annular, stratified and stratified-wavy flow regimes, an equivalent film thickness is calculated, and the pressure drop is calculated using a simplified geometry (see Figure 2.6). For each flow regime, different frictional pressure drop correlations are proposed. This technique guarantees a smooth transition from one flow regime to another.

2.3.6 Two-phase flow experiments

2.3.6.1 General information

This section gives an overview of various flow boiling experiments performed. It will focus on the experimental set-up, the measurement techniques used and the results and findings of the researchers. As an example of an experimental set-up, Figure 2.9 is given. In the set-up of Figure 2.9, the preheater is used to bring the refrigerant to the desired state. In the test section, the fluid boils. After the test section, the void fraction is measured. Then the fluid is condensed. The pump takes account of the pressure losses in the system. And the flow meter measures the mass flow rate. In each set-up, there are also pressure and temperature sensors, these are not indicated in this example. The major differences between various set-ups are:

- the tube diameter of the test section
- the way of applying heating power
- the void fraction measurement technique

Kandlikar [45] proposed the following ranges to make a distinction between tube sizes, based on the hydraulic diameter:

- conventional channels: $D_h \geq 3 \text{ mm}$
- mini-channels: $200 \mu \text{m} \leq D_h \leq 3 \text{ mm}$
- micro-channels: $10 \mu \text{m} \leq D_h \leq 200 \mu \text{m}$

Note that various classifications have been proposed and that the distinction is ambiguous. The following focuses on flow boiling in conventional channels.

The way of applying heating power can differ, it can even differ between the preheater and the test section in one experimental set-up. For example: the preheater uses electrical heating and the test section uses a countercurrent heat exchanger. The two methods have different properties. Electrical heating provides a constant heat flux along the test section, and to calculate local heat transfer coefficients, it is required that the wall temperature is measured at several locations along the test
section, which is relatively simple and inexpensive. Another advantage of electrical heating is that the heat flux can be adjusted quickly. The countercurrent heat exchanger provides a nearly uniform wall temperature at the inner perimeter of the tube. Typically, the refrigerant that is boiled flows in the core of the heat exchanger, while the hot fluid flows in the annulus. To calculate the heat flux and the heat transfer coefficient, thermocouples are put inside the annulus of the heat exchanger and on the outer wall of the core. This is less straightforward because the construction is more prone to errors and failures and it requires a higher labor cost. However, according to Thome [19], a constant wall temperature condition is similar to actual operation, and a constant heat flux leads to non-uniform wall temperatures in certain flow patterns.

Different techniques exist for measuring the void fraction. The capacitance sensor uses the capacitance of the cross-section as an indication of the void fraction [46]. Because the capacitance of the vapor and liquid phases differ, it can be used to discern between them. The time variation of this signal is related with the occurring flow pattern. For example, if stratified flow occurs, the signal is more or less constant. For plug of slug flow, the signal varies in time. Another technique uses a camera that counts the pixels corresponding with vapor and liquid, while the two-phase flow is illuminated by a laser sheet. This way, the void fraction can be calculated [21]. The latter method can only discern stratified, stratified-wavy and slug flows.

2.3.6.2 Experimental studies

Bandarra Filho et al. [34] did flow boiling experiments with R-404a and R-407c in horizontal copper tubes of 12.7 mm internal diameter. The test section was heated using electrical heating. To allow the flow to develop completely, a 1.5 m long tube was put between the preheater and the test section.
Both the preheater and the test section used tape electrical resistors to apply the heat flux. The obtained heat transfer coefficients were compared with the heat transfer correlation of Kandlikar [24]. First, the fluid dependent parameter was fitted for the two refrigerants. It was reported that the correlation had a good fit, and that both the nucleate and convective boiling regions were predicted reasonably.

Greco and Vanoli [36] investigated the local heat transfer coefficients of R-410a and R-404a in horizontal, smooth tubes with an internal diameter of 6 mm. The heat flux was applied by the Joule effect. At 8 places on the tube, there were 4 thermocouples: at the top, the bottom, the left and the right side of the tube. This way, they analyzed the circumferential variation of the heat transfer coefficient. It was concluded that when nucleate boiling is dominant, the heat transfer coefficient will be larger at the bottom of the tube. The liquid film at the bottom of the tube is thicker, which stimulates nucleate boiling because more place is available for bubbles to grow. However, when convective boiling is dominant, the heat transfer coefficients are higher at the top of the tube. This is because the thinner liquid layer at the top of the tube is a smaller thermal resistance for the heat flux. Also, a method to predict when the nucleate boiling is fully suppressed, was developed.

Kim, Seo and Chung [47] measured heat transfer characteristics of R-410a in electrically heated smooth and micro-fin tubes, with smooth tube diameters of 8.7 mm and 6.18 mm, and micro-fin tube diameters of 8.82 mm and 6.30 mm. The purpose was to investigate the heat transfer enhancement of micro-fin tubes when compared to smooth tubes. It was reported that the micro-fin tubes have higher heat transfer coefficients, but also higher pressure drops.

Shin, Kim and Ro [48] did two-phase flow boiling experiments with R-22, R-32, R-134a, R-290, R-600a and mixtures of these refrigerants and compared the heat transfer coefficients with the Gungor-Winterton correlation [22]. They used 7.7 mm inner diameter tubes that were electrically heated, and used Teflon® connections to insulate measuring instruments from the test section. In test-conditions in which overheating was expected, the heated length of the tube was diminished, while the heat flux was kept constant with a variable voltage transformer. The Gungor-Winterton correlation [22] gave an average deviation of 30.5 %.

Jung et al. [49] measured local heat transfer coefficients of R-22, R-114 and their mixtures with uniform heat flux, provided by electrical heating. The test section consisted of two tubes with an inner diameter of 9 mm that had a length of 4 m each, with a 180° U-turn bend in between. A variable length test section was created, using in total 8 different connections to the DC power supply. They observed that nucleate boiling was fully suppressed beyond certain vapor qualities in their test conditions. The influence of mixtures on the heat transfer coefficient was investigated. It was found that the heat transfer coefficient obtained with mixed refrigerants is lower than the value calculated by ideal mixing of the pure components’ heat transfer coefficients. It was stated that this is because full suppression of nucleate boiling is easier to achieve with mixtures. Similar experiments were
performed by Ross et al. [50].

Niederkruger et al. [51] performed nucleate boiling heat transfer experiments with pure components and mixtures of R-846 and R-12 in horizontal tubes of 14 mm inner diameter. The test section was segmented: it consisted out of 6 parts that could be heated independently. The thick-wall nickel tube has axial slits in it, to minimize the circumferential heat conduction. Each segment was heated separately by a resistance and contained at least one thermocouple (see Figure 2.10). By controlling the different segments, a uniform inner wall temperature could be obtained.

Kuo and Wang [52] investigated the two-phase heat transfer and pressure drop of R-22 and R-407c, which is a mixture of various refrigerants, blended to have similar properties as R-22. The experiments were done in a micro-fin tube with a nominal internal diameter of 9.52 mm. Both the preheater and the test section were inside the core of a double pipe heat exchanger, with hot water in the annulus. The heat flux was controlled by varying the mass flow rate and the inlet temperature of the hot water. The heat transfer coefficients and pressure drops were compared for the two refrigerants, and R-407c had lower local heat transfer coefficients and pressure gradients. It was concluded that these reductions may be attributed to the difference in flow patterns between the pure component and the mixture.

Yu et al. [53] boiled R-134a in horizontal smooth tubes of 10.7 mm internal diameter and micro-fin tubes of 11.1 mm nominal internal diameter. The copper tube was wrapped with an electric heater, to supply the heat flux. Flow patterns were determined by looking through sight glasses. The temperature was measured at 5 axial locations. At every location, thermocouples were placed in the middle of the tube and around the circumference. It was found that for certain test conditions in smooth tubes, there was a significant difference in circumferential heat transfer coefficients. For the micro-fin tubes, this difference was much smaller. This is because intermittent flow occurred in the micro-fin tube. The top and side portions of the tube were wet in this condition. Under the same test conditions, wavy flow occurred in the smooth tube. In general, different flow patterns

![Figure 2.10: Test section with 6 segments, thermocouples and heating resistances [51]](image)
lead to various heat transfer coefficient distributions around the circumference, and different flow patterns can exist for the same test conditions in smooth and micro-fin tubes. It was shown that in micro-fin tubes, the transition of flow patterns occurs earlier. The heat transfer coefficients were compared with Gungor-Winterton’s correlation, there was an average deviation of 19%.

Zou et al. [54] did a study on flow boiling of R-170 and R-290 mixtures in horizontal tubes with an internal diameter of 8 mm. The test section consisted of five test units in copper, welded together with stainless steel tubes having the same internal diameter, but a thin thickness of 1 mm to limit axial heat conduction. The heat flux was supplied with an electric heating wire coil, wrapped around the outer part of the tube. The reliability of the test section was validated by performing single-phase, liquid heat transfer experiments of turbulent flow and comparing the result with the correlation of Gnielinski [55]. It was stated that in refrigerant mixtures, it is harder to form bubbles than in pure refrigerants. In the nucleate boiling region, heat transfer coefficients are lower for mixtures. A new correlation was developed that can be used for pure refrigerants and mixtures.

Park et al. [56] measured the heat transfer coefficient and pressure drop of CO$_2$, R-22 and R-410a in horizontal tubes with an internal diameter of 6.1 mm. For CO$_2$ and R-410a, the flow patterns were observed. The local heat transfer coefficient was measured at constant wall temperature conditions, while the pressure drop and flow visualization were observed in adiabatic conditions. The test section was surrounded by a brass jacket, to provide a uniform temperature at the wall. Around this jacket, there was a tube circuit with a hot fluid that provided the heat flux. The flow patterns could be observed through the glass tube, immediately after the test section. It was found that the lower molecular weight and the higher reduced pressure of CO$_2$ stimulate nucleate boiling. At low vapor qualities, the heat transfer coefficients of CO$_2$ were much higher than those of the other refrigerants tested in the same conditions. Due to the higher vapor density of CO$_2$, its two-phase pressure drop is lower than that of the other refrigerants. If CO$_2$ would be used in low temperature applications, another advantage of this fluid would be that dryout (with the corresponding low heat transfer coefficients) would not occur.

Da Silva Lima et al. [33] did flow boiling experimental research of R-134a in tubes with an internal diameter of 13.84 mm. The purpose was to investigate the existence of a local minimum of the heat transfer coefficient and to investigate its dependence on experimental parameters. Furthermore, the results were compared with correlations. The test section was in the core of a double-tube heat exchanger and it was heated with hot water flowing in the annulus, see Figure 2.11. The temperature of the water was adjustable, in this way the heat flux could be adapted. By measuring the temperature of the hot water at several locations in the heat exchanger, the enthalpy could be determined and the heat flux follows. The saturation temperature was obtained by measuring the pressure at the inlet and at the outlet of the test section, and using a linear interpolation to find the pressure in the measurement section. This way the flow was not disturbed. At the location where the local
heat transfer coefficient was measured, 4 thermocouples were placed on the outer wall of the core (T-R* in Figure 2.11). The minimal heat transfer coefficient occurred at a particular combination of nucleate and convective boiling contributions. This minimum value increased with mass velocity and heat flux. It was found that the existing correlations tend to underpredict the heat transfer coefficient in these conditions.

Cho and Kim [38] did flow boiling experiments with CO$_2$ in smooth and micro-fin tubes of 5 mm and 9.52 mm outer diameter. The preheater and test section were heated by electrical heating. The heat transfer coefficients for smooth and micro-fin tubes had similar trends: they increased with heat flux, mass flux and evaporating temperature. The heat transfer coefficients of the 5 mm outer diameter tube were found to be the largest. As before, the micro-fin tubes stimulated annular flow and enhanced the heat transfer. In addition to that, micro-fin tubes have a larger heat exchanging surface. It was concluded that the benefit of a higher heat transfer coefficient outweighs the drawback of a higher pressure drop, 5 mm outer diameter micro-fin tubes are promising with CO$_2$ as a refrigerant.

Col [57] did flow boiling experiments at higher saturation temperatures than usual (25°C to 45°C), and compared these data points with existing correlations. The used refrigerants were R-22, R-134a, R-125 and R-410a. These were boiled in a smooth, horizontal tube with an internal diameter of 8 mm. The test section and the preheater consisted of counter-current tube-in-tube heat exchangers. To enhance the heat transfer of the hot water flowing in the annulus of the heat exchanger, a metal helix was wound around the test tube. It was found that nucleate boiling is more important at these higher evaporating pressures. The correlation of Gungor and Winterton underpredicted the heat transfer coefficient, but was able to predict the observed decrease of heat transfer with increasing vapor quality. By multiplying the heat transfer coefficient obtained with the Gungor-Winterton correlation [22] by a factor 1.2, the mean absolute deviation was only 9 %.

![Figure 2.11: Test section with a countercurrent heat exchanger](image)

Figure 2.11: Test section with a countercurrent heat exchanger [33], T and P stand for temperature and pressure sensor respectively, R and W for refrigerant and water
Grauso et al. [58] examined the flow boiling heat transfer of R-410a and CO$_2$ at low to medium reduced pressures in a 6 mm inner diameter, horizontal smooth tube. The main purpose was to investigate the effect of the reduced pressure on the heat transfer. The fluid first flowed through an adiabatic test section, and then through a diabatic one. In the adiabatic test section, the pressure drop was measured, while in the diabatic test section, the heat transfer coefficient was measured. After the diabatic test section, a sight-glass with a high-speed camera was installed to observe the flow pattern. Slug, intermittent and annular flow could be observed this way. The dryout flow pattern was observed indirectly by the strong decrease of heat transfer at the top of the tube. It was concluded that the heat transfer decreased with increasing vapor quality at high reduced pressure, high heat flux and low mass flux. At high mass fluxes, the heat transfer coefficients were more or less constant. The data points were also compared with various heat transfer correlations.

Oh et al. [39] boiled CO$_2$ in a horizontal tube with a 7.75 mm inner diameter at saturation temperatures between −5°C and 5°C, and compared the result with experimental data of R-134a and R-22. While the local heat transfer coefficients of R-134a and R-22 increased with vapor quality, it decreased for CO$_2$. In a large range of vapor qualities, the heat transfer coefficient of CO$_2$ was significantly higher than the heat transfer coefficient of the other two refrigerants. It was found that existing correlations underestimate the heat transfer coefficient of CO$_2$, because the heat transfer coefficient of this refrigerant is highly dependent on heat flux and saturation temperature. It was concluded that new correlations should be developed for CO$_2$.

The experimental conditions of the discussed experiments are listed in Table 2.2.
Table 2.2: An overview of the experimental conditions of the discussed experiments

<table>
<thead>
<tr>
<th>Investigators</th>
<th>Fluid</th>
<th>$D_i$ (mm)</th>
<th>$T_{sat}$ ($^\circ$C)</th>
<th>$P_{sat}$ (bar)</th>
<th>$G$ (kg/(m$^2$s))</th>
<th>$\dot{q}$ (kW/m$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bandarra et al. [34]</td>
<td>R-404a, R-407c</td>
<td>12.7</td>
<td>8</td>
<td></td>
<td>100 to 300</td>
<td>5 to 20</td>
</tr>
<tr>
<td>Greco and Vanoli [36]</td>
<td>R-404a, R-410a</td>
<td>6</td>
<td>-14.9 to 23.5</td>
<td>3 to 12</td>
<td>290 to 1100</td>
<td>11 to 39</td>
</tr>
<tr>
<td>Kim, Seo and Chung [47]</td>
<td>R-410a</td>
<td>6.18, 8.82</td>
<td>-15 to 5</td>
<td></td>
<td>70 to 211</td>
<td>5 to 15</td>
</tr>
<tr>
<td>Shin, Kim and Ro [48]</td>
<td>R-22, R-32, R-134a, R-290, R-600ac</td>
<td>7.7</td>
<td>12</td>
<td></td>
<td>424 to 742</td>
<td>10 to 30</td>
</tr>
<tr>
<td>Jung et al. [49]</td>
<td>R-22, R-114</td>
<td>9</td>
<td>/</td>
<td>2.6, 4</td>
<td>68 to 197</td>
<td>10 to 45</td>
</tr>
<tr>
<td>Niederkruger et al. [51]</td>
<td>R-846, R-12</td>
<td>14</td>
<td>/</td>
<td>5, 15, 25</td>
<td>80 to 400</td>
<td>$\leq$ 100</td>
</tr>
<tr>
<td>Kuo and Wang [52]</td>
<td>R-22, R-407c</td>
<td>9.52</td>
<td>1.66, 6.04</td>
<td>6</td>
<td>100 to 300</td>
<td>6 to 14</td>
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<tr>
<td>Yu et al. [53]</td>
<td>R-134a</td>
<td>10.7</td>
<td>6</td>
<td></td>
<td>163 to 408</td>
<td>2.2 to 56</td>
</tr>
<tr>
<td>Zou et al. [54]</td>
<td>R-170, R-290</td>
<td>8</td>
<td>/</td>
<td>3.5 to 5.7</td>
<td>64 to 103</td>
<td>13 to 66</td>
</tr>
<tr>
<td>Park et al. [56]</td>
<td>CO$_2$, R-22, R-410a</td>
<td>6.1</td>
<td>-30 to -15</td>
<td></td>
<td>10 to 400</td>
<td>5 to 15</td>
</tr>
<tr>
<td>Da Lima Silva et al. [33]</td>
<td>R-134a</td>
<td>13.8</td>
<td>5, 15, 20</td>
<td></td>
<td>300 to 500</td>
<td>8 to 18</td>
</tr>
<tr>
<td>Cho and Kim [38]</td>
<td>CO$_2$</td>
<td>4.5, 8.62</td>
<td>0 to 20</td>
<td></td>
<td>212 to 656</td>
<td>6 to 20</td>
</tr>
<tr>
<td>Col [57]</td>
<td>R-22, R-134a, R-125, R-410a</td>
<td>8</td>
<td>25 to 45</td>
<td></td>
<td>200 to 600</td>
<td>9 to 53</td>
</tr>
<tr>
<td>Grauso et al. [58]</td>
<td>R-410a, CO$_2$</td>
<td>6</td>
<td>5 to 42</td>
<td></td>
<td>150 to 500</td>
<td>5 to 20</td>
</tr>
<tr>
<td>Oh et al. [39]</td>
<td>CO$_2$</td>
<td>7.75</td>
<td>-5, 0, 5</td>
<td></td>
<td>200 to 500</td>
<td>10 to 40</td>
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Chapter 3

Experimental flow boiling test facility

This chapter provides a description of the experimental flow boiling test facility that is used for the heat transfer experiments. In the first part, the different components of the test facility are described. Thereafter, the measurement sensors are discussed. Subsequently, the data acquisition system is explained. The final section covers the dimensioning of the new components of the test facility.

3.1 Description of the flow boiling test facility

The aim of the test facility is to control the following properties in the test section: the mass flow rate, the inlet vapor quality, the outlet vapor quality, the heat flux and the saturation pressure. The schematic of the test facility is given in Figure 3.1. The test facility consists of two loops: the experimental loop and the cooling loop. In the experimental loop, the fluid is first preheated, such that it reaches the desired condition at the inlet of the test section. In the test section, the fluid is boiled (partially or completely). Then the fluid is expanded to a lower pressure, by the electronic expansion valve. Next, the fluid is condensed, to ensure proper functioning of the pump, since the pump can only deliver liquid phase. The fluid then ends up in the liquid receiver. This is a two-phase reservoir with liquid in the lower part, and vapor in the upper part. The bottom of the liquid receiver is connected to the subcooler, where the liquid is subcooled a few degrees. Then it ends up in the pump, which overcomes all pressure drops in the experimental loop and provides the desired mass flow rate. The cooling loop interacts with the experimental loop in the condenser, where it removes the condensation heat, and in the subcooler.

The test facility was designed for experiments in ORC conditions, for the fluids Solkatherm® SES36 and R-245fa. The choice of the saturation temperatures of the experiments leads to different saturation pressures for these two fluids. For the same saturation temperature, R-245fa has a higher saturation pressure, so R-245fa determined the maximum pressure of the experimental loop. This loop
CHAPTER 3. EXPERIMENTAL FLOW BOILING TEST FACILITY

was designed for a maximal pressure of 4 MPa and a maximal temperature of 150 °C. The test section was designed such that the tubes can be replaced easily. This way, experiments can be performed with different tube materials and various outer tube diameters, ranging from 0.5 inches (12.7 mm) to 1.5 inches (38 mm). More information about the test section can be found in section 3.1.1.2. The range of the mass velocities in the test section varies from 50 kg/(m² s) to 2000 kg/(m² s), depending on the diameter of the tube.

3.1.1 Experimental loop

The general functioning of the experimental loop was already described briefly. Here, the main components of Figure 3.1 are explained in more detail.

3.1.1.1 Preheater

The preheater is placed between the accumulator and the test section. It consists of four flange immersion heaters that are heated electrically, with a total power of 40 kW. The preheater can be regulated in two ways. In the first method, the outlet temperature of the preheater is regulated by the device itself, using an in-built PID-controller and a Pt-100 Resistance Temperature Detector. The second method allows the user to adapt the power of the preheater manually. The latter method will be used to control the test facility. There are two safety measures installed. A thermocouple is placed on the upper immersion heater, close to the outlet of the preheater. And a thermostat is placed on top of the device, it will cause the preheater to shut down when temperatures above 300 °C are reached. The preheater needs a volume flow rate of at least 2 L/min to ensure safe operation.

3.1.1.2 Test section

As stated before, the tube of the test section can be changed to tubes with other diameters or other materials. At the inlet of the test section, a capacitive sensor is installed. It estimates the void fraction, to give an indication of the vapor quality and it is used to detect the flow pattern. There are sight glasses installed at the inlet and the outlet of the test section, these can be used with high speed cameras to detect the flow pattern. The test section is also heated electrically, by the Joule effect. Both ends of the test section are connected to a DC power supply of 12 kW. An electrical current flows through the tube, depending on the voltage difference across the test section and the electrical resistance of the tube. The electrical energy is dissipated and heat is produced. The heating device can provide a voltage from 0 V to 72 V and a current from 0 A to 170 A. In the future, it will be necessary to use a resistor wrapped around the test section to provide the heat flux, because the resistance of the tube can be too small for other tube materials. Thermocouples are placed at 24 different axial locations along the test section. At each of these locations, three thermocouples are
Figure 3.1: Schematic of the experimental flow boiling test facility
installed. This leads to a total of 71 thermocouples (at one axial location, only two thermocouples were installed). At every axial location where thermocouples are installed, one is installed at the top of the tube, one at the bottom and one alternately at the left and right side, as can be seen in a cross sectional view of the test section in Figure 3.2, and in a side view in Figure 3.3. These thermocouples are used to measure the outer wall temperature of the test section very accurately. From this, the inner wall temperature can be calculated (see section 4.7). This information will then be used to calculate the local heat transfer coefficients. A Kapton® MT polyimide film is wrapped around the tube, to insulate it electrically. This film is very thin and a relatively good thermal conductor. The thermocouples are attached to this tape and thermal paste is spread on it to increase the thermal conductivity. Around the thermocouples, another film of Kapton® MT polyimide is placed, such that they remain in place. Finally, the test section is insulated with a 32 mm thick EPDM tube insulation to minimize heat losses.

3.1.1.3 Proportional relief valve

A proportional relief valve is installed after the test section and before the electronic expansion valve. If the pressure rises above 25 bar, this device gradually opens, preventing the other components of the test facility to be damaged. When the pressure is again below 25 bar, the device closes.

3.1.1.4 Electronic expansion valve

The electronic expansion valve is installed before the condenser. It is needed to reduce the saturation temperature of the experimental fluid, because otherwise the fluid in the cooling loop might heat up too much and it could start to evaporate. This would lead to an increased pressure in the cooling loop and could damage the circuit, because it was only designed for liquid flow. The expansion valve has a $k_{vs}$ value of 2.5 m$^3$/h, this value expresses the flow rate through the fully open

![Figure 3.2: Cross section of the test section](image-url)
valve at a differential pressure of 1 bar. The stroke of the valve can be set with an input voltage, 0 V corresponds with a closed valve and 10 V with a completely open valve. The relationship between the input voltage and the stroke is linear, so any voltage between 0 V and 10 V will set the valve in an intermediate position. During experiments, the stroke of the valve will be chosen such that the condensing temperature of the experimental fluid is sufficiently low and the mass flow rate is not limited by the expansion valve. The valve can be used for fluid temperatures between $-40^\circ C$ and $120^\circ C$.

3.1.1.5 Condenser

The condenser is a plate heat exchanger with 90 plates. The experimental fluid enters in the two-phase state and it should be condensed, such that it can be evaporated again in the test section. The secondary fluid circulates in the cooling loop, and it extracts heat from the primary (experimental) fluid. The secondary fluid is a water/glycol mixture. As the experimental fluid is condensed, the secondary fluid heats up and it will be cooled down again in the cooling loop, this is explained in section 3.1.2. The maximal cooling power of the condenser is 50 kW, and it can be adapted with a trunnion ball valve, which changes the mass flow rate of the water/glycol mixture.

3.1.1.6 Liquid receiver

A liquid receiver is installed between the condenser and the pump. It is needed to prevent cavitation at the suction line of the pump. The liquid receiver is a two-phase reservoir: it stores liquid and vapor. The bottom of the liquid receiver is connected to the pump inlet, such that only liquid ends up in the pump. The liquid receiver has a volume of 30 L. There are three sight glasses at different heights in the liquid receiver, such that the liquid level can be observed.

3.1.1.7 Subcooler

The subcooler is located between the liquid receiver and the pump. The subcooler has a cooling power of 2 kW. It is also connected to the cooling loop, and it ensures that the liquid is subcooled a few degrees. In the liquid receiver, the fluid is very close to its saturation pressure, because a two-phase state is present there. This means that there is only a very small margin with regard
to cavitation in the pump. Avoiding cavitation can be done in several ways, and one of them is thermally subcooling the working fluid with a subcooler after the liquid receiver [3]. This is further explained in section 3.4.3 and section 7.4.

3.1.1.8 Pump

The installed pump is a sliding vane pump with a permanent magnet drive system, driven by a variable speed drive. The advantages are that the experimental fluid lubricates the pump, no oil is required, and that there are no mechanical seals, which give rise to leakages in other pumps. The pump can provide a pressure increase up to 14 bar, can be used for temperatures up to 200°C and can withstand system pressures up to 25 bar. The pump is driven by an induction motor. At the nominal frequency, the motor speed is 1450 rpm and this corresponds with a volume flow rate between 310 L/h and 360 L/h, depending on the pressure increase. The pump characteristic at the nominal speed is shown in Figure 3.4. When the pressure increase by the pump increases, the volume flow rate diminishes due to backflow in the pump. In fact, the pump characteristic is almost a vertical line, because the volume flow rate on the x-axis represents only a small range. Physically, this means that the pump gives a more or less constant volume flow rate over the entire pressure range, at constant pump speeds. If the motor frequency is lowered, then the volume flow rate varies proportionally with this frequency (assuming that the pressure increase remains constant).

3.1.1.9 Accumulator

The accumulator is located after the outlet of the pump. Its main function is to dampen pressure and mass flow rate fluctuations. These originate from the fact that the pump is a positive displacement machine. The volume of the accumulator is 32 L. The accumulator has an internal bladder, which is pre-charged with nitrogen at 8 bar. When experiments are performed, the pressure of the experimental loop will be above this pressure (it will be about 10 bar). In that case, the bladder is compressed and some amount of the experimental fluid is stored inside the accumulator. One can understand that the accumulator stabilizes the system pressure as follows: if the system pressure would drop slightly, then the bladder would expand a bit. Some amount of the experimental fluid initially stored in the accumulator is brought back into the experimental loop, this results in a pressure increase. The amount of experimental fluid in the accumulator is dependent on the experimental condition, because it depends on the line pressure at the location of the accumulator.

3.1.2 Cooling loop

The cooling loop is used by several test facilities in the lab of the Applied Thermodynamics and Heat Transfer research group, and has already been described in other theses [59]. The cooling fluid is a
water/glycol mixture (70 %/30 % volume mixture). The cooling fluid is stored in a cold vessel, which is connected to a chiller, as can be seen in Figure 3.5. The desired temperature of the cooling fluid can be set, and a thermostat switches the chiller on or off accordingly. The mass flow rate of the cooling fluid through the condenser and the subcooler is adapted by controlling the 3-way valves.

3.2 Measuring equipment

In the test facility, three physical quantities are measured: the mass flow rate, the temperature and the pressure. The different sensors are described below.

![Figure 3.5: Schematic of the cooling loop](image)
3.2.1 Mass flow rate sensor

The mass flow rate sensor's working principle is based on the Coriolis effect. The fluid passes through a coil which is energized, the coil oscillates in a sine motion. This motion is observed by two sensors. The phase shift of this sine motion is dependent on and proportional to the mass flow rate through the coil. The sensor can also determine the density of the fluid by evaluating the frequency of the sine motion and measuring the fluid temperature with a Pt-500 sensor. The nominal mass flow rate of this sensor is 1200 kg/h, and it can be used for fluid temperatures between $-70^\circC$ and $230^\circC$.

3.2.2 Temperature sensors

Thermocouples are used to measure the outer wall temperatures in the test section. The working principle of a thermocouple relies on the Seebeck effect: a flow of current in a metal is associated with a flow of heat, or a temperature difference [59, 60]. A thermocouple consists of two different materials, joined together at two points or junctions. One junction is located at the place where the temperature is measured and the other is the reference junction. The thermocouple measures the temperature difference between its two junctions. Therefore, it is required that the reference temperature is measured. This is done with a Pt-100 Resistance Temperature Detector (RTD). This temperature sensor works based on the principle that the electrical resistance of a metal depends on its temperature [60]. All reference junctions and the Pt-100 sensor are put in a box that is thermally insulated from the surroundings. This way, influences from the environment are minimized and all reference temperatures are the same.

3.2.3 Pressure transducer

There are two kind of pressure transducers in the test facility: absolute and differential pressure transducers. The absolute pressure transducer measures the gauge pressure at a location: the pressure of the experimental fluid with respect to the atmospheric pressure. There are three pressure transducers of this kind: at the inlet of the test section, at the outlet of the condenser and at the inlet of the preheater. These sensors have a range of 0 bar to 45 bar. The differential pressure sensor is installed in the test section, it measures the pressure drop between the test section inlet and outlet, and it has a range of 35 kPa. By combining the measurements from the gauge pressure sensor and the differential pressure sensor, the pressure at every location in the test section can be calculated through interpolation. These pressure transducers are indicated in the schematic of the test facility in Figure 3.1.
3.3 Data acquisition

All electronic components are connected to the data acquisition system, which is connected to a personal computer. Its two functions are:

- read out and log measurement data of the sensors
- control and monitor the electronic devices

The data acquisition system consists of a CompactRIO platform and a Keithley device, both connected to the computer. On the computer, a program in the LabVIEW environment runs and communicates with both devices. The Keithley device is only used for the thermocouples, all other electronic components are connected to the CompactRIO platform.

The measuring devices represent their measurements with a voltage between 0 V and 10 V. From this, the actual physical quantity can be deduced. For example, the pressure sensor at the test section inlet. It has a range of 45 bar, 0 V corresponds with 0 barg and 10 V corresponds with 45 barg. The following electronic devices are controlled:

- the preheater
- the test section heater
- the variable speed drive
- the valves of the condenser and subcooler
- the electronic expansion valve

They are controlled by setting a voltage between 0 V and 10 V, these voltages correspond with the lower bound and the upper bound of the device, respectively.

From the LabVIEW program, an Excel sheet will be written with all measurement data and the set-points of the electronic devices. This information will be used in the data reduction and the uncertainty analysis, in Chapter 4 and Chapter 5.

3.4 Dimensioning of new components

When doing initial tests, a few problems emerged. Therefore, extra components needed to be installed in the test facility. These have already been described above, but the dimensioning of the components is explained in this section.
3.4.1 Electronic expansion valve

Without an electronic expansion valve, the experimental fluid is condensed at a temperature of approximately 120 °C. When the mass flow rate of the cooling fluid in the condenser is relatively low, it may start to evaporate. The cooling circuit was not designed for these conditions, and it would be damaged if boiling occurred. For this reason, an expansion valve, which decreases the condensing temperature and thus the temperature of the cooling fluid, was installed. The maximum value of the volume flow rate in the system is 1.35 m³/h. This is based on a mass flow rate of 0.1 kg/s and a density of the saturated vapor phase of Solkatherm® SES36 at $T = 120 \, ^\circ\text{C}$: $\rho_g = 0.074 \, \text{kg/m}^3$. Electronic expansion valves of Siemens are available with $k_{vs}$ values of 0.4 m³/h, 1.0 m³/h, 2.5 m³/h and 6.3 m³/h. The expansion valve with the $k_{vs}$ value of 2.5 m³/h was chosen, such that the maximum volume flow rate is allowed through the fully open valve with a relative small pressure drop, only 0.3 bar. By adapting the stroke of the expansion valve, a maximum pressure drop of 25 bar can be achieved, which is sufficient. To limit the condensing temperature of Solkatherm® SES36 to 80 °C, a pressure drop of 6 bar would be required in the expansion valve. This condensing temperature is regarded completely safe, since the water-glycol mixture only starts evaporating at a higher temperature (104 °C).

3.4.2 Liquid receiver

The liquid receiver was installed, after cavitation occurred in the pump when starting the test facility. It is installed together with the subcooler, and this way cavitation can be avoided. The volume of the liquid receiver is determined with the following volumes:

- volume of the preheater: $V_{ph} = 23.5\, \text{L}$
- volume of the condenser: $V_{cond} = 5\, \text{L}$
- maximum volume in the accumulator: $V_{acc} = 15\, \text{L}$

The other volumes of the system are negligible with respect to these volumes. The test facility will in the future also be used for another fluid, R-245fa, which has a higher saturation pressure than Solkatherm® SES36 at $T = 120 \, ^\circ\text{C}$, namely 19.3 bar. This fluid determines the maximum volume of the fluid in the accumulator. The volume of the compressed nitrogen in the accumulator in this condition is calculated as follows:

$$ p_1 \cdot V_1^k = p_2 \cdot V_2^k $$

with 1 the precharged condition, only nitrogen present in the accumulator ($p_1 = 8 \, \text{bar}$ and $V_1 = 32\, \text{L}$), 2 the condition with R-245fa at 19.3 bar, and $k = 1.4$ the specific heat ratio of nitrogen (assuming
that the process is isentropic). It follows that:

\[ V_2 = \left( \frac{p_1}{p_2} \right)^\frac{1}{k} \cdot V_1 = \left( \frac{8 \text{ bar}}{19.3 \text{ bar}} \right)^\frac{1}{1.4} \cdot 32 \text{ L} \approx 17 \text{ L} \]

The volume of R-245fa in the accumulator is then: 32 L - 17 L \approx 15 L. Because the accumulator only opens up at a system pressure of 8 bar, the accumulator is initially not filled with experimental fluid. So the liquid receiver is designed to hold at least 15 L of fluid. Next to that, the pump also needs an amount of liquid to ensure proper functioning, this amount is chosen as a fraction of the volumes of the condenser and the preheater:

\[ V_{tot} = V_{acc} + 0.5 \cdot (V_{ph} + V_{cond}) \approx 29 \text{ L} \]

Because this volume was not available, a liquid receiver with a volume of 30 L was chosen.

### Subcooler

In the pump manual, it can be found that the head at the suction line of the pump needs to be at least 0.5 m to 1 m above the saturation pressure of the fluid at that location. The calculation will be done with the most stringent condition, the latter one. This condition may be expressed in terms of a pressure difference with respect to the saturation pressure:

\[ \Delta p = \rho \cdot g \cdot h \]

with \( h \) the head expressed in meters, \( \rho \) the density of the experimental fluid in the pump and \( g \) the gravitational constant. When starting the cycle, the temperature of the fluid is equal to the room temperature, with a saturated liquid density of 1.38 kg/m³. During operation, the temperature in the liquid receiver is about 80 °C, with a saturated liquid density of 1.218 kg/m³. These result in the requirements of \( \Delta p \approx 13.5 \text{ kPa} \) and \( \Delta p \approx 12.0 \text{ kPa} \), respectively.

The amount of subcooling required to fulfill this pressure increase, can be calculated using CoolProp. The following equation needs to be solved for \( \Delta T_{sc} \), the amount of subcooling:

\[ \Delta p = p(T = T_{sat}, x = 0) - p(T = T_{sat} - \Delta T_{sc}, x = 0) \]  

(3.1)

with \( \Delta p \) as determined earlier and \( T_{sat} \) the saturation temperature in the liquid receiver, 20 °C and 80 °C for the two considered cases. The first term in equation 3.1 is the actual pressure at the pump inlet, and the second term is the saturation pressure at that location. Solving this equation results in \( \Delta T_{sc} = 7 \text{ K} \) at ambient temperature, and \( \Delta T_{sc} = 1.2 \text{ K} \) at 80°C. Now the cooling power of the subcooler can be calculated. Subcooling 7 K corresponds with an enthalpy difference of \( \Delta h = 7415 \text{ J/kg} \).
For a mass flow rate of the experimental fluid of 0.1 kg/s, the required cooling power is:

\[ \dot{Q}_{\text{cool}} = \dot{m} \cdot \Delta h = 0.1 \text{ kg/s} \cdot 7415 \text{ J/kg} \approx 742 \text{ W} \]

A subcooler with a cooling capacity of 2 kW is installed, which should be sufficient based on this reasoning. In the calculation, a few influences were neglected. First, there is a pressure drop in the tube connecting the subcooler to the pump. This effect partially cancels the subcooling effect. However, another factor that was not taken into account is the hydrostatic pressure effect in the liquid receiver. Depending on the liquid level, this effect leads to a pressure increase. It is concluded that most likely, the subcooling of 7 K is more than sufficient, also because a head difference of 1 m was used in these calculations.
Chapter 4

Data reduction

Some physical quantities cannot be measured directly, but they can be derived from the measurements of the mass flow rate, temperature and pressure in the test facility, and the heating power of the preheater and the test section heater. This is called the data reduction and it is explained in this chapter. In the last section, the heat loss of the test section is estimated.

4.1 Pressure

At the test section inlet, the pressure is known. The differential pressure sensor gives the pressure drop across the test section. By interpolating linearly between the inlet and outlet pressure, the local saturation pressure at each location is obtained. The interpolation is done with the following formula:

\[ p_{\text{sat},y} = p_{\text{ts},\text{in}} - \Delta p_{\text{tot}} \cdot \frac{y}{L_{\text{ts}}} \]  

(4.1)

where \( y \) is the distance measured from the inlet of the test section, and \( L_{\text{ts}} \) is the total length of the test section.

4.2 Vapor quality

At the entrance of the preheater, the thermodynamic state of the fluid is known, because the temperature and the pressure of the fluid are measured there. This means that the specific enthalpy at the entrance of the preheater is known, and it is evaluated using CoolProp [61]. An energy balance of the preheater leads to the following equation:

\[ \dot{Q}_{\text{ph}} = \dot{m} \cdot (h_{\text{ph, out}} - h_{\text{ph, in}}) \]  

(4.2)
The vapor quality at the outlet of the preheater is defined as:
\[ x_{ph, out} = \frac{h_{ph, out} - h_{ph, l, sat}}{h_{ph, g, sat} - h_{ph, l, sat}} \] (4.3)

where \( h_{ph, l, sat} \) and \( h_{ph, g, sat} \) are the specific enthalpy of the saturated liquid and saturated vapor at the outlet of the preheater. If equation 4.2 is substituted into equation 4.3, the following formula is obtained:
\[ x_{ph, out} = \frac{h_{ph, in} + \dot{Q}_{ph} \dot{m} - h_{ph, l, sat}}{h_{ph, g, sat} - h_{ph, l, sat}} \] (4.4)

At the inlet of the test section, the vapor quality is approximately equal to the outlet vapor quality of the preheater: \( x_{ts, in} \approx x_{ph, out} \), the difference between these values is caused by the frictional pressure drop in the tubes between the preheater and the test section. The vapor quality at the inlet of the test section is given by:
\[ x_{ts, in} = \frac{h_{ts, in} + \dot{Q}_{ts} \dot{m} - h_{ts, l, sat}}{h_{ts, g, sat} - h_{ts, l, sat}} \] (4.5)

With an energy balance of the test section, the outlet vapor quality may be calculated:
\[ \dot{Q}_{ts} = \dot{m}(h_{ts, out} - h_{ts, in}) \]

The vapor quality at the outlet of the test section is:
\[ x_{ts, out} = \frac{h_{ts, out} - h_{ts, l, sat}}{h_{ts, g, sat} - h_{ts, l, sat}} \]

with \( h_{ts, l, sat} \) and \( h_{ts, g, sat} \) the specific enthalpy of the saturated liquid and saturated vapor, evaluated at the outlet of the test section. In a similar way as before, the following equation for the vapor quality at the outlet of the test section is obtained:
\[ x_{ts, out} = \frac{h_{ts, in} + \dot{Q}_{ts} \dot{m} - h_{ts, l, sat}}{h_{ts, g, sat} - h_{ts, l, sat}} = \frac{h_{ph, in} + \dot{Q}_{ph} \dot{m}}{h_{ts, g, sat} - h_{ts, l, sat}} + \frac{\dot{Q}_{ts} \dot{m}}{h_{ts, g, sat} - h_{ts, l, sat}} - h_{ts, l, sat} \] (4.6)

The vapor quality at a location \( y \) in the test section, is obtained by interpolating linearly between the inlet and outlet specific enthalpies:
\[ x_{ts, y} = \frac{h_{ph, in} + \dot{Q}_{ph} \dot{m} + \frac{y}{L_{ts}} \cdot \dot{Q}_{ts} \dot{m} - h_{ts, l, sat}}{h_{ts, g, sat} - h_{ts, l, sat}} \] (4.7)

In these equations, the saturated liquid and saturated vapor specific enthalpies need to be evaluated at the local saturation pressure. This pressure can be calculated with equation 4.1.
4.3 Mass velocity

The mass velocity is defined as the mass flow rate per unit of cross-sectional area. Thus, it is calculated directly from the mass flow rate $\dot{m}$ and the tube’s inner diameter $D_i$:

$$G = \frac{\dot{m}}{\pi \cdot D_i^2}$$  \hspace{1cm} (4.8)

4.4 Heating power

The heating power in the test section, is the power that is dissipated in the electrical resistance of the test section:

$$\dot{Q}_{ts} = U \cdot I$$  \hspace{1cm} (4.9)

where I is the DC current through the test section, and U is the voltage difference over the test section.

4.5 Heat flux

The heat flux is defined as the heating power per surface unit of heat transfer. The heat is transferred to the fluid at the inner diameter of the test section. So, this surface is used to define the heat flux:

$$\dot{q} = \frac{\dot{Q}_{ts}}{\pi \cdot D_i \cdot L_{ts}}$$  \hspace{1cm} (4.10)

4.6 Volumetric heat flux

The volumetric heat flux is defined as the heating power per volume unit of heat generation. The heating power is generated in the tube of the test section by the Joule effect, so the volume of the tube is used to calculate the volumetric heat flux:

$$\dot{q}'' = \frac{\dot{Q}_{ts}}{L_{ts} \cdot \pi \cdot (D_o^2 - D_i^2) / 4}$$  \hspace{1cm} (4.11)
4.7 Heat transfer coefficient

The local heat transfer coefficient is obtained at several locations distributed axially and circumferentially along the test section. The local two-phase heat transfer coefficient is defined as:

\[ h_{tp} = \frac{\dot{q}}{T_{w,i} - T_b} \]  

(4.12)

where \( \dot{q} \) is the heat flux, \( T_{w,i} \) is the inner wall temperature and \( T_b \) is the bulk temperature of the fluid. The inner wall temperature is calculated from the outer wall temperature by assuming a one-dimensional, steady-state, cylindrical heat conduction problem. Because the tube is heated by the Joule effect, there is a contribution of both heat generation and heat conduction in the tube. By using a shell balance approach, a differential equation is obtained for the radial temperature profile in the tube [62]. Assuming that the outer wall of the tube is perfectly insulated from the surroundings and that its temperature is known, the required boundary conditions are obtained. This results in the following equation:

\[ T_{w,i} = T_{w,o} + \frac{\dot{q}'' \cdot (D_o^2 - D_i^2)}{16 \cdot \lambda_{ts}} \cdot \left( \frac{\dot{q}''}{8 \cdot \lambda_{ts} \cdot D_o^2} \right) \cdot \ln \left( \frac{D_o}{D_i} \right) \]  

(4.13)

where \( \dot{q}'' \) is the volumetric heat flux.

The bulk temperature of the fluid follows from the local saturation pressure, which is calculated with equation 4.1. There are three thermocouples at each axial location, for every thermocouple, the two-phase heat transfer coefficient is calculated. An average local heat transfer coefficient may then be calculated as follows:

\[ h_{tp} = \frac{h_{tp,\text{top}} + h_{tp,\text{bot}} + h_{tp,\text{side}}}{3} \]

where \( h_{tp,\text{top}}, h_{tp,\text{bot}} \) and \( h_{tp,\text{side}} \) are the local two-phase heat transfer coefficients at the top, the bottom and the side of the tube, respectively.

4.8 Estimation of heat loss

In the test section, some of the applied heating power is lost to the environment via conduction through the insulation and natural convection at the outside of the insulation. The amount of heat lost, depends on these two thermal resistances. In order to perform experiments in a correct manner, the amount of heat lost should be known and it should be as small as possible. In the following, a method to calculate this heat loss is explained. In Figure 4.1, a schematic overview of the cross section of the test section is shown. It consists of a tube and a layer of insulation. Let \( T_1 \) be the temperature between the tube and the insulation, \( T_2 \) the temperature at the outer diameter of the insulation and \( T_3 \) the ambient temperature. The ambient temperature and pressure are known. It
will be assumed that the heat loss $\dot{Q}_{\text{loss}}$ is known, and the wall temperature $T_1$ corresponding with this heat loss will be calculated. If an initial guess is made for $T_2$, the final solution can be found through iteration. For example:

$$T_2 = T_3 + 20 \text{K}$$

Now, the Nusselt number should be calculated, such that the heat transfer coefficient can be obtained. Then, the wall temperature $T_2$ may be recalculated. The Nusselt number with respect to the outer diameter of the insulation is defined as:

$$Nu_D = \frac{h \cdot D_2}{\lambda_{\text{air}}}$$  \hspace{1cm} (4.14)

In which $h$ is the convective heat transfer coefficient, $D_2$ is the diameter of the insulation and $\lambda_{\text{air}}$ is the thermal conductivity of the fluid, air in this case. Using the Nusselt correlation [63], the Nusselt number is calculated:

$$Nu_D = 0.502 \cdot (Gr_D \cdot Pr)^{0.25}$$  \hspace{1cm} (4.15)

In which $Gr_D$ is the Grashof number with respect to the insulation's outer diameter $D_2$ and $Pr$ is the Prandtl number. This correlation is valid for horizontal cylinders in air or liquids for $10^4 \leq Gr_D \cdot Pr \leq 10^8$ [63]. The Grashof number for cylinders is defined as:

$$Gr_D = \frac{g \cdot \beta \cdot (T_2 - T_3) \cdot D_2^3}{\nu^2}$$

In which $g$ is the gravitational acceleration, $\beta$ is the coefficient of thermal expansion, $T_2$ and $T_3$ are the previously defined temperatures, $D_2$ is the diameter of the insulation and $\nu$ is the kinematic viscosity of the fluid. All fluid properties (including the Prandtl number) are calculated at the bulk
temperature of the fluid:

\[ T_b = \frac{T_2 + T_3}{2} \]

Now, equation 4.15 can be evaluated. The result is inserted into equation 4.14 and the convective heat transfer coefficient is obtained. \( T_2 \) is recalculated:

\[ \dot{q}_{loss} = \frac{\dot{Q}_{loss}}{A_2} = h \cdot (T_2 - T_3) \]

In which \( \dot{q}_{loss} \) is the heat flux corresponding with the heat loss, \( \dot{Q}_{loss} \) is the heat loss, \( A_2 \) is the surface of the insulation \((A_2 = D_2 \cdot \pi \cdot L_{ts}, \text{with } L_{ts} \text{ the length of the test section})\). After iterations, the final value of \( T_2 \) is found. Now, \( T_1 \) can be calculated from a heat conduction problem:

\[ \dot{Q}_{loss} = \frac{T_1 - T_2}{R_{12}} \]

\[ R_{12} = \frac{D_2 - D_1}{2 \cdot \lambda_{ins} \cdot A_{lm}} \]

\[ A_{lm} = \frac{A_2 - A_1}{\ln(A_2/A_1)} \]

with \( R_{12} \) the thermal resistance of the insulation, \( A_2 \) and \( A_1 \) the heat transfer surfaces corresponding with \( D_2 \) and \( D_1 \), respectively and \( \lambda_{ins} \) the thermal conductivity of the insulation. This method was implemented, and the tube temperatures \( T_1 \) were calculated for different heat losses \( \dot{Q}_{loss} \). The result is depicted in Figure 4.2. The heat loss with a wall temperature of \( 120 \, ^\circ \text{C} \) is estimated to be \( 45.1 \, \text{W} \). It can be seen that with this model, the heat loss increases linearly with the tube temperature. This behavior was also observed in heat loss experiments [64].

In some investigations, this relation was obtained experimentally. This can be done as follows. The test section is heated up while there is no fluid in it yet. After some time, the temperature of the tube reaches steady-state, this is when the heat loss is equal to the heating power. So, by measuring the tube temperature and the heating power, the relation depicted in Figure 4.2 is obtained experimentally. This was not done due to time and practical constraints. When doing experiments, the heat loss is obtained by inserting the wall temperature in this relation [54, 64].
Figure 4.2: Heat loss versus tube temperature
Chapter 5

Accuracy of measurements

The purpose of the current research is to perform two-phase flow measurements. The local heat transfer coefficient and the two-phase pressure drop are both measured in various conditions. This is done using pressure transducers, mass flow meters and thermocouples. The working principle of these sensors was described in section 3.2. It is inherent to sensors that they do not measure physical quantities exactly, but they have some error. The error of a measurement is the difference between the actual and the measured value, as defined by Moffat [65]. The uncertainty is a possible value that the error may have [66]. In engineering experiments, it is customary to report estimates of the uncertainties of the measuring equipment and the measured physical quantities. With the knowledge of these uncertainties, the influence of scatter is described in an objective manner. However, not all physical quantities can be measured directly. Some of them, for example the enthalpy and the local heat transfer coefficient, are obtained by measuring other quantities. This was explained in Chapter 4. These quantities will also have some uncertainty, because the error of the measurements from which they are determined, propagates.

Suppose that a variable $X_i$ is measured, and that its uncertainty is $\delta X_i$. This may be represented as follows [65]:

$$X_i = X_{i,m} \pm \delta X_i$$

where $X_i$ is the real value of this quantity, $X_{i,m}$ is the measured value and $\delta X_i$ is a worst-case value of the uncertainty, which is determined statistically with confidence levels. For example, if in 95% of the cases, the absolute value of the error is smaller than the reported uncertainty $\delta X_i$, then $\delta X_i$ is two times the standard deviation of all possible measurements [65].

In the following, the uncertainties of the measuring equipment are discussed and an uncertainty analysis is done to determine the uncertainties of all important quantities.
5.1 Measurement uncertainties

In this section, an overview is given of the uncertainties that influence the uncertainties of the measurements.

5.1.1 Geometrical

The inner and outer diameter of the test section were measured with a caliper, with an uncertainty of $\delta D_i = \delta D_o = \pm 0.01 \text{ mm}$. This influences the uncertainty of the heat flux, which influences the uncertainty of the heat transfer coefficient. The location of the thermocouples in the test section, and the length of the test section, were measured with an uncertainty of $\delta L_{ts} = \delta y = \pm 0.05 \text{ mm}$.

5.1.2 Temperature

The thermocouples were calibrated by using a Triple Point of Water Cell (TWC) as a reference. This device creates the triple point of water internally, a thermodynamic state with a temperature that is known very precisely: $\pm 0.0005 \degree \text{ C}$ [59]. It generates temperatures between 0 \degree C and 100 \degree C, which are measured by the device with respect to the triple point of water. The calibration is done by comparing this measurement with the measurement of the thermocouple, and a linear fit between these temperatures is made for every thermocouple. The uncertainty of a thermocouple then consists of two parts: the uncertainty due to the deviation with the linear regression: $\delta T_{regr}$, and the uncertainty of the TWC device: $\delta T_{twc}$. Assuming that the uncertainties are independently distributed, the total uncertainty of a thermocouple is:

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

This way, the accuracy was determined for each thermocouple. The uncertainties of the 71 thermocouples range from $\pm 0.07 \text{ K}$ to $\pm 0.11 \text{ K}$, with an average uncertainty of $\pm 0.085 \text{ K}$.

5.1.3 Pressure

There are two important pressure measurements: the measurement of the absolute pressure at the inlet of the test section and the pressure difference across the test section. These are combined to calculate the local saturation pressures and temperatures in the test section, and the local saturation temperature is used to determine the local heat transfer coefficient (see equation 4.12). Thus, the uncertainty of the pressure measurements has an influence on the uncertainty of the local heat transfer coefficient. Next to that, the pressure sensor at the inlet of the preheater will result in an
uncertainty of the enthalpy at that location, and it will consequently influence the uncertainty of the vapor quality in the test section (see equation 4.7).

The uncertainty of the absolute pressure transducer is determined by two effects. The first effect is the accuracy. This quantifies the deviation of the sensor from perfect linear behavior. For both pressure transducers, it is \( \pm 0.04 \% \) FS BSL. Next to that, there is also a thermal error. The compensated temperature range of the sensor is between \(-40^\circ C\) and \(125^\circ C\). In this temperature range, the thermal error is \( \pm 0.24 \% \) FS TEB. Both uncertainties are expressed with respect to the full scale (FS) of the sensor. To obtain the absolute uncertainty, the pressure range needs to be multiplied with this uncertainty. The uncertainty of the differential pressure sensor has two additional effects that lead to an uncertainty: the line pressure and the age of the pressure transducer. The total uncertainty of the pressure sensor is then obtained as follows:

\[
\delta p_{tot} = \sqrt{\delta p_{acc}^2 + \delta p_{temp}^2 + \delta p_{line}^2 + \delta p_{age}^2}
\]

In Table 5.1, the uncertainties of the pressure transducers are summarized. Although the uncertainty of the absolute pressure transducer depends on only two effects, it is still significantly higher than the uncertainty of the differential pressure transducer. This is due to the full scale effect, the scale of absolute pressure sensor (45 bar) is much larger than that of the differential pressure sensor (35 kPa). The total uncertainty of the absolute pressure sensor is given by:

\[
\delta p_{tot, abs} = \sqrt{(0.04 \cdot 10^{-2} \cdot 45 \cdot 10^5)^2 + (0.24 \cdot 10^{-2} \cdot 45 \cdot 10^5)^2} \approx 10.95 \text{ kPa}
\]

The total uncertainty of the differential pressure sensor, at a line pressure of 9.83 bar (corresponding with a saturation temperature of 120\( ^\circ C\) for Solkatherm\(^\text{®}\) SES36) and without the age effect, because it is initially not present, is:

\[
\delta p_{tot, dif} = \sqrt{(0.04 \cdot 10^{-2})^2 + (1.13 \cdot 10^{-2})^2 + (0.06 \cdot 10^{-2} \cdot 9.83)^2 \cdot 35 \cdot 10^3} \approx 446 \text{ Pa}
\]

Table 5.1: The uncertainties of the absolute and differential pressure transducers

<table>
<thead>
<tr>
<th>Effect</th>
<th>Absolute</th>
<th>Differential</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Accuracy</td>
<td>0.04 %</td>
<td>0.04 %</td>
<td>FS BSL</td>
</tr>
<tr>
<td>Temperature</td>
<td>0.24 %</td>
<td>1.13 %</td>
<td>FS TEB</td>
</tr>
<tr>
<td>Line pressure</td>
<td>/</td>
<td>0.06 %</td>
<td>FS/bar</td>
</tr>
<tr>
<td>Age effect</td>
<td>/</td>
<td>0.1 %</td>
<td>FS/year</td>
</tr>
</tbody>
</table>
CHAPTER 5. ACCURACY OF MEASUREMENTS

5.1.4 Mass flow rate

The mass flow rate is measured by a Coriolis mass flow meter. This device was calibrated at three levels of the mass flow rate, by measuring the mass flow rate with a more accurate sensor. The uncertainty $\delta m_{\text{dif}}$ is the difference between the values measured by the two devices. The uncertainty of the device used for the calibration, $\delta m_{\text{cal}}$, needs to be added to the total uncertainty in the following manner:

$$\delta m_{\text{tot}} = \sqrt{\delta m_{\text{dif}}^2 + \delta m_{\text{cal}}^2}$$  \hspace{1cm} (5.1)

Equation 5.1 represents the uncertainty of the mass flow rate, at a certain mass flow rate. The uncertainty for other mass flow rates is obtained through interpolation. In Table 5.2, the results of the calibration are shown. This calibration was done with a measurement device that has an uncertainty of $\delta m_{\text{cal}} = \pm 0.035\%$. The relative uncertainty $\delta m_{\text{tot}} / m_{\text{tot}}$ is smaller than $\pm 0.04\%$ for mass flow rates below 592 kg/h, which corresponds with the experimental conditions.

5.1.5 Preheater

The preheater consists of four flange immersion heaters, each consisting of three heating elements. So, in total there are twelve heating elements, each with a heating power of 3333 W. The uncertainty of the heating power of each heating element is $+5\% / -10\% = +167 W / -333 W$. The uncertainty of the total heating power of the preheater is then: $+\sqrt{12 \cdot 167 W} / -\sqrt{12 \cdot 333 W} \approx +577 W / -1155 W$. This corresponds with $+1.5\% / -2.9\%$.

5.1.6 Test section heater

As stated before, the test section heater can provide a maximal output voltage of 72 V and a maximal output current of 170 A. The uncertainties of these quantities are smaller than $\pm 0.2\%$ and $\pm 0.1\%$, respectively. This results in maximal uncertainties of $\delta U = 0.144 V$ and $\delta I = 0.17 A$. These uncertainties propagate in the heat flux, the volumetric heat flux and the vapor quality.

Table 5.2: The result of the calibration of the mass flow meter

<table>
<thead>
<tr>
<th>Mass flow rate (kg/h)</th>
<th>Deviation (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>125</td>
<td>0.013</td>
</tr>
<tr>
<td>592</td>
<td>0.018</td>
</tr>
<tr>
<td>1059</td>
<td>0.03</td>
</tr>
</tbody>
</table>
5.1.7 Conclusion

The maximal uncertainties of the measuring equipment are summarized in Table 5.3.

5.2 Uncertainty analysis

Suppose that \( q \) is a quantity which is determined from the variables \( X_1, X_2 \ldots X_N: q = q(X_1, X_2, \ldots, X_N) \).

Then, the uncertainty of \( q \) is given by [65]:

\[
\delta q = \sqrt{\sum_i \left( \frac{\partial q}{\partial X_i} \delta X_i \right)^2}
\]

(5.2)

By making use of equation 5.2, the absolute uncertainties of all quantities are derived.

5.2.1 Pressure

The pressure in the test section is found by interpolating between the pressure at the inlet and the pressure at the outlet of the test section:

\[
p_{\text{sat}, y} = p_{\text{ts}, \text{in}} - \Delta p_{\text{tot}} \cdot \frac{y}{L_{\text{ts}}}
\]

(4.1 repeated)

These pressures are obtained from two pressure transducers. Their uncertainties will both influence the uncertainty of the pressure in the test section. Equation 5.2 is used to determine the uncertainty of the saturation pressure in the test section:

\[
\delta p_{\text{sat}, y} = \sqrt{\delta p_{\text{ts}, \text{in}}^2 + \left( \frac{y \cdot \delta \Delta p_{\text{tot}}}{L_{\text{ts}}} \right)^2 + \left( \frac{\Delta p_{\text{tot}} \cdot \delta y}{L_{\text{ts}}} \right)^2 + \left( \frac{-\Delta p_{\text{tot}} \cdot y \cdot \delta L_{\text{ts}}}{L_{\text{ts}}^2} \right)^2}
\]

In this equation, the uncertainty of the pressure measurement at the inlet of the test section is the dominant influence, all other terms are relatively small compared with the first term.

Table 5.3: Maximal uncertainties of the measuring equipment

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Device</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>Thermocouple</td>
<td>±0.11 K</td>
</tr>
<tr>
<td>Pressure</td>
<td>Absolute</td>
<td>±10.9 kPa</td>
</tr>
<tr>
<td>Pressure differential</td>
<td>Differential</td>
<td>±0.4 kPa</td>
</tr>
<tr>
<td>Mass flow rate</td>
<td>Coriolis</td>
<td>±0.04 %</td>
</tr>
<tr>
<td>Heating power</td>
<td>Voltmeter</td>
<td>±0.2 %</td>
</tr>
<tr>
<td>test section</td>
<td>Ampèremeter</td>
<td>±0.1 %</td>
</tr>
<tr>
<td>Heating power preheater</td>
<td>Wattmeter</td>
<td>±1.5 %/–2.9 %</td>
</tr>
</tbody>
</table>
5.2.2 Fluid properties

As stated before, relations between fluid properties are used for the determination of the heat transfer coefficient: the saturation temperature is calculated from the saturation pressure. The fluid properties are evaluated using CoolProp \[61\]. The fluid properties of Solkatherm® SES36 were determined experimentally by Fröba et al. \[10\]. In this research, it is stated that the uncertainty of the \( p_{\text{sat}}(T_{\text{sat}}) \) relation is \( \delta p_{\text{rel}} = 2 \text{kPa} \). The total uncertainty of the saturation pressure is in this case:

\[
\delta p_{\text{tot}} = \sqrt{\delta p_{\text{sat},y}^2 + \delta p_{\text{rel}}^2}
\]  

In equation 5.3, \( \delta p_{\text{sat},y} \) is the error of the saturation pressure due to the uncertainty of the pressure transducers, as defined in section 5.2.1. This results in the following worst-case value of the error of the bulk temperature.

\[
\delta T_b = \max(\{T_b(p_{\text{sat}}) - T_b(p_{\text{sat}} \pm \delta p_{\text{tot}})\})
\]

The uncertainty of the specific enthalpy can be derived in a similar manner. The worst-case uncertainty is obtained by evaluating the four combinations of the uncertainties of the pressure and temperature, and taking the maximum absolute value. This is notated as follows:

\[
\delta h = \max(\{|h(p, T) - h(p \pm \delta p, T \pm \delta T)|\})
\]

The specific enthalpy is evaluated at the inlet of the preheater, to determine the vapor qualities in the test section (see equation 4.7). The uncertainty of the latent heat of vaporization and of the saturated liquid enthalpy are evaluated as follows:

\[
\delta h = \max(\{|h(p) - h(p \pm \delta p)|\})
\]

These uncertainties are required to calculate the uncertainty of the local vapor quality.

5.2.3 Mass velocity

The mass velocity is defined as the mass flow rate per unit of cross-sectional area, as can be seen in equation 4.8. By applying equation 5.2, the following expression is obtained for the uncertainty of the mass velocity:

\[
\delta G = \sqrt{\left(\frac{4 \cdot \delta \dot{m}}{\pi \cdot D_i^2}\right)^2 + \left(\frac{-8 \cdot \dot{m} \cdot \delta D_i}{\pi \cdot D_i^3}\right)^2}
\]
CHAPTER 5. ACCURACY OF MEASUREMENTS

5.2.4 Heat flux

The heat flux in the test section is applied by means of the Joule effect. The heating power of the test section is calculated with the following formula:

$$\dot{Q}_{ts} = U \cdot I$$  \hspace{1cm} (4.9 repeated)

By applying equation 5.2, the following relation for the error of the heating power is obtained:

$$\delta \dot{Q}_{ts} = \sqrt{(I \cdot \delta U)^2 + (U \cdot \delta I)^2}$$

The heat flux is defined as the heating power per surface unit:

$$\dot{q} = \frac{\dot{Q}_{ts}}{\pi \cdot D_i \cdot L_{ts}}$$  \hspace{1cm} (4.10 repeated)

The uncertainty of the heat flux is again obtained by applying equation 5.2:

$$\delta \dot{q} = \sqrt{\left(\frac{\delta \dot{Q}_{ts}}{\pi \cdot D_i \cdot L_{ts}}\right)^2 + \left(-\dot{Q}_{ts} \cdot \delta D_i \pi \cdot D^2_i \cdot L_{ts}\right)^2 + \left(-\dot{Q}_{ts} \cdot \delta L_{ts} \pi \cdot D^2_i \cdot L_{ts}\right)^2}$$

5.2.5 Volumetric heat flux

The value of the volumetric heat flux is required to calculate the inner wall temperature. Therefore, its uncertainty is calculated as well (see equation 4.11):

$$\delta \dot{q}'' = \sqrt{\left(\left(\frac{4 \cdot \delta \dot{Q}_{ts}}{L_{ts} \cdot \pi \cdot (D_o^2 - D_i^2)}\right)^2 + \left(-\frac{8 \cdot \dot{Q}_{ts} \cdot D_o \cdot \delta D_o}{L_{ts} \cdot \pi \cdot (D_o^2 - D_i^2)}\right)^2 + \left(-\frac{8 \cdot \dot{Q}_{ts} \cdot D_i \cdot \delta D_i}{L_{ts} \cdot \pi \cdot (D_o^2 - D_i^2)}\right)^2 + \left(-\frac{4 \cdot \dot{Q}_{ts} \cdot \delta L_{ts}}{L_{ts}^2 \cdot \pi \cdot (D_o^2 - D_i^2)}\right)^2}}$$

5.2.6 Inner wall temperature

The uncertainty of the inner wall temperature is dependent on four parameters, it is derived from equation 4.13:

$$\delta T_{w,i} = \sqrt{\frac{\delta^2}{16 \cdot \lambda_{ts}} \left(\frac{D_o^2 - D_i^2 - 2 \cdot D_o^2 \cdot \ln(D_o/D_i) \cdot \delta \dot{q}''}{4 \cdot \lambda_{ts}}\right)^2 + \left(\frac{\dot{q}'' \cdot D_o \cdot \ln(D_o/D_i) \cdot \delta D_o}{4 \cdot \lambda_{ts}}\right)^2 + \left(\frac{\dot{q}'' \cdot (D_o^2 - D_i^2) \cdot \delta D_i}{8 \cdot D_i \cdot \lambda_{ts}}\right)^2}$$

The uncertainty of the thermal conductivity of the test section was neglected, because even for relatively large deviations, the influence is very small. In the investigation of Canière [59], the thermal conductivity was changed from 300 W/(m K) to 400 W/(m K), this led to a change of 0.15% of the
local heat transfer coefficient $h_{tp}$. In reality, the uncertainty of the thermal conductivity is much smaller (than 100 W/(m K)), and its influence is thus completely negligible.

5.2.7 Heat transfer coefficient

The uncertainty of the local heat transfer coefficient is evaluated using the following expression:

$$
\delta h_{tp} = \sqrt{\left(\frac{\delta \dot{q}}{T_{w,i} - T_p}\right)^2 + \left(-\dot{q} \cdot \delta T_{w,i}\right)^2 + \left(\frac{\dot{q} \cdot \delta T_b}{T_{w,i} - T_b}\right)^2}
$$

(5.4)

5.2.8 Vapor quality

The uncertainty of the local vapor quality is evaluated using equation 5.5. In this equation, the latent heat of vaporization is written as $h_{ts,lg} = h_{ts,g}^{\text{sat}} - h_{ts,l}^{\text{sat}}$.

$$
\delta x_{ts,y} = \sqrt{\left(\frac{\delta h_{ph,in}}{h_{ts,lg}}\right)^2 + \left(\frac{\delta \dot{Q}_{ph}}{\dot{m} \cdot h_{ts,lg}}\right)^2 + \left(\frac{y \cdot \dot{Q}_{ts}}{L_{ts} \cdot \dot{m} \cdot h_{ts,lg}}\right)^2 + \left(\frac{\delta h_{ts,l,\text{sat}}}{h_{ts,lg}}\right)^2 + \left(\frac{h_{ph,in} + Q_{ph}^{l} \cdot \dot{m}}{L_{ts} \cdot \delta \dot{m} \cdot h_{ts,lg}}\right)^2 + \left(\frac{y \cdot \dot{Q}_{ts}}{L_{ts} \cdot \dot{m} \cdot h_{ts,lg}}\right)^2}
$$

(5.5)

5.3 Uncertainty estimation

As can be seen in equation 5.4, the knowledge of the wall temperatures is needed to obtain the uncertainty of the local heat transfer coefficients. This uncertainty can only be obtained after doing experiments. Therefore, a heat transfer correlation was used to get an idea of the uncertainty of the local heat transfer coefficients. The correlation of Kanlikar [24] was used. The values of the local heat transfer coefficient are obtained this way. Using equations 4.12 and 4.13, the inner and outer wall temperatures are calculated. With this knowledge, all of the uncertainties of the measured quantities may be evaluated. In Table 5.4, the maximal absolute and relative uncertainties of the calculated quantities are displayed. The relative uncertainty of the vapor quality is calculated with respect to a reference vapor quality of $x = 1$. These uncertainties were calculated for experimental conditions corresponding with the Design of Experiments, as will be discussed in Chapter 6.

The relative uncertainty of the heat flux is completely independent of the experimental condition, because it only depends on the heating device and the geometrical uncertainties. The uncertainty of the mass velocity is relatively independent of the experimental conditions. There is a very weak dependence on the mass flow rate, due to the Coriolis mass flow meter’s calibration, but this is not
discernible in Table 5.4. The uncertainties of the vapor quality and the local two-phase heat transfer coefficients vary largely with the experimental conditions. The absolute uncertainty of the local heat transfer coefficient increases with the mass velocity, the heat flux and the saturation pressure. The influences of the heat flux, the mass velocity and the saturation pressure on the relative uncertainty of the local heat transfer coefficient, $\frac{\delta h_{tp}}{h_{tp}}$, were investigated. This was done by varying every parameter separately over the experimental range, while the other factors were at a reference level. The result is illustrated in Figure 5.1. It can be seen that the relative uncertainty decreases with increasing heat flux, increases slightly with increasing mass velocity and is more or less independent of the saturation pressure. For the heat flux, $\delta h_{tp}$ increases with increasing heat flux, but $h_{tp}$ itself increases even more, so the relative uncertainty decreases. For the mass velocity, the absolute uncertainty of the local heat transfer coefficient increases more than the increase of the local heat transfer coefficient. This results in an increase of the relative uncertainty, as can be observed in Figure 5.1b. In fact, the increase of mass velocity only results in an increase of the local heat transfer coefficient. Because the heat flux is constant in this case, the inner wall temperature decreases (see equation 4.12). The absolute uncertainty of the local heat transfer coefficient also depends on this temperature, but with a higher power. Therefore, the absolute uncertainty increases more than the local heat transfer coefficient, and the relative uncertainty increases. For the saturation pressure, both the local heat transfer coefficient and its uncertainty are relatively constant, and thus the relative uncertainty is more or less constant.

It can be concluded that the lowest relative errors are obtained for high heat fluxes and low mass velocities. The highest relative uncertainty occurred for $G = 300 \text{ kg/(m}^2\text{s)}$ and $\dot{q} = 5 \text{ kW/m}^2$, with $\frac{\delta h_{tp}}{h_{tp}} \approx 23.3\%$. However, this result was obtained using the Kandlikar [24] correlation, which may not be valid in these conditions. Next to that, the local heat transfer coefficient depends highly on the occurring flow pattern. It is possible that the trends of the relative uncertainty vary depending on the flow pattern.

![Graphs showing influences of heat flux, mass velocity, and saturation pressure on relative uncertainty of local heat transfer coefficient.](image)

**Figure 5.1:** Influences of the heat flux, mass velocity and saturation pressure on the relative uncertainty of the local heat transfer coefficient, reference levels are $\dot{q} = 10 \text{ kW/m}^2$, $G = 200 \text{ kg/(m}^2\text{s)}$, $p_{sat} = 9.83 \text{ bar}$
Table 5.4: Maximal uncertainties of the calculated quantities

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Value</th>
<th>Absolute uncertainty</th>
<th>Relative uncertainty</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$G$</td>
<td>100</td>
<td>±0.2</td>
<td>±0.04 %</td>
<td>kg/(m$^2$ s)</td>
</tr>
<tr>
<td></td>
<td>300</td>
<td>±0.6</td>
<td>±0.04 %</td>
<td></td>
</tr>
<tr>
<td>$\dot{q}$</td>
<td>$5 \cdot 10^3$</td>
<td>±22.9</td>
<td>±0.02 %</td>
<td>kW/m$^2$</td>
</tr>
<tr>
<td></td>
<td>$20 \cdot 10^3$</td>
<td>±91.4</td>
<td>±0.02 %</td>
<td></td>
</tr>
<tr>
<td>$x$</td>
<td>0-1</td>
<td>±0.01</td>
<td>±1.1 %</td>
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</tr>
<tr>
<td>$p_{sat}$</td>
<td>7.93</td>
<td>±0.109</td>
<td>±1.4 %</td>
<td>bar</td>
</tr>
<tr>
<td></td>
<td>9.83</td>
<td>±0.109</td>
<td>±1.1 %</td>
<td></td>
</tr>
<tr>
<td>$h_{sp}$</td>
<td>1484</td>
<td>±286</td>
<td>±19.3 %</td>
<td>W/(mK)</td>
</tr>
<tr>
<td></td>
<td>4837</td>
<td>±663</td>
<td>±13.7 %</td>
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</table>
Chapter 6

Design of Experiments

According to Anderson and Whitcomb, “Design of experiments is a planned approach for determining cause and effect relationships” [67]. Experiments are used to analyze the behavior of processes and systems. In general, such a process or system can be thought of as a black box that transforms some input into an output. The result of the output is dependent on controllable process variables and other variables that are uncontrollable [68]. The general model of a system is given in Figure 6.1. For two-phase flow experiments, the input is the fluid that will be evaporated. The controllable variables are the mass flow rate, the saturation pressure and the heat flux. The outputs are the local heat transfer coefficients and the two-phase pressure drop. An uncontrollable variable could be the diameter of the tube of the test section, although the diameter is adjustable in this case. The Design of Experiments (DOE) determines the combinations and values of the controllable variables that will we used in the experiments.

Different existing DOE techniques are discussed in the following section, and one of them is used to design the experiments. Thereafter, the flow pattern map [21] and the heat transfer correlation [69] of Wojtan et al. are applied in these experimental points.

![Figure 6.1: General model of a system [68]](image-url)
6.1 Different DOE techniques

Usually, the goal of an experiment is to determine what the influence is of certain factors on the output of the system. In Figure 6.1, these factors were called the controllable variables. To observe the influence of these factors, experiments are conducted at different levels of these factors. An experiment that has \( k \) different factors, each observed at 2 levels, is called a \( 2^k \) experiment. More generally, each factor of an experiment may have a certain amount of levels. The main effect is the influence of the level of a certain factor on the output of the process. The interaction effect expresses how much the difference in output between levels of a factor is dependent on the level of other factors. Consider the following example. An experiment is conducted with two factors A and B, each having a high and a low level. The main effect of the factor A would express how much the output differs between the high and low value of A, while the two-factor interaction effect quantifies how much this difference is dependent on the level of B [68].

6.1.1 Best-guess approach

In this design technique, the choice of the levels of the factors for the next experiment is made, based on the outcome of the current experiment. Typically, only one or two factors are varied between two experiments. This technique works well in cases in which the experimenters have a good theoretical and practical understanding of the test facility. An important disadvantage of this technique is that some ranges of the experimental variables may not be evaluated at all. It is possible that only local optima were evaluated [70].

6.1.2 One-factor-at-a-time

In the one-factor-at-a-time method, also known as the sequential design, each factor has a reference level. One factor is varied over all its levels, while the other factors remain at their baseline. This allows straightforward interpretation of the main effects of each factor. However, it does not capture any interaction effects between factors, because each time only one factor is varied. This technique is always less efficient than other techniques, it requires more data for the same precision of the estimation of the main effects [70, 71].

6.1.3 Factorial design

In this method, the different factors are varied together, in order to capture interaction effects. All possible combinations of the different levels of the factors form the set of experimental points (this method is also called a full factorial experiment). This is an efficient way to make use of the data. If each factor has two levels, it is called a \( 2^k \) factorial experiment. The \( 2^k \) factorial design is frequently
used in the beginning of an experimental work, because the effects of all factors are obtained. Based upon this information, the choice can be made to fix some factors at a level in the following experiments, because they do not have an important influence on the output. With a \(2^k\) factorial design, the number of experimental points increases exponentially with the number of factors, and this can be a drawback because time and resources might be limited. From a \(2^k\) factorial experiment, \(k\) main effects, \(\binom{k}{2}\) two-factor interactions, \(\binom{k}{3}\) three-factor interactions, ..., and \(1\) \(k\)-factor interaction are obtained. This leads to a total of \(2^k-1\) effects. Remark that for moderate \(k\) values, already a large amount of the interaction effects is of a high order. For example, \(k=6\) or a \(2^6\) factorial design has 6 main effects, 15 two-factor interactions and 42 interactions of three or more factors. The use of \(3^k\) experiments or experiments with more levels (with only quantitative factors) is discouraged by Montgomery [68], because then the main effect has to be fitted to a quadratic relationship. In that case, it is more efficient to add center points to the \(2^k\) design [68]. Several techniques exist to design more experimental points, in order to increase the fitting to a model. The central composite design (CCD) is one of them. It adds axial and center runs to a \(2^k\) factorial design, this is demonstrated in Figure 6.2. Note that the extra points represent the new extreme values of the levels of each factor.

6.1.4 Fractional factorial design

A fractional factorial design is a factorial design in which some combinations of factors are not present. From the previous paragraph, it followed that a full factorial experiment contains many information about the higher order interaction effects. The sparsity of effects principle states that in most systems, main effects and low order interaction effects are dominant [67]. This principle was proven by two different authors for a large database of experiments [72, 73]. If it is expected that only main effects and low order interaction effects are present, then the fractional factorial method can be chosen to save time or resources. This is in particular advantageous when the experiment has a great number of factors. For example, the one-half fractional factorial design contains only half of the experimental points of the full factorial design and it is written as \(2^{k-1}\). In a fractional factorial
design, it is not possible to discern between high order effects. The order from which it is impossible to discern between interaction effects, is expressed by the resolution. In practice, resolutions between 3 and 5 are most popular.

### 6.1.5 Space filling designs

These designs are frequently used in computer experiments. The idea is to spread out the experimental points uniformly over the entire experimental range, because it is not known from before what regions will be of interest. Two techniques that use this principle, are the Latin hypercube design and the sphere-packing design. In the Latin hypercube design, \( n \) experimental points are designed. Each experimental point corresponds with exactly one level of each factor, and there are as many experimental points as levels of each factor \([74]\). In Figure 6.3, a Latin hypercube design is given for two factors \( x_1 \) and \( x_2 \), with 10 levels between \(-1\) and 1 each. The sphere-packing design realizes a uniform spread of the experimental points by maximizing the minimal distance between these points. In space filling designs, each level of a factor is only present in one experimental point, therefore it is not possible to draw conclusions about the main effects \([68]\).

### 6.1.6 Conclusion

Several techniques have different advantages that are important in this experimental investigation about two-phase flow heat transfer. The best-guess approach and one-factor-at-a-time designs are not considered, because of the mentioned drawbacks. With the factorial designs, the main and all interaction effects can be investigated efficiently. However, its design leads to a large number of experiments. In the fractional factorial designs, there are less experimental points, but the higher order effects cannot be derived from these experiments. This is not necessarily a drawback, because the sparsity of effects principle states that these effects are on average less important. In the space filling designs, experiments are conducted in the entire experimental range, it is however not possible to observe the main effects. It needs to be taken into account that the experimental data will be used to develop a heat transfer correlation. It is therefore important that there are as much experimental points as possible. This way, the correlation is fitted on a broader experimental range, and it is more generally applicable. For this reason, and because it is important to observe the main effects, the \( 3^k \) factorial design will be applied. From a statistical point of view, it is better to use a \( 2^k \) design with center points, but this will not be done, because the \( 3^k \) factorial design will distribute the experimental points more evenly over the experimental range. Next to that, the CCD design changes the extreme values of the factors. This is not wanted, because the experiments are limited to a certain range (because of the test facility and the application in ORCs).
6.2 Experimental points

In this section, the Design of Experiments will be done for the fluid Solkatherm® SES36. The controllable variables are the mass flow rate, the saturation pressure or temperature and the heat flux. These are limited by the pump, the design of the test facility and the heating power of the test section, respectively. Experiments will be done at saturation temperatures up to 120°C, because this corresponds to the conditions in an evaporator of an ORC used for waste heat recuperation. The maximum mass flow rate follows from the maximum volume flow rate of the pump and the density of the fluid. The maximum volume flow rate of the installed pump is about 330 L/h.

For Solkatherm® SES36 at 120°C, the maximum mass flow rate is about 0.12 kg/s. For a one inch tube, this results in a mass velocity of 345 kg/(m²s). The volume flow rate and consequently the mass velocity, decrease with the pressure increase in the pump. Therefore, the mass velocity will be limited to 300 kg/(m²s) in this DOE. The test section has a heating power of 12 kW, with a one inch tube and a test section length of 2.5 m, this corresponds to a maximal heat flux of 72 kW/m². The heat flux will however be limited to 20 kW/m², because this corresponds with the highest occurring heat fluxes in low-grade waste heat recovery [75, 76]. In Table 6.1, an overview is given of the levels of the mass velocity, the levels of the saturation pressure and the corresponding saturation temperature and the levels of the heat flux that will be used to construct the $3^3$ full factorial DOE. In every experimental point, the mass velocity, the heat flux and the geometrical parameters are fixed. From equation 6.1, it can be understood that this implies that the enthalpy increase in the test section is fixed.

$$\dot{Q}_{13} = \dot{q} \cdot \pi \cdot D_i \cdot L_{13} = \dot{m} \cdot \Delta h = G \cdot \pi \cdot \frac{D_i^2}{4} \Delta h$$ (6.1)
The enthalpy in the test section increases linearly with the vapor quality. Because the enthalpy increase is fixed, the change of vapor quality is also fixed. In combinations of design parameters with low heat fluxes and high mass velocities, the fluid is only evaporated partially. Therefore, the combinations are subdivided in two or more parts with different inlet vapor qualities. This way, the entire range of vapor qualities between \( x = 0 \) and \( x = 1 \) is covered. The result of this DOE is displayed in Table 6.2, where the last column describes in how much parts an experimental point is subdivided. For example, the first experiment is split up in five parts (\( n = 5 \)). This means that one experiment will start with an inlet vapor quality of \( x_i = 0 \), the next one with an inlet vapor quality of \( x_i = 0.2 \) and the last one with an inlet vapor quality of \( x_i = 0.8 \). This leads to a total of 163 experiments.

Unfortunately, the construction of the test facility was not finished in time and the designed experiments have not been conducted yet. Throughout this master thesis, various practical problems were encountered, and each of them was solved separately. During the first experiments, cavitation occurred in the pump. This was solved by adding the liquid receiver and subcooler, as explained in section 3.4. After these components were ordered, delivered and installed, it was tried again to control the test facility. But a problem with the proportional relief valve occurred. Apparently, it was wrongly connected. This was quickly solved, but thereafter a problem with the microfilter strainer arose. It was clogged with some dirt that was present in the system. It is believed that all problems will be solved completely in the near future, and then the experiments designed here will be done.

### 6.3 Flow pattern map and heat transfer

The version of the diabatic flow pattern map for of Kattan, Thome and Favrat [30], modified by Wojtan et al. [21] was implemented. To be sure that it was implemented correctly, the result was compared with the flow pattern maps shown in the investigation of Wojtan et al. [21]. The flow pattern map for Solkatherm® SES36 at \( T_{\text{sat}} = 120^\circ\text{C} \), and \( \dot{q} = 10 \text{kW/m}^2 \) is shown in Figure 6.4 for three different mass velocities. These conditions correspond to three experimental points of Table 6.2 (namely experiments 8, 17 and 26). Flow pattern maps for other conditions of the DOE can be found in Appendix A. It can generally be stated that the following flow patterns occur consecutively for \( G = 100 \text{kg/(m}^2\text{s}) \) and \( G = 200 \text{kg/(m}^2\text{s}) \): slug and stratified-wavy, stratified-wavy, and dryout. For \( G = 300 \text{kg/(m}^2\text{s}) \) the flow patterns are: slug, intermittent, annular, dryout, and mist.

<table>
<thead>
<tr>
<th>( G ) ( (\text{kg/(m}^2\text{s)}) )</th>
<th>( p_{\text{sat}} ) ( (\text{bar}) )</th>
<th>( T_{\text{sat}} ) ( (^\circ\text{C}) )</th>
<th>( \dot{q} ) ( (\text{kW/m}^2) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>7.93</td>
<td>110</td>
<td>5</td>
</tr>
<tr>
<td>200</td>
<td>8.84</td>
<td>115</td>
<td>20</td>
</tr>
<tr>
<td>300</td>
<td>9.83</td>
<td>120</td>
<td>30</td>
</tr>
</tbody>
</table>
Table 6.2: The DOE for Solkatherm® SES36

<table>
<thead>
<tr>
<th>experiment</th>
<th>( G ) (kg/(m(^2) s))</th>
<th>( p_{\text{sat}} ) (bar)</th>
<th>( \dot{q} ) (kW/m(^2))</th>
<th>n</th>
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<tbody>
<tr>
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<td>7.93</td>
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<td>8.84</td>
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<td>300</td>
<td>8.84</td>
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<td>9.83</td>
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<td>300</td>
<td>9.83</td>
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<td>7</td>
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<tr>
<td>27</td>
<td>300</td>
<td>9.83</td>
<td>20</td>
<td>4</td>
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</table>
The heat transfer model of Wojtan et al. [69] was implemented for the same conditions as the flow pattern map (again experiments 8, 17 and 26). The result is illustrated in Figure 6.5. For the highest mass velocity (blue curve), the local heat transfer coefficient first increases and then decreases strongly. The highest heat transfer coefficients occur for the annular flow pattern. Similar results where found by Bandarra Filho et al. [34], for the highest mass velocities, annular flow was present and the local heat transfer coefficient varied significantly with the vapor quality. The strong decrease of the heat transfer coefficient in this curve corresponds to dryout flow. The small nearly horizontal part of the curve at the highest vapor qualities, corresponds to mist flow. It can be seen that convective boiling is dominant at high vapor qualities, because there is a strong increase of heat transfer there. The three curves merge at low vapor qualities, this indicates that there is only an influence of nucleate boiling there. This behavior was also observed by Wang et al. [37]. In the curves of the lowest mass velocities, the transition from dominant nucleate boiling to dominant convective boiling can be distinguished clearly. This occurs at the vapor quality of \( x \approx 0.5 \), or when the flow pattern changes from slug and stratified-wavy flow to stratified-wavy flow. Due to the lower mass velocities, the influence of convective boiling is less pronounced and the peak at higher vapor qualities is lower for \( G = 200 \, \text{kg}/(\text{m}^2 \, \text{s}) \), or even not present for \( G = 100 \, \text{kg}/(\text{m}^2 \, \text{s}) \). It can also be seen that dryout occurs at lower vapor qualities for higher mass velocities, which was observed by Da Silva Lima et al. [33] as well. Bandarra Filho et al. [34] also found that at low mass velocities, stratified-wavy flow occurred and the heat transfer coefficient remained more or less constant, which is also the case in Figure 6.5 for \( G = 100 \, \text{kg}/(\text{m}^2 \, \text{s}) \). The influences of the heat flux and the saturation temperature, are similar to the ones mentioned and explained in section 2.3.4, and the corresponding graphs can be found in Appendix B. Increasing the heat flux leads to an increase of the heat transfer coefficient for low vapor qualities and low mass velocities. Increasing the saturation temperature, leads to an increase of the local heat transfer coefficient at low vapor qualities and low mass velocities. At high mass velocities and vapor qualities, the local heat transfer coefficient decreases with increasing saturation temperature (compare Figures B.3 and B.4).

It is concluded that the trends of the local heat transfer coefficients of Solkatherm\textsuperscript{®} SES36 in ORC conditions, predicted by the heat transfer correlation of Wojtan et al. [69], agree with the trends of experimental data available in literature. However, this is not surprising, since the correlation was developed for the same conditions as the experimental data it was compared with. Therefore, the correlation predicts similar trends as present in the experimental data, although the ORC conditions are distinctly different. As stated before (in Chapter 1), at higher saturation temperatures, the thermodynamic properties tend to shift, which has an influence on the flow patterns and the heat transfer [12]. Moreover, the diameter in this DOE is not within the limit of the heat transfer correlation of Kattan, Thome and Favrat [30] (see Table 2.1) and it is also significantly larger than in the considered experimental data (see Table 2.2). Consequently, it is unlikely that this correlation will be valid as such in these conditions.
Figure 6.4: Flow pattern map of SES36 at $T_{sat} = 120^\circ$C, $\dot{q} = 10$ kW/m$^2$ for $G = 100$ kg/(m$^2$s) (green), $G = 200$ kg/(m$^2$s) (red) and $G = 300$ kg/(m$^2$s) (blue), $S$=stratified, SW=stratified-wavy, I=intermittent, A=annular, D=dryout, M=mist flow
Figure 6.5: Local heat transfer coefficients of SES36 at $T_{sat} = 120^\circ$C, $\dot{q} = 10$ kW/m$^2$ for $G = 100$ kg/(m$^2$s), $G = 200$ kg/(m$^2$s) and $G = 300$ kg/(m$^2$s),
Chapter 7

Control of the test facility

In the test facility, some parameters are controlled to conduct experiments at specified conditions that were determined during the Design of Experiments in Chapter 6. These parameters are the mass velocity, the heat flux, the vapor quality and the saturation pressure. Some other parameters need to be controlled within certain bounds, to ensure safe functioning of the test facility. For example, it is important to control the subcooling before the pump, because this is a measure to provide some margin against cavitation in the pump. In this chapter, the strategy used to control these parameters is described.

7.1 Mass flow rate control

The mass flow rate is varied by changing the pump’s frequency (see section 3.1.1.8). The volume and mass flow rate through the pump increase with the rotation speed of the pump. However, the pump cannot be controlled in an open loop by adjusting its frequency. This is because the volume flow rate at a certain pump frequency is dependent on the pressure increase by the pump, as can be seen in the pump characteristic, Figure 3.4. If the volume flow rate through the pump is increased by increasing the pump’s frequency, then the pressure drop in the system increases as well. The pump’s operating point changes to a higher pressure, which diminishes the volume flow rate slightly (due to backflow in the pump). Next to that, the pressure drop in the test facility also depends on the vapor quality [59]. Therefore, there is no one-to-one relationship between the pump’s frequency and the mass flow rate. Consequently, the pump should be controlled in a closed loop.

The desired mass flow rate is calculated from the mass velocity and the test section’s diameter. The Coriolis mass flow meter measures the actual mass flow rate. The difference between the desired and the actual mass flow rate gives rise to an error. Then, a new pump frequency is calculated to minimize this error. This process is illustrated in Figure 7.1.
CHAPTER 7. CONTROL OF THE TEST FACILITY

7.2 Vapor quality control

7.2.1 Inlet vapor quality

The inlet vapor quality is controlled by changing the heating power of the preheater. This follows from equation 4.5, where a heat balance was inserted in the definition of the vapor quality:

\[ x_{ts,in} = \frac{h_{ph,in} + \frac{Q_{ph}}{m} - h_{ts,l,sat}}{h_{ts,g,sat} - h_{ts,l,sat}} \]  

(4.5 repeated)

The closed loop control of the inlet vapor quality is done by determining all of the components of equation 4.5 and evaluating the actual inlet vapor quality. This value is compared with the desired vapor quality, and the power of the preheater is adjusted to minimize the error. The control scheme is depicted in Figure 7.2. Using CoolProp, the enthalpy of the fluid at the inlet of the preheater is determined from the temperature and pressure measurements at that location. The mass flow rate is measured by the Coriolis mass flow meter, and \( h_{ts,l,sat} \) and \( h_{ts,g,sat} \) are also determined by CoolProp with the measurement of the pressure at the inlet of the test section.

7.2.2 Outlet vapor quality

The outlet vapor quality is controlled in a similar way as the inlet vapor quality, by changing the power of the test section heater. The expression for the outlet vapor quality is:

\[ x_{ts,out} = \frac{h_{ts,in} + \frac{Q_{ts}}{m} - h_{ts,l,sat}}{h_{ts,g,sat} - h_{ts,l,sat}} \]  

(4.6 repeated)
The saturated enthalpies are determined with the measurement of the differential pressure drop in the test section.

7.3 Saturation pressure control

The test facility can be considered a large two-phase reservoir. The saturation pressure can thus be controlled by heating up or cooling down the system. In preliminary tests, the system pressure could easily increased by heating up the experimental fluid with the preheater.

Once the pressure has reached its desired value, after starting up the test facility, the pressure in the system is kept constant by balancing the heating power of the preheater and the test section heater, with the cooling power of the condenser and the subcooler. Because there is some heat loss to the environment, it is expected that the heating power will need the be larger than the cooling power. The power of the preheater and the test section heater cannot be changed freely, because they determine the vapor quality in the test section. Thus, the condenser and subcooler are used to control the saturation pressure in the test section. However, the cooling power of the subcooler has a lower limit, determined by the subcooling control.

7.4 Subcooling control

As mentioned before (in Chapter 3), it is important that the liquid is subcooled a few degrees before it enters the pump. The subcooling is the difference between the local temperature and the satu-
tion temperature of the fluid at a certain pressure. In fact, thermal subcooling is a method to reach the required Net Positive Suction Head (NPSH) of the pump in a system. If the NPSH available is smaller than the requirements, cavitation may occur in the pump. This can damage the pump and reduce the mass flow rate. Thus, it should always be avoided. Depending on the saturation pressure, subcooling the fluid 1.2 K to 7 K should be sufficient to avoid cavitation in the pump (this was calculated in section 3.4.3). The scheme of the subcooling control is shown in Figure 7.3. The temperature of the subcooled fluid is measured with a thermocouple at the outlet of the subcooler. It is then compared with the desired temperature, which is determined from the saturation temperature and the degrees of subcooling wanted. The saturation temperature is measured at the inlet of the subcooler. The temperature at that location is close to the saturation temperature, because it is connected to the two-phase liquid receiver. Avoiding cavitation results in a minimum requirement of the cooling power of the subcooler. By controlling the bypass valve of the subcooler, its cooling power is then adapted accordingly.

### 7.5 Starting-up the test facility

The different control loops are installed in parallel in the LabVIEW environment. When the test facility is started, first the subcooling control loop is started, such that the pump can operate safely. Then the mass flow rate control loop starts operating. This needs to be done before the preheater starts, because a volume flow rate of 2 L/min is recommended for safe operation of the preheater. Thereafter, the preheater is used to pressurize the system by means of heating up the working fluid. Once the saturation pressure is close to its desired value, the saturation pressure control loop starts and it makes sure that the pressure is kept at that value. Finally, the inlet and outlet vapor quality control loops are started subsequently. It is recommended to wait for steady state each time before starting a new control loop, because the system is expected to be quite unstable and hard to control precisely. When all control loops are in steady state, or close to steady state, their set points are fixed.

![Figure 7.3: Scheme of the subcooling control](image-url)
(because they might be changing constantly). At the moment all measured quantities are constant, the experiments can be done.

### 7.6 Control strategy

There are two options for minimizing the error of the control loops. First, it could be tried to control the system by adapting the set points of the devices manually, and minimizing the errors this way. Secondly, a proportional-integral-derivative (PID) controller could be implemented for every control loop. This method is now explained in more detail.

A PID controller consists of three control actions. Based on the error between the set point and the measured value of a process variable, a new input is generated for the device or actuator that controls this quantity, this is illustrated in Figure 7.4. The output of the PID controller is linearly dependent on three terms:

- the error
- the integral of the error, or the past values of the error
- the derivative of the error, takes possible future trends of the error into account

The output of the controller is obtained by multiplying each factor with its corresponding gain: the proportional gain $K_p$, integral gain $K_i$ and derivative gain $K_d$, respectively, and summing these terms. An appropriate choice of the three parameters leads to a stable controller which acts quickly, with a small steady-state error, this is the difference between the set point and the process variable as time goes to infinity. The interested reader is referred to Control Systems Engineering [77], where a thorough explanation about PIDs is given. Suitable values of the three PID parameters can be retrieved mathematically. This requires the knowledge of the transfer function of the process, which expresses its dynamic behavior. Deriving the transfer functions analytically is not straightforward in this case. However, the transfer functions can also be determined experimentally, by giving a step input to the device. The step response is then measured and fitted on a first order system with time

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![Figure 7.4: Schematic of a PID controller](image-url)
delay, as can be seen in Figure 7.5. This is done by drawing the tangent to the step response in its inflection point. The intersection of this tangent with the time axis gives the dead time \( t_d \), and its slope is \( K/\tau \). From these quantities, the three PID parameters can be calculated with the Ziegler-Nichols [78] method. For example, this method could be used to determine the transfer function of the preheater. First, a step input is given to the power of the preheater. Then the enthalpy difference with respect to its initial value is monitored at the outlet of the preheater. Based on this observation, the PID parameters are calculated.

It is expected that the transfer function of the preheater is dependent on the mass flow rate. Therefore, it might be necessary that the PID parameters are determined experimentally for various mass flow rates. It is not needed to tune the PID manually, as explained. This can be done using the auto-tuning function, which is available in LabVIEW. This function also makes use of the Ziegler-Nichols [78] method.

It is concluded that implementing PID controls in the test facility is feasible with the Ziegler-Nichols [78] method. This approach will not result in the most effective controller, but it allows a quick and autonomous control of the test facility. This method is not the only option, other methods for determining transfer functions experimentally exist, such as the method of Åström-Hägglund [79].

![Figure 7.5: Step response (blue curve) and its first order system with time delay approximation (red curve) [80]](image)
Chapter 8

Conclusion

The goal of this thesis was to complete the construction of the two-phase flow boiling test facility, to gain hands-on insight into how to control this system and to perform experiments in ORC conditions.

Therefore, as a first step, a thorough literature study was done, to familiarize with the subject and to be informed about the latest developments in the experimental research. The heat transfer mechanism, the principle behind existing heat transfer correlations and the way different factors influence the two-phase heat transfer, were investigated. A review was done about other two-phase flow boiling investigations, and their main results and remarks.

The next step was familiarizing with the experimental flow boiling test facility and its measuring equipment. New components had to be selected, due to the problems that were encountered. To avoid cavitation in the pump, the liquid receiver and subcooler were dimensioned and installed in the test facility. A suitable electronic expansion valve was chosen, such that the condensing temperature can be controlled actively, and that boiling of the cooling fluid is avoided. The method for calculating all physical quantities that are important in this investigation, the data reduction, was developed and the heat loss in the test section was calculated. At a saturation temperature of 120 °C, it was estimated to be 45 W. The uncertainties of the measuring equipment were reviewed, and an uncertainty analysis was done to determine the uncertainty of every measured quantity. The uncertainties were estimated by making use of a heat transfer correlation. The maximal relative uncertainties are 23.3 % for the local two-phase heat transfer coefficient, 0.2 % for the heat flux, 1.4 % for the saturation pressure and 1.1 % for the vapor quality.

The different methods for designing experiments were evaluated, and the DOE was done for Solka-therm® SES36. For these experimental points, a flow pattern map and a heat transfer correlation were implemented. The flow pattern map predicts the occurrence of slug and stratified-wavy, stratified-wavy and dryout flow for low mass velocities, and slug, intermittent, annular, dryout and
mist flow for high mass velocities. The result of the heat transfer correlation agreed with experimental data available in literature.

The control of the test facility was elaborated. Specific control loops for the mass flow rate, the inlet and outlet vapor quality, the saturation pressure and the subcooling were proposed, and a methodology for starting the test facility was suggested. Finally, an approach for implementing a PID control of the test facility was put forward.

As mentioned earlier, experiments could not be conducted because the test facility was not finished in time. The next steps in this project are controlling the test facility, and conducting experiments. These experiments will be done in waste heat recovery ORC conditions: relatively low saturation temperatures and a large test section diameter. The experiments will be used to develop two-phase flow heat transfer and pressure drop correlations, which are applied when designing ORC evaporators. By making use of these correlations, the cost and efficiency of ORC systems will be improved.
Bibliography


Appendices
Appendix A

Flow pattern maps

Figure A.1: Flow pattern map of SES36 at $T_{sat} = 120^\circ$C, $\dot{q} = 5$ kW/m$^2$ for $G = 100$ kg/(m$^2$ s) (= green), $G = 200$ kg/(m$^2$ s) (= red) and $G = 300$ kg/(m$^2$ s) (= blue), S= stratified, SW= stratified-wavy, I= intermittent, A= annular, D= dryout, M= mist flow
Figure A.2: Flow pattern map of SES36 at $T_{sat} = 120^\circ$C, $\dot{q} = 20$ kW/m$^2$ for $G = 100$ kg/(m$^2$s) (= green), $G = 200$ kg/(m$^2$s) (= red) and $G = 300$ kg/(m$^2$s) (= blue), S=stratified, SW=stratified-wavy, I=intermittent, A=annular, D=dryout, M=mist flow

Figure A.3: Flow pattern map of SES36 at $T_{sat} = 110^\circ$C, $\dot{q} = 10$ kW/m$^2$ for $G = 100$ kg/(m$^2$s) (= green), $G = 200$ kg/(m$^2$s) (= red) and $G = 300$ kg/(m$^2$s) (= blue), S=stratified, SW=stratified-wavy, I=intermittent, A=annular, D=dryout, M=mist flow
APPENDIX A. FLOW PATTERN MAPS

Figure A.4: Flow pattern map of SES36 at $T_{sat} = 100^\circ C$, $q = 10\,\text{kW/m}^2$ for $G = 100\,\text{kg/(m}^2\text{s)}$ (green), $G = 200\,\text{kg/(m}^2\text{s)}$ (red) and $G = 300\,\text{kg/(m}^2\text{s)}$ (blue), S=Stratified, SW=Stratified-Wavy, I=Intermittent, A=Annular, D=Dryout, M=Mist Flow.
Appendix B

Local heat transfer coefficient graphs

Figure B.1: Local heat transfer coefficients of SES36 at $T_{sat} = 120^\circ C$, $\dot{q} = 5\text{ kW/m}^2$ for $G = 100\text{ kg/(m}^2\text{s)}$, $G = 200\text{ kg/(m}^2\text{s)}$ and $G = 300\text{ kg/(m}^2\text{s)}$. 
Figure B.2: Local heat transfer coefficients of SES36 at $T_{sat} = 120^\circ C$, $\dot{q} = 20$ kW/m$^2$ for $G = 100$ kg/(m$^2$s), $G = 200$ kg/(m$^2$s) and $G = 300$ kg/(m$^2$s).

Figure B.3: Local heat transfer coefficients of SES36 at $T_{sat} = 110^\circ C$, $\dot{q} = 10$ kW/m$^2$ for $G = 100$ kg/(m$^2$s), $G = 200$ kg/(m$^2$s) and $G = 300$ kg/(m$^2$s),
Figure B.4: Local heat transfer coefficients of SES36 at $T_{sat} = 100^\circ C$, $\dot{q} = 10 \text{ kW/m}^2$ for $G = 100 \text{ kg/(m}^2\text{s)}$, $G = 200 \text{ kg/(m}^2\text{s)}$ and $G = 300 \text{ kg/(m}^2\text{s)}$,