Experimental Investigation of evaporation of propane (R290) in small pipes

Supervisors:
Prof. Luigi Pietro Maria Colombo
Prof. Trygve Eikeyk
Prof. Armin Hafner

Master thesis by:
Ehsan Allymehr, 845320

Academic year 2016-2017
Acknowledgments

I would like to thank my family for their support for my studies.
Extended Abstract

Introduction

Recent awareness of the environmental protection and global warming have shed the light on the advantages of new refrigeration processes and adopting natural refrigerants which are more environmentally friendly (zero ODP and low GWP (Bolaji & Huan, 2013)) and compact heat exchangers to improve energy efficiency.

Propane has long been considered an alternative refrigerant, with a great potential for compact heat exchanger applications because of it has saturation curve favorable for many processes. Moreover, its physical properties are close to those of R-22 making it a suitable candidate for a charge switch without significant variation in the heat transfer performance. This combines with a modest environmental impact, as ODP is zero and GWP is smaller than 3 over 100 years; as terms of comparison, for R410A and R134a GWP is equal to 2100 and 1400 respectively. However, the major concern with the use of hydrocarbons as refrigerants is flammability, in particular, since very low ignition concentration is required for propane (Granryd, 2001), charge minimization is a major design objective for wide spread usage of hydrocarbons as refrigerants. Actually, the charge can be theoretically half compared to R134a ($\Delta h_{lv}$ of propane is almost double) Nonetheless, the heat transfer performance has to be carefully investigated, especially in the case of small diameter pipes, since the literature is relatively lacking general predicting approaches.

In this study, at first a thorough investigation of recent papers dealing with heat transfer and pressure drop characteristics of propane has been carried out (Choi, Pamitran, Oh, & Oh, 2007; Choi, Pamitran, Oh, & Saito, 2009; Wang, Gong, Chen, Sun, & Wu, 2014). In spite of relative abundance, the results on some points (such as the effect of vapor quality) seem to be conflicting and the available data do not cover the typical range of operating conditions for HVAC application and process industry.

This is the motivation of the work which includes the design, construction and testing of an experimental apparatus aimed at studying the evaporation in forced flow of propane. The range of operating conditions includes mass flux $G = 250 - 505 \frac{kg}{m^2\cdot s}$, heat flux of $q = 15 - 62 \frac{kw}{m^2}$ at saturation temperatures of 0, 5, 10 °C.
Finally a pressure drop and heat transfer coefficient have been measured and compared with the predictions of the available models such as (Sun & Mishima, 2008) for pressure drop and (Shah, 1982) for heat transfer coefficient.

**Experimental test rig**

To measure the pressure drop and heat transfer coefficient in different conditions an experimental test rig was used. Test rig was rebuilt based on an existing setup and modified to achieve the accuracy required and to improve the range of operating conditions.

The schematic diagram of the test rig is shown in Figure A. The set-up consists of two loops: The first one is the propane loop; the second one is a cooling loop. A data logger system is installed on the set-up and connected to a personal computer for control and data acquisition.

![Figure A: schematic of test rig](image)

Propane flows in a closed cycle, starting from the condenser and subcooler it is ensured that the liquid is subcooled for this is essential for the proper operation of the pump and the flow meter. The pump is controlled via frequency change using an inverter, the flow
meter (Coriolis type, RHEOINK, RHM04) is placed after the pump. A glass sight gauge placed just downstream of the flow meter enables checking whether single phase flow occurs. Then, the propane flows through the electrical preheater, in order to set the desired vapor quality, after an adiabatic section to ensure fully developed conditions, propane flows through the test section, where electrical heating provides the desired vapour quality variation. At this stage the measurements of the heat transfer coefficient and the pressure drop take place. At the outlet of the test section glass sight enables visualizing the flow regime. Finally, propane moves through the receiver and goes back to the condenser.

To calculate the uncertainty method of (Moffat, 1985) was used. Accuracies of the related sensors, are reported in Table A, additionally an error of 0.2% was considered for the interpolation of thermodynamic data. Furthermore, in order to account for the heat leakage 2% error has been included in the analysis.

<table>
<thead>
<tr>
<th>Displayed text</th>
<th>Manufacturer</th>
<th>Model</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow Meter</td>
<td>RHEOINK</td>
<td>RHM04</td>
<td>±0.1%</td>
</tr>
<tr>
<td>$P_1$</td>
<td>WIKA</td>
<td>S10</td>
<td>±0.5%</td>
</tr>
<tr>
<td>$P_2$</td>
<td>WIKA</td>
<td>DPT-10</td>
<td>±0.15%</td>
</tr>
<tr>
<td>$T_4$ to $T_7$</td>
<td>0.5mm Type T</td>
<td></td>
<td>±0.2°C</td>
</tr>
<tr>
<td>$T_1$ to $T_3$ and $T_8$</td>
<td>0.5mm Type T</td>
<td></td>
<td>±0.2°C</td>
</tr>
<tr>
<td>Data logger</td>
<td>National Instruments</td>
<td>NI 9211</td>
<td>-</td>
</tr>
<tr>
<td>$Q_{\text{preheater}}$</td>
<td></td>
<td></td>
<td>±0.5%</td>
</tr>
<tr>
<td>$Q_{\text{test section}}$</td>
<td></td>
<td></td>
<td>±0.5%</td>
</tr>
</tbody>
</table>

*Table A: List of instruments and uncertainties*

The percentage error related to the vapor quality in the whole range of operating conditions was found to lie within 2.0% and 2.3%, while for the heat transfer coefficient was about 4.7%. The uncertainty of pressure drop measurement is equal to the accuracy of the pressure gauge (0.5%).

**Results**

Figure B shows the effect of mass flux on heat transfer coefficient at saturation temperature of 10°C. The dryout quality seems weakly dependent on $G$ with a tendency to increase as the mass flux lowers. In the post dry-out region the drop in the HTC appears
independent of $G$. Furthermore, up to the dry-out occurrence the HTC increases with the mass flux at constant average quality. Additionally, the rise in the HTC with the average quality is more pronounced as $G$ increases, this is explained by the relative importance of nucleation and convection as mechanisms of energy transfer.

Figure B: Effect of mass flux on HTC.

Figure C shows the effect of heat flux on the heat transfer coefficient. HTC rises with the increase of the heat flux in the low vapor quality region. This can be explained by the low vapor quality region is boiling dominant hence the HTC is essentially related to the heat flux. Furthermore, at higher heat fluxes dryout is initiated at lower qualities. This can be explained by considering that the higher heat flux promotes the evaporation of the liquid layer adjoining the wall, increasing the thermal resistance.

Figure C Effect of Heat flux on HTC ($T_{sat} = 10^\circ C$)
Results show that for increasing saturation temperatures the HTC increases, but very often this difference lies within the uncertainty of the HTC measurement, eventually, a clear effect of the saturation temperature has not been put in evidence.

Pressure drop and the effect of mass flux on it is shown in figure D. As expected the results show a strong dependency of pressure drop on mass flux. Total pressure drop is a combination of accelerational and frictional pressure drop.

Figure D: Effect of mass flux on pressure drop ($T_{sat} = 5^\circ C$)

Figure E shows that with increasing heat flux the pressure drop increases for all the range of vapor qualities. The increase of the pressure drop across test section can be explained by the increased vaporization, causing a higher void fraction and a higher gas velocity. Higher vaporization can also change create bubbles leading to a slug flow further increasing the pressure drop.

Figure E: Effect of heat flux on pressure drop ($T_{sat} = 5^\circ C$)
Figure F shows the effect of saturation temperature on pressure drop, there is a significant increase in the pressure drop with lowering temperatures. The pressure drop across test section for saturation temperature of 0°C is about 30% greater compared to the case at saturation temperature of 10°C. This can be explained by the decrease in the viscosity of the gas phase $\mu_g$, while liquid density $\rho_l$ and liquid viscosity $\mu_l$ increase; this causes in turn an increase in the superficial velocity of the gas phase and a higher shear stress which justifies the larger pressure drop.

![Figure F: Effect of saturation Temperature on pressure drop](image)

HTC and pressure drop prediction correlations

For prediction of pressure drop correlations of (Muller-Steinhagen & Heck, 1986), (Sun & Mishima, 2008), (Cavallini et al., 2002), (Tran, Chyu, Wambganss, & France, 2000), (Pamitran et al., 2010) were chosen.

To make the two data sets comparable, the accelerational pressure drop is subtracted from the total pressure drop since the correlations give the estimates of the frictional pressure drop. Obviously since that the tube is horizontal the gravitational pressure drop is equal to zero.

To measure the accuracy of correlations, three different statistical parameters were considered, MARD (Mean Absolute Relative Deviation) MRD (Mean Relative Deviation), and $\lambda$30%. They are defined as:

$$MRD = \frac{\sum (E - e)}{\bar{E}} \times 100$$
Extended Abstract

\[
MARD = \frac{\sum|(E - e)|}{E} \cdot \frac{n}{100}
\]

\[
\lambda_{30\%} = \frac{\text{number of predicted values with MRD < 30\%}}{\text{total number of predicted values}} \cdot 100
\]

Where E is the experimental results considering zero measurement error and e is the predicted value.

Table A shows the prediction performance for correlations of (Muller-Steinhagen & Heck, 1986), (Sun & Mishima, 2008) and (Cavallini et al., 2002). Results of the correlations of (Tran et al., 2000), (Pamitran et al., 2010) is not shown since the value of MARD exceeded 100%.

<table>
<thead>
<tr>
<th>MRD (%)</th>
<th>MARD (%)</th>
<th>(\lambda_{30%})</th>
</tr>
</thead>
<tbody>
<tr>
<td>(Muller-Steinhagen &amp; Heck, 1986)</td>
<td>32.4</td>
<td>36.4</td>
</tr>
<tr>
<td>(Sun &amp; Mishima, 2008)</td>
<td>5.6</td>
<td>28.7</td>
</tr>
<tr>
<td>(Cavallini et al., 2002)</td>
<td>56.7</td>
<td>61.3</td>
</tr>
</tbody>
</table>

Table A: Overview of the predicted pressure drops errors

The correlation by (Sun & Mishima, 2008) is the most reliable able to predict about 84% of data points within 30% relative error. In any case all correlations tend to over predict the pressure drop as all MRD values are larger than zero.

For prediction of heat transfer coefficient correlations of (Choi et al., 2007), (Liu & Winterton, 1991), (S. G. Kandlikar, 1990), (Tran et al., 1996), (Gungor & Winterton, 1986) and (Shah, 1982) were chosen.

(Shah, 1982) is the correlation with least MARD being at 100%. All the correlation tend to greatly overpredict the heat transfer coefficient. Reasons for this large error can be:

1- In the data base used to develop the correlations propane was not included though the effect of a different fluid is considered in some equations by the reduced pressure and molar mass.

2- The effect of the small diameter, included by using either the Weber or the Laplace number is not always accounted for.
3- There is a general difficulty with the prediction of heat transfer coefficient by mechanistic models. A combination of a new fluid and a smaller tube diameter can make a correlation that was not explicitly developed for these conditions to be completely unsuitable.

Conclusion

Two-phase pressure drop and heat transfer characteristics during flow boiling of propane in different experimental conditions were discussed in this study. Test conditions were \( G = 250 - 505 \frac{kg}{m^2s} \), \( T_{\text{sat}} = 0, 5, 10 \degree C, q'' = 15 - 62 \frac{kw}{m^2} \) corresponding to vapor quality variation \( \Delta x = 0.1 - 0.4 \). A smooth copper test tubes with \( D_t = 4.2 \ mm \) was used, test section length was equal to 0.5 m. Mean vapor quality between 0.2 and 1.0. Several existing pressure drop and heat transfer coefficient prediction methods were reviewed. From the analysis of the experimental data and the comparison with the literature findings the following conclusions can be drawn:

1- The heat transfer coefficient increases with the vapor quality till the dryout onset, occurring at average quality higher than 0.7 which increases as the mass flux is decrease at constant heat flux, highest observed heat transfer coefficient was 4.25 \( \frac{kw}{m^2k} \).

2- An increase in heat flux and mass flux causes an increase of the heat transfer coefficient. The influence of the heat flux is more prominent at low vapor quality where the heat transfer regime is boiling dominated while the effect of the mass flux is more significant at high vapor quality where the heat transfer regime is mainly convection dominated. Saturation temperature does not seem to have a tangible effect on the heat transfer coefficient.

3- Pressure drop increases with vapor quality, but after the dryout, the trend is reversed. Highest recorder pressure drop was around 67 \( \frac{kPa}{m} \) for a mass flux of 505 \( \frac{kg}{m^2s} \).

4- Both mass flux and heat flux increase the pressure drop in the whole range of tested vapor quality. However, as expected the effect of heat flux is relatively minor compared to the influence of mass flux.
4- Saturation temperature has a tangible effect on the pressure drop. The decrease of saturation temperature causes an increase in the pressure drop, explained by the increase of the vapor velocity.

5- The correlation of (Sun & Mishima, 2008) provides the most reliable prediction of the pressure drop, with a MRD 5.69% and is able to predict 83.96% of data points with less than 30% error. All the correlations tend to overpredict the pressure drop. Furthermore, none of them is able to account for the effect of saturation temperature in a meaningful way.

6- None of the correlations used for prediction of the heat transfer coefficient is reliable, all of them greatly overestimate the heat transfer coefficient.
4.3.1. Controlling mass flux .............................................................. 35
4.3.2. Controlling inlet vapor fraction ............................................. 36
4.3.3. Controlling saturation pressure (temperature) ......................... 36
4.3.4. Controlling heat flux in the test section ................................ 36
4.4. MEASUREMENT AND DATA LOGGING SYSTEM ......................... 36
4.4.1. Data logging and control system ......................................... 37
4.4.2. Temperature measurement ................................................. 39
4.4.3. Pressure measurement ..................................................... 40
4.5. DATA REDUCTION AND EXPERIMENTAL UNCERTAINTY .......... 40
4.5.1. Thermodynamic data ....................................................... 40
4.5.2. Pressure drop data reduction ............................................. 40
4.5.3. Heat transfer coefficient (HTC) data reduction ....................... 41
4.5.4. Methodology Error ....................................................... 42
4.6. VALIDATION ........................................................................ 42
5. RESULTS AND DISCUSSION ON HEAT TRANSFER COEFFICIENT 45
5.1. GENERAL CHARACTERISTICS OF HEAT TRANSFER COEFFICIENT 45
5.2. EFFECT OF SATURATION TEMPERATURE ON HEAT TRANSFER COEFFICIENT 47
5.3. EFFECT OF MASS FLUX ON HEAT TRANSFER COEFFICIENT .......... 48
5.4. EFFECT OF HEAT FLUX ON HEAT TRANSFER COEFFICIENT .... 50
6. RESULTS AND DISCUSSION ON PRESSURE DROP .......................... 51
6.1. GENERAL CHARACTERISTICS OF PRESSURE DROP OF PROPANE 51
6.2. EFFECT OF SATURATION TEMPERATURE ON PRESSURE DROP 52
6.3. EFFECT OF MASS FLUX ON PRESSURE DROP .......................... 53
6.4. EFFECT OF HEAT FLUX ON PRESSURE DROP .......................... 54
7. COMPARISON OF CORRELATIONS WITH TEST DATA ...................... 57
7.1. CORRELATIONS FOR FRICTIONAL PRESSURE DROP ................... 57
7.2. CORRELATIONS FOR HEAT TRANSFER COEFFICIENT ............... 61
8. CONCLUSIONS ....................................................................... 63
9. PROPOSAL FOR FURTHER WORK ............................................... 65

LIST OF TABLES ........................................................................ 66
10. LIST OF FIGURES ................................................................... 67
11. NOMENCLATURE .................................................................... 69
12. REFERENCES

71
Abstract

Characteristics of a two-phase evaporating flow of propane flowing in a small pipe utilizing an experimental test is presented. The effect of heat flux, mass flux and saturation temperature were studied on the heat transfer and pressure drop were studied. Test conditions were $G = 250 - 505 \frac{kg}{m^2s}$, $T_{sat} = 0,5, 10 \ ^\circ C$, $q'' = 15 - 62 \frac{kw}{m^2}$ corresponding to vapor quality variation $\Delta x = 0.1 - 0.4$. A smooth copper test tube with $D_t = 4.2 \ mm$ was used, test section length was equal to 0.5 m. Literature review was performed, and tests were categorized based on the inlet diameter of tube, tube material and surface finish, saturation temperature, mass flux, heat flux and the heating medium provided. Several existing pressures drop and heat transfer coefficient correlations were reviewed and compared to experimental results. Experimental results show an increase of heat transfer coefficient with mass flux, heat flux and vapor quality before dryout. Pressure drop increases with heat flux and mass flux, but it decreases with saturation temperature.

Keywords: Propane, R290, boiling heat transfer, pressure drop, microfinned tubes.
1. Introduction

Recent awareness of the environmental protection and global warming have shed the light on the advantages of new refrigeration processes and adopting natural refrigerants which are more environmentally friendly (zero ODP and low GWP (Bolaji & Huan, 2013)) and compact heat exchangers to improve energy efficiency.

Propane has long been considered an alternative refrigerant, with a great potential for compact heat exchanger applications because of its favorable saturation curve for many processes. Moreover, its physical properties are close to those of R-22 making it a suitable candidate for a charge switch without significant variation in the heat transfer performance. This with a modest environmental impact, as ODP is zero and GWP is smaller than 3 over 100 years; as terms of comparison, for R410A and R134a GWP is equal to 2100 and 1400 respectively. However, the major concern with the use of hydrocarbons as refrigerants is flammability, in particular, since very low ignition concentration is required for propane (Granryd, 2001), charge minimization is a major design objective for wide spread usage of hydrocarbons as refrigerants. Actually the charge can be theoretically half compared to R134a ($\Delta h_{tg}$ of propane is almost double) Nonetheless, the heat transfer performance has to be carefully investigated, especially in the case of small diameter pipes, since the literature is relatively lacking general predicting approaches.

In this study, at first a thorough investigation of recent papers dealing with heat transfer and pressure drop characteristics of propane has been carried out (Choi, Pamitran, Oh, & Oh, 2007; Choi, Pamitran, Oh, & Saito, 2009; Wang, Gong, Chen, Sun, & Wu, 2014). In spite of relative abundance, the results on some points (such as the effect of vapor quality) seem to be conflicting and the available data do not cover the typical range of operating conditions for HVAC application and process industry.

This is the motivation of the work which includes the design, construction and testing of an experimental apparatus aimed at studying the evaporation in forced flow of propane. The range of operating conditions includes mass flux $G = 250 - 505 \frac{kg}{m^2\cdot s}$, heat flux of $q = 15 - 62 \frac{kw}{m^2}$ at saturation temperatures of 0, 5, 10 °C.
Finally a pressure drop and heat transfer coefficient have been measured and compared with the predictions of the available models such as (Sun & Mishima, 2008) for pressure drop and (Shah, 1982) for heat transfer coefficient.
2. Review of existing modeling approaches

In this section, a general review of formulations for pressure drop and heat transfer coefficient is presented.

2.1. Fundamental Quantities

Due to the conflicting nomenclature used in different studies an overview of the fundamental quantities and their definition is presented.

Mass flux is defined as:

\[ G = \frac{\Gamma}{\Omega} \]

*Equation 2-1*

Where \( \Gamma \) is the mass flow rate \( [\frac{kg}{s}] \) and \( \Omega \) is the pipe cross-section area \( [\frac{1}{m^2}] \).

Apparent mass flux is considering a single phase either gas or liquid flowing alone with its mass flux and can be defined as:

\[ G_g^* = \frac{\Gamma_g}{\Omega} \quad \text{or} \quad G_l^* = \frac{\Gamma_l}{\Omega} \]

*Equation 2-2*

Superficial velocity or volume flux is defined as:

\[ J = \frac{Q}{\Omega} \]

*Equation 2-3*

Where \( Q \) is the volume flow rate \( [\frac{m^3}{s}] \).

A cross section average void fraction is defined is the surface area of flow which is occupied by the gas phase compared to the total flow area, it can be mathematically described as:

\[ \bar{\alpha} = \frac{\int a d\Omega}{\Omega} = \frac{\Omega_g}{\Omega} \]

*Equation 2-4*

Where \( \Omega_g \) is the cross-section area occupied by the gas phase.
2.2. Heat Transfer coefficient (HTC)

Either electrical heating or a chilled liquid (usually glycol water mixture) can be used for providing the heat required for the fluid to evaporate (or vice versa for condensation). For electrical heating, it can be reasoned that:

\[ h = \frac{Q}{s \times (T_w - T_{\text{sat}})} \]

Equation 2-5

Where \( Q \) is the total heat input and \( s \) is the heat transfer surface, \( T_w \) is the temperature at the wall.

Where \( K_w \) is the thermal conductivity of the wall material. It should be noted that due to small diameter in minichannels the difference between outside and inside temperature of the tube is negligible therefore in some papers this calculation has been omitted.

For the second case scenario (using a chilled fluid) usually the heat is provided in a double pipe with the refrigerant flowing in the inside tube while the chilled fluid is flowing in the outer tube, in this case the heat transfer coefficient is calculated as:

\[ h = \frac{Q}{\Delta T_{\text{lm}} \times s} \]

Equation 2-6

Where \( \Delta T_{\text{lm}} \) is the logarithmic mean temperature difference. As for the \( Q \) in this case:

\[ Q = [V_{\text{aux}} \rho_{\text{aux}} c_{p,\text{aux}} (T_{\text{aux, out}} - T_{\text{aux, in}})] \]

Equation 2-7

Where \( V_{\text{aux}} \) is the volume flow rate of auxiliary fluid providing the heat.

As vapor quality can affect the HTC and pressure drop characteristics it is important to know the vapor quality entering the test section. Vapor quality can be controlled through two different methods, usually heat is provided to the subcooled fluid in a pre-test section and using the energy balance equation vapor quality is:
Chapter 3: Literature review

\[ x = \frac{Q_{pre} - C_p * m(T_{in} - T_{sub})}{m * i_{lg}} \]

Equation 2-8

Where \( T_{sub} \) is the temperature in which the subcooled liquid enters the preheater.

Another method to control the vapor quality is having a heater chamber and draining liquid from the bottom while the vapor is extracted from above, using valve and mass flow meters the mass quality can be obtained. As in the case of equilibrium the mass quality can be considered the same as thermal quality.

2.3. Pressure drop

Since most of the modelling approaches are based on the separated flow model in the following reference will be made to such an approach. The reader will be warned in case of a change.

Basically in the separated flow model (SFM) the phases are considered as two separate one-dimensional streams with different velocities. The ratio of the gas (vapor) to liquid velocity is called the slip ratio. No attempt is made to describe the flow patterns, which is the main drawback of this approach, though it generally provides acceptable predictions in a wide range of operating conditions.

Two phase pressure gradient consists of three different parts including frictional, gravitational and accelerative components.

\[
\left( \frac{dP}{dZ} \right)_{TP} = \left( \frac{dP}{dZ} \right)_{fri} + \left( \frac{dP}{dZ} \right)_{grav} + \left( \frac{dP}{dZ} \right)_{acc}
\]

Equation 2-9

Accordingly, the gravitational pressure gradient reads:

\[
\left( \frac{dP}{dZ} \right)_{grav} = \frac{g * \sin(\theta)}{x} * \int_{0}^{x_o} [\bar{\alpha} \rho_g + (1 - \bar{\alpha}) \rho_l]
\]

Equation 2-10

Where \( \bar{\alpha} \) is the void fraction, expressed by a suitable correlation (see section 2.4.1). Gravitational pressure gradient is obviously zero in case of horizontal tubes.

The accelerative pressure gradient is:
Chapter 2: Review of existing models

\[
\left( \frac{dP}{dZ} \right)_{acc} = G^2 \left[ \frac{x^2 \cdot v_g}{\alpha} + \frac{(1 - x)^2 \cdot v_l}{1 - \alpha} - v_l \right] \quad \text{Equation 2-11}
\]

As far as the frictional pressure gradient is concerned, the formulation usually involves the use of a two-phase multiplier as specified in the dedicated section (see section 2.4.3)

2.4. Correlations

In this part of this study a review of the most used correlations for void fraction, two-phase pressure drop and heat transfer coefficients are presented.

2.4.1. Void fraction

Most of the available correlations are based on:

1. Separated Flow model (Slip ratio correlations)
2. Drift-Flux model (Distribution parameter and drift velocity)
3. General approaches based on dimensional analysis

In this work and generally for HVAC applications the mass flux of the refrigerant is less than 1000 \((\text{kg} \ (\text{m}^2 \ \text{s})^{-1})\) where the second approach gives the best results. According to (Woldesemayat & Ghajar, 2007) the following three models reported in Table 1 should be preferred.

<table>
<thead>
<tr>
<th>Author</th>
<th>void fraction correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>(Rouhani &amp; Axelsson, 1970)</td>
<td>( \frac{x}{\rho_g} \left{ 1.1 \left[ \frac{x}{\rho_g} + \frac{1 - x}{\rho_l} \right] + \frac{1.18}{G} \left[ \frac{\sigma \cdot g \cdot (\rho_g - \rho_l)}{\rho_l^2} \right] \right}^{-1} )</td>
</tr>
<tr>
<td>Toshiba (Coddington &amp; Macian, 2002)</td>
<td>( \frac{J_g}{1.08 \cdot J + 0.45} )</td>
</tr>
<tr>
<td>(Woldesemayat &amp; Ghajar, 2007)</td>
<td>( J_g \cdot \left[ 1 + \left( \frac{J_g}{J_g^0} \right)^{0.61} \right] + 2.9 \left[ g \cdot D \cdot \sigma \cdot (1 + \cos \theta) \cdot (\rho_l - \rho_g) \right]^{0.25} \cdot (1.22 + 2.22 \cdot \sin \theta) \cdot \frac{p_{atm}}{p_{syste}} )</td>
</tr>
</tbody>
</table>

\( J_g^0 \) is the critical mass flux for condensation

\( \sigma \) : interfacial tension

\( \rho \) : density

\( g \) : gravitational acceleration

\( D \) : pipe diameter

\( \theta \) : inclination angle

\( p \) : pressure

\( \rho_f \) : density of the refrigerant

\( \rho_l \) : density of the liquid

\( \rho_g \) : density of the gas

\( \rho_{atm} \) : atmospheric pressure

<table>
<thead>
<tr>
<th>Author</th>
<th>void fraction correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>(Rouhani &amp; Axelsson, 1970)</td>
<td>( \frac{x}{\rho_g} \left{ 1.1 \left[ \frac{x}{\rho_g} + \frac{1 - x}{\rho_l} \right] + \frac{1.18}{G} \left[ \frac{\sigma \cdot g \cdot (\rho_g - \rho_l)}{\rho_l^2} \right] \right}^{-1} )</td>
</tr>
<tr>
<td>Toshiba (Coddington &amp; Macian, 2002)</td>
<td>( \frac{J_g}{1.08 \cdot J + 0.45} )</td>
</tr>
<tr>
<td>(Woldesemayat &amp; Ghajar, 2007)</td>
<td>( J_g \cdot \left[ 1 + \left( \frac{J_g}{J_g^0} \right)^{0.61} \right] + 2.9 \left[ g \cdot D \cdot \sigma \cdot (1 + \cos \theta) \cdot (\rho_l - \rho_g) \right]^{0.25} \cdot (1.22 + 2.22 \cdot \sin \theta) \cdot \frac{p_{atm}}{p_{syste}} )</td>
</tr>
</tbody>
</table>

Table 1 Void fraction correlations

1. Correlation of (Rouhani & Axelsson, 1970) is very frequently adopted in the refrigerants in horizontal pipes.
2. Toshiba correlation which is very simple but provides one of the best predictions.
3. (Woldesemayat & Ghajar, 2007) if inclined pipes need to be considered.

2.4.2. Frictional pressure drop

(Xu, Fang, Su, Zhou, & Chen, 2012) have provided an extensive review of correlations for the frictional pressure drop developed in the last sixty years. For the goals of this work the most suitable models are briefly discussed in the following.

Pressure drop correlations usually make use of two-phase multiplier defined as follows:

\[
\phi_{lo}^2 = \frac{\left(\frac{dP}{dZ}\right)^{fric}}{\left(\frac{dP}{dZ}\right)_{lo}}
\]

\(\text{Equation 2-12}\)

Where the denominator represents the liquid only situation, meaning each phase flows alone in the pipe with total mass flux. (this can also be defined for gas only situation \(\phi_{go}^2\)). Therefore, the frictional pressure drop can be shown as:

\[
\left(\frac{dP}{dZ}\right)^{fric} = \frac{2f_{lo} G^2 v_l}{D} * \phi_{lo}^2
\]

\(\text{Equation 2-13}\)

We can say:

\[
\left(\frac{dP}{dZ}\right)^{fric} = \left(\frac{dP}{dZ}\right)_{lo} * \phi_{lo}^2
\]

\(\text{Equation 2-14}\)

Some correlations use the definition only liquid or only gas flow (\(\phi_{l}^2, \phi_{g}^2\)), in these cases meaning that the flow is liquid with its respective apparent mass flux, this can be defined as:

\[
\left(\frac{dP}{dZ}\right)^{fric} = \frac{2f_{l} G^2 v_l}{D} * \phi_{l}^2
\]

\(\text{Equation 2-15}\)

Such multipliers are frequently correlated to the Lockhart Martinelli parameter defined as:

\[
X_{lt}^2 = \frac{\left(\frac{dP}{dZ}\right)^{fric}_l}{\left(\frac{dP}{dZ}\right)^{fric}_g}
\]

\(\text{Equation 2-16}\)
And tt refers to flow regime, tt meaning both phases are turbulent. Therefore, these relations should be applied in the case where both phases are turbulent.

Or similar ones like $Y$ defined as:

$$
Y \triangleq \frac{(dP/dZ)_{g0}^{fric}}{(dP/dZ)_{l0}^{fric}} = \frac{f_{g0}}{f_{l0}} \cdot \frac{v_{g}}{v_{l}}
$$

Equation 2-17

To measure the accuracy of correlations, three different statistical parameters were considered, MARD (Mean Absolute Relative Deviation) MRD (Mean Relative Deviation), and $\lambda 30\%$. They are defined as:

$$
MARD = \frac{\Sigma (E - e)}{n} \cdot 100
$$

Equation 2-18

$$
MRD = \frac{\Sigma |(E - e)|}{n} \cdot 100
$$

Equation 2-19

$$
\lambda 30\% = \frac{\text{number of predicted values with } MRD < 30\%}{\text{total number of predicted values}} \cdot 100
$$

Equation 2-20

Where E is the experimental results considering zero measurement error and e is the predicted value.

2.4.2.1. Muller-Steinhagen & Heck

(Muller-Steinhagen & Heck, 1986) is reported as:

$$
\varphi_{l0}^2 = Y^2 x^3 + (1 - x)^{\frac{1}{3}} \cdot [1 + 2x(Y^2 - 1)]
$$

Equation 2-21

This correlation results the best predictor for evaporation with MARD of 25.9% and MRD is -8.8% meaning that the model slightly underestimated the pressure drop (Xu et al., 2012).

2.4.2.2. (Sun & Mishima, 2008)
For viscous flow ($Re_t < 2000$):

$$C = 26 \left(1 + \frac{Re_t}{1000}\right) \left[1 - \exp \left(-\frac{0.153}{0.8 + 0.27La}\right)\right]$$

For Turbulent flow:

$$C = 1.79 \left(\frac{Re_g}{Re_t}\right)^{0.4} \frac{1}{x}$$

$$\phi_i^2 = 1 + \frac{c}{X^{1.19}} + \frac{1}{X^2}$$

This correlation is worth noting because it includes the Laplace number:

$$La = \sqrt{\frac{\sigma}{g(\rho_t - \rho_g)D_t}}$$

Which accounts for surface tension, actually it is recognized (Satish G Kandlikar, 2002) that confinement significantly affects the performance of mini tubes. The reported values for MRD and MARD are -26.5% and 31.2% respectively (Xu et al., 2012).

On the other hand, at mass fluxes such that gravitational effects are negligible the Weber number, which compares inertia and surface tension, should be more appropriate to describe confinement is by Webber number. For more information regarding the effect of confinement see section 2.4.4.

Weber number is defined as:

$$We = \frac{G^2D}{\sigma \rho}$$

### 2.4.2.3. (Cavallini et al., 2002):

$$\phi_i^2 = (1 - x)^2 + x^2 * \frac{P_l f_{go}}{P_g f_{lo}} + \frac{1.262x^{0.6978}h}{We_{go}^{0.1458}}$$

Where:
Chapter 2: Review of existing models

\[ H = \left( \frac{\rho_l}{\rho_g} \right)^{0.3278} \cdot \left( \frac{\mu_g}{\mu_l} \right)^{-1.181} \cdot \left( 1 - \frac{\mu_g}{\mu_l} \right)^{3.477} \]

\[ W_{eg} = \frac{G_{tp}^2 D}{\rho_g \sigma} \]

The reported values for MRD and MARD are 13.9% and 35.3% respectively (Xu et al., 2012).

2.4.2.4. (Pamitran, Choi, Oh, & Hrnjak, 2010):

\[ \phi_i^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \]

Where:

\[ C = 3 \times 10^{-3} \times W e_{tp}^{-0.433} \times R e_{tp}^{1.23} \]

\[ W e_{tp} = \frac{G_{tp}^2 D}{\rho_{tp} \sigma} \]

\[ R e_{tp} = \frac{G_{tp} D}{\mu_{tp}} \]

Equation 2-26

In particular, (Pamitran et al., 2010) correlation is also based on data using propane as in the present study. The reported value of MARD for this correlation is higher than 100% (Xu et al., 2012).

2.4.3. Heat Transfer correlations

As it occurs for the pressure drop, in the last sixty years a large number of correlations for the prediction of the heat transfer coefficient during phase change of pure fluids flowing in pipes has been developed. Unfortunately due to the complexity of the phenomenon no one can be considered general, accordingly in this paper some of them have been chosen based on previous studies dealing with flow boiling of propane, preference has been given to correlations with lowest MARD, leading to the selection of the correlations listed in this section.

The correlation by (Choi et al., 2007) developed for refrigerants, is based on the model by (Chen, 1966), where the two mechanisms of nucleate boiling and convective heat transfer were assumed to have an additive effect, with different weights, determined by the parameters S and F respectively. For convective heat transfer parameter F was considered,
this parameter is always larger than 1, since in two-phase flow the velocities are always higher than single phase liquid only condition, therefore forced convective heat transfer is enhanced. Chen assumed that F is strictly a flow parameter and only a function of Lockhart and Martinelli.

Furthermore, a suppression factor S is applied to the pool boiling contribution, calculated from Forster and Zuber pool boiling equation. The suppression factor is always less than 1 because in flow boiling the wall superheating is lower compared to pool boiling because of the higher heat transfer coefficient. S is a function of a two-phase Reynolds number. It should be noted that this correlation only applies to vertical flows and water as the fluid.

2.4.3.1. (Chen, 1966)

\[ h_{tp} = S h_{nb} + F h_{lo} \]

where:

\[ h_{nb} = h_{froster-zuber} \]
\[ h_{lo} = h_{Dittus-Boelter} \]
\[ F = f(X_{tt}) \text{ and } S = f(Re_{tp}) \]

2.4.3.2. (Choi et al., 2007)

\[ h_{tp} = S h_{nbc} + F h_l \]

\[ S = 469.1689 (\varphi_1^2)^{-0.2093} B_o^{0.7402} \]
\[ h_{nbc} = 55 P_r^{0.12} (-0.434 \ln P_r)^{-0.55} M^{-0.5} q^{0.67} \text{ in } \text{Wm}^{-2} \]
\[ F = 0.042 \phi_1^2 + 0.958 \]

\[ h_l = \frac{0.023 k_l}{D} \left[ \frac{G (1 - x)D}{\mu} \right]^{0.4} \left[ \left( \frac{C_{pl} \mu_l}{k_l} \right) \right]^{0.4} \]

Where:

\[ \phi_1^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \]
\[ X^2 = \left( \frac{\mu_l}{\mu_g} \right)^{\frac{1}{2}} \left( \frac{1-x}{x} \right)^{\frac{2}{3}} \left( \frac{\rho_g}{\rho_l} \right) \]

\[ C(tt) = 20 \quad C(vt) = 12 \quad C(tv) = 10 \quad C(vv) = 5 \]

For the case of refrigerants in minichannels the correlation of (Choi et al., 2007) is developed by modifying the multiplier F, in order to account for several factors not included in (Chen, 1966) model, mainly:
Chapter 2: Review of existing models

1- The effect of heat flux by correlating S with the boiling number.

2- The extension to fluids other than water through the pool boiling model by (Cooper, 1984).

3- The consideration of viscous flow regime in the two phase multiplier, according to the (Chisholm, 1983).

2.4.3.3. (Liu & Winterton, 1991)

\[ h_{tp} = \sqrt{(F h_{D-B,lo})^2 + (S h_{cooper})^2} \]

\[ F = \left[ 1 + x Pr \left( \frac{\rho_l}{\rho_g} - 1 \right) \right]^{0.35} \]

\[ S = (1 + 0.055 Re_{lo}^{0.16} F^{0.1})^{-1} \]

\[ \text{where} \]

\[ h_{cooper} = 55 P_{Fr}^{0.12} (-\log_{10} P_{Fr})^{-0.55} M^{-0.5} q^{0.67} \]

q is in watts per square meter.

The third correlation to be discussed here is the formulation by (Liu & Winterton, 1991). based on the experimental results provided in (Wang et al., 2014) this correlation can predict 99.5% of the data points with lower than 30% error, therefore it can be considered a reliable analytical model for prediction of boiling heat transfer coefficient in small horizontal tubes for refrigerants.

Similar to the correlation by (Choi et al., 2007), this correlation uses the formulation provided by (Chen, 1966) while tweaking the coefficients for a better prediction of refrigerants.

2.4.3.4. Other Correlations

Furthermore, some other correlations were considered as they are very widely used for horizontal smooth tubes, a summary of these correlation are presented in Table 2.

(S. G. Kandlikar, 1990)

\[ \frac{h_{tp}}{h_t} = C_1 C_0 C_2 (25 F r_{lo})^{C_5} + C_3 B o^{C_4} F_{fl} \]

Where:
Chapter 3: Literature review

**Boiling Number**  \( Bo = \frac{q''}{h_{tg}g} \)

**Convection Number**  \( Co = \left( \frac{1-x}{x} \right)^{0.8} \times \left( \frac{p_g}{p_t} \right) \)

**Froude Number**  \( Fr_{tg} = \frac{g^2}{\rho_{tg}g} \cdot D \)

\( Fr_{tg} \) function of fluid. (for refrigerants \( \approx 1.3 - 1.6 \))

<table>
<thead>
<tr>
<th>( Co &gt; 0.65 )</th>
<th>( C_1 )</th>
<th>( C_2 )</th>
<th>( C_3 )</th>
<th>( C_4 )</th>
<th>( C_5 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.6683</td>
<td>-0.2</td>
<td>1058</td>
<td>0.7</td>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>1.136</td>
<td>-0.9</td>
<td>667.2</td>
<td>0.7</td>
<td>0.3</td>
<td></td>
</tr>
</tbody>
</table>

\( C_5 = 0 \) if vertical or horizontal and \( Fr_{tg} \) > 0.4

\[ h_t = 0.023k_t \frac{D}{\mu} \left[ \frac{G(1-x)D}{\mu} \right]^{0.8} \left( \frac{C_{pl} \mu_l}{k_t} \right)^{0.4} \]

(Shah, 1982)

\[ \psi = \frac{h_{tp}}{h_t} \]

If \( Co < 0.1 \)

\( \psi_{bs} = F. Bo^{0.5} \cdot \exp(2.47 \cdot Co^{-0.15}) \)

If \( 0.1 < Co < 1 \)

\( \psi_{bs} = F. Bo^{0.5} \cdot \exp(2.74 \cdot Co^{-0.1}) \)

\[ \psi_{cb} = \frac{1.8}{Co^{0.8}} \]

\( \psi = \text{MAX}(\psi_{bs}; \psi_{cb}) \)

(Tran, Wambgsass, & France, 1996)

\[ h_{tp} = (8.4 \cdot 10^{-5}) \times (Bo^2 \cdot We)^{0.3} \left( \frac{\rho_t}{\rho_g} \right)^{-0.4} \]

Where:

**Weber Number**  \( We = \frac{g^2d}{\rho_1\sigma} \)

(Gungor & Winterton, 1986)

\[ h_{tp} = Eh_t + Sh_{pool} \]

Where:

\[ E = 1 + 24 \cdot 10^3 \times Bo^{1.16} + 1.37 \left( \frac{1}{X_{tt}} \right)^{0.86} \]
2.4.4. Remarks on small diameter channels (pipes)

As most of the experiments and correlations are developed for macro channels, it is important to discuss the effect of the tube diameter. According to (Satish G Kandlikar, 2002) the dimensionless number used to classify macro, mini, and micro channels is the Laplace number (Equation 2-23)

\[
La \,<\, 0.1 \,\text{characterizes micro channels while } 0.1 \,<\, La \,<\, 1 \,\text{is the range of mini channels. Laplace number is sometimes also mentioned as Confinement (Co) number, but this was avoided in this study to prevent confusion with the convection number } [Co = \left( \frac{1-x}{x} \right)^{0.8} \times \left( \frac{\rho \mu}{\rho_1} \right)] .
\]

As the Laplace number in this study is around 0.5 confinement of the flow is expected to have a great significance. Many of the correlations able to predict heat transfer coefficients in working conditions with Laplace number above unity fail even if the Laplace number drops slightly below one; Therefore special attention has been paid to use correlation that include the surface tension in form of the Laplace number (or in some cases the Weber number).
3. Literature review

3.1. Heat transfer and pressure drop characteristics of propane

Due to propane’s potential as a substitute for traditional refrigerant, many studies have been performed regarding propane’s performance and characteristics in refrigeration cycles, with the most important points of interest being the heat transfer coefficient and pressure drop. The results are listed as follows.

(Shin, Kim, & Ro, 1997) studied the convective heat transfer of a variety of pure fluids and zeotropic mixtures including propane and R290/R600a mixture in steel tubes with an internal diameter of 7.7mm. The experiments were carried out with electric heating, in constant evaporation temperature of 12°C, heat flux 10 to 30 kW m⁻², and mass flux from 424 to 742 kg (m² s⁻¹) for pure propane and mass fluxes ranging between 268 to 583 kg (m² s⁻¹) for mixture. Results show that heat transfer coefficients depend strongly on heat flux at a low-quality region and become independent as vapor quality increases, furthermore results show that between the chosen fluids R290 and R600a hold the highest heat transfer coefficient. The experiment results for pure fluids were compared with the correlation of (Gungor & Winterton, 1986) which showed a standard deviation of 30.5%.

(Watel & Thonon, 2002) studied boiling of propane during a vertical up flow inside a compact serrated plate-fin exchange, the maximal heat rate exchanged between both fluids was equal to 70 kW. The vapor quality ranged from 0.15 to 0.9. Tests are carried out with propane mass fluxes between 12 and 71 kg (m² s⁻¹) for evaporation pressures between 0.46 and 1.17 MPa, results shows the separate effects of quality, mass flux, and pressure on heat transfer coefficient.

In (Lee, Yoon, Kim, & Bansal, 2005) the refrigeration characteristics (evaporation heat transfer coefficient and pressure drop) of R290, R-600a and R-1270 were experimentally obtained through a double horizontal tube (D_i=12.7 and 9.52mm) utilizing the water bath heating to provide the energy required and keep the tube temperature at 14°C. Furthermore, the results were compared with R22 gas to make assumptions for the suitability of these gases for heat transfer applications. In this test, the evaporation temperature was varied between -10°C to 10°C, mass flux ranged between 240 to 480 kg (m² s⁻¹). The results showed that heat transfer performance of tested HCs was better compared to R22, R-1270.
had the highest heat transfer coefficient followed by R600a and then R290. The diameter of the tube also makes a difference with the smaller tubes having a higher heat transfer. Like other papers, results showed that with higher mass flux, the heat transfer raises as well. Furthermore by comparing the experimental results with correlations it was found out that Kandlikar correlation (Kandlikar 1990) has the least error in predicting the heat transfer coefficient in between several correlations used in the research.

(Wen, Ho, & Hsieh, 2006) have discussed the heat transfer and pressure characteristics of R-290, R-600, and R-290/R-600 mixture (55% to 45%) in $D_t=2.46$ mm tube banks, with the test conditions as: heat flux ranging from 4-21 kW m$^{-2}$, mass flux ranging between 250-500 kg (m$^2$ s)$^{-1}$ and vapor quality ranging from 0 to 0.86. The results show that under the same heat and mass flux, propane’s heat transfer coefficient was 1.66-19.96 times greater than R134a’s, while the heat transfer coefficient of R290/R600 mixture was reported to be 1.32-1.50 times greater compared to R134a. As for the pressure drop, the result showed a 1.22 to 1.40 fold decrease in the values of pressure drop of R290 compared to R134a. Furthermore a new heat transfer correlation able to predict the experimental data for both pure refrigerants and refrigerant mixtures was created, with an absolute average deviation of 11.5% for refrigerants and refrigerants mixtures.

(Choi et al., 2009) discussed the convective boiling pressure drop and heat transfer of propane in horizontal stainless-steel smooth minichannels ($D_t=1.5$mm and 3mm), the data were obtained for heat fluxes ranging from 5–20 kW m$^2$, mass fluxes ranging from 50–400 kg (m$^2$ s)$^{-1}$, saturation temperatures of 10, 5 and 0 °C, and quality up to 1.0. Results show that mass flux has a strong effect on the pressure drop. An increase in the mass flux results in a higher flow velocity, which increases the frictional and accelerational pressure drops. Moreover, pressure drop increases as the heat flux increases. It is assumed that the increasing heat flux results in a higher vaporization, which increases the average fluid vapor quality and flow velocity. Analysis of the data shows that the effect of mass flux in the low vapor region is negligible while in the higher vapor quality region the increase of mass flux has resulted an increase in the heat transfer coefficient, this was attributed to the idea that in the small diameter channels the nucleate boiling is predominant while in the higher quality region convective boiling becomes more apparent. The results also show that the dry out happens in lower vapor quality for smaller diameter channels. Furthermore, a new correlation was developed to predict the pressure drop and heat transfer coefficient for
propane. A new pressure drop correlation was developed based on the Lockhart Martinelli method as a function of the two-phase Reynolds number, $Re_{TP}$, and the two-phase Weber number, $(We_{TP})$. A new factor $C$ was developed using a regression method with a comparison agreement of 10.84% mean deviation and 1.08% average deviation. A new boiling heat transfer coefficient correlation that is based on a superposition model for refrigerants in minichannels was presented with 9.93% mean deviation and -2.42% average deviation.

(Maqbool, Palm, & Khodabandeh, 2013) researched heat transfer and pressure drop characteristics of propane in a vertical 1.70 mm minichannel, results show that the heat transfer coefficients increase with the rise of heat flux and saturation temperature while the influence of mass flux and vapour quality is relatively negligible. Furthermore, the two-phase frictional pressure drop gradients, are higher for increased heat flux, mass flux and vapour quality. A reduction in the two-phase frictional pressure drop gradients is observed for higher saturation temperature.

In a comprehensive study (Wang et al., 2014) investigated two-phase heat transfer and pressure drop of propane during saturated flow boiling inside a horizontal tube, as these testing conditions are similar to what is going to be discussed in this thesis, this paper will be discussed more thoroughly here, to have a point of reference and comparisons. In Table 3 a more detailed summary of the test conditions is shown.

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Propane (R290)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube</td>
<td>Copper, smooth</td>
</tr>
<tr>
<td>Internal Diameter (D)</td>
<td>6 mm</td>
</tr>
<tr>
<td>Length of pressure drop section (L)</td>
<td>1550 mm</td>
</tr>
<tr>
<td>Saturation Temperature (T)</td>
<td>-35.0 to -1.9°C</td>
</tr>
<tr>
<td>Mass Flux (G)</td>
<td>62-104 kg (m² s)⁻¹</td>
</tr>
<tr>
<td>Heat Flux (q)</td>
<td>11.7-87.1 kW m²</td>
</tr>
<tr>
<td>Heating Medium</td>
<td>Electrical Heating</td>
</tr>
</tbody>
</table>

*Table 3 Test conditions for (Wang et al., 2014)*
It should be noted that the pressure drop measurement are done in separate adiabatic part of the test section. In here we are going to discuss the effects of different parameters on heat transfer coefficient and forgo the pressure drop measurement as the frictional pressure drops for adiabatic states are different from diabatic systems.

The results show that an increase in mass flux increases the heat transfer coefficient, while this is true in all vapor qualities this increase is very small in low vapor quality region (5%) it can be rather appreciable in higher vapor qualities (up to 20% increase) this was attributed to the fact that in the low vapor quality region the nucleate boiling prevails while the convective boiling contribution is weak. While in high vapor quality region, the flow pattern changes from stratified flow to fully developed annular flow, which in turn increases the heat transfer rate. As for the effect of heat flux, the results showed that in the same testing condition (Saturation temperature, mass flux and vapor quality) there is a linear increase between heat flux and heat transfer coefficient, it was reasoned that it may be like the situation in pool boiling where with increasing heat flux, bubble departure frequency and the nucleation sites number increase rapidly. Saturation temperatures has a peculiar effect on heat transfer coefficient, in low heat fluxes there is no apparent effect on heat transfer coefficient while in higher heat fluxes the difference can be appreciable with an increase of up to 50 percent in low quality region, however this data show that this increase starts to decline in higher quality region. Finally, for the effect of vapor quality, it was shown that the local heat transfer coefficient can either increase with quality, remain constant, or decrease with quality. This was considered mainly due to two important parameters Boiling number, Bo and liquid to vapor density ratio ($\rho_l/\rho_v$). They pointed out that for a high-density ratio, the convective effects dominate which leads to an increasing trend for the local heat transfer coefficient with the increase of vapor quality, while a high boiling number leads to a high nucleate boiling contribution, which tends to decrease as the vapor quality increases. This leads to a decreasing trend for the heat transfer coefficient with increasing vapor quality. Furthermore the experimental results were compared with five well-known correlations, (Liu & Winterton, 1991) correlation shows the best agreement, with a mean absolute relative deviation less than 10%, and over 99% of points within a deviation bandwidth of ±30%.

In a similar comprehensive study (Del Col, Bortolato, & Bortolin, 2014) managed to study the local evaporation and condensation heat transfer coefficients of propane in a 0.96mm diameter minichannel while also studying the pressure drop characteristics of it.
Flow boiling tests were carried out at 31°C saturation temperature and at mass flux between 100 Kg (m^2 s)^{-1} and 600 Kg (m^2 s)^{-1} the results showed that the boiling heat transfer coefficients for propane strongly depend on the heat flux while mass velocity and vapor quality have a minor influence., It was argued that this decrease may be due to a partial dryout at higher vapor qualities. All the models considered in the comparison underestimate the experimental data for heat transfer coefficient.

As discussed before a transition between flow patterns usually causes a large change in the value of heat transfer coefficient. Therefore it is important to have a flow pattern map suitable for prediction of flow patterns and their change. (Zhuang et al., 2015) conducted tests at saturation pressures from 0.2 MPa to 0.4 MPa for mass fluxes ranging from 70 to 180 Kg (m^2 s)^{-1} and two heat fluxes (20.16 and 37.1 kW m^2) in a smooth tube, using visualization of the flow they found out that the flow regime change follows the same trend as flow pattern maps for R134a, a modified transition equation was proposed from intermittent to annular flow.
As a point of comparison in this study Table 4 will compare the test condition in previous investigation in this subject.

<table>
<thead>
<tr>
<th>Refrigerant(s)</th>
<th>Tube</th>
<th>Internal Diameter [mm]</th>
<th>Saturation Temperature (or equivalent Pressure)</th>
<th>Mass flux [kg (m² s)⁻¹]</th>
<th>Heat Flux [kW m⁻²]</th>
<th>Heating Medium</th>
</tr>
</thead>
<tbody>
<tr>
<td>(Shin et al., 1997)</td>
<td>R22, R32, R134A, R290, and R600a &amp; mixtures of R32/R134a, R290/R600a, R32/R125</td>
<td>Stainless Steel</td>
<td>7.7</td>
<td>12°C</td>
<td>424–742 for R22, R32, R134a, R32/R134a, R32/R125; 265–583 for R290, R600a, R290/R600a</td>
<td>10 to 30</td>
</tr>
<tr>
<td>(Wen et al., 2006)</td>
<td>R290</td>
<td>Three-line serpentine, Copper Smooth</td>
<td>2.46</td>
<td>6°C</td>
<td>250–500</td>
<td>5 to 21</td>
</tr>
<tr>
<td>(Watel &amp; Thonon, 2002)</td>
<td>R290</td>
<td>Compact serrated plate-fin exchange</td>
<td>N/A</td>
<td>0.46 and 1.17 MPa</td>
<td>12-71</td>
<td>Max 70</td>
</tr>
<tr>
<td>(Lee et al., 2005)</td>
<td>R-290, R-600a, R-1270, R22</td>
<td>Copper Smooth</td>
<td>12.7, 9.52</td>
<td>-10.15 to 9.85°C</td>
<td>50-200</td>
<td>Water @ 287°C with mass flow 240 to 480 Kg h⁻¹</td>
</tr>
<tr>
<td>(Choi et al., 2009)</td>
<td>R290</td>
<td>Stainless steel</td>
<td>1.5, 3.0</td>
<td>0, 5, 10°C</td>
<td>50-400</td>
<td>5-20</td>
</tr>
<tr>
<td>(Maqbool et al., 2013)</td>
<td>R290</td>
<td>Stainless steel, vertical</td>
<td>1.9</td>
<td>23, 33, 43°C</td>
<td>100-500</td>
<td>5-280</td>
</tr>
<tr>
<td>(Wang et al., 2014)</td>
<td>R290</td>
<td>Copper Smooth</td>
<td>6.0</td>
<td>-35.0 to -1.9</td>
<td>62-104</td>
<td>11.7-87.1</td>
</tr>
<tr>
<td>(Del Col et al., 2014)</td>
<td>R290</td>
<td>Copper, Rough</td>
<td>0.96</td>
<td>31°C</td>
<td>100-600</td>
<td>20-200</td>
</tr>
<tr>
<td>(Zhuang et al., 2015)</td>
<td>R290</td>
<td>Copper Smooth</td>
<td>6.0</td>
<td>0.2 MPa to 0.4 MPa</td>
<td>70 to 180</td>
<td>20.16 and 37.1</td>
</tr>
</tbody>
</table>

*Table 4 Review of the test conditions in previous studies*
3.2. Mixtures of propane with other fluids and other important studies

In this part of literature review, we will look at other related works in not directly related to this study, but pertaining to neighboring fields, namely, mini-tubes performance, CO\textsubscript{2} refrigeration cycles and mixtures of propane with other fluids.

(Yan & Lin, 1998) did experiments with tubes of D\textsubscript{i}=2 mm regarding the heat transfer coefficient of evaporating R134a and compared it to larger pipes (D\textsubscript{i} \geq 8.0 mm) reported in the literature. The results showed an increase of 30–80% in the heat transfer coefficient for most situations.

As hydrocarbon gases are in competition with CO\textsubscript{2} to become the next generation of refrigerant fluids, it is worth taking a look at the literature on CO\textsubscript{2} for a fair comparison. (Choi et al., 2007) studied evaporation heat transfer of R-22, R-134a, and CO\textsubscript{2}, in stainless steel micro channels with D\textsubscript{i} =1.5 and 3mm. The test conditions were saturation temperatures 0-10°C, inlet vapor quality from 0 to 1, heat flux ranging from 5 to 20 kW m\textsuperscript{2} and mass flux from 50 to 400 kg (m\textsuperscript{2} s\textsuperscript{-1}). The tests were performed using electric heating. The results demonstrated that the effect of vapor quality and mass flux is insignificant in the low vapor quality region. Moreover, the heat transfer drop associated with dry-out occurred at lower vapor quality for higher mass flux. Taking R-22 as a benchmark, the ratio of the heat transfer coefficient resulted 0.8 for R-134a and 2.0 for CO\textsubscript{2}. Since the correlations available for macro-channels showed large deviations (in the order of 20%) a new model was developed to account for the diameter. The prediction ability improved with a mean deviation of 11.21% and a relative deviation of -0.72%.

(Thome, Cheng, Ribatski, & Vales, 2008) reported a review of flow boiling heat transfer, two-phase pressure drops and flow patterns of ammonia and hydrocarbons was performed. (Zou et al., 2009) studied flow boiling heat transfer of binary mixtures of R170/R290 for three different sets of composition. Test were performed in a horizontal stainless-steel tube with D\textsubscript{i}=7.7mm with electrical heating. The experiments were carried out at three different compositions of R170 and R290. The test conditions were: mass flux ranging from 63.6 to 102.5 kg (m\textsuperscript{2} s\textsuperscript{-1}), heat flux ranging between 13.1 and 65.5 kW m\textsuperscript{2}, evaporation pressure was tested within 0.35 and 0.57 MPa. The results showed that the heat transfer coefficients of R170/R290 mixtures increased with the increase of heat flux, mass flux and vapor quality. On the other
hand, evaporating pressure did not show an appreciable effect in heat transfer. Regarding the flow patterns, results showed that mass flux and vapor quality have a smaller influence on the flow pattern transitions for R170/R290 mixtures compared with pure substances. It was argued that the dryout onset takes place at higher vapor quality, resulting in a better overall heat transfer.

(Park & Jung, 2009) carried out an investigation on the use of R170/R290 mixture in the composition was varied from 0% to 10%. Test results showed that the COP decreases and the capacity increases with an increase in the concentration of R170. R170/R290 composition of 4%/96% showed capacity and similar to R22, therefore it was argued that this composition is suitable for a drop-in replacement of the refrigerant systems currently using R22.

(Grauso, Mastrullo, Mauro, & Vanoli, 2011) investigated the CO2/R290 (70%/30% wt.) Mixture heat transfer capabilities in 6 mm tubes. The mass flux in this test ranged between 200 and 35 kg (m² s)⁻¹, the heat flux between 10 and 20 kW m⁻² the evaporation temperature from 6.9 to 14°C. The results showed a strong decrease of heat transfer coefficient for the mixtures, in particular for the 70%/30% mixture the heat transfer coefficient decreased up to 3 times compared to pure CO₂. Moreover, the heat transfer coefficient results were only slightly dependent on the mass flux and the working temperature, while it was strongly influenced by the heat flux.

3.3. Conclusion on literature review

Based on the previous review, the following conclusions can be drawn about the flow boiling of propane:

1-The available correlations for the heat transfer coefficient usually underestimate the experimental results. This is more apparent for mini-tubes since the correlations have been usually developed for macro-tubes.

2-Flow visualizations show that propane undergoes the same patterns as R134a, the only change being the transition from annular to intermittent flow, which happens in lower vapor qualities.
3-As expected the heat transfer coefficient increases with both the heat flux and the mass flux. More specifically an increase in the heat flux has a strong effect at low vapor quality (boiling dominated region) whereas it's negligible at higher qualities. The contrary occurs for the influence of mass flux (high vapor quality region, convection dominant).

4- The pressure drop increases with the mass flux and the heat flux.

5- The effect of the saturation temperature is controversial. The effect of the saturation temperature seems to be negligible on the heat transfer coefficient while the pressure drop may lower with the increase of saturation temperature.

6-Based on the comparison with the experimental data the following correlations were found to be the most accurate: (Liu & Winterton, 1991) for the heat transfer coefficient, (Muller-Steinhagen & Heck, 1986) for the pressure drop and (Woldesemayat & Ghajar, 2007) for the void fraction.

However, due to the complexity of two phase flow nature, there are still major controversies in the results reported in the literature review. This is further complicated by the data range considered in by each study, which is limited to the scope of the study and the application of interest for the author. The present work aims at contributing to extend the database and to provide useful comparisons with the available models. In particular the internal diameter of the tested tube, equal to 4.2 mm, leads to a confinement number of about 0.5, i.e. in an intermediate range between mini and macro-tubes.
4. Experimental test rig

To measure the pressure drop and heat transfer coefficient in different conditions an experimental test rig was used. Test rig was rebuilt based on an existing setup and modified to achieve the accuracy required and to improve the range of operating conditions. In this chapter the experimental setup is described in detail. Furthermore, the uncertainty analysis is presented.

4.1. Test rig design

To achieve the goal of this research the experimental setup has the following features:

1- The vapor quality at the inlet must be controlled, this is realized by a preheater; with known inlet saturation temperature and mass flow the heating capacity of the preheater is calculated in order to set the defined vapor outlet.

2- The heat transfer rate of the test section must be controlled in order to enable setting the vapour quality variation, this is achieved by means of an energy balance.

3- The cooling capacity is provided by a condenser and a subcooler, each one connected to a chiller using ethylene glycol as cooling fluid.

4- The controlling of the mass flux is realized by a pump with an inverter, which is installed after the subcooler to ensure single phase liquid inlet.

5- Pressure drop is acquired through a gauge pressure sensor connected to the inlet and outlet of the test section.

6- Heat transfer coefficient is obtained by Equation 2-5. Saturation temperature is measured by the temperature sensor at the inlet of test section and the wall temperature is acquired by two pairs of temperature sensors connected to the inlet and outlet of the surface of the copper tube.

The schematic diagram of the test rig is shown in Figure 1. The set-up consists of two loops: The first one is the propane loop; the second one is a cooling loop. A data logger system is installed on the set-up and connected to a personal computer for control and data acquisition.
Propane flows in a closed cycle, starting from the condenser and subcooler it is ensured that the liquid is subcooled for this is essential for the proper operation of the pump and the flow meter. The pump is controlled via frequency change using an inverter, the flow meter (Coriolis type, RHEOINK, RHM04) is placed after the pump. A glass sight gauge placed just downstream of the flow meter enables checking whether single phase flow occurs. Then, the propane flows through the electrical preheater, in order to set the desired vapor quality, after an adiabatic section to ensure fully developed conditions, propane flows through the test section, where electrical heating provides the desired vapour quality variation. At this stage the measurements of the heat transfer coefficient and the pressure drop take place. At the outlet of the test section glass sight enables visualizing the flow regime. Finally, propane moves through the receiver and goes back to the condenser.
Chapter 4: Experimental test rig

The whole test rig is installed on an aluminum frame, provided with wheels for transportation. Furthermore, it is enclosed in a plastic case connected to a fan for proper venting of propane in case of a leakage. An overview of the test rig is shown in Figure 2.

As for the cooling loop, three chillers were used, both of them using ethylene glycol as the cooling agent. They were connected to the propane loop with two plate and fin heat exchangers.

4.1.1. Component selection

The components used for the test rig were chosen according to the requirements of the test conditions.
4.1.2. Pump

Considering that the maximum mass flux is \( G = 500 \frac{kg}{m^2s} \) and the internal diameter of the tube is \( D_i = 4.2 \text{ mm} \) the maximum flow rate in the test condition can be calculated as:

\[
\dot{m}_{\text{max}} = G_{\text{max}} S = 6.92 \times 10^{-3} \frac{kg}{s}
\]

Equation 4-1

Knowing that the density of the liquid propane is lowest at the highest test temperature \( T_{\text{sat}} = 10^\circ C \) we can write:

\[
V_{\text{max}} = \frac{\dot{m}_{\text{max}}}{\rho} = 0.75 \frac{L}{\text{min}}
\]

Equation 4-2

The selected pump is TUTHILL DGS.68EEET2NNSM423 which has a maximum flow rate of \( 1 \frac{L}{\text{min}} \).

4.1.3. Electrical heaters

To be on the safe side, the minimum requirement for the heat input is established on the maximum heat flux and mass flow rate.

For the preheater, assuming a maximum 10°C of subcooling it can be calculated as:

\[
P_{\text{pre heater}} = \Delta i_{\text{v@sat}10^\circ C} \dot{m}_{\text{max}} \Delta x + \dot{m}_{\text{max}} C_{p,\text{liq}} \Delta T = 2513 \text{ W}
\]

Equation 4-3

For the test section it can written:

\[
P_{\text{test section}} = \Delta i_{\text{v@sat}10^\circ C} \dot{m}_{\text{max}} \Delta x = 260.192 \text{ W}
\]

Equation 4-4

The resistance type used for the heaters is 2M-1-600, worth of \( 11 \frac{\Omega}{\text{m}} \).

Based on the experimental measurements of the voltage and the resistance, the total power output of the preheater and the test section are found to be:
Chapter 4: Experimental test rig

\[ R_{\text{preheater}} = 14,8 \, \Omega \]

\[ R_{\text{test section}} = 82,4 \, \Omega \]

\[ P = \frac{V^2}{R}, \quad \text{where} \quad V = 220 \, V \]

\[ P_{\text{pre heater}} = 3270 \, W, \quad P_{\text{test section}} = 587 \, W \quad \pm 0.5\% \]

4.1.4. Condenser and subcooler

Condensers should be able to cool down all the flow to subcooled liquid. Both condenser and subcooler are connected through a plate-and-fin heat exchanger to the propane test rig. The cooling agent in both chillers is ethylene glycol. Since for low temperature conditions the cooling capacity of the chiller diminishes, an additional subcooler is used in such extreme conditions.

<table>
<thead>
<tr>
<th>Model</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Condenser</strong></td>
<td>ATC K9</td>
</tr>
<tr>
<td><strong>Subcooler</strong></td>
<td>ATC K6</td>
</tr>
<tr>
<td><strong>Extra Subcooler</strong></td>
<td>Julabo FP50</td>
</tr>
</tbody>
</table>

*Table 5 List of the chillers used for condensation*

The cooling capacity of the mentioned chillers are shown in Figure 3.

Table 6 reports the data for cooling capacity of Julabo FP50:

<table>
<thead>
<tr>
<th>Working Temperature (°C)</th>
<th>20</th>
<th>0</th>
<th>-20</th>
<th>-30</th>
<th>-40</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooling Capacity (KW)</td>
<td>0.9</td>
<td>0.8</td>
<td>0.5</td>
<td>0.32</td>
<td>0.16</td>
</tr>
</tbody>
</table>

*Table 6 Cooling capacity for Julabo FP50 Chiller*
Figure 3 Cooling capacity of main chillers
4.1.5. Test Section

A smooth copper tube was tested. Tube internal diameter is $D_i = 4.2 \ mm$ and outer diameter of $5 \ mm$.

The heating element is attached to the copper block, there is a separate thermocouple in contact with the copper block for safety reasons. The test section is buried in a box filled with powder insulator (Perlite) and further insulated towards the ambient with foam, accordingly based on the estimate of the resulting thermal resistance heat leakage is negligible.

4.2. Test Conditions

Test conditions are summarized on Table 7.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Test Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant</td>
<td>Propane (R290)</td>
</tr>
<tr>
<td>Evaporating Temperature</td>
<td>$0, 5, 10^\circ C$</td>
</tr>
<tr>
<td>Mass Flux</td>
<td>$250-505 \ \frac{kg}{m^2 s}$</td>
</tr>
<tr>
<td>Inlet Quality</td>
<td>$0.2-0.9$</td>
</tr>
<tr>
<td>Heat flux</td>
<td>$15-61 \ \frac{kw}{m^2}$</td>
</tr>
<tr>
<td>Quality Change $\Delta x$</td>
<td>$0.1-0.4$</td>
</tr>
<tr>
<td>Test Section Length</td>
<td>$0.5 \ m$</td>
</tr>
<tr>
<td>Test section material</td>
<td>Copper, smooth and micro finned</td>
</tr>
<tr>
<td>Internal Diameter</td>
<td>$4.2 \ mm$</td>
</tr>
</tbody>
</table>

*Table 7 Test Conditions*

The test conditions are chosen based on the industrial application requirements.

4.3. Control systems

According to the test conditions there are parameters that need to be controlled and set to desired value, as it is discussed in the following.

4.3.1. Controlling mass flux

Control of the mass flux is realized by adjusting the rotation speed of the pump. This is achieved using an inverter to control the frequency of the electric field applied to the pump. A
mass flow meter (RHEOKIN RHM08) is installed after the pump. It is a Coriolis flow meter able to detect only the liquid flow, therefore a subcooler next to the condenser ensures that the pump is fed with the only liquid phase. A glass sight window is installed after the mass flow meter for visual inspection.

4.3.2. Controlling inlet vapor fraction

Heat input to the preheater is adjusted based on the saturation pressure, mass flux and amount of subcooling to achieve the desired vapor fraction. The equation used for calculation of power to preheater is:

\[ P_{pre\, heater} = \Delta h_{lv@T_{sat}} \dot{m} \Delta x + \dot{m} \Delta \left( C_{p,liq} T_{subcooling} \right) \]  

Equation 4-5

4.3.3. Controlling saturation pressure (temperature)

Since the working conditions of propane lie in the two-phase region and equilibrium may be assumed between the phases, temperature and pressure are corresponding to saturation. In any case the saturation temperature and pressure of the system are solely controlled by the condenser. Temperatures of the subcooler and condenser chiller are controlled by the operator.

4.3.4. Controlling heat flux in the test section

Heating for the test section is provided by an electrical resistance heater (2M-1-600). The control system for the heating of test section is similar to the one for the preheating section. It is based on the following expression of the heat flow rate.

\[ P_{test\, section} = \Delta h_{lv@T_{sat}} \dot{m} \Delta x \]  

Equation 4-6

4.4. Measurement and data logging system

To obtain desired results from the experimental test rig, data from all the sensors are acquired using a data logging system and saved in an excel format. Finally, pressure drop and heat transfer coefficient are obtained as postprocessing of the data.
4.4.1. Data logging and control system

Using LabView a control system and data logger has been implemented. The frequency of the pump and heat inputs are set, while the saturation temperature is adjusted manually by the operator acting directly on the condenser and subcooler. Furthermore, this interface reads the data acquired by the data logger (National Instrument NI9211) such as temperature, pressure sensors and mass flow rate. After reaching the steady state which is considered to be 2 minutes after surface temperature change is less than 1%, the data storage is initiated all the data are stored in an EXCEL file. An overview of the developed LabView interface is illustrated in Figure 4.

![LabView Interface developed for Control System and Data Acquisition](image)

The list of the instruments is presented in Table 8, with further information including manufacturer, uncertainty, connection type and size. More details about temperature and pressure measurements are given in the next section.
## Table 8 List of instruments

<table>
<thead>
<tr>
<th>Displayed text</th>
<th>Description</th>
<th>Connection Type</th>
<th>Output signal</th>
<th>Manufacturer</th>
<th>Model</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow Meter</td>
<td>Coriolis mass flowmeter</td>
<td>8 mm</td>
<td>4-20 mA</td>
<td>RHEOINK</td>
<td>RHM04</td>
<td>±0.1%</td>
</tr>
<tr>
<td>$P_1$</td>
<td>Absolute Pressure sensor</td>
<td>(1/8)''</td>
<td>4-20 mA, 2-wire</td>
<td>WIKA</td>
<td>S10</td>
<td>±0.5%</td>
</tr>
<tr>
<td>$P_2$</td>
<td>Gauge Pressure sensor</td>
<td>(1/8)''</td>
<td>4-20 mA, 2-wire</td>
<td>WIKA</td>
<td>DPT-10</td>
<td>±0.15%</td>
</tr>
<tr>
<td>$T_4$~$T_7$</td>
<td>Shielded thermocouple wire</td>
<td>Outer tube surface</td>
<td>0.5mm Type T</td>
<td>±0.2°C</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$T_1$~$T_3$ and $T_8$</td>
<td>Thermocouple</td>
<td>In propane cycle</td>
<td>0.5mm Type T</td>
<td>±0.2°C</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Data logger</td>
<td>-</td>
<td>-</td>
<td>National Instruments</td>
<td>NI 9211</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>$Q_{preheater}$</td>
<td>Preheater 3.27 kW</td>
<td></td>
<td></td>
<td></td>
<td>±0.5%</td>
<td></td>
</tr>
<tr>
<td>$Q_{test \text{ section}}$</td>
<td>Test-heater 0.58 kW</td>
<td></td>
<td></td>
<td></td>
<td>±0.5%</td>
<td></td>
</tr>
</tbody>
</table>
4.4.2. Temperature measurement

In order to find the heat transfer coefficient pipe wall temperature is needed. This is obtained through 4 thermocouples, two of them are installed in the inlet of the test section while the other two are installed in the end of test section, thermocouples are inserted 10 cm into the drilled copper block. An overview of the test section from top is shown in Figure 5.

![Figure 5 Overview of test section inlet](image)

The thermocouples are held in place by a copper block clamped around the test tube, copper blocks have a groove with adequate space for the thermocouples to be inserted. As the thermocouples are very thin and long inserting them requires considerable attention to avoid breaking.

Each pair is installed in a way that one is on top of the tube and the other one underneath the tube, this is to ensure an average wall temperature on the whole wall, as with certain flow patterns some parts of the wall may show large temperature difference (for example in stratified or slug flow patterns).

Thermocouples are Type T; because this class of sensors provides greater accuracy than other thermocouples in the temperature range of interest in this test. Dimensions of thermocouples are 0.5 mm in diameter and are 1500 mm long. Before beginning the tests,
temperature sensors were calibrated using an ice bath to set zero point and calibration charts from the manufacturer were implemented into the LabView code.

4.4.3. Pressure measurement

Two Pressure sensors were used in this test rig, one is an absolute pressure gauge installed on the inlet of preheater section, which gives the evaporation pressure of the system, the other pressure sensor is a differential pressure sensor and is connected through a T junction to the test tube. This T junction can be seen in Figure 5 just before the heating element.

4.5. Data reduction and Experimental Uncertainty

4.5.1. Thermodynamic data

In order to obtain the thermodynamic properties of propane in different test conditions, the RnLib library developed at NTNU was adopted. This library acts as an extension for Microsoft Excel.

4.5.2. Pressure drop data reduction

The pressure drop along the test section is directly measured. The results are presented in form of total pressure drop and frictional pressure drop (to be compared with correlations). To calculate the frictional pressure drop, accelerational pressure drop computed from Equation 2-11 is subtracted from the total pressure drop. Then, the pressure gradient is calculated, assuming a linear behavior of the pressure along the test section. Pressure drop data are reported for the average vapor fraction, evaluated as follows:

\[ x_{avg} = x_{in} + \Delta x \]  

*Equation 4-7*
Chapter 5: Results and discussion on heat transfer coefficient

Where:

\[ x_{in} = \frac{Q_{in} - (i_{sat, liq} - i_{liq@T,P\text{ in}l})\dot{m}}{(i_{sat, gas} - i_{sat, liq})\dot{m}} \quad \text{Equation 4-8} \]

\[ x = \frac{Q_{TEST}}{\dot{M}(i_{sat, gas} - i_{sat, liq})} \quad \text{Equation 4-9} \]

Since the pressure drop is directly measured, experimental uncertainty is equal to the accuracy of the pressure measurement sensor (±0.15%). Furthermore to calculate the uncertainty of the vapor quality method of (Moffat, 1985) was used. This method can be summarized as:

\[ \delta_{total} = \sqrt{\sum_i \delta_i^2} \quad \text{Equation 4-10} \]

Where \( \delta \) is the uncertainty of each instrument.

To calculate the uncertainty of the vapor quality; accuracies of the related sensors, reported in Table 8 were used, additionally an error of 0.2% was considered for the interpolation of thermodynamic data. The percentage error related to the vapor quality in the whole range of operating conditions was found to lie within 2.0% and 2.3%

4.5.3. Heat transfer coefficient (HTC) data reduction

In order to obtain the average HTC, two HTC were calculated at the inlet and outlet of the test section. Calculation steps are shown in the following:

\[ h_{in} = \frac{q''}{\frac{T_{4in} + T_{5in}}{2} - T_3} \quad \text{Equation 4-11} \]

\[ h_{out} = \frac{q''}{\frac{T_{6in} + T_{7in}}{2} - T_{sat@P_{out}}} \quad \text{Equation 4-12} \]

Where T4-T7 are wall temperatures and T3 is considered to be saturation temperature at inlet.
Chapter 5: Results and discussion on heat transfer coefficient

Since the thermocouples measure the outer wall temperature, the inner wall temperature required to evaluate the HTC is determined assuming a radial heat conduction through the pipe wall. Wall temperature ($T_{in}$) is calculated using the following equation.

$$T_{in} = T - \frac{q'' \cdot \ln \frac{d_o}{d_i}}{k_{copper}}$$  \hspace{1cm} \text{Equation 4-13}

Finally the average heat transfer coefficient is calculated as:

$$h = \frac{h_{in} + h_{out}}{2}$$  \hspace{1cm} \text{Equation 4-14}

Based on the same formulation of the uncertainty analysis shown in Equation 4-10 the maximum uncertainty of the heat transfer coefficient was found to be about 4.7%.

4.5.4. Methodology Error

The methodology error accounts for the unavoidable errors caused by disturbances not controllable or sometimes not even quantifiable. To name a few, the error caused by the interpolation of the thermodynamic data in the calculation and heat leakage from the test section casing can be considered methodology errors. In order to account for these errors, interpolation error of 0.2% was added to uncertainty analysis. Furthermore in order to account for the heat leakage 2% error has been included in the analysis.

4.6. Validation

As a preliminary validation tests were conducted with inlet vapor quality equal or higher than unity (super heated vapor) in order to get single phase flow. Accordingly, the frictional factor and the Nusselt number were determined from measurements and compared with well-known results from the literature, namely the correlations of (Dittus & Boelter, 1930) and (Gnielinski V., 1975), the calculation steps are shown as follows:
The data point from experimental results is corresponding to $T_{sat} = 10^\circ C, G = 250 \frac{k g}{m^2 s}, q'' = 30.78 \frac{kw}{m^2}$ on the smooth tube, therefore Nusselt and Reynolds number are calculated as:

$$Pr = \frac{C_p \mu_g}{k_f} = 0.693$$

$$Re_g = 1.33 \times 10^5$$

(Dittus & Boelter, 1930)

$$Nu_D = 0.023. Re^\frac{4}{5} Pr^{0.4} = 277.87$$

$$h = \frac{Nu. k_f}{D_l} = 1087.64 \frac{W}{m^2 k}$$

(Gnielinski V., 1975)

$$Nu_D = \frac{f}{8} \frac{1}{(Re - 10^3)Pr} \left[ 1 + 12.7 \left( \frac{f}{8} \right)^{0.5} (Pr^{\frac{2}{3}} - 1) \right]$$

$$f = (0.79 \ln(Re) - 1.64)^2 = 0.0169$$

$$Nu_D = 301.93$$

$$h = \frac{Nu. k_f}{D_l} = 1319 \frac{W}{m^2 k}$$

The results show $h = 1297 \frac{W}{m^2 k}$. Error between the (Dittus & Boelter, 1930) correlation and experimental results is 16% while the error for (Gnielinski V., 1975) is equal to -1.6%. accordingly, the test section seems suitable to provide the required measurements.
5. Results and discussion on heat transfer coefficient

In this chapter the data for the boiling heat transfer coefficient of propane in a copper tube are shown and analyzed to illustrate the effect of different parameters on the heat transfer performance. This section is divided into five parts: in first part the general features of the experimental heat transfer coefficient are explained, in the following sections the effects of evaporation temperature, mass flux, heat flux are taken into consideration.

5.1. General characteristics of heat transfer coefficient

Figure 6 illustrates the heat transfer coefficient against the vapor qualities as a significant example test condition were $T_{sat} = 5^\circ C$, $G = 379.41 \frac{kg}{s \ m^2}$, $q'' = 31.88 \frac{kw}{m^2}$. It should be noted that these conditions are averaged over the whole test range data, the deviation from average is minimal for saturation temperature and heat flux while for the mass flux it is noticeable, this is due to the inability of the pump to finely regulate the mass flux specially in low saturation temperatures.

The data are the average of the two heat transfer coefficients obtained at inlet and outlet of the test tube. The heat transfer coefficient for inlet and outlet of tube is shown separately in bottom part of Figure 6.
The results show an increase of the heat transfer coefficient with increasing vapor quality; this can be explained by considering that with higher vapor fractions the flow pattern...
tends toward the annular flow, where a layer of liquid is formed at the wall and by increasing vapor quality the liquid layer thins more and more, so the thermal resistance lowers i.e. the heat transfer coefficient increases. The highest HTC observed in the aforementioned test condition is \( h = 4.25 \frac{kw}{m^2k} \) where vapor fraction is \( x = 0.69 \). This trend is interrupted at dry out, meaning that the liquid film is completely evaporated (this doesn’t imply total evaporation but mist flow i.e. flow of vapor with liquid drops entrained) this occurrence causes a significant fall in the heat transfer coefficient. As it can be seen in the bottom part of Figure 6 this dry-out occurs sooner at the outlet, as the local quality is higher.

5.2 Effect of Saturation Temperature on heat transfer coefficient

In this study three different saturation temperatures were studied (0, 5, 10°C) it should be noted that due to some technical problems the data set for saturation temperature of 0°C is undersized compared to the other saturation temperatures, actually as already pointed out, the cooling capacity at high vapor fractions was limited due to the chiller capabilities, whereas the pump was unable to keep a stable flow at low qualities. Figure 7 illustrates the heat transfer coefficient for two different test conditions, \( G = 500 \frac{kg}{s \ m^2}, q'' = 31 \frac{kw}{m^2} \) and \( G = 379 \frac{kg}{s \ m^2}, q'' = 31 \frac{kw}{m^2} \). It is shown that for higher temperatures the heat transfer coefficient is higher, but very often this difference lies within the uncertainty of the HTC measurement, eventually, a clear effect of the saturation temperature has not been put in evidence.
Chapter 5: Results and discussion on heat transfer coefficient

5.3. Effect of mass flux on heat transfer coefficient

In order to illustrate the effect of mass flux the heat flux has been kept constant while changing the mass flux \((G = 250 - 502 \frac{kg}{s m^2})\), but it should be noted that it is not possible to
keep the same quality change for each value of the mass flux. Figure 8 shows the effect of mass flux on the heat transfer coefficient at saturation temperature of 10°C. The following remarks can be made:

1- The dryout quality seems weakly dependent on $G$ with a tendency to increase as the mass flux lowers.

2- In the post dry-out region the drop in the HTC appears independent of $G$.

3- Up to the dry-out occurrence the HTC increases with the mass flux at constant average quality. Furthermore, the rise in the HTC with the average quality is more pronounced as $G$ increases, this is explained by the relative importance of nucleation and convection as mechanisms of energy transfer. Actually, it is seen that the data at the lowest $G$ show a modest increase with the average quality, indicating a “boiling dominant” condition, whereas the data at the highest $G$ significantly increase with the average quality, indicating a “convective dominated” flow.

![Figure 8 Effect of mass flux on the HTC ($T_{Sat} = 10°C$)](image)
5.4. Effect of heat flux on heat transfer coefficient

In order to isolate the effect of heat flux on heat transfer coefficient experimental tests were performed by changing the heat flux \( (q'' = 15 - 61 \, \text{KW/m}^2) \) while keeping the mass flux constant. As shown in Figure 9 the heat transfer coefficient rises with the increase of the heat flux in the low vapor quality region. This can be explained by the low vapor quality region being boiling dominant hence the HTC is essentially related to the heat flux. Furthermore, at higher heat fluxes dryout is initiated at lower qualities. This is explained by considering that the higher heat flux promotes the evaporation of the liquid layer adjoining the wall, increasing the thermal resistance.

![Figure 9 Effect of Heat flux on the HTC (T_{Sat} = 10^\circ \text{C})](image)
6. Results and discussion on pressure drop

In this chapter the data for the pressure of propane in a copper tube with internal diameter of 4.2 mm are shown and analyzed to illustrate the effect of different parameters on the pressure drop characteristics of propane. This section is divided in five different parts; in the first section the general characteristics of the pressure drop of propane are explained; the subsequent sections deal with the effects of evaporation temperature, mass flux and heat flux.

6.1. General characteristics of pressure drop of propane

As a significant example, Figure 10 shows the data for pressure drop of propane at $T_{sat} = 5^\circ C, G = 379.41 \frac{kg}{s.m^2}, q'' = 31.88 \frac{kw}{m^2}$. Variation of the average quality was equal to $x = 0.17 - 1$.

The curve for pressure drop shows an increase from the lowest vapor quality up to vapor quality $x = 0.81$ which is corresponding to the point where dry out happens. Initiation of dry-out point can be either inferred from the heat transfer coefficient data or by visual confirmation from the glass sight placed at the end of the test section.
Chapter 6: Results and discussion on pressure drop

The trend of pressure drop curve can be explained accounting for the gas velocity variation. At low vapor quality the flow pattern seems to be intermittent or slug flow while at higher values it turns to annular flow. In the latter case, the liquid is in contact with the whole surface of the tube and since the liquid has a higher dynamic viscosity compared to the gas, pressure drop increases. As the evaporation goes on, the vapor velocity is higher and higher thus increasing the shear stress at the liquid-vapor interface. This increase in pressure drop continues until the dry-out occurs, meaning that there is no liquid film on the wall, and therefore the pressure drop decreases. It should be noted that across the dry out the decrease in pressure drop is not as drastic as the heat transfer coefficient drop.

6.2. Effect of saturation temperature on pressure drop

Figure 11 illustrates the effect of saturation temperature on pressure drop across the test section. In this study three different saturation temperatures were studied (0, 5, 10°C). The pressure drop is shown for two different sets of test conditions, \( G = 500 \frac{kg}{s \cdot m^2} \), \( q'' = 31 \frac{kw}{m^2} \) and \( G = 379.41 \frac{kg}{s \cdot m^2} \), \( q'' = 31.88 \frac{kw}{m^2} \). Unlike the case with heat transfer coefficient, pressure drop shows a strong dependence on saturation temperature. Actually, there is a significant increase in the pressure drop with lowering temperatures. The pressure drop across test section for saturation temperature of 0°C is about 30% greater compared to the case at saturation temperature of 10°C. This can be explained by the decrease in the viscosity of the gas phase \( \mu_g \), while liquid density \( \rho_l \) and liquid viscosity \( \mu_l \) increase; this causes in turn an increase in the superficial velocity of the gas phase and a higher shear stress which justifies the larger pressure drop.
6.3. Effect of mass flux on pressure drop

As a significant example for total pressure drop across the test section at $T_{sat} = 5^\circ C$ are illustrated in Figure 12 in a mass flux range of $G = 250 - 502 \frac{kg}{s \cdot m^2}$. As expected the results
show a strong dependency of pressure drop on mass flux. Total pressure drop is a combination of accelerational and frictional pressure drop.

6.4. Effect of heat flux on pressure drop

Figure 13 shows the data for pressure drop with saturation temperature of $T_{sat} = 5^\circ C$, and mass flux of $G \approx 250 \frac{kg}{m^2s}$ with varying the heat flux from $q'' = 15.6 - 62.44 \frac{kw}{m^2}$. Results show that with increasing heat flux the pressure drop increases for all the range of vapor qualities. In particular, compared to heat flux of $q'' = 15.6 \frac{kw}{m^2}$, a 4 times increase to $q'' = 62.44 \frac{kw}{m^2}$ causes about 20% increase in the pressure drop. The increase of the pressure drop across test section can be explained by the increased vaporization, causing a higher void fraction and a higher gas velocity. Higher vaporization can also change create bubbles leading to a slug flow further increasing the pressure drop.
Chapter 6: Results and discussion on pressure drop

Total pressure drop \([\text{kPa/m}]\)

\[x \text{ [-]}\]

\[\begin{align*}
G = 250.16 \text{ Kg/m}^2 \text{s}, q^\prime = 15.1 \text{ KW/m}^2 \\
G = 255.30 \text{ Kg/m}^2 \text{s}, q^\prime = 31.53 \text{ KW/m}^2 \\
G = 251.47 \text{ Kg/m}^2 \text{s}, q^\prime = 62.44 \text{ KW/m}^2
\end{align*}\]

Figure 13 Effect of heat flux on pressure drop \((T_{\text{sat}} = 5^\circ\text{C})\)
7. Comparison of correlations with test data

In this chapter a selection of correlations from the literature discussed in sections 2.4.2 and 2.4.3 are tested against the experimental data.

7.1. Correlations for frictional pressure drop

The selected correlations are (Muller-Steinhagen & Heck, 1986), (Sun & Mishima, 2008), (Cavallini et al., 2002), (Tran, Chyu, Wambega, & France, 2000), (Pamitran et al., 2010).

To make the two data sets comparable, the accelerational pressure drop is subtracted from the total pressure drop since the correlations give the estimates of the frictional pressure drop. Obviously since the tube is horizontal the gravitational pressure drop is equal to zero.

Table 9 shows the prediction performance for correlations of (Muller-Steinhagen & Heck, 1986), (Sun & Mishima, 2008) and (Cavallini et al., 2002). The errors are defined in section 2.4.2. Results of the correlations of (Tran et al., 2000), (Pamitran et al., 2010) is not shown since the value of MARD exceeded 100%.

<table>
<thead>
<tr>
<th></th>
<th>MRD (%)</th>
<th>MARD (%)</th>
<th>λ30%</th>
</tr>
</thead>
<tbody>
<tr>
<td>(Muller-Steinhagen &amp; Heck, 1986)</td>
<td>32.4</td>
<td>36.4</td>
<td>66.0</td>
</tr>
<tr>
<td>(Sun &amp; Mishima, 2008)</td>
<td>5.6</td>
<td>28.7</td>
<td>83.9</td>
</tr>
<tr>
<td>(Cavallini et al., 2002)</td>
<td>56.7</td>
<td>61.3</td>
<td>58.4</td>
</tr>
</tbody>
</table>

Table 9 Overview of the predicted pressure drops errors

Table 9 shows that the correlation by (Sun & Mishima, 2008) is the most reliable able to predict about 84% of data points within 30% relative error. In any case all correlations tend to over predict the pressure drop as all MRD values are larger than zero.
Figure 14 illustrates the behavior of predicted and measured frictional pressure gradient as a function of the average quality.

![Figure 14 Comparison of predicted and measured values for frictional pressure drop.](image)

*Test condition* $T_{\text{sat}} = 10^\circ C, G = 255 \frac{kg}{m^2 s}, q'' = 15.55 \frac{KW}{m^2}$

This diagram further shows that the correlation by *Sun & Mishima, 2008* follows the trend of experimental data more closely. In this set of data there is a large increase in the amount of predicted pressure drop by the correlation of *Sun & Mishima, 2008* after a vapor quality of about 0.7, such a large increase is due to the change of the formulation of the model due to the change in the flow regime; in this case the transition from laminar to turbulent occurs just near vapor quality of 0.7 and causes the inconsistency in the trend of predicted values of frictional pressure drop.

The correlation of *Muller-Steinhagen & Heck, 1986* overpredicts the pressure drop in all the tested range. This increase gets larger with the vapor quality and especially close to dry-out region. This is to be expected since none of the selected correlations accounts for the cry-out onset.

On the other hand, the correlation of *Cavallini et al., 2002* closely follows the trend of *Muller-Steinhagen & Heck, 1986* correlation with a further deviation at high vapor quality.
Figure 15, Figure 16 and Figure 17 show the parity plot between all of the predicted points by the aforementioned correlations and the experimental data for the frictional pressure drop.

Figure 15 Parity Plot for pressure drop prediction by (Sun & Mishima, 2008)
Chapter 7: Comparison of correlations with test data

Figure 16 Parity Plot for pressure drop prediction by (Muller-Steinhagen & Heck, 1986)

Figure 17 Parity Plot for pressure drop prediction by (Cavallini et al., 2002)
Chapter 7: Comparison of correlations with test data

The parity plot illustrates that all the correlations tend to overpredict the frictional pressure drop, especially for the higher saturation temperatures as they do not account properly for the effect of saturation temperature.

Table 10 illustrates the effect of saturation temperature on the accuracy of the predicted frictional pressure drop by the correlation of (Sun & Mishima, 2008).

<table>
<thead>
<tr>
<th>$T_{\text{sat}}$</th>
<th>10°C</th>
<th>5°C</th>
<th>0°C</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>MRD</td>
<td>20.6</td>
<td>2.6</td>
<td>-16.2</td>
<td>5.6</td>
</tr>
<tr>
<td>MARD</td>
<td>34.5</td>
<td>25.0</td>
<td>24.6</td>
<td>28.7</td>
</tr>
<tr>
<td>$\lambda_{30%}$</td>
<td>76.1</td>
<td>85.3</td>
<td>95.6</td>
<td>83.9</td>
</tr>
</tbody>
</table>

Table 10 Saturation temperature’s effect on frictional pressure drop prediction by (Sun & Mishima, 2008)

7.2. Correlations for heat transfer coefficient

The selected correlations are (Choi et al., 2007), (Liu & Winterton, 1991), (S. G. Kandlikar, 1990), (Tran et al., 1996), (Gungor & Winterton, 1986) and (Shah, 1982).

Table 11 shows the predictive performance in terms of the criteria defined in chapter 2.4.2. The only correlation with a MRD less than 100% is (Shah, 1982). All the correlation tend to greatly overpredict the heat transfer coefficient. The correlations by (Choi et al., 2007) and (Tran et al., 1996) are not included in the calculations since the error values were too large. Furthermore, there doesn’t seem to be a connection between the accuracy of the results and the saturation temperature as it was observed in the pressure drop correlations. Therefore, the correlations used in this study are unreliable to calculate the heat transfer coefficient.

<table>
<thead>
<tr>
<th></th>
<th>MRD (%)</th>
<th>MARD (%)</th>
<th>$\lambda_{30%}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>(Shah, 1982)</td>
<td>99.7</td>
<td>100.0</td>
<td>6.6</td>
</tr>
<tr>
<td>(Liu &amp; Winterton, 1991)</td>
<td>188.6</td>
<td>188.6</td>
<td>0</td>
</tr>
<tr>
<td>(S. G. Kandlikar, 1990)</td>
<td>137.9</td>
<td>137.9</td>
<td>0.9</td>
</tr>
<tr>
<td>(Gungor &amp; Winterton, 1986)</td>
<td>137.6</td>
<td>137.6</td>
<td>1.8</td>
</tr>
</tbody>
</table>

Table 11 overview of the predicted HTC by correlations

The inability of the correlations to predict the heat transfer coefficient might be explained as follows:
7- In the data base used to develop the correlations propane was not included though the effect of a different fluid is considered in some equations by the reduced pressure and molar mass.

8- The effect of the small diameter, included by using either the Weber or the Laplace number is not always accounted for.

9- There is a general difficulty with the prediction of heat transfer coefficient by mechanistic models. A combination of a new fluid and a smaller tube diameter can make a correlation that was not explicitly developed for these conditions to be completely unsuitable.
8. Conclusions

Two-phase pressure drop and heat transfer characteristics during flow boiling of propane in different experimental conditions were discussed in this study. Test conditions were $G = 250 - 505 \frac{kg}{m^2 s}$, $T_{\text{sat}} = 0, 5, 10 \degree C$, $q'' = 15 - 62 \frac{kW}{m^2}$ corresponding to vapor quality variation $\Delta x = 0.1 - 0.4$. A smooth copper test tubes with $D_t = 4.2 \text{ mm}$ was used, test section length was equal to $0.5 \text{ m}$. Mean vapor quality between 0.2 and 1.0. Several existing pressure drop and heat transfer coefficient prediction methods were reviewed. From the analysis of the experimental data and the comparison with the literature findings the following conclusions can be drawn:

5- The heat transfer coefficient increases with the vapor quality till the dryout onset, occurring at average quality higher than 0.7 which increases as the mass flux is decrease at constant heat flux, highest observed heat transfer coefficient was $4.25 \frac{kW}{m^2k}$.

6- An increase in heat flux and mass flux causes an increase of the heat transfer coefficient. The influence of the heat flux is more prominent at low vapor quality where the heat transfer regime is boiling dominated while the effect of the mass flux is more significant at high vapor quality where the heat transfer regime is mainly convection dominated. Saturation temperature does not seem to have a tangible effect on the heat transfer coefficient.

7- Pressure drop increases with vapor quality, but after the dryout, the trend is reversed.

8- Both mass flux and heat flux increase the pressure drop in the whole range of tested vapor quality. However, as expected the effect of heat flux is relatively minor compared to the influence of mass flux.

10- Saturation temperature has a tangible effect on the pressure drop. The decrease of saturation temperature causes an increase in the pressure drop, explained by the increase of the vapor velocity.
11- The correlation of (Sun & Mishima, 2008) provides the most reliable prediction of the pressure drop, with a MRD 5.69% and is able to predict 83.96% of data points with less than 30% error. All the correlations tend to overpredict the pressure drop. Furthermore, none of them is able to account for the effect of saturation temperature in a meaningful way.

12- None of the correlations used for prediction of the heat transfer coefficient is reliable, all of them greatly overestimate the heat transfer coefficient.
9. Proposal for further work

The following aspects should deserve attention for future investigation.

1- Flow visualization using a high-speed camera and a full glass sight. In order to draw an accurate flow pattern map.

2- Testing tubes of lower diameter to assess the effect of confinement.

3- Studying the effect of microfinned surface on pressure drop and heat transfer coefficient.

4- Extend the study to other hydrocarbons and mixtures.
List of Tables

Table 1 Void fraction correlations.................................................................12
Table 2 Summary of other correlations used for prediction of HTC................20
Table 3 Test conditions for (Wang et al., 2014)............................................21
Table 4 Review of the test conditions in previous studies .........................24
Table 5 List of the chillers used for condensation.......................................33
Table 6 Cooling capacity for Julabo FP50 Chiller....................................33
Table 7 Test Conditions............................................................................35
Table 8 List of instruments........................................................................38
Table 9 Overview of the predicted pressure drops errors..........................57
Table 10 Saturation temperature’s effect on frictional pressure drop prediction by (Sun & Mishima, 2008) ............................................................61
Table 11 overview of the predicted HTC by correlations .........................61
10. List of figures

Figure 1 Schematic design of test rig .................................................................30
Figure 2 Overview of test rig........................................................................31
Figure 3 Cooling capacity of main chillers ......................................................34
Figure 4 LabView Interface developed for Control System and Data Acquisition.....37
Figure 5 Overview of test section inlet............................................................39
Figure 6 Top: Average HTC; Bottom: HTC separately for inlet and outlet........46
Figure 7 HTC for different saturation temperatures........................................48
Figure 8 Effect of mass flux on the HTC ($T_{Sat} = 10^\circ C$) .........................49
Figure 9 Effect of Heat flux on the HTC ($T_{Sat} = 10^\circ C$)..............................50
Figure 10 General pressure drop characteristics............................................51
Figure 11 Effect of saturation temperature on pressure drop .........................53
Figure 12 Effect of mass flux on pressure drop ($T_{sat} = 5^\circ C$).....................54
Figure 13 Effect of heat flux on pressure drop ($T_{sat} = 5^\circ C$).....................55
Figure 14 Comparison of predicted and measured values for frictional pressure drop. 58
Figure 15 Parity Plot for pressure drop prediction by (Sun & Mishima, 2008) ......59
Figure 16 Parity Plot for pressure drop prediction by (Muller-Steinhagen & Heck, 1986)..............................................................60
Figure 17 Parity Plot for pressure drop prediction by (Cavallini et al., 2002).....60
# 11. Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_p$</td>
<td>Specific heat (KJ Kg$^{-1}$ K$^{-1}$)</td>
</tr>
<tr>
<td>$d$</td>
<td>Diameter (m)</td>
</tr>
<tr>
<td>$f$</td>
<td>Darcy Friction factor (-)</td>
</tr>
<tr>
<td>$F$</td>
<td>Chen parameter for convection enhancement (-)</td>
</tr>
<tr>
<td>$g$</td>
<td>gravity (m s$^{-2}$)</td>
</tr>
<tr>
<td>$G$</td>
<td>Mass Flux (kg m$^{-2}$ s$^{-1}$)</td>
</tr>
<tr>
<td>$h$</td>
<td>Heat transfer coefficient (Kw m$^{-2}$ K$^{-1}$)</td>
</tr>
<tr>
<td>$i_g$</td>
<td>Latent heat enthalpy (J Kg$^{-1}$)</td>
</tr>
<tr>
<td>$J$</td>
<td>Superficial velocity (m s$^{-1}$)</td>
</tr>
<tr>
<td>$K$</td>
<td>Thermal conductivity (Kw m$^{-1}$ K$^{-1}$)</td>
</tr>
<tr>
<td>$M$</td>
<td>Molecular weight (g mol$^{-1}$)</td>
</tr>
<tr>
<td>$m$</td>
<td>Mass flow rate (kg s$^{-1}$)</td>
</tr>
<tr>
<td>$P$</td>
<td>Pressure (KPa)</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl Number (-)</td>
</tr>
<tr>
<td>$Q$</td>
<td>Power input (Kw)</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds Number (-)</td>
</tr>
<tr>
<td>$S$</td>
<td>Heat Transfer Surface Area (m$^2$)</td>
</tr>
<tr>
<td>$S_s$</td>
<td>Chen parameter for suppression (-)</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature (°C)</td>
</tr>
<tr>
<td>$V$</td>
<td>Volume flow rate (m$^3$ s$^{-1}$)</td>
</tr>
<tr>
<td>$We$</td>
<td>Weber number</td>
</tr>
<tr>
<td>$x$</td>
<td>Average vapor quality (-)</td>
</tr>
<tr>
<td>$X$</td>
<td>Lockhart and Matinelli Parameter (-)</td>
</tr>
<tr>
<td>$Y$</td>
<td>Muller-Steinhagen &amp; Heck Parameter(-)</td>
</tr>
<tr>
<td>$Z$</td>
<td>Length of test section (m)</td>
</tr>
</tbody>
</table>

**Greek letters**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$</td>
<td>Void Fraction (-)</td>
</tr>
<tr>
<td>$\delta$</td>
<td>Error, Uncertainty (-)</td>
</tr>
<tr>
<td>$\nu$</td>
<td>Specific volume (m$^3$ kg$^{-1}$)</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Dynamic Viscosity (Pa.s)</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density (Kg m$^3$)</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>Surface tension (N m$^{-1}$)</td>
</tr>
<tr>
<td>$\phi$</td>
<td>Two-phase multiplier (-)</td>
</tr>
<tr>
<td>$\Omega$</td>
<td>Flow Surface Area (m$^2$)</td>
</tr>
</tbody>
</table>
Subscripts

Acc: Accelerational
Atm: Atmosphere
aux: auxiliary flow
fric: frictional
i: inside
in: inlet
Grav: gravitational
g: gas
go: gas only
lm: log mean
l: liquid
lo: liquid only
o: outside
Pre: preheating section
R: reduced
Sub: Sub cooling
Sat: saturation
TP: two phase
W: wall
12. References


pressure drop for evaporation and condensation of R134A in microfin tubes.  
https://doi.org/10.1016/j.ijrefrig.2012.08.019

https://www.bibsonomy.org/bibtex/23dd63148f7401039d79115650c4fb2d3/thorade


http://adsabs.harvard.edu/abs/1975STIA...7522028G


https://doi.org/10.1016/j.ijrefrig.2012.07.016

https://doi.org/10.1016/j.ijrefrig.2011.03.001


https://doi.org/10.1016/j.ijrefrig.2009.12.009


