Master Thesis

Study on bubble growth rate in a single microchannel heat exchanger with high-speed CMOS-camera

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STUDY ON BUBBLE GROWTH RATE IN A SINGLE MICROCHANNEL HEAT EXCHANGER WITH HIGH-SPEED CMOS-CAMERA

by
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Master Thesis at
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Zurich, April 2007
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3 SYMBOLES AND ABBREVIATIONS

3.1 GREEK LETTERS

$\alpha$ \hspace{0.5cm} $[m^2/s]$ \hspace{0.5cm} Temperature conductivity

$\delta$ \hspace{0.5cm} $[\mu m]$ \hspace{0.5cm} Wall boundary layer

$\eta$ \hspace{0.5cm} [-] \hspace{0.5cm} Fin efficiency

$\mu$ \hspace{0.5cm} $[Ns/m^2]$ \hspace{0.5cm} Dynamic viscosity

$\nu$ \hspace{0.5cm} $[m^2/s]$ \hspace{0.5cm} Kinematic viscosity

$\theta$ \hspace{0.5cm} [$^\circ$] \hspace{0.5cm} Contact angle at nucleation site

$\rho$ \hspace{0.5cm} $[kg/m^3]$ \hspace{0.5cm} Density

$\sigma$ \hspace{0.5cm} $[N/m]$ \hspace{0.5cm} Surface tension

$\Delta$ \hspace{0.5cm} [-] \hspace{0.5cm} Difference, Delta

3.2 LATIN LETTERS

$a$ \hspace{0.5cm} $[\mu m]$ \hspace{0.5cm} Half channel width

$b$ \hspace{0.5cm} $[\mu m]$ \hspace{0.5cm} Half channel depth

$A$ \hspace{0.5cm} $[\mu m^2]$ \hspace{0.5cm} Visible cross sectional area of bubble

$C$ \hspace{0.5cm} [-] \hspace{0.5cm} Constant

$c_p$ \hspace{0.5cm} $[J/kgK]$ \hspace{0.5cm} Specific heat capacity

$\frac{dr}{dt}$ \hspace{0.5cm} $[\mu m/s]$ \hspace{0.5cm} Bubble radius growth rate

$\frac{dp}{dx}, \frac{dp}{dz}$ \hspace{0.5cm} $[kPa/m]$ \hspace{0.5cm} Pressure drop across channel

$D$ \hspace{0.5cm} $[\mu m]$ \hspace{0.5cm} Diameter

$G$ \hspace{0.5cm} $[kg/m^2s]$ \hspace{0.5cm} Mass flux

$H$ \hspace{0.5cm} $[\mu m]$ \hspace{0.5cm} Height

$h$ \hspace{0.5cm} $[W/m^2K]$ \hspace{0.5cm} Heat transfer coefficient

$Ja$ \hspace{0.5cm} [-] \hspace{0.5cm} Jacob number

$k$ \hspace{0.5cm} $[W/Km]$ \hspace{0.5cm} Thermal conductivity
### 3. Symbols and Abbreviations

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L$</td>
<td>[µm]</td>
<td>Length</td>
</tr>
<tr>
<td>$M$</td>
<td>[-]</td>
<td>Number of data points</td>
</tr>
<tr>
<td>$m$</td>
<td>[-]</td>
<td>Fin parameter</td>
</tr>
<tr>
<td>$MAE$</td>
<td>[%]</td>
<td>Maximum absolute error</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>[kg/s]</td>
<td>Mass flow rate</td>
</tr>
<tr>
<td>$n$</td>
<td>[-]</td>
<td>Number of parallel micro-channels</td>
</tr>
<tr>
<td>$p$</td>
<td>[kPa]</td>
<td>Pressure</td>
</tr>
<tr>
<td>$P$</td>
<td>[W], [µm]</td>
<td>Electrical power, perimeter</td>
</tr>
<tr>
<td>$Pr$</td>
<td>[-]</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>$Q$</td>
<td>[W]</td>
<td>Heating power</td>
</tr>
<tr>
<td>$q$</td>
<td>[W/m²]</td>
<td>Heat flux per area</td>
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<tr>
<td>$r$</td>
<td>[µm]</td>
<td>Radius</td>
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<tr>
<td>$R$</td>
<td>[µm]</td>
<td>Bubble departure diameter</td>
</tr>
<tr>
<td>$R(t)$</td>
<td>[µm]</td>
<td>Bubble radius as a function of time</td>
</tr>
<tr>
<td>$\bar{R}$</td>
<td>[J/kgK]</td>
<td>Specific gas constant</td>
</tr>
<tr>
<td>$Re$</td>
<td>[-]</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>$t$</td>
<td>[s]</td>
<td>Time</td>
</tr>
<tr>
<td>$T$</td>
<td>[K]</td>
<td>Temperature</td>
</tr>
<tr>
<td>$\bar{u}$</td>
<td>[m/s]</td>
<td>Cross sectional area averaged flow velocity</td>
</tr>
<tr>
<td>$V$</td>
<td>[µm³], [V]</td>
<td>Volume or voltage</td>
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<tr>
<td>$\dot{V}$</td>
<td>[m³/s]</td>
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<tr>
<td>$\bar{V}$</td>
<td>[V]</td>
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<tr>
<td>$W$</td>
<td>[µm]</td>
<td>Width</td>
</tr>
<tr>
<td>$x$</td>
<td>[-]</td>
<td>Vapour quality</td>
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<th>Description</th>
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<tbody>
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<td>$b$</td>
<td>Bulk value or bubble</td>
</tr>
<tr>
<td>$BD$</td>
<td>Value at bubble departure</td>
</tr>
<tr>
<td>$c$</td>
<td>Nucleation site</td>
</tr>
<tr>
<td>$calibrate$</td>
<td>Measured value during calibration</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
</tr>
<tr>
<td>ch</td>
<td>Channel</td>
</tr>
<tr>
<td>CJC</td>
<td>Cold-junction-compensation</td>
</tr>
<tr>
<td>crit</td>
<td>Critical value</td>
</tr>
<tr>
<td>D</td>
<td>Value at bubble departure</td>
</tr>
<tr>
<td>e</td>
<td>Excess value or entrance region</td>
</tr>
<tr>
<td>eff</td>
<td>Net effective value</td>
</tr>
<tr>
<td>exp</td>
<td>Experimental value</td>
</tr>
<tr>
<td>Fluid</td>
<td>Absorbed by fluid</td>
</tr>
<tr>
<td>h</td>
<td>Hydraulic</td>
</tr>
<tr>
<td>Heater</td>
<td>Delivered by heater</td>
</tr>
<tr>
<td>in</td>
<td>Entrance of channel</td>
</tr>
<tr>
<td>l</td>
<td>Liquid</td>
</tr>
<tr>
<td>loss</td>
<td>Lost portion to surrounding</td>
</tr>
<tr>
<td>NIST</td>
<td>Value from NIST table</td>
</tr>
<tr>
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<td>Regarding shear stress</td>
</tr>
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<td>p</td>
<td>Regarding pressure</td>
</tr>
<tr>
<td>pred</td>
<td>Predicted value</td>
</tr>
<tr>
<td>r</td>
<td>At maximum nucleation site radius</td>
</tr>
<tr>
<td>s</td>
<td>Per surface</td>
</tr>
<tr>
<td>ρ</td>
<td>Regarding inertia</td>
</tr>
<tr>
<td>sat</td>
<td>Saturation value</td>
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<tr>
<td>sub</td>
<td>Subcooled value</td>
</tr>
<tr>
<td>σ</td>
<td>Regarding surface tension</td>
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<td>t</td>
<td>Thermal</td>
</tr>
<tr>
<td>tot</td>
<td>Total</td>
</tr>
<tr>
<td>v</td>
<td>Vapour</td>
</tr>
<tr>
<td>w</td>
<td>Wall</td>
</tr>
<tr>
<td>x</td>
<td>In channel direction</td>
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4 ABSTRACT

The present work investigates bubble dynamics in a flow boiling micro-channel heat exchanger with a hydraulic diameter of 68µm. The 50mm long channel is 44µm deep and 155µm wide and machined in an aluminium block. The block was equipped with a cartridge heater and k-type thermocouples to measure six wall temperatures and three fluid temperatures. De-ionized and degassed water was used with mass fluxes varying between 143.4 and 793.3kg/m²s. The electrical power input was varied between 188 and 272.9W respectively which translates into a heat input of 0.432W to 0.628W. Boiling curves and pressure dependence of temperature were recorded. Temperature and pressure measurements were synchronized with pictures of nucleating bubbles which were taken with a high speed camera with up to 67'796 frames per second. The bubble growth rate and the departure diameter were investigated for different heat inputs and mass fluxes. The experiments did not show a very clear trend regarding bubble growth rate. The values varied between 10.2µm/s and as much as 32'588µm/s. The test section was very sensitive regarding wear. Small geometrical changes had a significant impact on the boiling characteristics. Surprisingly the saturation temperature was 100°C in the experiments even though the pressure drop across the channel was two bars. The repeatability of certain experiments seemed also to be a problem. However, it was encountered that the bubble departure diameter followed an exponential trend as reported in literature.
5 INTRODUCTION

Over the last fifty years the integrated circuit industry has from infancy grown to become one of the largest industries. During the last 37 years the amount of transistors in microchips has doubled every 24 months. Such a development was first predicted by Gordon Moore, a cofounder of Intel cooperation who stated in 1967 that “The number of transistors incorporated in a chip will approximately double every 24 months.” [42]. Albeit originally cited in 1967, Moore’s appears valid until today. An integrated circuit in 1967 had less than 10’000 transistors, a modern chip today carries more than one billion transistors, shown in Fig. 5.1.

![Fig. 5.1 Prediction and actual numbers of transistors in integrated circuits by Moore [43]](image)

One predominant reason for this ongoing development relates to new production techniques that dramatically shrink the size of transistors. An empirical relation states that by doubling the transistor density the line width must be reduced by a factor of 0.7 [40]. From past dimensions of 130nm in late 60’s, measurements have been reduced to around 30nm. In addition, new fabrication methods and new fields of applications for transistors decreased production costs, increased demand and thereby lowered prices even more substantially. The cost of a transistor decreased from $ 1.00 in 1967 to less than $ 10^{-6} in 2002 [43]. The growing number of transistors in integrated circuits increased calculation speed from less than 0.1 MIPS (Million instructions per second) in 1972 to more than 10’000 MIPS in 2002 [43]. With improved performance comes an increasing demand for power. Although power supply voltage decreased from almost 20V in the year 1970 to about 1.2V in the year 2005 the overall power supply increased significantly. Fig. 5.2 shows the input power in Watts. As evidenced, the dissipated heat during operation also increased tremendously – reaching up to 130W with the newest
QuadCore generation from INTEL. Similar trends are also stated by the International Technology Roadmap for Semiconductors (ITRS) [23].

Consequently, the need for highly efficient and silently operating cooling techniques for processors becomes more serious as the surface temperature of current chips must not exceed 80-100°C for proper functionality and reliability. Without paying a big penalty in energy consumption, noise and increased size, the typical fan-fin-cooling method is not able to provide enough cooling power for actual processors anymore.

A new cooling method that has already been implemented is the heat pipe. This device is a double walled pipe which carries a liquid with a low saturation temperature. By heating up one end of the heat pipe the working fluid in the tube evaporates and diffuses to the cold end where it is condensed with a conventional heat sink mechanism. The liquid fluid runs down to the hot end of the heat pipe and a continuously working cooling circuit is maintained. However, the passive cooling of the heat pipe is quite limited as the pumping of working fluid relies on capillary forces. The size of the whole system is still in the scale order of magnitude of conventional fan-based cooling systems.

Another idea, which was proposed by Tuckerman and Pease [63] in 1981, is to use a single phase liquid heat exchanger attached to the chip. This technique encourages high efficiency due to the fluid dependent heat capacity. The hot working fluid is then conventionally cooled by fans away from the chip. To provide a large surface to reduce the thermal resistance from the heat exchanger to the fluid the design of the actual heat exchanger on top of the chip is very demanding. On the one hand, a large surface area between the working fluid and the chip is favourable; on the other, one does not want an unnecessary increase in pressure drop. Even though this technique may be useful in removing higher heat fluxes, the size of the system presents a considerable disadvantage, especially for portable devices wherein space is limited. It is noteworthy that in all portable applications the failure to achieve a large enough heat rejection surface amounts to a profound stumbling block. To maintain a high enough heat flux
away from the device by not exceeding a threshold wall temperature is crucial and
determined by the available size of the rejection surface. Nevertheless, a microchannel
heat exchanger seems to be the most promising idea for a cooling device involving
high-power chips in the future [39]. It is reported that cooling power of 7.9 MW/m²
with a maximum substrate temperature to water inlet temperature difference of 71°C
was achieved [63]. But the pressure drop was quite large with 380kPa with pin fin-
enhanced channels and 200kPa with plain channels.

The newest suggestion is based on a two-phase microchannel heat exchanger which is
directly attached to the back side of the microprocessor while a coolant is pumped
through the channels [25]. The capability to shrink these devices down with the help of
MEMS technology creates the opportunity to integrate the heat exchanger on the scale
of the transistors. Through the phase change phenomena one can make use of the latent
heat which makes this technique very interesting in terms of great heat removal. The
heat transfer coefficients (HTC) can be very high as the ratio of heat transfer surface to
unit of volume flow rate is very good due to the small dimensions. Furthermore, as the
latent heat is also used for removing dissipation heat, the mass flow rates can be
reduced compared to cooling devices that only make use of the sensible heat. Another
advantage is that the temperature of the working fluid stays the same at the boiling point
along the channel surface. Single-phase heat exchangers differ in that the liquid
temperature increases along the channel and the HTC decreases. The recent
development on electro-osmotic pumps allows a miniature closed-loop cooling system
[7]. The typical hydraulic diameters for micro-channels range from 10 to 200 microns
[25]. Complications arise with the formation of vapour bubbles in the channel which
lead to an increased pressure drop, occasional flow reversals and, due to instabilities, to
low critical heat flux values. Especially for multi-channel devices there occur periodic
fluctuations regarding pressure drop and wall temperature which can result in a critical
heat transfer coefficient. To prevent a burnout and to maintain a continuously cooling
power this value must not be exceeded.

Extensive research has been done to further understanding of the nucleation of vapour
bubbles on the small scale. Researchers have examined those instabilities in multi
channel devices which emerge due to sudden expansion of the forming vapour, see
Flynn et al. [14]. The difficulty here lies in the maintenance of a steady flow without a
big increase in pressure drop. Flynn et al. studied thermally insulated and thermally
coupled parallel microchannels and the resulting fluctuations in the mass flux through
the channels whether there is two-phase or single-phase flow in each of the channels.

The increased pressure drop due to rapid increase in specific volume for the vapour is
addressed in the work of Zhang et al. [68] and many others. Chen et al. [5] focused on
the critical heat transfer coefficients in dependence of the flow pattern. In addition there
have been several attempts at coming up with theoretical models that describe bubble
growth rate and bubble departure time. However, there is very little research that deals
with the nucleation and the growth rate of bubbles in single microchannels in
combination with a visualization tool that might equip one to actually witness the formation of the bubbles. This deficiency seems particularly surprising given first the inadequate comprehension of the nucleation phenomenon in the microchannels, and second, the reality that most models are based on empirical data predictive of bubble behaviour for a single configuration, data that may contradict the findings of other researchers.

In my work, I will address the formation of vapour and gas bubbles in a microchannel heat exchanger with a hydraulic diameter of about 68 µm. The flow rates that were applied varied between 143.3 kg/m²sec and 793.3 kg/m²sec. The electrical power input was varied from 188.0 to 272.9 Watts which translates into a heat input of 0.432 up to 0.628W after consideration of heat losses. The visualization of the bubbles was done with a high speed camera in combination with a microscope.

Due to the many difficulties that were encountered during the design of the experiment and the experiment itself in the micro scale, this work focuses on the experimental set-up and the iteration to achieve satisfying and repeatable experiments. Despite setbacks, the knowledge accrued by this study will be of great value to the Stanford Thermosciences research group. A current PhD student will benefit from my work as he plans to perform similar experiments and can make use of the experiences I had during my time at Stanford.
6 LITERATURE RESEARCH

6.1 DIFFERENT TYPES OF MICRO-CHANNEL HEAT EXCHANGERS

There are three different configurations of heat exchangers in literature. They differ in size of the channel, the number of channels (either parallel single- or multi-channel devices) and the material they are made of (usually silicon but also some are made of copper).

There are several different methods to classify the heat exchanger geometry. Although the distinction is always based on the hydraulic diameter of the device, the different categories of hydraulic diameters are determined by different approaches. The continuum physics still hold in micro-channel down to a few nanometers. This is only true for liquid single-phase flow where inertia is the governing force and surface tension is negligible. However, in two-phase flow the size matters much more. In macro-channels the nucleation and bubble growth is inertia-controlled, usually the bubble departs before it is constraint be the channel walls. This is fundamentally different in micro-channel heat exchangers. As the channel size is of the order of magnitude of a couple micrometers, the bubbles are strongly influenced by surface tensions and not by inertia. In addition the bubble is geometrically constrained by the channel walls, which even might suppress nucleation and bubble growth completely for extremely small channels of 10 to 15µm. Many classifications of heat exchangers are based on scaling analysis for forces. Depending on which force is the governing one, the category of heat exchangers is defined. Other researchers propose a classification based on empirical studies. They record the trend of some significant parameters like flow patterns or the heat transfer coefficient, and base their size wise classification of heat exchangers on significant changes in these trends.

Kandlikar et al. [25] as well as Mehendale et al. [39] propose the following classification of different heat exchangers, based on empirical studies

- Conventional channels $D_h > 3\text{mm}$
- Minichannels $200\mu\text{m} < D_h < 3\text{mm}$
- Microchannels $10\mu\text{m} < D_h < 200\mu\text{m}$
6.2 Governing Parameters

6.2.1 Pressure Drop in Single- and Multi-Channel Devices

The big difference between single- or multi-channel devices is that a multi-channel device undergoes severe pressure, mass flux and wall temperature fluctuations. Although there is pressure fluctuations in single-channel devices it seems to be larger in multi-channel devices. The averaged pressure drop in a multi-channel device as described by Zhang et al. [68] or a multi-channel device as described by Koo et al. [31] for a fixed flow rate and different heat input look the same.

![Fig. 6.1 Pressure drop in a 40-channel device [68]](image1)

![Fig. 6.2 Pressure drop in a single channel device [31]](image2)

It is characteristic that pressure drop decreases for higher heat inputs as viscosity of the fluid goes down. After the onset of boiling however pressure drop increases suddenly as vapour is formed in the channel. Due to the much lower density of vapour (about a factor of 1000 for water) and as the mass flux has to be conserved the mixture accelerates which yields a higher pressure drop. A qualitative analysis is provided by Flynn et al. [14]. He fixes the heat input and varies the mass flow rate. This is useful as soon as it comes to multi-channel as different channels usually see different mass fluxes due to redirection of the flow. In this case some channels start boiling due to lowered mass flux and other channels are still at single-phase flow.

![Fig. 6.3 Pressure drop for fixed heat input and varying mass flow rate [14]](image3)

If we take a closer look at the pressure drop in a multi-channel device without averaging out all the fluctuations we get a graph like shown in Fig. 6.4 from Chen et al. [63]. Even
this data is from a multi-channel device with a hydraulic diameter of 389µm similar behaviour was reported by Wu et al. [66] with a \(D_h=158.8\mu m\) and \(D_h=82.8\mu m\). The severe pressure fluctuation is a typical multi-channel phenomenon. It is created by channel interaction of channels with different flow patterns. One channel is already boiling which increases pressure drop there. That means flow is redirected to another channel which is still single flow. So boiling does even increase more in the one channel and reversal flow may occur as the vapour expands.

![Fig. 6.4 Pressure fluctuation in a 24-channel device with \(D_h=389\mu m\) [63]](image)

But also in single channel devices there is a pressure fluctuation which is linked to the formation of bubbles. Huh et al. [21] presented data that shows a pressure fluctuation with longer period compared to Chen [63]. Chen’s frequency is of the order of milliseconds whereas Wu and Huh report frequencies in the scale of seconds or hundreds of seconds as shown in Fig. 6.5.

![Fig. 6.5 Pressure fluctuations in a single-channel device with hydraulic diameter of 103.5µm [21]](image)

Other studies regarding pressure drop in single-channel devices were done by Lazarek and Black [33]. They evaluated the three components of pressure drop. Moryame and Inoue [44] measured pressure drop of R-113 boiling in narrow annular gaps of 35-110µm. Tong et al. [59] presented a broad study of pressure drop during sub cooled flow boiling in minichannels.
From the studies that are available in literature, the effect of the channel dimensions on the pressure drop of a two-phase mixture is not clearly established. Even though several authors provide pressure drop models for micro-channels they do not provide clear indication of the effects of small channels on the pressure drop. Some authors also take into account the wall roughness of the channel wall. This effect will become more important for smaller channel sizes.

### 6.2.2 Flow Pattern

The first flow pattern maps were developed for petrochemical industry for flow of oil and gas in large diameter pipes [1]. Later more and more adiabatic flow pattern maps were developed for example Hewitt et al. [19] and Taitel et al. [57]. Later on researchers focused on smaller hydraulic diameters. The survey by Fukano et al. [17] is representative in this sense and shows bubbly, slug and annular flow patterns. Triplet et al. [61] state that due to the increasing effect of surface tension for smaller channel sizes stratified flow is essentially absent in smaller diameter channels but slug (plug) and churn flow can be achieved over a wide range of flow rates. Hewitt [19] notes that evaporation and condensation has a big influence on the flow pattern in small channels. The effects of evaporation in micro-channels are two folds. The forming vapor increases the pressure drop due to an acceleration of the mixture. This increase can be quite large for high heat fluxes. Secondly the bubble size has the same dimension as the channel diameter. In this case the surface tension plays an important role and it happens that bubbles block the whole channel before they depart. The presence of vapor instead of liquid has also a strong effect on the local heat transfer. In general the heat transfer coefficient decreases with increasing quality and burn out may happen.

The flow patterns that occur in flow boiling in microchannels are bubbly flow, slug or elongated bubble flow, liquid ring flow and liquid lump flow as shown by Feng and Serizawa [13] in a 50µm diameter channel in Fig. 6.6. They took their images after the mixing of water and injected air.

![Flow patterns in a 50µm diameter micro-channel with air injected in water: (a) bubbly flow, (b) slug or elongated bubble flow, (c) liquid ring flow and (d) liquid lump flow [13]](image-url)

Fig. 6.6

---

6 Literature Research
It should be mentioned that different researchers reported same flow patterns under different names. As there are no strict definitions one should pay attention which term of the different paper refer to which flow pattern. The idea that surface forces play a key role in small channels and therefore eliminate stratification of the flow is supported by experiments from Triplett et al. [62] who stated that in their investigation it never occurred. They used rather large channels with inner diameters from 1.09 to 1.49mm for gas-liquid flows. Tran et al. [60] observed that the transition form slug flow (elongated bubble flow) to annular flow occurred at much higher vapor qualities than typical for macro-channels. They reported vapor qualities of \( x = 0.6 - 0.7 \) instead of \( x = 0.25 - 0.35 \) for macro-channel flows. They used a circular channel with R-12 and a diameter of 2.46mm. In cases when stratified flow was observed the channel dimension were those of mini-channels rather than micro-channels. Kasza et al. [28] reported stratified flow in their experiments with water at very low mass flux of 21 kg/m²’s and a rectangular channel of 2.5 x 6.0 mm. But they also saw bubbly flow and slug flow. Cornwell and Kew [9], [10] identified isolated bubble flow, confined bubble flow and annular slug flow regimes. Also Sheng and Palm [54] (diameter ranges from 1.0-4.0mm) and Mertz et al. (1-3mm in diameter) reported for their evaporating water experiments single bubble flow, confined bubble flow and annular flow. Coleman and Garimella [6] presented a flow map for their air-water and R-134a experiments in round and rectangular tubes of 1-4mm in diameter. Their flow pattern images are shown in Fig. 6.7.

![Flow regimes](image)

**Fig. 6.7** Description of different flow regimes encountered by Coleman and Garimella [6]

The work of Coleman and Garimella [6] supports the theory that the effect of different hydraulic diameters is negligible for large diameter tubes above 10mm but it is not for tubes with smaller diameters. This is based on the larger effect of the surface tension in
smaller scales. To conclude, one can expect bubbly, elongated bubble, annular and mist flows in microchannel evaporators.

**6.2.3 NUCLEATION IN FLOW BOILING**

In micro-channel heat exchangers only nucleate boiling just below the critical heat flux is favorable. Not all of the pool boiling regimes described by Dhir [12] in Fig. 6.8 can be observed in flow boiling. One encounters mostly partial, fully developed nucleate boiling and force convection flow boiling. The onset of nucleate boiling (ONB) depends on the flow conditions, especially the pressure drop which determines the saturation temperature and the nucleation site activity and the density and activity of possible nucleation sites. As the boiling process is highly stochastic all given numbers in literature regarding wall superheat temperature are averaged numbers which may vary from one experiment to another experiment.

![Boiling regime map from V. K.Dhir](image)

*Fig. 6.8 Boiling regime map from V. K.Dhir [12]*

The heat is transferred from the heated surface to the liquid according to Newton’s law of cooling

\[
q^* = h(T_s - T_{sat}) = h \cdot \Delta T_c
\]

where \(\Delta T_c = T_s - T_{sat}\) is termed the excess temperature. This determines the heat transfer coefficient \(h\). The saturation temperature of water for a certain pressure can be obtained from tables. Nucleation criterions were formulated by several researchers. Hsu [20] describes the nucleation in conventional size channels for pool and flow boiling. When a bubble covers the mouth of a cavity, the surrounding liquid temperature dictates whether the bubble grows or not. The vapour pressure inside the bubble must exceed the liquid pressure of the water so that the bubble nucleates and grows. As the equilibrium
pressure inside a bubble increases as the radius reduces, a greater wall superheat temperature is essential for nucleation over smaller cavities. The excess pressure inside the bubble is given by the following equation for static equilibrium of surface tension and pressure forces

\[ p_v - p_l = 2\sigma/r_b \]  

(6.2)

where \( p_v \) is the pressure inside the bubble, \( p_l \) the pressure of the surrounding liquid, \( \sigma \) the surface tension and \( r_b \) the bubble radius. If the bubble grows further depends on the lowest temperature at the bubble-liquid interface. If this temperature exceeds the saturation temperature for the corresponding vapour pressure inside the bubble, the bubble continues to grow. The nucleation over a cavity is shown in Fig. 6.9.

![Diagram of nucleation process](image)

Fig. 6.9  Left: temperature profile, right: stagnation streamline around a nucleating bubble [26]

Assuming that that the solid wall is the heated surface the lowest temperature is expected at \( y = y_b \) which is the nucleation criterion. The contact angle \( \theta_r \) was chosen differently by other investigators which lead to different expressions. Hsu [20] chose \( \theta_r = 53.1^\circ \) whereas Bergles and Rohsenow [2] used \( 90^\circ \) and Davis and Anderson [11] kept the contact angle as a variable in their expression for radii that fulfilled the nucleation criterion. Kandlikar et al. [26] came up with a critical nucleation size where bubble nucleation will start first. This radius is given by equation (6.3)

\[ r_{c, crit} = \frac{\delta_r \sin \theta_r \left( \frac{\Delta T_{sat}}{\Delta T_{sat} + \Delta T_{sub}} \right)}{2.2} \]  

(6.3)

where \( \Delta T_{sub} = T_{sat} - T_{bulk} \) and \( T_{sat} = T_w - T_{sat} \). The equation above as well as models from Hsu [20], Bergles and Rohsenow [2] and Davis and Anderson [11] were supported by experiments. This study about bubble nucleation is not available for microchannels but it should be applicable to them as well.

### 6.2.4 Heat Transfer Coefficient and Quality

There have been many studies about critical heat transfer coefficient (CHTC). This value is of particular interest as increasing the heat above the critical heat flux will not yield a higher heat transfer but a higher wall superheat temperature which results in dry
out. The understanding of the heat transfer coefficient and the parameters that determine its value is crucial for a reliable cooling device. Pressure fluctuations and instabilities lower the critical heat flux. An extensive study by Qu and Mudawar [49] compared eleven models for predicting the HTC from macro-, mini- and micro-channels with actual experimental data from mini-channel experiments with a hydraulic diameter of 356µm. The flow rates were varied between 135-402 kg/m²s and the inlet liquid temperature was either 30°C or 60°C. Qu and Mudawar [49] measured the HTC in dependence of the vapor quality. They used the fin analysis to obtain the mean heat transfer coefficient averaged over the heated perimeter of the channel with the dimensions shown in Table 6.1. The applicability of the fin analysis to micro-channels was discussed by Qu and Mudawar [48] earlier.

Table 6.1 Dimensions of micro-channel heat sink unit [49]

<table>
<thead>
<tr>
<th>$W_w$ (µm)</th>
<th>$W_{ch}$ (µm)</th>
<th>$H_{wj}$ (µm)</th>
<th>$H_{ch}$ (µm)</th>
<th>$H_{w2}$ (µm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>118</td>
<td>231</td>
<td>12'700</td>
<td>713</td>
<td>2462</td>
</tr>
</tbody>
</table>

The fin analysis yields the following energy balance

$$q_{eff}^* W_{cell} = h \left( T_{w,liq} - T_{sat,liq} \right) \left( W_{ch} + 2\eta H_{ch} \right)$$

(6.4)

where the left-hand side is the heat input and the right-hand side is the heat removal by flow boiling. The thin fin approximation is applied to the side walls by the fin efficiency

$$\eta = \frac{\tanh \left( m H_{ch} \right)}{m H_{ch}}$$

(6.5)

where $m$ is the fin parameter

$$m = \sqrt{\frac{h}{k \cdot W_w}}$$

(6.6)
The macro-channel correlations by Chen et al. [4] (a), Shah [52], [53] (b), Gungor et al. [18] (c), Kandlikar [27] (d) and Liu et al. [38] (e) do not match the experimental data. They do not predict the trend although. The comparison of experimental data at the axial position of thermocouple four at \( z_{4.4} \) and the correlations are shown in Fig. 6.11. The flow rate was \( G = 255 \text{ kg/m}^2\text{s} \) and \( T_{\text{in}} = 60^\circ\text{C} \).

![Comparison of experimental data and macro-channel correlations from (a) to (e) [49]](image)

It is noteworthy that all correlations predict an increasing \( h \) for an increasing quality. In mini- and micro-channels this trend is reversed. The higher the quality the smaller the heat transfer coefficient. The mean absolute error defined by

\[
MAE = \frac{1}{M} \sum \left| \frac{h_{\text{pred}} - h_{\text{exp}}}{h_{\text{exp}}} \right| \times 100\% 
\]

where \( M \) is the number of data points and \( MAE \) is the maximum absolute error which ranges from 35.1% to 53.7%. The bad prediction results from different reasons. Firstly, the macro-channel correlations are based on fairly big hydraulic diameters which mean turbulent flow is present. In micro-channels however \( Re_{\text{crit}} = 2000 \) Reynolds numbers are below the critical value of due to the small dimensions and the low flow rate. In the discussed study they varied from 60 to 300. Secondly, most macro-channel correlations assume nucleate boiling as the dominant heat transfer mechanism. It is widely accepted that the two governing modes for heat transfer are nucleate boiling and forced convection boiling [17]. For nucleate boiling the saturation temperature is reached near the heated wall and vapour bubbles form. The value of HTC is in this case dependent upon the heat flux but not that much on the mass flux and the vapour quality. This mode is usually connected with bubbly and slug flow pattern, contrary to annular flow for forced convection boiling. In the forced convection boiling region however, which is basically a single-phase heat transfer mode, the HTC is more dependent upon coolant mass velocity and vapour quality. In the study of Qu and Mudawar [49] the measured values of \( h_p \) propose that micro-channel flow boiling is mainly dominated by forced convection flow boiling instead of nucleate boiling.
The comparison of experimental data for \( G = 255 \text{ kg/m}^2\text{s} \) and \( T_{in} = 60^\circ\text{C} \) with the correlations for mini- and micro-channels by Steiner et al. [55] (f), Lazarek et al. [33] (g), Yu et al. [67] (h) and Warrier et al. [64] (i) is shown in Fig. 6.12.

![Fig. 6.12 Comparison of saturated flow boiling HTC with correlations (f) to (j) [49]](image)

Even though these correlations predict the measured HTC better they are still off by a maximum absolute error (MAE) of 36.2%. None of the above correlations can predict the trend of the HTC correctly which makes it necessary to develop more accurate models.

The authors Qu and Mudawar proposed an own correlation [50] which is based on forced convection flow boiling as the dominant heat transfer mechanism and annular flow. Clearly one can see the better agreement with the experimental recorded for \( G = 255 \text{ kg/m}^2\text{s} \) and \( T_{in} = 60^\circ\text{C} \) as shown in Fig. 6.13.

![Fig. 6.13 Comparison of model form Qu and Mudawar and experimental data for the HTC [50]](image)

### 6.2.5 Bubble Growth and Bubble Departure Diameter

There is very little literature available about bubble growth in micro-channels, even less for bubble growth in single-channel devices. Most investigators focused on pressure drop relations, HTC and flow pattern. As parallel micro-channels interact with each other by feedback through mass and heat flux [30] this configuration makes it even more difficult to extract the governing parameters for vapour bubble growth. Boiling in
one channel will probably remove much heat through wall conduction from the
neighbouring channel and might suppress nucleation. The hydrodynamic influence is
based on flow redirection to the sudden decrease in density when vapour forms. The
vapour is accelerated and pressure drop increase which redirects liquid flow to channels
that are still in single phase. The boiling channel encounters dry-out while the other
channels cool down so that nucleation is not possible either. Bubble dynamics in
homogenous superheated bulk mediums is governed by the Rayleigh equation.

\[
\frac{dr}{dt} = \left( \frac{2 \Delta p_v}{3 \rho_i} \right)^{1/2} \approx \left[ \frac{2}{3} \frac{(T_w - T_{sat}) \Delta h_v}{T_{sat} \rho_i v_{vl}} \right]^{1/2} 
\]

(6.8)

with \( \Delta p_v = p_v - p_l \), \( T_w - T_{sat} \) the difference of the wall and the saturation temperature
and \( v_{vl} \) the vapour-liquid specific volume.

Froster and Zuber [16] as well as Plesset and Zwick [47] presented solutions for the
time evolution of bubble radius. The growth rate is linear with time and the
proportionality constant is directly proportional to the square root of over-pressure
divided by liquid density. Over-pressure is vapour pressure inside the bubble minus
pressure of surrounding liquid. The vapour pressure is related to the superheat
temperature through the Clausius-Clapeyron equation. The next stage is governed by
thermal diffusion through the vapour to liquid interface of the bubble.

However, for bubble growth in heated walled channels it is more complicated as the
boundary conditions are different. Especially the thin boundary layer has significant
effect on the bubble growth as shown by Moore et al.[41]. In the experiments conducted
by Lee at al. [35] the bubble growth rate was much lower than predicted by the
Rayleigh equation. This might result from the fact, that the Rayleigh equation was
developed for pool boiling without any constraints for growing bubbles. This is
different in micro-channels with their small dimensions. Copper [8] suggested the
following model for micro-channels with heated walls which is based on the
evaporation of microlayer underneath the growing bubble

\[
R(t) = 2.5 \frac{Ja}{Pr_t^{1/2}} (\alpha_i t)^{1/2} 
\]

(6.9)

where

\[
Ja = \frac{\rho_c c_p \Delta P (T_w - T_{sat})}{\rho_l \Delta h_v} 
\]

(6.10)

and

\[
Pr_t = \frac{v_i}{\alpha_i} 
\]

(6.11)
According to Copper the bubble radius is a function of the square root of time. This behaviour is inconsistent with the results from Lee et al. [35] in Fig. 6.14 which suggests that the microlayer is not present in micro-channels. Furthermore Li et al. [37] and Lee at al. [35] report that bubble growth rate increases with increasing heat input and decreases with increasing mass flux. The experimental results in literature regarding bubble growth are pretty scattering. Until now there is no consistent model which predicts bubble growth correctly and takes into account the microlayer evaporation and the time-varying temperature and flow field around the bubble [12].

According to a dimensional analysis of Fogg et al. [15], the bubble departure size is influenced by the acting forces on the bubble. For microchannels the bubble is constraint be the side walls which mean the force balance in that lateral direction equals zero. Only forces in flow direction play a role in departure diameter. They identified the pressure gradient in axial direction at the front and back interface of the bubble as a significant factor in overcoming surface tension. Besides Fogg et al. also M. Lee et al. [34] report that the bubble departure size is inversely proportional to the Reynolds number in the channel as seen in Fig. 6.15.
The relation they suggest is shown in equation (6.12)

\[
\frac{V_{BD}}{H^3} = 2 \cdot 10^4 \exp\left(-5 \text{Re}^{0.25}\right) \tag{6.12}
\]

where \( H \) is the height of the channel. Lee at al. report that bubble departure size decreases with increasing heat flux. That is because with increasing heat flux their onset of nucleation moves upstream where a higher pressure exists which may suppress bubble growth. For low mass fluxes bubble size is biggest. Dominant forces are again surface tension and drag force. Applying the model of Levy [36] provides prediction of their result in 80% of the cases with an accuracy of \( \pm 20\% \). Levy’s model is shown in equation (6.13)

\[
R_B = C_1 \left(\frac{\sigma}{\rho c_d}\right)^{1/2} + C_2 = C_1 \left(\frac{\sigma L}{\Delta p}\right)^{1/2} + C_2 \tag{6.13}
\]

where \( R_B \) is the bubble departure radius, \( \sigma \) is the surface tension, \( L \) is the channel length and \( \Delta p \) is the pressure drop along the channel.

### 6.2.6 Geometries of Test Devices in Literature

Most of the micro-channel heat exchangers are made by MEMS-technology out of silicon. Due to time constraints it was impossible to design an experiment based on MEMS-technology. Besides a high thermal conductivity a rather soft material was needed which could be easily machined with conventional milling machines. Aluminum as well as copper seemed to fulfill these requirements. A comparison of their physical and thermophysical properties is shown in Table 6.2. Because of availability Aluminium was used instead of Copper.

<table>
<thead>
<tr>
<th>Material</th>
<th>Melting point [K]</th>
<th>( \rho ) [kg/m(^3)]</th>
<th>( c_p ) [J/kgK]</th>
<th>( K ) [W/mK]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminium Alloy 2024-T6</td>
<td>775</td>
<td>2702</td>
<td>903</td>
<td>237</td>
</tr>
<tr>
<td>Oxygen-free Copper</td>
<td>1358</td>
<td>385</td>
<td>385</td>
<td>401</td>
</tr>
</tbody>
</table>

Also some experiments in literature were conducted without silicon-based designs. Some designs with oxygen-free copper from Qu and Mudawar [51], Steinke and Kandlikar [56] and Bowers and Mudawar [3] are shown in Fig. 6.16.
In some designs the structure with the actual channels is separated from the rest [51], [56]. By doing so the heated part can be better isolated and several inserts with several channel geometries can be tested. Most of the designs have cartridge heaters that are controlled over input voltage. The channels are sealed from top with a transparent polycarbonate cover that allows optical access to the channel. Temperature measurements are performed with thermocouples. The design of our experiment was even simpler at the beginning but after several design iterations it got more similar to those in literature.

6.3 Goals of this Work

This work tries to fill the gap of bubble dynamics data in micro-channels. The goal was to get best results with a most simple experiment which could be quickly designed and used. In addition the experience gained during the design iteration and the actual experiment are meant to be used other scientists in the research group.

The use of a high speed CMOS-camera in combination with a microscope should allow high resolution pictures of bubble growth and departure diameter in two-phase flow boiling in a single-channel micro heat exchanger. Besides the acquisition of the data also the feasibility of such an easy to handle experiment should be demonstrated which allow further students to do ongoing research with the same device.


7 EXPERIMENTAL SETUP

One enormous obstacle in performing this experiment was the time frame. Simplifying the design seemed the paramount goal given that within 6 short months the entire set-up had to be built and tested and the experiments themselves conducted. At the same time, it had to guarantee enough accuracy and prove durable enough for re-use in other experiments that might follow in the wake of this thesis. The idea of using MEMS technology which is usually applied in micro-channel heat exchanger experiments had to be discarded due to such time constraints. Instead, the concept of an Aluminum based set-up was adopted. At a constant flow rate, a syringe pump delivered de-ionized water, which had been degassed before use. Before the fluid goes into the block it passes through the pressure transducer, a flow meter and a k-type thermocouple. The outlet was open to ambient pressure and the heating was applied by a cartridge heater. The test section was mounted on a telescope to which a high speed CMOS camera was attached. The data was recorded with a DAQ-system and the software LabView. A schematic of the experiment is shown in Fig. 7.1.

![Fig. 7.1 Schematic drawing of the set-up](image)

The following section will describe the components of the experimental set-up.

7.1 ALUMINIUM BLOCK WITH CHANNEL AND THERMOCOUPLES

Due to the aforementioned time constraints, a non-MEMS based experiment was chosen. The aluminum block was manufactured to the size specifications of 95mm x 43mm x 34.2 mm. Two holes for measuring the inlet and outlet water temperature with thermocouples were drilled. The channel itself is 50mm long with a width determined by the smallest end mill then available. The diameter of the end mill is 152µm whereas the depth was set with the CNC-machine to 50µm. Measurements of the dimensions are presented in 8.5.2. The inlet- and outlet-reservoir were needed to
minimize entrance effects in the channel. For the inlet- and outlet-tubing standard Swagelok fittings were used. The dimensions in millimeters are shown in Fig. 7.2.

Fig. 7.2 Dimensions of the Aluminium block, (a) top view with 50mm long channel and 18 boreholes for press fitting of the glass cover, (b) side view with two holes for k-type thermocouples, (c) front view with inlet hole and hole for cartridge heater

The details of the reservoir and the channel are shown in Fig. 7.3.

Fig. 7.3 Dimensions of the in-/outlet reservoir (a): top view of reservoir, (b) side cut of the reservoir with 50µm deep channel to the right
The top surface of the block was covered with a 1.5mm thick glass cover held down by an aluminium frame. A rubber frame around the glass plate was constructed to ensure superior clamping. The whole assembly is shown in Fig. 7.4.

For two reasons, however, this configuration failed to adequately seal the channel. Firstly, the top surface was too rough; even after applying fine mill tools. Secondly, the glass broke very easily as the screws had been tightened to the point of actually bending the aluminium frame. The bending of the top frame increased the pressure at the long edges of the glass plate which made the plate bend slightly, a phenomenon illustrated by exaggeration in the below. In an attempt to right this wrong, the plate was lifted off from the middle where the channel was and the sealing appeared to be at its worst.

The next iteration of the design addressed the problem of leakage. The stiffness of the glass cover accounted in part for the leakage that had occurred. Acrylic, a more flexible material for the cover, was decided upon in hopes of adjusting the roughness of the surface and thereby correcting the faulty seal. So too was the top surface of the block polished to improve sealing. In order to increase the clamping pressure without make the acrylic cover bend, the boreholes were moved closer to the channel as shown in Fig. 7.6.
Equipped with the acrylic cover, proper sealing could be achieved for pressures up to 40 psi. The rubber frame and the top aluminium frame were made redundant by the boreholes that had been drilled directly into the acrylic plate as seen in Fig. 7.7.

After first performing the flow boiling experiments, it became obvious that the residence time within the inlet port and the inlet reservoir were long enough to heat up the liquid to saturation temperature. The fluid partly vaporized before it entered the channel. In order to thermally isolate the heated section from the rest, parts of the block were cut out to provide isolation by way of an air gap. The heater was then installed at a transverse angle to the micro-channel as a means of reducing the area of the heated section. To get a more accurate temperature profile along the channel, seven more k-type thermocouples were installed. Except for the \( T_{\text{fluid 1}} \) and \( T_{\text{fluid out}} \), the thermocouples were about 2mm underneath the channel. The new design is shown in Fig. 7.8. One thermocouple was installed right above the heater, two more were placed above the air gap and the rest were situated upstream. The inlet is on the right, away from the heater. The distance for the thermocouples underneath the channel is measured from the beginning of the channel.
The block temperatures were high enough to reach the glass transition temperature of the acrylic cover plate at around 105°C. Therefore, polycarbonate was used for the cover plate as it can withstand temperatures up to 128°C.
Even though the temperature gradient along the channel was greater now the fluid would still boil in the inlet reservoir. To address this problem, one could either actively cool the inlet section or try to narrow the passage of material between the heated position and the upstream path of the channel. The latter was performed in the third iteration. The result is shown in Fig. 7.10.

The assembly is shown in Fig. 7.11.
7.2 **DAQ-SYSTEM (SBC-68, DAQ-CARD, LabVIEW)**

In accumulating the data, the DAQ-board SCB-68 was used in combination with the DAQ-card 6062E and the software LabView 7.1. The software took care of the acquisition of the thermocouples readings and the pressure transducer. Moreover, LabView synchronized the recording of the camera and the recording of the thermocouples as well as the pressure transducer data. The DAQ-board possesses 16 analog inputs and the capacity for integrated cold junction compensation. When acquiring analog signals one has to distinguish between non-referenced or floating signals and ground-referenced signals. Typical non-referenced signals are thermocouples or battery powered devices that are not connected to the building ground as shown in Fig. 7.12 (a). Each terminal is independent of the system ground. Typical grounded signal sources are devices that are plugged into wall outlets and thereby connected to building ground, such as generators and power supplies; refer to Fig. 7.12 (b).

![Fig. 7.12 Difference between a floating signal (a) and a ground-referenced signal (b)](image)

Furthermore, one distinguishes between differential (DIFF) measurements, referenced single-ended (RSE) measurements and non-referenced single-ended (NRSE) measurements. The differential measurement is similar to a floating signal in the sense that the measurement is made in reference to a floating, as opposed to a system, ground. Battery-powered instruments and handhelds provide examples of just such differential measurements as they are connected neither to the earth nor the building ground. A RSE measurement, on the other hand, is taken with respect to a fixed system ground – namely, the AIGND on the DAQ-board. In a NRSE measurement, signals are still interpreted with respect to a common ground called AISENSE, although their variance from the system ground, AIGND, can still be expected. One should always try to measure signals based on the DIFF method wherein two potentials are subtracted from each other thereby cancelling out noise that is present in the positive as well as in the negative potential signal and so increasing the accuracy. However, for acquiring one analogue signal in DIFF method one needs two channels which imply that only eight analogue signals can be measured in DIFF mode. For the described experiments up to 7 wall temperatures, three fluid temperatures and one pressure value were recorded. A
total of 11 analogue signals had to be acquired. The pressure transducer had to be in DIFF mode which required two channels. Fourteen channels remained for the temperature sensors which were not enough to perform the DIFF measurements that would have required 20 additional channels. For that reason the thermocouples were addressed as RSE. Details about the used thermocouples and the calibration can be found in 8.3.

### 7.3 High Speed Camera Phantom V7.3

In order to capture the fast process of bubble nucleation and departure, a high speed camera was used. As the camera was recently bought and was never before used, I had to implement it into the system and demonstrate its applicability for micro-channel bubble dynamic studies.

The camera, made by Vision Research, is a Phantom V7.3 with 4GB of internal flash memory. The CMOS-sensor measures 17.6 x 13.2mm and supports 8 and 14 bits of pixel depth with solutions of up to 800 x 600 pixels. The resolution is scaleable in increments of 32x8 pixels. The pixel size is 22µm. The maximum frame rate per second is 190’476 at 2µs exposure time. The Phantom recording software was triggered with LabView.

### 7.4 Microscope, Syringe Pump, Flow Meter, Pressure Transducer, White Light Source

The NIKON Eclipse TE2000-U is an inverted microscope with an external white light source. The magnification was adjustable between 4x to 30x. The high speed camera was mounted on a side-port of the microscope. For flow delivery, the syringe pump, PHD2000 from Harvard, was used with a 14.57mm diameter syringe. The working fluid was de-ionized water that was degassed for 15 minutes before usage. The infusion rate could be adjusted in increments of 0.1µl/min. The flow meter engaged, an ASL 1430-16 from Sensirion, is effective between 150nl/min and 1650µl/min with a response time of 20ms. The accuracy is about ±3% of the measured value. For pressure measurements, the pressure transducer, a PX 2300-50DI from Omega, was used. The range is 0-50psi with an accuracy of ±0.25% full scale. The applied white light was EXFO Methal Halide Source.

For calibration of syringe pump and pressure transducer, please refer to the next chapter.
8 CHARACTERIZATION

All the used devices except the flow meter were calibrated before use in this experiment. The syringe pump, in particular, failed to deliver the desired flow. Variations up to 48.3% were observed. The accuracy in delivered flow is crucial for the determination of the hydraulic diameter via the pressure drop. The isothermal experiments were mostly conducted with flow rates between 5µl/min and 60µl/min. For the non-isothermal experiments flow rates between 100µl/min and 400µl/min were applied. The choice of flow rate was based on an energy balance of the form

\[ P = \left( \bar{c}_{p,H,O} \cdot \Delta T + x \cdot \Delta h_v \right) \dot{m}_{tot} \]  

(8.1)

yields

\[ \dot{m}_{tot} = \frac{P}{\bar{c}_{p,H,O} \cdot \Delta T + x \cdot \Delta h_v} \]  

(8.2)

With \( P=130W \), \( \bar{c}_{p,H,O} = 4.19 \text{ kJ/kgK} \), \( \Delta T = 50K \), quality \( x=1 \) and vaporization energy for water \( \Delta h_v = 2257 \text{ kJ/kg} \). It was assumed that in practical cooling application the typical micro-channel heat exchanger distributed the flow through \( n = 30, 20, 10 \) channels and that all water would vaporize at a temperature of 100°C. Equation (8.2) yields a total mass flow rate of \( \dot{m}_{tot} = 5.27 \times 10^{-5} \text{ kg/s} \) which translates into a volume flow rate \( \dot{V}_{\text{single,n=30}} = 105.67 \mu l/min \). With only 20 or 10 channels, the volume flow rate increases to \( \dot{V}_{\text{single,n=20}} = 158.02 \mu l/min \) and \( \dot{V}_{\text{single,n=10}} = 317.02 \mu l/min \) respectively. In the following, the volume flow rate is given as a mass flux, calculated by dividing the mass flow rate by the cross sectional area of the channel. This method proves advantageous as one can compare different sized channels while the mass flux rate is normalized by the channel dimensions. In Table 8.1 a conversion of the most often used flow rates is shown based on the above water density.

<table>
<thead>
<tr>
<th>Volume flow rate ( \dot{V} ) [µl/min]</th>
<th>Mass flux ( G ) [kg/m²s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>12.21</td>
</tr>
<tr>
<td>10</td>
<td>24.43</td>
</tr>
<tr>
<td>20</td>
<td>48.86</td>
</tr>
<tr>
<td>30</td>
<td>73.28</td>
</tr>
<tr>
<td>40</td>
<td>97.71</td>
</tr>
<tr>
<td>50</td>
<td>122.14</td>
</tr>
<tr>
<td>60</td>
<td>146.57</td>
</tr>
<tr>
<td>80</td>
<td>195.42</td>
</tr>
<tr>
<td>100</td>
<td>244.28</td>
</tr>
</tbody>
</table>
8.1 CALIBRATION OF SYRINGE PUMP

The calibration was performed while the block and all accessories were applied to the system. Although the diameter of the syringe was initially corrected to 14.57mm, the syringe pump still persisted in delivering less flow than expected. The flow rate was checked with a flow meter that has a relative error of ±3%. The calibration chart is shown in Fig. 8.1.

![Calibration Chart]

One can see that the real mass flux is off by at least ten percent. Clearly, the relative error for larger mass fluxes tends towards a value of about 49%. For all following experiments the mass flux was adjusted until the flow meter showed the desired value.

8.2 CALIBRATION OF PRESSURE TRANSDUCER

For the calibration of the pressure transducer a handheld device was used which could be pressurized with a hand pump. Simultaneously, a voltmeter was applied to read the
voltage at the specified pressure. The calibration chart is shown in Fig. 8.2. The accuracy of the calibration device is ±0.05%. Noteworthy is the fact that the below graph depicts the absolute pressure necessary for determining the saturation temperature for the onset of boiling. For calculation of the channel dimensions later on only the pressure drop over the device is considered. This yields a similar graph as shown below with an offset of -1.013bar or -101.3kPa respectively.

Fig. 8.2 Linear relation between voltage and pressure for pressure transducer PX 2300-50DI

8.3 CALIBRATION OF THERMOCOUPLES

8.3.1 COLD-JUNCTION-COMPENSATION-MEASUREMENTS

It is imperative that one knows the temperature of the cold junction ($T_{ref}$) since the cold-junction-compensation-voltage is only valid for one particular value of $T_{ref}$. In my experiments the room temperature was the cold-junction temperature. If the ambient temperature does not vary much over the day then the error implied by assuming a constant cold-junction-temperature remains small. Fig. 8.3 charts the development of the temperature over 24 hours. The measurements were taken with the DAQ-board integrated IC-temperature-sensor. The accuracy is ±1°C over a 0°C to 110°C range. The black lines indicated the error in temperature measurement. The vertical grey dotted lines mark that time of day considered in averaging the ambient temperature.
Based on Fig. 8.3 the assumption of a constant ambient temperature of 23.1°C is valid within the 10am to 8pm timeframe.

**8.3.2 Determination of Cold-Junction-Compensation Voltage**

When two wires composed of dissimilar metals are joined at both ends and one of the ends is heated there is a continuous current which flows in the thermoelectric circuit. Thomas Seebeck made the latter discovery in 1821. If the circuit is broken the net open circuit voltage identifies itself as a function of the junction temperature and the composition of the two metals. The proportionality between temperature and voltage is related through the Seebeck constant, $\alpha$,

$$V = \alpha \Delta T$$  \hspace{1cm} (8.3)

which is characteristic for each combination of two different metals. If one wants to measure the potential difference, $V_1$, as shown in Fig. 8.4, by applying a voltmeter two new junctions, $J_2$ and $J_3$, are created which influence the reading by introducing additional junction voltages. If one maintains the temperature of $J_2$ and $J_3$ at the same temperature, $T_{\text{ref}}$, then the reading of the voltmeter will show

$$V = \alpha \left( T_1 - T_{\text{ref}} \right)$$  \hspace{1cm} (8.4)

If one knows $T_{\text{ref}}$ one can look up the junction voltages $J_2$ and $J_3$ in tables provided by the National Institute of Standards and Technology (NIST) and proceed from there to
calculate the corresponding junction voltages which must be added to the voltmeter reading to get the real potential, \( V_1 \). Applying the NIST-tables once more one can convert the voltage into the temperature \( T_1 \).

\[ T(U) = d_0 + d_1 \cdot U + d_2 \cdot U^2 + \ldots + d_n \cdot U^n \]  
\[ V(T) = \sum_{i=0}^{n} d_i(T)^i + a_0 e^{a_1(T-a_2)^2} \]  

\( \text{Fig. 8.4 Schematic of the different junctions when reading k-type thermocouple voltage} \)
Table 8.2 Conversion factors from NIST [45] for voltage to temperature conversion and vice versa

<table>
<thead>
<tr>
<th>Voltage to temperature $T(U)$</th>
<th>Temperature to voltage $U(T)$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$d_0$ to $d_0$ valid for 0°C to 500°C</td>
<td>$d_0$ to $d_0$ and $a_0$ to $a_2$ valid for 0°C to 1372°C</td>
</tr>
<tr>
<td>$d_0 = 0.000$</td>
<td>$d_0 = -1.760041 \cdot 10^{-2}$</td>
</tr>
<tr>
<td>$d_1 = 25.08355$</td>
<td>$d_1 = 3.892120 \cdot 10^{-2}$</td>
</tr>
<tr>
<td>$d_2 = 7.860106 \cdot 10^{-2}$</td>
<td>$d_2 = 1.855877 \cdot 10^{-5}$</td>
</tr>
<tr>
<td>$d_3 = -2.503131 \cdot 10^{-1}$</td>
<td>$d_3 = -9.945759 \cdot 10^{-8}$</td>
</tr>
<tr>
<td>$d_4 = 8.315270 \cdot 10^{-2}$</td>
<td>$d_4 = 3.184095 \cdot 10^{-10}$</td>
</tr>
<tr>
<td>$d_5 = -1.228034 \cdot 10^{-2}$</td>
<td>$d_5 = -5.607284 \cdot 10^{-13}$</td>
</tr>
<tr>
<td>$d_6 = 9.804036 \cdot 10^{-4}$</td>
<td>$d_6 = 5.607506 \cdot 10^{-16}$</td>
</tr>
<tr>
<td>$d_7 = -4.413030 \cdot 10^{-5}$</td>
<td>$d_7 = -3.202072 \cdot 10^{-19}$</td>
</tr>
<tr>
<td>$d_8 = 1.057734 \cdot 10^{-6}$</td>
<td>$d_8 = 9.715115 \cdot 10^{-23}$</td>
</tr>
<tr>
<td>$d_9 = -1.052755 \cdot 10^{-8}$</td>
<td>$d_9 = -1.210472 \cdot 10^{-26}$</td>
</tr>
</tbody>
</table>

The averaged thermocouple-voltage for three measurements in an ice bath was $V_{\text{calibrate},0^\circ C} \approx -0.919779 \, mV$ which corresponds to a theoretical temperature of $T_{\text{calibrate},0^\circ C} \approx -22.74^\circ C$. The corresponding voltage at 0°C according to the NIST-conversion would be $T_{NIST,0^\circ C} \approx 0.00^\circ C$. Accordingly, the CJC-voltage yields

$$V_{\text{CJC},0^\circ C} = V_{\text{NIST},0^\circ C} - V_{\text{calibrate},0^\circ C}$$

(8.7)

$$V_{\text{CJC},0^\circ C} = 0.00 \, mV - (-0.919779 \, mV) = 0.919779 \, mV$$

(8.8)

that had to be added to the actual voltage reading from the thermocouples by the DAQ-system to get the appropriate voltage which was converted to temperature with equation (8.5) in a next step.

The same was done for two measurements in boiling water which gave an averaged thermocouple-voltage, $V_{\text{calibrate},100^\circ C} \approx 3.167405 \, mV$, which corresponds to a theoretical temperature, $T_{\text{calibrate},100^\circ C} \approx 77.60^\circ C$. The voltage that conforms to 100°C according to the NIST-conversion would be $T_{NIST,100^\circ C} \approx 4.096230 \, mV$. The CJC-voltage is the difference between these two values.

$$V_{\text{CJC},100^\circ C} = V_{\text{NIST},100^\circ C} - V_{\text{calibrate},100^\circ C}$$

(8.9)

$$V_{\text{CJC},100^\circ C} = 4.096230 \, mV - 3.167405 \, mV = 0.928825 \, mV$$

(8.10)

The average from the calibration at 0°C and at 100°C that was used as the CJC-voltage in further experiments was added to the actual voltage reading before the conversion to the corresponding temperature value in °C.

$$\overline{V_{\text{CJC}}} = 0.924302 \, mV$$

(8.11)
8.4 **Flow Conditions**

The flow conditions that are present in the microchannel were determined for all applied flow rates. The thermal entrance length $L_{e,t}$ was determined by using the equation proposed by Kays and Crawford [29].

$$\left( \frac{L_{e,t}}{D_{h}} \right)_{\text{lam}} \approx 0.06 \text{Re}_{D_{h}} \cdot \text{Pr} \quad (8.12)$$

The hydrodynamic entrance length, $L_{e,h}$, was calculated based on the approximation of Langhaar [32]

$$\left( \frac{L_{e,h}}{D_{h}} \right)_{\text{lam}} \approx 0.05 \cdot \text{Re}_{D_{h}} \quad (8.13)$$

with

$$\text{Re}_{D_{h}} = \frac{\bar{u} \cdot D_{h}}{\nu} = \frac{\bar{u} \cdot D_{h} \cdot \rho}{\mu} \quad (8.14)$$

and

$$\text{Pr} = \frac{\nu}{\alpha} = \frac{\mu / \rho}{k / (\rho \cdot c_{p})} = \frac{\mu \cdot c_{p}}{k} \quad (8.15)$$

Even though relation (8.12) and (8.13) were originally derived for round ducts with round inlets they should give numbers in the right order of magnitude for a rectangular channel. Table 8.3 illustrates density, $\rho$, the dynamic viscosity, $\nu$, the thermal conductivity, $k$, the thermal capacity, $c_{p}$, and the Prandtl number, $Pr$, for different temperatures.

**Table 8.3** Parameters for liquid water at 20°C and 100°C

<table>
<thead>
<tr>
<th>Liquid water temperature [°C]</th>
<th>$\rho$ [kg/m$^3$]</th>
<th>$\nu$ [N/m/s]</th>
<th>$k$ [W/mK]</th>
<th>$c_{p}$ [kJ/kgK]</th>
<th>$Pr$ [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>998.23</td>
<td>$10^{-3}$</td>
<td>0.6</td>
<td>4.18</td>
<td>6.97</td>
</tr>
<tr>
<td>100</td>
<td>958.38</td>
<td>$2.8 \times 10^{-4}$</td>
<td>0.68</td>
<td>4.22</td>
<td>1.74</td>
</tr>
</tbody>
</table>
The resulting hydrodynamic and thermal entrance lengths shown in Table 8.4 are based on the hydraulic diameter, \( D_h = 68 \mu m \), the depth, \( d = 44 \mu m \), the width, \( w = 155 \mu m \), and the length, \( l = 50 mm \), of the channel.

<table>
<thead>
<tr>
<th>Mass flux [kg/m²s]</th>
<th>Liquid water temperature [°C]</th>
<th>Flow velocity [m/s]</th>
<th>( \text{Re}_D )</th>
<th>( \frac{L_{c,h}}{D_h} ) [%]</th>
<th>( L_{c,h} ) [µm]</th>
<th>( \frac{L_{c,t}}{D_h} ) [%]</th>
<th>( L_{c,t} ) [µm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>146.57</td>
<td>0.15</td>
<td>10</td>
<td>0.5</td>
<td>0.07</td>
<td>35</td>
<td>3.6</td>
<td>0.49</td>
</tr>
<tr>
<td>195.42</td>
<td>0.20</td>
<td>14</td>
<td>0.7</td>
<td>0.09</td>
<td>47</td>
<td>4.8</td>
<td>0.65</td>
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<tr>
<td>244.28</td>
<td>0.25</td>
<td>17</td>
<td>0.9</td>
<td>0.12</td>
<td>58</td>
<td>6.0</td>
<td>0.81</td>
</tr>
<tr>
<td>366.42</td>
<td>0.38</td>
<td>26</td>
<td>1.3</td>
<td>0.17</td>
<td>87</td>
<td>9.0</td>
<td>1.22</td>
</tr>
<tr>
<td>488.56</td>
<td>0.51</td>
<td>34</td>
<td>1.7</td>
<td>0.23</td>
<td>117</td>
<td>11.9</td>
<td>1.62</td>
</tr>
<tr>
<td>732.84</td>
<td>0.76</td>
<td>51</td>
<td>2.6</td>
<td>0.35</td>
<td>175</td>
<td>17.9</td>
<td>2.44</td>
</tr>
</tbody>
</table>

8.5 ISOETHERMAL MEASUREMENTS

8.5.1 ISOETHERMAL PRESSURE DROP

Many pressure drop measurements were taken to produce Fig. 8.12 with its pressure drop dependence on flow rate. Even though the reported data for pressure drop experiments confirms ones expectations, many experiments did not show the desired results. While the problems are not completely resolved, the most likely explanation for them is that little particles from the Teflon-sealing-tape at in- and outlet blocked the channel from time to time. The possibility that the cover plate deformed due to the applied forces by the screws is not supported by the data. Were such an occurrence to transpire it would manifest itself in low pressure drops during the first experiments and in later experiments increased ones as the shared plate would be subject to the potentially deforming forces for a longer period of time. The very stiff tubing seems an unlikely source or culprit for this inconsistency. The isothermal pressure drop experiments for different mass fluxes are shown in Fig. 8.5 to Fig. 8.11. The results vary from one experiment to another with up to 50%. For low mass fluxes, especially, it seems as the pressure drop varies most. For mass fluxes above 97.71kg/m²s the data is closer together and the variation is smaller. It varies between 15 and 25%. The error in the data is most likely systematic and unrelated to the pressure transducer which has an accuracy of 0.25%. 

<table>
<thead>
<tr>
<th>Mass flux [kg/m²s]</th>
<th>Liquid water temperature [°C]</th>
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<th>( \text{Re}_D )</th>
<th>( \frac{L_{c,h}}{D_h} ) [%]</th>
<th>( L_{c,h} ) [µm]</th>
<th>( \frac{L_{c,t}}{D_h} ) [%]</th>
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Many pressure drop measurements were taken to produce Fig. 8.12 with its pressure drop dependence on flow rate. Even though the reported data for pressure drop experiments confirms ones expectations, many experiments did not show the desired results. While the problems are not completely resolved, the most likely explanation for them is that little particles from the Teflon-sealing-tape at in- and outlet blocked the channel from time to time. The possibility that the cover plate deformed due to the applied forces by the screws is not supported by the data. Were such an occurrence to transpire it would manifest itself in low pressure drops during the first experiments and in later experiments increased ones as the shared plate would be subject to the potentially deforming forces for a longer period of time. The very stiff tubing seems an unlikely source or culprit for this inconsistency. The isothermal pressure drop experiments for different mass fluxes are shown in Fig. 8.5 to Fig. 8.11. The results vary from one experiment to another with up to 50%. For low mass fluxes, especially, it seems as the pressure drop varies most. For mass fluxes above 97.71kg/m²s the data is closer together and the variation is smaller. It varies between 15 and 25%. The error in the data is most likely systematic and unrelated to the pressure transducer which has an accuracy of 0.25%.
Fig. 8.5  Different isothermal pressure drop experiments for $G = 12.21 \text{kg/m}^2\text{s}$

Fig. 8.6  Different isothermal pressure drop experiments for $G = 24.43 \text{kg/m}^2\text{s}$

Fig. 8.7  Different isothermal pressure drop experiments for $G = 48.86 \text{kg/m}^2\text{s}$
Fig. 8.8  Different isothermal pressure drop experiments for $G = 73.28\text{kg/m}^2\text{s}$

Fig. 8.9  Different isothermal pressure drop experiments for $G = 97.71\text{kg/m}^2\text{s}$

Fig. 8.10  Different isothermal pressure drop experiments for flow rate $G = 122.14\text{kg/m}^2\text{s}$
In segue from the isothermal experiments to the boiling experiments, it should be mentioned that the cover plate deformed slowly due to the applied pressure via the screws. When reaching its glassy temperature, especially, the plate would slowly decrease the channel cross-sectional area which in turn increased the drop in pressure and so the saturation temperature too. Even though the polycarbonate plate was replaced regularly, the effect of reducing the channel cross sectional area might come to bear in the experiments. Moreover, a highly periodic behaviour of the pressure drop could be observed for several experiments. For details please refer to chapter 9.2 Boiling Curves.

8.5.2 Measurement of Channel Dimension: Optically and via Pressure Drop

There are at least two possible ways of measuring the channel dimensions; either optically or by gauging the pressure drop at a known flow rate. Both methods were applied and are described in this chapter. The width and height of the channel are crucial factors for the bubble dynamics. In order to verify the channel dimensions, the pressure drop was calculated with a model from White\textsuperscript{[65]} and compared with the measured values (Fig. 8.12).

\[
\dot{V} = \frac{4a^3b}{3 \cdot \mu} \left( \frac{dp}{dx} \right) \left[ 1 - \frac{192a}{\pi^2b} \sum_{i=1,3,5,\cdots}^{\infty} \left( \frac{\tanh \left( i\pi b / 2a \right)}{i^2} \right) \right]
\] (8.16)

The input parameters were the viscosity $\mu$, the half width $a$, the half depth $b$ and the flow rate $\dot{V}$.

First the channel was measured optically. Both the top surface of the block and the channel ground were focused and the translational distance of the stage was taken to be the channel depth. This resulted in a depth of $54 \mu m \pm 10 \mu m$. The width was measured with an eyepiece that had marks inscribed at a distance of $100 \mu m$. Counting the marks and applying the magnification factor yielded a width of $165 \mu m \pm 5 \mu m$ and resulted in a hydraulic diameter of $81 \mu m \pm 12 \mu m$. In order to verify this result, pressure drop
measurements were performed. Unfortunately the repeatability of the measurements could be characterized as anything but ideal. The pressure drop for one and the same mass flux differed by up to 54.1%. At low mass fluxes in particular the scatter was big, at higher mass fluxes the deviation shrank to a mere 18.5%. An average value of at least three pressure drop measurements was taken for each mass flux. In Fig. 8.12 one can see the measured pressure drop for different mass fluxes and the calculated pressure drop based on the dimensions acquired through optical means.

![Fig. 8.12 In red: calculated pressure drop and calculated hydraulic diameter based on width and height of channel measured optically, in black: actual measured pressure drop and resulting hydraulic diameter assumed that w = 155µm](image)

The hydraulic diameter based on optical measurement (w = 165µm and h = 54µm) was 81µm. The resulting pressure drop according to equation (8.16) was way too small. Conclusively, the width and the depth must have been smaller than observations by microscope would have had them.

The channel was machined by using an end mill with diameter of 152.4µm. Acknowledging that the instrument might be slightly off-centred, the width of the channel was assumed to be 155µm instead of 165µm as measured optically. The unknown depth would now result from equation (8.16) with the measured pressure drop values as input parameters. The resulting hydraulic diameter is shown in Fig. 8.12 as the black line with triangles. The values tend to attenuate about 67µm. If one ignores the outlier for the lowest mass flux of 12.21kg/m²s then the average hydraulic diameter is 67.95µm and the corresponding depth is 43.93µm.
9 TWO-PHASE FLOW MEASUREMENTS

9.1 TYPICAL TEMPERATURE PROFILES

As described earlier the test block was modified several times to improve the test conditions. It was most important to maintain a temperature profile along the channel that would allow boiling at saturation temperature towards the end of the test section while remaining below saturation temperature at the beginning of the channel. Otherwise, there would not have been a decisive observable onset of boiling. Even with the constructional modifications, the boiling process tended to be very stochastic. One day it would boil at a certain position $x$ in the channel and another time with, despite identical conditions, it would not. The temperature and pressure profile shown in Fig. 9.1 are taken for the last block-configuration at a mass flux of $G = 244.28\, \text{kg/m}^2\text{s}$ and an electrical power input of 85W.

![Temperature profile for the different thermocouples and development of pressure drop](image)

*Fig. 9.1 Temperature profile for the different thermocouples and development of pressure drop*
It is noteworthy that $T_{\text{wall} \ 2}$ is lower than $T_{\text{wall} \ 1}$ even though it is almost the same distance away from the heater. The reason might be that $T_{\text{wall} \ 1}$ is embedded in a relatively big piece of aluminium near the inlet port, whereas $T_{\text{wall} \ 2}$ is right underneath the channel where the heat transfer from the block to the fluid might play a more important role than near the inlet port. The other temperature profiles are consistent with expectations. The closer the thermocouples are to the heater, the higher the temperature. The 3rd modification of the aluminium block allowed the temperature to maintain a steady state profile along the channel. The temperature difference between $T_{\text{wall} \ 3}$, the thermocouple closest upstream to the heater, and $T_{\text{wall} \ 4}$, the thermocouple right above the channel, remained about 10°C. Theoretically, this difference was great enough to allow flow above $T_{\text{wall} \ 4}$ but not above $T_{\text{wall} \ 3}$.

The corresponding spatial temperature profile in dependence of time is shown in Fig. 9.2. All distances are measured positive in flow direction from the beginning of the channel. The position of the heater is marked with a bold back line about 37.5mm downstream from the inlet of the channel. One can clearly see that for one special time resolution, the thermocouple $T_{\text{wall} \ 4}$ always shows the highest wall temperature in the dataset. The fluid outlet temperature is almost identical to the sixth wall temperature.

![Fig. 9.2 Spatial temperature profile for $G = 244.28 \text{kg/m}^2\text{s}$ and electrical power input of 85W, distances measured from channel inlet, bold line is heater position](image)
9.2 BOILING CURVES

To determine the boiling curves for different mass fluxes, another basic measurement was taken before going into bubble dynamics. The boiling curves were measured for mass fluxes of 73.28, 146.57, 195.42 and 244.28 kg/m$^2$s. The lowest mass flux yielded scattered data points without conveying any clear trend. Such inconclusive findings most likely reflect the stochastic behavior of boiling and the unsteady delivery of flow by the syringe pump.

![Boiling curve for G = 73.28 kg/m$^2$s, values are fairly scattered, no trend observable](image1)

*Fig. 9.3* Boiling curve for $G = 73.28\text{kg/m}^2\text{s}$, values are fairly scattered, no trend observable

![Boiling curve for G = 146.57 kg/m$^2$s with linear approximations for phases liquid, liquid-vapour and vapour](image2)

*Fig. 9.4* Boiling curve for $G = 146.57\text{kg/m}^2\text{s}$ with linear approximations for phases liquid, liquid-vapour and vapour
For mass fluxes from 146.57 up to 244.28kg/m²s, one observes the typical plateau at the saturation temperature. Additional heat goes into vaporization of the water without increasing the temperature itself. The water is completely vaporised and the steam superheated over 190W for 146.57kg/m²s and 200W for 244.28kg/m²s respectively.

Comparing the saturation temperature from the boiling curves (in all cases 100°C) with the absolute pressure in the channel for these temperatures (a value calculated by deriving the pressure drop from Fig. 9.8 and adding ambient pressure of 101.320kPa) one can see that the water should actually not boil as the temperature is still below the saturation temperature for the corresponding absolute pressure. The saturation
temperature for the measured pressure in the channel of 131.3kPa is about 107.4°C but the boiling plateau is already at 100°C. Probably a pressure drop development that reaches ambient pressure in the channel already was the cause for that. This question is addressed in more detail in the next chapter.

9.3 NON-ISOTHERMAL PRESSURE DROP MEASUREMENTS

The pressure drop in the channel was determined for several flow temperatures for the fixed mass flux of \( G = 244.28 \text{ kg/m}^2\text{s} \). One can expect that the pressure drop decreases for increasing flow temperature due to lower viscosity until the onset of boiling where it increase again due to the formation of vapour and the acceleration of the flow. Because the relation between dynamic viscosity and temperature is not linear (Fig. 9.7), the pressure in dependence of temperature is also not linear.

![Fig. 9.7 Dynamic viscosity of liquid water for 101.3kPa as a function of temperature](image)

Fig. 9.7 shows the pressure drop for different heat inputs. One can clearly see the decrease in pressure drop towards the saturation temperature which is reached for about 170W of electrical power input into the cartridge heater. Up to a heat input of about 200W the pressure drop stays the same at which juncture the water has reached the saturation temperature of about 100°C. As soon as the heating power is increased above 200W the pressure drop increases dramatically due to the formation of vapour in the channel.
The equilibrium vapour pressure of water can be approximated by Clausius-Clapeyron

\[
\ln \left( \frac{p_{\text{sat},1}}{p_{\text{sat},2}} \right) = \frac{\Delta h}{R} \left( \frac{1}{T_{\text{sat},2}} - \frac{1}{T_{\text{sat},1}} \right)
\]  

(9.1)

with \( p_{\text{sat},1}, T_{\text{sat},1} \) the known and \( p_{\text{sat},2}, T_{\text{sat},2} \) the unknown pair of saturation pressure and temperature, \( \Delta h \) the enthalpy of evaporation and \( R \) the general gas constant. By solving for the unknown pressure \( p_{\text{sat},2} \) the equation (9.1) yields

\[
p_{\text{sat},2} = p_{\text{sat},1} \cdot e^{\frac{\Delta h}{R} \left( \frac{1}{T_{\text{sat},2}} - \frac{1}{T_{\text{sat},1}} \right)}
\]  

(9.2)

The solutions for temperatures up to 140°C for water are shown in Fig. 9.9.
There were severe pressure fluctuations during experiments which made it difficult to come up with an averaged value. The patterns that were mostly observed are discussed below. Fig. 9.10 shows a steady outlet temperature but the pressure transducer shows great changes even though it is single-phase flow. Even more startling are the results when $P=160W$ and the mass flux is 73.28kg/m$^2$s (Fig. 9.11). The pressure baseline slightly decreases after $t=40$ min.

![Fig. 9.10](image1.png)  
**Fig. 9.10** Temperature and pressure drop profile for $P=150W$ and $G = 73.28$ kg/m$^2$s

![Fig. 9.11](image2.png)  
**Fig. 9.11** Temperature and pressure drop profile for $P=160W$ and $G = 73.28$kg/m$^2$s

In Fig. 9.12 the pressure fluctuations are relatively high for low temperatures below 95°C. For higher temperatures however, they increase. The sharp pressure drop at $t=86$ min and at $t=117$ min suggest that a bubble of dissolved gases blocked the channel and was pushed through it thereby drastically decreasing the pressure drop. The temperature profile shows little bumps after $t=135$ min. The temperature goes up to 100.8°C and falls back down again to 99.9°C. Possible explanations are constrained by the uncertainty of the thermocouple readings with an error of about ±2°C. Even though the temperature variations are very small and might also be signal noise, the definite trend is obvious. Especially in combination with the corresponding pressure fluctuations which are in phase with the temperature bumps.
The fluctuations shown in Fig. 9.13 are most likely due to a dissolved gas bubble in the entrance region of the channel. Many times it was discovered that dissolved gases would coalesce to form a bubble large enough to fill out half of the inlet reservoir despite the fact that the de-ionized water was degassed before usage for at least 15 minutes. The sharp pressure drop at $t = 20 \text{ min}$ and $t = 53 \text{ min}$ indicates that the bubble was pushed through the channel and a new blockade started to amass.

For an electrical heat input of 190W both the temperature and the pressure drop fluctuate considerably over time. The pressure drop and the temperature increase are in phase. Fig. 9.14 shows how the temperature rises every time the pressure drop decreases, e.g. at $t = 15 \text{ min}$ and $t = 26 \text{ min}$. The temperature falls back to almost 100°C as indicated by the dotted baseline. It is most likely the case that a gas/vapour bubble constricted the flow in the in- or outlet reservoir encouraging the remaining smaller mass flux to more aggressively heat up in the channel. As soon as the bubble got pushed out of the way, the pressure dropped and cooler flow from the rest of the channel flushed through and allowing the temperature to decrease to 100°C. It took about 35 seconds for the temperature to fall from 105°C to 100°C and to rise again to 105°C. The time between two peaks at 100°C was about 12 minutes.
two-phase flow measurements

Fig. 9.14 Temperature and pressure drop profile for $P=190\text{W}$ and $G = 146.57\text{kg/m}^2\text{s}$

Fig. 9.15 and Fig. 9.16 show the typical in-phase behaviour of temperature increase and pressure drop. Each significant decrease in pressure drop shows a sudden temperature increase. Fig. 9.16 shows pressure patterns similar to those shown in Fig. 9.13 but for two-phase flow instead. It is noteworthy that for $t = 17.5\text{min}$ the temperature rises before the pressure does. A bubble most likely blocked the channel at least in part and increased the pressure drop at the channel inlet. But at the same time the absolute pressure behind the blockage as well as the mass flux may have been much lower which would explain the sudden temperature increase.

Fig. 9.15 Temperature and pressure drop profile for $P=220\text{W}$ and $G = 244.28\text{kg/m}^2\text{s}$

Fig. 9.16 Temperature and pressure drop profile for $P=210\text{W}$ and $G = 244.28\text{kg/m}^2\text{s}$

A similar behaviour was observed for $G = 244.28\text{kg/m}^2\text{s}$ as shown in Fig. 9.17. Interesting is the periodic manner in which the temperature increases and the pressure
drops. The period length varies between 10 and 13 minutes. This scale is in the range of several minutes whereas the time scales of similar phenomena reported in literature for single and multi-channel devices are of the order of hundreds of second [21], [66] and [51]. However their hydraulic diameters range from 82.8µm up to 348.95µm.

Another phenomenon confronted was reversed flow. This happens when the pressure fluctuations are so large that vapour is pushed upstream back into the reservoir. This usually happens at high heat fluxes. For electrical heat inputs above 200W this behaviour could be observed for \( G = 244.28 \text{ kg/m}^2\text{s} \) as shown in Fig. 9.18 to Fig. 9.20. The blue line represents the inlet temperature \( T_{\text{fluid in}} \) in the aluminium block. The thermocouple is about 40mm upstream of the entrance of the block. This and the fact that the outlet temperature is very steady when reverse flow occurred, indicates that the phenomena could not happen towards the end of the channel but more likely in the inlet reservoir. The pressure fluctuations are probably damped along the channel so that the flow in the outlet region of the block is not affected. The time between two temperature/pressure peaks is about 60 seconds which is about the same order of magnitude than what Wu et al. [66] reported for their experiments with 31 seconds and 140 seconds. The pressure fluctuations however are reported to be 13kPa only whereas in my experiments they reach values of up to 100kPa. Unfortunately there is no data available for the fluid temperature in the inlet ports of the channel as the corresponding thermocouple went dead after a series of experiments. Therefore we could only measure the temperature outside of the block before the fluid entered. Nevertheless as the presented pressure drop data was obtained with the original block design, we can be sure that boiling already occurred in the inlet reservoir as the heat would distribute uniformly within the block and vaporization would start in the inlet reservoir. This idea is supported by the inlet temperature fluctuation which was measured 40mm upstream the inlet to the block. As even the fluctuations measured outside the block reached already around 100°C, the temperature in the inlet reservoir is definitely at least 100°C which would cause boiling. As a consequence there is already two-phase flow in the inlet reservoir and the inlet port to the block. The idea of two-phase flow in the inlet port is supported that temperature fluctuations of up to 40°C were measured with a thermocouple 40mm upstream of the inlet. The absolute values of the pressure drop
fluctuations in vapour-phase are slightly higher than the steady-state pressure drop in the channel at two-phase flow at 100°C.

**Fig. 9.18** Temperature and pressure drop profile for $P=205\,W$ and $G = 244.28\,kg/m^2s$

Similar observations were made for $P=210\,W$ and $P=220\,W$ as shown in **Fig. 9.19** and **Fig. 9.20** respectively. The time from peak to peak both for temperature and pressure in for 210W as well as for 220W is approximately 60 seconds which is in good agreement with data presented by Wu et al. [66].

**Fig. 9.19** Temperature and pressure drop profile for $P=210\,W$ and $G = 244.28\,kg/m^2s$

**Fig. 9.20** Temperature and pressure drop profile for $P=220\,W$ and $G = 244.28\,kg/m^2s$
9.4 PRESSURE GRADIENT ALONG CHANNEL

The observation of a reduced saturation temperature is also supported by several other experiments. Two examples with a mass flux of 73.28 kg/m$^2$s, electrical power input of 210 W and a mass flux of 244.28 kg/m$^2$s, electrical power input of 240 W are shown in Fig. 9.21 and Fig. 9.22.

One can clearly see the plateau at 100°C between $t = 40$ min and $t = 46$ min which indicates boiling. Additional heat does not increase temperature but isothermally vaporizes all of the water. The value of 100°C is surprising as the pressure drop is between 10 and 23 kPa which means that the saturation temperature according to Clausius-Clapeyron (9.1) should be between 102.6°C and 105.4°C. Instead we observe boiling at a temperature that is typical for atmospheric pressure.
In the figure above the isothermal plateau is also at 100°C even though the pressure drop indicates an even higher saturation temperature between 120.0°C and 124.6°C. The pressure transducer was checked after the experiment and worked fine. The thermocouple was checked and seemed to work alright too. It is more reasonable that something blocked the channel which had the effect of a choke that limited the flow and increased the pressure drop upstream. Downstream from the choke, the total pressure was less than upstream and corresponded to the saturation temperature of 100°C. Obviously the pressure drop increased to a more or less stable value of around 80kPa and then suddenly increased even more before boiling was observed. This indicates that due to out gassing a bubble formed at around \( t = 8 \text{ min} \) in the inlet reservoir and constricted the flow.

The described pressure drop patterns above and the boiling curves in chapter 9.2 poses question about the possible pressure drop development along the channel. Unfortunately only the inlet pressure could be measured so that only speculations can be given and a most possible scenario can be presented. In general there are three different possibilities.

1. Linear pressure profile along the whole channel
2. Little single-phase pressure drop in liquid, big pressure drop in two-phase flow
3. Choked flow, all the pressure drop occurs at beginning of channel

The following analysis is based on the experimental data from Fig. 9.22 with \( P = 240W \) and \( G = 244.28 \text{ kg} / \text{m}^2 \text{s} \) at the time \( t = 10 \text{ min} \) and a pressure drop of 104.32kPa which results in an absolute pressure of 119.01kPa.
The gray line is the temperature-profile along the channel extracted from measurements shown in Fig. 9.22 at the specific time $t = 10\, \text{min}$. The data points show an average value calculated with two data points before 10 minutes and two data points after the 10 minutes passed. That means that the profile is time-averaged over 2 seconds with an uncertainty of the thermocouples of $\pm 2^\circ\text{C}$. The red line indicates the development of a linear pressure profile assuming that pressure drop only occurs in the channel but not in inlet and outlet passages. This value is also averaged over five consecutive values or a period of 2 seconds. The maximum value of 205.7\, kPa is the value at $t = 10\, \text{min}$ from Fig. 9.22. The blue line shows the development of the saturation temperature according to the pressure drop in red. It was calculated based on the Clausius-Clapeyron equation (9.1). Taking the argument that boiling occurs as soon as the liquid reaches saturation temperature for the corresponding pressure one would assume that boiling should have started at the intersection of the blue line and the grey line at $x = 46.2\, \text{mm}$. However Fig. 9.22 indicates that boiling occurred at an outlet temperature of 100°C. Assuming that we have an infinite heat transfer coefficient $h$ so that $\Delta T$ between the wall and the fluid is zero, boiling can therefore occur as soon as the wall reaches the 100°C which is at $x = 36.0\, \text{mm}$ as shown in Fig. 9.24. Automatically the pressure drop adjusts and ambient pressure is established at $x = 36.0\, \text{mm}$. Assuming that pressure cannot fall below ambient pressure, the pressure drop in the vapour phase must be very
small. This might be possible if the available liquid that vaporizes is very little. Taking
into account the uncertainty of the thermocouple of \( \pm 2^\circ C \), the wall would have reached
the desired temperature at \( x = 34.2 \text{mm} \) already. The experiments with the high speed
camera showed that boiling always occurred upstream of the heater rather than
downstream which seems to be in agreement with the x-location indicated below.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{fig924.png}
\caption{Grey is the temperature-profile and red the pressure-profile assuming a linear pressure drop.
Provided boiling occurs at 100°C, the pressure drop profile changes and the saturation profile
as well, boiling is now possible at \( x = 36.0 \text{mm} \)}
\end{figure}

It is also possible, as mentioned above that the pressure drop mainly occurred in the
two-phase flow region and only very little in the liquid single-phase flow as shown in
Fig. 9.25. The single-phase flow pressure drop was measured during an experiment at a
temperature of about 80°C which matches the averaged temperature of \(~81^\circ C\) between
position \( x = 0 \text{mm} \) and \( x = 36 \text{mm} \) pretty well. The averaged dynamic viscosity
for \(~81^\circ C\) was assumed to be \( \nu = 0.0004 \cdot 10^{-3} \text{Ns/m}^2 \). The saturation temperature
decreases slowly according to the equilibrium vapour pressure until \( x \approx 36 \text{mm} \) and then
drops suddenly down to 100°C. Another possibility would be that the flow is
constricted or choked by particles so that the pressure drop occurs right at the beginning
of the channel. In this case, the fluid would warm up until it reached saturation
temperature for the corresponding absolute pressure in the channel and would boil. Of
course, any pressure profile is possible which provides ambient pressure downstream of
\( x = 36.0 \text{mm} \). Unfortunately there does not exist a possibility yet which measures in situ
the pressure drop along the channel which would be needed to come to a clear
conclusion.
9.5 HEAT LOSSES

The heat loss (9.3) was calculated by applying a certain power input to the block and waiting until steady state was reached. Then water was fed through the block and the difference in of inlet fluid temperature and outlet fluid temperature was used in the energy equation (9.4) to calculate the transferred heat. The working fluid was always single-phase. The development is shown in Fig. 9.26 for a mass flux of 244.28 kg/m²s. One can observe the linear dependence of $\Delta T$ regarding heat input. The heat losses are extremely high as no insulation of the aluminum block was used. Most losses are radiation and natural convection losses to the ambient air. The average heat loss for single phase flow was 99.77%. The heat loss for two-phase flow is very hard to determine as measurement of vapour temperature and quality is not easy. The heat loss in two-phase flow may be approximated by extrapolating the single-phase heat loss with an error of 20%.

\[ Q_{\text{loss}} = Q_{\text{Heater}} - Q_{\text{Fluid}} \]  

(9.3)

with

\[ Q_{\text{Fluid}} = \bar{c}_p \Delta T \cdot \dot{m} \]  

(9.4)

where $\bar{c}_p$ is the averaged heat capacity over the range of $\Delta T$ and $\dot{m}$ the mass flow rate.
Fig. 9.26  In grey the temperature increase for a given heat input, in red the relative heat loss, all data for mass flux of $G = 244.28 \text{kg/m}^2\text{s}$.
Bubble dynamic studies were performed with a high speed camera attached to the microscope. The recording was triggered with LabView that also took temperature and pressure measurements. The volume flow rate was controlled in real-time with a separate computer. Before pictures were taken the block was heated up and allowed to reach steady state while the syringe pump was running. Steady-state was defined as a temperature change of the wall temperatures less than 1°C in one minute. Magnifications of 4, 6, 10 and 30 were used with the microscope. The light source was white light. The resolution of the camera was varied between 512 x 128 and 512 x 64. Shutter times of 500µs down to 15µs were applied which yields 2’000 frames per second and 67’796 respectively. Before each run the ambient temperature was checked and the CJC-voltage was adjusted in LabView.

The recorded videos were saved as multi-page tiff files which allowed easy handling and analysis. As the conducted experiments produced more than 60GB of recorded data, a simple and effective method to extract bubble departure time and bubble growth rate had to be found. Unfortunately the Matlab edge detection scheme could not be used as the roughness of the channel bottom made it impossible to get clear enough images. Especially reflections on the channel’s ground did create a background that was highly non-uniform as shown in Fig. 10.1.

Even a change of the lightening did not allow taking pictures that had a contrast high enough for automated edge detection. The pictures above cover a time span of 21.94µs. Every couple microseconds the camera recorded significantly darker images with less contrast as seen in the last row above. This made it even more difficult for an automated image analysis. For the mentioned reason the interface tracking was done manually with the camera software. Each recorded video was analyzed for interesting bubble growth scenes. The video was converted from the camera manufacturer file extension *.cin to a
multi-page tiff file *.tif. The tif-file could now be saved as a sequence of single pictures. As for most experiments the shutter time was set rather too short than too long in order to catch any detail of interest, the produced sequence of tif-images had to be filtered to reduce data. The reduced number of tif-images were combined to one video again and processed with the camera software.

All the bubble measurements were done manually. The departure time as well as the growth rate was extracted by sense of visual judgment. The error in determining the bubble diameter is about $\pm 4\mu m$.

A systematic uncertainty related to the bubble dynamics is, that the pressure drop development along the channel is unknown. As described in chapter 9.4, it is most likely that the pressure drop reached ambient pressure before leaving the channel otherwise the temperature plateau would not have been at 100°C but at a higher temperature according to the vapour pressure. Assuming that the same situation prevailed while taking bubble dynamics data, it is impossible to calculate the wall superheat temperature as the local pressure is unknown and the numbers given in Table 10.1 are speculative. Even though the calculation according to Clausius-Clapeyron would yield in many experiments to a sub-cooled liquid flow in the channel, bubble nucleation was observed. One could still argue that this is not boiling but degassing that has been encountered.

10.1 BUBBLE GROWTH

As discussed in the chapter Literature Research 6.2.5, there have been different attempts to model bubble growth in micro-channels. First approaches came from macroscale experiments as the Rayleigh equation

$$\frac{dr}{dt} = \left( \frac{2}{3} \frac{\Delta p_w}{\rho_l} \right)^{1/2} \approx \left[ \frac{2}{3} \frac{(T_w - T_{sat}) \Delta h_v}{T_{sat} \rho_l \rho_v} \right]^{1/2} \quad (10.1)$$

which gives a correlation for homogenous superheated bulk mediums. In macroscale boiling however, the magnitude of the relevant forces are quite different compared to microscale. This is because of the strong effect of buoyancy and inertia in larger scales where bubbles usually do not reach the other wall before they depart. In microscale, buoyancy is less important as the bubble is usually constrained by the geometry of the channel in an early stage of the growing process. In return, surface tension plays a major role which also determines the bubble departure diameter. Further complexity to microscale boiling is added by the boundary condition of usually three heated walls and the thin boundary layer that influences bubble growth. A scale analysis of the forces in microchannels with a hydraulic diameter of $47\mu m$ was conducted by Fogg et al. [15]. They distinguished between four different forces acting on the bubble and stated the force balance in channel direction $x$

$$F_x = F_\sigma + F_p + F_\mu + F_p = 0 \quad (10.2)$$
The qualitative surface tension is
\[ F_\sigma \sim -\sigma P_c \cos \theta \] (10.3)
with \( \sigma \) the coefficient of surface tension, the contact perimeter \( P_c \) and the contact angle \( \theta \).

The liquid inertia is
\[ F_\rho \sim \rho u_{in}^2 A_{ch} \] (10.4)
with the liquid density \( \rho \), the entrance velocity \( u_{in} \) and the cross sectional area of the channel \( A_{ch} \).

The viscous or drag forces on the bubble depend in the velocity gradient at the bubble surface and its area. Assuming only significant flow around the sides of the bubble but not above or below the bubble, due to the high aspect ratio channels they used, the viscous force can be modelled as flow between two flat plates
\[ F_\mu \sim \mu_l L_b H_{ch} \frac{du}{dr} \approx \mu_l L_b u_{in} \frac{A_{ch}}{A_{ch} - A_b} \] (10.5)
with \( \mu_l \) the liquid viscosity, \( L_b \) the bubble length, \( H_{ch} \) the channel height and \( A_b \) the bubble cross sectional area.

The pressure force scales as follows
\[ F_p = \frac{dP}{dx} L_b W_b H_b \approx \frac{\mu_l C u_{in} L_{ch} W_b}{2 H_{ch} \left( 1 - \frac{W_b}{W_{ch}} \right)} \] (10.6)

Inserting values of their experimental conditions yield
\[ \frac{F_p}{F_\sigma} = 105 \] (10.7)
\[ \frac{F_\mu}{F_\sigma} = 0.36 \] (10.8)
\[ \frac{F_\rho}{F_\mu} = 290 \] (10.9)

which identifies the pressure force as the dominant force that overcomes surface tension. Viscous forces and inertia play a minor role as opposed to macrochannel experiments.

It is therefore not surprising that equation (10.1) gives values that do not match experimental data from micro-channel experiments. The predicted growth rate is way too large compared to observed trends reported by Lee et al. [35].
In general one would expect a linear bubble radius development that increases with larger heat input and decreases with higher mass flux. This behaviour is partly supported by data from Lee et al. [35] shown in Fig. 10.2. Obviously the data at the bottom does not follow the trend. The bubble grows fastest even though the heat input is lowest.

![Fig. 10.2](image)

**Fig. 10.2** Increasing bubble growth for higher heat inputs in a 41.3μm single channel reported by Lee et al. [35], the data at the bottom does not match the trend as the bubble grows extremely fast for the lowest heat input.

However, there are also opposite trends in literature which show smaller bubble growth for higher heat input which seems rather against intuition. This behaviour is reported by Li et al. [37] and shown in Fig. 10.3. The investigators gave two linear empirical approximations for bubble growth

\[ R(t)_{G=269\text{kg/m}^2, q=143\text{kW/m}^2} = 6.40 + 0.334t \] (10.10)

\[ R(t)_{G=269\text{kg/m}^2, q=303\text{kW/m}^2} = 5.19 + 0.199t \] (10.11)

which indicate faster bubble growth for lower heat flux.
In order to correctly predict the bubble growth in microchannels, investigators derived new correlations. Copper [8] proposed a relation based on an evaporating microlayer underneath the bubble with heated walls.

\[ R(t) = 2.5 \frac{Ja}{Pr^{1/2}} (\alpha t)^{1/2} \]  

(10.12)

\[ Ja = \frac{\rho_v c_{p,\ell} (T_w - T_{sat})}{\rho_i \Delta h_i} \]  

(10.13)

The bubble growth in equation (10.12) is proportional to the square root of time. This behaviour is in contrast to the linear bubble development reported by Lee et al. [35]. However one experiment from the series shown in Fig. 10.3 by Li et al. [37] proves a bubble growth which follows a function dependent on the square root of time. The corresponding data is shown below.
The foregoing discussion showed how discordant the investigators are about bubble growth phenomena in micro-channels. This makes it difficult to formulate expectations for the experiments that I conducted.

In the following it is assumed that the pressure drop develops as modeled for choked flow in chapter 9.4. This is reasonable as the recorded bubbles were always encountered upstream of the heater. The accomplished experiments are presented in Table 10.1. If the bubble growth did not follow a linear trend it was approximated by two linear lines, each with an own growth rate. The number behind the slash represents the averaged growth rate. It is remarkable that not a single experiment suggested a trend proportional to the square root of time as proposed by Copper [8].

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The two-dimensional bubble departure diameter $D$ that is reported is considered to be

$$D = \sqrt{\frac{4A}{\pi}}$$  \hspace{1cm} (10.14)
where $A$ is the visible cross sectional area. In the following the bubble dynamic study is divided in two groups. One group has similar mass fluxes but different heat inputs and the other group has similar heat inputs but different mass fluxes. The next paragraph only presents the data whereas the chapter 10.2 and 10.3 offer conclusive results.

### 10.1.1 Constant Mass Flux and Varying Heat Input

In Fig. 10.5 the bubble growth rate is shown for a mass flux $G \sim 260\text{kg/m}^2\text{s}$ and electrical power inputs between 190.8W and 224.2W. All bubble growth rates are linear as reported by literature. It seems reasonable that the blue line has a lower slope than the black one at the same electrical power input its mass flux is larger. In general the larger the mass flux at same heat input, the lower the growth rate should be. But the rest of the lines do not seem to follow this trend. In contrast they tend to go to one growth rate, independent of the heat input.

Fig. 10.5 Bubble growth data for $G \sim 260\text{kg/m}^2\text{s}$ and different heat inputs

Fig. 10.6 shows the growth rate for $G \sim 291\text{kg/m}^2\text{s}$ and a fixed power input of 203.3W and 206.0W respectively. Even though all parameters remain the same and even though all four experiments were conducted at one day after another, the results vary significantly. The orange line (EN7) has a slightly higher slope than the black line (EN3) which seems reasonable as its heat input is also slightly higher. But the large difference of departure diameter cannot be explained by the small difference in heat input. The red (EN11) and grey lines (EN10) do not correspond at all with the black and orange line. A close up view of the red and grey line is shown in Fig. 10.7. Both lines seem to show the same trend. This suggests that there was inherent difference between the black/orange- and the red/grey-experiments. Both lines (red, grey) tend to a
departure diameter of 120 µm. Interestingly both lines follow more an exponential trend than a linear trend.

Fig. 10.6 Bubble growth data for $G \approx 291 \text{kg/m}^2\text{s}$ and different heat inputs

Going to higher mass fluxes would suggest that the growth rate decreases if heat input stays constant. Fig. 10.8 however shows no such trend. It is remarkable that for all
different heat inputs the linear slope stays the same. The departure diameter is about 45-50µm for all heat inputs.

![Bubble growth data for G~365kg/m²s and different heat inputs](image)

Fig. 10.8  Bubble growth data for G~365kg/m²s and different heat inputs

For $G \sim 420$kg/m²s the bubble diameter develops as shown in Fig. 10.9 There are two trends observable. Those represented by the black (EN13), grey (EN14) and red (EN15) line and those represented by the orange (EN25) and green line (EN26) respectively. It seems reasonable that the bubble radii of the black, grey and red line grow slower than the other two as their electrical power input is lower and their mass fluxes are even higher. The difference in growth rate and departure diameter however is surprising.
For a nearly constant mass flux and identical heat input, the growth rate should be the same. However Fig. 10.10 does not match with the prediction. The bubble growth rate for the grey and red line seems almost the same even though the departure diameter and the starting diameter are quite different. Surprisingly the black line is steeper despite the lower heat input.
10.1.2   **CONSTANT HEAT INPUT AND VARYING MASS FLUX**

The data in Fig. 10.11 spreads over a wide range of growth rates even though the mass flux is only slightly changed for a constant electrical power input. It makes sense that the grey line (EN12) is steeper than the black one (EN9) as its heat input is higher but it should not be the case that the green (EN10) and blue (EN11) line differ so much. A close up view of the green and blue line is shown in Fig. 10.7. The fact that the orange (EN6) line is steeper than the red, grey and black one seems reasonable as its heat input is high at a fairly low flow rate.

![Fig. 10.11 Bubble growth data for P~202W and different mass fluxes](image)

One would assume that the higher the heat input the faster the bubble grows. This trend is not supported by Fig. 10.12. The opposite is observed; the lower the mass flux the slower the bubble growth which appears incorrect. Moreover it is strange that the difference in time scale is more than three orders of magnitude between the black (EN2) and the grey (EN13) line as shown in Fig. 10.13. Another trend that also seems unusual can be deduced by taking the grey, red and orange line. Their heat inputs are almost the same but their mass fluxes vary significantly. The departure diameter increases for increasing mass fluxes from the orange (EN4), over the red (EN5) up to the grey line. This goes against intuition, as the drag force on the bubble due to increased mass flux should decrease the departure diameter. Maybe the flow was choked and the measured mass flux was not maintained throughout the whole channel which would explain why the grey line is so steep and the departure diameter much too large.
Fig. 10.12 Bubble growth data for P~210W and different mass fluxes

Fig. 10.13 Detail of bubble growth data for P~210W, x axis rescaled up to 0.25 seconds

A similar trend to Fig. 10.12 can be observed in Fig. 10.14. The higher the mass flux the faster the bubble grows in radius which seems strange. The departure diameter varies between 40µm and almost 100µm without following a distinct trend. This behaviour might also be explained with a choked channel.
Fig. 10.14 Bubble growth data for P=222W and different mass fluxes

The bubble growth represented by the black (EN18) and the grey line (EN19) are similar which makes sense as heat input as well as mass flux are identical. The difference in departure diameter however cannot be explained. The fact that the red line (EN31) is steeper than the grey and black one makes sense as the mass flux is lower and the bubble therefore grows faster. The departure diameter seems small compared to the grey line. In general one would expect the departure diameter to decrease with increasing mass flux as the drag force on the bubble increases.
The data for an electrical heat input of about 272W is shown in Fig. 10.16. The trend that at lower mass flux the bubble growth speed should increase is observed. Only the data for the mass flux of 793.3kg/m$^2$s does not fit in there. These growth rates are the highest even though the mass flux is also the highest.
10.2 **Bubble Growth Rate**

The values given in the following are all averaged values. There were three experiments with growth rates of more than 30,000 µm/s which are not considered in the following graph.

![Graph showing bubble growth rate](image)

*Fig. 10.17 The dots represent the averaged bubble growth rate at a specific combination of mass flux and power input.*

The diamonds represent the average bubble growth rate for one specific pair of heat input and mass flux. The stems are drawn for a better orientation as they indicate the position of the data points in the x-y-plane. In the figure above no consequent trend is observable. If one ignores some points along a constant mass flux line, there seems to be an increase in bubble growth rate for increasing power input. Nevertheless, this trend is not very decisive and it would need more data points as a proof. Intuitionally one
would assume an increase in bubble growth rate for increasing heat input and decreasing mass flux which is also reported in literature by Lee et al. [34].

10.3 Departure Diameter

The forces acting on the bubble dictate the departure diameter. For the force balance one should be able to use the macroscale approach. The complexity that occurs with a microscale force analysis is based on the geometrical constraints and the magnitude of the relevant forces which differs between the two scales. In Microsystems surface forces become dominant compared to body forces such as buoyancy, which is an important physical mechanism in macroscale bubble dynamics.

Levy [36] developed a model to predict the bubble departure size $D_b$ in macrochannels based on a force balance between buoyancy, surface tension and flow drag

$$
D_b \propto \frac{\sigma}{\sqrt{g \left( \rho_l - \rho_g \right) + C \tau_w / H_b}}
$$

where $\tau_w$ and $H_b$ are the wall shear stress and the bubble height respectively. As the wall shear stress is proportional to the square of the flow rate, the bubble departure size is mainly the result of a balance between buoyancy, surface tension forces and the flow drag. This is different to microchannels where buoyancy is negligible as the bubble fills up the entire space between the bottom and the top surface. Therefore bubbles depart in microchannel when the flow drag force, which depends on Reynolds number only, is sufficiently high.

An increasing bubble departure diameter with temperature and a decreasing with increasing mass flux was reported by Thorncroft et al. [58]. As soon as the pressure drop across the bubble, created by the bulk fluid drag force, overcomes the surface tension, the bubble detaches. Several researchers [34], [15] proposed an exponential relation of bubble departure volume and Reynolds number and bubble departure diameter and Reynolds number respectively, as seen in

Fig. 10.18 Bubble departure volume normalized by channel height versus Reynolds number for different values of wall superheat, reported by Lee et al. [34]
Fig. 10.19 shows the non-dimensional departure volume and the non-dimensional departure diameter from the conducted experiments as a function of the Reynolds number. The normalization was done with the depth of the channel. Both data sets tend to follow exponential trends as indicated in the figure below.

Unfortunately the data points above do not follow the exponential trend as much as reported in literature by other investigators. In addition the reported data in literature is for Reynolds numbers up to 0.6 whereas the conducted experiments had Reynolds numbers up to 196.

An empirical relation for isothermal bubble departure diameters was proposed by Fogg et al. [15]

\[
\frac{D_b}{W_{ch}} = 0.058 \frac{W_{ch}}{H_{ch} \, Re_H} + 0.18
\]  

(10.16)

which is gives reasonable values only for Reynolds numbers between 0 and 0.35. Another relation for non-isothermal bubble departure diameters is suggested by Lee et al. [34]

\[
\frac{V_D}{H^2} = 2 \cdot 10^4 e^{-5Re_{c25}}
\]  

(10.17)

which applies also only for Reynolds numbers up to 0.6. For higher values the exponential trend is washed out and follows a linear trend.
The reasons why the recorded data does not follow the exponential trend very accurately are manifold. There were many uncontrollable variables in the experiments such as nucleation site activity, roughness of channel and pressure drop development across the channel which influenced the values of the bubble departure diameter. Moreover the uncertainty in temperature measurements and the problems with degassing surely had an influence too on the reliability of the data presented above.
11 **Encountered Challenges during Experiments**

Rigorous is the nature of flow boiling experiments in a micro-channel device. What might not be deemed a problem in a macro-channel application can present as an insurmountable hurdle when dealing in the small scale. During my experimental work, I encountered many problems which I here aim to discuss in greater detail. Hopefully, my experiences will not have been in vain and the Stanford Thermosciences research group of which I have been a part will benefit as equally from my setbacks as from my achievements.

11.1 **Challenges during Test Section Design**

Three design iterations were necessary in producing the desired performance. The sealing of the channel posed a grave challenge. No level of accuracy of the milling machine, no tool available seemed would suffice in achieving a proper seal. Polishing the top surface of the block in combination with a polycarbonate cover instead of glass finally provided a leakage-free device. Also, the uniformity of the applied heat created difficulty. The primarily chosen configuration applied the heat throughout the whole block with a very slight temperature gradient along the channel. It was discovered that the residential time in the inlet reservoir was too long for all applied mass fluxes of up to $793.3 \text{ kg/m}^2\text{s}$ since the water at such a level would problematically start boiling before ever entering the test section. In order to achieve a wall temperature pressured gradient along the channel, material was taken away along the channel to build high thermal resistance so that heat would concentrate instead of spreading throughout the whole block. Also, active cooling of the entrance section was considered but seemed to be too complicated.

After extensive use of the test device, it was observed that the water in the channel leaked underneath the acrylic cover. As it did not leave the control volume, it did not change the mass balance. It did however find its way parallel to the micro-channel. Observable too were retrospectively telling particles deposited by the water on the block surface after extensive boiling experiments. Further polishing would have been necessary to provide proper sealing again but at the same time the hydraulic diameter would have changed.

11.2 **Problems with K-Type Thermocouples Measurements**

For the six wall temperatures as well as for the three fluid temperatures, k-type thermocouples were used. It became clear that the uncertainty of k-type thermocouples precludes the accurate tracking of bubble nucleation. The fluctuations of $\pm 2^\circ\text{C}$ made it
impossible to reveal the effect of vapour formation on the wall temperature. In addition to this, the large thermal mass of the test section increased the time constant and averaged out local cooling phenomena so that they could not be tracked down. Tangential to this topic of thermal mass, it seems fitting to remark here that those thermocouples applied about 1.5mm underneath the channel experienced reduced reaction time. Complex too was the process of correctly calibrating the thermocouples. Due to an insufficiency of DIFF-capable analogue channels, the calibrations had to be performed without the aid of the Cold-Junction-Compensation-sensor on the DAQ-board.

11.3 PRESSURE DROP MEASUREMENTS AND FLUCTUATIONS

Extensive isothermal pressure drop measurements were performed to determine the channel depth. However severe pressure drop scatter was observed. Variations of up to 52% were encountered. It is still not understood what mechanism caused this wide variation in collected data. Another unexplained issue pertained to the period of time necessary for reaching steady state conditions in the channel. It seemed surprising that at least 30 minutes were needed to reach a more or less steady state pressure drop. Running times up to 60 minutes were sometimes necessary. It is believed that the simple design of the test section which allowed the cover to be demounted was also part of the uncertainty in pressure drop measurements. Little particles or abrasions from the metal screws as well as from the aluminium block might have been trapped in the channel which strongly influenced the measured pressure drop. Similar pressure drop measurement behaviour was encountered during flow boiling experiments.

11.4 ANALYSIS OF RECORDED HIGH SPEED SEQUENCES

In order to deduce the bubble dynamics in the flow boiling micro-channel experiment, a high speed camera was used. Due to the fabrication process of the test section, the bottom of the channel was very rough; the side walls in no way visible as sharp contrasting lines. This was a major problem in analysing the recorded sequences. It made it impossible to use standard edge detection codes like the one provided by Matlab. It became obvious that the fluorescence method might have been a solution to this problem. On the other hand, the fluorescence particles might have agglomerated in the channel during boiling and caused a change of the geometrical parameters of the test section.
Flow boiling in a single micro-channel heat exchanger was investigated. A high speed camera was used to visualize bubble nucleation processes. The test section was conventionally machined with an end mill in an aluminium block. The length of the micro-channel was 50mm. The width was set by the used end mill to 155µm. Pressure drop measurements suggested a depth of 44µm yielding a hydraulic diameter of 68µm. Mass fluxes of de-ionized and degassed water between 143.4 and 793.3kg/m²s were applied. The electrical power input was varied between 188 and 272.9W which equalled a net heat input between 0.432 and 0.628W. The block was equipped with a cartridge heater and k-type thermocouples to measure six wall temperatures and three fluid temperatures.

Boiling curves and pressure dependence of temperature were recorded. Temperature and pressure measurements were synchronized with pictures of nucleating bubbles which were taken with a high speed camera with up to 67’796 frames per second. The bubble growth rate was investigated for different heat inputs and mass fluxes but no clear trend was observable. Higher heat inputs did not automatically yield higher bubble growth rates, nor did higher mass fluxes result in smaller growth rates. Instead, the latter rates varied between 10.2µm/s and as much as 32’588µm/s. However, the finding reported in literature with regards to bubble departure diameter and the bubble departure volume was by experiments tested and further supported.

It was observed that all bubbles indeed start to nucleate on the side walls of the channels. It was evident that leakage flows of water entered the channel underneath the polycarbonate cover at the side walls. It is possible that evaporation had already occurred underneath the cover out of the channel and initiated nucleation when partially vaporized water re-entered the channel at the walls.

The test section was very sensitive regarding wear. Small geometrical changes had a significant impact on the boiling characteristics. This fact and the highly stochastic nature of boiling made it hard to guarantee repeatability of boiling experiments.

Surprisingly, the saturation temperature was 100°C in many experiments even though the pressure drop across the channel was two bars and would have suggested a much higher saturation temperature. It is believed that the pressure drop was highly non-linear across the channel and that the ambient pressure corresponding to a saturation temperature of 100°C had already been reached within the channel. This might have been caused by the polycarbonate cover which decreased the channel size in some experiments as it reached glassy state due to the high wall temperatures.
During the design of the test section one should pay particular attention to the thermal properties of the device. It is crucial in the experiments that a temperature gradient is maintained along the channel. There are at least two different methods for achieving that. Firstly, one could remove enough material from the side of the channel to create a higher thermal resistance so that heat conduction is decreased. Inherently problematic to this approach however are the tensions that arise due to different thermal expansion coefficients of the aluminium and the polycarbonate cover. This can lead to the deformation of the block which in turn affects the test device and the pressure drop therein. Secondly, the entrance region could be actively cooled.

In addition, the volume of the inlet tubing within the aluminium block should be minimized in order to reduce the residential time in the entrance region and minimize preheating.

The sealing of the channel with an acrylic or polycarbonate cover should be achievable when the top surface is polished. The holes for the screws have to be positioned as close to the channel as possible to be able to build up a high pressure with the screws for proper sealing.

The channel should be marked with ticks to allow the determination of the region where the high speed sequences were recorded. A laser-based method would probably be most suitable for the marking.

In order to control the heating power more accurately, a digitally controlled power supply unit should be used for the cartridge heater.

The temperature measurements should be more accurate. With this aluminium configuration the problem of the large thermal mass of the block averages out all fluctuations due to bubble nucleation. The fact that there is at least 1mm of aluminium between the channel bottom and the thermocouple tip makes it especially hard to track temperature developments accurately.

Another suggestion encourages one to measure water temperatures along the channel and not just at inlet and outlet position. This could be done by using a measurement technique based on temperature sensitive fluorescence.

In order to minimize uncertainties in pressure measurements, it might be useful to set-up the experiment as a closed-loop system. In that case one could also control the outlet pressure and actively regulate the pressure gradient along the channel.

An automated edge detection scheme seems unfeasible in combination with an aluminium-based test section. The channel roughness is too pronounced and counter-
productively too conducive to reflection and scattering for any automated edge detection scheme to successfully process the data in a reliable way.
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