FLOW BOILING OF A HIGHLY VISCOUS POLYMER SOLUTION

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Abstract

Flow boiling experiments with mixtures of polydimethyl siloxanes and n-pentane are carried out in a vertical annular channel. The mixtures are fed to the test section with a subcooling of 30 – 65 K. The flow condition at the inlet can be characterized as creeping flow with Re-numbers as small as 0.01. Strong radial temperature gradients prevail throughout the entire evaporation process, indicating a laminar flow character even under boiling conditions.

Analysing the experiments, heat transfer to a homogeneous liquid phase, subcooled boiling, fully developed nucleate boiling, convective heat transfer to a two-phase flow and flash effects due to pressure loss can be distinguished.

Yet heat transfer can be described as a first approximation by a simple model which is based on laminar single phase heat transfer and incorporates an additional convective term reflecting the augmentation of convection through the gas phase.

Nomenclature

A	cross section of the flow channel (m²)	$\dot{M}, \dot{m}$	mass flow (kg/s), mass flux (kg/m²s)
c	specific heat (J/kg K)	p	pressure
d	diameter of the flow channel (m)	$\dot{Q}, \dot{q}$	heat rate (W), heat flux (W/m²)
α	heat transfer coefficient (W/m²K)	T	temperature
H, h	enthalpy (J), specific enthalpy (J/kg)	z	axial coordinate
λ	thermal conductivity (W/m K)	ϕ	volume fraction of polymer in the liquid phase

Indices

b	bulk	m	mixture
g	gas phase	o	outer wall
h	hydraulic	sp	single phase
i	inner wall	w	wall
lg	evaporation		

Dimensionless groups

Nu	Nusselt number	Pe	Peclet number	Re	Reynolds number

1. Introduction

Flow boiling of highly viscous fluids is used in many technical installations in order to separate high-molecular substances from solvents and unreacted monomers. Nevertheless only very few publications exist on this topic. Ritter (1969) examined the evaporation of ethylbenzene from mixtures with polystyrene. More recently, Junhong et al. (1995) presented an experimental study using mixtures of sodium carboxymethylcellulose (CMC) and water. These systems

characteristically consist of a non-volatile (polymer-) component which is responsible for the high viscosity and of a second low-molecular component which is evaporated. Ritter (1969) recognized the laminar flow character and fitted his data to global heat transfer relations for laminar flow. Junhong et al. (1995) pointed out the importance of viscosity and presented a correlation based on the mechanisms of nucleate boiling and convective flow boiling which incorporates the fluid viscosity as an independent variable.

In this paper, an experimental study using polydimethyl siloxane (PDMS, silicone oil) and n-pentane as a model system, is presented.

2. Experimental setup

The experiments were carried out in a vertical annular channel with an overall heated length of 2.100 mm. The test section (8) consists of seven identical segments. Heat is supplied to each segment individually by electric heating of the inner cylinder. The inner cylinder has a diameter of 10 mm, whereas the outer diameter of the flow channel is 20 mm. Each segment is equipped with a sight glass section allowing the visual observation of the flow. In order not to influence the flow by heat transfer via the sight glasses, the test section is positioned in a temperature controlled chamber (16). The outlet of the flow channel is connected via an adapter piece (9) with an inner diameter of 20 mm to the vacuum chamber (10). The adapter piece is equipped with a choke valve and a compensator. Further details of the test section have been published by Wienecke et al. (2005).
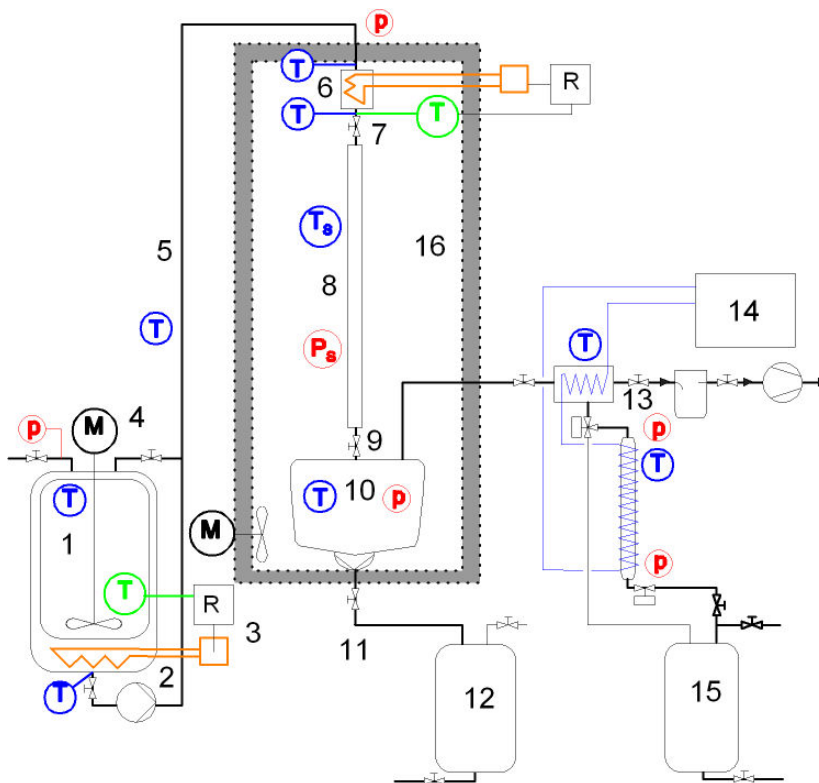

Figure 2.1: *Flow sheet showing the arrangement of the test rig: (1) storage tank with (2) feed pump, (3) heating facility and (4) stirrer for mixing oil and solvent, (5) heated supply pipe, (6) heat exchanger with flow mixer, (7) choke valve, (8) test section, (9) choke valve with compensator, (10) separator with discharging pump and vapour pipe, (11) installation for taking samples, (12) oil collector, (13) condensation system for the vapour solvent, (14) refrigeration machine, (15) solvent collector, (16) temperature controlled chamber*

In each segment the wall temperature of the heated surface is measured by means of 0.5 mm thermocouples which are soldered into the inner tube. Fluid temperatures are measured by an arrangement of three thermocouples (diameter 0.25 mm) extending 5 mm parallel to the flow (see Figure 2.2).
The thermocouples are placed at distances of 1 mm, 2.5 mm and 4 mm from the heated surface. This setup eliminates the influence of heat conduction inside the thermocouple on the measurement.
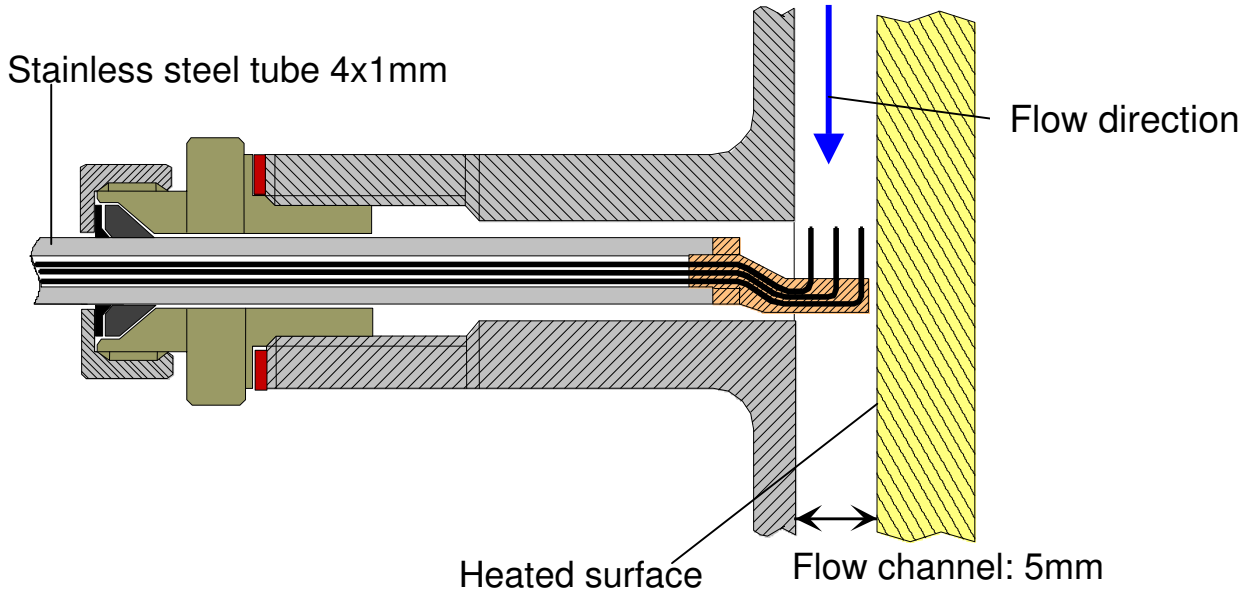

Figure 2.2: *Measurement of fluid temperature with an arrangement of three thermocouples*

The accuracy of the thermocouple measurement is ± 0.2 K and the position of the thermocouple tip in the flow channel could be determined with an accuracy of ± 0.05 mm.
Pressure transducers are placed at the same axial position as the temperature measurement on the opposite side of the annulus. Their accuracy is ± 27 mbar.
For each segment the heat supply is determined by measurement of the electric currant with an accuracy of ± 1%.

3. Experimental results

Through the sight glasses, the same flow patterns as in boiling flow of low viscosity media were observed: single phase flow, bubble flow, slug flow and annular flow. The range of experimental parameters is listed in Table 1:

Table 1: range of experimental parameters

	Low	High
Volume flow rate	3 l/h	15 l/h
Heat flux	2.7 kW/m²	9 kW/m²
Solvent concentration	9%	27%
Viscosity PDMS	100 Pas	1.000 Pas

PDMS supplied by GE Bayer Silicones (M 100.000 with a viscosity of 100 Pas at room temperature and M 1.000.000 with 1.000 Pas) and commercial n-pentane of 99% purity were used in these experiments.

The PDMS/n-pentane solution was preheated to a temperature of 50°C. Therefore, the fluid was always in a subcooled state when entering the flow channel. This is emphasized by the fact that the highly viscous flow causes a considerable pressure drop. Typically, the entrance pressure ranges between 3 and 7 bar absolute pressure, yielding a boiling point of 80 to 115°C at the tube inlet.
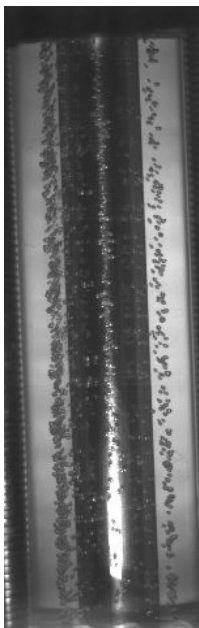
Second segment of the test section
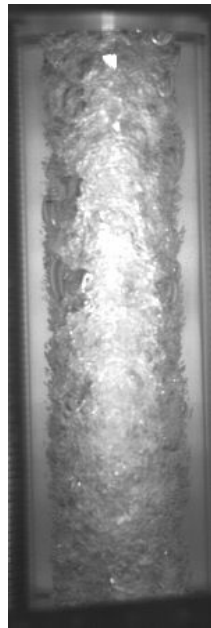
Third segment of the test section

Figure 3.1: *Photographs showing subcooled boiling in the flow channel*

Figures 3.2 and 3.3 show the course of the quantities measured along the flow channel for the high parameter settings (volume flow rate 15 l/h, heat flux 9 kW/m², solvent concentration 27% and PDMS viscosity 1.000 Pas). The vertical grid lines mark the seven segments of the flow channel. The eighth measuring position is situated in the adapter piece between the test section and the vacuum chamber.
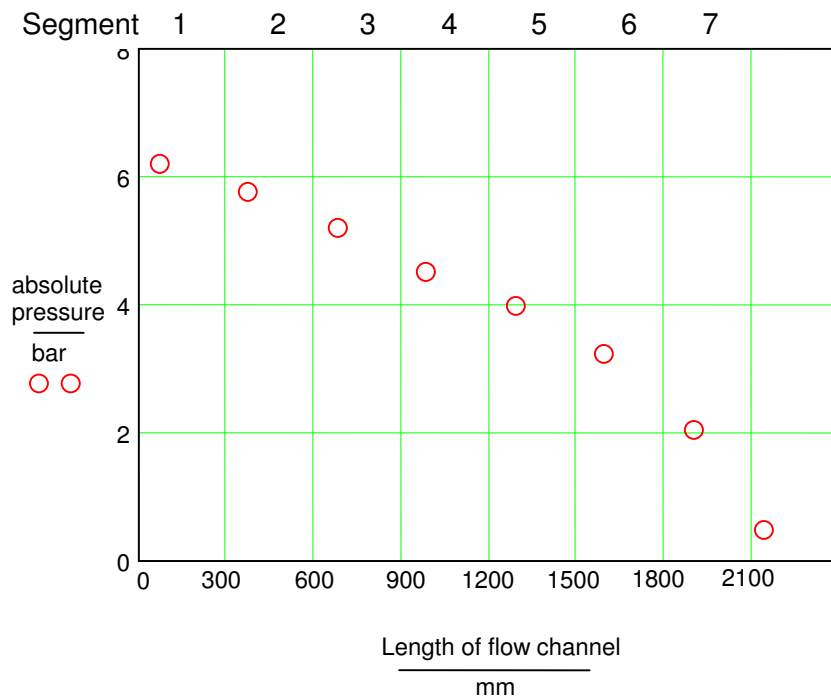

Figure 3.2: *Pressure profile*

The axial pressure gradient increases along the flow path, reflecting the increasing polymer concentration. In fact, the effect of the solvent evaporation on the viscosity of the liquid phase is much stronger than that of the increasing temperature.
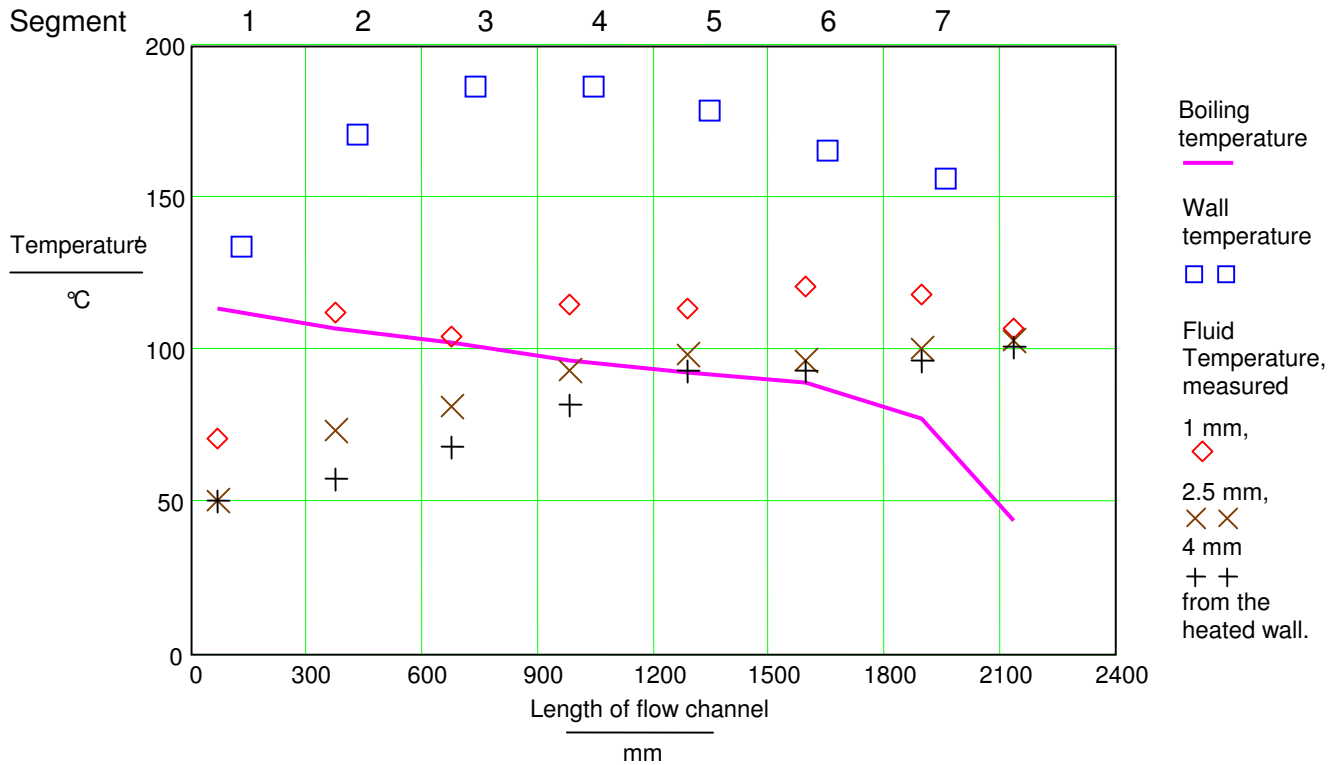

Figure 3.3: *Temperature profiles along the flow channel*

The temperature profiles (Figure 3.3) show steep radial temperature gradients in the fluid with a peak after one third of the flow path, where the temperature difference across the channel is greater than 100 K. Only at the exit of the tube the radial temperature gradients become rather small.

4. Theory

The Re-number at the tube entrance is on the order of 0.01. Thus, forces of inertia are negligible compared to viscous forces and the flow can be characterized as creeping flow. Therefore, the Re-number is meaningless for the presented problem. However, it points out the strong contrast to industrial flow boiling applications using water or refrigerants, with a fully turbulent flow and a Re-number of 10^5.

As illustrated in chapter 3, strong radial temperature gradients prevail from the entrance to the exit of the tube even under boiling conditions. Therefore, it can be concluded that the flow has a laminar character in the bubbly regime as well. Heat transfer to a laminar flow is consequently taken as a base case for our considerations.

The heat transfer coefficient is defined as follows:

$$\alpha := \frac{\dot{q}}{(T_w - T_b)} \tag{1}$$

Given constant physical properties, the exact analytical solution to the problem of single phase heat transfer to laminar flow in an annulus is provided by Lundberg et al. 1963. Due to the length of the exact solution, an approximation (2) is used in this paper. This asymptotic approximation is formally identical to the approximation by Gnielinski (1997) for round tubes.

$$Pe := \frac{\dot{M}_m \cdot c_m \cdot d_h}{\lambda_m \cdot A}$$

$$Nu := \frac{\alpha_{sp} \cdot d_h}{\lambda_m} = \left[6.181^3 + 1 + \left[1.67 \left(Pe \cdot \frac{d_h}{z} \right)^{\frac{1}{3}} - 1 \right]^3 \right]^{\frac{1}{3}} \tag{2}$$

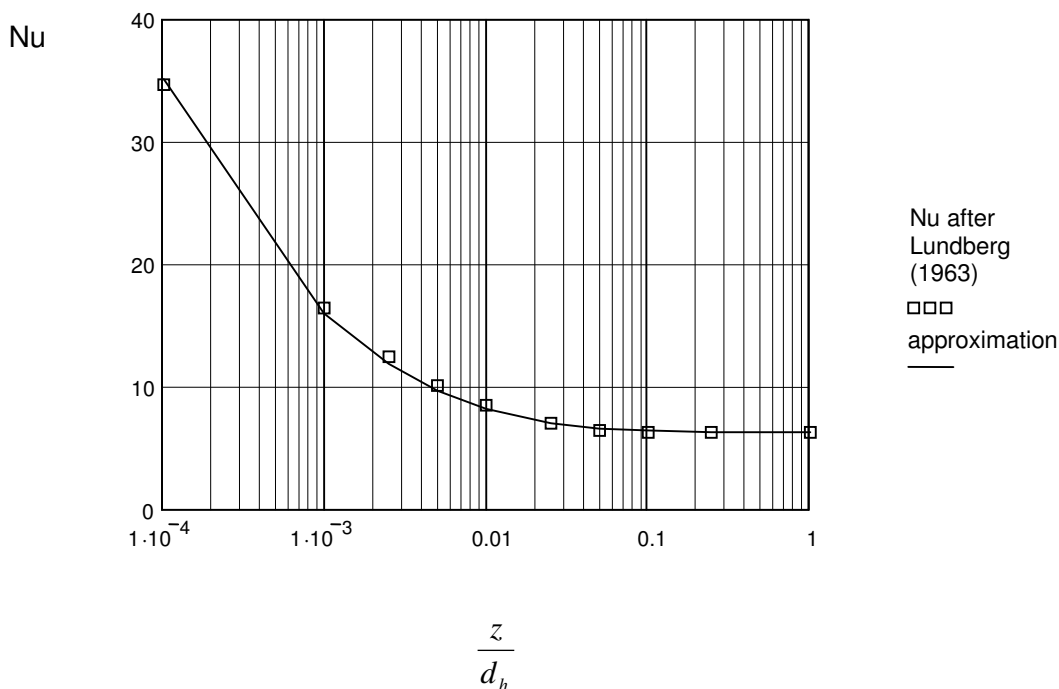It closely approximates the values given by Lundberg et al. (1963):

Figure 4.1: *Single phase heat transfer for an annulus with $d_i / d_o = 0.5$*

Major difficulties are encountered in determining the bulk fluid temperature from the measured temperature profiles, since the radial distribution of mass flow is not known. Because of the laminar character of the flow, heat transfer to a laminar single phase flow is taken as a benchmark. In this case, the bulk fluid temperature linearly increases along the flow path and the mass flux distribution across the flow channel can be calculated. The bulk fluid temperature was calculated based on the measured radial temperature profiles (Figure 3.3) and on the mass flux distribution of a single phase flow. The result is shown in Figure 4.2.

$$T_b = \frac{1}{\dot{M}} \cdot \int_{r_i}^{r_o} T(r) \cdot 2\pi r \cdot \dot{m}(r) dr \tag{3}$$

It is noticeable that, as long as the flow is subcooled, these bulk fluid temperatures are well above the bulk temperatures for single phase flow. Since this is physically impossible, it can be concluded, that the radial velocity profile must be different from single phase flow. In particular the flow is slowed down close to the heated surface because of a locally reduced pentane content which produces an increase in viscosity.

In the following figures (4.2 to 4.6) the vertical grid lines mark seven segments of the test section. Beginning with the fifth segment, the bulk fluid temperature is lower than the single phase temperature and does not increase significantly, indicating that the supplied energy leads mainly to evaporation of pentane.
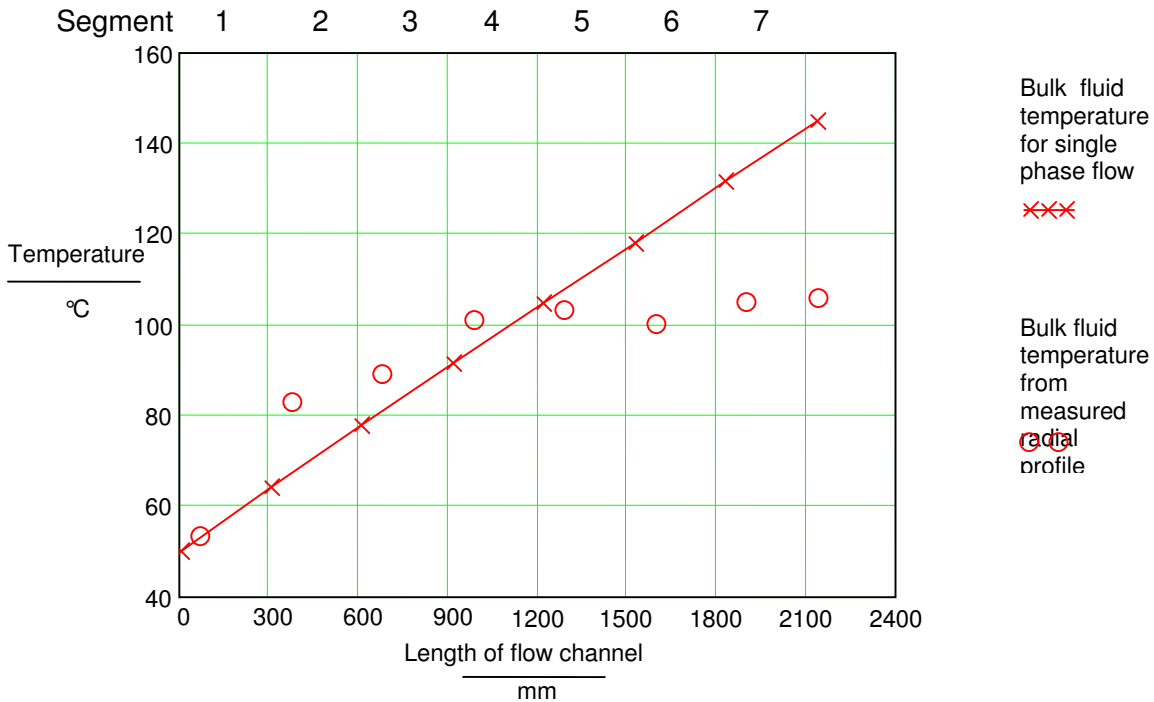

Figure 4.2: *Bulk fluid temperature*

With respect to the wall temperature, it can be stated that wall temperatures are slightly higher in the domain of subcooled boiling than in the case of single phase heat transfer. This can be understood taking into account the distortion of the mass flux profile as concluded above. Starting at the fourth segment, the measured wall temperatures are significantly lower compared to a single phase flow. This indicates an improved heat transfer (Figure 4.3).
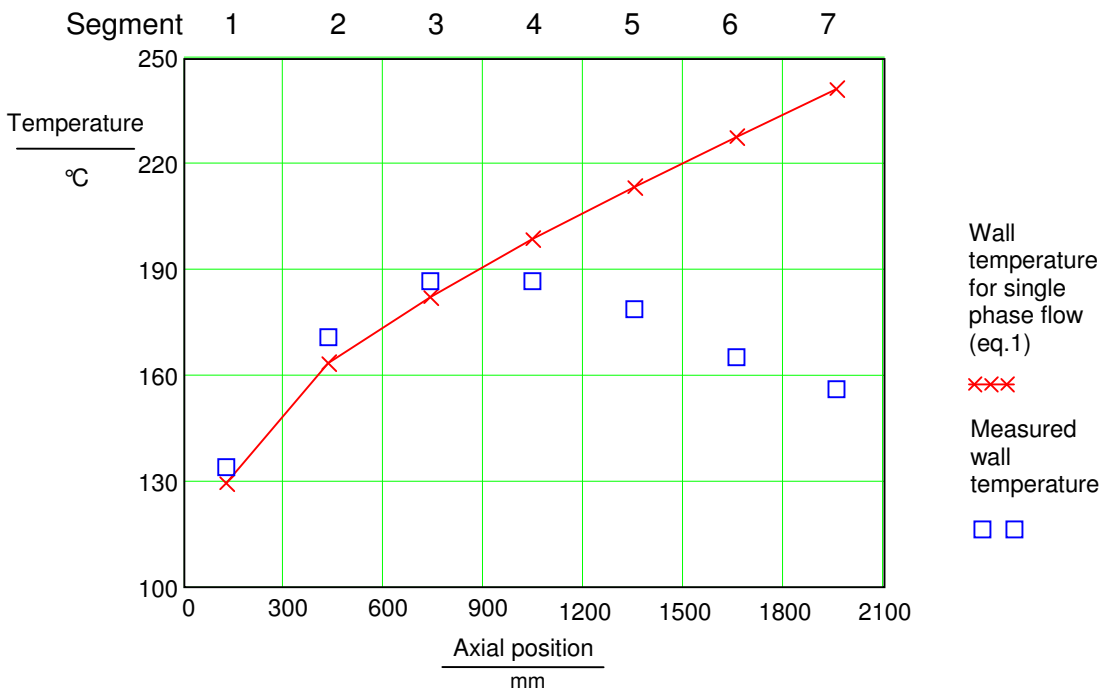

Figure 4.3: *Wall temperature*

In order to create a simple model for highly viscous flow boiling the bulk temperature is assumed to be equal to the bulk temperature in single phase flow as long as it is below the boiling point. This is justified as long as the vapour bubbles recondense in the subcooled liquid. The photographs in chapter 3 show, that this is not exactly true. There is some net vapour content in the subcooled domain as well. But lacking detailed information on the flow field, single phase flow is used as a first approximation. As heat transfer is not enhanced in the subcooled boiling domain, single phase heat transfer is a good approximation.

The vapour pressure of the mixture is dependent not only on the local pressure and temperature but also on the solvent concentration. It can be determined using the Flory-Huggins equation (4) with the volume fraction ϕ of pentane in the liquid phase and the interaction parameter X.

$$p_{vap} = p_{vap,pentane} \cdot (1-\phi) \cdot \exp(\phi + X\varphi^2) \qquad (4)$$

As the mixture reaches the boiling point, the flow transforms to a fully developed bubble flow or foamy flow. Our model assumption is, that in this domain the flow can be described by a uniform foam viscosity. Consequently, the mean temperature can be determined analogous to single phase laminar flow.

The volume fraction ϕ of the polymer in the liquid phase can then be derived from an energy balance which can be determined for each segment of the flow channel. The evaporated amount of liquid is

$$\Delta \dot{M}_{lg} = \frac{\dot{Q} + \dot{H}_{in} - \dot{H}_{out}}{h_{lg}} \qquad (5)$$

Using equation (1) and the calculated bulk temperatures, local heat transfer coefficients were determined for each segment at the point of temperature measurement (Figure 4.4). The heat transfer coefficient of segments two and three, is lower than for single phase laminar flow, because the wall temperature is higher and the bulk temperature equal to or lower than in single phase laminar flow.
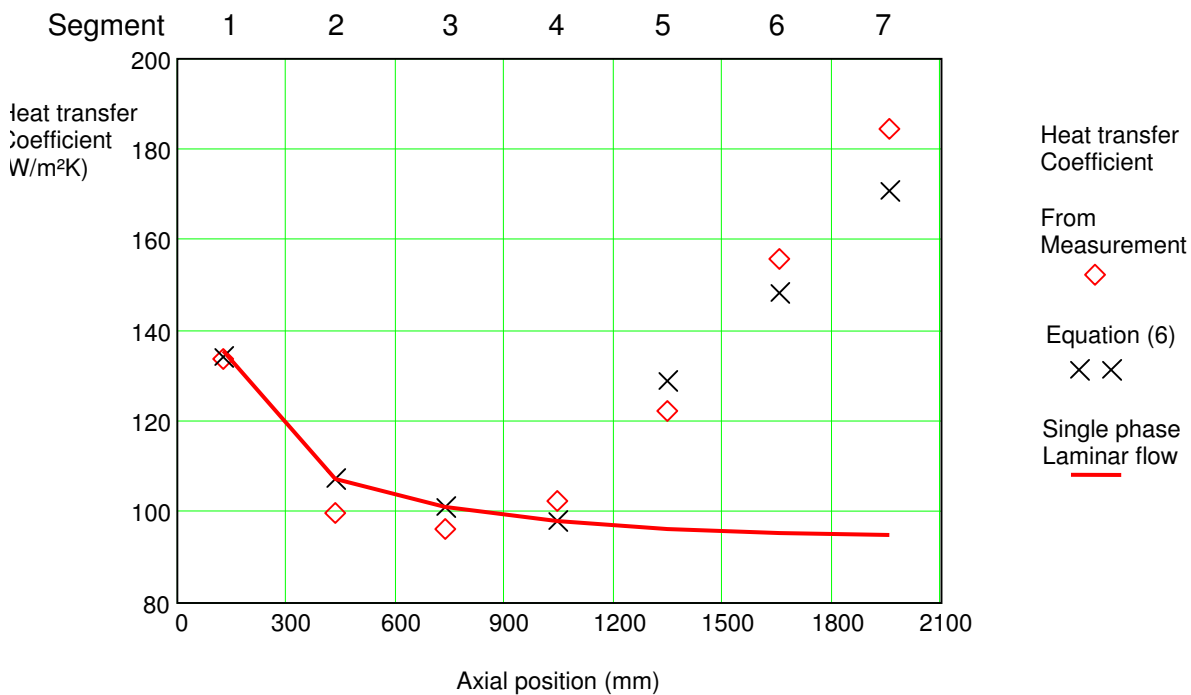

Figure 4.4: *Heat transfer coefficient*

The dependence of the heat transfer coefficient on the specific heat flux is a characteristic feature of nucleate boiling. If heat transfer coefficients are plotted against the axial coordinate for different heat fluxes the heat transfer coefficients diverge. This can be interpreted as a symptom of nucleate boiling. Yet the improvement of heat transfer with increasing heat flux is rather small. An increase of heat flux produces mainly increased temperature differences.
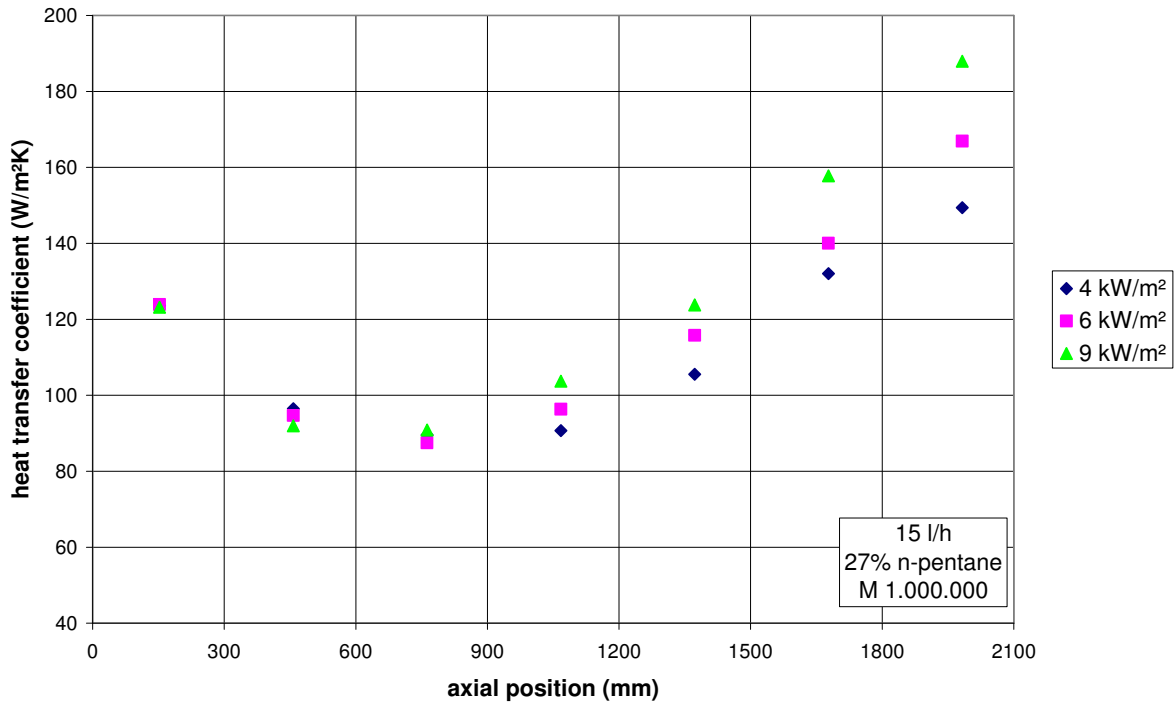

Figure 4.5: *Heat transfer coefficient for different heat fluxes*

On the other hand, our data suggest that heat transfer improves with increasing gas mass flow. The evaporated amount of n-pentane was determined using an energy balance. If the heat transfer coefficient is plotted against the gas mass flow, which was determined by a heat and mass balance, the differences between the varying heat fluxes more or less level out (Figure 4.6).
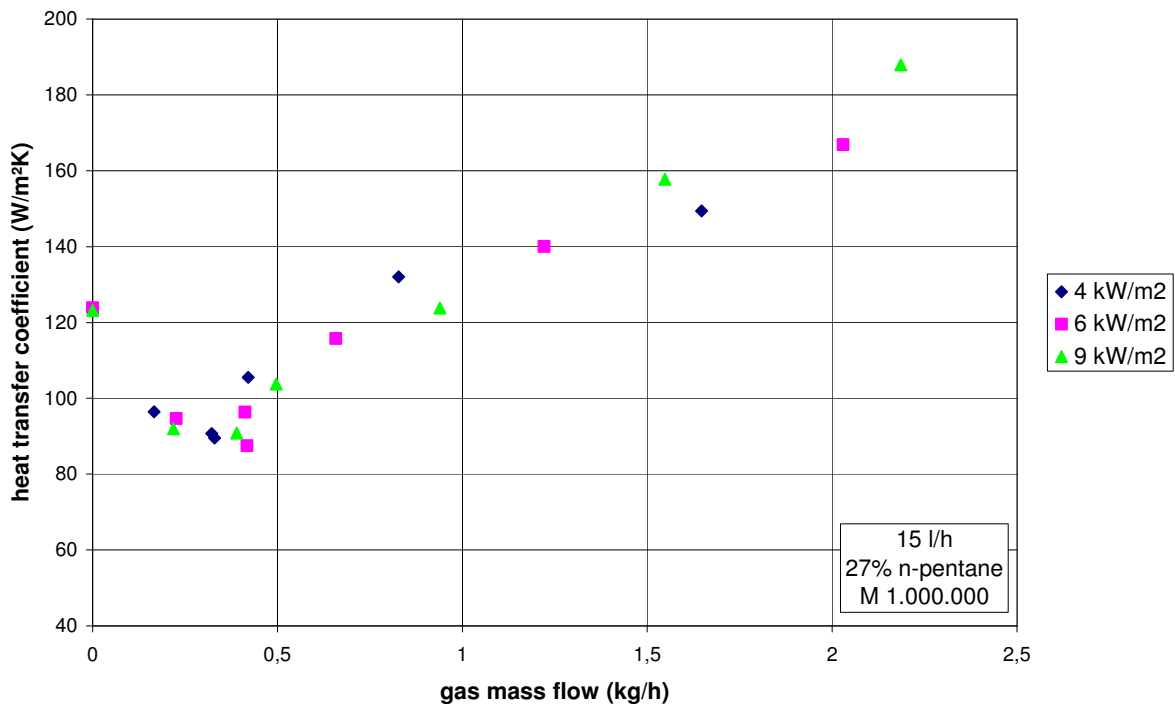

Figure 4.6: *Heat transfer coefficient versus gas mass flow*

According to this interpretation, an increased heat flux produces an increase in evaporation and consequently an increase in the gas mass flow. Thus, heat transfer is promoted primarily by additional convection which is produced by the gas flow. The strong temperature gradient, which also exists in the fully developed bubble flow or foamy flow, indicates that the flow still has a laminar character. Yet, the gas mass flow appears to contribute to convective heat transfer.

In order to describe this behaviour, a Peclet-number is expressed for the gas flow and introduced as an additional term to equation (2) accompanied by an approximation constant:

$$Pe_g := \frac{\dot{M}_g \cdot c_g \cdot d_h}{\lambda_m \cdot A}$$

$$\alpha_{mod} = \frac{\lambda_m}{d_h} \cdot \left[\left[6.181^3 + 1 + \left[1.67 \left(Pe \cdot \frac{d_h}{z} \right)^{\frac{1}{3}} - 1 \right]^3 \right]^{\frac{1}{3}} + 0.008 \cdot Pe_g \right] \tag{6}$$

The heat transfer coefficients shown in Figure 4.5 and 4.6 can be predicted by this simple approach with an absolute error of 4%.
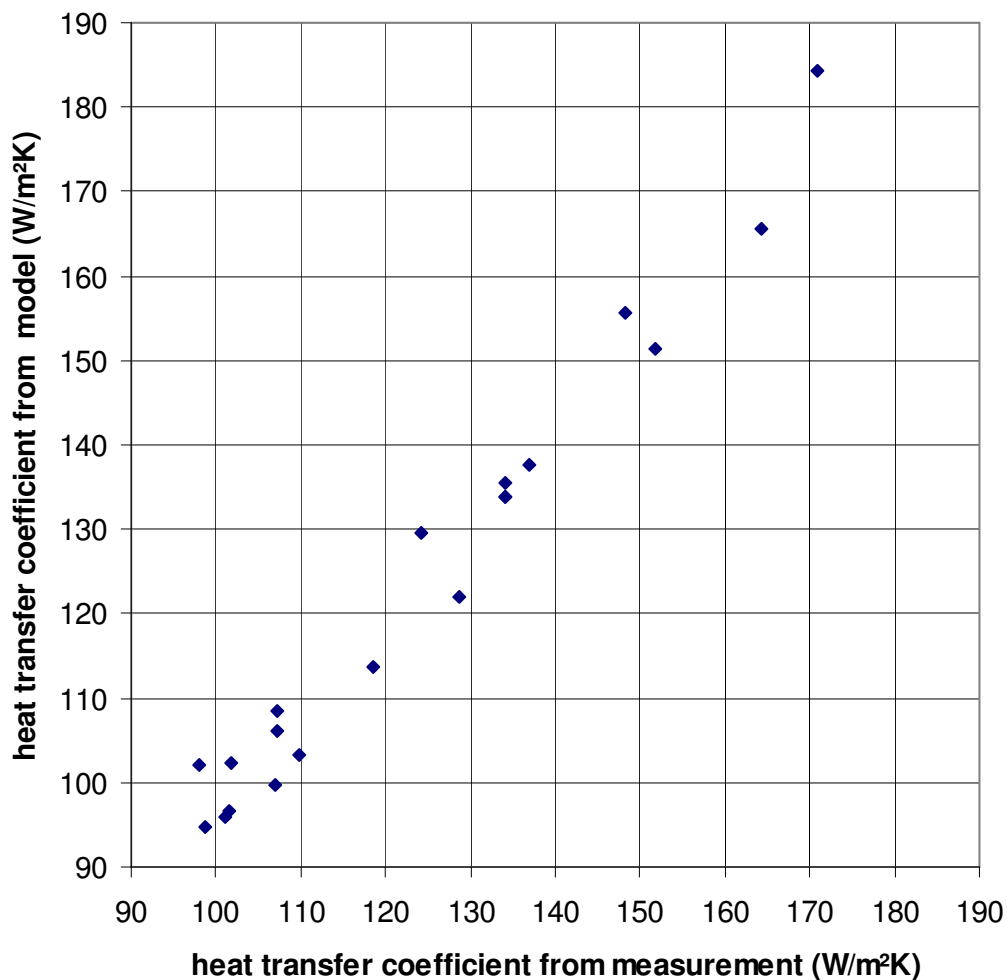

Figure 4.7: *Predicted heat transfer coefficient versus measurement*

To further extend the validity of the model, additional data with varying mass flux and pentane concentration must be evaluated which subject of future work.

5. References

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