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Study on flow characteristics and pressure distribution along a heated channel in subcooled flow boiling

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Abstract

In the present work, an experimental investigation is presented which is focused on the pressure drop along the heated as well as the unheated zone of a flow channel. The experiments were performed by using a 16 mm stainless steel ID tube with a 350 mm heated length followed by a 500 mm unheated extension downstream. Pressure drops and pressure distribution were measured along the flow channel, with water, at mass flow velocities of 6342–9513 kg/m^2 s and at heat fluxes of 3.37–4.1 MW/m^2. The whole experiments were conducted near atmospheric pressure by keeping the outlet of the flow channel open to the atmospheric pressure. The experimental results show a reduction of the pressure at the outlet of the heated zone at the onset of significant void (OSV) conditions, caused by the increasing of the average velocity at that zone. That reduction of the pressure, at the outlet of the heated zone, decreases as the inlet (and also the outlet) temperature increases, due to the increasing of the length of the two phase flow at the unheated zone. That phenomenon is backed by photographing the flow in the unheated zone.

Keywords: Subcooled boiling; Pressure drop; Flow boiling

1. Introduction

The pressure drop in subcooled flow boiling region is an important factor in considering the design of a high heat flux systems such as nuclear reactors, electronic systems, heat exchangers etc. The conditions in which the pressure drop begins to increase during the transient from forced convection heat transfer to subcooled flow boiling, related to the onset of significant void (OSV), were investigated in many works in the past. In a few works, a direct measurement of the pressure drop along the heated channel was conducted in order to determine those conditions of OSV which are leading to flow excursion. Those investigations were focused on the pressure drop along the heated zone and ignored the pressure drop along the outlet zone. That extension of the flow channel is usually an unheated zone where the two phase flow exists.

In an unheated channel ($q'' = 0$), the flow in the channel is single phase, and the curve of pressure-drop vs. flow-rate in the channel has a positive slope (internal pressure drop), as shown in Region #1 of Fig. 1. When a constant heating rate is supplied to the channel at a high flow rate (Region #1), the slope of the curve is still positive. When the flow rate is reduced, the internal pressure drop decreases until a point where the flow in the channel becomes two phase (interface between Region #1 and #2). At that point, the slope of the pressure drop curves becomes negative (S curve). In Region #2, where the slope of the pressure drop is negative the flow is unstable (in a constant driving force). The increasing of the pressure drop causing the decreasing of the flow rate and the increasing of void creation, which leads to Region #3. In Region #3 vapor flow through the channel, which means a low cooling rate and the channel wall will reach burnout.

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Nomenclature

Acronyms
- BD bubble departure point
- OFI onset of flow instability
- ONB onset of nucleate boiling
- OSV O onset of significant void

Symbols
- C_o radial void distribution parameter
- C_p specific heat per unit mass
- D_H hydraulic diameter
- d test section diameter
- G mass flux
- k thermal conductivity
- L heated length of the channel
- p pressure
- Δp pressure difference
- Q volumetric flow rate
- q'' heat flux
- T_o outlet temperature of the heated test section
- T_{sat} saturation temperature

Subscripts
- ext external
- f liquid or bulk mean liquid
- in(1) inlet computed conditions using measurements of point 1
- in(2) inlet computed conditions using measurements of point 2
- in-1 difference between the inlet and point 1
- in-2 difference between the inlet and point 2
- int internal
- out(1) measured at the outlet point no. 1
- out(2) measured at the outlet point no. 2
- sat saturation

In many works, that changing point of the curve slope was determined as an unstable point which led to flow excursion. The flow excursion was first analyzed by Ledinegg [1]. The criterion for the instability was given by

$$\left(\frac{\partial \Delta p}{\partial G}\right)_{\text{int}} < \left(\frac{\partial \Delta p}{\partial G}\right)_{\text{ext}} \tag{1}$$

where $\left(\frac{\partial \Delta p}{\partial G}\right)_{\text{int}}$ slope of internal pressure drop vs. mass velocity, $\left(\frac{\partial \Delta p}{\partial G}\right)_{\text{ext}}$ slope of external pressure drop vs. mass velocity.

This means that the flow excursion occurs, when the slope of the curve of the internal (including the heated test section) pressure-drop vs. flow-rate becomes smaller (more negative) than the external pressure-drop vs. flow-rate curve (or pump characteristic). In many works, flow instability is defined as the occurrence of minimum in the demand curve. That was also the way in which the conditions for achieving flow instability were examined experimentally in a flow system. For a system containing a large number of heated flow channels, connected in parallel at the inlet and at the outlet plane, the external pressure drop is almost constant and the minimum point of the S curve is the point of the onset of flow instability (OFI). Bowring [2] tried to give a relationship between the flow velocity, the subcooling, the pressure and the heat flux in which the OSV occurs. Whittle and Forgan [3] examined the pressure loss in rectangular and circular channels under subcooled boiling conditions. They tried to give a new interpretation to the correlation of Bowring and connected between the OSV and the OFI. They found that for a given L/d ratio of the channel, the minimum in the S curve occurred at fixed values of the ratio between the channel temperature rise to the inlet subcooling. Duffey and Hughes [4] as well as other researchers developed models to predict the pressure drop in subcooled boiling in order to predict the minimum point in the S curve. Those models used a correlation for prediction of the OSV in the channel and a void fraction profile as function of the distance from the point of the OSV. All of those models, as well as the experiments, end in the minimum point of the S curve. Lee et al. [5] examined various models to predict OSV under subcooled boiling conditions. Their results show that the OFI point is almost coincident with OSV. Hoffman and Wang [6] presented a semi-empirical model for prediction of the pressure drop in subcooled flow boiling of water, in tubes subject to uniform high heat flux. That model is based on various correlations from the literature which presented the pressure drop in the single phase liquid regime, the partial developed boiling regime and fully developed boiling regime. In order to model the non-equilibrium volumetric void fraction in the fully developed boiling regime, they used the drift flux model developed by Kroeger and Zuber [7]. They found that the model predicts too large void fraction at low pressure, and they had to modify it by adjusting the distribution parameter (C_o) in

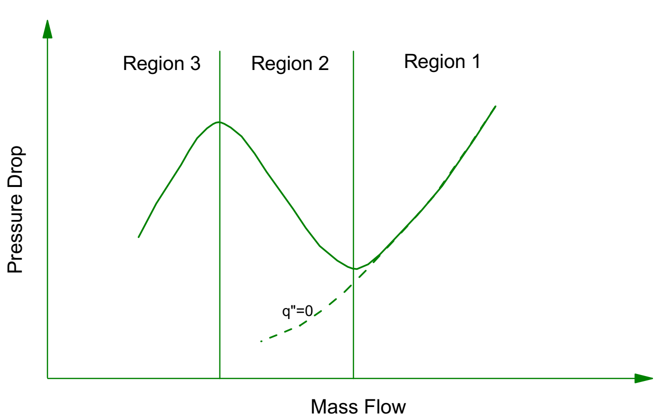

Fig. 1. Pressure-drop vs. flow-rate.

the void fraction model. The model of Hoffman and Wang was compared with various experimental results relating to experiments with small channel diameter up to 4.85 mm. One of those comparisons with experimental results of Owens and Schrock [8] is presented in Fig. 2. The arrows in the figure are presenting the onset of nucleate boiling (ONB) and the bubbles departure (BD) points. Those points were calculated by the authors [6] which used the Bergles and Rohsenow [9] correlation and a modified Saha and Zuber [10] correlation, respectively. As can be seen from Fig. 2, the ONB which was calculated by the correlation of Bergles and Rohsenow, is upstream to the inlet of the channel.

Celata et al. [11] presented results of experimental research on the onset of subcooled boiling in water forced convective flow. From the measurement of the pressure drop along the heated test channel ($d = 8$ mm, $L = 100$ mm), they evaluated the heat flux at which the subcooled boiling occurs. They focus their research on the point and the thermohydraulic conditions at which the pressure drop changes slope and begins to increase. Those experiments were conducted at a pressure of 1–3.5 MPa and a velocity of 5–10 m/s.

All of the above investigators focused on the heated zone of the flow channel and the pressure drop along this zone. They ignored the influence of the exit zone of the channel, which is, always unheated and is acting as a condensation zone. An overall view on the flow channel leads to a better understanding of the pressure distribution along it. It also leads to a correct understanding of the channel behavior as part of a multi channel cooling system in the case of OFI.

In the present work, experiments are presented in order to investigate the behavior of the pressure drop and the pressure distribution along the heated channel, and at the downstream unheated zone. All the experiments were conducted using subcooled flow boiling at low pressure and high heat flux.

This article is based on broader thesis research [12], which describes the experiments and related models in detail. The portion of the experiments that offer new and unique results are presented.

2. Experimental apparatus and test section

A schematic of the experimental loop is shown in Fig. 3. The test loop consists of a water tank of 350 l, a centrifugal pump, a bypass, flow control valves, two orifice flowmeters and a preheater. The loop is made of type 304 stainless steel and filled with deionized water. The water circulated by the pump through the test section and the bypass. The maximum volumetric flow rate of the pump through the test section is 11 m^3/h. The total flow-rate and the flow-rate through the test section were measured by the two orifice flowmeters. The water from the test section and the bypass returned back to the water tank. The exit pressure was always near atmospheric pressure by keeping the water tank open to the atmosphere and due to a ventilation chimney which was connected to the exit of the test section as shown in Fig. 4. Electric pressure transducers were connected to pressure taps along the flow channel and measured the pressure continuously during the tests. The test section was made up of 16 mm ID, stainless steel tube as shown in Fig. 5. The heated and the unheated lengths of the test section were 350–500 mm, respectively. The unheated section of the channel was downstream of the heated zone. In one of the experiments, part of the exit zone was replaced by a Pyrex tube at the same diameter.

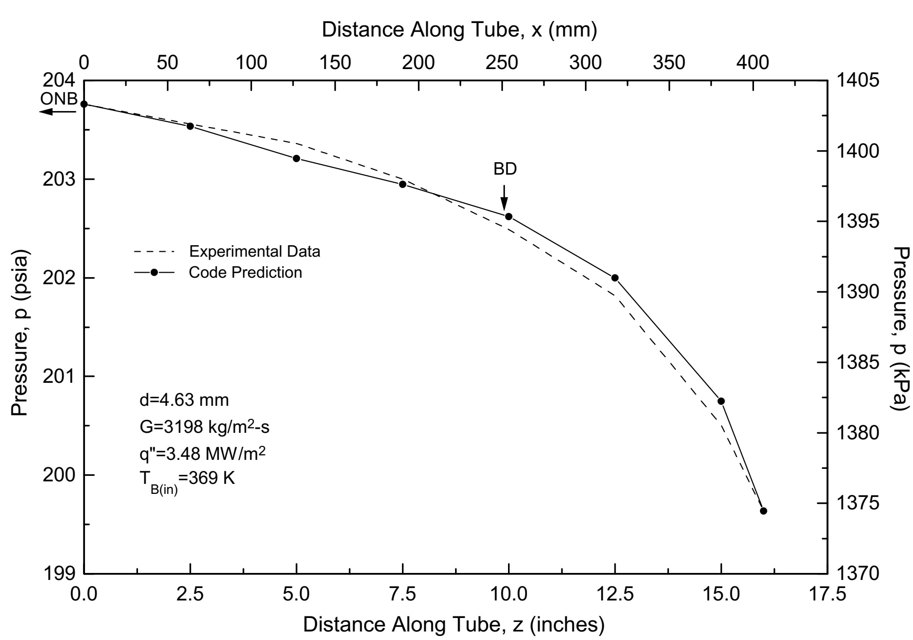

Fig. 2. Comparison of the predicted pressure distribution along the heated channel with a set of experimental results [6].

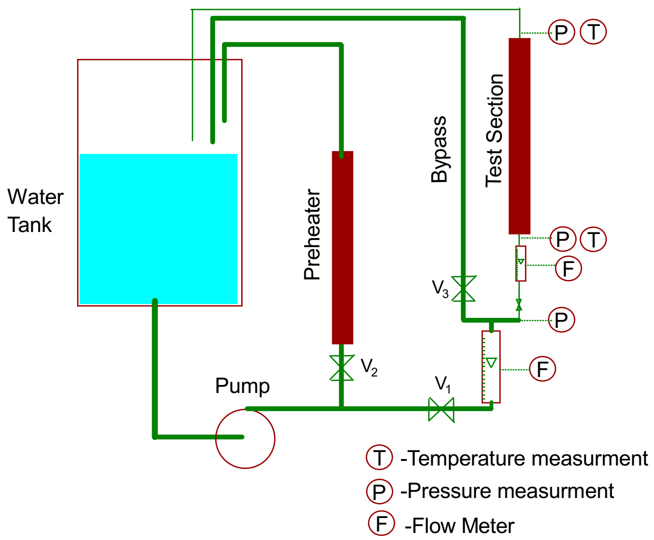

Fig. 3. The test loop.

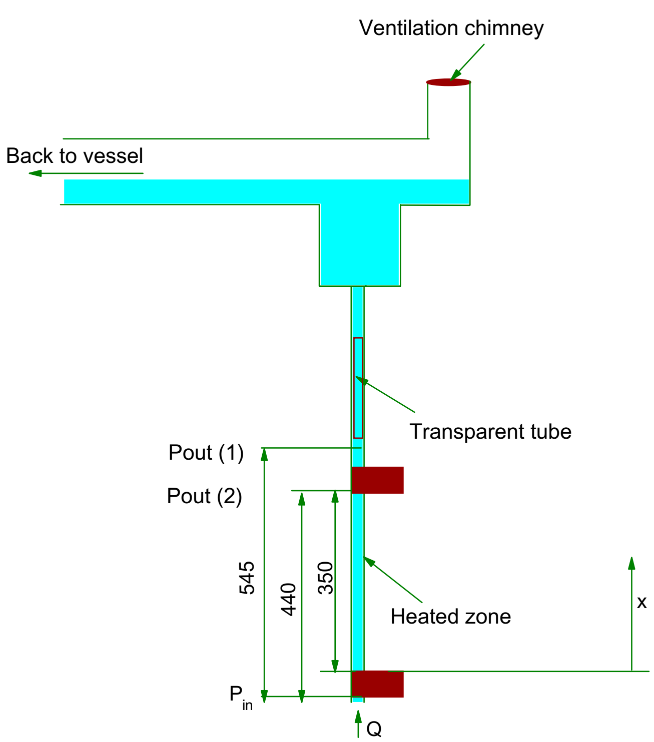

Fig. 4. The test section and the exit zone.

That transparent tube used to photograph the two phase flow during the test. Copper electrodes were silver soldered to each end of the heated zone. The test section and the bypass were connected to an inner plenum and the water entered to the heated zone through a developing region of about 150 mm. The test section is heated electrically by a DC power supply of 20 V and 4000 A. The inlet and outlet temperatures of the flowing water through the channel, were measured continuously by thermocouples. The pressure drop and the pressure distribution were measured by using the pressure taps, which were located at the inlet and at the outlet of the heated zone of the channel, as shown in Fig. 5. One pressure tap was located at the inlet of the channel (in) and two pressure taps (1 and 2) were located at the outlet. One of them (2) was immediately at the end of the heated length (upstream the copper electrode) and the second one (1) 105 mm downstream (downstream the copper electrode).

3. Test procedure

The experimental procedure consists of the following steps. The water in the tank were circulated through the pre-heater to get the beginning inlet temperature to the test section. When the inlet temperature was achieved, the flow rate through the test section was set to the required value. Once the flow rate was set, the heat flux was delivered to the test channel by increasing the current through the test section to the required power. The flow rate through the test section was constant during each experiment due to a high head pump and a high flow resistance at the valves. As shown in Fig. 3, the water which flow through the test section, returned back to the water tank. That caused the inlet temperature to the test section to increase monotonously during the test. The experiment was ended when the exit temperature exceeded the saturation temperature. In that way the experiments scanned the whole flow regimes in the test section, beginning in one phase flow of liquid through the OSV to the beginning of saturated boiling. The rate of the inlet temperature increase was about 2 °C/min, which ensured quasi steady state measurements. During the experiment the pressure drop between the inlet of the channel and the two exit points 1 and 2 was measured as well as the absolute pressure at those two points.

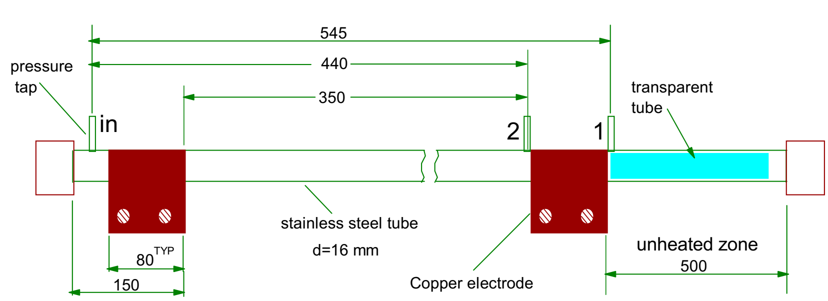

Fig. 5. The test section.

In one of the tests, part of the unheated zone was replaced by a Pyrex tube, in order to photograph the flow at the exit of the channel.

4. Results and discussion

Fig. 6 shows the measured pressure drop and the outlet pressure at the heated zone as function of the outlet temperature, of the heated test section (T_o). The open circles show the pressure drop between the inlet to point 1, and the solid circles show the pressure at point 1. Similarly the open squares show the pressure drop between the inlet to point 2, and the solid squares show the pressure at point 2.

The inlet pressures were calculated by adding the pressure drop and the pressure itself at point 1 and 2, which are shown as $P_{in}(1)$ and $P_{in}(2)$, respectively. As can be seen in Fig. 6, the pressure drop $\Delta P_{in\text{-}1}$ and $\Delta P_{in\text{-}2}$ have a different behavior. $\Delta P_{in\text{-}2}$ remains constant until outlet temperature of about 75 °C and then increases gradually until outlet temperature of about 82 °C. From that point on it continues to increase at a smaller slope. $\Delta P_{in\text{-}1}$ begins to increase at higher outlet temperature and increases in a very sharp slope. Then it continues to increase with a very small slope.

Looking at Fig. 6, one can see that the sudden increase of the pressure drop at point (2), occurs while the inlet pressure does not change. The increase of the pressure drop is accompanied by a sudden decrease of the local pressure, due to a local increasing of the average velocity. The void creation at the exit of the heated zone reduces the average density of the vapor–water mixture. For a constant mass flux this reduction causes an acceleration of the fluid at that location. If the vapor condenses downstream in the unheated region, the average density resumes its initial value. The local decrease of density causes local acceleration with pressure reduction due to Bernoulli Law. This phenomenon occurs at point (2) when the subcooling is quite high ($T_o = 75$ °C, $T_o - T_{sat} \approx 25$ °C) and therefore, the vapor condenses downstream of the unheated zone, with no effect at point (1). When the outlet temperature increases, the phenomenon moves downstream to point (1). The condensation time of the bubbles, depends on the outlet temperature. When the outlet tem-

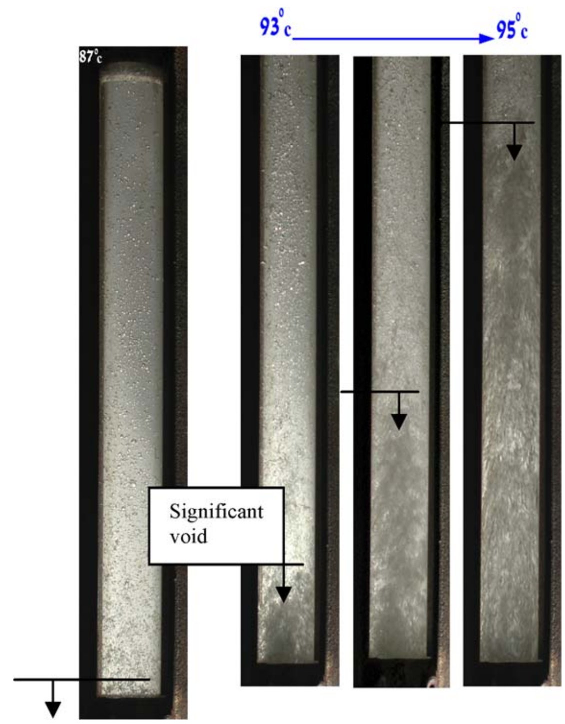

Fig. 7. Photographs of the outlet zone at various outlet temperatures.

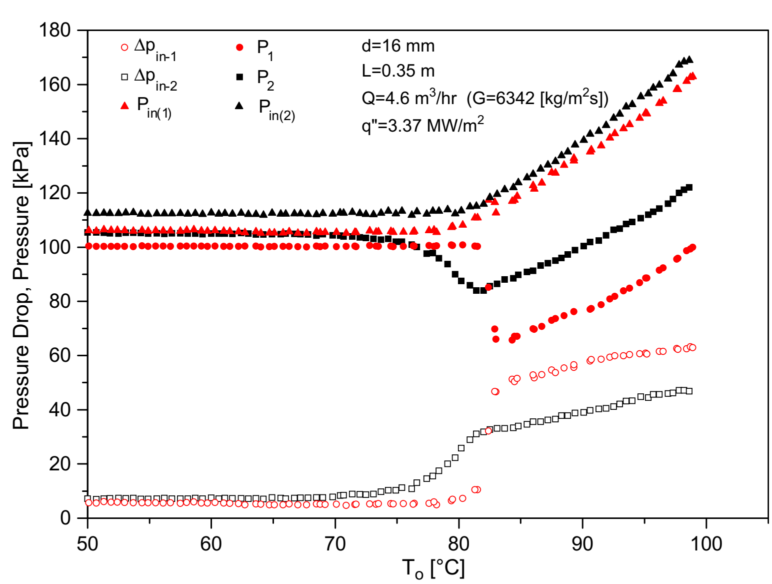

Fig. 6. Pressure and pressure drop measurements at points 1 and 2 vs. outlet temperature.

perature increases, the existence time of the bubbles and the void becomes longer; and the two phase zone extends towards the unheated part of the channel. That explains why the pressure begins to decrease at point 1 at higher outlet temperature than at point 2. The void wave reached point 1 just when the outlet temperature reached the value of about 82 °C. At that temperature the average velocity at point 1 increased sharply which caused a sharp reduction of the pressure. The inlet pressure didn't increase when the pressure drop on the heated zone began to increase; and, that is because the pressure drop increasing was due to a local phenomenon at the outlet measurement point.

Fig. 7 presents photographs of the void front advancing at the unheated zone. It can be seen that the significant void front advances when the outlet temperature increases. Figs. 8–11 present measured results of the outlet pressure and the pressure drop referring to point 2 at various flow rates and heat fluxes. In those figures one can see that the pressure drop and the pressure behavior at the exit of the heated zone are the same as in Fig. 6. The point at which the outlet pressure, at the exit of the heated zone, begins to decrease is the OSV. Fig. 12 presents a typical result of the outlet pressure versus the Stanton Number at the outlet of the heated zone. The ordinate of this figure presents the difference between the measured pressure at point (2) and the pressure measured when the flow was one phase ($P_{out} - P_{out,OP}$). It can be seen that at $St_{Nr} \approx 0.00369$ the outlet pressure begins to decrease.

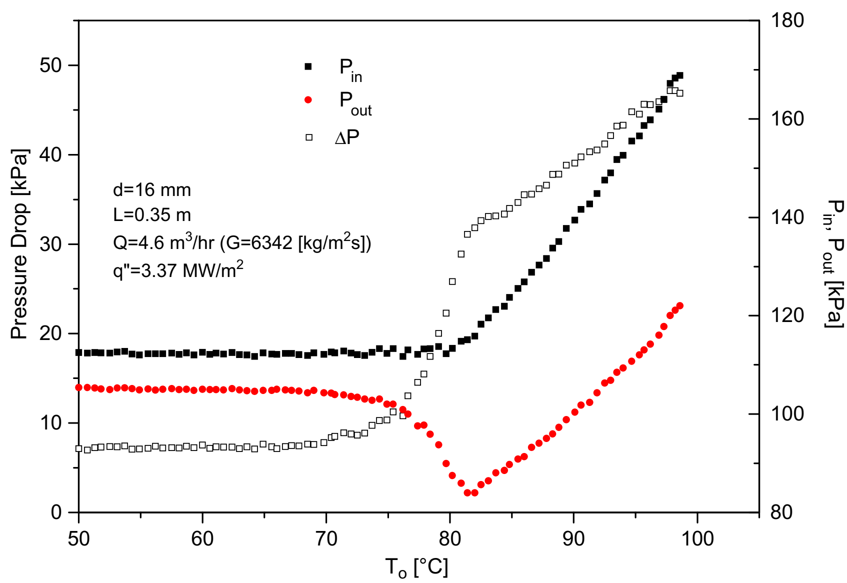

Fig. 8. Pressure drop, inlet and outlet pressure vs. outlet temperature $Q = 4.6\ m^3/h$, $q'' = 3.37\ MW/m^2$.

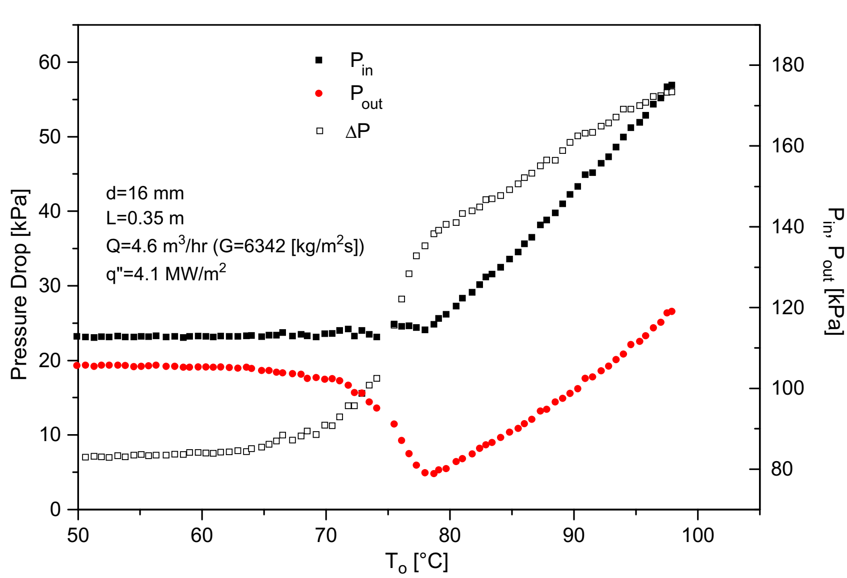

Fig. 9. Pressure drop, inlet and outlet pressure vs. outlet temperature $Q = 4.6\ m^3/h$, $q'' = 4.1\ MW/m^2$.

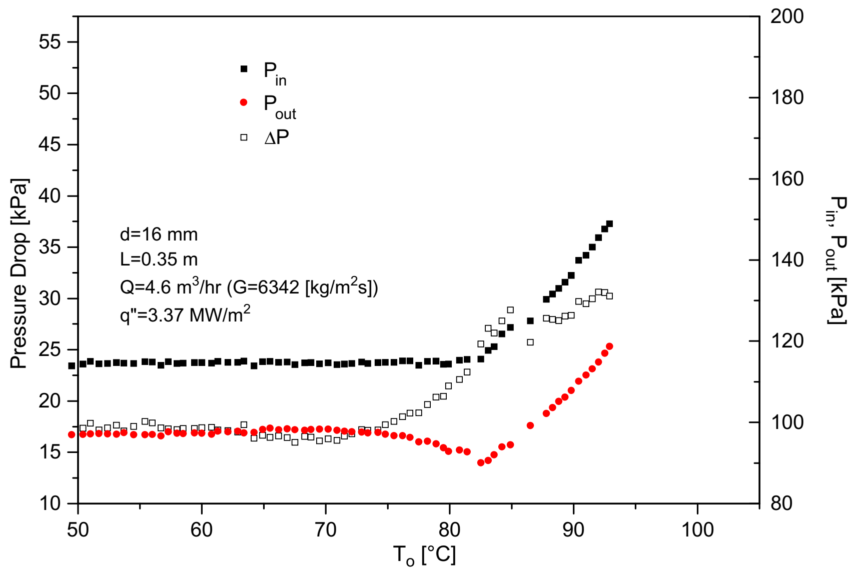

Fig. 10. Pressure drop, inlet and outlet pressure vs. outlet temperature $Q = 6.9\ \mathrm{m^3/h}$, $q'' = 3.37\ \mathrm{MW/m^2}$.

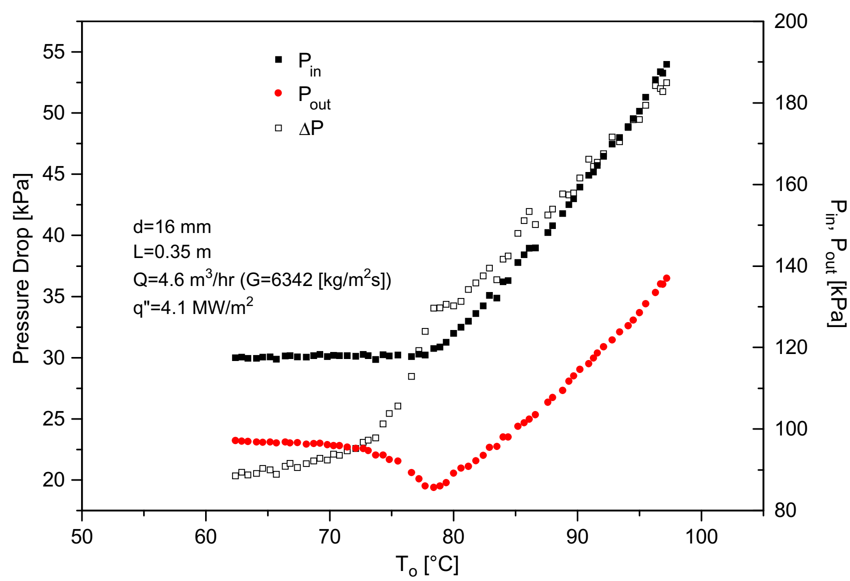

Fig. 11. Pressure drop, inlet and outlet pressure vs. outlet temperature $Q = 4.6\ \mathrm{m^3/h}$, $q'' = 4.1\ \mathrm{MW/m^2}$.

Fig. 13 summaries the outlet pressure measurements at the exit of the heated zone from the experiments, which are presented in Figs. 8–11. The results are presented versus the outlet Stanton number. In that figure the Stanton numbers at which the OSV occurs (St_{OSV}), at each experiment, are tabulated. According to Saha and Zuber [10] at Peclet number greater than 70,000 the Stanton number at OSV (St_{OSV}) is a constant number and is equal to 0.0065 where:

$$St_{\mathrm{Nr}} = \frac{q''}{GC_p(T_{\mathrm{sat}} - T_{\mathrm{o}})} \tag{2}$$

$$Pe = \frac{GD_{\mathrm{H}}C_p}{k_{\mathrm{f}}} \tag{3}$$

That statement was modified by Hoffman and Wang [6] and they extended the range of St_{OSV} between 0.0039 and 0.0065. A comparison between Saha and Zuber correlation and the experimental results of the outlet pressure correspond to the OSV is presented in Fig. 13. It can be seen that the Stanton number at OSV which was predicted by Saha and Zuber correlation is much higher than the experimental results. A better prediction of the experimental results achieved by using the Hoffman and Wang range of St_{OSV}. All of the experimental results which are tabulated in Fig. 13, agree with the lower limit of that range.

5. Concluding remarks

Experiments of pressure drop and pressure distribution measurements were performed of subcooled flow boiling at atmospheric pressure up to flow rate of $G = 9513\ \mathrm{kg/m^2\,s}$

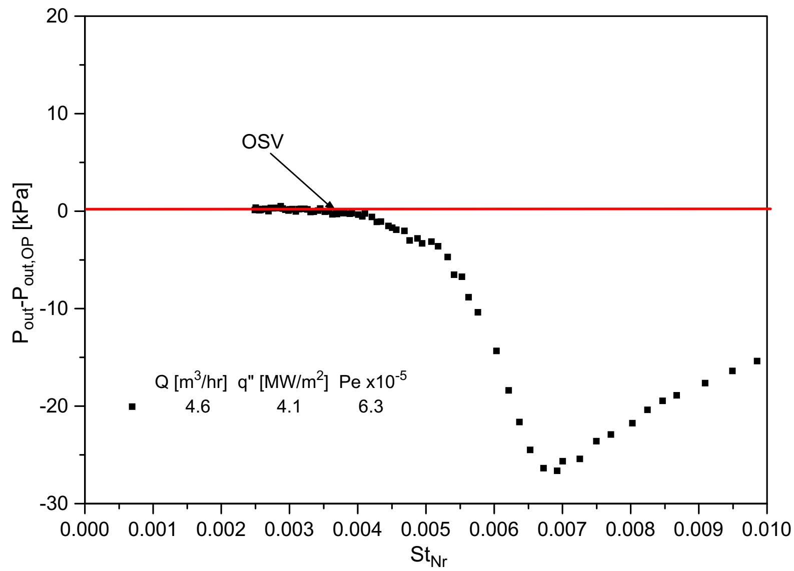

Fig. 12. Typical result of the outlet pressure versus the Stanton number at the outlet of the heated zone.

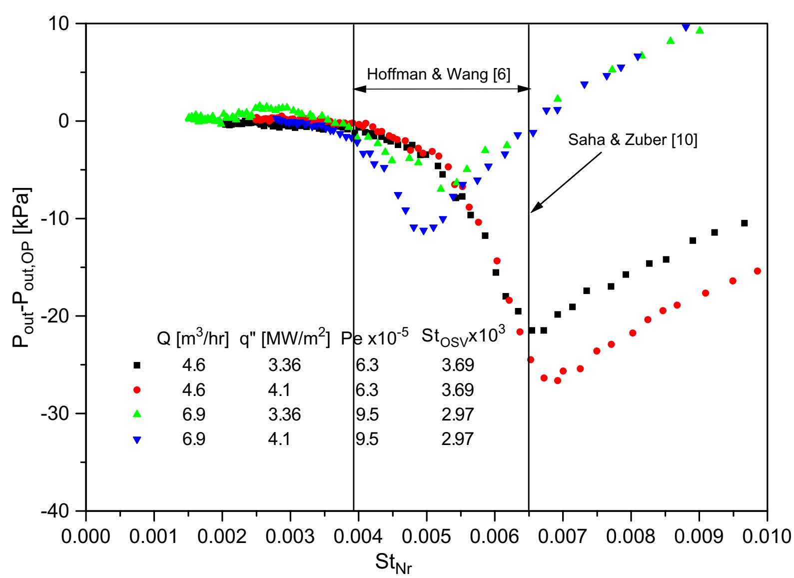

Fig. 13. Outlet pressure at various flow heat flux and flow rate versus the outlet Stanton number.

and heat flux of $q'' = 4.1$ MW/m^2. Using water in a vertical uniformly heated channel ($d = 16$ mm and $L = 350$ mm). The measurements were focused on the pressure behavior at the exit of the heated zone of the channel. Photographs of the void at that zone were taken in order to explain the pressure measurement behavior. It was found that the significant void front advances when the outlet temperature increases and that influences the overall pressure drop along the channel and the pressure at the exit of the heated zone.

The results obtained from the present study are summarized as follows:

(1) In subcooled flow boiling, the onset of a significant void in the heated channel, causes a reduction of the pressure at the outlet of the heated zone. That void existence depends on the outlet temperature. At the conditions of OSV, the vapor immediately condense downstream the heated zone.
(2) Photographs of the void front downstream the heated zone show that the significant void front advances when the outlet temperature increases and it influences the pressure distribution behavior along the test channel.
(3) From the measurement of the pressure at the outlet of the heated zone, the onset of the significant void can be observed.
(4) Comparison of the conditions at which the OSV occurs with correlations from the literature shows an over prediction of those correlations.

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