Unsteady-state fluctuations analysis during bubble growth in a “rectangular” microchannel

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\textbf{A B S T R A C T}

Boiling in microchannels shows great potential for cooling systems and compact heat removal applications. However for confidence in this cooling technique, it is essential that any excursions from typical flow boiling are understood and predicted. Confined bubble growth can cause pressure fluctuations which interfere with bubble nucleation and growth and can also lead to flow reversal and instances of temperature excursions. Boiling experiments are performed in a single rectangular microchannel of hydraulic diameter 771 \,\mu\text{m}, using n-Pentane as the working fluid. A heating technique was incorporated on the exterior walls of the microchannel; a transparent, metallic, conductive deposit, which allows simultaneous uniform heating and visualisation to be achieved. In conjunction with obtaining high-speed imaging, an infrared camera is used to record the temperature profile at the microchannel wall, and sensitive pressure sensors are used to record the pressure drop across the microchannel over time. During flow boiling in the microchannel periodic and non-periodic fluctuations in both the channel pressure drop and channel temperature profile over time are apparent. In this paper we provide a full analysis of the temperature measurements and pressure data obtained during the growth of a vapour bubble in the microchannel. An augmentation of the heat transfer coefficient of over 216\% has been achieved during periodic two-phase flow boiling in the microchannel. However overpressure (over 410\% increase) in the microchannel occurs at corresponding instances to the heat transfer enhancement. The two time steps during the periodic bubble dynamics, namely the bubble expansion time period and the waiting time period in-between the bubble expansion fluctuations, are also investigated and modelled. It was determined that both the bubble dynamics and the channel wall heating time period are responsible for the pressure and temperature fluctuation time periods observed.

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1. Introduction

Flow boiling in microchannels is an efficient mode of heat transfer for a variety of applications, particularly in compact evaporators for the automobile industry and for use in cooling microelectronics components. It is also important to point out here that there are significant differences between macroscale flows and microscale ones. Whilst much research has been performed into determining microchannel heat transfer and pressure correlations, some correlations have been shown to differ drastically when applied to other fluids than those initially developed for, as noted by Su et al. [1].

There are observations of instabilities in both the temperature and pressure during flow boiling. The potential damage these flow instabilities could cause if the heat is not properly dissipated due to instances of vapour blockage and channel dry-out, with a decrease in the critical heat flux [2], is of paramount importance. Two-phase and boiling flow instabilities are complex, due to phase change and the presence of several interfaces. There has since been a lot of research devoted to flow boiling instabilities in microchannels [3–6]. Hetsroni et al. [7] reported oscillation amplitudes in pressure drop at a varying mass flux and heat flux during flow boiling in triangular silicon microchannels of hydraulic diameter 130 \,\mu\text{m}. The measured pressure fluctuations were attributed to the periodic growth and collapse of vapour fractions.

In recent years, a number of studies provided evidence to the sensitivity of two-phase microchannel systems to flow instabilities. Oscillating pressure drops and wall temperatures, and visualizations showing cyclical backflow, were encountered in many experiments. Bergles and Kandlikar [2] classified these as compressible volume instabilities, relating them to the presence of compressibility upstream of the heated channels. Relative to a stable mode, an unstable system presents entirely different flow features, and may bring about substantial differences in the heat transfer efficiency.
Another work that evidenced pressure drop fluctuations, by Chang and Pan [8], also observed bubble slug oscillating growth leading to instances of reversed flow in microchannels. They found that the magnitude of the pressure drop oscillations can be used as an index for the appearance of reversed flow.

Surface temperature rises are expected to occur at equivalent times to a microchannel filling with vapour. Since the heat flux at the channel walls is usually removed by the fluid boiling, the surface temperature of the vapour-filled channel will rise dramatically. Fluctuations in the pressure drop across the channel accompany the temperature fluctuations occurring during the bubble dynamic process. Temperature fluctuations during a bubble expansion process, i.e. where a nucleated bubble grows, is confined, elongates and is ‘swept away’ by incoming liquid flow into the channel, are recorded simultaneously in an experimental work by Wang and Cheng [9]. Other researchers [10] have also cited bubble dynamics as fundamental in the explanation of the sharp pressure peaks observed during flow boiling in microchannels. The trigger mechanism of flow boiling instabilities in a single microchannel is the venting of a vapour slug due to its sudden rapid expansion. Barber et al. [10] have related the sharp periodic pressure fluctuations witnessed during confined bubble growth in a microchannel to the bubble dynamics, and instances of vapour blockage in the microchannel. This experimental work was achievable through the simultaneous acquisition with a high-speed camera, an infrared camera and data measurements. Li et al. [11] have also investigated bubble dynamics in microchannels. They related the bubble growth in the axial direction both forwards and backwards, whilst its length increases exponentially, to the evaporation of the thin liquid film between the bubble and heating wall. They produced a theoretical model of the exponential bubble growth as a relationship between the evaporation effect and the pressure effect around the bubble.

Temperature measurements at the wall of a microchannel are not trivial to perform, mainly due to the small scales involved. Even though there are now commercially available thin and fast thermocouples, it is crucial that there is no disruption to the internal wall conditions of the microchannels and no interference with the bubble dynamics inside. Infrared thermography has been previously used in the literature [12] in determining heat transfer measurements. Hetroni et al. [13] conducted an experimental study where they coupled an infrared (IR) camera with high-speed imaging for temperature and flow pattern analysis inside mini and microchannels. The IR technique has also been successfully applied to study boiling [14], mainly due to its fast time response. The IR technique is a non-contact, non-destructive test method. Its unobtrusive nature makes it very appealing for temperature determination at the microscale. Xu et al. [15] show line plot variations in a chip wall temperature and in the heat transfer coefficient across the heated length of a chip. Their test section was a silicon wafer consisting of ten triangular microchannels with a hydraulic diameter of 155 μm, with acetone as the working fluid. A thorough survey on infrared thermography for convective heat transfer measurement has been presented by Astarita et al. in 2000 [16]. Other thermal detection techniques, such as Thermochromic Liquid Crystals (TLCs) have been utilised to identify temperature fluctuations during boiling. Liquid crystal thermography has a unique capability to record two dimensional temperature fields at frequencies up to 500 Hz. Kenning [17] used Thermochromic Liquid Crystals to observe temperature distributions on the back of a thin stainless steel heated plate during nucleate pool boiling of water. Hohmann and Stephan [18] used thermochromic crystals with much the same technique as Kenning, but they measured the temperature distribution underneath an evaporating meniscus so as to investigate the heat transfer. However the utilization of TLCs limits the simultaneous flow visualization of the boiling phenomena with a high-speed camera, due to the layer of black paint required with TLCs. A positive feature of IR cameras with photon detectors is their quick time response, in the order of micro-seconds, since radiation travels at the speed of light, compared to the time response of TLCs whose response has been evaluated to be in the range of a few to a hundred milliseconds. However, TLCs in their unsealed state have a better spatial resolution than IR cameras.

In this paper IR thermography is used to determine the microchannel wall temperature profile over time during flow boiling.

**Nomenclature**

- \( A \) channel surface area (m²)
- \( B \) bubble area (m²)
- \( B_i \) Biot number (–)
- \( C_o \) Confinement number (–)
- \( C_p \) specific heat capacity (J kg⁻¹ K⁻¹)
- \( d_h \) channel hydraulic diameter (m)
- \( d \) channel diameter (m)
- \( E \) power (J s⁻¹)
- \( G \) mass flux (kg m⁻² s⁻¹)
- \( h \) heat transfer coefficient (W m⁻² K⁻¹)
- \( k \) thermal conductivity (W m⁻¹ K⁻¹)
- \( l \) channel heated length (m)
- \( L \) characteristic length (m)
- \( L \_v \) latent heat of vapourisation (J kg⁻¹)
- \( m \) liquid mass flowrate (kg s⁻¹)
- \( P \) pressure (bar)
- \( Q_e \) heat flux density at the channel wall (W m⁻²)
- \( Q_r \) heat flux density at the triple contact line (W m⁻²)
- \( r \) thickness of glass channel (m)
- \( s \) Laplace variable (s⁻¹)
- \( t \) time period (s)
- \( T \) temperature (°C)
- \( U \) velocity (m s⁻¹)
- \( V \) bubble volume (m³)
- \( w \) channel width (m)
- \( W_p \) wetted perimeter (m)

**Greek**

- \( \alpha \) thermal diffusivity (m² s⁻¹)
- \( \Delta P \) pressure drop (ΔP = Pᵣᵢ – Pᵢₒ) (bar)
- \( \varepsilon \) channel emissivity (–)
- \( \mu \) viscosity (kg m⁻¹ s⁻¹)
- \( \rho \) density (kg m⁻³)
- \( \sigma \) surface tension (N m⁻¹)
- \( \omega \) Stefan–Boltzmann constant \((5.67 \times 10^{-8}) \) W m⁻² K⁻⁴

**Subscripts**

- \( \text{avg} \) average value
- \( \text{c} \) convective
- \( \text{elec} \) electrical
- \( \text{i} \) inner dimension
- \( \text{in} \) inlet conditions
- \( \text{o} \) outer dimension
- \( \text{out} \) outlet conditions
- \( \text{sat} \) saturation
- \( \text{wall} \) wall condition
- \( L \) liquid
- \( V \) vapour
The correlation of pressure and temperature data during periodic fluctuations is presented. The vapour bubble expansion and microchannel wall heating times steps are calculated and subsequently modelled.

2. Experimental apparatus

2.1. Working fluid

The working fluid chosen for the experimental campaigns is n-Pentane. This fluid is chosen for its low saturation temperature ($T_{sat} = 35.5\, ^\circ\text{C}$ at atmospheric conditions) and hence it has a low power requirement to induce boiling.

The physical properties of the fluid as seen in Table 1 allow the Confinement ($Co$) number to be determined. The Confinement number as proposed by Kew and Cornwell [19] can be utilised when the bubble diameter approaches the channel diameter, and is thus ‘confined’. The transition criterion for the use of the confinement number as proposed by Kew and Cornwell $Co = 2.1$, hence implying that confinement effects will be present in the channel, and differences from macrochannel phenomena will be seen. The Confinement number for n-Pentane in a $d_i = 771 \, \mu m$ channel in the experimental set-up as described in this paper is $Co = 2.1$, hence implying that confinement effects are present. To calculate the hydraulic diameter, $d_h$, of the microchannel Eq. (1) was used, where $A = w_d + \frac{a_i}{2}$, and $W_p = 2w_i + \pi d_i$, it should be noted here that the curved edges of the microchannel are taken into account in the calculation of both the channel wetted perimeter and the channel surface area.

$$d_h = \frac{4A}{W_p}$$  \hspace{1cm} (1)

The cross-section and sizing of the rectangular microchannels used can be seen schematically in Fig. 1. For a hydraulic diameter microchannel $771 \, \mu m$, the dimensions are: $d_i = 400 \, \mu m$, $w_i = 8000 \, \mu m$ and $r = 400 \, \mu m$, with a cross-sectional aspect ratio of 20, with an approximate heated channel length, $l$, of 70 mm. (It should be noted here that all dimensions are provided to an accuracy of ±10% by the manufacturer.)

2.2. Experimental set-up

The novel aspect of this research is the simultaneous data acquisition and visualisation. This has been achieved due to a uniform, transparent, metallic deposit of tantalum (Ta) on the exterior walls of the rectangular microchannels investigated. This tantalum deposit is both transparent and conductive, with an electrical resistance of $1.3 \times 10^{-6} \, \Omega \, m$ and a deposit resistance of $7 \, k\Omega$ at the thickness sputtered (0.2 nm), hence enabling simultaneous uniform heating and visualisation of the microchannel flow. The deposit is also transparent to the IR camera, and so we are able to record the temperature profile at the exterior microchannel wall with an IR camera.

The tantalum thin film, micro-heater was sputtered on a borosilicate glass channel. The borosilicate glass has a low thermal conductivity, so axial conduction in the micro-heater film is negligible. So the effective heat flux absorbed by the working fluid in the microchannels can be determined, along with the conduction heat loss through the channel walls and radiative losses inside the closed experimental loop. The applied heat flux to the microchannel is calculated using the external heated surface area of the microchannel, including the curved surface area at the microchannel edges. The tantalum was deposited at a uniform thickness over the entire external surface area of the microchannel. The heat losses, diffusion time and Biot number are all calculated and explained in further detail in Sections 2.5 and 3.1 of this paper.

The experimental apparatus has been designed to induce boiling, to measure parameters across the microchannel such as the pressure drop and the temperature profile, and to visualise and record the phenomena occurring inside the vertically aligned microchannel test section. An annotated photograph of the microchannel and its electrical heating is provided in Fig. 2.

The final flow loop consists of a syringe pump providing a constant liquid mass flowrate into the system, an interchangeable microchannel vertical test section, a condensation system, a flow visualisation system using a high-speed camera, an IR camera to record the temperature profile across the microchannel, and a data acquisition system. This is all housed inside a temperature regulated box of volume 1 m$^3$, at a regulated temperature of $34\, ^\circ\text{C}$ (at 1 bar).
atmospheric pressure) which is just below the saturation temperature of n-Pentane (35.5 °C). The pressure sensors used (Honeywell 24PC differential series) are accurate to ±0.25% span. The IR technique is a non-contact, non-destructive test method. Its unobtrusive nature makes it very appealing for temperature determination at the microscale. It was decided that mapping of the temperature using an IR camera would prove to be a simple, effective and powerful tool. A FLIR Systems ThermaCam A40 infrared camera is used to visualise and record the temperature profile at the microchannel exterior wall. The IR camera used has a spatial resolution in the order of 100 μm, with a thermal sensitivity of 80 mK at 30 °C, an accuracy of ±1 °C for temperatures in the range 40 °C–120 °C, a resolution of 320 × 240 pixels, and it is Stirring cooled to 70 K. The camera operates at 50 Hz, therefore 50 frames per second. The two-phase flow boiling was also simultaneously visualised with a high-speed camera, with frame rates up to 2000 fps. Further descriptions and diagrams of the experimental apparatus can be found in Barber et al. [3].

2.3. Experimental procedure

The working fluid is degassed before entry into the flow loop to remove any non-condensables. The degassing is achieved by boiling the n-Pentane liquid in the reservoir several times using an imbedded 320 W cartridge heater for 1 h, which is over five times the required time to boil the volume of n-Pentane liquid present in the reservoir. The inlet liquid mass flowrate is held constant by the syringe pump during an experimental run. It is also possible to apply a range of heat flux to the test section via electrical resistance and the power regulator that is connected by silver epoxy and wires to the microchannel. Steady state is reached after about 20 min in each test run. Bubble nucleation, expansion and coalescence to slug flow, and instances of channel dry-out are all observed with the use of a high-speed camera and macro lens. Pressure readings at the inlet and outlet of the microchannel in conjunction with temperature profiles, recorded at the microchannel wall with an IR camera, provide simultaneous data.

2.4. Pressure drop data

The pressure sensors are located upstream and downstream of the microchannel, and so the measured pressure drop is the summation of the pressure drops across the inlet and outlet manifolds, the microchannel, and the pressure drop resulting from the inlet and outlet contraction and expansion. The acquisition frequency for all the pressure data is 133 Hz.

2.5. Heat transfer data

The power provided to the fluid (E_p) was calculated based on the power applied to the microchannel deposit (E_{elec}) and corrected for losses to the environment (E_{base}), i.e. E_p = E_{elec} - E_{base}. These losses include both convective and radiative heat transfer, i.e. E_{losses} = E_p + E_{rad}. The convective losses (E_c) were calculated using Eq. (2), where h_c is the convective heat transfer coefficient. The value obtained for E_c is 0.08 W, for a T_{in} value of 34 °C and a T_{wall} value of 48.9 °C.

\[ E_c = h_c A (T_{wall} - T_{in}) \]  

(2)

The radiative losses (E_{rad}) were calculated using Eq. (3), where ε is the emissivity of the microchannel deposit (ε = 0.76), and α is the Stefan–Boltzmann constant (5.67 × 10^(-8) W m^(-2) K^(-4)). The value obtained for E_{rad} is 0.10 W, again with the same T_{in} and T_{wall} values as used in Eq. (2).

\[ E_{rad} = \varepsilon \alpha A (T_{wall}^4 - T_{in}^4) \]  

(3)

The power actually provided to the microchannel (E_p) could then be calculated based on: E_p = E_{elec} - E_c - E_{rad}.

The heat flux (Q') actually provided to the fluid via the deposit at the microchannel wall, the calculation is based on E_p where E_p = 2.82 W, and hence Q' = 2.33 kW/m². Percentage losses from the microchannel could then be calculated based on: E_p/E_{elec}. These losses were of the magnitude of 6% or less, for a worst case scenario. Hence implying that the majority of the heating provided at the microchannel exterior wall (over 94%) was transmitted directly to the flowing liquid inside the microchannel, with only small losses to the surroundings.

3. Results

Two types of fluctuations are observed during two-phase flow boiling; both periodic and non-periodic fluctuations, these are found to depend on the heat flux and the inlet liquid mass flux. Increasing both the heat and mass flux, promotes a move from periodic pressure and temperature fluctuations to chaotic fluctuations, as seen in Fig. 3a and b. At large heat flux and/or mass flux, the periodicity of the instabilities are hard to determine, and therefore a quantitative comparison between periodic and non-periodic instabilities is difficult, hence only a periodic pressure and temperature case is presented in this paper, similar to that observed in Fig. 3a but at different flow conditions. The simultaneous measurement techniques of pressure, temperature and flow visualisation used in the experiments allowed the correlation between the periodic pressure and temperature fluctuations to be achieved.

3.1. IR measurements

The temperature data presented in this paper was achieved with an infrared camera. It is important to highlight here that
the microchannel of 3.52 kg m⁻²·s⁻¹. Based on a mass flux through the microchannel is 12.15 s, which is smaller than the diffusion time calculated at the microchannel wall. This implies that the sensitivity and reaction time of the temperature at the inner microchannel wall, i.e. the temperature in direct contact with the fluid, is comparable to the instabilities characteristic time during fast bubble expansion. This will be discussed in further detail in Section 3.3.

Following the calculation of the diffusion time, a calculation of the Biot number (Bi), which relates the heat transfer resistance inside and at the surface of a heated object, will allow us to better understand the heat transfer occurring at the microchannel wall. Comparing the conduction resistance to the convection resistance, a Biot number of 0.02 was calculated using Eq. (5), where L is a characteristic length, namely the microchannel wall thickness (r) in this case

\[ Bi = \frac{h_r \times L}{k} \]  

(5)

Since Bi < 0.1, it can be said that the heat conduction inside the object is much faster than the heat conduction away from its surface, and temperature gradients are negligible inside it. From this, we can state that the exterior wall temperature field is approximately the temperature field of the interior wall of the microchannel. To verify this we can calculate the temperature change in the borosilicate glass from one side to the other side, using Eq. (6), and we obtain \( \Delta T = 0.82 ^\circC \)

\[ \Delta T = \frac{d_r \times Q''}{k} \]  

(6)

The local convective heat transfer coefficient, h, can be determined from the IR thermography obtained at the microchannel wall, using Eq. (7), where one-dimensional heat conduction is considered. In Eq. (7), \( Q'' \) is the corrected heat flux provided at the microchannel wall, and \( (T_{wall} - T_{sat}) \) is the difference between the channel wall temperature and the saturation temperature of the bulk liquid. To have a good determination of the one-dimensional heat transfer coefficient, an average of \( T_{sat} \) is calculated based on the pressure change from the inlet (z = 0) to the outlet (z = L) of the microchannel. \( T_{sat} \) is a function of the average pressure in the channel, i.e.

\[ T_{sat} = \left( \frac{T_{in} + T_{out}}{2} \right) \]

\[ h(z) = \frac{Q''}{(T_{wall}(z) - T_{sat})} \]

\[ h = \frac{1}{T} \int_{0}^{L} h(z) dz \]  

(7)

3.1.1. IR raw data measurements

Infrared raw data and temperature profiles are presented in Figs. 4 and 5 for a 771 \( \mu \)m hydraulic diameter microchannel using n-Pentane as the working fluid. The IR camera measures emitted radiation and the fact that this radiation is a function of the object temperature makes it possible to access the local surface temperature at the microchannel wall. However, the radiation measured by the IR camera also depends on the emissivity of the deposit at the microchannel wall, and on any reflected radiation from the surroundings. To measure the wall temperature accurately, it was necessary to compensate for the effects of a number of different radiation sources. Several object parameters are required to be input into the camera, such as the emissivity of the object, the reflected temperature, the distance between the object and the camera and the relative humidity. A further experiment was performed to determine the emissivity of the microchannel deposit, by comparing the channel deposit to a known emissivity of a black painted heated object. In this way it was possible to determine the
emissivity of the channel deposit over a range of heater temperatures. The emissivity of the coated microchannel was found to be: $\varepsilon = 0.76$. The tantalum deposit was assumed to be uniform, and so the heat flux applied at the microchannel wall, once corrected for losses, was also assumed to be constant and uniform across and along the microchannel. This was verified by acquiring IR sequences during single-phase liquid flow in the microchannel with an applied heat flux at the wall. From these sequences wall temperature profile data was extracted and it was evident that the temperature profile across and along the channel was constant even though there was an applied heat flux at the microchannel wall, since the heat flux provided was smaller than that required to induce boiling of the working fluid.

The IR camera image shown to the left of Fig. 4 is an example of the raw temperature data directly obtained from the infrared camera experimental set-up. The plot on the right of Fig. 4 illustrates these two temperatures (channel centre-line and channel surface area) over time. The magnitudes of six of the peaks lie between approximately 52–59.5 °C, and there is one peak at 64.8 °C, with fluctuation time periods between 10 and 15 s. It can be seen in Fig. 4, that the average channel centre-line temperature is slightly greater than the average channel surface area temperature. The microchannel centre-line wall temperature profile is at certain instances 3 °C greater than the microchannel surface area wall temperature profile in Fig. 4. This can be explained by considering the rectangular geometry of the microchannel and appreciating that although a uniform heat flux is applied over the entire channel surface area, the centre of the channel will be hotter than the channel edges since heat will dissipate outwards towards the edges, see Fig. 5a. The plots in Fig. 5a and b illustrate the temperature profile across the microchannel width and along the microchannel length at a particular time. From Fig. 5 it is possible to see a change in wall temperature along the channel length illustrating the presence of two-phase flow boiling, which is evident when comparing this result to the constant, flat wall temperature.

![Fig. 4. Left, raw IR image and right. A plot of the temporal variation of the average temperature (Tavg) taken both across the entire channel heated surface and along the channel centre-line. Experiment conditions: $d_h = 771 \mu m$, $G = 3.52 \text{ kg m}^{-2} \text{s}^{-1}$, $U_m = 5.76 \text{ mm s}^{-1}$ and $Q = 2.33 \text{ kW/m}^2$.](image)

![Fig. 5. (a) Left, temperature distribution of $T_{\text{wall}}$ across the microchannel width and (b) right, temperature distribution of $T_{\text{wall}}$ along the channel length during two-phase flow boiling of pentane. Experiment conditions: $d_h = 771 \mu m$, $G = 3.52 \text{ kg m}^{-2} \text{s}^{-1}$ and $Q = 2.33 \text{ kW/m}^2$.](image)
profile obtained from the IR camera during single-phase liquid flow.

Particularly notable in the temperature profile plot of Fig. 5, along the channel length, is a peak in the temperature at 54.7 °C occurring approximately 53 mm along the channel length. This peak or temperature hotspot can be explained by the boiling phenomena occurring inside the channel as recorded by the high-speed imaging: a vapour bubble grows and blocks the inlet flow, as the bubble diameter approaches the channel diameter towards the outlet. A pictorial explanation of this blockage phenomenon can be seen annotated in Fig. 5.

3.2. Bubble dynamics

High-speed imaging was obtained simultaneously to the IR temperature profile and pressure drop data across the microchannel. Several bubble growths to slugs are observed in the image sequences performed at varying frame rates from 100 to 2000 fps. The high-speed sequence presented in Fig. 6 is at the same flow conditions as that seen in Figs. 4 and 5, i.e. a mass flux $G = 3.52 \text{ kg m}^{-2} \text{s}^{-1}$ and a heat flux $Q = 2.33 \text{ kW/m}^2$. A small bubble is nucleated in the centre of the microchannel at the inlet; it quickly becomes confined by the channel walls and undergoes elongated slug growth. It should be noted here that there is already a vapour slug proceeding through the microchannel ahead of the newly nucleated vapour bubble. During the elongated slug growth phase, the liquid–vapour interfaces of the bubble expand towards both the channel inlet and outlet creating an overpressure in the microchannel, evident in the resulting pressure signal fluctuations at the inlet and outlet. The n-Pentane bubble growth is very fast, and the bubble area over time in Fig. 6 can be calculated, see Fig. 7.

The plot in Fig. 7 shows the log-linear relationship of the bubble area over time, data extracted from Fig. 6. Experiment conditions: $d_i = 771 \mu\text{m}$, $G = 3.52 \text{ kg m}^{-2} \text{s}^{-1}$, $U_{in} = 5.76 \text{ mm s}^{-1}$ and $Q = 2.33 \text{ kW/m}^2$.

3.3. Correlation of pressure and temperature fluctuations

Data was obtained over a range of heat flux and mass flux. The correlation between the average temperature profile at the microchannel wall and the pressure at the microchannel inlet and outlet was achieved for all periodic boiling cases investigated. As
previously stated in Section 3.1, the heat transfer coefficient can be calculated from the temperature data obtained from the IR thermography at the microchannel wall. The correlation presented in Figs. 8–10 is again for the same flow conditions as that in the previous Figs. 4–6, a mass flux $G = 3.52 \text{ kg m}^{-2} \text{s}^{-1}$ and a heat flux $Q_0 = 2.33 \text{ kW/m}^2$. The temperature data is extracted from the IR sequence over the bubble growth area at the base of the microchannel as highlighted in Fig. 6. The first correlation of the data is presented over a time period of 100 s in Fig. 8, moving to data correlated over the first 17 s in Fig. 9, and finally data correlated over 10 s during just one temperature peak in Fig. 10.

Figs. 8–10 illustrate the correlation achieved between the temperature at the exterior microchannel wall, the local heat transfer coefficient and the pressure at both the microchannel inlet ($P_{\text{in}}$) and outlet ($P_{\text{out}}$) over time. As stated previously the pressure fluctuations observed in the microchannel can be related back to the bubble dynamics, for further explanation see our previous paper Barber et al. [10]. The vapour bubble growth leading to channel confinement creates sharp pressure fluctuations in the microchannel at both the channel inlet and outlet. These pressure fluctuations are periodic as can be seen in Figs. 8 and 9. The periodic microchannel wall temperature can be correlated to the pressure fluctuations. The peak in the pressure fluctuation represents an instance of vapour blockage in the channel, understandably when the channel is predominantly filled with vapour due to channel blockage, the temperature of the microchannel wall will also rise. Since with predominantly vapour flow in the microchannel, the
heat removal rate from the wall is lower than instances when there is predominantly liquid flow in the microchannel.

In Fig. 9, the two time-steps have been annotated on the plot for both the bubble growth rate and the channel wall heating time period. The top plot of Fig. 9 illustrates the local heat transfer coefficient over 17 s. One peak in heat transfer can be seen between t = 4.9 s and t = 14.9 s, with an average local heat transfer coefficient value during this peak of approximately 490 W m⁻² K⁻¹. This average value of the non-stationary local heat transfer coefficient shows an enhancement of approximately 288%, as observed at the baseline of the plot. The baseline value of approximately 170 W m⁻² K⁻¹ for the stationary local heat transfer coefficient shows good agreement with a typical value calculated for a forced convection heat transfer coefficient of n-Pentane in a rectangular cross-section channel, based on a laminar flow Nusselt number correlation for microchannels (see the article by Wu and Cheng [20] for a similar Nusselt number calculation). A similar calculation of the heat transfer enhancement achievable over 100 s, as shown in the top plot of Fig. 8, was performed. In Fig. 8 seven complete fluctuation periods of the heat transfer coefficient can be seen. An integral calculation of the average heat transfer coefficient during this 100 s period provides a value of 432 W m⁻² K⁻¹, this value can be considered as a more global representation of the heat transfer enhancement achievable, than that which was calculated previously for a single fluctuation in Fig. 9. However it is important to note here, that whilst an enhancement of approximately 216% for the heat transfer coefficient is achievable globally, there are also drawbacks to such a heat transfer enhancement, mainly in terms of the accompanying pressure fluctuations and over pressure phenomena. Pressure fluctuations as great as ±33 mbar, which is over 410% greater than the typical pressure baseline values of 7 mbar during liquid only flow, are reached during the instances of heat transfer enhancement.

The close-up plot in Fig. 10 over 10 s shows that during the increase in the microchannel wall temperature over 7.95 s, the heat transfer coefficient decreases from approximately 1319 W m⁻² K⁻¹ to 212 W m⁻² K⁻¹, whilst the pressure data remain at constant baseline values of approximately 8 mbar and 0 mbar for P_in and P_out respectively. In Fig. 11, the same periodic pressure fluctuation can be seen over just 1400 ms; the wall temperature peaks at 46.9 °C, and then decreases suddenly to approximately 39.0 °C over just 700 ms, before continuing to a minimum at 37.6 °C, 900 ms later. This temperature increase at the wall followed by a sudden decrease can be understood to be a sort of activation energy accumulated at the wall, where energy is stored and is then transferred to the liquid, which at a certain point induces a fast vapour expansion of the liquid–vapour interfaces of the growing bubble. During this time period the heat transfer coefficient decreases; if this heat transfer technique was integrated in a compact cooling system, any instances of poor heat transfer are important to determine and predict.

Two different time steps are evidenced in the fluctuations shown in Figs. 8–10. Firstly the fast vapour expansion (t_bubble) that results in the sharp pressure peaks, and secondly the time period (t_period) in between the pressure fluctuations which relates directly to the wall temperature increasing and the heat transfer coefficient decreasing. These time steps need to be analysed separately to understand their occurrence due to the flow conditions and the bubble dynamics occurring inside the microchannel geometry. The first time step, which is of the order of 80 ms, relates to the bubble dynamics (see Fig. 9). After a bubble is nucleated, it quickly grows until it becomes confined by the microchannel walls, initially by the microchannel d, dimension, which is 400 µm for the 771 µm hydraulic diameter microchannel. At a certain moment the confined bubble expands rapidly towards both the microchannel inlet and outlet during elongated bubble growth in the channel, see Fig. 6. This confinement followed by the sudden vapour expansion results in the pressure fluctuations observed at the microchannel inlet and outlet. The moment at which the confined bubble expands suddenly can be directly related to the temperature maximum at the microchannel wall, before the sudden decrease in temperature as the vapour slug begins to be purged from the microchannel, with fresh liquid readmitted into the microchannel at the inlet, i.e. the temperature decreases when there is predominantly liquid only flow in the microchannel. This fast time period for the bubble expansion can be seen highlighted in Fig. 11, where the pressure fluctuations over 1400 ms are presented, as well as the corresponding wall temperature and heat transfer coefficient data.

The second time step relates to the steady increase in the temperature at the microchannel wall, and also to the constant...
to the microchannel is applied at the microchannel wall, and since all faces of the channel are sputtered with the conductive, metallic tantalum deposit, the heat flux ($Q^*$) applied is uniform across and along the channel. Initially the bubble growth in the channel is radial, whilst the bubble is unconfined. Once the bubble approaches the channel inner diameter ($d_i$), partial confinement occurs, until the bubble also grows to the width of the channel, ($w_i$), see Fig. 12. At this point the bubble is fully confined and the bubble cross-sectional area $B$ can be assumed to be equivalent to the channel cross-section, i.e. $d_i \times w_i$. The time step models presented here involve two different time periods, namely $t_{\text{bubble}}$ and $t_{\text{period}}$ as seen annotated in Fig. 9, and as described previously in Section 3.3.

The schematics in Fig. 12 parts a and b, illustrate the bubble growth in the microchannel; the bubble shape is initially radial (part a) until it becomes fully confined by the channel cross-section (part b). Heat input to the bubble ($Q^*$) is focussed at the contact line of the bubble, and the value for $Q^*$ at the contact line can be approximated roughly to twice that of $Q^*$ as seen at the channel wall [21].

### 4.1. Bubble expansion time period

The bubble expansion time period ($t_{\text{bubble}}$) is small, and is of the order of 80 ms in the plot shown in Fig. 11. If we consider the bubble volume ($V$) as: $V = dB$, where $d_i$ is the inner channel diameter and $B$ is the bubble area in contact with the microchannel wall, i.e. $B$ is the bubble area once the bubble has already become partially confined. We can set the power imposed at the channel wall $E_p$ equal to the bubble growth over time as seen in Eqs. 8 and 9, with $\rho_v$ and $L_v$ as the vapour density and latent heat respectively, where $Q^*$ is the heat flux provided at the contact line of the bubble (incorporating the heated bubble perimeter)

$$\rho_v L_v \left( \frac{dB}{dt} \right) = E_p$$

$$\rho_v L_v d_i \left( \frac{dB}{dt} \right) = 2Q^* \times B$$

Rearranging and integrating Eq. (9) gives Eq. (10). To determine the constants of integration, we have assumed that at: $t = \tau$, $B(t) = B_0$, where $B_0$ is not zero but an area value of the bubble before it is confined by the channel dimensions, i.e. before the bubble grows large enough to make contact with the heated channel walls. $B_0$ is also a function of the inlet mass flowrate, $m_{\text{in}}$, since the inlet liquid velocity through the channel affects the convective flow, and the displacement of the bubble in the microchannel

$$B(t) = B_0 \exp \left( \frac{2Q^* \times \left( t - \tau \right)}{\rho_v L_v d_i} \right)$$

The area of the bubble observed in Fig. 6 has been calculated and is plotted in Fig. 7 as it varies over time. The characteristic time $t_\tau = \frac{\rho_v L_v d_i}{2Q^*}$ for the bubble expansion, obtained from the relationship shown in Eq. (10) where $Q^* = 2Q'$, gives a characteristic time of 43 ms ± 16x, which is qualitatively in agreement with the observed case at the flow conditions shown. A more realistic model taking into account the real 2D shape of the bubble is needed, so as to have a better accuracy on the bubble expansion time.

### 4.2. Wall and liquid heating time period

The time period between the pressure fluctuations ($t_{\text{period}}$), i.e. whilst the temperature at the microchannel wall increases, is between 8 and 16 s, see Fig. 8, is approximately two orders of magnitude larger than the bubble expansion time period. This time period in-between the fluctuation instabilities is understood to be due to the time required for the microchannel wall to heat.

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**Fig. 11.** $T_{\text{wall}}$ and the heat transfer coefficient, correlated with the microchannel pressure fluctuations at both the inlet and outlet taken over 1400 ms, taken from Fig. 9, and illustrating the fast bubble expansion time step. Experiment conditions: $d_i = 771 \, \mu\text{m}$, n-Pentane, $G = 3.52 \, \text{kg m}^{-2} \text{s}^{-1}$, $U_m = 5.76 \, \text{mm s}^{-1}$ and $Q^* = 2.33 \, \text{kW/m}^2$.

Baseline pressure values, and this time step encompasses a much longer time period of between 8 and 16 s. This time is perceived to be the time required to heat the microchannel wall and also the heating time necessary for the liquid to increase from its inlet temperature to its saturation temperature. Once the saturation temperature of the liquid is reached, and a nucleated bubble in the channel starts to grow, the bubble becomes confined, and at a certain moment the sudden vapour expansion occurs.

### 4. Time step models

The geometry of the microchannel is rectangular ($d_i \times w_i$) with a high aspect ratio of 20 as illustrated previously in Fig. 1. Heat input
The time period is hence heavily dependent on the thermal properties (heat capacity \( C_p \), thermal conductivity \( k \)) and thickness of the borosilicate glass wall. Assuming that the rear face of the channel is cooled by the n-Pentane liquid with a convective exchange coefficient \( h \) and the temperature \( T_{in} \), the problem is modelled as presented in Fig. 13. The convective exchange on the front face between the air and the surface is negligible.

One-directional heat transfer in the \( x \)-direction is considered here. A heat flux step (Heaviside function), \( Q_{in}^{*}(t,0) \), is provided at the outer microchannel wall. The outer and inner wall temperatures are denoted as \( T_{out}(t,r) \) and \( T_{in}(t,r) \) respectively. The problem is to find the temperature variation with time of the internal face \( T_{in}(t,0) \) and to see if this is comparable to the temperature increase over time during the waiting period in-between the flow instabilities as evidenced in Figs. 8–10. The modelling of this problem can be realised with the thermal quadrupoles method \([22]\). The Laplace temperature-vector corresponding to the Laplace transformation of the inner channel temperature and to the heat flux imposed on the inner wall face is linked to the same vector composed of the outer temperature and to the heat flux penetrating the outer channel wall, by the matrix equation seen in Eq. \((11)\), where \( A, B, C \) and \( D \) are also defined in Eq. \((11)\) and \( s \) is the Laplace variable \((1/s)\). The borosilicate glass and the heat exchange are modelled with \( 2 \times 2 \) matrix. In the Laplace space, the problem is written as follows:

\[
\begin{bmatrix}
\phi_{in}(s,0) \\
\phi_{out}(s,0)
\end{bmatrix} = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \times \begin{bmatrix} 1 \\ 0 \end{bmatrix} \times \begin{bmatrix} Q_{in}^{*}(s,r) \\
\phi_{out}(s,r)
\end{bmatrix}
\]

It should be noted that at ‘in’ \( x = 0 \), and at ‘out’ \( x = r \) (where \( r \) is the thickness of the glass wall). \( \phi_{in}(s,0) \) is the Laplace transform of \( T_{in}(t,0) - T_{0}, \phi_{out}(s,0) = 0; \phi_{in}(s,0) \) and \( \phi_{out}(s,r) \) are respectively the Laplace transform of \( Q_{in}^{*}(t,0) \) and \( Q_{out}(t,r) \) variables \( A, B, C \) and \( D \) are defined:

\[
\begin{align*}
A &= \cosh \left( \frac{s}{\sqrt{2}} \right) \\
B &= \frac{1}{K\sqrt{2}} \sinh \left( \frac{s}{\sqrt{2}} \right) \\
C &= k\sqrt{2} \sinh \left( \frac{s}{\sqrt{2}} \right) \\
D &= A
\end{align*}
\]

The Laplace transform of a step function is \( 1/s \). So the precedent expression in Eq. \((11)\) can be written as Eq. \((12)\) as follows:

\[
\begin{bmatrix}
\phi_{in}^{*}(s,0) \\
\phi_{out}(s,0)
\end{bmatrix} = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \times \begin{bmatrix} 1 \\ 0 \end{bmatrix} \times \begin{bmatrix} 0 \\
\phi_{out}(s,r)
\end{bmatrix}
\]

Finally, we can obtain the expression of the inner temperature in Laplace space, Eq. \((13)\):

\[
\phi_{in}^{*}(s,0) = \frac{A/h + B Q''}{C/h + A} \times \phi_{out}(s,r)
\]

After an inverse Laplace transform, we obtain using this quadrupole method the following outer wall temperature presented in Fig. 14. Fig. 14 is a plot comparing the experimental temperature profile.
obtained by the IR camera over time with the modelled temperature profile obtained by the quadrupole method, as explained previously.

The time period presented in Fig. 14 is over 11 s as the wall temperature profile increases from approximately 39.0 °C to 47.2 °C. We obtain a good agreement (±0.5 °C) for the temperature evolution over time, indicating that the time period between two instabilities is driven by the heat capacity of the glass. This affirms our initial postulation that the time period in-between the fluctuation instabilities is understood to be due to the time required for the microchannel wall to heat up (\(\mu C_f\)).

5. Conclusion

In this paper simultaneous flow visualisation and data measurements of pressure and temperature have been performed, with the use of both a high-speed and an IR camera, to investigate flow boiling instabilities of n-Pentane in a single microchannel of hydraulic diameter 771 µm. The periodic oscillations of channel wall temperature, the local heat transfer coefficient and the inlet and outlet pressure were recorded and correlated. In the periodic flow boiling regime presented in this paper, bubbles were seen to nucleate, grow and quickly elongate upstream and downstream during rapid vapour expansion, causing the increase in the wall temperature and the pressure at the inlet and outlet to fluctuate. This type of fast vapour expansion instability is specific to microchannels due to the confined space available for bubble growth. The temperature at the microchannel wall was seen to increase whilst heat was also provided to the bulk liquid, whose inlet bulk temperature increased to the saturation temperature. Inversely, at periods where there were large temperature excursions, the heat transfer coefficient was at a minimal value. The two time periods of the flow instability, namely \(t_{\text{bubble}}\) and \(t_{\text{period}}\), have been analysed and modelled. Since we obtain a good agreement between the experimental times periods and the modelled ones for both the bubble expansion instability and for the channel wall heating time period in between the fluctuations, we can affirm that the frequency of the instabilities in a microchannel are driven by both the bubble dynamics and the channel wall thermal properties. An augmentation of the heat transfer coefficient was observed during two-phase periodic flow boiling. The enhancement of the heat transfer was a global phenomena, however it is important to note that in conjunction with the heat transfer enhancement were accompanying pressure fluctuations, with overpressure in the microchannel being as great as 33 mbar (over a 410% increase) at certain instances.

References