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Flow boiling of ethanol/water binary mixture in a square mini-channel

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Abstract: Two-phase flow heat transfer was examined in a single 5 mm inner hydraulic diameter square channel in a vertical orientation. The channel uses a resistive coating to allow for transparent heating of the walls. Transparency of the heating enabled high speed visualization of the boiling phenomena at various heat and mass fluxes. The pressure is monitored at the inlet and outlet of the channel. Infra-red thermography is used to map the external wall temperature of the channel and the local heat transfer coefficient is estimated from the local wall temperature and the saturation temperature of the liquid.

Ethanol, deionized water and a 5% v/v ethanol/water mixture were used as working fluids. Three mass fluxes (0.33, 0.66 and 1.00 kg/m²s) were tested as well as three heat fluxes (2.8, 4.2 and 6.1 kW/m²). Experiments were conducted in a controlled temperature environment (40°C). The addition of ethanol into water (5% v/v ethanol/water mixture) was found to enhance heat transfer resulting in higher heat transfer coefficients than for either of its pure components.

Keywords: flow boiling, heat transfer, mixtures, micro-channels

1. Introduction

Two-phase flow heat transfer is a method of heat transfer that has been under significant investigation over the past few decades. It is well known that heat transfer with phase change can result in increased heat transfer coefficients compared to single-phase heat transfer. These higher heat transfer coefficients occur naturally in some heat exchangers – primarily condensers and evaporators – which are widely used in industry. Recently, however, several studies were carried out focusing on two-phase flow heat transfer in the microscale. It has been suggested that two-phase flow increases heat transfer efficiency in microscale systems, and that it could thus be expected to serve as a competent cooling mechanism, for example in the cooling of microelectronic devices. Progress in the last few decades in the electronics sector created a requirement for increased cooling efficiencies. Moreover, Micro-Electro-Mechanical Systems (MEMS), micro chemical reactors and compact heat exchangers, as well as other micro systems would benefit from increased heat transfer in the microscale. Hence, two-phase flow heat transfer has received significant attention from the international scientific community with the identification of the controlling mechanisms at the microscale being the main objective.

As reported by various studies [1-6], several factors have been found to affect the heat transfer of micro-cooling systems and enhanced heat transfer rates can be achieved by varying these factors. The main factors recognised are: the hydraulic diameter of the channels used (Yan and Lin [1], Lin et al. [2]), their geometry (Tran et al. [3]), as well as the mass flux and the heat flux applied to the system. Moreover, the working fluid itself was found to be a determining factor in the heat transfer capability of a micro-system. A significant portion of research had been focused on different refrigerants, water and organic fluids. The thermophysical properties of an operating fluid were suggested to be important factors in heat transfer, since at the microscale surface tension and capillary forces are believed to have significant influence on heat transfer (Thome [4]). Moreover,

the latent heat of vaporisation and fluid viscosity are indicated to have a significant role in microscale flow boiling, according to Peng et al. [5] who suggest that liquids with higher liquid to vapour density ratio, greater thermal diffusion coefficients and larger latent heats require higher heat flux for the initiation of nucleation. After the onset of nucleation, liquids that have lower surface tension and lower latent heat of vaporisation contribute to higher heat transfer coefficients, as stated by Greco and Vanoli [6].

A number of decades ago, chlorofluorocarbon refrigerants were the fluids of choice for cooling purposes, due to their thermal properties and stable structures. However, the impact of the fluids on the environment has been identified and investigated in recent times and hence, replacement liquids with similar heat transfer efficiencies have been considered, including mixtures. Miscellaneous multi-component mixtures have been used as working fluids in studies in order to determine their potential for heat transfer in micro-scale systems compared to their pure components or other pure liquids. Water is one of the most commonly used operating fluids as it is easily accessible and also a component in many organic binary mixtures. Kaya et al. [7] used water and examined the effect of high mass flux on critical heat flux (CHF) in fluid boiling, observing that CHF increased with increasing mass flux. It was also observed that high heat removal rates could be achieved (more than 300 MW/m^2) by combining short heated tube length, high mass flux and small hydraulic diameter. Yu et al. [8] noticed as well that CHF quality was dependant on mass flux, while for water used in a small tube ($d_h = 2.98 \text{ mm}$). However, it was recognised that boiling heat transfer of water is essentially only a function of heat flux but not mass flux in this study. This trend could potentially be translated as the domination of nucleate to convective heat transfer in small channels. Nevertheless, Kosar et al. [9] (channel $d_h = 227 \mu\text{m}$, deionised water) pointed out that both nucleate and convective heat transfer mechanisms were dominant, depending on the mass and heat flux applied to the system. Sumith et al. [10] (water, $d_h = 1.45 \text{ mm}$) suggested that liquid film evaporation was the predominant heat transfer mechanism.

Pool boiling of binary mixtures of water with each of 2-propanol and methanol was investigated by McGillis et al. [11] and the CHF achieved, when small additions of alcohol were made to pure water, was found to be significantly higher than that of pure water. Fu et al. [12] experimented using ethanol/water mixtures in a diverging microchannel with microcavities. They showed that two-phase flow heat transfer coefficient was strongly dependant on the composition of the mixture, as well as on the wall superheat. Moreover, they concluded that addition of ethanol into water significantly increased the CHF recorded, with the maximum CHF being achieved for a 0.1 ethanol mole fraction. More specifically, the CHF of a mixture of 10% ethanol in water was found to be 110% that of pure water, and 234% that of pure ethanol. This increase in CHF was considered to be a result of the Marangoni effect occurring due to the addition of ethanol.

Mixtures of water with ethanol, propanol and butanol were examined by Suzuki et al. [13] in a subcooled flow boiling system to determine the effect that the addition of an organic fluid into water would have on heat transfer. It was observed that the maximum heat flux of the system increased with the addition of the organic fluid and that no dry-out was recorded under these circumstances. The maximum heat flux was found to be about 10% higher for the 10 wt% ethanol/water mixture compared to that for pure water, as was the case for Fu et al [12] above.

In this study, new aspects of flow boiling heat transfer will be examined for a 5% v/v ethanol/water mixture and its pure components (ethanol, water). The heat transfer coefficients of the three fluids, as well as the pressure drop of the system will be determined in order to investigate the effect that the addition of ethanol in water has on heat transfer rates and its potential as an enhanced fluid. The flow patterns were also assessed to provide a complete view of the prevailing phenomena.

2. Experimental apparatus and procedure

2.1. Test section

In this study a square cross-section channel of 5 mm inner hydraulic diameter orientated with its axis vertical was used as the test section. The channel was manufactured from borosilicate glass and the outer surface was sputtered with tantalum, allowing the channel to become electrically conductive while remaining transparent. Thus, heat could be applied to the system using a power supply and at the same time the flow patterns within the channel could be monitored with the use of a high speed camera. The wall thickness of the channel was measured to be 0.7 mm and the test section heated length was 72 mm.

The working fluids used for the experiments were a 5% v/v ethanol/water mixture and its pure components: deionised water and ethanol. Hence, a comparison of the heat transfer of the mixture

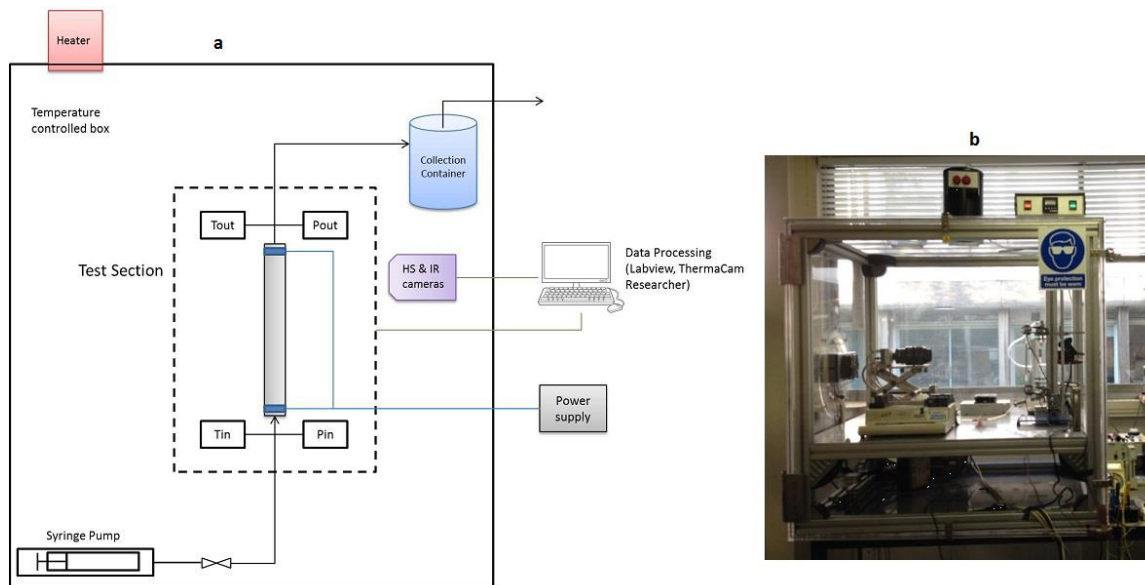


Figure 1. Experimental setup (a) schematic diagram, (b) picture (showing the Plexiglas box with the equipment within)

with the pure components could be obtained.

2.2. Experimental setup

The experimental setup consisted of a syringe pump that supplied the liquid to the channel and a DC power supply that provided the necessary power to heat up the system, through a transparent resistive coating on the outside of the test chamber. The experiments were conducted in a 1 m³ (1×1×1 m³) Plexiglas box which allowed for the control of the ambient temperature. The temperature was controlled by a PID controller mounted on the top of the Plexiglas box as depicted in Figure 1, in order to ensure minimum heat losses from the test section to the environment.

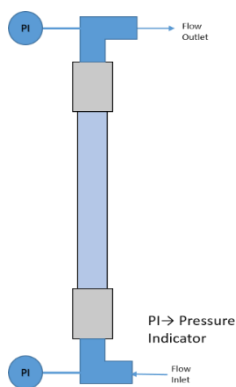


Figure 2. Schematic diagram of pressure sensors positioning

The temperatures at the inlet and outlet of the channel were measured and recorded by two K-type thermocouples and the PID controller was connected with a third K-type thermocouple that was positioned inside the box. The pressure drop across the channel was also obtained during the experiments using pressure sensors at the inlet and outlet of the channel (Figure 2). Two Omega PXM209 pressure transducers were used, measuring at a frequency of 6 Hz and recorded using National Instruments[®] software. A Mikrotron

MC1310 high speed camera (set at 500 fps) and a Thermovision[®] 900 infrared (IR) camera (6 fps) were simultaneously recording the growth and movement of vapour bubbles within the channel and the temperature profile of the outer channel wall. Experiments were conducted with three mass fluxes (0.33 kg/m² s, 0.66 kg/m² s and 1.00 kg/m² s) and three heat fluxes (2.8 kW/m², 4.2 kW/m² and 6.1 kW/m²) for all the working fluids. The inlet temperature of the fluid was maintained constant at 40°C, the same as the controlled environmental temperature, and with an atmospheric pressure of 1 atm. Thus, the subcooling temperature for each fluid was different, being 60.0°C for water, 77.4°C for ethanol and 54.0°C for the 5% v/v ethanol/water mixture.

3. Data reduction

Local heat transfer coefficients were obtained from the spatial temperature measurements at the wall and the local liquid saturation temperature by the following method.

3.1. Heat losses

The power supplied (Q_{sup}) to the system was computed using the voltage (V) and current (I) applied to the channel, using the equation: $Q_{sup} = VI$.

(1)

For the experiments run with single-phase flow, the effective heat transferred to the liquid (Q_{eff}) was calculated as:

$$Q_{eff} = \dot{m}c_{P,L}(T_{out,avg} - T_{in,avg}) \quad (2)$$

where \dot{m} is the mass flux (kg/s), $c_{P,L}$ is the specific heat capacity of the liquid (J/kg K), $T_{in,avg}$ and $T_{out,avg}$ are the inlet and outlet temperatures respectively (averaged over time).

$$\text{The heat flux was then calculated as } q = Q_{eff}/A_{w,in} \text{ (kW/m}^2\text{)} \quad (3)$$

where $A_{w,in}$ was the inner channel wall surface area (m²).

For the cases where two-phase flow was present in the channel, the effective heat transfer to the fluid (Q_{eff}) was calculated from the heat supplied, Q_{sup} , minus the heat losses to the environment:

$$Q_{eff} = Q_{sup} - Q_{conv} - Q_{rad} \quad (4)$$

The heat loss was estimated taking into account the convective (Q_{conv}) and radiative (Q_{rad}) heat losses.

Convective heat loss was calculated by the following equation:

$$Q_{conv} = h_{conv}A_{w,out}(T_{w,avg} - T_{box}) \quad (5)$$

where h_{conv} is the convective heat transfer coefficient of air and it is estimated with the use of empirical correlations for natural convection. $A_{w,out}$ is the outer channel wall surface area, $T_{w,avg}$ is the

average wall temperature over time and for the overall channel surface, and T_{box} is the temperature around the test section, hence the temperature in the Plexiglas box.

The radiative heat loss was then calculated from:

$$Q_{rad} = \varepsilon \sigma A_{w,out} (T_{w,avg}^4 - T_{box}^4) \quad (6)$$

where ε is the emissivity of the microchannel deposit and σ is the Stefan-Boltzmann constant ($5.6703 \times 10^{-8} \text{ W/m}^2\text{K}^4$). The channel walls emissivity was experimentally estimated and fell into a range of 0.76 to 0.80. Hence an emissivity of 0.78 was used for the channel wall.

3.2 Heat transfer coefficient

Due to the limited field of view of the IR camera, infrared images of only the top 2.8 cm of the channel were recorded and hence only the wall temperatures and local heat transfer coefficients for that part could be obtained (the total length of the channel test section was 7.2cm). The top part of the channel was chosen to be analysed since as it was seen that boiling was present in that part of the channel for all the cases where two-phase flow was achieved for all the fluids.

The Biot number (Bi) was calculated in an attempt to identify whether the temperature on the inside wall of the channel differs from the external surface temperature by comparing the conduction resistance to the convection resistance. The Biot number is:

$$Bi = \frac{h_c \delta_{wall}}{k_{glass}} \quad (7)$$

where h_c is the convective heat transfer coefficient, calculated from the Shah and London [14] correlation for fully developed laminar flow with uniform heat flux at the wall in a square channel, δ_{wall} is the channel thickness and k_{glass} is the channel thermal conductivity. The Biot number was calculated to be 0.035 and since $Bi \ll 1$, the difference between the inner and outer surface

temperature may be assumed to be negligible. Thus, the outer channel wall temperature can be used to calculate the local heat transfer coefficient, h_w .

$$h_w = \frac{q_{eff}}{T_{w,local} - T_{sat}} \quad (8)$$

where $T_{w,local}$ is the local wall temperature and T_{sat} is the saturation temperature of the working fluid. This equation was used when two-phase flow was present in the channel (See Table 1 for when two-phase flow is present and this method is used). T_{sat} for the pure fluids was calculated as a function of temporal average inlet and outlet pressures and it was found that the saturation temperature only changed by 0.5K ($\pm 1\%$) across the channel length. Hence, a constant saturation temperature was used for all the locations of the channel (average of inlet and outlet). For the 5% v/v ethanol/water mixture a similar behaviour was assumed, and a constant saturation temperature used, that was calculated from the vapour-liquid equilibrium data.

Table 1. Fluids behaviour at different conditions

	Ethanol			Water			5% Ethanol/Water		
q (kW/m ²)	2.8	4.2	6.1	2.8	4.2	6.1	2.8	4.2	6.1
\dot{m} (kg/m ² .s)									
0.33	Two-phase	Two-phase	Two-phase	Single-phase	Two-phase	Two-phase	Two-phase	Two-phase	Two-phase
0.66	Two-phase	Two-phase	Two-phase	Single-phase	Two-phase	Two-phase	Single-phase	Two-phase	Two-phase
1.00	Two-phase	Two-phase	Two-phase	Single-phase	Single-phase	Two-phase	Single-phase	Single-phase	Two-phase

In **single-phase** flow cases the following equation was used to calculate the local heat transfer coefficients:

$$h_L = \frac{q_{eff}}{T_{w,local} - T_{l,local}} \quad (9)$$

where $T_{l,local}$ is the local liquid temperature.

The inlet and outlet fluid temperatures in this study were obtained by two thermocouples at the inlet and the outlet of the channel respectively. There were no other thermocouples along the length of the channel due to the damage they could potentially cause to the channel. Also, they could possibly induce disturbances to the flow patterns and hence, inaccuracies to the experimental results. The local liquid temperature was calculated, assuming constant average heat flux and therefore a linear variation in temperature from the inlet to the outlet, as shown by the energy balance:

$$4zq_{eff} = c_p \dot{m} d_h (T_{l,local} - T_{in}) \quad (10)$$

where z is a local position on the channel, d_h is the inner hydraulic diameter of the channel, T_{in} is the inlet liquid temperature. The local liquid temperature is therefore:

$$T_{l,local} = \frac{4q}{c_p \dot{m} d_h} z + T_{in} \quad (11)$$

$T_{w,local}$ is recorded using the IR camera.

3.3 Uncertainties

The experimental uncertainties arise primarily from equipment limitations. The main uncertainties were in the syringe pump velocity, the supplied power and the temperature measurements (mainly IR readings), that lead to uncertainties in the calculated values of mass and heat fluxes, as well as the heat transfer coefficients values. An uncertainty analysis was conducted and Table 2 summarises the results in percentage terms.

Table 2. Experimental measurement and calculations uncertainties

Parameter	Uncertainty (%)
Pressure	±0.2
Pump volumetric flow rate	±0.5

Voltage	± 0.8
Current	± 0.8
IR camera temperature	± 1
Mass flux	± 3
Heat flux	± 7.3
Pressure drop	± 0.5
Heat transfer coefficient	± 20

4. Results and discussion

4.1. High speed visualization results

In this set of experiments the liquid always entered subcooled at the bottom of the channel at the same inlet temperature (40°C). Therefore, boiling only started further along the length of the channel and, since the subcooling and the properties of each fluid were different, the nucleation position was also different for each fluid for the same operating conditions (Table 3). The nucleation positions were recognised by examining the high speed visual images obtained, with the use of imaging software (ImageJ).

Nucleation Position from entry [mm] (± 0.5 mm)										
Working Fluid	Mass flux [kg/m ² s]	0.33			0.66			1.00		
	Heat flux [kW/m ²]	2.8	4.2	6.1	2.8	4.2	6.1	2.8	4.2	6.1
Ethanol		8.2	6.2	0	12.3	43.0	9.5	0	0	20.3
Water		n/a	53.5	39.0	n/a	52.0	53.4	n/a	n/a	60.7
5% Ethanol/Water		47.0	19.5	47.1	n/a	53.0	47.0	n/a	n/a	45.0

Table 3. Nucleation positions for all the fluids at the different operating conditions

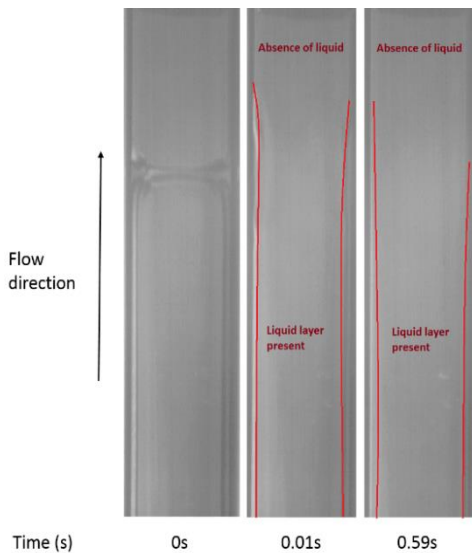


Figure 3. Ethanol boiling at $G=0.33\text{kg/m}^2\text{s}$ and $q=6.1\text{kW/m}^2$, top 2.8cm of the channel length is featured here

Ethanol was found to be the fluid for which a lower amount of heat was required for the initiation of boiling, as it was expected based on the properties of the fluids. Moreover, it was observed that bubble nucleation was more easily initiated for the ethanol/water mixture than for pure water, suggesting that the addition of even a small amount (5% v/v) of alcohol in water was adequate to influence the properties of the mixture significantly. Boiling was also found to be more vigorous in the case of pure ethanol and the ethanol/water mixture. However, when mass flux was decreased and heat flux increased, there was dry-out observed at the top of the channel, as it can be seen in Figure 3 for ethanol. Dry-out was also observed for water at $0.33\text{ kg/m}^2\text{s}$ and 6.1 kW/m^2 but not for the mixture. The lack of dry-out for the mixture suggested a more reliable heat transfer system for cooling high heat flux components.

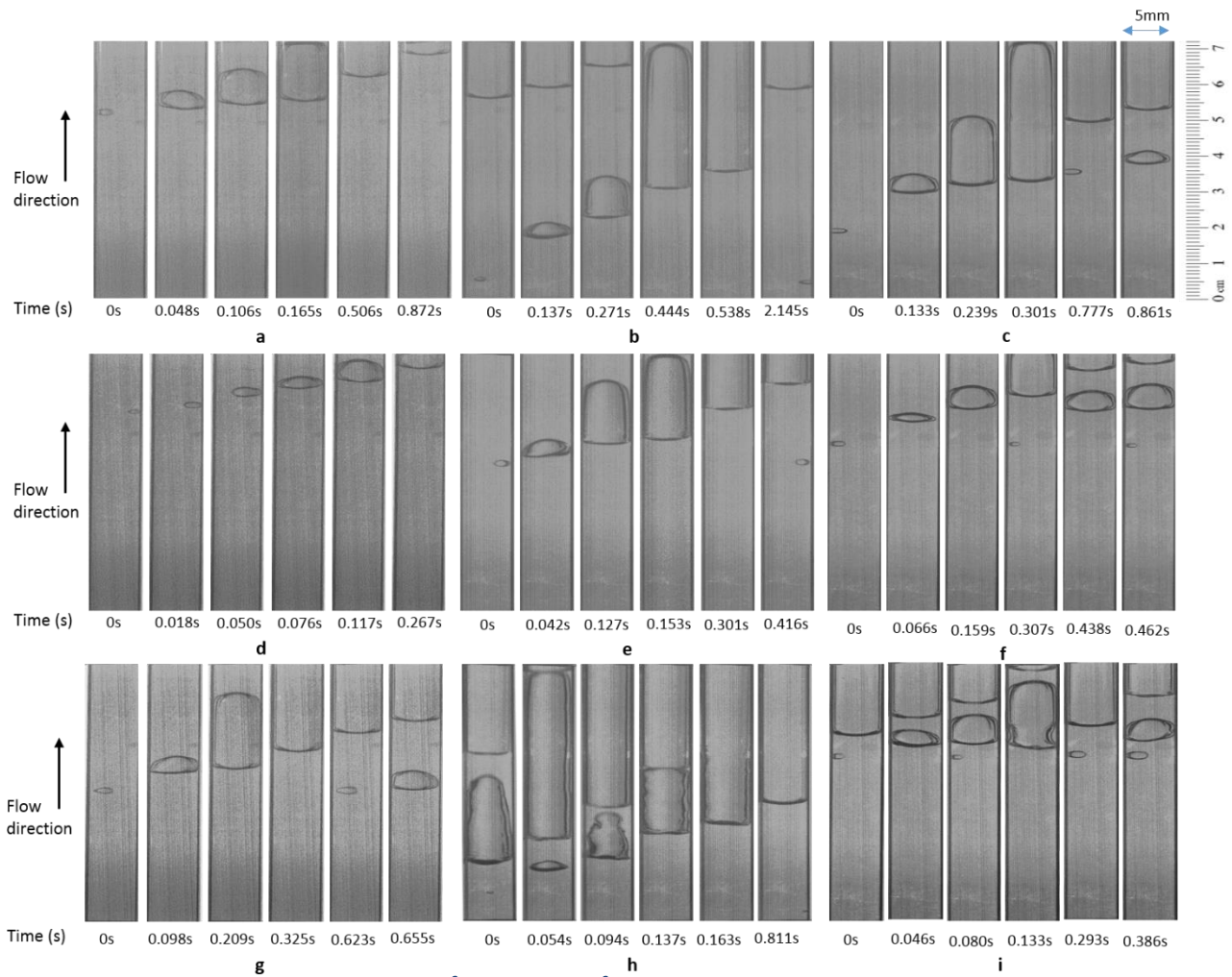


Figure 4. Flow regimes of boiling at $G=0.33\text{kg/m}^2\text{s}$ and 4.2kW/m^2 for (a) water, (b) ethanol and (c) 5% v/v ethanol/water mixture, at $G=0.66\text{kg/m}^2\text{s}$ and 4.2kW/m^2 for (d) water, (e) ethanol and (f) 5% v/v ethanol/water mixture, and at $G=0.66\text{kg/m}^2\text{s}$ and 6.1kW/m^2 for (g) water, (h) ethanol and (i) 5% v/v ethanol/water mixture, all the test section (7.2cm) is featured here

The boiling patterns observed varied depending on the operating conditions and the working fluid, but the main patterns recorded, as depicted in Figure 4, were bubbly, slug and elongated slug/annular flow. These observations are in agreement with previous studies such as Huh and Kim [15] and Mishima and Hibiki [16]. Moreover, it was observed that in all the cases a small bubble was initially formed that rapidly expanded into slug/annular flow. The sudden expansion of the vapour would lead to a momentary blockage or slowdown of liquid inflow and thus, to an increase in pressure drop. Liquid inflow was then restored and the channel was completely covered with liquid until the nucleation temperature was reached again. This repetitive bubble recoil, dewetting and rewetting, or ebullition cycle as referred to by Carey [17], is the mechanism that allows the heat of

vaporisation of the liquid to be used as a heat transfer mechanism and possibly increase the heat transfer efficiency of two-phase flow compared to single-phase flow.

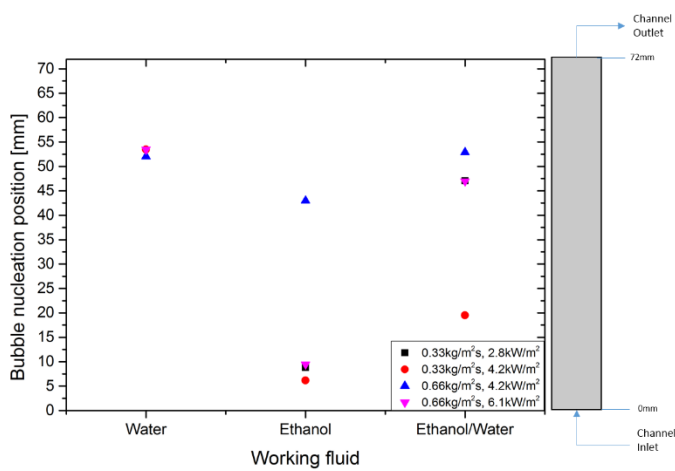


Figure 5. Nucleation positions for all the fluids at different mass and heat fluxes

that a rise of mass flux would enhance convective heat transfer, dropping the wall temperature at a higher rate and hence, an increase in the heat supply would be needed to initiate bubble nucleation. On the other hand, increasing heat flux while keeping a constant mass flux would result in nucleation starting closer to the channel inlet since the increased fluid temperature would assist the activation of nucleation sites.

Finally, it was noticed that for an increased mass flux and constant heat flux, boiling would start further along the channel. In some cases, an increased heat flux was required for the commencement of nucleation see Figure 5, which is in accordance with literature (Bertsch et al. [18]). This phenomenon could be explained by the fact

4.2. Wall temperature fluctuations for the different fluids

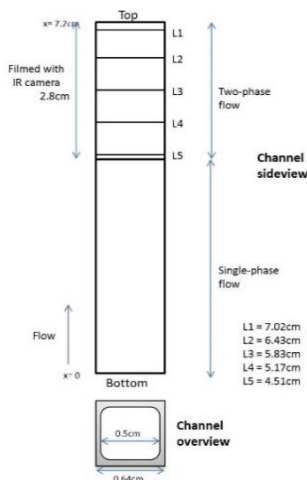


Figure 6. Schematic diagram of the local heat transfer coefficients on the channel

The temperature profile of the top 2.8 cm of the channel outer wall was recorded with the IR camera at 6fps (frames per second) and temperature fluctuations over time were observed, which are believed to be the result of boiling in the channel. Only part of the channel was thermally mapped due to limited field of view and focal distance of the IR camera. The temperature data at five positions, seen in Figure 6, at the top of the channel were analysed ($T_{wall,L1}$, $T_{wall,L2}$, $T_{wall,L3}$, $T_{wall,L4}$ and $T_{wall,L5}$).

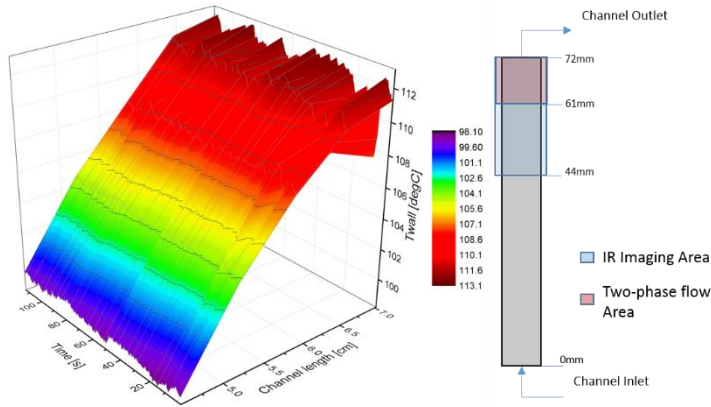


Figure 7. A 3-dimensional plot of wall temperature against channel length of water at $G=1.00\text{kg/m}^2\text{s}$, $q=6.1\text{kW/m}^2$

From the experiments conducted, it was concluded that when the system is either single-phase or at positions upstream of boiling, the wall temperature was constant over time with no fluctuations present. On the other hand, during two-phase flow,

fluctuations of wall temperature were noticed and given that the heat was uniformly applied on the channel, it could be suggested that these fluctuations were entirely a result of boiling. The aforementioned phenomena can be observed in Figure 7, where until a certain length along the channel ($x=60.7\text{ mm}$) there is no boiling present and hence there are no fluctuations in the wall temperature, while further along the channel the temperature starts fluctuating. Also, it could be implied that the fluctuations of the temperature were a result of the alternating boiling regimes within the channel (vapour occupying channel followed by its rewetting). It was seen that the temperature increased when the channel was filled with vapour and it decreased when liquid was covering the heated channel wall again.

Comparing the wall temperature for the different fluids (Figure 8), it can be seen that in the case of the mixture the temperature fluctuations were substantially smaller and almost insignificant,

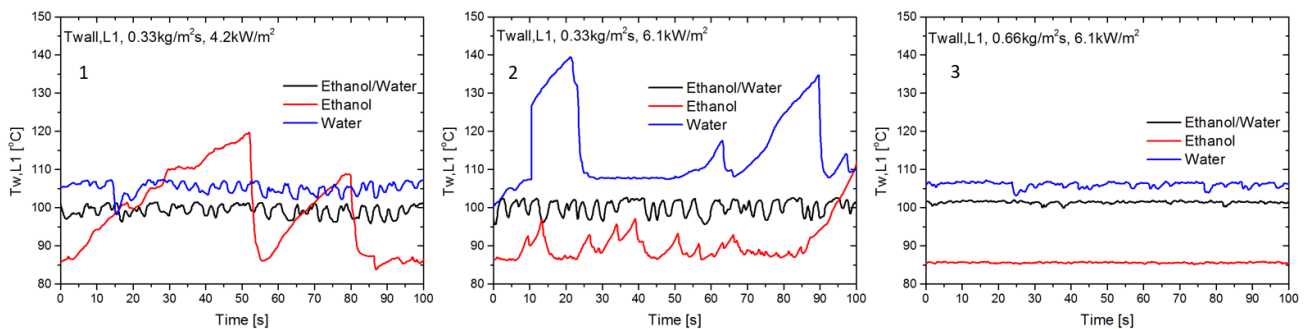


Figure 8. Channel wall temperature fluctuation of the 5% v/v ethanol/water mixture, ethanol and water at (1) $G=0.33\text{kg/m}^2\text{s}$, $q=4.2\text{kW/m}^2$, (2) $G=0.33\text{kg/m}^2\text{s}$, $q=6.1\text{kW/m}^2$ and (3) $G=0.66\text{kg/m}^2\text{s}$, $q=6.1\text{kW/m}^2$ at position channel position L1

compared to either the pure liquids.

Furthermore, it appeared that increasing heat flux, while keeping a constant mass flux, led to higher amplitude fluctuations for all the fluids. On the other hand, when mass flux was increased (at constant heat flux) it was evident that the amplitude of fluctuations was lower. Also, at experiments where two-phase flow was present, it was noticed that the wall temperature was lower further along the channel ($T_{wall,L1} < T_{wall,L5}$) which suggested that heat was removed at a higher rate, comparing to single-phase flow, where the reverse observation was made ($T_{wall,L1} > T_{wall,L5}$) (Figure 9). Finally, in the case of highest heat flux and lowest mass flux (6.1 kW/m^2 , $0.33 \text{ kg/m}^2\text{s}$) when ethanol and water were used as the working fluids, immense increases in wall temperatures were observed (Figure 9 (2,3)), which implies the presence of dry-out at the channel wall.

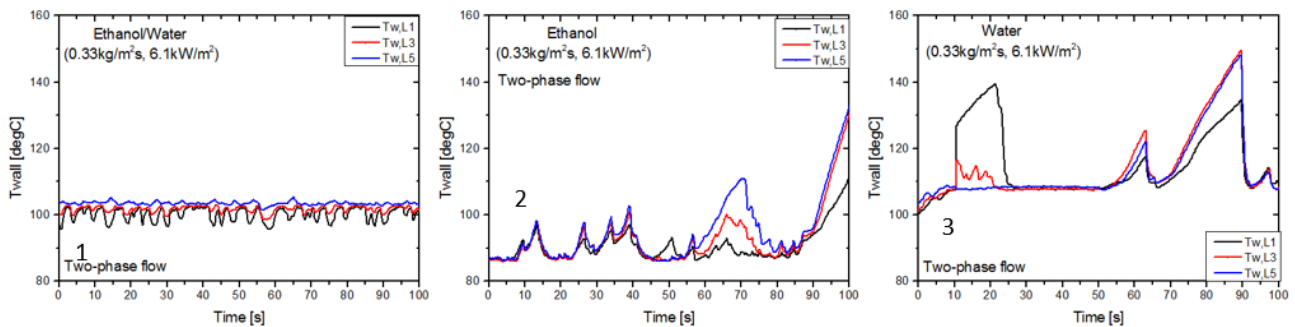


Figure 9. Channel wall temperature fluctuation at $G=0.33 \text{ kg/m}^2\text{s}$, $q=6.1 \text{ kW/m}^2$ for (1) ethanol/water, (2) ethanol and (3) water

4.3 Pressure drop

The pressure drop across the test section was also monitored in this study, in order to investigate the relationship between heat transfer and pressure effects. It was found that the pressure drop oscillated over time when boiling was present in the channel, and the magnitude and frequency of the fluctuations varied for different fluids and operating conditions. As was briefly explained in section 4.1, an increase in pressure drop was observed in the channel during bubble nucleation and especially when sudden vapour expansion occurred. The expansion of a bubble towards both the inlet and outlet caused a blockage of the liquid flow and led to the highest magnitude pressure drops, which were

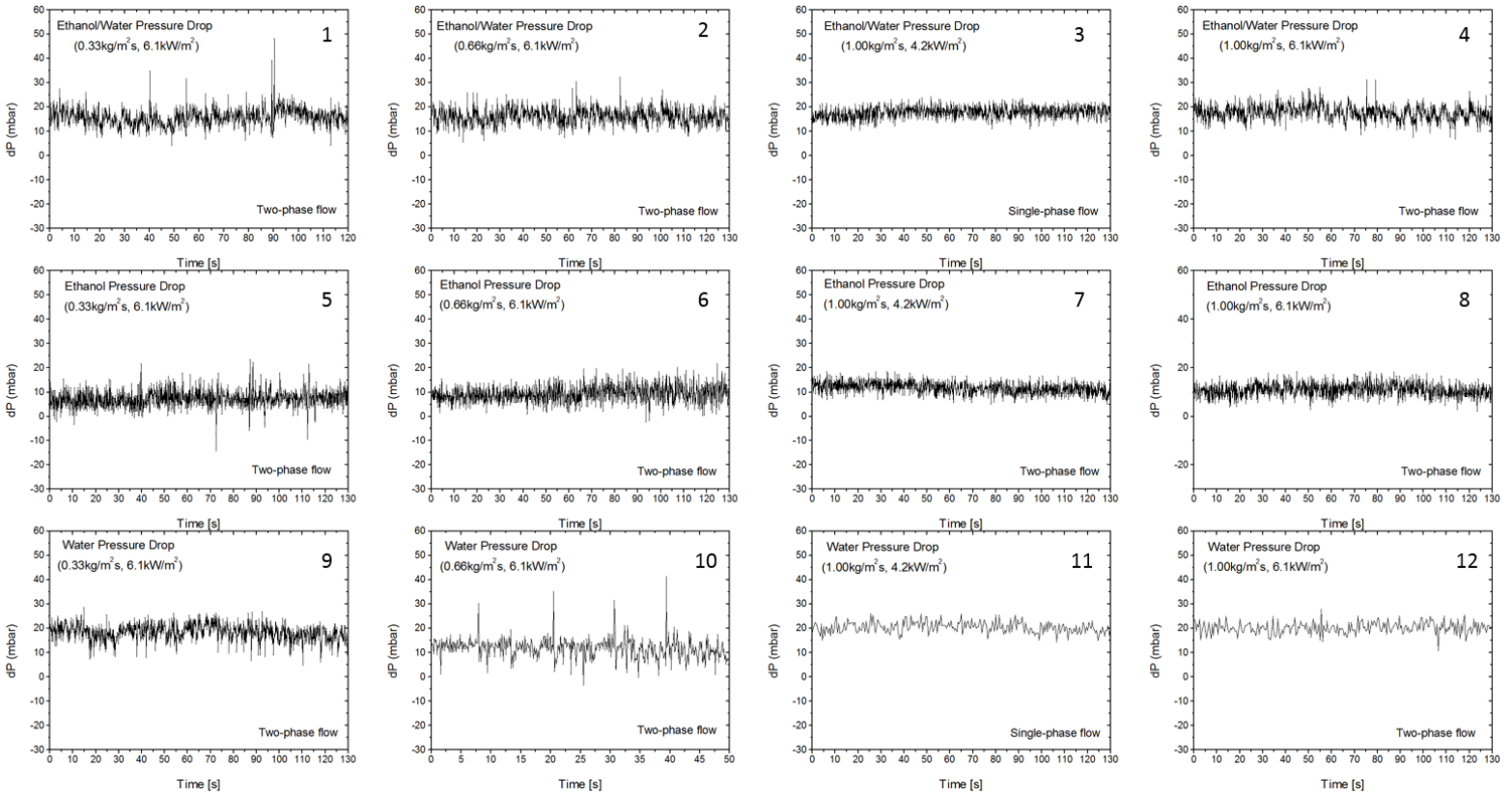


Figure 10. Pressure drop along the channel for:

5% v/v ethanol/water mixture at (1) $G=0.33\text{kg/m}^2\text{s}$, $q=6.1\text{kW/m}^2$, (2) $G=0.66\text{kg/m}^2\text{s}$, $q=6.1\text{kW/m}^2$, (3) $G=1.00\text{kg/m}^2\text{s}$, $q=4.2\text{kW/m}^2$ and (4) $G=1.00\text{kg/m}^2\text{s}$, $q=6.1\text{kW/m}^2$

Ethanol at (5) $G=0.33\text{kg/m}^2\text{s}$, $q=6.1\text{kW/m}^2$, (6) $G=0.66\text{kg/m}^2\text{s}$, $q=6.1\text{kW/m}^2$, (7) $G=1.00\text{kg/m}^2\text{s}$, $q=4.2\text{kW/m}^2$ and (8) $G=1.00\text{kg/m}^2\text{s}$, $q=6.1\text{kW/m}^2$

Water at (9) $G=0.33\text{kg/m}^2\text{s}$, $q=6.1\text{kW/m}^2$, (10) $G=0.66\text{kg/m}^2\text{s}$, $q=6.1\text{kW/m}^2$, (11) $G=1.00\text{kg/m}^2\text{s}$, $q=4.2\text{kW/m}^2$ and (12) $G=1.00\text{kg/m}^2\text{s}$, $q=6.1\text{kW/m}^2$

observed for mass and heat flux of $0.33\text{ kg/m}^2\text{s}$ and 6.1 kW/m^2 .

From Figure 10, it can be seen that there are two type types of pressure drop fluctuations present: low amplitude, high frequency ones and high amplitude, low frequency ones. The second type of fluctuations is more evident in the cases of higher heat flux and smaller mass fluxes where boiling

was more vigorous and the vapour expansion incidents were more violent. In Figure 10 (1, 5 and 10) the thigh peaks in pressure drop due to the sudden expansion of vapour can be seen more clearly. The smaller amplitude oscillations are believed to be the result of bubble flow due to the movement and collisions of the bubbles.

4.4 Local heat transfer coefficients

Local heat transfer coefficients were calculated as a function of time for five positions in the upper section of the channel, see Figure 11. It was concluded from the experiments performed that the higher heat transfer coefficients were for the ethanol/water mixture compared to its pure components, ethanol and water, at the same conditions. Higher heat transfer coefficients were observed in the cases of two-phase flow and the highest ones were noticed for the cases with the lowest mass flux and highest heat flux.

Moreover, it was seen that increasing the heat flux resulted in increased heat transfer coefficients, see Figure 11. However, increasing mass flux appeared to lead to a decrease in the heat transfer coefficients for all the fluids for the two lower heat fluxes (2.8 and 4.2 kW/m²). At the highest heat flux (6.1kW/m²), the increase of mass flux led to decrease heat transfer coefficients only for ethanol/water and it did not influence the ones obtained for pure ethanol and water in a clear way (Figure 11). Also, examining the calculated local heat transfer coefficients in synchronisation with the high speed imaging, it was observed that higher h_L values were obtained when slug/annular flow was present in the channel, as opposed to bubble flow being present. It could thus be suggested that in these cases another heat transfer mechanism was activated, such as liquid film evaporation working along with forced convection.

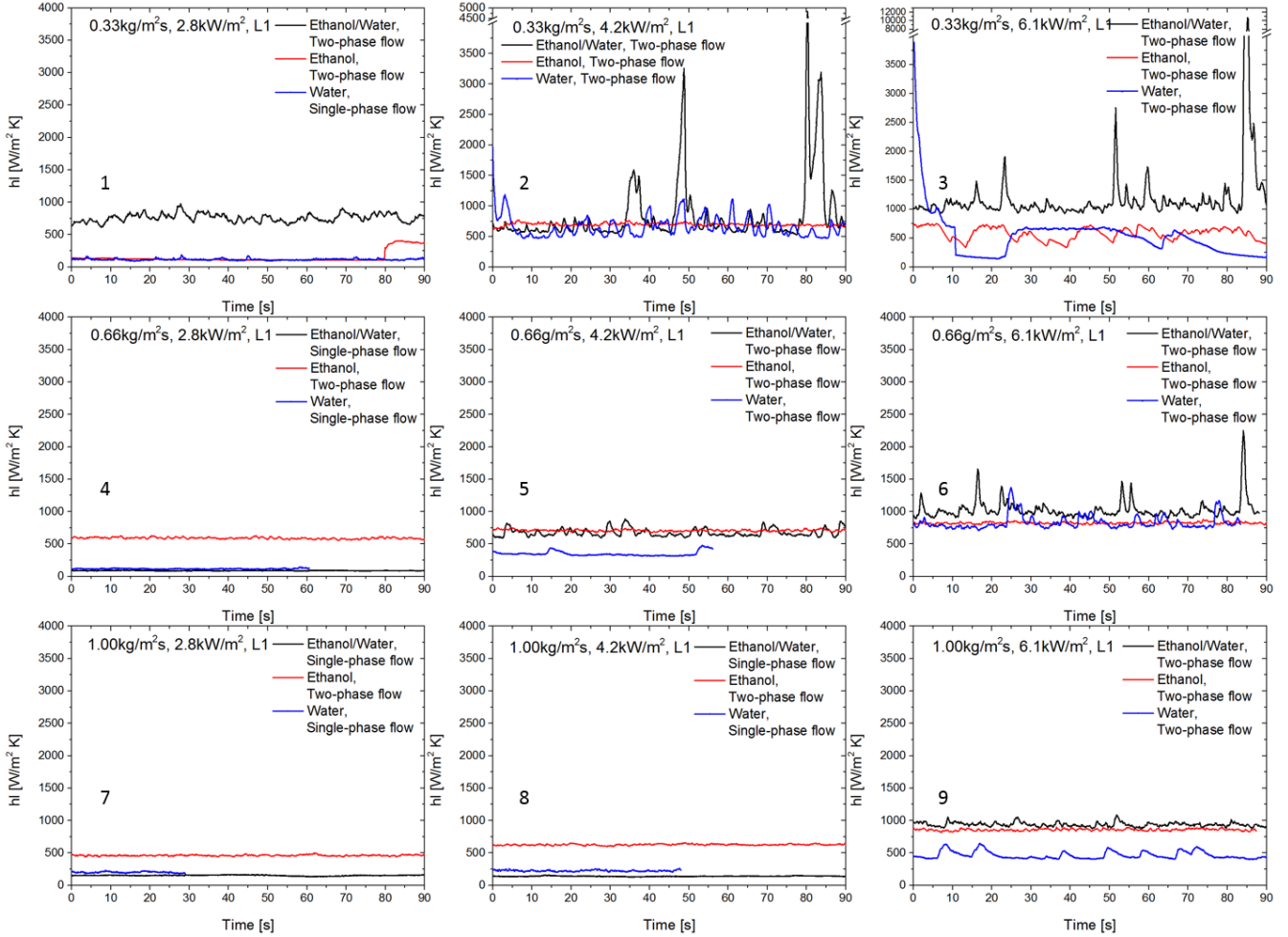


Figure 11. Local heat transfer coefficient over time of 5% v/v ethanol/water mixture, water and ethanol for:
 $G=0.33\text{kg/m}^2\text{s}$ at (1) $q=2.8\text{kW/m}^2$, (2) $q=4.2\text{kW/m}^2$, (3) $q=6.1\text{kW/m}^2$
 $G=0.66\text{kg/m}^2\text{s}$ at (4) $q=2.8\text{kW/m}^2$, (5) $q=4.2\text{kW/m}^2$, (6) $q=6.1\text{kW/m}^2$
 $G=1.00\text{kg/m}^2\text{s}$ at (7) $q=2.8\text{kW/m}^2$, (8) $q=4.2\text{kW/m}^2$, (9) $q=6.1\text{kW/m}^2$

Temporally averaged local heat transfer coefficients were also calculated to allow for a numerical comparison between the different fluids. These average local heat transfer coefficients at the lowest mass flux and highest heat flux ($0.33\text{kg/m}^2\text{s}$ and 6.1kW/m^2) at position L1 are: $420\text{W/m}^2\text{K}$ for ethanol, $540\text{W/m}^2\text{K}$ for water and $1200\text{W/m}^2\text{K}$ for the ethanol/water mixture (three times higher than that of ethanol and roughly twice as high as that of water). However, in the case of both the highest mass and heat flux ($1.00\text{kg/m}^2\text{s}$ and 6.1kW/m^2), the ethanol and the ethanol/water mixture h_L values were closer together, $940\text{W/m}^2\text{K}$ for ethanol/water and $860\text{W/m}^2\text{K}$ for ethanol, while the value of the water heat transfer coefficient was substantially lower ($460\text{W/m}^2\text{K}$). Principally, the heat transfer coefficients values for the fluids were closer together when mass flux was increased

and, given the uncertainty of the calculated values ($\pm 20\%$), it was not possible to distinguish which fluid was more efficient for heat transfer at higher mass fluxes. Also, the higher heat transfer coefficients values were calculated to be at the highest position of the channel (L1), while values closer to the channel inlet were lower (see Figure 12).

The trace for pure ethanol appears to be the most stable, with both pure water and the mixture having reasonably random fluctuations and it was noticed that higher heat transfer coefficients could be achieved, avoiding a possible dry-out that was found to occur for ethanol and water. The

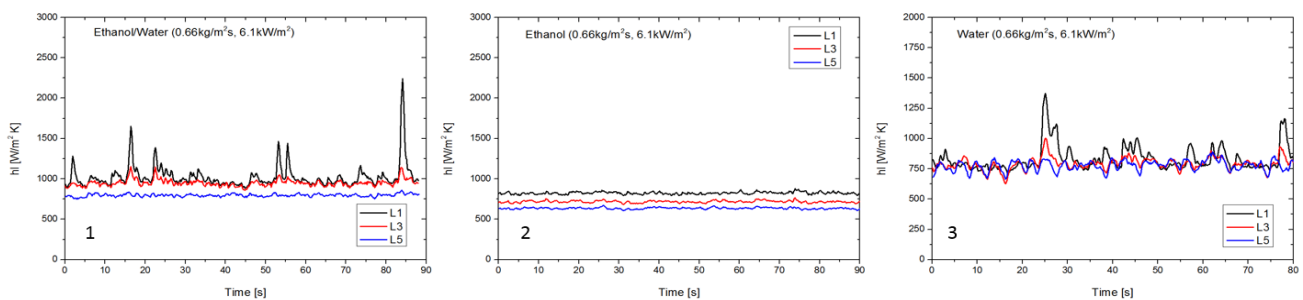


Figure 12. Local heat transfer coefficient over time of 5% v/v ethanol/water mixture, water and ethanol at $G=0.66\text{kg/m}^2\text{s}$ and $q=6.1\text{kW/m}^2$ for (1) ethanol/water (2) ethanol and (3) water

ethanol/water mixture is a mixture, for which boiling/evaporation occurs at different rates for each component (more volatile component evaporates first). Hence, a concentration gradient occurs, which leads to Marangoni forces being induced and thus, to the movement of bulk liquid towards the contact line (Fu et al. [12]). This behaviour results in a better liquid distribution in the channel, which in turn helps prevent dry-out from happening.

5. Conclusions

In this study, different aspects of two-phase flow heat transfer for a 5% v/v ethanol/water mixture and its pure components (water and ethanol) were investigated. High speed imaging of the process was obtained, along with the channel wall temperature and the pressure drop across the channel, while local heat transfer coefficients were also calculated.

Several boiling patterns were identified, that were common for all the fluids, but varied depending on the operating conditions used in each experiment. The boiling regimes observed were: bubble, slug and elongated slug/annular flow.

The channel wall temperature profile was also recorded over time and it was analysed to determine the effect boiling had on it, and more specifically, how different boiling regimes influenced its fluctuations. It was concluded that channel wall temperature fluctuated over time with two-phase flow, and that this was solely a result of boiling occurring in the channel. However, it was seen that for the ethanol/water mixture, the amplitude of the fluctuations was significantly decreased, allowing for a more stable heat transfer process over time.

Lastly, heat transfer of the test section was found to have been enhanced by the addition of a small amount of ethanol into water (5% v/v ethanol/water mixture), resulting in higher heat transfer coefficients than both of its pure components. The heat transfer coefficients fluctuations (over time) were compared to the high speed imaging of the process, and the reverse flow of liquid followed by the rewetting of the channel walls (passing of slug through the channel) was found to increase heat transfer rates. Finally, the use of the mixture can help to ensure the prevention of a wall dry-out, which could lead in a potential critical failure in a cooling system.

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