CONVECTIVE BOILING WITH ELECTROHYDRODYNAMIC ENHANCEMENT: THE INFLUENCE OF INLET QUALITY

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This work investigates the influence of alternating current electric fields on the flow patterns, associated heat transfer, and power penalty during convective boiling of HFE7000. A single-pass counter-flow heat exchanger is employed, whereby heated water flowing in the shell side transfers heat to the two-phase HFE7000 fluid flowing within the tube side. In a novel design feature, optical transparency is achieved by using a sapphire central tube surrounded by a Perspex water jacket containing the heated water. A stainless steel rod running concentrically through the tube acts as an electrode while the outer surface of the sapphire tube is coated with a thin layer of indium tin oxide forming an electrically conductive and optically transparent ground to establish an electric field across the working fluid. This unique test setup facilitates visualization of the flow patterns caused by the electrohydrodynamic (EHD) forces and allows high-speed videography of the HFE7000 while boiling. Tests were performed at a low mass flux (100 kg/m²s) and fixed average heat flux (12 kW/m²) for inlet qualities of 2%, 15%, 30%, and 45% and applied voltages of V = 0, 4, and 8 kV. The results show that the average heat transfer coefficient improves with applied voltage over the entire range of qualities tested. However, as inlet quality increases the heat transfer enhancement tends to decrease, as does the electrical power required for EHD (EHD penalty). Conversely, pumping losses were seen to increase as inlet quality increases.

KEY WORDS: electrohydrodynamics (EHD), convective boiling, heat transfer enhancement, flow patterns, HFE7000

1. INTRODUCTION

Heat transfer enhancement methods are in continual development to improve heat exchanger effectiveness. One such method, electrohydrodynamics (EHD) can offer increased heat transfer rates while the high-voltage low-current phenomenon allows electric fields to be established that can radically alter the flow patterns with low electrical power requirement (Cotton et al., 2012; Di Marco, 2012) and low pressure drop penalties (McGranaghan and Robinson, 2014a). However, the mechanisms by which EHD influences the fluid flow and heat transfer within a closed heat exchanger are still not thoroughly understood, although significant progress is ongoing (Cotton et al., 2005, 2012; Sadek et al., 2006, 2008; Ng, 2010; Di Marco, 2012; McGranaghan and Robinson, 2014a,b). For two-phase flows subject to EHD, Jones (1979), Chang and Watson (1994), Cotton et al. (2012), and Di Marco (2012), among others, posed the following expression for the forces induced on a dielectric medium within an electric field:

\[ f_e = \rho_e E - \frac{1}{2} E^2 \nabla \varepsilon - \frac{1}{2} \nabla \left[ \rho E^2 \left( \frac{\delta \varepsilon}{\delta \rho} \right)_T \right] \]  

(1)

The three terms in Eq. (1) represent the electrophoretic, dielectrophoretic, and electrostrictive components of the EHD body force. The first term is the only force that depends on the polarity of the electric field and is known as the...
electrophoretic or Coulomb force. It is related to the force acting on the net free charge in the fluid. The second term is related to the spatial gradient of the permittivity and is termed the dielectrophoretic force. In a two-phase scenario where there is a large permittivity change across the thin vapor/liquid interface, this force can be significant according to Cotton et al. (2005). It was believed by Cotton et al. (2005) and others to be the most dominant force during flow boiling since the coupling of the inertial and EHD forces can cause liquid and vapor redistribution as well as migration of bubbles or droplets. The last term is the electrostrictive force and arises due to inhomogeneity in the electric field strength and variation in the dielectric constant with density. It is generally considered insignificant in two-phase heat transfer (Cotton et al., 2005), although this is a matter of some debate.

Recently, with regard to improving heat transfer in shell and tube heat exchangers, researchers have experimentally investigated the EHD technique with convective boiling in tubes while using visualization techniques, such as high-speed videography, to elucidate the behavior of the fluid within tubular test sections of similar geometry to that used in this study (Singh et al., 1995; Bryan and Seyed-Yagoobi, 2000; Cotton, 2000; Cotton et al., 2001, 2005; Sadek, 2004; Sadek et al., 2006, 2008; Di Marco, 2012). Previous research, such as that of Cotton et al. (2005, 2012) and McGranaghan and Robinson (2014a) have shown that both direct current (DC) and alternating current (AC) electric fields can have a significant influence on the two-phase heat transfer. It has been postulated that within tubular geometries featuring concentric electrodes the EHD forces act radially on the axially flowing fluid in such a way as to redistribute the phases and cause flow regime transitions that are beneficial with regard to heat transfer. The EHD forces were seen to attract fluid from the stratified layer toward the central electrode and expel it upward and outward within the tube, causing increased liquid disturbance and rewetting of dry tube walls and thus improving heat transfer (McGranaghan and Robinson, 2014a). More recently, McGranaghan and Robinson (2014a) have shown that EHD does not solely cause flow regime transitions but that unique flow regimes can be established under the action of electric fields. Two dimensionless numbers, the EHD number and the Masuda number, which relate the inertial forces to the EHD forces, are important in predicting when EHD forces can dominate over the inertial forces for a given flow rate (Cotton et al., 2012).

Cotton et al. (2001) noted a special case at frequencies of 60 Hz, which they referred to as “oscillatory entrained droplet EHD two-phase flow.” In this case, an annular flow regime is clearly visible but additionally with droplets as large as 2 mm diameter oscillating at 120 Hz within the vapor core. Cotton et al. (2001) ascribed the phenomenon to the continuous construction and destruction of two separate flow regimes, namely, stratified flow and intermittent annular or entrained droplet flow.
In a subsequent study on condensing flows, Sadek et al. (2008) also investigated the effect of high-voltage frequency on two-phase flow regimes. Sadek et al. (2008) utilized a concentric electrode layout and voltages in the range of 0–8 kV and square waves at frequencies between 4 and 1 kHz. They found that at lower frequencies (4 Hz ≤ f ≤ 10 Hz) the liquid could respond to the forces before the AC wave completed its cycle. Therefore, an increase in frequency in this range resulted in increased extraction from the liquid layer and redistribution of the liquid to the top of the tube. In the intermediate range of frequencies (10 Hz < f < 100 Hz) the time period of the wave was less than the time necessary for the liquid extraction cycle; however, liquid interaction with the electrode was high. An increase in the frequency in this region was found not to affect the flow above the electrode, but rather to cause higher intensity liquid/vapor interactions below the electrode. Increasing the frequency further (f ≥ 100 Hz) meant the fluid could not respond quickly enough to the rapidly varying voltage cycle; therefore, the liquid/vapor interactions decreased.

In a study on EHD in condensation, Gidwani et al. (2002) also found that EHD was more effective in stratified wavy and stratified flow patterns. This was due to the dominance of the EHD forces over the momentum flux, resulting in higher fluid extraction and crossed motions within the tube. They noted EHD enhancement fell from 3.5- to 2.4-fold as the quality increased from 10% to 50%; over the same interval the pressure drop also decreased from 4.1- to 2.7-fold.

Ng (2010) also studied the mechanisms of EHD two-phase flow structures, and showed that at lower qualities, liquid is extracted from the lower stratum and attracted to the central electrode due to the EHD body force. Then, once the electrode is surrounded by the liquid, twisted liquid cones are formed, which emanate radially from the surface of the electrode to the tube wall surfaces. Using pulsed width control of the DC EHD voltage, Ng (2010) found that the twisted cones were a transitory feature. The twisted cones help expel liquid from the central electrode to drier areas of the tube, as well as being a possible gravity-independent phenomenon, which may be applicable in microgravity environments.

Although EHD is predominantly employed for its enhancement role, a number of researchers have noticed that the EHD effect could, in certain conditions, inhibit the convective heat transfer coefficient. Feng and Seyed-Yagoobi (2001) found that a transition from EHD enhancement to EHD suppression occurred near the annular-to-mist transition region. This was supported by Bryan and Seyed-Yagoobi (2000), who also performed experiments with R134a and observed a similar suppression with EHD under certain conditions, noting that the EHD force could effectively strip the liquid layer from the tube wall, thus reducing the heat transfer. EHD boiling suppression was also observed by Ogata and Yabe (1993) and Kawahira et al. (1990).

In the previous works of Cotton et al. (2005, 2012) and Sadek et al. (2006) the tube and shell heat exchanger test sections used were limited by the fact that the working fluid flowed on the tube side and the tube needed to be electrically and thermally conductive, necessitating the use of opaque metallic sections to measure thermal hydraulic behavior under the influence of EHD. Visualization was often performed at a viewing window at the test section exit, which is not ideal since conditions are adiabatic and may not give details of flow regime development along the test section. With regard to the flow regime development, they were reconstructed based on the fluctuating wall temperature histories and were thus only hypothesized. In the recent investigation by McGranaghan and Robinson (2014a) a unique test section was used, which offered the possibility of utilizing a tube and shell fluid–fluid heat exchanger with EHD while still having visual access to the entire fluid under study within the tube section. In the study, the influence of EHD on flow boiling of HFE7000 was investigated for a fixed mass flux and inlet quality for a range of electric field strengths.

In a more recent study, McGranaghan and Robinson (2014b) used a similar test section, although without the water jacket as the heat source. Here, the indium tin oxide (ITO) coating was used as a resistive heating element and the ground for establishing the electric field. With thermocouples located at the top and bottom of a sapphire tube along its length, local heat transfer coefficient distributions were obtained for a range of applied electric field strengths for low inlet quality. For the predominantly stratified flow, the study showed that the influence EHD had on the local heat and mass transfer at the top of the tube was considerably different from that which occurred at the bottom. This indicates that the baseline flow regime, which strongly depends on the quality, is an important factor when considering the influence of EHD in two-phase flow scenarios.

This study aims to contribute to the knowledge regarding EHD augmented flow boiling by investigating the influence of inlet vapor quality on a tube and shell heat exchanger average heat transfer coefficient while also assessing
both electric and pressure power penalties. Because the heat exchanger is transparent, while being electrically and thermally conductive, the focus will be on understanding the observed heat transfer behavior, both with and without EHD, within the context of the observed two-phase flow regimes, which are recorded using high-speed videography.

2. EXPERIMENTAL METHODOLOGY

2.1 Experimental Apparatus

A schematic of the test facility is shown in Fig. 1 and consists of a closed loop filled with HFE7000 as the working fluid. A gear pump circulates the fluid through a flow control valve, a pre-heating section, the test section, and a condenser. The pre-heater section consists of two heaters, one (DIR1) is electrical and the other (HEX1) is a compact plate heat exchanger that receives hot water from the first secondary loop (SEC1). For a given working fluid flowrate, temperature control of these secondary heating loops determines the quality of the HFE7000 to the test section inlet via the expression

\[ x = \frac{\left( Q_{\text{DIR}} + Q_{\text{HEX1}} \right) / m_{\text{WF}} - c_{p\text{WF}} (T_{15} - T_{13})}{h_{fg}} \]  

(2)

The working fluid then passes via flexible piping to the test section, which is a horizontally mounted counter-flow shell and tube heat exchanger. The fluid passes through the tube side, while hot water circulated from the secondary heated water loop (SEC2) passes along the shell side, transferring heat to the working fluid and thus increasing its quality along the test section. After leaving the test section, the working fluid is condensed by another compact heat exchanger (HEX2) supplied by cold water from a chiller unit. Various ancillary valves on the loop allow for drainage, pressure control, and removal of non-condensable gas. A list of the experimental parameters is given in Table 1.

**FIG. 1:** Schematic of the flow loop (numbers indicate thermocouple locations).
TABLE 1: Summary of experimental conditions in the test section

<table>
<thead>
<tr>
<th>Parameter</th>
<th>HFE7000</th>
<th>Heating</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flux</td>
<td>100 kg/m²s</td>
<td></td>
</tr>
<tr>
<td>Heat flux</td>
<td>12.4 kW/m²</td>
<td></td>
</tr>
<tr>
<td>Pressure</td>
<td>1.0 bar (abs)</td>
<td></td>
</tr>
<tr>
<td>Inlet quality</td>
<td>2%, 15%, 30%, and 45%</td>
<td></td>
</tr>
<tr>
<td>High voltage</td>
<td>0, 4, 8 kV</td>
<td></td>
</tr>
<tr>
<td>High-voltage frequency</td>
<td>60 Hz</td>
<td></td>
</tr>
</tbody>
</table>

The shell and tube test section is novel in that it is optically transparent, allowing visualization of the working fluid flow regime under both EHD and non-EHD conditions. The inner tube is a 480-mm-long sapphire tube (outside diameter: 10 mm; inside diameter: 8 mm) supported by two machined polypropylene supports that connect it to the other loop pipework. A circular rod of 3-mm-diameter stainless steel runs concentrically inside the tube and acts as a high-voltage electrode. A transparent conductive layer of ITO was deposited on the outside of the sapphire tube in order to connect it to ground and establish the electric field. The transparent square cross-section (18 mm × 18 mm) Perspex shell section is located against machined faces on the polypropylene supports and then assembled around the sapphire tube, as depicted in Fig. 2. The front and rear Perspex sheets form a viewing window to observe the fluid flow regime. The Perspex jacket holds 10 T-type thermocouples, five along the top and five along the bottom at 120 mm spacing, each protruding into the SEC2 water stream.

Several T-type thermocouples (150 mm long by 1.5 mm in diameter) were used to measure temperatures (see Fig. 1); nine at various locations in the rig and 10 at the water side of the test section. The latter were spaced 120 mm apart, top and bottom, subdividing the test section into four regions designated as \( y/L = 1/8, 3/8, 5/8, \) and \( 7/8 \), respectively. At the center of each zone, top and bottom, tube wall temperatures were recorded by eight T-type thermocouples, 150 mm by 0.5 mm in diameter, inserted into 0.5-mm-deep pockets machined into the wall of the sapphire tube. All thermocouples were calibrated in a water calibration bath against a precision reference probe. Four flowmeters were used; one turbine type (0–4.5 L/min; Titan Instruments 945, USA) for the working fluid in the main loop, and three turbine types (Gems FT-110, 1–10 L/min, Gems, UK) were used in the secondary and chiller loops. A differential pressure gauge (Taylor DP504T, Taylor, USA) was used to measure the pressure drop across the test section.

High voltage to the electrode was supplied from a Matsusada (Japan) AMT series 0–10 kV high-voltage amplifier, while the primary sinusoidal signal was delivered from a Hewlett Packard 3325B (USA) function generator. The data acquisition system consisted of two National Instruments Compact DAQ 9172 8-slot chassis units with three types of modules used depending on measurement type. Two NI9219 4-channel modules (National Instruments, USA) were used for the embedded test section thermocouples set at the highest resolution of \( \pm 125 \) mV, giving a minimum

![FIG. 2: Drawing and cross section of the test section and photograph with insulation removed.](image-url)
accuracy of ±0.18% (typically better than 0.09°C) of temperature reading. Seven NI9211 4-channel thermocouple modules (National Instruments) were used for the remainder of the thermocouples. These units have an accuracy of 0.07°C over the entire range of T-type thermocouples. Two NI9215 4-channel modules (National Instruments) with an accuracy of 0.029 V over a range of ±10 V were used for the high level 0–10 V analog signals such as pressure, flowmeter output, etc.

In addition to the data logging information, a Hot Shot 1280 (NAC Image Technology, USA) black and white high-speed camera with its associated software was used for recording high-speed video and photographic stills of each experiment. Lighting was provided by four pulseless light-emitting diode cluster lamps (Fawoo LH-03, S. Korea) of 240 lm, two in the front and two in the back. Images were recorded at a frame rate of 2000 frames/s.

2.2 Data Reduction and Error Analysis

This study focuses on the overall thermal–hydraulic performance of the EHD augmented heat exchanger. Since the overall power supplied to the working fluid was kept constant, the main performance indicator is the average heat transfer coefficient $h$

$$h = \frac{Q''}{\Delta T}$$

(3)

The average heat flux is determined from the total heat supplied to the working fluid by the water on the shell side of the heat exchanger and the sapphire tube inner surface area

$$Q = Q''S = \dot{m}c_p(T_{w \text{ in}} - T_{w \text{ out}})$$

(4)

where $Q''$ is the average heat flux and $S$ is the inner surface area of the sapphire tube. The superheat is calculated using the arithmetic average of the test section wall temperatures and the saturation temperature

$$\Delta T_{\text{sup}} = \bar{T}_{\text{wall}} - T_{\text{sat}}$$

(5)

An uncertainty analysis based on the methodology of Kline and McClintock (1953) was carried out on all instrumentation. The expression

$$\Delta f = \sqrt{\sum_{i=1}^{n} \left(\frac{\partial f}{\partial x_i}\right)^2 \Delta x_i}$$

(6)

forms the basis of the uncertainty calculation, where $f$ is the uncertainty, and $x_i$ is the parameter on which $f$ depends. The basic formula for a two-instrument reading takes into account the accuracy of both thermocouples giving

$$\sigma_f = \sqrt{\sigma_x^2 + \sigma_y^2}$$

(7)

where $\sigma_f$ is the final uncertainty (percentage), and $\sigma_x$ and $\sigma_y$ represent the published error of both thermocouples (percentage). The final percentage uncertainties of the measured parameters are given in Table 2. The uncertainty of the heat transfer coefficient was better than 15% and was due primarily to the uncertainty of the heat transfer [Eq. (3)] due to the temperature drop across the water being of the order of 1–2°C with thermocouples calibrated to ±0.1°C.

### Table 2: Percentage uncertainties of the measured parameters

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Uncertainty (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Quality</td>
<td></td>
</tr>
<tr>
<td>$x_{\text{in}} = 0.02$</td>
<td>±17.4</td>
</tr>
<tr>
<td>$x_{\text{in}} = 0.15$</td>
<td>±10.2</td>
</tr>
<tr>
<td>$x_{\text{in}} = 0.3$</td>
<td>±7.2</td>
</tr>
<tr>
<td>$x_{\text{in}} = 0.45$</td>
<td>±6.4</td>
</tr>
<tr>
<td>Heat flux SEC1 water</td>
<td>±17.1</td>
</tr>
<tr>
<td>Heat flux SEC2 water</td>
<td>±12.2</td>
</tr>
<tr>
<td>Heat flux condenser water</td>
<td>±16.5</td>
</tr>
<tr>
<td>Heat transfer coefficient</td>
<td>±12.3</td>
</tr>
</tbody>
</table>

Interfacial Phenomena and Heat Transfer
The power penalties associated with the EHD augmented flows were the electrical power and the hydraulic pumping power calculated, respectively, as

$$P_{EHD} = IV$$  \hspace{1cm} (8)$$

and

$$P_{pumping} = \dot{V} \Delta P$$  \hspace{1cm} (9)$$

where a homogeneous density model was used to estimate the density in determining the volumetric flow rate. Thermal gradients in the sapphire tube were also considered using the Biot number ($Bi$)

$$Bi = \frac{hL}{k}$$  \hspace{1cm} (10)$$

where $h$ is the convective heat transfer coefficient on the inside of the tube; $L$ is the characteristic length, in this case the tube thickness; and $k$ is the tube thermal conductivity, taken as 40 W/m K. The Biot number for the tube in the radial direction was found to be 0.03, and since it is less than 0.1 the convective heat transfer dominates over conduction (Incropera, 1996). Given the low Biot number of the tube, for the purposes of this study the temperature across the tube wall in the radial direction was assumed to be uniform.

Axial heat transfer through the tube wall was also examined. The primary driver is $\Delta T$ in the lengthwise direction and Fourier’s law was used to estimate the heat transfer in the axial direction. This is given as

$$q = k \frac{dT}{dx}$$  \hspace{1cm} (11)$$

where $k$ is the thermal conductivity of sapphire; $dT$ is the difference in temperature; and $dx$ is the length of the tube. The maximum thermal gradient between the ends of the tube was $\sim 3^\circ C$, which resulted in negligibly small axial heat transfer compared with that being transferred across the tube wall to the refrigerant. Heat losses from the test section to the surrounding air were estimated at less than 5% of the heat input.

3. RESULTS AND DISCUSSION

3.1 Field Free or 0 kV Condition

The field-free (0 kV) flow case is first described in order to form a baseline to which the EHD cases can be compared. Taking into account the relative magnitude between inertial and EHD forces, where $\frac{E_{hd}}{Re^2} \sim 1$ and/or $\frac{M_d}{Re^2} \sim 1$, a mass flux of $G = 100 \text{ kg/m}^2\text{s}$ was selected to provide a flow pattern where the EHD should influence the flow patterns within the heat exchanger. A constant heat flux of 12.4 kW/m$^2$ was maintained across the test section irrespective of the heat transfer coefficient by adjusting the water inlet temperature, which resulted in a total heat transfer of 150 W.

The heat transfer and pressure drop behavior are given in Fig. 3(a), while Fig. 3(b) gives the standard deviation of the local wall superheat ($\sigma_{\Delta T}$). Standard deviation values lower than $\sim 0.1^\circ C$ are indicative of constant temperature over time, such as would be the case for a constantly wet or constantly dry surface (Cotton et al., 2012). The higher the $\sigma_{\Delta T}$ value, the higher will be the superheat fluctuations, which indicates partial wetting and is related to the heat transfer since it is indicative of the flow regime interaction with the heated wall. Finally, Fig. 4 presents an illustration of the flow regimes for different inlet qualities based on the high-speed video footage. Also, it should be mentioned that since the mass flux and total heat transfer of 150 W are fixed parameters, the quality increase from the inlet to the exit is also fixed, which was about $\Delta x \sim 30\%$ for each test performed.

For the lowest inlet quality tested, $x_{in} = 2\%$, observing the high-speed video revealed a flow regime that was predominantly stratified wavy with nucleate boiling in the stratified layer and intermittent liquid bridging events (Fig. 5, $V = 0$ kV). The liquid bridging periodically wets the top surface since, based on the quality and mass flux, the predominantly stratified-wavy flow regime is transitioning to slug flow. The constant temperature on the bottom is indicated in Fig. 3(b), where $\sigma_{\Delta T} < 0.1^\circ C$ at the inlet and exit. The intermittent wetting of the top surface is evident from $\sigma_{\Delta T} = 0.16^\circ C$ at the entrance, which reduces toward the exit due to the increased quality, and thus lower stratified liquid level. Figure 3(a) shows that the average heat transfer coefficient is $\sim 1000 \text{ W/m}^2\text{K}$. 

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With increasing inlet quality, and thus average quality within the test section, Fig. 3(a) shows that the heat transfer coefficient increases marginally up to \( x_{\text{in}} = 30\% \), after which it drops. The initial increase in the heat transfer coefficient appears to be due to the wetting characteristics of the top wall. As the quality increases the length and duration of the liquid bridging events tend to increase, and this is evidenced by the increase in \( \sigma \Delta T \) at both the entrance and exit. This improved wetting has the effect of increasing the heat transfer coefficient. However, at \( x_{\text{in}} = 45\% \) it is noted that \( \sigma \Delta T < 0.1^\circ\text{C} \), which indicates that the top surface is continually dry, thus causing the heat transfer to drop due to the diminishing heat transfer at the exit region. With regard to the pressure drop, Fig. 3(a) shows a continuously increasing pressure drop penalty with increasing test section quality, increasing about 3-fold over the range of inlet qualities tested.

### 3.2 \( V = 4 \text{ kV} \)

Figures 6 and 7 illustrate the influence of EHD on the thermal hydraulics of the heat exchanger, while Fig. 8 illustrates its influence on the flow regimes. For \( V = 4 \text{ kV} \) there is clear enhancement of the heat transfer. For \( x_{\text{in}} = 2\% \) the enhancement is 24% and this drops to about 12% at \( x_{\text{in}} = 45\% \). Figures 7(a) and 8(a) give some indication as to why this enhancement occurs.

For the \( V = 4 \text{ kV} \) and \( x_{\text{in}} = 2\%–15\% \) cases the electrostatic forces tend to keep nucleated bubbles within the stratified layer, causing them to slide, coalesce, and form larger bubbles that oscillate transverse to the flow [Fig. 5, \( V = 4 \text{ kV} \); Fig. 6(a)]. The sliding and oscillating bubbles cause better mixing within the stratified layer, which...
improves the overall heat transfer. Nearing the exit of the test section, liquid extraction is evident, which pulls a curtain of liquid toward the electrode. This has the effect of thinning the stratified layer, which also improves heat transfer. In this region bubble nucleation is no longer evident. With regard to the top surface there is clear indication of reduction in $\sigma_{\Delta T}$ compared with the 0 kV case, which is indicative of different wetting characteristics. The high-speed videos show that this is in fact the case. As discussed by McGranaghan and Robinson (2014a) and depicted in Fig. 5, this is due to bubble clusters being formed, which grow to such an extent as to wet the top surface. This is a different mechanism compared with the long duration slugs caused by flow regime transitions associated with the 0 kV case.

For the $V = 4$ kV and $x_{\text{in}} = 30\%–45\%$ cases the heat transfer coefficient tends to remain more or less the same as with the lower inlet qualities. This occurs even though the flow regimes tend to be quite different. As depicted in

**FIG. 4:** Reconstruction of flow regimes (illustrations not to scale).

**FIG. 5:** Images of flow regimes at location $y/L = 3/8$ for two inlet qualities and three voltage levels.
FIG. 6: (a) Heat transfer coefficient versus inlet quality; (b) pressure drop versus inlet quality for $V = 0, 4,$ and $8 \text{kV}$.

Figs. 6 and 7, the combined influence of the increased quality and applied voltage tends to redistribute the liquid more evenly around the periphery of the electrode and tube. Interestingly, for the $4 \text{kV}$ case the EHD influence does not have a profound effect on the pressure drop penalty as shown in Fig. 6(b).

3.3 $V = 8 \text{kV}$

For the $V = 8 \text{kV}$ and $x_{in} = 2\%$ test, Fig. 6(a) shows the highest level of heat transfer of all tests performed, with a heat transfer coefficient of $\sim 2000 \text{ W/m}^2\text{K}$. This is a 2-fold improvement in the heat transfer coefficient compared with the field-free case, which is substantial. Referring to Fig. 6(b) it is evident that this enhancement occurs with a very moderate ($\sim 30\%$) increase in the pressure drop, which highlights the capacity of EHD to augment heat transfer with small hydraulic penalties. Figure 7(b) shows that the bottom surface can be considered continually wet, which is confirmed by the high-speed video footage and is illustrated in Fig. 8(b). Figure 7(b) also indicates that $\sigma_{\Delta T} < 0.1^\circ\text{C}$ for the top surface and this is due to the fact that the EHD forces are strong enough to redistribute the liquid all around the tube surface. Although the mechanisms are different, the continual wetting of the entire tube is akin to an annular flow, and as a result the heat transfer coefficient is greatly enhanced compared with the stratified-wavy/slug flow associated with the $0 \text{kV}$ case. With regard to the flow regime, the EHD forces tend to draw liquid to the electrode and subsequently propel it outward to the upper and side regions of the tube in jet-like structures. As the test section quality is increased, the overall amount of liquid present in the system decreases to the extent that it is difficult for the EHD forces to extract liquid from the bottom layer. As a result, the wetting becomes increasingly more intermittent as is evident from the increase in $\sigma_{\Delta T}$ of the top surface for both the inlet and exit. The end result is a monotonic decrease in the heat transfer coefficient with vapor quality. For the highest quality tested the enhancement drops to $30\%$ with a similar percentage increase in the pressure drop.
FIG. 7: Superheat standard deviation versus local inlet and outlet quality for (a) $V = 4 \text{kV}$ and (b) $V = 8 \text{kV}$.

FIG. 8: Reconstruction of flow regimes for (a) $V = 4 \text{kV}$ and (b) $V = 8 \text{kV}$. 
An overall view of the test results is shown in Fig. 8. For the field-free case (0 kV) the general increase in the heat transfer coefficient with inlet quality up to \( x_{in} = 30\% \) and subsequent decrease is purely dependent on conventional inertial liquid/vapor forces within the tube. The addition of 4 kV is seen to augment this basic pattern, where the 4 kV force initially assists heat transfer by acting on the bubbles at lower quality. This augmentation process then changes as the EHD forces initiate a radical departure in the stratified layer flow regime to the attracted curtain flow pattern and to the wavy thin film–type flows. As the quality increases, the 4 kV EHD force precipitates a flow regime change to an induced annular type of flow, which is responsible for a further increase in heat transfer. At a quality of 45\%, a drop in heat transfer is noticed since the amount of liquid available for redistribution is less, and dry out along the upper tube becomes more prevalent.

At 8 kV EHD applied voltage, the flow regime at low quality experiences large changes in bubble activity, especially growth phenomena, oscillations, and wiping, as well as coalescence and climbing, causing wetting of the tube top, which all contribute to deliver high heat transfer coefficients at the tube top and bottom. Liquid jets are also a feature of these high EHD voltages and contribute to the redistribution of the fluid phases. However, as the inlet quality increases the amount of liquid available is seen to decrease, and nucleation also decreases, but the high EHD forces are effective in assisting redistribution of liquid from the bottom layer to wet the top of the tube, again creating an induced annular flow. As the quality increases further, this redistribution effect reduces in effectiveness because of the increased dry conditions at the tube top.

As the inlet quality increases, the pressure drop increases, leading to a higher pumping power penalty. This is due to the significant difference in velocity between the liquid and the generated vapor, resulting in higher shear and increased frictional drag within the test section (Müller-Steinhagen and Heck, 1986; Reeser et al., 2014). From the four graphs given in Figs. 9(a) and 9(b) and Figs. 10(a) and 10(b), a gradual rise in pumping losses with increasing

**FIG. 9**: EHD losses and pumping power losses against voltage for (a) \( x_{in} = 2\% \) and (b) \( x_{in} = 15\% \).
vapor quality and also with increasing EHD voltage is clearly seen. However, while pumping losses increase with inlet quality, the EHD power requirements decrease. The same information is portrayed more clearly in Fig. 11(a), where EHD power levels decrease with inlet quality, while in Fig. 11(b) pumping losses gradually increase. The decreasing of the required EHD power as the inlet quality increases bears a similar likeness to Fig. 6(a), where the EHD-induced improvements in the heat transfer coefficient also begin to decrease as the inlet quality increases. It is postulated that the reason for this is the lower amount of liquid within the tube for the EHD forces to act upon, thus affecting not only the heat transfer but also the pressure drop, which explains the similarity in the trends. EHD power demands and associated pressure drop losses are low in all cases—in the present study accounting for at most around 2.1% (3.2 W) of the thermal power transport (150 W).

4. CONCLUSIONS

The influence of electrostatic forces on the flow and heat transfer of flow boiling HFE7000 has been investigated. In particular, the influence of the EHD for relatively low mass flux (100 kg/m²s) and heat flux (12 kW/m²) flow boiling has been studied. The results indicate that the enhancement of the heat transfer depends on the applied voltage level and the vapor quality. For $V = 4$ kV the overall flow regime was quite different from the baseline $V = 0$ kV and the heat transfer was enhanced by between 12% and 24%, with the enhancement generally decreasing with vapor quality. For the $V = 8$ kV case the enhancement levels reached 100% for the lowest quality tested and decreased with increased quality to 30% for the highest inlet quality. For the 4 kV case the pressure drop penalty was not severe, while for the 8 kV case the pressure drop penalty was more substantial, ranging from 30% to 60% over the $V = 0$ kV case. However, the power penalty was small overall, at around 2% of the transported thermal power.
FIG. 11: Effect of voltage on (a) EHD losses and (b) pumping power losses against inlet quality.

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