Heat transfer model for gas–liquid slug flows under constant flux

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ABSTRACT

This paper investigates the mechanisms leading to enhanced heat and/or mass transfer rates in two-phase non-boiling slug flows. The problem is analyzed in a minichannel geometry subjected to a constant heat flux boundary. Local Nusselt numbers, obtained using Infrared thermography are analyzed in both entrance and fully developed flow regions. These novel measurements highlight the physics governing slug-flow heat transfer and results indicate that optimized slug geometries can yield up to an order of magnitude heat transfer enhancement. Finally, based on the physics identified, a heat transfer model is developed which is also applicable to similar mass transfer problems.

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1. Introduction

In recent years there has been significant drive in almost all industries to reduce the size of heat and mass exchange devices. In heat transfer applications, this has been primarily driven by advances in microprocessor performance, as the computing industry is currently curtailed by increased heat flux levels due to reduced die size and increased numbers of transistors. This problem of elevated heat flux has led numerous researchers to investigate many novel air cooled heat sink designs [1–4] and to examine the efficiency of blade geometry in the miniature fans required [5,6]. Also, the advent of microchannel technology has lead to the development of numerous liquid cooling devices, such as cold plates for processor cooling, and compact heat exchangers for industrial applications. These liquid cooling devices yield significantly enhanced heat transfer coefficients over their air based counterparts and hence, allow for further advances in processor performance and the miniaturization of industrial processes. Various designs of these devices were suggested and prototyped by Mudawar [7] including: spray cooling [8]; jet cooling [9]; various designs of cold plates [10] and microchannels [11] to name but a few.

The latter of these in particular, investigating microchannel heat exchangers, has almost become a research discipline in itself. Numerous designs have been suggested to increase heat transfer rates, ranging from different channel layouts to boiling induced two-phase flows. The concept of microchannel cooling is simply based on increasing the convective surface area to coolant volume ratio. However, despite some early suggestions that an increase in Nusselt number above the classical macroscale level may be achieved in microchannels, it has since being proven that heat transfer at the microscale, ~250 μm or greater, collapses well with the long established analytical solutions applicable to macroscale systems. Also, laminar flows dominate at the microscale and the pressure required to pump fluids significantly increases as channel scales reduce. Hence, existing pumping technologies cannot provide the required pressure drop and flow rate to achieve enhanced heat transfer through turbulent flow in practical environments. The Graetz solution [12,13] in the developed region for laminar flow shows that no gains in fully developed flow Nusselt number are achievable for increases in flow rate, although the capacity to remove more heat, in line with the energy equation, is possible. Hence, much recent microchannel research has been focused on using physical structures to induce local turbulent spots or the use of two-phase flows, both of which enhance the Nusselt number beyond its laminar level. The current study focuses on the latter of these.

Many different two-phase flow regimes have been studied from both pressure drop and heat transfer perspectives. The most commonly studied two-phase flows are typically encountered in boiling heat transfer where the coolant changes phase within the heat exchange device. This mechanism of heat transfer provides high local heat transfer coefficients but the process is complex, difficult to accurately control, suffers from dry out issues and requires high pressure sealed vessels for containment. Comprehensive reviews on such technologies have been reported by Thome [14] and Kandlikar [15]. Alternatively, two-phase flows without boiling have rarely been investigated for use in microscale heat exchange devices. Instead, the bulk of studies analyzing non-boiling two-phase flows are focused on large scale systems such as that studied...
by Kim and Ghajar [16], Hetsroni and Rozenbilt [17]. The flow regimes encountered in these large scale systems were characterized by bubbly, slug, mist, annular, wavy or stratified flows. Additionally, many authors have formulated charts in an attempt to identify which type of flow regime is most likely under prescribed gas and liquid flow rates [16–18]. Throughout these studies there is little consideration given to the local flow field. For example, the length of liquid slugs is not typically measured within the slug flow regime even though it has been noted to be of importance [19–22]. This is especially significant in microscale heat or mass transfer devices where capillary forces dominate and the slug flow regime, as it is depicted in Fig. 1, is most often observed. Importantly, the definition of slug flow in microscale systems differs somewhat from that in macroscale systems. In microscale, liquid slugs and gas bubbles occupy the entire channel cross-section whereas in macroscale they may only occupy a percentage of this due to buoyancy mismatches between the gas and liquid phases.

The focus of the current study is on this non-boiling two-phase slug flow regime. Such flows are generated by segmenting a continuous liquid stream with an injected gaseous phase to create well-ordered trains of segmented liquid slugs and gas bubbles. As seen from Fig. 1, a large number of different configurations are possible for the same gas and liquid flow rates depending on the ratio of slug length to channel diameter ($L_s/D$). In gas–liquid slug flows, the gaseous phase has a negligible contribution to heat transfer since, at the same volumetric flow rate, its thermal capacity is <0.1% of the liquid phase when water and air are used. Also of interest for comparison with this flow regime, is the theoretical Graetz solution for single phase flow in the limit of Prandtl number approaching zero. This theoretical result is referred to as the ‘Plug Flow’ limit and has been shown by Bejan through analytical considerations [23], to be capable of enhancing the Nusselt number in the developed region from 4.36 for typical single phase convection to 8 for plug flow under an isoflux boundary. In reality plug flows can only be realized by the use of an inviscid fluid or one with an infinite thermal conductivity. The slug flow depicted in Fig. 1 on the other hand, is more realistic in microscale devices where typical coolants are used since viscous forces dominate at low Reynolds numbers. Two distinctly different mechanisms when compared to single phase convection could account for increased heat transfer rates within this flow regime. These were noted by Oliver and Wright [24] and include an increased velocity of the liquid phase due to the addition of a segmenting gaseous phase and an internal circulation within liquid slugs, as was recently measured by King et al. [25] using Particle Image Velocimetry in microchannels. Muzychka et al [26] have shown analytically that flow segmentation, resulting in increased fluid velocity alone, cannot account for increased heat transfer rates over single phase or long solid plug flows. Hence the physical mechanism driving enhanced heat transfer must either result from the internal circulation, or a modified velocity profile within liquid slugs. Fig. 2 illustrates the circulation within each slug and the difference in slug shapes that result from passing the fluid through channels with (a) hydrophobic and (b) hydrophilic surfaces.

Although most studies do not control the slug length, this has been recognized as a key factor in characterizing the enhanced heat transfer rates offered by segmented flows. A small number of authors have experimentally [19–22] and more recently numerically [27–29] investigated the influence of slug length on heat and mass transfer characteristics. Table 1 summarizes the experimental variables of these studies and an in-depth analysis of their results was recently published by the authors [26]. The focus of the experimental studies listed in Table 1 was on bulk tube heat transfer with an isothermal wall boundary condition, although the numerical studies do show local heat transfer rates. To date, only one experimental study has been undertaken to examine local heat transfer with an isoflux boundary condition but this was limited in scope [22]. Hence, a key question is: what Nusselt number enhancement, if any, can be achieved in segmented flows and how does this vary with changes in slug length, void fraction and Reynolds number?
The current paper addresses this deficit in the literature by answering these open questions. An experimental facility is built to examine the problem in tube flow with a constant wall heat flux. A controlled slug flow regime is produced using water as the liquid phase and air as the segregating gaseous phase. Dimensionless heat transfer results in terms of Nusselt numbers, as defined by the constant wall heat flux. Dimensionless heat transfer rates for such developed regions of internal channel flows. The specific problem under investigation analyzes thermal boundary layer enhancement over single phase flow while short slugs resulted in developed regions. It was found that long slugs showed little enhancement over single phase flow while short slugs resulted in almost an order of magnitude enhancement.

2. Analytical models

This section highlights the relevant analytical expressions used for characterizing single phase heat transfer in both the entrance and fully developed regions of internal channel flows. The specific problem under investigation analyzes thermal boundary layer development in a hydrodynamically developed laminar flow with constant wall heat flux. Dimensionless heat transfer rates for such flows are characterized by local Nusselt numbers, as defined by the following equation:

$$N_{ux} = \frac{hD}{k} = \frac{q'D}{k(T_w - T_m)}$$

(1)

In addition, the dimensionless position downstream of the heated section entrance is characterized by the inverse Graetz number. This parameter is referred to as $x^*$ throughout and is defined by the following equation:

$$x^* = \frac{x}{U_D D} = \frac{x}{DRePr} = \frac{x}{DPe}$$

(2)

For fully developed laminar tube flow, the velocity profile at any cross-section is known; hence the energy conservation equation can be solved to provide an exact solution to the temperature field. Such a solution was first reported by Graetz [12,13] and allowed for the later development of simplified piecewise analytical expressions defining the variation of local Nusselt number with dimensionless position [30]. These solutions are separated into two regions where an expression is defined for the entrance region while the Nusselt number in fully developed flow is constant. Muzychka and Yovanovich [31] combined such expressions using the addition of asymptotic limits approach and arrived at the expression of Eq. (3). This defines the variation of local Nusselt number with dimensionless position where a constant wall heat flux boundary condition is applied and the fluid Prandtl number, $Pr$, is greater than unity.

$$N_{ux_{plug}} = \left[ \frac{(1.302)}{x^{0.886}} + (4.36) \right]^{1/5}$$

(3)

Also of interest to the current study is the theoretical solution obtained in the limit of $Pr$ approaching zero. This is known as the plug flow limit and can be thought of as either an inviscid fluid stream or a solid rod passing through a channel. Hence, fluid flows as a solid plug with a uniform velocity profile across the channel cross-section. An analytical expression describing this solution, again in terms of asymptotic limits was defined by Muzychka et al. [32], Eq. (4)

$$N_{ux_{plug}} = \left[ \frac{(0.886)}{x^{0.886}} + (7.96) \right]^{1/2}$$

(4)

In order to evaluate local heat transfer rates in liquid–gas slug flow systems, it is also necessary to introduce a ‘void fraction’ parameter. This is required since the liquid phase is the primary contributor toward heat transfer but is only in contact with a percentage of the total heat transfer surface, since voids or gas bubbles are present. Hence, heat transfer rates should be normalized using this parameter which defines the percentage of the heat transfer channel volume filled with gas at any operating condition. This can be calculated using the expression of Eq. (5), where $Q_g$ and $Q_l$ are the gas and liquid input volumetric flow rates, respectively.

$$\varepsilon = \frac{Q_g}{Q_g + Q_l}$$

(5)

When analyzing results in subsequent sections, the single phase heat transfer prediction of Eq. (3) will be used to validate the

Table 1

<table>
<thead>
<tr>
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<tr>
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<td>1.6−20</td>
<td>10.5−61</td>
<td>1.92−6.1</td>
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<td>260</td>
<td>128</td>
<td>40</td>
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<td>3–220</td>
<td>0.7–15.8</td>
<td>1396–2135</td>
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<tr>
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<td>1700</td>
<td>1000, 10,000</td>
<td>5.5</td>
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<td>$Pe$</td>
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<td>51,000−374,000</td>
<td>4210−17,800</td>
<td>7678–11,472</td>
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<td>3.85–31.2</td>
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<tr>
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<td>0.00025–0.00052</td>
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<td>0.5</td>
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<td>0.205–0.480</td>
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experimental set-up by comparing with results from benchmark single phase convection tests. Also, the models of Eqs. (3) and (4) along with the void fraction of Eq. (5) will be used to gauge the magnitude of enhancement offered by liquid–gas slug flows, and in developing a model for the heat transfer results obtained.

3. Experimental facility and data processing

The experimental facility used during this study is shown schematic in Fig. 3. This was designed and constructed to: (1) enable a continuous stream of slugs to be produced, whose length could be accurately controlled and measured; (2) provide a heated test section which subjects the internal flow to a constant wall heat flux boundary condition; (3) obtain high resolution local measurements of the heated section surface temperature and (4) develop an analysis tool to interpret temperature measurements and average these over the recording period. Each of these aspects of the facility will be discussed throughout this section along with the methodology employed for extracting results.

Two high precision Harvard PHD 2000 Programmable syringe pumps were used to set the gas and liquid volumetric flow rates delivered from 100 ml capacity Hamilton glass syringes (Model 11000TLL). Glass syringes were used as these were found to provide steadier flow than was attainable using equivalent plastic syringes. The flow from these entered transparent tubing and was combined in T-junctions where well-ordered segmented slug trains were generated. The liquid slugs and gas bubbles were visualized, both prior to entering and downstream of, the heated section through transparent tubing to ensure that consistent slug trains were maintained and that no coalescence of slugs occurred. Images such as that shown in Fig. 4 were recorded at the exit to the heated section and analyzed using Matlab (Version R2008a) to obtain an accurate measure of liquid slug lengths. This resulted in obtaining an average and standard deviation for the slug length from a series of 10 images recorded simultaneously with tube surface temperatures for each test conducted. A number of different internal bore T-junctions were used to generate different length slugs for fixed liquid and gas flow rates. In all cases water was used as the liquid phase and air as the segmenting gas phase. Finally, it was also noted that there was no noticeable change in this hydrophilic condition, or in heat transfer characteristics, during the testing period of 3 months. This suggests that results are independent of surface contamination by atmospheric air.

External surface temperature measurements of the heated section were obtained using a high resolution Infrared (IR) thermography system. This consisted of a FLIR Systems ThermaCam Merlin series IR camera set-up. Such measurements required the exterior surface of the stainless steel test section to be sprayed matt black to enhance its surface emissivity. The actual emissivity of this surface was evaluated prior to testing by simultaneously comparing IR thermography measurements with equivalent measurements from four K-type thermocouples. These were calibrated to ±0.1 K and mounted on the tube surface. Calibration was achieved by pumping single phase liquid at a minimum of three known temperatures through the tube at a sufficiently high flow rate to ensure that the temperature drop between thermocouple probes, due to natural convection losses, was less than 0.1 K. This procedure resulted in determining that the test section had a surface emissivity of 0.95, which is typical of a matt black surface. Additionally, during testing both the IR camera and thermocouples were used to measure the surface temperature of the tube, hence allowing for a continuous calibration check of the thermal imaging set-up in-situ.

In attaining local temperature measurements throughout experiments, the IR camera was placed at a fixed position with its field-of-view extending over approximately 100 mm length of the tube, as is shown in Fig. 5(a). Hence, all local temperature measurements were obtained within this region. Also, in order to avoid thermal reflections from surrounding bodies, this section of the apparatus was further enclosed within a matt black box. Recording of thermal images began immediately prior to supplying electrical current to the test section for heating. Images were recorded for up
to 15 min and at a frequency of 0.2–1 Hz depending on the flow rate. Such extended recording times were necessary to let the heat transfer process reach a quasi steady-state condition where the wall temperature clearly oscillates periodically about a constant mean value. This periodic oscillation is due to the inherently unsteady heat transfer process associated with the periodic passing of liquid slugs and gas bubbles. In general, this quasi steady-state condition was achieved in approximately 1 min. Also noted is that the IR camera recording frequency was typically lower than the slug passing frequency, \( F \), see Table 2. Hence the exact oscillatory nature of the flow could not be captured. However, when recordings were made over long time periods and averaged, the system was capable of accurately capturing average heat transfer rates.

After thermal images were recorded, it was necessary to develop an analysis code in Matlab (Version R2008a) for extracting the transient tube surface temperature profile and averaging this over time. Fig. 5 illustrates the procedure following by this code. Firstly, a single image without heating, and incorporating a ruler, is used for scaling, image (a). In image (b), a region of interest is highlighted as shown by the dashed lines. The code uses an edge detection algorithm to accurately identify the heated tube edges and subsequently identify its centerline. Surface temperatures are then extracted along this line at each pixel location and resulted in one temperature measurement every 300 \( \mu \)m along the tube length. This latter step was looped for each IR recording over the test duration and resulted in obtaining the temperature contour map shown in (c). The abscissa of this contour plot represents distance along the tube from the inlet and the ordinate represents each image number in recorded sequence, i.e. representative of the time domain and related through the recording frequency. The color contour map represents the tube surface temperature in degree Celsius with magnitude indicated by the legend shown. The user then inputs a range of image numbers between which the tube surface temperature is considered to be in the quasi steady-state condition discussed in the previous section. An average temperature is then calculated for each axial location. The output of this averaging technique is shown in Fig. 5(d) which reports the mean surface temperature profile along the tube length. Also shown are curves marked ‘upper’ and ‘lower’, these represent three standard deviations of the periodically fluctuating temperature along the length of the tube. Importantly, this fluctuation does not represent an error, but rather is reflective of the unsteady nature of the heat transfer process when using segmented flows. Finally, the entire system was also calibrated using single phase flow theory, where good agreement was found for five different flow rates, ranging over three orders of magnitude of inverse Graetz number, \( x^* \). Results from these calibration experiments along with those obtained from 40 individual slug flow set-ups are discussed in the following section. Table 2 presents a brief summation of the range of experimental parameters examined throughout the study.

Before presenting results, an estimate of the error associated with the experimental measurements is made using the method of Kline and McClintock [33], which is based on the expression outlined in Eq. (6)

\[
W_E = \sqrt{\frac{dE_{y_1}}{dy_1}w_1^2 + \frac{dE_{y_2}}{dy_2}w_2^2 + \cdots + \frac{dE_{y_n}}{dy_n}w_n^2}
\]  

Here, ‘W’ represents the error associated with any calculated parameter ‘E’, while ‘w’ represents the error associated with any measured variable ‘y’. The errors associated with most of the
measured parameters were found to be dependent upon the parameters’ magnitude. However, maximum errors were of order ±50 µm for tube diameter, ±100 µm for all other length scales, ±0.5% for prescribed volumetric flow rates, ±0.5% for supplied heat flux, ±1% for tabulated fluid properties and ±0.1 K for temperature measurements. From these intrinsic errors, the total measurement error for Nu, of Eq. (1), and x’ of Eq. (2), were calculated by substitution into Eq. (6). This resulted in experimental errors which varied between 8.5% and 6.8% over the range of Nu considered and 4.7% to 1.1% over the range of x’ values considered. Such errors were at a maximum in the early entrance region and quickly diminished towards their lower values as the flow developed. Uncertainty calculations also showed that errors in void fraction measurements through Eq. (5) were negligible due to the precision of the pumps employed. However, when measuring void fraction based on inlet volumetric flow rate conditions, errors can result due to an expansion of the gas phase between the location of the slug producing mechanism and the heated test section. This can result from variations in both the pressure and temperature of the gas. An estimate of the maximum pressure drop between these locations was made from single phase correlations at the highest flow rate considered and corresponded to ~1200 Pa, while the maximum temperature rise in the test section was 15 ºC. Such small variations can be shown through the ideal gas law to result in maximum void fraction errors of only 4% for the range of parameters considered. It will be seen in the following section that these uncertainty levels are far less than the enhancements observed during experimentation. Hence, all error bars are omitted from the graphs presented so as to enhance their clarity.

4. Results and discussion

This section details the heat transfer results obtained when analyzing the effect of segmenting a continuous Graetz type flow into discrete liquid slugs separated by gas bubbles and heated by a constant wall flux boundary condition. The discussion of results begins by first showing the augmentation of heat transfer rates attainable in a dimensioned format and then generalizing these results by presenting them in dimensionless format. The physics of the flow are detailed throughout and correlations based upon the described flow physics, and the analytical solutions presented earlier, are determined. These are put forward to enable accurate predictions of heat transfer in the entrance and fully developed flow regions of two-phase non-boiling slug flows. Importantly, these can also be applied to similar mass transfer problems as the same physics govern this process also.

The first set of results presented in Fig. 6 show a dimensioned plot of the time averaged wall temperature rise versus distance from the entrance of the heated section. The theoretical bulk fluid and tube wall temperatures based upon an enthalpy balance and the continuous Graetz flow solution of Eq. (3) are plotted for reference. In addition to these, three experimental results for segmented slug flows are also presented where the slug length to diameter ratio is varied from 1.6 to 14.3. The plot highlights the effect that changing slug length has on the resulting time averaged wall temperature for constant input boundary conditions of wall heat flux; inlet flow rates of air and water; and hence void fraction. The most striking observation is the obvious change in the wall temperature profile within the entrance region. It is seen that the ratio of slug length to tube diameter, L/D, has a profound effect on this profile and also on the fully developed temperature difference between the tube wall and the bulk fluid. The general trends observed are that: shorter slugs provide augmented heat transfer rates throughout the test section; moderate length slugs can result in a degradation of heat transfer rates within the early entrance region but augmented in fully developed flow; and long slugs can result in a degradation of heat transfer rates throughout the entire system. It is noted that such significant deviations in heat transfer rates within the entrance region were not predicted by the numerical studies analyzing a constant heat flux boundary [28,29]. Also, none of the experimental studies analyzing isothermal wall boundaries were able to obtain local heat transfer measurements such as those obtained herein [19–21].

In order to explain the trends in wall temperature observed, the result obtained for L/D = 5.7 or slug length of 8.6 mm will be focused upon as this result clearly highlights both entrance and fully developed flow phenomena. In the early entrance region, the tube wall temperature increases above that of continuous flow, then at a downstream distance of approximately 9 mm or one slug length, the wall temperature flat lines until approximately 22 mm. This phenomenon is due to the internal circulation within moving slugs that simply results from a combination of viscous dissipation and mass conservation. This causes fluid from tube centerline to be forced towards the tube wall at the slug leading edge and visa versa at the trailing edge. Given that in laminar flow a parabolic velocity profile should develop in the center of such slugs, one circulation length can be approximated by twice the slug length plus a single tube diameter. This is the distance a slug would move downstream before an element of fluid would complete one circulation around the slug and is based upon (1) the no-slip condition at the wall and (2) a typical centerline velocity for parabolic tube flow equal to twice the mean slug velocity. Hence, the thermal boundary layer at the slug leading edge will continuously be renewed with fluid at the inlet temperature until it has reached a downstream distance of one full circulation length, provided of course that Pr > 0. The result is that the wall temperature increases to a value where the supplied heat flux can be dissipated and holds constant until heated fluid from the early entrance region makes its way to the slug leading edge. In Fig. 6, the circulation length corresponds to a distance of approximately 20 mm which correlates well with the position where the wall temperature begins to rise again. Following this, a similar phenomenon is seen to occur downstream but its amplitude is damped significantly after each cycle until the flow fully develops and the wall temperature begins to rise linearly with distance.

The most sensible way of generalizing such results is to present them in dimensionless form as shown in Fig. 7 where the local Nusselt number, Eq. (1), is plotted against dimensionless position or inverse Graetz parameter, Eq. (2). In this plot, the dimensionless
position is calculated based on the inlet liquid Reynolds number only and hence does not account for the increased fluid velocity due to segmentation. This allows a direct comparison of heat transfer rates achievable when using a constant mass flux of water. The first point to note here is the excellent agreement found between the Graetz solution for single phase flow and experimental values since all local Nu measurements fall within 4% of this solution. With regard to slug flows, the plot clearly shows that these can degrade heat transfer performance within the entrance region but will typically augment heat transfer in fully developed flows, a finding which is of significant importance in applications where the heat exchanger length is of order the thermal development length. In addition, the aforementioned internal circulation effect is clearly evident as the local Nu is seen to undergo a decaying oscillation about its fully developed value.

The first peak of this oscillation corresponds to a downstream distance from the thermal entrance of just over one circulation length which is also equal to the period of oscillation. Furthermore, it is seen that the amplitude of the oscillation varies inversely with slug length, a phenomenon which is believed to be due to smaller slugs having lower thermal mass and so reaching fully developed flow much quicker. It is also worth noting, that in general, the thermal entrance length for slug flows is much shorter than for a continuous flow. In fact, it is found that the only parameter which affects the entrance length is the liquid slug length since the local Nu reaches its steady state value after only traveling one slug length and oscillates about this value thereafter. Finally, it is also clearly evident that the slug length has a significant effect on fully developed Nu as short slugs can greatly enhance this asymptotic value over that typical of a single phase flow.

Up to this point, the experimental data has been presented for direct comparison with a similar continuous flow system but in order to gain a comprehensive understanding of the flow physics governing segmented flow heat transfer, the data must be reduced appropriately. To achieve this, both Re and Nu are normalized by the percentage contact area of the active cooling fluid, i.e. liquid slugs, since the thermal capacity of the gaseous phase is insignificant. This percentage contact area is defined as one minus the void fraction, \((1 - \varepsilon)\) see Eq. (5) for its calculation.

The experimental data is plotted in this format in Fig. 8 where the physics of the flow can now be accurately compared with solutions for unsegmented Poiseuille and plug flow. Reporting the data in this format highlights some very interesting findings regarding the flow physics. The first of these relates to the thermal development region where it becomes clear that their heat transfer characteristics approach that of a continuous Poiseuille type flow for long slugs whereas it approaches the theoretical plug flow limit for short slugs. Furthermore, over the range of experimental testing conducted, it was confirmed that slugs of length equal to one tube diameter fall upon the plug flow limit and indeed this limit can be exceeded with slugs of even shorter lengths, see Figs. 9 and 10. This finding is in contrast with the numerical studies analyzing local Nu within the entrance region [28,29], as these showed that slug flow characteristics in this region always fall on the plug flow limit regardless of slug length. This experimental finding can be explained by noting that the velocity profile in long slugs is identical to that of a continuous Poiseuille flow for the majority of their length, whereas in short slugs, flatter velocity profiles at the slug leading and trailing ends dominate the heat transfer process. Furthermore, the observation of increased Nu above the plug flow limit is hypothesized to be plausible due to an impinging type flow phenomenon existing at the slug leading edge which results from the internal circulation. This impingement point occurs at the leading edge of the slug where it contacts with the wall. Also noted is that the angle of impinging flow will be influenced by the wall surface properties, i.e. hydrophobic or hydrophilic, as was illustrated in Fig. 2.

With regard to fully developed Nusselt numbers, \(Nu_{Dev}\), it is clearly seen that shorter plugs provide far higher values than

![Fig. 7. Dimensionless plot showing measured and predicted local Nusselt numbers versus inverse Graetz number, \(x^*\), for measured single phase, plug and slug flows. Re and \(\varepsilon\) held constant for slug flow tests, equal to 112.7 and 0.33, respectively. Re range for single phase flow was 56–1127.](image)

![Fig. 8. Dimensionless plot showing effect of slug length after normalizing local Nu and Re with the liquid wetting fraction \((1 - \varepsilon)\), Re and \(\varepsilon\) held constant at 112.7 and 0.33, respectively.](image)

![Fig. 9. Plot highlighting the effect of slug length to diameter ratio on fully developed Nu magnitude.](image)
longer ones. The maximum augmentation shown in Fig. 8 is a 600% enhancement over the fully developed Poiseuille flow limit or a 300% enhancement over the theoretical limit of fully developed plug flow. However, for longer slugs it is seen that \( N_{\text{UDev}} \) reduces significantly and appears to be approaching the Poiseuille flow limit. This enhancement in \( N_{\text{UDev}} \) results from the aforementioned internal circulation, as it provides constant mixing of the slug volume. Hence, the shorter the slug, the more isothermal its temperature will be. It is believed that the thermal mass of the slug is the dominating factor in this regard.

Now that the physics of the slug flows under analysis have been explained, the development of a correlation to accurately predict local \( Nu \) will be focused upon. This is achieved by deriving separate expressions for the developing and fully developed asymptotic limits and later combining these through the blending approach of Churchill and Usagi [34]. Firstly, results within the thermal entrance region from the wide range of experimental investigations conducted showed that the local \( Nu \) varies between the Poiseuille and plug flow limits as slug length varies. This behavior was found to be best correlated using a weighted mean method between these two limits where a slug of length equal to one diameter falls on the plug flow limit, while an infinite length slug falls on the Poiseuille limit. The local \( Nu \) then varies linearly between these limits for slugs of intermediate lengths. Hence, a correlation defining local \( Nu \) within the entrance region is defined in the following equation:

\[
Nu_{\text{U;}(\text{Ent})} = Nu_{\text{U}(\text{Pos;}\text{Ent})} + D/L_s \left( Nu_{\text{U}(\text{Pos;}\text{Eng})} - Nu_{\text{U}(\text{Pos;}\text{Ent})} \right)
\]  

(7)

With respect to the fully developed flow limit, slug length again has a significant effect on Nusselt number, \( N_{\text{UDev}} \). In order to obtain an accurate representation of how \( N_{\text{UDev}} \) varies with liquid slug length, an extensive range of tests were conducted with control parameters ranging between the limits defined in Table 2. These tests investigated extensive ranges in Reynolds number, void fraction and liquid slug length. Overall, the flow physics observed throughout these were identical to those already described through Figs. 6–8. In all, 40 slug flow tests were conducted and examined to determine the effect of slug length on \( N_{\text{UDev}} \). In order to determine the correct relationship, it was first noted that in the limit of an infinite length slug, \( N_{\text{UDev}} \) should approach that of a Poiseuille flow. Hence, the difference between the Poiseuille flow limit and the measured slug flow values are considered for correlating, and are presented in Fig. 9. This plot clearly highlights the effect of liquid slug aspect ratio on the increase in \( N_{\text{UDev}} \) over the Poiseuille flow limit, where the power law trend identified accurately represents the experimental data. Hence, the fully developed asymptotic limit is given by Eq. (8) where \( N_{\text{UDev}} \) enhancements, over the Poiseuille flow limit, are observed to be inversely proportional to the square root of slug length to diameter ratio. Also worth noting from Fig. 9, is that there is little scatter in the experimental findings from the expression of Eq. (8) when \( L_s/D \) is below ~3. Above this however, some scatter is observed. It is believed that this scatter is primarily due to errors resulting from the method used in determining \( N_{\text{UDev}} \) when the flow had not reached its fully developed limit within the experimental test section. An example of this can be seen from Fig. 8 when \( L_s/D \) was equal to 14.7. For such cases, the value of \( N_{\text{UDev}} \) was estimated to equal the mean of the peaks and troughs observed

\[
Nu_{\text{U}(\text{Dev})} = Nu_{\text{U}(\text{Pos;}\text{Dev})} + 25(L_s/D)^{1/2}
\]  

(8)

These limits are now combined in Eq. (9) with a blending parameter, \( n \), equal to 10 used to ensure an abrupt transition from entrance to fully developed flow. The accuracy of this correlation is seen in Fig. 10 by comparing predictions to a number of the experimental measurements obtained. It is seen that very good agreement was found in both the thermal entrance and the fully developed flow regions. However, for longer slugs where \( Nu \) oscillated within the transition region, the correlation was less accurate in this region. Hence, the proposed correlation can be employed with a high level of accuracy for values of \( x/L_s < 1 \) and \( x/L_s > 5 \). However, within the range \( 1 < x/L_s < 5 \) the proposed correlation results in up a ±20% deviation based on the experimental data. Of course, this correlation does require knowledge of the liquid slug length. In practical applications this would have to be either directly measured within the system, or alternatively it could be predicted from recently published [35] relationships between pressure drop and slug aspect ratio in gas–liquid slug flows without boiling

\[
Nu_{\text{U}(\text{Ent})} = \left( Nu_{\text{U}(\text{Pos;}\text{Ent})} \right)^{1/n} + \left( Nu_{\text{U}(\text{Dev})} \right)^{1/n} \right)^{n-10}
\]  

(9)

Finally, as was noted at the beginning of this section, the derived correlation is not limited to modeling heat transfer problems but can also be applied to similar mass transfer problems. This is achieved by replacing the Nusselt number throughout with its analogous mass transfer equivalent, the Sherwood number, and likewise changing the Prandtl number used in the definition of inverse Graetz parameter, Eq. (2), with the Schmidt number. It is noted that such an analogy should provide an excellent mass transfer model within the entrance region, Eq. (7), since this was defined from theoretical considerations alone. The model put forward for fully developed flow on the other hand, Eq. (8), incorporates an empirically derived coefficient which may change if applied to mass transfer problems. Further experiments are required to determine if this is the case but the power law relation between the transfer rate and slug aspect ratio should remain unchanged.

5. Conclusions

The current study reported novel experimental insights into the flow physics associated with segmented slug flows under a constant wall heat flux boundary condition. High spatial resolution temperature measurements were obtained which allowed for flow characteristics within the entrance, transition and fully developed regions to be accurately defined. It was shown that heat transfer characteristics in the entrance region of slug flows are governed by a combination of conventional single phase and plug flow behaviors with short slugs approaching the theoretical plug flow limit and long slugs approaching the single phase flow limit.
Another interesting finding, with regard to the thermal entrance length, was that slug flows reach their fully developed asymptotic limit at a distance of one slug length regardless of flow Reynolds number. However, in transitioning from the entrance region to fully developed slug flow, local Nusselt number oscillate about the fully developed limit with a period equal to the length of one internal circulation. Finally, in the fully developed flow limit, it was found that Nu_{dev} can be augmented by up to an order of magnitude over the single phase limit by utilizing slugs of length to diameter ratio close to unity. A correlation based on the described flow physics was proposed to aid in predicting local Nu over the entire range of inverse Graetz numbers but it was noted that some error may be encountered at the transition region due to the observed oscillations. Overall, the findings reported herein provide a greater understanding of the physics associated with non-boiling slug flows and correlations are proposed which can be used for predictive purposes in either heat or mass transfer applications.

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