Heat Transfer Characteristics of Liquid-Gas Taylor Flows incorporating Microencapsulated Phase Change Materials

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Abstract. This paper presents an investigation on the heat transfer characteristics associated with liquid-gas Taylor flows in mini channels incorporating microencapsulated phase change materials (MPCM). Taylor flows have been shown to result in heat transfer enhancements due to the fluid recirculation experienced within liquid slugs which is attributable to the alternating liquid slug and gas bubble flow structure. Microencapsulated phase change materials (MPCM) also offer significant potential with increased thermal capacity due to the latent heat required to cause phase change. The primary aim of this work was to examine the overall heat transfer potential associated with combining these two novel liquid cooling technologies. By investigating the local heat transfer characteristics, the augmentation/degradation over single phase liquid cooling was quantified while examining the effects of dimensionless variables, including Reynolds number, liquid slug length and gas void fraction. An experimental test facility was developed which had a heated test section and allowed MPCM-air Taylor flows to be subjected to a constant heat flux boundary condition. Infrared thermography was used to record high resolution experimental wall temperature measurements and determine local heat transfer coefficients from the thermal entrance point. 30.2\% mass particle concentration of the MPCM suspension fluid was examined as it provided the maximum latent heat for absorption. Results demonstrate a significant reduction in experimental wall temperatures associated with MPCM-air Taylor flows when compared with the Graetz solution for conventional single phase coolants. Total enhancement in the thermally developed region is observed to be a combination of the individual contributions due to recirculation within the liquid slugs and also absorption of latent heat. Overall, the study highlights the potential heat transfer enhancements that are attainable within heat exchange devices employing MPCM Taylor flows.

1. Introduction

Two phase flows consist of a dispersed immiscible phase within a base fluid and they can take many different geometric configurations which are driven by balances between interfacial tension with viscous, inertial and gravitational forces. Liquid-gas flows are the primary focus of this work, specifically investigating non-boiling flows. In mini and micro channels, where the hydraulic diameter is less than the Capillary length of the fluid, slug or Taylor flow regime is typically observed. This is a well ordered segmented flow comprised of alternating liquid slugs and gas bubbles. Closely controlled flow rates can be used to develop trains with varying slug lengths where the gas bubble is separated

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from the channel wall by a liquid film [1]. Non-boiling flows are simply generated by pumping both immiscible phases into the channel which allows for greater control. A key component of such flows is the void fraction which is the fraction of channel occupied by the gas phase. This can typically be controlled by the inlet volumetric quality, or simply the ratio of gas flow rate to total flow rate within the system.

The primary benefit associated with Taylor flows is significantly enhanced transport phenomena. The separation of the two phases into discrete fluid packets significantly affects fluid dynamics and as a result, the heat transfer rates attainable. One of the most notable effects is the enhancement of radial transport observed in laminar flows due to flow confinement within alternate slugs and bubbles. If viewed from a moving wall reference frame, this radial convection is observed as an internal circulation [2]. This fluid motion is presented in figure 1 and results in modified velocity profiles that have been shown to increase heat transfer rates. Many authors have investigated specifically the enhancement associated with liquid-gas Taylor flows, both experimentally [3, 4] and numerically [5, 6]. A study [7] which examined liquid-gas Taylor flows, confirmed liquid slug length to be one of the most important parameters related to heat transfer enhancement. Increased performance is observed with a reduction in slug length but data needs to be reduced to account for the gaseous voids within the channel. A comprehensive model was also put forward to predict for local heat transfer rates. A later study [8] expanded the applicability of this model to a greater range of liquid phases.

**Figure 1.** Schematic highlighting fluid recirculation within liquid slugs when observed from a moving wall reference frame.

Alternative investigations into heat transfer enhancements have looked into the incorporation of specifically engineered suspension fluids. These are typically either micro or nano scale in size and suspended within a base fluid. However, recent advances in microencapsulation technologies have enabled the development of microencapsulated phase change material (MPCM) suspensions. These are suspensions where a solid core is encapsulated within a polymer shell and suspended within a base fluid as demonstrated in figure 2. MPCM suspensions offer significantly enhanced convective area for heat transfer to the core while the resultant flows have only a modest increase in viscosity. Phase change particles absorb latent heat of fusion as they melt and have been reported to demonstrate enhancement in thermal management applications [9]. Latent heat absorption results in an apparent increase in specific heat capacity at the phase transition temperature. This increase in specific heat capacity can be many times greater than the sensible heat of the suspension which increases its apparent thermal capacity. This technology enables devices which are subjected to isoflux conditions to theoretically maintain a near constant temperature within the fluid during the phase change process.

**Figure 2.** Microencapsulated phase change material (MPCM) suspension.
Many investigations have examined the heat transfer characteristics of microencapsulated phase change material suspensions, particularly under laminar flow conditions. Studies looking at local heat transfer rates are typically numerical [10, 11]. Experimental investigations are commonly focused on global performance and 2 – 4 fold increases in Nusselt number have been reported [12]. Other investigations have demonstrated significant deviation in local Nusselt number values from the conventional Graetz solution [13]. Many authors have identified the concentration of particles within the base fluid as being the one of the most dominant parameters [14]. Microencapsulated phase change materials are conventionally a separate field of research from Taylor flows with no literature available on segmented type MPCM or particulate liquid-gas flows. A large gap exists to segment MPCM flows with a gas phase as it would theoretically combine the benefits associated with both flow types. These include the increased heat transfer rates observed for Taylor flows with the increased thermal capacity attributable to the phase change process. The goal of this investigation is to combine these flow technologies and study the overall thermal performance.

The investigation addresses a significant deficit in the literature. Previous publications have investigated local heat transfer rates associated with Taylor flows and MPCM particulate flows and this work combines both those technologies. An experimental study was conducted to investigate liquid-gas Taylor flows incorporating MPCM materials and analyse the effects associated with the primary Taylor flow parameters. These include liquid slug length and gas void fraction. Overall, this investigation aims to further enhance the fundamental understanding associated with two phase flows and provide steps in developing future thermal solutions.

2. Phase Change Characteristics of MPCM Suspensions

Microencapsulated phase change materials are solid micron sized particles that are dispersed within a liquid phase to form a suspension. The presence of particulates result in modified bulk thermophysical properties associated with the base fluid. It is necessary to be able to predict such properties when using analytical models or investigating MPCM flows. These suspensions are dominated by the phase change process and this is principally defined by the onset melt temperature and latent heat of fusion. The melt temperature is simply the point at which phase change occurs within the particles while latent heat represents the quantity of heat which can be absorbed throughout the transition. These properties are determined with the use of differential scanning calorimetry (DSC). Figure 3 demonstrates a DSC plot of the MPCM particles used in this investigation, where a heating/cooling rate of 5°C/min was used. It clearly indicates the onset melt temperature, \( T_1 \), and the latent heat of fusion, \( h_{fs} \), which are equal to 36.1°C and 160.7 J/g, respectively.

![DSC plot of MPCM particles](image)

**Figure 3.** Differential scanning calorimetry (DSC) plot of MPCM particles, highlighting the onset melt temperature and latent heat of fusion.

Developing a specific heat capacity model is required when analysing MPCM flows as it embodies both sensible and latent heat capacity of the suspension. It is typically modelled as a piece wise equation where latent heat is absorbed as an effective enhancement in specific heat capacity over the
melt temperature range [15]. Sensible heat capacity is employed for temperatures below and above $T_1$ and $T_2$, respectively, which is described using a mass weighted average expression (1).

$$c_{p,mpcm} = c_{p,pf} + (1 - \phi_m)c_{p,bf}$$

Typically, specific heat capacity of the particles can be lower than the base fluid, hence, the primary benefit associated with MPCM flows is latent heat attributed to the solid-liquid phase transition. The latent heat of the MPCM particles is measured using DSC. This is obtained by integrating the change in specific heat capacity with respect to temperature defining the phase change. Hence, the effective increase in specific heat capacity over the melt range, $c_{p,pc}$, as given by (2).

$$c_{p,pc} = \frac{h_{fs}}{T_2 - T_1}$$

A piece wise expression is used to describe the specific heat capacity, with a step change observed over the phase change region and this technique of accounting for the latent heat of MPCM suspensions is widely used by many authors [16, 17]. The use of different profiles within the phase change region, such as a sine wave was also suggested [16].

3. Theoretical Models

The primary benefit associated with Taylor flows is the potential they offer to enhance transport phenomena or more specifically heat transfer. Local dimensionless heat transfer rates for such flows are characterised by local Nusselt number (3).

$$Nu_x = \frac{h D}{k} = \frac{q^* D}{k(T_{xw} - T_{x,bm})}$$

This is a function of dimensionless longitudinal position or inverse Graetz number (4).

$$x^* = \frac{x}{D Re Pr} = \frac{x}{d Pe}$$

The Graetz solution describes local heat transfer values occurring in hydrodynamically developed flows [18, 19] but this was simplified to a piecewise solution [20]. The solution is derived from an energy balance and fully defines the temperature field which results in an expression governing variation of local Nusselt number with dimensionless position and spans both the thermally developing and fully developed regions. A further simplification to the piecewise expression was achieved with asymptotic limits [21] where an expression is defined for the thermal entrance region while fully developed values are constant. These limits were combined using the addition of asymptotic limits approach [22] to develop (5).

$$Nu_{x,point} = \left[ \frac{1.302}{x^{1/3}} \right]^5 + (4.36)^5$$
\[ \varepsilon_{\beta} \approx \beta = \frac{Q_g}{Q_g + Q_s} \]  

(6)

When considering heat exchange systems incorporating MPCM slurries, phase change within the particles leads to deviations of theoretical predictions from conventional single phase flows. The primary difference is observed in the bulk mean temperature as illustrated in figure 4. This is due to absorption of latent heat as the phase change particles melt. For conventional liquid cooling flows subjected to a constant heat flux, bulk mean temperature increases linearly which can be calculated using a simple energy balance. However, with MPCM flows this is not the case due to the change in specific heat capacity over the melt region. Phase change occurs over a temperature range, \( T_1 \) to \( T_2 \).

![Figure 4. Theoretical bulk mean temperature profile of MPCM suspension flows.](image)

The bulk mean temperature increase with respect to axial distance can be divided into three distinct regions: pre-melt region where \( T < T_1 \), melt region where \( T_1 < T < T_2 \) and post-melt region where \( T_2 < T \). Within the pre- and post-melt regions, temperature rise is dictated by sensible heating based on the specific heat capacity of the suspension but the effective phase change heat capacity, \( c_{p,pc} \), within the melt region results in a lower temperature increase along the channel. An energy balance can be used to develop analytical correlations describing these distinct regions (7).

\[
T_{\text{x,bm}} = \begin{cases} 
T_{\text{in}} + \frac{q'' \pi D x}{c_{p,\text{mpcm}} \dot{m}} & x < L_1 \\
T_1 + \frac{q'' \pi D(x - L_1)}{c_{p,\text{mpcm}} + \phi_{\text{m}} c_{p,pc} \dot{m}} & L_1 < x < L_2 \\
T_2 + \frac{q'' \pi D(x - L_2)}{c_{p,\text{mpcm}} \dot{m}} & L_2 < x 
\end{cases}
\]  

(7)

The energy balance is based on the applied heat flux, \( q'' \), Diameter, \( D \), specific heat capacity, \( c_p \), and the mass flow rate, \( \dot{m} \). \( L_1 \) and \( L_2 \) are the axial distances downstream from the thermal entrance point which indicates onset and end of the phase change process, (8, 9), respectively. They are defined using a balance based on temperatures \( T_1 \) and \( T_2 \).

\[
L_1 = \frac{(T_1 - T_{\text{in}}) \rho c_{p,\text{mpcm}}}{q'' \pi D}
\]  

(8)
By employing the Graetz solution (5), to predict wall temperatures in conjunction with (7), a reduction in local wall temperature rise is predicted. This highlights a common issue with MPCM suspension flows where particles nearest the wall reach the melt temperature, \( T_1 \), before the remainder of the channel, hence, this results in a moving melt boundary, known as the ‘Stefan Problem’. It leads to propagation of a phase change boundary moving down stream with transition also dependent on the radial position [23]. This means that the phase melt range occurs at the wall before the bulk mean of the fluid reaches the phase change temperature, hence, the Graetz solution may not accurately predict experimental wall temperatures within the phase change region. Previous authors [13, 24] have shown deviation in wall temperatures from the theoretical Graetz solution but none have offered a correlation to allow its prediction.

4. Experimentation

The primary aim of this work was to investigate the thermal characteristics of liquid-gas Taylor flows incorporating microencapsulated phase change material (MPCM) suspension as the liquid phase. In order to accomplish this, an experimental setup was designed and constructed which is presented in figure 5. The rig was required to enable the development of well ordered Taylor flows and subject the flow to an isoflux boundary condition while recording experimental wall temperatures. There are three distinct regions: 1) slug generation where the well ordered segmented flow was developed, 2) thermal region where the flow is subjected to the constant heat flux boundary condition, and 3) slug visualisation where both liquid slug and gas bubble lengths can be accurately measured downstream of the test section. The MPCM particulate solution was custom made by Thies Technology. Particles contained a methyl stearate core encapsulated by a polyurethane shell and were dispersed in water. 30.2\% mass concentration of particles within the base fluid was used throughout the experimental investigation and the thermo-physical properties of this suspension are presented in Table 1.

Steady flows were generated such that flow rates of the respective phases could be accurately controlled and this was achieved by employing two syringe pumps, pumping both a gaseous and liquid phase respectively. High precision Harvard PHD 2000 programmable syringe pumps were used and capable of accurately and independently setting liquid and gas volumetric flow rates. These were delivered from 100ml capacity Hamilton gas tight syringes. Flow from each pump was combined in a T-junction before entering the experimental test section. A range of liquid slug and gas bubble lengths were achieved by using T-junctions with varying internal diameters. Variations in the flow structure were also achieved by changing the relative flow rate magnitudes which resulted in varying volumetric quality, \( \beta \). A settling period was also allowed when generating the Taylor flows due to compressibility of the gas phase but resulted in steady flows with constant slug and bubble lengths.
Table 1. Thermo-physical properties of MPCM suspensions

|        | μ  | ρ  | γ  | c_ρ | k  | h_f | T_1 | T_2 |
|--------|----|----|----|-----|----|-----|-----|-----|
| Particles | mPa s | kg/m^3 | mN/m | J/kg K | W/m K | kJ/kg | °C | °C |
| 30.2% | 7.75 | 965.2 | 55.06 | 3642.4 | 0.486 | 48.3 | 36.1 | 38.1 |

The experimental test section allowed the flow to be subjected to a constant heat flux boundary condition while recording wall temperature measurements. The heated test section consisted of a 1.5mm internal bore stainless steel tube with a wall thickness of only 250μm. The thermal boundary condition was created by Joule heating using a D.C power supply. Electrical connections to the tube were achieved using custom made sharp edged copper ring contacts to ensure that the thermal entrance point could be clearly identified. 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. Infrared measurements also required the external surface of the test section to be sprayed matt black to increase its surface emissivity. The value of the emissivity was determined by comparing experimental images with four thermocouple readings which were placed directly on the tube surface while single flow was pumped through the system. This was pumped at sufficiently high flow rates as to minimise the difference between thermocouples and the emissivity was found to be 0.95 which is typical for a matt black finish. The heated test section was also placed within an enclosure in order to shield it from its environment. For each test, thermal images were recorded for up to 6 minutes at a frequency of 1Hz.

An analysis code in Matlab was used for extracting the transient temperature profiles of the external tube surface from thermal images and it was a four step process as illustrated in figure 6. The first image highlights scaling which was obtained from images recorded prior to heating. Next, the region of interest was identified and an edge definition algorithm was used to locate the tube walls and hence, identify its centreline. Experimental temperature measurements were recorded at each pixel location along the tube centreline which resulted in a data point approximately every 300μm. Temperature data was extracted from each experimental image as highlighted in figure 6 (c). The recorded image number is plotted versus the tube axial distance where image numbers are representative of a time domain through the recording frequency. The temperature in °C is represented by colour contours as indicated in the legend. The desired range of image numbers are selected where the flow has reached steady state and mean time averaged temperature profile was outputted as shown in image (d). Upper and lower lines represent three standard deviations of the periodically fluctuating temperature along the length of the test section. This fluctuation is indicative of the alternating flow structure associated with Taylor flows.
The entire system was calibrated using single phase flows as water was pumped through the system for a range of different flow rates with a uniform heat flux being applied. This allowed the electrical resistance of the stainless steel test section to be calculated and hence, heat flux values by assuming the water flow exactly matched the Graetz solution. Experimental wall temperatures are observed to closely follow the analytical predictions, although, slight deviation is noted immediately downstream of the thermal entrance point due to heating effects associated with the copper electrodes.

5. Results and Discussion

Dimensionless heat transfer rates were investigated for MPCM flows which were segmented by a gas phase. Only slurries with a concentration of 30.2% were analysed since they provide the maximum enhancement over Graetz flow predictions. This concentration was held constant, as was the applied heat flux at 6.23kW/m², in order to quantify the effect of primary Taylor flow parameters, i.e. dimensionless slug length, \( L^* \), and length void fraction, \( \varepsilon_{L/\beta} \). The thermo-physical characteristics presented in Table 1 were used throughout the analysis.

Experimental wall temperatures were recorded and MPCM characteristics were compared with the Graetz solution for Hagen Poiseuille flow. Figure 7 (a) compares experimental wall temperatures of single phase and two phase flows for the same mass flow rate of MPCM suspension, \( Q_{mpcm} = 6ml/min \). Volumetric gas quality, \( \beta \), was increased from 0 for single phase to 0.5 for the MPCM-air flow. The

**Figure 7.** Local wall temperatures (a) and Local \( N_u \) values (b) comparing MPCM, with and without segmentation.
inlet T-junction resulted in a dimensionless liquid slug length, \( L_s^* = 3.5 \). Single phase MPCM flow results in reduced wall temperatures when compared with the theoretical Graetz solution for Poiseuille flow (5) and this agrees with previously published work [13]. It was also observed that the Taylor flow regime provides a further reduction in wall temperatures even when data is not reduced to take account of the gaseous voids in the flow. These are similar to observation made for short slug lengths by [7, 8] which investigated heat transfer characteristics associated with liquid-gas flows without phase change. Figure 7 (b) is developed by converting local wall temperatures into dimensionless Nusselt numbers and plotted against inverse Graetz number. The data is also reduced to account for the proportion of the channel occupied by the liquid phase, i.e. \((1 - \varepsilon_L \beta)\). Two phase flow utilising MPCM suspension demonstrates elevated \( Nu \) values in both the thermal entrance A) and thermally developed B) regions. These are similar to trends seen for liquid-gas flows [7] but significant deviation is observed when compared with the correlation put forward by the same authors, this is indicated by a dashed line in figure 7 (b). A significant increase in dimensionless heat transfer is seen in the thermal entrance region while the thermally developed asymptote appears to be significantly below predictions. It is also observed that two phase MPCM flows exhibit the \( Nu \) enhancement peak C) which was previously noted for conventional MPCM flows [13]. However, in this case a reduction peak was not observed in this measurement as it occurred beyond the experimental field-of-view of the IR thermography system. An increased deviation in local \( Nu \) values over the phase change region is apparent when comparing the different flow regimes. A shift in dimensionless position of the enhancement peak is also observed due to the higher axial velocities, hence, lower Reynolds numbers associated with Taylor flows for similar mass flow rates of MPCM suspension.

**Figure 8.** Experimental \( Nu \) vs \( x^* \) data for increasing flow rates of MPCM-gas flows.

Initially, the influence of MPCM \( Re \) number on dimensionless heat transfer was examined. While maintaining a constant volumetric gas quality, the volume flow rate of the MPCM suspension was increased from 3\( ml/min \) to 8\( ml/min \) for the same inlet T-junction. The resultant slug lengths varied from 3.2 – 3.9 diameters due to slight variations in the flow structure. Those variations are attributable to the slug-bubble curvature becoming elongated with increased two phase velocity, hence, reducing the volume of fluid within the slug core. Experimental local Nusselt numbers for this range of flow rates are plotted in figure 8. All plots demonstrate very good agreement, suggesting the effect of changing Reynolds number is adequately captured by the conventional inverse Graetz number scaling. Dimensionless profiles collapse very well in the thermal entrance region. All flow rates examined result in the same thermally developed asymptote but differences in oscillation magnitude are observed. Oscillations experienced along the \( Nu \) profile are due to the segmented nature of two phase flows. The dependency on dimensionless velocity suggests a transient effect which is related to heat
conduction through the liquid film. This is most likely a Fourier effect and similar to damping observed in previous work [8] for flows with thick liquid films. Dimensionless profiles in figure 8 also partially capture the reduction peak typically observed in MPCM flows.

![Figure 9](image)

**Figure 9.** Local wall temperature (a) and local $Nu$ vs $x^*$ (b) data of MPCM-gas flows with increasing volumetric gas quality.

Investigating the influence of remaining Taylor flow parameters, the flow rate of MPCM was held constant at 6ml/min. Three different size inlet T-junctions were used to attain a range of dimensionless slug lengths, $L_s^*$, while the volume flow rate of gas was varied from 2 – 10ml/min. Varying volumetric gas quality directly resulted in varied length void fraction. Figure 9 (a) and (b) present experimental results for min and max gas flow rates which were developed using the smallest inlet T-junction. Increasing the volume flow rate of air within the system results in a larger length void fraction but reduced liquid slug length. Figure 9 (a) shows the experimental wall temperatures for MPCM-air flows with $L_s^*$ equal to 2.97 and 5.66 while $\epsilon_L/\epsilon_B$ values are 0.57 and 0.25, respectively. Wall temperatures shown in figure 9 (a) are not reduced to account for void fraction and both plots demonstrate similar profiles to the two phase flow characteristics already shown in figure 7 (a). Although reduced slug lengths result in enhanced heat transfer, increased void is undesirable within a system to maximise the overall heat transfer. The shorter slug length results in higher experimental wall temperatures as it has larger voids within the channel. Figure 9 (b) captures the same wall temperatures converted to dimensionless form, i.e. $Nu$ vs $x^*$. The graph illustrates the dependence on
dimensionless liquid slug length within the thermal entrance region and thermally developed values, before phase change occurs. A reduction in $L_s^*$ results in enhanced $Nu$ values in both the entrance and fully developed regions. Thermally developed asymptotic limits can be interpreted from oscillations in the plot. Significant differences between the thermally developed asymptote and the peak $Nu$ values can also be observed, suggesting that it is a function of the primary Taylor flow parameters.

![Image of Local Number plot](image)

**Figure 10.** Local Number plot, highlighting the effects of void fraction, $\varepsilon_L(\beta)$, on enhancement due to phase change, $\Delta Nu_{enh}$.

The total enhancement in dimensionless heat transfer for MPCM-air flows is a summation of enhancements observed in MPCM Poiseuille flow plus an enhancement due to flow recirculation within the slugs. Results in figure 9 (b) highlight differences in $Nu$ enhancement, $\Delta Nu_{enh}$, due to phase change, i.e. peak height above the thermally developed asymptotic limit and this is clearly identified in figure 10. This plot shows two profiles with similar dimensionless slug lengths, $L_s^*$ equal to 5.7 and 6.6, respectively which were developed using two different inlet T-junctions. Good agreement is observed between both profiles in the thermal entrance region. Similar slug lengths result in similar thermally developed $Nu$ asymptotes but they have significantly different volume gas qualities, 0.25 and 0.625, respectively which is observed to affect peak enhancement. Increased gas quality reduces the $x^*$ distance to each peak, as previously identified but it also reduces the magnitude of peak enhancement. Enhancement due to phase change, $\Delta Nu_{enh}$, becomes a function of length void fraction. Phase change particles act as surfactants within the Taylor flow and preferentially gather at the interfaces in the flow so much of the MPCM capsules are at the leading and trailing slug interface. It is believed that these particle concentrations at the interface could lead to differences in $\Delta Nu_{enh}$ observed in figure 10. It is also believed that phase change particle concentrations within the liquid film result in the elevated heat transfer rates observed in the thermal region, above the correlation of [7]. Increased amounts of surfactants or particles result in a hardening of the interface which leads to changes in the velocity profile. This modification is believed to result in some appreciable volumetric rate within the film that contributes to the heat transfer. Further work is required to fully understand the fluid dynamics within this film region and effect of phase change particles.

6. Conclusions

This investigation presented and discussed the heat transfer characteristics of MPCM particulate Taylor flows subjected to an isoflux boundary condition. Data shows segmented MPCM flows provide further enhancement than that observed for MPCM Poiseuille flows. Dimensionless liquid slug length and void fraction have been shown to have a significant effect on dimensionless heat transfer values. Total heat transfer enhancement observed with these flows is a summation of enhancement due to recirculation within the slugs and enhancement due to phase change within the particles. $\Delta Nu_{enh}$ has
been shown to be dependent on $\varepsilon_{L(\beta)}$ which is believed to be due to particles concentrations at the interface. Nusselt number enhancement above previous predictions within the thermal entrance region was observed for MPCM-air flows, which is believed to be attributable to modified velocity profiles and increased flow within the film. Overall, this investigation on MPCM Taylor flows has shown significant enhancements in local heat transfer rates over conventional Poiseuille flows and even higher local $Nu$ values than predictions for other liquid-gas flows within the phase change region.

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References
[1] Bretherton F P 1961 Journal of Fluid Mechanics 10(2) 166 – 188
[2] Taylor G I 1961 Journal of Fluid Mechanics 10(2) 161 – 165
[3] Horvath C, Soloman B A and Engasser J M 1973 Industrial & Engineering Chemistry Fundamentals 12 431 – 439
[4] Vrentas J, Duda L J, Lehmkuhl G D 1978 Industrial & Engineering Chemistry Fundamentals 17(1) 39 – 45
[5] Lakehal D, Larrignon G and Narayanan C 2008 Microfluidics and Nanofluidics 4(4) 261 – 271
[6] Narayanan C and Lakehal D 2008 Journal of Heat Transfer 130(7) 074502
[7] Walsh P A, Walsh E J and Muzychka Y S 2010 International Journal of Heat and Mass Transfer 53 3193 – 3201
[8] Howard J A, Walsh P A and Walsh E J 2011 International Journal of Heat and Mass Transfer 54 4752 – 4761
[9] Medrano M, Yilmaz M O, Nogues M, Martorell I, Roca J and Cabeza L F 2009 Applied Energy 86(10) 2047 – 2055
[10] Goel M, Roy S K and Sengupta S 1994 International Journal of Heat and Mass Transfer 37(4) 593 – 604
[11] Alisetti E L and Roy S K 2000 Journal of Thermophysics and Heat Transfer 14(1) 115 – 118
[12] Charunyakorn P, Sengupta S and Roy S K 1991 International Journal of Heat and Mass Transfer 37(4) 819 – 833
[13] Howard J A and Walsh P A 2013 Heat Transfer Engineering 34(2–3) 223 – 234
[14] Yamagishi Y, Takeuchi H, Pyatenko A T and Kayukawa N 1999 AIChE Journal 45(4) 696 – 707
[15] Kasza K E and Chen M M 1985 Journal of Solar Energy Engineering 107(3) 229 – 236
[16] Hu X and Zhang Y 2002 International Journal of Heat and Mass Transfer 45(15) 3163 – 3172
[17] Roy S K and Avanic B L 2001 International Communications in Heat and Mass Transfer 28(7) 895 – 904
[18] Graetz L 1883 Annual Review of Physical Chemistry 18 79 – 94
[19] Graetz L 1883 Annual Review of Physical Chemistry 25 337 – 357
[20] Shah R K and London A L 1978 Laminar Flow Forced Convection in Ducts: Sussplement 1 to Advances in Heat Transfer (New York: Academic Press)
[21] Muzychka Y S and Yovanovich M M 2004 ASME Journal of Heat Transfer 126 54 – 61
[22] Churchill S W and Usagi R 1972 AIChE Journal 18(6) 1121 – 1128
[23] Choi E, Cho Y I and Lorsch H G 1994 International Journal of Heat and Mass Transfer 37(2) 207 – 215
[24] Zeng R, Wang X, Chen B, Zhang Y, Niu J, Wang X and Di H 2009 Applied Energy 86(12) 2661 – 2670