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NON-BOILING HEAT TRANSFER IN HORIZONTAL AND NEAR HORIZONTAL UPWARD INCLINED GAS-LIQUID TWO PHASE FLOW

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ABSTRACT

Heat transfer in non-boiling gas-liquid two phase flow finds its practical application in chemical and petroleum industries. So far, majority of the research dedicated to study heat transfer in non-boiling two phase flow is limited to horizontal and vertical pipe orientations with very little attention given to the study of this phenomenon in inclined systems. To contribute and further enhance the general understanding of heat transfer in non-boiling two phase flow, the main focus of this work is to experimentally measure local and average convective heat transfer coefficients for different flow patterns in horizontal and near horizontal upward inclined two phase flow. In total, 368 experiments are carried out in a 12.5 mm I.D. schedule 10S stainless steel pipe at 0, +5, +10 and +20 degrees pipe orientations using air-water as fluid combination. For each pipe orientation, the superficial gas and liquid Reynolds number is varied from 200 to 19,000 and 2000 to 18,000, respectively and the measured values of the averaged heat transfer coefficient were found to be in a range of 1300 W/m^{2}K to 8000 W/m^{2}K. The two phase heat transfer coefficients are compared among the above mentioned orientations. It is found that the two phase heat transfer coefficient increases from 0º to +5º and +10º degree and then decreases at +20 degree.

INTRODUCTION

Non-boiling two phase flow is a common and important occurrence in industries, especially in the oil industry. The temperature of the hydrocarbon fluids flowing in the pipe during transportation changes due to the difference in temperature of the surface and the oil reservoir. This results in wax deposition in the inner walls of the pipe blocking fluid flow thereby causing severe mechanical problems. Increasingly, inclined pipes are being used in the transportation of the fluids. Due to the lack of literature regarding flow regimes and thermo-fluid dynamics in inclined pipes, experimental work in this study is carried to fill the gap in literature and enhance understanding of the two phase heat transfer phenomenon in inclined non-boiling systems. The available experimental data in two phase flow literature for horizontal and near horizontal upward inclined pipes is that of Ghajar and Tang [1], Tang and Ghajar [2] and Hestroni et al. [3, 4]. It is important to note that the experiments conducted by Tang and Ghajar [2] deal with pipe diameter of 25 mm and that of Hestroni et al. [3, 4] are for pipe diameters 25 mm and 50 mm. Both of these studies have observed a significant enhancement in two phase heat transfer with the increase in inclination from the horizontal. The current experiments analyses the two phase heat transfer in a pipe of diameter 12.5 mm and the results agree with that observed by the above mentioned authors to some extent. The results of these different experiments show that the heat transfer coefficient also depends on the pipe diameter along with flow pattern and pipe inclination. The experimental data from these investigations can be used to develop heat transfer correlations accounting for the above mentioned factors. In the present study the data is collected for 0°, +5°, +10°, and +20° using air-water mixture and is measured at similar mass flow rates for all orientations. The flow patterns observed in different inclinations are mapped to have a better understanding of the effect of pipe inclination on the transition between different flow patterns. The experimental data of two phase heat transfer coefficient is analyzed as a function of gas and liquid flow rates and their magnitude is compared for different pipe orientations.

NOMENCLATURE

dz	[m]	Differential change in axial direction
D	[m]	Pipe diameter
G	[kg/m^{2}s]	Mass flux
h	[W/m^{2}K]	Heat transfer coefficient
$\bar{h}$	[W/m^{2}K]	Circumferentially averaged heat transfer coefficient
L	[m]	Pipe length
Nu	[-]	Nusselt number
N_{ST}	[-]	No. of thermocouple stations
Re	[-]	Reynolds number
T	[ºC,K]	Temperature
U	[m/s]	Phase velocity
x	[-]	Two phase flow quality
z	[m]	Axial direction

Special characters

μ	[Pa-s]	Phase dynamic viscosity
ρ	[kg/m^3]	Phase density
θ	[deg.]	Pipe orientation
Δ		Differential operator

Subscripts

b	Bulk
G	Gas
i	Pipe inlet
L	Liquid
o	Pipe outlet
S	Superficial
TP	Two phase

EXPERIMENTAL SETUP

The experimental set up used for measuring two phase convective heat transfer coefficient as shown in Figure 1 consists of 12.5 mm I.D. schedule 10 S steel pipe of roughness 0.0152 mm. The setup also consists of a 12.67 mm I.D. transparent polycarbonate pipe that can be used for flow visualization and measurement of void fraction and pressure drop. The fluid combination used for generating two phase flow is compressed air-distilled water. The air is supplied through an Ingersoll Rand T-30 Model 2545 compressor, passed through a regulator and filter-lubricator circuit before it is fetched to the water submerged coil heat exchanger. Next, the air is passed through Coriolis mass flow meter and controlled by the Parker needle valve (Model 6A-NLL-NE-SS-V) before it is mixed with water in the static mixer. The liquid phase, i.e., distilled water is stored in a 55 gallon tank and is circulated in the system using a Bell and Gosset (series 1535, model number 3445 D10) centrifugal pump and passed through an Aqua-Pure AP12-T purifier followed by the flow through a ITT model BCF 4063 shell and tube heat exchanger. The water is then directed to flow meter through Emerson (Micro Motion Elite Series model number CMF 100) Coriolis mass flow meter and then allowed to mix with air in a static mixer. The water mass flow rate is controlled by a gate valve placed after the water mass flow meter. The CO1-T type thermocouples with an accuracy of ±1ºC are used to measure wall temperatures at seven different stations spaced 127 mm apart along the pipe length. The thermocouple probes (TMQSS-06U-6) used to measure temperature at the pipe inlet ($T_{i,b}$) and outlet ($T_{o,b}$) are inserted inside through pipe wall till it almost touched the other end of the pipe wall in order to ensure that the probes are always in contact with the two phase mixture.

The thermocouples at each station (four thermocouples at each station) are arranged radially along the pipe circumference as shown in Figure 2. The uniform heat flux in a range of 7500 W/m^2 to 57,000 W/m^2 is supplied by Lincoln DC-600 welder having a maximum current supply of 750 Amp. In order to ensure no heat transfer takes place from the system to surrounding, a 0.076 m (3 in.) thick Micro-Lok Fiber Glass insulation with thermal conductivity of 0.042 W/mºC is used. The local inside wall temperature, wall heat flux and convective heat transfer coefficient is calculated using a finite difference formulation based data reduction program developed by Ghajar and Kim [5]. The two phase convective heat transfer coefficient is represented by the average of the measured local values at each station as shown in Eq. (1).

$$h_{TP} = \frac{1}{L}\int \bar{h}\,dz = \frac{1}{L}\sum_{j=1}^{N_{ST}} \bar{h}_j \Delta z_j \tag{1}$$

The uncertainty of the experimental data for single phase heat transfer coefficient is calculated using Kline and McClintock [6] uncertainty analysis. The validity of the single phase heat transfer data is also confirmed by comparing it against the correlations of Gnielinski [7], Ghajar and Tam [8] and Sieder and Tate [9]. As shown in Figure 3 the measured single phase heat transfer coefficient is found to be within ±10% of the predicted values by Seider and Tate [9]. The average and maximum deviation in measurement of single phase heat transfer coefficient with respect to the correlations of Gnielinski [7] and Ghajar and Tam [8] was found to be 4.8%, 14.82% and 0.86%, 3.7%, respectively.

High uncertainty in the measured heat transfer coefficient is found in the annular flow region. This is expected as the heat transfer in annular flow region becomes difficult to measure due to a small temperature difference between inlet and outlet. It is also to be noted that for bubbly and slug flows, the uncertainty values are considerably lower. For the flows which are affected by both gravity and inertia in upward inclination, like wavy flows, the uncertainty varies to a greater extent. In intermittent flows, including slug wavy or wavy annular flows, the flow boundaries are hugely affected by the change in orientation and gravitation. Heat balance error also seems to affect the uncertainty. In slug flows the heat balance error tends to be at maximum of 3.40% while in annular flows the heat balance error is found to be up to 12% for higher pipe orientations. The heat balance error for intermittent flows is in a range of 5.65% to 10% for the low, moderate or high mass flow rates of water. Table 1 summarizes the minimum and maximum uncertainty in the measured two phase heat transfer coefficient for different flow patterns and pipe orientations.

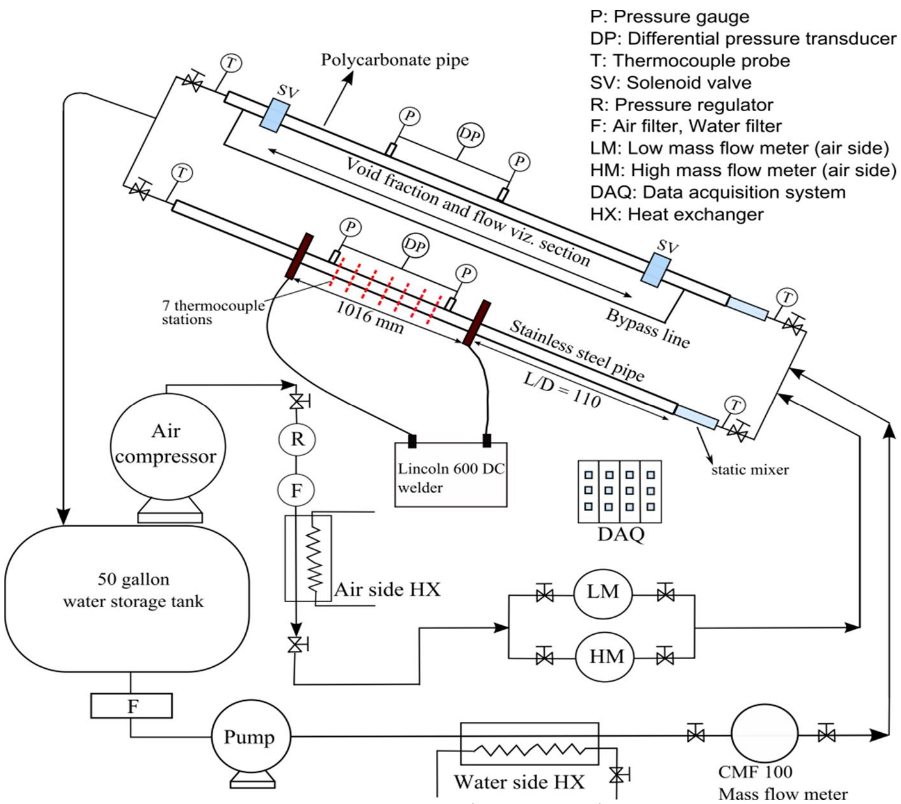


Figure 1 Experimental Setup used for heat transfer measurements.

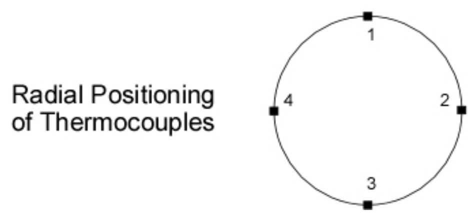


Figure 2 Radially arranged thermocouples at pipe circumference.

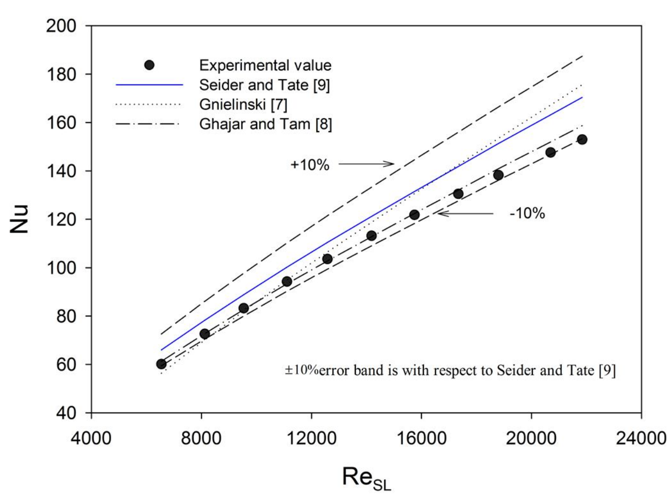


Figure 3 Uncertainty in measurement of single phase Nusselt number

Table 1 Minimum and maximum uncertainty in two phase heat transfer coefficient for different flow patterns and pipe orientations.

Flow pattern		5°	10°	20°	0°
Stratified	Min %	-	-	-	27
	Max %	-	-	-	10
Intermittent	Min %	13.56	12.03	13.13	25
	Max %	30.26	28.12	25.85	11
Slug	Min %	12.04	12.27	12.30	13
	Max %	16.87	15.97	16.47	9
Bubbly	Min %	11.66	10.89	10.86	12
	Max %	11.06	11.64	11.79	10
Wavy	Min %	14.72	15.40	15.08	-
	Max %	23.20	22.01	21.42	-
Annular	Min %	15.57	16.86	15.85	-
	Max %	34.92	32.76	29.58	-

RESULTS AND DISCUSSION

Flow Patterns and Flow Pattern Maps

The study of the effect of pipe orientation on two phase flow is important since it is found that the variations in most of the two phase flow parameters for constant gas and liquid flow rates are related to the flow patterns which in turn are directly affected by the change in pipe orientation. The different two phase flow patterns are generated due to the compressibility nature of the gas phase and significantly different physical properties of the two phases. A careful mapping of the flow patterns is essential for the estimation of the sequence of appearance of different flow patterns with change in the gas and liquid flow rates. It should be noted that the definitions of flow patterns and their transitions are highly qualitative in nature and are mostly based on the individual's perception of the physical structure of two phase flow patterns. Other methods such as power spectral analysis of the pressure drop signal and probabilistic flow regime mapping are introduced recently in the two phase flow literature however; these methods are very much specific to the flow conditions and are indirectly based on some reference to the visual flow pattern observation. In the present study, the key flow patterns observed in horizontal and upward inclined two phase flow are bubbly, slug, intermittent, stratified and annular flow regimes as shown in Figure 4. These flow patterns are generated by varying the gas and liquid flow rates (superficial gas and liquid Reynolds numbers) in a range of 0.003 kg/min to 0.2 kg/min (Re_{SG} = 200 to 20,000) and 1.5 kg/min to 12.5 kg/min (Re_{SL} = 2000 to 18,000), respectively. The superficial gas and liquid Reynolds number is defined in terms of superficial phase velocity, phase density and viscosity and pipe diameter as represented in Eqs. (2) and (3). The flow visualization is carried out in transparent polycarbonate pipe using Nikon D3100 camera and 200mm f/5.6 lens with a shutter speed of 1/4000 s.

$$\mathrm{Re}_{SG} = \frac{\rho_G D U_{SG}}{\mu_G} = \frac{GxD}{\mu_G} \tag{2}$$

$$\mathrm{Re}_{SL} = \frac{\rho_L D U_{SL}}{\mu_L} = \frac{G(1-x)D}{\mu_L} \tag{3}$$

The bubbly flow regime is characterized by the dispersion of small gas phase bubbles in the continuous liquid medium near the pipe top wall while the slug flow is identified as a flow structure consisting of elongated gas slugs of varying lengths and frequency that flow alternate to a liquid plug. The stratified flow is featured by the gas phase flowing parallel to the liquid phase at the bottom wall of the pipe. The annular flow is observed in form of liquid film flowing in contact with the pipe wall that surrounds a fast moving gas core. Theoretically, the annular flow is defined as a flow of two phases flowing separately. However, in practice, the annular flow is accompanied by significant entrainment of the liquid droplets into the fast moving gas core. This entrainment is quite noticeable for high liquid flow rates and gradually reduces with decrease in the liquid flow rates. It's expected that the two phase annular flow with liquid entrainment will increase the two phase heat transfer due to added turbulence in comparison to the quasi equilibrium (no entrainment) annular two phase flow that may occur for very low liquid flow rates.

In the present study, the intermittent flow pattern is identified based on the pulsating, chaotic and wavy nature of the two phase flow that lacks a specific alignment of the gas phase with respect to the liquid phase. Thus the flow structure tagged as intermittent flow in the present study comprises of slug-wavy, stratified-wavy and annular-wavy flow patterns for moderate liquid and moderate gas flow rates, low liquid and moderate gas flow rates and low to moderate liquid and high gas flow rates, respectively.

The slug wavy flow is marked by vigorously moving distorted slug and is characterized by the entrainment of tiny bubbles in both elongated gas bubble and the liquid slug. The slug wavy flow may be referred to as a chaotic version of slug flow that helps to enhance two phase heat transfer. The definition of stratified wavy flow that appears only in horizontal pipe orientation is the wavy and unstable gas liquid interface. In this subcategory of intermittent flow, the increase in liquid and gas flow rates caused the gas-liquid interface to become wavy with momentarily touching (splashing) of the liquid phase to the pipe top wall. In wavy annular flow, the liquid film thickness at pipe bottom wall is considerable compared to that at the pipe top wall. The liquid film at the pipe top wall is supported mostly due to the continuous splashing of the liquid however, sometimes a few dry spots are also observed.

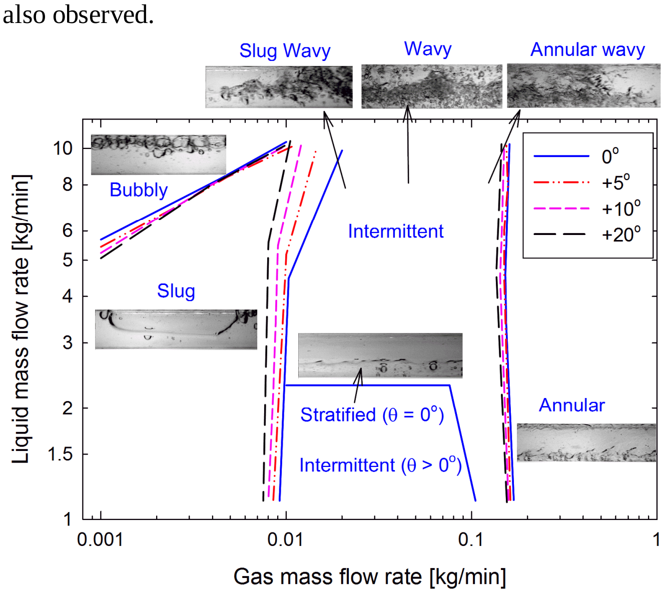


Figure 4 Flow regime map for horizontal and upward inclined two phase flow.

As shown in Figure 4, for all pipe orientations considered in this study, the bubbly flow exists for low gas and high liquid flow rates. For horizontal pipe orientation, at low liquid flow rates, increase in the gas flow rate causes the flow pattern to shift from stratified to intermittent and finally to annular flow. The intermittent flow between stratified and annular flow may be classified as wavy annular flow having physical structure described above. Whereas, for moderate liquid flow rates, the flow pattern shifts from slug to intermittent to annular flow as the gas flow rate is increased from low to moderate to high flow rates. It is also evident from this flow pattern map that the transition boundaries between different flow patterns with the exception of boundary between intermittent and annular flow are influenced by the change in pipe orientation. It should be noticed that for horizontal flow, there is no stratified flow pattern for gas mass flow rates lower than 0.01 kg/min and in upward inclined pipe orientations.

A peculiar phenomenon observed in upward inclined two phase flow is the 'flow reversal'. Flow reversal in upward pipe inclinations results from the interaction of buoyancy, inertia and gravity forces and is a consequence of dominant gravity force acting on the liquid phase that tends to pull the liquid phase back while the gas phase moves in the upward direction. In the present study, flow reversal is observed for low values of liquid and gas flow rates typically $Re_{SL} < 5000$ and $Re_{SG} < 2000$. The flow reversal is observed by visual observation when the liquid layer near the pipe wall (bottom) is found to move in the direction opposite (downward) to that of the mean flow (upward). Based on the flow visualization it is found that the flow reversal phenomenon depends on the pipe orientation to a considerable extent and the range of gas flow rates over which the flow reversal is directly proportional to the pipe orientation. The two phase flow literature reports experimental work related to flow reversal for vertical upward flow with hardly any information available for upward inclined systems. Typically, two phase flow literature acknowledges the existence of flow reversal by finding the frictional pressure drop minimum after which the frictional pressure drop increases consistently with increase in the gas flow rates indicating end of flow reversal regime.

Table 2 Distribution of data for different flow patterns at different pipe orientations.

Flow pattern	0º	+5º	+10º	+20º
Stratified	5	-	-	-
Slug	28	17	13	12
Intermittent	58	34	36	30
Bubbly	4	3	6	7
Wavy	-	9	11	12
Annular	-	29	25	29
Total	95	92	91	90

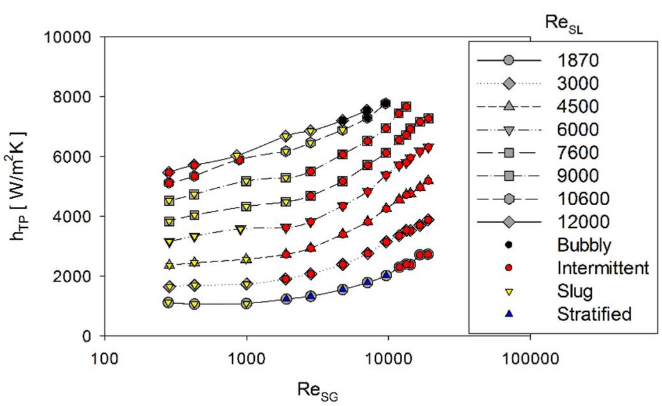
A total of 368 data points of two phase heat transfer measurements are carried out with about 90 data points in each orientation. As seen from Table 2, majority of these data points are populated in intermittent, slug and annular flow regimes. Due to the limitations on the mass flow meter, very few data for bubbly flow could be measured.

Effect of Flow Patterns and Pipe Orientation on Two Phase Heat Transfer Coefficient

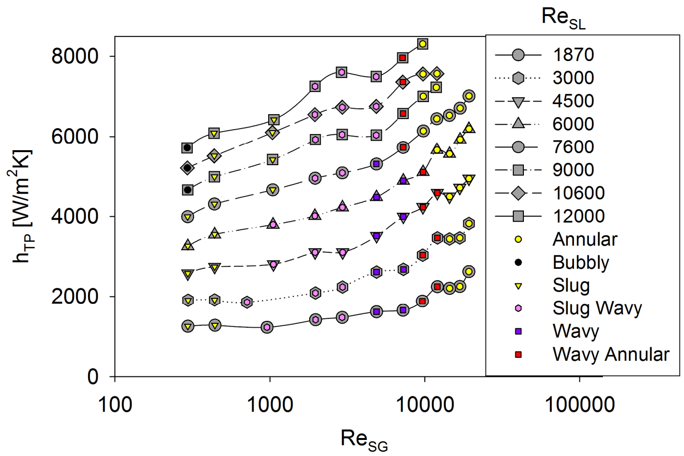
Considering the dependence of flow structure on the pipe orientation, this experimental study attempts to study the combined effect of the flow pattern and change in pipe orientation on two phase heat transfer coefficient. This combined effect is studied by varying the inclination of the pipe in the upward direction with measurements carried out at similar gas and liquid mass flow rates at all orientations. Figure 5 shows the variation of two phase heat transfer coefficient measured for varying gas and liquid flow rates. The liquid flow rate ranged from 1.5 to 5 kg/min ($2000 < Re_{SL} < 12,500$) and the gas flow rate ranged from 0.003 to 0.2 kg/min ($200 < Re_{SG} < 20,000$). The flow pattern corresponding to each combination of Re_{SG} and Re_{SL} is also identified for better understanding of the flow pattern effect on h_{TP}. It is observed that the heat transfer coefficient increases with the increase of the superficial liquid and gas Reynolds number. In case of horizontal two phase flow, it is found that the two phase heat transfer coefficient remains virtually unchanged in the slug flow regime while increases sharply with increase in Re_{SG} ($Re_{SG} > 3000$) in intermittent flow regime due to added turbulence and enhanced mixing of the two phases. In case of upward inclined systems, the two phase heat transfer coefficient remains almost constant due to the flow reversal effect that causes the liquid film to travel in the downward direction and hence doesn't contribute significantly to the increase in h_{TP}.

For $1000 < Re_{SG} < 10,000$ and $Re_{SL} = 2600$ stratified flow is observed in the horizontal orientation which is shear driven in nature. As the flow is co-current, no resistance is offered to the inertial force of the liquid. The gas layer moves on top of the wavy and unstable liquid layer creating a disturbance wave that increases the heat transfer rate. For $Re_{SG} < 1000$ and $Re_{SL} < 12,100$ slug flow is observed in horizontal and upward inclined systems. Bhagwat et al. [10] and Oliver and Wright [11] observed that the two phase heat transfer coefficient in slug flow regime is a function of slug length and slug frequency. The magnitude of h_{TP} is observed to increase with increasing slug frequency that corresponds to short length slugs and decreases with decrease in slug frequency associated with large length slugs. This relation between h_{TP} and slug length and slug frequency is anticipated since fast moving slugs would result into lower residence time of the gas phase in the test section (smaller contact time of the gas phase with the pipe wall) and hence increase in the heat transfer coefficient. The opposite must hold true for slow moving long slugs.

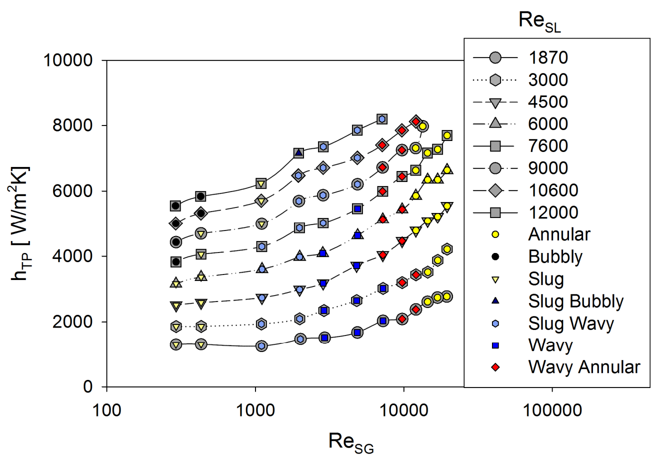
For 1000 < Re_{SG} < 10,000 and Re_{SL} > 2600, the flow is intermittent (slug wavy) in nature. The increasing trend of h_{TP} shown in the Figure 5 at all orientations reflects the observations reported above. In slug wavy type of a flow for Re_{SL} > 6000, the liquid phase seems to repeatedly wet the top surface of the pipe and thereby increasing the h_{TP}. This observation is in agreement with that of Tang and Ghajar [2]. The wetting of top surface is decreased in wavy region where the high velocity gas phase appears to suppress the wave growth on the liquid surface thereby preventing the liquid splashing on the pipe top wall and hence a slight drop in h_{TP} is observed. This effect is pronounced for +5° and then gradually decreases in +10° and +20°. For instance, in +5° for 1000 < Re_{SG} < 5000 and Re_{SL} > 7600, we see momentarily peaking of h_{TP} in slug wavy region and then a drop when the flow pattern changes to wavy region. This drop occurs because the frequent wetting of top surface with the waves on liquid surface is not possible in wavy flow regime. In +10°, this effect is reduced as it is observed for Re_{SL} > 9000, the transition in the flow occurs from slug wavy to wavy annular with the increase in Re_{SG}. In +20°, the effect is not observed as gravitational effects come into picture suppressing the lateral growth (across pipe cross section) of the wave in slug wavy region, enabling the flow pattern to go into wavy region with the increase in Re_{SG}.



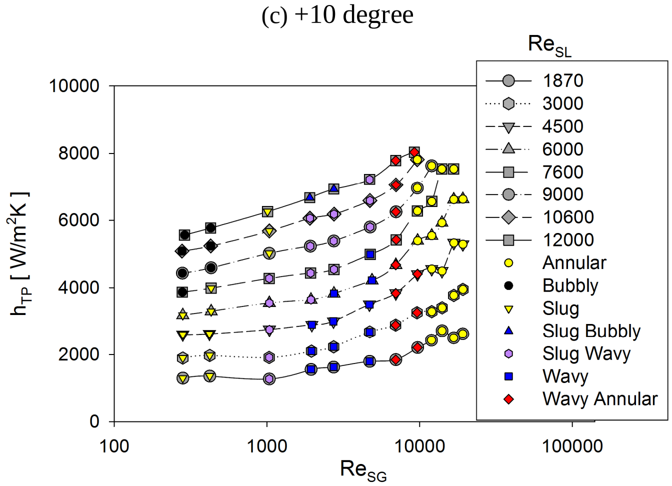
(a) 0 degree



(b) +5 degree



(c) +10 degree



(d) +20 degree

Figure 5: Variation of two phase heat transfer coefficient at fixed Re_{SL} and increasing Re_{SG} (a) 0 degree, (b) +5 degree, (c) +10 degree, (d) +20 degree.

As disused so far, it can be concluded that the heat transfer in upward inclination is heavily dependent on the physical structure of flow pattern and also the pipe inclination. For a slug flow we see maximum heat transfer in +5° as a result of wetting of the top surface. This phenomenon also occurs for +10° flow in wavy region. For +20° flow, the waves cannot wet the top surface as pipe orientation and gravitational forces act to suppress the wave rise. The comparison percentage increase in the heat transfer between different orientations with reference to horizontal flow is shown in Figure 6. It would also be interesting to know the level of enhancement in terms of numbers to get a clear understanding of the effect of the pipe orientation on h_{TP}. For slug/slug wavy flow regime the maximum increase of 15% was observed in +5° pipe orientation. As the flow pattern is changed to wavy and annular regime a consistent increase in h_{TP} of about 10% in +10° is observed whereas for +20° it was between 0.5% and 5%. For low values of gas flow rates (Re_{SG} < 5000), the lower values of h_{TP} and large percentage change in h_{TP} are essentially due to the existence of flow reversal in upward inclined systems. It is also observed that as the flow pattern changes from wavy to wavy annular/annular, the increase of heat transfer in +5º and +20º (about 5%) which is not as much as it is in +10º (10%) accounting for the maximization of heat transfer in +10° for most of the populated data.

In Figure 7 we observe that for slug flows the effect of pipe orientation on two phase heat transfer coefficient is negligible at low gas flow rates (flow reversal region), but for Re_{SG} > 4500, the effect of pipe orientation on h_{TP} is noticeable for different flow patterns. At +20° pipe orientation, a steep increase of h_{TP} is observed at Re_{SG} = 17,000 after the flow transits to wavy annular. This increase in h_{TP} is probably because unlike in wavy region, a thin liquid film is always in contact with the pipe upper wall that aids two phase heat transfer in wavy annular flow. A slight increase in h_{TP} is observed further this point for low values of Re_{SL}. This average increase in h_{TP} observed in the present study for Re_{SL} = 4500 is about 6%. At higher liquid flow rates (Re_{SL} > 10,000) we observe the effect of slug-wavy flow on h_{TP} in the inclined orientations. For +5° it dramatically increases the h_{TP} and falls as the flow changes from slug wavy to wavy annular. Also, as the Re_{SL} increases gradually, the difference between h_{TP} at inclined pipe orientations and horizontal is gradually increased which is clearly illustrated in Figure 7. For high values of gas flow rates (Re_{SG} > 5000), waves are developed on the liquid surface and there is not a significant difference in the flow structure. However, a minute observation of the wavy liquid surface reveals that in case of horizontal flow 'rolling waves' are observed that are in contact with the pipe upper wall quite often in comparison to the 'wave undercutting' observed in upward inclined flows. In wave undercutting process the wavy liquid surface is tore off by the fast moving gas and the torn liquid ligament moves in a direction opposite to that of the mean flow. However, in case of 'rolling wave' process, the gas phase rolls the liquid crests in the downstream direction possibly without detaching it from the liquid surface. This perhaps promotes a local acceleration of the liquid phase that enhances two phase heat transfer coefficient in horizontal orientation. In upward inclination the 'wave undercutting' affects the orientations above +10° reducing the two phase heat transfer coefficient.

With increasing liquid flow rates it is found that the two phase heat transfer coefficient in inclined system supersedes the h_{TP} in horizontal orientation to a greater degree. This is mostly because that at high liquid flow rates, the translational velocity of the gas slugs (in slug and wavy slug flow) or disturbance waves (in wavy annular and annular flow) is aided by the buoyancy acting on the gas phase that increases the slug/disturbance wave frequency and hence enhances two phase heat transfer coefficient with increase in the pipe orientation. However, it is found that at +20º pipe orientation, h_{TP} is consistently less than that observed in +10º inclination.

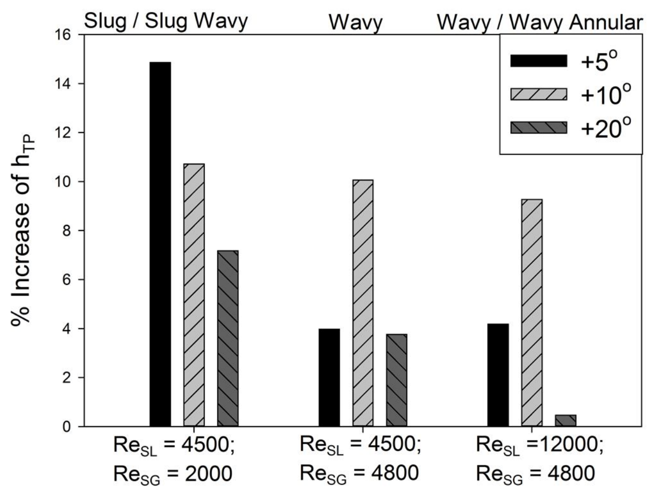
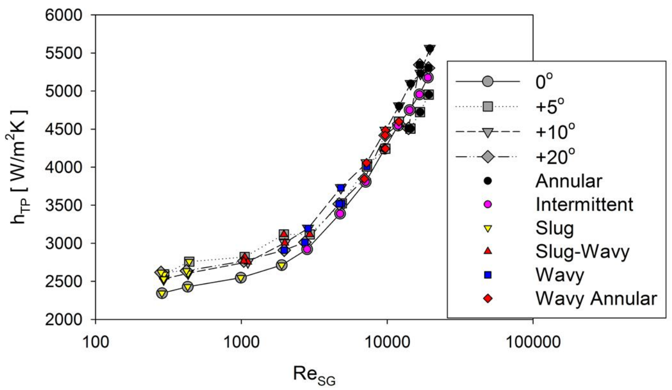


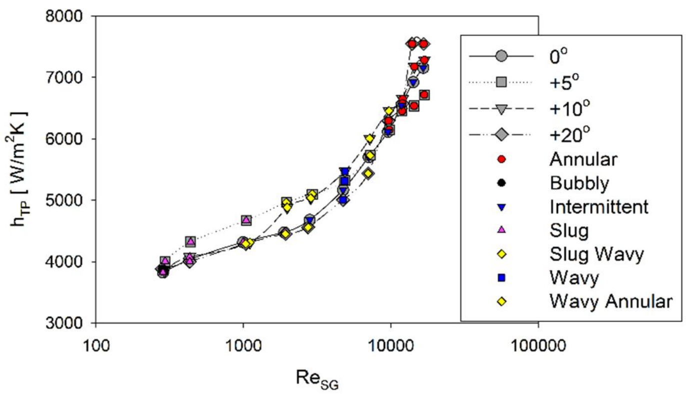
Figure 6: Percentage change in h_{TP} for different inclinations with reference to horizontal orientation.

In order to verify if the two phase heat transfer coefficient decreases consistently between +10º and +20º, some measurements are also carried out at an intermediate orientation of +15º. These data points populated in +15° revealed that the heat transfer coefficient is maximum at +10° and from there on decreases with further increase in the pipe orientation. This can be attributed to the fact that as the inclination increases above +10° gravitational effects dominate the buoyancy effects to decrease the heat transfer. This can be seen in Figure 8.

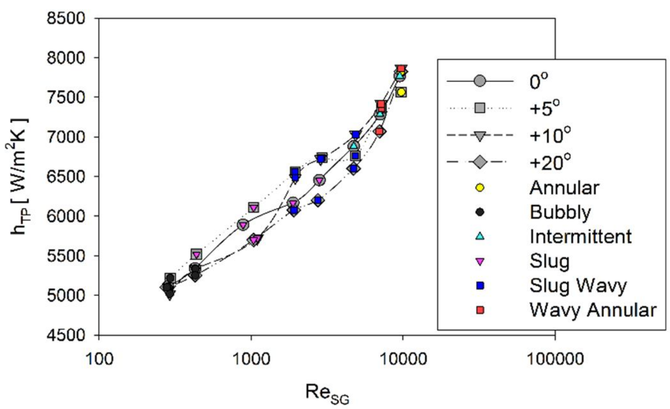
Conclusively, it can be said that for low liquid flow rates involving slug/slug wavy flow +5° is predominantly greater than horizontal, +10º and +20º. For medium liquid and gas flow rates with wavy and wavy annular two phase flow, heat transfer coefficient in +10º is greater than all the orientations considered in this study. For medium to high liquid flow rates (7000 < Re_{SL} < 13000), and all gas flow rates h_{TP} increases from 0º to +5º and then from +5º to +10º and eventually decreases from +10º to +20º. It would be interesting to check the trends of h_{TP} at intermediate pipe orientations such as +3°, +7° and orientations above +20° from horizontal to verify if the flips in trend of two phase heat transfer coefficient observed between 0° and +5° and +10° and +20° occur on a random or systematic basis.



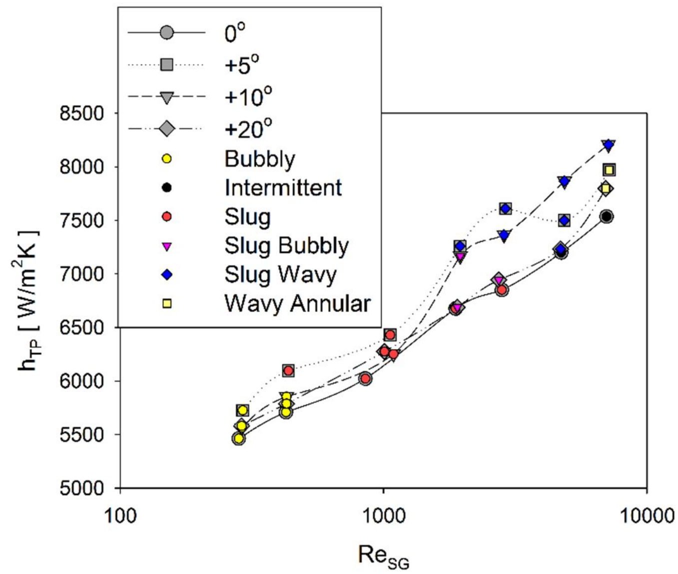
(a) Re_{SL} = 4500



(b) Re_{SL} = 7600



(c) Re_{SL} = 10,600



(d) Re_{SL} = 12,000

Figure 7: Comparison of two phase heat transfer coefficient at different liquid superficial Reynolds number (a) Re_{SL} = 4500, (b) Re_{SL} = 7600, (c) Re_{SL} = 10,600, (d) Re_{SL} = 12,000.

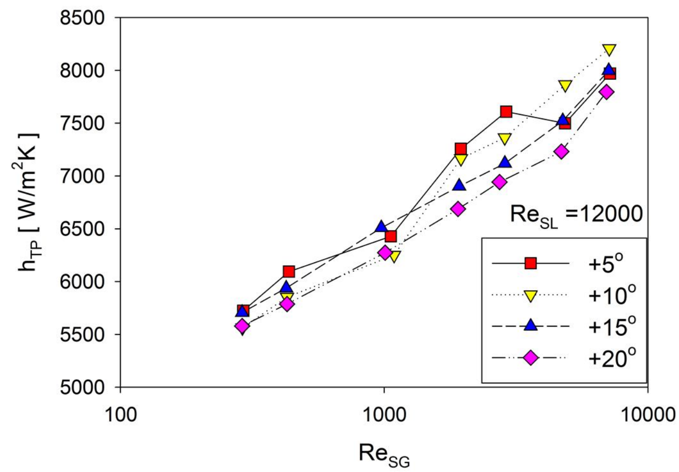


Figure 8: Comparison between upward inclined pipe orientations

CONCLUSIONS

In this study new data is presented for non-boiling two phase heat transfer coefficient in horizontal and near horizontal upward inclinations. The experimental work is carried out in a 12.5 mm I.D. pipe using air-water fluid combination. The general trends of two phase heat transfer coefficient with change in gas and liquid flow rates and pipe orientation have been presented. The relative magnitude of h_{TP} for varying two phase flow conditions is discussed with the help of physical structure of the flow patterns. It is found that the variation of h_{TP} is dependent on both flow patterns and pipe orientation. Overall it is found that there is an increase in two phase heat transfer coefficient in upward inclinations compared to that in horizontal two phase flow. The two phase heat transfer coefficient maximizes at +10° for most of the data involving intermittent, wavy and annular region. Based on this experimental data it would be interesting to check the trends of h_{TP} at intermediate pipe orientations between 0º and +5º and above +20°. Moreover, further investigation is required to check the relative magnitudes of two phase heat transfer coefficients for different fluid combinations and pipe diameter.

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