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SYSTEMATIC HEAT TRANSFER MEASUREMENTS FOR AIR-WATER TWO-PHASE FLOW IN A HORIZONTAL AND SLIGHTLY UPWARD INCLINED PIPE

Afshin J. Ghajar, Jae-yong Kim, Kapil Malhotra, and Steve Trimble
School of Mechanical and Aerospace Engineering, Oklahoma State University, Stillwater, OK 74078, USA
ghajar@ceat.okstate.edu

Abstract. *Local heat transfer coefficients and flow parameters were measured for air-water flow in a pipe in the horizontal and slightly upward inclined positions (2°, 5°, and 7°). The test section was a 25.4 mm stainless steel schedule 10S pipe with a length to diameter ratio of 100. For this systematic study, a total of 121 data points were taken on horizontal position by carefully coordinating the liquid and gas superficial Reynolds number combinations. Theses superficial Reynolds numbers were duplicated for each inclination angle. The heat transfer data were measured under a uniform wall heat flux boundary condition ranging from about 3000 to 10,600 W/m^2. The superficial Reynolds numbers ranged from about 820 to 26,000 for water and from about 560 to 48,000 for air. Comparison of heat transfer data for two-phase gas-liquid flow revealed that the heat transfer results were significantly dependent on the liquid and gas superficial Reynolds numbers, flow pattern, and inclination angle. The experimental data indicated that even in a slightly upward inclined pipe, there is a significant effect on the two-phase heat transfer of air-water flow.*

keywords: *two-phase flow, gas-liquid flow, heat transfer, horizontal flow, upward-inclined flow*

1. Introduction

Gas-liquid two-phase flow in pipes is commonly observed in many industrial applications, such as oil wells and pipelines, solar collectors, chemical reactors, and nuclear reactors, and its hydrodynamic and thermal conditions are dependent upon the interaction between the two phases. Therefore, it is very important to understand heat transfer in gas-liquid two-phase flow for economical and optimized operation in those industrial applications.

A comprehensive discussion of the available experimental data and heat transfer correlations for forced convective heat transfer during gas-liquid two-phase flow in vertical and horizontal pipes, including flow patterns and fluid combinations is provided by Kim et al., 1999. However, due to the complex nature of the two-phase gas-liquid flow, no systematic investigation has been conducted to document the influence of flow pattern and inclination angle on the two-phase heat transfer. The only available information on the effect of inclination in the literature is from Hetsroni et al., 1998, and their study was qualitative in nature and limited to slug flow. Their experimental work measured the local heat transfer, using infrared thermography, as a function of slug frequency, slug length and height, inclination angle, and Froude number. Inclination angles were limited to 2° and 5°. They concluded that there was a drastic increase in heat transfer with only slight increases in inclination angle. The authors provided no quantitative information to support the observed increase in the heat transfer. Later, Trimble et al., 2002, quantitatively investigated the effect of inclination on heat transfer in slug flow. In their experimental study, the 2° and 5° data showed an average increase over the horizontal position of about 10 % and 20 %, respectively. However, their investigation was exploratory in nature and was not conducted systematically. In addition it was limited to only one flow pattern (slug flow).

The objectives of this study were to extend the knowledge base by gathering quality non-boiling, two-phase, two-component heat transfer data in the horizontal and inclined positions with various flow patterns, and analyze their behavior in order to develop a general overall heat transfer coefficient correlation for gas-liquid two-phase flow regardless of flow orientation. In order to achieve this goal, the nature of the heat transfer in air-water two-phase flow was investigated by comparing the two-phase heat transfer data that were obtained by systematically varying the air or water flow rates (flow pattern) and the pipe inclination angle.

2. Experimental Setup

A schematic diagram of the overall experimental setup for heat transfer measurements is shown in Fig. 1. The test section is a 25.4 mm straight standard stainless steel schedule 10S pipe with a length to diameter ratio of 100. The setup rests atop a 9 m long aluminum I-beam that is supported by a pivoting foot and a stationary foot that incorporates a small electric screw jack.

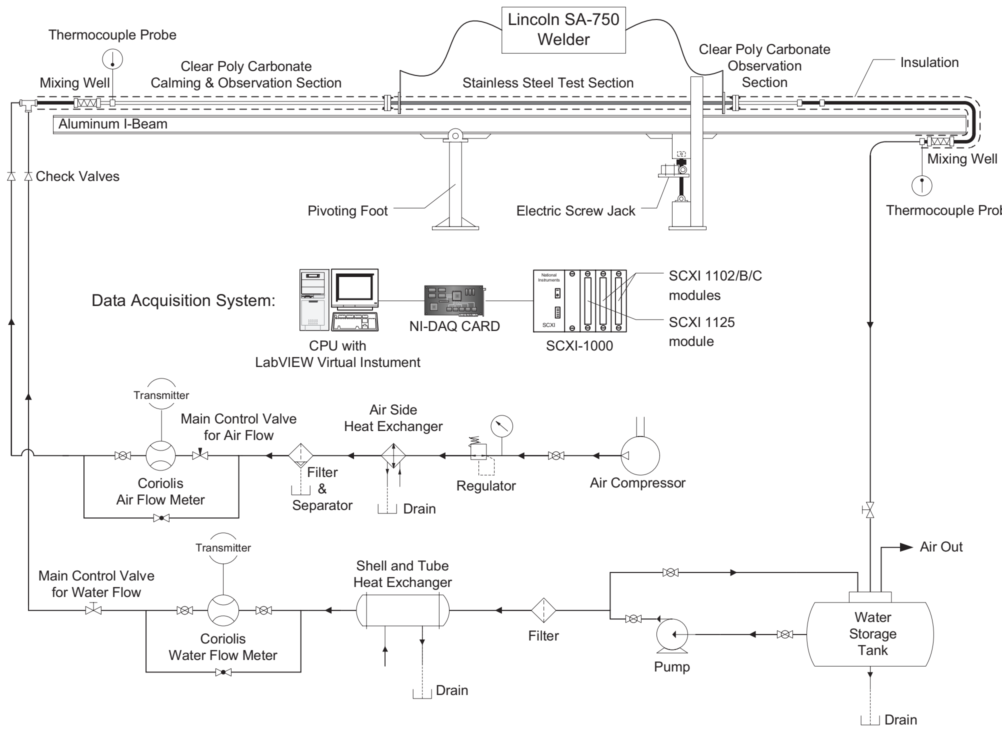


Figure 1: THE SCHEMATIC OF THE EXPERIMENTAL SETUP

In order to apply uniform wall heat flux boundary condition to the test section, copper plates were silver soldered to the inlet and exit of the test section. The uniform wall heat flux boundary condition was maintained by a Lincoln SA-750 welder. The entire length of the test section was wrapped using fiberglass pipe wrap insulation, followed by a thin polymer vapor seal to prevent moisture penetration.

In order to develop various two-phase flow patterns (by controlling the flow rates of gas and liquid), a two-phase gas and liquid flow mixer was used as shown in Fig. 2. The mixer consisted of a perforated stainless steel tube (6.35 mm I.D.) inserted into the liquid stream by means of a tee and a compression fitting. The end of the stainless steel tube was silver-soldered. Four holes (3 rows of 1.587 mm, 4 rows of 3.175 mm, and 8 rows of 3.968 mm) were positioned at 90° intervals around the perimeter of the tube and this pattern was repeated at fifteen equally spaced axial locations along the length of the stainless steel tube. The two-phase flow leaving mixer entered the transparent calming section.

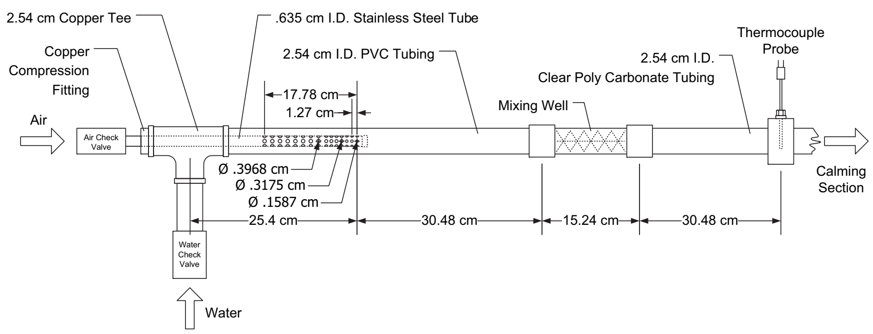


Figure 2: THE AIR-WATER MIXING SECTION

The calming section [clear polycarbonate pipe with 25.4 mm I.D. and $L/D = 88$] served as a flow developing and turbulence reduction device, and flow pattern observation section. One end of the calming section is connected to the test section with an acrylic flange and the other end of the calming section is connected to the gas-liquid mixer. For the horizontal flow measurements, the test section, and the observation section (refer to Fig. 1) were carefully leveled to eliminate the effect of inclination on these measurements.

T-type thermocouple wires were cemented with Omegabond 101, an epoxy adhesive with high thermal conductivity and electrical resistivity, to the outside wall of the stainless steel test section as shown in Fig. 3. OMEGA EXPP-T-20-TWSH extension wires were used for relay to the data acquisition system. Thermocouples were placed on the outer surface of the tube wall at uniform intervals of 254 mm from the entrance to the exit of the test section. There were 10 thermocouple stations in the test section. All stations had four thermocouples, and they were labeled looking at the tail of the fluid flow with peripheral location "A" being at the top of the tube, "B" being 90° in the clockwise direction, "C" at the bottom of the tube, and "D" being 90° from the bottom in the clockwise sense (refer to Fig. 3). All the thermocouples were monitored with a National Instruments data acquisition system. The experimental data were averaged over a user chosen length of time (typically 20 samples/channel with a sampling rate of 400 scans/sec) before the heat transfer measurements were actually recorded. The average system stabilization time period was from 30 to 60 min after the system attained steady state. The inlet liquid and gas temperatures and the exit bulk temperature were measured by Omega TMQSS-125U-6 thermocouple probes. The thermocouple probe for the exit bulk temperature was placed after the mixing well. Calibration of thermocouples and thermocouple probes showed that they were accurate within ±5 °C. The operating pressures inside the experimental setup were monitored with a pressure transducer.

To ensure a uniform fluid bulk temperature at the inlet and exit of the test section, a mixing well was utilized. An alternating polypropylene baffle type static mixer for both gas and liquid phases was used. This mixer provided an overlapping baffled passage forcing the fluid to encounter flow reversal and swirling regions. The mixing well at the exit of the test section was placed below the clear polycarbonate observation section (after the test section), and before the liquid storage tank (refer to Fig. 1). Since the cross-sectional flow passage of the mixing section was substantially smaller than the test section, it had the potential of increasing the system back-pressure. Thus, in order to reduce the potential back-pressure problem, which might affect the flow pattern inside of the test section, the mixing well was placed below and after the test section and the clear observation sections. The outlet bulk temperature was measured immediately after the mixing well.

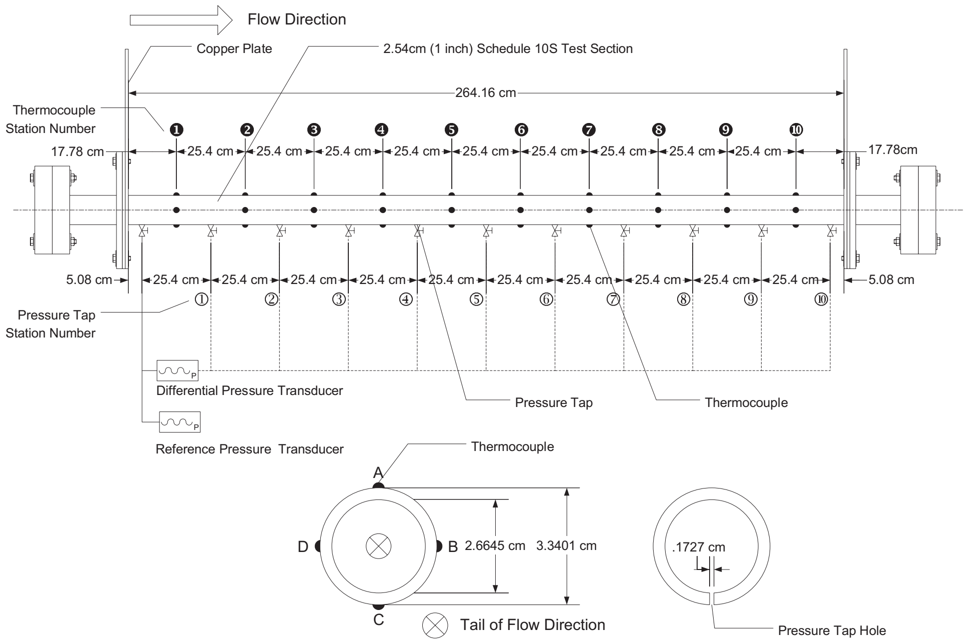


Figure 3: THE STAINLESS STEEL TEST SECTION

The fluids used in the test loop are air and water. The water is distilled and stored in a 55-gallon cylindrical polyethylene tank. A Bell & Gosset series 1535 coupled centrifugal pump was used to pump the water through an Aqua-Pure AP12T water filter. An ITT Standard model BCF 4063 one shell and two-tube pass heat exchanger removes the pump heat and the heat added during the test to maintain a constant inlet water temperature. From the heat exchanger, the water passes through a Micro Motion Coriolis flow meter (model CMF125) connected to a digital Field-Mount Transmitter (model RFT9739) that conditions the flow information for the data acquisition system. Once the water passes through the Coriolis flow meter it then passes through a 25.4 mm, twelve-turn gate valve that regulates the amount of flow that entered the test section. From this point, the water travels through a 25.4 mm flexible hose, through a one-way check valve, and into the test section. Air is supplied via an Ingersoll-Rand T30 (model 2545) industrial air compressor mounted outside the laboratory and isolated to reduce vibration onto the laboratory floor. The air passes through a copper coil submerged in a vessel of water to lower the temperature of the air to room temperature. The air is then filtered and condensate removed in a coalescing filter. The air flow is measured by a Micro Motion Coriolis flow meter (model CMF100) connected to a digital Field-Mount Transmitter (model RFT9739) and regulated by a needle valve. Air is delivered to the test section by flexible tubing. The water and air mixture is returned to the reservoir where it is separated and the water recycled.

The heat transfer measurements at uniform wall heat flux boundary condition were carried out by measuring the local outside wall temperatures at 10 stations along the axis of the tube and the inlet and outlet bulk temperatures in addition to other measurements such as the flow rates of gas and liquid, room temperature, voltage drop across the test section, and current carried by the test section. The peripheral heat transfer coefficient (local average) was calculated based on the knowledge of the pipe inside wall surface temperature and inside wall heat flux obtained from a data reduction program developed exclusively for this type of experiments (Ghajar and Zurigat, 1991). The local average peripheral values for inside wall temperature, inside wall heat flux, and heat transfer coefficient were then obtained by averaging all the appropriate individual local peripheral values at each axial location. The large variation in the circumferential wall temperature distribution, which is typical for two-phase gas-liquid flow in horizontal and slightly inclined tubes, leads to different heat transfer coefficients depending on which circumferential wall temperature was selected for calculations. In two-phase heat transfer experiments, in order to overcome the unbalanced circumferential heat transfer coefficient, Eq. (1) was used to calculate an overall mean two-phase heat transfer coefficient ($h_{TP_{EXP}}$) for each test run.

$$h_{TP_{EXP}} = \frac{1}{N}\sum_{i}^{N}\left[\frac{\frac{1}{M}\sum_{j}^{M}\dot{q}''_{i,j}}{\frac{1}{M}\sum_{j}^{M}(T_W)_{i,j} - (T_B)_i}\right] \tag{1}$$

where $\dot{q}''$ is heat flux; T_W is temperature of pipe inner-wall; T_B is fluid bulk temperature; M is number of thermocouples in a station; N is number of thermocouple station; $_i$ is index of thermocouple station; and $_j$ is index of thermocouple in a station.

The data reduction program used a finite-difference formulation to determine the inside wall temperature and the inside wall heat flux from measurements of the outside wall temperature, the heat generation within the pipe wall, and the thermophysical properties of the pipe material (electrical resistivity and thermal conductivity). In these calculations, axial conduction was assumed negligible, but peripheral and radial conduction of heat in the tube wall were included. In addition, the bulk fluid temperature was assumed to increase linearly from the inlet to outlet.

A National Instruments data acquisition system was used to record and store the data measured during these experiments. The data acquisition system is housed in an AC powered four-slot SCXI 1000 Chassis that serves as a low noise environment for signal conditioning. Three NI SCXI control modules are housed inside the chassis. There are two SCXI 1102/B/C modules and one SCXI 1125 module. From these three modules, input signals for all 40 thermocouples, the two thermocouple probes, voltmeter, and flow meters are gathered and recorded. The computer interface used to record the data is a LabVIEW Virtual Instrument (VI) program written for this specific application.

The reliability of the flow circulation system and of the experimental procedures was checked by making several single-phase calibration runs with distilled water. The single-phase heat transfer experimental data were checked against the well established single-phase heat transfer correlations (Kim and Ghajar, 2002) in the Reynolds number range from 3000 to 30,000. In most instances, the majority of the experimental results were well within ±10 % of the predicted results (Kim and Ghajar, 2002; Durant, 2003). In addition to the single-phase calibration runs, a series of two-phase, air-water, slug flow tests were also performed for comparison against the two-phase experimental slug flow data of Kim and Ghajar, 2002, and Trimble et al., 2002. The results of these comparisons, for majority of the cases were also well within the ±10 % deviation range.

The uncertainty analysis of the overall experimental procedures using the method of Kline and McClintock, 1953, showed that there is a maximum of 11.5 % uncertainty for heat transfer coefficient calculations. Experiments under the same conditions were conducted periodically to ensure the repeatability of the results. The maximum difference between the duplicated experimental runs were within ±10 %. More details of experimental setup and data reduction procedures can be found from Durant, 2003.

The heat transfer data obtained with the present experimental setup were measured under a uniform wall heat flux boundary condition that ranged from 2730 to 10690 W/m^2 and the resulting mean two-phase heat transfer coefficients (h_{TP}) ranged from 513 to 4419 W/m^2·K for horizontal flow. For these experiments, the liquid superficial Reynolds number (Re_{SL}) ranged from 821 to 26,043 (water mass flow rates from about 1.18 to 42.5 kg/min) and the gas superficial Reynolds numbers (Re_{SG}) ranged from 560 to 47,718 (gas mass flow rates from about 0.013 to 1.13 kg/min).

3. Flow Patterns

Due to the multitude of flow patterns and the various interpretations accorded to them by different investigators, no uniform procedure exists at present for describing and classifying them. In this study, the flow pattern identification for the experimental data was based on the procedures suggested by Kim and Ghajar, 2002, and visual observations deemed appropriate. All observations for the flow pattern judgments were made at two locations, just before the test section (about $L/D = 93$ in calming section, see Fig. 1) and right after the test section. Leaving the liquid flow rate fixed, flow patterns were observed for various air flow rates. The liquid flow rate was then adjusted and the process was repeated. If the observed flow patterns differed at the two locations of before and after the test section, experimental data was not taken and the flow rates of gas and liquid were readjusted for consistent flow pattern observations. Flow pattern data were obtained with the pipe at horizontal position and at 2°, 5°, and 7° upward inclined position. These experimental data were plotted and compared using their corresponding values of mass flow rates of air and water and the flow patterns. The digital images of each flow pattern at each inclination angle were also compared with each other in order to identify the inclination effect on the flow pattern. The different flow patterns depicted on Fig. 4 illustrate the capability of our experimental setup in producing multitude of flow patterns. The shaded regions represent the boundaries of these flow patterns. Also shown on Fig. 4 with symbols is the distribution of the heat transfer data that were obtained systematically in our experimental setup with the pipe in the horizontal position. As can be seen from Fig. 4, we did not collect heat transfer data at low air and water flow rate combinations (water flow rates of less than about 5 kg/min and air flow rates of less than about 0.5 kg/min). At these low water and air flow rates and heating there is a strong possibility of either dry-out or local boiling which could damage the test section. Figure 5 shows photographs of the representative flow patterns that were observed in our experimental setup with the pipe in the horizontal position and no heating (isothermal runs).

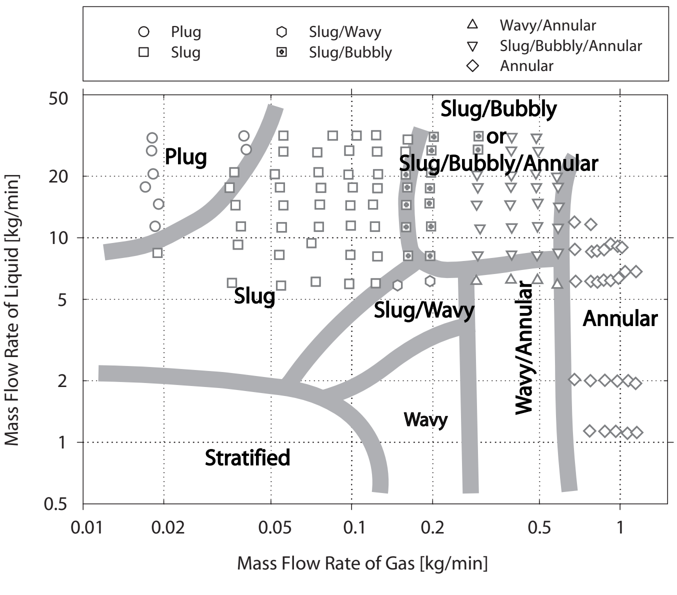


Figure 4: THE FLOW PATTERN MAP FOR HORIZONTAL FLOW

4. Heat Transfer Results

In this section we present an overview of the different trends that we have observed in the heat transfer behavior of the two-phase air-water flow in horizontal and inclined pipes for a variety of flow patterns. The two-phase heat transfer data were obtained by systematically varying the air or water flow rates and the pipe inclination angle.

Figure 6 provides an overview of the pronounced influence of the flow pattern, superficial liquid Reynolds number (water flow rate) and superficial gas Reynolds number (air flow rate) on the two-phase mean heat transfer coefficient in horizontal pipe flows. The results presented in Fig. 6(a) clearly show that two-phase mean heat transfer coefficients are strongly influenced by the liquid superficial Reynolds number (Re_{SL}). As shown

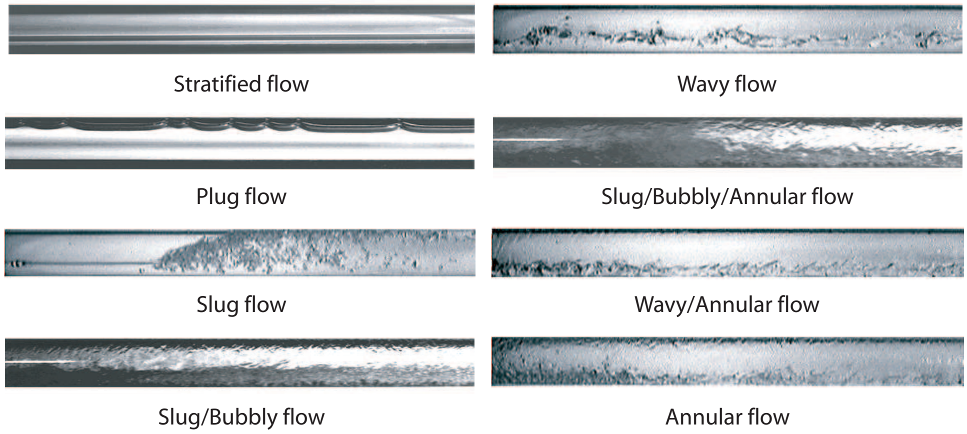


Figure 5: PHOTOGRAPHS OF FLOW PATTERNS (HORIZONTAL FLOW & ISOTHERMAL)

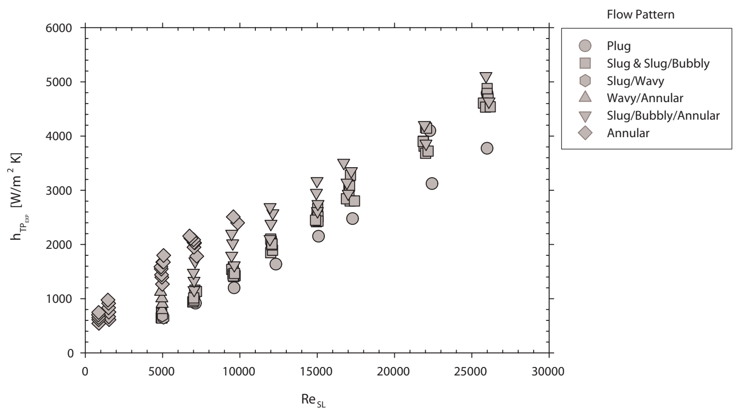
in Fig. 6(a), the heat transfer coefficient increases proportionally as Re_{SL} increases. In addition, for a fixed Re_{SL}, the two-phase mean heat transfer coefficients are also influenced by the gas superficial Reynolds number (Re_{SG}) and each flow pattern shows it's own distinguished heat transfer trend as shown in Fig. 6(b). Typically, heat transfer increases at low Re_{SG} (the regime of plug flow), and then slightly decreases at the mid range of Re_{SG} (the regime of slug and slug-type transitional flows), and again increases at the high Re_{SG} (the regime of annular flow).

To complicate matters even further, we also studied the effect of inclination angle on two-phase heat transfer in pipe flows for different flow patterns. To demonstrate the effect of inclination we duplicated the runs from the results presented in Fig. 6 by varying the inclination angle of the pipe, going from the horizontal position to each 2°, 5°, and 7° upward inclined positions. Figure 7 shows the heat transfer results for these cases. The figure shows that not only Re_{SL}, Re_{SG}, and flow pattern but also inclination affect the two-phase heat transfer. In general, the typical trend of heat transfer shown in Fig. 6(b) was also repeated in the inclined cases. However, the results clearly show that a slight change in the inclination angle has a significant effect on the two-phase heat transfer, especially in the mid range of Re_{SG}.

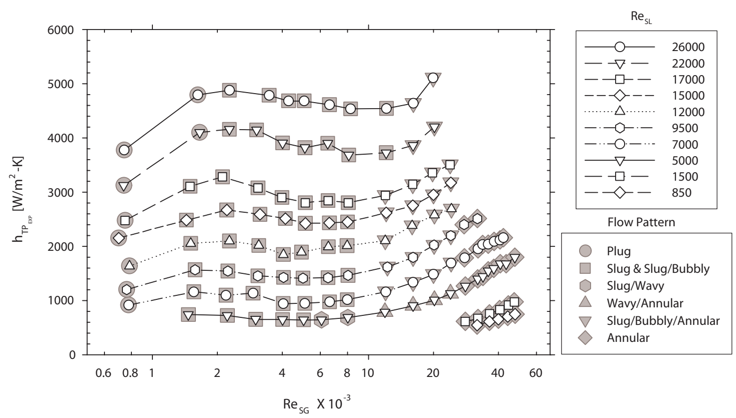
In order to conduct a more detailed comparison, the data matching the flow patterns between horizontal and inclined flows were selected and compared to see how much heat transfer increased in the inclined cases. Note that as the test section was inclined in the upward position, the flow patterns at certain cases were changed; for example, wavy-type transitional flow patterns in horizontal flow were changed to slug-type transitional flow patterns. A total of 68 horizontal flow data points were compared with their corresponding inclined flow data. The detailed results are shown in Table 1. As shown in the table, slug flow shows the biggest effect on two-phase heat transfer due to inclination. At the 5° upward inclined position, slug flow had an average increase of 45.3 % against horizontal flow. In contrast, annular flow, which is the flow mainly driven by inertia forces of gas phase, shows little effect on heat transfer due to inclination at 2° position.

Certain flow patterns, such as plug flow, slug/bubbly/annular (SBA) flow, and annular flow, showed that the heat transfer rate increased as the test setup was inclined from 0° upto 7°. However, the other flow patterns, which are slug flow and slug/bubbly (SB) flow, had the maximum increase at the 5° inclination position, and then the effect of inclination was decreased at 7°. Most of all, the effect of inclination on the heat transfer of two-phase gas-liquid flow is significant in the slug and slug/bubbly flow patterns, which had an increase in the heat transfer which was much more than the average increase of 20 % compared to the horizontal flow. These observations are well presented in Fig. 8. The comparison results presented in Table 1 and Fig. 8 indicate that the slug and slug/bubbly flows show a much more pronounced enhancement in the two-phase heat transfer at all inclination angles in comparison to the other flow patterns shown (plug flow, slug/bubbly/annular flow, annular flow). The difference between the two groups of flow patterns has to do with the degree of mixing between each phase and the inertia force carried by each phase against the buoyancy force.

For a more detailed look at the effect of inclination on heat transfer in two-phase gas-liquid flow, the increase in h_{TP} versus Re_{SL} for each flow pattern is presented in Fig. 9. As shown in the figure, except for the case of annular flow, all other flow patterns indicated that the effect of the inclination at low Re_{SL} was significantly high and then decreased with increasing Re_{SL}. In the case of slug flow, the increase in the heat transfer was as much as 94 % at Re_{SL} of around 5000 and at the 5° inclined position. However, it dropped to around 13 % at Re_{SL} of around 25,000. This drop can be expressed as a drastic change in the effect of inclination on the heat transfer. The other flow patterns, except annular flow, show a similar trend as that of slug flow. These trends



(a) h_{TP} vs. Re_{SL}



(b) h_{TP} vs. Re_{SG}

Figure 6: VARIATION OF h_{TP} OF HORIZONTAL FLOW

show that the increase of the inertia force in the fluid phases suppresses the effect of inclination.

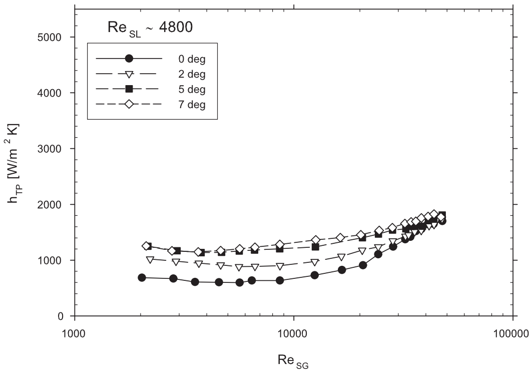
The effect of inclination on heat transfer in non-boiling two-phase gas-liquid flow has been presented in many ways to better understand the mechanisms involved. As presented in this section, heat transfer in non-boiling two-phase gas-liquid flow is influenced by each of Re_{SL}, Re_{SG}, flow pattern, and inclination angle in a very complicated way. With increasing Re_{SL}, heat transfer proportionally increased regardless of the rest of the factors. By varying Re_{SG}, the distinguished trends of heat transfer by flow patterns were observed. Furthermore, significant changes were observed in the two-phase heat transfer of air-water flow with a slight upward inclination of the pipe from the horizontal position.

5. Conclusions

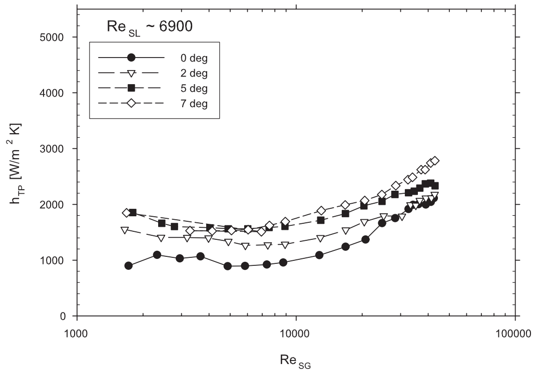
The purpose of this study was to further develop the knowledge and understanding of heat transfer in non-boiling two-phase, two-component flow. For this purpose air-water flow heat transfer experiments were conducted in a circular pipe in the horizontal and slightly upward inclined positions at 2°, 5°, and 7° under uniform wall heat flux boundary condition.

In the case of heat transfer in horizontal flow, the heat transfer rate proportionally increased as Re_{SL} increased. However, in more detail, it is observed that heat transfer shows distinguished trends depending on the flow pattern and Re_{SG}. The effect of inclination was significant on heat transfer in two-phase flow and had different characteristics depending on the flow pattern. The data showed that heat transfer coefficient increased upto around 90 % for slug flow at 5° inclined position and at low Re_{SL} range. However, the effect of inclination was quickly diminished as Re_{SL} increased. In contrast, annular flow showed little effect on heat transfer due to inclination at lower inclination angle and Re_{SL}. However, annular flow showed that the heat transfer rate increased with increasing inclination angle and Re_{SL} compared to the horizontal flow.

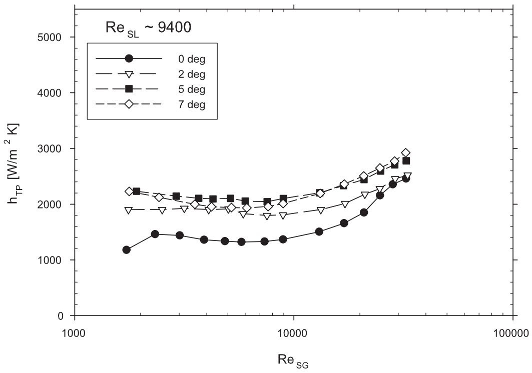
This study will ultimately lead to the development of a general overall heat transfer coefficient correlation for gas-liquid two-phase flow regardless of flow orientation. On the way to proceed to our ultimate goal, the



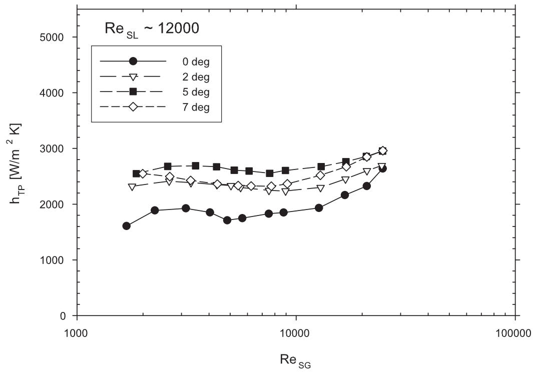
(a) $\mathrm{Re}_{SL} \approx 4800$



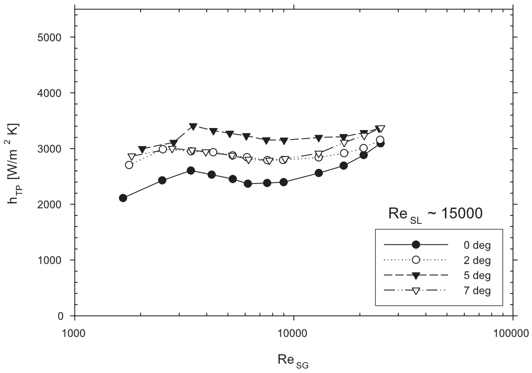
(b) $\mathrm{Re}_{SL} \approx 6900$



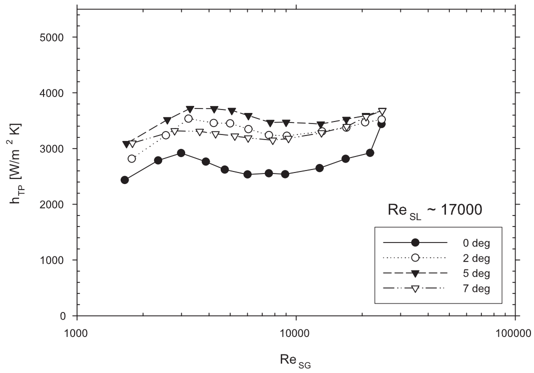
(c) $\mathrm{Re}_{SL} \approx 9400$



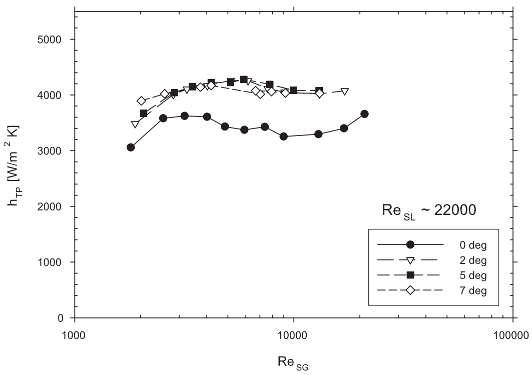
(d) $\mathrm{Re}_{SL} \approx 12000$



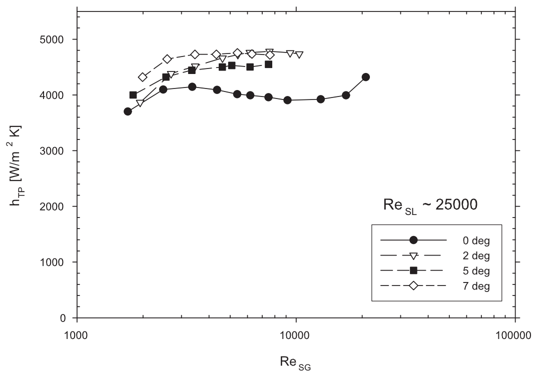
(e) $\mathrm{Re}_{SL} \approx 15000$



(f) $\mathrm{Re}_{SL} \approx 17000$



(g) $\mathrm{Re}_{SL} \approx 22000$



(h) $\mathrm{Re}_{SL} \approx 25000$

Figure 7: INCLINATION EFFECTS ON h_{TP}

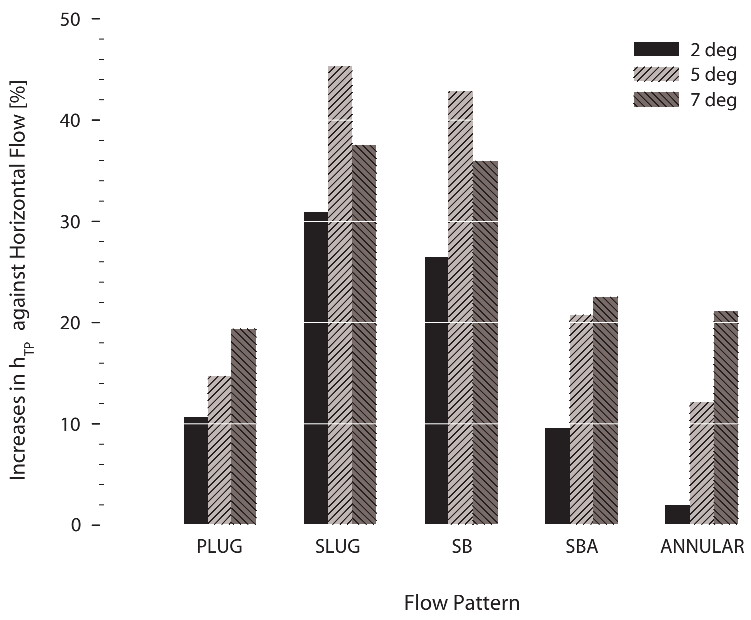
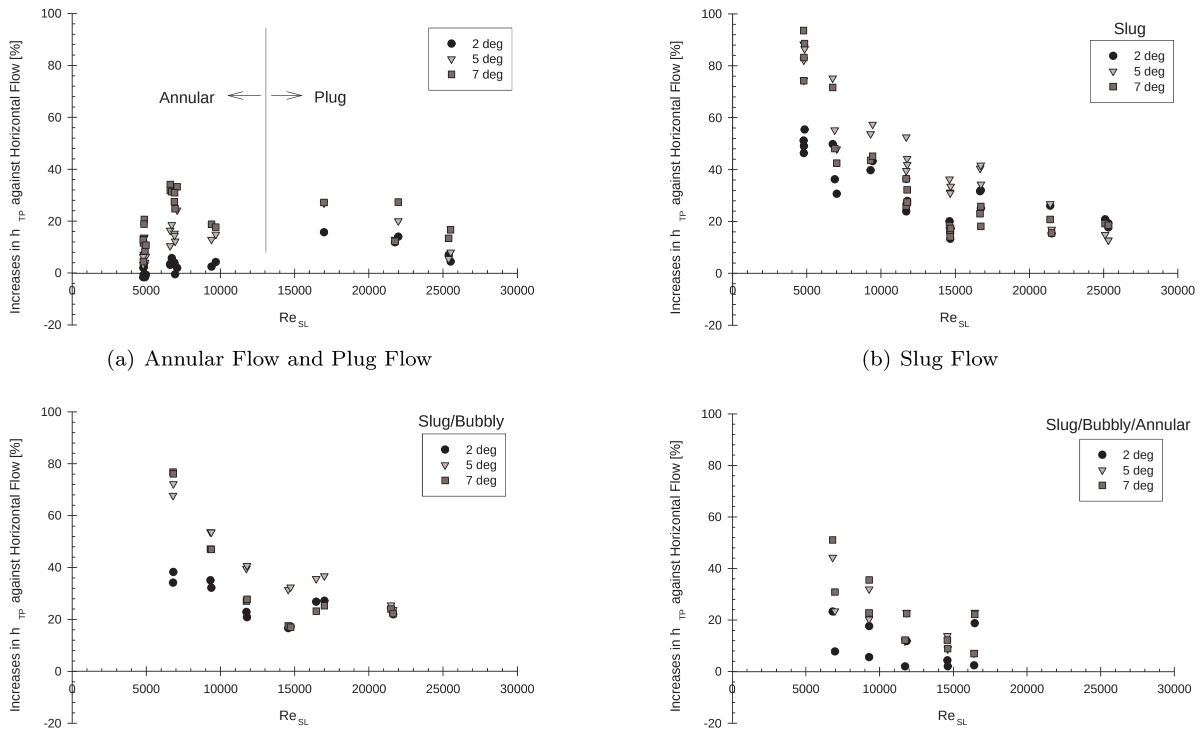


Figure 8: INCREASES OF h_{TP} AGAINST HORIZONTAL FLOW



(a) Annular Flow and Plug Flow

(b) Slug Flow

(c) Slug/Bubbly Flow

(d) Slug/Bubbly/Annular Flow

Figure 9: INCREASES OF h_{TP} AGAINST HORIZONTAL FLOW BY FLOW PATTERN

Table 1: INCREASES OF h_{TP} AGAINST HORIZONTAL FLOW

Pattern (No. Data)		**Horizontal**			**2°**	**5°**		**7°**		
		Re_{SL}	Re_{SG}	h_{TP} [W/m^2-K]	0° to 2° [%]	0° to 5° [%]	2° to 5° [%]	0° to 7° [%]	2° to 7° [%]	5° to 7° [%]
Plug (5)	MIN	16979	1651	2435	4.4	5.5	−1.4	12.3	0.4	−0.5
	MAX	25510	2535	4096	15.7	26.9	9.7	27.3	11.8	8.0
	AVG	-	-	-	10.6	14.7	3.6	19.4	7.9	4.2
Slug (25)	MIN	4777	2026	605	13.3	12.8	−5.4	14.0	−6.6	−13.1
	MAX	25321	7479	4013	55.4	88.4	24.6	93.6	28.1	5.1
	AVG	-	-	-	30.8	45.3	10.6	37.5	4.4	−5.5
SB[a] (12)	MIN	6801	7346	921	16.6	23.7	0.9	17.0	−2.8	−11.6
	MAX	21640	12942	3295	38.3	72.2	25.0	76.8	31.8	5.4
	AVG	-	-	-	26.5	42.8	12.7	35.9	7.0	−5.1
SBA[b] (10)	MIN	6810	20723	1369	2.0	7.1	3.5	7.0	2.9	−1.5
	MAX	16453	24879	3439	23.3	44.3	17.0	51.1	22.5	6.0
	AVG	-	-	-	9.6	20.7	10.1	22.5	11.7	1.3
Annular (16)	MIN	4793	28281	1374	−1.6	3.2	4.2	4.4	2.3	−1.8
	MAX	9678	47578	2461	5.7	24.2	21.9	34.1	30.7	19.3
	AVG	-	-	-	2.0	12.1	9.9	21.1	18.7	7.9

[a]Slug/Bubbly
[b]Slug/Bubbly/Annular

data collected in this study has been successfully applied to the Kim and Ghajar, 2002's general heat transfer correlation for horizontal slug flow (Ghajar et al., 2004b) and horizontal-inclined annular flow (Ghajar et al., 2004a), respectively. However, the development of the general correlation still requires further analysis of the data and better understanding of the two-phase flow heat transfer characteristics. This work is in progress.

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