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An Experimental Study of Fluidelastic Instability and Drag Force
on a Tube in Two-Phase Cross Flow

A dissertation submitted in partial satisfaction of the
requirements for the degree Doctor of Philosophy
in Mechanical Engineering

by

Youngcheol Joo

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MASTER

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Nomenclature

A	flow area (m^2) or amplitude
C	coefficient or constant
c	damping coefficient
D	tube diameter (m)
F	force (N)
f	tube frequency (Hz)
g	gravitational acceleration
Gr	Grashof number
h	fluid level height (m)
I	integral under the curve
K	instability constant in Connors' equation
k	spring constant
m	mass per unit length (kg/m)
P	tube pitch (m) or pressure (N/m^2)
P^*	dynamic pressure (N/m^2)
Q	flow rate (m^3/s)
Re	Reynolds number
V	flow velocity (m/s)
x	tube displacement (m)
$\dot{x}$	tube velocity (m/s)
$\ddot{x}$	tube acceleration (m/s^2)

Greek Letters

- α void fraction
- Δ the difference in hydrostatic head of water from that at the lower stagnation point (mm) or the difference
- ζ damping ratio
- θ the angular position measured from the lower stagnation point of the tube
- ν kinematic viscosity (m^2/s)
- ρ density of a fluid (kg/m^3)

Subscripts

- add* added
- D* drag
- e* effective
- fluid* fluid
- g* gas phase (air)
- i* *i*th peak
- L* lift
- l* liquid phase (water)
- n* natural
- r* reference gap flow parameter
- rel* relative
- s* structure
- t* total
- tube* tube
- turb* turbulence

0,1,2 constants

1 ϕ single phase

2 ϕ two-phase

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ABSTRACT OF THE DISSERTATION

An Experimental Study of Fluidelastic Instability and Drag Force on a Tube in Two-Phase Cross Flow

by

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Two-phase cross flow over heat exchanger tubes creates vibrations which contribute greatly to the wear on the tubes. Fluidelastic instability is a major mechanism by which tubes can fail. In this work, the fluidelastic instability of a tube placed in an array subjected to two-phase cross flow has been studied. For the determination of fluidelastic instability, a triangular tube array was used. The tubes were made of acrylic and were 2.2 cm or 2.37 cm in diameter and 20 cm in length. Eighteen tubes and 4 half tubes formed 5 rows with a pitch to diameter ratio of 1.4. All of the tubes except the test tube were rigidly supported at the test section wall. The test tube was flexibly supported with two cantilever beams. By installing cantilever beams horizontally and vertically, drag and lift direction tube vibration were studied. Parameters of tube mass, structural stiffness, natural frequency, and pitch to diameter ratio were varied. The drag coefficients on

a rigidly held tube in an array subjected to two-phase cross flow were measured. The tube in an array was located at displaced positions as well as at the normal position in order to study the variation of fluid force as the tube vibrates. In the experiments, gap Reynolds numbers up to 1×10^5 were obtained, while void fraction was varied from zero to 0.5.

The drag coefficients in two-phase flow are much higher than those in single phase flow. The ratio of two-phase to single phase drag coefficient decreases as Reynolds number increases. The drag coefficient on a tube in an array increases as the tube is displaced in the direction of flow. The drag coefficient increases rapidly when the tube is displaced more than a certain critical distance. The fluidelastic instability occurs when the mean displacement of a tube exceeds the critical distance. The total damping ratio and the effective spring constant were measured by experiments and were predicted by calculating fluid force components proportional to tube velocity and tube displacement. The total damping ratio increases with the increase of liquid gap velocity in single phase flow and with the increase of void fraction in two-phase flow. However, the effective spring constant decreases rapidly with the increase of liquid gap velocity or void fraction just prior to the onset of instability. It is mainly due to the rapid increase of drag force in the displaced region prior to the onset of instability. The fluidelastic instability is generated by rapid decrease of the effective spring constant because the amplitude of tube vibration is inversely proportional to the spring constant.

Chapter 1

INTRODUCTION

1.1 Background

Two-phase cross flow over tubes in many shell-and-tube heat exchangers, steam generators, condensers, and other heat exchanger equipment, causes flow-induced vibrations. These flow-induced vibrations have been recognized as a major factor in the wear and eventual failure of the tubes in the heat exchangers. Tube failures are described by three modes (figure 1.1): (1) if the stress levels in the tube root are above the fatigue limit, a circumferential crack will grow about the tube; (2) large amplitude vibrations result in clashing of adjacent tubes at midspan, and this will cause wear flats on neighboring tubes; (3) if there is clearance between a tube and its support, large amplitude tube motion can wear a groove in the tube at a support (Blevins, 1990).

Several mechanisms which induce vibrations have been identified, including turbulence, vortex shedding, and fluidelastic instability. Among them the fluidelastic instability induces the largest vibration amplitude. The second and the third tube failure modes can be attributed to this mechanism. When a tube array is subjected to cross flow, the RMS amplitude or the maximum amplitude of a tube increases slowly with flow velocity, but at a certain flow velocity, it shows a rapid increase. The flow velocity at which a large increase in the amplitude occurs is defined as the critical velocity for the fluidelastic instability.

Figure 1.1 Tube support and failure modes: (a) circumferential crack at tube sheet; (b) flat worn at midspan by clashing; (c) groove worn at support

1.1.1 Fluidelastic Instability in Two-Phase Cross Flow

Connors (1970)' paper is one of the well documented studies of flow induced vibrations in single phase cross flows. He developed a stability criterion to predict the onset of instability based on the displacement mechanism (section 1.1.2). The critical velocity above which large amplitude vibrations occurred was given by

$$\frac{V_r}{f_n D} = K \left(\frac{2\pi\zeta m}{\rho D^2} \right)^{1/2} \quad (1.1)$$

where V_r is the critical gap velocity, f_n is the natural frequency of the cylinder, D is the diameter of the cylinder, K is an instability constant, ζ is the cylinder damping ratio in still fluid, ρ is the fluid density, and m is the mass per unit length of the tube, including added mass. His stability criterion was then verified by dynamic stability tests on flexibly mounted tubes. The experiments were carried out in a 30 HP open circuit wind tunnel with a 20 cm by 20 cm test section. The tubes were 20 cm in length and 2.54 cm in diameter. The tubes themselves were rigid and the desired flexibility and damping were provided by an external spring mounting. The tubes were designed to have resonant frequencies in the 10–40 Hz or 200 Hz range. The vibrational amplitude and forces were measured with strain gages which were attached at each end of the model tube assembly. The onset of vibrational instability of the tube was determined from the maximum of the vibration amplitude–flow velocity curve. The experiments were conducted with both single and multiple row tube configurations. From his experimental data, he recommended a value of 9.9 for the instability constant K .

After Connors developed the criterion for the fluidelastic instability, many experiments have been performed to obtain the instability constant K in single phase flow. However, experimental data for the critical flow velocity obtained in various studies are not in agreement, and different values of instability constant are suggested. Chen (1987) has given several reasons of the disagreement in various studies. The major reason is that different parameters were used by different investigators; some used in-vacuum parameters, some used in-fluid parameters, and some used in-flow parameters. Even with the same experimental data, the instability constant would be different by using two different sets of parameters. He has argued that for the fluidelastic instability, the velocity mechanism (section 1.1.2) as well as the displacement mechanism should be considered. Moreover, he pointed that the critical flow velocities were not determined with the same method in various studies. Chen has also noted that because critical flow velocities of tube arrays depend on tube arrangement, spacing, and other parameters, it is difficult to develop a universal stability criterion applicable to all cases.

Studies of fluidelastic instability of a tube bundle under two-phase cross flow are very limited. Pettigrew et al. (1988b)'s paper is one of the key papers in this field. In their study, normal triangular, parallel triangular and normal square arrays (8 rows by 6 columns) were formed with pitch to diameter ratios of 1.47 and 1.32. Half tubes were installed on each side of the tube bundle to minimize wall effects. Stainless steel tubes—60 cm long, 1.3 cm in diameter and 1.07 mm in wall thickness were used. They tested bundles with all flexible tubes and bundles with one flexible tube surrounded by rigid tubes. The flexible tubes were cantilevered

and had a natural frequency in air of 33 Hz. They made rigidly held tubes by supporting the free end of the cantilevered tubes to the end cover. This type of support increased the natural frequency of the tubes to 160 Hz. The experimental apparatus used by Pettigrew et al. was capable of a water flow rate up to 0.3 m³/s and an air flow rate up to 0.5 m³/s. The tube vibration response was measured with strain gages installed inside the tube near the supported end of flexible tubes. Two pairs of diametrically opposed strain gages were installed in each tube 90° from each other to measure vibration in the flow direction (drag) and in the direction normal to the flow (lift). The onset of instability was determined from the RMS amplitude of the vibration-flow rate curve. Experiments were performed at all void fractions, but mostly at high values (0.75—0.99). They obtained the two-phase fluid properties such as density, mass flux and void fraction by assuming the flow to be homogeneous. According to this assumption, the average void fraction was calculated as

$$\alpha = \frac{Q_g}{Q_g + Q_l} \quad (1.2)$$

where α is a void fraction and Q_g and Q_l are gas and liquid volume flow rates respectively. However, the correct definition of average void fraction should have been

$$\alpha = \frac{A_g}{A_g + A_l} \quad (1.3)$$

where A_g and A_l are areas of gas and liquid respectively. The value obtained from equation (1.3) is different than that from equation (1.2) because air velocity is

larger than water velocity. For a more accurate analysis, measured void fraction of a water-air mixture should be used or should be calculated using well tested correlations. In their figures the instability threshold points for the all flexible tube bundle were much distinct than those for a flexible tube in a rigid tube array. They found that the critical flow velocity was significantly lower for the all-flexible tube bundles and the amplitude of lift direction vibration was larger than that in the drag direction vibration at the onset of instability. They observed that the tube response spectrum at instability showed a narrower and higher peak than that in stable mode of operation. For the Connors' criterion, they used the damping ratio obtained in rigid tube bundles at roughly half the mass flux for the instability. A value of 4 for the instability constant K was recommended.

Remy (1982) measured the vibration response of a tube in tube bundles in single phase flow and air-water two-phase flow. One hundred tubes of diameter 10 mm, 1 mm thick and 1 m long were positioned in a square array with a pitch to diameter ratio of 1.44. One end of the tubes was expanded, while the other was held in a tubular plate by means of a sliding fixture. The natural frequency of the tube was 56.6 Hz in air and 49.7 Hz in water. Inside the tube, a biaxial accelerometer was installed with a glue or clips. Inside the non-instrumented tubes, a lead mass was fitted to compensate for the accelerometer mass. The onset of instability was determined from the RMS amplitude of the acceleration-flow rate curve. The maximum water flow rate was 0.014 m³/s, and the air flow rate could reach 0.3 m³/s. Void fraction was measured with an optical fiber probe having a support diameter of 3 mm, allowing it to be inserted into the bundle. The

probe delivered 5 volts if it was in contact with air and zero volts if it was in water. Void fraction was determined by taking the voltage time average. He observed that the excitation due to two-phase flow was greater than that due to single phase flow. However, in two-phase flow, there was no threshold for the rapid increase of excitation as observed in the single phase tests. He recommended grouping together the fluidelastic instability constant and damping in order to avoid the issue of damping. For single phase flow of water, Remy recommended a value of 4 for the instability constant K and a value of 1.17 for $K\sqrt{2\pi\zeta}$. However, he could not recommend any values for two-phase flow.

Heilker and Vincent (1981) measured instability and damping of tube bundles in single and two-phase flows. Tubes 0.91 m in length were supported in a horizontal position with O'rings at each end in removable end plates, the flow being upward across the tube bank. Tubes of 19 mm and 22 mm diameter, having a pitch of 25.4 mm and 30 mm respectively were employed. They were positioned in triangular and square arrays. Selected tubes were instrumented internally to measure the mid-span vibration motion in two directions—with flow (drag direction) and transverse to the flow (lift direction). Biaxial piezoelectric accelerometer and semi-conductor strain gages were mounted inside the tubes to determine midspan deflections under flow conditions and to create desired natural frequency. The instability of the test tube was determined from the RMS midspan deflection–fluid pressure ($\rho V_r^2/(2g)$) curve. The maximum water flow rate was 0.32 m³/s. The two-phase experiments were conducted at void fractions higher than 0.72. They observed that the damping at low flow rate showed values similar to those in a

stationary fluid, but as the flow rate was increased, the damping value decreased both for single phase and two-phase flows. They did not specify the damping values that were used in the calculation of Connors' criterion, but it is believed that they used the damping values at onset of instability. They recommended a value of 5 for the instability constant K .

Axisa et al. (1985) measured damping and instability of tube bundles in steam-water two-phase flow. Inconel 600 tubes with 19.05 mm diameter, 1.09 mm thickness and 1.19 m in length were used. They were positioned in a 10×6 square array, normal triangular array and parallel triangular array with a pitch to diameter ratio of 1.44. One end of the tubes was clamped to a thick support plate. The other end was clamped to another plate through sleeves. The sleeve position was adjusted to tune the first mode of vibration of every tube to the same frequency—about 74 Hz in air. Vibration sensors consisted of biaxial piezoelectric accelerometers and strain gages mounted in pairs to measure motions in the lift and drag directions. Accelerometers were mounted inside tubes at about third of the length and strain gages were welded inside tubes, close to the clamped ends. The instability was determined from the maximum amplitude-gap flow velocity curve. Steam was produced in an experimental nuclear reactor and flow rates up to 28 kg/s at 25 bar and 210 °C, and water flow rates up to 670 kg/s were obtained. Experiments were performed at void fractions higher than 0.85. They observed that, at instability, lift direction vibration response was larger than that in drag direction. The normal square and the parallel triangular bundles exhibited similar behavior in both turbulent excitation response and fluidelastic instability. For

Connors' criterion, damping ratios were calculated with the half-power bandwidth method, but they did not specify which flow rates were used for this calculation. Axisa et al. recommended a fluidelastic instability constant between 4 and 5.

Marn and Catton (1992) numerically studied the fluidelastic instability of a flexible tube in a rigid tube array subjected to single phase flow. They calculated the fluid force components proportional to tube acceleration, velocity and displacement. Each component was related to the structural mass, damping and stiffness, respectively. They considered two criteria for the instability: dynamic instability and divergence. When the total damping (structural damping plus fluid force component proportional to tube velocity) became negative, the system would react to applied pressure so forcefully that it would actually overshoot the stable position on the other side. This mode was named as the dynamic instability. When the total stiffness (structural stiffness plus fluid force component proportional to tube displacement) became negative, the system became statically unstable (divergence). In their calculation, the dynamic instability occurred at lower flow velocity than that for divergence.

1.1.2 Mechanisms of Fluidelastic Instability

Two mechanisms of fluidelastic instability have generally been proposed in the literature: the displacement mechanism and the velocity mechanism. The displacement mechanism explains the fluidelastic instability of an all flexible tube bundle. If a tube in an array is slightly displaced, the flow pattern changes and the steady fluid force on the tube changes. The fluid flow pattern in a tube array

is a function of the position of the tubes relative to each other. When the tubes vibrate, they tend to vibrate in oval orbits somewhat synchronized with neighboring tubes. As the flow velocity increases, the fluid force can reduce total damping. When the total damping (including structural damping) becomes negative, the tubes become unstable. The weak point of the displacement mechanism is that this mechanism cannot explain the instability of a flexible tube in a bundle of rigid tubes, which is observed in experiments.

The velocity mechanism assumes that there exists a small but finite time lag between changes in tube position and fluid force. Because of the time lag, the fluid force has a component proportional to tube velocity, which reduces the total damping of a tube. When the total damping becomes negative, the tubes become unstable (Blevins, 1990). This mechanism can explain the instability of a flexible tube in a bundle of rigid tubes.

Both of these mechanisms assume that the fluidelastic instability occurs when the total damping ratio becomes negative. Only few experimental investigations have documented the decrease of total damping as the flow rate approaches the critical flow rate. The study of Weaver and El-Kashlan (1981) found that total damping approaches zero at the onset of instability in single phase flow of air, while other studies (Pettigrew et al. 1988b and Heilker and Vincent 1981) did not observe such results in their experiments conducted in two-phase flow. None of these studies provide a clear insight regarding the variation of total damping near the onset of instability.

Another commonly accepted explanation of the cause of the fluidelastic instability is that the instability occurs when the energy input by the fluid force exceeds the energy expended in damping. This explanation implies that the total damping ratio does not necessarily have to be negative to cause the instability. As long as the energy input exceeds the energy loss, the instability will occur. Displacement and velocity mechanisms contradict this explanation.

1.1.3 Damping Ratio

Damping ratio is one of the important parameters needed for the investigation of the mechanism of fluidelastic instability. There are basically two methods which are generally used to calculate damping: the free decay method and the half-power bandwidth method (figure 1.2). If the structure is excited to an amplitude A_0 and then the excitation is removed, the amplitude decays slowly in time. The damping ratio can be calculated from the equation

$$\zeta = \frac{1}{2\pi} \ln \frac{A_i}{A_{i+1}} \quad (1.4)$$

where A_i and A_{i+1} are the amplitudes of the i th peak and $i+1$ th peak respectively. This is the free decay method. The response of a single-degree-of-freedom structure to steady-state sinusoidal excitation is sinusoidal at the excitation frequency. If the response amplitude is plotted as a function of frequency, it will show one peak. Damping ratio can then be calculated from the equation

$$\zeta = \frac{\Delta f}{2f_n} \quad (1.5)$$

(a) free decay method

(b) half power bandwidth method

Figure 1.2 Methods of measuring the damping ratio

where Δf is the bandwidth of the frequency response at $1/\sqrt{2}$ times the peak amplitude and f_n is the natural frequency which is almost the same as the first dominant frequency. This is the half-power bandwidth method. Usually, damping measurements have been made in stationary fluid using the free decay method and in flowing fluid using the half-power bandwidth method.

Carlucci and Brown (1983) measured the damping ratio of a cylinder exposed to air-water two-phase axial flow. A cylinder of 25.4 mm in diameter and 1.24 m in length was mounted centrally within a plastic tube of diameter 40 mm. Water or air-water two-phase mixture flowed vertically upward. Usually, the half-power bandwidth method was used to determine the damping ratio. The excitation of the cylinder was caused by flow-induced vibration. Sometimes the cylinder was excited randomly by a servo-controlled electrodynamic shaker which was connected to the midpoint of the cylinder. When flow excitation was not sufficient to produce a reasonable signal, the cylinder was impacted periodically to yield a well-defined time decay signal. The damping ratios from both methods were almost similar. They found that the two-phase damping ratio increased with increasing void fraction up to a void fraction of about 0.3, reached a maximum value at void fractions between 0.3 to 0.6, and decreased as the void fraction increased beyond 0.6. They divided the total damping ratio into the structural, viscous, flow-dependent and two-phase damping components.

Pettigrew et al. (1988a) measured the damping ratio of a tube in tube bundles subjected to two-phase cross flow. The same setup as used for the instability experiments (Pettigrew et al., 1988b) was used. They employed the half-power

bandwidth method to obtain damping values in two-phase flows. The least square regression analysis technique was used to fit the best single degree of freedom frequency response curve to the vibration response spectrum. Damping values were obtained from such curve fits. Pettigrew et al. observed that more reliable and lower damping values were obtained with one flexible tube surrounded by rigid tubes than with all flexible tubes. They found that the effect of void fraction was dominant and the effect of pitch to diameter ratio and of mass flux were generally weak. The damping was maximum at void fractions between 0.4 and 0.8. In their measurements, the drag direction damping was larger than the lift direction damping.

Hara (1988) measured the damping ratio of a rod in two-phase stagnant mixture. The test section consisted of a circular rod fixed vertically and centrally within a cylindrical container. The circular rod was 500 mm in length and fixed rigidly at the bottom end and free at the top. The test rods were 5 mm and 20 mm in diameter and containers of diameter 20 mm to 200 mm were used to change the gap between the rod and the container. The container was filled with water to a level h_l . Air was injected from the bottom of the test setup to generate air bubbles of a mean diameter of 5 mm. Void fraction was calculated from

$$\alpha = \frac{h_{2\phi} - h_l}{h_{2\phi}} \quad (1.6)$$

where $h_{2\phi}$ is the height of the water and air mixture. The damping ratio was evaluated by the half-power bandwidth method. The structural damping ratio, ζ_s , was evaluated from the experiments done in air. Subtracting the structural

damping ratio from the total damping ratio, ζ_t , the two-phase fluid damping ratio, $\zeta_{2\phi}$, was obtained as

$$\zeta_{2\phi} = \zeta_t - \zeta_s \quad (1.7)$$

He found that two-phase fluid damping increased with void fraction up to values in the range from 0.3 to 0.6. The small size bubbles in water generated higher damping than did the large size bubbles.

Blevins (1990) developed a model for the fluid damping ratio of a cylinder in a cross flow. He assumed the tube to vibrate in a harmonic motion. The vibration amplitude was assumed to be small and the tube was assumed to oscillate at its natural frequency. When the structural velocity ($\dot{x}$) was small relative to the mean fluid velocity V , an expression for the fluid damping in a cross flow was obtained as

$$\zeta_{fluid} = \frac{1}{4\pi} \frac{V}{f_n D} \frac{\rho D^2}{m} C_D \quad (1.8)$$

where ρ is the density of the fluid, m is the total mass of the tube, f_n is the natural frequency of the tube in the fluid, and C_D is the drag coefficient of the tube in cross flow.

1.1.4 Drag Coefficient and Void Fraction

Drag coefficient is one of the important parameters needed to understand the mechanism of fluidelastic instability of tubes subjected to cross flow. Little work has been done on the drag coefficient of a blunt body subjected to two-phase flow. Yokosawa et al. (1986) measured the drag on a single tube under two-phase cross

flow in the Reynolds number range of 4,000 to 300,000 and for low void fractions (0 to 0.1). In the experiments cylinders of 10, 20, 30 or 40 mm in diameter and 58 mm in length were used. The cylinders were made of brass and were held rigidly or flexibly. For the rigidly mounted tubes, they determined the drag force from pressure distributions on the tubes. For the flexibly mounted tubes, cantilever beam supports which were instrumented with strain gages were used. The drag coefficients from both methods were almost similar. They found that the drag coefficient decreased with increasing void fraction for two-phase Reynolds numbers sufficiently below the single phase critical Reynolds number. But, for Reynolds numbers above the critical value, the drag coefficient was found to gradually increase with increasing void fraction, although it remained significantly less than the value for subcritical, single phase flow. However, they did not measure how the drag coefficient varied for void fractions higher than 0.1. Also, their tube length to diameter ratios (1.45 and 2.9) were relatively smaller than those used in other studies (present study: 9.1).

In the first part of the same study (Inoue et al., 1986), the void fraction distribution around the tube was measured with an electrical impedance probe inserted in the flow in the Reynolds number range of 5,000 to 80,000. They found that high void fraction regions with the local void fraction about 3 to 4 times higher than the freestream value were produced near the separation point. As the mean velocity in the main flow increased, this peak void fraction also increased and the location of the peak void fraction came close to the cylinder. However, they did not measure void fractions upstream of a tube. From photographs, they noted

liquid-rich layers in the front and in the rear of the tube. As the mean velocity increased, the liquid layer became thicker in the front of the tube and thinner in the rear of the tube.

Pettigrew et al. (1988b) studied the flow pattern through the tube bundles also. By taking photographs of the flow at the end of the bundle, they observed that two-phase flow tended to stream through flow paths between the tubes. Relatively stagnant zones of mostly liquid were observed immediately upstream and downstream of each tube. These zones appeared to be symmetrical about the equator of the tube. Considering the non-stationary nature of two-phase mixtures, the boundaries of the flow paths were reasonably well defined.

Hara (1984) measured the vibration of a single tube which was installed horizontally in two-phase cross flow. The solid test tube, 30 mm in diameter and 60 mm in length, was held flexibly by flat-plate springs. The natural frequency of the tube-spring system was 3.4 Hz in still water. He observed that for low liquid Reynolds numbers (lower than 12,100), tube RMS vibrational acceleration increased with void fraction. For higher Reynolds numbers (19,000 and 20,700), the vibrational acceleration was very large in single phase flow, but for a void fractions up to 0.07 it decreased with the injection of air. With further increase of void fraction, the vibrational acceleration increased again. For Reynolds numbers between 12,100 and 19,000, the vibrational acceleration increased or remained constant from the single phase value up to a void fraction of 0.01, and then decreased with increasing void fraction up to 0.07. With further increase of void fraction, the vibrational acceleration increased again in a manner similar to that

for higher Reynolds numbers. He found that the reduction of vibrational acceleration occurred when tube vibrations were resonant with vortex shedding. He thought that the major mechanisms for vibration reduction are: (1) the detuning of synchronization between cylinder oscillation and vortex shedding through strong disturbance of the cylinder boundary layer caused by injection of a small amount of air bubbles in the flow and (2) randomness in two-phase flow induced unsteady fluid forces.

Most previous studies showed similar results even though different tube arrangements with different tube-supporting techniques and apparatuses were used. The excitation in two-phase flow is larger than that in single phase flow, but the onset of instability is less clear in two-phase flow. Lift direction excitation is much larger than the drag direction excitation at the flow rate of instability. The damping ratio reaches a maximum value at a void fraction of 0.3 to 0.6. The values ranging from 4.0 to 9.9 have been recommended for the fluidelastic instability constant, K , of Connors' criterion. However damping ratios obtained at different flow conditions were applied in the calculation of Connors' constant. The mechanisms proposed for the fluidelastic instability assume that damping is zero at the onset of instability, but most of the experimental studies do not confirm such results.

1.2 Objective of the Present Study

Mass, damping, spring constant and fluid force are four major components of interest in the study of tube vibrations induced by a fluid flowing across a tube bundle. In order to understand the fluidelastic instability of tubes which are subjected to two-phase cross flow, the investigation and understanding of these four components is necessary. These four components are influenced by several other parameters; such as tube diameter, tube mass, pitch to diameter ratio, natural frequency, flow rate, void fraction, types of tube array, methods in which tubes are supported, and whether the tubes surrounding the test tube are supported rigidly or flexibly, etc.. Most of the previous studies concentrated on how these parameters affected the critical flow conditions for the fluidelastic instability and on obtaining the instability map. However, the past research did not focus on how these parameters affected the four components of tube vibration when the fluidelastic instability occurred. As such the mechanism of the instability is not clearly explained yet. The objective of the present study is to explain the mechanism of fluidelastic instability in tubes subjected to single and two-phase cross flow.

To investigate the damping and spring constant of a tube system at onset of instability, the amplitude of tube vibration, the damping ratio, the first dominant frequency, and tube mean displacement will be measured. The parameters of tube mass, structural spring constant, and pitch to diameter ratio will be varied.

In a vibrating tube system, the fluid force is coupled to the tube through the

damping and the spring constant. It is difficult to measure fluid force directly in such a system. Therefore, the drag force, which is the dominant component of the fluid force, on a rigidly supported tube will be studied. The drag coefficient of a single tube and a tube in an array which are subjected to two-phase cross flow will be measured. The tube in an array will be located at displaced positions as well as at the normal position in order to study the variation of fluid force as the tube vibrates. The fluid force component proportional to the tube velocity, which is coupled with the damping ratio, and the component proportional to the tube displacement, which is coupled with the spring constant, will be determined.

The total damping ratio and effective spring constant of a vibrating tube system can be obtained by subtracting the fluid force components from the structural damping ratio and the structural spring constant as well as from the measurements of frequency and response spectrum. By investigating the total damping ratio and the effective spring constant at the onset of instability, an effort will be made to explain the mechanism of the fluidelastic instability.

Chapter 2

EXPERIMENTAL APPARATUS AND PROCEDURE

2.1 Two-Phase Flow Loop

A schematic diagram of the flow loop for air-water two-phase flow is shown in figure 2.1. It is primarily formed with 11.4 cm diameter PVC pipe. A $0.063 \text{ m}^3/\text{s}$ (1000 GPM) capacity pump supplies water from the secondary reservoir to the pressure vessel. The capacities of the secondary reservoir and the pressure vessel are 3.9 m^3 and 2.5 m^3 respectively. A by-pass line is connected from the main pipe to the secondary reservoir. A butterfly valve is installed in the main pipe and a gate valve is installed in the by-pass line. Water flow rate from the pump to the pressure vessel is controlled with a combination of these valves. Free surface of water in the pressure vessel is pressurized with utility air. A pressure regulator keeps the air pressure constant. This in turn allows a constant flow rate to be maintained from the vessel to the remaining sections of the loop. The pressure in the pressure vessel is regulated from 0.14 MPa at low flow rate (about $0.012 \text{ m}^3/\text{s}$) to 0.19 MPa at high flow rate (about $0.02 \text{ m}^3/\text{s}$). Water exits the pressure vessel through a globe valve and passes through a turbine flow meter. During the measurements, water flow rate is controlled with this globe valve.

The liquid stream then turns upward to flow vertically through the test section assembly. When two-phase flow is desired, utility air is introduced through one

Figure 2.1 Schematic diagram of the flow loop

or more injection holes placed upstream of the test section. Air entering an injection hole passes through a flow meter with a pressure gage. A correction to the measured air flow rate is made for the pressure being different than that at which flow meter was calibrated. Air bubbles are broken up by a series of wire meshes which homogenize the mixture before it enters the diffuser. After expansion in the diffuser, which expands at an angle of 7 degrees, the flow is stabilized in the flow developing section. It then passes through the test section, where the tube bundle is installed. The flow developing section and the test section have the same dimensions: 30 cm high with a 12.5 cm $\times$ 20 cm cross section. The flow exits through a converging section which has dimensions identical to the diffuser. The test section assembly—diffuser, flow developing section, test section, and converging section—are made with transparent half inch thick plexiglass plates. The flow exiting the test section is released to the secondary reservoir, where air is allowed to escape. The uncertainty in air and water flow rate are calculated to be $\pm 2\%$ and $\pm 0.4\%$ respectively.

2.2 The Tube Bundle and Test Tube

In the test section, acrylic tubes are installed horizontally to form a normal triangular array (figure 2.2). The tubes, 2.2 cm in diameter and 20 cm in length, are arranged with pitch to diameter ratio of 1.4. Half tubes are installed on each side of the tube bundle to minimize the wall effect. Eighteen tubes and 4 half tubes form 5 rows. All tubes are rigidly supported at the test section wall except the test tube.

Figure 2.2 A normal triangular tube array

The hollow acrylic test tube, 27 cm in length, is positioned at the center of the fourth row, and extends through holes in the test section wall. The tube is flexibly supported with two cantilever beams which are held outside of the test section wall. The holes in the test section wall are large enough to give the test tube sufficient room to vibrate. Easily removable caps cover the cantilever beams and holes to prevent leaking. The test tube can vibrate in only one direction—normal to the cantilever beams. The cantilever beams are installed horizontally when drag direction vibrations are measured. When lift direction vibrations are measured, the cantilever beams are installed vertically (figure 2.3).

The cantilever beams which support the test tube are made from spring steel plates. Spring constants can be varied by using spring steel plates of different thicknesses. The beams are 7 mm in width and 53.5 mm in length. Steel plates of three different thicknesses were used: 0.56 mm, 0.71 mm and 0.89 mm. The natural frequencies in water of these tube-cantilever beam systems are 8.5 Hz, 12.2 Hz and 17.4 Hz, respectively. In a few experiments, a solid aluminum rod was inserted in the hollow acrylic test tube in order to change the tube mass. The tube mass was changed from 0.062 kg to 0.207 kg. When the aluminum rod inserted tube is supported with 0.71 mm thickness cantilever beams, the natural frequency in water is 10.0 Hz. Uncertainty in the measured frequency is expected to be less than $\pm 3\%$.

In order to study the effect of pitch to diameter ratio, the tubes of 2.37 cm in diameter were used. By arranging the tubes with the same pitch as before (3.08 cm), pitch to diameter ratio of 1.3 was obtained. When the solid acrylic test

Figure 2.3 Test tube mounting

tube, 0.126 kg in mass, is supported with 0.71 mm thickness cantilever beams, the natural frequency in water is 10.0 Hz.

Strain gages were mounted on the top and bottom sides of each cantilever beam. They were installed 6 mm away from the beam supporting plates. The strain gage output was read from the strain gage input module of the data acquisition system. Vibration responses were calculated with a personal computer which is connected to the data acquisition system.

2.3 Fluidelastic Instability Measurements

In experiments conducted to determine onset of fluidelastic instability, water was filled in the test section assembly up to the converging section and a certain time period was allowed to still the water. The strain gage signal was measured with still water. Since the strain gage module could not be set at zero for the still water signal, this value was used to compensate the measured response during flow. Thereafter, the pump was started and the strain gage signal was measured again. Measurements were repeated with increasing water flow rate until instability occurred. Onset of instability was identified by a rapid increase in the amplitude of the strain gage signal. For two-phase flow experiments, the water flow rate was set constant at some lower flow rate than the single phase instability value. The vibration response was measured with increasing air flow rate until instability occurred. Time averaged mean value, RMS amplitude and vibration response spectrum were obtained from the strain gage voltage-time data.

The strain gages were calibrated before and after measurements. Known forces

were applied to the test tube, and strain gage signals were recorded at each time. Reproducibility of the strain gage signals by repeating an experiment was observed. A linear relationship between applied force-strain gage signal was observed. From these data, each strain gage signal was converted to the force acting on the tube.

2.4 Damping Ratio Measurements

The structural damping ratio was measured in air with free decay method. After test tube was installed on the cantilever beams, the tube was pulled upward by a string and was released suddenly. It vibrated with some damping, and the output of the strain gage signal was recorded. The damping ratio and the natural frequency in air were determined from the logarithmic decrement shape of the signal.

For the measurement of the damping ratio of a tube in stagnant two-phase fluid, water was filled in the test section assembly and air was introduced from the air injection holes at the bottom of the test section assembly. The free surface was located 50 cm above the test tube in order to minimize the turbulence and bubble accumulation effect at the water-air interface. When the damping ratio was measured with free decay method, it was hard to obtain a smooth decay shape of strain gage signal because air bubbles shake the tube too much. Therefore the half power bandwidth method was used. The tube was vibrated by the air bubbles. The response spectrum had one peak, and the damping ratio and the natural frequency were obtained from the spectrum. When the damping ratio and

natural frequency in water were measured, in order to shake the tube the possible smallest amount of air was introduced from the air injection holes. The tube vibration was very small to see with naked eyes. However, the response spectrum was very clear, and the damping ratio and the natural frequency were obtained from the spectrum.

For the measurement of the damping ratio of a tube in an array subjected to fluid flow, the half power bandwidth method was used. Damping ratios in fluid flow condition have much larger uncertainties than those in stagnant fluid conditions because the response spectrums in fluid flow are much less smooth than those in stagnant fluid. The damping ratios were also calculated theoretically using equation (1.8).

2.5 Drag Coefficient of a Rigidly and Flexibly Mounted Single Tube

The acrylic test tube of 2.2 cm in diameter was mounted horizontally in the test section. The tube was instrumented with pressure taps to measure pressure distribution and in turn the drag coefficient. The taps are located at angles of 0° , -30° , 60° , $\pm 90^\circ$, $\pm 120^\circ$, and 180° from the lower stagnation point. Each tap was 3.2 mm in diameter. A 3.2 mm outer diameter tygon tubing was inserted inside each tap. Another 1 mm inner diameter, 2 mm outer diameter teflon tubing was inserted in the tygon tube to avoid air bubbles from entering the tube during experiments. Outside the test section, the tubing is again expanded through a reducing union to a 4.2 mm inner diameter, 6.4 mm outer diameter polyethylene

tubing. Taps at the lower stagnation point were connected to each of the remaining taps through a manometer. Each line had a valve for the expulsion of any bubbles in the line. The orientation of the pressure taps is shown in figure 2.4.

The drag coefficient was calculated from the pressure distribution by using the following equation:

$$C_D = \frac{gD}{V_l^2} \left(\frac{\pi\alpha}{2} + \int_0^{2\pi} C_1 \frac{\Delta}{D} \cos\theta d\theta \right) \quad (2.1)$$

and the lift coefficient was calculated by using the equation:

$$C_L = -\frac{gD}{V_l^2} \int_0^{2\pi} C_1 \frac{\Delta}{D} \sin\theta d\theta \quad (2.2)$$

where V_l is the superficial liquid velocity at free stream, Δ is the difference in hydrostatic head of water from that at the lower stagnation point, g is the gravitational acceleration, θ is the angular position measured from the lower stagnation point, and C_1 is a constant which depends upon the angle that the manometers are inclined from the horizontal. The uncertainty in the measurement of drag and lift coefficient is calculated to be less than $\pm 10\%$. The drag coefficients were measured by varying from 430 to 21,900 the liquid Reynolds number (Re_l) based on superficial liquid velocity.

For the measurement of drag coefficient of a flexibly mounted tube, the test tube with pressure taps was mounted on the cantilever beams as described in section 2.2. The pressure distribution was measured in exactly same manner as for a rigidly mounted tube.

Single tube

Tube in an array

Figure 2.4 Pressure tap distribution

2.6 Drag Coefficient of a Rigidly and Flexibly Mounted Tube in an Array

The experiments were performed not only for a single tube but also for a tube in a triangular array and for a tube in an array but displaced from its original position. The test tube, 22 mm in diameter, had pressure taps at angles of 0° , $\pm 30^\circ$, $\pm 60^\circ$, $\pm 120^\circ$, $\pm 150^\circ$, and 180° . Both ends of test tube were inserted into tube holders, which are 27.7 mm in width and height and the same thickness with test section wall of 12.7 mm (half inch). The tube holder has a hole of 22 mm in diameter, and test tube was inserted into it. The holder, which mounts test tube, was inserted into the rectangular hole in test section wall. The test section which has tube array as shown in figure 2.2 has rectangular slots of 27.7 mm in width and 31.7 mm in height at the test tube position. A 4 mm wide gap exists in vertical direction between the slot and the tube holder. Stainless steel plates were inserted into this gap to fix the holder rigidly. The thicknesses of plates were 1 mm, 3 mm, and 4 mm. The uncertainty in plate thickness is less than 0.7%. With combination of these plates, the test tube can be located at its normal position, at the position of 3 mm higher than the normal position, and at 4 mm higher than the normal position in the array. Drag coefficients of a tube at each position were measured in the same manner as for a single tube.

For a tube in an array, superficial liquid gap velocity $V_{l,r}$ is used in equation

(2.1) instead of V_i , which is defined as

$$V_{i,r} = \frac{P}{P - D} V_i \quad (2.3)$$

where P is the pitch of tubes. In the experiment, the liquid gap Reynolds number was varied from 32,900 to 61,600 while the Reynolds number based on free stream velocity was varied from 9,400 to 17,600.

For the flexibly mounted tube in an array, pressure distribution were not taken because the displacement of tube in an array was found to affect the measured drag. However, during the experiments for the determination of fluidelastic instability (section 2.3), drag forces could be obtained from the mean value of strain gage signal.

2.7 Void Fraction Measurements

The plexiglass test section allowed the use of a gamma densitometer to measure the time-averaged void fraction at various positions. The gamma densitometer consists of a cesium-137 source and an NaI detector. The gamma beam created by the source is collimated by passing through a small opening in the source's lead housing. It then passes through the test section, and it is again collimated by passing through a small opening in the detector's lead housing. Gamma rays are absorbed in the detector, producing electronic pulses. A 1024-channel analyzer sorts the pulses into different channels, depending on the energy of each pulse, and stores them. The number of pulses is plotted versus channel number, and the integral under the curve (I) is evaluated within the region of interest amongst all

the channels. The value of I is recorded and used to calculate the void fraction using the following equation:

$$\alpha = \frac{\ln(I/I_l)}{\ln(I_g/I_l)} \quad (2.4)$$

where I_l and I_g are the integrals under the curves for water and air respectively. The uncertainty of void fraction measurement with gamma densitometer itself is $\pm 3\%$. However, when the void fraction is measured very near the tube, the uncertainty in the positioning of the beam also becomes important. For the measurement of void fraction in a tube array, the maximum value of overall uncertainty is calculated to be $\pm 20\%$. This uncertainty occurs very near the tube wall. The uncertainty of data away from the tube is much less.

The free stream void fractions in the test section were measured upstream of a tube bundle. From these data, an empirical correlation was obtained as

$$19.65(1 - \alpha)^3 \left(\frac{V_g(1 - \alpha)}{\alpha} - V_l \right)^2 - 1 = 0 \quad (2.5)$$

where V_g is the superficial gas velocity. The velocity is measured in meters per second. The free stream void fractions in a given experiment were calculated by using this correlation.

Chapter 3

RESULTS

3.1 Single Phase Flow Instability During Tube Vibration in the Drag Direction

The vibration responses of the test tube during tube vibration in the drag direction under single phase flow of water were measured. Figure 3.1 shows the RMS amplitudes of the empty test tubes which were supported with 0.56 mm and 0.71 mm thickness cantilever beams and of an aluminum rod inserted test tube which was installed with 0.71 mm thickness cantilever beams in an array of pitch to diameter ratio of 1.4. Figure 3.1 also shows the RMS amplitude of test tube which was installed with 0.71 mm thickness cantilever beams in an array of pitch to diameter ratio of 1.3. For the empty tube supported with 0.56 mm thickness cantilever beams in an array of pitch to diameter ratio of 1.4, the RMS amplitude increases slowly with the liquid gap velocity up to 1.296 m/s. At a liquid gap velocity of 1.337 m/s, the RMS amplitude increases a little faster than before. With further small increase in the liquid gap velocity (about 1%), the amplitude increases by a large amount and tube hits upper rigid tubes. Liquid gap velocity of 1.296 m/s represents the onset of instability. Onset of instability was observed at a liquid gap velocity of 2.188 m/s for the empty tube supported with 0.71 mm thickness cantilever beams, at a liquid gap velocity of 2.965 m/s for the aluminum rod inserted tube supported with 0.71 mm thickness cantilever beams, and at a

Figure 3.1 RMS amplitude change with liquid gap velocity in single phase flow

liquid gap velocity of 2.253 m/s for the tube with pitch to diameter ratio of 1.3. For the array of pitch to diameter ratio of 1.4, the critical gap velocity increases as the cantilever beam thickness increases and as the tube mass increases. The natural frequency is proportional to the square root of cantilever beam stiffness and inversely proportional to the square root of tube mass. Therefore in one case critical gap velocity increases as the natural frequency increases, and in the other case it decreases as the natural frequency increases. This phenomena implies that the stiffness of the system as well as the total mass and the natural frequency are independent parameters to affect the onset of fluidelastic instability. In Connors' criterion the conditions for the onset of instability are described with total mass and natural frequency and not stiffness of the system. For the same natural frequency of the tube system, the critical gap velocity increases as the pitch to diameter ratio increases. The effect of stiffness of the system and the pitch to diameter ratio will be discussed in section 4.5.

In figure 3.2, the time averaged mean displacements are plotted. The mean displacement is observed to increase smoothly with liquid gap velocity. However, the slope of the curve that fitted through the data for mean displacement increases rapidly just before the onset of instability.

The vibration response spectra near the onset of instability are shown in figure 3.3. The spectra are for an empty test tube held with 0.56 mm thickness cantilever beams in an array of pitch to diameter ratio of 1.4. Figure 3.3 (a) shows the response spectrum at a liquid gap velocity of 1.134 m/s, which is smaller than the critical gap velocity for the fluidelastic instability. The response spectrum shows

Figure 3.2 Mean displacement change with liquid gap velocity in single phase flow

Figure 3.3 Vibration response spectrum in single phase flow
(Empty tube mounted on 0.56 mm thickness cantilever beams)

a narrow band unique peak at a frequency of 8.1 Hz. Figure 3.3 (b) is a vibration response spectrum at a liquid gap velocity of 1.296 m/s—the critical velocity for the instability. The response spectrum shows a much broad band peak. The first dominant frequency decreases to 6.0 Hz. The liquid gap velocity corresponding to figure 3.3 (c) was 3% larger than that of figure 3.3 (b). With this small increase of liquid gap velocity, the response spectrum shows much different shape than that prior to onset of instability. It has the largest value at a frequency very close to 0 Hz.

The first dominant frequencies of vibration response are plotted as a function of liquid gap velocity in figure 3.4. The first dominant frequency decreases slowly with increase of liquid gap velocity. However, just prior to the onset of instability, it shows a rapid decrease with further increase of liquid gap velocity.

3.2 Two-Phase Flow Instability During Tube Vibration in the Drag Direction

The vibration response of the test tube in air-water two-phase flow was measured at superficial liquid gap velocities lower than the critical liquid gap velocity for the instability under single phase flow conditions. Figure 3.5 is a typical graph of RMS amplitude as a function of void fraction under two-phase flow conditions. The empty test tube was installed with 0.89 mm thickness cantilever beams in an array of pitch to diameter ratio of 1.4. The superficial liquid gap velocity was fixed at 2.8 m/s and tube response was measured as the superficial gas gap

Figure 3.4 The first dominant frequency change with liquid gap velocity in single phase flow

Figure 3.5 RMS amplitude change with void fraction in two-phase flow

velocity was increased. The RMS amplitude increases slowly with the increase of the superficial gas gap velocity or void fraction. However, at a void fraction of 0.18, RMS amplitude increases rapidly with further increase of void fraction. This is the condition for the onset of instability under two-phase flow conditions. In comparison to the single phase case (figure 3.1), the threshold point for onset of instability is not pronounced in two-phase flow. As the superficial liquid gap velocity decreased, the instability occurred at higher void fraction. However, the threshold point for the onset of instability became less pronounced as the superficial liquid gap velocity decreased or void fraction increased. When the liquid gap velocities were smaller than 1.9 m/s, it was hard to determine the threshold point for the onset of instability.

As seen in figure 3.5, a small amount of air in the flow is found to slightly decrease the RMS amplitude. With further increase in the void fraction up to 0.08, the RMS amplitude is found to return to its single phase value, and then continue to increase. At lower superficial liquid gap velocity, the same phenomena was observed, but the RMS amplitude attained its single phase value at a lower void fraction. For the superficial liquid gap velocity of 1.12 m/s, the RMS amplitude recovered to its single phase value at a void fraction of 0.02. This is in good agreement with Hara (1984)'s observation.

The determination of the first dominant frequency for two-phase flow is not as straightforward as for single phase flow, because the response spectrum curves for two-phase flow are less smoother than those for the single phase case. Figure 3.6 is a typical plot of the first dominant frequency as a function of void fraction. The

Figure 3.6 The first dominant frequency change with void fraction in two-phase flow

cantilever beam thickness for the plotted data was 0.89 mm, and the superficial liquid gap velocity was 2.8 m/s. Prior to onset of instability, the first dominant frequency decreases slowly with increase of void fraction. However, upon onset of instability a rapid decrease on frequency is observed with further increase of void fraction.

Onset of instability in two-phase flow is a function of superficial liquid gap velocity and void fraction. In figure 3.7, the flow conditions at onset of instability are plotted with the superficial gas gap velocity as the abscissa and the superficial liquid gap velocity as the ordinate. As the superficial gas gap velocity increases, instability occurs at lower superficial liquid gap velocities. The region under the curves is the stability zone, while the region above the curves represents the instability zone. The instability for liquid gap velocities higher than 2.8 m/s for the empty tube supported with 0.89 mm thickness cantilever beams in an array of pitch to diameter ratio of 1.4 could not be measured because of the limitation of the experimental setup.

3.3 Instability During Vibrations in the Lift Direction

The tube response when cantilever beams were installed vertically to allow the tube to vibrate only in the lift direction (normal to the flow) was also measured. Figures 3.8 and 3.9 show the measured RMS amplitude and the first dominant frequency respectively as a function of void fraction. Empty tube was supported with 0.89 mm thickness cantilever beams in an array of pitch to diameter ratio of 1.4. For single phase water flow, the RMS amplitude is observed to increase

Figure 3.7 Two-phase flow conditions for the onset of instability

Figure 3.8 Tube vibration in the lift direction in two-phase flow

Figure 3.9 The first dominant frequency of a tube vibration in the lift direction

gradually with liquid gap velocity. At a given superficial liquid gap velocity, as the superficial gas gap velocity or void fraction is increased RMS amplitude remains fairly constant or decreases slightly before showing a gradual increase. The data do not show any rapid increase in the amplitude as was observed at the onset of instability when the tube was allowed to vibrate in the drag direction. It should be noted here that the calibration factor for the RMS values plotted in figure 3.8 is different than for the cases when tube was allowed to vibrate only in the drag direction. Similarly no rapid decrease in the first dominant frequency (synonymous with the onset of instability) is observed. In fact for all of the superficial liquid gap velocities the first dominant frequency is observed to increase with void fraction. It appears that forces originating from turbulence in the flow are responsible for the observed vibrational response in the lift direction.

3.4 Damping Ratio

The damping ratios when the flexibly supported tube was allowed to vibrate in the flow direction were calculated from the response spectrum data and from equation (1.8). The drag coefficient in equation (1.8) was obtained from the drag force which was deduced from the mean value of strain gage signal. In figure 3.10, the damping ratios obtained from these two methods are plotted as a function of void fraction when tube vibrated in a stable mode. The data plotted in figure 3.10 are for the case when aluminum rod was inserted in the hollow tube and the superficial liquid gap velocity was fixed at 1.68 m/s. The damping ratios obtained from these two methods show similar values.

Figure 3.10 Damping ratio of a tube in an array in two-phase flow

The damping ratios for the void fractions close to the critical void fraction for the onset of instability could not be measured because the response spectra were not clear as shown in figure 3.3 (b). However, it could be noticed that the damping ratio at the onset of instability was not zero, which were used in several previous studies as a criterion for the onset of fluidelastic instability.

The damping ratios in a stagnant two-phase mixture are plotted in figure 3.11. The empty tube mounted on 0.56 mm thickness cantilever beams and the aluminum rod inserted tube mounted on 0.71 mm thickness cantilever beams were measured. The natural frequencies in water of each tube are 9.2 Hz and 10.0 Hz respectively. Water was filled in the test section and air was introduced from the air injection holes at the bottom of the test section assembly. The damping ratio was calculated from the response spectrum of the vibrating tube. Damping ratios of both systems are seen to increase as the void fraction increases. The damping ratio of empty tube is about two times larger than that of aluminum rod inserted tube. The difference of damping ratios can be explained from equation (1.8). The natural frequencies of two systems are almost similar. However, total mass, including the added mass, for the empty tube is about half of that of aluminum inserted tube. Damping ratio is inversely proportional to the total mass in equation (1.8). Damping ratios at void fractions higher than 0.2 for the empty tube and 0.32 for the aluminum rod inserted tube could not be measured because of the erratic nature of the response signal. The damping ratio obtained in a flowing two-phase mixture are much larger than those obtained in a nearly stagnant two-phase mixture.

Figure 3.11 Damping ratio of a tube in an array in stagnant two-phase mixture

An empirical correlation which best fits all of the measured damping ratios in stable flow conditions is obtained as

$$\zeta_{fluid} = \zeta_{stillwater} + \frac{1}{140} \left(\frac{V}{f_n D} \right) \left(\frac{1}{1 - \alpha} \right)^3 \quad (3.1)$$

where ζ_{fluid} and $\zeta_{stillwater}$ are the damping ratios in fluid flow and in still water respectively. Ninety percents of measured data were found to be within $\pm 18\%$ of the values calculated from this correlation. As shown in figure 3.10, the damping ratio predicted from this empirical correlation and that from theoretical equation were very similar.

To obtain the fluid damping, the structural damping should be subtracted from the total damping. For the present system, the structural damping ratio was found to be 0.008 ± 0.001 which is much smaller than the fluid damping ratio.

3.5 Drag Coefficient on a Single Tube

The pressure distribution around the perimeter of a tube were measured using manometers. Figure 3.12 shows one of the typical pressure distribution profiles around a single tube at a Reynolds number of 8,200 and a void fraction of 0.25. The pressure plotted is the difference in pressure from that at the lower stagnation point (the upstream stagnation point). The pressure on the surface of the tube is maximum at the lower stagnation point and decreases along the tube perimeter. The pressure is minimum at 90° , where the flow velocity is expected to be maximum. After passing the 90° point, pressure increases again as flow lines expand. However, the pressure at the 180° point does not return to its value at 0° because

Figure 3.12 Pressure distribution around a single tube

of a wake behind the tube. The pressure distribution is symmetrical about the vertical axis passing through the center of the tube and parallel to the flow.

The drag coefficient was obtained by taking the integral of the pressure difference multiplied by the cosine of the angle around the tube as given in equation (2.1). In figure 3.13, the drag coefficients for single rigidly and flexibly mounted tubes are plotted as a function of void fraction for various Reynolds numbers. The open symbols represent data for the rigidly mounted tube whereas the solid symbols are for data obtained on a flexibly mounted tube. The data obtained on rigidly and flexibly mounted tubes nearly overlap. At low Reynolds numbers addition of gas dramatically increases the drag coefficient. However, at high Reynolds numbers the effect of presence of second phase on drag diminishes. At a Reynolds number of 430, the drag coefficient for a void fraction of 0.36 is 717, which is about 600 times the value for single phase flow. However, for a Reynolds number of 8,200 and a void fraction of 0.36, the drag coefficient is 4.3. This value is only about four times the single phase value.

Interestingly at a Reynolds number of 15,400 and 21,900, the drag coefficient for a void fraction of 0.046 is only 0.81, which is less than (about 0.6 times) the single phase value of 1.3. The pressure distribution data for these Reynolds numbers showed on the frontal region of the tube a behavior similar to that observed for low Reynolds numbers. However, small amount of gas in the flow was found to have little or no effect on the wake region. The reduction in the drag coefficient at low void fraction was also observed in the study of Yokosawa et al. (1986). With further increase in the void fraction, the drag coefficient is found to exceed again

Figure 3.13 Drag coefficient over a single tube

the value for single phase flow.

The lift coefficient was obtained by taking the integral of the pressure difference multiplied by the sine of the angle around the tube (equation (2.2)), even though it was expected to be zero from theory. The magnitude of the lift coefficient is indicative of the extent of uncertainty in the drag data. The lift coefficient was found to be at least an order of magnitude smaller than the drag coefficient at high velocities, while it was at least two orders of magnitude smaller than the drag coefficient at lower velocities.

3.6 Drag Coefficient on a Tube in an Array

The pressure distributions around the perimeter of the tube in a triangular array were also measured. In figures 3.14 and 3.15, the drag coefficients obtained from these pressure distribution data are plotted as a function of free stream void fraction and as a function of liquid gap Reynolds number respectively. The test tube was located at its normal position. The drag coefficient is seen to increase as the void fraction increases but decreases as the liquid gap Reynolds number increases. The prediction from the curve drawn through the data of Zukauskas and Ulinskas (1990) which was obtained for single phase flow across a tube in a triangular array are also plotted in figure 3.15. Present data for single phase flow are found to be about 35% lower than the data of Zukauskas and Ulinskas.

The pressure distributions around the perimeter of a tube located at the positions of 3 mm and 4 mm higher than the normal position were also measured.

Figure 3.14 Drag coefficient over a tube in an array

Figure 3.15 Drag coefficient over a tube in an array

Figure 3.16 shows the typical shapes of pressure distribution. ΔP is the difference in pressure around a tube with that at the lower stagnation point. P^* is a dynamic pressure, which is defined as $\frac{1}{2}\rho V_r^2$. The flow condition is single phase flow and liquid gap Reynolds number is 41,100. The pressure distributions on the upstream portion of the tube are almost the same for all positions. However, pressure on the downstream portion of tubes displaced 3 mm and 4 mm in the vertical direction are lower than that on tube placed at the normal position. As the test tube moves up, the gap between test tube and upper tubes becomes smaller and flow velocity over the downstream portion of the tube increases (see figure 3.17). As the velocity increases, the pressure decreases.

The decrease of pressure in the downstream portion manifests an increase in the drag force on tubes displaced in the vertical direction or along the flow direction. In figure 3.18, the drag coefficients on tubes at each location are plotted as a function of void fraction. The liquid gap Reynolds number was 41,100. The drag coefficients of tubes located at 3 mm and 4 mm above the normal position are larger than those when tube is placed on its normal position. The curves of drag coefficient for each displacement are almost parallel and the shift between each curve is almost same. This implies that a sudden drag coefficient change was occurred between 3 mm and 4 mm high.

3.7 Void Fraction Profile

Figure 3.19 (a) shows void fraction profile near a single tube. The void fraction profile is along the direction of flow. For the plotted data the freestream void

Figure 3.16 The pressure distribution around a tube in an array, which is located at different positions

Superficial liquid gap velocity in normal position: $V_{r,1}$

Superficial liquid gap velocity in displaced position: $V_{r,2}$

$$V_{r,2} = \frac{8.8}{5.4} V_{r,1}$$

$$= 1.63 V_{r,1}$$

Figure 3.17 Tubes at normal position and at displaced position

Figure 3.18 Drag coefficient of a tube at different locations

Figure 3.19 (a) Vertical void fraction profile near a single tube at low Reynolds number

fraction was 0.25, and the liquid Reynolds number was 430. Approaching the tube from below, the void fraction increases gradually until just below the lower stagnation point. A decrease in void fraction is detected immediately downstream of the cylinder. Thereafter, the void fraction gradually returns to the freestream value. For the same flow conditions, the circumferential void profile at various radii are plotted in figure 3.19 (b). The gas rich region on the upstream portion and a liquid rich region on the downstream portion of the tube can be seen clearly in the figure.

Figures 3.20 (a) and (b) show the vertical and the circumferential void fraction profiles at a liquid Reynolds number of 8,200, and a freestream void fraction of 0.25. The void fraction is minimum very near the lower stagnation point (0°), and then increases along the circumference. It attain a maximum value between 90° and 135° , and then decreases up to the downstream stagnation point. Thus liquid rich regions occupy both upstream and downstream stagnation point regions.

At low flow rates boundary layer near the forward stagnation point is relatively thicker and bubbles accumulate there. With increase in velocity the boundary layer thins and bubbles simply follow the stream lines and only a little accumulation of bubbles occur near the forward stagnation point. In the region downstream of the 90° , the bubbles start to accumulate as the fluid starts to slow down and pressure starts to recover. However only a few of the bubbles are entrained in the wake behind the cylinder for all flows and as a result a liquid rich region is observed near the downstream stagnation point. The latter observation is consistent with the result of Inoue et al. (1986). However Inoue et al. did not observe a gas

Figure 3.19 (b) Circumferential void fraction profile near a single tube at low Reynolds number

Figure 3.20 (a) Vertical void fraction profile near a single tube at high Reynolds number

Figure 3.20 (b) Circumferential void fraction profile near a single tube at high Reynolds number

rich region near the forward stagnation point. The reason being that Reynolds numbers in their experiments were always higher than 5,000.

A circumferential void fraction profile around a tube in a normal triangular array is plotted in figure 3.21. The liquid gap Reynolds number is 41,000 and the freestream void fraction is 0.15. The void fractions at two different radii were measured. For the position near the tube (radius of 13.5 mm), void fraction is minimum at the lower stagnation point (0°), and then increases along the circumference. It attains a maximum value at an angle of 150° , and then decreases until 180° . This profile at 13.5 mm radius is very similar to that for a single tube shown in figure 3.20 (b). The void fraction profile at 15.4 mm radius has two peaks of about the same magnitude at 30° and 150° . This is because 30° and 150° points on a radius of 15.4 mm are the midpoints between two adjacent tubes and represent the smallest flow area. From their work Pettigrew et al. (1988b) noted that flow pattern was symmetrical about the tube equator. However, from the present data it can be seen that upstream and downstream void fraction profiles in the vicinity of the tube are asymmetrical.

Figure 3.21 Void fraction distribution in a triangular tube array

Chapter 4

DISCUSSION

4.1 Correlation of Two-Phase Drag Coefficient

Pressure distributions around a single tube showed that for Reynolds numbers less than about 10,000, addition of gas in the fluid resulted in a larger and rapid decrease of pressure in the frontal portion of the tube. In the downstream direction of the tube, however, the pressure did not recover to its single phase value. Although it is difficult to separate the pressure drop due to acceleration of the flow and that due to viscous dissipation, it appears that presence of bubble in the flow tends to squeeze the flow lines in the frontal portion whereas in the section downstream of the equator, the bubble tends to inhibit the expansion of the flow lines. The viscous dissipation increases because of the relative velocity that exists between gas and liquid. The interfacial area between gas and liquid also increases with increase in gas flow rate or void fraction. The net effect being that drag force or drag coefficient on the tube increases with increase of void fraction.

The pressure distributions around a tube with different flow conditions have been nondimensionallized. For single phase flow, the pressure differences were divided by the dynamic pressure, $\frac{1}{2}\rho_l V_r^2$. The dimensionless pressure curves for different flow rates overlapped each other. For two-phase flow, the dynamic pressure was defined with the two-phase mixture value ($\frac{1}{2}\rho_{2\phi} V_{2\phi,r}^2$) or with the superficial liquid value ($\frac{1}{2}\rho_l (V_r/(1-\alpha))^2$). In either case the results were almost similar.

For the case of two-phase flow on a single tube, the curves again overlapped each other. However, for the case of two-phase flow on a tube in an array, higher void fraction shows larger dimensionless pressure in the region from 60° to 120° (figure 4.1). This implies that the stream lines were not squeezed much in two-phase flow on a tube in an array, while those were squeezed much in two-phase flow on a single tube. It also implies that the two-phase mixture flowed in flow paths in a tube bundle and the velocity in the cross section of the path was somewhat constant. Pettigrew et al. (1988b) also observed the well defined two-phase mixture flow paths in the tube bundle. This makes the use of equation (2.5) possible for the calculation of mean gap void fraction in the tube bundle.

Void fractions in the gap between two tubes in an array were calculated by using equation (2.5). Gas and liquid superficial gap velocities were substituted in the equation for the calculation. The void fractions in the gap between the tubes were about 30% larger than those in the free stream. Void profiles such as those shown in figure 3.21 indeed show an increase in the average void fraction in the flow passage.

The ratio of two-phase to single phase drag coefficients for a single tube and for a tube in an array are plotted in figure 4.2, as a function of Reynolds number. The plotted data are for void fractions in the range 0.25 to 0.4. For a single tube, liquid Reynolds number and free stream void fraction were used whereas for a tube in an array liquid gap Reynolds number and average gap void fraction were used. It is seen that the ratio of two-phase to single phase drag coefficients decreases with Reynolds number and appears to reach an asymptotic value after

Figure 4.1 Pressure distribution around a tube in an array subjected to two-phase flow ($P^* = \frac{1}{2}\rho_{2\phi}V_{2\phi,r}^2$)

Figure 4.2 The ratio of two-phase to single phase drag coefficient for a single tube and for a tube array

a Reynolds number of about 20,000. Consistent with the data at lower Reynolds numbers, the asymptotic value of the ratio is larger at higher void fraction. The clear distinction between data for a tube in an array and a single tube is not observed as the array data appear to be simply an extension of the single tube data.

For single phase flow, the Reynolds number is a key correlation parameter for the drag coefficient. Density difference between gas and liquid leads to a buoyancy force which manifests itself as a velocity difference between gas and liquid. The velocity difference between the two phases leads to additional viscous dissipation. As such Grashof number is considered to be an important parameter. Grashof number is defined as

$$Gr = \frac{(\rho_l - \rho_g)gD^3}{\rho_l\nu_l^2} \quad (4.1)$$

where ν_l is the kinematic viscosity of water. Although in the present experiment Gr was not varied, there is ample precedence in the literature (see for example Drucker et al. 1984 and Kianjah and Dhir 1989) to use this group to describe the effect on drag and heat transfer of addition of a discontinuous phase in a continuous phase. The presence of voids in the fluid not only influence the viscous dissipation but also affect the flow lines as was discussed earlier. Thus the key dimensionless parameters considered to be important in correlating the two-phase drag data are: α , Gr , and Re . In figure 4.3, the data for the ratio of two-phase to single phase drag coefficient obtained on a single tube and on a tube in an array are plotted

Figure 4.3 Empirical correlation of the ratio of two-phase to single phase drag coefficient (----- equation (4.2))

as a function of $\alpha Gr/Re^2$. All of the data are correlated to within $\pm 30\%$ as

$$\frac{C_{D2\phi}}{C_{D1\phi}} = 1 + 3\left(\frac{\alpha Gr}{Re^2}\right) + 12\alpha^2 \quad \text{for } 2 \times 10^{-3} < \frac{\alpha Gr}{Re^2} < 4 \times 10^2 \quad (4.2)$$

where $C_{D2\phi}$ and $C_{D1\phi}$ are two-phase and single phase drag coefficients respectively. As Reynolds number becomes larger or liquid inertia increases, the effect of void on viscous dissipation in the flow diminishes and eventually the ratio of two-phase to single phase becomes nearly independent of $\alpha Gr/Re^2$. However the effect of void fraction in altering the flow field continues to persist. For single phase drag coefficient in a single tube, Roshko's data (1961) were used as single phase data taken in the present work, which was found to compare well with the data of Roshko. Single phase drag coefficient on a tube in a bundle were correlated as

$$C_{D1\phi} = \frac{5.5}{Re^{0.3}} \quad (4.3)$$

It is interesting to note that Drucker et al. correlated their ratio of bubbly flow to single phase heat transfer data with $(\alpha Gr/Re^2)^{\frac{1}{2}}$. Their data was in the range $10^{-1} < \alpha Gr/Re^2 < 10$ with $\alpha < 0.15$. It is possible that they did not recognize the existence of α^2 term because its contribution to their data was relatively not significant.

4.2 Change in Drag Coefficient with Tube Displacement

In figure 4.4, the drag coefficients on a tube in an array are plotted as a function of tube displacement. The ratio of increase of drag coefficient with tube displacement ($\frac{\Delta C_D}{\Delta x}$) is seen to increase rapidly when the displacement exceeds 3 mm. This trend is very distinct in single phase flow, but weakens a little as the void fraction increases. For single phase flow, $\frac{\Delta C_D}{\Delta x}$ in the range from 3 mm to 4 mm is about 4 times larger than that for displacement less than 3 mm. But for two-phase flow with void fraction of 0.3, increase in $\frac{\Delta C_D}{\Delta x}$ for displacements larger than 3 mm is only 1.2 times.

The gradient of drag coefficient with tube displacement ($\frac{dC_D}{dx}$) is needed for the analysis of the fluidelastic instability mechanism (section 4.4). In order to obtain $\frac{dC_D}{dx}$, the drag coefficient was assumed to have the following form,

$$C_D(x) = C_D(0mm) + C_1 x^{C_2} \quad (4.4)$$

where C_1 and C_2 are empirical constants. These constants were determined by substituting the drag coefficients for the tube displaced 3 mm and 4 mm from its normal position. Figures 4.5 and 4.6 show the drag coefficient and $\frac{dC_D}{dx}$ obtained from equation (4.4) as a function of tube displacement respectively. $\frac{dC_D}{dx}$ increases with tube displacement in both single phase and two-phase flows. However, in single phase, $\frac{dC_D}{dx}$ shows very rapid increase when the displacement exceeds 3 mm, while in two-phase flow $\frac{dC_D}{dx}$ increases slowly in that displacement range. The

Figure 4.4 Change in drag coefficient with tube displacement

Figure 4.5 Drag coefficient with tube displacement

Figure 4.6 The gradient of drag coefficient with tube displacement

uncertainty in the drag coefficient itself is less than $\pm 10\%$. However, when the gradient of drag coefficient is calculated, the uncertainty is enlarged. For the single phase flow, the uncertainty is calculated to be about $\pm 44\%$.

4.3 Single and Two-Phase Flow Instability

In the present study, single phase flow instabilities were observed for four different tube-cantilever beam systems when the tube was allowed to vibrate in the flow direction. In the past the flow conditions at onset of instability have been written in terms of two dimensionless groups: $V_r/(f_n D)$ and $2\pi\zeta m/(\rho D^2)$. V_r is the reference gap flow velocity, which is defined as

$$V_r = \frac{P}{P-D} V_l \quad (4.5)$$

where V_l is a freestream velocity. The mass, m , per unit length is

$$m = m_{tube} + m_{add} \quad (4.6)$$

where m_{tube} is the mass of the test tube, and m_{add} is the added mass which is calculated as

$$m_{add} = \frac{\rho \pi D^2}{4} \left[\frac{(D_e/D)^2 + 1}{(D_e/D)^2 - 1} \right] \quad (4.7)$$

For a triangular array (Pettigrew, 1988a)

$$D_e/D = (0.96 + 0.5 \frac{P}{D}) \frac{P}{D} \quad (4.8)$$

Single phase instability data for drag direction vibrations are plotted in figure 4.7 with the data obtained from other studies. The data of present study generally lie near the upper band of the data reported in the literature. The possible reason for this will be discussed in section 4.6. It should be noted that while plotting data obtained in this work the damping ratios were calculated from the equation (1.8). The flow condition and the drag coefficient at onset of instability were used in equation (1.8) to evaluate the damping ratios.

For correlation of two-phase flow data, the two-phase flow density, $\rho_{2\phi}$, is defined as

$$\rho_{2\phi} = \rho_g \alpha + \rho_l (1 - \alpha) \quad (4.9)$$

where ρ_g and ρ_l are the densities of air and water. The mass, m , is calculated for each void fraction because added mass changes with void fraction. For air-water mixture, two-phase mean gap velocity, $V_{2\phi,r}$, is defined as

$$V_{2\phi,r} = \frac{P}{P - D} \times \frac{Q_g + Q_l}{A} \quad (4.10)$$

where A is the cross sectional area. In figure 4.8, the flow conditions at onset of instability are plotted as a function of the dimensionless groups based on Connors' equation. The curve is drawn using equation (1.1) with a value of 6.8 for the constant K . The data of present study show a large scatter from the predicted values from Connors' criterion. Moreover, for the data set of each tube-cantilever beam system, the dimensionless gap velocities show somewhat constant values with varying mass damping parameter. This phenomena implies that the mechanism from which Connors' criterion was derived has some problem to be modified.

Figure 4.7 Onset of instability in single phase flow (comparison with other studies)

Figure 4.8 Onset of instability in two-phase flow

In figure 4.9, the data representing the onset of fluidelastic instability are plotted along with the data from other studies. The data of present study are in good agreement with the data of Heilker and Vincent (1981) and Axisa et al. (1985), but are higher than that of Pettigrew et al. (1988b).

4.4 Mechanism of Fluidelastic Instability

The tube-cantilever beam system of present study (figure 2.3) can be simplified as a mass-damper-spring system. The equation of motion can be written as

$$\begin{aligned} m\ddot{x} + c_s\dot{x} + k_sx &= \Sigma F \\ &= F_D + F_{turb} \end{aligned} \quad (4.11)$$

where m is the total mass including added mass, c_s and k_s are the structural damping coefficient and the structural spring constant respectively. F is the fluid force acting on the tube, which has the components of the drag force (F_D) and the forces caused by turbulence and vortex shedding (F_{turb}). The fluid force caused by vortex shedding is expected to be small in two-phase flow. Drag force can be written as

$$F_D = \frac{1}{2}\rho V_{rel}^2 C_D D \quad (4.12)$$

where V_{rel} is relative flow velocity, and

$$V_{rel} = V - \dot{x} \quad (4.13)$$

and

$$C_D = C_{D_0} + \frac{dC_D}{dx}x \quad (4.14)$$

Figure 4.9 Onset of instability in two-phase flow (comparison with other studies)

where C_{D_0} is the drag coefficient when the tube is at its normal position. Since x is measured from zero, dx has been replaced by x in equation (4.14) (see figure 4.10 (a)). The term $\frac{dC_D}{dx}$ exists only for a tube in an array and it does not exist for a single tube. Also it should be noted that $\frac{dC_D}{dx}$ represents the total change in C_D and includes the contributions of all the independent parameters. After substituting for C_D and V_{rel} , equation (4.12) becomes

$$F_D = \frac{1}{2}\rho V^2 C_{D_0} D - \rho V C_{D_0} D \dot{x} + \frac{1}{2}\rho C_{D_0} D \dot{x}^2 + \frac{1}{2}\rho V^2 D \frac{dC_D}{dx} x - \rho V D \frac{dC_D}{dx} x \dot{x} + \frac{1}{2}\rho D \frac{dC_D}{dx} x \dot{x}^2 \quad (4.15)$$

The first term on right hand side is the static drag force and the second and third terms are damping due to fluid flow. The fourth, the fifth and the sixth terms are fluidelastic force which exists only in a tube array. As such equation (4.11) can be written as

$$m\ddot{x} + (c_s + \rho V C_{D_0} D - \frac{1}{2}\rho C_{D_0} D \dot{x})\dot{x} + k_s x = F_{turb} + \frac{1}{2}\rho V^2 C_{D_0} D + \frac{1}{2}\rho V^2 D \frac{dC_D}{dx} x - \rho V D \frac{dC_D}{dx} x \dot{x} + \frac{1}{2}\rho D \frac{dC_D}{dx} x \dot{x}^2 \quad (4.16)$$

On the right hand side of above equation, the fourth and fifth terms are negligibly small in comparison to the third term. By erasing those terms, above equation can be written as

Figure 4.10 Tube vibration in drag and lift directions

$$m\ddot{x} + (c_s + \rho V C_{D_0} D - \frac{1}{2} \rho C_{D_0} D \dot{x}) \dot{x} + (k_s - \frac{1}{2} \rho V^2 D \frac{dC_D}{dx}) x = F_{turb} + \frac{1}{2} \rho V^2 C_{D_0} D \quad (4.17)$$

If we define

$$x_1 = x - \frac{\frac{1}{2} \rho V^2 C_{D_0} D}{k_s - \frac{1}{2} \rho V^2 D \frac{dC_D}{dx}} \quad (4.18)$$

and substitute in equation (4.17) yields

$$m\ddot{x} + (c_s + \rho V C_{D_0} D - \frac{1}{2} \rho C_{D_0} D \dot{x}) \dot{x} + (k_s - \frac{1}{2} \rho V^2 D \frac{dC_D}{dx}) x = F_{turb} \quad (4.19)$$

For simplicity, in writing equation (4.19) x_1 has been replaced by x .

On equation (4.19) the total damping coefficient (c_t) is defined as

$$c_t = c_s + \rho V C_{D_0} D - \frac{1}{2} \rho C_{D_0} D \dot{x} \quad (4.20)$$

The structural damping coefficient, c_s , is negligibly small in comparison to the other terms. The second term, $\rho V C_{D_0} D$, is identical with equation (1.8), which showed good agreement with experimental results when the tube vibrated in a stable mode. As the flow rate increases, $\rho V C_{D_0} D$ increases continuously while $\frac{1}{2} \rho C_{D_0} D \dot{x}$ increases in magnitude with change in sign in one cycle. However, the relative value of $\frac{1}{2} \rho C_{D_0} D \dot{x}$ to $\rho V C_{D_0} D$, $(\dot{x}/2V)$, is small even at the onset of instability. For the case of empty tube mounted on 0.56 mm thickness cantilever

beams subjected to single phase flow, maximum value of $\dot{x}/2V$ is 0.01. Therefore the total damping ratio at the onset of instability is not zero, which were used in several previous studies as a criterion for the onset of fluidelastic instability.

On the other hand, from equation (4.19), the effective spring constant (k_e) can be defined as

$$k_e = k_s - \frac{1}{2}\rho V^2 D \frac{dC_D}{dx} \quad (4.21)$$

The fluidelastic force ($\frac{1}{2}\rho V^2 D \frac{dC_D}{dx} x$) increases as the flow velocity increases. This causes the effective spring constant to decrease. The effective spring constant was calculated with the data obtained from figure 4.6. In figure 4.11, the first dominant frequencies in single phase flow predicted from the calculated effective spring constant and obtained from the measurement are plotted. For the empty tubes supported with 0.56 mm and 0.71 mm thickness cantilever beams, the predictions are in good agreement with measured data. The first dominant frequencies decrease very rapidly prior to the onset of instability. It is mainly due to the rapid increase of $\frac{dC_D}{dx}$ in the displaced region prior to the onset of instability and due to the increase of fluidelastic force proportional to the square of the liquid gap velocity. However, for the aluminum rod inserted tube supported with 0.71 mm thickness cantilever beams, the first dominant frequency obtained from the theory shows lower value than that from the measurement. The probable reason is the enlargement of uncertainty in the calculation of drag coefficient gradient which was described in section 4.2. The measured data are located within the uncertainty limits. Anyway, the general trends, the rapid decrease of the first dominant

Figure 4.11 The first dominant frequencies in single phase flow obtained from the measurement and from the theory

frequency prior to the onset of instability, are agreed in the prediction and the data. The uncertainty of the effective spring constant is calculated to be about $\pm 50\%$, which is mainly due to the calculation of the gradient of drag coefficient and tube displacement.

In two-phase case, the added mass as well as the effective spring constant decreases as the void fraction increases. In figure 4.12 the effective spring constants in two-phase flow predicted from the theory and deduced from the measured first dominant frequency are plotted. The plotted data are for the case of empty tubes supported with 0.56 mm and 0.71 mm thickness cantilever beams. The predictions are in agreement with the measurement data. The effective spring constant decreases as the void fraction increases. Compare to the case of single phase flow, the decrease of the effective spring constant prior to the onset of instability is less rapid.

The rapid decrease of effective spring constant makes the rapid increase of amplitude of the tube vibration because the amplitude is inversely proportional to the spring constant. The rapid increase of amplitude of tube vibration with small increase of flow velocity means fluidelastic instability is occurred. Therefore the fluidelastic instability is occurred due to the rapid decrease of effective spring constant.

The rapid decrease of effective spring constant prior to the onset of instability implies that the mean displacement of a tube should also increases rapidly prior to the instability. Moreover, when the mean displacement increases, the tube is easier to move up because the drag force increases with the tube displacement.

Figure 4.12 The effective spring constants in two-phase flow obtained from the measurement and from the theory

The mean displacement increase rate change prior to the instability was observed in figure 3.2.

4.5 Mean Displacement of Tube at Onset of Instability

The drag forces on the tube at onset of instability were calculated from the mean values of the strain gage signals. For each tube-cantilever beam system, drag forces at onset of instability were almost constant. The values of the drag force ranged from 10 N/m to 60 N/m and were found to depend not only on the mass of the tube but also on the natural frequency of the system.

The spring constant of each cantilever beam was determined from the natural frequency measured in air. The values were about 10% smaller than those calculated from the geometrical dimensions of the cantilever beam. The displacement of the test tube for a certain applied force could be deduced by knowing the spring constant. In figure 4.13, the mean displacements of test tube at onset of instability are plotted. The displacement of the test tube due to the buoyancy force and the tube weight were taken into consideration while plotting the data. For each tube-cantilever beam system, the displacement of test tube at onset of instability are again almost similar. It shows a little increase with the increase of void fraction.

In figure 4.13, the mean displacement at onset of instability of each tube increases a little as the void fraction increases. In two-phase flow the critical superficial liquid gap velocity (V) is smaller than the critical liquid gap velocity for the single phase flow. Moreover, in the range from 3 mm to 4 mm, where

Figure 4.13 Mean displacement at onset of instability for different tube-cantilever beam systems

the most onset of instability occurred, $\frac{dC_D}{dx}$ for the two-phase flow is smaller than that for the single phase flow (see figure 4.6). In order to decrease the effective spring constant the same amount with single phase flow, a larger $\frac{dC_D}{dx}$ is needed in two-phase flow. A larger $\frac{dC_D}{dx}$ is obtained as the tube displaced more than that for the single phase case.

Moreover, in figure 4.13 the mean displacement at onset of instability for thick cantilever beams are larger than those for thin cantilever beams. The spring constant of a thick cantilever beam is larger than that of a thin cantilever beam. In order to decrease the effective spring constant to the same ratio with thin cantilever beams, a larger flow rate (V) and a larger $\frac{dC_D}{dx}$ are needed for the thick cantilever beams. A larger $\frac{dC_D}{dx}$ is obtained when the displacement of the tube from its mean position is large.

The mean displacements for the tubes in an array of pitch to diameter ratio of 1.4 range from 3.3 mm to 4.4 mm. For the tube in an array of pitch to diameter ratio of 1.3, the mean displacements are about 2.2 mm, which are smaller than those for the case of pitch to diameter ratio of 1.4. The distance between two tubes in an array of pitch to diameter ratio of 1.3 is smaller than that of pitch to diameter ratio of 1.4. Figure 4.14 shows that, for all of the cases, the fluidelastic instability occurred when the gap between the test tube and upper tube was about 5–6 mm. Therefore, the critical gap between test tube and upper tube for the fluidelastic instability exists.

In figure 4.6, $\frac{dC_D}{dx}$ increases very rapidly in single phase flow in the range from

Figure 4.14 Gap between test tube and upper tube at onset of instability for different tube-cantilever beam systems

3 mm to 4 mm, while it increases slowly in two-phase flow. This phenomena explains why the threshold point for onset of instability in two-phase flow is less pronounced than that in single phase flow, which was observed in section 3.2. Because in two-phase flow $\frac{dC_D}{dx}$ increases slowly compare to single phase flow case, the effective spring constant also decreases slowly. This in turn causes the less rapid increase of amplitude of tube vibration at the instability compare to the case for single phase flow. Therefore the threshold point for onset of instability in two-phase flow is less pronounced than that in single phase flow.

The fluidelastic instability was not observed when the tube vibrated in the lift direction. In order for the fluidelastic instability to occur in the drag direction, a large value of $\frac{dC_D}{dx}$ was needed. This was obtained when the distance between the test tube and upper tube was smaller than the critical gap for the fluidelastic instability. However, when the tube vibrates in the lift direction, the mean displacement is always zero (figure 4.10 (b)). The distance between the test tube and the side tubes could not be reduced enough to induce a large value of $\frac{dC_L}{dy}$. Therefore the fluidelastic instability in the lift direction could not occur in the present study.

4.6 Fluidelastic Instability of a Tube with Movement in Two Directions

In figures 4.7 and 4.9, the present study showed the critical flow velocities higher than those of most other studies. The present study allowed the test tube

to vibrate in only the direction of flow, while in most of the other studies the test tube was allowed to move in two directions. When the test tube can move in two directions the fluidelastic instability occurs at a lower flow rate than that for the tube is allowed to vibrate in one direction. Let's consider a tube in an array of pitch to diameter ratio of 1.4, which is displaced 2.8 mm in the upward direction with a certain flow rate (figure 4.15). If the tube vibrates in the vertical direction only, the gap between the test tube and the upper tube is 6.4 mm, and the fluidelastic instability cannot occur. However, if the tube can move in two directions, and if the tube moves 1 mm in the horizontal direction, the gap is 5.9 mm which is smaller than the critical gap for the fluidelastic instability. Therefore, the onset of instability of a tube which can move in two directions occurs at a lower flow rate than that of a tube which can vibrate in one direction.

4.7 Fluidelastic Instability of an All Flexible Tube Bundle

In a heat exchanger, all tubes are flexible and free to vibrate in two directions. When a flexible tube bundle is subjected to cross flow, the vibration center of each tube (the mean displacement of a tube in the present study) will be moved the same amount in the downstream direction (figure 4.16). As the flow rate increases, the vibration centers will move further, but the distance between the vibration centers of each tube will be the same. It is reported that the tubes generally vibrate in oval orbits somewhat synchronized with neighboring tubes (Blevins, 1990). As the flow rate increases, the amplitude (or the diameter of an

Figure 4.15 Tube with movement in two directions

(a) Flexible tube bundle

(b) Rigid tube bundle

Figure 4.16 Tube in an all flexible tube bundle and in a rigid tube bundle

oval) will increase, and the minimum distance between two neighboring tubes in one cycle will become smaller. When the two neighboring tubes approach closer than the critical gap for the fluidelastic instability, the instability will occur for those tubes.

If the fluidelastic instability occurs for two tubes in an array, the instability will be propagated to the neighboring tubes very rapidly. Even though the minimum distance in one cycle between certain two tubes is larger than the critical gap, the instability may occur by the propagation of the instability initiated at other tubes. If tube vibration is measured for the tubes which go into the unstable mode by the propagation of the instability, the threshold point for the instability will be more distinct than that measured in the tubes which initiate the instability. Because the test tube in all flexible tube bundle may not be a tube which initiate the instability, it is expected that the threshold point for the instability measured in an all flexible tube bundle is more distinct than that measured for the test tube in rigid surrounding tubes. It was observed in the experimental results of other studies. In the study of Pettigrew et al. (1988b), the threshold points for the all flexible tube bundle were much distinct than those for a flexible tube in a rigid tube array.

It is not certain whether the critical flow velocity of all flexible tube bundle is smaller than that of a flexible tube in a rigid tube array or not. In all flexible tubes, for the same amount of tube amplitude, the gap between two tubes is much smaller than that in the case of a single flexible tube because both neighboring tubes come closer. However, if the dummy tubes are fixed rigidly, only the mean position of a

flexible tube moves in the downstream direction. Thus, the flexible tube vibrates in a smaller distances. It is not certain which effect is larger. The experimental results of other studies are in contradiction of each other also. Pettigrew et al. (1988b) observed that in two-phase cross flow the critical flow velocity in all flexible tube bundle is much lower than that of a flexible tube in a rigid tube array. However, Gorman (1977) and Lever and Weaver (1982) observed that in single phase flow the critical velocities in the two cases were almost similar.

Chapter 5

CONCLUSIONS

1. The drag coefficients in two-phase flow are much higher than those in single phase flow. The ratio of two-phase to single phase drag coefficient decreases as Reynolds number becomes large or inertia of the base flow increases. This ratio has been correlated with α^2 and $\alpha Gr / Re^2$.
2. The drag coefficient on a tube in an array increases as the tube is displaced in the direction of flow. The drag coefficient increases rapidly when the tube is displaced more than a certain critical distance (about 3 mm in the present study). This trend is very distinct in single phase flow, but the effect weakens a little as the void fraction increases.
3. The fluidelastic instability occurs when the mean displacement of a tube exceeds a certain value. For the tube arrays of same pitch to diameter ratio, the mean displacements at onset of instability are almost similar. Moreover, the mean displacement for the onset of instability are similar to the critical distance beyond which the rapid increase of drag coefficient occurs.
4. The total damping ratio increases with the increase of liquid gap velocity in single phase flow and with the increase of void fraction in two-phase flow. However, the effective spring constant decreases rapidly with the increase of liquid gap velocity or void fraction just prior to the onset of

instability. It is mainly due to the rapid increase of drag force in the displaced region prior to the onset of instability.

5. The fluidelastic instability is generated by rapid decrease of the effective spring constant because the amplitude of tube vibration is inversely proportional to the spring constant. This mechanism is contradictory to previous mechanisms which explained that negative damping causes the fluidelastic instability.

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Appendix A

Uncertainty Analysis

1. Amplitude

1.1 Strain Gage Input Module

The manual of the ACRO-941 Strain Gage Input Module states that the accuracy of the module is $\pm 0.1\%$. Since the input gain range is ± 40 mV, the uncertainty is calculated to be ± 0.04 mV. The followings are the voltage range at onset of instability and the uncertainties related to the strain gage input module for each tube-cantilever beam system.

Tube, Cantilever thickness and P/D		Voltage Range	Uncertainty
Empty tube,	0.56 mm, 1.4	4 mV	$\pm 1.0\%$
Empty tube,	0.71 mm, 1.4	7 mV	$\pm 0.6\%$
Empty tube,	0.89 mm, 1.4	4 mV	$\pm 1.0\%$
Aluminum ins. tube,	0.89 mm, 1.4	5 mV	$\pm 0.8\%$
Solid tube,	0.71 mm, 1.3	3 mV	$\pm 1.3\%$

1.2 Strain Gage

The largest uncertainty source of the measurement with strain gages is the temperature change. In the present study, the other uncertainties such as those due to soldering and due to bonding did not affect the measurement, because the strain gage reading was always compensated by that at still water. All electrical conductors display resistivity change as temperature varies. Since the strain gage input module which is connected to a bonded strain gage will interpret any resistance change in the gage as an applied mechanical strain, the resistance change

which occur only as a function of temperature should be subtracted from the read-out value. The resistance change due to temperature variation is called *thermal output*. Some resistance alloys exhibit very small resistance changes with temperature, and *constantan*, the strain gage alloy, is in this class. The engineering data sheet of the CEA-06-125UN-120 strain gage, which was used in present study, shows that the thermal output of the strain gage is almost zero in the temperature variation range of present study (20°C – 30°C).

However, the cantilever beam, which the strain gages are mounted on, is subjected to change in length with temperature. This length change, which is not related to mechanical force, affects strain gage reading. The uncertainty of a strain gage reading (ΔE) caused by temperature variation is calculated as follows.

- Thermal expansion coefficient of spring steel: $11 \times 10^{-6} [^{\circ}\text{C}^{-1}]$
- Gage factor of a strain gage, GF : 2
- Power voltage of the ACRO data acquisition system, E : 5 [V]

Microstrain of a strain gage is calculated as

$$\epsilon = \frac{1}{GF} \frac{\Delta R}{R} \quad (\text{A.1})$$

where R is resistance. With the temperature change of 1°C ,

$$\pm 11 \times 10^{-6} = \frac{1}{2} \frac{\Delta R}{R} \quad (\text{A.2})$$

$$\frac{\Delta R}{R} = \pm 22 \times 10^{-6} \quad (\text{A.3})$$

The voltage change of a strain gage in Wheastone bridge is calculated with following equation,

$$\frac{\Delta E}{E} = \frac{\frac{\Delta R}{R}}{4 + 2(\frac{\Delta R}{R})} \quad (A.4)$$

Therefore, the uncertainty is

$$\begin{aligned} \Delta E &= 5 \times \frac{\pm 22 \times 10^{-6}}{4 \pm 2 \times 22 \times 10^{-6}} \\ &= \pm 27.5 \times 10^{-6} \text{ [V/}^{\circ}\text{C]} \\ &= \pm 0.0275 \text{ [mV/}^{\circ}\text{C]} \end{aligned} \quad (A.5)$$

In the present study, the dynamic strain (RMS amplitude) and the static strain (mean displacement) have been measured. Onset of instability is determined by the dynamic strain, and the drag force is determined by the static strain. Dynamic strain is affected little by the temperature change and only static strain is affected by the temperature. Moreover, in most measurement the strain gages mounted on top and bottom side of a cantilever beam have been connected as a half bridge connection. In this connection the resistance change of a strain gage due to length change of a cantilever beam is canceled out by that of a strain gage mounted on the other side. Therefore, only static strain measured with the quarter bridge connection, which uses one strain gage, is affected by temperature. When the water temperature was changed 5°C during an experiment, the total uncertainty of static strain was calculated as follows.

$$\frac{\Delta E}{E} = \pm \frac{\sqrt{(0.0275 \times 5)^2 + 0.04^2}}{11}$$

$$= \pm 1.3\% \quad (\text{A.6})$$

2. Water Flow Rate

In this experiment, First Rate Flow Totalizer of ITT Barton Co. were used as a flow meter. The manual states that the accuracy of the totalizer is $\pm 0.25\%$ for 2500 counts. The value of $\pm 0.25\%$ can be accepted as an uncertainty of this flow meter. During an experiment, water volume flow rate (Q_l) was stabilized to within $\pm 0.3\%$. Total uncertainty of water flow rate was calculated as follows.

$$\begin{aligned} \frac{\Delta Q_l}{Q_l} &= \pm \sqrt{0.0025^2 + 0.003^2} \\ &= \pm 0.4\% \end{aligned} \quad (\text{A.7})$$

3. Air Flow Rate

Dwyer Rate-Master Flowmeters of 1000 scfh capacity ($0.00787 \text{ m}^3/\text{s}$) and 100 scfh capacity were used to measure air volume flow rate. During an experiment, the float oscillated up and down within $\pm 2\%$ of air flow rate, and this value was accepted as the uncertainty.

4. Void Fraction

The void fraction measurements were made using the gamma densitometer and were evaluated with the following equation:

$$\alpha = \frac{\ln(I/I_l)}{\ln(I_g/I_l)} \quad (\text{A.8})$$

The major source of uncertainty in the void fraction measurement is the inaccuracy in the measurements of I , I_g , and I_l . The resulting uncertainty in void fraction,

$\frac{\Delta\alpha}{\alpha}$, is obtained as follows:

$$\Delta\alpha = \sqrt{\left(\frac{\partial\alpha}{\partial I}\Delta I\right)^2 + \left(\frac{\partial\alpha}{\partial I_l}\Delta I_l\right)^2 + \left(\frac{\partial\alpha}{\partial I_g}\Delta I_g\right)^2} \quad (\text{A.9})$$

Substitute

$$\frac{\partial\alpha}{\partial I} = \frac{1}{I \ln(I_g/I_l)} \quad (\text{A.10})$$

$$\frac{\partial\alpha}{\partial I_l} = \frac{\alpha - 1}{I_l \ln(I_g/I_l)} \quad (\text{A.11})$$

$$\frac{\partial\alpha}{\partial I_g} = \frac{-\alpha}{I_g \ln(I_g/I_l)} \quad (\text{A.12})$$

to get

$$\Delta\alpha = \frac{1}{\ln(I_g/I_l)} \sqrt{\left(\frac{\Delta I}{I}\right)^2 + \left(\frac{(\alpha - 1)\Delta I_l}{I_l}\right)^2 + \left(\frac{-\alpha\Delta I_g}{I_g}\right)^2} \quad (\text{A.13})$$

$\Delta\alpha$ was calculated as ± 0.006 – 0.007 in most cases. When α was 0.2 and the water flow rate was $0.018 \text{ m}^3/\text{s}$, $\frac{\Delta\alpha}{\alpha}$ was calculated as $\pm 3\%$. However, at low void fraction this value was large because denominator was small; $\frac{\Delta\alpha}{\alpha}$ was $\pm 12\%$ when α was 0.04 .

When the void fraction is measured very near the tube, the uncertainty in the positioning also becomes important. If the distance between the beam and the tube center is r , the total uncertainty in the void fraction can be calculated as

$$\Delta\alpha_{\text{total}} = \sqrt{(\Delta\alpha)^2 + \left(\frac{\partial\alpha}{\partial r}\Delta r\right)^2} \quad (\text{A.14})$$

For the measurement of void fraction in a tube array, the maximum uncertainty of $\frac{\Delta\alpha_{total}}{\alpha}$ was calculated to be $\pm 20\%$. This uncertainty occurred very near the tube wall. The uncertainty of data away from the tube was much less.

For the measurement of the free stream void fraction, the void fractions at three points in upstream were measured. The variation of void fraction at each point was $\pm 8\%$.

5. Natural Frequency

The natural frequency in water of a tube mounted on cantilever beams was determined with two methods. The first method counts peak number in a free decay shape of amplitude-time graph. The uncertainty of this measurement was $\pm 0.6\%$, but the variation of each measurement was $+1.8\%$ and -3.3% . The other method considers the first dominant frequency of tube vibration as the natural frequency. Water was filled in the test section and small amount of air were introduced at the bottom of test section assembly. The air bubbles vibrated the test tube and the response spectrum was obtained from the tube vibration. The uncertainty of this method was $\pm 3\%$.

6. Drag Coefficient

The drag coefficient of a rigidly supported tube was calculated as:

$$C_D = \frac{gD}{V_l^2} \left(\frac{\pi\alpha}{2} + \int_0^{2\pi} C_1 \frac{\Delta^*}{D} \cos\phi d\phi \right) \quad (A.15)$$

where Δ^* is the difference in pressure from that at the lower stagnation point. This method has the possibilities for uncertainty in velocity, void fraction, and

integration of pressure. Using:

$$\frac{\partial C_D}{\partial V_l} = \frac{-2C_D}{V_l} \quad (\text{A.16})$$

$$\frac{\partial C_D}{\partial \alpha} = \frac{\pi g D}{2V_l^2} \quad (\text{A.17})$$

$$\frac{\partial C_D}{\partial (\int_0^{2\pi} C_1 \frac{\Delta^*}{D} \cos \phi d\phi)} = \frac{gD}{V_l^2} \quad (\text{A.18})$$

to get

$$\Delta C_D = \sqrt{\left(-\frac{2C_D}{V_l} \Delta V_l\right)^2 + \left(\frac{\pi g D}{2V_l^2} \Delta \alpha\right)^2 + \left(\frac{gD}{V_l^2} \Delta \left(\int_0^{2\pi} C_1 \frac{\Delta^*}{D} \cos \phi d\phi\right)\right)^2} \quad (\text{A.19})$$

The uncertainty $\frac{\Delta C_D}{C_D}$ was calculated to be less than $\pm 10\%$.

For the measurement with strain gages, strain gage reading and flow rate are source of uncertainty. The drag coefficient is calculated as

$$C_D = \frac{F_D}{\frac{1}{2} \rho_l V_l^2 A} \quad (\text{A.20})$$

Therefore

$$\frac{\partial C_D}{\partial F_D} = \frac{2}{\rho_l V_l^2 A} \quad (\text{A.21})$$

$$\frac{\partial C_D}{\partial V_l} = \frac{-4F_D}{\rho_l V_l^3 A} \quad (\text{A.22})$$

giving an uncertainty in drag coefficient of

$$\Delta C_D = \sqrt{\left(\frac{2}{\rho_l V_l^2 A} \Delta F_D\right)^2 + \left(\frac{-4F_D}{\rho_l V_l^3 A} \Delta V_l\right)^2} \quad (\text{A.23})$$

and $\frac{\Delta C_D}{C_D}$ was calculated to $\pm 1.5\%$. However, this method has other source of uncertainty such as tube vibration and flow shape change due to tube displacement.

7. Gradient of Drag Coefficient

For the calculation of the effective spring constant, the value of $\frac{dC_D}{dx}$ was needed. There are two major sources of uncertainty in the $\frac{dC_D}{dx}$, which are the uncertainty in the gradient of drag coefficient itself and the uncertainty in the predicting tube position.

The gradient of drag coefficient was calculated as

$$\frac{\Delta C_D}{\Delta x} = \frac{C_{D,2} - C_{D,1}}{x_2 - x_1} \quad (\text{A.24})$$

where $C_{D,1}$ and $C_{D,2}$ are drag coefficients at the displacement of x_1 and x_2 respectively. The maximum value of gradient is obtained from

$$\left(\frac{\Delta C_D}{\Delta x}\right)_{\max} = \frac{C_{D,2,\max} - C_{D,1}}{x_2 - x_1} \quad (\text{A.25})$$

and the minimum value is obtained from

$$\left(\frac{\Delta C_D}{\Delta x}\right)_{\min} = \frac{C_{D,2,\min} - C_{D,1}}{x_2 - x_1} \quad (\text{A.26})$$

$C_{D,\max}$ and $C_{D,\min}$ are 10% larger and smaller values than the mean C_D because the uncertainty in drag coefficient is $\pm 10\%$. The uncertainty in the tube position is calculated about $\pm 10\%$, because the spring constants determined from the natural frequency measured in air and calculated from the geometrical dimensions of the

cantilever beam are differed about 10%. For the case of single phase flow, the uncertainty in $\frac{dC_D}{dx}$ was calculated to be about $\pm 50\%$.

Appendix B

Drag Coefficient of a Tube in an Array

Figure B.1 Drag coefficient of a tube at different locations ($Re_r = 32900$)

Figure B.2 Drag coefficient of a tube at different locations ($Re_L = 51600$)

Figure B.3 Drag coefficient of a tube at different locations ($Re_L = 61600$)

Appendix C

RMS Amplitude and Mean Displacement of Test Tube

Table C.1: RMS amplitude and mean displacement of the empty tube supported with 0.56 mm thickness cantilever beams

Mass of tube = 0.062 kg

Spring constant of cantilever beams = 643 N/m

Pitch to diameter ratio = 1.4

Calibration data = 0.53818 N/mV

Unit of RMS amplitude and mean displacement: [mV]

Unit of liquid gap velocity: [m/s]

Void fraction: free stream value

(a) Single phase flow

liquid gap velocity	RMS amplitude	mean displacement
1.134	0.14215361	1.98418558
1.296	0.17468458	2.71195495
1.337	0.20279993	3.00937307
1.355	hit the upper tube	

Table C.1 Contd.

(b) Superficial liquid gap velocity: 1.296 m/s

void fraction	RMS amplitude	mean displacement
0	0.17468458	2.71195495
0.009	0.21756503	3.00881708
0.034	0.30895054	3.32109678
0.046	0.31006601	3.33816993
0.054	0.44357789	3.55274618
0.056	hit the upper tube	

(c) Superficial liquid gap velocity: 1.134 m/s

void fraction	RMS amplitude	mean displacement
0	0.14215361	1.98418558
0.050	0.23226103	2.26242316
0.073	0.33463824	2.50179613
0.094	0.44856107	2.62388480
0.107	0.51720226	2.81177223
0.124	0.67005932	3.05831754

Table C.2: RMS amplitude and mean displacement of the empty tube supported with 0.71 mm thickness cantilever beams

Mass of tube = 0.062 kg
 Spring constant of cantilever beams = 1406 N/m
 Pitch to diameter ratio = 1.4
 Calibration data = 0.95614 N/mV
 Unit of RMS amplitude and mean displacement: [mV]
 Unit of liquid gap velocity: [m/s]
 Void fraction: free stream value

(a) Single phase flow

liquid gap velocity	RMS amplitude	mean displacement
1.620	0.17010574	1.84628564
1.782	0.18875371	2.24495157
1.863	0.18412891	2.33177151
1.944	0.20149222	2.65210571
2.025	0.20567596	2.86611618
2.106	0.23359309	3.19886889
2.188	0.30399328	3.87173976
2.204	hit the upper tube	

(b) Superficial liquid gap velocity: 2.025 m/s

void fraction	RMS amplitude	mean displacement
0	0.21972622	2.86631481
0.070	0.35752347	3.53674070
0.116	0.55708814	4.56444947
0.122	0.73483056	4.83010070

Table C.2 Contd.

(c) Superficial liquid gap velocity: 1.944 m/s

void fraction	RMS amplitude	mean displacement
0	0.20149222	2.65210571
0.104	0.50955385	3.77281084
0.135	0.59541506	4.52984491
0.148	0.67394233	4.82309548
0.156	0.76349854	5.19504180
0.159	hit the upper tube	

(d) Superficial liquid gap velocity: 1.782 m/s

void fraction	RMS amplitude	mean displacement
0	0.18875371	2.24495157
0.107	0.46306649	2.94441077
0.144	0.56881076	3.65653869
0.160	0.59285629	4.02754900
0.185	0.66071552	4.71839163
0.199	0.89692307	5.50382969

(e) Superficial liquid gap velocity: 1.620 m/s

void fraction	RMS amplitude	mean displacement
0	0.17010574	1.84628564
0.147	0.62941617	3.08316904
0.206	0.88089257	4.20716172
0.222	0.97409654	4.59201842
0.237	1.16140759	5.15614682
0.252	1.64771676	5.85257703

Table C.3: RMS amplitude and mean displacement of the empty tube supported with 0.89 mm thickness cantilever beams

Mass of tube = 0.062 kg
 Spring constant of cantilever beams = 2621 N/m
 Pitch to diameter ratio = 1.4
 Unit of RMS amplitude and mean displacement: [mV]
 Unit of liquid gap velocity: [m/s]
 Void fraction: free stream value

(a) Superficial liquid gap velocity: 2.800 m/s

Calibration data: not available

void fraction	RMS amplitude	mean displacement
0	0.27569893	3.79326112
0.010	0.24009494	3.67451692
0.024	0.23691303	3.82865739
0.047	0.24735372	4.12640166
0.106	0.31801441	5.15591319
0.180	0.48180038	7.38240941
0.193	0.76823747	8.14768059

(b) Superficial liquid gap velocity: 2.520 m/s

Calibration data = 1.67692 N/mV

void fraction	RMS amplitude	mean displacement
0	0.20490170	2.83137375
0.137	0.45594174	4.24639046
0.214	0.59215736	5.99720585
0.228	0.63170379	6.69690668
0.250	0.73471099	7.60019028

Table C.3 Contd.

(c) Superficial liquid gap velocity: 2.240 m/s

Calibration data = 1.67692 N/mV

void fraction	RMS amplitude	mean displacement
0	0.18911631	2.17884773
0.239	0.64942986	5.11204332
0.280	0.77880341	6.14457506
0.306	0.88047487	7.42555230
0.320	0.98846477	7.58978885
0.324	1.65066826	8.49093669

(d) Superficial liquid gap velocity: 1.960 m/s

Calibration data = 1.67692 N/mV

void fraction	RMS amplitude	mean displacement
0	0.18560204	2.16370314
0.144	0.48699257	3.01954239
0.269	0.83303142	5.02418559
0.342	0.96570492	6.46191448
0.359	1.11835217	6.83645957
0.380	7.32487440	9.56730060

Table C.4: RMS amplitude and mean displacement of the aluminum rod inserted tube supported with 0.71 mm thickness cantilever beams

Mass of tube = 0.207 kg
 Spring constant of cantilever beams = 1406 N/m
 Pitch to diameter ratio = 1.4
 Calibration data = 1.07862 N/mV
 Unit of RMS amplitude and mean displacement: [mV]
 Unit of liquid gap velocity: [m/s]
 Void fraction: free stream value

(a) Single phase flow

liquid gap velocity	RMS amplitude	mean displacement
0.692	0.12232290	0.57747984
1.960	0.17068866	3.10692430
2.240	0.20732623	3.81033039
2.520	0.26244476	4.47642040
2.693	0.29459411	4.92992830
2.800	0.33717626	5.33377683
2.815	0.33772969	5.29621363
2.965	0.37607077	5.73314893
3.780	2.54460549	8.29728639

(b) Superficial liquid gap velocity: 2.520 m/s

void fraction	RMS amplitude	mean displacement
0	0.26111388	4.52462399
0.007	0.23228666	4.37686407
0.014	0.23670813	4.43890106
0.028	0.22546248	4.59722650
0.053	0.25755596	4.89622890
0.097	0.36180690	6.43071472
0.120	0.42901757	7.22419369
0.132	0.76548845	7.75658255

Table C.4 Contd.

(c) Superficial liquid gap velocity: 2.240 m/s

void fraction	RMS amplitude	mean displacement
0	0.21778136	4.00948203
0.007	0.19826297	3.83445752
0.030	0.18142182	3.94464386
0.058	0.21847296	4.14748681
0.096	0.30068469	4.66767252
0.126	0.35202119	5.30968440
0.164	0.41351765	6.23714888
0.202	0.61337972	7.73258698

(d) Superficial liquid gap velocity: 1.680 m/s

void fraction	RMS amplitude	mean displacement
0	0.14381184	2.42637434
0.009	0.13084053	2.32345631
0.035	0.13305201	2.35854348
0.068	0.22418305	2.47299393
0.150	0.52989972	3.03021612
0.214	0.85001165	3.97970440
0.259	1.22875905	4.71204307
0.296	1.67503691	5.93792727
0.337	1.90605497	6.85827973
0.372	2.76390982	8.50060275

Table C.5: RMS amplitude and mean displacement of the tube in an array of pitch to diameter ratio of 1.3

Mass of tube = 0.126 kg
 Spring constant of cantilever beams = 1406 N/m
 Pitch to diameter ratio = 1.3
 Calibration data = 0.88219 N/mV
 Unit of RMS amplitude and mean displacement: [mV]
 Unit of liquid gap velocity: [m/s]
 Void fraction: free stream value

(a) Single phase flow

liquid gap velocity	RMS amplitude	mean displacement
1.733	0.13228177	2.25418888
2.080	0.16921249	2.93859659
2.253	0.20151150	3.49975167
2.305	hit the upper tube	

(b) Superficial liquid gap velocity: 2.080 m/s

void fraction	RMS amplitude	mean displacement
0	0.17496748	2.92088187
0.010	0.17997088	3.13355172
0.022	0.18233384	3.14385307
0.039	0.24750622	3.26000393
0.051	0.33303282	3.78910363

Table C.5 Contd.

(c) Superficial liquid gap velocity: 1.733 m/s

Calibration data = 0.97129 N/mV

void fraction	RMS amplitude	mean displacement
0	0.13825734	2.10289063
0.048	0.26701567	2.33491387
0.088	0.52636296	2.61174919
0.106	0.62889206	2.91923001
0.128	0.72077489	3.31748307
0.148	0.74740326	3.80507911
0.178	0.92737782	4.82345021

Table C.6: RMS amplitude and mean displacement of the tube in an array supported with 0.89 mm thickness cantilever beams in lift direction vibration

Mass of tube = 0.062 kg
 Spring constant of cantilever beams = 2621 N/m
 Pitch to diameter ratio = 1.4
 Unit of RMS amplitude and mean displacement: [mV]
 Unit of liquid gap velocity: [m/s]
 Void fraction: free stream value

(a) Single phase flow

liquid gap velocity	RMS amplitude	mean displacement
0.728	0.10029913	0.21341324
1.400	0.16433255	0.14079309
2.100	0.28110027	0.05293715
2.520	0.37963399	0.14324320
2.800	0.44127533	0.10941506

(b) Superficial liquid gap velocity: 2.800 m/s

void fraction	RMS amplitude	mean displacement
0	0.44127533	0.10941506
0.056	0.46762922	0.13064205
0.108	0.45903271	0.21622253
0.180	0.42902732	0.25198901
0.273	0.37185353	0.30086327
0.375	0.39611667	0.30205369
0.400	0.42944133	0.46607101
0.488	0.48977312	0.32519091

Table C.6 Contd.

(c) Superficial liquid gap velocity: 2.520 m/s

void fraction	RMS amplitude	mean displacement
0	0.34014484	0.02877891
0.060	0.34921852	0.01748359
0.135	0.37867692	-
0.250	0.34467664	0.22526705
0.325	0.36094466	0.19160354
0.400	0.44881755	0.18801415
0.472	0.54951704	0.17361128

(d) Superficial liquid gap velocity: 2.240 m/s

void fraction	RMS amplitude	mean displacement
0	0.28276122	0.01426137
0.064	0.27698949	0.16192651
0.135	0.30257297	0.27885616
0.268	0.31098771	0.33314884
0.350	0.36905387	0.31771970
0.408	0.46202236	0.32721222
0.468	0.53720051	0.31575144

Table C.6 Contd.

(e) Superficial liquid gap velocity: 1.960 m/s

void fraction	RMS amplitude	mean displacement
0	0.22987157	0.16023133
0.069	0.21497187	0.14210522
0.148	0.25723338	0.30581796
0.280	0.31956822	0.34769166
0.365	0.43688449	0.32873428
0.413	0.51480889	0.32468856
0.494	0.58770788	0.30101359

(f) Superficial liquid gap velocity: 1.400 m/s

void fraction	RMS amplitude	mean displacement
0	0.15310523	0.03333060
0.077	0.14974599	0.09388796
0.168	0.25242791	0.19237379
0.306	0.41334039	0.23062018
0.379	0.49511927	0.24209753
0.453	0.55480981	0.23548821
0.519	0.59690726	0.24325029

Table C.7: RMS amplitude and mean displacement of the tube in an array supported with 0.71 mm thickness cantilever beams in lift direction vibration

Mass of tube = 0.062 kg
 Spring constant of cantilever beams = 1406 N/m
 Pitch to diameter ratio = 1.4
 Unit of RMS amplitude and mean displacement: [mV]
 Unit of liquid gap velocity: [m/s]
 Void fraction: free stream value

(a) Single phase flow

liquid gap velocity	RMS amplitude	mean displacement
0.728	0.10371564	0.13454008
1.120	0.14459783	0.09113521
1.400	0.19206151	0.06111813
1.470	0.22453152	0.12264597
1.540	0.24830724	0.10411537
1.680	0.28529626	—
1.960	0.38093901	0.03722703
2.100	0.43629110	0.00875049
2.240	0.50829643	0.05447352
2.520	0.68914676	0.10067397
2.660	0.83407253	0.07086449

(b) Superficial liquid gap velocity: 2.100 m/s

void fraction	RMS amplitude	mean displacement
0	0.42213312	0.11066943
0.008	0.38797593	0.03905450
0.041	0.36806563	0.08820933
0.077	0.33672184	0.14307279
0.147	0.33359495	0.20294100
0.264	0.31987682	0.29570789
0.339	0.38717002	0.30859942
0.424	0.52184719	0.27442164
0.424	0.60389286	0.23940188

Table C.7 Contd.

(c) Superficial liquid gap velocity: 1.750 m/s

void fraction	RMS amplitude	mean displacement
0	0.26950097	0.06764859
0.024	0.25532180	0.07732183
0.047	0.24164365	0.12302894
0.090	0.23418763	0.22936112
0.163	0.26133057	0.36929411
0.308	0.39193279	0.46090860
0.356	0.44108769	0.54641301
0.440	0.57231390	0.89703781
0.507	0.72036493	—

(d) Superficial liquid gap velocity: 1.400 m/s

void fraction	RMS amplitude	mean displacement
0	0.20624019	0.24296331
0.008	0.19737574	0.16633987
0.019	0.19241543	0.15213966
0.038	0.18396403	0.15726888
0.056	0.17932904	0.19079506
0.087	0.18592720	0.21869254
0.120	0.23615815	0.22582161
0.164	0.30641478	0.27350163
0.266	0.47209784	0.28436673
0.332	0.53173888	0.17252547
0.345	0.56949854	0.22779939
0.368	0.59684020	0.24692929
0.373	0.54564697	0.24354369
0.456	0.65809685	0.24258161
0.456	0.61681503	0.15795321

Appendix D

Void Fraction Distribution in a Triangular Tube Array

Tube radius = 11 mm

Pitch to diameter ratio = 1.4

Superficial liquid gap velocity = 1.620 m/s

Superficial gas gap velocity = 0.459 m/s

Free stream void fraction = 0.18

Table D.1: Void fraction profile around a tube in an array at radius of 13.5 mm

Angle	minimum	maximum	average
-150	0.2823	0.2823	0.2823
-120	0.2567	0.2616	0.2592
-90	0.1715	0.1809	0.1762
-60	0.1451	0.1576	0.1514
-30	0.1270	0.1318	0.1294
0	0.0566	0.0721	0.0643
30	0.1073	0.1116	0.1095
60	0.1317	0.1398	0.1358
90	0.1769	0.1785	0.1777
120	0.2386	0.2386	0.2386
150	0.2457	0.2580	0.2519
180	0.1774	0.1774	0.1774

Table D.2: Void fraction profile around a tube in an array at radius of 15.4 mm

Angle	minimum	maximum	average
-150	0.2360	0.2436	0.2398
-120	0.1921	0.2052	0.1987
-90	0.1465	0.1506	0.1486
-60	0.1276	0.1295	0.1286
-30	0.2089	0.2228	0.2159
0	0.0872	0.1007	0.0940
30	0.2042	0.2082	0.2062
60	0.1243	0.1277	0.1260
90	0.1265	0.1347	0.1306
120	0.1466	0.1495	0.1480
150	0.1857	0.1901	0.1894
180	0.1406	0.1528	0.1467

Appendix E

Pressure Distribution Around a Tube in an Array

Angle of manometer inclination = 0°

Density of water = 997 kg/m^3

Density of Freon-113 = 1565 kg/m^3

Unit: [mm]

Void fraction: free stream value

Table E.1: Difference in pressure from that at the lower stagnation point, normal position, $V_{r,l} = 1.296 \text{ m/s}$

Angle	single phase	$\alpha = 0.05$	$\alpha = 0.1$	$\alpha = 0.2$	$\alpha = 0.3$
-150	130.5	133.5	137.5	179.5	255.5
-120	165	171	187	234	302
-60	175	183	190	210	250
-30	104.5	109.5	116.5	147.5	190.5
30	91	95	100	111	156
60	188	192	197	224	262
120	163	167	180	227	299
150	131	135	141	183	256
180	118	124	134	176	255
C_D	0.275	0.290	0.330	0.527	0.831

Table E.2: Difference in pressure from that at the lower stagnation point, normal position, $V_{r,l} = 1.620 \text{ m/s}$

Angle	single-phase			$\alpha = 0.05$		$\alpha = 0.1$		$\alpha = 0.2$		$\alpha = 0.3$	
	185.5	181	176.5	194.5	195	213.5	210	285.5	285	408.5	395
-150	240	251	251	256	255	278	275	354	361	464	467
-60	243.5	250.5	262	278	276.5	286	284.5	328	335.5	386	400.5
-30	160	148	103	170	166	189	181	241.5	236	308	304
30	131	136	136	139	148.5	147	157	189	189	243	244.5
60	261	269	280	290	290	300	301	348	347	406	409
120	234	225	225	249	242	272	252	348	329	441	429
150	186	176	167	197	191	217	210	289	285	402	406
180	185	163.5	163.5	199	213	214	213	277	259	378	363
C_D	0.270	0.229	0.246	0.273	0.274	0.323	0.297	0.488	0.459	0.774	0.735

Table E.3: Difference in pressure from that at the lower stagnation point, normal position, $V_{r,l} = 2.025 \text{ m/s}$

Angle	single-phase			$\alpha = 0.05$		$\alpha = 0.1$		$\alpha = 0.2$		$\alpha = 0.3$	
	284	276	268	314	304	344	335	462	452	629	637
-150	284	276	268	314	304	344	335	462	452	629	637
-120	375	363	385	388	384	417	411	538	548	701.5	727
-60	377	400.5	408.5	425	431.5	447	466.5	535	533.5	612	633.5
-30	227	225	177	258	256	291	290	384	378	488	479
30	197	192	190.5	208.5	211.5	230	238	301	310	401.5	396
60	386	427	416	438	455	459	484	538	556	634	653
120	347	324	340	366	347	394	393.5	543	521	690	670
150	291	266	259	312	295	341	313	456	463	622	642
180	261	267	245	286	288	323	328	434	450	592.5	610
C_D	0.265	0.217	0.238	0.268	0.235	0.306	0.264	0.473	0.470	0.722	0.748

Table E.4: Difference in pressure from that at the lower stagnation point, normal position, $V_{\infty} = 2.431$ m/s

Angle	single-phase			$\alpha = 0.05$		$\alpha = 0.1$		$\alpha = 0.2$		$\alpha = 0.3$	
	389	381	371	435	437	486	478	665	656	910	868
-150	389	381	371	435	437	486	478	665	656	910	868
-120	526	501	516	512	528	557	579	746	747	997	952
-60	538.5	563.5	585.5	615.5	624	663.5	669	765.5	768	892.5	854
-30	331.5	314	272	361	368.5	408	421.5	544	559.5	681	692.5
30	259.5	259.5	266	296.5	—	338.5	—	438.5	—	563.5	—
60	560	597	595	648	593	682	648	805	777	924	909
120	496	496	484	501	—	591	—	757	—	950	—
150	394	364	352.5	419	435	478	481	664	652	915	857
180	370	370	348.5	418	418	469	469	650	650	847	847
C_D	0.235	0.216	0.207	0.232	0.252	0.282	0.287	0.465	0.456	0.718	0.672

Table E.5: Difference in pressure from that at the lower stagnation point, 3 mm displaced, $V_{r,l} = 1.620$ m/s

Angle	single-phase		$\alpha = 0.05$		$\alpha = 0.1$		$\alpha = 0.2$		$\alpha = 0.3$	
-150	198	186	224	215.5	248	234.5	319.5	306	428	411
-120	250.5	248	271.5	272.5	291	290	350	349	465.5	447
-60	240	232	278	281	294	297	325	330	392	386
-30	143.5	157	163	174	182	190.5	224	236	280	292
30	117	116	136	131	139.5	139	175	171	213	204.5
60	282	281	299.5	290	309.5	297	344	343	391.5	394
120	269.5	276.5	293.5	299	297	306	346.5	359	445.5	546
150	196	196	216	216	243.5	242	324	319	426	410
180	164	176.5	182	197	202.5	221.5	272	288	381	391
C_D	0.311	0.308	0.339	0.348	0.394	0.403	0.580	0.577	0.883	0.907

Table E.6: Difference in pressure from that at the lower stagnation point, 3 mm displaced, $V_{r,l} = 2.025$ m/s

Angle	single-phase		$\alpha = 0.05$		$\alpha = 0.1$		$\alpha = 0.2$		$\alpha = 0.3$	
-150	295	287	338.5	319	370.5	360.5	505	487	655	644
-120	380	399	413	412	452	442	543	550	703	693
-60	374	399	437	431	457.5	463	511	514	601	612.5
-30	216	226	252	257.5	282	286	353.5	369	433.5	450
30	177	178	202	200.5	222	221	271.5	271	347	339
60	420	415	462.5	454	480	477	551	550	634.5	629
120	423	430	446.5	448	464	476	566	557	701	697.5
150	301	314	335	333	367	373	507	494	671.5	651
180	246.5	264	281.5	295	319	328	453	466	610	613
C_D	0.304	0.315	0.326	0.325	0.372	0.374	0.595	0.574	0.857	0.827

Table E.7: Difference in pressure from that at the lower stagnation point, 3 mm displaced, $V_{r,l} = 2.431$ m/s

Angle	single phase	$\alpha = 0.05$	$\alpha = 0.1$	$\alpha = 0.2$	$\alpha = 0.3$
-150	401	447	526	689	920.5
-120	548	585	627	774	977
-60	525	634	672	767	853
-30	318	374	420	535	642
30	239	279	312	392	489.5
60	576	644	684	786.5	918
120	579	622	660	794	983
150	411	471	531	708	926.5
180	359.5	415	483	655	867
C_D	0.294	0.304	0.367	0.541	0.800

Table E.8: Difference in pressure from that at the lower stagnation point, 4 mm displaced, $V_{r,l} = 1.296$ m/s

Angle	single-phase		$\alpha = 0.05$		$\alpha = 0.1$		$\alpha = 0.2$		$\alpha = 0.3$	
-150	150	150	167	167	177	182	222	223.5	286	285
-120	173	176	179	182	192	193	230	232	292	287
-60	171	172	182	183	186	191	207.5	207.5	239	239
-30	104	97.5	108	97.5	115	97.5	144	123.5	172	161
30	83.5	82	87	88	90	92	102	107	133.5	130
60	186	192.5	192	199	197	205	221	225.5	255.5	249
120	193.5	197	198.5	207	203	213	233	238	287.5	286
150	153	151.5	163	161	174	177	219	220	284	300
180	140	133	146	140	153.5	149	195	188	249.5	246
C_D	0.426	0.423	0.471	0.475	0.518	0.544	0.704	0.721	0.968	1.007

Table E.9: Difference in pressure from that at the lower stagnation point, 4 mm displaced, $V_{r,l} = 1.620$ m/s

Angle	single-phase		$\alpha = 0.05$		$\alpha = 0.1$		$\alpha = 0.2$		$\alpha = 0.3$	
-150	219	220	261	249	277.5	275	343	346	447	449
-120	254	248	277	273	297	291	352.5	349	461.5	446
-60	238	234	281	282	296	283	329	322	382	383
-30	150.5	147	169	168.5	188	186	234	225	283	267
30	112.5	120	132.5	137.5	140.5	145.5	170.5	173.5	217.5	210
60	293.5	268	300	302	308	308	345	346.5	397	383.5
120	288.5	295	309.5	308	318	314	362	362	450.5	461
150	218	232	248.5	246	269	270	339	342	443	453
180	187.5	193	212	210.5	229	228.5	296	294.5	401	393.5
C_D	0.383	0.417	0.447	0.422	0.485	0.482	0.646	0.657	0.936	0.966

Table E.10: Difference in pressure from that at the lower stagnation point, 4 mm displaced, $V_{r,l} = 2.025$ m/s

Angle	single-phase		$\alpha = 0.05$		$\alpha = 0.1$		$\alpha = 0.2$		$\alpha = 0.3$	
-150	334	335	377	377	428	411	536	530	697	680.5
-120	399	410	419	420.5	446	448	545	548	702	703
-60	396	408	430	438	448	473	511	519	611	593
-30	226	223.5	263	255.5	288	278.5	365	349	445	433
30	172	177	199	207	222	223	278	278	340.5	345
60	435	425.5	456	468	477	485.5	533	546	634	629
120	443.5	437	463.5	460	477	473	556	559	690	704
150	331	339	368	370.5	405	405	530	540	682	699
180	282	296.5	312	315	345	348.5	480	457	630	607
C_D	0.370	0.385	0.400	0.395	0.453	0.437	0.640	0.631	0.893	0.895

Table E.11: Difference in pressure from that at the lower stagnation point, 4 mm displaced, $V_{r,i} = 2.431$ m/s

Angle	single-phase		$\alpha = 0.05$		$\alpha = 0.1$		$\alpha = 0.2$		$\alpha = 0.3$	
-150	474	463	507	537	608	575.5	753	751	958	956
-120	544	551	594	601.5	638	632	780	794	975	990.5
-60	544	534	630	640	668	674	762	763	853	857
-30	293	307	359	368	410	392	514	525	607.5	611
30	232	237.5	282	289	313	314	397	404	482	488
60	597	593.5	646	655	676	678	775	783.5	890	872
120	609.5	529.5	650	634	672.5	653	810	785	967.5	982
150	468	470	517	522	556	591	743.5	757	938	971
180	409	394	442.5	440	491.5	479	665.5	663	867.5	863
C_D	0.384	0.366	0.378	0.378	0.430	0.424	0.605	0.596	0.842	0.860

Table E.12: Difference in pressure from that at the lower stagnation point, single tube, $V_l = 0.606 \text{ m/s}$

Angle	single-phase	$\alpha = 0.05$	$\alpha = 0.1$	$\alpha = 0.2$	$\alpha = 0.3$
-150	102	102	106.5	135	175.5
-120	114.5	116.5	134	169.5	227
-60	135	155	174	228	294.5
-30	44	52.5	73	85	103
30	36.5	36.5	36.5	45.5	70
60	118	122	148	192.5	273
120	106.5	109.5	122	156.5	218
150	98	92	107	140	195
180	87	83	101.5	120.5	169
C_D	1.393	1.233	1.367	1.835	2.492