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SOME ASPECTS OF HEAT-EXCHANGER TUBE DAMPING IN TWO-PHASE MIXTURES

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Two-phase flow exists in many shell-and-tube heat exchangers. To avoid excessive flow-induced vibration, it is necessary to have information on damping. A simple experiment was undertaken to study the effect of several parameters such as void fraction, surface tension, tube frequency and confinement on damping in two-phase mixtures. The experiment consisted of a cantilevered tube immersed in a two-phase mixture generated by bubbling air through water. It is found that void fraction and flow regime are dominant parameters. Surface tension is also important. The results are presented in detail in the paper.

1. INTRODUCTION

TWO-PHASE cross-flow exists in many shell-and-tube heat exchangers. To avoid excessive flow-induced vibration, it is necessary to carry out a vibration analysis of heat-exchanger tubes at the design stage. Tube damping information is required for vibration calculations.

This paper describes a simple experiment to study damping of heat-exchanger tubes in two-phase mixtures. A single cantilever tube was subjected to various air–water mixtures up to 30% void fraction. The effects of void fraction, surface tension, tube frequency and degree of confinement were investigated. The results are presented and discussed in this paper. It is hoped that this information will lead to a better understanding of the energy-dissipation mechanisms that govern damping in two-phase flow.

2. BACKGROUND INFORMATION

There is very little information available on damping in two-phase flow. Carlucci (1980) and Carlucci & Brown (1983) studied damping of cylinders subjected to confined axial two-phase flow. They found that damping is highly dependent on void fraction, as shown in Figure 1. The damping ratio, ζ, reached a maximum of 3–4% at void fractions between 20 and 60%. They also found that damping is directly related to the ratio of hydrodynamic mass to cylinder mass, m_h/m. Damping was not much dependent on mass flux, until the flow regime was affected. Cylinder frequency was not a dominant parameter.

More recently, we have studied damping of tube bundles subjected to two-phase (air–water) cross-flow (Pettigrew *et al.* 1989). Triangular tube bundles and square tube bundles of pitch over diameter ratios, P/D, of 1·22 and 1·47 were tested. The tubes were cantilevered and had a natural frequency of roughly 30 Hz. The damping results, given in

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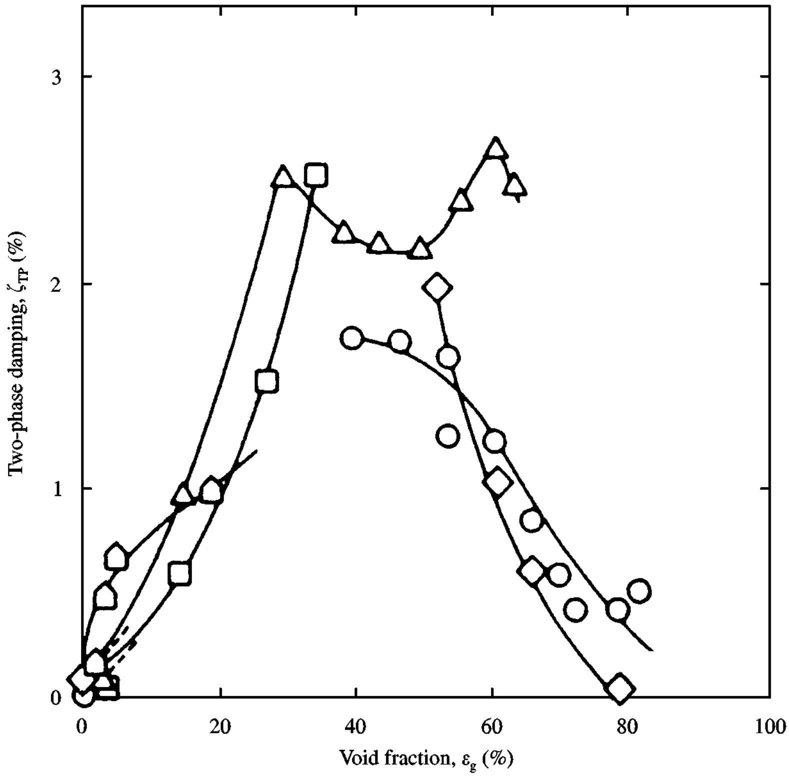

Figure 1. Damping of a cylinder in confined air–water axial flow; mass flux: ⌂ = 0, ◇ = 500, ○ = 1000, △ = 3000, □ = 5000 kg m^{-2} s^{-1} (Carlucci 1980).

Figure 2, show that damping is largest between 40 and 70% void fraction, where ζ reaches about 4%.

Axisa *et al.* (1984, 1985, 1986, 1988) have measured damping of tube bundles subjected to both air–water and steam–water flow for void fractions generally higher than 85%. Normal-square, normal-triangular and rotated-triangular tube bundles, all of $P/D = 1{\cdot}44$, were tested. The tubes were clamped at both ends and had a frequency of about 72 Hz. They found damping to decrease drastically with void fraction beyond 85%. This behaviour is similar to that observed by Carlucci (1980) and by Pettigrew *et al.* (1989) as discussed in the preceding paragraphs. Interestingly, damping in air–water appears to be more than 50% larger than in steam–water. There is some scatter in the results, as shown in Figure 3. However, the reader should understand that damping tests in high temperature and pressure steam–water are extremely difficult to do. Thus, such damping results are very scarce. Furthermore, some scatter in results is generally encountered in the study of many two-phase flow parameters. It is inherent to the nature of two-phase flow phenomena. Thus, for the time being, some judgment is required to make appropriate use of the limited available results.

In summary, previous work shows that damping in two-phase flow is strongly dependent on void fraction or flow regime, directly related to the ratio of hydrodynamic mass to cylinder mass, and weakly related to frequency, confinement and mass flux. However, the

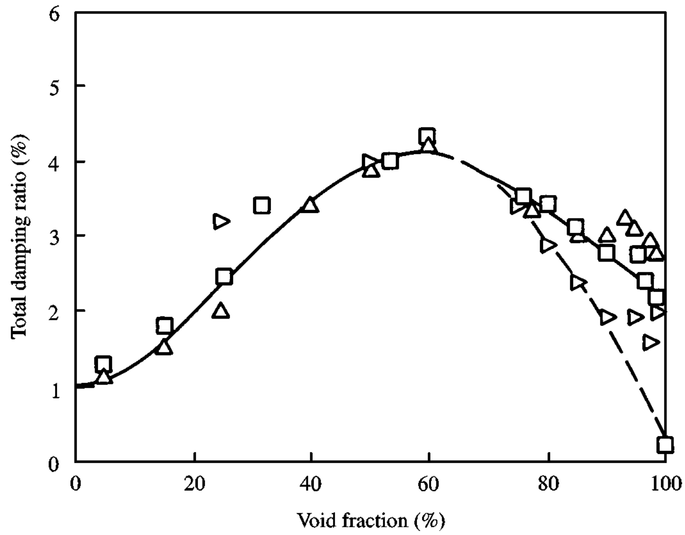

Figure 2. Total damping for tube bundles of $P/D = 1{\cdot}47$ in two-phase cross-flow (Pettigrew *et al.* 1989): ▷, normal triangular; △, rotated triangular; □, normal square bundle.

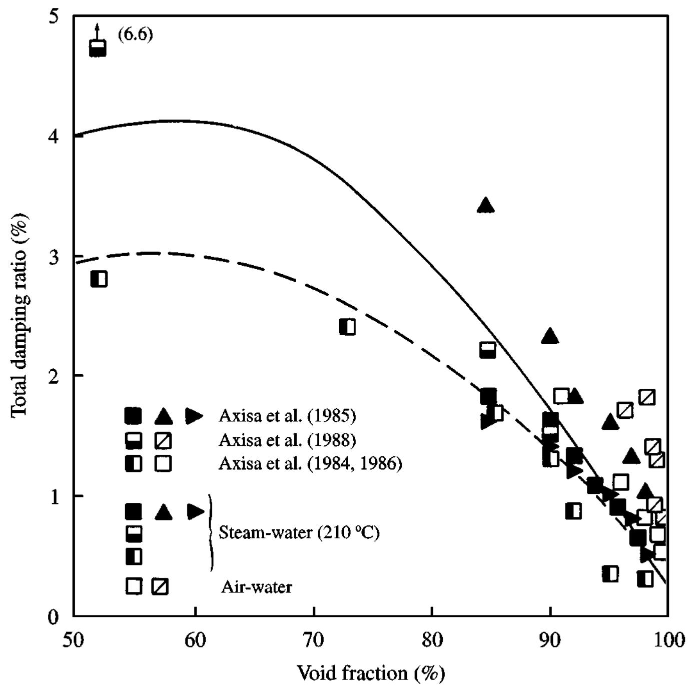

Figure 3. Damping of tube bundles in two-phase cross-flow: air–water versus steam–water results; □, normal-square; △, rotated triangular; ▷, normal-triangular bundle. Curves: ___, average of many data points in air–water for $P/D = 1{\cdot}47$ from Pettigrew *et al.* (1989); _ _ _, steam–water data at 210°C for $P/D = 1{\cdot}44$ from Axisa *et al.* (1984, 1985, 1986, 1988).

difference between air–water and steam–water remains unexplained. In two-phase flow, the fluid forces that govern the flow regime, and thus the nature of the flow (i.e. bubble size, etc.), are largely related to surface tension. Therefore, we speculated that fluid damping forces in a two-phase flow are dependent on surface tension. Interestingly, surface tension in air–water at 20°C is twice that of steam–water at 210°C. Consequently, we conducted a simple experiment on a cantilevered tube subjected to two-phase mixtures to investigate the effect of surface tension at low void fraction. This experiment is discussed in detail in this paper. The effects of void fraction, tube frequency and confinement were also investigated.

The intent of this study was not to provide two-phase damping information that could directly be applied to the design of industrial components. Rather, it was a simple experiment which allowed many tests to be done, to study a broad range of parameters. The intent was to identify the relevant parameters that govern damping, with the hope that this would lead to a better understanding of the energy dissipation mechanisms in two-phase mixtures. This paper should be viewed as a stepping stone towards the development of a model to formulate two-phase damping in process equipment.

3. EXPERIMENTAL CONSIDERATIONS

The experimental rig consisted of a vertical cantilevered tube immersed in a two-phase mixture, as shown in Figure 4. A stainless-steel Type 304 tube of outside diameter

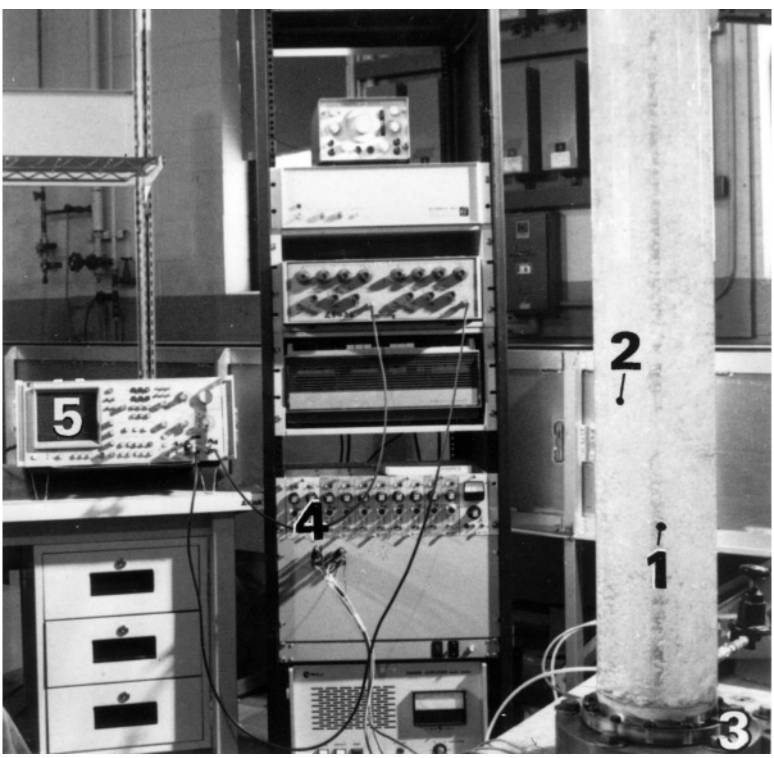

Figure 4. Experimental apparatus: 1, tube; 2, acrylic cylinder; 3, tubesheet; 4, strain-gauge bridge; 5, real-time analyser.

$D = 13{\cdot}0$ mm and of wall thickness $w = 1{\cdot}1$ mm was used. The tube length, l, was originally 1·2 m, which gave a natural frequency, f, in water of about 7 Hz. The tube was shortened a section at a time to increase the natural frequency.

The tube was solidly clamped at one end by brazing it into a steel tubesheet, 70 mm thick and 280 mm in diameter. This was done to minimize the extraneous tube-joint damping. A plug installed at the upper end prevented water from filling the tube. The tube and tubesheet assembly were bolted to a large concrete block.

The tube was surrounded by a concentric acrylic tubular cylinder. The majority of the tests were done with a tubular cylinder of inside diameter, $D_e = 152$ mm. This large inside diameter essentially simulated an unconfined flow field. Two smaller diameter tubes of $D_e = 51$ and 25 mm were used for some tests to study the effect of confinement.

The two-phase mixture was generated by bubbling air through water. The air was injected through a circular sintered (100 μm) steel plate attached to the tubesheet at the bottom of the test section. Void fractions of up to 25%, sometimes 30%, could be simulated by varying the air flow. As shown in Figure 5, the flow regime was essentially bubbly flow.

The surface tension was varied by the addition of a surfactant, namely Triton CF-32, an amine-polyglycol condensate. Surface tension, σ, is inversely proportional to the logarithm of the additive concentration. The surface tension in distilled water at 20°C is 0·070 N/m. Tests were conducted at surface tensions of 0·070, 0·060, 0·055, 0·045 and 0·035 N/m by increasing the concentration of the additive. The surface tension was measured on samples of water with a laboratory surface tension analyser.

Initially, we simply estimated void fraction from the difference in height between the column of water at rest and the two-phase mixture. This worked well for distilled water. However, with the addition of the surfactant, the height of the two-phase mixture became ill-defined because of foam formation. Thus, another technique was needed. The void fraction, ε_g, was deduced from the difference in pressure, $P_2 - P_1$, between pressure taps, a known height, h, apart on the test section. Since the density of water, ρ, is much larger than air in this case, it can be shown that $\varepsilon_g = (P_2 - P_1)/(\rho g h)$ where g is the acceleration due to gravity. The pressures were simply measured with manometers made of plastic tubing. This method worked well, except when the flow became very turbulent at high void fractions. Measurements using both techniques, where possible, were in good agreement.

Both the logarithmic decrement technique and the random vibration response technique were used to measure damping. For the logarithmic decrement technique, the tube was displaced transversely near the end with a quick-release device. This device was supported by a separate column to prevent structural coupling between the tube and the quick-release device. The resulting transverse vibration was measured with two pairs of strain gauges mounted at 90° from each other, inside the tube, near the tubesheet. One strain gauge pair measured vibration in the direction of the initial displacement imposed by the quick-release device. The other pair was used to make sure any orthogonal component of vibration was minimum. Usually, three or four logarithmic decrement measurements were done for each test condition. Repeatability was generally within $\pm$ 5%.

At higher void fraction, the random turbulence generated in the air–water mixture was sufficient to excite measurable tube vibration. Assuming that the resulting excitation is essentially broadband random, it is possible to deduce damping from the vibration response spectrum, since it is essentially the same as the frequency response function. As explained in Pettigrew *et al.* (1985), a least-squares regression analysis technique is used to obtain the best-fit curve for the frequency response function (see Figure 6). Damping can be obtained

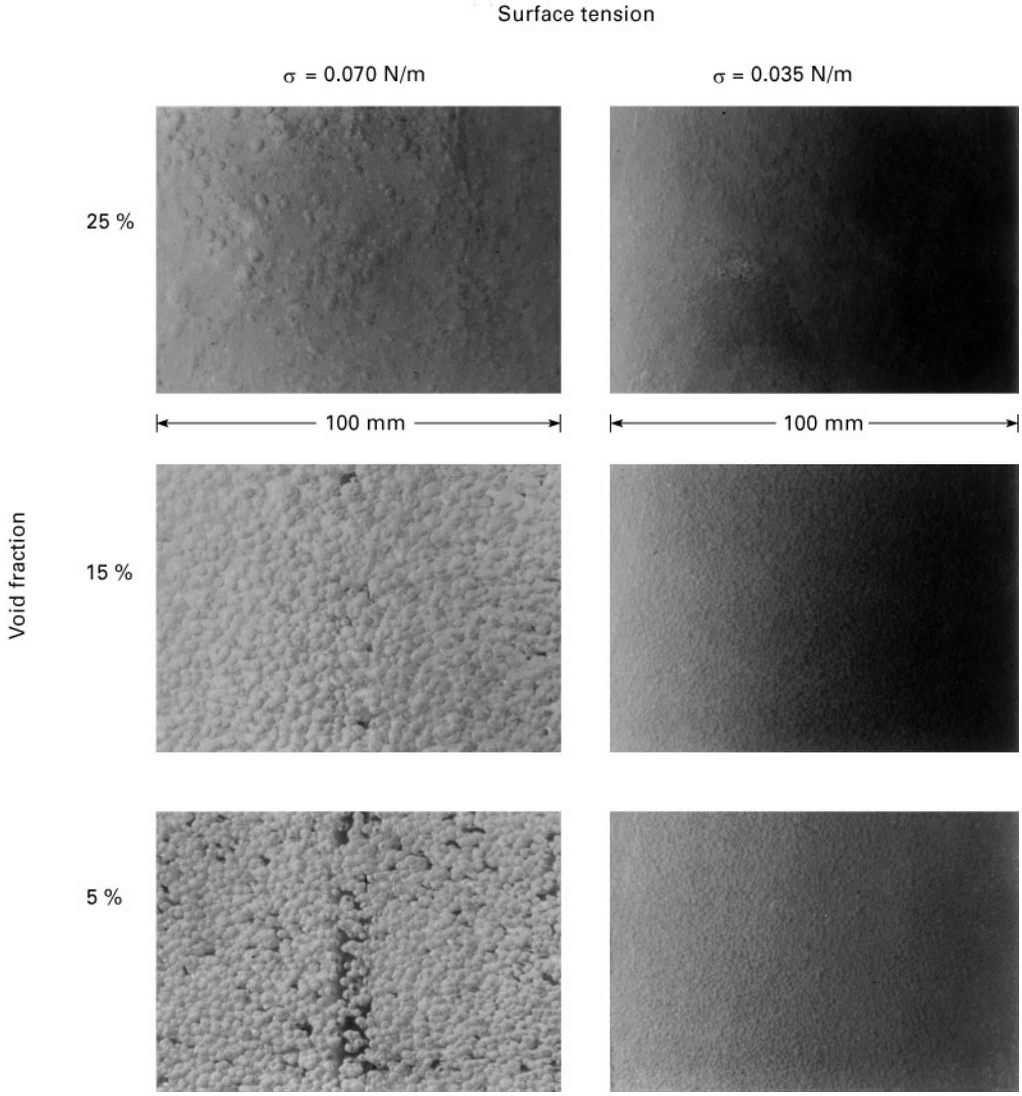

Figure 5. Photographs of flow regime in air–water mixtures showing the effect of surface tension and void fraction.

directly from this curve assuming a single-degree-of-freedom system. In some cases we measured damping using both techniques. The results were in good agreement.

Wherever possible, the logarithmic decrement technique was used, since it is faster and simpler. The random response method was used when the vibration response due to turbulence was too high and interfered with the logarithmic decrement measurements.

Damping measurements were taken in air to estimate the contribution of structural damping. Damping ratios of about 0·01% were measured, which is two orders of magnitude lower than typical test results. Thus, structural damping is very small compared to fluid damping, which is desirable, since only the latter is of interest here.

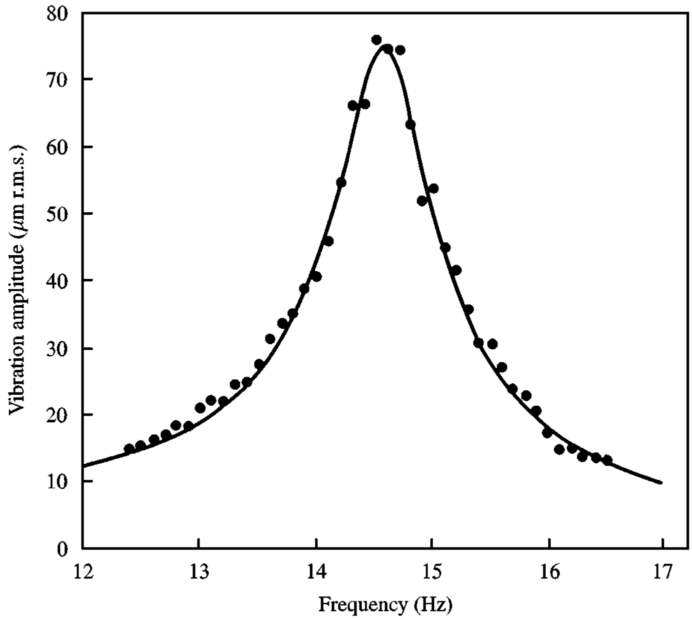

Figure 6. Random response method to evaluate damping: best-fit frequency response curve: —, curve fit; ●, experimental data. Void fraction = 15%, ζ = 2·7%, and f = 14·6 Hz.

We carried out nearly 200 tests covering the following nominal range of parameters: void fraction ε_g, 0–30%; tube frequency f, 7–230 Hz; surface tension σ, 0·035–0·070 N/m; and degree of confinement, D_e/D, 1.9–11.7. The results are presented in Figures 7–10.

4. ANALYSIS OF RESULTS

Damping in water (i.e. $\varepsilon_g = 0\%$) is mostly due to viscous damping. Damping in two-phase mixtures is generally larger than that due to viscous damping alone based on two-phase parameters. Thus, there appears to be a two-phase mechanism of damping in addition to viscous damping. Therefore, we propose that fluid damping, ζ, in two-phase mixtures is due to a viscous component, ζ_V, and a two-phase component, ζ_{TP}, or

$$\zeta = \zeta_V + \zeta_{TP}. \tag{1}$$

This approach was originally suggested by Carlucci (1980).

The viscous component of damping in two-phase mixtures is taken to be analogous to viscous damping in single-phase fluids, as discussed in Pettigrew *et al.* (1986):

$$\zeta_V = \frac{100\pi}{\sqrt{8}}\left(\frac{\rho_{TP}D^2}{m}\right)\left(\frac{2\nu_{TP}}{\pi f D^2}\right)^{1/2}\left\{\frac{[1+(D/D_e)^3]}{[1-(D/D_e)^2]^2}\right\}, \tag{2}$$

where ζ_V is the viscous damping ratio in percent (of the critical damping), ρ_{TP} is the density of the two-phase mixture, ν_{TP} is the equivalent kinematic two-phase viscosity, D is the tube diameter, m is the tube mass per unit length including the hydrodynamic mass, m_h, f is the tube frequency, and D_e is the inside diameter of the containment tube. This formulation follows the fundamental work done by Chen *et al.* (1976) on this topic. Following

MacAdams *et al.* (1942), the equivalent two-phase viscosity is defined as

$$v_{TP} = \frac{v_l}{1 + \varepsilon_g(v_l/v_g - 1)}, \tag{3}$$

where v_l and v_g are the kinetic viscosities of the liquid phase and gas phase, respectively, and ε_g is the void fraction.

The measured total damping values, ζ_T, include some structural damping, ζ_S. Rogers *et al.* (1984) and Yeh & Chen (1978) show that ζ_T can be formulated in terms of the different damping components by

$$\zeta_T = \zeta_S\left(\frac{m_t}{m_t + m_h}\right)^{1/2} + \zeta_V + \zeta_{TP}. \tag{4}$$

To obtain the two-phase component of damping alone, structural and viscous components of damping are subtracted as follows:

$$\zeta_{TP} = \zeta_{\mathrm{T}} - \zeta_V - \zeta_S\left(\frac{m_t}{m_t + m_h}\right)^{1/2}. \tag{5}$$

This process is illustrated for a typical damping value. For $\varepsilon_g = 20\%$, $f = 7{\cdot}2$ Hz, $D_e/D = 11{\cdot}7$ and $\sigma = 0{\cdot}070$ N/m, the total damping ratio, ζ_T, was measured to be 4·8%. For these test conditions, $\rho_{TP} = 800\,\mathrm{kg/m^3}$, $v_l = 10^{-6}\,\mathrm{m^2/s}$, $v_g = 1{\cdot}8\times10^{-4}\,\mathrm{m^2/s}$, $m_t = 0{\cdot}30$ kg/m, and $m_h = 0{\cdot}13$ kg/m. From equation (3) $v_{TP} = 1{\cdot}25\times10^{-6}\,\mathrm{m^2/s}$ and from equation (2) the viscous component of damping ζ_v is calculated to be 1·13%. As discussed earlier, $\zeta_S = 0{\cdot}01\%$; thus, using equation (5), $\zeta_{TP} = 3{\cdot}58\%$ is obtained. Note that the structural damping has little effect on the results since it is very small, as it was purposely minimized in this experiment. The two-phase component of damping, ζ_{TP}, is the pertinent parameter in the following discussions.

5. RESULTS

5.1. Effect of Void Fraction

The effect of void fraction on two-phase damping is dominant, as expected. This is shown in Figure 7(a, b, c, d) for tube frequencies of 7, 28, 54 and 110 Hz, respectively. Damping increases significantly between 0 and 25% void fraction. In most cases, damping appears to reach a plateau above roughly 20% void fraction, which is similar to the results of Figures 1 and 2.

The maximum two-phase damping ratio varies between 2 and 4% depending on frequency and surface tension, as discussed later. We do not yet understand the basic energy-dissipation mechanisms in two-phase mixtures. Obviously, the fluid forces that control the structure of two-phase flow increase with void fraction. It is reasonable to assume that the energy-dissipating forces are related to the fluid forces in two-phase flow. Fluid forces, flow regime, damping forces, void fraction and surface tension are obviously all related to one another.

5.2. Effect of Surface Tension

Two-phase damping ratios are presented against surface tension in Figure 8(a, b, c) for tube frequencies of 7, 28 and 54 Hz, respectively.

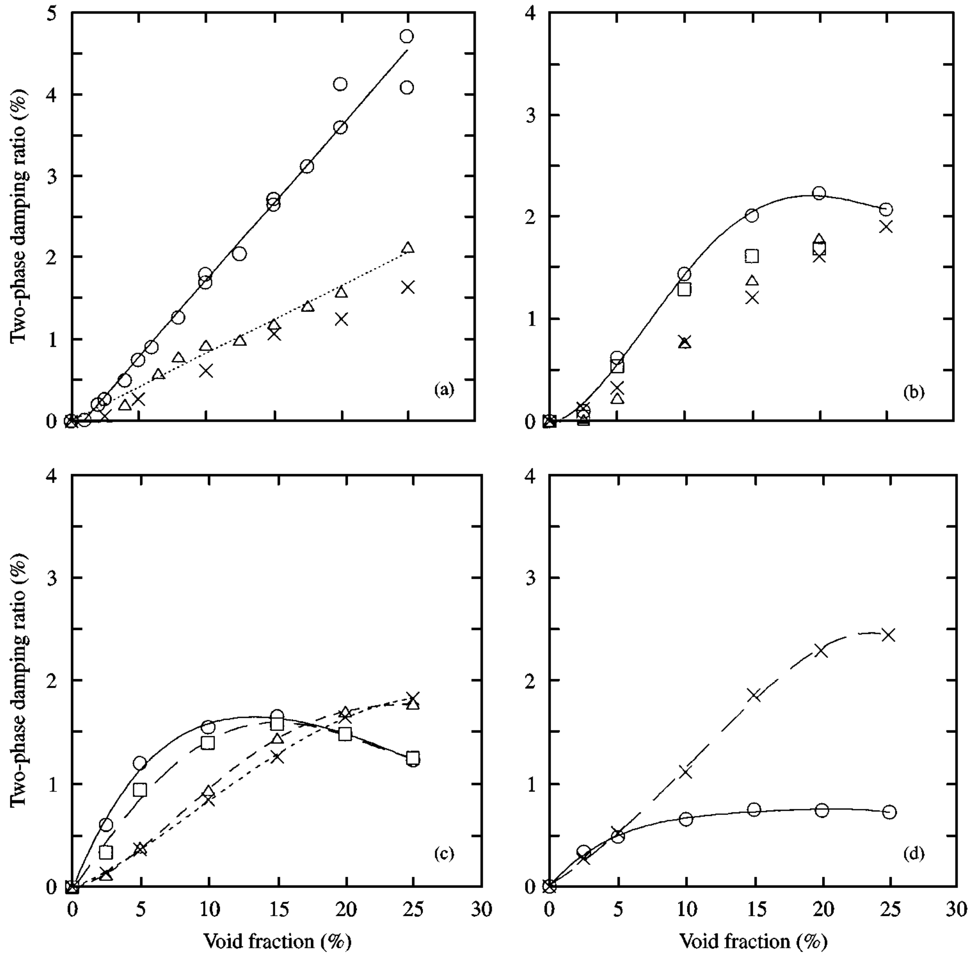

Figure 7. Effect of void fraction on two-phase damping for tube frequencies of (a) 7 Hz, (b) 28 Hz, (c) 54 Hz, (d) 110 Hz. Surface tension, σ (N/m): ○, 0·070; □, 0·055; △, 0·045; ×, 0·035.

Generally, damping increases with surface tension. Although this relationship is not simple, we attempted to correlate the damping ratio, ζ_{TP}, as a function of surface tension, σ, such that

$$\zeta_{TP} \propto \sigma^n, \tag{6}$$

where n is an exponent, found to vary between $-0{\cdot}6$ and 2·0. However, for tube frequencies of 7 and 28 Hz, the exponent n is generally not far from unity, indicating a direct dependence on the surface tension.

This model is based on the assumption that the fluid forces governing the structure (i.e. bubble size and distribution) of the two-phase mixture are related to surface tension and that energy dissipation occurs as this structure is broken down or reformed (i.e. bubbles forming or breaking). However, this relationship is very much affected not only by void fraction, as expected, but also by frequency in a most unexpected way. For instance, the exponent n decreases from 1·4 for 7 Hz to $-0{\cdot}6$ for 54 Hz at a void fraction $\varepsilon_g = 25\%$, whereas for $\varepsilon_g = 5\%$, n increases from 1·4 to 2·0.

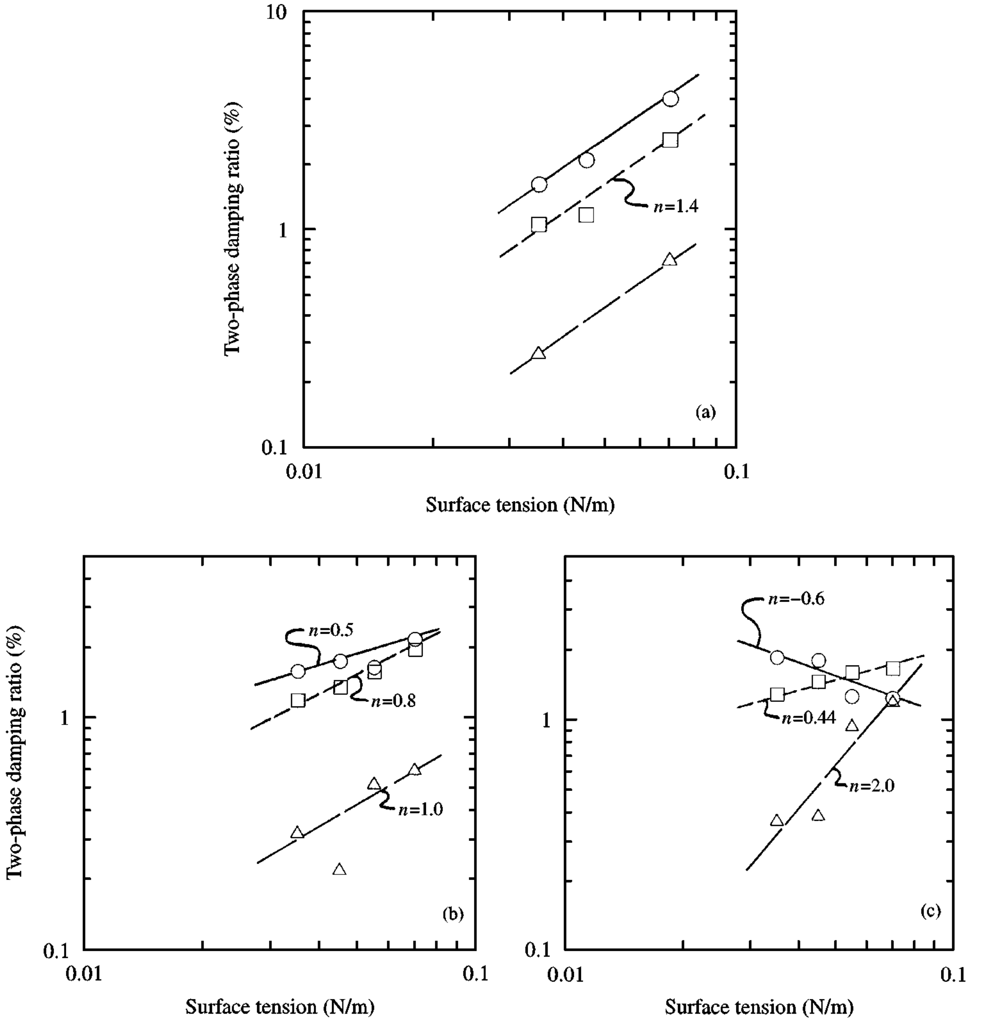

Figure 8. Effect of surface tension on damping (i.e. $\zeta \propto \sigma^n$) for tube frequencies of: (a) 7 Hz; (b) 28 Hz;; (c) 54 Hz. Void fraction: △, 5%; □, 15%; ○, 25%.

The damping versus surface tension relationship is obviously much more complicated than we originally hoped. It appears that damping is related to the scale of the two-phase flow structure (i.e. higher-frequency damping is related to smaller-scale phenomena or flow structure). For example, smaller bubbles may not break so easily, and thus do not dissipate as much energy as larger bubbles. The reader must appreciate that the above is very speculative. This topic requires further thought.

5.3. Effect of Frequency

In an attempt to isolate the effect of frequency, the damping information is presented in terms of frequency in Figure 9(a, b, c, d) for void fractions of 5, 10, 15 and 25%, respectively.

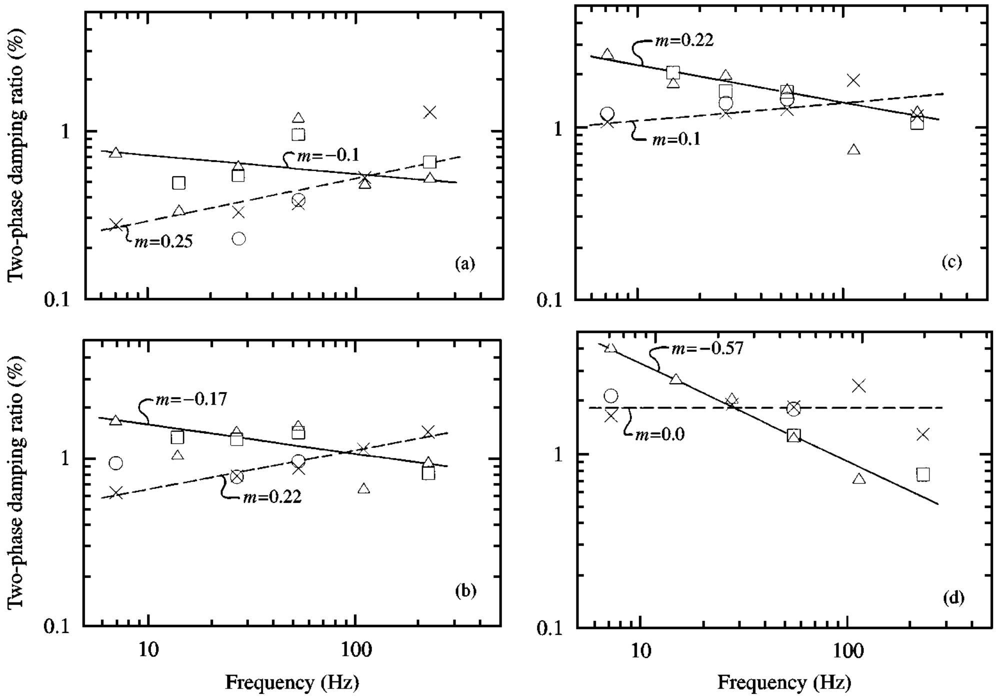

Figure 9. Effect of frequency on damping (i.e., $\tau_{TP} \propto f^m$). Surface tension: △, 0·070; □, 0·055; ○, 0·045, ×, 0·035 N/m. Void fraction: (a) 5%; (b) 10%; (c) 15%; (d) 25%.

Generally, two-phase damping is not too dependent on frequency. A simple relationship of the form

$$\zeta \propto f^m \tag{7}$$

was used to correlate the data. This is analogous to the viscous damping theory for which the exponent $m = -0{\cdot}5$. However, the situation is quite different for two-phase damping. The frequency exponent, m, tends to decrease slightly with void fraction. It is also slightly lower for higher surface tension. For example, m decreases from 0·25 to 0 for a void fraction increase of 5–25% at a surface tension of $\sigma = 0{\cdot}035$ N/m. Similarly, m decreases from $-0{\cdot}10$ to $-0{\cdot}57$ at $\sigma = 0{\cdot}070$ N/m [see Figure 9(a, d)]. There is no obvious explanation for this behaviour. Two-phase damping is undoubtedly a very complex energy-dissipation mechanism.

5.4. Effect of Confinement

The effect of confinement was studied by decreasing the inside diameter, D_e, of the test-section from 152 to 51 and 25 mm. The results, shown in Figure 10(a, b) do not indicate well-defined trends. Generally, confinement does not appear to play a major role on damping for the limited range of confinement studied here (i.e. D_e/D of 1·9, 3·7 and 11·7). In Figure 10(a), for $\varepsilon_g \geqslant 20\%$, the damping values for $D_e/D = 1{\cdot}9$ and 3·9 are somewhat lower than for $D_e/D = 11{\cdot}7$. This is unexpected, since standard damping theory would predict the

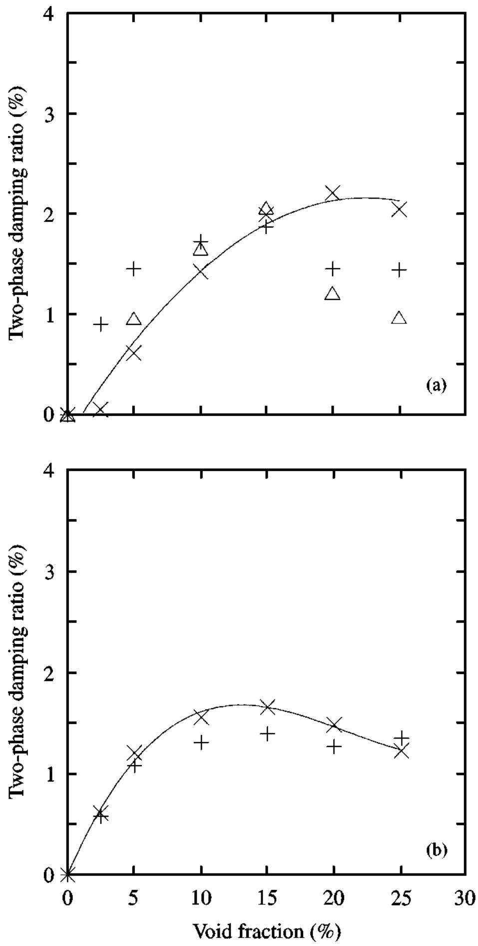

Figure 10. Effect of confinement on two-phase damping. Confinement, D_e/D: $\triangle$, 1·9; $+$, 3·9; $\times$, 12. Tube frequency: (a) 29 Hz; (b) 54 Hz.

opposite. A possible explanation is that confinement can also have a very significant effect on flow regime, which in turn can affect the damping behaviour. In practice, much narrower confinements are possible (i.e. $D_e/D < 1{\cdot}5$), and should be studied in future tests.

5.5. Effect of Vibration Amplitude

The peak vibration amplitude during the logarithmic decrement tests varied between 50 and 500 μm. This covers the range of maximum-tolerable vibration amplitude in real heat exchangers. The effect of vibration amplitude on damping was investigated as shown in Figure 11 for a typical test. We found damping to be linear over the range of two-phase mixtures studied.

We monitored vibration amplitude during damping measurements using the random vibration response method. The vibration response is largely affected by surface tension. As shown in Figure 12, vibration response is a factor of 10 larger at the higher surface tension

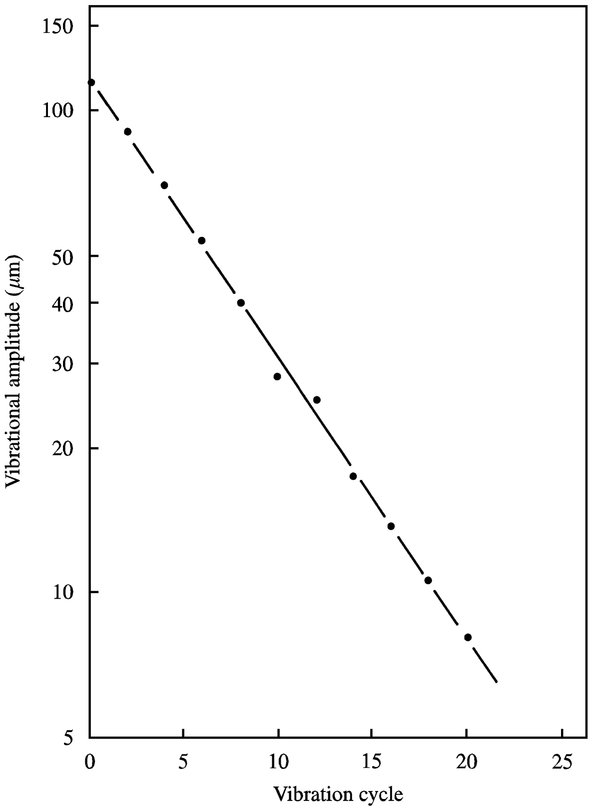

Figure 11. Vibration amplitude decrease versus cycle of motion in logarithmic decrement test showing linearity. Test data: $f = 54$ Hz; $\varepsilon_g = 20\%$; $\sigma = 0{\cdot}045$ N/m; $D_e/D = 11{\cdot}7$; $\zeta = 2{\cdot}11\%$.

(i.e. $\sigma = 0{\cdot}035$ versus $\sigma = 0{\cdot}070$ N/m). This may be explained in terms of the structure of two-phase mixtures. As shown in Figure 5, the structure is much coarser, and thus hydraulically noisier at the higher surface tension. This finding is of great practical significance. Vibration response to random turbulence excitation in realistic two-phase flows may be very different from that predicted from information derived from air–water tests. For example, steam–water flow at higher temperatures may be much smoother, thereby generating less vibration excitation.

5.6. Hydrodynamic Mass

The random response method to evaluate damping also provided an accurate estimate of tube frequency. Hydrodynamic mass, m_h, can be deduced from the tube frequency using the following relationship (Carlucci 1980):

$$m_h = m_t[(f_a/f)^2 - 1], \tag{8}$$

where m_t is the mass of the tube alone, f_a and f are the tube frequencies in air and in the two-phase mixture, respectively. The results are shown in Figure 13. Hydrodynamic mass decreases linearly with void fraction, as expected. The rate at which it decreases is greater

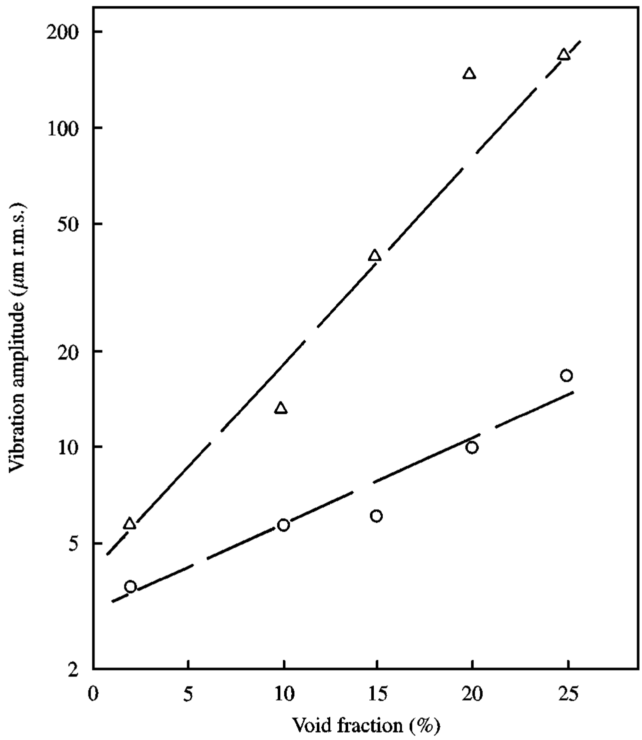

Figure 12. Effect of surface tension on vibration response to mixture turbulence. Frequency = 7 Hz. Surface tension (N/m): ○, 0·035; △, 0·070.

than that expected from homogeneous two-phase mixture density. This trend has been observed before by Carlucci (1980) in two-phase annular flow.

6. DISCUSSION

As this paper was prepared, further work was done on two-phase damping. In particular, a design guideline was developed for damping of tube bundles in two-phase cross-flow (Pettigrew *et al.* 1994). Also, a review paper on two-phase flow-induced vibration was prepared (Pettigrew & Taylor 1994). Probably the most relevant work related to two-phase damping was done on a rotated triangular tube bundle subjected to two-phase Freon 22 cross-flow (Pettigrew *et al.* 1995). It shows that, for this configuration, damping is similar to that in air–water, although the surface tension in Freon 22 is smaller by a factor of nine. Thus the effect of surface tension on damping cannot be generalized to all two-phase flow situations. It appears that the nature of the flow regime is very important. In the experiment described here the flow is essentially axial to the tube, the void fraction is relatively low, and the flow regime is definitely bubbly, as shown in Figure 5. On the other hand, the flow regime in the two-phase cross-flow experiment reported by Pettigrew *et al.* (1995) was very different because of the tortuous flow path through the tube bundle and because of the broad range of void fraction and mass fluxes.

Thus it would seem that this paper is raising more questions than it answers. Further work in this area will require a thorough understanding of flow regime. The true nature of

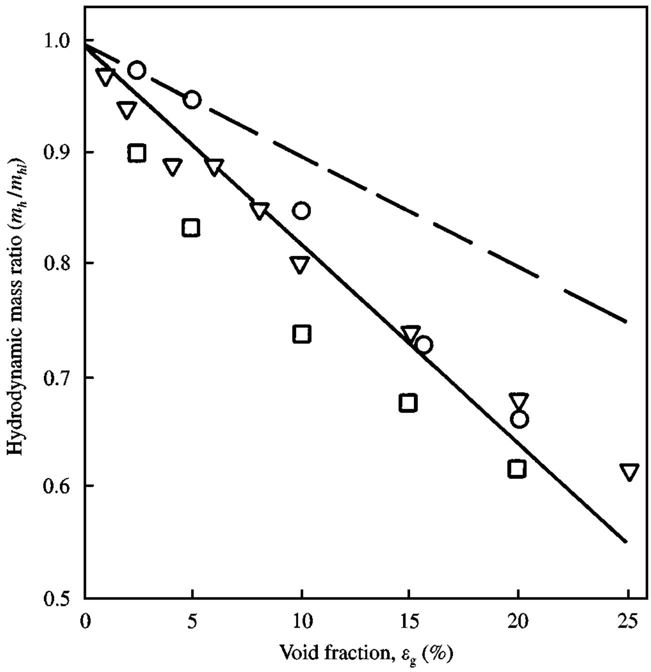

Figure 13. Effect of void fraction on hydrodynamic mass for $\sigma = 0{\cdot}070$ N/m and $D_e/D = 11{\cdot}7$. Tube frequency: ∇, 7 Hz; ○, 28 Hz; □, 220 Hz. Curves: ——, $m_h/m_{hl} = 1{\cdot}0 - 1{\cdot}8\varepsilon_g$; - - -, $m_h/m_{hl} = 1 - \varepsilon_g$, i.e. proportional to mixture density.

energy dissipation mechanisms in two-phase mixtures is still unknown. It is hoped that this paper will stimulate and help further research on this topic.

7. CONCLUDING REMARKS

A simple experiment was done to study damping of heat-exchanger tubes in two-phase mixtures up to 25% void fraction. Although the relationships between damping and void fraction, surface tension and tube frequency appear very complex, the following has been found.

(i) The effect of void fraction on two-phase damping is dominant. Generally, damping increases with void fraction (within the range of void fraction considered herein).

(ii) Surface tension is an important parameter. Two-phase damping generally increases with surface tension.

(iii) Generally, the dependence of two-phase damping on tube frequency is weak.

(iv) Vibration response to random turbulence excitation in two-phase mixtures is strongly related to surface tension.

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APPENDIX: NOMENCLATURE

D	tube diameter
D_e	test section inside diameter
f	tube frequency in fluid
g	acceleration due to gravity
l	tube length
m	tube mass per unit length including m_h

m_h	hydrodynamic mass per unit length
P	tube pitch
w	tube wall thickness
ε_g	void fraction, %
ζ	damping ratio, %
ν	kinematic viscosity
σ	surface tension

Subscripts

a	in air
g	gas phase
h	hydrodynamic
l	liquid phase
t	tube
s	structural
T	total
TP	two-phase
V	viscous