Journal of Fluids and Structures 17 (2003) 351–364

JOURNAL OF FLUIDS AND STRUCTURES

www.elsevier.nl/locate/jnlabr/yjfls

Scaling of damping induced by bubbly flow across tubes

F. Baj[a], E. de Langre[b,*]

[a] Département Modélisation de Systèmes et Structures, Commissariat à l'Energie Atomique Saclay, 91191 Gif-sur-Yvette France
[b] Département de Mécanique, LadHyX, CNRS-École Polytechnique, 91128 Palaiseau, France

Received 26 March 2002; accepted 10 September 2002

Abstract

The damping of tubes subjected to two-phase air–water bubbly cross-flow is investigated with the use of an experimental database from several authors. A new definition of damping in stagnant flow is proposed using an extrapolation of the measured values at low dimensionless flow velocities. This approach yields values of damping substantially lower than those currently defined in the literature. They are found to vary continuously with void fraction, within the bubbly flow regime. These data are used to compare several models of the equivalent viscosity of a two-phase mixture. The effect of the flow velocity is then analysed up to fluidelastic instability. It is observed that, using scaling factors based on the characteristics of the liquid phase, fluidelastic effects of bubbly flows are closely related to those known in single-phase flows.

1. Introduction

Many industrial components operate with two-phase flows across tube bundles, such as heat exchangers and nuclear steam generators. Better performance often requires higher flow velocities, while reduced structural support is desirable to minimize manufacturing costs. High flow velocities may lead to severe flow-induced vibrations and fatigue or fretting-wear. Flow-induced vibrations in tube bundles are generally considered as the consequence of two distinct mechanisms (Chen, 1987) namely random buffeting forces caused by fluctuations inherent to the flow and fluidelastic forces which depend on the motion of the vibrating tubes. Through their influence on damping, fluidelastic forces are the cause of instabilities. In order to avoid excessive flow-induced vibrations, it is essential to acquire a better knowledge of the behaviour of damping in tube bundles subjected to two-phase cross-flow, for two reasons: (i) in predictive analysis, the calculation of the response of a tube subjected to fluid forces requires knowledge of damping in flow conditions, (ii) criteria for fluidelastic instability are generally formulated in terms of a reduced flow velocity and a dimensionless mass-damping parameter (Blevins, 1990) which requires a value of damping in stagnant fluid, even for two-phase mixtures. More fundamentally, a recent paper (Pettigrew and Knowles, 1997) concluded that "the true nature of energy dissipation mechanisms in two-phase mixtures is still unknown".

Two-phase air–water mixtures are considered in this paper. Though this mixture is not necessarily representative of the steam–water mixture that flows in heat exchangers, it is commonly used to investigate fluid–structure effects in two-phase flows. Following most flow-induced vibration analysis (Blevins, 1990; Chen, 1991; Pettigrew and Taylor, 1994), it is convenient here to use the homogeneous model where both liquid and gas phases are assumed to have the same velocity. More refined models will be discussed further. When considering test sections such as in Fig. 1, the

*Corresponding author. Tel.: +33-1-69-33-36-01; fax: +33-1-69-33-30-30.
E-mail address: delangre@ladhyx.polytechnique.fr (E. de Langre).

0889-9746/03/$ - see front matter
PII: S0889-9746(02)00142-1

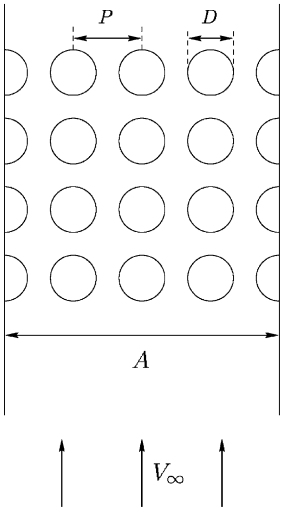


Fig. 1. Main characteristics of a tube bundle in a test section.

homogeneous void fraction is defined as the ratio of the gas flow rate to the total flow rate of the two-phase mixture, namely

$$\alpha = \frac{Q_g}{Q_l + Q_g}, \tag{1}$$

where Q_g and Q_l are the volume flow rates of gas and liquid. The mixture density ρ is given by

$$\rho = \alpha\rho_g + (1-\alpha)\rho_l \tag{2}$$

and the upstream and gap velocities, V_∞ and V, respectively, are defined as

$$V_\infty = \frac{\rho_g Q_g + \rho_l Q_l}{\rho A}, \quad V = V_\infty \frac{P}{P-D}, \tag{3}$$

where A is the test section area, D is the diameter of the tube and P is the pitch of the array.

The total damping ξ of a tube in fluid may be divided into a structural component, ξ^s, and a fluid component, ξ^f, also called added damping (Carlucci and Brown, 1983)

$$\xi = \xi^s + \xi^f. \tag{4}$$

The added damping in quiescent fluid, which we shall refer to as "quiescent fluid damping", ξ^f_o, may be easily defined for single-phase fluids. It is the part of the system damping that originates from the effects of the viscosity of the surrounding nonmoving fluid. It may be experimentally obtained from the response of a tube to a known excitation or estimated from analytical considerations for a vibrating tube in a confined viscous fluid. When the Stokes number, $St = fD^2/\nu$, is large enough ($St \geqslant 2100$), the solution of Stokes equations yields (Rogers et al., 1984)

$$\xi^f_o = \xi^v = \frac{\pi}{\sqrt{8}}\left(\frac{\rho D^2}{m}\right)\left(\frac{2}{\pi\,\mathrm{St}}\right)^{1/2}\frac{1+\gamma^3}{(1-\gamma^2)^2}, \tag{5}$$

where $\gamma = D/D_e$ is the ratio between the diameter of the tube D and the diameter of the flow boundary D_e, m is the total mass per unit length including the hydrodynamic mass, f is the frequency of the tube motion and ν is the kinematic viscosity. Comparisons with experimental results show good agreement (Pettigrew et al., 1986).

Experimentally, it is not feasible to maintain a stagnant liquid–gas mixture, the densities of each phase being different. The concept of quiescent fluid damping was nevertheless extended for two-phase mixtures, using experimental data at low flow velocities (Carlucci and Brown, 1983). Experimental damping in two-phase air–water was observed to be much higher than the calculated damping in air or in water flow, and even in two-phase flow using the equivalent two-phase viscosity

$$\nu = \frac{\nu_l}{1 + \alpha((\nu_l/\nu_g) - 1)}, \tag{6}$$

proposed by McAdams et al. (1942). Therefore, it has been assumed in the literature (Carlucci and Brown, 1983) that a mechanism of damping specific to two-phase mixtures exists in addition to viscosity so that the total fluid damping in almost quiescent fluid should read

$$\xi_o^f = \xi^v + \xi^{tp}. \tag{7}$$

Using large sets of experimental data, Pettigrew and Taylor (1997) proposed a lower bound of ξ^{tp}, convenient for design purposes,

$$\xi^{tp} = 4f(\alpha)\left(\frac{\rho_l D^2}{m}\right)\frac{1+\gamma^3}{(1-\gamma^2)^2}, \tag{8}$$

where the void fraction function $f(\alpha)$ is taken as 1.0 for α from 0.4 to 0.7, $\alpha/0.4$ for $\alpha<0.4$, and $1.0-(\alpha-0.7)/0.3$ for $\alpha>0.7$.

In this paper, we propose an alternative approach for the definition of quiescent fluid damping in two-phase bubbly mixtures and of its behaviour with flow velocity. In Section 2, we describe the experimental database used throughout the paper. Damping at low reduced velocities is considered in Section 3 and the behaviour with flow velocity in Section 4. The physical meaning of the results and their use are discussed in Section 5.

2. Experimental database

Several experimental results on flow-induced vibrations in two-phase flow may be found in the literature, see for instance a review in de Langre and Villard (1998). Most of these tests were not primarily aimed at the measurement of damping. We shall therefore use for our purpose specific tests in air–water, from Taylor et al. (1988) and Taylor (1994), Axisa et al. (1989) and a new experimental programme described further in Baj (1998) and Baj and de Langre (1999). The general characteristics of these tests are given in Table 1.

Taylor et al. (1988) measured damping of a cantilever tube in a tube row. Detailed data from this test programme may be found in Taylor (1994) and the retained values, in the range of void fraction [20%, 80%], are given in Table 2.

As described in Axisa et al. (1989) tests in two-phase air–water mixtures have been conducted on a flexible tube inserted in a rigid bundle. The central tube is mounted on a flexible plate which allows vibrations in the lift direction only. Here, we consider tests in the range of void fraction [20%, 80%]. A result from a test in water ($\alpha = 0$) on the same

Table 1
General characteristics of tube bundles

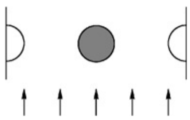
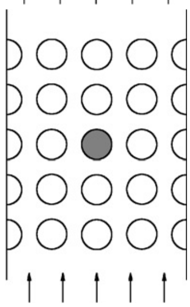
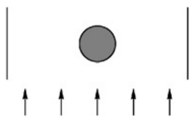
Reference	Bundle	D (m)	P/D	f_{air} (Hz)	ξ^s %
Taylor (1994)		0.03	3	128	0.2
Axisa et al. (1989)		0.03	1.5	33.1	0.15
Baj and de Langre (1999)		0.0133	∞	33	0.2

Table 2
Tests by Taylor (1994)

α (%)	V (m s^{-1})	f (Hz)	ξ (%)
20	0.94	104	3.1
20	1.87	106	4.5
25	1.00	106	3.5
25	2.00	107	4.4
25	4.00	109	5.7
30	0.53	106	2.9
35	0.58	106	3.1
35	1.15	108	3.1
35	2.31	109	4.2
35	4.61	110	4.4
50	0.75	109	2.6
50	1.50	110	2.8
50	3.00	112	3.6
50	5.99	112	3.9
70	9.97	112	4.0
75	1.50	113	2.7
75	3.00	114	2.1
75	5.98	114	3.0
80	1.87	114	2.1
80	3.73	115	2.1
80	7.47	114	2.7

system is also considered from Hadj-Sadok et al. (1995). The test results are given in Table 3. Data points with strong vortex shedding effects have been excluded.

Finally, a new set of tests is considered, which have been carried out on a single cantilever tube subjected to air–water cross-flow (Baj and de Langre, 1999). The tube is horizontally mounted and subjected to vertically upward flow, see Fig. 2. The test section has a rectangular cross-section of 70×100 mm^2. The tube is mounted on a flexible plate which allows vibration in the lift direction only. The data may be found in Table 4.

All these test data may now be analysed in terms of flow regimes according to the pattern map developed by Ulbrich and Mewes (1994) for flow across tube bundles. Fig. 3 shows that the entire set of experimental data considered here falls into the domain of bubbly flow.

3. Damping in quiescent fluid

In this section, we consider how the concept of quiescent fluid damping in bubbly flow may be defined from the set of experimental data. We still assume that the measured total damping ratio ξ is composed of a structural component ξ^s and a fluid component ξ^f as in single-phase flow, but the latter is now considered as a global entity in contrast with previous approaches.

In order to compare results from one configuration to another, the fluid damping ξ^f needs to be related to parameters such as the tube mass per unit length, m_t, the hydrodynamic mass per unit length, m_h, the confinement and the diameter of the tube. Carlucci (1980) observed that damping varies with the ratio of hydrodynamic mass over the total cylinder mass, $m = m_t + m_h$. The ratio $\rho_l D^2/m$ was used in the normalization of the two-phase damping by Pettigrew and Taylor (1994). In contrast, we consider here the ratio $\rho D^2/m$ to account for the effective density of the surrounding mixture, using ρ instead of ρ_l. In the case of tube bundles or tube rows, confinement needs also to be taken into account. Chen et al. (1976) showed that confinement affects, respectively, the hydrodynamic mass $m_h = \beta\pi\rho D^2/4$ and the fluid

Table 3
Tests by Axisa et al. (1989)

α (%)	V (m/s)	f (Hz)	ξ (%)	α (%)	V (m/s)	f (Hz)	ξ (%)
0	0.00	18.2	0.60	45	1.36	22.1	10.2
0	0.28	18.1	0.73	45	2.73	22.2	11.1
0	0.56	18.3	0.99	45	3.18	22.3	9.53
0	0.66	18.3	1.07	45	3.64	22.4	7.69
0	0.69	18.2	1.33	45	4.10	22.8	6.10
0	0.83	18.1	1.49	45	4.55	22.9	1.77
0	0.90	18.0	1.87	45	5.00	23.2	0.93
0	1.39	18.0	3.21				
0	1.52	17.3	2.71	55	3.89	24.9	11.7
0	1.66	17.1	1.53	55	4.44	24.9	10.7
0	1.86	16.9	0.57	55	5.00	24.9	6.23
0	1.94	17.7	0.50	55	5.56	25.1	5.23
				55	6.10	25.5	1.63
25	1.00	20.2	8.80	55	6.67	25.6	0.35
25	1.67	19.7	10.5				
25	2.00	19.6	9.29	65	3.57	26.5	8.73
25	2.33	19.3	6.41	65	4.29	26.8	11.0
25	2.47	19.2	5.43	65	5.00	27.2	10.5
25	2.67	19.1	2.53	65	5.71	27.2	12.4
				65	6.43	26.9	7.92
35	1.15	21.0	10.7	65	7.14	26.2	3.52
35	2.30	20.6	9.80	65	7.86	27.4	0.90
35	2.69	20.0	8.10				
35	2.85	20.5	7.10	75	5.00	28.5	10.8
35	3.08	20.3	5.74	75	6.00	28.6	10.9
35	3.46	20.5	2.57	75	9.00	26.9	7.56
35	3.65	20.7	0.85	75	9.50	26.3	4.98
35	3.85	20.6	0.33	75	10.0	27.4	2.74
				75	12.0	30.7	0.92

damping with multiplicative coefficients β and δ given by

$$\beta = \frac{1+\gamma^2}{1-\gamma^2}, \quad \delta = \frac{1+\gamma^3}{(1-\gamma^2)^2} \tag{9}$$

with $\gamma = D/D_e$ where $D_e = (1.07 + 0.56P/D)P$ for square bundles where P is the pitch of the bundle, see Rogers et al. (1984). We also assume that the dependence of fluid damping in two-phase flow with the tube diameter is linear as in single-phase flow, Eq. (5), so that the fluid damping ratio ξ^f should be referred to a reference diameter, D_{ref}, chosen arbitrarily. Here, we use $D_{\mathrm{ref}} = 30$ mm, see also de Langre and Villard (1998).

From these considerations, we may define a normalized fluid damping ratio as

$$\xi_n^f = \frac{\xi^f}{(\rho D^2/m)(D_{\mathrm{ref}}/D)\delta}. \tag{10}$$

This differs from the normalization by Pettigrew and Taylor (1997) by the use of the mixture density ρ instead of the liquid density ρ_l and by a diameter referencing through D_{ref}. Considering now tests at low reduced velocities (Taylor, 1994; Baj and de Langre, 1999), the dependence of the normalized fluid damping ξ_n^f with the reduced velocity V_R is shown in Fig. 4, for void fraction ranges [20%, 30%] to [70%, 80%]. For each range of void fraction, the normalized fluid damping is found to increase with the reduced flow velocity. Note here that low reduced velocities $V_R = V/fD$ are obtained either by low-velocity tests (Baj and de Langre, 1999) or high-frequency tests (Taylor, 1994). Despite the fact that quiescent fluid does not exist for mixtures such as air–water, a value of quiescent fluid damping $(\xi_n^f)_o$ may be easily defined by extrapolating the data to $V_R = 0$ in each range of void fraction, Fig. 4. A test in quiescent water from Axisa et al. (1989) is also considered to obtain some data at $\alpha = 0$. It is remarkable that the value of the normalized fluid

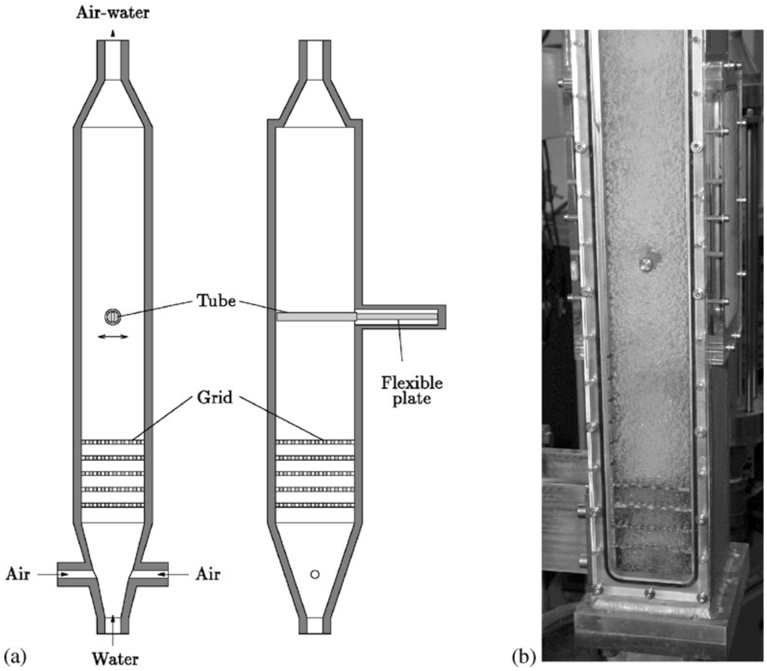


Fig. 2. Experimental apparatus (Baj and de Langre, 1999).

Table 4
Tests by Baj and de Langre (1999)

α (%)	V (m/s)	f (Hz)	ξ (%)
55	0.19	28.8	5.6
55	0.31	29.1	6.1
55	0.39	29.1	6.1
55	0.51	29.3	5.9
65	0.19	29.1	5.0
65	0.31	29.4	5.6
65	0.39	29.5	6.7
65	0.59	29.6	6.0
75	0.20	29.2	4.5
75	0.39	29.6	4.7
75	0.60	29.9	5.6
75	0.72	30.1	5.5

damping in quiescent water seems to agree well with that obtained by extrapolating the air–water data to zero void fraction, Fig. 5.

If we now assume that the extrapolated quiescent fluid damping in two-phase mixtures is exclusively due to some viscous phenomenon, as in single-phase fluid, it is possible to define an apparent vibrational viscosity using Eq. (5) and the results of Fig. 5. The behaviour of this apparent dynamic viscosity with void fraction is plotted in Fig. 6, in comparison with other commonly used models of viscosity by McAdams, Dukler or Cicchitti, see Collier (1981), given in Eqs. (6), (11) and (12), respectively,

$$\mu = \alpha\mu_g + (1-\alpha)\mu_l, \tag{11}$$

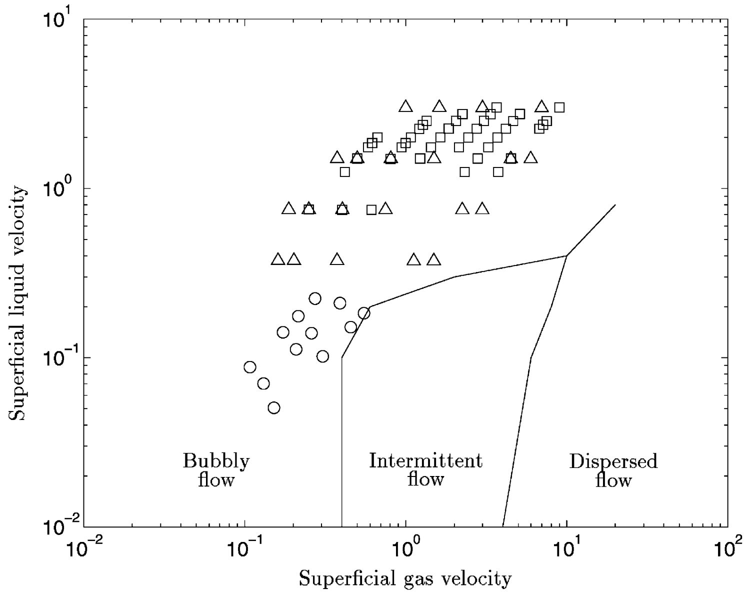


Fig. 3. Test data fall in the bubbly region of the pattern map by Ulbrich and Mewes (1994): △, Taylor et al. (1988); □, Axisa et al. (1989); ○, Baj and de Langre (1999).

$$\rho\mu = \alpha\rho_g\mu_g + (1-\alpha)\rho_l\mu_l, \tag{12}$$

using $\mu_g = 1.5\,10^{-5}\ \mathrm{m^2\,s^{-1}}$ in air and $\mu_l = 10^{-3}\ \mathrm{m^2\,s^{-1}}$ in water.

There is a clear difference between our measured vibrational viscosity and that predicted by more classical models, which were designed to simulate other viscous effects. Note also the difference between these earlier models. This difference is precisely the reason why Pettigrew and Taylor (1994, 1997) introduced an additional two-phase damping coefficient to obtain adequate values of predicted damping.

In the case of vertical bubbly flow parallel to vibrating tubes, a similar behaviour of damping with void fraction has been observed by Hara (1988). This effect was satisfactorily modelled by considering that, in bubbly flow, global dissipation is actually increased by the viscous interaction between vibrating columns of bubbles and the surrounding liquid (Hara, 1993). In other terms, the viscous force acting on the tube was assumed to be multiplied by a factor of $(1 + S_b/S)$, where S_b is some equivalent surface of the bubble columns and S is that of the tube.

This model may be qualitatively adapted to the present case of bubbly flow across a tube by simply considering bubbles around the tube instead of columns of bubbles. Let us consider a section of fluid in the plane perpendicular to the tube. When flow confinement is small, as in the experiments of Baj and of Taylor, the area of fluid influenced by the tube motion is of the order of the tube cross-section, namely $S_T = \pi D^2/4$. Along a length h of the tube, the volume of bubbles is therefore

$$V_b = \alpha S_T h. \tag{13}$$

It may also be written as

$$V_b = N_b \frac{4}{3}\pi\left(\frac{D_b}{2}\right)^3, \tag{14}$$

where N_b is the number of bubbles and D_b is their diameter, supposed to be uniform in our model. A simple model of bubble size (de Langre and Villard, 1998) is $D_b = 0.1D/\sqrt{1-\alpha}$. The interface between the liquid phase and the bubbles has a total area of

$$S_b = N_b 4\pi\left(\frac{D_b}{2}\right)^2, \tag{15}$$

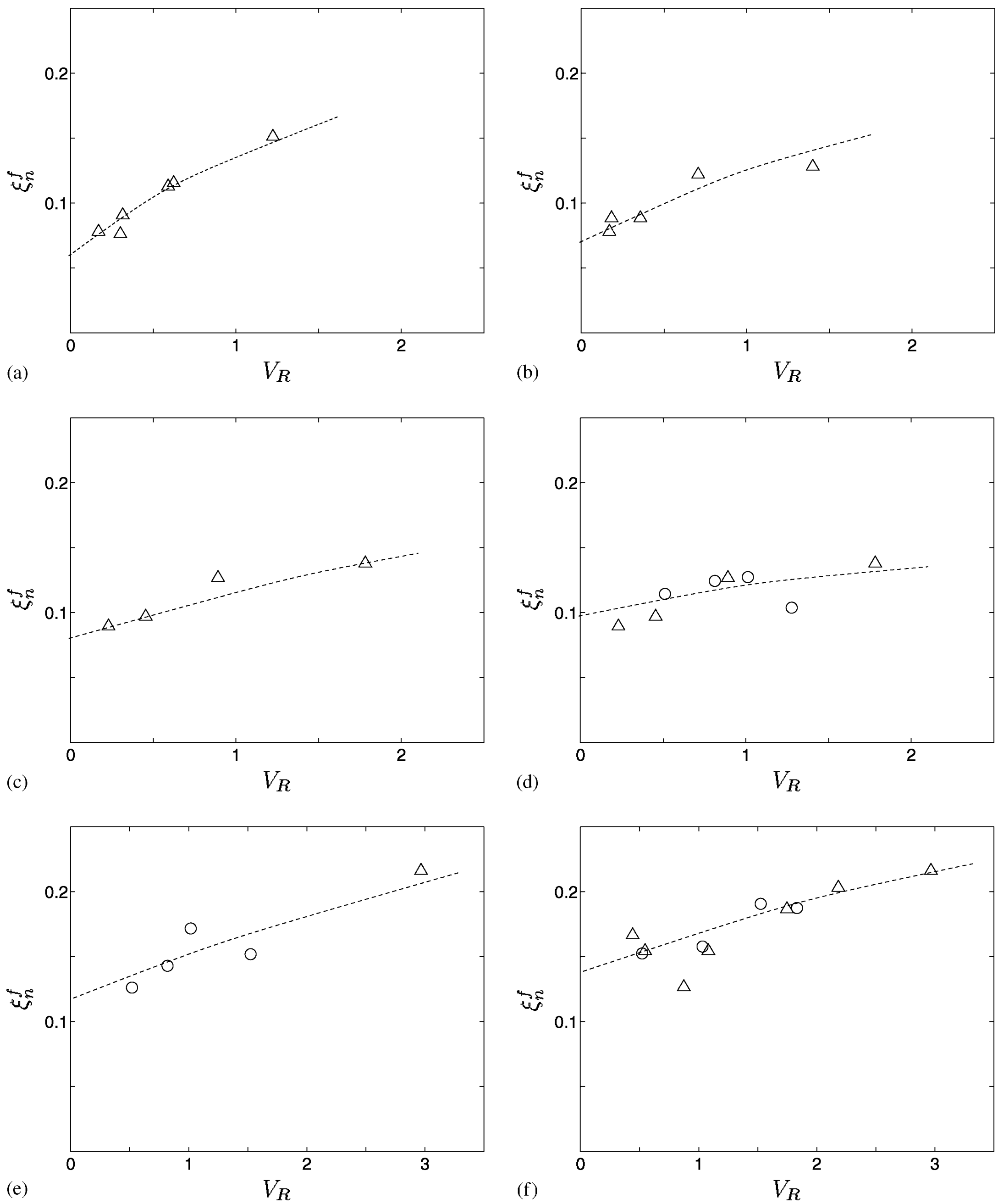


Fig. 4. Normalized fluid damping at low reduced velocity for several ranges of void fraction α: (a) 20–30%; (b) 30–40%; (c) 40–50%; (d) 50–60%; (e) 60–70%; (f) 70–80%. △, Taylor et al. (1988); ○, Baj and de Langre (1999); (.....), quadratic interpolation.

while that of the tube is

$$S = \pi D h. \tag{16}$$

Combining the foregoing equations yields

$$\frac{S_b}{S} = 15\alpha\sqrt{1-\alpha}. \tag{17}$$

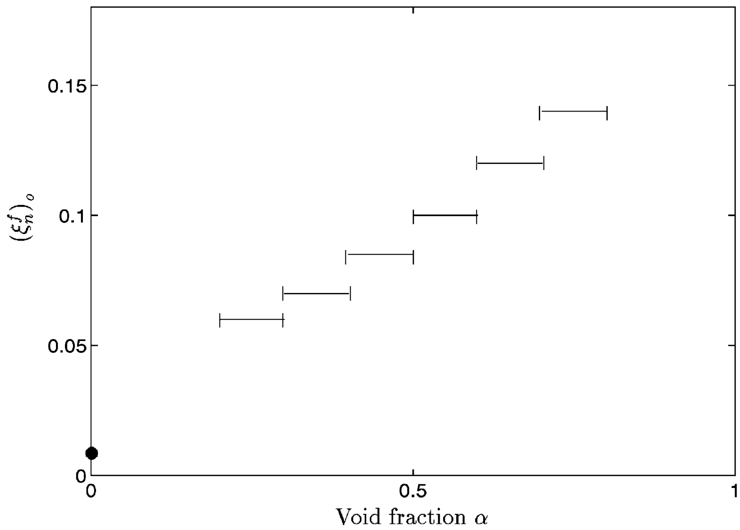


Fig. 5. Continuity of the normalized quiescent fluid damping with void fraction ranges: •, water; —, air–water.

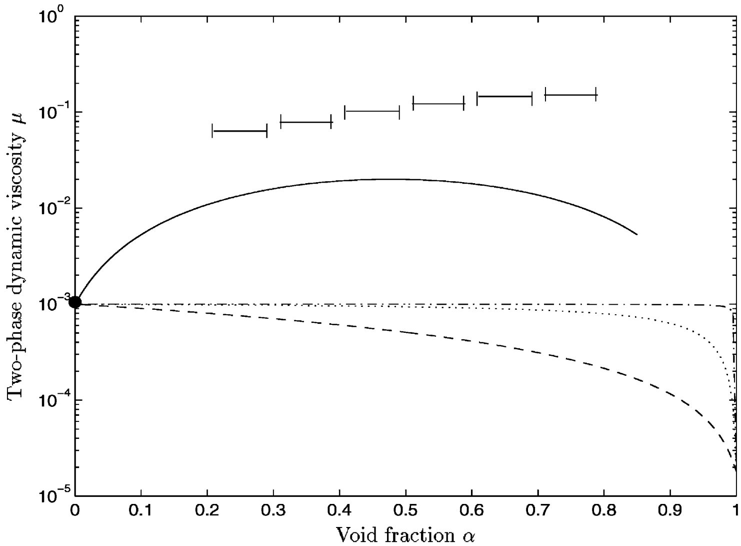


Fig. 6. Equivalent dynamic viscosity μ of a two-phase mixture. Experimental data derived from damping: •, water; —, air–water. Models: (....), McAdams, Eq. (6); (- - -), Dukler, Eq. (11); (–.–), Cicchitti, Eq. (12); (—), Eq. (19) derived from the model of Hara (1988).

The ratio of the average bubble diameter to the tube diameter is therefore fixed in our model. Following Hara (1988), we now assume that the damping increases with void fraction as $(1 + S_b/S)$. As the viscous damping varies as $\nu^{1/2}$ in terms of viscosity, Eq. (5), this results in an apparent two-phase viscosity of

$$\nu = \nu_l \left(1 + \frac{S_b}{S}\right)^2. \tag{18}$$

Using the approximation $\rho \simeq (1 - \alpha)\rho_l$, we have finally

$$\mu = \mu_l (1 - \alpha)\left(1 + 15\alpha\sqrt{1 - \alpha}\right)^2. \tag{19}$$

This equation, when plotted in Fig. 6, is found to give a good qualitative approximation of the effect of void fraction on the apparent viscosity of a bubbly mixture, in terms of vibration damping.

4. Effect of the flow velocity on damping

In this section, we now consider the effect of the velocity of the mixture on damping. This effect is studied here on the same normalized damping ratio, Eq. (10), assuming that the confinement effect, Eq. (9), does not depend on the flow velocity. As in Section 3, several ranges of void fraction are considered. The behaviour of normalized fluid damping with the reduced flow velocity is shown in Fig. 7, considering all tests from the database.

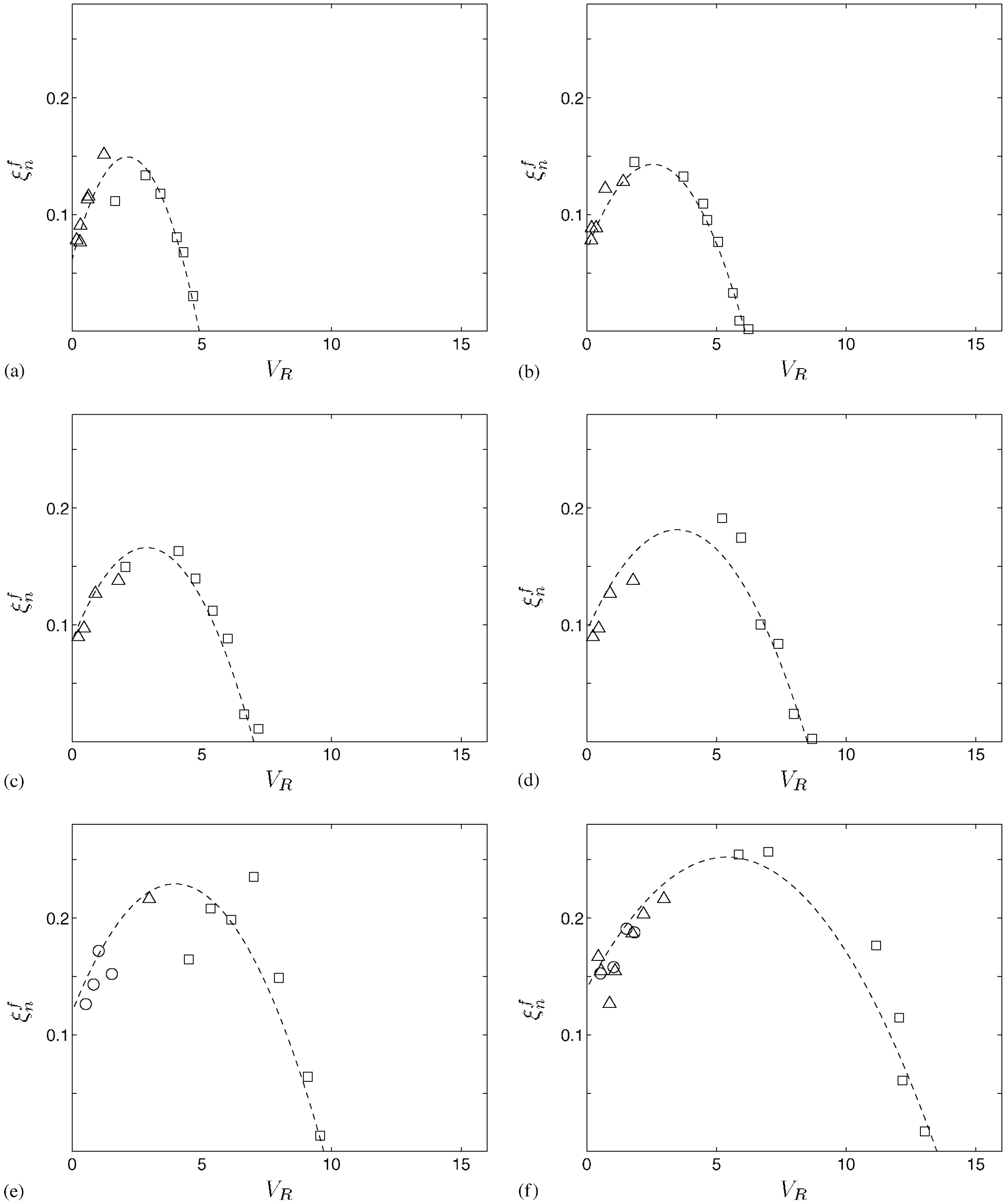


Fig. 7. Normalized fluid damping versus reduced velocity for several range of void fraction α: (a) 20–30%; (b) 30–40%; (c) 40–50%; (d) 50–60%; (e) 60–70%; (f) 70–80%. △, Taylor et al. (1988); □, Axisa et al. (1989); ○, Baj and de Langre (1999); (– – –), quadratic interpolation of the data.

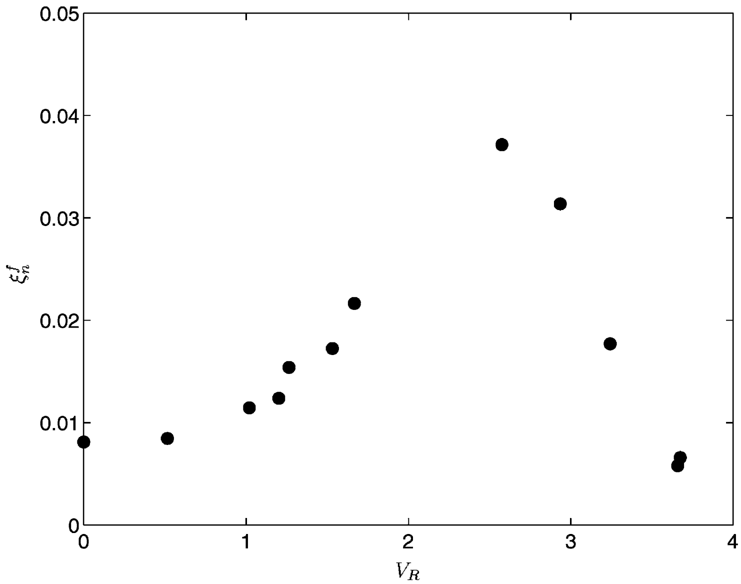


Fig. 8. Normalized damping in water: •, Axisa et al. (1989).

It appears that the reduced velocity does affect damping on the whole range of its values, typically from $V_R = 0$ to 10. Whereas two-phase fluid damping is often considered to be constant for low velocities (Pettigrew et al., 1989), it can be seen in Fig. 7 that damping significantly increases at low reduced velocities as noted in the preceding section. The maximum value of normalized fluid damping is obtained between the third and the half of the maximum velocity considered for each void fraction. This maximum value is typically twice the extrapolated value at zero velocity.

A simple quadratic interpolation of the data emphasizes similarities of the behaviour for all ranges of void fraction. In comparison, Fig. 8 shows the behaviour of fluid damping in water flow (Axisa et al., 1989). Clearly, a similarity exists with the two-phase data of Fig. 7, the damping increasing then decreasing with increasing reduced velocity.

Let us now analyse the fluidelastic effects alone by removing the damping in quiescent fluid as defined in Section 3, Fig. 5, from the total fluid damping, thus defining a fluidelastic damping coefficient

$$\xi^{fe} = \xi^f - \xi_o^f. \tag{20}$$

For the sake of clarity, only data points from Axisa et al. (1989) in Figs. 7 and 8 are now plotted in terms of this fluidelastic damping ξ^{fe} versus reduced velocity V_R, Fig. 9.

The self-similarity of these behaviours, as noted above, leads us to propose new scaling factors on both axes, using physical arguments. Fluidelastic effects are by nature caused by the reaction of the flowing fluid to the tube motion. In the particular case of air–water mixtures, because of the mass ratio between the phases, most of the momentum originates in the motion of the liquid. We therefore propose to use for the scaling of fluidelastic effects new quantities referring to the liquid phase only. The proposed scaling velocity is the superficial liquid velocity J_l as in Nakamura et al. (1995) or Inada et al. (1996), $J_l = (1-\alpha)V$, so that the reduced superficial liquid velocity is

$$(J_l)_R = (1-\alpha)\frac{V}{fD}. \tag{21}$$

Similarly, the fluidelastic part of damping is now referred to the liquid density ρ_l instead of ρ and reads

$$\xi_n^{fe} = \frac{\xi^{fe}}{(\rho_l D^2/m)(D_{\mathrm{ref}}/D)\delta}. \tag{22}$$

Using these variables, data points from all tests of the database are plotted in Fig. 10. A first result is that all two-phase flow data follow the same behaviour, regardless of void fraction. This shows that the scaling factors defined by Eqs. (21) and (22) have some relevance. A second result is that this common behaviour is qualitatively similar to that of the single-phase flow test, suggesting a similar fluidelastic mechanism.

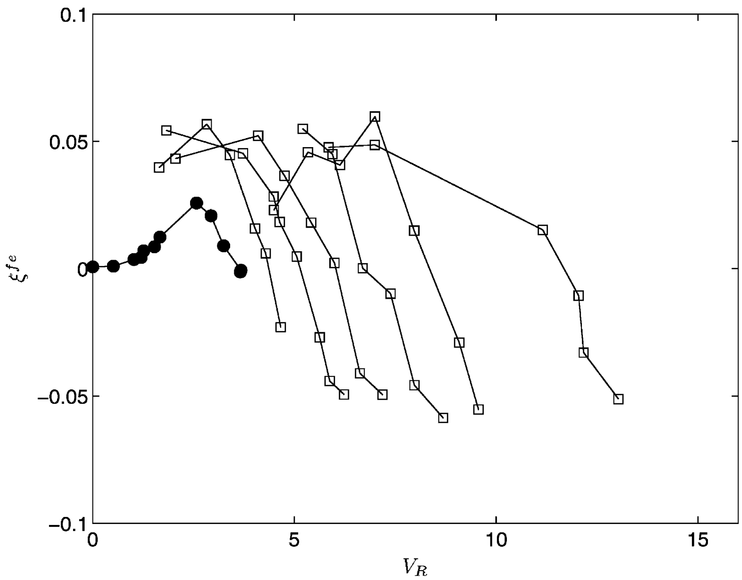


Fig. 9. Fluidelastic effects on damping: •, water (Axisa et al., 1989); □, air–water (Axisa et al., 1989).

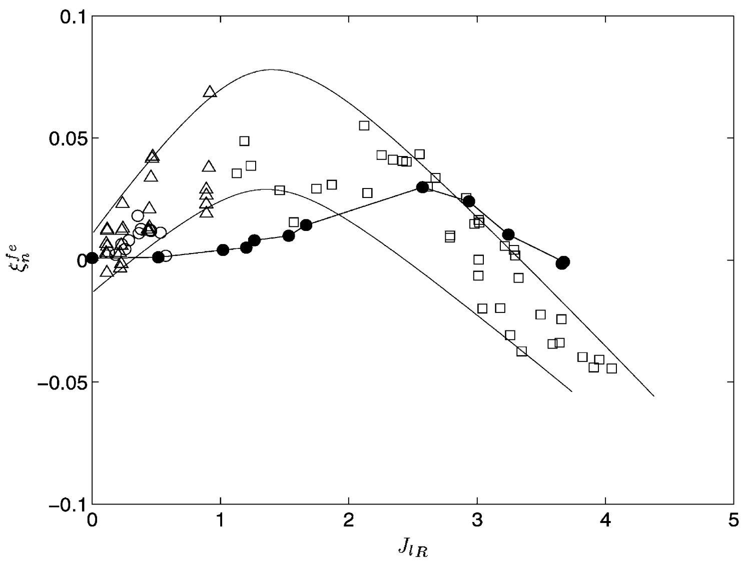


Fig. 10. Scaling of fluidelastic effects in two-phase flow: •, water (Axisa et al., 1989); ○, air–water (Baj and de Langre, 1999); △, air–water (Taylor et al., 1988); □, air–water (Axisa et al., 1989).

5. Discussion

In the preceding sections, we have defined a dimensionless damping in quiescent fluid, Eq. (10), and a dimensionless fluidelastic damping dependent on the reduced superficial liquid velocity, Eqs. (21) and (22). The mixture characteristics (ρ) have been used in the first scaling while the liquid characteristics (ρ_l, J_l) have been used in the second. These scalings differ from those of other authors (Pettigrew and Taylor, 1994; Inada et al., 1996). The first one allows a dependence of damping with the actual average mass of the medium surrounding the tube and, in general, is thought to be more adequate to the comparison of mixtures such as air–water and steam–water. The second one is more related to the dynamics of the moving medium, where momentum is dominated by the effect of the liquid. More precisely, Fig. 10

suggests that the superficial liquid velocity would be the relevant quantity to define the convection delay that causes instability, according to several analytical models (Price, 1995).

More refined descriptions exist for two-phase mixtures, such as drift-flux models or slip models. These may be used to redefine void fractions and liquid or gas velocities as done by several authors (Inada et al., 1996; Feenstra et al., 1995). The use of a drift-flux model to reanalyse our data showed only a slight improvement of the collapse of data points.

It is quite natural to assume that the effects of two-phase mixtures on tubes are qualitatively different from those of single phase fluids. Indeed, for the random buffeting excitation exerted by such mixtures, many authors pointed out specific excitation mechanisms probably due to the variations in term of phase at a given location (Taylor et al., 1996; Nakamura et al., 1995; de Langre and Villard, 1998). Conversely, the results of the present paper seem to indicate that the mechanisms controlling damping effects in two-phase bubbly flows are similar to those known in single phase. Still fluid damping, when defined by an extrapolation at zero velocity, is found to be in striking continuity from water to air–water bubbly mixtures. Fluidelastic effects on damping, when referred to the superficial liquid velocity, may be closely compared with those known in single phase flow.

6. Conclusion

The knowledge of the behaviour of modal characteristics, and particularly damping, of a tube inserted in a bundle subjected to two-phase cross-flow is required for predictive calculation, especially for the prediction of the onset of fluidelastic instability. An analysis of damping in quiescent fluid and the fluidelastic behaviour of damping have been carried out using several series of water and bubbly air–water tests which cover a range of void fraction from 0% to 80%. The main conclusions are the following.

(a) As stagnant fluid does not exist with two-phase mixtures such as air–water, quiescent fluid damping needs to be defined by extrapolation towards zero of the damping measured at low reduced velocities.

(b) As damping increases with reduced velocity at low reduced velocity, the value of quiescent fluid damping thus obtained is generally significantly lower than that currently used considering values at half the critical velocity for instability.

(c) Assuming that the damping in quiescent fluid as defined here is due to some viscous mechanism, it is shown that the resulting apparent two-phase vibrational viscosity is far higher than that obtained from classical models. It may be approximated by adapting the model of Hara (1988).

(d) The study of the behaviour of normalized fluid damping shows that fluidelastic phenomena occur over the whole range of reduced velocity, from the lowest velocities up to fluidelastic instability. In fact, normalized fluid damping first increases with reduced velocity then decreases towards very low values which are characteristic of the fluidelastic instability phenomenon.

(e) Using adequate scaling factors related to the characteristics of the liquid phase only, it is observed that fluidelastic effects are qualitatively similar in water and in air–water bubbly flow.

This analysis shows that damping in bubbly mixtures follows the same laws as in single-phase flows, when considering scale factors derived from physical considerations. This suggests that, in contrast with random buffeting effects, there is no mechanism of damping unique to two-phase bubbly flow across tubes.

References

Axisa, F., Villard, B., Antunes, J., 1989. Fluidelastic instability of a flexible tube in a rigid normal square array subjected to uniform two-phase cross-flow. In: Proceedings International Conference on Engineering Hydro-Aeroelasticity, Prague, Czech Republic, pp. 118–123.

Baj, F., 1998. Amortissement et instabilité fluide-élastique d'un faisceau de tubes sous écoulement diphasique. Ph.D. Dissertation, Université Paris VI, Paris, France.

Baj, F., de Langre, E., 1999. An analysis of fluid damping in two-phase air–water cross-flow. In: M.J. Pettigrew et al. (Eds.), Flow-Induced Vibration, PVP-Vol. 389, Boston, MA, USA. ASME, New York, pp. 81–88.

Blevins, R.D., 1990. Flow-Induced Vibration, 2nd Edition. Van Nostrand Reinhold, New York.

Carlucci, L.N., 1980. Damping and hydrodynamic mass of a cylinder in simulated two-phase flow. Journal of Mechanical Design 102, 597–602.

Carlucci, L.N., Brown, J.D., 1983. Experimental studies of damping and hydrodynamic mass of a cylinder in confined two-phase flow. ASME Journal of Vibration, Acoustics, Stress, and Reliability in Design 105, 83–89.

Chen, S.S., 1991. Flow-induced vibrations in two-phase flow. Journal of Pressure Vessel Technology 113, 234–241.

Chen, S.S., 1987. A general theory for dynamic instability of tube arrays in cross-flow. Journal of Fluids and Structures 1, 35–53.

Chen, S.S., Wanbsganss, M.W., Jendrzejczyk, J.A., 1976. Added mass and damping of a vibration rod in confined viscous fluids. Journal of Applied Mechanics 43, 325–329.

Collier, J.G., 1981. Convective Boiling and Condensation, 2nd Edition. McGraw-Hill, New York.

Feenstra, P.A., Judd, R.L., Weaver, R.L., 1995. Fluidelastic instability in a tube array subjected to two-phase R-11 cross-flow. Journal of Fluids and Structures 9, 747–771.

Hadj-Sadok, C., de Langre, E., Granger, S., 1995. Inverse methods for the measurement of fluid-elastic forces in tube bundles. In: Bearman, P.W. (Ed.), Proceedings Sixth International Conference on Flow-Induced Vibrations. Rotterdam, Balkema, London, UK, pp. 363–371.

Hara, F., 1988. Two-phase fluid damping in a vibrating circular structure. In: Proceedings of the ASME PVP Conference, PVP-Vol. 133, Pittsburgh, PA, USA. ASME, New York, pp. 1–8.

Hara, F., 1993. A review of damping of two-phase flows. In: Proceedings of the ASME PVP Conference, Seismic Engineering, PVP-Vol. 256-2, Denver, CO, USA. ASME, New York, pp. 87–101.

Inada, F., Kawamura, K., Yasuo, A., 1996. Fluid-elastic force measurements acting on a tube bundle in two-phase cross-flow. In: Pettigrew, M.J. (Ed.), Proceedings of the ASME PVP Conference, Flow-Induced Vibration, Vol. PVP-Vol. 328, Montreal, Que., Canada. ASME, New York, pp. 81–87.

de Langre, E., Villard, B., 1998. An upper bound on buffeting forces caused by two-phase flows across tubes. Journal of Fluids and Structures 12, 1005–1023.

McAdams, W.H., Woods, W.K., Heroman, L.C., 1942. Vaporization inside horizontal tubes-II-benzene–oil mixtures. Transaction ASME 64, 193–200.

Nakamura, T., Fujita, K., Kawanishi, K., Yamagushi, N., Tsuge, A., 1995. Study on the vibrational characteristics of a tube array caused by two-phase flow. Part 1: random vibration. Journal of Fluids and Structures 9, 519–538.

Pettigrew, M.J., Rogers, R.J., Axisa, F., 1986. Damping of multispan heat exchanger tubes. Part 2 : in liquids. In: Chen, S.S. et al. (Ed.), Proceedings Pressure Vessel and Piping Conference. ASME, New York, Chicago, IL, USA, pp. 89–98.

Pettigrew, M.J., Taylor, C.E., Kim, B.S., 1989. Vibration of tube bundles in two-phase cross-flow. Part 1: hydrodynamic mass and damping. ASME Journal of Pressure Vessel Technology 111, 466–477.

Pettigrew, M.J., Knowles, G.D., 1997. Some aspects of heat-exchanger tube damping in two-phase mixtures. Journal of Fluids and Structures 11, 929–945.

Pettigrew, M.J., Taylor, C.E., 1994. Two-phase flow-induced vibration: an overview. ASME Journal of Pressure Vessel Technology 116, 233–253.

Pettigrew, M.J., Taylor, C.E., 1997. Damping of heat exchanger tubes in two-phase flow. In: Païdoussis, M.P. et al. (Eds.), Fluid-Structure Interactions, Aeroelasticity, Flow-Induced Vibration and Noise, AD-Vol. 53-2, Dallas, TX, USA. ASME, New York, pp. 407–418.

Price, S.J., 1995. A review of theoretical models for fluidelastic instability of cylinder arrays in cross-flow. Journal of Fluids and Structures 9, 463–518.

Rogers, R.J., Taylor, C.E., Pettigrew, M.J., 1984. Fluid effects on multi-span heat-exchanger tube vibration. In: Chung, H., Bernstein, M.D. (Eds.), Proceedings Pressure Vessel and Piping Conference. ASME, New York, San Antonio, TX, USA, pp. 17–26.

Taylor, C.E., Pettigrew, M.J., Axisa, F., Villard, B., 1988. Experimental determination of single and two-phase cross-flow induced forces on tube rows. ASME Journal of Pressure Vessel Technology 110, 22–28.

Taylor, C.E., 1994. Random excitation forces acting on tube arrays subjected to water and air–water cross-flow. Ph.D. Dissertation, University of Toronto, Toronto, Canada.

Taylor, C.E., Pettigrew, M.J., Currie, I.G., 1996. Random excitation forces on tube bundles subjected to two-phase cross-flow. ASME Journal of Pressure Vessel Technology 118, 265–277.

Ulbrich, R., Mewes, D., 1994. Vertical, upward gas–liquid two-phase flow across a tube bundle. International Journal of Multiphase Flow 20 (2), 249–272.