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The Measurement of Fluidelastic Effects at Low Reduced Velocities Using Piezoelectric Actuators

Fluidelastic effects which are responsible for fluidelastic instabilities may be indirectly measured through the analysis of the vibrating motion of a system under flow. In this paper, piezoelectric actuators are used to increase the vibratory level when buffeting forces which excite tube vibration are low, and to improve the measurement of fluidelastic forces. The proposed method based on an added excitation allows the study of the added mass and provides a better accuracy on the measurement of the vibrating characteristics and, thereby, of the fluidelastic forces. This added excitation method is compared with a standard indirect approach on a tube underwater cross-flow. The influence of the level of piezoelectric excitation forces is discussed, as well as the range of application of this technique.

1 Introduction

Tube bundles are used in heat exchangers and may vibrate as a consequence of the external cross-flow. The time-dependent fluid forces acting on the tubes are commonly divided into random buffeting forces and motion-dependent fluid forces (Chen, 1987; Pettigrew and Taylor, 1994). The latter, also called fluidelastic forces, depend on the tube displacement, velocity, and acceleration. Since theoretical and numerical predictions of these unsteady fluid dynamic forces remain difficult, experimental approaches have been developed.

Two approaches are currently used in order to measure fluidelastic forces in tube bundles and are referred to as direct and indirect methods. In the direct method (Tanaka and Takahara, 1981; Chen et al., 1996), the fluidelastic forces are directly measured by the use of force cells when the tube is artificially forced to vibrate. The main disadvantage of this approach is its poor signal-to-noise ratio when other unsteady fluid forces dominate (Hadj-Sadok et al., 1995). In the indirect method (Pettigrew et al., 1989; Granger et al., 1993), the flow-induced displacement of a flexible tube is measured and fluidelastic forces are indirectly derived from the evolutions of the modal frequency and damping as the flow velocity is varied. A most common technique is to use the vibrating response to the buffeting forces. Provided these are sufficiently broad-banded, accurate measurements of the modal parameters, and thereby of the fluidelastic forces, may be obtained (Pettigrew et al., 1989; Granger et al., 1993). Yet, the measurement of the fluidelastic forces becomes impossible when the structure reaches fluidelastic instability, thus giving an upper limit to the range of application of the indirect method. On the other hand, the signal-to-noise ratio is poor at low flow velocities where buffeting forces are not sufficient to make the tube vibrate significantly and to allow a proper modal identification. This lower limit of application is the concern of the present paper.

In the past ten years, piezoelectric actuators have been shown to be efficient for in-situ applications of controlled forces in vibrating systems (Baz and Ro, 1991; Kaneko and Hirota, 1992). We propose here to extend the indirect method to lower values of flow velocities through the use of additional excitation forces imposed by such piezoelectric actuators (Crawley and De Luis, 1987; Preumont, 1997).

The additional excitation method presented here (referred to as AEM) takes advantage of the use of additional forces generated by piezoelectric actuators in three ways:

(*a*) The vibratory level of the tube is artificially raised so that the signal-to-noise ratio becomes acceptable, thus compensating for the insufficient buffeting forces.

(*b*) As this added excitation is known, the modal parameters of the system may be estimated based on the transfer function between the vibratory response and this excitation. This is known to be more accurate than a modal identification on the sole PSD of the response as in the standard indirect method. As both unknown buffeting forces and known added forces simultaneously excite the tube, a specific procedure involving a coherence analysis is needed for the computation of the transfer function.

(*c*) The modal identification of this frequency response function provides three coefficients: the modal frequency, the reduced damping, and the modal participation coefficient. Thus, the fluidelastic coefficients corresponding to the added stiffness, the added damping, and the added mass may be estimated. This is not the case of the indirect method where only two fluidelastic coefficients may be derived.

To evidence the necessity of specific procedures, experimental tests are performed with a flexible tube equipped with piezoelectric actuators and inserted in a bundle under water cross-flow. At a given flow velocity, the level of the added excitation is varied to study its influence on the coherence functions. Then, at the same flow velocity, the results obtained with the additional excitation method are compared with those given by the standard indirect method. Finally, measurements of fluidelastic forces are investigated at low flow velocities using the additional excitation method. A comparison with the results obtained using the standard indirect method for higher flow velocities is made in order to display the complementarity of the two methods on a wide range of flow velocities. Comparison with experimental data of Granger et al. (1993) shows the usefulness of the method to derive accurate values of fluidelastic coefficients.

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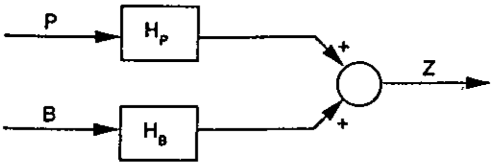

Fig. 1 Multi-excitation diagram

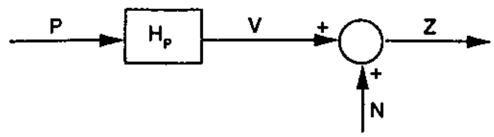

Fig. 2 Response with output noise diagram

2 Theoretical Background

In the indirect method, the characteristics of fluidelastic forces are indirectly derived from the response of the structure through a modal identification of the vibrating tube (Granger, 1990). Fluid forces acting on the structure are divided here into two groups (Chen, 1987), namely, buffeting forces (F_B) and fluidelastic forces (F_{FE}) which are motion-dependent. The dynamic equation of the vibrating system reads

$$M_s\ddot{q}(t) + C_s\dot{q}(t) + K_sq(t) = f_B(t) + f_{FE}(\ddot{q}(t), \dot{q}(t), q(t)) \quad (1)$$

where M_s, C_s, and K_s are, respectively, the modal mass, damping and stiffness matrices of the structure, q are modal coordinates, f_B and f_{FE} being the modal projections of F_B and F_{FE}. When the modal fluidelastic forces are linearized in terms of modal displacement, velocity, and acceleration, Eq. (1) becomes

$$(M_s + M_a)\ddot{q}(t) + (C_s + C_a)\dot{q}(t) + (K_s + K_a)q(t) = f_B(t) \quad (2)$$

where M_a, C_a, and K_a are the modal added mass, damping, and stiffness matrices. Buffeting forces are assumed to be broad-banded random excitations such as those resulting from turbulence (Axisa et al., 1990) or two-phase flow (Taylor et al., 1996; de Langre and Villard, 1998). The modal frequency and damping may then be derived from the tube response with a PSD-fitting or other techniques (Granger, 1990). Thus, the standard indirect method can provide only two fluidelastic coefficients. Usually, the added mass is supposed to be constant when flow velocity is varied and the measured frequency and damping give the fluidelastic stiffness and damping coefficients of Eq. (2) (Granger et al., 1993).

In the additional excitation method (AEM) presented here, piezoelectric forces are added and Eq. (2) becomes

$$(M_s + M_a)\ddot{q}(t) + (C_s + C_a)\dot{q}(t) + (K_s + K_a)q(t) = f_B(t) + f_P(t) \quad (3)$$

where f_P are the modal piezoelectric forces. The piezoelectric forces are added to raise the level of vibration in order to obtain a good signal-to-noise ratio when the buffeting forces are low. Finally, the system of Eq. (3) which is multi-excited can be represented in a bloc diagram as in Fig. 1, where the signals B and P are, respectively, the buffeting and piezoelectric excitation forces, Z the experimental displacement of the tube, and H_B, H_P the corresponding transfer functions.

The piezoelectric excitation forces are not directly measured. Only the voltage applied across the piezoelectric actuators may be easily measured. Therefore, if the voltage-expansion relationship of the actuators is supposed to be linear (Preumont, 1997), the actuators input signal may be calibrated in terms of forces acting on the structure. Then, as these piezoelectric excitation forces may be measured, the modal estimation may be done on the transfer function H_P between the response and the piezoelectric excitation. This approach is more accurate than a fitting on the PSD of the response when only buffeting forces are used, because the random error of the estimates of frequency response functions is smaller than the random error of the estimates of auto-spectra (Bendat and Piersol, 1993).

The actual accuracy of the transfer function H_P depends strongly on the treatment of the part of the response due to the buffeting forces which remain unknown. If buffeting forces and the piezoelectric excitation are uncorrelated, this part of the response may be considered as an external noise N as pictured in Fig. 2, and thus be eliminated by the use of adequate filtering. A most commonly used filtering technique is that of the H1-estimate of the frequency response function (Bendat and Piersol, 1993), which allows computing the transfer function as

$$H_P(\omega) = \frac{\phi_{ZP}(\omega)}{\phi_{PP}(\omega)} \quad (4)$$

where ϕ_{ZP} and ϕ_{PP} are, respectively, the cross-spectrum between the signals Z and P, and the auto-spectrum of the signal P. The H1-estimate of the transfer function is commonly used in order to eliminate noise of low level on the output signal. So, in order to improve the measurements with the AEM, the part of the response due to the additional forces needs to be higher in comparison with the part of the response due to the buffeting forces. As the causality between the excitation and the response is measured by the coherence function, the foregoing hypothesis may be checked in practice through a coherence analysis. The coherence function between two signals P and Z is written as (Bendat and Piersol, 1993)

$$\gamma^2_{ZP}(\omega) = \frac{|\phi_{ZP}(\omega)|^2}{\phi_{ZZ}(\omega)\phi_{PP}(\omega)} \quad \text{with} \quad 0 \leq \gamma_{ZP} \leq 1 \quad (5)$$

When the measured coherence function γ_{ZP} is expressed as a function of the auto-spectrum ϕ_{NN}, the influence of the external response N on the output signal Z may be estimated. Bendat

Nomenclature

C = damping matrix
D = diameter of tube
F = force
H = transfer function
K = stiffness matrix
M = mass matrix
U = pitch flow velocity
U_r = reduced velocity
f = frequency
q = modal coordinates
s = Laplace's variable
α = modal participation coefficient of H
ϵ = strain gage deformation
ϕ = auto/cross-spectrum
γ = ordinary coherence
ψ = modal projection coefficient
ω = circular frequency
σ = standard deviation
ξ = reduced damping

Subscripts

B = buffeting forces
FE = fluidelastic forces
N = noise
P = piezoelectric
V = output signal without noise
Z = response of strain gage
a = fluidelastic coefficient
o = reference parameter
s = structure

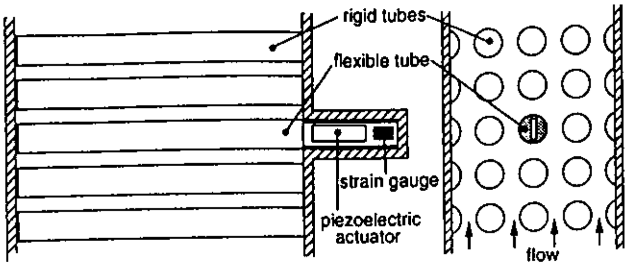

Fig. 3 Experimental square array of tubes

and Piersol (1993) give the ordinary coherence γ_{ZP} as a function of the ordinary coherence γ_{VP}, which is closed to one, between the response without noise (noted V in Fig. 2) and the piezoelectric excitation

$$\gamma_{ZP}^2(\omega) = \frac{\gamma_{VP}^2(\omega)}{1 + \dfrac{\phi_{NN}(\omega)}{\phi_{VV}(\omega)}} \quad (6)$$

Equation (6) shows that the quality of the measured coherence function γ_{ZP} is decreased by the external response N. Thus, the coherence function γ_{ZP} may be an indirect indicator in order to evaluate the accuracy of the measurements using the AEM: when γ_{ZP} is poor, the signal V, which is the part of the response Z due to the additional excitation forces is small in comparison with the signal N, which is the part of the response Z due to the buffeting excitation forces.

The modal estimation of H_P gives three modal coefficients per mode which are the modal frequency f, the reduced damping ξ, and a new result, the modal participation coefficient α_P (Eq. (14), Appendix). These coefficients may be associated with the mass, damping, and stiffness of a studied mode of the fluid-structure system by using

$$m_s + m_a = \frac{\psi_P \psi_Z}{\alpha_P} \quad (7)$$

$$c_s + c_a = 2\frac{\psi_P \psi_Z}{\alpha_P}(2\pi f)\xi \quad (8)$$

$$k_s + k_a = \frac{\psi_P \psi_Z}{\alpha_P}(2\pi f)^2 \quad (9)$$

where ψ_P and ψ_Z, which may be derived from the modal base, are, respectively, the modal projection coefficients of the added excitation forces and of the response. Finally, the AEM provides more information on the fluidelastic forces than the indirect method. The AEM allows to check the hypothesis of the standard indirect method which assumes that the added mass is constant.

3 Experimental Setup

We consider here a square bundle of tubes (Fig. 3), the central tube of which is flexible. The array is similar to that reported in Hadj-Sadok et al. (1995) and includes 15 stainless steel cylinders (3 columns and 5 rows) and 10 half-cylinders of diameter 30 mm with pitch-to-diameter ratio of 1.5. It is confined in a 180 × 300 mm^2 vertical channel. The pitch flow velocity is varied from 0 up to 1.53 m/s.

The flexible system (Fig. 4) is made of the tube under flow attached to a flexible plate which allows vibration in the lift direction only. Thus, the first bending mode in still water is about 18.7 Hz. The displacement of the vibrating tube is derived from a strain gage bonded at the base of the flexible plate.

Two PZT actuators (Physik Instrumente, PIC151, 0.25 × 22 × 50 mm^3) are symmetrically bonded on the plate (Figs. 4 and 5). The bond (Epoxy Technology, Epo-Tek417 Silver epoxy) is electrically conductive and watertightness is realized with a joint (Le Joint Français, PR395P) around the flexible beam. The tube is excited in open-loop by the two piezoelectric actuators, which are supplied with power by two amplifiers (Physik Instrumente, E507.00). The excitation signal is a gaussian white noise delivered by a frequency analyzer HP3582 and the electric field is applied across one piezoelectric actuator in one direction and reversed on the second. If the actuators are perfectly bonded on the beam, Crawley and De Luis (1987) show that the two piezoelectric actuators deform the structure in pure bending, and this bending is caused by two moments acting at the ends of the actuators. The projection of these moments on the modal base allows to calibrate the excitation signal across the actuators in terms of moments acting at the ends of the actuators. A finite element analysis provides here the modal base of the tube. Then, the calibration coefficient of the actuators is 1.03 × 10^{-3} Nm/V.

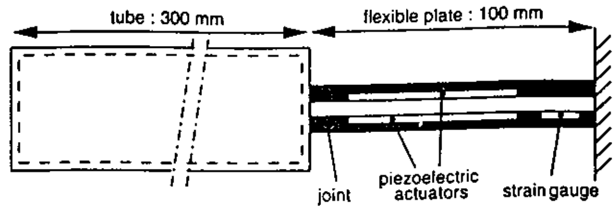

Fig. 4 Flexible tube with bonded piezoelectric actuators

4 Measurements at a Given Flow Velocity

4.1 Influence of the Piezoelectric Excitation Level. At a given flow velocity (U = 0.35 m/s), the piezoelectric excitation level is now varied to evidence the necessity of a coherence analysis. Two excitation levels are considered, where the standard deviations of the moments acting on the beam are, respectively, $\sigma_p = 4 \times 10^{-3}$ Nm and $\sigma_p = 20 \times 10^{-3}$ Nm. The corresponding rms displacements at the end of the tube are σ_s = 6.9 μm and σ_s = 35.9 μm. The associated coherence functions and frequency response functions shown in Figs. 6 and 7 are calculated on 80 averages with a frequency resolution of 0.0488 Hz (as well as the following auto-spectra).

The coherence functions of both excitation levels shown in Fig. 6 illustrate the importance of a coherence analysis as described in Eq. (6). The coherence function for the second excitation level (Fig. 6 (b)) is higher than the coherence function for the first excitation level (Fig. 6(a)). Thus, when the excitation level is increased, the ordinary coherence around the first bending mode at 18.7 Hz is improved. Both coherence functions

Fig. 5 Flexible plate with bonded piezoelectric actuators

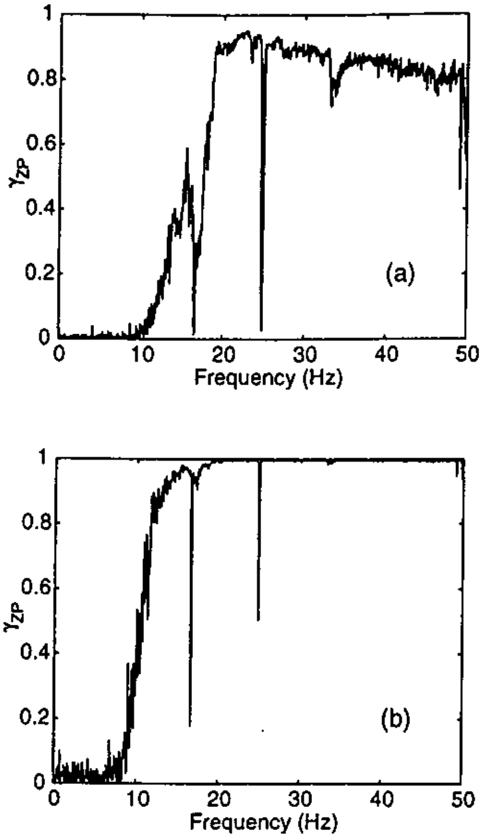

Fig. 6 Coherence functions – U = 0.35 m/s; (a) $\sigma_p = 4 \times 10^{-3}$ Nm, (b) $\sigma_p = 20 \times 10^{-3}$ Nm

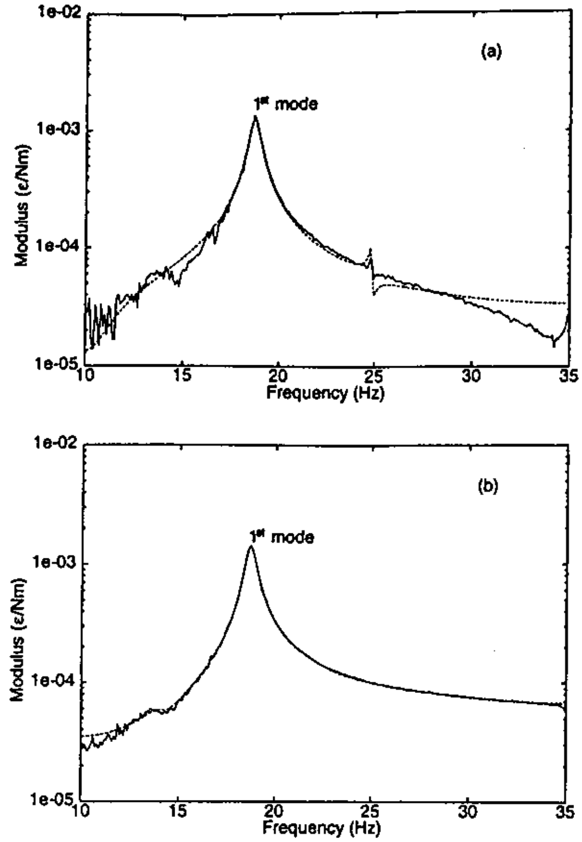

Fig. 7 Transfer functions – U = 0.35 m/s; (a) $\sigma_p = 4 \times 10^{-3}$ Nm, (b) $\sigma_p = 20 \times 10^{-3}$ Nm; ——: experiment, - - -: Prony fitting

are closed to zero on the frequency range 0–10 Hz (this point is analyzed in the Appendix). As this limit is always less than the frequency of the first mode, we can always choose a valid frequency range for the identification of the modal characteristics. Finally, the coherence analysis has two objectives: first, the coherence function must be high enough around the frequency of the studied mode in order to validate the piezoelectric excitation level chosen; and second, the coherence analysis allows to choose the frequency range for the modal identification. In practice, the level of piezoelectric excitation is chosen so that the coherence function is closed to one in the range of frequencies under consideration. Moreover, a high level of coherence is a proof that the piezoelectric actuators behave as linear systems (Bendat and Piersol, 1993).

The modal parameters such as frequency and damping are derived from a time-domain Prony fitting (Granger, 1990) on the frequency range 10–35 Hz. In the case of a single flexible tube inserted in a rigid bundle, this was shown to lead to results identical to that of a PSD-fitting (Hadj-Sadok et al., 1995). The comparison of the results given by the Prony fitting on both frequency response functions (Fig. 7) shows a better fit for a higher excitation level. The Prony fittings confirm the conclusions of the coherence analysis on the added excitation level.

4.2 Comparison With the Indirect Method. With the same tube, the standard indirect method is now used at the previous pitch flow velocity (0.35 m/s) in order to evaluate the improvements due to the additional excitation method. The piezoelectric excitation level is set to zero ($\sigma_p = 0$), and the structure is only excited by the buffeting forces. The standard deviation of the response at the end of the tube is $\sigma_s = 2.7$ μm. Therefore, the frequency response function H_P, Eq. (4), may not be calculated and only the auto-spectrum of the response (Fig. 8) is available. The comparison with the auto-spectrum of the response, when $\sigma_p = 20 \times 10^{-3}$ Nm (Fig. 9), shows the increase of the vibratory level due to the added excitation.

The signal-to-noise ratio with the standard indirect method is poor, as shown in Fig. 8, due to the low level of buffeting forces. Moreover, two peaks appear at 17 and 25 Hz, which may be identified as pump harmonic excitations. The peak corresponding to the first bending mode is affected by the noise that distorts the estimation of its modal characteristics. By comparing Figs. 7(b) and 9, it is clearly shown that the use of the transfer functions with the H1-estimate of Eq. (4) eliminates the external pump harmonic excitations. The differences obtained in the estimations of the frequency and damping by the standard indirect method and the AEM are shown in Table 1.

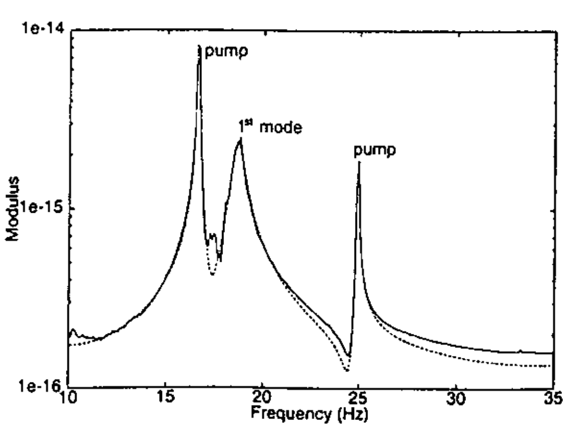

Fig. 8 Indirect method auto-spectrum – U = 0.35 m/s – σ_p = 0; ——: experiment, - - -: Prony fitting

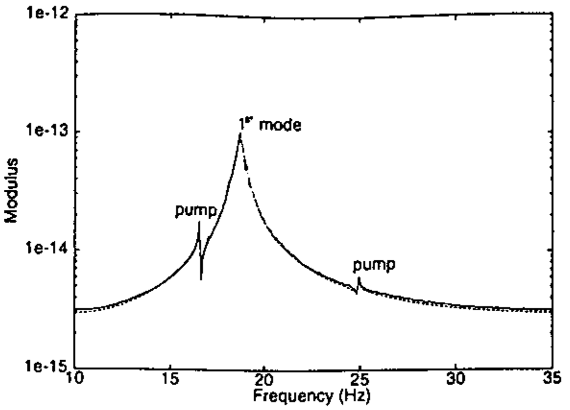

Fig. 9 AEM auto-spectrum - U = 0.35 m/s - $\sigma_p = 20 \times 10^{-3}$ Nm; ——: experiment, - - - : Prony fitting

5 Application of the AEM for Low Flow Velocities

A measurement in still fluid ($U = 0$) is done using the piezoelectric actuators to excite the tube (Table 2). When the pitch flow velocity is varied from 0.08 up to 0.66 m/s, measurements of the modal characteristics (frequency, damping, and modal participation coefficient) are done using the additional excitation method (Table 2). For higher pitch flow velocities, from 0.69 up to 1.53 m/s, buffeting excitation forces are significant and the classical indirect method may be used (Table 3) with the same procedure as in Hadj-Sadok et al. (1995). In that case, piezoelectric actuators are not supplied with power ($\sigma_p = 0$). In Tables 2 and 3, the reduced velocity U_r is given by $U_r = U/(f.D)$. The results given in Table 2 when U = 0.36 m/s are slightly different from those given in Table 1 by the AEM when U = 0.35 m/s as the tube was dismounted between the two series of tests. Figure 10 shows the complementarity of the two methods to derive the modal frequency and damping when pitch flow velocity is varied from 0 up to 1.53 m/s.

6 Discussion

With the same flexible tube equipped with piezoelectric actuators, fluidelastic stiffness and damping forces may be measured using the additional excitation method and the standard indirect method on a reduced velocity range that includes still fluid.

Fluidelastic effects on damping may be represented by a dimensionless coefficient (Chen et al., 1996; Granger et al., 1993) defined as

$$C_d = \frac{8\pi m_e}{\rho D^2 U_r}\left(\xi - \xi_o \frac{f_o}{f}\right) \tag{10}$$

where m_e is the equivalent mass per unit length, and f_o, ξ_o, the reference frequency and damping in still water. In Fig. 11, we compare the dimensionless fluidelastic damping coefficients $Cd.U_r$ obtained in the present experiments with those of Granger et al. (1993) on a similar tube bundle. A good similarity is found between the two sets of experimental results, except for a constant shift in value. This may be attributed, in the data of Granger et al. (1993), to the use of two different methods to derive modal parameters, namely, a free decay method for U_r = 0, an auto-spectrum fitting for $U_r > 2$. Our data show a continuous evolution due to the use of the added excitation method. Figure 10 illustrates that there is no discontinuity of

Table 1 Comparison between the indirect method and the AEM at U = 0.35 m/s

Method	f (Hz)	ξ (%)
Indirect Method	18.64	1.57
AEM	18.70	1.45

Table 2 Results with the AEM

U (m/s)	U_r	f (Hz)	ξ (%)	α_P ($N^{-1}.m^{-1}.s^{-2}$)
0	0	18.75	1.44	0.583
0.08	0.14	18.76	1.44	0.590
0.12	0.21	18.75	1.44	0.585
0.19	0.34	18.75	1.45	0.583
0.26	0.46	18.76	1.45	0.583
0.36	0.64	18.76	1.46	0.583
0.46	0.82	18.75	1.44	0.583
0.54	0.96	18.73	1.49	0.587
0.66	1.17	18.74	1.56	0.587

Table 3 Results with the standard indirect method

U (m/s)	U_r	f (Hz)	ξ (%)
0.69	1.23	18.69	1.67
0.90	1.62	18.49	2.18
0.97	1.76	18.35	2.03
1.05	1.91	18.32	1.16
1.25	2.27	18.38	1.75
1.38	2.52	18.28	2.48
1.53	2.81	18.11	3.04

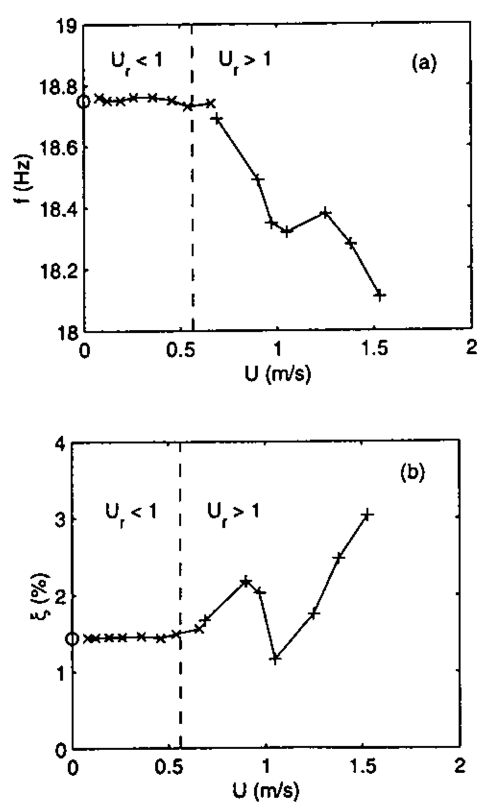

Fig. 10 Evolution of modal parameters with flow velocity; (a) frequency, (b) damping; o: AEM (U = 0), x——x: AEM (U > 0), +——+: indirect method

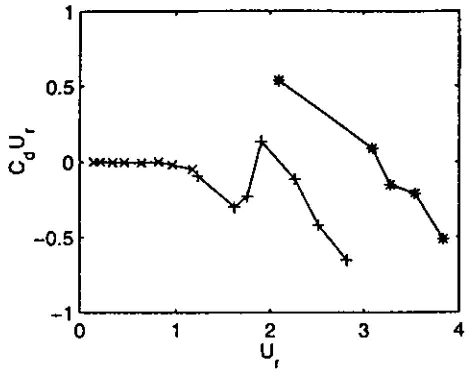

Fig. 11 Dimensionless damping coefficient; x——x: AEM, +——+: indirect method, *——*: Granger et al. (1993)

the frequency and the reduced damping between $U = 0$ and $U > 0$ when the same experimental method is used. Discontinuities between still fluid and low flow velocities of the damping may only be a consequence of using two different experimental procedures.

As the modal participation coefficient α_P is constant (Table 2), the AEM proves that for low flow velocities, the fluidelastic phenomena has no effect on the added mass. The use of both AEM and standard indirect method allows to build the evolution of fluidelastic effects with a good level of accuracy on a large range of reduced velocities. It clearly appears on Fig. 10 that the fluidelastic forces have some influence only for a reduced velocity higher than one. This is fully consistent with previously published models (Lever and Weaver, 1982; Price and Païdoussis, 1984) where the fluidelastic mechanism is thought to originate in a phase lag between the tube motion and the reacting force by the fluid. This phase lag was related to the ratio between the time of tube oscillation and the time of fluid convection. This ratio is identical to the reduced velocity $(T/(D.U) = U/(f.D) = U_r)$. Therefore, $U_r = 1$ is an order of magnitude of the onset of fluidelastic effects.

7 Conclusions

In this paper, we have shown that piezoelectric actuators may be used to significantly improve the measurement of fluidelastic effects in the case of a flexible tube under water cross-flow. These actuators provide an additional excitation which allows to extend the standard indirect method to lower value of flow velocity, where buffeting forces are not sufficient to make the tube vibrate. Moreover, as the content of additional excitation is known, a specific procedure may be used to derive the transfer function and use it for the modal identification, which is more accurate than a PSD-fitting. This procedure allows to calculate the effect of fluidelastic forces on the added mass which is not the case of the standard indirect method. It should be noted that the proposed method is limited to a single degree of freedom system and assumes that nonlinear effects are negligible in the tube behavior.

When this additional excitation method is used on a single tube mock-up, it is found that no fluidelastic effect exists at the transition between still fluid and very low flow velocities. The fluidelastic effects remain negligible up to a reduced velocity of about one. The proposed methodology is not restricted to reduced velocities of less than one. For instance, in the case of two-phase flow, it is known that buffeting forces are significantly lower than in single phase, though fluidelastic forces exist (Pettigrew and Taylor, 1994; de Langre and Villard, 1998). Added excitation might be of a great use to improve the accuracy on fluidelastic coefficients even at higher flow velocities.

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References

Axisa, F., Antunes, J., and Villard, B., 1990, "Random Excitation of Heat Exchanger Tubes in Cross-Flow," *Journal of Fluids and Structures*, Vol. 4, pp. 321–341.

Baz, A., and Ro, J., 1991, "Active Control of Flow-Induced Vibrations of a Flexible Cylinder Using Direct Velocity Feedback," *Journal of Sound and Vibration*, Vol. 146, No. 1, pp. 33–45.

Bendat, J.S., and Piersol, A.G., 1993, *Engineering Applications of Correlation and Spectral Analysis*, Wiley-Interscience, 2nd Ed., New York, NY.

Chen, S.S., 1987, "A General Theory for Dynamic Instability of Tube Arrays in Cross-Flow," *Journal of Fluids and Structures*, Vol. 1, pp. 35–53.

Chen, S.S., Cai, Y., and Zhu, S., 1996, "Flow-Induced Vibration of Tubes in Cross-Flow," ASME *Journal of Offshore Mechanics and Arctic Engineering*, Vol. 118, pp. 253–258.

Crawley, E.F., and De Luis, J., 1987, "Use of Piezoelectric Actuators as Elements of Intelligent Structures," *American Institute of Aeronautics and Astronautics Journal*, Vol. 25, No. 10, pp. 1373–1385.

de Langre, E., and Villard, B., 1998, "An Upper Bound on Random Buffeting Forces Caused by Two-Phase Flows Across Tubes," *Journal of Fluids and Structures*, Vol. 12, pp. 1005–1023.

Granger, S., 1990, "A New Signal Processing Method for Investigating Fluidelastic Phenomena," *Journal of Fluids and Structures*, Vol. 4, pp. 73–97.

Granger, S., Campistron, R., and Lebret, J., 1993, "Motion-Dependent Excitation Mechanisms in a Square In-Line Tube Bundle Subject to Water Cross-Flow: An Experimental Modal Analysis," *Journal of Fluids and Structures*, Vol. 7, pp. 521–550.

Hadj-Sadok, C., de Langre, E., and Granger, S., 1995, "Inverse Methods for the Measurement of Fluid-Elastic Forces in Tube Bundles," *Flow-Induced Vibration*, Proceedings, 6th International Conference, London, England, Bearman Ed., Rotterdam, pp. 363–371.

Kaneko, S., and Hirota, K., 1992, "A Study on Active Control of Leakage Flow Induced Vibrations," Proceedings, *Control of Flow-Induced Vibration*, Vol. III, pp. 63–69.

Lever, J.H., and Weaver, D.S., 1982, "A Theoretical Model for Fluidelastic Instability in Heat Exchanger in Tube Bundles," ASME JOURNAL OF PRESSURE VESSEL TECHNOLOGY, Vol. 14, pp. 147–158.

Pettigrew, M.J., Taylor, C.E., and Kim, B.S., 1989, "Vibration of Tubes Bundles in Two-Phase Cross-Flow—Part 1: Hydrodynamic Mass and Damping," ASME JOURNAL OF PRESSURE VESSEL TECHNOLOGY, Vol. 111, pp. 466–477.

Pettigrew, M.J., and Taylor, C.E., 1994, "Two-Phase Flow-Induced Vibration: An Overview," ASME JOURNAL OF PRESSURE VESSEL TECHNOLOGY, Vol. 116, pp. 233–253.

Preumont, A., 1997, *Vibration Control of Active Structures: An Introduction*, Presses Polytechniques et Universitaires Romandes, Lausanne Switzerland.

Price, S.J., and Païdoussis, M.P., 1984, "An Improved Mathematical Model for the Stability of Cylinder Rows Subject to Cross-Flow," *Journal of Sound and Vibration*, Vol. 97, pp. 615–640.

Tanaka, H., and Takahara, S., 1981, "Fluid Elastic Vibration of Tube Array in Cross-Flow," *Journal of Sound and Vibration*, Vol. 77, No. 1, pp. 19–37.

Taylor, C.E., Pettigrew, M.J., and Currie, I.G., 1996, "Random Excitation Forces in Tube Bundles Subjected to Two-Phase Cross-Flow," ASME JOURNAL OF PRESSURE VESSEL TECHNOLOGY, Vol. 118, pp. 265–277.

APPENDIX

On the frequency range 0–10 Hz, the ordinary coherence functions of both excitation levels (Fig. 6) are closed to zero. Measurement instruments are known to affect coherence in low frequency range. The present observation may also be explained with an analysis on transfer functions forms. For the present mock-up, the second-order form of the transfer functions H_B and H_P can be expressed in the Laplace domain. The classical form of the transfer function H_B is

$$H_B(s) = \sum_{i=1}^{\infty} \frac{\alpha_B^i}{(s^2 + 2\xi_i\omega_i s + \omega_i^2)}. \qquad (11)$$

The frequency response function H_P may be identified in three steps. First, the transfer function H_P considering an infinity of modes has the same form as Eq. (11). Second, as the bending of the flexible tube is caused by two moments acting at the ends of the actuators (Crawley and De Luis, 1987), only the part

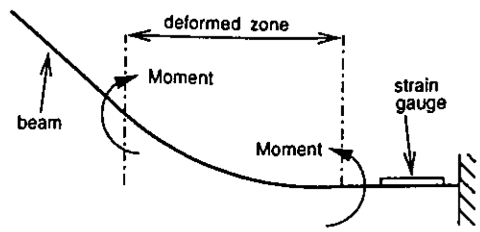

Fig. 12 Two piezoelectric moments acting in bending on a beam

of the tube under the actuators is deformed (Fig. 12). The corresponding static deformation is not measured by the strain gage. Thus, the transfer function H_P is equal to zero at $s = 0$

$$H_P(0) = \sum_{i=1}^{\infty} \frac{\alpha_P^i}{\omega_i^2} = 0 \qquad (12)$$

Finally, the transfer function H_P may be written as

$$H_P(s) = \sum_{i=1}^{\infty} \frac{\alpha_P^i}{(s^2 + 2\xi_i\omega_i s + \omega_i^2)} - \sum_{i=1}^{\infty} \frac{\alpha_P^i}{\omega_i^2} \qquad (13)$$

which reads

$$H_P(s) = -\sum_{i=1}^{\infty} \frac{\alpha_P^i(s^2 + 2\xi_i\omega_i s)}{\omega_i^2(s^2 + 2\xi_i\omega_i s + \omega_i^2)}. \qquad (14)$$

Equation (14) shows that the frequency response function H_P has the same form as a high-pass filter; thus, low frequencies are filtered by H_P. Furthermore, the transfer function H_B given in Eq. (11) may be identified as a low-pass filter; therefore, H_B is constant for low frequencies. Finally, for low frequencies, the response of the tube is mainly due to the buffeting excitation. Thus, considering frequencies under 10 Hz, the ordinary coherence γ_{ZP} is closed to zero as shown in Fig. 6 and the piezoelectric excitation cannot improve the quality of the coherence function on this frequency range. One solution to avoid this problem at low frequencies may be the use of a piezoelectric film sensor, such as PVDF polymer, collocated with the actuators (Preumont, 1997).