Measurement of Critical Velocities for Fluidelastic Instability Using Vibration Control

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Fluidelastic effects may be responsible for instabilities of heat exchanger tubes when the fluid flow reaches the critical velocity. The fluidelastic phenomenon is usually studied on experimental mock-ups, which may display only one critical velocity. In this paper, a method based on active vibration control is proposed in order to derive several critical velocities for fluidelastic instability corresponding to several different values of damping, which is artificially varied on the same mock-up. Experimental tests are performed on a flexible tube equipped with piezoelectric actuators in a rigid array under air-water cross-flow. It is shown that the reduced critical velocities thus obtained fit well in a classical stability map. [S0739-3717(00)01603-2]

1 Introduction

Heat exchanger tubes may exhibit, under particular conditions, large amplitude vibrations, which may lead to failures. One of the origins of these vibrations has been identified as fluidelastic dynamic instability, a subject of intensive research over the past thirty years [1–3]. At a critical value of flow velocity, a sudden increase of vibration amplitude that is caused by a fall toward zero of the damping of the fluid-structure system [4] may be displayed. This instability is observed in wholly flexible arrays but also at a different threshold when all tubes are fixed except one and allowed to move in the lift direction [5,6]. In that case, the instability has been attributed to a velocity dependent force rather than a displacement dependent force. In this paper, we shall only consider one flexible tube. Since theoretical and numerical predictions of the fluidelastic instability remain difficult [6–8], experimental approaches have been developed. Thus, studies have been carried out in order to display critical velocities when the array characteristics and the fluid properties are varied. It has been observed that tube damping, which depends on void fraction and interactions with the supports, had a crucial effect on the instability threshold. An approach to this problem has been to map the reduced critical velocity as a function of a nondimensional parameter such as the mass-damping parameter [1]. To build up such a map, each point requires the use of a given mechanical system, with its frequency and its damping, which is tested up to fluidelastic instability.

On the other hand, vibration control has been shown to be efficient for in situ applications of controlled forces in vibrating system under fluid flow [9,10]. We propose here a method to derive several values of the critical velocity corresponding to several values of damping, which is varied by the mean of active vibration control [11]. This allows to build the instability chart in a systematic way. Meskell and Fitzpatrick [12] also varied the damping of a single flexible tube in an array under air cross-flow using an electromagnetic shaker connected to an electrical resistance. The proposed method was based on passive control [11] which only dissipates mechanical energy. This passive control mechanism can only add damping to the structure under control. This is not the case of active control where damping may be added or subtracted.

The actual damping brought by the active control may be measured, but may also be estimated from the characteristics of the active control-loop [11]. The latter approach is in fact necessary in the case of two-phase flow where the modal characteristics of the fluid-structure system in a still two-phase mixture may not be measured. A method is described here to derive the damping brought by the controller as a function of the characteristics of the control-loop in the case of a two-phase flow. Experimental tests are performed with a flexible tube equipped with piezoelectric actuators and inserted in a tube bundle. Several critical velocities for fluidelastic instability are measured using vibration control on four homogeneous void fractions. The results are compared with a design guide-line using the mass-damping parameter and the reduced velocity [1].

2 Active Control Measurement Method

2.1 Calibration Necessity. Active vibration control allows the modal damping of a flexible structure to vary. Damping may be artificially added or subtracted to the initial damping. The main objective of the present active control method is to set the modal damping of the fluid-structure system to a required value for the measurement of fluidelastic effects. Active vibration control is physically based on a control-loop, which may be represented in a bloc diagram as in Fig. 1. The mechanical system, here the fluid-structure system, and the controller are respectively represented by the transfer functions $H(s)$ and $G(s)$ where $G(s)=gD(s)$ and g is the scalar gain. When g is varied, the efficiency of the controller is varied. Thus, a given added damping, which may be positive or negative, corresponds to a given scalar gain g. No damping is added to the structure when this last gain is set to zero.

2.2 Experimental Calibration. To calibrate the control-loop, the added damping for a given scalar gain may be derived from two tests. In a first test, the modal damping of the system with no control may be measured through a modal analysis. A second test with a given scalar gain provides the effective damping of the system under control. The added damping corresponding to the given scalar gain is thus obtained from the difference between the two measured damping values. To calibrate the control loop, the second test needs to be repeated for each scalar gain

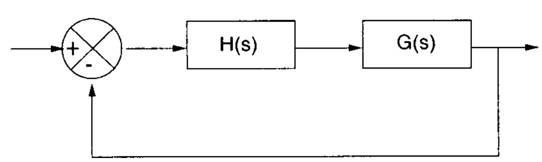

Fig. 1 Block diagram of the closed-loop

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because the correspondence between the scalar gain and the added damping is generally not linear on a large scalar gain range [11].

2.3 Numerical Calibration. A SISO control-loop (Single Input, Single Output) consists of a sensor which provides the input information on the movement of the structure, and an actuator which exerts the output effort of control on the structure. The frequency response function between the sensor and the actuator, called frequency response function in open-loop H_o, may be measured. The structure is then directly excited by the actuator of the control-loop. H_o may be defined as

$$H_o = H(s)G(s) \tag{1}$$

The modal characteristics of the structure with no control, which are the poles of $H(s)$, may be derived from the frequency response function in open-loop $H_o(s)$ as shown in Eq. (1).

On the other hand, when the structure is under control, the transfer function in closed-loop H_c may be defined as a function of H_o [11]. It reads

$$H_c = \frac{H_o(s)}{1+H_o(s)} = \frac{H(s)G(s)}{1+H(s)G(s)} \tag{2}$$

From Eq. (2), if the transfer function in open-loop H_o is measured, the transfer function in closed-loop H_c may be estimated without any other specific measurement. The calibration of the control-loop is here made using the frequency response function in open-loop H_o which is experimentally estimated. The modal characteristics under control, namely the poles of the transfer function in closed-loop H_c, are calculated for a given scalar gain g by finding the roots of its determinant. From Eq. (2), the equation $(1+H_o(s)=0)$ is solved to find the poles of the system under control. By doing so, the numerical calibration needs only one experiment in open-loop to characterize the control-loop on a large scalar gain range.

2.4 Case of Two-Phase Flow. In a two-phase flow, the modal characteristics in still fluid may not be measured due to the unstable state of a gas-liquid mixture. The calibration of the control-loop described above needs to be numerical. We propose below a procedure based on the measurements of the transfer functions in open-loop in still gas and in still liquid separately, which constitutes the first step of the required calibration. The structure is excited by the actuator of the control-loop. Second, the transfer function in open-loop in two-phase mixture H_o is derived from both transfer functions previously estimated. In a third step, the poles of the frequency response function in closed-loop H_c are calculated from the frequency response function in open-loop H_o.

2.5 Calibration Procedure in Two-Phase Flow. The calibration procedure of the control-loop in a two-phase mixture may be detailed in three steps:

(a) Experiments in open-loop; The structure is excited by the actuators separately in still gas and in still liquid, using for instance a white-noise excitation signal. Thus, the frequency response function in open-loop between the response of the sensor and the excitation due to the actuator is measured respectively in gas and in liquid. The modal parameters in still gas and in still liquid, such as frequencies, dampings and modal participations, may then be estimated using a Prony fitting [13] or a frequency fitting including the truncated transfer function form [14]

$$H_o(s) = -\sum_{i=1}^{m} \frac{\beta_i(s^2+2\zeta_i\omega_i s)}{s^2+2\zeta_i\omega_i s+\omega_i^2} \tag{3}$$

where the modal participation β_i includes the ratio between the modal projection coefficients of the sensor response and the actuator force with the modal mass. The modal participation expresses the efficiency of the control-loop. Note that all the electronic components of the control-loop which may include filters are not explicitly described in Eq. (3) but are taken into account in the fit of the frequency response function.

(b) Computation of the modal parameters in a still two-phase mixture, the modal characteristics, which may not be measured in a still two-phase mixture, are here derived from the experimental results in still gas and in still liquid using the homogeneous equilibrium model [15]. The frequency in a still two-phase mixture f_{tp} may be estimated from the frequencies in gas f_g and in liquid f_l by

$$f_{tp} = \frac{f_g f_l}{\sqrt{f_g^2(1-\varepsilon_g)+f_l^2\varepsilon_g}} \tag{4}$$

where ε_g is the homogeneous void fraction. Similarly, the modal mass in a two phase mixture M_{tp} is expressed as a function of the modal masses in gas M_g and in liquid M_l by

$$M_{tp} = M_g\varepsilon_g + M_l(1-\varepsilon_g) \tag{5}$$

The modal participation β_{tp}, which is equivalent to the inverse of a modal mass, Eq. (3), is estimated using the modal participations in gas β_g and in liquid β_l by

$$\beta_{tp} = \frac{\beta_g\beta_l}{\beta_g(1-\varepsilon_g)+\beta_l\varepsilon_g}. \tag{6}$$

The total fluid damping is calculated using state of the art models [3]. Finally, the transfer function in open-loop in a two-phase mixture is built up again using the modal characteristics calculated above.

(c) Numerical simulation of the control in still fluid; this simulation allows to derive the modal parameters in closed-loop for a given scalar gain from the modal characteristics in open-loop.

3 Experimental Application

3.1 Experimental Setup. We consider here a square bundle of tubes (Fig. 2), the central tube of which is flexible. The array, similar to that reported in [14,16–18], 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. The lowest natural frequency of the rigid tubes is about 900 Hz. It is confined in a $180\times300\ \mathrm{mm}^2$ vertical channel. The flexible system (Fig. 3) is made of the tube under flow attached to a flexible plate, which allows vibration in the lift direction only. Thus, the first bending

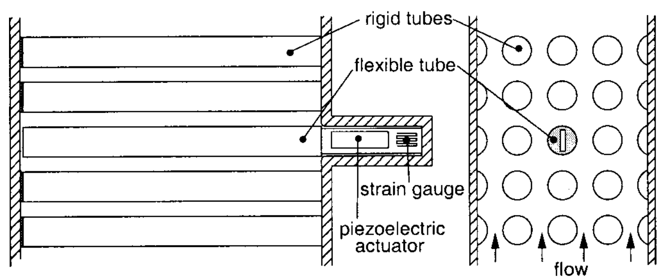

Fig. 2 Experimental square array of tubes

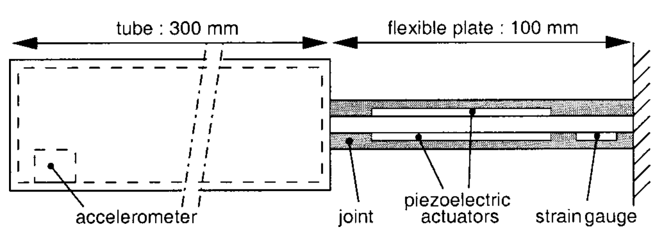

Fig. 3 Flexible tube with bonded piezoelectric actuators

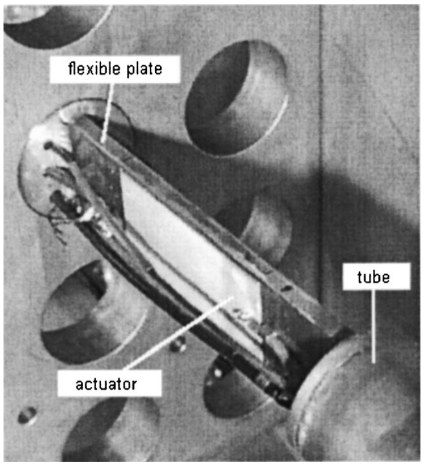

Fig. 4 Flexible plate equipped with piezoelectric actuators

mode in still water is about 18.7 Hz. The displacement of the vibrating tube is derived from a strain gauge bonded at the base of the flexible plate.

Two PZT actuators (Physik Instrumente, PIC151 material, $0.25\times22\times50\,\mathrm{mm}^3$) are symmetrically bonded on the plate (Figs. 3 and 4). 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 may be excited in open-loop or controlled in closed-loop using both piezoelectric actuators, which are supplied with power by two amplifiers (Physik Instrumente, E507.00). The electric field is applied across one piezoelectric actuator in one direction and reversed on the second. Thus, the two PZT actuators are assumed to act in pure bending on the structure [19]. The sensor of the control-loop is an accelerometer Endevco 2222C bonded at the end of the tube. The acceleration signal is integrated using a charge amplifier Brüel & Kjaer 2635 in order to obtain a direct velocity feedback law [11]. The scalar gain g is set using an amplifier Gearing & Watson. We can note that in the range of scalar gain used, we did not observe any instability of the structure due to the controller, though the direct velocity feedback technique is known to potentially bring instabilities [9].

3.2 Calibration of the Control-Loop. The procedure described in section 2 is now applied.

First, the frequency response functions in still air and in still water are estimated in open-loop using a frequency fitting on the first three modes (Table 1). It is shown that three modes are sufficient to estimate the modal parameters in closed-loop for our problem.

Second, the modal characteristics (frequency, damping and modal participation) in air-water mixture are calculated from the models described above for each void fraction (Table 2).

Third, for a given scalar gain and for a given void fraction, the new frequency and damping of the first bending mode at $U=0$ are calculated (Table 3). For each void fraction, positive and negative scalar gains are chosen. When the scalar gain is about $\pm300\,\mathrm{V.s.m}^{-1}$ and near the critical velocity, the voltage across the piezoelectric actuators is about 400 V which is close to the maximum of voltage acceptable by the actuators. When $g=0$, the controller has no effect on the structure, the modal characteristics in closed-loop are the same as in open-loop (Table 2).

Table 1 Results of the experiments in open-loop

Fluid	Mode	f_i (Hz)	ξ_i $(\%)$	β_i $(m.s^{-2}.V^{-1})$
AIR	1	28.87	1.0	-0.020
	2	312.8	0.61	-0.017
	3	1235	0.41	0.225
WATER	1	18.80	1.42	-0.0083
	2	227.2	0.72	-0.0092
	3	836.2	0.65	0.143

Table 2 Calculated modal parameters in still two-phase mixture

ε_g $(\%)$	Mode	f_i (Hz)	ξ_i $(\%)$	β_i $(m.s^{-2}.V^{-1})$
15	1	19.67	3.22	-0.0092
	2	235.7	0.72	-0.0097
	3	872.4	0.65	0.151
25	1	20.32	4.00	-0.0098
	2	241.9	0.72	-0.0102
	3	899.3	0.65	0.157
35	1	21.04	6.00	-0.0105
	2	248.7	0.72	-0.0108
	3	928.8	0.65	0.164
55	1	22.74	7.52	-0.0123
	2	264.1	0.72	-0.0121
	3	997.9	0.65	0.179

3.3 Fluidelastic Instability. The critical velocities of the structure are investigated with four homogeneous void fractions (15-25-35-55 percent) in air-water cross-flow when the damping of this tube is varied using vibration control. The vibration response curves are plotted on Fig. 5 for each void fraction. The flow velocity U is defined using the homogeneous equilibrium model [15].

In practice, the definition of the critical velocity for fluidelastic instability in a test in two-phase flow is a complex problem, which has been addressed by Pettigrew et al. [15]. Following these authors, we shall use a criterion on a critical RMS vibratory level of 1 mm at the end of the tube (Table 4). The stability threshold are shown in Fig. 6 as a function of the mass-damping parameters A_r and the reduced velocities U_r of Eq. (7).

$$A_r=\frac{2\pi m_{tp}\xi}{\rho_{tp}D^2}\qquad U_{rc}=\frac{U_c}{f_{tp}D} \tag{7}$$

where m_{tp} is the equivalent mass per unit length in two-phase mixture.

3.4 Discussion. The influence of the controller on the frequencies is quite limited as the frequencies remain constant when the scalar gain is varied in Table 3. The direct velocity feedback is

Table 3 Frequencies and dampings of the first bending mode in closed-loop

ε_g $(\%)$	g $(V.s.m^{-1})$	f (Hz)	ξ $(\%)$
15	-300	19.78	2.2
	0	19.67	3.2
	+300	19.55	4.2
25	-300	20.43	3.0
	0	20.32	4.0
	+300	20.20	5.0
35	-300	21.15	4.9
	0	21.04	6.0
	+300	20.92	7.1
55	-200	22.82	6.7
	0	22.74	7.5
	+300	22.61	8.7

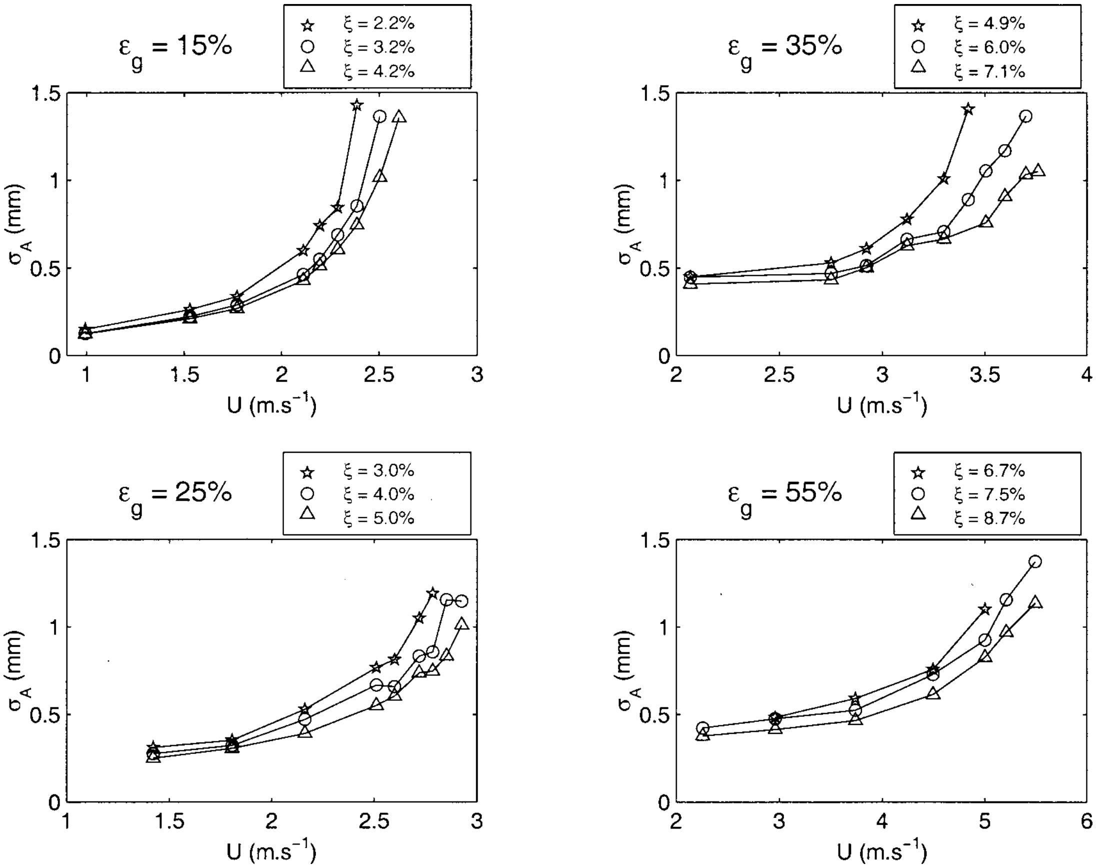

Fig. 5 Vibratory levels for the void fractions 15-25-35-55 percent

known to modify only the damping of the structure [11] which was the initial objective in the design of the control-loop.

The possibility of slip of the gas phase in vertical up-flow will result in overly high estimates of void fraction by the homogeneous equilibrium model. This would make the two-phase added mass estimation too low and the frequency estimate too high, particularly at high void fractions.

The critical velocities are estimated using the criterion on vibration amplitude proposed by Pettigrew et al. (1989). The use of other criteria would slightly shift our points in Fig. 6, without altering our conclusions.

In the map of Fig. 6, the results obtained may be compared with the general instability criterion given by: $U_{rc}=3A_r^{0.5}$. This criterion, proposed by Pettigrew and Taylor [3], is a conservative recommendation for the design of tube bundle of pitch-to-diameter ratio between 1.4 and 1.5. The critical reduced velocities measured as the mass-damping parameter is artificially varied using vibration control are in agreement with this guide-line. The reduced critical velocities obtained are over the design guide-line because a single flexible tube is more stable than a wholly flexible tube bundle [5].

For the four void fractions considered here, the reduced damping of the first bending mode is varied approximately of 0.02

Table 4 Critical velocities and nondimensional parameters

ε_g (%)	g ($V.s.m^{-1}$)	U_c ($m.s^{-1}$)	A_r	U_{rc}
	-300	2.30	0.25	4.10
15	0	2.41	0.36	4.30
	+300	2.45	0.47	4.37
	-300	2.69	0.36	4.42
25	0	2.82	0.48	4.63
	+300	2.93	0.60	4.81
	-300	3.30	0.63	5.19
35	0	3.44	0.76	5.41
	+300	3.65	0.89	5.74
	-200	4.88	1.05	7.70
55	0	5.07	1.18	7.35
	+300	5.24	1.36	7.60

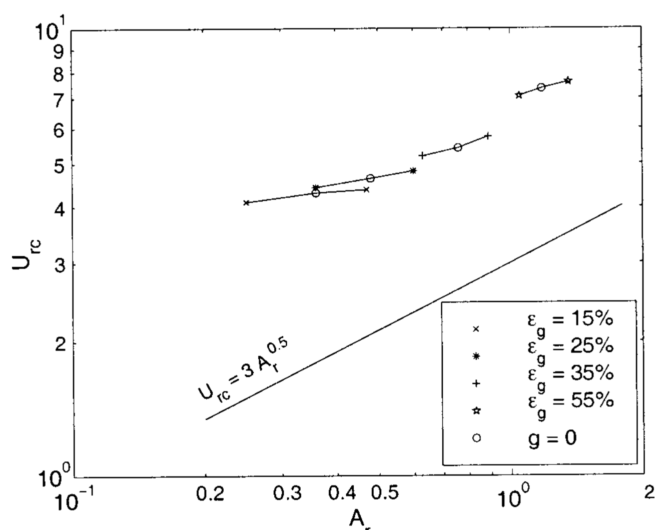

Fig. 6 Stability map

(Table 3) which is significant. Nevertheless, the range of application of the method is here limited by the power of the piezoelectric actuators. Note that the efficiency of our test facility in terms of added damping, in two-phase flow, is equivalent to that of Meskell and Fitzpatrick [12], in air flow.

4 Conclusion

The active vibration control method proposed in this paper is shown to be efficient in order to display several critical velocities on a given tube bundle, corresponding to several values of damping.

The present method allows to increase the number of tests without any changes on the mock-up by the sole modification of the scalar gain of the control-loop. The direct velocity feedback allows to set here the damping of the fluid-structure system, but other feedback laws may be used. For instance, it may be interesting to change the frequency of the tubes using an appropriate feedback law, and by doing so to vary artificially the reduced velocities.

Finally, active vibration control in the measurement of fluidelastic effects has two main interests:

(a) to decrease the costs of the tests (less tests facilities and less testing time).
(b) to provide new results by exploring the range of higher flow velocities.

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Nomenclature

A_r = mass-damping parameter
D = diameter of the tube
G = transfer function of the compensator
H = transfer function of the system
M = modal mass
U = pitch flow velocity
U_r = reduced velocity
f = frequency
g = scalar gain
m = equivalent modal mass per unit length
s = Laplace's variable
β = modal participation coefficient of H
ϵ_g = homogeneous void fraction
ω = circular frequency
ρ = density
σ_A = standard deviation at the end of the tube
ξ = reduced damping

Subscripts

c = closed-loop
g = gas
l = liquid
o = open-loop
r = reduced parameter
tp = two-phase

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