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Active Vibration Control for the Measurement of Fluidelastic Effects

A new method based on active vibration control is proposed to investigate fluidelastic coupling effects beyond fluidelastic instability. This active control method allows to extend the range of flow velocity explored for single input-single output control systems. The method is applied on a flexible tube inserted in a rigid bundle in water and air-water cross-flows. This structure becomes unstable for high flow velocities, fluidelastic forces then causing the damping of the fluid-structure system to fall towards zero. The active control method allows to carry out tests beyond the fluidelastic instability. The flow velocity range explored is doubled in two-phase flow. [DOI: 10.1115/1.1561451]

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 instability, a subject of constant research efforts in 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 towards zero of the damping of the fluid-structure system may be observed [4]. Since theoretical and numerical predictions of the fluidelastic instability remain difficult [5–8], experimental approaches have been developed.

A first approach is to display critical velocities when the array characteristics and the fluid properties are varied. This leads to the construction of stability map such as in Connors [1]. A second approach, based on fluidelastic forces, has been proposed by Tanaka et al. [9,10] where the fluidelastic forces are studied as a function of flow velocity. This allows a more complete description of the fluidelastic phenomenon over a large flow velocity range. In that case, two experimental methods are commonly then used to investigate fluidelastic forces: the direct method [2,9,10], which allows the measurement of the fluid forces acting on all tubes induced by the harmonic motion of a driven tube, and the indirect method [4,11], where fluidelastic forces are derived from the evolution of the modal characteristic of the fluid-structure system.

At the same time, vibration control has been shown to be efficient for in situ applications of controlled forces in vibrating system under fluid flow by Baz and Ro [12] and Kaneko and Hirota [13]. Moreover, Meskell and Fitzpatrick [14] have used a passive technique and an electromagnetic shaker to control a tube under cross-flow and to identify fluidelastic instability.

In [15], we have used active vibration control to build an instability map. The damping of the fluid-structure system was then artificially varied to derive several critical velocities for fluidelastic instability. We propose here a new method based on active vibration control in the indirect method in order to derive the fluidelastic forces at instability. The originality of this active control method (ACM) is based on the use of active vibration control for the identification of fluidelastic coupling, which effects on the controller are taken into account. Modifications of the dynamic system induced by the active control may be measured, but may also be estimated from the characteristics of the active control-loop as shown by Preumont [16]. We propose here a method based on experiments and calculations in order to derive the fluidelastic forces using active vibration control and the indirect method. The principle of the method remains general and is not specific to tube bundles.

Experimental tests are performed with a flexible tube equipped with an electromagnetic shaker and inserted in a tube bundle in water and air-water cross-flows. The ACM combined with the classical indirect method [4,11] and the added excitation method [17], which is available at low flow velocities, gives a complete description of fluidelastic coupling on a large flow velocity range.

2 Active Control Method

Active vibration control allows the variation of the modal characteristics of a flexible structure. More particularly, damping may be artificially added or subtracted to the initial value. The main objective of the present active control method (ACM) is to set the modal damping of the fluid-structure system to any positive required value. An active control technique based on transfer function, which is supposed to be efficient for single input-single output (SISO) control system [16], is used here.

2.1 The Active Control Technique. Active vibration control, using transfer functions, 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 $H_s(s)$ and $H_k(s)$ where $H_k(s)=g\ h_k(s)$ and g is the scalar gain. When g is varied, the efficiency of the controller varies. Thus, a given added damping corresponds to a given scalar gain g.

A SISO control-loop 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 the frequency response function in open-loop $H_o(s)$, may be measured. The structure is then directly excited by the actuator of the control-loop. The function $H_o(s)$ is defined as

$$H_o(s)=H_k(s)H_s(s) \tag{1}$$

The modal characteristics of the structure with no control, which are the poles of $H_s(s)$, may be derived from $H_o(s)$ as

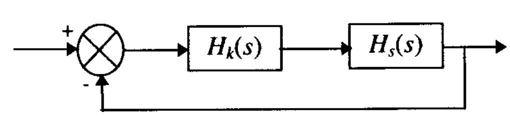

Fig. 1 Block diagram of the control loop

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shown in Eq. (1). On the other hand, when the structure is under control, the transfer function in closed-loop $H_c(s)$ may be defined as a function of $H_o(s)$ [16]. It reads

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

From Eq. (2), if $H_o(s)$ is known, $H_c(s)$ may be estimated without any other specific measurement. The modal characteristics of the system under control, namely the poles of $H_c(s)$, may be 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 then solved to find the poles of the system under control.

2.2 Fluidelastic Coupling and Active Vibration Control. The structure is subjected to fluidelastic coupling and to coupling induced by the controller. The controller acts on the fluid-structure system which is illustrated in Fig. 2. The fluid-structure system alone and the fluid-structure system under control correspond respectively to the system in open-loop and to the system in closed-loop. Here, the system in open-loop (i.e., H_o) is partly unknown as the fluidelastic coupling forces are the object of the tests and modify the modal characteristics of the system. Finally, $H_c(s)$ can not be calculated using Eq. (2).

We propose here to determine experimentally the modal characteristics of the system under control which is subjected to fluid flow. In that case, the structure is excited by turbulence. The modal characteristics of the structure are estimated on the spectral density of its vibratory response induced by the fluid excitation. We propose now to calculate $H_o(s)$ using $H_c(s)$. From Eq. (2), we have

$$H_o(s)=\frac{H_c(s)}{1-H_c(s)} \tag{3}$$

The determination of the frequency and damping in open-loop is divided into four steps:

1. Calculation of $H_o(s)$ in still fluid $(V=0)$

This step is the definition of the reference system, which is not subjected to fluidelastic coupling, the flow velocity V being zero. The transfer function in open-loop and in still fluid, noted $H_o^0(s)$, may be expressed as a function of its zeros z_i and its poles p_i.

$$H_o^0(s)=K\frac{\prod_{i=1}^{b}(s-z_i)}{\prod_{i=1}^{a}(s-p_i)} \tag{4}$$

2. Calculation of $H_c(s)$ in still fluid $(V=0)$

This step allows the calculation of the (*a-m*) poles which are not affected by fluidelastic coupling. These poles may be poles of the structure and poles of the electronic control loop. We distinguish the *m* structural poles which are subjected to fluidelastic effects from the poles which are not. The transfer function in closed-loop and in still fluid, noted $H_c^0(s)$, is calculated using Eq. (2) as $H_o^0(s)$ is known from step 1.

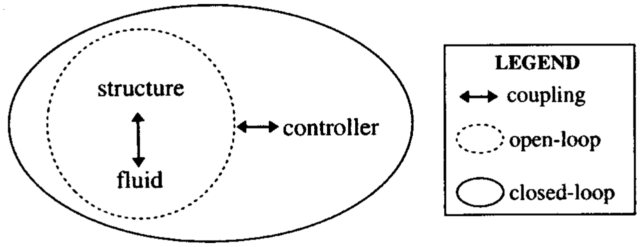


Fig. 2 Fluidelastic coupling and active control diagram

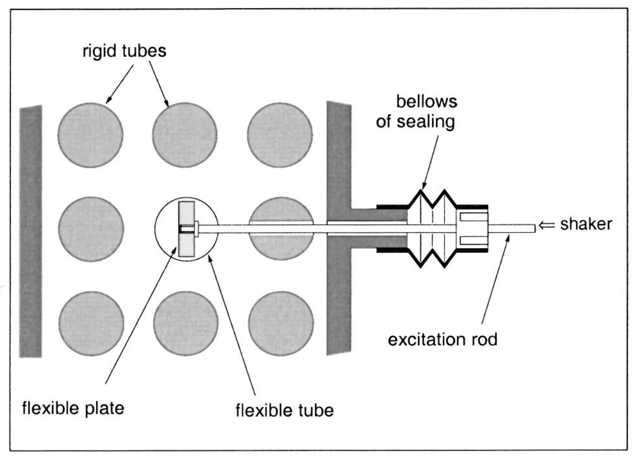
$$H_c^0(s)=K\frac{\prod_{i=1}^{b}(s-z_i)}{\prod_{i=1}^{m}(s-p_i^k)\prod_{i=m+1}^{a}(s-p_i^k)} \tag{5}$$

3. Determination of $H_c(s)$ with $V>0$

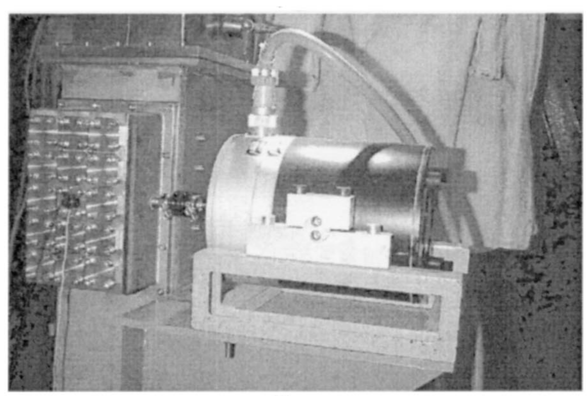
The structural poles which are supposed to be affected by fluidelastic coupling are experimentally derived in closed-loop using the spectral density of the response of the structure subjected to turbulent fluid forces; From steps 1 and 2, $H_c^0(s)$ is modified in $H_c^V(s)$ by replacing the *m* structural poles p_i^k of Eq. (5) by the *m* experimentally estimated poles p_i^{fk}, so that

$$H_c^V(s)=K\frac{\prod_{i=1}^{b}(s-z_i)}{\prod_{i=1}^{m}(s-p_i^{fk})\prod_{i=m+1}^{a}(s-p_i^k)} \tag{6}$$

4. Calculation of $H_o(s)$ with $V>0$



(*a*)



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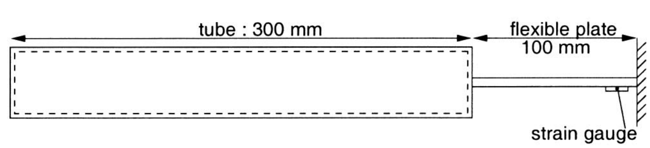

(*c*)

Fig. 3 Experimental setup—(*a*) sight partial of the tube bundle, (*b*) the vertical channel and the shaker, (*c*) flexible tube

This last step allows to determine the modal characteristics of the system subjected to fluidelastic coupling only. Using Eq. (3), the transfer function in open-loop and under fluid flow, $H_o^V(s)$, may be calculated.

$$H_o^V(s)=K\frac{\prod_{i=1}^{b}(s-z_i)}{\prod_{i=1}^{m}(s-p_i^f)\prod_{i=m+1}^{a}(s-p_i)} \tag{7}$$

The comparison of the reference structural poles p_i of Eq. (4) and the modified poles p_i^f of Eq. (7) allows the determination of the fluidelastic effects acting on the structure at a given flow velocity. The frequency and the damping may be easily derived from the value of the corresponding pole using

$$p=-\xi\omega+j\omega\sqrt{1-\xi^2} \tag{8}$$

The indirect estimation of fluidelastic effects using the evolution of frequency and damping are detailed in [4] and [17].

3 Experimental Application

3.1 Application of the ACM to Tube Arrays. In single-phase flow, the modal parameters in still fluid, such as frequency, damping and modal participation, may be estimated using a Prony fitting [18] or a frequency fitting [17]. In two-phase flow, the modal characteristics in still fluid may not be measured due to the unstable state of a gas-liquid mixture. The first step of the ACM proposed here needs to be numerical. We may use the procedure proposed by Caillaud et al. [15], which is based on the measurements of the transfer functions in open-loop in still gas and in still liquid separately. The structure is excited by the actuator of the control-loop. The transfer function in open-loop in still two-phase mixture $H_o^0(s)$ is derived from both transfer functions previously estimated. The frequencies and modal participations are derived from the experimental results in still gas and in still liquid using the homogeneous equilibrium model [3]. The total fluid damping is calculated using state of the art models (here [3]).

3.2 Experimental Setup. We consider here a square bundle, the central tube of which is flexible (Fig. 3(*a*)). The array, similar to that reported in [11,15,17,19–21], 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 confirmed in a $180\times300\ \mathrm{mm}^2$ vertical channel. The flexible system (Fig. 3(*c*)) is made of a 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 38 Hz. The displacement of the vibrating tube is derived from a strain gauge bonded at the base of the flexible plate.

The actuator is an electromagnetic shaker (Prodera, EX220) attached to the flexible plate (Figs. 3(*a*) and (*b*)). The tube may be excited in open-loop or controlled in closed-loop by this shaker. The sensor of the control-loop is an accelerometer Endevco 2222C bonded at the middle of the tube. The acceleration signal is integrated using a charge amplifier Brüel & Kjaer 2635 in order to obtain a direct velocity feedback law [16]. The scalar grain 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 as shown by Baz and Ro [12].

3.3 Results in Water Cross-Flow. Two methods are used here for the experimental determination of fluidelastic effects. The usual indirect method [4,11] is used when the tube under cross-flow is stable. The ACM is applied beyond the critical velocity for fluidelastic instability. In Fig. 4, the vibration amplitude at the free end of the tube σ_A, the frequency f and the reduced damping ξ of the first bending mode are given as a function of pitch-flow velocity V.

Up to $V=4$ m/s, the indirect method is used. The tube becomes unstable as the reduced damping is falling towards zero. The control gain g is then set to 1 N.s/m; The vibration amplitude is divided by two and we can proceed with the tests. When the vibration amplitude at the end of the tube σ_A is near 0.8 mm, the control gain g is increased again.

The modal characteristics (f, ξ) of the fluid-structure system in closed-loop using the ACM are estimated on the spectral density of response of the tube using a frequency fitting. The damping in closed-loop remains positive. The modal characteristics of the fluid-structure system in open-loop are calculated using the ACM. The damping of the fluid-structure system becomes negative for $V>5$ m/s. The reduced damping in closed-loop for the last measurement ($V=5.8$ m/s) is 2.72% which corresponds to a reduced damping of -1.25% in open-loop. For higher velocities, the very high level of turbulence buffeting and fluidelastic forces do not allow the ACM to be used as the shaker is limited in power.

The results on the frequency and the reduced damping in open-loop are continuous between both experimental methods and when the control gain is varied.

3.4 Results in Air-Water Cross-Flow. The tests are realized using both indirect method and ACM as in water cross-flow. The homogeneous void fraction ε_g is set to 15, 25, 35, 55, 70, and 85%. For each void fraction, the frequency (Fig. 5) and the reduced damping (Fig. 6) are given. As in water cross-flow, the results are continuous except on the frequency for $\varepsilon_g=85\%$.

3.5 Discussion. The results obtained illustrate the efficiency of the proposed method. In two-phase flow, the range of flow velocity explored is doubled. In water cross-flow, the value of this multiplication factor is about 1.3 as buffeting excitation level is higher.

The continuity of the results, between both experimental methods and when the control gain is varied, validates the ACM. This new method has also been validated on the stable region by comparison with the indirect method, which is known to be accurate, in [21].

For low void fractions (15 and 25%), we observe an oscillation of damping for high flow velocities (Fig. 6). For $\varepsilon_g=25\%$, the damping is negative when $6<V<8.1$ m/s and $9.5<V<10.1$ m/s. Finally, when flow velocities are between 8 and 9.5 m/s, the system is restabilized. These oscillations may also be observed on the frequency (Fig. 5) for $\varepsilon_g=25\%$. This oscillation of damping has not been reported before.

4 Conclusion

A new method based on active vibration control and the indirect method is proposed to measure fluidelastic effects. As active control techniques may be used to stabilize unstable structures under fluid flow, the ACM presented here may be extended to structures which show strong fluidelastic coupling.

This method is applied to the case of tube bundles under single-phase flow and two-phase flow. Its efficiency is shown on an experimental array, the central tube of which is flexible, in water and in air-water cross-flows. Fluidelastic coupling is investigated beyond fluidelastic instability and the flow velocity range explored is doubled in two-phase flow.

The method proposed here is only available for Single Input-Single Output systems. For instance, it can not be applied to a flexible tube bundle. In that case, the problem is quite complex for two main reasons: the fluidelastic coupling between tubes must be taken into account and more actuators may be added to the system. Finally, the present approach, which is only applicable to a

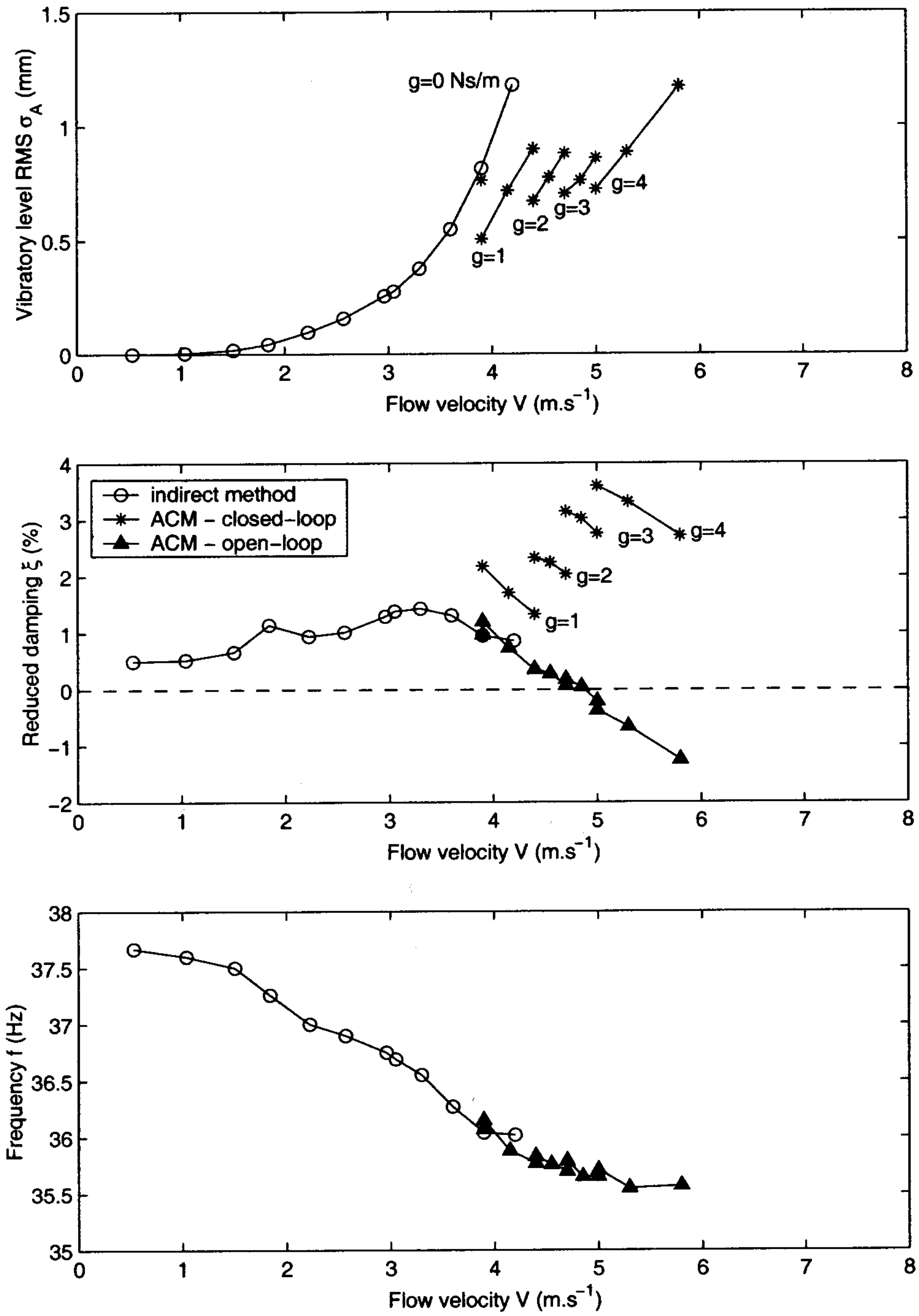

Fig. 4 Experimental results in water cross-flow

single flexible cylinder, is quite limited for industrial applications such as steam generators tubes.

Nomenclature

f = frequency
g = scalar gain of controller
H = transfer function
p = pole of transfer function
s = Laplace's variable
V = pitch flow velocity
z = zero of transfer function
ε_g = homogeneous void fraction
σ_A = standard deviation at free end of tube
ξ = reduced damping

Subscripts

0 = structure in still fluid-$V=0$
c = closed-loop
f = fluid
k = controller
o = open-loop
s = structure
V = structure in cross-flow-$V \neq 0$

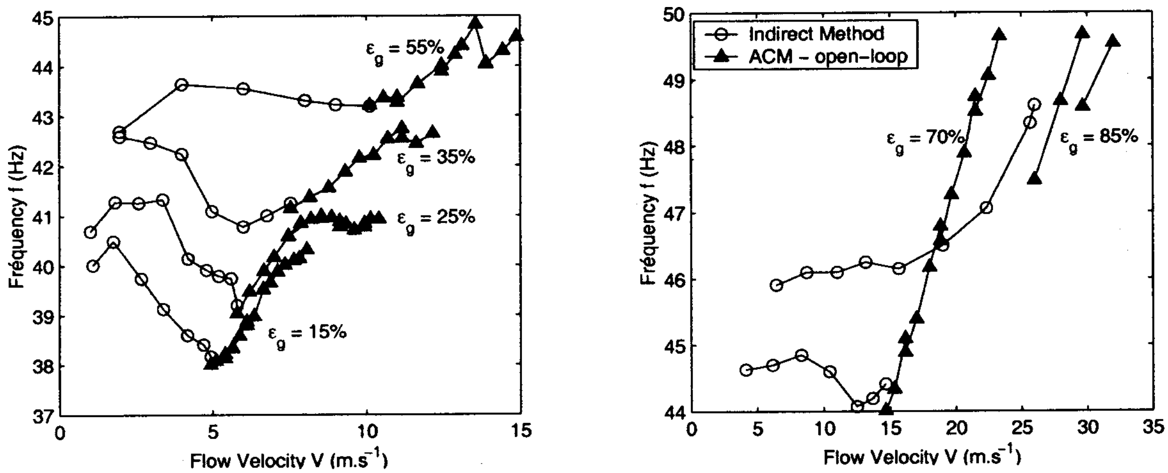

Fig. 5 Experimental frequencies in air-water cross-flow

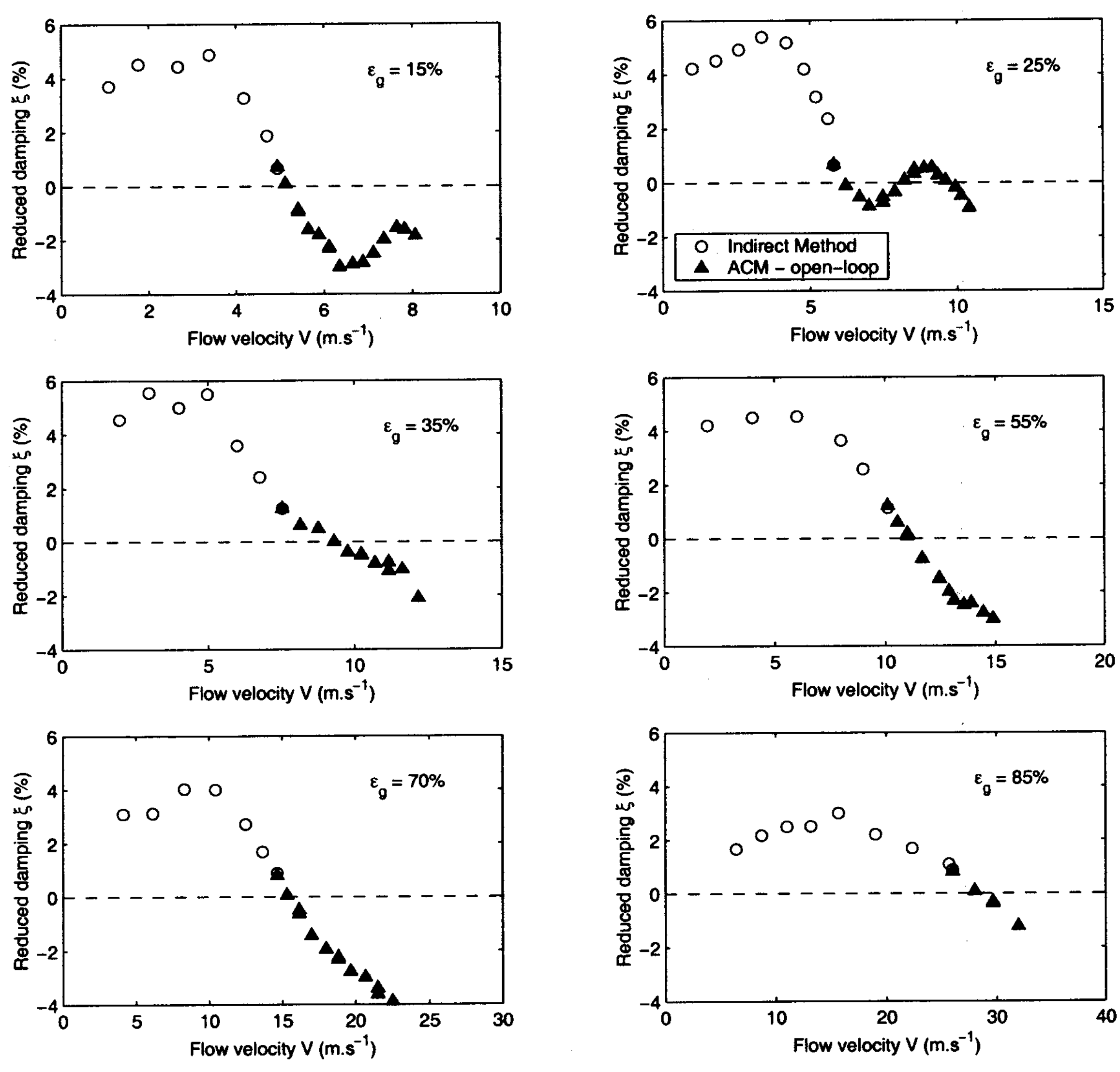

Fig. 6 Experimental reduced damping in air-water cross-flow

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