ACOUSTIC DAMPING IN CAVITATION-INDUCED TWO-PHASE FLOW

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ABSTRACT

In recent experiments during which a globe valve and a single hole orifice were tested in cavitation regime, a correlation between the acoustic damping coefficient and the speed of sound was observed. The results are proposed as a contribution to two-phase flow vibrations, and theoretical data are provided to support the correlation observed. Further work is needed to determine whether this correlation is dependent on the pressure drop device tested or the test rig or not.

1. INTRODUCTION

Two-phase flow damping is a concern in many fields of flow-induced vibrations, as for instance predicting conditions of fluid-elastic instability of tube bundles in Steam Generators (see the reviews in Weaver et al. 2000; Gravelle et al., 2006; Anscutter et al., 2006). Yet the damping is usually taken into account by using an average coefficient associated with the broadness of resonance peaks, and not by using well defined properties of the fluid. This is especially the case in pipe vibrations, where two-phase flows are known to increase the global damping coefficient of the coupled fluid/structure system and to reduce the speed of sound (Hassis, 1999).

Several studies have recently been made to evaluate the pressure fluctuations generated downstream of pressure drop device in cavitation conditions (Testud et al. 2005; Caillaud et al, 2006; Testud et al. 2007). A striking feature of two-phase flows during the experiments was that the speed of sound was prone to spontaneously evolve, without any apparent variation of the experimental conditions. The reason of this spontaneous evolution could not be determined at the time of the tests. By force, the pressure spectra obtained were labeled by the speed of sound, and, comparing data obtained with the same speed of sound, a fair collapse of the pressure Power Spectrum Densities (PSD) was obtained (Testud et al. 2005). Furthermore, it was observed that the PSD exhibited significantly lower values in the frequency range 100-1000 Hz when the speed of sound was low, as if self-quenching effects were at stake.

One comes to the conclusion that some relation exists between the speed of sound and an acoustic attenuation effect in the presence of two-phase flow. The purpose of the present paper is to support this proposition with experimental data.

2. EXPERIMENTAL PROCEDURE

2.1 Test rig description

The test section, shown in Fig. 1, consists out of an open loop with a hydraulically smooth steel pipe of inner diameter $D = 74$ mm and wall thickness $t_p = 8$ mm. The water is injected from a tank located 17 m upstream from the orifice. The nitrogen pressure in the tank above the water is controlled by a feedback system to maintain a constant pressure. The water is released at atmospheric pressure 20 m downstream of the orifice. The temperature is kept equal to 310 ± 1 K during all experiments.

The test rig is equipped with several series of three pressure sensors equally spaced, so that identification of propagating pressure waves can easily be achieved.

A first series of experiments was performed with a thin single hole orifice with a hole diameter equal to 22 mm. A second series of experiments was performed with a globe-style valve. All the experiments were made with upstream pressures varying from 10 bars to 50 bars and downstream pressures of the order of 3 bars, so that cavitation always occurred (see details of the experiments in Caillaud et al, 2006 and Testud et al., 2007).

2.2 Acoustic pressure waves identification

Due to the relatively low value of the speed of sound in gas systems, the identification of forward and backward propagating waves from pressure sensors demands the steady fluid velocity be taken into account (see for example Holland, 2000). In water pipes, the flow velocity seldom exceeds 10 m/s whereas the speed of sound is of the order of 1400 m/s, so that simpler expressions of the propagating waves can be used.

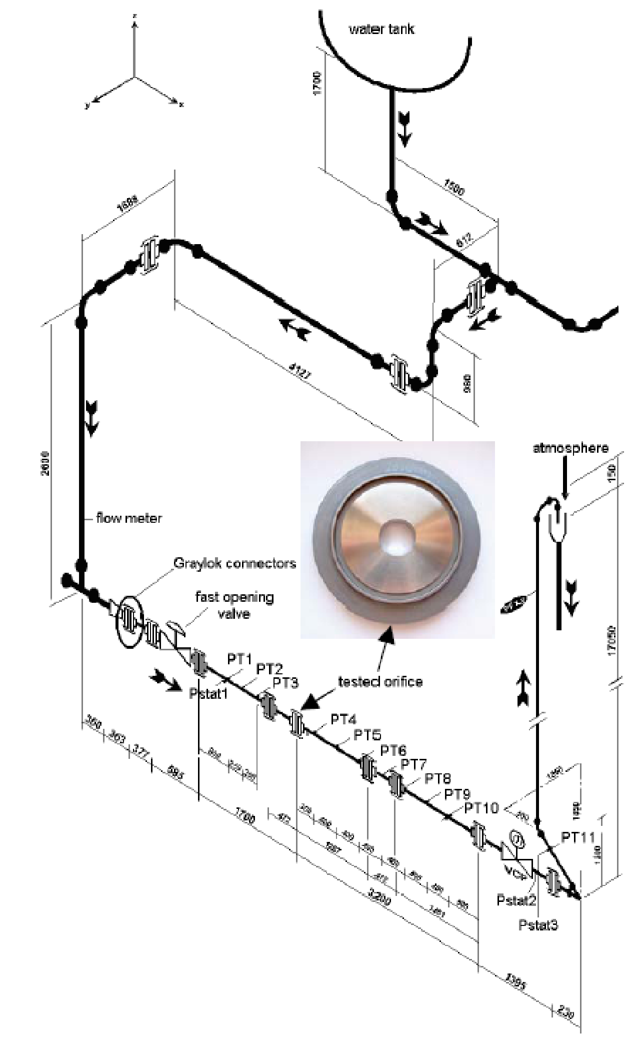

Figure 1: experimental facility used during the first series of tests

Let three pressure sensors be considered, indexed by 1, 2 and 3, and such that the distance between the sensors 1 and 2 and the distance between the sensors 2 and 3 are identical. Denoting L this distance and c the speed of sound in the pipe, the forward propagating spectrum p_+ exhibits a time delay τ equal to L/c from one sensor to the next, i.e., in the time domain:

$$p_+(x_1, t - \tau) = p_+(x_2, t) = p_+(x_3, t + \tau) \qquad (1)$$

Similar expressions hold for the backward propagating pressure p_-:

$$p_-(x_1, t + \tau) = p_-(x_2, t) = p_-(x_3, t - \tau) \qquad (2)$$

Let the steady pressure fluctuations be described by their PSD, and let a sensor denoted "ref" be used as a phase reference.

The relations (1) become

$$e^{-j\omega\tau}\, C[p_+(x_1), p_{ref}, \omega] = C[p_+(x_2), p_{ref}, \omega]$$
$$C[p_+(x_2), p_{ref}, \omega] = e^{j\omega\tau}\, C[p_+(x_3), p_{ref}, \omega]$$

where $C(p_i, p_j, \omega)$ is the cross spectrum of the pressures p_i and p_j at the circular frequency ω (in Pa²/Hz). In a similar way, the relations (2) become

$$e^{j\omega\tau}\, C[p_-(x_1), p_{ref}, \omega] = C[p_-(x_2), p_{ref}, \omega]$$
$$C[p_-(x_2), p_{ref}, \omega] = e^{-j\omega\tau}\, C[p_-(x_3), p_{ref}, \omega]$$

Identification of the forward and backward traveling pressure waves is achieved by combining the above equations and noting that the acoustic pressure is the summation of the forward and backward propagating waves. Considering for instance the sensors 1 and 2, one has:

$$C[p_+(x_2), p_{ref}, \omega] + C[p_-(x_2), p_{ref}, \omega] = C[p_2, p_{ref}, \omega]$$
$$e^{j\omega\tau} C[p_+(x_2), p_{ref}, \omega] + e^{-j\omega\tau} C[p_-(x_2), p_{ref}, \omega] = C[p_1, p_{ref}, \omega]$$

Inverting the above equation system, the propagating waves can be expressed as:

$$C[p_+(x_2), p_{ref}, \omega] = \frac{-j(C(p_1, p_{ref}, \omega) - e^{-j\omega\tau} C(p_2, p_{ref}, \omega)}{2\sin\omega\tau}$$
$$C[p_-(x_2), p_{ref}, \omega] = \frac{j(C(p_1, p_{ref}, \omega) - e^{j\omega\tau} C(p_2, p_{ref}, \omega)}{2\sin\omega\tau}$$

The time of flight τ can be determined by expressing the cross spectra at points 1 and 3:

$$e^{j\omega\tau} C[p_+(x_2), p_{ref}, \omega] + e^{-j\omega\tau} C[p_-(x_2), p_{ref}, \omega] = C[p_1, p_{ref}, \omega]$$

$$e^{-j\omega\tau} C[p_+(x_2), p_{ref}, \omega] + e^{j\omega\tau} C[p_-(x_2), p_{ref}, \omega] = C[p_3, p_{ref}, \omega]$$

and summing up the expressions so that a cosine is obtained:

$$\cos\omega\tau = \frac{C[p_1, p_{ref}, \omega] + C[p_3, p_{ref}, \omega]}{2C[p_2, p_{ref}, \omega]} \qquad (3)$$

The relation (3) can be used to check that the pressure measurements do not incorporate local turbulence, and to evaluate the quality of the acoustic propagation. It is also used to evaluate the value of the speed of sound. Examples are given in Fig. 2 for the single orifice case, where the sensors distance was equal to 40 mm; the speed of sound evolved spontaneously from 1420 m/s to 650 m/s during the test with a flow velocity equal to 26 m/s in the hole.

3. ACOUSTIC PROPERTIES OF TWO-PHASE FLOW

In the present section, some theoretical properties of two-phase flows relative to acoustic propagation are summarized.

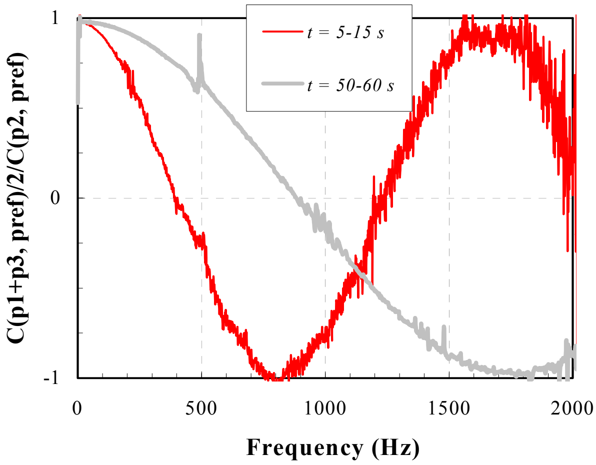

Figure 2: example of cosine combination for determining the speed of sound

3.1 Speed of sound and void fraction

In single-phase flow, the speed of sound c in a water-filled pipe can be theoretically determined with the help of the formula of Young (Lighthill, 1978) for the compliance of the pipe:

$$\frac{1}{\rho_w c^2} = \frac{1}{\rho_w c_w^{\ 2}} + \frac{D\,(1-\zeta^2)}{t_p\ E} \tag{1}$$

where ζ is the Poisson ratio, ρ_w the volume density of water, t_p the thickness of the pipe wall, E the Young modulus of the pipe, c_w the speed of sound in pure water. Taking $\zeta = 0.3$ for steel, $c_w = 1523$ m/s at $T = 310$ K, $E = 2\ 10^{11}$ Pa, $t_p = 8$ mm and $\rho_w = 10^3$ kg/m^3, a value of $c = 1454$ m/s is obtained, in close agreement with the measurements of the speed of sound obtained upstream of the orifice.

The density of a mixture of gas and water is:

$$\rho_{2\varphi} = \beta\rho_g\ +\ (1\ - \beta)\rho_w$$

where $\rho_{2\varphi}$ is the volume density of the mixture, ρ_g is the volume density of gas and β the void fraction.

Assuming an homogeneous behavior of each phase and a 'frozen' constant mass ratio between phases, the compliances combine according to:

$$\frac{1}{\rho_{2\varphi} c_{2\varphi}^2} = \frac{\beta}{\rho_g c_g^2} + \frac{1-\beta}{\rho_w c_w^2}$$

where c_g is the speed of sound in the gas and ρ_g is the density of the gas. The speed of sound in the gas is taken equal to 340 m/s.

The estimation of the gas density can be achieved by assuming the gas to be ideal and the compression to be an adiabatic transform. Denoting P the static pressure and γ the Poisson constant of the gas, taken here equal to 1.4, one gets:

$$c_g^2 = \frac{\gamma P}{\rho_g}$$

The resolution of the three former equations is straightforward, and the two phase speed of sound can be determined as a function of the void fraction β (see Fig. 3).

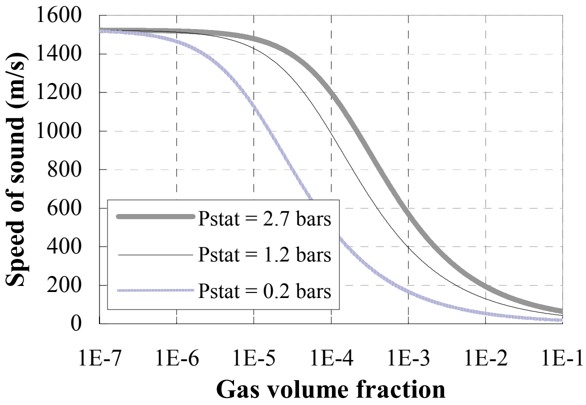

Figure 3: two-phase speed of sound vs. void fraction

As can be seen, a very small void fraction is enough to significantly decrease the speed of sound. As the initial void fraction was not measured during the experiments, one comes to the conclusion that this void fraction spontaneously evolved, and that it is the cause of the evolution of the speed of sound during the single orifice tests.

3.2 The Wijngaarden model of attenuation

The issue is now to determine how the acoustic plane waves can be attenuated when traveling through a two-phase flow. This can be achieved by energy balance during one cycle of compression/expansion of the fluid, assuming that dissipation occurs within the gas bubbles.

The elastic energy U_{bubble} of a single bubble with a volume V submitted to a fluctuating pressure with a PSD equal to $C_{pp}(\omega)$ can be written in the linear approximation as:

$$U_{bubble} = V\ C_{pp}(\omega)\ /\ \rho_g c_g^2$$

The bubble does not expand instantaneously when the pressure increases, and this delay generates energy dissipation during one cycle. Using the Chapman and Plesset data provided by Van Wijngaarden (1972), the energy loss can be described by a logarithmic decrement coefficient Λ, with a value close to 0.4 for bubbles with a radius varying from 1 µm to 1 mm. More specifically, during one cycle of compression/ expansion, the energy decrease rate is equal to Λ/T times the bubble energy. This expression can be extended to all frequencies according to:

$P_{bubble} = 2\pi\ \omega\ \Lambda\ V\ C_p(\omega)\ /\ \rho_g c_g^2$

P_{bubble} being the dissipated power spectrum density for one bubble (in W/Hz). Summing up the contributions of all bubbles in one volume unit, one gets the volumic dissipated power as a function of the void fraction β:

$P_{vol} = 2\pi\ \omega\ \Lambda\ \beta\ C_p(\omega)\ /\ \rho_g c_g^2$

The issue is now to apply this law to the propagation of acoustic pressure waves. Considering a forward propagating wave p_+, and assuming the steady flow velocity be negligible compared to the speed of sound, the acoustic energy flow can be written $p_+^2/\rho c$ in the time domain (see for instance Morse and Ingard, 1968). Using the PSD formalism, and demanding the acoustic energy loss per unit length along the pipe axis be equal to the dissipation, one gets:

$$\frac{dC_{p_+p_+}(\omega)}{dx} = -\frac{2\pi\omega\Lambda\beta}{c_{2\varphi}}\frac{\rho_{2\varphi}c_{2\varphi}^2}{\rho_g c_g^2}C_{p_+p_+}(\omega)$$

Hence, the attenuation of a propagating acoustic wave appears to be proportional to the frequency. Introducing a dimensionless damping coefficient α by:

$$\alpha = 2\pi\Lambda\beta\frac{\rho_{2\varphi}c_{2\varphi}^2}{\rho_g c_g^2}, \qquad (4)$$

the propagation of a pressure wave is described by

$$C_{p_+p_+}(x,\omega) = C_{p_+p_+}(0,\omega)e^{(j-\alpha)\frac{\omega x}{c_{2\varphi}}}. \qquad (5)$$

Hence, one needs two series of three sensors at two different locations to evaluate the attenuation coefficient α.

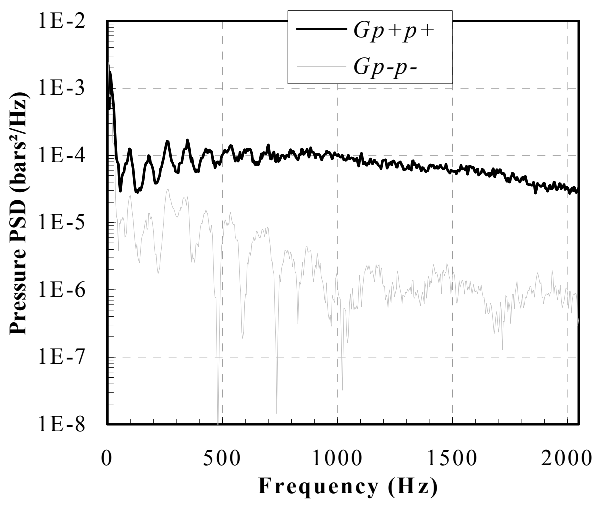

Figure 4: example of identification of forward and backward propagating pressure waves

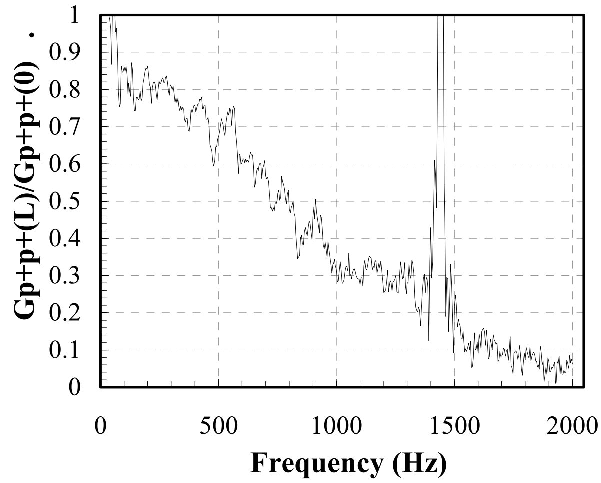

Figure 5: example of evaluation of the attenuation as a function of the frequency

4. RESULTS

Estimation of the acoustic attenuation within the pipe is obtained by determining the PSD of forward propagating pressure waves according to the formulas of section 2.2 at different locations (see an illustration in Fig. 4). The attenuation coefficient α is then estimated by plotting the ratio of the forward propagating pressure waves at the two locations, and estimating the slope of the ratio, as shown in Fig. 5.

Most of the tests performed with the single hole orifice could not be used for evaluating the acoustic attenuation, because the first and second series of pressure sensors were too close, so that the accuracy was poor. Furthermore, the speed of sound appeared to be significantly different between the first and the second series, which suggests that the pressure sensors were in the downstream area where two phase flows still evolved. Only a few measurements can be used, keeping in mind that the uncertainty associated with them is high. More precise measurements were obtained with the globe style valve.

The results are given in Fig. 6 for the globe valve in three different operating conditions, depending on the pressures upstream and downstream, and for the single orifice when the measurements were accurate enough to estimate the acoustic damping. As can be seen, all data collapse on the same curve.

Applying Equ. (4) does not bring a fair agreement with the data. A value of Λ of the order of 0.004 instead of 0.4 should be used to fit approximately the data, and only for speed of sound higher than 600 m/s. This suggests that very small bubbles were involved in the experiments, and that their diameter evolved with the operating conditions.

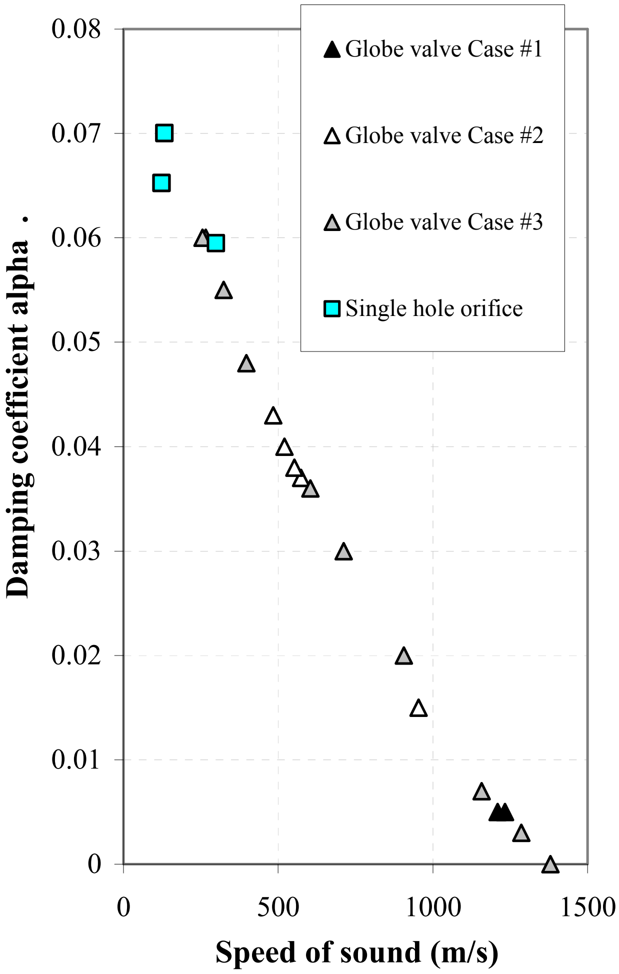

Figure 6: attenuation coefficient as a function of the speed of sound

5. PERSPECTIVES

An acoustic attenuation vs. speed of sound curve was proposed in two phase flows. Whether the results obtained can be applied to any type of two phase flow in water cannot be told by now. The kind of pressure drop device used, the arrangement of the test rig or the pressure conditions may have generated one special type of micro-bubbles that leads to the data collapse of Fig. 6. As very few data seems to be at hand in this field, the present paper should be considered as a contribution to two-phase flow vibrations.

6. ACKNOWLEDGEMENTS

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