International working group on fast reactors specia-
list flow induced vibration in LMFBR design. Argonne,
Ill., USA, 20-23 September 1977
CEA-CONF--4155

FLOW INDUCED VIBRATIONS OF SECONDARY PIPING OF L.M.F.B.R.

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- SUMMARY -

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This paper presents a method for evaluating the characteristics of vibrations caused by internal flow in three-dimensional piping systems conveying high density fluids.

The excitation of the circuit is mainly caused by the flow singularities, and it is shown that the problem may be reduced to calculate the response of the circuit to an acoustical pressure discontinuity, localised at each flow singularity. The paper is divided into two main parts :

- First part is devoted to the theoretical formulation of the coupled acoustical-mechanical problem and to its numerical solution by the french computer code TEDEL.

- Second part describes an experimental test of the method. The tested piping system consists of a stainless steel tube circuit comprising four 90° bends, conveying water. Vibrations are excited by a half closed gate valve. Satisfactory results are obtained concerning both the frequencies of resonance of the circuit and the level of the vibrations observed.

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1 - INTRODUCTION -

1-1 - Statement of the problem -

Internal flow induces vibrations of LMFBR secondary piping systems which are mainly caused by the presence of flow singularities (e.g. sudden enlargements, bends, valves ...). Indeed, the unsteady flow field, taking place downstream from such singularities, generates both "local" wall pressure fluctuations and acoustic waves (generally plane waves). The latter are transmitted through the whole circuit. The "source problem" has been studied at Ref. 1 where local pressure fluctuations and associated acoustic sources have been characterized for several current singularities.

The present paper is devoted to the "response problem" of a piping system to such sources. In the first part, a theoretical formulation for the modal characteristics and for transfer functions computation of piping systems conveying fluid, is described. In the second part, the results of an experimental test are compared to the values obtained with the french computer code TEDEL, which uses this formulation.

1-2 - Modal characteristics calculation of piping systems conveying high density fluids -

The effect of internal axial permanent flow on cylindrical structures has already been analysed in the literature (e.g. Ref. 2). Results are usually presented by plotting the resonance frequencies in the complex plane, as functions of the flow velocity (V). The general shape of such plots is sketched in the diagram given below :

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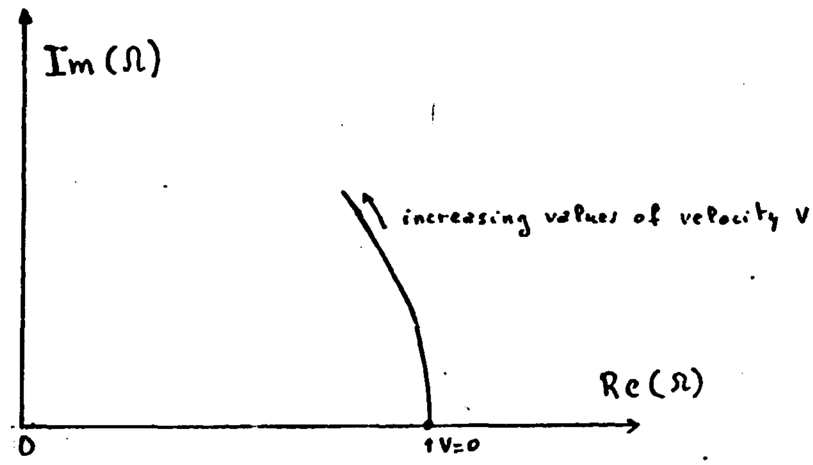

For most nuclear reactor circuits, the range of interest is usually limited to the very initial part of the curve (near V = 0). Therefore, the main additional effect of the fluid flow is to damp the vibrations without changing significantly their frequency. Moreover in actual circuits, the damping rate due to this effect is often less than that caused by other effects in particular imperfect connections between structures.

Consequently it is a good approximation to perform a modal calculation of the structure-steady fluid system and to use an estimate of the damping rate derived from independent considerations (calculated or experimental data). Furthermore, in the case of piping systems the mechanical-acoustical calculation is simplified by using the fact that generally the acoustical-wave lengths of vibrations are very large with respect to the cross dimensions of the tubes.

2 - DESCRIPTION OF THE THEORETICAL FORMULATION -

2-1 - Equations of the problem -

As the structure is concerned, the beam hypothesis (no strain of the tube cross section) may be well applied. Tube displacements are characterized by a displacement vector $\vec{\delta}$ (s, t) and a rotation vector $\vec{\alpha}$ (s, t) of the neutral fiber and normal cross section, which depend upon the curvilinear abscissa : s.

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In the simple case of a straight tube with open extremities and constant cross section, only the transverse component of the displacement δ_t (s) acts on the fluid. Since the acoustical wave lengths are large, compared to the cross dimensions, fluid moves with an uniform velocity $\dot{\delta}_t(s)$ through the whole cross section S(s)*. Hence the fluid effect can be taken into account by adding the mass per unit length of fluid : $\rho_0 S$ (ρ_0 = fluid density), for transverse motions of the tube.

When the circuit presents geometrical singularities (namely : bends, changes in the cross section, closed extremities, junctions between several tubes) wall motions excite longitudinal plane waves (variations of the acoustical mass flow rate are generated and located at the singular points of the circuit). Conversely, a plane wave system, present in the circuit, induces forces acting on the walls of the singularities. This plane wave effect must be added to the transversal effect described above, and one is led to solve a monodimensional acoustical-mechanical problem.

Let R(s) be the local radius of curvature, $\frac{dS}{ds}$ (s) the local variation rate of the cross section, $\vec{l}$ and $\vec{t}$ the local unit vector system associated with the osculatory plane of the neutral fiber. The local coupled equations can be written as :

$$(1)\quad \left| \begin{array}{l} K_m \vec{\delta} + M_m \ddot{\vec{\delta}} + \rho_0 S \ddot{\vec{\delta}}_t - \frac{p}{R} S\vec{t} + p \frac{dS}{ds} \vec{l} = 0 \\ \frac{\partial}{\partial s} \left(S \frac{\partial p}{\partial s} \right) - \frac{S}{c^2} \frac{\partial^2 p}{\partial t^2} - \rho_0 S \frac{\ddot{\vec{\delta}} \cdot \vec{t}}{R} + \rho_0 \frac{dS}{ds} \ddot{\vec{\delta}} \cdot \vec{l} = 0 \end{array} \right.$$

K_m and M_m represent the stiffness and mass operators of the tube in absence of fluid, p is the acoustical pressure averaged over the cross section S(s), c is the sound velocity.

From these equations, simplified expressions may be easily derived for several particular cases, like a junction between tubes, a local variation in the curvature or the cross section of the tube.

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* That is not rigorously verified near the extremities of the tube. However needed corrections remain small, provided that L/D ≫ 1 (L = length of the tube D = diameter of the cross section).

2-2 - Description of the mechanical version of the computer code TEDEL -

a) Short description of the mechanical version of TEDEL :

TEDEL is aimed to compute three dimensional piping systems. The structure is modelized by connecting straight beams or more complex elements (e.g. bends, T junctions, etc. ...). Computations of several types may be performed, in particular computation of the resonance frequencies and mode shapes, and computation of the time response to any excitation.

The code uses the classical finite element method. The elements have two nodes and there are six variables per node (three displacements and three rotations). The form function which is used is a cubic polynom.

b) Modifications introduced for taking into account the fluid effects :

To be introduced into the TEDEL formalism, the Fourier transform of system (1) must be written as :

$$(K - \omega^2 M)\ \Delta = 0 \qquad (2)$$

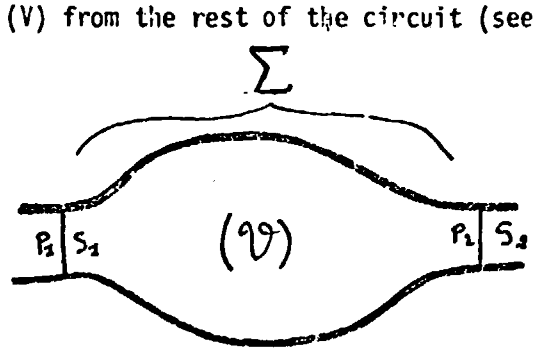
Here K and M are symmetrical stiffness and mass operators (ω = pulsation). For that purpose a supplementary variable : $\pi = \frac{p}{\omega^2}$ is needed. Hence this leads to consider eight variables per node (three displacements, three rotations, π and p). Discretisation of the modified system (1), by the classical finite element method, provides elementary symmetrical stiffness and mass 16 x 16 matrices. These matrices involve a 12 x 12 submatrix and a few additional terms related to the fluid-structure interaction.

In a further step elementary matrices are assembled in a classical way and the boundary conditions are introduced. As the fluid is concerned such conditions may be a closed extremity of the circuit or a node, either for the acoustical pressure or for the acoustical mass flow rate. These last operations provide the final K and M matrices of the whole system. Last step consists in deriving resonance frequencies and mode shapes by using classical algorithms.

c) Calculation of the transfer functions and characterization of the sources :

Starting from the computed modal characteristics, the code allows to obtain any desired transfer function of the system, provided that modal damping coefficients have been introduced into the code as user's data. To perform such a computation the nature of the source to be considered, must be known. Ref. 1 shows that the main sources of vibrations are the flow singularities present in the circuit.

Local and acoustical effects of such singularities have been outlined above. The local pressure fluctuations generate forces on the walls (Σ) of the locally perturbed volume (V), which takes place next downstream from the singularity. For the first resonance modes of the system, the strains over (Σ) are small enough to be neglected. In addition, acoustical wave lengths are also large in respect with the dimensions of the perturbed volume. Therefore, by integrating the equation of the fluid motion, it can be shown that the force field acting on (Σ) is equivalent to the force field acting on the cross sections (Si) which separate (V) from the rest of the circuit (see the diagram below) :



As the force field acting on (Si) depends only upon the acoustical pressure averaged over (Si), the local excitation problem can be reduced to an acoustical plane wave problem. Therefore excitation effects on the whole circuit, caused by a singularity, can be derived from the characteristics of the acoustical source which is associated with the singularity. In the case of low Mach number flows, such as liquid flows encountered in nuclear reactor circuits, their acoustical sources may be characterized by a discontinuity of the acoustical pressure (see Ref. 1).

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TEDEL program allows to calculate the transfer functions $H(\nu, s)$ associated with a unit pressure discontinuity, located anywhere in the circuit . Using such functions, it is also possible to compute the power spectral density $\mathcal{S}_R$ of the response (displacement, stress, acceleration, pressure) at any point of the piping system, to several decorrelated singularities, according to the relation :

$$(3) \quad \mathcal{S}_R(\nu, s) = \sum_{i=1}^{N} \left[H_i(\nu, s) \right]^2 \mathcal{S}_{\Delta p_i}(\nu)$$

$\mathcal{S}_{\Delta p_i}(\nu)$ represents the power spectral density of the pressure discontinuity associated with the singularity indexed by i.
$\mathcal{S}_{\Delta p_i}(\nu)$ can be estimated from the results given at Ref. 1.

3 - DESCRIPTION AND RESULTS OF AN EXPERIMENTAL TEST -

3-1 - Description of the experiment -

The tested piping system consists of an stainless steal tube circuit 2 mm thick comprising four 90° bends, conveying water (diameter of the tubes is D = 18 cm, curvature radius of the bends is R = 27.5 cm, R/D = 1.54). The tubes are joined together by very stiff flanges. The rig is supported by rubber extensible springs (Steal mass of the whole is M_s = 250 kg, fluid mass is M_f = 300 kg). This pipe is inserted in the "GASCOGNE" water loop. Connections are made by using rubber bellows which isolate mechanically the rig from the rest of the circuit. In the same way two cavities with a compressed air level are located up and downstream from the rig, providing an acoustical isolation of it (see Fig. 1).

Vibrations are induced by a half closed gate valve, inserted between the downstream bellows and the downstream cavity. The level of the acoustical source associated with the valve is much higher than that from any other source in the whole circuit, at least in the low frequency range, which is of interest here (0, 10 Hz). Pipe motions are measured by accelerometers, and pressure fluctuations by wall pressure transducers (their location is given in Fig. 1). Pressure measurements are used in particular for evaluating the spectral characteristics of the acoustical source.

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3-2 - Modal analysis of the piping by TEDEL -

The modelisation takes into account the stiffness of the springs and that of the bellows (which have been estimated by static tests) and also the effect of the flanges. The latter introduce a reduction of the flexibility of the bends which is worth to be emphasized. This effect has been calculated separately, by using the three-dimensional shell computer code TRICO.

The fluid boundary conditions are assumed to be fluctuating pressure nodes, located at each cavity. TEDEL provides the resonance frequencies and the mode shapes (displacement and pressure) of the system, in the 0 - 10 Hz frequency range. Mode shapes fall into two categories :

- Parallel to the pipe plan :

$\nu_1 = 3.82$ Hz $\quad \nu_2 = 5.88$ Hz $\quad \nu_3 = 6.94$ Hz $\quad \nu_4 = 9.4$ Hz

Figure 2 shows the shape of the first two modes. In particular, along the straight portions of the circuit a linear evolution of the fluctuating pressure is noted. This is a consequence of the large value of the acoustical wave lengths in respect with the length of the pipe. A discontinuity in the slope of the pressure occurs at each bend, this is equivalent to a discontinuity in the fluctuating mass flow rate.

The first mode corresponds to a swing motion of the bended part of the pipe. For the next modes, an important strain of the straight parts can be noted. For all these low frequency modes, stiffness of support springs and bellows, reveal as important parameters.

- Normal to the pipe plan :

It is expected that such modes are not excited by the source. Indeed, the level of such resonances is always very low in the experimental data.

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3-3 - Acoustical source associated with the valve -

Tests performed with the valve in full opened position, show that the noise coming from the whole loop remains imperfectly softened by the cavities (especially in the low frequency range of interest here). Moreover some pressure transducers are sensitive to local turbulence. Interpretation of the results is much simpler when the valve is half closed because in that case, the background due to disturbing sources and local turbulence becomes much lower than the noise due to the valve. Indeed, the fluid mean velocity is much higher at the valve location than everywhere in the whole circuit.

Various levels of the valve gate have been tested, and for each level several mass flow rates have been studied. The power spectral density of the acoustical pressure discontinuity associated with the valve can be easily obtained from the spectra of the pressure transducers which are located in the straight tube, upstream from the valve : TEDEL shows that for the present experiment the transfer functions of the pressure sensors does not differ significantly from the pure acoustical transfer functions of the whole circuit. In the 0 - 10 Hz interval these functions vary slowly with the frequency. Then the pressure discontinuity spectrum $\mathcal{S}_{\Delta p}(\nu)$ is deduced simply by dividing the experimental spectrum by the squared modulus of the corresponding transfer function. This is done with the adimensional quantities :

Strouhal number $$s = \frac{\nu d}{V}$$

Adimensional power spectrum $$\mathcal{F}_{\Delta p}(s) = \frac{\mathcal{S}_{\Delta p(\nu)}}{(1/2\,\rho_0 V^2)^2}\,\frac{V}{D}$$

(D = tube diameter, V = mean velocity of the flow at the valve). It is worth to emphasize that $\mathcal{F}_{\Delta p}(s)$ doesn't depend upon the mass flow rate and only is a function of the valve gate level. Consequently, $\mathcal{F}_{\Delta p}(s)$ turns out to be a specific quantity which characterises the singularity. The shape of $\mathcal{F}_{\Delta p}(s)$ is given in the diagram below.

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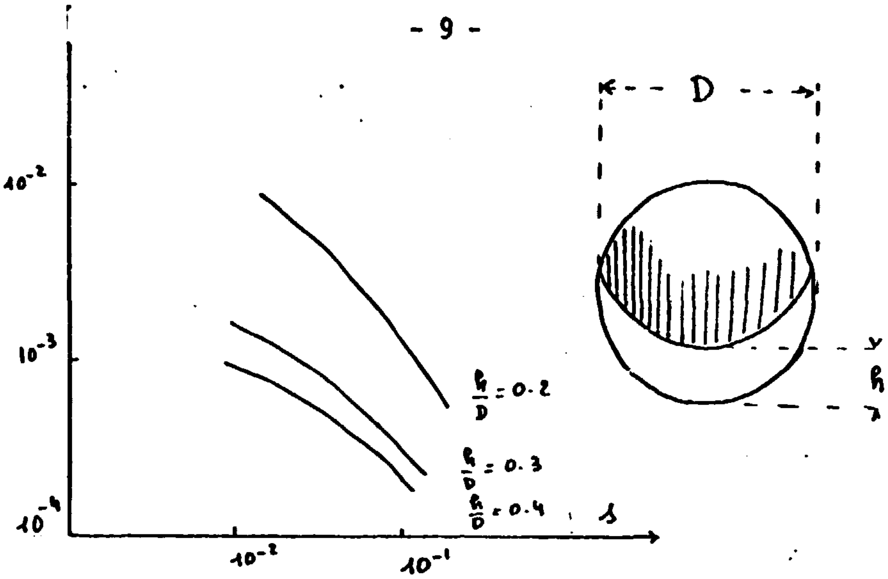

These results are in general agreement with other measurements, performed in various experimental conditions (especially in air or water test loops) and for various singularities (see Ref. 1).

3-4 - Computation of the response of the circuit to the source, comparison with experiment -

Experimental damping coefficients ε are about 10^{-2}. They are introduced in TEDEL for calculating the transfer functions corresponding to a unit pressure discontinuity, which is located at the valve position. Then the spectrum of the source (see § 3) is used to derive the P.S.D. of vibrations, in the whole circuit. Fig. 3 and 4 summarize the comparison between the measured and computed spectra for three accelerometers and two values of the valve gate opening.

4 - CONCLUSION -

There is a good agreement between computation and experiment, in spite of some small differencies concerning the frequency of the resonances. This is probably caused by an imperfect knowledge concerning the actual stiffness of flanged bends and pressurized bellows. Nevertheless, it appears that a quantitative estimation of piping vibrations can be obtained with a sufficient accuracy, by using the TEDEL program and the source data presented in Ref. 1. However complementary studies are still needed to improve knowledge about damping effects in piping systems conveying fluid.

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- BIBLIOGRAPHY -

1 GIBERT R.J. - VILLARD B. - "PRESSURE FLUCTUATIONS INDUCED BY FLUID FLOW IN SINGULAR POINTS OF INDUSTRIAL CIRCUITS"
(4^{th} SMIRT CONFERENCE B 3/5 - 1977).

2 PAIDOUSSIS M.P. - "VIBRATIONS OF TUBES CONTAINING FLOWING FLUID"
(CONF. ERMENONVILLE sur l'Aéro-hydroélasticité - Eyrolles Ed. 1972).

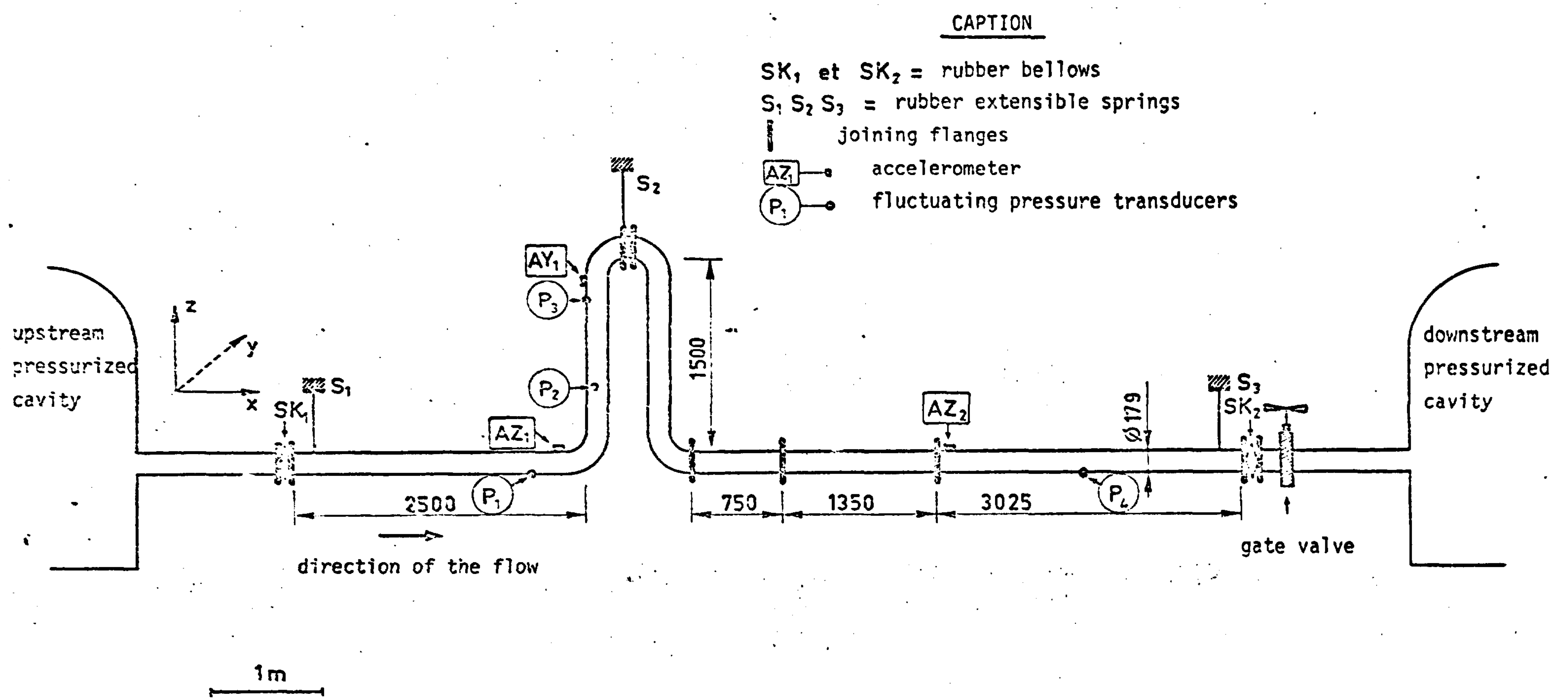

FIG. 1 CONFIGURATION OF THE CIRCUIT

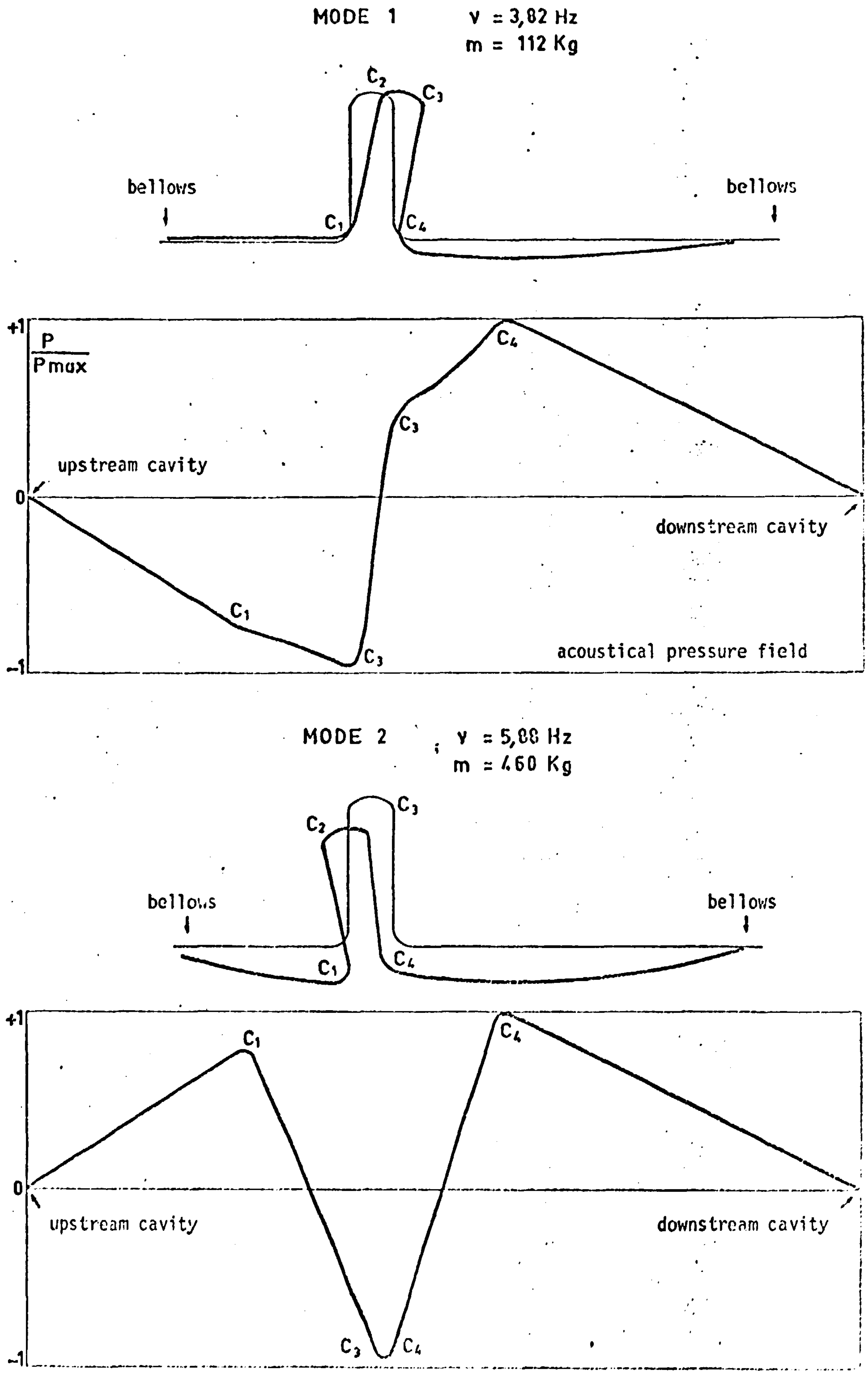

FIG. 2 COMPUTED MODAL SHAPES

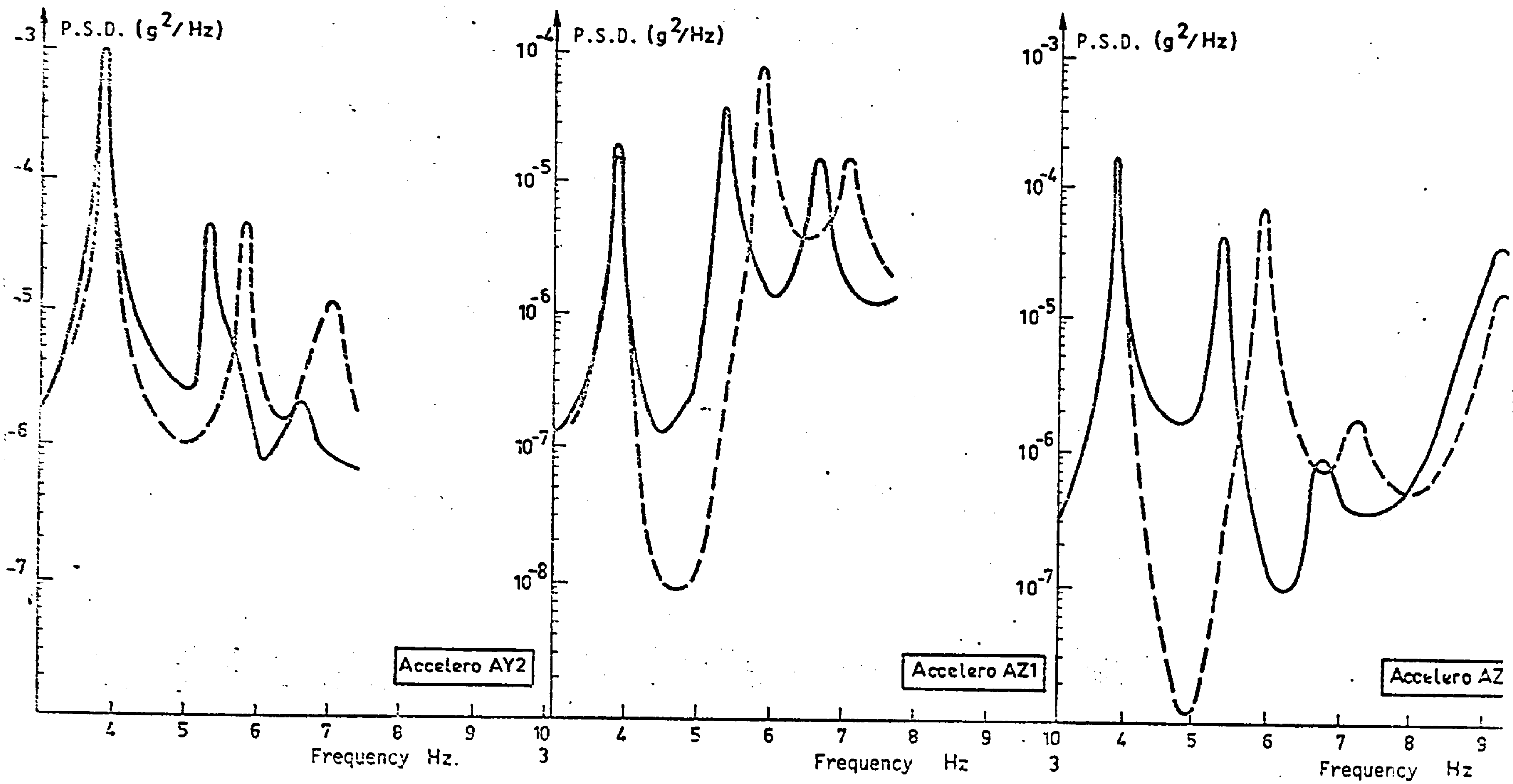

FIG. 3 COMPARISON BETWEEN EXPERIMENTAL AND COMPUTED RESULTS (VALVE'S GATE APERTURE H/D = 0.4)

—— Experimental power spectral density

– – – Computed power spectral density

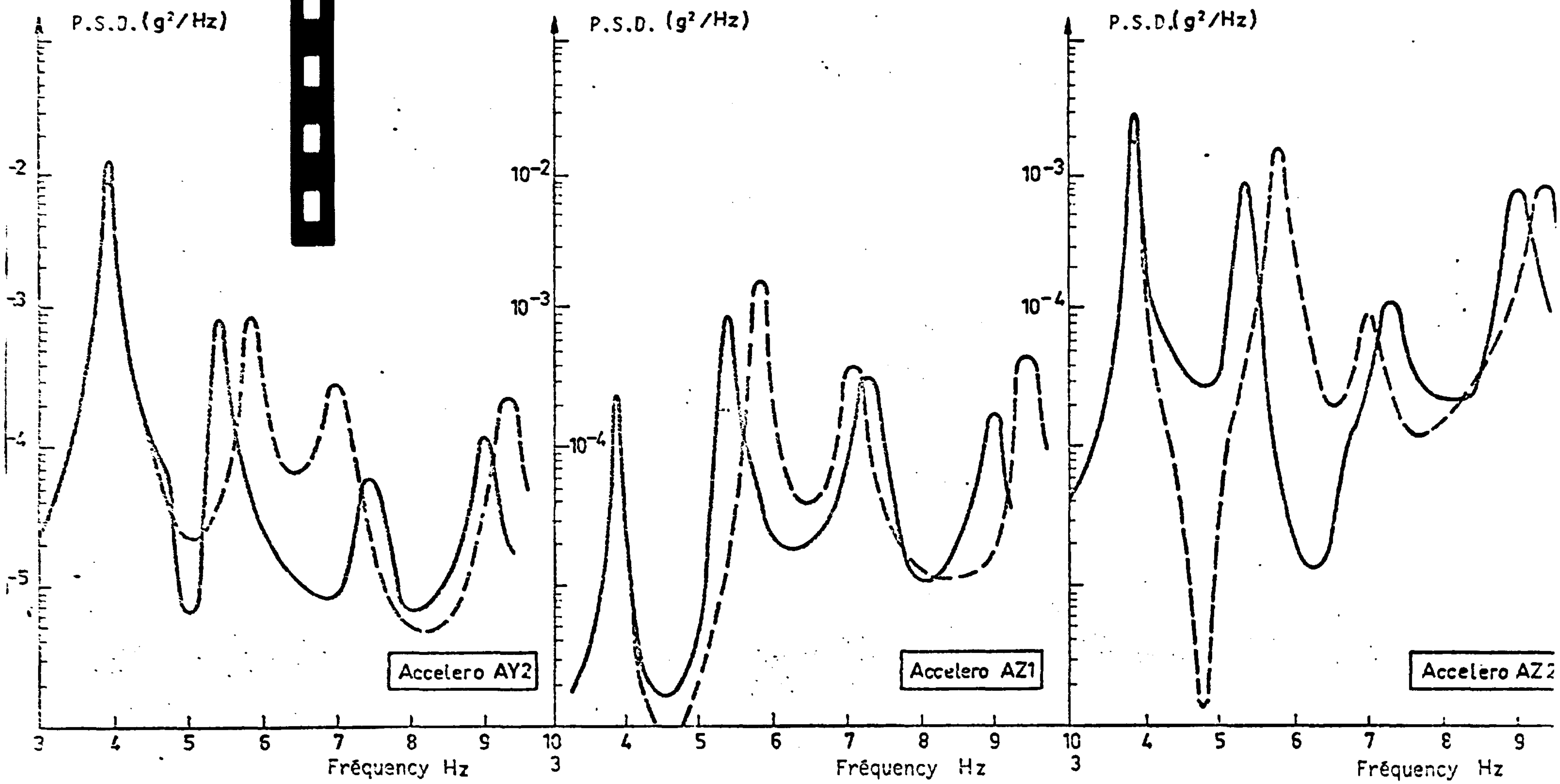

FIG.4 COMPARISON BETWEEN EXPERIMENTAL AND COMPUTED RESULTS (VALVE'S GATE APERTURE H/D = 0.21)
Experimental power spectral density
Computed power spectral density