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NON-LINEAR VIBRATIONS INDUCED BY FLUIDELASTIC FORCES IN TUBE BUNDLES

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ABSTRACT

We present in this paper computations of the response of a loosely supported tube to fluidelastic forces. Several models of forces are considered, including negative damping, coupling forces and Price and Paidoussis' model. Unidirectionnal and bidirectionnal motions are studied, special attention being paid to the evolution of dynamic parameters influencing wear and to the changes in the dynamic regimes. The influence of the coefficient of friction is also analysed. A corrective methodology is proposed for the use of the negative damping model in non-linear computations.

NOMENCLATURE

a	reduced pitch,
$A(\ )$	damping operator,
A_n	modal damping coefficient,
C_L, C_D	tube lift and drag coeffcients,
D	tube diameter,
e	excentricity of tube in the support,
f, f_n	mode frequency,
f'_n	mode instantaneous frequency,
f^n_I, f^n_F	modal impact and fluidelastic forces,
g	clearance of support,
H	hitting motion,
k	coupling forces parameter,
K	instability threshold parameter,

Case 4 : Unidirectional vibrations with five loose supports. Influence of the models of fluidelastic forces

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k	coupling forces parameter,
K	instability threshold parameter,
$K(\ )$	stiffness operator,
K_c	impact stiffness,
K_n	modal stiffness,
L_n	modal length,
m_e	tube equivalent lineic mass,
$M(\ )$	mass operator,
M_n	modal mass,
q_n, $\dot{q}_n$, $\ddot{q}_n$	modal displacement, velocity and acceleration,
R	rubbing motion,
t	time,
T	transverse motion,
V	flow gap velocity,
V_c, V^n_c	critical gap velocity,
$\bar{X}$, $\dot{\bar{X}}$, $\ddot{\bar{X}}$	tube displacement, velocity and acceleration,
$\bar{X}_T$, $\dot{\bar{X}}_T$	tube tangentiel displacement and velocity,
α_{ij}, α'_{ij}, α''_{ij}	coupling coefficients,
Δt	time step,
ξ, ξ_n	modal damping,
ξ^α_n ξ^*_n	modified modal damping,
μ	coefficient of friction,
ρ	flow volumetric mass,
φ_n	modal shape,
ω_n	modal circular frequency,
ω'_n	modal instantaneous circular frequency,
τ	time lag.

thereby, their consequences. This effect is all the more important in that the post-instability regime is different from the free harmonic vibration regime. It is more particularly the case when several loose suports are considered and for high velocity ratios (case 4).

When the model of Price and Païdoussis was used (case 1), it was observed

1. INTRODUCTION

Flow-induced vibrations of heat exchangers have often been reported to be the cause of numerous tube failures (Païdoussis, 1979). Due to the industrial and economic consequences of such failures, a considerable amont of research work has been done in the past decades on these phenomena. Different kind of flow-induced forces have been identified and shown to potentially lead to intense vibrations.

Among them, random forces which are due to turbulence and two-phase flow have been extensively measured and modeled (Axisa et al., 1990, Chen and Jendrejczyk, 1987, Granger, 1990, and others). The main other type of force, which is the object of the present paper is fluidelastic force. In can be roughly defined as the flow related forces that depend on the tube motion.

Today's knowledge of fluidelastic forces seems at the same time highly advanced if one is concerned with experimental analysis of instability theseshold, and still scarce if one is concerned with the actual physical phenomenon leading to such forces. In fact, the onset of instability in a tube bundle can now be estimated using design guidelines such as those of Pettigrew and Taylor (1991). Unfortunately, in real heat exchangers, tubes are loosely supported and according to the above guidelines they may vibrate in a post-instability situation. In that case, the destabilizing fluidelastic forces are balanced by stabilizing effects such as impacts and friction at the supports. The prediction of the wear associated with such vibrations is therefore of major practical importance, and there is indeed a need for fluidelastic forces models, not just for instability threshold guidelines.

Computations of the post-instability regimes of vibrations, using different models of fluidelastic forces, have already been reported by Axisa et al. (1988), Antunes et al. (1990), Fricker (1988), Langford and Connors (1991), Paidoussis et al. (1991).

The specific concern of the present paper is to analyse the ability of several models of fluidelastic forces to be used in non-linear computations of post-instability tube motions, and explore the sensitivity of dynamic regimes and quantities affecting wear to flow velocity and support characteristics. For the sake of clarity, random forces and permanent dray force associated with the flow will not be considered in the present study, though they might be expected to contribute significantly to the wear process.

This study will be presented in four parts. The first one deals with a brief review of some current models of fluidelastic forces in tube bundles. In the second part, the methodology for the computation of non-linear tube dynamics is presented. The third part presents several results of computations on

$$\left(\frac{V}{V_c}\right)' = \sqrt{\frac{f_n'}{f_n}}\left(\frac{V}{V_c}\right) \quad (16)$$

The main differences between this procedure and that proposed b Fricker post-instability behaviour of tubes with loose supports.

Finally, in a fourth part, a discussion is made concerning the pratical use of fluidelastic forces models and recommandation for industrial cases are proposed.

2. MODELS OF FLUIDELASTIC FORCES IN TUBE BUNDLES

In the past decades, a considerable amont of experimental data on the onset of fluidelastic instability was produced (see reviews in Chen (1977), Blevins (1990), Pettigrew and Taylor (1991)).

In the same time, several models of the fluidelastic phenomenon were proposed. These, in most cases, were only applied to the prediction of critical flow velocities. The relationship between the flow and tube parameters at instability is usually expressed as

$$\frac{V_c}{fD} = \mathcal{F}\left(\frac{2\Pi\,\xi\,m_e}{\rho\,D^2}\right) \qquad (1)$$

where V_c is the critical gap velocity, ρ is the flow volumetric mass, D, m_e, ξ and f being the tube diameter, equivalent mass per unit length, damping and mode frequency, $\mathcal{F}(\ )$ being a function which depends on the model considered.

For design purposes, a guideline was recently proposed by Pettigrew and Taylor (1991), using most of the existing data points, as

$$\frac{V_c}{fD} = 3\left(\frac{2\Pi\,\xi\,m_e}{\rho\,D^2}\right)^{.5} \qquad (2)$$

Such a result, though its pratical consequence for designers is evident, does not give a model of fluidelastic forces or indicates the kind of mechanism that controls the generation of fluidelastic forces. When one is concerned with post-instability behaviour of loosely-supported tubes, a critical velocity formula, such as (2), is not sufficient. A consistent fluidelastic forces model would need to have the following characteristics :

a) The fluidelastic instability should be analysed in termes of forces acting on the tube, not in terms of critical velocity only.

b) These forces should explicitly depend on the motion of the tube, which is not restricted to be harmonic.

c) In the case of harmonic motion these forces should lead to critical values of velocities consistent with the experimental data.

b) The derivation of the relationship between this negative damping (or energy input rate), the flow parameters and the tube motion. This is a problem of fluidelasticity.

From this point of view, several of the most commonly used models are now studied. For the sake of clarity, the analytical expression of the forces are presented in the case of uniform cross flow through a bundle made of identical straight tubes with identical modes in the two transverse directions. Moreover, instability is considered only in one transverse direction. In fact, streamwise fluidelastic forces may modify the energy balance of the system, though most probably less than in the transverse direction.

2.1. Connors-Blevins Model

A quasi-static mechanism is assumed to couple the motions of neighbouring tubes. The lineic forces acting on two fluidelastically coupled tubes are given by (Blevins, 1990)

$$\begin{cases} F_1^x = - k \rho V^2 y_2 \\ F_2^y = + k \rho V^2 x_1 \end{cases} \qquad (3)$$

where the subscripts denote the tube number, x and y are transverse directions and k is a parameter of the model.

In the case of free vibrations at the natural frequency of a given mode, the critical flow velocity may be derived from equation (3)

$$\frac{V_c^n}{f_n D} = 2 \sqrt{\frac{\pi}{k}} \left(\frac{2\pi \, m_n \, \xi_n}{L_n \, \rho \, D^2} \right)^{.5} \qquad (4)$$

where $L_n = \int_0^L \varphi_n^2 \, dx$ and f_n, m_n, ξ_n are the modal frequency, mass and damping. Starting from equation (3), a negative damping representation of the fluidelastic forces may be obtained, if the motion of each mode is assumed to be harmonic, at frequency f_n :

$$\xi_n^* = \xi_n \left[1 - \left(\frac{V}{V_c^n} \right)^2 \right] \qquad (5)$$

This model has been widly used in computations (Axisa et al., 1988, Fricker, 1988) and in mechanical simulations of fluidelastic forces by a velocity feedback loop by Antunes et al. (1990).

Its use might nevertheless be questionnable, as it expresses forces which

negative damping or rate of energy input of the unstable mode. Experimental and theoretical work is needed to have a better knowledge of this relationship.

are scaled on the harmonic modal free vibration, not on the actual dynamic motion. Actually, it has often been noted that the motion frequency might be significantly different from the natural frequency, as soon as impact non-linearities occur. In fact, the motion might not even be harmonic at all (De Langre et al., 1990, Païdoussis et al., 1991).

In order to take into account the frequency shift in non-linear post-instability motion, equation (5) was modified by Fricker (1988). The practical use of such corrective methods will be discussed in the last section of this paper.

There are two expression, of fluidelastic forces associated with Connors-Blevins modal, which may therefore be used in post-instability analysis :

a) the negative damping model, equation (5), where the fluidelastic forces are dependent on the motion of the tube itself.
b) the coupling forces model, equation (3), where these forces depend on the motion of a neighbouring tube.

2.2. <u>Chen's Model</u>

A more general form of coupling forces between tubes was proposed by Chen (1977). It includes inertial, damping and stiffness terms. The lineic forces on tube i due motion of the surrounding j tubes is expressed as follows

$$F_i = \sum_j \left[- \rho D^2 \alpha_{ij} \ddot{x}_j - \rho DV \alpha'_{ij} \dot{x}_j + \rho V^2 \alpha''_{ij} x_j \right] \qquad (6)$$

where α_{ij}, α'_{ij}, α''_{ij} are coupling coefficients. Most of these coefficients may in fact only be derived from specific experiments, such as those of Taneka et al. (1982).

Even when these coefficients are known, which is seldown the case, the pratical use of equation (6) for non-linear computations is rather cumbersome.

2.3. <u>Price and Païdoussis' Model</u>

Fluidelastic forces acting on a tube are assumed to depend on the tube motion itself, through the lift and drag coefficients (Price and Païdoussis, 1984). In the linear (i.e. small displacement) form of the model, the instantaneous lineic forces are given by

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$$F^{*}(t) = \frac{1}{2}\rho\frac{V^2}{a^2}D\left[\frac{\partial C_L}{\partial x}x(t-\tau) - C_D\frac{\dot{x}}{V}\right] \tag{7}$$

where $a = (P - D)/P$, C_L and C_D are lift and drag coefficients and τ is a time lag which is of the order of D/V. A more general expression of theses forces, which is suitable for large amplitude motion was used by Païdoussis et al. (1981).

A critical velocity criteria may be easily derived from equation (7) :

$$\left(\frac{V_c^n}{\omega_n D}\right)^2 \sin\left(\frac{\omega_n D}{V_c^n}\right) = -\frac{4\, m_n\, \xi_n}{\frac{\partial C_L}{\partial x} L_n\, \rho\, D^2} \tag{8}$$

Such expression was found to agree reasonnably well with experimental data (Price and Païdoussis, 1984).

Moreover, its is quite noticable that :

a) The lift and drag coefficients may be derived from quasi-static fluid forces measurement.
b) The fluidelastic force, equation (7), is expressed in terms of the tube motion, which is not restricted to be harmonic.
c) Only the motion of one tube is to be considered.

Therefore, this model is quite suitable for post-instability non-linear computations. In Païdoussis et al. (1991), such computations were presented using one mode in each direction only, which is quite restrictive, in terms of non-linear regimes. Nevertheless, interesting bifurcations, eventually leading to chaos were observed.

2.4. Lever and weaver's Model

By the use of a simplified analysis of the fluctuating flow in the tube, a fluidelastic instability model was proposed by Lever and Weaver (1982). In the case of an harmonic motion of the tube, a stability criterion was derived, which was found to be consistent with experimental data (Lever and Weaver, 1986). Moreover, the assumed flow pattern was in fact observed in practice (Weaver and Abd-rabbo, 1984, Andjelic et al., 1990).

In this model, the fluidelastic forces have so far been only derived in the case of harmonic motions, and are not suitable for non-linear computations, except in a simplified negative damping form.

As the basic hypothesis of the model are not associated with harmonic motion of the tube, it would be quite interesting to derive a more general formulation of the fluidelastic forces. The use of this model is therefore beyond the scope of the present paper.

3. METHOD FOR THE COMPUTATION OF NON-LINEAR TUBE DYNAMICS

3.1. Dynamic equation on Modal Basis

The methodology used in the present paper for the computation of vibrations of a tube with impact non-linearities, is identical to that described in Axisa et al. (1988), Antunes et Al. (1990 b), De Langre et al. (1991).

The tube to support control forces are computed with the use of an elastic impact force and a frictional force, modelled by Coulomb's equations. The time-domain dynamic equations are projected on the modal basis of the tube

$$M_n \ddot{q}_n + A_n \dot{q}_n + k_n q_n = f_I^n + f_F^n \qquad (9)$$

where M_n, A_n and K_n are modal parameters, q_n is the modal displacement, f_I^n and f_F^n being the modal projections of the impact and fluidelastic forces respectively. The set of modal dynamic equations is integrated with the use of an explicit algorithm. The tube motion is then analysed in terms of mean values of the quantities which influence wear. These are assumed to be :

- the impact force $F_N(t)$,
- the wear work rate $\dot{W} = F_N |\dot{X}_T|$,

where $\dot{X}_T$ is the sliding velocity at impact.

3.2. Fluidelastic forces on Modal basis

In the first part of this paper, it was shown that several models are suitable for non-linear post-instability computations.

If the negative damping modal is considered, equation (5) may be expressed in terms of the modal fluidelastic force to be used in equation (9)

Tanaka, H., Takahara, S., Ohta, K., 1982. "Flow induced vibrations of tube away with various pitch to diameter ratios", ASME pressure vessel and i in

$$f_F^n\left(\dot{q}_n\right) = k\,\rho\,V^2\,\frac{L_n}{2\Pi\,f_n}\,\dot{q}_n \qquad (10)$$

For stable modes, i.e. for those where $V_c^n > V$, the corresponding fluidelastic forces may be negleted, as was shown by Beaufils (1990).

For the coupling forces model, equation (3), the motion of two tubes need to be computed. In that case, the set of modal equations of the two tubes is

$$\begin{cases} M_n\left(\ddot{q}_n\right)_1 + A_n\left(\dot{q}_n\right)_1 + k_n\left(q_n\right)_1 = \left(f_I^n\right)_1 + \left(f_F^n\right)_1 \\ M_n\left(\ddot{q}_n\right)_2 + A_n\left(\dot{q}_n\right)_2 + k_n\left(q_n\right)_2 = \left(f_I^n\right)_2 + \left(f_F^n\right)_2 \end{cases} \qquad (11)$$

where the fluidelastic coupling forces are

$$\begin{cases} \left(f_F^n\right)_1 = -\,k\,\rho\,V^2\,L_n\left(q_n\right)_2 \\ \left(f_F^n\right)_2 = k\,\rho\,V^2\,L_n\left(q_n\right)_1 \end{cases} \qquad (12)$$

In fact, in the case of a non uniform flow, or non identical tubes, there might be non-zero coupling coefficients between all the modes of the two tubes. They may be easily derived from the projection of the lineic forces, equation (3), on the modal basis. On the other hand, computations done by the authors showed that the coupling forces between the most unstable modes can be expected to be largely predominant, and that it is not necessary to take into account all couplings. In the computations presented in this paper, the coupling forces, equation (12), were only considered on the unstable modes.

When Price and Païdoussis model, is used, it is only necessary to consider one tube. The modal projection of fluidelastic forces may easily be derived from equation (7) :

$$f_F^n(t) = \frac{1}{2}\,\rho\,\frac{V^2}{a^2}\,D\,L_n\left[\frac{\partial C_n}{\partial x}\,q_n(t-\tau) - C_D\,\dot{q}_n\,(t)\right] \qquad (13)$$

For the three models considered here, it is seen that the modal basis representation is quite convenient for the expression of fluidelastic forces. In fact, the use of equations (10), (12) and (13) is much easier then the use of nodal forces in a finite element model. Moreover, the restriction of those fluidelastic forces on the unstable modes only is quite straightforward in the modal basis representation of the motion.

3.3. <u>Computation of the post-instability regimes</u>

With the above models, the post-instability regime of vibration may be obtained with the use of the following procedure :

a) A non-zero initial condition, such as an impact on the tube, is used to initiate the growth of motion due to instability.
b) The system is then let free to vibrate, until a steady state regime is achieved. In such a regime, the energy input by fluidelastic forces is balanced by the energy output though damping and friction.
c) The average characteristics of the regime may then be computed, using adequate sampling (De Langre et al. 1991).

As it is known that such system may have several regimes of vibrations (Axisa et al. 1988, Antunes et al. 1990 a, Fricker 1991, Païdoussis 1991), it is necessary, for each value of the steady flow velocity, to do several computations with different initial conditions.

4. APPLICATION TO LOOSELY SUPPORTED TUBES

The models and procedures described in the preceding sections are now applied to the computation of the non-linear vibration of a tube in post-instability conditions. Particular attention is paid on the comparison between fluidelastic forces models, as well as on characteristics of the non-linear regimes. The parameters to be varied are velocity ratio (V/V_c), shape and number of supports, and impact parameters. The straight tubes considered for all cases is the same as in Axisa et al. (1988), De Langre et al. (1991). It has length L = 2.26 m, diameter D = 22.2 mm, thickness e = 1.27 mm, Young's modulus $E = 2.10^{11}$ Pa and equivalent mass per unit length me = .885 kg/m. As impact stiffness is taken as 10^6 N m^{-1} or 4.10^6 N m^{-1}, only the first 7 flexural modes are required in each direction. A common value of damping in still fluid ξ =.01 was assumed for all modes. In fact, here, fluid damping, is not considered as a part of the fluidelastic forces but as a part of the total model damping. The time history is computed with a time step of 4.10^{-5} s, using more than 250 000 time steps in each case.

Case 1 : Unidirectional vibration with one loose support. Influence of the model of the fluidelastic forces

The tube is assumed to have a loose support at midspan with a ± 5 mm clearance and an impact stiffness of $K_c = 10^6$ N.m^{-1}. This support acts in the same direction as the fluidelastic forces, so that only a unidirectional motion needs to be considered. As the wear work rate $\dot{W}$ is nul in such a case, the computed quantity to be considered is the impact force.

For the use of the negative damping model, only one tube is analysed. For the coupling forces model, a symetric system of two tubes is used, where only one direction is analysed for each tube and the support acts similarly on both.

For Price and Païdoussis model, a single tube is considered, and the values of dimension less drag and lift coefficients are $C_D = 2.3$, $\frac{\partial C_L}{\partial x} \times D = -73$ (Price et al. 1984).

The flow velocity V is varied and is scaled with the critical velocity of the first mode, as obtained from each model. This reference velocity is identical for the negative damping and coupling forces model. The results of the computations for the three models are summarized in figure 1 (average impact force versus velocity ratio) and figure 2 (phase diagrams at the impact point).

The response of the tube, considering first the negative damping model displays the following characteristics :

a) The average impact force at the supports is found to increase irregularly.
b) A step like variation of the impact force, of one order of magnitude, is observed for $V/V_c = 3$. This step phenomenon was previously observed in experimental and numerical studies (Axisa et al. 1988, Antunes et al. 1990, Fricker 1991).
c) The above step variation in impact force occurs simultenaously with a change of shape of the phase diagram at the impact location. The motion is seen to change from a periodic regime to a new periodic regime through a non-periodic one.
d) At a velocity ratio greater than about 5., no steady state regime can be found for the system. The computed impact force increases steadily with time. This mechanism was observed experimentally by Antunes et al. (1990). It is not a numerical divergence of the time stepping algorithm but a global instability of the mechanical system, where the stable modes dissipation is unable to balance the energy input of the unstable mode.

The results obtained with the Price Paidoussis model may be analysed in a similar may :

a) A step like variation of impact force is observed at a velocity ratio of about 4 (figure 1).
b) This variation coincides with the change in shape of the phase diagram (figure 2).
c) A mechanical instability is observed at a velocity ratio of about 6.5.

When the coupling forces modal is considered, it is found that :

a) The motion of the two coupled tubes are almost identical in shape and in terms of mean values, though they differ when the detailed time history is considered.
b) The mean impact force of one of the tube (figure 1) displays a step variation at a velocity ratio of about 5, simultaneously with the change on the phase diagramm.
c) The mechanical instability is also observed, but at a much greater velocity ratio of about 10.

The preceding results clearly point out the fact that there is a great similarity in the response of the system to the three kind of fluidelastic forces considered. The main difference lies in the scale of velocity ratios where changes in dynamic regimes occur. The step like impact force variation is observed at respective velocity ratios of 3, 4 and 5, while the global instability is found at ratios of 5, 6.5 and 10 respectively. This strongly suggests that the results obtained with the three models may in fact be derived from those of the negative damping model, through an adequate change in the scale of velocity ratio. This specific point will be discussed in the last section of this paper in view of the results of all the test cases.

It should also be noted that the negative damping modal, which is the only one that does not take into account explicitely the motion dependent nature of the fluidelastic forces, leads to much higher impact force levels at a given velocity ratio.

<u>Case 2 : Bidirectional vibration with a circular loose support. Influence ofthe model of fluidelastic forces</u>

The support at midspan of the tube is now considered to have the shape of a circular hole. In order to have a bidirectional motion of the tube, this support is assumed to have an excentricity equal to half the clearance. The fluidelastic forces are only taken into account in the direction perpendicular to the excentricity (figure 3a). The impact stiffness is now $K_c = 4.10^6$ $N.m^{-1}$ and the friction coefficient is $\mu = .2$. For the use of the coupling forces model, two tubes are considered with circular supports, the excentricity of which are perpendicular to the fluidelastic coupling forces, in order to preserve the symetry of the system (figure 3b). The results are analysed in terms of wear work rate, which is the variable of pratical importance, and in terms of the shape of the trajectory of the tube inside the circular support.

When the negative damping model is used, it can be seen (figure 4), that the trajectory of the tube in its support significantly changes as the velocity ratio is increased. The regular elliptic motion (henceforth called "Rubbing") which exists at low velocities progressively fills the clearance circle, but is eventually replaced by a quasi-unidirectionnal motion (henceforth called "Hitting"). As in the preceding that case, the system is seen to have dynamic regimes which are quite sensitive to the velocity ratio.

In terms of wear work rate it is seen in figure 5 that the regular increase with the velocity ratio is only slightly perturbated by the change in the regime of motion.

Considering now the coupling forces model, a quite similar pattern of evolution can be observed in figure 5 : the rubbing motion is replaced by a hitting motion, this event occuring here at a higher velocity ratio ($V/V_c = 4$ instead of 3).

On the other hand, the evolution of wear work rate displays the same features than with the preceding model, but with a change in the scale of the velocity ratio. This is quite consistent with the conclusions of the case 1 computations. It is worth noting that the change from rubbing (R) to hitting (H) regime is associated, in the coupling forces model, with a plateau in the rate of increase of the wear work rate (figure 5).

It can be understood that the hitting motion generates less tangential velocity at the impact, so that the wear work rate can be expected to be quite different.

<u>Case 3</u> : <u>Bidirectional vibration with a circular loose support. Influence of the coefficient of friction</u>

Considering the dynamic regimes that were observed in the preceding section, it can be expected that the friction coefficient, μ, may have some influence on the response of the system. In a parameter study, this coefficient is varied from $\mu = .1$ to $.4$ all others parameters being kept identical to those of case 2. As the results of the fluidelastic forces models were found to be

qualititavely similar, only the coupling forces model is used here.

In figure 6a, it can be seen that the friction coefficient has no significant influence on the mean impact force. If the wear work rate is considered, figure 6b, considerable differences may be found, of more than one order of magnitude, when μ is varied. In fact, when the velocity ratio is increased, the mean work rate, using $\mu = .1$, steadily increases up to hundreds of milliwatts. For higher values of μ, the wear work rate almost stabilizes in a range of tens of milliwatts.

To understand such a behaviour, it is necessary to analyse the trajectory of the tube in its support, figure 7. Up to a velocity ratio of about 3, the motion of the tube is almost identical for $\mu = .1$ and $\mu = .2$.

For higher velocity ratios, the trajectories are quite different :

a) For $\mu = .1$, the rubbing motion progressivly fills up the clearance circle. The tube is almost always in contact with the support, and the rubbing induces a considerable amont of wear work rate.

b) For $\mu = .2$, the rubbing motion is transformed to a hitting one when the velocity ratio is increased. For higher values of μ, a similar change is observed, at slighty lower values of V/V_c. Clearly, the hitting motion induces much less mean work rate than the rubbing one.

The two kind of motions observed, rubbing and hitting, seem to be more or less stable, depending on the parameters of the system. In fact, in some of the computations, the two regimes were sometimes observed in a given time-history. In those cases, one of them was found to last a few seconds or less, then to bifurcate to the other one where it stayed. In the results, only the characteristics of the latter were reported.

The mechanism controlling the stability of these regimes of vibrations is not yet clearly identified. It can nevertheless be assumed that a higher coefficient of friction tends to reduce the rebound angle after impact, because of the reduction in tangential velocity, and thereby to destabilize the rubbing motion and stabilize the hitting one.

Such a trend was also observed by the authors on other systems, such as forced ones (Beaufils, 1990). It was also reported recently by Fisher et al. (1991), for a tube with several loose supports, though the effect was less noticeable. It seems to be a general feature for these systems that higher friction coefficients induce less wear work rate.

Case 4 : Unidirectional vibrations with five loose supports. Influence of the models of fluidelastic forces

To test now the influence of the number of loose supports, the simple case of unidirectional motion, as in case 1, is assumed. Five equally spaced supports, identical to that of case 1, are considered, all other parameters being unchanged.

Computations are made with the negative damping model and the coupling forces model. Figure 8 displays the evolution of the mean impact force at midspan versus the velocity ratio.

The following points may be outlined :

a) The evolutions obtained with the two models are rather similar in shape.
b) No step-like variation of the impact force could be observed in the results, though the phase diagrams depict rather complicated evolutions.
c) There is a considerable difference in terms of the level of impact force obtained with the two models, specially for high velocity ratio. In fact, the mechanical instability, as in case 1, is observed at $V/V_c = 5.3$ for the negative damping model, while it was not found, with the coupling forces model, even at $V/V_c = 25$.

5. DISCUSSION

5.1. The motion dependent nature of fluidelastic forces

In the preceding computations of post-instability vibrations, it was observed that the two models of fluidelastic forces associated with Connor's equation led to quite different results, in terms of the characteristics of the vibratory regimes. The only difference between these two models lies in their dependance on the motion of the system.

With negative damping the force is scaled on the free vibration of the mode considered, equation (6).

With quasi-static coupling between two tubes, no reference is made, in the fluidelastic forces, to the frequency of the free vibration. Therefore as stated before and as can be observed in all the cases presented in this paper, the negative damping model clearly overestimates these fluidelastic forces, and thereby, their consequences. This effect is all the more important in that the post-instability regime is different from the free harmonic vibration regime. It is more particularly the case when several loose suports are considered and for high velocity ratios (case 4).

When the model of Price and Païdoussis was used (case 1), it was observed that, because of its motion dependent nature, it lead to results quite consistent with those of the coupling forces model. Therefore, it seems that, notwithstanding the physical justification of the models, it is of importance that, for there use in post-instability computations, the fluidelastic forces depend explicitly on the real motion of the system.

5.2. A corrective procedure for the negative damping model

As stated above, it is over-conservative to use a negative modal damping as defined by equation (5).

In order to take into account the characteristics of the real motion in this method which is quite simple to use, Fricker (1991) has introduced a modified damping

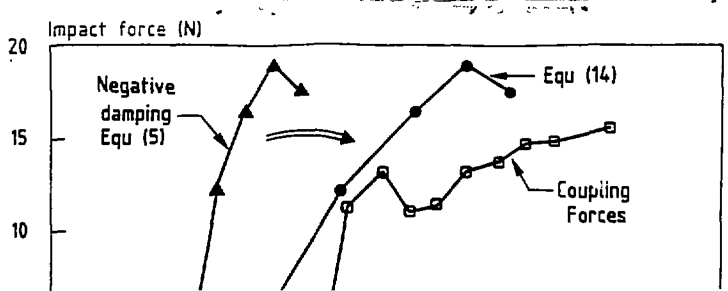
$$\xi_n^a = \xi_n \left(1 - \frac{f_n}{f_n'} \left(\frac{V}{V_c^n}\right)^2\right) \qquad (14)$$

where f_n' is the apparent frequency of motion of the mode considered. In Fricker (1988) and (1991), equation (14) is applied by recalculating the damping during the time-history, estimating the apparent frequency by

$$2\pi\, f_n' = \frac{(\dot{q}_n)_{RMS}}{(q_n)_{RMS}} \qquad (15)$$

Antunes (1991) proposed a procedure which is of a more practical use :

a) The response of the system is computed, using the negative damping of equation (5), for given values of velocity ratios.
b) The modified frequency f_n' is estimated, if possible, by analysing the peak in the PSD of the first mode displacement.
c) The results obtained from step (a), such as wear work rate, are associated to a new set of velocity ratios, defined by



brief review of some current models of fluidelastic forces in tube bundles. In the second part, the methodology for the computation of non-linear tube dynamics is presented. The third part presents several results of computations on

$$\left(\frac{V}{V_c}\right)' = \sqrt{\frac{f_n'}{f_n}}\left(\frac{V}{V_c}\right) \qquad (16)$$

The main differences between this procedure and that proposed by Fricker are that :

- no rescaling of damping is needed during computation,
- if the motion is strongly non harmonic this can be seen on the PSD spectrum,
- results may be derived for high velocity ratios which would lead to global mechanical instabilities, using equation (15) and progressive recalculation of damping.

In figure 9 this procedure was applied to the results obtained with the negative damping model in the preceding test cases. It is seen that the corrected results obtained are consistent with those of the coupling forces modal, which is quite satisfactory.

Such a procedure is expected not to give statisfactory results in cases where several modes are unstable (in that case, Fricker's procedure applies) or when the motion is strongly non harmonic or chaotic (in which case no corrective procedure can be used).

5.3. Models of fluidelastic forces

It was observed, in all the cases presented in this paper, that a similar pattern of evolution of post-instability regimes could be found in the results of all models. This strongly suggests, as stated before, that the main difference between the results of these models lies in the relationship between the velocity ratio and the effective energy input, i.e. the equivalent negative damping. In fact, for all these models, this relationship is well defined in the case of a free harmonic motion and is usually expressed in terms of a stability criterion (equation 1). In non-linear post-instability regimes, this relationship is not explicity known and is quite dependent on the system considered. From a practical point of view, it is convenient to separate the problem of non-linear vibrations induced by fluidelastic forces in two distinct problems :

a) The computation of the non-linear vibrations induced by the negative damping. This is a problem of non-linear mechanics.

b) These forces should explicitly depend on the motion of the tube, which is not restricted to be harmonic.
c) In the case of harmonic motion these forces should lead to critical values of velocities consistent with the experimental data.

b) The derivation of the relationship between this negative damping (or energy input rate), the flow parameters and the tube motion. This is a problem of fluidelasticity.

In that sense, experimental and numerical studies using negative damping as a simulation of fluidelastic forces are thereby justified, but their application in terms of flow velocity should be done with special attention.

5.4. <u>Non-linear regimes</u>

The existence of a large variety of non-linear regimes, as observed in the test cases, is a problem of practical importance. In fact it was shown that changes of an order of magnitude could be observed in the parameters influencing wear, coincidently with changes in regimes. The mechanisms underlying these behaviour and the sensitivity towards different parameters are not yet clearly understood. For predictive analysis, it seems today reasonnable to use the following procedure in post-instability computations, to deal with this problem.

a) All computations should be done with at least two different values of initial conditions and friction coefficient to check for possible changes in regimes.
b) The simulated time-history should be sufficiently long to check for the stability of the regimes.

6. CONCLUSION

The predictive analysis of post-instability vibrations with impacts is of actual practical importance for heat exchangers tube bundles. In this paper, several models and methodologies were considered and applied on various test cases. In the results it was observed that

a) It is quite important that, in the fluidelastic forces models, the forces are expressed in terms of the real motion of the tube, not in terms of its free vibrations. In that sense, the negative damping model, as usually defined, was found to be overconservative. It is not recommended for the analysis of multisupported tubes.
b) When such a motion dependance of the forces is taken into account, the different models considered here give consistent results.
c) The main difference between these models seems to lie in the relationship between the flow velocity and the actual equivalent

This model has been widly used in computations (Axisa et al., 1988, Fricker, 1988) and in mechanical simulations of fluidelastic forces by a velocity feedback loop by Antunes et al. (1990).

Its use might nevertheless be questionnable, as it expresses forces which

negative damping or rate of energy input of the unstable mode. Experimental and theoretical work is needed to have a better knowledge of this relationship.

d) A corrective procedure is proposed to adapt the negative damping model to post-instability cases.

e) A large variety of dynamic regimes was observed, which were found to be quite sensitive to the friction parameter. Particular attention is to be paid on this problem in predictive analysis, as changes of regimes may induce considerable changes in wear work rate.

f) The measurement of fluidelastic forces remains a major point to assess realistic wear work rate prediction in post-instability situations.

7. ACKNOWLEDGEMENTS

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Fluidelastic forces acting on a tube are assumed to depend on the tube motion itself, through the lift and drag coefficients (Price and Païdoussis, 1984). In the linear (i.e. small displacement) form of the model, the instantaneous lineic forces are given by

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was found to be consistent with experimental data (Lever and Weaver, 1986). Moreover, the assumed flow pattern was in fact observed in practice (Weaver and Abd-rabbo, 1984, Andjelic et al., 1990).

In this model, the fluidelastic forces have so far been only derived in the case of harmonic motions, and are not suitable for non-linear computations,

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In the first part of this paper, it was shown that several models are suitable for non-linear post-instability computations.

If the negative damping modal is considered, equation (5) may be expressed in terms of the modal fluidelastic force to be used in equation (9)

Tanaka. H., Takahara, S., Ohta, K., 1982. "Flow induced vibrations of tube away with various pitch to diameter ratios", ASME pressure vessel and piping conference, Orlando.

For the three models considered here, it is seen that the modal basis representation is quite convenient for the expression of fluidelastic forces. In fact, the use of equations (10), (12) and (13) is much easier then the use of nodal forces in a finite element model. Moreover, the restriction of those fluidelastic forces on the unstable modes only is quite straightforward in the modal basis representation of the motion.

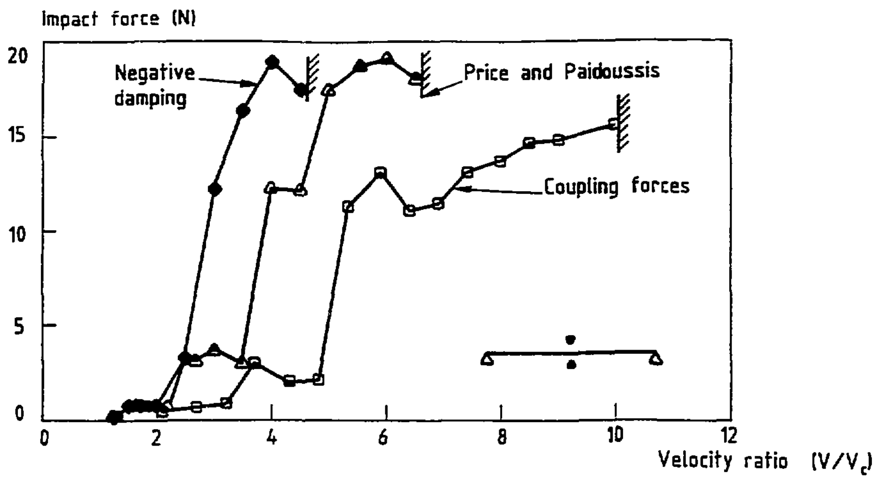

Fig. 1 - Case 1. Average Impact Force

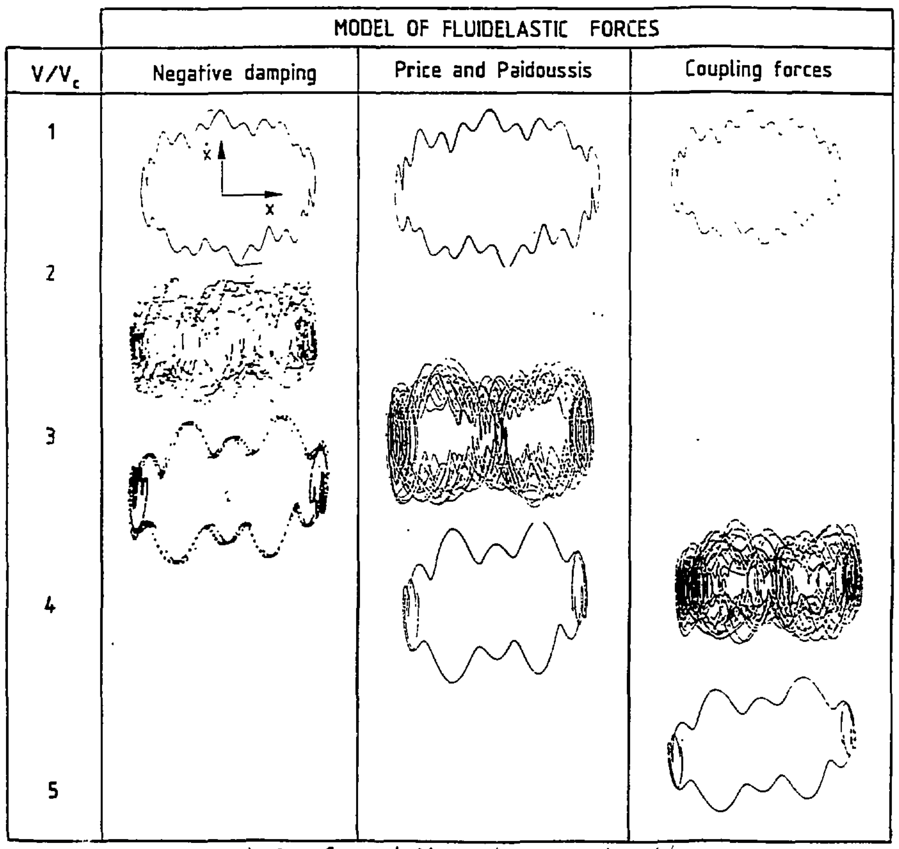

Fig. 2 - Case 1 Phase diagram at midspan

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was assumed for all modes. In fact, here, fluid damping, is not considered as a part of the fluidelastic forces but as a part of the total model damping. The time history is computed with a time step of 4.10^{-5} s, using more than 250 000 time steps in each case.

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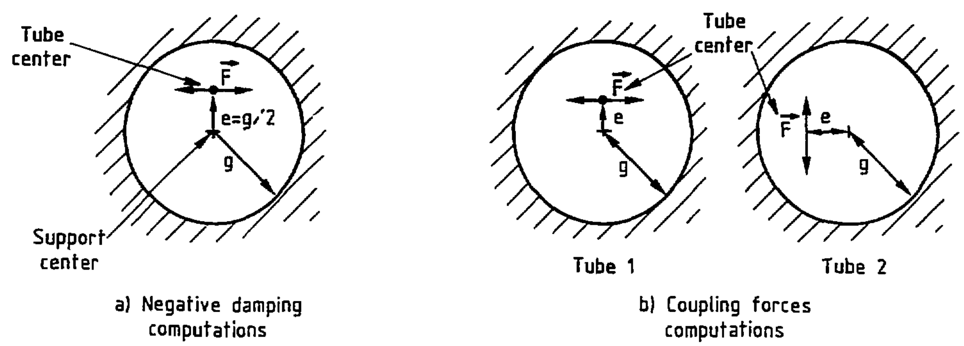

Fig. 3 - Case 2 :Configurations of tubes in the clearance circle
g : clearance, e : excentricity, $\vec{F}$: Fluidelastic forces

ra io grea er an a ou ., no stea y state regime can be found for the system. The computed impact force increases steadily with time. This mechanism was observed experimentally by Antunes et al. (1990). It is not a numerical divergence of the time stepping algorithm but a global instability of the mechanical system, where the stable modes dissipation is unable to balance the energy input of the unstable mode.

The results obtained with the Price Paidoussis model may be analysed in a

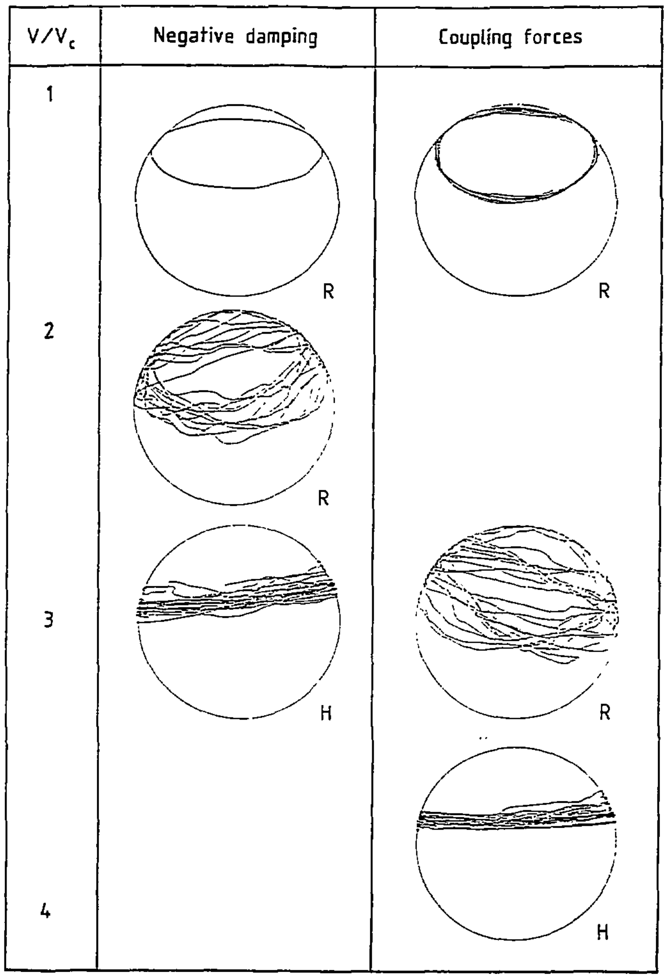

Fig. 4 - Case 2. Bidirectional motion with a circular loose support
Tube motion in the clearance circle
(R) Rubbing motion
(H) Hitting motion

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Case 2 : Bidirectional vibration with a circular loose support. Influence ofthe model of fluidelastic forces

The support at midspan of the tube is now considered to have the shape of a circular hole. In order to have a bidirectional motion of the tube, this support is assumed to have an excentricity equal to half the clearance. The fluidelastic

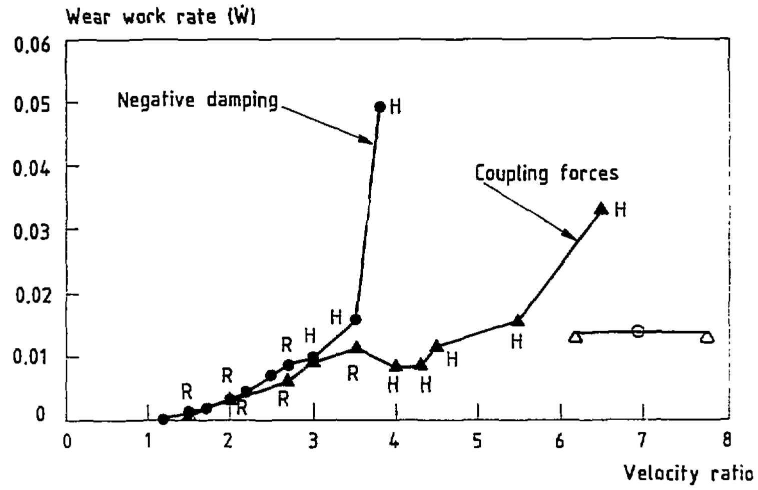

Fig. 5 - Case 2. Bidirectional motion with a circular loose support
(R) Rubbing motion
(H) Hitting motion

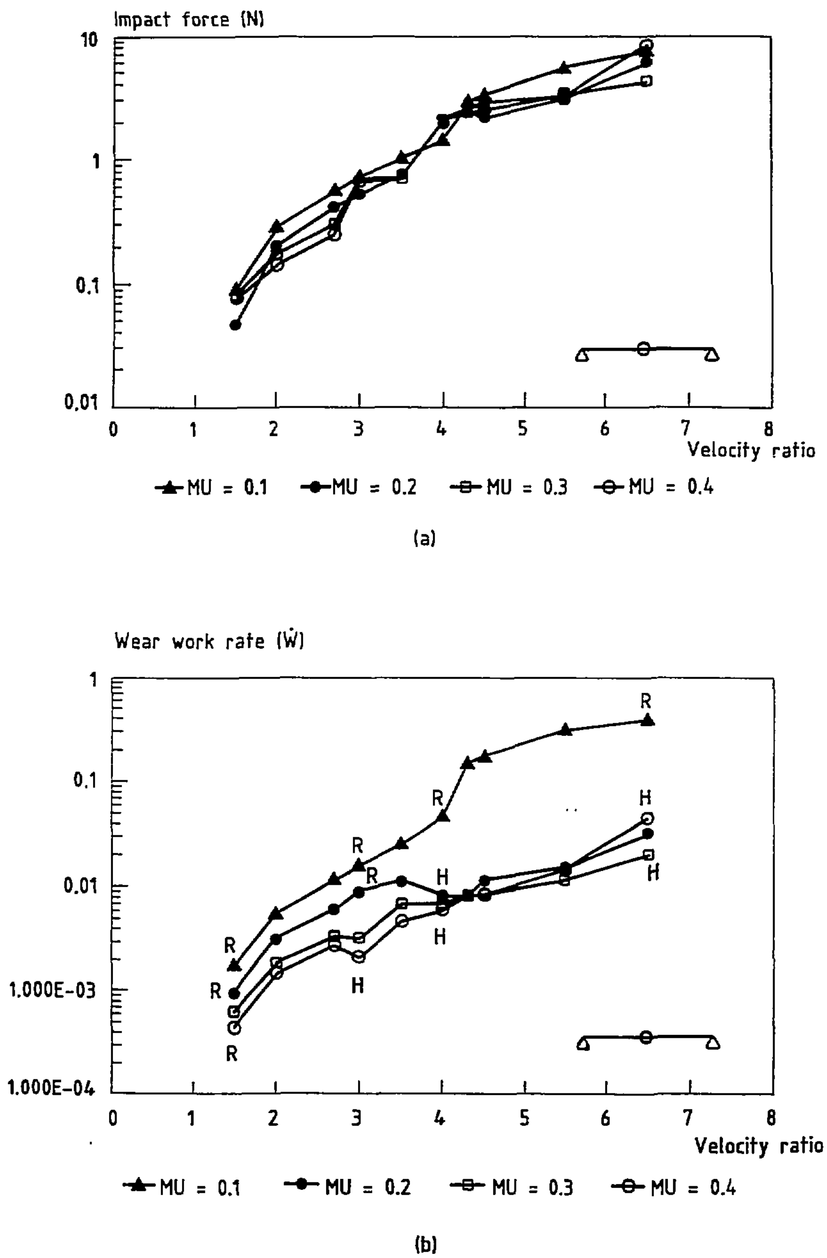

Fig. 6 - Case 3. Influence of the friction parameter
(R) Rubbing motion
(H) Hitting motion

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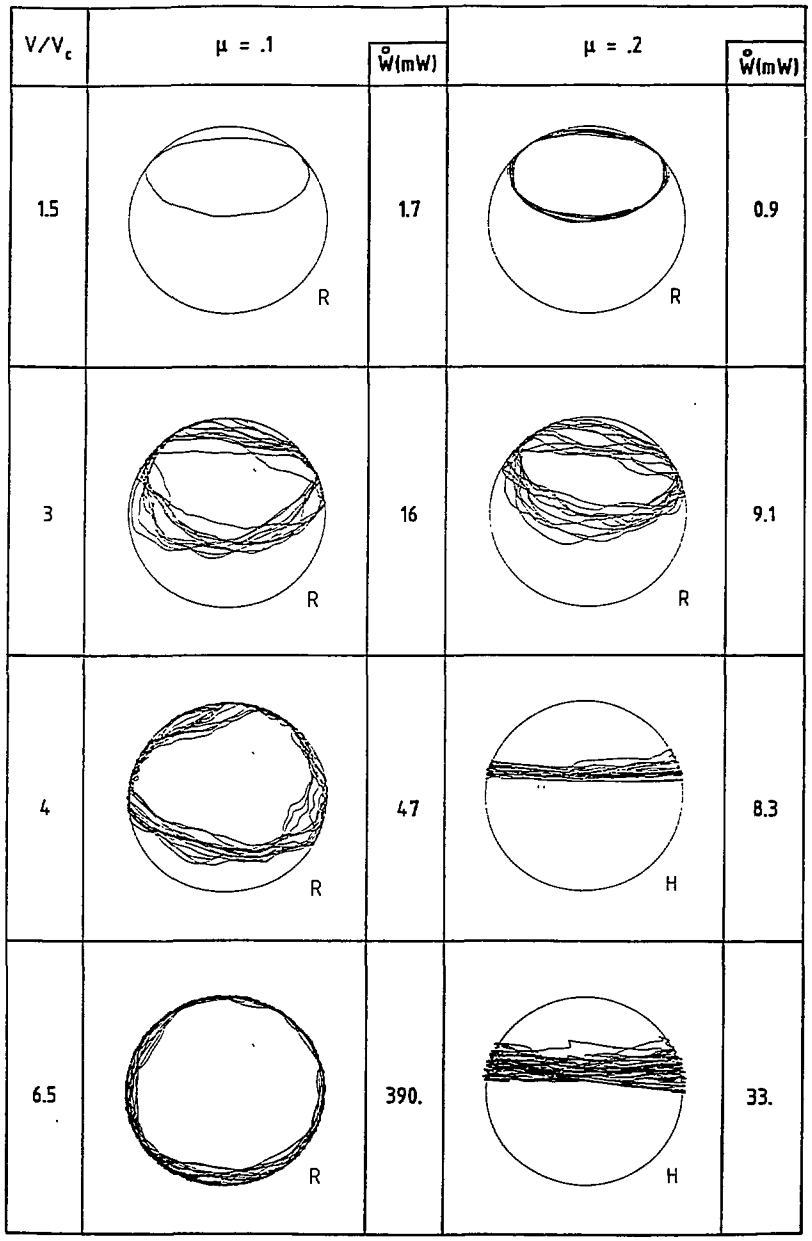

Fig. 7 - Case 3. Influence of the coefficient of friction on the tube trajectory in its support
(R) Rubbing motion (H) Hitting motion

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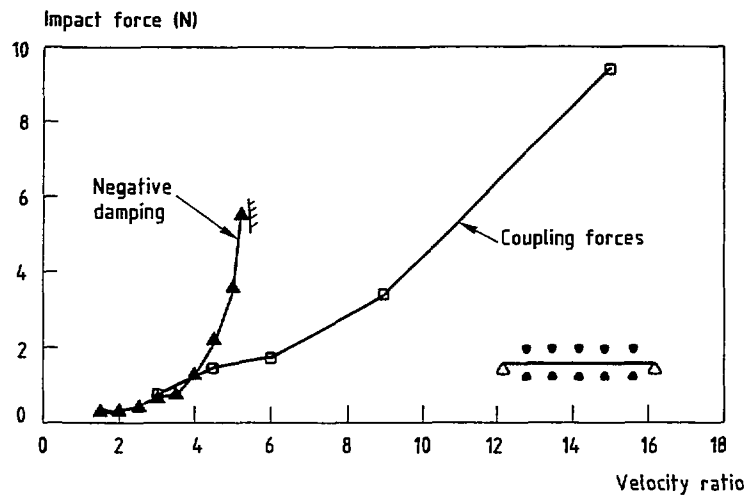

Fig. 8 - Case 4. Unidirectional vibration with five loose supports

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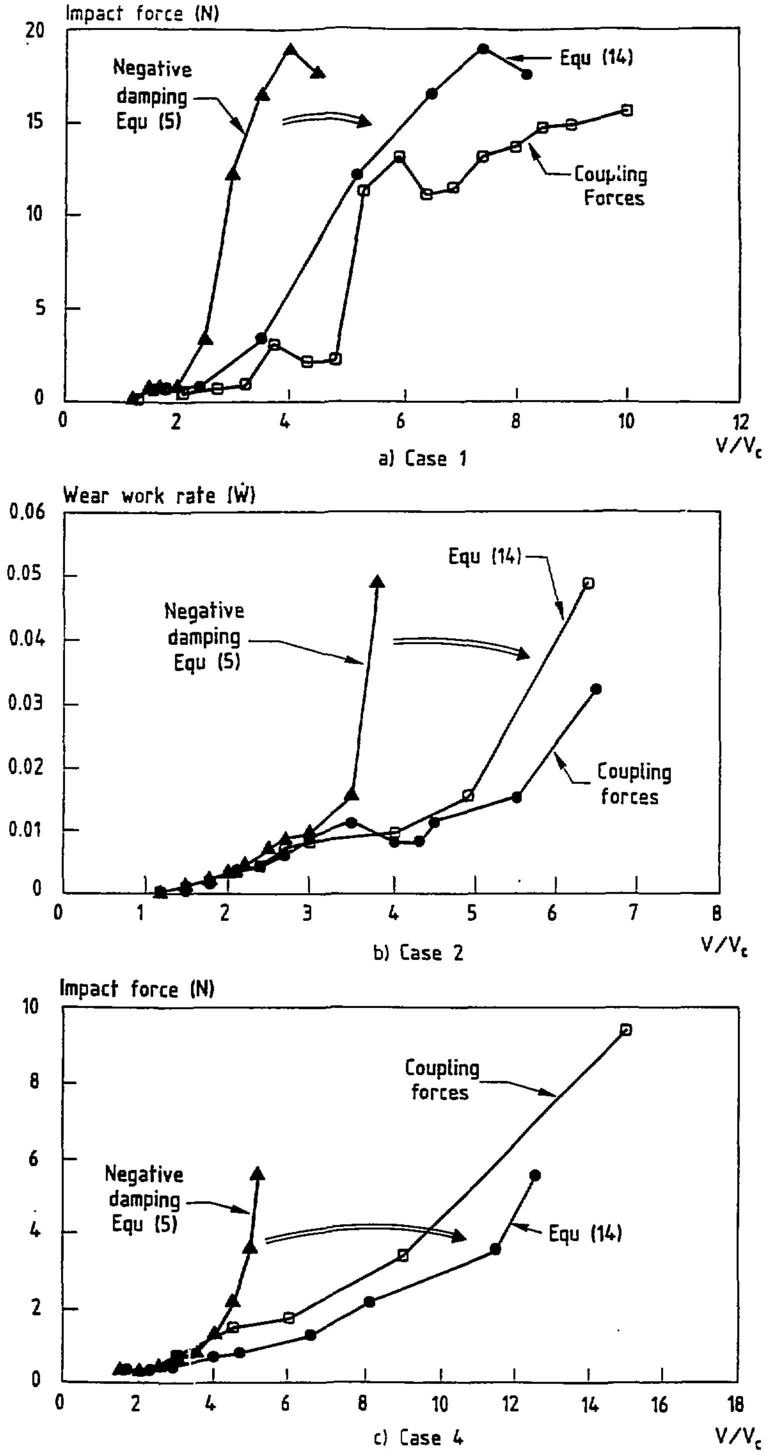

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