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A DECADE OF PROGRESS IN FLOW-INDUCED VIBRATION

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ABSTRACT

In this paper an attempt is made to discuss on a qualitative basis the FIV risks of typical reactor components and the methods employed in design evaluations, according to the present state of knowledge concerning the basic physical mechanisms. Instead of trying to provide the reader with an exhaustive list of practical cases of FIV incidents which occured in the nuclear power industry during the last decade, attention is focused on various generic problems which have motivated a large amount of research and which generated significant progress in the physical comprehension and design oriented analyses.

1 INTRODUCTION

Risks of excessive vibrations in nuclear components

Since its early beginning nuclear power engineering was faced with excessive flow-induced vibration problems. Indeed, nuclear reactors necessarily include flexible components permanently subjected to high flow velocities. Resonance frequencies less than one Herz are not uncommon. In liquid cooled reactors, mean flows up to 10m/s occur, at least in some places. In gas cooled reactors, mean flows of 30 m/s with gas densities of 30 kg/m^3 are present. Based on such orders of magnitude, it is immediatly realized that Flow Induced Vibrations (FIV) are of real concern in highly diversified areas of modern engineering. In the nuclear power industry, the problem is certainly not overwhelming, as evidenced by the statistics on nuclear plant shut-downs, which report a relatively small number of incidents caused by FIV. Accordingly, FIV problems are generally managed as a side aspect of the design, and it is remarkable, though a bit disapointing, that many of the failures caused by FIV which were reported in the 1970's and earlier, still occured over the 1980's in old and new designs. This clearly indicates that besides much valuable progress, important gaps in knowledge obviously persist and that deficiencies also exist in the process of transfering results of research into the design procedures. Whatever the present state-of-the art really is, most of the practical FIV problems are still cured or tried to be cured using a large amount of empirism.

Nevertheless, there is little doubt that flow-induced vibration phenomena in nuclear power station components will continue to deserve considerable attention because of obvious reasons of cost, safety and even of the emotional mediatic consequences of any incident occuring in the nuclear industry. To conclude this section, it may be worthwhile mentionning that besides being motivated by practical aims, the FIV area also has much of appeal as a scientific domain for research in applied mechanics, as it deals with at least two complex disciplines, namely fluid mechanics of turbulent flows and dynamics of complex structures, coupled together in a very subtle way. Then, it is expected that the FIV area will continue to be faced with new challenging problems of great scientific and industrial interest.

Scope of the present review

For sake of clarity and conciseness, is is found convenient to arrange the presentation according to a few physical mechanisms of basic importance in FIV problems:

Section 2 outlines fluid-structure coupling mechanisms which result from the interaction between a vibrating structure and a still fluid. In principle, such problems are mastered by theory since a long time. However, they are still motivating a substantial amount of research to improve the capabilities of the computer programs aimed at analysing the nuclear components.

Section 3 is devoted to the forced vibrations excited by the random fluctuations of the cooling flows. During the 1980's, significant progress was achieved to measure flow induced forces. Furthermore, the end of the decade was also marked by the development of new numerical techniques to calculate the fluctuating components of turbulent flows.

Section 4 is devoted to a few fluidelastic instability mechanisms which are still of major concern in nuclear components. Attention is focused on tube arrays in cross-flow and on axial annular flows. During the 1980's, much effort was paid to model the fluidelastic forces, in contrast with earlier work, which was mostly restricted to the experimental determination of linear instability maps.

Section 5 outlines impact-sliding interaction and fretting wear problems, which are of practical concern in several tubular components such as tube bundles of heat-exchangers, fuel and control rod assemblies etc... Indeed, as the nuclear industry is becoming more mature, expertise in the area evolves progressively from the initial major concern of avoiding short term failures, towards the prediction of flow induced risks which develop in the longer term, and which involve nonlinear vibration. Clearly, the development of such predictive methods is much more demanding in terms of modeling of excitation mechanisms and of dynamical responses, than the design recommandations aimed at avoiding prompt failure. It is expected that such trends will be amplified over the 1990's.

2 FLUID-STUCTURE COUPLING

Physical aspects

Small oscillations of quiescent fluid, induced by a vibrating wall are described by the continuity and Navier-Stokes equations, linearized around the quiescent state. To each of the quiescent fluid parameters, a distinct physical aspect of the fluid-structure interaction is associated, namely the density, ρ, is related to the inertial coupling and the speed of sound, c, to the acousto-mechanical coupling, which are both energetically conservative, whereas the viscosity, ν, is related to energy dissipation. When there is a free level, coupling with sloshing modes may also be of importance. By using dimensional analysis, it is easy to derive a few dimensionless parameters which characterize the relative importance of these effects. This can be conveniently illustrated by considering a coaxial cylindrical annular space of width, h, and mean radius, R. Analytical formulation of the problem is reviewed in particular, in Chen (1987a). Besides of being directly related to ρ, fluid added mass coefficients are proportional to the confinment parameter, $C = R/h$. Fluid compressibility can be discarded when the compressibility number $\lambda = \omega R/c$, is sufficiently smaller than unity. Viscosity effects are governed by the Stokes number, $S\nu = \omega h^2/\nu$. When $S\nu$ is greater than unity, viscosity produces a small amount of damping, proportional to $(S\nu)^{-1/2}$. At the opposite, when the Stokes number is very small viscous forces play a leading part, providing large damping proportionnal to $S\nu^{-1}$. Gravity effects are scaled by the Froude number $\mathcal{F} = \omega^2 R/g$, in which g denotes the acceleration due to gravity. As soon as $\mathcal{F}$ becomes sufficiently higher than unity, the coupling with sloshing modes can be discarded. It has to be noted that any other fluid geometry provided with two quite distinct length scales would behave qualitatively in the same way as the simple geometry just discussed. Hence, fluid confinement is a key parameter in fluid-structure interaction problems. This is also true as fluidelastic vibrations are concerned, simply because it turns out that fluid-elastic forces are proportional to the added mass coefficients.

Numerical models of complex fluid-structure systems

Vibration analyses of nuclear components require the use of large computer programs, able to treat fluid-structure coupled systems with complex geometries. Generally, analysis is started by assuming that the system is linear and conservative in nature. Then it is discretized according to a symmetrical variational principle, which makes use of two fluid variables, namely the acoustic pressure p and $\pi = \rho\Phi$, where Φ is the potential of the fluid displacement, see, Gibert et al.(1978), Combescure et al.(1980), Ohayon and Valid (1981). This formulation produces the same standard matrix system as the one used to describe the mechanical vibrations of structures without fluid, except that the acousto-mechanical vectors are mixing displacement and stress variables. In Gibert et al.(1978), the particular case of plane waves in piping systems was considered. Furthermore, Axisa and Gibert (1982) extended the formalism to nonlinear problems such as cavitation and transient response to explosive water-sodium reactions.

As shown in Jeanpierre et al.(1982), Aïta et al.(1983,1985) and Brabant et al.(1987) it may be advisable to use the finite element method in association with a modal synthesis technique to analyse fluid-structure systems of complex geometry. Indeed, such methods are very efficient in reducing the size of the system to be computed and in easing the physical understanding of the numerical results obtained. A typical example is provided by the analysis of the Superphenix internals, presented in Brabant et al. (1987). Here, it is sufficient to give a few figures which illustrate the size of the problem. The 3D calculations corresponded to the finite element mesh shown in Fig.1, where a quarter of the reactor is modelized, including one primary pump and two adjacent intermediate heat exchangers. The penetrations through the *redans*

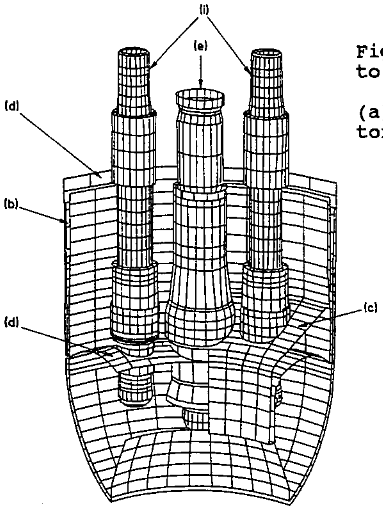

Figure 1 Finite element mesh of the SuperPhenix internals, used to perform seismic and vibration 3D analyses

(a) Main vessel, (b) thermal baffles, (c) conical redan, (d) toroidal redan, (e) primary pump, (i) intermediate heat exchangers

and the thermal baffles were also modeled. Sodium was considered in the analysis, but for the sake of simplicity the sloshing modes were discarded by assuming pressure nodes at the sodium free level. In a first step, modes of individual components were computed according to thin axisymmetric shell models. This resulted in 29 modes for the primary pump, 57 modes for the heat exchanger, at frequencies less than 40 Hz, and 246 modes for the vessels and thermal baffles with frequencies less than 20 Hz. Then, a modal synthesis method was used to compute the modes of the whole system, which resulted in 159 modes at frequencies less than 10 Hz. The first computed resonance was at 0.336 Hz. A few other calculations were also performed to test the sensivity of the results to the free level effect. Not surprisingly, the coupling between sloshing and structural modes was found to be important, especially at frequencies less than one Hertz. Calculations were also made in air for comparison with tests carried out on the reactor, see Aïta et al. (1983).

On the other hand, one is often faced with the problem of large fluid volumes when modeling reactor internals. Such volumes require specific simplified formulations, based on physical reasoning, in order to avoid too large or meaningless finite element meshes, and to properly connect distinct large volumes to one another. To deal with large fluid volumes, it can be of interest to mix finite and boundary element techniques, see Brebbia (1984), Hamdi (1987), Hwang and Ting (1987), Lazzeri (1987) Shiojiri and Hagiwara (1990). Finally, the problem of taking into account fluid-structure interaction is complicated further when numerous components are immersed in the fluid volume to be modeled. This is the case in particular for the core of many reactors. Then, homogenization techniques can be very helpful for replacing the actual system by an equivalent medium, see Schumann and Baner (1981), Planchard (1985), Brochard et al. (1987). However, such techniques are obviously more suited to deal with global modes of cores than with individual modes of tube rods. Hence, they remain of limited use for FIV analyses, where the main concern is about the risk of damaging individual components. The problem of the dense cluster of fluid-structure coupled modes of tightly packed tube arrays is discussed in Chen (1987a).

3 TURBULENT EXCITATION MECHANISM

Basic mechanisms

Random fluctuations are unavoidably present in most of the industrial flows. In single phase, they are related to turbulence. Depending on the flow geometry, the fluctuations have a broadband spectrum in which most of the power density is confined below some cutoff frequency, or they display spectra which are markedly peaked at a single or a few discrete frequencies, proportional to the mean flow velocity. The most classical example is the periodic vortex shedding observed when a single tube is subjected to cross-flow.

Determination of random excitation by turbulence

Measurement techniques

Turbulent buffeting is by far the most common FIV excitation mechanism, operative practically at any flow velocity. Hence, it can be foreseen that requirements to compute the response of nuclear components to this mechanism will be reinforced in the present decade, at least to analyse service life problems. Up to now, the only practical way of determining the turbulent force spectra rely on experiments and semi-empirical modeling. Over the 1980's significant progress was achieved in flow-induced force measurement and data processing techniques, see Mulcahy (1980,1984a), Tanaka and Takahara (1981), Goyder and Teh (1984), Taylor et al. (1986), Chen and Jendrzejczyk (1987b), Nakamura et al.(1991). The studies just quoted use *direct methods* consisting in measuring the fluid forces acting on the structure set in static or quasi-static equilibrium. On the other hand, *indirect methods* which consist in letting the structure vibrate freely in the flow to extract the modal parameters and the forcing function from the dynamical response were also significantly improved with respect to the 1970's, in particular by progressing in the data processing techniques, both in the time and in the frequency domains, see Granger et al. (1990), Esmonde et al.(1990). Generally, the information obtained is limited to the resultant forces exerted on the structure and no attempt is made to derive the spatial distribution of local forces, except in a very few studies in which wall pressures have been measured, see for instance, Blevins et al.(1981), Mulcahy et al.(1980,1981), Hobson (1991), Nakamura et al. (1991).

Tube arrays in cross-flow

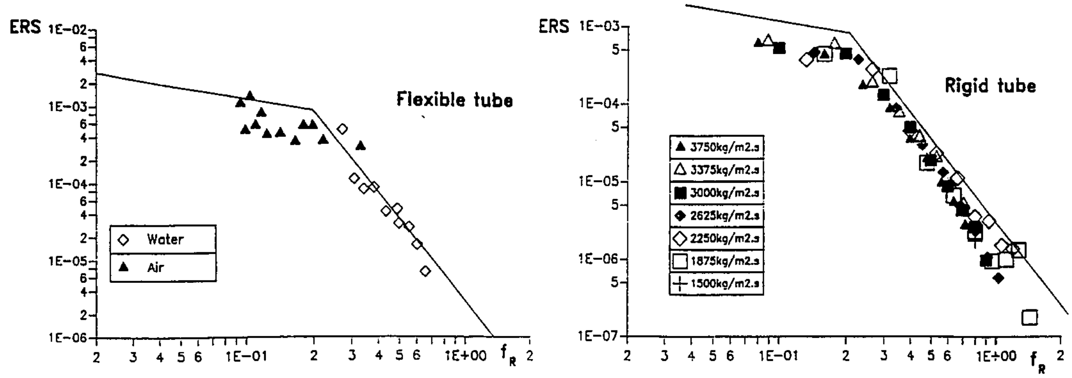

Figure 2 Equivalent reduced spectrum (ERS), of tubes bundles subjected to single phase cross-flows

Random excitation mechanisms of tube arrays in cross-flow has been reviewed in Païdoussis (1983), Axisa et al.(1988,1992), Taylor et al.(1988,1992) and Pettigrew and Taylor(1993). From these studies it can be stated that in single phase, simple scaling factors can be used to reduce the experimental data, namely the reduced frequency, fD/V_P, and the steady force per unit of length $0.5\rho V_P^2 D$, where f is the frequency, D the tube diameter and V_P is the pitch cross-flow velocity. In Axisa et al.(1988) a convenient procedure was given in order to avoid the experimental difficulty in measuring the correlation length, λ_c, of fluctuating forces along the tube span, of length, L, which is valid when $\lambda_c/L \ll 1$, as it is usually the case. Furthermore, it was found that forces induced by turbulence are not strongly dependent upon the array geometry. Then, based on a fairly large set of experimental data, a unique equivalent reduced force spectrum, (ERS), was derived, which may be used for design purposes, see the solid line in Fig.2, taken from Axisa and Villard (1992). Fig.2 also allows for comparison between measurements made by using the direct and the indirect methods, which are found to agree satisfactorily with each other.
Two-phase flows also generate random fluctuating forces, which can lead to excessive vibration. Not surprisingly, excitation mechanism in two-phase is much more complex than in single phase because it depends on the properties of the two-phase mixture, which have to be specified according to some empirical flow models, may be including even more qualitative information such as two-phase flow regimes. Furthermore, as pointed out in Pettigrew and Taylor (1993), research in two-phase induced vibration is a relatively new area of study and up to now, there is no widely accepted scaling factors for data reduction. As a first working hypothesis, it is tempting to start with using the homogeneous flow model

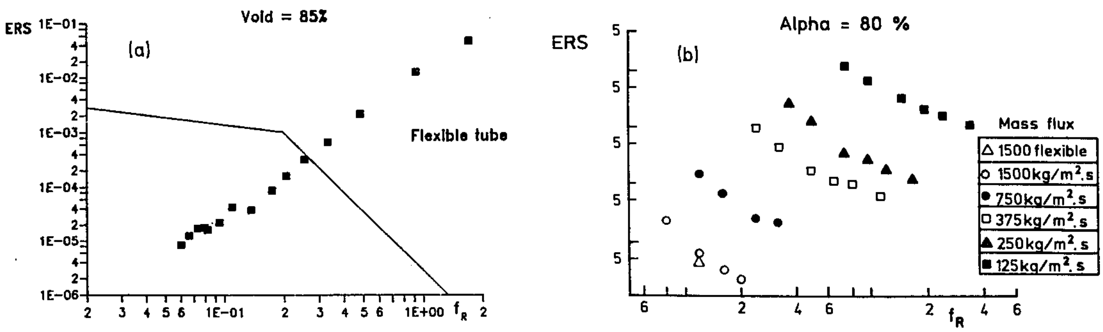

Figure 3 Equivalent random force spectrum in air water, reduced according to the single phase scaling factors (a) flexible tube, (b) rigid tube.

without slip between the two phases and to use the same scaling factors as in single phase. Figure 3, taken from Axisa and Villard (1992), shows typical equivalent reduced force spectra, derived according to this procedure for the same tube array as in Fig 2, when subjected to an air-water cross-flow with homogeneous void fraction $\alpha_H = 0.85$. Fig 3a, refers to data obtained with a flexible tube, at a fixed resonance frequency and at distinct homogeneous flow velocities V_H. Fig. 3b refers to data obtained with a rigid tube, at fixed V_H and distinct frequencies and α_H. It is obvious that the single phase scaling factors are physically irrelevant. In Figure 4 two-phase data obtained in air-water, steam-water and freon are shown to collapse nicely on a unique curve when the reduced quantity $ERS \times f_r^2$ is plotted versus the homogeneous mass flux G_H, resulting in a G_H^{-4}dependency. From this, it was possible to derive an empirical relationship to characterize the random forces versus void fraction. Similar, though distinct relationships are proposed in Taylor (1992).

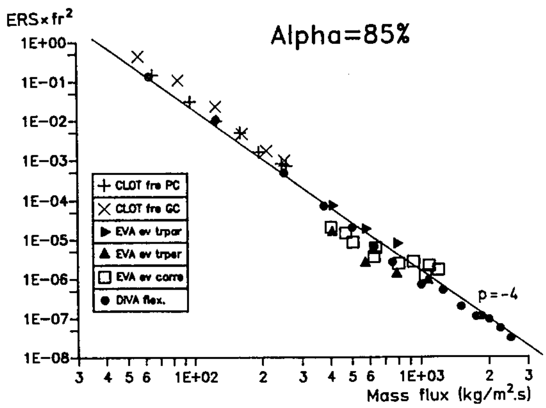

Figure 4 Random force spectra obtained in a few different two-phase fluids, CLOT: freon, EVA: steam-water, DIVA: air-water

Tubular structures in axial flows

Axial flows are prevailing in several parts of nuclear reactors. Piping systems can be the site of various undesirable FIV phenomena, excited by the fluid they convey. The effects of vibration on the integrity of piping systems is a real concern for both the nuclear and the non nuclear piping systems of a nuclear plant. Unfortunately, place is insufficient to discuss further this topic here which is overviewed in Axisa (1993).

External axial flows also are of practical interest. In particular, the fuel is often contained in slender straight cylindrical tubes, externally subjected to the primary coolant. There is generally little concern about such structures, at least when the flow is not confined too much and when it remains purely axial. Indeed, it is generally agreed that turbulence is a much weaker excitation mechanism in axial flow than in cross-flow. Moreover, in such systems, fluidelastic coupling induces extra damping which increases with flow-velocity, see Wambsganss and Chen (1971), Connors et al. (1982), Tanaka et al.(1988). Nevertheless, large uncertainties still remain concerning the actual tube response to axial flow turbulence. In Henry et al.(1979) several empirical relationships were investigated and relative discrepancies as large as two decades were found in the predicted rms responses of the tube amplitude. It would be of interest to improve knowledge in this area, because it may be more important than previously believed in relation to long term fretting wear problems. So far, excitation by two-phase axial flow in relation with nuclear fuel is even less documented, see Pettigrew and Taylor (1993).

Numerical simulations of turbulent flows

A quite challenging alternative to such experimental approaches is the development of numerical methods to calculate turbulent flows. Simulation techniques made considerable progress over the last decade, which can be evoked here only in a superficial way.

Whatever the drawbacks may be for the author to mention very specialized and difficult topics in which he has little intimacy, it is felt useful not to ignore the subject. Indeed, it is hoped that

this area of research will become sufficiently mature in the next future to be successfully applied to at least some of the basic configurations found in various nuclear reactor components, such as annular axial flows. Two promising approaches, referred to as direct and large eddy simulations, were recently reviewed in relation to turbulence induced forces by Hansen et al.(1987). They consist in solving the full Navier-Stokes equations on a computational grid. In the direct method, the grid is made sufficiently fine as to allow resolution of the small (dissipative) scales of the turbulent flow. Hence, obvious practical difficulties arise from the computer limited capabilities to deal at the same time with the small and the large scales of turbulence, the latter being of major interest in exciting the first resonances of the structures. As a consequence, its use remains presently limited to simple configurations like channel flows and to relatively small Reynolds numbers. In large eddy simulation techniques, attention is precisely focused on the behaviour of the large, geometry dependent, turbulent eddies and their interaction with the walls, whereas small scale turbulence is modeled according to suitable semi-empirical relationships which close the system. Both approaches led to a very productive effort over the past decade, see in particular Moin and Kim (1985). Pruit et al.(1992), performed a quite impressive simulation of the turbulence inside the bundle of a PWR steam generator, leading to calculated force spectra in good agreement with the experimental data by Chen and Jendrzejczyk (1987) and Axisa et al.(1988,1992).
To close this section, it has to be stressed that even if numerical tools for computing fluid-structure coupled systems in the presence of turbulent flows are expected to achieve substantial progress in a predictable future, it is still rather clear that experimental work will remain a key aspect to understand FIV mechanisms and to solve practical problems. At this respect, it has to be stressed that there is a basic advantage of maintening a fair balance between studies carried out on mock-ups which are representative of the practical problem under investigation and more generic and fundamentally oriented experiments, conducted on idealized laboratory rigs, which often reveal more suited than the former ones to model the FIV mechanisms of interest.

4 FLUIDELASTIC INSTABILITIES

Basic mechanisms

Fluidelastic forces arise when structures vibrate in the presence of a flowing fluid. They lead to an exchange of energy between the two systems. Depending on the direction taken by the flow of the exchanged energy, fluidelastic coupling has a stabilizing or a destabilizing effect on the vibrating system. Linear theory allows for defining coupling terms which can be interpreted as fluidelastic stiffness and damping matricial operators, related to the mean flow velocity, V, and to the fluid added mass coefficients. Then, the linear stability of the system can be analysed in a standard way, by inspecting the complex eigenvalues of it as functions of V. In most cases, calculation can be conveniently carried out by projecting first the equations on a set of a few modes of the conservative system (V=0). Nonlinear analyses also are feasible numerically, by using a time step integration routine. Basic difficulty, here, is to formulate the fluidelastic operators which are relevant in the problem of interest, i.e. to determine the fluidelastic forces. Up to now, there is no general theory or computational technique for solving this kind of problem. It is related to the calculation of turbulent flows, complicated further by the presence of oscillating boundary conditions. On the other hand, the experimental approach is also uneasy, especially when the goal is to measure the fluidelastic forces and not only to determine the stability charts.

Fluidelastic instability of tube arrays in cross-flow

It is notoriously known, that fluidelastic instability of heat exchanger tube bundles subjected to cross-flow is a major concern when designing such components against FIV risks. Since the pioneering work by Roberts (1966), several empirical criteria have been proposed to formulate the critical flow velocity for this instability and to relate it to simple formulations of the fluid-elastic forces. The most famous relationship, broadly known as the Connors' equation is expressed in terms of the dimensionless pitch velocity V_P/fD and the dimensionless mass-damping parameter $2\pi\zeta m/\rho D^2$, where ζ is the modal reduced damping and m the equivalent mass per unit length of the tube,

$$V_P/fD = K\sqrt{2\pi\zeta m/\rho D^2} \qquad (1)$$

K is a coefficient which may depend upon the array geometry. This model can be derived by assuming that in a fully flexible array, neighbouring tubes are antisymmetrically coupled by fluidelastic stiffness forces proportional to ρV_P^2, see Blevins (1990). Since these early studies, it was recognized that the Connors-Blevins model oversimplifies reality. In particular, it was verified that a tube vibrating within a rigid array can also become unstable. This motivated the development of new semi-empirical models based on single degree of freedom vibrating systems and simplified flow induced force calculations or measurements, see Lever and Weaver (1986), Price and Païdoussis (1987). Such models provided new enlightening insights into this difficult problem. In particular, it was recognized that several distinct stability boundaries are likely to take place instead of the line defined by the eq. (1). Moreover, nonlinear instabilities could be easily anticipated from such models, which were actually identified in experiments by Andjelic et al (1989) and by Chen (1988). Based on such studies, new stability criteria were proposed by a few authors, as reviewed in Chen (1987a), Weaver and Fitzpatrick (1987), Pettigrew and Taylor (1991). As a consequence, the designer is presently left in a rather unconfortable situation, because of the abundance of models and design guides which are still open to controversy. Pettigrew et al. (1989,1991) produced a new piece of information to the debate by pointing out that most of the discrepancies between the experimental data and eq.(1) can be due to experimental errors and relative inconsistencies in data reduction procedures adopted by different researchers. This conclusion is based on an elaborated critical review of a very substantial set of data in single and two-phase cross-flows, which is available in the open literature. From the designer point of view, a major conclusion arising from this work certainly is that less than 10% of the data would be left in the unstable domain defined by the Connors' stability criterion with K = 3, and this in a very large range of reduced mass-damping parameters from 5×10^{-2} to 5×10^{3}, see Fig.5 taken from

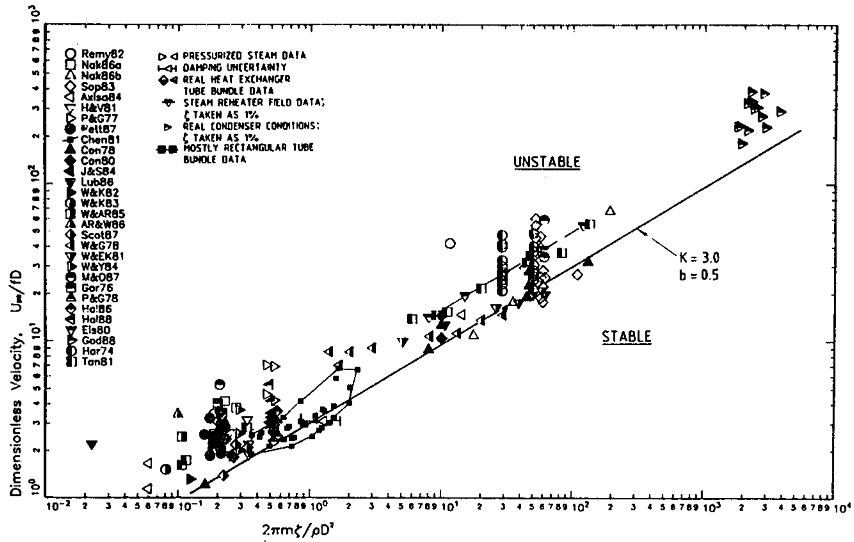

Figure 5 Fluidelastic instability data for all flexible tube bundles, with kind permission of M.J. Pettigrew

Pettigrew and Taylor 1991. However, it may be noted that as useful such a statistical approach may be, it is still not sufficient to model the fluidelastic forces underlying the instability. To the author's opinion, much effort has to be made in the future to produce experimental data in this area, which would extend those presented in the pioneering work by Tanaka et al.(1981). Indeed, knowledge of fluidelastic forces is necessary not only to understand better the physical mechanisms but also to perform design analyses of fatigue and wear of tubes vibrating in a fluidelastic nonlinear regime, as outlined in section 5 of this paper.

Annular leakage flow instabilities

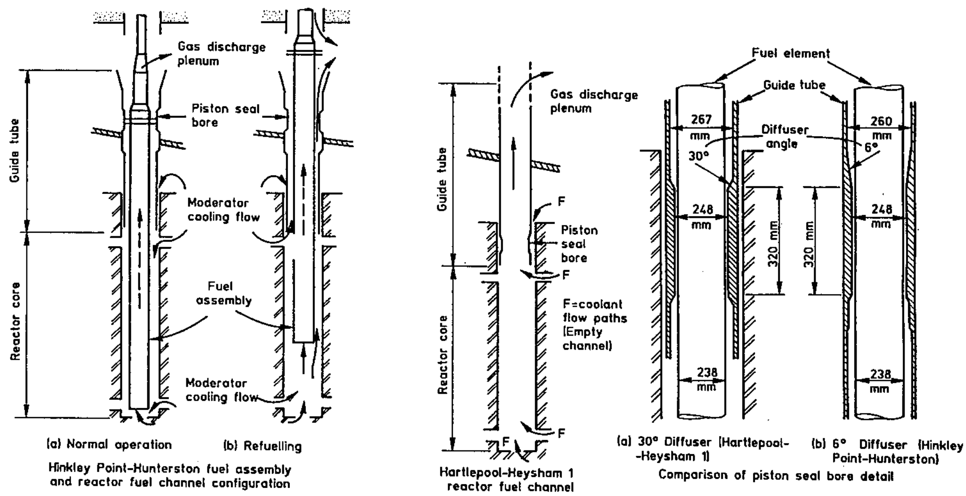

Figure 6 Fuel stringers of AGRs, with kind permission of D. Hobson

The occurence of fluidelastic instabilities in nuclear components, due to highly confined axial flows were clearly identified on Advanced Gas cooled Reactors (AGR) developed in Great Britain up to the mid 1980's. The component of interest is a fuel assembly which looks like a slender articulated cylinder 22 m long, suspended in a cylindrical channel in the graphite bricks of the core, see Fig. 6. Instability was observed to occur during refuelling operation, when the component is subjected to an annular leakage flow of carbon dioxide at 40 Bar. This problem is documented in an extensive series of papers, see in particular Hobson (1982,1984), Parkin et al.(1980-1987). Analytical modeling of leakage flow instability was formulated on a semi-empirical basis first by Hobson (1982). The basic idea was to recognize that pressure loss in a narrow annular space becomes time dependent because of the vibration of the system.

Leakage flow instabilities were extensively investigated over the 1980's, from several theoretical and experimental points of view, in relation to geometries of relevance in AGRs, and to more generic configurations. In particular, Hobson's model was refined to cope with multidegree of freedom systems, continuous pressure loss coefficients and nonlinearities caused by impacts of the fuel assembly on the channel, Hobson (1984), Parkin et al. (1987). Hobson and Jedwab (1988), and Hobson (1991), carried out experiments which allowed measurements to be made of flow induced forces exerted on the vibrating fuel assembly, set in concentric and eccentric configurations. Mulcahy (1984,1987), investigated the instabilities of a cylindrical annular joint, provided with a localized annular flow restriction. In concentric configurations negative damping instability was obtained provided that the flow restriction was located upstream in the annular space, in good qualitative agreement with Hobson (1982,1984). Divergence occured for upstream and for downstream restrictions. The effect of relative eccentricity between the two cylinders was found rather marginal. Mulcahy (1988) studied the one-dimensional leakage flow instabilities of Single Degree-of-Freedom (SDOF) systems, based on semi-empirical formulations similar to those adopted by Hobson. Analytical calculations were performed to emphasize the relative importance of several aspects of the problem such as position of the flow restricting device, change in cross-section of the flow channel, and wall friction. Inada and Hayama (1990a,b) analysed a Two-Degree-of-Freedom (TDOF) system, which consisted of a thin plate separating two tapered narrow one-dimensional channels of air flow. Divergence, SDOF and TDOF flutter instabilities were predicted by the model in close agreement with the experimental results.

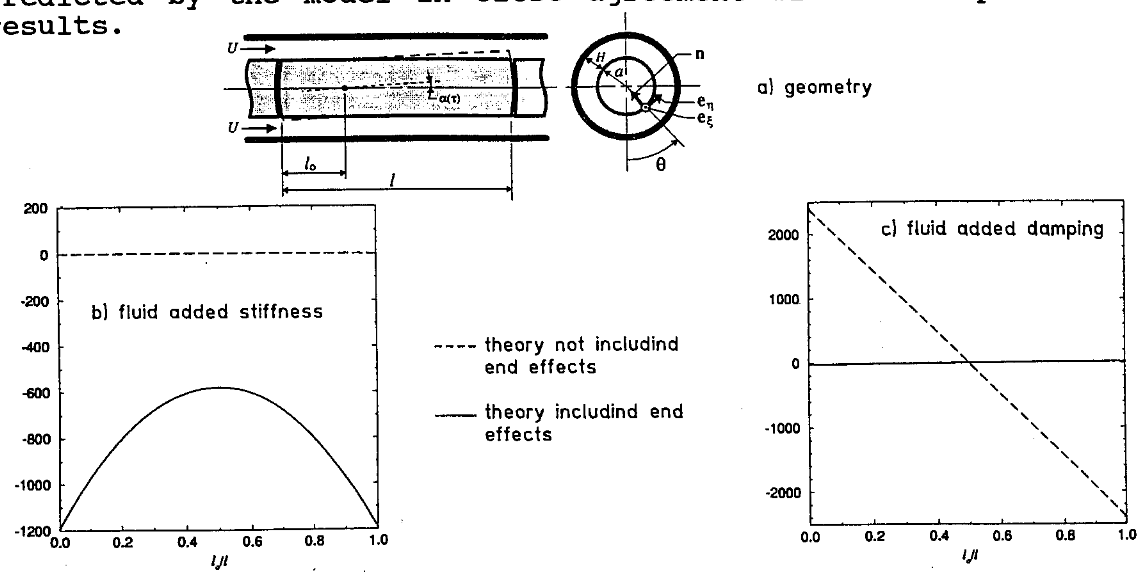

Figure 7 Fluidelastic parameters in annular axial flow, plotted vs hinge location lo/l

The problem of determining the fluidelastic forces based on a rather general computational technique was investigated at McGill University by Mateescu, Païdoussis and co-workers, starting first from potential flow theory, simplified in such a way as to make the problem tractable analytically. The physical system they considered consisted of an axisymmetric rigid cylindrical body maintained at one point by a rotational spring and dashpot, so that it could undergo rocking oscillations, see Fig.7. This center-body was coaxially mounted inside a uniform (or convergent or diffuser shaped) duct. The main flow upstream and downstream from the vibrating center-body was smoothly shaped by two fixed ogives. The fluidelastic forces exerted on the oscillating body were computed, and calculated pressure fluctuations were compared to experiments Mateescu et al. (1988,1989), Païdoussis et al. (1991a,b). According to potential flow calculations by Mateescu and Païdoussis (1984, 1985), it was found that,

(a) negative damping instability of the center-body can occur, unless the support point is sufficiently far upstream.

(b) divergence instability is impossible, for a center-body shaped as a uniform cylinder. It becomes possible for a diffuser, at least if the support point is located downstream from the middle section of the centre body.

A simplified formulation including viscous effects did not change such qualitative conclusions, though viscosity has a significant stabilizing effect on the system El Chebaïr et al.(1988). Comparison with experiments, Mateescu and Païdoussis (1985), Mateescu et al.(1988), was found good for pressures sufficiently far from the extremities of the oscillating center-body, and even excellent when a theoretical computational model dealing with turbulent flows was used, Mateescu et al. (1989). The fact that large discrepancies occured for pressures taken near the ends of the center-body was accepted as an unavoidable deficiency of the mathematical models which did not take into account the displacement discontinuities between the fixed ogives and the oscillating center-body. However, de Langre and Bélanger (1992) reviewed the potential flow theory using a finite element technique which allows to perform 3D analyses and to model in a more realistic way the end conditions of the McGill experiment. As a first result, the conclusions of the preceeding work on the stability of the system changed completely because the unsteady pressure at the extremities of the centre-body is of paramount importance in the problem. Calculation taking into account end effects is found to fit much better the experimental data obtained at McGill on the amplitude and phase profiles of the unsteady pressure along the cylindrical center-body, than calculation performed without end effects. Starting from a flow calculation similar to Hobson's model, taking into account turbulent dissipation and end effects, Pérotin and Granger (1992) arrived at the same conclusion. Thus, it is believed that the quite appealing subject of fluidelastic instabilities in annular leakage flows will still motivate very exciting research work before disclosing all its secrets and tricks. Nevertheless, it can reasonably be expected that suitable computational procedures will be made available to the designer in a near future. Finally, fluidelastic vibration of cylindrical structures have been extensively studied, even in the case of poor confinment, see, Païdoussis (1987).

5 IMPACT-SLIDING INTERACTION AND FRETTING WEAR

Worldwide experience on heat exchangers of various types clearly demonstrated that FIV can easily become a very serious problem, if not addressed with sufficient care at the design stage. This is particularily true as the steam generators (SG) of nuclear power stations are concerned. Maintaining the service life of SGs over more than 30 years under highly reliable conditions requires a complex design to prevent various damaging processes, including those related to FIV. Over the 1980's, it became generally accepted that vibration of multisupported tubes depend largely on the non linear interaction at loose supports. Accordingly, the predictive methodology leading to the assessment of a steam generator design with respect to vibration problems evolved considerably over the last decade. The major steps are summarized in Fig 8, taken from Axisa et al.(1989). Use is made of numerical methods to compute the vibratory response of multi-supported tubes in association with semi-empirical relationships for quantifying FIV mechanisms and tube damaging processes, in particular wear. Efficient algorithms to compute the tube dynamics have been developed first by Rogers and Pick (1977). A major aspect of the problem, pointed out in Axisa et al.(1986), is that fluidelastic instability is far less predictible than assumed in standard linear analyses. Indeed, in an actual steam generator low-frequency modes may develop, due to tube-support gaps, and become unstable at comparatively low flow velocities. Experiments were found to agree with such a point of view, see Chen et al.(1984), Antunes et al.(1985), Fricker (1988,1991). In Axisa et al.(1986) the fluidelastic vibrations nonlinearly stabilized by impacting on a loose support were investigated. Several distinct dynamical regimes, both periodic and chaotic, were clearly identified see Fig.9. Similar responses were further investigated to characterize chaos trough the analytical tools of non linear dynamics by Païdoussis and Moon(1989), and by Cai and Chen (1992).

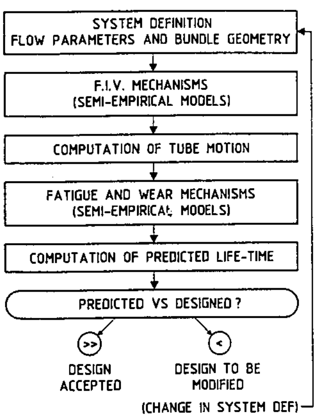

Figure 8 Flow chart of design predictive analyses

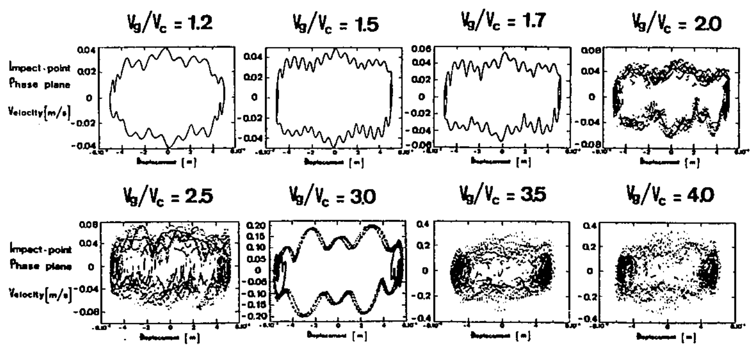

Figure 9 Phase portrait of a tube vibrating in nonlinear fluidelastic regimes and limited by a loose support at midspan; instability rate is an increasing function of V_g/V_c.

From a more practical point of view, the topological nature of tube response is likely less important than the time averaged quantities such as wear work rate, which control tube damaging. Then a few authors paid much effort to investigate the behaviour of such averaged quantities. Antunes et al.(1990), Vento et al. (1992) conducted experiments in which fluidelastic instability was simulated by using an electronic feed-back loop. Axisa et al.(1989), and Axisa and Izquierdo (1992) described other experiments in which tubes were excited either by cross flow turbulence or by electromagnetic shakers. Fig.10, taken from Axisa et al.(1989), indicates that reasonably good agreement between computed and measured wear work rates can be achieved, even for tubes bundles subjected to random excitation by turbulence. On the other hand, Antunes et al. (1988), de Langre et al.(1991,1992), Fisher et al.(1992), investigated the importance of various parameters entering in the numerical simulations, namely the duration of time averaging process in relation with chaoticity of the tube response, the effect of dry friction in relation with orbital motions in fluidelastic regime, and the influence of the semi-empirical model used to determine the fluidelastic forces. As it could be easily expected, the last point was found to be of major importance. As a whole, Such numerical and experimental studies provided very valuable information to validate the whole methodology sketched in Fig.8.

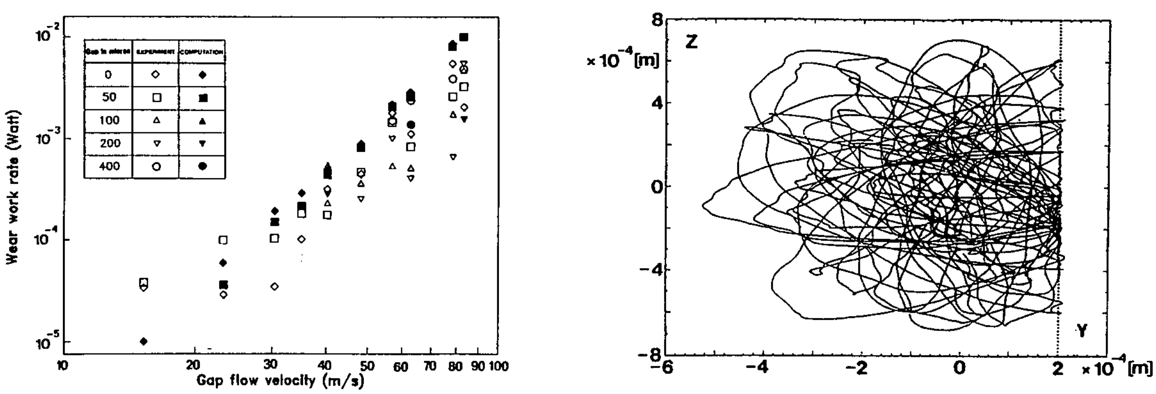

Figure 10 Calculation and measurement of wear work rate for a tube excited by cross flow turbulence and impacting at a antivibration bar, as shown on the Lissajous plot.

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