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Interactive Fluid-Structural Dynamic Problems in Power Engineering

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Interactive Fluid-Structural Dynamic Problems in Power Engineering

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FOREWORD

Dynamic analyses of fluid systems within rigid boundaries and dynamic analyses of flexible structures in the absence of fluid have both been developed to highly sophisticated states that include closed-form analytical solutions, finite difference and finite element numerical solutions or simulation, as well as scale model testing and dimensional analysis. However, the analysis of coupled fluid-flexible structural system is relatively new to the power generation industry, even though energy conversion almost always involves the interaction of the working fluid and the structure confining or conveying it.

The mathematical models for both fluid and structure systems were formulated years ago as governing differential equations for both steady and unsteady states. Solutions to these differential equations, both in closed form and by numerical techniques, have been obtained in many cases of engineering importance. Although the governing physics remains unchanged, the combination of a fluid region in contact with flexible structural boundaries forms a coupled system whose dynamic response is often completely different from that of either the fluid or the structure alone — as researchers in the dynamics of heat exchanger tube banks long are aware of.

In the past five years, many papers have appeared describing methods for solving the coupled fluid-structural dynamic problem. Most of these were addressed specifically to the safety analysis of light water nuclear reactors and ignored any earlier work. This symposium, which is sponsored by the Subcommittee on Fluid-Structure Systems of the Pressure Vessel and Piping Division, will clarify some of the confusion existing in the current state of the art. Emphasis has been placed on understanding the physics of coupled fluid-structure systems commonly encountered in the power generation industry and on clarifying the different methods of solving the problem. Papers that solve specific problems are included mainly as illustrative examples. Papers related to traditional tube bank dynamics are not included in this volume because tube bank dynamics is a highly specialized subject that deserves a separate symposium.

Readers who are interested in fluid-structure interaction may be interested in an earlier publication, *Dynamics of Fluid-Structure Systems in the Energy Industry,* edited by M. K. Au-Yang and S. J. Brown (ASME Special Publication PVP-39, 1979).

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FLUID-STRUCTURE INTERACTION – A SURVEY WITH EMPHASIS ON ITS APPLICATION TO NUCLEAR STEAM SYSTEM DESIGN

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ABSTRACT

The development of fluid-structure interaction methodologies as applied to nuclear steam system design is reviewed and classified according to whether the system is strongly or weakly coupled. Different numerical methods are discussed and compared. Finally, the effect of fluid-structure interaction on flow-induced vibration, seismic response, and response to loss-of-coolant accidents in an NSS is discussed.

INTRODUCTION

Even though fluid-structure interaction seems to be a very recent technology to a nuclear steam system (NSS) designer, its history can actually be traced back to 1843, when Stokes studied the uniform acceleration of an infinite cylinder, which is a structure in its simplest form, in an infinite fluid medium. He concluded that the only effect of the fluid on the motion of the cylinder is to increase its effective mass by an amount equal to the mass of fluid it displaces (1). The elegance of Stokes' result lies in its simplicity – it enables an analyst who does not specialize in fluid mechanics to calculate the acceleration of a solid in a fluid. His basic idea of added effective mass (called virtual mass in books on fluid dynamics) greatly influences even the latest development in the theory of lfuid-structure interaction. This will be apparent in the latter part of this paper.

Yet despite its effectiveness, Stokes' virtual mass equation was derived for a very specific problem, the uniform acceleration of a solid in an infinite fluid medium. In the absence of any better analytical technique, aerospace and NSS engineers in the early 1950's applied the virtual mass equation to calculate the natural frequencies of liquid fuel tanks and nuclear reactor internal components. Subsequent field test data had shown that they missed the mark by huge margins.

Modern fluid-elastic structure interaction apparently began in the 1950's. The earliest applications, however, are not to NSS design but to large liquid fuel tanks in aerospace vehicles (2). The earliest documented study specifically for NSS application appears to be the famous Fritz and Kiss work at Knolls Atomic Power Laboratory (3,4). Their report, available only from the U. S. government clearing house, lay relatively unnoticed for almost 10 years.

Fritz and Kiss' report dealt only with the rocking motion of coaxial cylinders; therefore, it is still a fluid-solid interaction study except that the fluid medium is finite, and the acceleration of the solid is nonuniform. One of the earliest papers on the dynamics of coupled coaxial fluid-elastic shell systems was published by Krajcinovic in 1974 (5). Since the mid-1970's fluid-structure interaction has drawn the attention of most NSS designers, particularly as related to safety issues. A vast number of papers has been published on this topic in the past four years. As one might expect, the earlier papers dealt with fundamental studies based on simplified models and analytical solutions. This has recently been superceded by papers dealing with solution techniques and applications to complex NSS systems.

In the following sections, we first review fluid-structure interaction problems from a broader viewpoint, then specialized problems related to hypothetical accidents in nuclear reactors. Problems related to the dynamics of tube banks and nuclear fuel bundles, although belonging to the same topic and certainly very important to NSS design, are not discussed here because they are a special field in themselves. Readers who are interested in this area are referred to an excellent review article by Nahavandi and Chen (6).

STRONGLY COUPLED SYSTEMS

In common with the science of interacting fields, a fluid-structure system can be classified as strongly or weakly coupled. A strongly coupled fluid-structure system is one in which the flow field induced by the structural motion (called induced field from now on) and the original flow field (hereafter called incident field) cannot be linearly superimposed on each other. This is usually caused by large structural displacements, resulting in large induced fluid velocity and completely distorted incident flow field. Perhaps the best known examples of strongly coupled (or interacting) fluid-structure systems are steam generator tube banks.

Less familiar to NSS designers is the huge field of aerolasticity, which can also be classified under strongly coupled fluid-structure systems and is beyond the scope of the present discussion.

To illustrate the complexity of a strongly coupled fluid-structure system, we write down the governing equations assuming that the fluid is non-viscous, non-conductive, and single-phased and that no heat is generated within the fluid itself. These are as follows:

<u>Eauations of Fluid Dynamics</u>

Continuity equation: $$\frac{\partial \rho}{\partial t} + \nabla(\rho \underset{\sim}{V}) = 0 \qquad (1)$$

Momentum equation: $$\frac{\partial \underset{\sim}{V}}{\partial t} + \underset{\sim}{V} \cdot \nabla \underset{\sim}{V} + \frac{1}{\rho} \nabla p - \underset{\sim}{F} = 0 \qquad (2)$$

Energy equation: $$\frac{\partial E}{\partial t} + \nabla \cdot (E\underset{\sim}{V} + p\underset{\sim}{V}) - \rho \underset{\sim}{F} \cdot \underset{\sim}{V} = 0 \qquad (3)$$

where $E = \rho e + \frac{1}{2}\rho V^2$, ρ is the density of the fluid, $\underset{\sim}{V}$ is the fluid density vector, $\underset{\sim}{F}$ is the body force on the fluid, e is the internal energy, and p is the pressure.

<u>Equations of Structural Dynamics</u>

$$[m]\{\ddot{q}\} + (i\nu + 1)[k]\{q\} = \{p\Delta A\} \qquad (4)$$

where [m] is the mass matrix, [k] is the stiffness matrix, {q} is a column matrix corresponding to the structural displacements, ν is the damping coefficient, and ΔA is the element area on which the fluid pressure acts. In equation 4, we assume that the only force acting on the structure is the fluid pressure.

The equations of fluid dynamics (1-3) and structural dynamics (4) are coupled by the requirement that at the fluid structure interface, the fluid velocity normal to the structural surface must be equal to the normal component of the structural velocity:

$$q_n = V_n. \tag{5}$$

Equations 1-5, which have been written in compact vector and matrix notations, are far more complex than they appear to be. They would be even more formidable if viscosity, thermal conductivity, and heat addition or dissipation were included in the analysis. To date, no analytical techniques for dealing with such complete systems have been developed. Simplifications are often possible in most cases, including the weak coupling approximation discussed below.

WEAKLY COUPLED SYSTEMS

In contrast to a strongly coupled fluid-structure system, a weakly coupled system is one in which the flow field induced by the structural motion can be regarded as a small perturbation of, and therefore can be linearly superimposed onto, the original (incident) field. The best known example of this is small-amplitude vibration of shell structures in a fluid. Because a pressurized water reactor (PWR) is designed to respond only slightly to a sudden loss of coolant accident (LOCA), most LOCA problems also belong to this category. Using the weak coupling approximation, many investigators had been able to predict the natural frequencies of a thin shell vibrating in a heavy fluid, and their results had been verified by experiments (7-11).

Fluid Priority Approach to Weakly Coupled Systems

To see the simplifications made possible by the weak coupling approximation, we discuss here the steps followed by Dienes to solve the PWR LOCA problem (12). First, the pressure is computed by a thermal hydraulic code, SOLA-DF, which is a finite difference code based on the equations of two-phase fluid dynamics. In the case of a single-phase, non-viscous, non-conductive fluid, these equations are the same as equations 1-3. In computing the pressure, the structural boundaries are assumed to be stationary. This pressure distribution is then input as the forcing function to a structural analysis code called FLX, which then computes the response. This response is returned to the SOLA code as a correction term in the velocity. Thus, e.g., instead of V in equations 1-3, we have

$$\underset{\sim}{V} + \hat{r}\dot{w}$$

where $\hat{r}\dot{w}$ is the radial velocity of the core barrel. Equation 1 becomes, e.g.,

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \underset{\sim}{V}) = -\nabla \cdot (\rho \hat{r}\dot{w}).$$

This is the continuity equation with a source term in it. The equations of fluid dynamics with a source term are again solved by the SOLA code. The resultant pressure can then be input again into the FLX code to obtain the corrected structural response.

In the SOLA-FLX scheme, the iteration above is carried out at every fluid time step. Furthermore, two iterations are required to yield the structural response.*

Since fluid-structure interaction is accounted for in the equations of fluid dynamics in the scheme above, it is a fluid priority approach.

Structure Priority Approach to Weakly Coupled Systems

In most problems related to reactor safety, including LOCAs, the fluid involved is single-phased (15). Under this condition the problem can be further

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*The Westinghouse priprietary computer code MULTIFLEX (13,14) is based on a scheme similar to that of SOLA-FLX.

simplified. Based on the weak coupling condition, we can rewrite equation 4 as

$$[m]\{\ddot{q}\} + (i\nu + 1)[k]\{q\} = \{p_o \Delta A\} + \{p\Delta A\} \tag{6}$$

where p_o is the incident pressure and p is the induced pressure. Furthermore, since the induced field is small, the equations of fluid dynamics (1-3) reduce to a single acoustic equation for induced pressure p*:

$$\nabla^2 p + \frac{1}{C^2}\frac{\partial^2 p}{\partial t^2} = 0 \tag{7}$$

where C is the speed of sound in the fluid. Equation 6 is coupled to 7 by the same boundary condition whereby at the fluid-structure interface, the normal components of the fluid and structural velocities must be equal (equation 5). Using Bernoulli's equation, this can be reduced to

$$\left.\frac{\partial p}{\partial n}\right|_S = -\rho\,\frac{\partial^2 q n}{\partial t^2} \tag{8}$$

where n is the outward normal to the structural surface S.

One of the biggest achievements in fluid-structure interaction studies is the introduction of the hydrodynamic mass, or added mass, concept. It was shown by several authors (2-10) that equations 7 and 8 lead to the expression for induced pressure p in the form

$$p = -[M]\, q_n \,. \tag{9}$$

Substituting into equation 6, we obtain

$$\Big([m] + [M]\delta_{33}\Big)\{\ddot{q}\} + (i\nu + 1)\{q\} = \{p_o\Delta A\} \tag{10}$$

where δ_{33} denotes the fact that [M] acts only in the direction normal to the surface of the structure.

Equation 10 differs from the usual equation of structural dynamics only in the addition of the term $[M]\delta_{33}$ to the physical mass matrix. For this reason [M] is known as the hydrodynamic mass or added mass matrix. In general, [M] is time-dependent. If the compressibility effect is ignored, equation 7 reduces to the Laplace equation,

$$\nabla^2 p = 0, \tag{11}$$

and the resulting hydrodynamic mass matrix is time-independent (17). This greatly simplifies the analytical procedure of a fluid structure system, while at the same time providing a basis for comparison for the more general case in which the compressibility effect is included. Basically, it decouples the calculation of the hydraulic forcing function from the response analysis and accounts for the effect of fluid-structure interaction by a separately computed, time-independent, hydrodynamic mass matrix.

The incompressible fluid assumption is justified if the time for the acoustic wave to traverse a characteristic length of the fluid structure system is much smaller than the dominant modal period of vibration of the structure. In a PWR, this length can be taken as the length of the downcomer, or about 300 inches. Assuming that the velocity of sound is 3000 fps, this characteristic time is about 0.008 second. The dominant modal frequencies of a reactor system are between 10 and 30 Hz, corresponding to natural periods between 0.033 and 0.1 second. Thus, we see that for LOCA studies, compressibility effects are only marginally negligible.

LOCA-induced compressible fluid-structure interaction has been studied by the present authors (15), using a time history modal superposition method

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*For a more detailed discussion of the conditions for the acoustic approximation, readers are directed to reference 16.

commonly employed in structural dynamic analysis. The basic idea is that a fluid-structure system responds to a forcing function only at its natural modes, each with a characteristic natural frequency. Thus, instead of dealing with a continuously time-dependent hydrodynamic mass matrix, one is faced with a finite, discrete set of hydrodynamic matrices, each corresponding to a normal mode of the system, and each can be separately computed as in the case of incompressible fluid.

Contrary to the SOLA-FLX approach, the hydrodynamic mass concept accounts for the effect of fluid-structure interaction in the structural response part of the calculation and is termed a structural priority approach.

Fluid Vs Structural Priority Approach: Comparison of the Two Methods

Both of these methods attempt to solve fluid-structure interaction problems with minimum modifications to the existing thermal-hydraulic and structural analysis codes. When the fluid geometry is more complex, or when it is multi-phased, the fluid priority approach is the obvious choice. In most reactor-related accident problems, the fluid is single-phased and the geometry of its boundary is relatively simple. Furthermore, it is usually the structural response that is directly related to the integrity of the hardware. For these applications, the structure priority approach, along with the time history modal superposition method, offers the optimum computational efficiency and ease of application. Not only can the coupling effect be accounted for separately, but the hydraulic and structure codes do not need to interface at each time step as required in the SOLA-FLX system. While the SOLA-FLX system takes two iterations to obtain the first corrected value of structural response, the structural priority approach requires only one. Indeed, the whole idea resembles the original Stokes study on the acceleration of a solid in an infinite fluid medium.

COMPUTATIONAL METHODS

In any branch of analytical science, there are currently four principal methods of computing the results:

1. Closed form solutions, sometimes together with empirical parameters that can be determined only by experiment — Obviously, this method has only limited engineering applications. However, when a phenomenon is first studied, the model is usually highly simplified so that closed-form solutions, which are more amenable to physical interpretation, can be obtained.

2. Series expansion — This is a powerful tool in numerical analysis. The normal mode method commonly used in structural dynamic analysis is a series expansion method. In addition to yielding the final answer, the series solution method sometimes yields rich information on the physics of the problem.

3. Finite difference — This is probably the most commonly used numerical technique in acoustics and fluid dynamics and it was, until the advent of the finite element method, widely used in structural analysis as well.

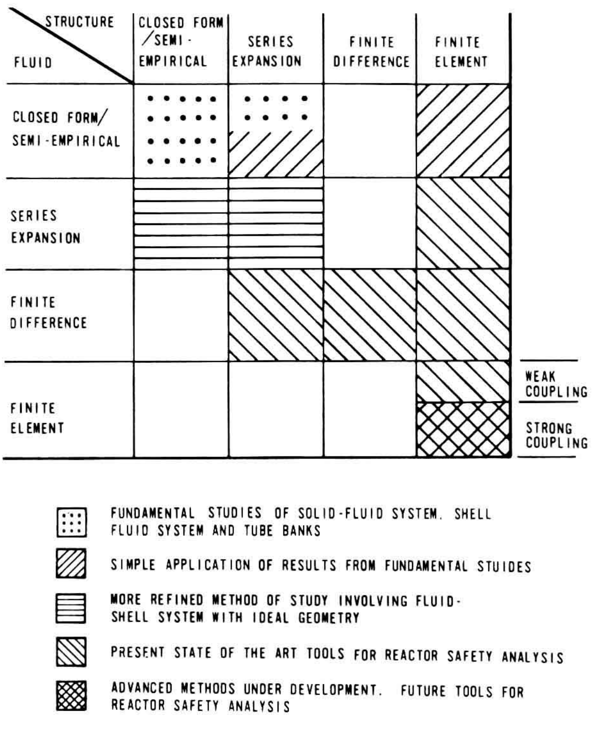
4. Finite element — This is undoubtedly the numerical tool in structural analysis. Virtually all commercially available codes in structural mechanics are based on this method, sometimes coupled with the series expansion technique. Application of the finite element method to acoustics and fluid dynamics problems, however, is still largely in the experimental stage.

In the numerical analysis of a fluid-structure system, any of these four techniques can be used in either the fluid or the structure part of the system, resulting in 16 possible methods of analysis. Table 1 shows this matrix of possible numerical methods, together with representative published papers or computer

Table 1. Computational Technique — Some Representative Work

Fluid \ Structure	Closed Form/ Semi-Empirical	Series Expansion	Finite Difference	Finite Element
Closed form/ semi-empirical	Stokes(1) Fritz/Kiss(3,4) Abramson/Kana(2) Krajcinovic(5) Au-Yang(7) Chen/Rosenburg(8)	Horvay/Bowers(9) Scavuzzo(18) Yeh/Chen(10)		McDonald(19)
Series expansion	Au-Yang(11) Housner/Herr(20)	Au-Yang(11) Ball/Citerley(21)		Au-Yang/Galford(15)
Finite difference		YAQU1/CYLDY2(22) FLUX/CYLDY2(23)	SOLA-FLX(12) FLEXWALL(24) F-FIX/FLX(25) MULTIFLEX(13,14)	PELE-IC(26) ICECO/WHAM(27) STEALTH/WHAM(28,29)
Finite element				Weak Coupling Nahavandi(30) Shaaban(17) Levy/Wilkinson(31) Yu(32), MacNeal(33) Brown/Hsu(34) Everstine(35) Kalinowski(36) Strong Coupling Belytschko/Kennedy(37 Donea(38), ANL(39)

Table 2. Computational Technique — Areas of Application



codes reflecting each technique. As one descends the main diagonal of the matrix, increasingly more sophisticated numerical techniques are encountered, until we reach the finite element-finite element technique. This method, which can be used to solve strongly coupled as well as weakly coupled problems, is undoubtedly the ultimate tool in solving a coupled fluid-structure problem. The latest development in reactor safety studies follows this approach. This is not to undervalue the contribution of researchers using other analytical techniques. Much of the understanding of a fluid structure system comes from closed form or series expansion analysis. In fact, in the extremely complicated area of tube bank dynamics, a large portion of the current work still belongs to square one: closed form/empirical solution for both the fluid and the structure.

The simplest method in the finite element-finite element approach is to compute the hydrodynamic mass matrix by a separate finite element computer code, and then input the hydrodynamic mass matrix into a general purpose structural analysis program for subsequent dynamic analysis (17,30,31). However, some authors employ the solid elements in general purpose structural codes to compute the mass matrix by inputting prescribed sets of "structural" properties, which in essence change the solid element into a fluid element. This approach, sometimes called the "mocked" fluid element approach, has been proven quite successful, in simple problems at least, with such general purpose structural codes as NASTRAN and SAP (32-36). Both of these approaches, which can be used with either the direct time integration or modal superposition method, are applicable only to weakly coupled systems.

For strongly coupled systems, the hydrodynamic mass concept does not apply. The equations of fluid dynamics and structural mechanics must be solved simultaneously. This is perhaps the most ambitious approach to the coupled fluid-structural dynamic problem and is currently being pursued by several authors (37,38).

Table 2 shows the areas of application of the different computational techniques. Note that not all 16 possibilities have been explored. For example, the finite difference method is seldom used to solve the structural mehcanics part of the problem except in certain cases when it is used to solve the equations of fluid dynamics. Then this method is used to solve the structure problem also, to facilitate matching of the fluid-structure boundary. On the contrary, an analyst seldom chooses the finite element method to solve the fluid mechanics part of the problem unless he also uses the same method to solve the structure problem.

EFFECT OF FLUID-STRUCTURE INTERACTION ON NSS

Having described the different methodologies in analyzing a coupled fluid-structure system, we now proceed to discuss their impact on NSS design in the three areas of flow-induced vibration, seismic excitation, and LOCA. Only a few years ago, NSS designers calculated the forcing function on and the structural response of an NSS separately without taking into account the coupling effect between the fluid and the structure. It is only within the past few years that fluid-structure interaction has been recognized as having a large effect on the structural response and must be included in the analysis.

Flow-Induced Vibration

It has long been recognized that under certain conditions, steam generator tube banks form strongly coupled systems with the surrounding fluid. So much has been published in this area that it is virtually a separate subject and thus is not discussed here. The reader is directed to an excellent survey report by Nahavandi and Chen (6) and a textbook by Blevins (40).

We discuss instead reactor internal components, such as the thermal shield, core support barrel, and the core basket or surveillance specimen holder tube. These components are susceptible to flow-induced vibrations caused by turbulent eddies and reactor coolant pump-induced acoustic waves. Figure 1 shows the power spectral density of the dynamic pressure acting on the thermal shield of a commercial PWR. One can readily see that the forcing function consists of a continuous spectrum that increases exponentially with decreasing frequency, and a discrete set of spectra at the coolant pump blade-passing frequencies. Since the latter is discrete, the components can be designed to avoid resonance with the

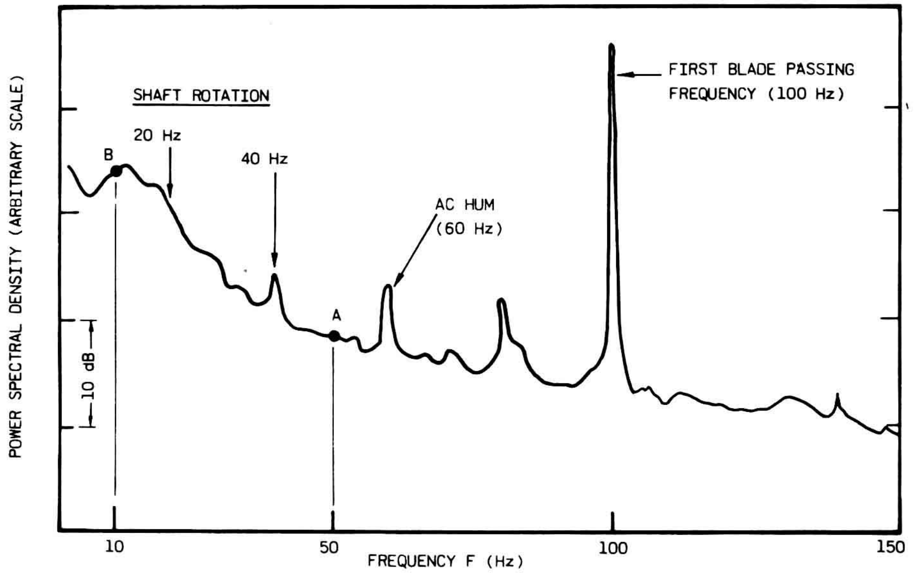


Figure 1. Dynamic Pressure in a PWR Downcomer

A structure with a natural frequency of 10 Hz will be excited by a much higher forcing function (B) than a structure with a natural frequency of 50 Hz (A).

reactor coolant pump during normal operations. For this, accurate determination of the components' natural frequencies is necessary, which in turn requires that fluid-structure interaction be included in the analysis.

Since turbulence energy has a continuous spectrum, one cannot design the components to avoid being excited by it. However, it is important to be able to estimate the turbulence-induced vibration amplitudes of different reactor internal components. This can be most easily carried out by the structural priority approach, assuming that the fluid and the structural component form a weakly coupled system. This is true if the pressure induced by structural motion can be linearly superimposed onto the incident pressure caused by turbulent eddies and the coolant pump (see equation 6).

Figure 2 depicts a test setup to experimentally verify the superposition rule of the incident and induced pressures for small-amplitude, flow-induced vibration of cylindrical shell structures. First, a thin-walled test cylinder was coaxially placed inside another thick-walled cylinder. Water was pumped into the annular gap between the two cylinders through four inlets 90 degrees apart around the circumference and one quarter of the axial length of the cylinders from the top. The water then made a 90-degree turn, flowed down the annular gap, and exited through an opening at the bottom of the outer cyilnder. During the test, the inner cylinder was also filled with water. The dimensions of the stainles steel inner cylinder were as follows: length 111.76 cm (45 in.), OD 55-88 cm (22 in.), thickness 0.3175 cm (0.125 in.). The dimensions of the aluminum outer cylinder were length 111.76 cm (45 in.), ID 60.96 cm (24 in.), and thickness 2.54 cm (1.0 in.). The outer cylinder was further reinforced by four heavy aluminum rings equally spaced axially to ensure rigidity. Dynamic pressure transducers and accelerometers were installed at various locations on both cylinders.

The test cylinder was then replaced by one of identical outer dimensions but with a wall thickness of 2.54 cm (1.0 in.) to simulate a rigid shell. Pressure transducers were installed at identical locations, and water was pumped through the loop at the same rate as in the thin-shell test.

Figure 3 shows the power spectral densities (PSDs) of the dynamic pressure recorded by one of the transducers during these two tests. This transducer was

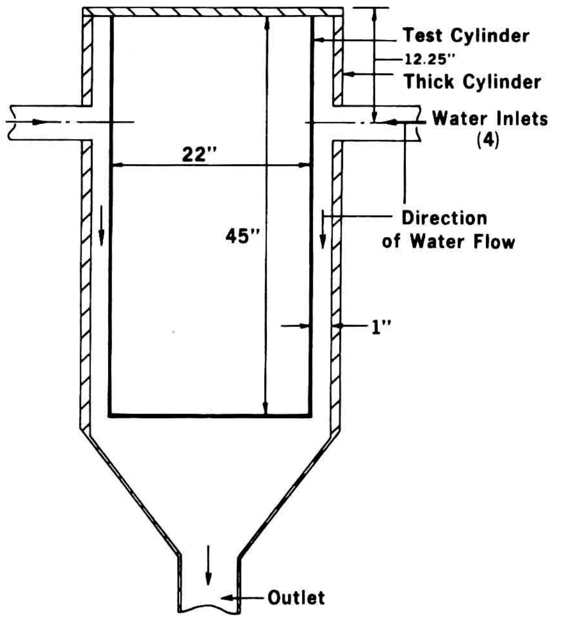


Figure 2. Test Setup to Verify Superposition of Incident and Induced Pressure

installed on the inner cylinder just opposite the center of one of the inlet ducts. In the thick-shell test, the pressure PSD consisted of a continuous, exponentially decaying function of frequency typical of turbulence-induced pressure. In the thin-shell test, the pressure PSD was almost the same except that superimposed ont the continuous spectrum was a series of small discrete peaks at frequencies corresponding to the in-water natural frequencies of the thin shell. These spectral peaks corresponded to the pressure induced by the vibrating thin

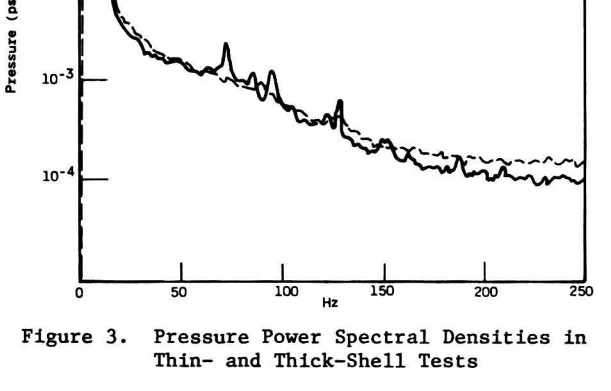


Figure 3. Pressure Power Spectral Densities in Thin- and Thick-Shell Tests

shell and were suppressed in the thick-shell test because of the much smaller amplitude of vibration. PSD readings from other pressure transducers showed the same trend.

Using the weak coupling, structural priority approach, several authors have shown that the natural frequencies of large shell structures, in particular, are drastically reduced by narrow fluid gaps surrounding them (7,9). Laboratory and field measurements agreed with their analytical derivation. As an example, when measured in air, the core support barrel of a typical PWR may have a fundamental shell mode frequency of 50 Hz. When measured in the field with coolant in the downcomer, this may decrease to 10 Hz. Since the turbulence-induced forcing function increases rapidly as the frequency decreases, this means that fluid-structure interaction will cause the core support barrel to respond to a much more intense forcing function.

In second generation PWRs, the thermal shield is replaced by the core basket assembly inside the core support barrel. One reason for the change is undoubtedly the flow-induced vibration problems experienced by the thermal shields of early PWRs. It was thought that by moving the thermal shield to the inside of the core support barrel, it would be protected from excitation due to turbulence. Yet because the two components are fluid-elastically coupled, they respond to external excitation as a whole. As a result, whether the thermal shield is inside or outside the core barrel has only a small effect on its response to turbulence- or pump-induced pressure pulses.

LOCA-Induced Response

In steady-state flow-induced vibration, the effect of fluid-structure interaction manifests itself entirely in the frequency shifts of the structure, causing the latter to respond to different energy bands in the spectrum of the forcing function. In transient response, as that induced by a LOCA, fluid-structure interaction affects the final response in two ways: frequency shifts as mentioned above, and the direct effect due to an increase in the effective mass of the structure. If the forcing function is a true impulse of infinitesimal duration, its power spectrum will be a constant of frequency. The response of the structure to impulsive forcing function is therefore not affected by frequency shifts, but by the effective masses (physical and hydrodynamic) of the system. It has been mentioned before that a PWR LOCA can be treated as a weakly coupled system involving a single-phase fluid, in which the forcing function can be separately computed without taking into account the effect of coupling. Since the spectra of this forcing function vary from case to case, there is no universal trend in the effect of frequency shifts on the structural response as in turbulence-induced vibration. Therefore, we can only discuss the effect of hydrodynamic mass loading on the structural response. For this, we use impulsive forcing function together with a highly simplified reactor model in which the core and the reactor vessel are assumed to form a double pendulum with only two degrees of freedom. Unit-impulsive moments are assumed to act on the core and the vessel in either the same or opposite directions.

The relative displacement between the core and vessel, $\theta_a - \theta_b$, is a measure of the moment induced by the impulses at the upper flange of the core. Figures 4 through 8 show the time history of $\theta_a - \theta_b$ due to impulses of different phases. It is immediately apparent that as far as the internal load is concerned, hydrodynamic mass loading has the greatest effect when the forcing functions acting on the two structural components are out of phase. In the present benchmark study, hydrodynamic masses between the core and the vessel reduce the internal load by 28% (Figure 2), 15.5% when the reactor vessel is constrained (Figure 8). When the force acts only on the core or only on the vessel, Figures 5 and 6 show that it reduces the internal load by only about 14%, which is approximately the same as the case in which the reactor is constrained. However, when the forces acting on the core and on the vessel are in phase, then hydrodynamic mass loading has very little effect on the internal load, as shown in Figure 7. This is physically explainable: The in-phase force favors the in-phase mode, which is affected only slightly by hydrodynamic effect. The out-of-phase forces favor the out-of-phase mode, which is affected significantly by hydrodynamic mass coupling.

The rotation of the reactor about its support, θ_b is a measure of the external nozzle load induced by the impulses. Figures 9 through 12 show the time history of θ_b. Unlike the case of internal load, hydrodynamic mass has only a small effect on the nozzle load, except in the case when the force acts only on the core. When this happens, the hydrodynamic mass effect reduces the nozzle load by about 16%. In all other cases, the change is no more than a few percent and is probably caused by interaction of the two modes (which have different frequencies with hydrodynamic mass coupling) rather than hydrodynamic mass loading.

Based on these studies we expect that if, during the initial moments of peak loads, the LOCA induces moments on the core and vessel that are of opposite signs, hydrodynamic masses will have a significant effect on the internal load while leaving the nozzle load relatively unaffected. On the other hand, if the moments are of the same sign, neither the nozzle load nor the internal load will be affected significantly. If the LOCA force acts mainly on the internal only, both both the nozzle and internal loads will be moderately reduced by including fluid-structure interaction in the analysis.

These conclusions are for impulsive forcing functions only. Actual LOCA forcing functions contian spectral peaks. Thus, in practice, resonance effects may or may not reverse this trend.

Response to Seismic Excitation

In comparison with flow-induced vibration and LOCA-induced response, seismic analysis is even further from being an exact science. Any present seismic analysis is no more than an attempt to obtain a crude estimate of the upper bound to the response. The reason is that the input "forcing function," i.e., the response acceleration spectrum, is no more than an envelope of all the measured ground acceleration.

Figure 13 shows a typical response acceleration spectrum for 1% damping. Unlike flow-induced vibration and LOCA-induced response, this "forcing function" does not process sharp peaks and valleys. The reason, of course, is that the response spectrum represents the envelope of these peaks. Thus, we expect that frequency shifts would not give rise to sharp resonant effects as in the other two cases. This is especially true since only the beam mode response is of importance in seismic analysis. Table 3 shows the three lowest beam modes of a typical PWR core/vessel system. One can readily see that the changes in the coupled frequencies due to fluid-structure interaction, unlike the case of shell modes, are only moderate and do not significantly affect the amplitude of the spectrum (Figure 13) to which the structure will respond. Therefore, the only other effect due to fluid-structure coupling is due to fluid loading.

Table 3. Natural Frequencies of a Reactor System

	Coupled frequencies, Hz	
Mode	No fluid coupling	With fluid coupling
In-phase rocking	11.9	11.6
Out-of-phase rocking	21.9	18.8
Translational bending	31.0	27.8

From our previous study of LOCA-induced response, we conclude that fluid loading is significant only if the forcing functions acting on the core and on the vessel are in opposite directions. In response to ground acceleration, however, the "forcing functions" on the core and on the vessel are necessarily in the same direction. Hence, we do not expect that fluid loading has a significant effect on the structural response to ground motion, at least with regard to the beam mode.

In conclusion, then, one can say that fluid-structure interaction has only small effects on the response of a reactor system to ground motion.

CLOSURE

The phenomenon of fluid-structure interaction is reviewed in general and in particular with regard to its impact on NSS design. The difference between strongly and weakly coupled fluid-structure systems is discussed, and in the latter case, it is shown that one can solve the problem in a sequential manner, from either a fluid mechanics or a structural dynamics point of view. It is pointed out that in most problems related to nuclear reactor safety, the weak coupling assumption is justified, thus permitting existing computer codes in structural and thermal-hydraulic analysis to be used with minimum modification. Finally, the effect of fluid-structure interaction on flow-induced vibration, seismic response, and loss-of-coolant accident analysis of a PWR is discussed.

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