AN EVALUATION OF FLOW-INDUCED VIBRATION
PREDICTION TECHNIQUES FOR IN-REACTOR COMPONENTS

by

T. M. Mulcahy and P. Turula
Components Technology Division

NOTICE
This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Energy Research and Development Administration, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

Base Technology

May 1975

DISTRIBUTION OF THIS DOCUMENT UNLIMITED

ABSTRACT

Selected in-reactor components of a hydraulic and structural dynamic scale model of the U. S. Energy Research and Development Administration experimental Fast Test Reactor have been studied in an effort to develop and evaluate techniques for predicting vibration behavior of elastic structures exposed to a moving fluid. Existing analysis methods are used to compute the natural frequencies and modal shapes of submerged beam and shell type components. Component response is calculated, assuming as fluid forcing mechanisms both vortex shedding and random excitations characterized by the available hydraulic data. The free and force vibration response predictions are compared with extensive model flow and shaker test data.

INTRODUCTION

Although higher reactor coolant flow rates are beneficial from the standpoint of heat transfer and power efficiency, they require improved reactor designs from the standpoint of flow-induced vibrations (FIV). Many past structural failures of reactor components have occurred because of inadequate design for FIV [e.g., 15]. Currently a high level of concern is being given to reactor FIV problems. The work discussed herein represents some of the effort to assure an adequate design for the U. S. ERDA's Fast Flux Test Facility (FFTF) liquid metal breeder reactor, which is currently under construction.

The Hydraulic Core Mockup (HCM), a 0.285 geometric scale model of the FFTF was designed, constructed, and tested by the Hanford Engineering Development Laboratory to verify experimentally the hydraulic design and assess the potential for flow-induced vibrations of components in the upper plenum of the reactor vessel. To determine the vibrations, the components were instrumented with (piezoelectric) accelerometers at locations where extreme motion was expected.

Argonne National Laboratory was requested to predict analytically, utilizing existing FIV analysis methods and information, the free and forced vibration response of most of the instrumented upper plenum HCM components. The analytical predictions made were not envisioned as a cross check of experimental data, but rather a basis for evaluating FIV analysis methods when applied to typical reactor components. Because the flow-induced vibrations of a body generally are highly nonlinear and dependent upon body geometry, most of the analytical prediction methods [11] are based on experimental data and deal with simplified flow fields and simplified solid boundary geometries: uniform two-dimensional or axisymmetric

flow fields and isolated long circular cylinders are typical. Although many upper plenum reactor components are circular cylinders, they usually are not isolated nor infinite in length. Furthermore the flow fields usually are not uniform or axisymmetric. As a result, many simplifying assumptions had to be made to predict the FIV response of reactor components. Because the analysis methods have questionable applicability, the comparisons of the predictions with test data were desirable to provide a basis for confidence in existing analysis methods or to provide the basis for further development of analysis methods.

The approximate radial location of the reactor components which were analyzed and a schematic of a typical radial flow field cross section are shown in Fig. 1. The arrows indicate the general direction of the flow. Their length is proportional to the fluid velocity. Of course, the flow is not axisymmetric; the components do not lie in the same plane; and more than one Temperature-Liquid Level Monitor (T/LLM), Closed Loop Instrument Line (CLIRA), Control Rod Drive Mechanism (CRDM), and Low Level Flux Monitor (LLFM) are in the reactor. Only one thermal liner shell exists. For the purpose of analysis, information [4] was available for defining the local flow field around a specific component.

VIBRATION ANALYSIS

The determination of the natural frequencies of the individual components is an important step in FIV reactor design. Presently, one of the most effective means of designing against the occurrence of severe flow-induced vibration problems is to separate structural natural frequencies from expected excitation frequencies. Two main difficulties exist in making accurate predictions of the free vibrations of reactor components. First, the boundary conditions often are difficult to define. Because the components' supports usually provide thermal expansion clearance fits, the boundary conditions are amplitude dependent. The direct solution of a problem with nonlinear boundary conditions is formidable and is the subject of current research [12]. Herein linear structural analysis and limiting sets of boundary conditions are employed in an attempt to provide bounds on the actual structural response. Second, the coolant fluids utilized in most U.S. experimental and commercial reactors are relatively dense and their mass cannot be neglected in FIV analysis as is often done when air is the coolant fluid. As for structures vibrating in air, the damping effects of fluid generally are negligible in determining free vibrations [13].

Methods for the prediction of the free vibration of elastic structures in a vacuum are well developed. They have been employed with much success for structures vibrating in light fluids, such as air. Analytical solutions are available for many simple body geometries and computer codes, such as NASTRAN, provide a means of analysis for complex body geometries. Analysis methods for the free vibrations of elastic structures in dense fluids, such as water, are not as well developed. Although the theory is developed, at least for nonflowing fluids, the analysis is complicated greatly because of the strong interaction between the fluid and structure. Further, the

vibrations of adjacent structures may be coupled by dense fluids, whereas the coupling in light fluids is less likely except when the separation distances are small.

Typically the natural vibration frequencies of isolated structures immersed in water are 5-40% lower than those of the same structure in a vacuum. However, when two structures are separated by small fluid gaps, the differences of frequencies in a vacuum and water can differ by very large factors, depending on the size of the fluid gaps [7]. Some analytical fluid-solid interaction solutions exist for simple structures in dense fluids [11], but more solutions need to be developed. Computer codes, such as NASTRAN, do provide means to solve the combined fluid-solid interaction problem, but the solutions are time consuming and costly. Further, more confidence in numerical fluid-solid interaction results needs to be gained by comparisons between experimental and analytical solution data.

As a result of the lack of direct utilitarian methods for solving the fluid-solid interaction problem, the free vibrations of components in a dense fluid most often are determined approximately. The component is assumed to be vibrating in a vacuum and additional mass, "added mass" is assigned a priori to account for the effect of the surrounding fluid. The added mass effect is approximated utilizing available analytical results. This approach is employed herein.

Prediction of flow induced vibration response is limited by the state-of-the-art in describing the fluid forcing functions associated with fluid excitation mechanisms and the damping provided by the fluid. Current research indicates that often the excitation mechanisms are highly complex and non-linearly coupled with the motion of the solid [2]. Some research has been directed at characterizing fluid damping [13]; but the results are not conclusive and difficult to interpret, especially when the fluid is flowing. Since well defined and agreed upon descriptions of the forcing functions and damping mechanisms are not available, herein linearized approximations are employed in an attempt to bound a components response.

The linearized representation of the fluid forcing functions employed in the present analysis are given in subsequent sections, along with comments on their applicability for reactor component analysis. The fluid damping is assumed to be describable by an equivalent viscous damping ratio. A similar assumption is considered adequate for structures vibrating in air [14]. Much testing has been done on structures in air and adequate information exists for specifying damping ratios. However, specific information on the equivalent viscous damping ratios for particular components in dense fluids usually is not available. The only recourse in a forced vibration analysis is to assume a range of damping ratios which lead to bounds on the displacements. At least, typical air damping ratios should lead to upper bounds on the displacements of the structure.

FLUID EXCITATION MECHANISMS

Many excitation mechanisms could be responsible for HCM component vibration. Some examples are cross flow vortex shedding, parallel flow turbulent boundary layers, structural borne noise, fluid borne noise, wakes from adjacent components, fluidelastic coupling between adjacent components, etc. Vortex shedding was of major concern in the HCM because of existing flow conditions in the upper plenum, and because of the extreme component vibrations which usually accompany such excitation. The scale modeling of the HCM was chosen, in particular, to simulate cross flow vortex shedding.

Although all the mechanisms cited above were of concern, in most cases not enough information has been developed for use in a forced vibration analysis. An analytical characterization was possible for only the first two mechanisms, cited above, when each was assumed to act alone. Even then, the forcing functions were based on gross assumptions. Since the response of a member subject simultaneously to cross and parallel flows is not known, no attempt was made to superimpose results obtained separately.

Cross Flow

For rigid stationary circular cylinders in uniform cross flow, a widely accepted empirical representation of the fluid flow induced lift direction force per unit length p has evolved [1,2] for a large range of Reynolds numbers, N_R .

$$\text{For } 2 \times 10^4 < N_R \lesssim 4 \times 10^5$$

$$p = \frac{C_L}{2} \rho_f v^2 d \sin 2\pi f_v t \quad (1)$$

where v is the flow velocity, d the tube diameter, ρ_f the fluid mass density, f_v the vortex shedding frequency, and C_L the lift coefficient.

The f_v is related to v via the Strouhal number N_S

$$N_S = \frac{f_v d}{v} \quad (2)$$

where $N_S \approx 0.2$ for the N_R range specified. The C_L varies with Reynolds number and particular experimental setups, but it is generally observed to be less than 2.0 and most often less than 1.

For $4 \times 10^5 \lesssim N_R \lesssim 6 \times 10^6$, the wake turbulence appears random in character over a band of frequencies [3], the width of which depends on the Reynolds number, surface roughness, free stream turbulence, etc. Dominant N_S have been found to be less than 0.3, but some data exists for which $N_S \approx 0.45$ [1]. The average lift coefficient has been observed to be smaller by a factor of 1 to 10 for the latter Reynolds number range. At present, there exists questions as to whether the lift force is random or deterministic. Some evidence exists [16] that indicates the pressure fluctuations and resultant lift force are deterministic. That is, there may be no relation between the wake turbulence and the lift force as for the lower Reynolds number range.

For vibrating deformable circular cylinders in a uniform flow field, the fluid forces due to vortex shedding are the subject of current research for all N_R . Much ambiguity exists in the literature. However, the forces for an oscillating cylinder have been observed to be similar in magnitude to forces on a corresponding stationary cylinders, unless the frequency of the vortex shedding and a structural natural frequency are close in value. If the fluid and structural frequency coincide, the force is greatly magnified, the phase relation between the force and the cylinder displacement is very sensitive to flow velocity, the length of the cylinder over which the vortex shedding is in phase is greatly effected, etc. None of these variations nor the possibly random forces for $4 \times 10^5 \lesssim N_R \lesssim 6 \times 10^6$ have been modeled well enough to include in an analysis for predicting vibration response. Such fundamental work is needed since several of the components may be in the higher Reynolds number range.

Little or no information exists for vortex induced forces for members which are yawed to the direction of the flow, members which are subject to nonuniform flow, members which are close to adjacent bodies, members which vary abruptly in cross section, etc. These conditions are common in the HCM. Obviously then, gross assumptions had to be made in representing forces due to vortex shedding.

For making predictions under cross flow conditions, p was assumed in the form of (1) with $C_L = 2$ and $N_S = 0.2$ to 0.3 . The velocity was assumed uniform at the maximum velocity measured [4] over the length for which any cross flow was apparent. Implicit in these assumptions was that vortex shedding was in phase along the length of the component where cross flow was assumed to exist. The p assumed was expected to give conservative predictions of response except where vortex shedding and structural natural frequencies were close in value or random vortex shedding was the source of excitation.

Parallel Flow

At present, analytical prediction methods are available only for uniform circular rods subject to uniform flow in a symmetric annular region along the entire length of the rod. Excitation is due to the pressure fluctuations developed in the turbulent boundary layer around the rod. Because of the restrictive nature of the solutions available, all analyses that were made for the HCM components included gross assumptions regarding the component geometry and flow conditions. As for cross flow, little or no information exists for more appropriate cases where the flow is nonuniform along or perpendicular to the member, where members are close to other bodies, where members have varying cross sections, etc. In an attempt to predict conservative results the uniform geometries assumed always were more flexible and less massive than that of the actual component, and the velocity of the uniform parallel flow assumed was always greater than any velocity observed in model tests [4].

Specific HCM pressure disturbances due to upstream bodies or system noise, so-called far field effects, were not known and could not be included in an analysis. Characterization of forces due to far field effects in the HCM would require considerably more experimental effort than the direct measurement of response. An attempt to bracket the response of the HCM components was made by employing two representations of the pressure fluctuations [6,7]. The experimental apparatus used to characterize the pressure fluctuations in [7] was designed to eliminate all far field effects and all local disturbances except those due to the uniform flow. The expression developed for the root mean square deflection in a single vibration mode

$$y_{\text{RMS}}(x, v) = \frac{0.0180 K(d d_n)^{1.5} v^2 \phi_1(x)}{\ell^{0.5} f_o^{1.5} (M_A + m) (\zeta_o + a_1 v + a_2 v^2)^{0.5} \left[1 - \frac{\beta_1 M_A v^2 \ell^2}{EI} \right]^{0.75}} \quad (3)$$

where

$$K = 2.56 \times 10^{-3} \text{ lb-sec}^{2.5}/\text{ft}^{5.5}$$

$$a_1 = 2.4 \times 10^{-4} \text{ sec}/\text{ft}$$

$$a_2 = 3.4 \times 10^{-6} \text{ sec}^2/\text{ft}^2$$

$$\beta_1 = 0.101 \text{ for simply-supported rod and } 0.0246 \text{ for a fixed-fixed rod}$$

$$\phi_1(x) = \text{mode shape}$$

$$\zeta_o = \text{damping in stagnant fluid}$$

$$v = \text{flow velocity, ft}/\text{sec}$$

$$\ell = \text{length, in.}$$

$$M_A = \text{added mass of fluid per unit length lb-sec}^2/\text{in.}^2$$

$$m = \text{cylinder mass per unit length lb-sec}^2/\text{in.}^2$$

$$f_o = \text{fundamental vibration frequency in static fluid, Hz}$$

$$E = \text{elastic modulus, lb}/\text{in.}^2$$

$$I = \text{cross sectional moment of inertia, in.}^4$$

$$d = \text{cylinder outside diameter, in.}$$

d_n = hydraulic diameter, in.

y_{RMS} = displacement in inches RMS

was expected to give an estimate of response for a quiet system. Predictions based on (1) of [6]

$$y = \frac{5d}{\alpha_1^4} \left[\frac{u^{1.6} \epsilon^{1.8} N_R^{0.25}}{1 + u^2} \right] \left[\left(\frac{d_n}{d} \right)^{0.4} \right] \left[\frac{\beta^{2/3}}{1 + 4\beta} \right] \quad (4)$$

where

α_1 = first mode eigenvalue

$\epsilon = \ell/d$

$\beta = M_A / (M_A + m)$

$u = (M_A/EI)^{1/2} v \ell$

were expected to give an estimate of response when significant far field noise exists, since significant system disturbances were present in experimental apparatus employed in characterizing (4). For the flow velocity and component geometries assumed, the range of response predicted via (3) and (4) should bracket the actual component response due to parallel flow.

ADDED MASS AND DAMPING

The effect of adjacent bodies on the vibrational response was of concern for the thermal liner, CLIRA, and the CRDM. In the case of the thermal liner shell, no applicable information was available to predict the expected large effect on the vibration response of the small annular fluid filled gap created by the surrounding main vessel. An analysis is outlined below which constitutes one approach for estimating the added mass effect of the gap and the thermal liner frequencies.

The CRDM is surrounded by the circular cylindrical guide tube mounted on the instrument tree, the core duct below the water level, and the intervening fluid filled gaps. Because of the variation in the CRDM's diameter, the width of the gap varies along the length. The added mass/unit length was approximated by the relation [7]

$$M_A = C_M \rho_f v_D \quad (5)$$

where

$$C_M = \left(\frac{1 + \gamma^2}{\gamma^2 - 1} \right)$$

and v_D is the volume/unit length of the fluid displaced by the stationary boundaries of the CRDM, and for example γ is the ratio of inside diameter of the control rod guide tube to that of the local diameter of the CRDM. Relation (5) was developed for determining the added mass for a rigid, uniform diameter rod oscillating in a rigid, uniform diameter cylindrical cavity filled with fluid. The same relation (5), with $\gamma \rightarrow \infty$, was employed to estimate the added mass of the other members which essentially were immersed in an infinite fluid.

At the time of the analysis HCM shaker test data had not yet been made to determine either the natural frequency or damping of HCM components.

Therefore, equivalent viscous modal damping of 0-5% critical damping was assumed, depending upon the component. Damping of 1% was considered small whereas 5% was considered large.

PREDICTED HCM COMPONENT RESPONSE

Since the HCM test facility was designed to operate at 250°F, the following values of material properties were assumed for all calculations:

[9,10]

Elastic modulus of Type 304 SS:	$27.5 \times 10^6 \text{ lb/in.}^2$
Density of Type 304 SS:	0.284 lb/in.^3
Kinematic viscosity of water:	$2.3 - 2.8 \times 10^{-6} \text{ ft}^2/\text{sec}$
Density of water:	0.0341 lb/in.^3

The response predictions for each component will be presented separately. Except where noted, forced vibration response is made for normal operating flow conditions.

T/LLM

Essentially the T/LLM is a tube cantilevered from the false head into the reactor upper plenum water pool and is subject to both cross flow and parallel flow induced forces. See Figs. 1 and 2(a). The stiffness and mass of the 10.5 inches of the component immediately below the false head varies in step like fashion along the length because of four reductions in outside diameter of the tube and a perforated region around the water level. For the analytical model the moment of inertia of the perforated region was assumed to be that of a uniform tube having the metal volume and inside diameter of the T/LLM. Several concentrated masses simulating instrumentation also are present. Their masses were averaged over the step region in which they were located. The remaining 44.5 in. of the tube has a uniform stiffness and mass.

For both cross flow and parallel flow, the tube was expected to vibrate predominantly at its lowest natural frequency. A Rayleigh analysis was made to obtain an approximation to the fundamental bending frequency of the tube. The mode shape assumed was

$$y = 1 - \cos \frac{\pi x}{2\ell} \quad (6)$$

where y is the lateral deflection, ℓ the total length of the component, and x the axial position with $x = 0$ at the fixed end. Static analysis of the false head subjected to a concentrated moment showed that it could be considered rigid with respect to T/LLM deflections. The fundamental natural frequency was found to be 19.4 Hz when account was made for all the variations in component cross section, mass of the contained fluid, and apparent mass of the surrounding fluid, given by (5) with $\gamma = \infty$. Also, the natural frequencies were determined for a more conservative model to be used in forced response calculations. For a uniform beam, with stiffness and mass properties of the last 44.5 inches of the actual component, the lowest natural frequency was found to be approximately 15 Hz.

The maximum cross flow velocity observed [4] in the region of the T/LLM was 5 ft/sec over the 6 inches immediately below the suppressor plate. For the 2.25 inch diameter of the component in the region of cross flow the Reynolds number

$$N_R = \frac{vd}{\nu} \quad (7)$$

equals 4×10^5 , where ν is the kinematic viscosity of water. The vortex shedding frequencies calculated according to (2) were in the range of 5.3 - 8.0 Hz. Since these forcing frequencies are greatly separated from the predicted lowest natural structural frequency, a very small response to vortex shedding excitation was expected. The maximum amplitude at the accelerometer was determined to be approximately 3 mils, assuming that a concentrated force acted 17 inches below the fixed end of the conservative model discussed above. The concentrated force was chosen equal to the resultant of a distributed load p , given by (1), acting over the six inches where vortex shedding was expected to occur.

A parallel flow response prediction was made for the 15 Hz uniform beam discussed above, assuming a velocity of 4 ft/sec [4]. A computer program [7] was employed to solve the cantilever beam problem. Methods in the program are the same as those employed in deriving (3). The extremely small deflection of 0.02 mils RMS was predicted for 1% of critical damping.

LLFM

The LLFM in the HCM consists of a uniform stiffness tube which extends from the false head to the level of the top of the core making an angle of approximately 8.5 degrees with the vertical. See Figs. 1 and 3(a). Mass has been added to the lower ~ 56 inches of the HCM LLFM to simulate instrument mass and that part of the FTR LLFM below the level of the core which is not included in the HCM LLFM model. Fluid surrounds the lower ~ 55 inches of the LLFM and the added fluid mass was determined from (5) with $\gamma = \infty$. Static analysis of the false head indicated it could be considered rigid in comparison to the LLFM. Therefore the LLFM was assumed fixed at the false head. A pinned and shimmed connection is provided for the bottom of the HCM LLFM at the level of the core. The model used to calculate natural frequency and response, shown in Fig. 3(b), includes a pinned bottom connection.

The NASTRAN computer code (Level 15) was utilized to calculate the structural natural frequencies and mode shapes. For the model shown in Fig. 3(b), the three lowest frequencies were found to be ~ 30 , 99, and 208 Hz.

Flow-induced loading appeared [4] to be limited to that due to cross flow immediately below the suppressor plate, with the remainder of the length of the LLFM immersed in essentially stagnant fluid. A maximum cross flow velocity of 80 in./sec across the 2 inch diameter of the LLFM

resulted in a Reynolds number of 4.75×10^5 , according to (7), which corresponded to a vortex shedding frequency of 8 to 12 Hz, according to (2), assuming a Strouhal number of 0.2 to 0.3. This forcing frequency was well removed from the lowest structural natural frequency, and response was expected to be small. Assuming that p given in (1) acted over the 6 inches immediately below the suppressor plate, maximum deflections of less than 1 mil were predicted at $x = 32$ in. by NASTRAN code analysis for the model in Fig. 3(b) assuming zero damping. Since the forcing and structural frequencies are largely separated, any damping value will give similar results.

CLIRA

The CLIRA consists of several different sized tubes joined together to form a beam which is seated in the core support structure and extends up through the core, the upper plenum water pool, and the false head where it is rigidly connected to the top of a stand pipe. See Fig. 1 and 4(a). Since internal coolant flow in the CLIRA was provided by a separate heat exchanger, the CLIRA was assumed full of water to the top of the stand pipe. The added mass of the CLIRA above the core and below the water level was determined via (5) with $\gamma \rightarrow \infty$. Below the core, the added mass of the adjacent ducts also was determined via (5) by assuming the flow tube was surrounded by a rigid circular cylinder. The radius of the cylinder was assumed equal to the average radius to the adjacent hexagonal ducts. No information was available to account for the partial openings or gaps between surrounding ducts.

Lateral motion restraint was assumed to exist at the top of the core and ~ 10 inches below the top of the core where load pads contacted adjacent fuel ducts. Additional lateral restraint was assumed to be provided by interaction between tube No. 3 and the standpipe, since clearances between the two are only 15 mils (on the radius) along most of the standpipe length. The support conditions chosen for the analytical modal analyses are shown in Fig. 4(b).

The NASTRAN computer code was used to obtain the natural frequencies and mode shapes for various support conditions. All cases investigated assumed a simple support between the CLIRA and the core support structure, and a fixed support between the false head and the standpipe. Boundary conditions were modified by changing the stiffness, k , of the springs simulating the core restraint, or by including or not including the optional simple support simulating contact between the standpipe and tube No. 3. The first four modal frequencies determined are given in Table 1. Obviously, the free and therefore the forced vibration response is very dependent upon the actual boundary conditions.

Forced vibration analysis was performed separately for both the apparent cross and parallel flow loading conditions which have been observed in model studies [4]. Crossflow induced loading due to vortex shedding appeared to be limited to the approximately 6 inch fluid layer immediately below the suppressor plate where the maximum flow velocity was approximately 5 ft/sec. Nearly parallel flow occurs over the rest of the CLIRA in the upper plenum pool. The associated Reynolds number of the crossflow, according to (7), was calculated to be 2.46×10^5 for the CLIRA tube No. 2 diameter of 1.360 inches. This corresponded, according to (2), to a vortex shedding frequency range of 8 to 14 Hz.

Since the frequency of vortex shedding was well removed from the natural frequencies corresponding to the support conditions of Cases 3 to 5 given in Table 1, the corresponding forced response was expected to be small. For Cases 1 and 2, the magnitude of the response was expected to be considerably higher, since the forcing frequency is at or near the lowest structural natural frequency. Because no information was available on damping, conservative values of effective modal damping (1% to 2% of critical damping) were assumed. The distributed lift force per unit length

p given by (1) was assumed to act over the 6 inch length of the CLIRA immediately below the suppressor plate. The calculated maximum response is given in Table 2 for selected damping values. The form of the general response was assumed to be the superposition of the first nine mode shapes. Since the lowest structural natural frequency of Case 1 was lower than the vortex shedding at normal flow conditions, Case 1 response was calculated at the below normal flow velocity corresponding to synchronization of the vortex and structural natural frequency. Case 5 response can be estimated from the above results to be on the order of several mils for similar damping assumptions.

A conservative uniform beam idealization of the CLIRA for parallel flow analysis was made, and both (3) and (4) were used to calculate response. The stiffness and mass (including contained and added fluid mass) of the uniform beam is defined to be that of tube No. 1 ($I = 0.03914 \text{ in.}^4$, weight/length = 0.140 lb/in.) which is the lightest most flexible part of the CLIRA. The length of the beam was defined to be the distance from the top of the core to the top of the standpipe (77 inches). The length of the beam was not defined to be the distance from the top of the standpipe to the core basket because vortex excitation will be the dominant mechanism in the case of an inoperative core restraint. Both fixed-fixed and simple-simple end support conditions were investigated. The maximum velocity measured in model testing near the CLIRA was 15 ft/sec [4]; a uniform velocity of 20 ft/sec was assumed in the analysis. Table 2 summarizes the results. For the expected conditions of a fixed CLIRA support at the core, the calculated response was very small. In addition a displacement of 0.004 in. resulted in a maximum stress of only 650 lb/in.^2 .

CRDM

The CRDM is a very thin, flexible component, consisting of many different size joined tubes, which extends approximately 106 inches from the top of the support tube to the upper portion of the control rod duct in the core. Its main function is to provide a means to raise and lower the relatively massive absorber section from above the false head. See Figs. 1 and 5(a). The main tubes surrounding the CRDM are the upper and lower supports, the shield plug above the suppressor plate, the control rod guide tube on the instrument tree, and the control rod duct. Nearly every section of the CRDM contains mass simulation tubes or mass due to accelerometers and cables. The CRDM was assumed filled with fluid to the water level.

Various boundary constraints were considered possible since: the CRDM is loose at the top, but hangs under its own weight; it is not directly attached to the support coupling, but has only 20 mils diametral clearance; it is not directly attached to the dashpot cup, but has only 25 and 64 mils diametral clearance at the top and bottom, respectively; the dashpot cup may or may not rest on the instrument tree control rod guide tube, depending upon vibration level or flow pressure; and the absorber section may or may not contact the control rod ducts (121 mils diametral clearance), depending upon the level of vibration and/or the level of static deflection caused by the fluid flowing in the duct. In essence the CRDM was assumed likely to rattle on its supports. To assess possible frequency and amplitude content of the rattling, the model with the several optional boundary conditions shown in Fig. 5(b) was analyzed.

The frequencies reported in Table 3 are expected to be those most likely to occur for the forcing function assumed. For other forcing functions, different support conditions may be active. In all cases

reported, the assumption was made that a simple support condition exists at the support coupling. For Cases 1-4, the dashpot was assumed to move with the CRDM, but not in Case 5. For Case 1, the additional assumptions were made that the upper end is fixed and the lower end is free to move laterally but with no rotation. For Case 2, the additional assumptions made were that the upper end is fixed and the lower end is completely free. For Case 3, the additional assumptions were made that the upper end is simply supported and the lower end is completely free. For Cases 4 and 5 the additional assumptions were made that the upper end is simply supported and the absorber section has a lateral spring support of 10 lb/in. and a rotational spring support of 46 lb·in./degree.

Flow induced loading appeared to be limited to that due to a maximum cross flow of 8 ft/sec [4] in the approximately 5 in. region directly below the suppressor plate. Parallel flow induced forces in the guide tubes and control rod ducts were assumed negligible because the rate of flow is about one-fifth that in the fuel rod ducts. The Reynolds number of the maximum flow across the 0.625 inch diameter of the CRDM was calculated to be $\sim 1.75 \times 10^5$, according to (7) with a corresponding vortex shedding frequency range of $\sim (30 - 46)$ Hz. Therefore several natural frequencies of the CRDM will coincide with the vortex shedding frequency at different times during flow startup if the excitation mechanism is active. Further, the associated response will be highly dependent upon damping. An effective damping of 5% critical damping was assumed for making predictions. This is considered to be conservative damping, because of all the expected impacting at supports and the narrow fluid gaps. Since the calculations were made, data on damping of similar components has become available [17] which support the assumption.

For a p given by (1), displacements were calculated for coincidence of the vortex and structural natural frequencies at the upper accelerometer, the lower accelerometer, and at the top of the very flexible section which connects to the absorber section for the first three cases. The results are shown in Table 4. Approximate displacements at other damping ratios can be determined by linear extrapolation. For any of the cases reported, the maximum stress calculated was never less than 10,000 psi for the flexible section of the CRDM at the support coupling, or less than 22,000 psi for the flexible section of the CRDM above the absorber.

Thermal Liner

The thermal liner is a cylindrical shell (inside radius = 33.94 in., wall thickness = 0.25 in.) which extends upward 82 inches from the top of the core support structure, where it is fixed, to above the level of the vortex suppressor plate and water pool in the upper plenum, where it is free. It is situated concentrically with respect to the main vessel cylindrical shell (inside diameter = 34.625 in., wall thickness = 0.625 in.) and separated by a very small fluid filled gap of nominally 0.44 inches (see Figure 1).

Predicting the natural frequencies of a cantilevered cylindrical shell in vacuum or even containing a fluid is tedious, but analysis techniques do exist. However, solutions are not available in the literature for concentric shells separated by fluid filled gaps. In general, vibrations are coupled and only natural frequencies of the system of the two shells and the fluid filled gap are meaningful. Added or apparent fluid masses for an individual shell are not meaningful unless one of the shells can be assumed rigid.

An estimate of the thermal liner natural frequency was made by first calculating the natural frequency for motion in a vacuum, utilizing the

NASTRAN computer code, and then applying approximate added mass correction factors obtained by assuming the main vessel rigid in comparison to the thermal liner. The added mass factors were determined from available analytical solutions for an infinitely long cylindrical shell with circumferential simple supports placed periodically along the length of the shell [18]. By defining the distances between simple supports to be twice as long as the thermal liner, the fluid boundary conditions are the same as for the cantilevered free thermal liner. Although the longitudinal shell mode shapes are somewhat different than those of the cantilevered thermal liner, the differences are expected to be small for such short shells. Results are given in Table 5 for vibration modes with a single half wave length in the axial direction ($m = 1$) and several ($n = 0 - 6$) in the circumferential direction. In addition, the analysis of the thermal liner frequencies shows that the added mass effect on frequency due to the fluid in the gap was an order of magnitude larger than that due to the fluid interior to the liner. Further, for the gap sizes considered a 10% change in gap size resulted in a 40% change in added mass. Thus even if methods of prediction can be developed for concentric shells, information on the effects of as-fabricated tolerances, such as out of roundness, also must be developed.

COMPARISON OF PREDICTED AND ACTUAL HCM VIBRATION RESPONSE

Extensive acceleration power spectral density records (PSD's) obtained during flow testing were made available to ANL. Comparison of predicted and actual lower structural natural frequencies was readily accomplished from the PSD's because of the sharpness (low damping) of the resonant response. Also, the PSD enabled comparison of the predicted and actual component response. However, comparison of assumed and actual component equivalent viscous damping ratios was not made from PSD's. The half-power band-width method could have been employed to estimate damping from the acceleration PSD's for some of the components structural natural frequencies; however, half power points are often difficult to define in the presence of background noise and extraneous system response in the same frequency range. Shaker test data is considered essential in evaluating component damping, especially in evaluating the flow dependency of component damping. Some shaker test data was also made available to ANL for comparison purposes.

As an example of the ease of identifying structural natural frequencies, a typical maximum flow rate PSD record for the CLIRA 1202 is shown in Fig. 6. The structural natural frequencies at 30, 85, and 165 are quite discernible on this record as well as on PSD's at lower flow rates. However, at the lowest flow rates (25 - 50% of maximum flow) the peaks are somewhat masked by background noise because of the generally small response of the components. Comparison with the higher flow rate records was necessary to identify the structural natural frequencies on the lowest flow rate records. In general, most of the components response increased with flow rate, see Fig. 7, at each structural natural frequency. An exception was the response of the T/LLM which peaked slightly at approximately 75% of maximum flow rate. These results indicate that the prevalent excitation source has a broad random frequency spectrum similar to what one can expect from general flow turbulence.

The T/LLM PSD records show two natural frequencies at 16 and 103 Hz. These two frequencies are assumed to be the two lowest natural frequencies on the basis that the ratio of these frequencies is close to that for a uniform cantilevered beam. Shaker testing found the two lowest natural frequencies to be 17.3 Hz and 105 Hz with 3.6% and 2.0% of critical damping, respectively. The calculated lowest natural frequencies were in the same range 15 - 19 Hz. The measured response, primarily at the lowest natural frequency, had a maximum value of ~ 0.07 mils. See Fig. 7. This is larger than maximum predicted response to parallel flow (~ 0.02 mils) at the lowest natural frequency assuming 1% of critical damping. Thus, the component excitation is probably due to both parallel and cross flow fluid turbulence excitation. No excitation by periodic vortex shedding was discernible as predicted.

The LLFM PSD records showed sharp resonance response at 30 Hz, 95 - 100 Hz, 140 Hz, and 190 - 200 Hz. All but the 140 Hz frequency was predicted as a component natural frequency. This frequency may have been that of an adjacent component. Shaker testing found the two lowest natural frequencies to be 31 - 35 Hz and 100 Hz with 1.9% and 2.6% of critical damping, respectively. The predicted response of 1 mil at the vortex shedding frequency of 8 - 12 Hz was not observed. The maximum measured response of 1 mil, primarily at 30 Hz, apparently was due to random cross flow turbulence, since no parallel flow was present around this component [4].

Two phases of HCM testing were performed and in the first phase of testing one CLIRA (3406) was monitored for a core with adjacent fuel ducts and the CLIRA firmly compacted together. In the second phase of testing two CLIRA's were monitored (1406 and 1202) for a less compacted core. The PSD records show strong resonance motion for CLIRA 3406 at approximately 45 Hz, 119 Hz, 190 - 220 Hz and possibly 70 Hz and 103 Hz. Case 5 natural

frequency predictions listed in Table 1, which assume large clamping forces at the core, correlate well with those of CLIRA 3406. For CLIRA 1202 the resonant motion frequencies on the PSD records appeared at 30 Hz, 85 Hz, 165 Hz, and 215 Hz. These frequencies appear to be between Case 3 and 4 natural frequency predictions of Table 1 which assume small core clamping. Also, Case 3 modified to include the optional simple support may lead to a good natural frequency correlation with CLIRA 1202 frequencies. Unexpectedly, the observed natural frequencies of CLIRA 1406, with a supposed loosely clamped core, were 43 Hz, 205 Hz, 250 Hz, and possibly 120 Hz. These frequencies correlate much better with the Case 5 predictions which assume a tightly clamped core than they do for predictions which assume a loosely clamped core. Shaker testing found a lowest natural frequency of 45 Hz and 1.5 - 1.9% of critical damping for CLIRA 1406. These test data and predictions are indicative of the variability of reactor component boundary conditions and the need to consider all possible conditions in the structural design of the components.

The parallel flow response predictions made for the CLIRA's (Table 2), with a conservative assumption of 1% of critical damping, do bound the observed response whereas no motion was discernible at the periodic cross flow vortex shedding frequency predicted. Thus parallel flow turbulence is the most likely excitation source for the CLIRA's.

Extensive testing was done for the CRDM because impacting was observed on the accelerometer signals located in the dashpot: impacting was occurring at $\sim (1-2)$ times per second for various flows and vertical positions of the CRDM. The primary cause was traced to impacting between the dashpot cup and the piston. Also, some impacting between the drive line and the support coupling was observed to occur after the dashpot cup was locked to the piston. The numerous test run PSD records showed many

dominant resonant motions peaks whose values varied somewhat from record to record. This variation was attributable to the slightly changed boundary conditions which were specified from test to test, but most of the variation was suspected to be due to the numerous ill defined boundary conditions described previously in the response prediction section. The predominant bands of resonant frequency response, as observed on the PSD records, are given in Table 3. There is good correlation of the data with the modal frequency predictions for a combination of Case 5, motion of the CRDM without the dashpot cup, and any of the other cases. The only frequency not observed in the accelerometer data was the low frequencies around 1 Hz. However, such modes could be greatly suppressed if the absorber section was statically deflected against the control rod duct by the flow. Also, such modes could have occurred, and in fact caused the observed impacting, without being identifiable on the accelerometer records because of the inherent difficulty of measuring such low frequency motion with piezoelectric accelerometers. No shaker tests were performed for the CRDM.

The amplitude of the response at the upper accelerometer was determined by integration of the acceleration PSD for the accelerometer located in the dashpot piston. The major part (90%) of the response was determined to occur in the 5 Hz frequency range with values between 2.5 - 10.5 mils RMS for flow rates between (1200 - 3700) GPM. The possibility of periodic vortex shedding or more generally turbulent cross flow was ruled out as the basic excitation mechanism on the basis of tests for which the fluid surface was lowered below the top of the control rod guide tube: cross flow could not impinge upon the CRDM. Elimination of cross flow only caused an approximately 25% reduction in response. Evidently the small parallel flow along the complex geometry of the CRDM in the control rod guide tube was the main source of excitation.

The response of the thermal liner was also monitored. However, the acceleration PSD did not show any dominant resonant peaks. Apparently, for the flow conditions tested, the resonant frequencies of the thermal liner are not readily excitable. Also, no shaker tests of the thermal liner were performed. An approximate model of the HCM liner and fluid gap is planned for testing to assess the ability to predict natural frequencies. The damping provided by the small fluid gap also will be investigated.

CONCLUSIONS

The most obvious conclusion which can be drawn was that the response of none of the components studied showed any significant motion which could be attributed to the predicted cross flow periodic vortex shedding excitation at any of the flow rates tested. The reason periodic cross flow vortex shedding excitation did not occur can only be speculated upon. As noted, in describing the forcing functions employed in making the predictions, there remains doubt as to whether the forces on a vibrating component due to vortex shedding are random or deterministic in the Reynolds number transition range ($4 \times 10^5 - 6 \times 10^6$), and most of the maximum cross flow velocities impinging upon the HCM components were in the transition range. However, most of the cross flows were not in this range for the lower flow rates tested. A more likely explanation of the absence of periodic cross flow excitation is that cross flow of sufficient spatial homogeneity to produce periodic vortex shedding is not present in the reactor flow.

No conclusion can be drawn as to the utility of the parallel flow forcing functions in predicting vibration response, mainly because the actual response due to parallel flow could not be separated from other sources of excitation. In one case reasonable predictions were made. For the CLIRA, which was subject primarily to parallel flow, a crude parallel flow prediction was made which bounded the components actual response. However, the response of the CRDM was shown by test to be primarily due to parallel flow, but predictions were not made because (1) the parallel flow was considered too small to cause significant excitation by a turbulent boundary layer, and (2) the many abrupt changes in cross sections along the length of the CRDM made application of the parallel flow forcing function highly questionable, since it was derived from the pressures

exerted by a turbulent boundary layer developed along an infinitely long circular cylinder. Even though conclusions cannot be drawn as to the worthiness of the parallel flow predictions, one can conclude from the test data that if parallel flow is the primary source of excitation, then the response of the components will be quite small. The exception is a component like the CRDM where although parallel flow is predominant, the source of excitation is probably due to the gap modulation as the CRDM vibrates inside the close fitting tubes in which it is inserted.

The most promising result from the comparison of the predictions with the test data was that for circular pipe-like components, the natural frequency in water could be predicted or bounded within acceptable accuracy even when the boundary conditions and spatial variation in the cross sections of the tubes were complicated.

For future design purposes, the apparent need is to ascertain whether periodic vortex shedding can actually occur for the type of turbulent flows which occur in reactor plenums. Elimination of it as a possible excitation source would alleviate a good deal of worry among reactor designers. A case could be made for developing a forcing function for predicting the apparent turbulence related buffeting which was observed in the NCM testing; however, the level of the responses observed were small and such an effort may not be justifiable. An effort directed toward understanding the fluid flow excitation mechanisms for components like the CRDM definitely would be justifiable from the standpoint of reactor safety alone.

REFERENCES

1. Y. N. Chen, "Fluctuating Lift Forces of the Karman Vortex Streets on Single Circular Cylinders and in Tube Bundles, Part 2 Lift Forces of Single Cylinders," J. Eng. Ind. 94, 613-622 (May 1972).
2. C. R. Gerlach and F. T. Dodge, "An Engineering Approach to Tube Flow-Induced Vibrations," Flow-Induced Vibration in Heat Exchangers, ASME, (1970), p. 18.
3. G. W. Jones, Jr., "Unsteady Lift Forces Generated by Vortex Shedding About a Large Stationary, and Oscillating Cylinder at High Reynolds Numbers," ASME Paper 68-PE-36.
4. L. M. Polentz, D. L. Ballard, and F. H. Nunamaker, "FFTF Outlet Region Feature Model Localized Pool Velocities," Hanford Engineering Development Laboratory Report HEDL-TME-48 (Dec 1971).
5. S. S. Chen and G. S. Rosenberg, "Dynamics of a Coupled Shell/Fluid System," to appear in Nuc. Eng. Des.
6. M. P. Paidoussis, "An Experimental Study of Vibration of Flexible Cylinders Induced by Nominally Axial Flow," Nuc. Sci. Eng. 35, 127-138 (1969).
7. S. S. Chen and M. W. Wambsganss, "Parallel-Flow-Induced Vibrations of Fuel Rods," Nuc. Eng. Des. 18, 253-278 (1972).
8. G. B. Warburton and J. Higgs, "Natural Frequencies of Thin Cantilevered Cylindrical Shells," J. Sound Vib. 11(3), 335-338 (1970).
9. Nuclear Systems Materials Handbook, Hanford Engineering Development Laboratory, TID 26666 (1975).
10. R. L. Daugherty and A. C. Ingersoll, "Fluid Mechanics," McGraw-Hill (1954), p. 15.
11. T. M. Mulcahy and S. S. Chen, "Annotated Bibliography on Flow Induced Vibrations," Argonne National Laboratory Technical Memorandum ANL-CT-74-05 (Jan 1974).

12. G. Bohm, "Analytical Problems Associated with Core Support Structures of PWR," Nuc. Eng. Des. 18, 305-321 (1972).
13. R. King, M. S. Prosser, D. J. Johns, "On Vortex Excitation of Model Piles in Water," J. Sound Vib. 29, 169-188 (1973).
14. S. H. Crandall, "The Role of Damping in Vibration Theory," J. Sound Vib. 11, 3-18 (1970).
15. M. W. Wambsganss, "Vibration of Reactor Core Components," Reactor and Fuel Processing Technology 10, 208-219 (1967).
16. L. KH. Blyumina and K. K. Fedyaevski, "Periodic Shedding of Vortices from the Surface of a Cylinder and the Forces Caused by These Vortices for Super-Critical Flow Conditions," Fluid Dynamics 3, 67-70 (1968).
17. A. Morronne, "Damping Values of Nuclear Power Plant Components," Nuc. Eng. Des. 26, 343-363 (1974).

Table 1

First Four Modal Frequencies of the CLIRA for Various Possible Boundary Conditions

Case No.	k (lb/in.)	Optional Simple Support	Modal Frequency, Hz			
			<u>f₁</u>	<u>f₂</u>	<u>f₃</u>	<u>f₄</u>
1	0.00	No	10.1	31.8	66.5	110.8
2	50.0	No	15.6	33.3	67.0	111.0
3	10 ³	No	28.9	65.9	77.4	116.8
4	10 ⁹	No	37.3	99.5	194.7	203.8
5	10 ⁹	Yes	48.1	128.1	203.8	230.9

Table 2

Maximum Deflection of the CLIRA due to Crossflow Vortex Shedding

<u>Case No.</u>	<u>Damping (%)</u>	<u>Deflection (in.)</u>	<u>x (in.)</u>
1	1	0.586	71.90
	2	0.331	71.90
2	2	0.284	64.25
3	1	0.015	43.85
4	1	0.010	41.3

Maximum Deflections of the Parallel Flow Model of the CLIRA

<u>End Support Condition</u>	<u>Displacement (in.)</u>	
	<u>Equation (3)</u>	<u>Equation (4)</u>
Simple-simple	0.004	0.022
Fixed-fixed	0.002	0.004

Table 3

Modal Frequencies (in Hz) for the CRDM

Analytical Predictions

<u>Frequency</u>	<u>Case 1</u>	<u>Case 2</u>	<u>Case 3</u>	<u>Case 4</u>	<u>Case 5</u>
f_1	0.65	0.46	0.43	2.5	2.5
f_2		1.60	1.52		
	4.1			5.5	5.5
f_3	17.1	6.54	6.45	7.8	8.9
f_4	32.9	19.43	19.25	19.7	20.2
f_5	63.4	33.29	32.65	33.3	39.3
f_6	103.6	63.48	62.78	63.5	70.7
f_7	154.2	103.61	102.85	103.6	127.0
f_8	221.3	154.2	153.40	154.2	186.6
f_9	296.2	221.4	219.83	221.3	267.0

Experimental Data

	<u>f_1</u>	<u>f_2</u>	<u>f_3</u>	<u>f_4</u>	<u>f_5</u>	<u>f_6</u>	<u>f_7</u>	<u>f_8</u>	<u>f_9</u>
Dominant Frequency Peaks (Hz)	-	5	-	18	40	70	107	150	205
Variation with Test Runs (%)	-	10	-	10	10	10	7	7	5

Table 4

CRDM Response (mils) Assuming 5% Effective Modal Damping

<u>Position</u>	<u>Case 1</u>	<u>Case 2</u>	<u>Case 3</u>
Upper Accelerometer (Located in Dashpot Piston)			
f_2	20	10	10
f_3	-	10	11
f_4	10	9	9
f_5	3	3	3
Lower Accelerometer (Located in Upper Part of Absorber Section)			
f_1	8	-	15
f_3	13	17	25
f_4	5	16	46
f_5	2	6	64
Absorber Coupling			
f_1	8	5	15
f_3	2	25	25
f_4	61	46	46
f_5	67	62	64

Table 5

Natural Frequencies for the Thermal Liner for One Axial Half Wave
($m = 1$) and Several (n) Circumferential Half Waves

<u>n</u>	<u>Frequency (Hz)</u>	
	<u>In Vacuum</u>	<u>In Fluid</u>
0	-	-
1	208.0	6.8
2	98.8	5.7
3	56.3	4.6
4	44.3	4.8
5	53.7	7.2
6	73.0	11.6

Fig. 1. HCM Upper Plenum Components and a Typical Flow Field.

(a)

(b)

Fig. 2. (a) Simplified Sketch of T/LLM, and (b) Model Used in the Analysis.

(a)

(b)

Fig. 3. (a) Simplified sketch of HCM LLFM, and (b) model used in the analysis.

(a)

(b)

Fig. 4. (a) Simplified Sketch of CLIRA, and (b) Model Used in the Analysis.

Fig. 5. (a) Simplified drawing of HCM Control Rod Drive Mechanism, and
(b) Model Used for Free- and Forced-vibration Analysis.

Fig. 6. Radial Direction Acceleration PSD for CLIRA 1202 at 3700 GPM Flow at 100°F (HEDL Phase II Testing - Test Run No. 55611-2F1).

Fig. 7. Maximum Response of the LLFM, T/LLM, and CLIRA's (1202 and 1406) for Various Flow Rates During Phase II Testing at 100°F.