**FLOW-INDUCED VIBRATION
FOR LIGHT-WATER REACTORS**

**PROGRESS REPORT
(APRIL 1978 — DECEMBER 1979)**

MARCH 1980

J. F. SCHARDT

**GENERAL ELECTRIC COMPANY
NUCLEAR ENGINEERING DIVISION
SAN JOSE, CALIFORNIA 95125**

**PREPARED FOR THE
U.S. DEPARTMENT OF ENERGY
UNDER CONTRACT EN-77-C-02-4175**

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ABSTRACT

Flow-Induced Vibration for Light Water Reactors (FIV for LWRs) is a four-year program designed to improve the FIV performance of light water reactors through the development of design criteria, analytical models for predicting behavior of components, general scaling laws to improve the accuracy of reduced-scale tests, and the identification of high FIV risk areas. The program is managed by the General Electric Nuclear Power Systems Engineering Department and has three major contributors: General Electric Nuclear Power Systems Engineering Department (NPSED), General Electric Corporate Research and Development (CR&D) and Argonne National Laboratory (ANL). The program commenced December 1, 1976, but was suspended on September 30, 1978, due to a shift in Department of Energy (DOE) priorities away from LWR productivity/availability. It was reinitiated as of August 1, 1979. This progress report summarizes the accomplishments achieved during the period from April 1978 to December 1979.

INTRODUCTION

The Flow-Induced Vibration for Light Water Reactors Program is being conducted for the Division of Nuclear Power Development, US DOE. The goal of the program is to increase nuclear plant availability through substantially reducing downtime caused by FIV failure of components. It is a four-year balanced effort of fundamental studies, analyses, tests of idealized conditions, and realistic tests of reactor components, all leading to the preparation of design guides and criteria for LWRs. The program is broken into four major tasks:

- Task 1 - Fundamental Studies (CR&D and ANL)
- Task 2 - Model and Full-Size Tests (NTD)
- Task 3 - Design Methods, Guides and Criteria (NTD)
- Task 4 - Program Administration (NTD)

A brief summary of the objectives of these tasks and a description of the technical progress achieved over the period from April 1, 1978 through December 31, 1979, is provided in the following sections. Additional background information can be obtained from *Flow-Induced Vibration for Light Water Reactors Progress Report: December 1976 - May 1977* (DOE Document Number C00/4175-1).

TASK 1 - FUNDAMENTAL STUDIES

The purpose of Task 1 is to develop a better fundamental understanding of FIV phenomena as related to light water reactors (LWRs). A combination of tests and analyses considers geometries and FIV mechanisms common to the LWR. Particular emphasis is given to characterizing the FIV forcing functions for use in establishing predictive models. The task is divided into four major areas:

Task 1.1 - External Flow over Tube Banks - Cross Flow (CR&D)

Task 1.2 - External Flow over Tube Banks - Parallel Flow (ANL)

Task 1.3 - Leakage Flow Mechanisms (CR&D)

Task 1.4 - Random Vibration Effects (ANL)

Progress has been achieved on Tasks 1.1 and 1.2 over the report period and is described on the following pages. Task 1.4 has been completed; the final results are documented in DOE report COO/4175-10, "Random High-Cycle Fatigue Analysis."

TASK 1.1 - EXTERNAL FLOW OVER TUBE BANKS - CROSS FLOW (CR&D)

Task 1.1, which is being performed by General Electric Corporate Research and Development, is designed to develop a better understanding of the response of tube arrays to fluid cross flow. It consists of six activities:

Task 1.1a - Preliminary Array Definition Experiments

Task 1.1b - Water-Based Tests at High Reynolds Numbers

Task 1.1c - Buffeting of Cylinder Arrays in Turbulent Cross Flow

Task 1.1d - Fluidelastic Instability Boundaries

Task 1.1e - Proof Test

Task 1.1f - Design Guides

Progress achieved over the report period on these activities is described below.

Task 1.1a - Preliminary Array Definition Experiments (S. D. Savkar, R.M.C. So)

Task Description

The objective of this task was to develop an experimental scheme capable of directly measuring the unsteady forces induced on cylinders immersed in a turbulent cross flow. The measurement scheme will be used in several of the Task 1.1 test programs.

Work Completed This Report Period

Task 1.1a was successfully completed during a previous report period. Work is now terminated on this activity.

Task 1.1b - Water-Based Tests at High Reynolds Number (R.M.C So)

Task Description

Task 1.1b consists of a series of experiments performed in water, using the Garfield Thomas Water Tunnel at Pennsylvania State University, to study the buffeting of a cylinder in turbulent cross flow. The primary objective is to measure and correlate the unsteady forces induced by the crossflow on the cylinder as a function of Reynolds number and turbulence. The program is designed to complement the crossflow-in-air activities studied in Task 1.1c.

Work Completed This Report Period

The first activity in the test program, performed prior to the installation of the instrumented cylinder, was to measure, using hot film anemometers, the flow characteristics in the test section at the location where the cylinder was to be installed. The linearized hot film anemometer signals were passed through an analog sum and difference amplifier and true RMS meter to determine the mean velocity profile, $V(r)$, as well as the turbulent intensity, u'/V , for the various turbulence-generating grids to be used in the test program (1-in., 2-in., and 3-in. hole diameter grids). Results showed the mean velocity to be uniform across the central 10-in. core of the 12-in.-diameter flow tunnel, and essentially independent of the grid size. The turbulent intensity was 0.5% with a radial variation of 0.07% when no grid was used, and $9.5\% \pm 1\%$ with a radial variation of less than 1% for each turbulence grid when located at a distance $X/M = 10$ upstream of the measurement probe (X = distance in inches, M = grid hole size).

Upon completion of the flow characterization studies, cross-flow experiments were performed on single cylinders in which the cylinder's diameter and active span, as well as the Reynolds number and turbulence intensity, were varied. The purpose was to better understand the relationship between the fluid characteristics and the resultant forces acting on the cylinder. Steady forces were determined by measuring the static pressures on the cylinder at numerous

azimuthal locations; unsteady forces were determined by measuring the shear loads at the cylinder supports using piezoelectric load cells. Key results are summarized below.

The steady drag coefficient was determined by inputting static pressure data into the following relationships:

$$2RD = \int_0^{2\pi} P \cos \theta R d\theta$$

$$C_d = \frac{D}{1/2 \rho V^2}$$

where θ is the cylinder azimuthal position, R is the cylinder radius, P the wall static pressure, D the drag force, ρ the fluid density, and C_d the steady drag coefficient. Sample drag coefficient results are shown in Figure 1.1-1 for a 1.5-in.-diameter cylinder. As is well established, the effect of the free stream turbulence generated by the grids is to shift the subcritical-to-supercritical transition of the flow over the cylinder (that is, the point at which the steady drag coefficient makes a dramatic reduction) to a lower Reynolds number.

The RMS Strouhal and buffeting lift and drag coefficients were calculated from the unsteady force measurements, using the following relations:

$$C_L' = \frac{L'}{\rho V^2 \ell R}$$

$$C_D' = \frac{D'}{\rho V^2 \ell R}$$

where ℓ is the active span, L' and D' the total RMS fluctuating lift and drag measured by the load cells, and C_L' and C_D' the fluctuating lift and drag coefficients. One of the key observations from this study to date is the

close similarity between the fluctuating lift coefficient and the steady state drag coefficient. This is illustrated in Figure 1.1-2 for no grid and 2-in. grid data. As in the steady drag, the unsteady lift has three distinguishable regions of response. The low Reynolds number region may be called subcritical, analogous to that for the steady drag. In this region, the turbulence from the grid does not seem to significantly affect the fluctuating force response. In the vicinity of $Re = 10^5$ is the "transition region," where the lift coefficient decreases dramatically. Here, the effect of increased turbulence is very strong, causing the transition to shift to a lower Reynolds number. Finally, in the post-transition region, the fluctuating lift coefficient appears to flatten somewhat, the data suggesting the level to be somewhat directly related to the free stream turbulence.

The effect of the cylinder active span length (length of cylinder exposed to the flow) on the fluctuating lift coefficient was studied by performing tests on cylinders with active span lengths of 1.5 in., 3 in., 4.5 in., and 8 in. Results are plotted in Figure 1.1-3. It can be seen that as the active span is increased, a lower value of C'_L is obtained for the same Reynolds number. This appears to be caused by a lack of spanwise correlation in the pressure field, possibly due to variations along the span in the circumferential position of the boundary layer separation.

The Strouhal frequencies were determined from the auto-power spectra of the fluctuating lift signals. At low Reynolds numbers, one distinct peak existed for each spectrum, and the Strouhal frequencies could be identified without ambiguity. However, as the Reynolds number increased, the spectra tended to flatten, such that no discernible peaks were evident beyond $Re = 3 \times 10^5$. The measured Strouhal number, $S_t = f_L \cdot 2R/V$ (where f_L is the Strouhal lift frequency obtained from the power spectrum) is plotted as a function of Reynolds number for the no-grid case in Figure 1.1-4. It can be seen that S_t remains fairly constant at 0.21 for Re less than 2×10^5 , but abruptly increases to approximately 0.5 for Re equal to 3×10^5 . Much the same trend is shown by the data of Richter and Naudascher.^{1.1-1} Their measurements with the 25% flow blockage are also shown in Figure 1.1-4 for comparison.

The spectra of the buffeting lift forces are generally similar for the four different active spans tested. Therefore, only one set of spectra for the Reynolds number range investigated is reported, that being for the 4.5-in. active span. The output from the Hewlett Packard Fourier Analyzer, $E(f)$, which represents force squared per Hertz for the ordinate and Hertz for the abscissa, has been normalized to reflect a fluctuating lift coefficient spectrum, where the abscissa is now the reduced frequency. The basis for the normalization is the following:

$$C_L'^2 = \frac{L'^2}{(\rho V^2 \ell R)^2} = \int_0^\infty \frac{E(f D/V)}{(\rho V^2 \ell R)^2} \frac{V}{D} d \frac{fD}{V}$$

Thus, the outputs of the spectrum analyzer, $E(f)$ and f , were normalized by $(\rho V^2 R/(V/D))$ and V/D , respectively, to produce the dimensionless quantities, ϕ and S , as defined below:

$$\phi = \frac{(V/D) E\left(\frac{fD}{V}\right)}{(\rho V^2 \ell R)^2}$$

$$S = \frac{fD}{V}$$

The normalized spectra are plotted in Figure 1.1-5 for the Reynolds number range, 3.45×10^4 to 34.51×10^5 . It can be seen that at low Reynolds numbers, the force response is Strouhal-like. However, as Re increases, a slow broadening of the spectrum is observed. At $Re = 13.79 \times 10^4$, the distinct peak on the spectrum disappears, and thereafter, the spectral response resembles that of the upstream turbulence. Also note the reduction in amplitude of the peak. From a high of $\phi \approx 9$ at $Re = 3.45 \times 10^4$, it reduces to $\phi \approx 0.02$ at $Re = 13.79 \times 10^4$. This is consistent with the large decrease in C_L' shown in Figure 1.1-3.

Task 1.1c — Buffeting of Cylinder Arrays in a Turbulent Cross Flow (S. Savkar)**Task Description**

Task 1.1c consists of a series of experiments performed in air to study the buffeting of single cylinders and cylinder arrays in a turbulent cross flow. The experiments are directed towards characterizing the forces exerted on the cylinder as a function of such parameters as Reynolds number, upstream turbulence, Strouhal number and cylinder array geometry. The program utilizes the force measurement scheme developed in Task 1.1a to characterize the cylinder forces.

Work Completed This Report Period

The design of the tube array buffeting apparatus and much of the fabrication of the apparatus has been completed. The overall scope of the test program is outlined in Figure 1.1-6 and the schematic of the apparatus assembly is illustrated in Figure 1.1-7.

Task 1.1d — Fluidelastic Instability Boundaries (T. F. Balsa)**Task Description**

A combined test and analysis program is directed toward establishing the nature and boundary of fluidelastic instability of arrays as a function of array geometry. A near-term objective is the numerical prediction of the forces on an oscillating single cylinder at high Reynolds numbers (10^4 to 10^7). These predictions should be useful in establishing whether the forces can produce a hydroelastic instability or simply excite cylinder motion through their fluctuating or unsteady component.

Work Completed This Report Period

The conceptual design of an apparatus in which to critically examine the results of the theory previously reported^{1.1-2} is essentially completed. The apparatus would be the same as that used in Task 1.1c except that the load cell mounted measuring cylinder would be replaced with a variable spring mounted/controlled damping cylinder. A schematic of the apparatus and the range of parameters to be tested are shown in Figure 1.1-8. A schematic of the eddy current damper developed is shown in Figure 1.1-9.

References

- 1.1-1 A. Richter and E. Naudascher, "Fluctuating Forces on a Rigid Circular Cylinder in a Confined Flow," J. Fluid Mech., 1976, Vol. 78, p. 56.
- 1.1-2 J. F. Schardt, "Flow Induced Vibration for Light Water Reactors, Quarterly Progress Report, January-March 1978," GEAP-24163, November 1978.

Figure 1.1-1. Variation of Mean Drag with Reynolds Number for 1.5-in.-Diameter Cylinder

Figure 1.1-2. Comparison of the Behavior of the Mean Drag and rms Lift Coefficient for the 1.5-in.-Diameter by 4.5-in.-Span Cylinder in a Uniform and Turbulent Stream

Figure 1.1-3. Effects of Span on the Variation of Buffeting Lift with Reynolds Number (Δ = Measurements from Top Load Cell; $\blacktriangle, \bullet, \circ, \diamond$ = Measurements from Bottom Load Cell)

Figure 1.1-4. Variation of Strouhal Number with Reynolds Number for 1.5-in.-Diameter by 4.5-in.-Span Cylinder

Figure 1.1-5. Power Spectral Density Function of Buffeting Lift at Different Reynolds Numbers for the 1.5-in.-diameter By 4.5-in.-span Cylinder

- TEST ONE ROW OF TUBES – $1.2 \leq s/d \leq 1.7$
- TEST THREE (5 DEEP) STAGGERED ARRAYS – $1.2 \leq s/d \leq 1.7$
- TEST THREE (5 DEEP) INLINE ARRAYS – $1.2 \leq s/d \leq 1.7$
 - $10^4 < Re \leq 2 \times 10^5$
 - MEASURE LOADS ON ONE ACTIVE CYLINDER
 - PRESSURES ON ONE CYLINDER
 - HOT WIRE MEASUREMENTS IN THE GAP
 - UPSIREAM FLOW WITH GRID TURBULENCE

Figure 1.1-6. Tube Array Buffeting Experimental Scope

1.1-14

Figure 1.1-7. Test Section Assembly - Tube Array Buffeting Experiments

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1.1-15

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Figure 1.1-8. Fluid Elastic Instability Apparatus Conceptual Design

Figure 1.1-9. Isotropic Two-Dimensional Damper

TASK 1.2 - EXTERNAL FLOW OVER TUBE BANKS - PARALLEL FLOW (ANL)

Task 1.2, which is performed by Argonne National Laboratory, is designed to develop a better understanding of the response of tube arrays subjected to parallel flow. The specific objectives include the following:

- (1) Determine, by a systematic series of experiments, the dependence of amplitude of motion and forces on Reynolds number, upstream flow conditions, array geometry, and weak cross flow for tube arrays in essentially parallel flows;
- (2) Establish weak cross-flow-induced fluidelastic stability boundaries: and
- (3) Evolve design methodology to estimate the fluid forces and resulting vibration response.

The program is divided into five subtasks:

Task 1.2a - Review of Available Data from Flow-Induced Vibration Tests and Assessment of the State of the Art in Understanding/Modeling Parallel Flow-Induced Vibration Phenomena and/or Response

Task 1.2b - Experimental Study of Single Cylinders in Nominally Parallel Flows Typically Encountered in Channels and Reactor Plenums

Task 1.2c - Experimental Study of Fluidelastic Interaction that Can Result in Coupled Response and/or Instabilities in Rod/Shell Systems, Tube Bundles, Tubes in Asymmetric Flow Channels, and Systems with Nonuniform Cross Sections

Task 1.2d - Experimental Study of Tube Arrays in Parallel Flow

Task 1.2e - Development of Mathematical Models and Participation in
Design Guides Evaluation (Task 3)

The primary test fluid will be water. Incoming flow conditions, together with a weak cross flow, will be used to establish the effects of the parameters of interest. The flow will be varied over a sufficient range to clearly establish the role of Reynolds number. Besides acquiring data of significance to the parallel flow FIV of tube arrays, analytical tools which can be evolved into design guides will be devised.

A description of the progress achieved on these subtasks over the periods from April 1978 to September 1978 and from October 1979 to December 1979 follows.

Task 1.2a - State-of-the-Art Assessment

Task Description

Task 1.2a consists of a day-to-day review of available parallel FIV literature applicable to light water reactors in order to maintain a current knowledge of the state of the art.

Work Completed This Report Period

No work, other than continued general review of the FIV literature, was performed on this activity during the report period.

Task 1.2b - Single Cylinders in Nominally Parallel Flow (T. M. Mulcahy,
T. T. Yeh, W. H. Lin, W. P. Lawrence)

Task 1.2b consists of two complementary parallel FIV programs on isolated cylinders. Experiments are performed in water using the ANL 500 gpm water loop and in air using the University of Illinois at Urbana open cycle air loop at the Thermal Hydraulics Laboratory. Work performed on these programs is described below.

Activity 1 - Annular Region (Water)

Task Description

Parallel FIV experiments are performed on isolated cylinders in water to determine the cylinder response, wall pressure statistics, cylinder resonances, and fluid damping as a function of the independent parameters with the ultimate goal of developing an analytical response prediction model characterized by the test data. The independent test parameters include a wide range of Reynolds numbers, hydraulic diameters, cylinder structural boundary conditions, and most importantly, upstream flow disturbances. The creation and characterization of upstream flow disturbances as a test parameter will be the unique contribution of this work.

The specific objectives are to:

- (1) provide test data to develop an integrated analytical-experimental model for predicting response of a rod located symmetrically in a cylindrical region subjected to fully developed flow with and without upstream flow disturbances; and
- (2) establish sensitivity (scaling) of wall pressures with respect to rod vibration, Reynolds number, hydraulic diameters, and upstream disturbances (turbulence).

Work Completed This Report Period

Turbulence Measurements

Turbulence flow characterizations in the annular flows have been measured. As indicated in the previous quarterly report (C00/4175-9), each turbulence generator can be installed at two positions upstream of the test rod to produce different turbulent flows. Location A, 7 cm upstream from the lower support fin of the rod, was chosen for obtaining a maximum turbulence effect

and location B, 35 cm upstream from the support fin, was selected for producing a lower intensity turbulence. It was expected that the turbulence produced by grids located at A was likely to behave as a highly nonuniform jet-like flow, while the turbulence generated at location B would be more homogeneous.

The turbulent flow was characterized by the mean velocity distribution $U(r)$, the turbulence intensity T_u , the turbulence energy spectra $\phi_{uu}(f)$, and the turbulence integral time scale I and length scale L . Analytically, these parameters are defined as

$$T_u = [\langle u^2 \rangle]^{1/2} / \bar{U} ,$$

where the mean square of the turbulent velocity fluctuations is given by

$$\langle u^2 \rangle = 2 \int_0^{\infty} \phi_{uu}(f) df ;$$

$$I = \int_0^{\tau_m} R(\tau) d\tau ,$$

where the normalized autocorrelation function is

$$R(\tau) = \frac{\int_{-\infty}^{\infty} \phi_{uu}(f) e^{i2\pi f\tau} df}{\langle u^2 \rangle} ;$$

and

$$L = \bar{U} I ,$$

where the spatially averaged mean velocity measured by the turbine flow meters is

$$U = \frac{8}{D^2 - d^2} \int_{d/2}^{D/2} U(r) r dr ,$$

For the nonhomogeneous turbulence which predominates in the tests, I gives an order of magnitude estimation of the average time period over which the flow past a particular fixed point is correlated, while L gives a similar approximation to the length of the turbulent eddies in the axial direction. Both measures are most meaningful, particularly L , for homogeneous turbulence where the eddies remain nearly constant in size, shape and distribution everywhere in the flow.

The characterizations of the turbulent flow produced by the turbulence generating grids for the test section without a rod have already been reported in the quarterly report for October-December 1977 (C00/4175-5). Those for the smallest rod/largest annular region are illustrated by the $\bar{U} = 8$ m/sec data shown in Figures 1.2-1 through 1.2-6 for several different grids. These data were measured at a radial position close to the rod wall and half-way across the annulus, $(2r - d)/D_h = 0.086$ and 0.5 , respectively. The effect of the support fins and the transition from the pipe to the annulus is significant and can be assessed from the no grid data of Figures 1.2-1 through 1.2-4. The mean velocity distributions of Figure 1.2-1 show that fully developed annular flow (Brighton and Jones^{1.2-1}) was not attained by $z/D_h = 26$, almost the length of the rod. Further, nonhomogeneous turbulence up to 14% in intensity is produced at the entrance and decays to nominal levels of 8-11%, several percent higher than the nominal levels observed for the open pipe.

Grids positioned at location B, farthest upstream from the test rod, were found to be ineffective in producing turbulence intensities any greater than that which already existed for the open annulus (Figure 1.2-2). However, significant amounts of turbulence could be introduced into the annular region

when the grid was positioned at location A, immediately upstream of the rod. The turbulence introduced decayed most rapidly close to the rod wall (Figure 1.2-3). Away from the rod wall, grid effects were observable for most of the length of the rod, to $z/D_h = 25$, and differences in the character of the turbulence between the different grids were discernible (Figures 1.2-5 and 1.2-6). The integral scales were of the order of 0.5 cm, the same size as for the grid in the pipe.

Flow characteristics for the smallest annular region are shown in Figures 1.2-7 through 1.2-9, again for $\bar{U} = 8$ m/s. Obviously, the grids, even in location A immediately upstream of the rod, were not able to create any more turbulence in the annular region than existed for the open annulus (Figure 1.2-7), which was approximately 20% at the test section entrance. However, for this smaller annulus, the turbulence intensity decayed very rapidly, in one-fifth the rod length ($z/D_h = 20$), to a constant value of $\sim 9\%$, just slightly larger than the smallest measured in the larger annular region. The power spectral densities of the turbulence, shown in Figure 1.2-8, are more typical of those expected for fully turbulent flow, and the integral scales (Figure 1.2-9) are significantly smaller in comparison to the larger annular region and the open pipe.

The turbulence measurements showed that upstream disturbances, such as the turbulence grids, could increase the turbulence levels of the annular region in the structural component frequency range below 500 Hz. However, significant increases above the turbulence levels already existing in the open annular regions were possible only for the annular region with a relatively large hydraulic diameter (3 cm as compared to 1 cm), and only when high intensity turbulence ($\sim 20\%$) was introduced immediately upstream of the 1.37-cm-diameter rod. The high intensity and close location were required because the transition from a pipe to an annular region with support fins alone produced high levels of nonhomogeneous turbulence (9 to 12% intensity for the larger annulus and up to 20% for the smaller annulus). Not only did the large entrance turbulence intensity tend to mask the effects of other disturbances for the smaller annulus, but the wall shear flow appeared to dominate the

character of the flow throughout the annulus, such that any upstream disturbance quickly decayed out in the first 20% of the rod length. The effects of the wall shear flow were present in the larger annulus but only at locations close to the rod.

Another interesting finding of the turbulence measurements was that the average eddy size was on the order of magnitude of the rod diameter and was smallest for the high intensity grid-produced turbulence. Speculating that the turbulence eddy size is related to the correlation length of the fluctuating wall pressure field, then the grid-generated upstream disturbance is a somewhat ineffective means of rod excitation. However, disturbances which could produce any larger eddy sizes in such small annuli are hard to conceive. Correlation lengths which match the fundamental vibration mode wavelength would be optimal for a random excitation source. The mismatch in size which actually occurs is the probable reason that such very high intensity turbulence is required to produce significant vibration levels in parallel flow.

Vibration Measurements

Two Endevco 22 Picomin piezoelectric accelerometers, with sensitive axes 90° apart, were affixed to the inside of the test rod using a special mounting pellet. They were used to measure rod acceleration in transverse directions (x and y) at the midspan of the rod. The displacements of the test rod were subsequently obtained from the acceleration data. Both the root mean square value and the power spectrum of displacement, x_{rms} and ϕ_{xx} , respectively, were obtained for the same test parameters used in the turbulence measurement. Analytically, the power spectrum of displacement x is related to the mean square displacement response by

$$\langle x^2 \rangle = 2 \int_0^{\infty} \phi_{xx}(f) df .$$

The x_{rms} calculation was limited to the narrow frequency band which contained the fundamental modal response of the rod (i.e., 20 to 40 Hz).

From the data of the earlier tests, it was noticed that most of the displacement signals contained significant electronic background noise and that the response signal-to-noise ratios were very poor. The large unwanted signal picked up by the accelerometers in the high frequency range (500 to 10,000 Hz) made it necessary to operate the operational amplifiers at a minimum sensitivity scale to avoid oversaturation of the amplifiers. Unfortunately, at these settings, the signals were buried in the background noise. To improve the signal-to-noise ratio, a special low pass active filter (Endevco Model 35257-501, -5% point at 500 Hz) was employed. The use of such filtering was justified because (1) the largest displacement is at the fundamental mode frequency of ~ 27 Hz; and (2) coupling between the high frequency signal and the fundamental vibration mode was not important. Figure 1.2-10 shows the typical noise reduction of the active filter in the frequency range of interest. The filtered data gave a similar rms response, although the PSD can be clearly isolated from the background noise, exhibiting a well-defined peak near the modal frequencies. A brief study of the characteristics of the filtered data and the active filter suggested an active filter with a lower cutoff frequency (Endevco Model 35257-201, -5% point at 200 Hz) be used.

As indicated in the last quarterly report, the swept-sine excitation method used in still fluid to determine that the modal damping ratio was not suitable for a flowing fluid condition because the flow-induced vibration response was usually as large as that of the swept-sine excitation. The damping ratios during flow conditions were, therefore, determined by a transfer function technique in conjunction with a narrow band random excitation. The excitation used in the technique was exactly the same as that used in the swept-sine method described in the last report, except that a narrow-band random (20 to 35 Hz) excitation was applied to the test tod. The equation of motion of the fundamental mode can be described by

$$\ddot{x} + 2\xi\omega_n \dot{x} + \omega_n^2 x = h_m + h_f \quad (1)$$

where x is the displacement; ω_n and ξ are the natural frequency and the modal damping ratio, respectively; and h_m and h_f are the generalized narrow band random excitation force and the generalized random fluid force, respectively. Using the Fourier transform technique, equation (1) can be rewritten as

$$T_{xm}(\omega) = A(\omega)[1 + T_{fm}(\omega)]$$

where $T_{xm}(\omega)$ and $T_{fm}(\omega)$ are the transfer functions between x and h_m and between h_f and h_m , respectively; and

$$A(\omega) = [\omega_n^2 - \omega^2 + i2\xi\omega\omega_n]^{-1}$$

is the response function to be determined.

Since the two forcing functions, h_f and h_m , were produced by two different independent sources, it was considered justifiable to assume that there was no correlation between the two; that is, the transfer function $T_{fm}(\omega) = 0$, and $A(\omega) = T_{xm}(\omega)$. Thus, the modal damping ratio ξ and the natural frequency ω_n can be determined from the transfer function between the applied excitation h_m and the displacement x .

The first two measured modal frequencies of the rod in air (29.8 Hz and 99.5 Hz) were found to be slightly lower than calculated values (31.6 Hz and 102.4 Hz) where fixed-simple end conditions were assumed. This indicates that the upper support cannot be assumed to be completely rigid. Predictions for the rod in water were made according to

$$f_n = \frac{\beta^2}{2\pi\ell^2} \sqrt{\frac{E_s I_s}{\rho A + m_a}}$$

where the measured values $\ell = 1.22$ m, $E_s I_s = 297.7$ Pa m⁴, and $\rho A = 0.81$ kg/m were employed along with the non-flowing fluid added mass approximation $m_a = \rho_f d^2 \pi \alpha / 4 = 0.1478$ kg/m. The results are presented in Table 1.2-1.

Selected flow test data for the annular region with the largest hydraulic diameter, $D_h = 3.71$ cm, are shown in Figures 1.2-11 and 1.2-12. The rod response shown in Figure 1.2-11 was found to be nearly the same for the case of no grid or when any grid was positioned at location B, farthest upstream from the rod. Also, this response was small in comparison to the response obtained when the same grids were positioned at location A, Figure 1.2-11. Quite the opposite from location B, different grids at location A produced significant variations in the levels of response. For the largest flow velocity, differences of a factor of ten were observed. As expected,^{1.2-2} the vibration amplitude increased with increasing flow velocity. A note of warning is also included in the data of Figure 1.2-12. One of the grids produced cavitation at the higher flow velocities tested and the associated response became very large. Not only should this situation be avoided in practice, but should be accountable in analyzing response data. In a normally quiet loop the noise associated with cavitation reveals its existence, but in inherently noisy loops more effort may be required to detect its presence.

Slight decreases in the rod natural frequency occurred as the flow velocity increased and subjected the rod to small compressive forces (Figure 1.2-13). The changes in frequency were different for different grids, but the total changes were so small as to be inconsequential. For no flow, the measured frequency in the largest annulus was 27.5 Hz as compared to the analytic prediction of 28.7 Hz (Table 1.2-1). This corresponds to the slightly higher prediction in air already discussed. Of more significance is the variation of the modal damping with flow velocity shown in Figure 1.2-14. As before,^{1.2-2} a nearly linear increase in the damping ratio was observed, but there appears also to be a trend toward higher damping for those grids which produce higher turbulence. To firmly establish this trend, tests would have to be performed on rods with smaller mass and initial (no flow) damping.

The rod response for the smallest annular region, $D_h = 1.17$ cm, is depicted in Figure 1.2-15. None of the turbulence grids, regardless of their location, produced a rod response significantly different from that obtained for the

open annulus. The response is also considerably smaller than that for the larger annulus at the same flow velocity. In other respects, the rod response for the smaller and larger annuli were similar. Changes in structural frequencies with increases in flow velocities occurred but were small and inconsequential. Increases in damping were very nearly the same as shown in Figure 1.2-14 for the larger annulus, except that the scatter caused by different grids was not present.

The rod response correlated well with the turbulence measurements previously discussed. Test runs in which the turbulence levels in the annulus were similar also had similar rod response amplitudes. Two examples are the case of the largest annulus with different grids at the farthest upstream location, and the case of the smallest annulus for any grid in any location. In both instances, changes in grids produced very little change in the annular turbulence (particularly near the tube surface) and consequently little change in response. Where sizable increases in turbulence were produced, significant differences in response occurred. This was observed for the largest annulus with different grids immediately upstream of the test rod, where large differences were observed both in turbulence intensity and vibration response.

Not only does the rod response appear to depend on the intensity of turbulence produced by the upstream disturbance, but the relative size of the hydraulic diameter to the length of the rod is apparently important as well. For the same grid, grid location, and flow velocity, the response of the rod for the smaller annulus was significantly smaller than that of the larger annulus, even though the structural and flow damping was nearly the same for both annuli. For instance, at $\bar{U} = 8$ m/sec, $X_{rms} = 45$ μ m for $D_h = 3.71$ cm while $X_{rms} = 15$ μ m for $D_h = 1.17$ cm. This reduction in response can be attributed in a small part to the increase in added fluid mass, but mainly appears to be linked with the more rapid decay of upstream turbulence which was previously observed to occur for the smallest annulus (Figures 1.2-4 and 1.2-7). Essentially, the substantial upstream disturbances at the small annulus entrance cannot propagate far enough along the rod length to affect the rod response.

Wall Pressure Measurements

An instrumented rod for the measurement of wall pressure fluctuations, independent of the response testing discussed above, was designed and fabricated. The pressure transducers (Entran piezoresistive) for use in these wall pressure measurements were pre-calibrated before installation and tested for strain sensitivity. It was found that the strain-induced pressures varied as the direction of bending of the test element was changed. With static deflections of 0.003, 0.008, and 0.024 inches at rod center, the apparent compression strain induced pressures were 2.75×10^{-3} psi, 5.51×10^{-3} psi and 1.1×10^{-2} psi, respectively. These deflections indicated an approximate apparent pressure of 6.68×10^{-4} psi per mil deflection of the center of the test element. Since the lowest fluctuating pressure for this program (0.01 psi) is expected to produce a rod deflection on the order of 0.002 in., the strain-induced pressure was considered to be within acceptable limits (e.g., an effective pressure of 10^{-3} psi) and the pressure transducer mounting was determined to be adequate.

Methods of pressure transducer signal subtraction and data reduction were developed. For reliable pressure signal subtraction, pressure transducers should have the same sensitivities and the outputs should be linear. Bench testing with plane acoustic waves to determine the accuracy of eliminating far field noise in an annular region by subtraction of signals from diametrically opposed pressure transducers was performed. For a broad band random pressure field of 0 to 1.5 kHz and a total pressure of 0.5 psi rms, pressure signal differences of 0.012 psi rms were obtained. This indicated that 96 to 99 percent of the acoustically induced pressure field for the parallel flow test pressure measurements could be eliminated. Since the signal of interest is expected to be of the order of 0.01 psi, the far-field noise of the test section must be below pressure levels of 0.1 psi rms to recover data from the noise.

To improve the reliability of the pressure signals, the test loop was modified to eliminate some of the far field acoustic noise and pump

fluctuations. The modifications included utilizing the pumps from another facility (located approximately 100 feet away from the test section and separated from the test section by a 10,000-gallon accumulator tank).

Turbulent boundary-layer wall pressure data were obtained for three flow channel annuli and four different upstream flow spoilers (Grids A through D of the turbulence study). As shown in Figure 1.2-16, the rod instrumented for measuring wall pressure fluctuations contained pairs of pressure transducers, positioned at four axial locations. The utilization of diametrically opposite transducer pairs allowed for application of a subtraction technique^{1.2-3} in the data processing to eliminate the acoustic noise associated with the far field. Due to the failure of transducer 7, pressure data are reported at only three axial locations.

The data processing was accomplished on a digital spectrum analyzer (HP 5451C) and included computations of correlation functions, mean-square (or power) spectral density functions (PSDs) and rms pressure coefficients. A low-pass filter with a 10 kHz cutoff frequency was used in all data processing. The pressure difference from two diametrically-opposite transducers is obtained using a digital subtraction operation built into the computer program of the analyzer.

a. Mean-Square Spectral Function

Analytically, the mean-square spectrum, $\phi_{mn}(f)$ of the pressure-difference signals, $p_{mn} = p_m - p_n$ is given as

$$\int_0^{\infty} \phi_{mn}(f) df = \frac{1}{2} \langle (p_m - p_n)^2 \rangle$$

where the mean-square of the pressure difference $\langle (p_m - p_n)^2 \rangle = 2\langle p^2 \rangle$ for a circumferentially homogeneous case.

Typical results are given in Figures 1.2-17 and 1.2-18. In Figure 1.2-17, mean-square spectra of the pressure-difference signals from transducers 5 and 6, for the annulus with a 0.96 in. hydraulic diameter and no upstream spoilers, are plotted for four different flow speeds. In Figure 1.2-18, mean-square spectra of pressure-difference signals from transducers at three locations (Figure 1.2-16) for the 0.46-in. hydraulic diameter annulus, with no upstream spoilers, and a mean axial flow velocity of 15 m/sec, are plotted. It can be seen from Figure 1.2-17 that the turbulent wall pressure level increases with increasing mean flow speed, and that the shapes of the spectra are similar. Figure 1.2-18 shows that the pressure level at the first location (x_{1-2} , 15.71 cm from the flow entrance) is higher than that at the other two locations, and that the spectrum from the first transducer pair differs from that of the second and third pairs. This implies that the flow near entrance is not homogeneous with respect to the axial space. At the second and third locations, the flow field is almost homogeneous with respect to axial space so that the spectra of the pressure signals are similar in shape and magnitude for frequencies greater than 200 Hz. The peaks shown in the spectra are caused by the structure-borne noise and the transducers responding to rod strain. In particular, the dominant frequency peaks in the signal from transducer pair 3-4 corresponds to the first, and successive, natural vibration modes of the rod. Due to inherent differences in their installation in the rod, certain of the transducers are more sensitive than others to bending strain. In the subtraction process, the output due to test rod straining can be doubled as the tensile and compression strains add together. The "doubling" is further amplified through the squaring process associated with the computation of mean-square spectral density functions.

Scaling of the mean-square spectra of wall pressure was studied extensively. Normalization of the intensity with the dimensionless parameter $\rho^2 u^3 d_h$, and plotting the resulting curves versus dimensionless frequency fd/u collapses the data reasonably well, as shown

in Figure 1.2-19 (ρ , u , d_h , f and d are, respectively, the fluid density, mean flow velocity, hydraulic diameter, frequency, and rod diameter). The curves shown in Figure 1.2-19 are for wall pressures measured at an axial position of $x/\ell = 0.625$ for three different hydraulic diameters with no upstream spoiler.

Results obtained from earlier tests at ANL^{1.2-3} and from airflow tests at the University of Illinois^{1.2-4} are also plotted for comparison. In general, all the data agree quite well when fd/u is greater than 0.4; however, the values of the present results, except in the low frequency range, are slightly less than those of the earlier tests at ANL and of airflow tests at the University of Illinois. The lower values may be due to the "size effect" of the transducer. Inadequate spatial resolution of transducers with non-zero sensing area results in attenuation of high-frequency, small-scale pressure fluctuations owing to a cancellation of pressure fluctuations on the face of the transducer.

Based on the assumption of a spatially-homogeneous, convecting wall pressure field that obeys Corcos' phenomenological model for cross-spectral density, the rms displacement response of a rod in nominally axial flow is given as^{1.2-5}

$$y_{\text{rms}}(x) = \frac{(0.018)d^{1.5}\phi_1(x)\phi_0^{0.5}}{(M+m)\ell^{0.5}f_1^{1.5}\zeta_1^{0.5}}$$

where d is rod frequency, ϕ_1 is fundamental mode, M is added mass of fluid per unit length, m is rod mass per unit length, ℓ is rod length, f_1 is fundamental mode frequency, ζ_1 is fundamental mode damping factor, and $\phi_0 = \phi_0(u)$ is the magnitude of the power spectral density of the turbulent wall pressure fluctuations in the low frequency range and is assumed to be a constant for a given mean axial flow velocity. By using this equation and the data obtained

in the structural vibration tests of this program (Figure 1.2-15 and Reference 1.2-6), the averaged magnitude of the wall pressure PSD, Φ_o , required to give the measured values of rms rod response was computed and is given in Figure 1.2-19 for comparison. This analysis exercise is based on the assumption that the rod response is only caused by the turbulent boundary layer pressure fluctuations. As can be seen from Figure 1.2-19, the calculated value falls within the envelope of the pressure data but must be considered to be on the "high side." This indicates that sources of rod excitation in addition to turbulent wall pressure fluctuations may be contributing to rod motion.

Analysis of the data showed the intensity of the wall pressure PSDs to vary with mean flow velocity to a power of 2.2 to 3.7 for the no-spoiler case and to a power of 2.2 to 4.4 for the case of grid A. In general, the value of the exponent increased with decreasing hydraulic diameter and with decreasing transducer pair location. The changes in the exponents were different for different upstream conditions, but the total changes are so small as to be inconsequential.

b. RMS Pressure Coefficient

The RMS pressure coefficient is defined as

$$C_p = \frac{\sqrt{p^2}}{\frac{1}{2} \rho u^2}$$

where $\sqrt{p^2}$ is the root-mean-square value of wall pressure, and ρ and u are fluid density and mean flow velocity, respectively. The rms values of wall pressure were obtained via integration of the mean-square spectra of pressure difference signals over the frequency domain. A low-pass filter with cutoff frequency of 10 kHz

was used to exclude pressure signals from the high frequency region, and the extraneous structural straining effects at low frequency were accounted for by subtracting the strain induced energy occurring at the natural frequencies of the rod vibration.

In Figures 1.2-20 through 1.2-22, rms wall pressure coefficients are plotted, together with results from the literature, as a function of Reynolds number, $u d_h / \nu$, where ν is kinematic viscosity of water. As shown in Figure 1.2-20, the rms coefficient of pressure measured at the third transducer location ($x/l = 0.625$) without upstream spoiler is less than the results of earlier ANL tests (specifically transducer pair 2-5) for the case of no upstream spoiler. However, the discrepancy between the two is much reduced at higher Reynolds numbers ($Re \geq 2 \times 10^5$). The results of Wilson/Jones data and that of Willmarth, et al.,^{1.2-7} when plotted on the same figure, lie in the low Reynolds number ($Re < 2 \times 10^5$) region on the curve, and, for the lower Reynolds numbers, are below the RMS pressure coefficients of the present study. However, in the case of Willmarth, et al., contributions to the wall pressure fluctuations from frequencies below 250 Hz were excluded from the data; and consequently, the rms pressure coefficients computed from their data may be expected to be smaller.

The influence of upstream spoilers on the values of rms pressure coefficients is shown in Figures 1.2-21 and 1.2-22. Near the flow entrance (i.e., at $x/l = 0.125$), the rms pressure coefficients are affected by the upstream spoilers, as shown in Figure 1.2-21. The grid spoiler has a greater effect than the other types of spoilers. Further from flow entrance (namely, at $x/l = 0.625$), the effect of the upstream spoilers on rms pressure coefficients is not so apparent, as shown in Figure 1.2-22. This implies the upstream disturbances created by the flow spoilers decay very fast in the axial direction so that their effect on rms wall pressure is dominant only near the entrance of flow. This result is consistent with turbulence measurements previously reported.^{1.2-6}

The ratios of rms wall pressure fluctuations to the wall shear stress are plotted in Figure 1.2-23 for data obtained at $x/l = 0.625$ for three different hydraulic diameters and no upstream spoiler. The value of this ratio varies between 1 and 3, depending upon the value of hydraulic diameter (the smaller the hydraulic diameter, the lower the value of this ratio). The present results are again less than those obtained from earlier tests at ANL, but the result for $d_h = 0.96$ in. is close to that of Corcos.^{1.2-8}

c. Convection Property of Flow Field

The convective velocity of the flow field can be obtained from cross-correlations between two pressure difference signals from transducer pairs located at two different axial locations. Attempts were made to cross-correlate adjacent transducer pairs with the result that no correlation between any two pairs was obtainable. This result was not unexpected since the separation of any two transducer pairs in present tests is sufficiently great that an eddy of certain size sensed at one transducer location will have decayed away before reaching the next transducer location. The fact that no correlation was achieved provides further indication that the acoustic noise has been successfully subtracted. Having a long-wave length, the acoustic noise would be expected to be correlated over the axial separation of the transducers.

Activity 2 - Annular Region (Air) (B. G. Jones, R. J. Wilson)

Task Description

Two fundamental parallel FIV studies are being conducted, one in which both the wall and fluid field fluctuating pressures will be measured simultaneously, and the other in which both the wall fluctuating pressures and the turbulent velocity field will be measured simultaneously. The first will allow determination of the stochastic structure of the pressure field and a look for

correlation relationships between the fluid stochastic pressure field and that at the wall. The second study will allow determination of the turbulent velocity field and examination of its relationship to the stochastic wall pressures. These studies will be conducted in air flows in which a simply supported central rod is mounted in a circular tube to provide a concentric annular flow field.

The research program is centered around four interrelated experimental objectives to be carried out in a concentric single pin annular geometry turbulent air flow:

- (1) Measure turbulent velocity intensity levels and mean velocity profiles in the flow region;
- (2) Measure, and compare with existing analytical data, pressure correlation in the flow region;
- (3) Measure the correlation between the fluctuating flow field pressure and wall pressure fluctuations on the rod; and
- (4) Measure the correlation between the fluctuating flow field velocity and wall pressure fluctuations on the rod.

This fundamental study should result in a better quantitative and qualitative understanding of the flow phenomena which generate the wall pressure field.

Work Completed This Report Period

A single wire hot-film sensor was used to obtain preliminary velocity data. The mean velocity profile is shown in Figure 1.2-24. The local turbulence intensity across the flow field is shown in Figure 1.2-25. Both results agree quite well with experimental results of Brighton and Jones^{1.2-1} and the analytical results of Schumann.^{1.2-9}

The energy spectrum of the turbulent velocity field is shown in Figure 1.2-26. These results show that fully developed turbulent flow exists in the test section. It also shows that the turbulence level is not abnormally high and, thus, the pressure levels should be consistent with other data of similar flow geometries.

The wall pressure transducers were fully calibrated and mounted in the test section. On the initial run of the flow system, a large, 6800-Hz sine wave masked out the rest of the signal. This 6800-Hz signal was generated by the Helmholtz cavity used to pass the wall pressures to the pressure transducers. To alleviate this problem, the cavities were filled with porous foam which was flush with the wall and the transducers were recalibrated. This completely destroyed the 6800-Hz signal without loss of sensitivity or frequency response.

The fluctuating pressure measured at any point in the flow field, including the walls, consists of both an acoustic pressure fluctuation and a local turbulent fluid pressure fluctuation. In the case where the acoustic pressure wave is large in magnitude with respect to the turbulent fluid fluctuation, it is virtually impossible to directly obtain the correct auto-correlation, PSD and rms pressure level. Since the acoustic noise arises from flow field turbulence far upstream (i.e. flow by obstructions, through restrictions and through expansions), the frequency bandwidth lies within the frequency bandwidth of the local hydrostatic pressure fluctuations which are to be measured. This makes it impossible to separate the hydrostatic and acoustic pressure signals from a single sensor.

Various loop modifications were made in an attempt to reduce the acoustic noise level. These modifications included the addition of an acoustic noise muffler, eliminating three short radius elbows, channeling of flow through the remaining elbows, and the addition of an exit diffuser. However, even with the modifications, the resulting acoustic noise was sufficiently high that it still masked the turbulent boundary layer wall pressure fluctuations. This necessitated the use of more than one sensor to separate the super-imposed acoustic and hydrostatic pressure information.

A data analysis technique involving the use of three wall pressure sensors (two on the flow channel wall and one on the cylinder wall) was devised.

The analysis method is based on the following assumptions:

- (1) The acoustic pressure is equal in magnitude and phase at all points in any axial plane;
- (2) There is no correlation between the hydrostatic pressure and the electronic instrument noise;
- (3) There is no correlation between the hydrostatic pressures measured at two diametrically opposite positions in a given axial plane; and
- (4) There is no correlation between any two transducer electronic noise signals.

The method allows for expressing the covariance of the hydrostatic pressure at a point in terms of the covariances of the three possible pressure difference signals and that of the electronic noise associated with the sensor located at the point of interest.

The RMS pressure level is usually nondimensionalized by either the dynamic pressure, $q_\infty = 1/2 \rho U_{\max}^2$ or the wall shear stress, τ_w . Typical ranges as reported by Willmarth^{1,2-10} are 4.5 to 7.5×10^{-3} for p_{rms}/q and 2.0 to 3.5 for p_{rms}/τ_w . These results are for measurements on flat plates, on the outer surface of a rod in a wind tunnel and circular pipe flow.

The RMS wall pressure level normalized by the dynamic pressure is shown in Figure 1.2-27 for different flow velocities. This data was acquired by use of the signal analysis method previously outlined. The maximum velocity was used in place of the mean velocity, as it most closely parallels the use of the mean velocity at large distances from the wall in the boundary layer work which was used for comparison.

The data of Figure 1.2-27 shows that the RMS level is lower on the inner wall than on the outer wall. This is in agreement with the predicted result of Schumann^{1.2-9} and Brighton and Jones,^{1.2-1} which showed that the difference increased as R_1/R_2 decreased.

The PSD of the wall pressure, as measured with the signal subtraction analysis method, is shown in Figure 1.2-28 for both the inner wall and the outer wall for a mean flow velocity of 200 ft/sec. The PSD, $F_{pp}(f)$, is nondimensionalized by the fluid density, ρ , the maximum velocity, U_{max} , and the hydraulic diameter d_h . This is the nondimensionalization used by Willmarth and Wooldridge^{1.2-11} and is used here for comparison purposes and to show that the spectra adequately collapse for different flow velocities using these variables.

As can be seen in Figure 1.2-28, the peak at 3.4 kHz is still in existence using the signal subtraction data analysis method. It is denoted by the solid symbols to infer that it exists in the data but is acoustic in origin as noted above. The signal subtraction would not eliminate this acoustic contribution to the sensor signal. The only explanations for this are either that the sensors were out of phase at this frequency or that the acoustic pressure wave was originating too close to the point of measurement to be of the same magnitude and phase at all radial and azimuthal locations in the axial plane of the wall pressure transducers. Since the phase response of the transducers was checked and no discrepancies were evident at this frequency, it was assumed that the latter explanation was the case even though no source could be identified.

Since the 3.4-kHz peak was due to acoustic contamination, it was removed from all measurements by assuming the PSD of the hydrostatic pressure fluctuations was smooth and continuous across the frequency range of the acoustic resonance peak. All of the PSDs of the pressure measurements were corrected to eliminate this resonance peak.

The solid line drawn in Figure 1.2-28 is the assumed normalized PSD used by Chen and Wambsganss.^{1.2-12} This is a representative PSD taken from the

data of Bakewell^{1.2-13} for a body of revolution in water flow, Clinch^{1.2-14} for water pipe flow, Willmarth and Wooldridge^{1.2-11} for air flow on a flat plate and Wambsganss and Zaleski^{1.2-15} for annular water flow. The agreement between the assumed PSD and that of the outer wall is quite good. However, the measured inner wall PSD is somewhat lower in the low frequency end of the PSD than the assumed PSD.

Figures 1.2-29 and 1.2-30 show the PSD of the pressure fluctuations at the inner and outer walls, respectively, for five mean flow velocities. The PSD's collapse in magnitude for the nondimensionalization previously noted and in frequency for the Strouhal number based on the hydraulic diameter (i.e., fd_h/U_{max}). The inner wall PSD has a somewhat lower magnitude in the low frequency range than the outer wall PSD. This could be due to the different curvatures of the two walls which yield different effective flow volumes. The point of maximum velocity lies nearer to the inner wall than to the outer wall. It would appear that the largest eddies would be of the order of the distance from the wall to the point of maximum velocity. If this is the case, there would be more larger scale, lower frequency fluctuations near the outer wall than there would be near the inner wall.

The PSD of the inner wall has a much sharper cutoff frequency than that of the outer wall PSD. The cutoff frequency corresponds to $fd_h/U_{max} = 2.5$, which is in agreement with the information given by Shin.^{1.2-16} The outer wall pressure PSD also cutsoff near $fd_h/U_{max} = 2.5$, but is not as pronounced as that of the inner wall.

Both the inner and outer wall pressure PSDs are very flat in the low frequency end of the spectrum. The measurements of Figures 1.2-28 through 1.2-30 are for frequencies as low as 10 Hz (high pass filters were set at 2 Hz).

It is felt that the low frequency end of the PSDs of Figures 1.2-28 through 1.2-30 accurately portray the frequency content of the hydrodynamic pressure

fluctuations, as the system was designed specifically for the low frequency measurements and there is a high confidence level in the data at low frequencies. Thus, it would appear that the PSDs used for input to vibration models should be flat in the low frequency range, similar to the model used by Chen and Wambsganss.^{1.2-12} This information would appear to be crucial to vibration models, as this is the frequency range in which the fundamental frequencies of flow-induced vibrations occur. Since the origin of the driving mechanism lies in the fluid turbulence, it would seem that the frequency range of the turbulence which is of most importance to the excitation of flow-induced vibrations would be of the same order of magnitude as that of the fundamental structural vibration.

Both the inner wall and outer wall pressure PSDs collapse at high frequencies. The slope is approximately $f^{-9/3}$ above the cutoff frequency. This slope is consistent with the PSD information given by Chen and Wambsganss^{1.2-12} which shows the high frequency fluctuations dropping off at about $f^{-9/3}$ for several different wall pressure measurements.

It should be noted that these PSDs have not been corrected for high frequency attenuation as discussed by Willmarth.^{1.2-10} A correction would have the effect of increasing the high frequency end of the spectra which would shift the slope slightly toward $f^{-7/3}$.

In summary, the pressure signal analysis method presented does a good job of separating the fluctuating hydrostatic pressure signal from the combined hydrostatic and acoustic pressure information when the flow field and acoustic pressure field meet the criteria set forth in the analysis development. The PSD information presented at low frequency appears to be an addition of new information in this region of the spectrum. It also shows that the wall pressure fluctuations are significantly different on the two walls of the flow channel. This infers that data from different flow geometries cannot be interchanged for input to vibration models without inserting an unknown error which may be of significant magnitude. For example, the data presented above

shows that the energy in the low frequency end of the PSD is about a factor of 2 larger on the outer wall than on the inner wall. The implication is that the magnitude of the fluctuating hydrostatic pressure in the important low frequency end of the spectrum (the important fluid fluctuations would be expected to be in the frequency range of the rod vibrations themselves - less than a few hundred hertz) can vary widely in magnitude for different flow geometries.

Task 1.2c - Fluidelastic Interaction in Parallel Flow (W. H. Lin, T. T. Yeh, T. M. Mulcahy, B. G. Jones, W. Nagel)

Task Description

To develop a better understanding of the fluidelastic interaction that can result in coupled response and/or instabilities in parallel flow systems, a single rod asymmetrically positioned in both a circular and square flow channel is being investigated. Wall pressure fluctuations and rod response will be measured for a range of Reynolds numbers, hydraulic diameters, reduced velocities, and damping in order to provide the basis for developing a mathematical model to predict response and scaling relations. As with the annular flow experiments of Task 1.2b, complementary studies, performed in water in the ANL 500-gpm flow facility and in air using the University of Illinois open cycle air loop, will be carried out.

Work Completed This Report Period

Activity 1 - Water Tests (ANL)

Fabrication of the test model, with the exception of upstream flow channel extensions, has been completed. The model is designed to allow for changing the asymmetric location of the rod without having to remove it from the test section. Consequently, the structural damping, which can be expected to be heavily dependent on the support conditions, should remain the same. Instrumentation mounts for measuring acceleration and internal solenoid coils for external excitation have been constructed.

Activity 2 - Air Tests (Univ. Ill.)

Continued study of parallel flow-induced rod vibration was conducted in air flow through an eccentric annulus. During this reporting period, detailed measurements of wall pressure and flow static pressure fluctuations as well as transverse and axial fluctuating velocities were performed and correlated at several locations in the flow geometry. Initial examination of these results, without using multiple probe coherence techniques to reduce noise contamination, has been completed. The study of this flow configuration was performed to ascertain, primarily, what perturbation the eccentric rod location would make on the results obtained for the annular flow (Task 1.2b). Simple signal subtraction techniques were employed to reduce the effects of acoustic noise in the results. As a result of not employing the more accurate, but considerably more time consuming, coherence methods, the spatial trends and absolute intensity and covariance results may not be correct. However, the data includes a relatively detailed set of spatial measuring locations and several cross correlation estimates so that future studies can be more selective, thus reducing the amount of data required to clarify trends and effects. A few preliminary results using the full coherence technique have been obtained, but not in sufficient detail to comment on the significance with respect to magnitude and trends of improved data processing.

The primary objective of the next quarter is to complete the assessment of the validity and accuracy of the present results and to draw firm conclusions relative to the effects of eccentric versus concentric annular flows with respect to wall pressure fluctuations. It is anticipated that these results will directly lead to specific dependencies which will require further study:

- (1) flow rate variation;
- (2) annular radius ratio variation; and
- (3) eccentricity variation.

Additionally, power spectral density dependencies on these parameters will be studied to determine the appropriate scaling laws for data collapse and intensity level predictions.

Task 1.2d - Experimental Study of Tube Arrays in Parallel Flow

Task Description

Task 1.2d consists of an experimental study of the wall pressures and response of rods in seven-tube and four-by-four tube arrays. As in Task 1.2b, the ultimate goal is to develop an analytical response prediction model characterized by the test data. The independent test parameters include a wide range of Reynolds numbers and upstream flow disturbances.

Work Completed This Report Period

To obtain insight into the behavior of rod bundles in parallel flow, a seven-rod array on an equilateral triangular pitch and a four-by-four rod array on a square pitch will be tested. The fabrication of the test hardware for the seven-rod array is nearly completed. Two locations for the instrumented rod have been selected for the seven-rod tests; one is in the center of the array and the other is in a peripheral position. Test specifications and procedures have been defined and test element fabrication is in progress.

Task 12.e - Development of Mathematical Models and Design Guides

Task Description

Experimental results from subtask activities 1.2b and 1.2d will be used to evaluate and improve prediction methods for parallel flow induced vibration. The methods will be further evaluated via comparisons of predicted response with results from model and full-sized tests (Task 2). Critical flow velocities associated with the onset of fluidelastic instabilities will be expressed in terms of system parameters. Validated design guides will be developed for inclusion in an FIV Handbook.

Work Completed This Report Period

No work except continuing review of the FIV literature was performed during this report period.

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Table 1.2-1
ANALYTIC PREDICTION OF FREQUENCIES
FOR FIXED SIMPLE END CONDITION

	<u>Vibration Mode</u>	
	<u>1</u>	<u>2</u>
Eigenvalue	3.93	7.07
f_n (Hz), air	31.6	102.4
f_n (Hz), $D_n = 3.71$ cm	28.7	93.0
f_n (Hz), $D_n = 1.17$ cm	27.4	88.7

Figure 1.2-1. Mean Velocity Distribution at $z/D_h = 26$ ($z/l = 0.8$) for the Largest Annular Region, $D_h = 3.71$ cm, and No Grid Compared to Data of Brighton^(2.1-1)

Figure 1.2-2. Turbulence Intensity Axial Decay for Points Close to the Rod in the Largest Annular Region and Different Grids

Figure 1.2-3. Turbulence Intensity Axial Decay for Points Close to the Rod in the Largest Annular Region and Different Grids

Figure 1.2-4. Turbulence Intensity Axial Decay for Points Halfway Across the Largest Annulus and Different Grids Positioned at Location A

Figure 1.2-5. Turbulence Power Spectral Densities for Points Close to the Rod at Two Axial Positions in the Largest Annular Region and Different Grids Positioned at Location A

Figure 1.2-6. Turbulence Power Spectral Densities for Points Halfway Across the Largest Annulus at Two Axial Locations and Different Grids Positioned at Location A

Figure 1.2-7. Turbulence Intensity Axial Decay for Points Halfway Across the Smallest Annulus and Different Grids at Location A

Figure 1.2-8. Turbulence Power Spectral Densities for Points Halfway Across the Smallest Annulus at Two Axial Locations and Different Grids Positioned at Location A

Figure 1.2-9. Turbulence Integral Scale Axial Variations for Points Halfway Across the Smallest Annulus and Different Grids at Location A

Figure 1.2-10. Typical Noise Reduction of the Active Filter

Figure 1.2-11. Comparison of the Rod Displacement Variation with Mean Velocity for the Largest Annular Region with No Grid, with Grid A at Location A, and with Grid A at Location B

Figure 1.2-12. Rod Displacement as a Function of Mean Velocity for the Largest Annular Region and Different Grids Positioned at Location A

Figure 1.2-13. Fundamental Rod Frequency at Different Mean Velocities for the Largest Annulus and Different Grids at Location A.

Figure 1.2-14. Variation in Rod Modal Damping with Flow Velocity in the Largest Annulus for Different Grids at Location A

Figure 1.2-15. Comparison of the Rod Displacement Variation with Mean Flow Velocity for the Smallest Annular Region and Different Grids at Location A and No Grid

Figure 1.2-16. Pressure Transducer Locations on Central Rod

Figure 1.2-17. Wall Pressure PSD; $d_h = 0.96$ in., $x/l = 0.625$

Figure 1.2-18. Wall Pressure PSD; $d_h = 0.46$ in., $u = 49.2$ ft/sec

Figure 1.2-19. Normalized Wall Pressure PSD

Figure 1.2-20. RMS Pressure Coefficient vs. Reynolds Number

Figure 1.2-21. RMS Pressure Coefficient vs. Reynolds Number

Figure 1.2-22. RMS Pressure Coefficient vs. Reynolds Number

Figure 1.2-23. Normalized RMS Pressure vs. Reynolds Number

Figure 1.2-24. Velocity Profile

Figure 1.2-25. Local Turbulence Intensity

Figure 1.2-26. Velocity Spectra

Figure 1.2-27. Wall Pressure Intensity Level as a Function of Mean Flow Velocity

Figure 1.2-28. PSD of the Hydrostatic Wall Pressure and the In-stream Pressure Showing the 3.4 kHz Acoustic Peak Which Could Not Be Electronically Removed

Figure 1.2-29. PSD of the Hydrostatic Pressure Fluctuations at the Inner Wall for Several Different Mean Flow Velocities

Figure 1.2-30. PSD of the Hydrostatic Pressure Fluctuations at the Outer Wall for Several Different Mean Flow Velocities

TASK 1.3 - LEAKAGE FLOW MECHANISMS (CR&D)

Task 1.3 performed by General Electric Corporate Research and Development, is designed to establish a better understanding of leakage-type flow-induced vibrations for geometries common to light water reactors. The technical objectives include the following:

- Mathematically model and classify components involving leakage flows. Identify flow/geometric and structural features leading to an instability.
- Experimentally illustrate the general classification with critically selected experiments.
- Contribute input to design guides for critical reduced velocity $U/f_n L$ as a function of flow/geometric and structural features to eliminate or reduce possibility of self-induced vibrations. Proof test the design guide by application to a selected LWR component.

Task Description

Task 1.3 is divided into four subtasks:

Task 1.3a - Modeling and Classification of Classes of Components with Leakage Flow-Induced Vibration Susceptibility

Task 1.3b - Simple Experiments to Illustrate Concepts and Governing Parameters Identified by Mathematical Models

Task 1.3c - Proof Test of Evolving Design Guides on a Selected LWR Component

Task 1.3d - Leakage Flow Induced Vibration Design Guide

Work Completed This Report Period

No work was performed on this activity during the report period.

TASK 1.4 - RANDOM VIBRATION EFFECTS (ANL)

Task 1.4 consists primarily of a literature survey to review existing random vibration fatigue damage rules and experimental data related to interpretation of FIV data. Based on this review, a suitable random fatigue damage criteria will be developed and recommended for application to LWR components subjected to flow-induced vibration.

Task Description

Task 1.4 is broken into three subtasks:

Task 1.4a - Survey of Literature

Task 1.4b - Derivation of Procedure/Criteria

Task 1.4c - Design Guides

Work Performed This Report Period

Task 1.4a, which consists of a technical literature review to identify the existing methods of random fatigue damage analysis applicable to fatigue life prediction of LWR components experiencing flow-induced vibration, has been completed and a topical report issued. Results were summarized in FIV for LWR Progress Report January - March 1978 (COO/4175-9). Efforts are now being directed to the application of the random fatigue damage analysis techniques to the evaluation of actual LWR component FIV data (specifically, the lower plenum FIV data generated in the (HF)² 60° sector model).

TASK 2 - MODEL AND FULL-SIZE COMPONENT TESTS

Task 2 is devoted to the FIV testing of realistic model and prototypical geometries, rather than the idealized geometries studied in Task 1. Model and full-scale testing of several LWR components will be performed to extend the Task 1 results and provide input for the development and verification of empirical relations, scaling laws, and predictive models generated as part of Task 3. The components tested will, in general, be from General Electric's most recent boiling water reactor product line, the BWR/6. Task 2 is divided into five component test programs:

Task 2.1 - Lower Plenum

Task 2.2 - Fuel Assembly

Task 2.3 - Jet Pump

Task 2.4 - Lower Pressure Coolant Injection Line

Task 2.5 - LWR Component (undefined)

Progress achieved for each of these component tests for the report period is described on the following pages.

TASK 2.1 - LOWER PLENUM (NTD)

Task 2.1 is an experimental FIV study of tubes in an array. The array is a mix of tubes with different diameters. The flow environment combines regions of cross flow and parallel flow. The results from this task will be evaluated using results from the fundamental studies (Tasks 1.1 and 1.2) in order to highlight the important FIV mechanisms in real LWR components. The first steps in this study were to perform FIV tests on tenth-scale and quarter-scale models of a BWR lower plenum. This provided preliminary data and the experience necessary to run a productive full-scale model test. The final step in this process is to measure vibration in a LWR before power generation startup. This will verify the more extensive full-scale model results.

Task 2.1 consists of four subtasks:

Task 2.1a - Lower Plenum One-Tenth Scale FIV Test

Task 2.1b - Lower Plenum Quarter-Scale FIV Test

Task 2.1c - Lower Plenum Full-Scale FIV Test

Task 2.1d - Lower Plenum Operational LWR Field FIV Test

Progress achieved on Task 2.1 is described on the following pages.

Task 2.1a - Lower Plenum One-Tenth Scale FIV Test (M.A. DeCoster, S. L. Kushman)

Task Description

A one-tenth-scale model of a BWR lower plenum was constructed on clear cast epoxy to study the flow patterns in the lower plenum via flow visualization in 1974. Later, velocity probes were inserted into the flow stream to obtain quantitative measurements of the flow distribution. In 1976, flow-induced vibration tests were initiated to determine the vibrational characteristics of control rod guide tubes in the lower plenum. The tests were completed late in 1976.

Work Performed This Report Period

The one-tenth-scale lower plenum model test and analysis program was completed during a previous report period. Results were documented in GE report GEAP-24124, "Flow-Induced Vibration for Light Water Reactors Progress Report (June - September 1977)."

Task 2.1b - Lower Plenum Quarter-Scale FIV Test (M.A. DeCoster)

Task Description

A quarter-scale model of a BWR lower plenum was constructed by GE in 1974 to study flow distribution and vibration levels of lower plenum components. The model was situated at the Colorado State University Hydro-machinery Laboratory. Flow and shake tests performed in 1974, 1975, and 1976 yielded extensive flow distribution vibration data for various designs. The final tests were completed in the last quarter of 1976.

Work Performed This Report Period

The quarter-scale lower plenum model test and analysis program was completed prior to the commencement of the Flow-Induced Vibration for Light Water Reactors program. Results were documented in GEAP-24124.

Task 2.1c - Lower Plenum Plenum Full-Scale FIV Test (M.A. DeCoster)

Task Description

Task 2.1c consist of the FIV testing of prototype LWR components in the lower plenum of the 60° sector model at the High Flow Hydraulic Test Facility. The test components were made at the same facility and to the same standards as actual reactor components. The components are full scale but the sector model contains one-sixth the number of components in a reactor. The components in the lower plenum are cylinders of various diameters in an array pattern. Water flows around the outside of these components.

The test program consists of a series of shake tests followed by flow tests. In the shake tests, a single component is forced to vibrate by an electromagnetic shaker or impact hammer. The input force and resulting motion are measured, and the information is processed to define the structural characteristics of the component. In the subsequent flow tests, the vibration response of the components, as well as certain aspects of the FIV forces, are measured for a variety of operational and nonoperational flow conditions.

Work Performed This Report Period

Both the shake tests and flow tests were completed on the BWR/6 lower plenum components in the High Flow Hydraulic Test Facility, (HF)², sector model. The shake and flow tests were completed late in 1979. The lower plenum components and layout are summarized in Figures 2.1-1 and 2.1-2.

During the shake tests, 156 vibration sensors on the in-core guide tube and stabilizer grid assembly were monitored. A single shaker was used with sinusoidal force inputs from 1 to 20 lb swept from 2 to 500 Hz. The assembly was tested in air and under water. Impact tests using an instrumented hammer were also performed on the in-core assembly and differential pressure line in air.

The flow tests were done in two parts. The first part was completed on October 14, 1979. In this part, 316 vibration sensors were monitored in all the lower plenum components (control rod guide tube, control rod drive housing, in-core stabilizer grid, in-core guide tube, and the differential pressure line). The 316 sensors were split into two groups of 160 for simultaneous recording of the data. Two flow conditions were simulated (one- and two-recirculation-pump operation of the reactor). Because one of the test facility pumps was not working properly, the maximum flow rates attainable were 120% of reactor conditions in the two-pump mode and 111% of reactor conditions in the one-pump mode. The second part of the flow tests was performed after all the test facility pumps were working correctly. For this test phase, only the one pump operation mode was studied, and the vibration monitoring

was limited to 160 sensors, consisting primarily of strain gages and a few accelerometers. Test data was acquired for flow rates up to 133% of reactor conditions in the one-pump mode.

Data reduction is continuing. A comprehensive test report is being prepared preliminary results are presented on the following pages.

Shake Test Results

The primary purpose of the shake tests was to verify that the in-core assembly in the sector model was structurally similar to a reactor. Because only a 60-degree sector of the stabilizer grid could be reproduced in the sector model, a special wall connection in the sector model was required in order for the fundamental modes and frequencies to match those of a full-size reactor. The shake test results are summarized in Table 2.1-1. They show that the sector model assembly natural frequencies were very close to the theoretical predictions in the model and reactor. This means that the partial in-core assembly in the sector model is a good structural simulation of the complete assembly in a reactor.

Flow Test Results

The primary purpose of the flow tests was to determine the FIV characteristics of each vibrating component at reactor operating conditions. A survey of strain amplitude measurements for the two-pump mode, 100% of reactor velocity flow conditions, as well as a sample time history, probability density, and spectrum, are presented in Figures 2.1-3 through 2.1-9.

It should be noted that these are true RMS amplitudes from a 2-minute data record with a data bandwidth of 0 to 500 Hz. An interesting observation from these strain data is that the in-core guide tube vibration response appears to be fairly strongly related to position within the plenum (those closest to the jet pump) entering flow having the highest levels (Figure 2.1-4), while the control rod guide tube response is somewhat independent of position (Figure 2.1-6).

In general, the vibration amplitude varied as the jet pump exit velocity raised to a power between two and three (see the sample plot in Figure 2.1-10). Table 2.1-2 lists the largest strain amplitude in each component under both the one-pump mode and two-pump mode (at 100% of reactor conditions). This table also contains the maximum strains measured in the components in the quarter-scale model. The vibration results of the quarter-scale model and full-scale test at $(HF)^2$ are essentially the same.

Table 2.1-1

COMPARISON OF MEASURED AND CALCULATED FREQUENCIES OF THE IN-CORE ASSEMBLY

<u>Measured Frequency</u>		<u>Measured Damping (%)</u>		<u>Calculated Frequency, Sector Model</u>		<u>Calculated Frequency, Reactor</u>
<u>Wet</u>	<u>Dry</u>	<u>Wet</u>	<u>Dry</u>	<u>Wet</u>	<u>Dry</u>	<u>Wet</u>
5.8	10.7	8.4	0.51	5.9	9.5	6.1
14.9	18.3	10.5	0.37	14.0	22.8	13.8, 14.9
18.8	28.9	0.22	0.54	18.8	30.2	18.0
25.0	48.8	0.18	0.55			
	56.0		0.51			
41.3	64.5	0.46	2.55	40.4	64.7	
47.0		0.40				
52.0		0.87				

Table 2.1-2

LARGEST RMS MICROSTRAIN IN LOWER PLENUM COMPONENTS

<u>Component</u>	<u>Full-Scale</u>		<u>Quarter-Scale</u>	
	<u>Two-Pump Mode</u>	<u>One-Pump Mode</u>	<u>Two-Pump Mode</u>	<u>One-Pump Mode</u>
In-core guide tube	10.7	16.4	7.8	16.9
In-core stabilizer grid tension	3.2	5.4	1.0	
In-core stabilizer grid bending	10.3	16.4	3.2	
Control rod guide tube	5.9	32.0	3.9	5.0
Control rod drive housing	11.5	21.1	0.7	
Control rod	7.4	3.8		
Differential pressure line	6.0	16.2	3.9	12.6

Figure 2.1-1. Lower Plenum in the (HF)² Sector Model (Cutaway Side View)

Figure 2.1-2. Lower Plenum in the $(HF)^2$ Sector Model (Top View)

Figure 2.1-3. RMS Microstrain Measured in the In-Core Stabilizer Grid

Figure 2.1-4. RMS Microstrain Measured in In-core Guide Tubes

Figure 2.1-5. RMS Microstrain Measured in the Differential Pressure Line

Figure 2.1-6. RMS Microstrain in the Control Rod Guide Tubes

Figure 2.1-7. RMS Microstrain Measured in Control Rod Housings

Figure 2.1-8. RMS Microstrain Measured in Control Rods

Figure 2.1-9. Typical Data from Lower Plenum Test

Figure 2.1-10. RMS Strain Amplitude Versus Flow Rate Example

TASK 2.2 - FUEL ASSEMBLY (NTD)

Task 2.2 is designed to fully investigate the FIV characteristics of a prototype fuel assembly. The primary objectives are to:

- (1) Establish the FIV character of a typical fuel assembly including mode shapes, frequencies, character of response, and assessment of the stability margin as a function of system damping and flow;
- (2) Determine the relationship between the FIV characteristics and coolant properties; i.e., coolant temperature (ambient versus LWR operational conditions) and quality (single-phase water versus two-phase water-steam); and
- (3) Proof-test the concepts and forcing function estimates devised in Task 1.

Task 2.2 will be useful in verifying and extending the fundamental studies on parallel flow over tube banks of Task 1.1. It will provide input to the analytical model development of Task 3.1 and the scaling development (importance of temperature) of Task 3.2.

Task 2.2 is divided into three test programs:

Task 2.2a - Fuel Assembly Cold FIV Test

Task 2.2b - Fuel Assembly Hot Test (optional)

Task 2.2c - Fuel Assembly Operational LWR Data Acquisition (optional)

A description of progress achieved during this report period is given in the following pages.

Task 2.2a - Fuel Assembly Cold FIV Test (L.S. Dorfman)

Task Description

Task 2.2a consists of the FIV testing of a prototype 8 x 8 fuel assembly over a wide range of hydraulic conditions using the General Electric Large Tank Hydraulic Loop. The assembly is heavily instrumented with accelerometers, pressure transducers, and strain gages to determine the dynamic characteristics of the fuel assembly components as well as to describe the forcing mechanisms. The test is performed at a water temperature of approximately 75°F.

The fuel assembly (Figure 2.2-1) is made up of 64 Zircaloy rods, each approximately 16-in. long, 1/2-in. in diameter, and 32-mils thick. The rods are spaced and supported in a square (8 x 8) array by a lower and upper tie-plate. The lower tieplate has a nosepiece which distributes coolant flow to the fuel rods. Seven spacers, evenly spaced along the rods, maintain rod-to-rod spacing. Each spacer provides a three-spring support for the fuel rod: two stiff springs contacting the rod at points azimuthally separated by 90 degrees, and a third flexible spring on the opposite side of the rod, azimuthally separated from the other two springs by 135 degrees. Three-point contact is maintained by establishing an initial preload (i.e., sizing the springs such that they are compressed when the fuel rod is first installed in the spacer). Finally, the fuel bundle is enclosed by a square channel box, causing the coolant flow to be almost exclusively parallel in nature.

Work Performed This Report Period

The cold flow-induced vibration testing of a prototypical 8 x 8 fuel assembly was completed early in the FIV for LWR Program, and documented in previous progress reports. Results showed the vibration response of the fuel rods, which are excited primarily by the parallel flow of the coolant, to be broad-band (30 to 100 Hz) with predominant frequencies at 45, 53, and 58 Hz and very low-level (typically less than 1.3 mil peak displacement) up to

136% of reactor volumetric flow. However, one of the parameters which was not addressed by this early test program was the effect of spacer preload on the vibration response of the fuel rods. This has recently become an important consideration, because of the desire for improved uranium utilization through extended exposure, and extending exposure would most likely produce a reduction in spacer preloads through radiation-enhanced relaxation. The effect of low preload could be to allow FIV-induced liftoff of the fuel rod from the spacer, resulting in rod-spacer relative motion and subsequent cladding wear.

A test program has been initiated to address this fuel FIV performance issue. The specific objective is to characterize the fuel rod FIV response as a function of the spacer preload conditions. Presently in the planning stage, the test will consist of monitoring the FIV response of an instrumented fuel rod at high flow rates as the preloads on the spacers are varied. The test will be performed in the Fuel Flow Induced Vibration Test Loop, located on the (HF)² test site.

Figure 2.2-1. Fuel Assembly

TASK 2.3 - JET PUMPS

Task 2.3 is designed to fully investigate and quantify the FIV characteristics of a jet pump assembly. A typical jet pump assembly is shown in Figure 2.3-1. There are 20 jet pumps in the current BWR design. Jet pump primary components are a mixer and diffuser which are joined together at a thermal expansion slip joint. Approximately one-third of the reactor core flow is taken from the vessel, pumped through a piping manifold system, and returned to the vessel, where it is discharged from the jet pump nozzle at high velocity (≈ 200 ft/sec) into the initial stage of the jet pump mixer. The nozzle flow entrains surrounding water into the jet pump throat where the two flows mix and then diffuse, to be finally discharged into the lower core plenum at velocities up to 40 ft/sec. The jet pump has many characteristics that can contribute to flow-induced vibrations (e.g., high-velocity flow in small pipes, high-velocity flow in a pipe turning 180° within a small radius, and high-velocity jets exiting into a confined region). However, the two characteristics which are judged to be the most significant contributors are, in order of magnitude:

- Leakage flow through the mixer-diffuser slip joint interface.
- Mixing of the high-velocity drive flow with the entrained suction flow in the mixer.

Jet pump tests will be performed hot (550°F) and cold ($40^\circ\text{F} - 200^\circ\text{F}$), under operational and nonoperational conditions, at rated and above-rated flows. Specific goals include the following: (1) identify the vibration characteristics of a typical jet pump (natural frequencies, mode shapes, damping); (2) characterize the primary forcing functions; (3) establish a jet pump response map (i.e., response versus flow conditions); (4) relate jet pump stability to system damping, flow conditions, geometry, etc., and (5) determine the effect of fluid temperature on the FIV characteristics. This work will be directly tied to the fundamental studies performed in Task 1 (in particular, the leakage flow mechanisms work of Task 1.3) for input into the Design Guide

Development work of Task 3.1. In addition, it will be utilized in developing temperature scaling relationships in Task 3.2.

Task 2.3 is divided into three subtasks:

Task 2.3a — Jet Pump Cold FIV Test

Task 2.3b — Jet Pump Hot FIV Test

Task 2.3c — Jet Pump Mechanistic Test

Task 2.3a — Jet Pump Cold FIV Test (J. F. Schardt)

Task Description

Four prototype BWR/6-238 jet pumps (two jet pump pairs) are flow-tested over a wide range of hydraulic conditions, using the High Flow Hydraulic Facility 60° Sector Test Stand (Figure 2.2-2). The jet pumps are heavily instrumented with accelerometers, pressure transducers, strain gages, and displacement sensors so as to determine the dynamic characteristics and vibration response, as well as to characterize the forcing function. Tests are performed at temperatures ranging from 40°F to 200°F.

Work Performed This Report Period

The major portion of the test program has been completed. This includes extensive shake tests to ascertain the system natural frequencies and mode shapes, followed by flow tests to characterize the FIV response. The test procedures and results are described on the following pages.

Shake Tests

Shake tests were performed on one of the two jet pump pairs in the (HF)² 60° sector model. Two different shake test configurations were used; (1) force applied in the radial direction at each of the mixer elbows; and (2) force applied in the circumferential direction at the top of the riser transition piece. Force locations were chosen based on the results of modal

analyses, as well as consideration of available space in the sector model for setting up the shaker equipment. The driving force was provided by 100-lb electrodynamic shakers mechanically attached to the jet pump assembly. Both sinusoidal (sweeps and dwells) and white noise inputs were utilized. The forces input to the jet pumps were monitored by PCB integrated circuit piezoelectric force gages.

The hardware for the circumferential and radial shake tests is shown in Figures 2.3-3 and 2.3-4, respectively. The shakers were suspended by elastic cords, rather than rigidly fixed to the vessel wall. This technique provided greater ease of alignment and allowed for the avoidance of data contamination as well as linkage failure which can occur when the shaker is rigidly fixed. Under this arrangement, force inputs on the order of 30 lb were achievable over the entire frequency range of interest (20 to 200 Hz).

Two shakers were used simultaneously for the radial tests to excite both in-phase and out-of-phase jet pump modes. As is shown in Figure 2.3-4, they were suspended in the annulus between the 60° sector simulated vessel wall and the outer vessel wall. This was done to keep them dry during the water shake tests, since only the sector vessel was filled during this phase of the test program. Because of the limited space available in the annular region, the shakers could not be oriented for direct application of the radial force into the jet pump elbow. Instead, the shakers had to be rotated 90° and a linkage used to transmit the load. This linkage, which was designed to allow rotation about a stationary point without adding damping or noise to the system, proved to be very successful.

Only one shaker was used in the circumferential test, the force being applied at the top of the riser transition piece. While the application of two shakers may have been preferable, the problems involved, such as the space limitations and having to immerse the shakers in water, outweighed the limited benefits.

The vibration response of the jet pumps was monitored using accelerometers at the locations shown in Figure 2.3-5. The sensors were oriented to pick up

radial, circumferential and vertical modes of vibration. Sensor locations were selected based on a jet pump modal analysis. The accelerometers used are uniaxial, biaxial, and triaxial piezoresistive devices manufactured by Entran Devices, Inc. The sensing element is a cantilever beam with four strain gages mounted on it to measure beam bending in a Wheatstone bridge electrical configuration. In general, the sensors are damped to 70% of critical. They are designed to measure accelerations between 0.01 and 200 g over a frequency range of DC to 1000 Hz. They are hermetically sealed and the leads are protected by 0.125-in.-diameter stainless steel hardline casings which are capacitance-welded to the jet pumps with stainless steel straps as they are routed to the vessel instrument ports.

The vibration signals were stored on magnetic tape for later analysis via the (HF)² computer-controlled data acquisition system. They were multiplexed, digitized, and then recorded in pulse code modulated (PCM) form. Data analysis was performed using the (HF)² FASST analysis system. This is a minicomputer-controlled signal processing system which is capable of providing printed and plotted results in a reasonably short time, and can also provide on-line viewing of data analysis in real-time. The output from the analysis included time histories, power spectral densities, cross correlations, probability densities, and RMS and peak values.

Examples of results generated from the shake test program are given in Figures 2.3-6 through 2.3-15. They included 20- to 200-Hz response spectra in the radial and circumferential directions, which came from sinusoidal sweeps, and the first three jet pump modes in water, identified from sinusoidal dwells. Also included are the first three predicted mode shapes, based on a finite element analysis using the structural dynamics code, SAP4G06. There is good agreement between the measured and predicted mode shapes for the first two modes. However, the third experimental mode is asymmetric and therefore deviates from the analytical prediction. In all three modes, the measured frequencies are within 15% of the predicted values.

The jet pump vibration response was found to be very sensitive to the conditions at the mixer-diffuser joint. Initially, the jet pumps were installed

such that the mixer and diffuser were not in contact at the slip joint, separated by a 4-mil nominal gap. However, shortly after commencement of shake tests, this gap was observed to disappear, with contact occurring in the radial direction (a condition that has been observed after many of the jet pump FIV tests). For subsequent shake tests, the jet pumps tended to behave as a linear system when excited radially, with the slip joint acting as a pinned joint between mixer and diffuser. The radial mode shapes and natural frequencies were similar to those calculated from the dynamics model, as one might expect, since the model assumes pinned boundary conditions at the slip joint. However, when excited circumferentially (perpendicular to the surface of mixer-diffuser contact), the jet pump response was somewhat nonlinear, with substantial slippage and banging occurring between mixer and diffuser at the slip joint. The resonant frequencies related to the circumferential modes tended to be affected by the excitation force level, and two of the first five mode shapes were asymmetric. Surprisingly, the predicted natural frequencies were still reasonably close to the analytical predictions.

Flow Tests

Flow-induced vibration tests were performed on the two jet pump pair assemblies (four jet pumps plus two risers) in the (HF)² 60° sector model immediately following completion of the shake test program. The prototypical reactor components were subjected to a wide range of hydraulic conditions to fully characterize the FIV response. Parameters which were varied include the drive flow, M-ratio (ratio of suction flow to drive flow), and temperature. Drive flow and M-ratio were varied, since these are known to have a direct effect on the vibration intensity of the jet pumps and are typically used to suction their vibration characteristics. The water temperature was varied simply to better understand how the vibration response scales with this parameter.

To define the vibration response as well as the forcing mechanisms during the flow tests, the jet pumps were instrumented with a combination of accelerometers, strain gages, and dynamic pressure transducers. The accelerometers

were the same piezoresistive devices used in the shake test, and were located at the slip joint regions and the mixer elbows (the points of maximum response), as shown in Figure 2.3-16. The strain gages were Ailtech weldables, attached at the locations shown in Figure 2.3-17 for the purpose of helping in the identification of the modes and displacement responses to the fluid excitations. Pressure transducers were installed on all four jet pumps at the locations shown in Figures 2.3-18 through 2.3-20 to satisfy the following objectives:

- (1) Jet Pump 2 was heavily instrumented with pressure sensors to measure the pressure fluctuations applied to the inner surface of the mixer and diffuser due to the combined drive and suction flows. The transducers are located on the mixer at distances $0.5D_m$, D_m , $2D_m$, $3D_m$, $5D_m$ and $8D_m$ from the suction flow inlet (D_m = mixer inlet ID) and on the diffuser at the distances D_d and $4D_d$ from the top of the diffuser, just below the collar (D_d = ID at top of diffuser). The greatest concentration of sensors is just downstream from the mixer inlet. It is here that the most unsteady mixing between suction and drive flow occurs, resulting in high turbulence intensities and consequent time-varying pressure forces on the mixer wall. Further downstream (approximately five mixer diameters from the mixer inlet), the fluctuating pressures are expected to be more homogeneous and much less intense.

At each axial position chosen for pressure measurements, four pressure sensors were mounted flush with the inner wall, two diametrically opposite to each other in the radial direction and two diametrically opposite in the circumferential direction. By subtracting signals from diametrically opposite sensors, the acoustic noise is removed. In addition, it provides information useful in estimating the net radial or circumferential force being applied at that axial location.

- (2) Jet Pump 3's mixer and diffuser were instrumented with an array of pressure sensors (Figure 2.3-19) in order to examine the spatial correlation of the pressure field both axially and circumferentially. Sensor locations were chosen to determine the spatial correlation at distances of 1D, 2D, 4D and 8D, both axially and circumferentially. A value of D equal to 0.75 in. was chosen as the smallest practical distance, considering the physical size of the pressure sensors themselves. The spatial correlation was expected to be significant only at separation distances less than the jet pump inlet diameter (6.4 in.).
- (3) All four jet pumps were instrumented with pressure transducers on the diffusers at the mixer diffuser slip joint. The objective was to better understand and to characterize the pressures in the slip joint region as the jet pumps make the transition from low-level to high-level vibration intensities (i.e., as the M-ratio is decreased to induce high-level vibrations).

The dynamic pressure fluctuation measurements were made with hermetically sealed flush-mounted integrated circuit piezoelectric pressure sensors manufactured by PCB Piezotronics, Inc. The sensing element is a high-sensitivity multiplate quartz crystal. A low-noise amplifier within the sensor converts the charge produced by the crystal to a low impedance analog output signal proportional to the measurand and compatible with most readout instruments. The sensor is capable of measuring pressure fluctuations from 0.01 to 100 psi over a frequency range of 3 to 1000 Hz. The sensor leads are enclosed in 0.125-in. OD stainless steel tubes to ensure watertight integrity.

The vibration data was acquired and stored using the computerized (HF)² Data Acquisition System. To date, much of the acceleration and strain data have been analyzed via the (HF)² FASST system and will be summarized below. The dynamic pressure data remains to be studied.

Hydraulic Characteristics

It was useful for later vibration analysis purposes to identify the relationship between M-ratio, drive flow, slip joint differential pressure and

temperature for the (HF)² jet pumps (all these parameters were carefully monitored and recorded throughout the test program). It was found that the hydraulic data could be collapsed by dividing the slip joint differential pressure by the fluid kinetic energy (Figure 2.3-21). By identifying through a process of trial and error a function which reasonably fit the collapsed data of Figure 2.3-21, the following formula relating the key thermal-hydraulic parameters was established:

$$\frac{SJDP}{\rho \left(\frac{W_D}{A} \right)^2 (1+M)^2} = e^{-1.55(M-2.8) - 1.55}$$

where

- W_D = Jet pump drive flow (normalized to 100% design operating flow of 4221 gpm)
 ρ = Water density normalized to 180°F density (1.883 slugs/ft³)
 A = Cross-sectional area of the mixer throat normalized to BWR/6-238 (32/17 in.²)
 M = M-ratio (ratio of suction to drive flow)
 SJDP = Slip joint differential pressure

As Figure 2.3-21 demonstrates, this formula is a good representation of the collapsed data over the M-ratio range of interest (approximately 1 to 2.8).

FIV Characteristics

The two (HF)² jet pump pairs exhibited FIV characteristics similar to those observed during tests performed previously on single jet pumps at the PG&E Moss Landing Large Steam-Water Test Facility. Probably the most significant similarity was the existence of three separate, somewhat distinct vibration regimes. For the (HF)² jet pumps, these may be characterized as follows:

(1) Low-Level Vibration Regime (<0.5g_{rms})

The response is fairly random and the energy is spread over a 30- to 200-Hz frequency band. The vibrations are caused by the turbulent buffeting of the internal flow.

(2) Transition Regime

The response is mid-level (approximately $0.5g_{\text{rms}}$ to $1.5g_{\text{rms}}$), and the energy is concentrated over smaller frequency bands, particularly in the 100- to 120-Hz region.

(3) High-Level Regime

The response is greater than $1.5g_{\text{rms}}$ and the energy is concentrated primarily at specific frequencies, seemingly preferring the 100- to 120-Hz region, followed at increased intensities by the 50- to 60-Hz and 200- to 240-Hz regions. The response is caused by leakage flow through the mixer-diffuser slip joint which apparently leads to fluid-elastic coupling.

The acceleration levels referred to in the definitions of the vibration regimes above are for the mixer at the slip joint (typically where the maximum jet pump accelerations occur). PSDs illustrating the gradual transition from one regime to another are shown in Figure 2.3-22.

The jet pump vibration response was found to be governed primarily by three interrelated parameters: (1) slip joint differential pressure; (2) water temperature; and (3) drive flow. If the water temperature and drive flow were maintained at constant levels, the three vibration regimes could be induced by simply sweeping the SJDP over a sufficiently wide range. At low pressure differentials, the vibrations would be low-level and relatively insensitive to changes in SJDP. As the pressure differential were swept upward, the intensity would eventually begin to rise, first gradually, as the jet pump entered the second regime, and, shortly thereafter, rapidly, as it entered the third regime. It appears that, at sufficiently high SJDPs, this third regime would tend to flatten, becoming somewhat insensitive to further pressure differential increases.

The effects of drive flow on the vibration response are somewhat erratic. Increased drive flow will increase the vibration intensity in the low-level

regime and the saturation level (level of flattening out) in the high-level regime. However, it has little effect on the transition from the first to second regime and appears to actually delay transition to the third. These characteristics are illustrated in Figure 2.3-23.

Test results seem to indicate that water temperature does not significantly affect the vibration intensity within each regime, but greatly affects the transition from one regime to another. As the temperature increases, the transition SJDP decreases, apparently in a linear relationship with the log of the fluid kinematic viscosity. For example, the relationship between the SJDP required to induce lg_{rms} at the slip joint (a vibration level midway in the transition from low to high-level vibration) and the kinematic viscosity for the (HF)² jet pumps was found to be:

$$SJDP = 40 (1 + 0.25 \lg v)$$

where v is the kinematic viscosity of the water. This relationship is derived from the (HF)² 70°F and 180°F results, as well as those from a test on a jet pump of similar design, performed at 532°F at the Moss Landing Pacific Gas & Electric Station, where the GE Large Steam-Water Test Facility is located. The data is shown in Figures 2.3-24 and 2.3-25. The obvious conclusion is that it is much easier to induce high-level (third regime) vibrations in jet pumps at reactor temperature (532°F) than at the (HF)² temperatures (70° to 180°F). This conclusion should also apply to any other reactor components having similar slip joint/piping geometries.

The (HF)² jet pumps exhibited a fairly significant change in behavior during the early stages of the test program. For example, initially the transition from low to high-level vibration under operating drive flow and 180°F water temperature commenced at an M-ratio of about 2.4 (SJDP ≈ 1 psi) and was very gradual. However, after about 100 hours of operation, the transition was observed to commence at about 1.9 (SJDP ≈ 20 psi) and to be much more dramatic (more like a step change), as Figure 2.3-26 illustrates. This modification in behavior (shift in transition to higher SJDP), which persisted through the remainder of the FIV test program, is apparently due to geometric changes

in the mixer-diffuser slip joint region. Initially, at the beginning of FIV testing, Jet Pumps 3 and 4 had a 3-mil minimum slip joint gap. Jet Pumps 1 and 2 had no gap (they were lost during shake testing); however, the shake test data analysis revealed the bearing loads between the mixers and diffusers at the slip joints to be low. Conversely, post-FIV test inspections revealed all four jet pumps to have mixer-diffuser radial contact at the slip joint, with preloads ranging from 200 to 850 lb. This gradual loss of gap and subsequent loading which occurred in the slip joint has been observed in other jet pump test programs and apparently is characteristic of what happens in the reactor during the early stages of startup (data indicates that a steady state "shakedown" condition should be reached after several days of operation, in which there should be no further changes to the slip joint loading or, correspondingly, in the vibrational characteristics of the jet pumps). The conclusions reached concerning the impact the loss of gap has on the FIV response is that it will tend to significantly shift the transition to high-level vibration to higher slip joint pressures, but the ultimate transition will be much more dramatic (i.e., the slope of the Response-SJDP curve in the second and third regimes will be much higher). The net effect in most cases should be an increase in safety margin.

Other observations from data analyzed to date follow:

- (1) The vibration response in the radial direction tends to be more intense than in the circumferential direction. (Note: mixer-diffuser slip joint contact is in the radial direction; this may or may not be a factor.) This is particularly true for the high-level sinusoidal vibration regime.
- (2) When the vibrations are low-level random, the mixer and diffuser RMS accelerations at the slip joint tend to be approximately equal. However, as the vibration intensities increase, the mixer, which is 4.5 times stiffer than the diffuser, will tend to vibrate at higher magnitudes than the diffuser. This is illustrated in Figure 2.3-27. In the high-level vibration regime, the ratio between mixer and diffuser response was observed to be as much as 3:1.

Task 2.3b — Jet Pump Hot FIV Test (L. LaCroix)

Task Description

Flow tests are performed on single prototypical jet pumps under reactor temperature and pressure (550°F, 1050 psia) at the Pacific Gas & Electric Moss Landing Power Station, where the GE Steam Water Test Facility is located. The test program is designed to establish a better understanding of the relationship between the vibration response of the component and the key fluid and geometric parameters, particularly in the slip joint region. The program is separated into two activities, each described below.

- (1) The first activity consists of the thorough FIV testing of a BWR/5 201-in. jet pump. This jet pump is of interest to the FIV for LWR program because its geometric and fluid characteristics (lower fundamental frequency and higher slip joint pressure drop than other designs) make it more susceptible to slip joint-type vibration stabilities. Sufficient vibration instrumentation will be utilized to characterize the natural frequencies, mode shapes, primary forcing functions, and system response for various flow conditions.
- (2) The second activity consists of the very limited FIV testing of BWR 218-in. and 251-in. jet pumps. The data, consisting primarily of vibration response as a function of flow condition, are acquired through the addition of several vibration sensors to jet pumps scheduled for hydraulic performance testing at the Moss Landing facility. The additional data will provide significant information concerning the effect of geometry (diffuser length, slip joint geometry, etc.) on the dynamic response of this LWR component.

Work Performed This Report Period

All flow-induced vibration testing of single prototypical jet pumps under reactor temperature and pressure in the Pacific Gas & Electric Moss Landing Steam Water Test Facility has been completed. Much of the vibration data has been reduced and analyzed. Topical reports documenting this work will be issued shortly.

Figure 2.3-1. Jet Pump Recirculation System

Figure 2.3-2. (HF)² Jet Pump Pairs A and B

Figure 2.3-3. Tangential Shake Test Configuration

Figure 2.3-4. Radial Shake Test Configuration

Figure 2.3-5. Jet Pump Pair A — Accelerometers for Mode Shapes

Figure 2.3-6. Response Spectrum in Radial Direction: 20-100 Hz Sine Sweep; 10-lb Force at Mixer Elbows; Sensor 1Y (EMR Channel 2) (Water)

Figure 2.3-7. Response Spectrum in Tangential Direction: 20-100 Hz Sine Sweep; 15, 20 and 30-lb Forces at Top of Riser; Sensor 12 (EMR Channel 3) (Water)

Figure 2.3-8. Response Spectrum in Radial Direction: 20-100 Hz Sine Sweep; 10-lb Force at Mixer Elbow; Sensor 1Y (EMR Channel 2) (Air)

Figure 2.3-9. Response Spectrum in Tangential Direction: 20-100 Hz Sine Sweep; 15 and 30-lb Forces at Top of Riser; Sensor 12 (EMR Channel 3) (Air)

Figure 2.3-10. $(HF)^2$ Jet Pump Mode 1 at 29 Hz (Radial Direction); Test Number = DS5; Force Direction = Radial; Environment = Water at 70°F

Figure 2.3-11. $(HF)^2$ Jet Pump Mode 2 at 35-37 Hz (Tangential Direction); Test Numbers = LS5 and LS8; Force Direction = Tangential; Environment = Water at 70°F

Figure 2.3-12. (HF)² Jet Pump Mode 3 at 41 Hz (Tangential Direction); Test Number = M50; Force Direction = Tangential; Environment = Water at 70°F

Figure 2.3-13. $(HF)^2$ Jet Pump Mode Shape Prediction
(First Mode, 30.8 Hz, Tangential)

Figure 2.3-14. $(HF)^2$ Jet Pump Mode Shape Prediction
(Second Mode, 31.3 Hz, Radial and Vertical)

Figure 2.3-15. $(HF)^2$ Jet Pump Mode Shape Prediction
(Third Mode, 38.7 Hz, Radial and Vertical)

Figure 2.3-16. Accelerometers to Measure FIV Response (on All Jet Pumps)

* STRAIN GAGE NUMBERING SYSTEM IS THAT REPORTED IN NEDE-23840, "VIBRATION ANALYSIS FOR THE REACTOR INTERNALS OF BWR/6 238 PLANTS" (NOTE : DIFFUSER STRAIN GAGES 4 and 8 (ON VESSEL SIDE OF TAILPIPES) ARE NOT SHOWN).

Figure 2.3-17. Strain Gage Locations to Satisfy Plant Preoperational and Startup Test Requirements

Figure 2.3-18. Jet Pump 2 Pressure Sensors

Figure 2.3-19. Jet Pump 3 Pressure Sensor Array (for Spatial Correlation)

Figure 2.3-20. Pressure Transducers at Slip Joint to Measure Dynamic Pressures
(on All Jet Pumps)

Figure 2.3-21 Exponential Curve Fit to $(HF)^2$ Jet Pump 1 Hydraulic Parameters (SJDP, Drive Flow, M-Ratio)

Figure 2.3-22. PSDs for Jet Pump 2 Mixer Radial Acceleration at Slip Joint for Several M-Ratios

Figure 2.3-23. Jet Pump 1 Mixer Radial Acceleration as a Function of Slip Joint Pressure Differential (Water Temperature = 180°F)

Figure 2.3-24. Jet Pump Mixer Slip Joint Response at Operating Drive Flow as a Function of Temperature

Figure 2.3-25. Kinematic Viscosity versus Slip Joint Differential Pressure at Onset of High Level Vibration (1g_{rms})

Figure 2.3-26. Jet Pump 2 Diffuser (Slip Joint) Radial Response at 100% Drive Flow versus M-Ratio; Beginning and Middle of Test Program

Figure 2.3-27. Averaged Mixer and Diffuser Response at Slip Joint Versus Drive Flow at Design Operating M-Ratio

TASK 2.4 - LOW PRESSURE COOLANT INJECTION COUPLING

The low pressure coolant injection (LPCI) system is part of the boiling water reactor emergency core cooling system. Not used during normal operation, its sole purpose is to restore and maintain the desired water level in the reactor vessel after a loss-of-coolant accident (LOCA). The overall system is shown in Figure 2.4-1; the internal coupling hardware is shown in Figure 2.4-2. The coupling hardware consists primarily of two 10-in.-ID elbows, joined together by a 26-in.-long sleeve. Collars which thread on the two elbows capture the sleeve during installation and hold it in place (to accommodate differential thermal expansion, the sleeve is not welded to the elbows). A nominal gap of 8 mils exists between the sleeve and each collar.

The LPCI is subject to FIV under both normal plant operation and emergency (LOCA) modes. Under normal operating conditions, when there is no flow through the LPCI, downcomer annulus flow buffets the LPCI coupling. In addition, a steady-state pressure drop across the collar-sleeve slip joints induces leakage flow which can result in fluid-elastic coupling. In the LOCA mode, the high flow velocities through the coupling produce turbulent buffeting of the inner walls. Also, as in the normal operating mode, slip joint differential pressures exist which can induce some leakage flow.

Task Description

Task 2.4 consists of flow-testing a prototypical BWR/6 238-in. LPCI coupling in the High Flow Hydraulic Facility's General Purpose Test Tank. The coupling is heavily instrumented with a combination of accelerometers, pressure transducers, and strain gages, as well as many thermal-hydraulic sensors to ascertain the vibration characteristics and leakage flow through the collar-sleeve slip joints as a function of the following parameters:

- (1) Slip joint differential pressure;
- (2) Internal (LOCA) flow;

- (3) External (normal plant operation flow);
- (4) LPCI coupling alignment; and
- (5) Slip joints sealing (none, labyrinth seal, and piston ring).

This test program should provide some fundamental understanding of the flow-induced vibration characteristics of large piping configurations with slip joints.

Work Performed This Report Period

All test activities have been completed, and efforts are now directed toward data reduction and analysis, as well as final test documentation. Reported below are brief descriptions of the test hardware and facilities, vibration instrumentation, and procedures utilized, as well as preliminary results from data analysis performed to date.

Test Hardware and Facilities

The tests were performed on prototypical BWR/6-238 LPCI coupling hardware in the (HF)² General Purpose Test Tank (Figure 2.4-3). Support hardware was added to allow for reasonable simulation of the surrounding reactor structural environment, as well as the flow conditions. The fixturing was made adjustable so that the alignment between the two coupling elbows could be varied radially and vertically to simulate differential thermal expansion effects. The two facility arrangements used in the test program — one for normal plant operation (external downcomer flow) and the other for LOCA operation (internal flow) — are shown in Figures 2.4-4 and 2.4-5, respectively, and described below:

Normal Plant Operation

Under normal plant operation simulation, the external downcomer flow (Maximum Velocity $\approx$ 15 ft/sec) is provided by the three (HF)² B-Loop

pumps (1250 Hp motors, total flow capacity of approximately 16,000 gpm). A secondary loop provides for the desired pressure drop across the slip joints and monitors the leakage flow. There is no flow in the coupling, other than that due to leakage through the slip joints.

LOCA Operation

Under LOCA operation simulation (internal coupling flow), the (HF)² B-loop pumps are connected to the LPCI upper elbow via a thermal sleeve and provide the internal coupling flow. The slip joint differential pressure is controlled by establishing a pressure-regulated nitrogen environment external to the LPCI coupling. The secondary loop is used to maintain the water level of the pool in the vessel-shroud annulus below the coupling and nitrogen environment, thus providing a means of monitoring the slip joint leakage flow (changes in the elevation of the pool will have to be caused by the leakage flow).

The vibration sensors utilized in this test program are shown in Figures 2.4-6 and 2.4-7. Included are the following:

- a. Vibra Metrics Model 384812 integrated circuit piezoelectric (ICP) accelerometers, located primarily on the elbows and collars, as well as on the support structures.
- b. PCB Piezotronics Model 112M108 ICP dynamic pressure transducers, located primarily on the collars to help identify the character of the forcing mechanism.
- c. Ailtech Model MG128/14-09-30-304SS and MG128/12-11-30-304SS weldable strain gages, located at the thermal sleeve and lower elbow ring to monitor the maximum system strains and aid in the understanding of the coupling's displacement response.

Test Conditions

The test program covered the following structural and thermal-hydraulic conditions:

Temperature	60°F to 180°F
Flow	
Normal (external)	0 to $\approx$ 16 fps (14,300 gpm)
LOCA (internal)	0 to $\approx$ 67 fps (15,000 gpm)
Offset (normal/LOCA)	
Vertical	$\approx$ 0.5 in./1.5 in. (0 in. is solid elbow sleeve elbow contact)
Radial	0.26 in./1.1 in. (0 in. is perfect upper-lower elbow alignment)
Circumferential	0 in./0 in.
Slip joint differential pressure	0 to 70 psi
Leakage flow restrictors	
None	
Labyrinth seal	
Piston ring	
Stiffness Modifications	
None	
Lower elbow brace	

Results

Initial results for an LPCI coupling with no flow restrictions at the slip joint (considered to be the base case) indicated the vibration response to be governed primarily by the slip joint differential pressure (SJDP) and possibly the offset conditions. In the normal operating configuration, where there was

very little radial misalignment, an upward sweep in SJDP eventually produced an abrupt five-fold increase in vibration intensity (to approximately 1 g_{rms} for the lower elbow), the character of the vibration changing from low-level random to high-level sinusoidal (36 Hz). Strain time histories representative of behavior both before and after transition are shown in Figures 2.4-8 and 2.4-9. Once high-level vibration had been achieved, the SJDP would have to be reduced to some level below the transition inception point to cause "drop-out" to low-level vibration. The inception and drop-out SJDPs appeared to be inversely related to the water temperature, as Figure 2.4-10 indicates (this is consistent with results from the jet pump tests reported in Task 2.3). The downcomer flow tended to suppress the high-level vibration, the transition SJDP being lowest when there was no external flow. As for the LOCA configuration with internal flow, there was no evidence of any high-level vibration regime, although this may be due to the radial offset (≈ 1.1 in.) which was initially established for this test condition.

The relationship between SJDP and slip joint leakage flow at 67°F and 181°F for the base case is shown in Figure 2.4-11. As expected, the leakage flow increased with temperature.

Having confirmed the transition to high-level vibration to be possible under certain thermal-hydraulic conditions, several vibration suppression techniques were investigated. The first was to machine a five-groove labyrinth at each end of the LPCI sleeve, where it inserts into the collar (Figure 2.4-12). Previous experience on other reactor components had revealed this to be effective in reducing the leakage flow through slip joints, and, as a consequence, improving the FIV characteristics. Leakage-only flow tests were performed on the modified sleeve design in the normal operating (no radial offset) setup mode, since it was under these conditions that the non-labyrinth seal base case had experienced the highest vibration levels. The test results were somewhat mixed. There was no significant reduction in slip joint leakage flow from that observed for the non-labyrinth seal case (Figure 2.4-13). However, the maximum vibration intensities were noticeably less, there no longer being an observable transition to high-level vibrations over the range of SJDPs studied. Also missing was the 35-Hz response component so predominant

in the base case. It should be noted that hardware inspections performed after completion of the labyrinth seal test revealed the sleeve to be bound in the collars, a condition which did not exist at the beginning of the test. The fact that the sleeve was no longer reasonably concentric with respect to each collar, but rather was actually loaded against the collar surface, may explain both the higher-than-anticipated leakage flow as well as the absence of a high-level vibration regime. Further data analysis is required before a conclusive judgment can be made.

A second method investigated for the suppression of high-level vibrations was to install piston rings on the sleeves where they insert into the collar (Figure 2.4-14). As with the labyrinth seal, the purpose was to restrict the leakage flow through the slip joints. Leakage flow-only tests at normal operating offsets for the first piston ring design considered produced results which were very similar to the labyrinth seal test. As Figure 2.4-15 indicates, the piston rings were not effective in reducing leakage flow. However, they did seem to suppress the transition to high-level vibrations over the SJDP range tested. The maximum acceleration observed was a $0.3 g_{rms}$ harmonic motion on the upper elbow at a SJDP of 12 psi, caused by 50-Hz and 54-Hz beating. This was not observed at other SJDP, with the possible exception of 18 to 20 psi. In general, the vibration response was very insensitive to the SJDP.

After studying the results of the piston ring test, in which it was concluded that the piston ring was, among other things, too flexible to properly seal the slip joints, stiffer, more carefully manufactured piston rings were installed on the LPCI sleeve. Additionally, a strut was attached between the lower elbow and shroud wall, raising the fundamental frequency by about 70% (to ≈ 85 Hz). This arrangement was then subjected to the full complement of flow tests, including downcomer (external) flow sweeps, LOCA (internal) flow sweeps, and SJDP sweeps, as well as variations in water temperature and upper elbow-to-lower elbow alignment. Preliminary results are summarized below:

- (1) The piston rings were very effective in reducing the leakage flow (Figure 2.4-16). The maximum leakage flow experienced was 12 gpm at 50 psi.

- (2) Vibration levels were, in general, low-level random, typically an order of magnitude reduction from the base case high-level results. A typical acceleration time history and PSD are given in Figure 2.4-17. The response level tended to increase with increased internal flow (Figure 2.4-18) and be insensitive to changes in external flow. It also appeared to be unaffected by SJDP (Figure 2.4-19), although there was some indications that decreasing SJDP increased the response energy in the 60- to 120-Hz range. There was absolutely no evidence of any low-to-high level transition regime over the SJDP range tested. Changes in temperature or offset conditions did not significantly affect the results.

- ① CONTAINMENT
- ② DRYWELL
- ③ RPV
- ④ SYSTEM PUMP
- ⑤ SUPPRESSION POOL
- ⑥ HEAT EXCHANGERS

Figure 2.4-1. Residual Heat Removal System, Low Pressure Coolant Injection Function

Figure 2.4-2. Low Pressure Coolant Injection System

Figure 2.4-3. (HF)² General Purpose Test Tank

Figure 2.4-4. LPCI Coupling Normal Condition Simulation

Figure 2.4-5. LPCI Coupling LOCA Condition Simulation

Figure 2.4-6. LPCI Vibration Instrumentation, Lower Elbow Region

Figure 2.4-7. LPCI Vibration Instrumentation, Upper Elbow Region

Figure 2.4-8. Lower Elbow Ring Strain Time History - Low Level Vibration Regime (Base Case)

Figure 2.4-9. Lower Elbow Ring Strain Time History - High Level Vibration Regime (Base Case)

Figure 2.4-10. SJD P Onset and Dropout Levels versus Temperature and Kinematic Viscosity

Figure 2.4-11. Base Case (No Slip Joint Seal) Leakage Flow for Two Temperatures (Normal Plant Configuration)

Figure 2.4-12. LPCI Coupling Sleeve Five-Groove Labyrinth Seal

Figure 2.4-13. Leakage Flow as a Function of Differential Pressure: Labyrinth Seal versus No Slip Joint Seal (Leakage Flow Only at 70°F)

Figure 2.4-14. Cross Section of LPCI Ball Joint (Including Piston Ring)

Figure 2.4-15. Leakage Flow as a Function of Slip Joint Differential Pressure: Piston Ring 1 Design vs No Slip Joint Seal (Leakage Flow Only Test)

Figure 2.4-16: Leakage Flow as a Function of Slip Joint Differential Pressure: Improved Piston Ring (2) vs No Slip Joint Seal

Figure 2.4-17. Lower Collar Radial Response to 10,500 gpm Internal Flow at Local Offset Conditions, 120°F (Improved Piston Rings on Sleeves)

2.4-25

GEAP-24248

Figure 2.4-18. Lower Elbow Acceleration Response versus Internal Flow for Improved Piston Ring ($\Delta P = 20$ psi)

Figure 2.4-19. Lower Elbow Acceleration Response versus Slip Joint Differential Pressure for Improved Piston Ring (Downcomer Flow = 11,000 gpm, Temperature = 180°F)

TASK 3 - LWR FIV DESIGN METHODS, GUIDES AND CRITERIA

The primary purpose of Task 3 is the development of FIV guides and criteria which can be utilized by engineers for the design of LWR components. This is accomplished through the generation of analytical models and empirical relations, using the results of the fundamental studies, full-scale component tests, and actual plant vibration data generated in Tasks 1 and 2. Additionally, Task 3 is designed to establish scaling laws for FIV testing of reactor components, as well as to provide analytical support for the model testing of Task 2. The technical objectives are to:

- (1) Develop analytical methods and computer codes for predicting the FIV response of reactor components;
- (2) Develop empirical relations, scaling laws, and practical design guides for reactor design utilizing the information generated in Tasks 1 and 2; and
- (3) Verify analytical methods by comparison with full-scale development tests and actual plant vibration data as available.

Task 3 is divided into three separate activities:

Task 3.1 - Predictive Model Development

Task 3.2 - Scaling

Task 3.3 - Correlation with Full-Scale Tests and Plant Data

Task 3.1 will develop the analytical methods and computer codes for predicting the FIV response of reactor components. Task 3.2 will indicate which parameters are critical (i.e., must be duplicated or accounted for) when performing

scale-model testing of reactor components. Task 3.3 will verify the results of the predictive models and scaling laws, using vibration data from prototype LWR tests.

Progress achieved on these activities is described on the following pages.

TASK 3.1 - PREDICTIVE MODEL DEVELOPMENT (NTD)

Task 3.1 is devoted to the generation and verification of analytical methods for predicting the FIV response of reactor component structures. The desired output from the task is one or more FIV design guides.

The basic approach used in Task 3.1 is to characterize the FIV problem into two categories, one to determine the amplitude of response, and the other to establish critical stability parameters and their range of influence. For response problems, the approach taken is to stochastically model the excitation and random response. Wherever possible, existing analytical tools, as well as special analytical techniques developed through the fundamental studies, are used for the generation of predictive FIV methodology. For the stability problem, reliance will be placed primarily on empirical results. Computer results are written to calculate the FIV response of structures, knowing structural and damping characteristics. Empirical formalisms and predictive models are verified against experimental results from Task 2. Stability curves are generated for structures to aid in safe-point operations. Specific problems to be addressed and the relevant brief details are given in Table 3.1-1.

Task 3.1a - Random Decrement Method (M.A. DeCoster)

Task Description

Damping is a very important parameter in evaluating the response of any vibrating structure. Unfortunately, it can be difficult to quantify, particularly for some of the complex reactor components under operational conditions. Therefore, methods are being sought to accurately measure the damping of structures in the actual reactor environment (i.e., after plant installation). The method presently under consideration is called random decrement. It is designed to identify modal parameters such as frequencies and damping for any component from a simple vibration response time history obtained during normal

operations. This is accomplished by time-averaging specifically chosen blocks of data to form a randomdec signature, and then fitting one or more damped sinusoids to the signature via the modified difference Levenberg-Marquardt algorithm.

Work Performed This Report Period

A computer program was developed to identify natural frequencies and damping values of structures using the random decrement method. Several test cases were analyzed on the Honeywell 6000 computer, using both synthetic time histories as well as vibration from operating plants. Results, some of which are reported in "FIV for LWR Progress Report, January-March 1978," were inconclusive. Natural frequency calculations were good. However, damping value calculations were in some instances suspect, although there was insufficient information to base a definite judgment. The primary conclusion was that further evaluation is necessary. As a consequence, the following actions are being taken:

- (1) Plans are being formulated to proof-test the technique via a small-scale bench test. The program will consist of recording FIV data for random decrement analysis, using hardware in which the damping is very carefully characterized and controlled. The random decrement results will then be compared to the actual values.
- (2) The random decrement program is being loaded onto the (HF)² Data Analysis Computer System for use in analyzing the component FIV data generated during the first half of the FIV for LWR program. This should provide a good test of the usefulness of this method in the handling of plant vibration data.

Task 3.1b - Slip Joint Mechanistic Model (M.R. Torres)

Task Description

The internal flow velocities required to incur detrimental levels of flow-induced vibration in pipes are usually quite high for most practical applications. However, if submerged piping structures are coupled together with annular slip joints to accommodate static load deflections or thermal growth, a practice utilized quite freely in industry, a geometry is created which is very susceptible to high-level FIV. This was demonstrated during the early stages of the FIV for LWR Program through small scale experiments on a pair of submerged cantilever pipes joined together at their free end with internal flow (documented in "FIV for LWR Progress Report, June-September 1977," Document No. C00/4175-2).

An indepth investigation of existing technical literature has been performed which reveals extensive information to exist on the dynamic stability of pipes with internal or external flow, as well as on the dynamic characteristics of submerged structures (e.g., virtual mass, stiffness, frequency mode shape). However, sparse technical reporting on the subject of slip joint geometries is available, revealing that more fundamental work is warranted, particularly as related to the development of analytical models to predict the response of reactor components with slip joints. Therefore, it is the objective of this task to generate a mathematical model which can be used to predict the level and onset of slip joint vibration as a function of piping structural parameters, details of the slip joint geometry and hydraulic parameters.

Work Performed This Report Period

A mathematical model is currently under derivation. The approach taken is to solve the Navier-Stokes equation within the annular slip joint and determine the interaction force between the two cantilever pipes as a function of the motion of the pipes at the slip joint. The equations of motion of each

cantilever pipe are coupled with this interaction (slip joint) force. Solution of these fluid-structure equations is then required.

After several simplifying assumptions with regard to the Navier-Stokes equations within the slip joint and modeling each cantilever pipe as a single-degree-of-freedom system, governing cantilever pipe-slip joint equations were derived.

An example of one of these governing equations of motion in non-dimensional form is provided below.

Governing cantilever pipe(s)-slip joint equation of motion for Pipe 1

$$\begin{aligned}
 \eta_1 + 2\beta\dot{\eta}_1 + \eta_1 = & -A \left(\frac{\delta v}{\delta} \right)^2 \frac{L^2}{\delta^2} \frac{(\eta_1 - \dot{\eta}_2)}{\gamma_1} \left[\frac{1 + 3(\eta_1 - \eta_2)^2}{(1 - (\eta_1 - \eta_2)^2)^3} \right] \\
 & - B \frac{L^2}{\delta^2} \frac{(\eta_1 - \eta_2)^2}{\gamma_1} \left[\frac{\eta_1 - \eta_2}{(1 - (\eta_1 - \eta_2)^2)^2} \right] \\
 & - C \frac{L^2}{\delta^2} \frac{(\dot{\eta}_1 - \dot{\eta}_2)}{\gamma_1} \left[\frac{g_1}{1 + \eta_1 - \eta_2} + \frac{g_2}{1 - \eta_1 + \eta_2} \right] \\
 & - E \frac{2}{3} \frac{L^2}{\delta^2} \frac{(\eta_1 - \eta_2)}{\gamma_1} \left[\frac{1}{1 - (\eta_1 - \eta_2)^2} \right] \\
 & - H \frac{L^2}{\delta^2} [f(g_1, g_2)]
 \end{aligned}$$

where

$$\eta_1 = \frac{x_1}{\delta}$$

$$\eta_2 = \frac{x_2}{\delta}$$

x_1 = Transverse displacement of pipe 1 at slip joint (see Figure 3.1-1)

x_2 = Transverse displacement of pipe 2 at slip joint (see Figure 3.1-1)

where

$$\eta_i = \frac{x_i}{\delta}, \quad \dot{\eta}_i = \frac{\partial \eta_i}{\partial \tau}, \quad \ddot{\eta}_i = \frac{\partial^2 \eta_i}{\partial \tau^2}$$

x_i = Transverse displacement of pipe i

δ = Average radial gap at slip joint

$\tau = w_i \cdot t$

t = Time

w_i = Fundamental frequency of pipe i in the absence of a slip joint

$$\gamma_i = \frac{3E_i I_i}{2LD\delta\rho w_i^2 \ell_i^3}$$

E_i = Modulus of elasticity for pipe i

I_i = Moment of inertia for pipe i

D = Slip joint diameter

ρ = Fluid density

L = Slip joint length/2

ℓ_i = Length of pipe i

$$\delta_v = \sqrt{\frac{2v}{w_i}}$$

v = Viscosity

g_i = Functional parameter which must be solved. This parameter has units of velocity

f = algebraic function

A, B, C, E, H = constants

The terms on the right hand side are the coupling force terms due to the presence of a slip joint. Notice the first term is a viscous damping term, the second a hydrodynamic stiffness (kinetic energy) term and, the third term possibly the source of the high level vibration (negative damping). The fourth expression, with acceleration values, is obviously the virtual mass term. It is important to note that these coupling slip joint forces are sensitive to the ratio of slip joint length to slip joint gap by a power of two. Since some of these terms may contribute to stabilization and others to destabilization of the pipes vibrational response, the exact role of (L/δ) cannot be directly extracted from these equations.

Future Work

Current plans are to solve the governing equations of motion numerically for a few cases. A computer program is in process but requires further rewriting. The goal is to correlate these analytical model (computer) results with experimental results obtained from the slip joint mechanistic model previously reported.

Table 3.1-1
SUMMARY OF TASKS

<u>Problem</u>	<u>Analytical Approach</u>	<u>From Task</u>	<u>Details</u>	<u>Desired Output</u>
1. Closely packed plenum structures (guide tube clusters)	Random vibration theory; empirical formalism	1(1.1,1.2,1.4)	Excitation force characteristics, damping, hydrodynamic mass coefficients	Predictive analytical model, scaling laws and design guides
2. Multirod arrays in single-phase turbulent parallel flow	Random vibration theory; empirical formalism	1(1.1,1.2,1.3,1.4) 2(2.2)	Excitation force characteristics, damping, hydrodynamic mass coefficients	Predictive analytical model, scaling laws and design guides
3. Jet pump vibration	Random vibration and stability theory	1(1.3,1.4) 2(2.2)	Leakage and mixing phenomena information, critical stability parameters, force and damping characteristics	Stability curves relating critical parameters, predictive analytical
4. Improved structural dynamic element models of LWR internals	Finite-element computer simulation	2 3	Test results (frequencies, mode shapes, etc.) to verify and modify structural models	Specialized computer programs

Figure 3.1-1. Cantilever Pipes With Slip Joint

TASK 4 - PROGRAM ADMINISTRATION

Task Description

The purpose of Task 4 is to provide the overall management of the program, including coordination, reports and reviews. In addition, it is to ensure that the program's fundamental studies, model and full-size component tests, and analytical method development are responsive to the PWR as well as the BWR needs in the area of flow-induced vibrations.

Work Performed This Report Period

Initial solicitation of PWR manufacturers for comments on the FIV for LWR Program has been accomplished. Conclusions, derived from the contacts, include the following:

- (1) The PWR manufacturers consider FIV to be a significant problem.
- (2) While there was general agreement with the present content of the program, it is felt that the program should be expanded, primarily to include additional fundamental studies, but also to include testing of PWR components, possibly in the GE High Flow Hydraulic Facility.
- (3) The need for improved communications was emphasized. Means suggested to accomplish this was through regularly scheduled review meetings and workshops involving all LWR manufacturers.

A letter report summarizing the results has been prepared.

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