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Vibration and Wear in Steam Generator Tubes Following Chemical Cleaning

Final Report

Prepared by W. I. Enderlin, D. E. Fitzsimmons

Pacific Northwest Laboratory
Operated by
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EXECUTIVE SUMMARY

The buildup of magnetite in the steam generators of some pressurized water reactors (PWRs) has led operators to propose chemical cleaning to remove this product. In some cases, the volume of magnetite formed by the corrosion of the carbon steel has been sufficient to cause "denting" or reduction of the outer diameter of the tubes where they pass through the support plates. Moreover, when the magnetite is removed by a chemical cleaning process, the diameter of the hole in the tube support plate is increased. The U.S. Nuclear Regulatory Commission (NRC) has expressed concern that the resulting increased clearances may allow an increased level of flow-induced vibrations in chemically cleaned steam generators that may, in turn, lead to high tube wear rates and unacceptable levels of tube failure.

The Pacific Northwest Laboratory (PNL) is supporting the NRC staff in addressing the effects of increased tube/tube-support clearances. The objective of PNL's work is to provide NRC with criteria with which to evaluate licensees' specific proposals for chemical cleaning of steam generators.

Research has been performed in both the U.S. and Canada on vibration and wear of PWR steam generator tubes. However, most of the work performed in the U.S. is currently classified as proprietary by its sponsors and is unavailable for general use.

In Canada, P. L. Ko and co-workers have been investigating wear in CANDU steam generators for over 10 years. Their work has included experimentation to determine the fretting wear mechanisms involved, development of analytical techniques to predict impact forces at the tube supports, and the correlation of tube wear and tube motion parameters. However, this work was not predicated on tube motions and tube support impact forces that were experienced within an actual reactor operating environment. Rather, a computer code (VIBIC) was developed to simulate the motion of a cantilevered beam impacting against supports with clearance, and to predict the midspan displacements and the support impact forces. The difference between the experimental and analytical results using this code was found to be greater for those tubes tested in the presence of water than for those tested in air. The VIBIC code considers only tube motion in air. The fluid damping effect in the tube/support clearance space and the film effect of the water at the contact surface had not been included in the VIBIC simulation used in Ko's work.

In the U.S., the Electric Power Research Institute (EPRI) is currently sponsoring a research project to quantify tube/tube-support plate interaction characteristics of multispan steam generator tubes and to gain a better understanding of their dynamic characteristics. Measurements were recorded in air and in cold stagnant water environments for different tube alignment conditions and for different support clearances. The data were recorded over a range of sinusoidal and random excitations and then digitally analyzed to compute statistical force and sliding distance parameters. These data provide the basis for wear tests being conducted in an autoclave by Kraftwerk Union AG in Germany.

The investigation undertaken by PNL extends the previous research. Flow tests at prototypic Reynolds number, with tube clearances at the flow entrance region representative of as-built and of post-cleaned conditions, were performed at elevated temperature and pressure, using a full-length scale model of a steam generator tube bundle. The tests were performed to 1) establish the forcing boundary conditions and 2) determine if an environment conducive to fretting wear actually exists under both tube clearance conditions at the instrumented support location.

From the flow test results, PNL researchers concluded that a tube clearance of 10 mils or greater in excess of design clearance would not tend to increase tube wear rate at the instrumented tube support location. The tube support plate was shown to be considerably less active at this clearance than at the design clearance when fluid flows at 400°F ranged from 50% to 150% of the flow required for prototypic Reynolds number. Tube motion was elliptic under these conditions, and vibration amplitudes were greater than for the design clearance case, suggesting that the tube in this case was restrained by its stiffness rather than by the support plate. It was evident that frequent tube contact did occur at design clearance conditions. Based on these results, a forcing function for accelerated wear tests cannot be defined. It was further concluded that the data did not justify further testing at tube clearance conditions in excess of 10 mils over design clearance, and that accelerated wear testing was not justified.

The tube excitation force expected under field conditions would be even less than what was imposed by the test conditions. Moreover, based on the results of chemical cleaning tests performed by other researchers, the post-cleaning tube/tube-support plate clearance is expected to remain at no less than 10 mils over design clearance, and will likely be even larger. Hence, PNL concluded that under normal operating conditions there is little potential for increased tube wear rate as a result of chemical cleaning at tube support locations within a steam generator where conditions are similar to those at the instrumented tube support location.

ACKNOWLEDGMENTS

This project was funded by the U.S. Nuclear Regulatory Commission under NRC FIN B2858. Dr. Joseph Muscara was the NRC Project Manager. Mr. Lowell Strope of PNL was responsible for instrumentation and data acquisition. Mr. John Baugh of PNL was responsible for data reduction and evaluation. Ms. J. A. Bamberger of PNL performed design calculations and flow loop modification and participated in flow loop operations.

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VIBRATION AND WEAR IN STEAM GENERATOR TUBES FOLLOWING CHEMICAL CLEANING

FINAL REPORT

1.0 INTRODUCTION

The buildup of magnetite in the steam generators of some pressurized water reactors (PWRs) has led operators to propose chemical cleaning to remove this product. In some cases, the volume of magnetite formed by the corrosion of the carbon steel has been sufficient to cause "denting" or reduction of the outer diameter of the tubes where they pass through the support plates. A profilometry examination of the retired Surry 2A steam generator, conducted at Hanford by Babcock and Wilcox, revealed an average dent in the 40- to 50-mil range, with some tube diameter reduction down to 0.470 inch. In addition, when the magnetite is removed by a chemical cleaning process, the diameter of the hole in the tube support plate is increased even further.

The U.S. Nuclear Regulatory Commission (NRC) has expressed concern that an increased clearance between the tube and the tube support plate may result in an increased level of flow-induced vibrations in chemically cleaned steam generators, in cases where the tube support plates are exposed to the cleaning process. This may, in turn, lead to high tube wear rates and unacceptable levels of tube failure.

The Pacific Northwest Laboratory (PNL) is supporting the NRC staff in addressing the effects of increased tube/tube-support clearances. The objective of PNL's work is to provide NRC with criteria with which to evaluate licensees' specific proposals for chemical cleaning of steam generators.

1.1 BACKGROUND

Chemical cleaning experiments conducted in the laboratory (Schneidmiller and Stitler 1983) demonstrated that the diameter of the hole in the support plate would be increased by at least 25 mils in the case of 100% effective cleaning at incipient denting, and could be in excess of this. To date, the only steam generator to undergo chemical cleaning is at the Millstone II power station. Cleaning in this case was confined to the sludge pile. Care was taken to isolate the tube support plates from the cleaning process. There has been no experience with chemical cleaning of tube crevices in an actual steam generator.

Research has been performed in both the U.S. and Canada on vibration and wear of PWR steam generator tubes. However, most of the work performed in the U.S. is currently classified as proprietary by its sponsors or is incomplete, and the data and final results are unavailable for general use.

In Canada, P. L. Ko and co-workers have been investigating wear in CANDU steam generators for over 10 years, and have documented their work in several significant articles (Ko, Tromp, and Weckwerth 1982; Ko 1973; Ko 1979). Their work has included experimentation to determine the fretting wear mechanisms involved, development of analytical techniques to predict impact forces at the tube supports, and the correlation of tube wear and tube motion parameters. Ko et al. developed a bench-scale tube fretting apparatus in which they extensively studied the effects of various tube/support plate parameters on tube wear. These researchers found that high wear was caused not by the high-force components such as impact motion that have low probability of occurrence, but instead by the high probability intermediate-range force components (usually combined impact and rubbing motions). The amount of wear was observed to increase exponentially with excitation frequency and to increase approximately linearly with diametrical clearance and excitation amplitude, up to a point. However, the data to substantiate the latter were extremely limited. Moreover, this work was not predicated on tube motions and tube support impact forces that were experienced within an actual reactor operating environment. Rather, a computer code (VIBIC) was developed to simulate the motion of a cantilevered beam impacting against supports with clearance, and to predict the midspan displacements and the support impact forces. The difference between the experimental and analytical results using this code was found to be greater for those tubes tested in the presence of water than for those tested in air. The VIBIC code considers only tube motion in air. The fluid damping effect in the tube/support clearance space and the film effect of the water at the contact surface had not been included in the VIBIC simulation at the time of this work (Ko and Rogers 1979).

In the U.S., the Electric Power Research Institute (EPRI) is currently sponsoring a research project to quantify tube/tube-support plate interaction characteristics of multispan steam generator tubes and to gain a better understanding of their dynamic characteristics (Haslinger and Smith 1983). The project employs a single-tube test fixture with three instrumented supports (one egg crate and two cylindrical-hole type). Measurements were recorded in air and in cold stagnant water environments for different tube alignment conditions and for different support clearances. The data were recorded over a range of sinusoidal and random excitations and then digitally analyzed to compute statistical force and sliding distance parameters. These data provide the basis for wear tests being conducted in an autoclave by Kraftwerk Union AG in Germany.

1.2 SCOPE

The PNL investigation described in this report extends the previous research in that flow tests at prototypic Reynolds number, with tube clearances at the flow entrance region representative of as-built and of post-cleaned conditions, were performed at elevated temperature and pressure, using a full-length scale model of a steam generator tube bundle. The tests were performed to 1) establish the forcing boundary conditions and 2) determine if an environment conducive to fretting wear actually exists under both tube clearance conditions at the instrumented support location.

In Section 2.0 the methodology used in the PNL study is discussed. The analysis-before-test is described in Section 3.0. Section 4.0 documents the actual test performance. The conclusions PNL reached from this study are presented in Section 5.0.

2.0 METHODOLOGY

PNL's approach to this investigation was to:

- perform an analysis-before-test (ABT)
- design the hardware and facility modifications required to support flow testing and accelerated wear tests
- procure, fabricate and install required hardware and instrumentation and perform functional checkout
- perform flow tests
- perform accelerated wear tests if appropriate
- evaluate test results to determine the potential for premature tube failure as a result of chemical cleaning of tube crevices.

Each of these activities is documented in the following subsections.

2.1 ANALYSIS-BEFORE-TEST

The objectives of the ABT were to 1) identify important parameters in the tube vibration and wear process; 2) determine criteria for the design of apparatus for flow testing and accelerated wear testing; 3) determine ranges of test parameters; 4) establish test procedures; and 5) develop analysis methods.

2.2 DESIGN

Based on the criteria developed in the ABT, a test assembly (Figure 2.1) was designed to investigate the forcing function boundary conditions for a small tube bundle with a square tube pitch that is representative of inservice steam generators. Modifications to an existing flow loop (Figure 2.2) at PNL to accommodate the installation of a test assembly were also designed. The resulting system represented the straight section of the steam generator tubes. The design of the test assembly provided for an instrumented tube support plate (Figure 2.3) that would accommodate testing at various tube/-tube-support plate clearances at the tube under study.

An apparatus (Figure 2.4) was also designed to accommodate accelerated wear testing inside an autoclave at elevated temperature and pressure, based on the boundary conditions established by the flow loop tests.

2.3 PROCUREMENT, FABRICATION, AND INSTALLATION

Materials and instrumentation were procured subsequent to finalizing the designs for the flow loop modification and the accelerated wear test apparatus.

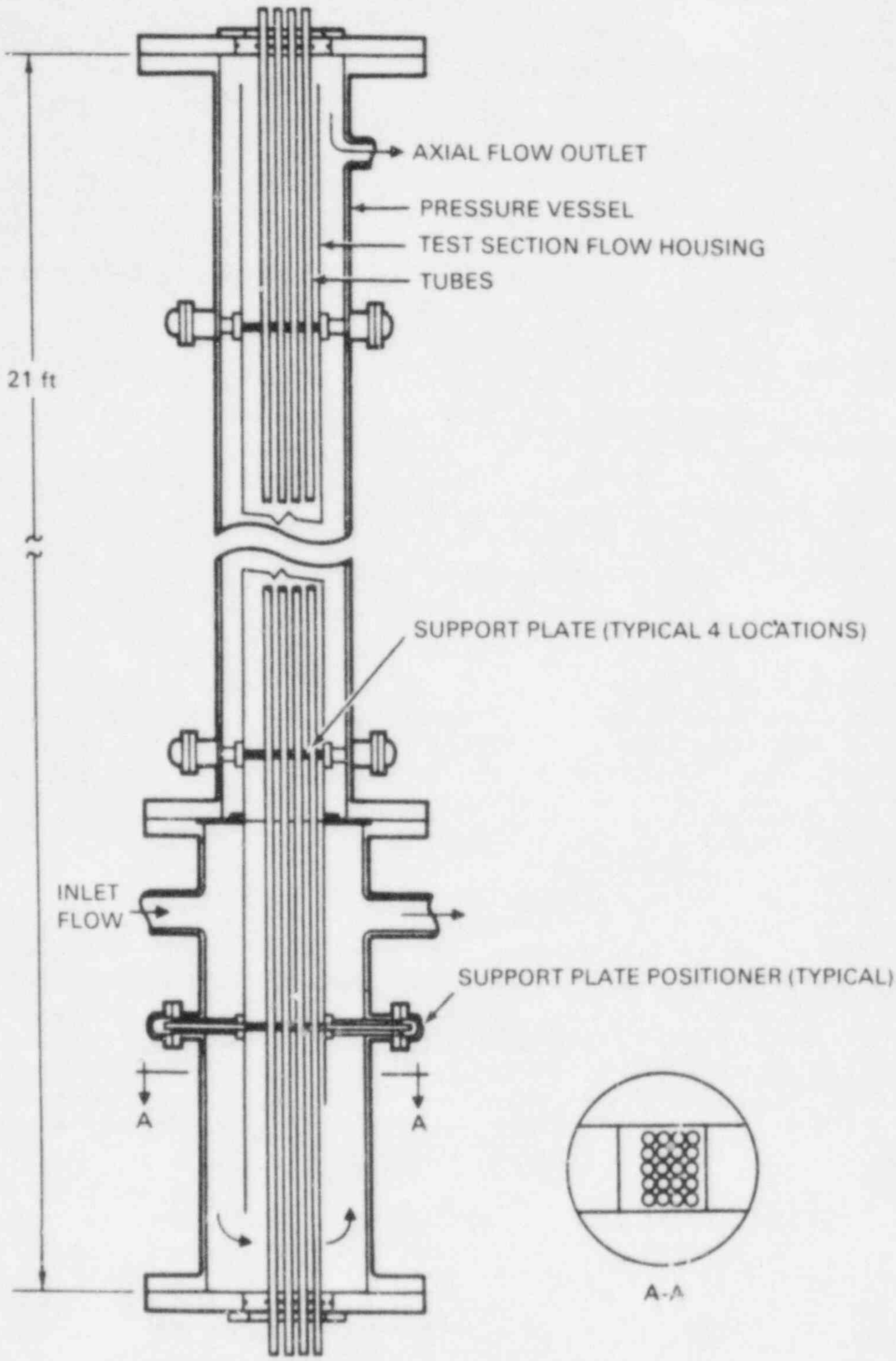

FIGURE 2.1. Full-Length Scale Model of Steam Generator

Source: Enderlin and Baugh (1985, p. 4)

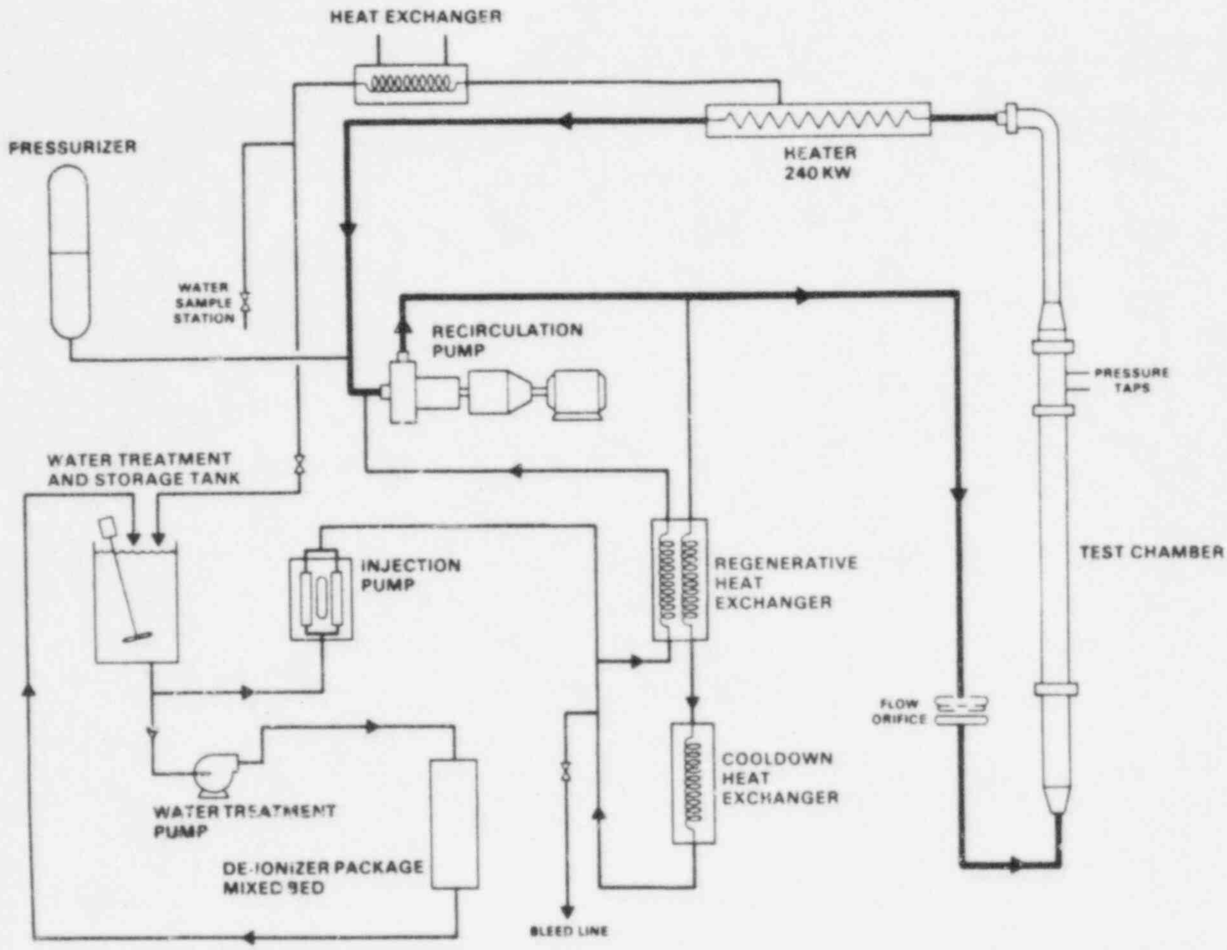

FIGURE 2.2. Flow Loop Schematic

Source: Enderlin and Baugh (1985, p. 5)

The flow loop test assembly and associated piping were fabricated off-site. Inspection and acceptance of these components was performed at the vendor's location and upon arrival at PNL by a representative from PNL. Installation of these components was by PNL Craft Services.

The accelerated wear test apparatus was fabricated onsite.

2.4 FLOW TESTING

Flow tests were conducted in a wet environment at low temperature and pressure and at elevated temperature and pressure for various tube/tube-support plate clearance conditions to establish the forcing function boundary conditions to be used in the accelerated wear tests. Flow sweep tests were conducted at low temperature and pressure to determine the onset of hydrodynamic instability under each of the clearance conditions considered.

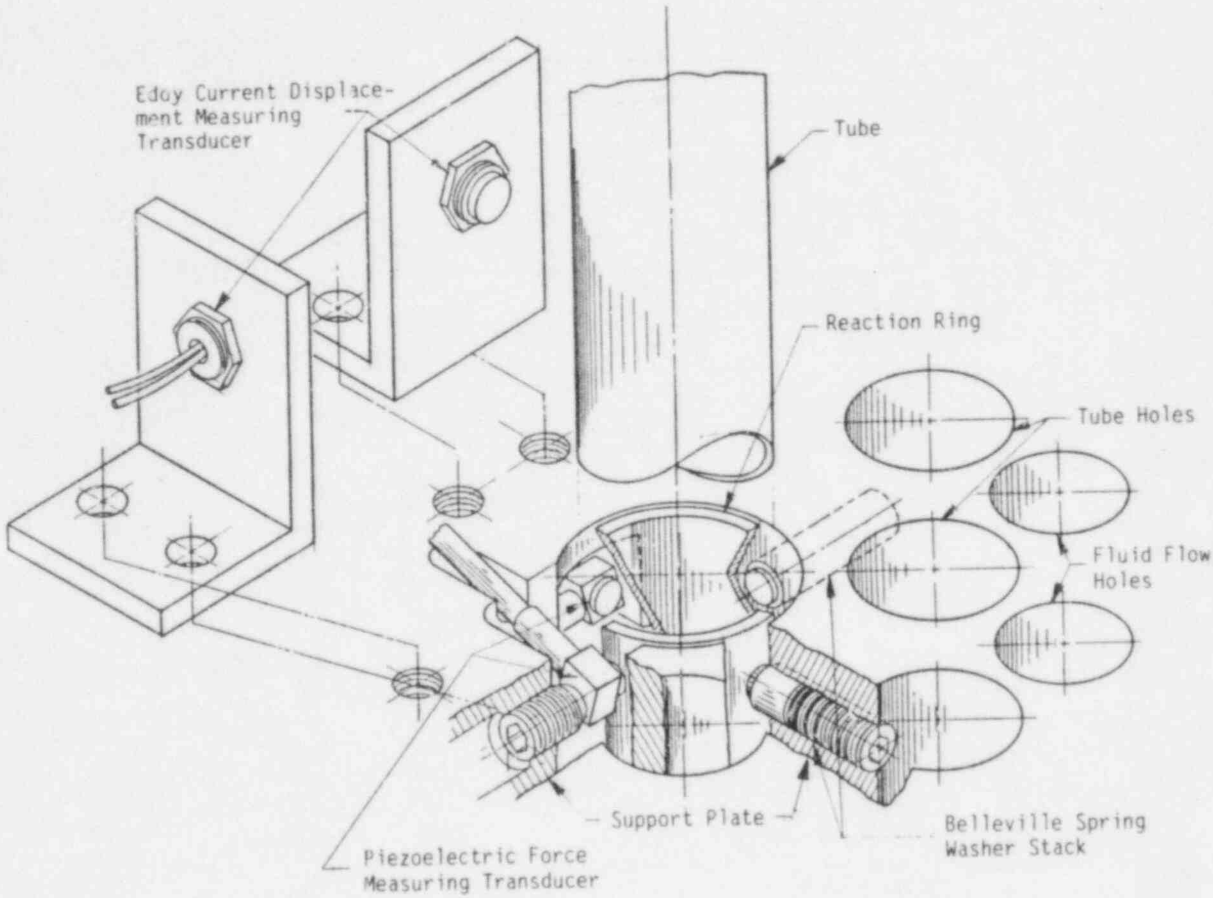

FIGURE 2.3. Instrumented Tube Support

2.5 ACCELERATED WEAR TESTS

In the event the flow tests demonstrated that an environment does, in fact, exist for increased tube wear rate at the increased tube clearance conditions considered, and that a forcing function could be determined, accelerated wear tests would be performed at elevated temperature and pressure using the accelerated wear test apparatus (Figure 2.4). Vibration would be imparted to the tube by a vibrator mounted on the top end of the tube specimen.

Initial parametric wear tests would be performed to demonstrate satisfactory performance of the vibrator under actual test conditions and to determine the relationship between tube/tube-support contact force and tube wear rate. These tests would provide a basis for extrapolating the results of the accelerated wear tests to obtain the predicted tube wear rate under actual operating conditions.

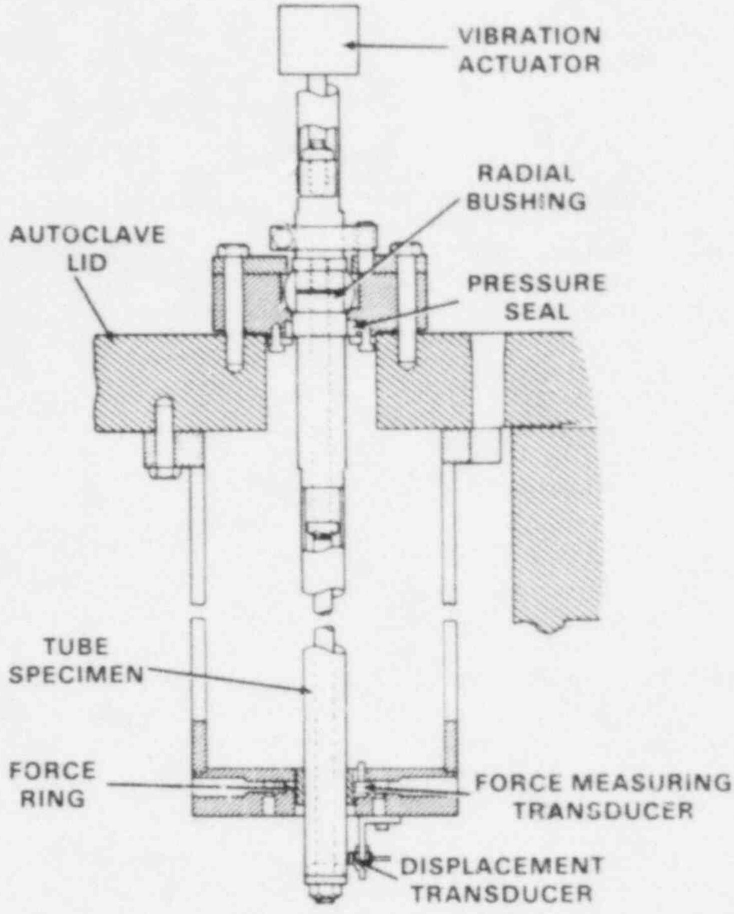

FIGURE 2.4. Wear Test Apparatus

Upon satisfactory completion of the initial parametric wear tests, the vibrator would be adjusted to impart the type of tube motion that was experienced in the flow loop. Tube wear would be accelerated by conducting the wear tests at a vibration frequency greater than what was experienced in the flow loop. The other test parameters would be matched to those experienced in the flow loop; however, the fluid would not be flowing. Tube wear for a given number of cycles would be determined by measuring the reduction in tube outside diameter using an optical noncontact gauging apparatus (Figure 2.5).

2.6 DATA ANALYSIS

A data analysis plan was prepared. Software for interfacing the data acquisition system with the Apollo computer, together with software for reducing the data generated in the flow loop, was developed (Enderlin and Baugh 1985).

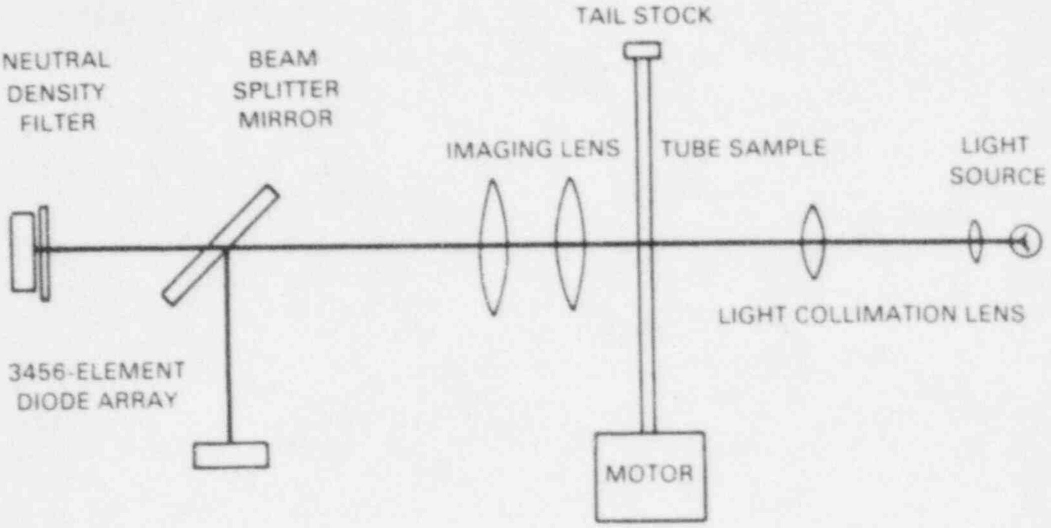

FIGURE 2.5. Optical Noncontact Gauging Station

In the event wear tests would be performed, tube wear rates (mil/1000 hr) for the accelerated wear environment would be determined for each gap condition considered. These wear rates would be extrapolated to predicted wear rates under actual operating conditions after chemical cleaning, using the relationship between tube/tube-support contact force and tube wear rate established in the initial parametric wear tests.

3.0 ANALYSIS-BEFORE-TEST

This section describes the equipment and instrumentation used to conduct the flow tests and acquire the data. The equipment to be used to conduct accelerated wear tests, if appropriate, is also detailed.

3.1 FLOW TEST EQUIPMENT

Flow-induced vibration experiments using the steam generator test assembly were conducted in an existing flow vibration test facility at PNL. The flow test assembly and the flow loop are described in the following sections.

3.1.1 Flow Test Assembly

A test assembly was constructed to model the Westinghouse steam generator design. The assembly models the length of the steam generator tube bundle between the tubesheet and the fifth tube support plate and a width of five tube rows in the region immediately adjacent to the bundle wrapper. The depth of the test assembly tube array is four rows for the square pitch layout of the Westinghouse reference design. Table 3.1 lists the principal dimensions of the test assembly, which is illustrated in Figure 2.1.

A flow housing, which encloses the test bundle, provides a downcomer and an inlet window that models the referenced steam generator inlet. In addition, the housing channels the flow to simulate both the crossflow and the axial flow occurring in the modeled section of the steam generator.

The flow housing and test bundle assembly are enclosed in a pressure vessel that is designed to contain the flow loop water at operating pressure and temperature. At each end of the pressure vessel the tubes pass through tubesheet flanges that seal the fluid while enabling access to the tube bores for inserting instrumentation. The pressure vessel also contains ports that provide access to the tube support plates for their positioning.

The principal design conditions for the test assembly are 420°F and 650 psig. The normal operating conditions are limited to 400°F and 550 psig.

3.1.2 Tube Array

The Westinghouse steam generator design employs a bundle geometry having a square pattern of tubes. In this arrangement, centers of adjacent tubes are located at the vertices of squares. This pattern is oriented in the test assembly model so that a plane containing the axes of adjacent tubes in the array is parallel to the direction of the approach flow. This orientation results in an "in-line" array with the tubes of succeeding rows falling immediately behind the tubes of the preceding rows as shown in Figure 2.1, Section A-A.

TABLE 3.1. Principal Flow Test Assembly Dimensions

	Westinghouse
Tube OD x wall, in.	0.875 x 0.050
Tube array pitch, in.	1.281
Number of tubes, pattern	20 square
Flow housing dimensions: axial flow housing width x depth, in.	7.162 x 6.772
crossflow housing width, in.	6.404
Wrapper wall to near tube axis, in.	1.91
Side wall to near tube axis, axial flow, in.	1.019
Rear wall to near tube axis, axial flow, in.	1.019
Side wall to near tube axis, crossflow, in.	0.640
Lower tubesheet to 1st support centerline, in.	50-1/8
1st support centerline to 2nd support centerline, in.	50-1/2
2nd support centerline to 3rd support centerline, in.	50-1/2
3rd support centerline to 4th support centerline, in.	50-1/2
4th support centerline to upper tubesheet, in.	50-1/8
Support plate thickness, in.	0.750
Downcomer width x depth, in.	6.404 x 2.56
Bundle inlet port width x height, in.	6.404 x 14

The tube array models the full-scale tube diameter of 0.875 in. and tube pitch of 1.281 in., and incorporates a group of tubes five rows wide and four rows deep.

3.1.3 Flow Housing

Flow supplied to the test assembly pressure vessel enters into a downcomer channel, which is part of the inlet flow housing. The downcomer represents a segment of the annular flow passage formed by the space between the steam generator pressure vessel ID and the bundle wrapper OD, and is modeled so as to provide a flow field entering the bundle that is representative of the steam generator.

The lower portion of the downcomer opens into the bundle region through a port with a vertical dimension of 14 in. This space is equal to the gap between the bottom of the bundle wrapper and the tubesheet of the steam generator.

Flow entering the bundle separates into a crossflow fraction and an axial flow fraction. The crossflow fraction passes through the bundle confined by the flow housing side walls and exits through an outlet nozzle opposite the inlet nozzle in the pressure vessel. The axial flow fraction passes up along the bundle, confined within the bundle flow housing, and exits near the bottom of the upper pressure vessel. The axial flow/crossflow ratio is controlled by valves located in the outlet piping lines.

The walls of the flow housing consist of the front wall, which represents the inner surface of the bundle wrapper, and the remaining walls, which do not have real counterparts in the actual steam generator. The nominal dimension between the front wall and the near tube axis was determined from the prototype steam generator dimension to be 1.910 in.

The side and rear walls are positioned to minimize the influence of the walls on the bundle flow distribution. In the axial flow region, the equal hydraulic radius method was applied. Using this method, the hydraulic radius, which is the area of the flow cross section divided by the wetted perimeter, was calculated for a tube array unit flow cell. The tube array unit cell is the region enclosed by joining the centers of four adjacent tubes. The cell boundaries and the surface of the tube sectors form the boundary of the flow cross section while the surface perimeter of the tube sectors form the wetted perimeter. By adjusting the wall-to-tube spacing, the hydraulic radius for the wall unit flow cell was made equal to that of the array unit cell. As a result, the average axial flow velocity along the wall will be nominally the same as the average velocity within the bundle, thereby minimizing the wall effect. The appropriate wall-to-tube axis spacing was determined to be 1.019 in., which resulted in the flow housing sidewall-to-sidewall dimension of 7.162 in. x 6.772 in. front wall-to-rear wall.

In the inlet crossflow region of the bundle, the sidewall spacing was established by equating the head loss calculated for flow through the sidewall channel to the head loss calculated for flow through the bundle, assuming equal minimum gap velocities. Here, the head loss results from changes in velocity; the decrease in velocity caused by flow area expansion produces the predominate loss. These expansion losses take the form

$$h_L = K_L \frac{\left(1 - \frac{A_1}{A_2}\right)^2 V_1^2}{2g}$$

where h_L = head loss, ft of fluid

K_L = loss of coefficient, dimensionless

A_1 = contracted flow area, ft^2

A_2 = expanded flow area, ft^2

V_1 = velocity at contracted flow area, ft/sec

g = acceleration of gravity, ft/sec^2.

It was assumed that the loss coefficient for flow through the tube-sidewall gap was equal to the coefficient for flow through the tube-tube gaps. This is reasonable because the shapes of the expansion geometries are similar when the sidewall is considered a plane of symmetry. Further, in the region of rapid expansion, the coefficient is relatively unchanging. In the square pitch tube array, the number of expansion losses encountered when passing through the bundle is equal to the number encountered along the sidewall. Thus, the sidewall spacing was selected so the contraction-expansion area ratio was equal to that of the bundle. The sidewall-to-tube axis spacing was calculated to be 0.640 in., which resulted in the flow housing sidewall-to sidewall dimension of 6.404 in. in the crossflow region. To provide this, spacing filler plates were mounted on the sidewalls of the 7.162-in.-wide flow housing.

3.1.4 Tube Support

The test assembly has simulated tube support plates at axial locations that model the design and axial spacing for the prototype steam generator. The support plates are made of 0.75-in.-thick carbon steel drilled for the 20 tube holes and 16 flow holes. The flow holes have a diameter of 0.750 in. The tube holes for all but the monitored tube at the instrumented support plate were sized at 0.920 in. to represent a 20-mil oversize condition. The support plates were securely positioned in the flow housing by adjustable mounting fixtures installed in the wall of the pressure vessel. The mounting fixtures permit the adjustment of each plate's lateral position 0.125 in. in either direction relative to the direction of flow at the inlet. After adjustment, the fixtures rigidly secure the support plates to the pressure housing.

3.1.5 Pressure Vessel

The test bundle and flow housing were installed in an ASME Code-stamped pressure vessel that includes the inlet and both cross- and axial-flow outlet nozzles. The vessel was constructed of stainless steel and made in two sections; the lower section contains the inlet flow housing and the upper section contains the axial flow housing. Nozzles along the vessel, located at the planes of the support plates, provide access for the plate mounting fixtures.

The vessel end closures consist of flanges, each containing a tandem pair of tubesheet type flat head inserts. Each of these circular inserts contains the hole pattern for the tube array, and the holes each contain O-ring seals to seal the tubes as they penetrate the insert. In this manner, the tube ends extend outside the pressure boundary. The pair of inserts is each sealed in the bore of the closure flange and is retained by a retainer ring. A space between the two inserts, formed by a spacer ring, allows for the circulation of coolant to cool the O-ring seals. To model the density of the steam generator primary-side fluid within the tubes, the tubes were filled with sunflower seed oil.

3.2 INSTRUMENTATION

The instrumentation used in the flow loop tests monitored the displacement and force of a single tube relative to the tube support plate. The force and displacement transducers were independent systems positioned around the tube in a biaxial arrangement. The two transducers in each system were positioned at 90 degrees to each other and 45 degrees to the direction of flow. The plane of measurement in each system was parallel to the support plate and perpendicular to the axis of the tube.

3.2.1 Data Acquisition System

The data acquisition system used was a MegaDAC 2200C manufactured by Optim Electronics of Gaithersburg, Maryland. The unit is a programmable dynamic data acquisition system capable of sampling rates up to 20,000 samples per second. The system has 32 analog input channels with a range of up to ±10 volts. Mass storage for the device is a magnetic tape cartridge capable of storing 30 megabytes of digitized data. The unit is equipped with RS-232C and IEEE-488 standard interfaces for communication with other digital devices.

The configuration of the system for the flow loop tests used four channels, two for displacement and two for force. The displacement channels were set for a range of ±2 volts to accommodate ±0.050 in. of displacement. The force channels were set to a range of ±0.2 volt corresponding to ±20 lbf. The sampling rate used was 20,000 samples per second, giving each channel a rate of 5,000 samples per second. Acquired data was stored on the magnetic tape cartridge and later transferred to a large data processor through the RS-232C interface.

3.2.2 Oscilloscopes

Two dual trace oscilloscopes were used to provide photographic records of the force and displacement waveforms. The oscilloscopes were configured to provide either amplitude versus time waveforms or lissajous patterns.

3.2.3 Filters

In order to filter out high frequency components that interfered with the data of interest, low-pass filters were used. The filters used were dual low-pass filters with cutoff frequencies that are adjustable from 20 Hz to 20 MHz. During the tests, these filters were adjusted for a cutoff frequency of 100 Hz.

3.2.4 Force Measuring System

The force transducers in this system were piezoelectric devices manufactured by Kistler Instrument Corporation of Amherst, New York. The units, Kistler Model 910M113, are roughly 0.250 in. in diameter and 0.250 in. high with a hermetically sealed, high temperature leadwire. Maximum range of these transducers is 600 lbf and nominal sensitivity is 18 pc/lb.

Charge amplifiers were used to condition the force transducers. The charge amplifiers used were PCB Model 463A manufactured by PCB Piezotronics, Inc., of Buffalo, New York.

3.2.5 Displacement Measuring System

The displacement measuring system used for these tests was an eddy-current device manufactured by Kaman Instrumentation Corporation of Colorado Springs, Colorado. The system, Kaman Model KD-1925, has a range of 0.050 in. and a sensitivity of 0.040 volt/0.001 in. The transducer is a high temperature unit roughly 0.250 in. in diameter with a hermetically sealed leadwire. A diagram of the instrumentation system is shown in Figure 3.1.

3.3 FLOW TEST MODELING

3.3.1 Hydrodynamic Similitude

Similarity of flow fields is in general based on three dimensionless parameters (Sabersky, Acosta, and Hauptmann 1971): the Euler number, the Froude number, and the Reynolds number. The Strouhal number is not considered because it is only a particular property of the flow field, not a defining parameter (Batchelor 1967). In addition, the Mach number, which is significant near sonic velocities, may also be discarded. Of the three dimensionless parameters generally used, two may be eliminated in certain situations. For example, when no free surfaces are present, the Froude number may be ignored. The Euler number is important only when cavitation is of concern. Therefore, equating Reynolds numbers alone is sufficient to produce a hydrodynamically similar, although distorted, model of the steam generator. Thus,

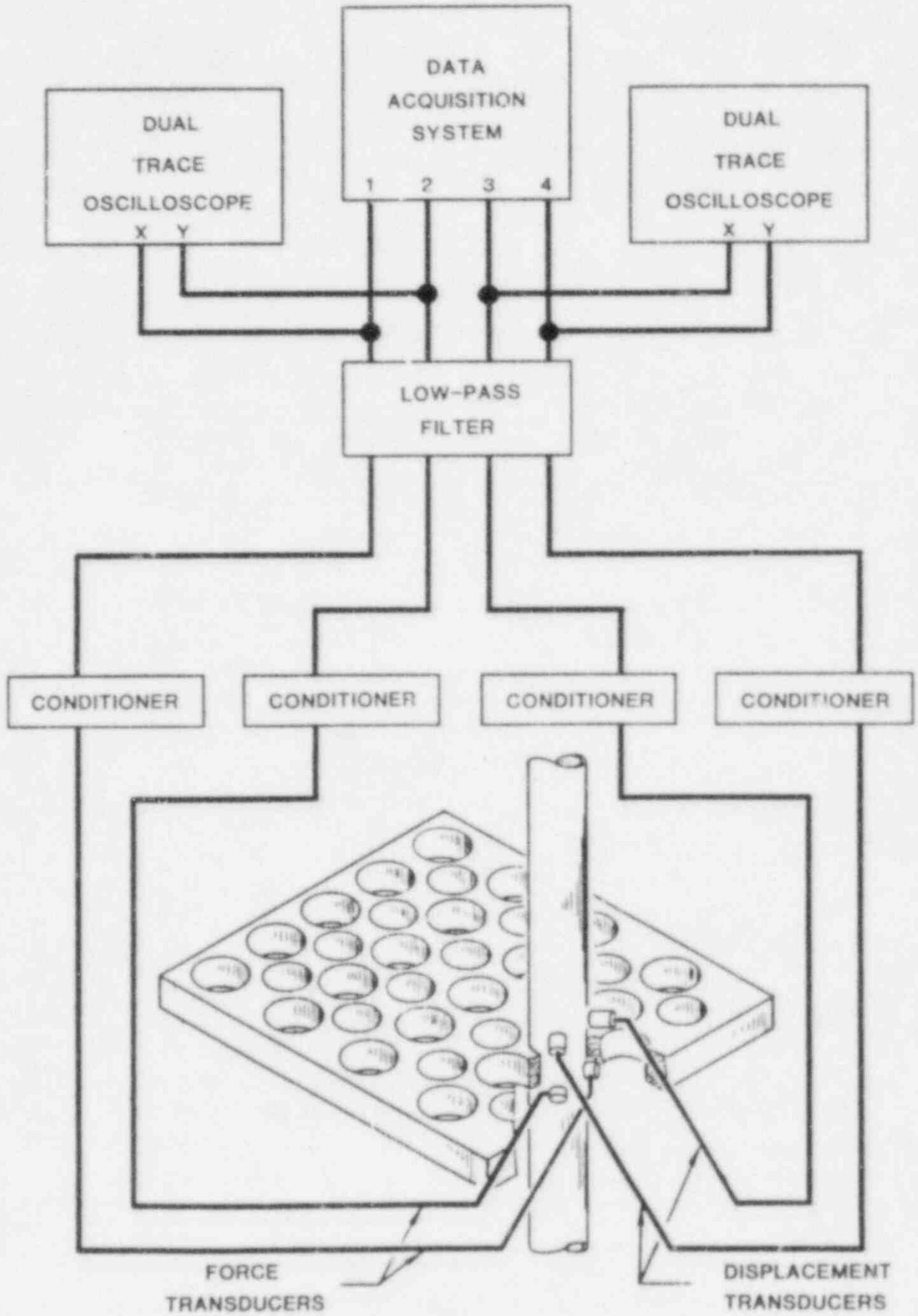

FIGURE 3.1. Flow Loop Instrumentation

$$R_m = R$$

$$\frac{V_m L_m \rho_m}{\mu_m} = \frac{VL\rho}{\mu}$$

where R = Reynolds number
V = velocity of the fluid
ρ = density of the fluid
L = a geometric property
μ = absolute viscosity of the fluid
and m denotes the model.

Flow conditions for the test assembly model were based on a Westinghouse Electric Corporation steam generator prototype, typical of that used at the Trojan Nuclear Power Station of the Portland General Electric Company. Table 3.2 lists the principal geometric and performance design data for the prototype steam generator. The Reynolds number for the prototype full power inlet total flow condition was the basis for the equal Reynolds number criterion of the model. In vibration studies where cylinder crossflow is the principal interest, the characteristic Reynolds number $Re = \rho \frac{VL}{\mu}$ uses the free stream velocity upstream of the cylinder for V and the cylinder diameter for L. Using the equal Reynolds number criterion, the flow rate for the model was calculated by setting the model to have a Reynolds number equal to that of the prototype.

$$(Re)_m = (Re)_p = (\rho\frac{VL}{\mu})_m = (\rho\frac{VL}{\mu})_p$$

Because the tube (cylinder) diameter of the model was full scale, $L = L_p$; therefore,

$$(\rho\frac{V}{\mu})_m = (\rho\frac{V}{\mu})_p$$

Both the fluid density ρ and the fluid viscosity μ are temperature-dependent and, to a minor degree, pressure-dependent. A heat balance applied to the full load design data revealed the total flow inlet temperature of the prototype to be 511°F. This temperature and an inlet pressure of 920 psia were used for Reynolds number calculations for the prototype. For the model, the reference fluid conditions were the design operating conditions of 400°F and 500 psia. Substituting density and viscosity values at the indicated fluid conditions into the Reynolds number equality determined the velocity relationship for the model at 400°F to be

$$V_m = 1.175\ V_p$$

where

$$V = \frac{Q\ total}{A\ inlet}$$

The volumetric total flow rate for the prototype was found to be 5193 ft^3/min at 511°F, calculated from the full load total flow design data.

TABLE 3.2. Steam Generator Bulk Design Data

Item	Trojan (W)
Geometric Data	
Height, ft	67.7
Outer diameter, ft (upper/lower)	14.6/11.2
Maximum bundle diameter, in.	119.665
Wrapper inner diameter, tube centers, in.	123.5
Downcomer annulus width, in.	2.56
Tube bundle length, ft	40
Number of U tubes	3388
Tube outer diameter, in.	0.875
Tube wall thickness, in.	0.050
Tube array	Square
Tube spacing pitch, in.	1.281
Tube support plate hole diameter, in.	0.900
Tube support plate flow hole diameter, in.	0.758
Tube support plate thickness, in.	0.75
Tube support plate spacing, in.	50.5
Tube support plate-to-wrapper clearance, in.	0.38
Primary Side Design	
Maximum pressure, psia	2485
Maximum temperature, °F	650
Total heating area, ft^2	51,500
Secondary Side Design	
Maximum pressure, psia	1,185
Maximum temperature, °F	600
Full load pressure, psia	910
Full load temperature, °F	533
Full load steam flow, lb/hr	3.77×10^6
Full load total flow, lb/hr	15.08×10^6
Feedwater temperature, °F	440

The flow area location used to establish the free stream velocity for the steam generator was the inlet flow window formed by the circumference of the tube bundle wrapper and the height of the space between the tubesheet and the bottom of the wrapper. The flow area is

$$(A_{in})_p = \pi \; d \; h = \pi \; (123.5) \text{ in. } (14) \text{ in. } \left(\frac{1}{144}\right) \frac{ft^2}{in.^2} = 12.01 \; ft^2$$

Solving for the prototype free stream velocity gives

$$V_p = \frac{5193 \; ft^3/min}{12.01 \; ft^2} = 137.7 \; ft/min$$

For the model, the flow area location comparable to the prototype was the inlet window 6.404 in. wide and 14 in. high, which yielded a flow area

$$(A_{in})_m = w \; h = (6.404) \text{ in. } (14) \text{ in. } \left(\frac{1}{144}\right) \frac{ft^2}{in.^2} = 0.6226 \; ft^2$$

The resulting total volumetric flow rate for the model at 400°F became

$$(Q_{total})_m = (V_{in})_m \; (A_{in})_m = 1.175 \; (V_{in})_p \; (A_{in})_m$$

$$= 1.175 \; (137.7) \; \frac{ft}{min} \; (0.6226) \; ft^2 \; (7.48) \; \frac{gal}{ft^3} = 753.5 \; gpm$$

As noted earlier, the test assembly was arranged to measure and control both the axial flow and crossflow exiting the model. The axial flow rate was determined from the average axial mass velocity of the steam generator. The axial cross-sectional flow area for the steam generator is

$$(A_{axial})_p = A_{wrapper} - A_{tubes} = \frac{\pi}{4(144)} \frac{ft^2}{in.^2} [(123.5)^2$$

$$- 6776(0.875)^2] \text{ in.}^2 = 54.89 \; ft^2$$

The axial mass velocity is

$$(G_{axial})_p = \frac{\dot{m}}{A_{axial}} = \frac{(15.08 \times 10^6)}{54.89 \text{ ft}^2} \frac{\text{lb}}{\text{hr}} = 2.747 \times 10^5 \text{ lb/hr-ft}^2$$

which, when evaluated at 511°F, results in an average axial velocity

$$(V_{axial})_m = A_{shroud} - A_{tubes} = \frac{1}{144} \frac{\text{ft}^2}{\text{in.}^2} [(7.162)(6.772)$$

$$- 20 \frac{\pi}{4}(0.875)^2] \text{ in.}^2 = 0.2533 \text{ ft}^2$$

and the axial flow rate for the model becomes

$$(Q_{axial})_m = (V_{axial})_m (A_{axial})_m = 1.175(V_{axial})_p (A_{axial})_m$$

$$= 1.175 \ (94.60) \frac{\text{ft}}{\text{min}} (0.2533) \text{ ft}^2 \ (7.48) \frac{\text{gal}}{\text{ft}^3} = 210.6 \text{ gpm}$$

An independent determination of the axial flow rate was provided by M. B. Carver (1982), in which local mass velocity vector mapping was shown for the prototype steam generator under full power conditions. The mass velocity vectors adjacent to the wrapper wall and within the nearby bundle were used to calculate axial flow rates in those regimes within the model. The resulting axial flow rate for the model was found to be 204.4 gpm, which is a good check on the flow rate determined on an average velocity basis.

The crossflow for the model is determined by difference.

$$(Q_{cross})_m = (Q_{total})_m - (Q_{axial})_m = 753.5 - 210.6 = 533.9 \text{ gpm}$$

In conducting the experiments, the crossflow fraction of the total flow was increased about 10%, which ensured a conservative treatment of flow measurement uncertainties and crossflow vibrations within the steam generator.

The tube support plate location adjustments were made at 100°F. The flow rates were chosen to permit drag forces at 100°F to equal those experienced by the model at 400°F.

Set $D_{100} = D_m = (\frac{1}{2} C_D \rho V^2)_{100} = (\frac{1}{2} C_D \rho V^2)_m$ and assume that $(C_D)_{100} = (C_D)_m$, then

$$\frac{D_{100}}{D_m} = \frac{(\rho V^2)_{100}}{(\rho V^2)_m} = 1$$

$$V_{100} = \left(\frac{\rho_m}{\rho_{100}}\right)^{\frac{1}{3}} V_m = 0.866\ V_m$$

or

$$= (0.866)(1.175\ V_p) = 1.017\ V_p$$

4.0 TEST PERFORMANCE

Flow tests were performed under a design tube/tube-support clearance condition (12.5-mil annular clearance) and under a 10-mil-over-design-clearance condition (17.5-mil annular clearance), which was intended to simulate a conservative post-chemical cleaned condition. At the beginning of the test series for each clearance condition, the horizontal position of the instrumented support plate was adjusted in the crossflow direction until maximum dynamic tube displacement was indicated on the oscilloscope with the fluid crossflow velocity adjusted to produce prototypic fluid drag conditions on the tube under study at 100°F water temperature. Flow tests were then conducted at 400°F and 550 psia with a water inlet velocity of 162 fpm, the conditions for prototypic Reynolds number at 100% reactor power. Flow tests were conducted also at fluid velocities corresponding to the 50%, 75%, 125%, and 150% power reactor levels. After the completion of both test series, prototypic fluid drag conditions were again established at 100°F water temperature under design clearance conditions. The tube under study was then withdrawn through the top of the test vessel until the tube displacement signal disappeared, indicating that the bottom of the tube had cleared the instrumented support ring. The established flow conditions were maintained, and the force signals were recorded after the tube had been removed.

4.1 DATA REDUCTION AND EVALUATION

Raw data was obtained in the form of x and y forces and displacements at discrete time intervals for each of the tube clearance conditions considered. Thus,

$$\underset{\sim}{F}_i = \begin{Bmatrix} F_{x_1}, F_{x_2}, F_{x_3}, \ldots, F_{x_n} \\ F_{y_1}, F_{y_2}, F_{y_3}, \ldots, F_{y_n} \end{Bmatrix} \text{ for n data points}$$

$$\underset{\sim}{D}_i = \begin{Bmatrix} D_{x_1}, D_{x_2}, D_{x_3}, \ldots, D_{x_n} \\ D_{y_1}, D_{y_2}, D_{y_3}, \ldots, D_{y_n} \end{Bmatrix} \text{ for n data points}$$

where i denotes the clearances and ~ denotes an array.

Because these forces were 52% greater than what was anticipated for an actual steam generator, they were reduced by a factor of 1.52 or, in pseudo-code,

```
DO j = 1 TO n
  FX(j) = FX(j)/1.52
  FY(j) = FY(j)/1.52
END
```

This calculation was performed for each clearance condition considered. From the resulting forces and displacements, normal and tangential forces are computed by proper rotation of the x and y axes at each point in time, as shown in Figure 4.1.

The derivation for the rotation of coordinate systems is as follows:

$$F_{x_n} = F_x \cos\theta$$

$$F_{y_n} = F_y \sin\theta$$

$$F_{x_t} = -F_x \sin\theta$$

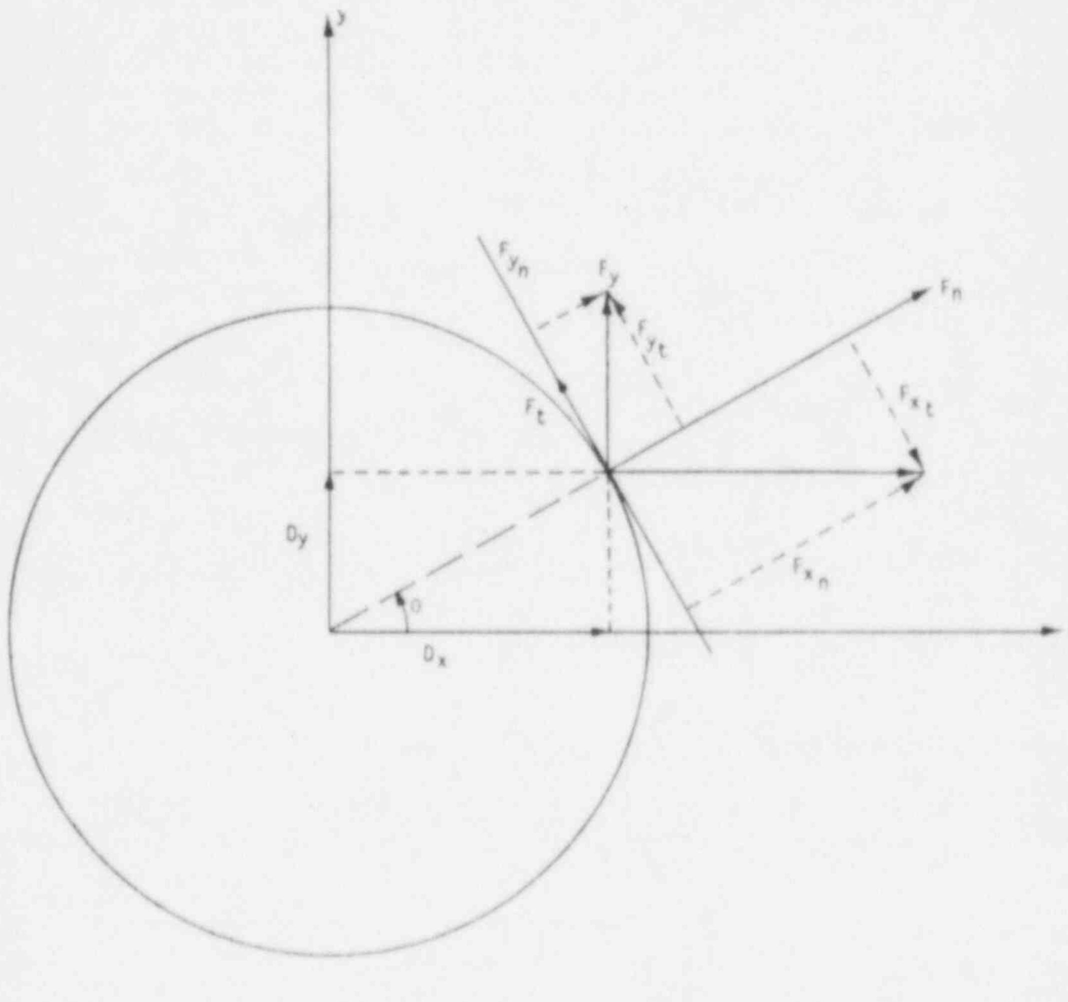

FIGURE 4.1. Rotation of Coordinate System

$$F_{y_t} = F_y \cos\theta$$

$$F_n = F_{x_n} + F_{y_n} = F_x \cos\theta + F_y \sin\theta$$

$$F_t = F_{x_t} + F_{y_t} = -F_x \sin\theta + F_y \cos\theta$$

or, in matrix form,

$$\begin{Bmatrix} F_n \\ F_t \end{Bmatrix} = \begin{bmatrix} \cos\theta & \sin\theta \\ -\sin\theta & \cos\theta \end{bmatrix} \begin{Bmatrix} F_x \\ F_y \end{Bmatrix}$$

where $\theta = \tan^{-1}(D_y/D_x)$.

This matrix relationship is valid for any set of x-y coordinates. Incorporation of this equation into the data reduction software is achieved by the algorithm

```
DO j = 1 TO n
 THETA(j) = ARCTAN (DY(j)/DX(j))
 FN(j) = FX(j)*COS(THETA) + FY(j)*SIN(THETA)
 FT(j) = -FX(j)*SIN(THETA) + FY(j)*COS(THETA)
END
```

The frequency of rotation, ω, was determined by plotting the value of θ versus time. The total number of cycles (one cycle is equal to 2π radians) divided by the elapsed time gives the frequency of rotation of the tube specimen, which is expected to correspond to that of the steam generator.

The following information was obtained for each of the clearance condition tested:

- normal and tangential force histories, which were reduced by the appropriate factor to satisfy hydrodynamic similitude requirements
- the type of motion (e.g., circular, elliptic, impact)
- the expected frequency of rotation of tubes in the steam generator.

Based on this information, the following plots were produced for each test:

- X displacement versus Y displacement
- X force versus Y force
- X displacement versus time

- Y displacement versus time
- X force versus time
- Y force versus time
- normal force versus time for 10-mil oversize case at the 100% power flow condition only.

4.2 INTERPRETATION OF RESULTS

The computer plots strongly indicate that the tube support was considerably less active under simulated post-cleaned conditions than at the design clearance condition. For the larger clearance, Figures 4.2 and 4.3 show tube displacement for the 100% power case to be an elliptical pattern with the major axis in the direction of flow, and displacement along the X and Y axes being about 3 mils. Intersection of the circle in Figure 4.2 would denote impact of the tube with the reaction ring. In comparison, for the design clearance case at 100% power, tube displacements are immeasurably small (less than 1 mil), as shown in Figure 4.4.

Moreover, in the larger clearance case, tube displacement became more active as fluid velocity was increased, as shown in Figures 4.5 and 4.6 for the 150% reactor power case. The total displacement of the tube from the null (no flow) position increased while the force remained relatively constant at about 0.5 lb and the contact envelope in Figure 4.5 was not intersected, which indicates that the tube was not confined by the support plate. For the design clearance case, the tube did not become more active as the fluid velocity was increased, but the magnitude of the force tended to increase. Further, the displacement from null position was only about one-tenth as far as for the oversized clearance case, indicating that the tube in this case was confined by the support plate.

There is also a decided lack of correlation between force and displacement histories over the same time interval in the case of the larger clearance. Moreover, negative forces are indicated in the plot of normal forces shown in Figure 4.7, which is an impossibility in the case of the tube contacting the support plate. These conditions suggest that the force signal is probably driven by something other than the tube contacting the support plate. There is reason to believe that, in this case, force is transmitted to the force sensors via hydraulic coupling, as forces in the range of 0.25 lb to 0.8 lb were detected after the tube had been withdrawn from the force ring (Figures 4.8 and 4.9). Plots of X force, Y force, X displacement, and Y displacement with time are shown in Figures 4.10, 4.11, 4.12, and 4.13 for the 100% power case at oversized clearance conditions. The time intervals for these plots are concurrent with each other and are also concurrent with the displacement history shown in Figure 4.3.

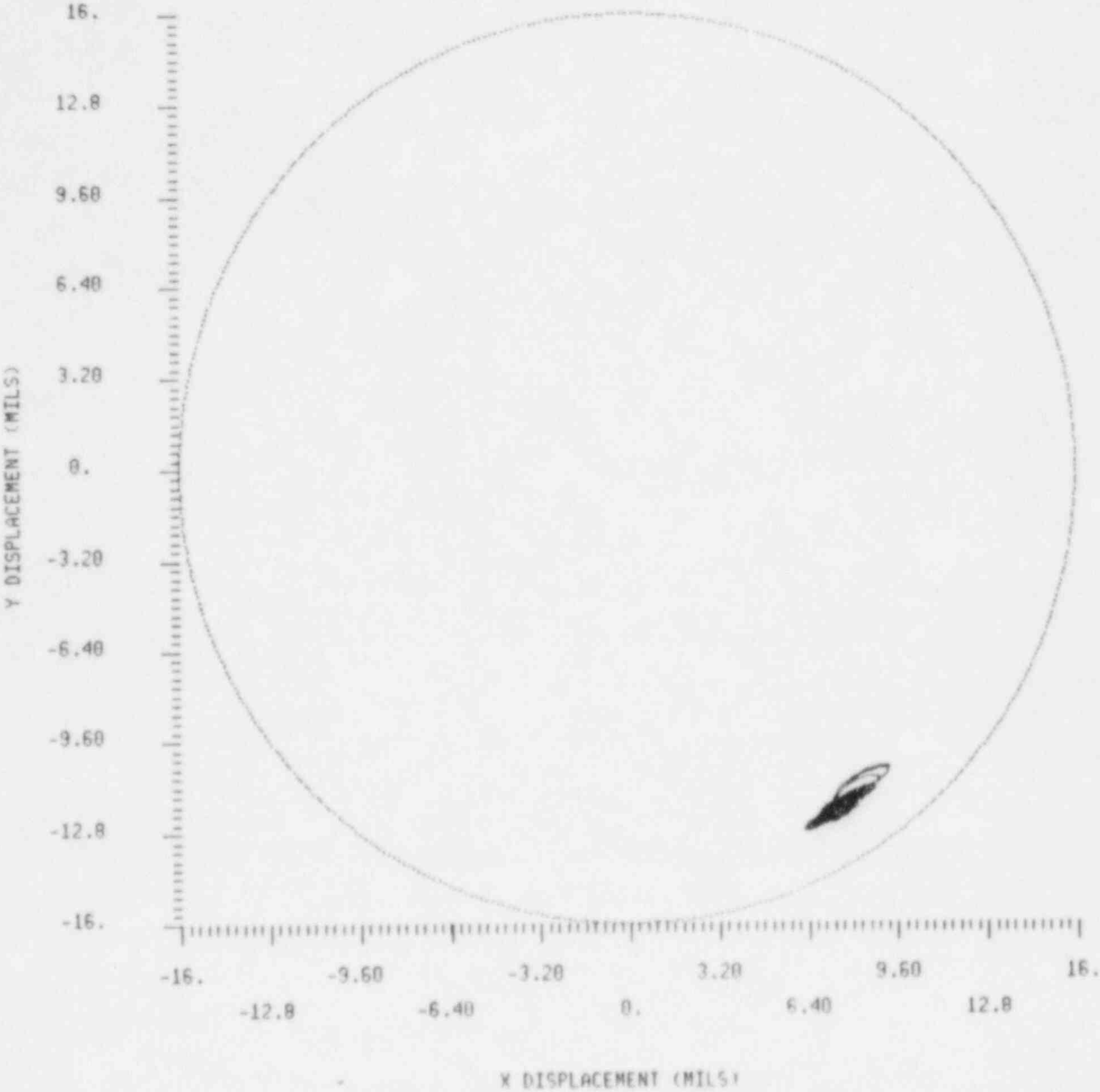

FIGURE 4.2. Contact Envelope, Y-Displacement Versus X-Displacement, 10 Mils Over Design Clearance, 100% Reactor Power

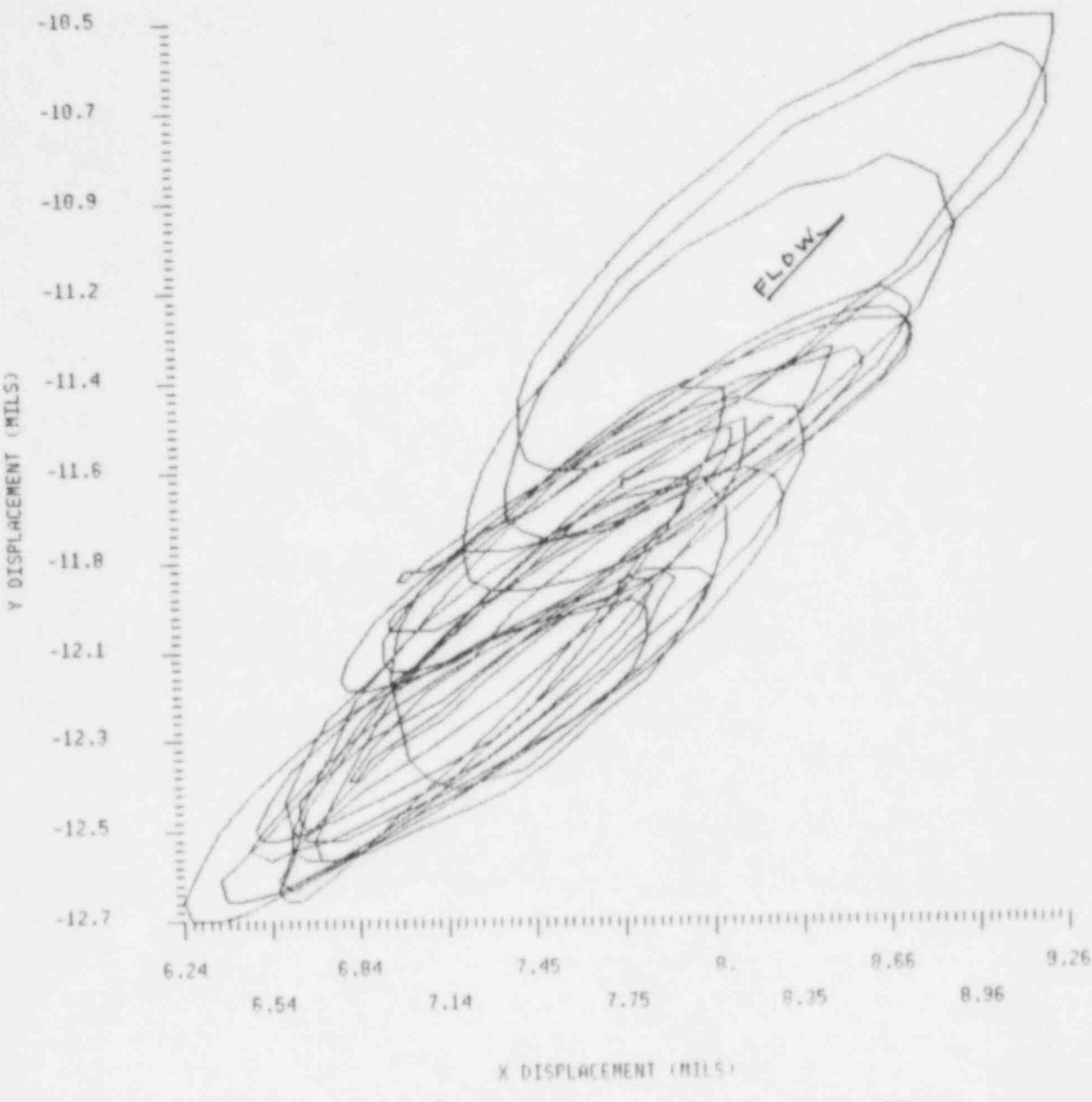

FIGURE 4.3. Y-Displacement Versus X-Displacement, 10 Mils Over Design Clearance, 100% Reactor Power

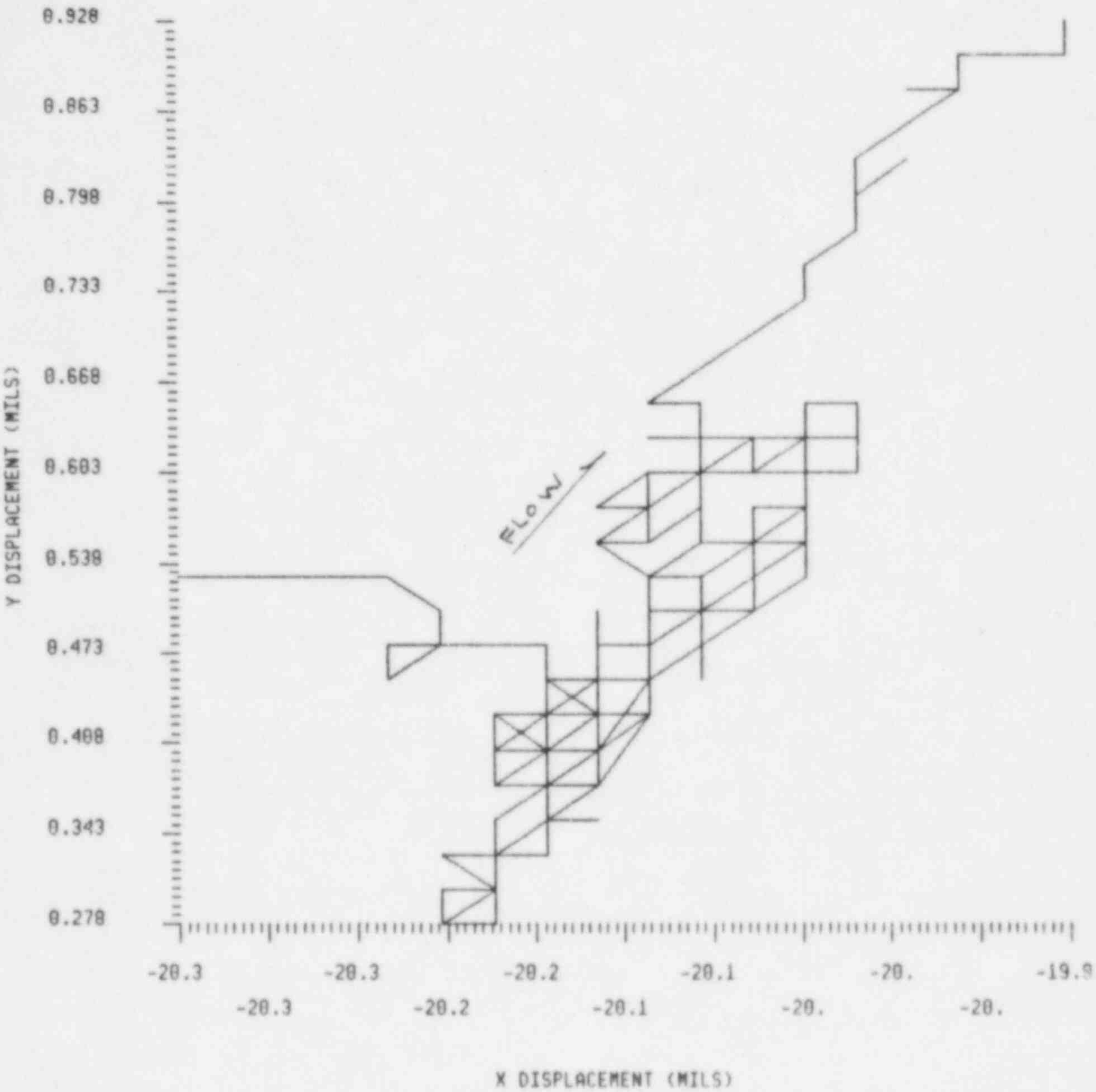

FIGURE 4.4. Y-Displacement Versus X-Displacement at Design Clearance

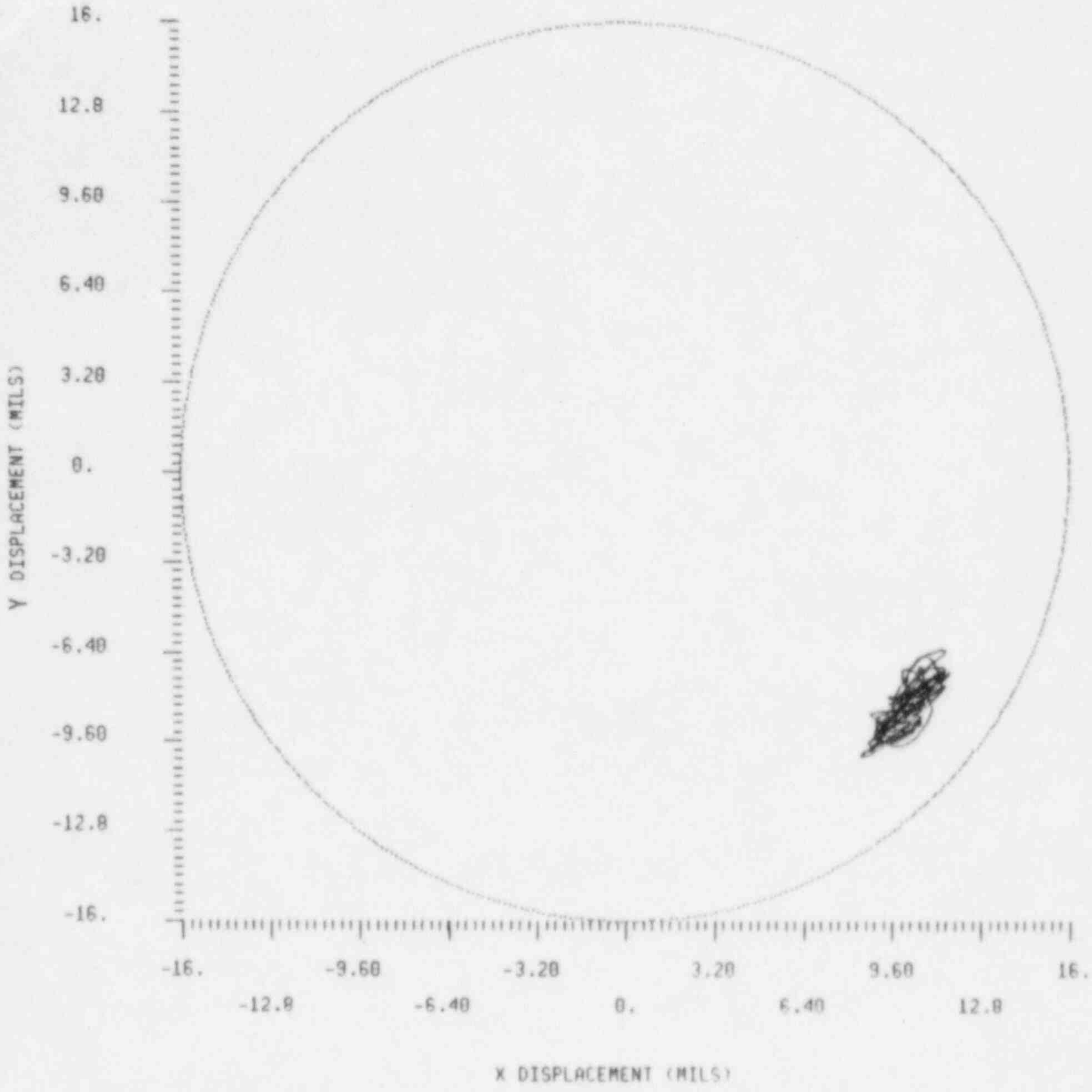

FIGURE 4.5. Contact Envelope, Y-Displacement Versus X-Displacement, 10 Mils Over Design Clearance, 150% Reactor Power

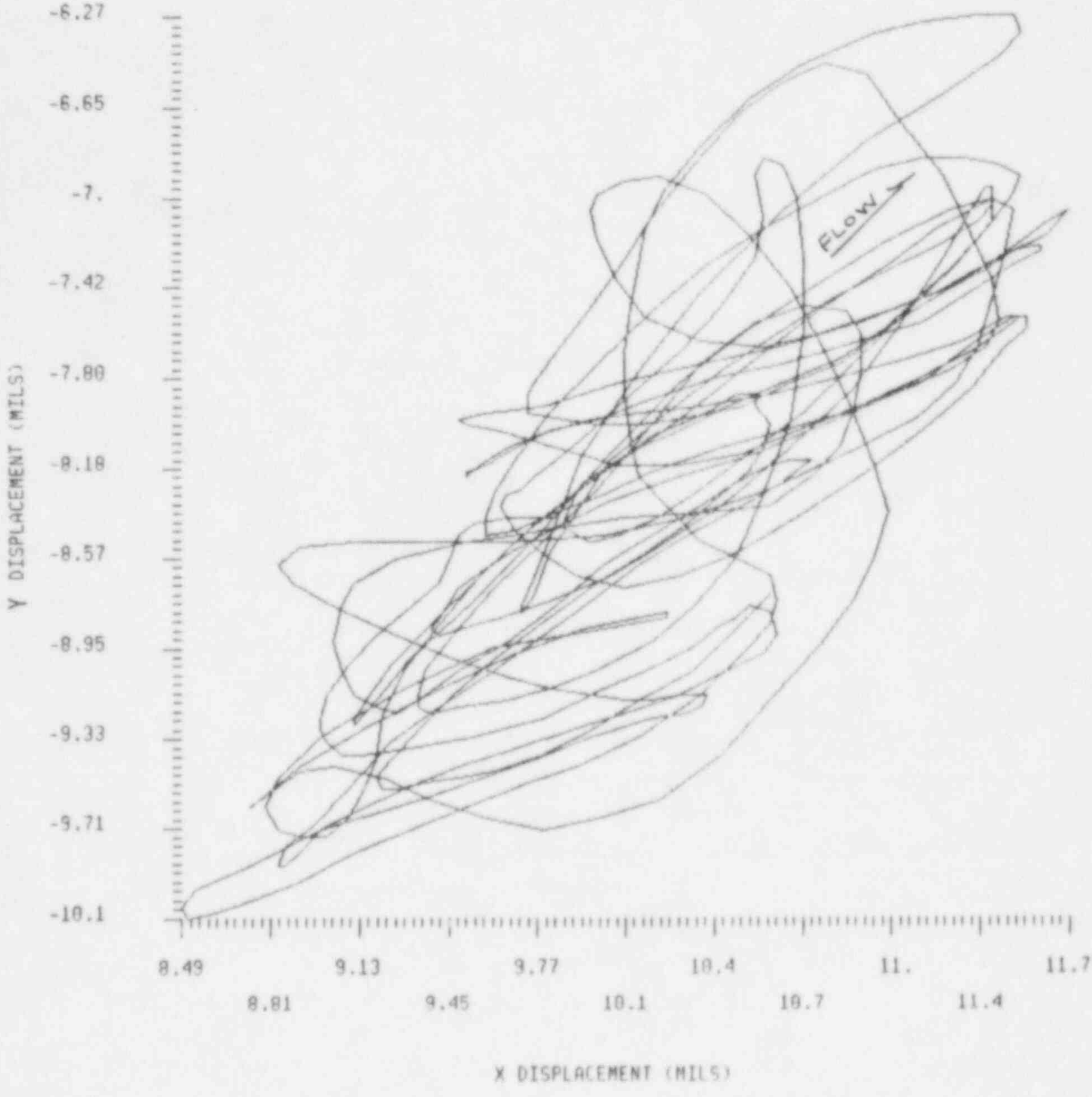

FIGURE 4.6. Y-Displacement Versus X-Displacement, 10 Mils Over Design Clearance, 150% Reactor Power

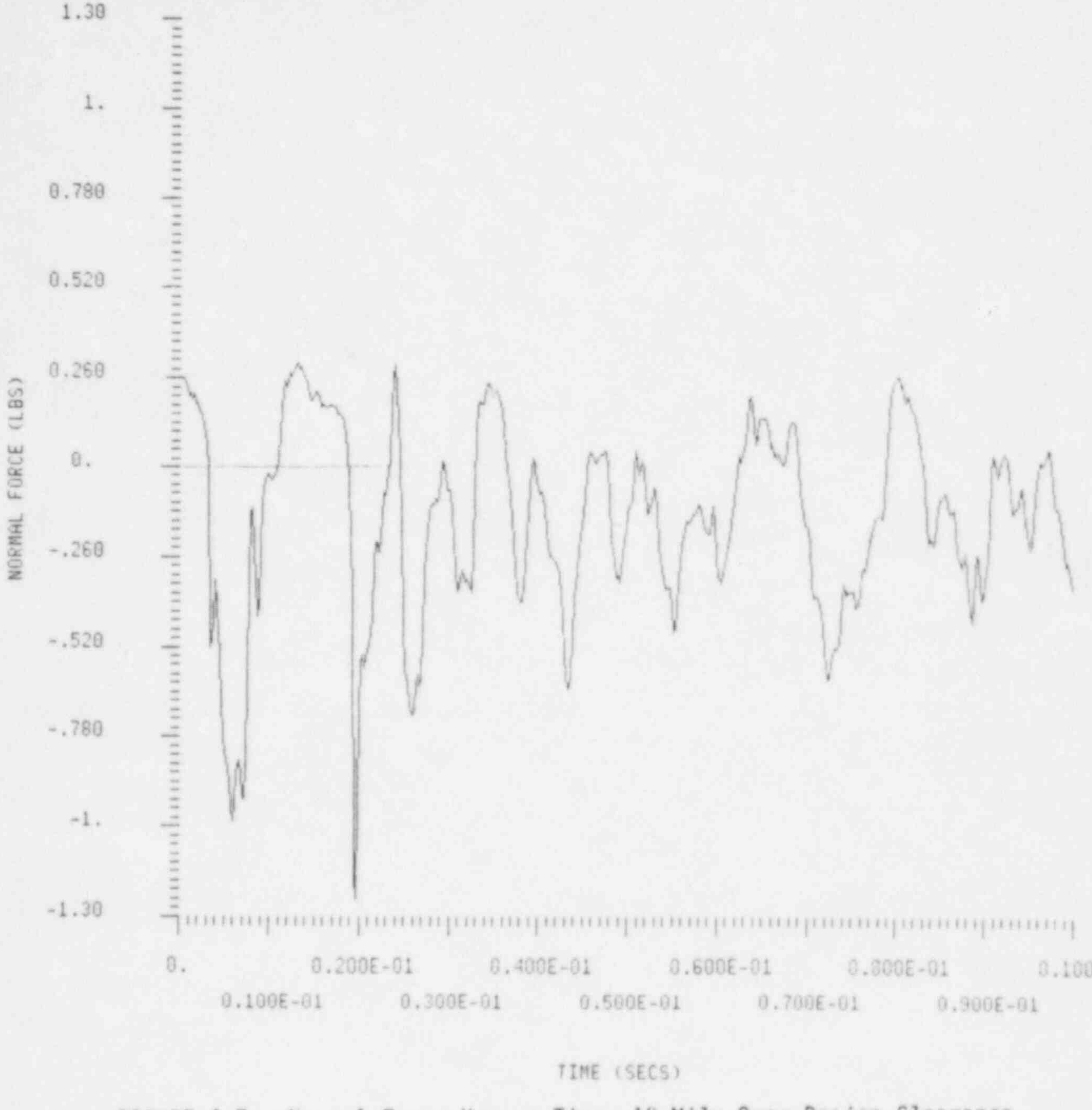

FIGURE 4.7. Normal Force Versus Time, 10 Mils Over Design Clearance

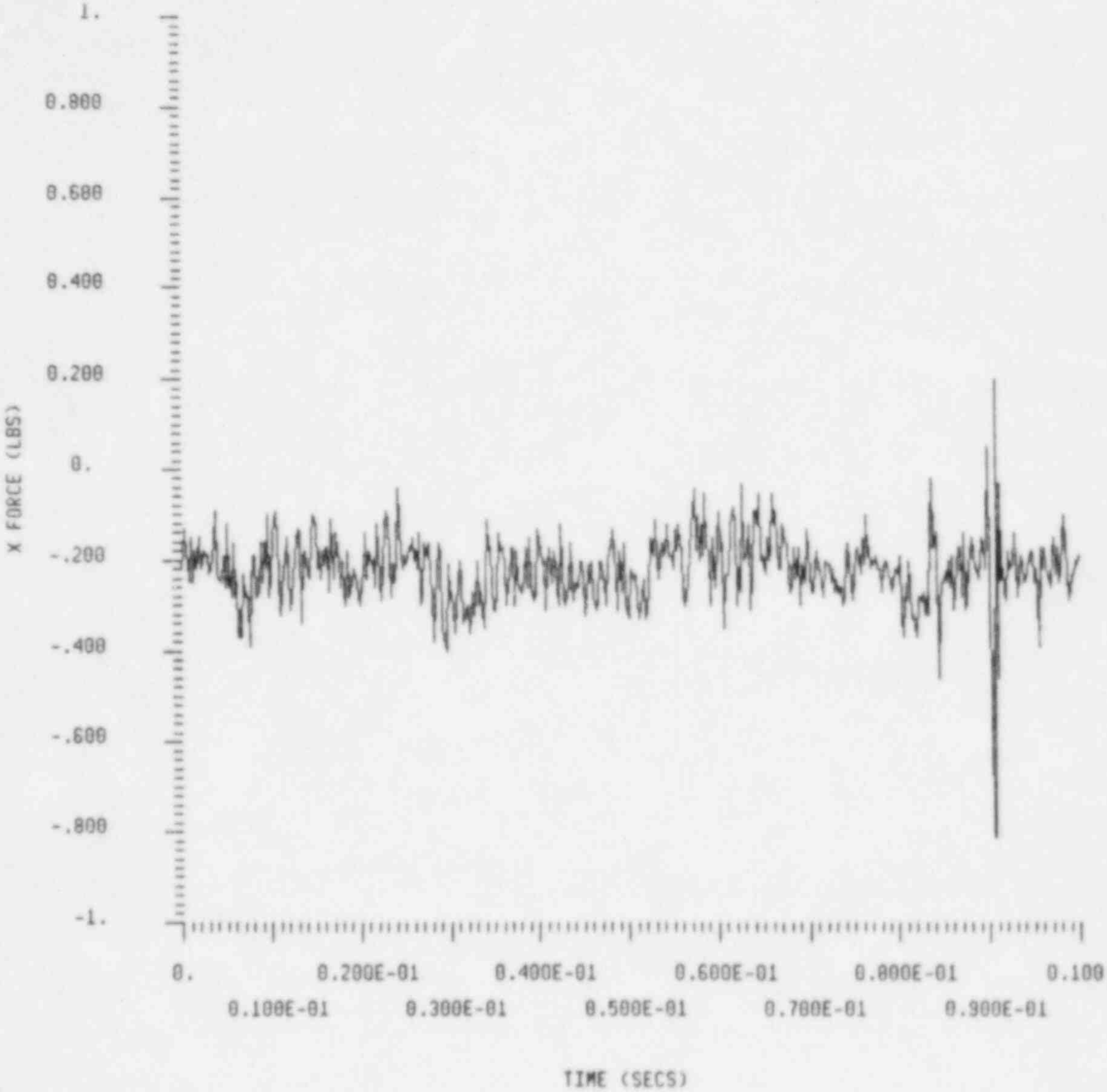

FIGURE 4.8. X-Force Versus Time - No Tube

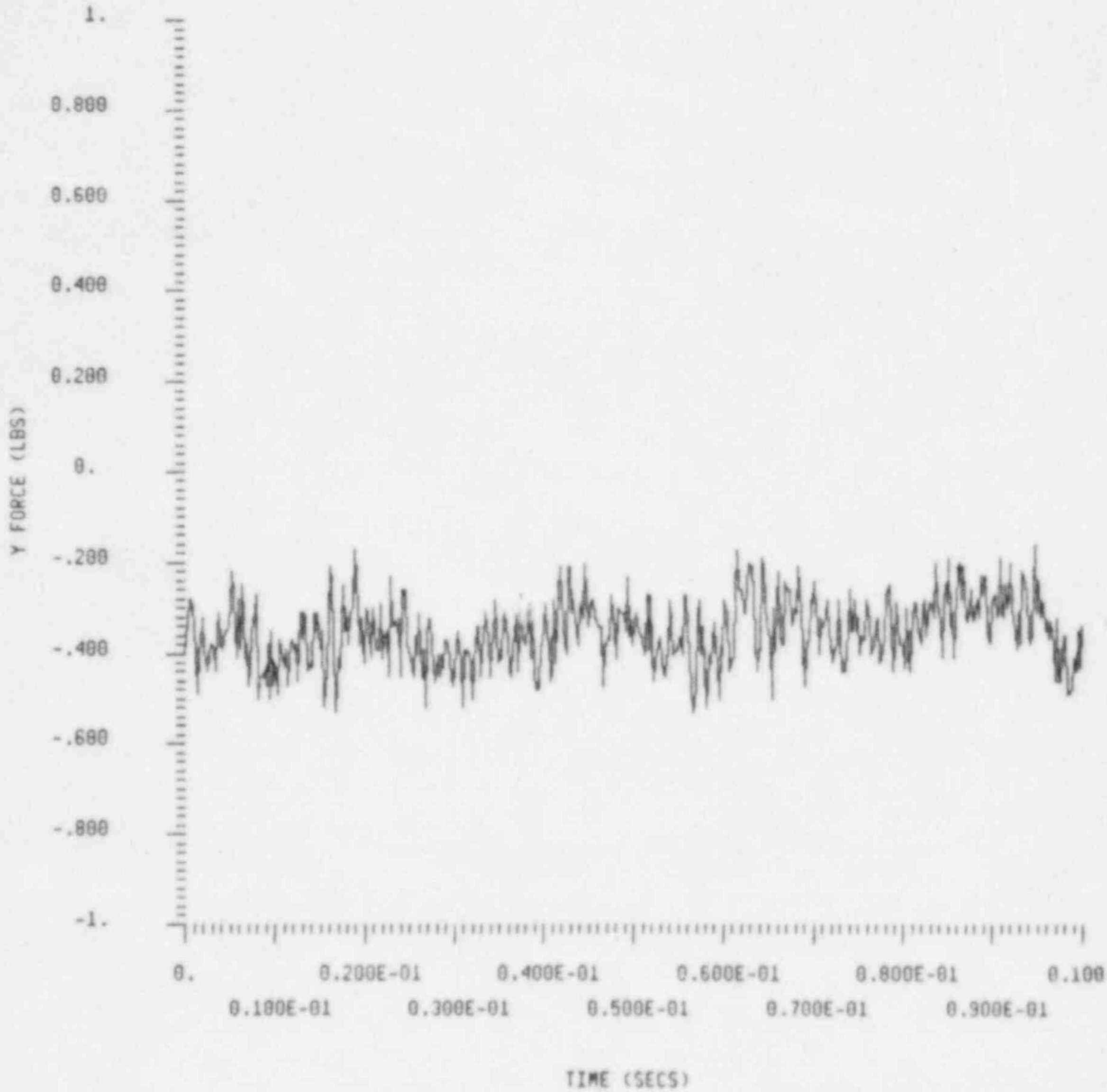

FIGURE 4.9. Y-Force Versus Time - No Tube

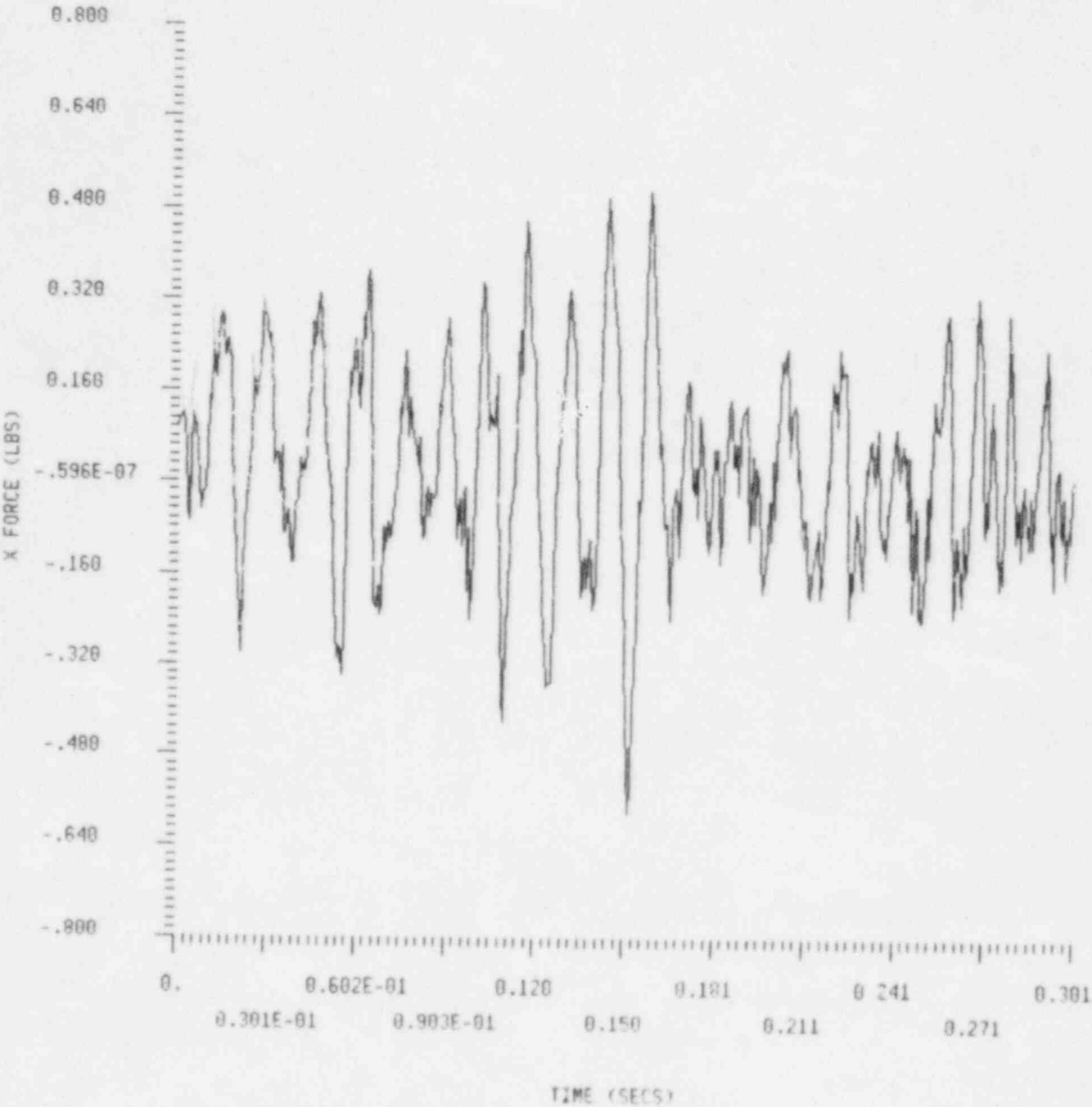

FIGURE 4.10. X-Force Versus Time, 10 Mils Over Design Clearance, 100% Reactor Power

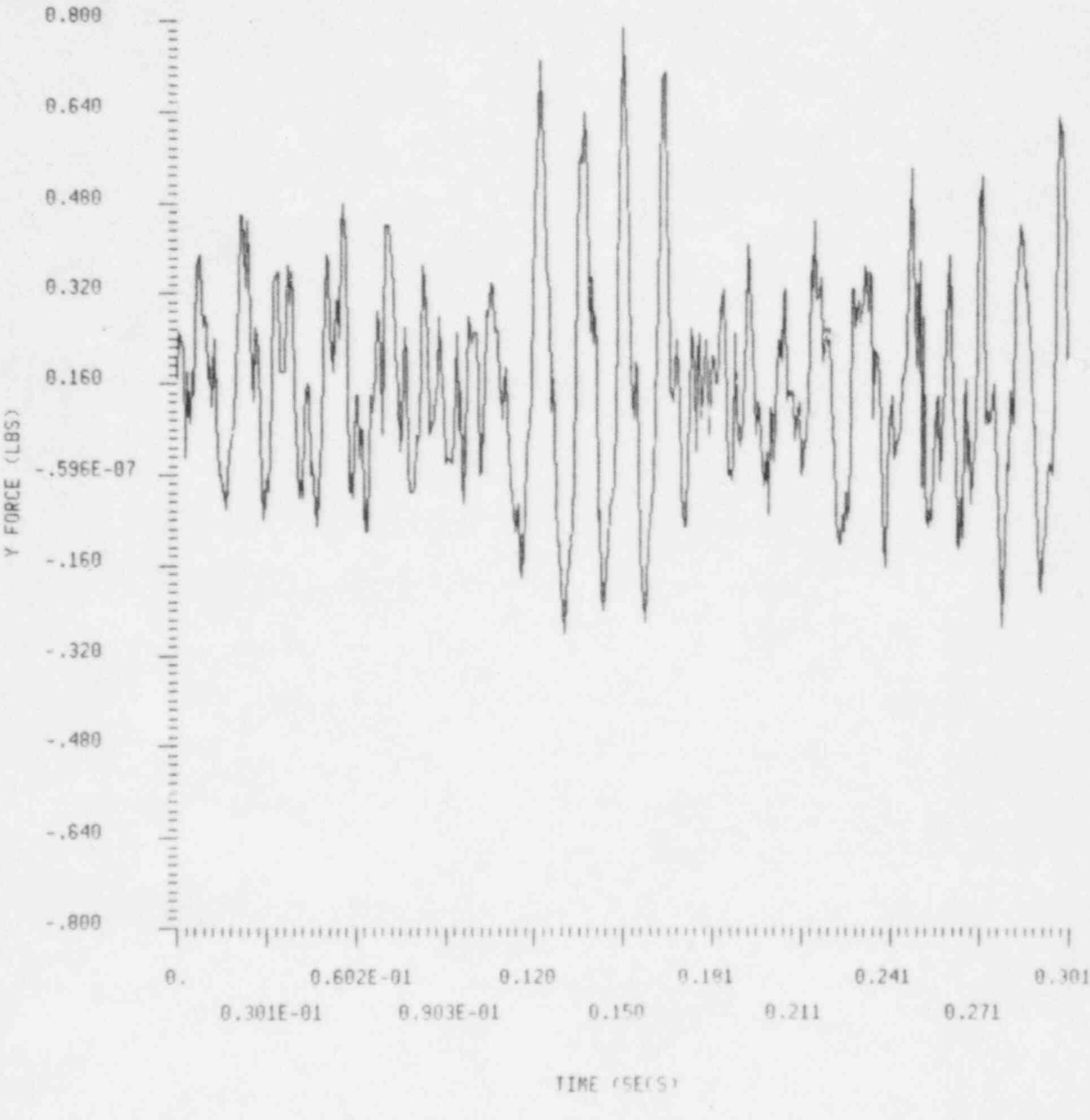

FIGURE 4.11. Y-Force Versus Time, 10 Mils Over Design Clearance, 100% Reactor Power

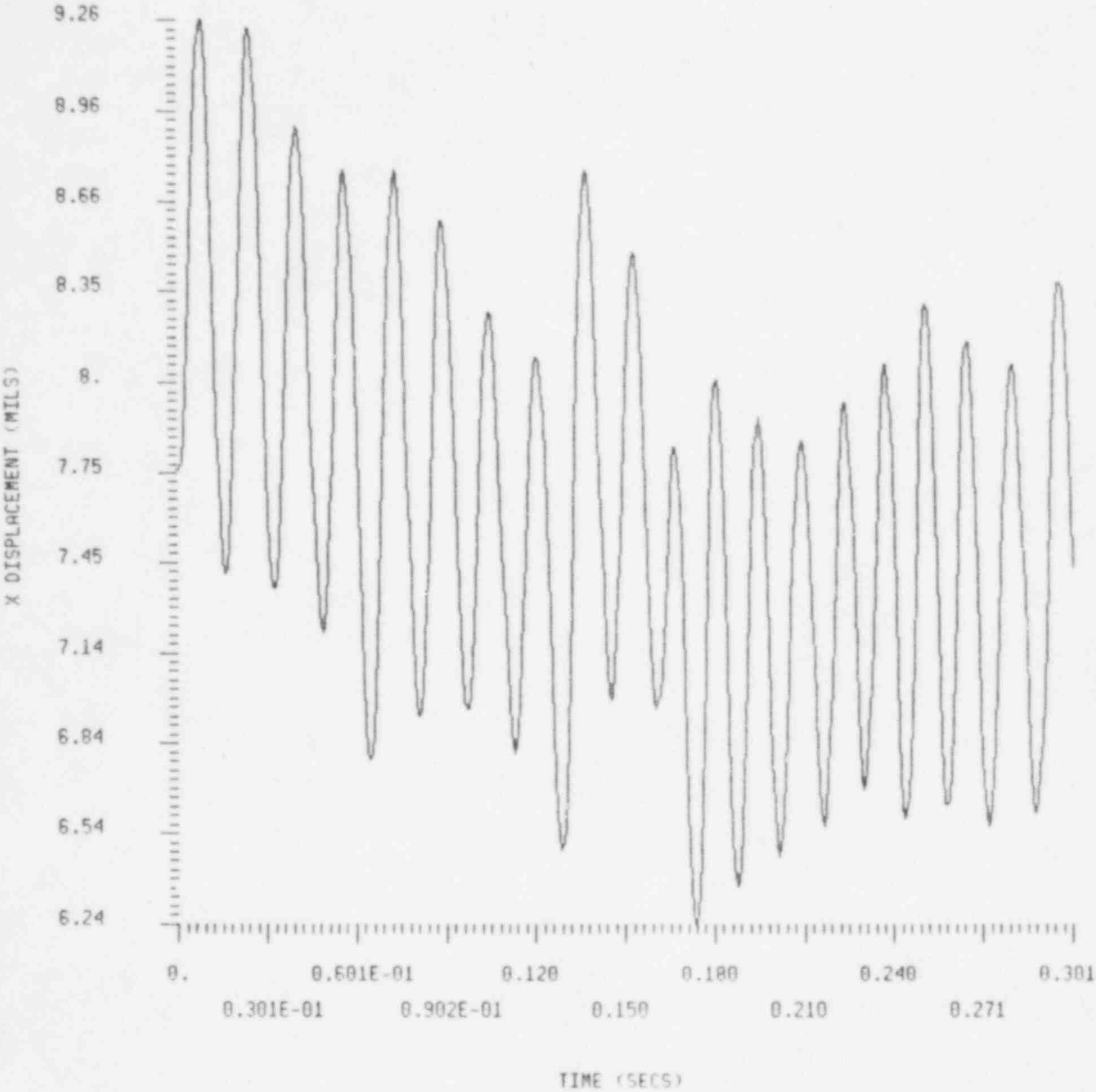

FIGURE 4.12. X-Displacement Versus Time, 10 Mils Over Design Clearance, 100% Reactor Power

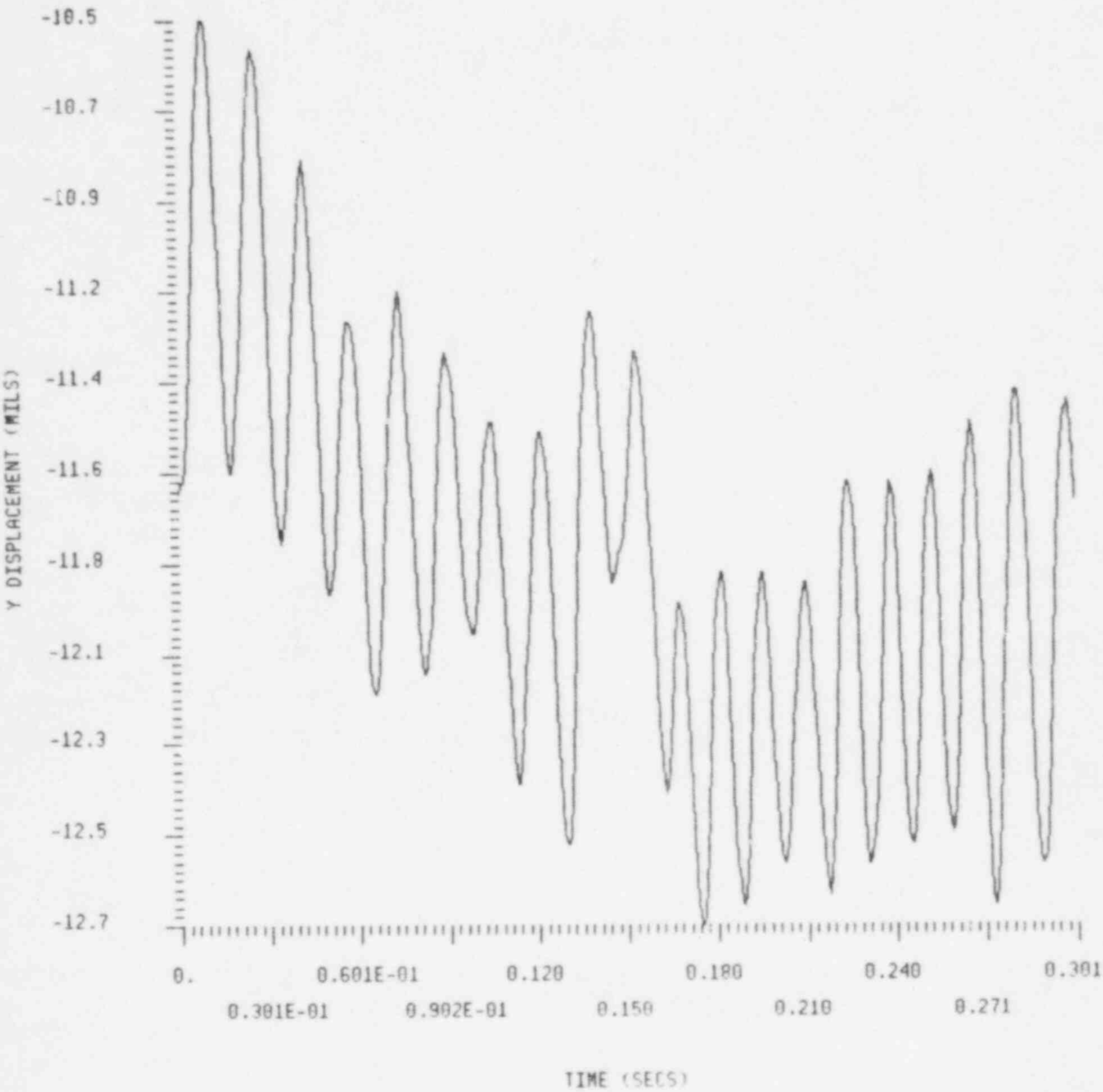

FIGURE 4.13. Y-Displacement Versus Time, 10 Mils Over Design Clearance, 100% Reactor Power

5.0 CONCLUSIONS

From the flow test results, PNL researchers concluded that a tube clearance of 10 mils or greater in excess of design clearance would not tend to increase tube wear rate at the instrumented tube support location. The tube support plate was shown to be considerably less active at this clearance than at the design clearance when fluid flows at 400°F ranged from 50% to 150% of the flow required for prototypic Reynolds number. Tube motion was elliptic under these conditions, and vibration amplitudes were greater than for the design clearance case, suggesting that the tube in this case was restrained by its stiffness rather than by the support plate. It was evident that frequent tube contact did occur at design clearance conditions. Based on these results, a forcing function for accelerated wear tests cannot be defined. It was further concluded that the data does not justify further testing at tube clearance conditions in excess of 10 mils over design clearance, or performing accelerated wear tests as previously planned (Enderlin and Baugh 1985).

In addition, the results of chemical cleaning tests performed by other researchers suggest that the post-cleaning tube/tube-support plate clearance is expected to remain at no less than 10 mils over design clearance, and will likely be even larger. Hence, PNL concluded that under normal operating conditions there is little potential for increased tube wear rate as a result of chemical cleaning at tube support locations within a steam generator where conditions are similar to those at the instrumented tube support location. Consequently, based on the results of the investigation, PNL cannot recommend a maximum allowable tube/tube-support clearance criterion for chemical cleaning.

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Chemical cleaning has been proposed to remove magnetite buildup in some pressurized water reactor steam generators. The U.S. Nuclear Regulatory Commission (NRC) has expressed concern that such cleaning would combine with the tube denting caused by magnetite formation to enlarge tube/tube-support plate clearances, increasing the level of flow-induced vibrations that could lead to unacceptably high tube wear and failure rates. In support of NRC, the Pacific Northwest Laboratory investigated whether such increased clearances would exacerbate tube fretting wear. Using a full-length scale model of a steam generator tube bundle, flow tests were conducted at an instrumented location through clearances representing as-built and post-cleaned tube conditions. Test results indicated little potential for increased tube wear as a result of chemical cleaning, under normal operating conditions at tube support locations similar to that tested.

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VIBRATION AND WEAR IN STEAM GENERATOR TUBES FOLLOWING CHEMICAL CLEANING MARCH 1986