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DYNAMICS OF TUBES IN FLUID
WITH TUBE-BAFFLE INTERACTION

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ABSTRACT

Three series of tests are performed to evaluate the effects of tube to tube-support-plate (TSP) clearance on tube dynamic characteristics and instability phenomena for tube arrays in crossflow. Test results show that, for relatively large clearances, tubes may possess "TSP-inactive modes" in which the tubes rattle inside some of the tube-support-plate holes, and that the natural frequencies of TSP-inactive modes are lower than those of "TSP-active modes," in which the support plates provide "knife-edge" type support. Tube response characteristics associated with TSP-inactive modes are sensitive to tube-to-TSP clearance, TSP thickness, excitation amplitude, tube alignment, and the fluid inside the clearance. In addition, tube response is intrinsically nonlinear, with the dominance of TSP-inactive or TSP-active modes depending on the magnitudes of different system parameters. In general, such a system is difficult to model; only a full-scale test can provide all the necessary characteristics. A tube array supported by TSPs with relatively large clearances may be subjected to dynamic instability in some of the TSP-inactive modes; tube response characteristics and impact forces on TSPs for a tube row are studied in detail in this report. Tube displacements associated with the instability of a TSP-inactive mode are small; however, impacts of the tube against TSPs may result in significant damage in a relatively short time. More detailed studies are needed to characterize the damage associated with the TSP-inactive modes resulting from dynamic instability and fluid-force excitations.

I. INTRODUCTION

Heat exchanger tubes are frequently supported by tube support plates (TSP) at intervals along their length. To facilitate assembly and relative motion caused by thermal expansion, tube-support-plate (TSP) holes are made larger than tube diameters. It is not uncommon that the tube holes are drilled 0.4 mm to 0.8 mm over the outside diameter of the tubes. In special

situations, tube holes may be drilled smaller than 0.4 mm or larger than 0.8 mm.

Current design practices consider the heat exchanger tubes to be simply supported at the tube support plates without clearances. For small clearances, this assumption is expected to be applicable; the tube will respond as a continuous beam supported by all TSPs and anti-nodes of all modes do not appear at the TSPs. This type of mode is called "TSP-active." Several studies show that this assumption is indeed applicable, although there are some variations of resonant frequencies [1-4]. With this assumption, the simple beam theory can be used to predict the tube natural frequencies as well as tube responses to different excitations [5,6].

When the clearance is relatively large, the tube will rattle inside some of the TSP holes for small-amplitude oscillations. This type of mode, in which some TSPs do not provide effective supports, is called "TSP-inactive."

For intermediate clearances, the tube response will depend strongly on the excitation amplitude. For low excitations, the tube will vibrate within the gap (TSP-inactive mode). For large excitations, the tube will contact all TSPs for most of the time (TSP-active mode). In reality, the tube response will be composed of both TSP-active and TSP-inactive modes. The dynamic responses for both modes have been reported [7-10].

Tubes exposed to flow will be subjected to different types of excitation. The role of clearance between the tube and tube support plates depends on different types of fluid excitation forces and flow velocity. Only very limited studies have been reported to study the effect of the clearance on tube response subjected to crossflow. Gorman [11] has concluded that a clearance of ~ 0.46 mm has the effect of significantly increasing the approach flow velocity required to initiate instability, while Weaver and Schneider [12] have found that with a clearance of 0.51 mm, the tube may become unstable in the mode in which some tube support plates do not provide effective support, but is suppressed when tube-support impacting occurs. The role of the clearance on fluidelastic instability is not well understood.

This report presents three series of tests to evaluate the effects of tube to tube-support-plate clearance:

- A single tube on multiple supports.
- Scaling effect of tube-support interaction.
- A tube row in crossflow.

Results of this study show that TSP-inactive modes can be very important for large clearances and cannot be ignored. Therefore the current design practice to consider only TSP-active modes is not always reliable.

II. DYNAMIC CHARACTERISTICS OF A TUBE ON MULTIPLE SUPPORTS

A. Experimental Apparatus and Measurement Technique

The test model features a 19.05 mm (0.75 in.) outer diameter x 1.59 mm (62.5 mil) thick x 711.2 cm (280.0 in.) long carbon steel tube; two sets of 10.16 cm (4 in.) thick brass tubesheets; and a series of nine 19.05 mm (0.75 in.) thick cold rolled steel plates. The locations of the baffle plates TSPs are shown in Fig. 1. When centered, the diametral clearance between the tube and a baffle plate hole is 0.584 mm (23 mils).

The test model is mounted on W 12 x 79 steel beam (304.30 mm deep x 304.30 mm wide x 19.05 mm flange thick x 12.70 web thick) with a 254 mm wide x 228.6 mm deep aluminum cage to enclose the test model and contain water. The test rig is mounted horizontally and is sufficiently rigid that significant interaction between tube and test rig caused by mechanical or fluid coupling is avoided (see Fig. 2). The baffle plates can be adjusted laterally to obtain various initial clearances.

A means to excite the tube in the horizontal direction is provided by a noncontacting electromagnetic exciter assembly consisting of a pair of coils mounted about 23 cm from TSP 3 between TSPs 3 and 4. The response of the tube is measured by a biaxial accelerometer located inside the tube at the midspan of TSPs 4 and 5. In addition, two displacement transducers are located very close to TSP 3 to measure the tube displacements in the vertical and horizontal direction at that location.

B. Test Procedures and Experimental Data

The tube is excited by a sinusoidal force with a constant amplitude. The frequency of excitation force is swept through the frequency range of interest. Tube acceleration signals are recorded as a function of excitation frequency and are used to determine tube natural frequency and damping.

Under steady-state excitation, the current to the coil is set at many different values to investigate the effect of excitation amplitude on natural frequencies, mode shapes, and damping. Figure 3 shows the steady-state response of tube acceleration in the horizontal direction as a function of frequency for a series of excitation forces for the tube vibrating in air.

Based on the peaks of the response curves, the natural frequencies of the tube can be identified approximately; these frequencies are given in Table 1 for different excitation levels.

Fig. 2. Test model of a continuous tube. ANL Neg. No. 113-82-61 #5.

Fig. 3. Tube acceleration as a function of excitation frequency for various excitation amplitudes in air.

Table 1. Natural Frequencies (Hz) for a Tube on Multiple Supports in Air

Method	Mode				
	TSP-inactive	TSP-active			
Analysis	70.3 ⁺	167.0	171.3	186.5	
Experiment					
0.018 Newtons	69 ⁺	168	173	180	191
0.036 Newtons	70 ⁺	166	169	178	
0.285 Newtons	72 ⁺	165	168	178	187
1.09 Newtons	38 [*]	164	168		
1.90 Newtons	46 [*]		166		

TSPs 3 and 4 inactive

TSP 4 inactive

Once the natural frequencies are known, mode shapes can be determined using the following procedure:

- A sinusoidal current with its frequency equal to one of the natural frequencies is applied to the coils; thus, the tubes are set in resonance.
- The signal from the internal accelerometer is displayed on an oscilloscope.
- An accelerometer, whose signal is also displayed on the oscilloscope, is hand-held at different positions of the tube.
- The phase difference between those signals can be seen from the oscilloscope; the phase is either 0° or 180° when the tube is in resonance.
- By moving the accelerometer to various places, the approximate mode shape can be determined from the phase.

Figure 4 shows the experimentally determined modes for different frequencies. These modes are clearly identifiable. Note that, in the first mode (38 Hz), both TSPs 3 and 4 are not active, and in the second mode (69 Hz), the fourth support plate is inactive. These modes are TSP-inactive modes. In other modes, all TSPs correspond to the nodal point of the modes; these modes are TSP-active modes.

Calculations also have been made for the tube corresponding to two different support conditions:

- Fixed at tube sheets and hinged at all other supports.
- Fixed at tube sheets, no support at TSP 4 and hinged at all the other supports.

In Table 1, the frequencies and mode shapes from the analytical results are compared with the dominant response modes obtained experimentally. The frequencies agree reasonably well with the experimental data; the mode shapes also are in agreement with the test results.

The frequencies given in Table 1 depend on the excitation levels. In the excitation range in this experiment, the dominant frequencies for TSP-active modes decrease with increasing excitation amplitude, while the natural frequencies of TSP-inactive modes increase with increasing excitation amplitude.

Tube responses at low frequencies are of particular interest. These modes are associated with the TSP-inactive modes. The response characteristics depend on excitation levels. Figure 5 shows the response curves in air for different excitation amplitudes. At low excitation levels (see Fig. 5a), the dominant frequency is at about 69 Hz; this is associated with the TSP-inactive mode, in which TSP 4 is inactive. At higher excitation levels, such as 1.089 Newtons (see Fig. 5b), the dominant frequency is shifted to 40 Hz, which is the TSP-inactive mode in which TSPs 3 and 4 are

Fig. 4. Experimentally determined mode shapes in air.

Fig. 5. Tube response curves at low frequencies.

inactive. At even higher excitation levels, the dominant peak frequencies are shifted again. At very large excitation levels (see Fig. 5d), the peaks do not correspond to any of the TSP-inactive modes.

Figure 6 shows the tube displacement components at TSP 3 in the horizontal and vertical directions and Figure 7 shows the dominant frequency. At very small excitation levels, the dominant mode is the TSP-inactive mode of ~ 69 Hz. When the excitation is larger than 1 Newton, the dominant frequency is shifted to the TSP-inactive mode of ~ 40 Hz. There also is a drastic increase of tube displacement at TSP 3 when the excitation is ~ 1 Newton. At excitation levels larger than 1 Newton, the tube impacts against TSP 3; the double peak displacement is basically the clearance of the diametral gap between the tube and tube support plate. It is noted that the displacement component in the horizontal direction is about 1 order of magnitude larger than that in the vertical direction. This is because the excitation is applied only in the horizontal direction.

Figure 8 shows the frequency response of tube acceleration in water for different excitation levels. The general characteristics in water are similar to those in air. Two effects are noted:

- Because of the added mass, the response frequencies in water are lower than the corresponding values in air.
- Because of the fluid damping attributed to the fluid between the tube and TSP clearance, the damping in water is larger than that in air.

The effect of damping for TSP-inactive modes is more significant. As can be seen from Fig. 8, the tube response peaks in water are not as sharp as those in air for TSP-inactive modes.

In all previous tests, all TSPs are aligned with the tube such that there is practically no contact force between the tube and TSPs except the dead weight of the tube; this case is called "no preload." Tests in water are also performed for misalignment by moving TSPs 3 and 4 in the same horizontal direction such that they are exerting the same magnitude of force on the tube. Two force levels are employed:

- A static force of 3.20 Newtons, called light preload.
- A static force of 8.36 Newtons, called heavy preload.

The tube is subjected to random excitation for various force levels (four pairs of coils are used). Figure 9 shows the frequency spectra of tube acceleration for light preload and heavy preload and Figure 10 shows the horizontal rms tube displacement for three different conditions. These results show that:

- Misalignment is important for tube response; the tube responds differently to different amounts of misalignment.

Fig. 6. RMS tube displacement components at TSP 3 as a function of excitation force.

Fig. 7. Dominant response frequency as a function of excitation force.

Fig. 8. Tube acceleration as a function of excitation frequency for various excitation amplitudes in water.

Fig. 9. Frequency spectra of tube acceleration in water for light preload and heavy preload.

Fig. 10. Tube rms displacement component in the horizontal direction for three different support conditions.

- At low excitation levels, the tube with large preload has a smaller displacement; however, at larger excitation levels, the tube will "lift off" from TSPs 3 and 4 and the misalignment is no longer so important.
- The force-displacement curves given in Fig. 10 illustrate the nonlinear behavior of the tube; for a tube with preload, the slope of the force-displacement changes when the tube starts to lift off from TSPs 3 and 4.

C. Concluding Remarks

The results of this test have illustrated several important tube characteristics:

- Tubes supported by tube support plates with diametral clearances can be excited in resonance in TSP-inactive modes, in which some TSPs do not provide effective support for the tube.
- The natural frequency of a TSP-inactive mode can be significantly lower than those of TSP-active modes, in which TSPs do provide support for the tube.
- The dominant TSP-inactive mode for a tube subject to an excitation depend on the amplitude and frequencies of the excitation as well as other parameters.
- The damping associated with the fluid in the clearance between the tube and TSPs is much more significant for TSP-inactive modes.
- Misalignment plays an important role for tube response. For small excitation levels, misalignment reduces tube response amplitudes.
- Tubes supported by TSPs with clearance are highly nonlinear.
- Natural frequencies of TSP-inactive modes increase with excitation amplitude while those of TSP-active modes decrease with excitation amplitude.

In this test, the diametral clearance is 0.584 mm (23 mils). The existence of TSP-inactive modes is firmly established. Therefore, the common practice of considering multiple span tubes with "knife-edge" supports is not acceptable for relatively large clearances.

III. SCALING EFFECT OF TUBE/SUPPORT INTERACTION

A schematic of the tests is shown in Fig. 11. Two tubes are tested: (1) a tube with 1.595 cm OD and 2.74 mm wall thickness that is prototypic of an LMFBR steam generator tube and (2) an approximate 0.4 scale model of the prototype tube with 0.643 cm OD and 0.80 mm wall thickness. The geometries

PROTOTYPE

CROLOY TUBE

1.595 cm (0.628") OD

2.74 mm (0.108") WALL THICKNESS

SCALE MODEL

CARBON STEEL TUBE

0.643 cm (0.253") OD

0.80 mm (0.0313") WALL THICKNESS

Fig. 11. Tube/support configuration of the tests for scaling effect.

of TSPs 1 to 5 are shown in Fig. 11; the thickness of the prototype support plates is 3.19 cm. The lengths given in Fig. 11 denote the dimensions for the prototype; the lengths of the scaled model are 0.4 scale of the prototype. Note that the tubes and tube support plates in the two tests are assembled in the same test channel as that discussed in Section II.

The tubes are excited at midspan between TSPs 1 and 2 with an electromagnetic exciter assembly. The tube displacement is measured with a displacement transducer located at the midspan between TSPs 2 and 3. Two types of excitation are employed: broad-band random excitation; and steady-state swept-sine excitation. Tests are conducted in water only.

In the scale model, after the completion of the test for the geometrically scaled TSP clearance of 0.28 mm at the land portion, the clearance is then increased to 0.43 mm, while all other parameters are kept constant. This test is to demonstrate the effect of different clearance. Note that the clearance of the scale model of 0.28 mm corresponds to 0.69 mm in the prototype. The over-sized hole of the scale model with a clearance of 0.43 mm corresponds to the prototype of 1.08 mm.

Figure 12 shows the steady-state tube response amplitude for the two tubes for different excitation amplitudes. In these tests, the displacement is given in rms values and in the same scale. The numbers given in the figure, such as 3A, denote the excitation current in the coils. Figure 13 shows the tube response of the prototype to random excitation for different excitation amplitudes.

Figures 12 and 13 show that there are two groups of frequencies:

- At low excitation amplitudes, the responses are dominated by a group of low frequencies. These frequencies correspond to the case in which some of the TSPs do not contact with the tube; i.e., some TSPs are inactive. In these modes, the tube vibrates within the clearance of the support.
- At high excitation amplitudes, the responses are dominated by a group of high frequencies. These frequencies correspond to the case in which all plates are active; i.e., the plates provide support for the tube.

In the intermediate ranges of excitation amplitudes, both groups of frequencies will exist. The system is nonlinear.

Although the response curves of the prototype and scale model are not identical, the general response characteristics are similar: both low and high frequencies exist; and the frequency ratios of the prototype and scale model for both groups of frequencies follows the scaling law.

Fig. 12. Steady-state response curves of the prototype and the scale model in water at different excitation amplitudes.

Fig. 13. Tube response frequency spectra to random excitation for the prototype.

Figure 14 shows the response of the scale model with oversized holes to swept-sine excitation. Comparison of Figs. 12b and 14 shows that the dynamic characteristics are not the same.

The mode shapes of the scale model with the two different sizes of TSP clearance for the low frequencies also are investigated; the results are given in Fig. 15. For the geometrically scaled clearance (0.28 mm), TSP 2 is inactive. For the oversized clearance (0.43 mm), TSPs 1, 2, and 3 are inactive. Based on these tests, it is clear that the tube with geometrically scaled clearance simulates the prototype better than the oversized clearance.

The damping values of one of the TSP-inactive modes also were measured using the bandwidth method; the results are given in Table 2. Because of the clearance between the TSP and tube, the damping increases with excitation amplitude. It should be pointed out that the system is nonlinear and the damping obtained by the bandwidth method is not very accurate because of the impact at the TSP.

Based on this study, the geometrically scaled clearance simulates the prototype better. However, the responses of the prototype and the scale model cannot be correlated directly for some excitation amplitudes because of the nonlinear behavior. Therefore, for tubes supported by multiple TSPs with relatively large clearances, tests should be conducted on the prototype tube and TSPs. Great care should be exercised to apply the data obtained from the scale model to the prototype.

IV. TUBE ROWS IN CROSSFLOW

A. Test Section

The experiment is performed in the rectangular flow channel, 12.0 cm (4.75 in.) wide and 25.9 cm (10.2 in.) high, of a test chamber connected to a water loop with a maximum flow rate of $0.052 \text{ m}^3/\text{s}$ (700 gpm). Each tube element is suspended as a simply supported beam on two O-rings mounted 91.4 cm (36 in.) apart (see Fig. 16). The O-rings are seated in compression plates. The tube is submerged in fluid between the two O-ring supports (C1 to C2) but subjected to flow at the middle portion only.

Two types of tubes are used:

- Tubes 1, 2, 4, and 5 are stainless steel tubes with 1.59 cm (5/8 in.) OD, 0.32 cm (1/8 in.) wall thickness, and 94.6 cm (37-1/2 in.) length.
- Tube 3 is a brass tube with 1.59 cm (5/8 in.) OD, 1.59 mm (1/16 in.) wall thickness, and 131.8 cm (51-7/8 in.) length.

Fig. 14. Tube response curves of the scale model with over-sized holes.

(a) GEOMETRICALLY SCALED CLEARANCE

(b) OVERSIZED CLEARANCE

Fig. 15. Mode shapes of the 0.4 scale model of low-frequency modes, in which some of support plates are inactive.

Table 2. Damping of the Prototype Tube for One of the
TSP-Inactive Modes

Excitation Current (amperes)	Damping (%)
1	1
2	2.2
3	3.4
4	4.3
5	4.5
6	8.5
7	12.5

Fig. 16. Schematic of a tube row.

The overhung portion for tubes 1, 2, 4, and 5 is 1.6 cm, and for tube 3 is 1.6 cm at one end and 38.8 cm at the other end. The overhung portion of the tubes between C2 and C3 may be in air or in quiescent water. To raise their natural frequencies in the lift direction, Tubes 1-2 and 4-5 are welded together outside the flow region.

Tube 3 also is "supported" by a brass baffle plate (TSP); its thickness is 1.27 cm (1/2 in.). The diametral clearance (gap) between the tube and the TSP is set at different values to study the effect of clearance on tube response: 0, 0.25 (10 mil), 0.38 (15 mil), 0.50 (20 mil), 0.64 (25 mil), 1.02 (40 mil), 1.27 (50 mil), 2.54 (100 mil) and ∞ mm. The position of the TSP can be adjusted to simulate the misalignment of TSP with respect to Tube 3. Figure 17 shows the overhung end assembled in the TSP with a diametral clearance of 1.02 mm (40 mil).

The TSP at the overhung end also can be replaced by a set of four impact transducers, as shown in Fig. 18. The impact transducer stiffness itself is 1.75×10^{10} N/m (10^8 lb/in.). The four transducers can be adjusted to give proper clearance between the tube and the transducers.

Two displacement transducers, A and B (see Fig. 16), are located at two places to measure the displacement of Tube 3; Transducer A is close to the TSP (see Fig. 19) and Transducer B is about at the mid-span of the overhung portion. Because the dominant displacement is in the lift direction, both transducers are measuring the lift-direction displacement.

B. Test Procedures

The objectives are to measure the response characteristics of Tube 3 for different support conditions. The stiffnesses of Tubes 1, 2, 4, and 5 are much larger than that of Tube 3; therefore, Tubes 1, 2, 4, and 5 can be considered as rigid tubes. The four series of tests are summarized in Table 3.

1. Tube Supported by Baffle Plate

The test section (see Fig. 16) is installed in the water loop vertically with the overhung portion pointing upward. The span between the two O-rings is submerged in water. The overhung portion, C2 to C3, can be in air or in water. The following tests are carried out for the two different conditions in order to understand the effect of water.

Test A1: The span between C2 and C3 is in air; therefore, there is no water between the tube and TSP at C3.

Test A2: The span between C2 and C3 is submerged in quiescent water; therefore, water is in the clearance between the tube and TSP at C3.

Fig. 17. Baffle plate at the overhung end with a diametral clearance of 1.02 mm. ANL Neg. No. 113-83-39 #3.

Fig. 18. Impact force transducers installed at the overhung end.
ANL Neg. No. 113-83-38 #6.

Fig. 19. Displacement transducer close to the support C3. ANL Neg. No. 113-83-40 #6.

Table 3. Four Series of Tests in Crossflow

Test	Support Condition at C3	Fluid for Span between C2 and C3	Diametral Clearance (mm)		
A1	Baffle plate (TSP)	Air	0	0.51	1.27
			0.25	0.64	2.54
			0.38	1.02	∞
A2	Baffle plate (TSP)	Water	0.25	0.64	
			0.38	1.02	
			0.51	1.27	
B1	Impact force transducers	Air	0.3	0.64	2.54
			0.38	0.76	
			0.51	1.27	
B2	Impact force transducers	Water	0.64		
			1.27		

a. Tube Characteristics in Air and Quiescent Water

Testing in air and in quiescent water is performed to determine the natural frequency and damping of Tube 3. The tube is excited by impact at the overhung end.

b. Tube Responses in Flowing Fluid with Tube 3 Centered at the Baffle Plate Support

For a specific TSP clearance, the TSP is adjusted to be in alignment with Tube 3. Then the flow velocity is increased at small intervals. At each flow velocity, the tube displacement signals are recorded for several minutes for subsequent analysis. Because of the tube displacement caused by fluid drag force, at each different flow velocity, the TSP has to be adjusted to realign with the clearance. In these tests, the tube is centered in the TSP hole for all flow velocities. The test is terminated when the instability occurs that is associated with the mode in which the TSP is active. This type of test is performed for all sizes of clearances.

c. Tube Responses in Flowing Fluid for Different Misalignment with 0.64 mm Diametral Clearance

Four tests are performed for Tube 3 supported by the TSP with 0.64 mm diametral clearance:

- Tube 3 slightly touches one side of the TSP in the lift direction (light preload).
- Tube 3 strongly touches one side of the TSP in the lift direction (heavy preload).
- Tube 3 is aligned with the TSP at zero flow velocity.
- Tube 3 is aligned with the TSP at the flow velocity equal to 2.4 m/s.

In these tests, once the TSP is set in a particular position, no adjustment is made at different flow velocities. Tube displacements are measured for different flow velocities. The tests are terminated when the instability associated with the TSP-active mode occurs.

2. Tube Supported by Impact Force Transducers

After completion of the tests for the baffle-plate support, four impact force transducers were installed to replace the TSP at C3 (see Fig. 16). The diametral clearance between the tube and the flat surfaces of the two opposite force transducers can be adjusted to any selected value. Because the tube motion is predominantly in the lift direction, the clearance between the tube and force transducers in the drag direction is adjusted to a value such that no impact will occur. Two series of tests are made:

- Test B1: The Span between C2 and C3 is in air. The diametral clearance between the tube and force transducers in the lift

direction is tested for the following values: 0.3 mm (12 mil), 0.38 mm (15 mil), 0.51 mm (20 mil), 0.64 mm (25 mil), 0.76 mm (30 mil), 1.27 mm (50 mil), and 2.54 mm (100 mil).

- Test B2: The span between C2 and C3 is in water. The diametral clearance is tested for 0.64 mm (25 mil) and 1.27 mm (50 mil).

The test procedure is basically the same as that in Tests A1 and A2. Once the instability associated with the TSP-active mode occurs, testing is stopped to avoid damage to the tube and force transducers.

C. Test Results

Because of the clearance at support C3, the tube can respond in different modes: TSP-inactive mode - support at C3 is inactive; and TSP-active mode - support at C3 is active. The two modes are shown in Fig. 20. The natural frequencies and modal damping ratios for these two modes under different test conditions are given in Table 4. Note that the frequency of the TSP-active mode is much higher than that of the TSP-inactive mode. In addition, the modal damping ratios for the TSP-inactive mode depend on the diametral clearance.

1. Test A1

Figure 21 shows the rms tube displacements as a function of flow velocity for two extreme cases:

- ∞ diametral clearance: This corresponds to the case of no support at C3. The tube responds in the TSP-inactive mode.
- 0 diametral clearance: This corresponds to the case of a hinged support at C3. The tube responds in the TSP-active mode.

These two cases represent the two ideal conditions. For other diametral clearances, the response will consist of a combination of both TSP-inactive and TSP-active modes.

For ∞ diametral clearance, tube displacement increases drastically with flow velocity at 1.49 m/s (4.9 ft/sec); this is the critical flow velocity for the TSP-inactive mode. In this mode, tube displaced at Transducer A is larger than that at Transducer B; the displacement ratio of the two transducers is about 0.58. For zero diametral clearance, the rms tube displacement at Transducer A is small because it is very close to the support and the displacement at Transducer B is much larger. The ratio of rms displacements at the two transducers is about equal to 3.67 and the critical flow velocity is equal to 2.2 m/s.

TSP-INACTIVE MODE

TSP-ACTIVE MODE

Fig. 20. TSP-inactive and TSP-active modes.

Table 4. Natural Frequencies and Modal Damping Ratios
under Test Conditions

Test Conditions	TSP-inactive Mode		TSP-active Mode	
	Natural Frequency (Hz)	Modal Damping Ratio (%)	Natural Frequency (Hz)	Modal Damping Ratio (%)
In air (C1 to C3 in air)	26.5	2.0	45.4	1.5
C1 to C2 in water C2 to C3 in air	24.0	2.2	39.3	1.8
C1 to C3 in water	22.7	7.2 (0.38 mm) 4.6 (0.64 mm) 3.2 (1.02 mm) 2.7 (1.27 mm) 2.6 (2.54 mm) 2.4 (∞ mm)	38.9	2.0

Fig. 21. Tube displacements of TSP-inactive and TSP-active modes as a function of flow velocity.

Figure 22 shows the frequency spectra of the transducer B for the two cases as a function of flow velocity. For the TSP-inactive mode, its frequency increases insignificantly with the flow velocity and varies from 24 to 25 Hz. At the critical flow velocity, the oscillation frequency is 24 Hz. For the TSP-active mode, the dominant frequency varies from about 38 Hz to 41 Hz. At the critical flow velocity, the oscillation frequency is about 39 Hz. In both cases, there is only one frequency peak in the frequency spectra of the tube displacement.

Tube motion at the TSP is of particular interest because it is related to wear and fatigue at the support location. All data presented in the following text, except as noted, are taken from Transducer A.

Figures 23 to 25 show the rms tube displacements and Figs. 26 and 27 show typical frequency spectra as a function of flow velocity. Tube response is sensitive to the alignment of tube with respect to the TSP hole. In Fig. 23, the TSP is aligned with the tube at all flow velocities; i.e., the tube is always located at the center of the TSP hole. This is accomplished by adjusting the TSP to compensate for the drag-induced displacement. Tube response characteristics can be divided into three regions:

- Low-amplitude oscillations: For flow velocity less than about 1.5 m/s, tube motion is small and the tube responds in the TSP-inactive mode.
- Instability of the TSP-inactive mode: As the flow velocity increases to over 1.5 m/s, the TSP-inactive mode becomes unstable. Once the tube loses stability, large-amplitude oscillations occur and the tube impacts against the TSP. Therefore, the maximum peak-to-peak displacement at TSP C3 is that of the diametral clearance. In this range of flow velocity, tube displacement at C3 is almost independent of the flow velocity.
- Instability of the TSP-active mode: With further increase in flow velocity, the tube becomes unstable in the TSP-active mode. The dominant response frequency is that of the TSP-active mode.

Figure 24 shows the rms tube displacements for the following different conditions with the diametral clearance equal to 0.64 mm:

- Alignment at all flows: the tube is aligned with the TSP at all flow velocities.
- Light preload: the tube slightly touches against the TSP in the lift direction at zero flow.
- Heavy preload: the TSP is pushed against the tube in the lift direction at zero flow.

Fig. 22. Frequency spectra of TSP-inactive and TSP-active modes as a function of flow velocity.

Fig. 23. RMS tube displacements as a function of flow velocity for various diametral gaps (Test A1).

Fig. 24. RMS tube displacements as a function of flow velocity for different alignments with a diametral gap of 0.64 mm (Test A1).

Fig. 25. RMS tube displacements as a function of flow velocity for Test A1 with a diametral gap of 1.02 mm.

Fig. 26. Frequency spectra of tube displacement for the diametral gap equal to 1.27 mm (Test A1).

Fig. 27. Frequency spectra of tube displacement for the diametral gap equal to 0.64 mm (aligned at zero flow).

- Alignment at zero flow: the tube is aligned at zero flow velocity.
- Alignment at 2.44 m/s: the tube is aligned at 2.44 m/s.

Except for the first condition, in which adjustment has to be made at different flow velocities, no adjustment of the TSP is made at different flow velocities. Figure 24 shows that the tube response characteristics depend on the tube support conditions, in particular, in the region in which the TSP-inactive mode is unstable. In general, when the tube is in alignment with the TSP clearance, the tube displacement is the largest. The alignment of tubes in the heat exchanger depends on several factors: initial fitup of the tubes in the various support plates, including variations in the clearance between the tube and support; changes in mechanical fitup of the tube at operating conditions; and the steady drag and lift forces. The alignment is difficult to predict; therefore, the tube response associated with the instability of the TSP-inactive modes also is difficult to predict.

Figure 25 shows the tube displacements at Transducers A and B with the diametral clearance equal to 1.02 mm. In this case, the tube is in alignment with the clearance at all flow velocities. In the stable region, because tube displacement is small, the TSP is inactive and tube displacement at Transducer A is larger than that at B. In the unstable region of the TSP-inactive mode, the displacement at Transducer B is again smaller than that at Transducer A. Note that in this region the displacement is attributed to the motion-dependent fluid force. Because the tube displacement associated with the TSP-inactive mode is limited by the TSP, the fluid force coefficients are practically constant for the whole region regardless of the flow velocity. Therefore, large tube displacement at Transducer B cannot develop. This is contrary to the case of forced excitation. In the case of forced excitation, when the excitation force increases, although the tube displacement is limited at Transducer A, the displacement at Transducer B can increase significantly with the flow velocity. Based on the different characteristics of the tube response, measuring tube displacements at different locations can be used as a means to determine the dominant excitation mechanism.

When the tube is aligned with the TSP hole, at different flow velocities, there is a dominant frequency (see Fig. 26). In the higher flow velocity range in which the TSP-inactive mode is unstable, tube impact with the TSP becomes larger and the frequency spectra becomes broader. Once the tube becomes unstable in the TSP-active mode, the dominant frequency is practically the same as the case of a tube supported by TSP without clearance.

Figure 27 shows the frequency spectra for the case in which the tube is aligned with the TSP clearance at zero flow. As the flow velocity

increases, the tube becomes unstable in the TSP-inactive mode. At a little higher flow velocity, fluid drag force forces the tube to contact the TSP and the TSP-inactive mode is stabilized. At the higher flow velocity, the tube loses stability in the TSP-active mode.

Figures 28 and 29 show the time histories for two different clearances at several flow velocities. In the region in which the TSP-inactive mode is unstable, the displacement doesn't increase with the flow velocity. In this region, the tube may be stabilized by drag force (see Fig. 29) and the tube displacement is reduced. Once the tube becomes unstable in the TSP-active mode, tube displacement becomes much larger.

From the tube response curves and frequency spectra, the critical flow velocities associated with the TSP-inactive and TSP-active modes can be determined. The critical flow velocity for the TSP-inactive mode is independent of the TSP clearance. The critical flow is ~ 1.49 m/s and the oscillation frequency at instability is 24 Hz.

The critical flow velocity for the TSP-active mode depends on the diametral clearance of the baffle plate and the alignment of the baffle-plate support. Figure 30 shows the critical flow velocity and oscillation frequency at instability as a function of diametral clearance for the case in which the tube is aligned with the TSP. The critical flow velocity is larger for larger clearance. This agrees with the data obtained by Gorman [11]. However, the oscillation frequency is larger for clearance between 0.25 to 1.25 mm.

Table 5 shows the critical flow velocities and oscillation frequencies for diametral clearance equal to 0.64 mm. The critical flow velocities for the TSP-active mode vary from 2.24 to 3.55 m/s and the oscillation frequencies at instability vary from 38.4 to 41.8 Hz. Among the different support conditions, critical flow velocity and oscillation frequency are the smallest for zero clearance and the largest for alignment at 2.44 m/s.

The alignment of tube with respect to TSP hole is important to the instability of the TSP-inactive mode and the TSP-active mode. Misalignment of the tube may suppress the instability of the TSP-inactive mode and increase the critical flow velocity of the TSP-active mode. Even for the simple TSP in this test, tube response characteristics are complex. In practical situations, where tubes are supported by a series of TSPs, the tube response characteristics and the critical flow velocity are even more complicated.

2. Test A2

In this test, the tube span between C2 and C3 is submerged in water. For small-amplitude oscillations the natural frequencies are approximately independent of diametral clearance. However, the modal

Fig. 28. Tube displacement time histories for the diametral gap equal to 0.51 mm (Test A1).

Fig. 29. Tube displacement time histories for Test A1 with the diametral gap equal to 0.64 mm (alignment at zero flow).

Fig. 30. The critical flow velocity associated with the TSP-active mode and oscillation frequency as a function of diametral gap (Test A1).

Table 5. Critical Flow Velocity and Oscillation Frequency for
for Diametral Clearance Equal to 0.64 mm (Test A1)

Support Conditions	Critical Flow Velocity (m/s)	Oscillation Frequency at Instability (Hz)
No clearance	2.24	38.4
Heavy preload	2.68	39.8
Alignment at all flow velocities	3.02	41.5
Alignment at zero flow	3.05	41.5
Light preload	3.32	41.5
Alignment at 2.44 m/s	3.55	41.8

damping ratio for the TSP-inactive mode is a function of the diametral clearance because of the fluid viscous effect; modal damping of the TSP-inactive mode increases with reducing clearance. The results are given in Table 4.

Figures 31 and 32 show the tube displacement as a function of flow velocity for different diametral clearances and different alignments with a diametral gap of 0.64 mm. The results are similar to those of Test A1. However, there are two differences:

- In Test A1, because there is no water in the gap between the tube and TSP, the tube can move freely. In Test A2, the fluid is inside the clearance. Therefore, the rms tube displacements in Test A2 are smaller than those in Test A1.

- Once the tube becomes unstable in the TSP-inactive mode, the fluid in the gap plays an important role such that the displacements do not increase as drastically as those in Test A1.

Figures 33 and 34 show typical frequency spectra of tube displacements. The frequency spectra are similar to those in Figs. 26 and 27 for Test A1. However, the dominant frequency peaks are not as sharp as those in Test A1.

Figure 35 shows the time histories for 0.51 mm diametral clearance. The displacement associated with the TSP-inactive mode is in general not as steady as that in air (see Fig. 28). This is probably attributed to the fluid in the gap.

The critical flow velocity for the TSP-inactive mode can be determined from the response curve and frequency spectra. For small diametral clearances, tube displacement does not increase drastically with the increase in flow velocity; the critical flow velocity is not well defined. From Fig. 31, it is noted that the critical flow velocity for the TSP-inactive mode increases with decreasing diametral clearance; this is attributed to the increase in the modal damping ratio as presented in Table 4. The critical flow velocity varies from 1.4 m/s for ∞ clearance to about 1.59 m/s for 0.25 mm diametral clearance.

The critical flow velocities for the TSP-active mode are summarized in Table 6. The critical flow velocity increases with the diametral gap. For large diametral clearances, the tube does not become unstable for flow velocities up to 4.1 m/s, which is the maximum flow velocity obtainable in the test loop. For other clearances, the critical flow velocities and oscillation frequencies given in Test A2 are, in general, larger than those in Test A1.

Fig. 33. Frequency spectra of tube displacement for the diametral gap equal to 0.64 mm (aligned at zero flow).

Fig. 34. Frequency spectra of tube displacement for the diametral gap equal to 0.64 mm (aligned at zero flow).

Fig. 35. Tube displacement time histories for Test A2 with a diametral gap equal to 0.51 mm.

Table 6. Critical Flow Velocity and Oscillation Frequency
for the TSP-Active Mode (Test A2)

Diametral Clearance (mm)	Critical Flow Velocity (m/s)	Oscillation Frequency at Instability (Hz)
0.25	2.53	40.0
0.38	3.52	42.3
0.64		
Alignment at all flow	3.81	42.8
Alignment at zero flow	3.75	42.8
Alignment at 2.44 m/s	3.34	42.6
1.02	3.87	43.0
1.27	> 4.1	
2.54	> 4.1	

3. Test B1

Figure 36 shows the rms tube displacements as a function of flow velocity for different diametral gaps. The response characteristics are similar to those of Tests A1 and A2. To prevent the damage of the force transducers, the flow velocity is limited to a value smaller than the critical flow velocity of the TSP-active mode. In the region in which the TSP-inactive mode is unstable, the tube displacement is almost independent of the flow velocity and increases with the diametral clearance.

Figure 37 shows typical frequency spectra as a function of flow velocity. In general there is a dominant frequency at each flow velocity; these dominant frequencies are given in Fig. 38. The dominant response frequencies increase with flow velocity for a specific diametral gap. Among the different tests, the dominant response frequency decreases with the diametral gap for smaller gaps (0.3 mm to 0.64 mm) and increases with the diametral gap for larger gaps (0.76 mm to 2.54 mm).

Figures 39 through 42 show the time histories of tube displacements and impact forces. Some general characteristics are noted:

- In the region in which the TSP-inactive mode is unstable, tube displacement close to the baffle plate varies very little with the flow velocity. However, the impact force and impact frequency increase significantly with the flow velocity (see Figs. 39 and 40). At lower flow velocities, the impact between the tube and baffle plate is intermittent and its peak magnitude is much smaller than 1 Newton. At higher flow velocities, the tube impacts almost periodically with the baffle plate and its peak magnitude may be larger than 1 Newton.

- For the tube located centrally in the gap, the impact forces measured from the two force transducers are approximately the same (see Figs. 39, 41a, and 42a). When the tube is offset to one side, the tube does not impact with the other transducer. However, there are double impacts against one of the transducers during each cycle of vibration (see Figs. 40 and 41b).

- The tube displacement close to the baffle plate correlates with the impact force fairly well; the impacts occur corresponding to the tube displacement peaks.

- For a given flow velocity, the tube displacement and impact force depend on diametral gap. For larger gaps, tube motion is more steady and the impact force is larger. For example, a comparison of Figs. 41c and 39 shows that the impact forces for 2.54 mm at 2.83 m/s are much larger than those for 0.64 mm at 2.97 m/s.

- Once the tube becomes unstable in the TSP-active mode (see Fig. 42), tube motion is very steady and the impact forces are significantly

Fig. 36. RMS tube displacements as a function of flow velocity for different diametral gaps (Test B1).

Fig. 37. Frequency spectra of tube displacement for the diametral gap equal to 0.38 mm (Test B1).

Fig. 38. Dominant tube response frequencies at different flow velocities for different diametral gaps (Test B1).

Fig. 39. Time histories of tube displacement and impact forces for the diametral gap equal to 0.64 mm (Test B1, tube centrally located).

Fig. 40. Time histories of tube displacement and impact forces for diametral gap equal to 0.64 mm (Test B1, tube offset to force transducer 2).

(a) TUBE CENTRALLY LOCATED (GAP = 0.64 mm, $V = 2.15$ m/s)

(b) TUBE OFFSET TO FORCE TRANSDUCER 2 (GAP = 0.64 mm, $V = 2.82$ m/s)

(c) TUBE CENTRALLY LOCATED (GAP = 2.54 mm, $V = 2.83$ m/s)

0 0.05 0.10 0.15 0.20 0.25 0.30
TIME, s

Fig. 41. Time histories of tube displacement and impact forces for two different diametral gaps (Test B1).

(a) TUBE CENTRALLY LOCATED (DIAMETRAL GAP = 0.51 mm, $V = 3.05$ m/s)

(b) TUBE OFFSET TO IMPACT FORCE TRANSDUCER 2 (DIAMETRAL GAP = 0.64 mm, $V = 3.17$ m/s)

Fig. 42. Time histories of tube displacement and impact forces for two different diametral gaps (Test B1).

larger than those associated with the instability of the TSP-inactive mode. In addition, the offset of the tube appears to have little effect on the impact forces associated with the TSP-active mode.

Wear rate of the tube at the TSP location is related to the impact force characteristics. Three parameters are analyzed:

- Average impact force: The impact force is in general a triangular pulse (see Fig. 41) and varies with flow velocity and clearance. The averaged impact force, which is obtained by converting to rectangular pulses and then averaging the magnitude of the rectangular pulses, is analyzed for different gaps.
- Number of impacts per cycle of oscillation: The impact number is counted from the time history record of the force transducers.
- Work-rate parameter: The impact force multiplied by the number of impacts per unit time is related to work rate of the impact force.

Average impact force, number of impacts per cycle of oscillation and work-rate parameter are given in Figs. 43 to 45 for the flow-velocity region in which the TSP-inactive mode is unstable. Some general characteristics are noted:

- For diametral gaps equal to 0.3 mm to 0.76 mm, the average impact forces are practically independent of the flow velocity. However, the impact-force magnitude increases with the diametral clearance.

- The number of impacts increases with the flow velocity and varies insignificantly with the diametral clearance for 0.3 to 0.76 mm. For large gaps (1.27 and 2.54 mm), the number of impacts is about 2 per cycle of oscillation.

- The work-rate parameter increases with flow velocity and diametral clearance.

- Figure 44 shows that the impact forces and number of impacts on the two force transducers are not identical; this is attributed to the fact that the tube is located closer to Force Transducer 2. Note that force magnitude is larger, but number of impacts is smaller for Transducer 1. However, the work-rate parameter is about the same for the two force transducers.

- Figure 45 shows the results for two cases---tube centrally located and tube offset to one side. For the offset case, the tube never contacts Force Transducer 1. The impact force is small but the number of impacts is large for the offset case. The work-rate parameter for the offset case is about half that of the centrally located tube.

Fig. 43. Average impact force, number of impacts per Hertz and work-rate parameter for Test B1: (a) diametral gaps 0.3 mm to 0.76 mm.

Fig. 43. Average impact force, number of impacts per Hertz and work-rate parameter for Test B1: (b) diametral gaps 1.27 mm and 2.54 mm (Test B1).

Fig. 44. Impact force, number of impacts per Hertz and work-rate parameter for the diametral gap equal to 0.38 mm (Test B1).

Fig. 45. Impact force, number of impacts per Hertz and work-rate parameter for the diametral gap equal to 0.64 mm (Test B1).

Figure 46 shows the histograms of impact forces for two clearances 0.64 mm and 2.54 mm at several flow velocities. For 0.64 mm, the force distribution is similar to the forced excitation studied by Ko and Basista [13]. However, for 2.54 mm, the force distributions vary with the flow velocity.

4. Test B2

Tube response characteristics in Test B2 are basically the same as those obtained from Test B1. Figures 47 and 48 compare the rms tube displacements and time histories of tube displacement and impact forces for two diametral gaps--0.64 mm and 1.27 mm. The effect of water in the gap is the same as those in Test A2. Several effects are noted:

- Once the tube becomes unstable in the TSP-inactive mode, the rms tube displacement for Test B1 is larger than that in Test B2. This is attributed to fluid in the gap.
- Without fluid, tube motion appears to be more steady (see Fig. 48).
- The impact forces in Test B1 are larger than those in Test B2. The fluid in the gap tends to reduce the impact.

The average impact force, number of impacts, and work-rate parameter are given in Figs. 49 and 50. The effects of fluid in the clearance are noticeable:

- 1) The average impact force is small with fluid in the clearance.
- 2) The number of impacts is significantly reduced.
- 3) The work-rate parameter is much smaller, and is practically independent of flow velocity.

The fluid in the clearance plays an important role in wear rate. It tends to reduce the wear rate drastically for fluidelastic instability of a TSP-inactive mode.

V. CONCLUSIONS

This report presents the results of three series of tests on tube vibration in fluid:

- A single continuous tube supported by baffle plates with clearance.
- A prototypic LMFBR steam generator tube and its 0.4 scale model.
- A tube row with tube/baffle interaction in crossflow.

Test results provide some new insights into the dynamic characteristics of tube/baffle interaction.

The existence of TSP-inactive modes has been firmly established for TSPs with relatively large diametral clearances. Under such conditions, tube response characteristics are drastically different from those of tubes

Fig. 46. Histograms of impact force for diametral gaps equal to 0.64 mm and 2.54 mm (Test B1).

Fig. 47. RMS tube displacements as a function of flow velocity for diametral gap equal to 0.64 mm and 1.27 mm (Tests B1 and B2).

TEST B1

TEST B2

Fig. 48. Time histories of tube displacement and impact forces for the diametral gap equal to 1.27 mm (Tests B1 and B2).

Fig. 49. Average impact force, number of impacts per Hertz and work-rate parameter for the diametral gap equal to 1.27 mm (Tests B1 and B2).

Fig. 50. Average impact forces, number of impacts per Hertz and work-rate parameter for diametral gap equal to 0.64 mm (Tests B1 and B2).

with knife-edge supports. Current design practices of assuming heat exchanger tubes to be simply supported at TSPs without clearance are not conservative.

Scale-model testing of tubes on multiple TSPs with clearance may not provide satisfactory solutions because of nonlinear response behavior that results in scaling law and similitude requirements not being directly applicable. Great care should be exercised in applying the data from a scale model to a prototype. Testing on a full-scale model is recommended whenever possible. In a case where scale-model testing is a necessity, a geometrically scaled TSP clearance provides better results.

Tubes supported by TSPs with clearance are intrinsically nonlinear. Tube response characteristics depend on excitation magnitude and other system parameters, such as alignment, preload, and operating conditions. Prediction of the response of heat exchanger tubes is difficult because of the uncertainty in characterizing the initial condition of the tube with respect to TSPs.

The role of TSP-inactive modes in fluidelastic instability has been demonstrated. In the past, fluidelastic instability of TSP-inactive modes has not been considered in practice. Some of the tube damage may be attributed to the fluidelastic instability of TSP-inactive mode, in particular, for those tubes supported by TSPs with relatively large clearances.

Tube response characteristics associated with the instability of a TSP-inactive mode are different from other types of excitation and those of TSP-active modes. When a tube becomes unstable in a TSP-inactive mode, because of the constraints at TSPs, large tube oscillations will not occur. With the increase of flow velocity, tube displacement remains almost constant until the instability of other modes occurs. However, the peak impact force at TSPs and number of impacts per cycle of oscillation--and, consequently, the work-rate parameter--do increase with the flow velocity. In addition, tube displacement, impact force, and number of impacts at a given flow velocity increase with the diametral clearance in the practical cases. Therefore, it is beneficial to limit the TSP clearances to sizes as small as practically possible.

The fluid in the clearance between the tube and TSPs affects tube response characteristics: it increases system damping and reduces tube response. For the instability of a TSP-inactive mode, the fluid in the gap reduces tube displacement and impact force so wear rate is reduced. The detailed fluid dynamics associated with the tube/baffle interaction is important to quantify the wear of heat exchanger tubes. More detailed analysis and experiments are needed to understand these effects.

The alignment of the tube is important for TSP-inactive modes. Misalignment of the tube is beneficial; it reduces tube displacement and work-rate parameter. Therefore, tubes located centrally at the TSP holes represent the worst case.

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REFERENCES

1. J. F. Sebald and W. D. Nobles, "Control of Tube Vibration in Steam Surface Condensers," Proc. Am. Power Conf., Vol. XXIV, 630-643 (1962).
2. P. M. Moretti and R. L. Lowery, "Heat Exchanger Tube Vibration Characteristics in a 'No Flow' Condition," Final Report: Tubular Exchanger Manufacturers Association Experimental Program, Oklahoma State University (1973).
3. Y. S. Shin, J. A. Jendrzejczyk, and M. W. Wambsganss, "Effect of Tube/Support Interaction on the Tube Vibration of a Tube on Multiple Supports," Argonne National Laboratory, Technical Memorandum ANL-CT-77-5 (1977).
4. R. J. Rogers and R. J. Pick, "On the Dynamic Spatial Response of a Heat Exchanger Tube with Intermittent Baffle Contacts," Nucl. Eng. Des. 36, 81-90 (1976).
5. Y. S. Shin, D. E. Sass, and J. A. Jendrzejczyk, "Vibro-Impact Response of a Tube with Tube-Baffle Interaction," Argonne National Laboratory, Technical Memorandum ANL-CT-78-11 (1978).
6. P. L. Ko and R. J. Rogers, "Analytical and Experimental Studies of Tube/Support Interaction in Multi-Span Heat Exchanger Tubes," Nucl. Eng. Des. 65, 399-409 (1981).
7. S. S. Chen, G. S. Rosenberg, and M. W. Wambsganss, "On Tube-Baffle Impact During Heat Exchanger Tube Vibration," Flow-Induced Vibration in Heat Exchanger, D. D. Reiff (ed.), ASME, pp. 27-35 (1970).
8. E. C. Ting, S. S. Chen, and M. W. Wambsganss, "Dynamics of Component-Support Impact: An Elastic Analysis," Nucl. Eng. Des. 52, 235-244 (1979).
9. M. Wahle, "Determination of Flow Induced Nonlinear Vibrations of Prestressed Heat Exchanger Tubes," Heat Exchangers, J. Taborek et al. (eds.), Hemisphere Publishing Co., pp. 339-353 (1981).
10. R. Jiyavan, "Analysis of Cantilever Pipes in Transverse Fluid Flow with Motion Limiting Stops at the Free End," Paper No. G5/8, 7th SMIRT Conference, Chicago, Ill., USA (1983).
11. D. J. Gorman, "The Effects of Tube-to-Baffle Plate Clearances on the Liquid Crossflow Induced Vibration of Heat Exchanger Tube Bundles," Presented at Sixth Symposium on Engineering Applications of Mechanics, Calgary, Alberta, Canada, June 6-9, 1982.
12. D. S. Weaver and W. Schneider, "The Effect of Flat Bar Supports on the Cross Flow Induced Response of Heat Exchanger U-Tubes," ASME Paper No. 82-JPGC-NE-12 (1982).

13. P. L. Ko and H. Basista, "Correlation of Support Impact Force and Fretting-Wear for a Heat Exchanger Tube," presented at ASME Pressure Vessel and Piping Conference, Portland, Oregon, June 1983.

APPENDIX: FORCE CALIBRATION

The force calibration apparatus features a carbon steel tube fastened to two rigid supports with two piezoelectric force transducers and a pair of drive coils placed at the center of the tube (see Fig. A.1). The distance between the coil and tube is 1.52 mm. The coils are driven at current levels from 2 to 16 amp at 25, 50, 75 and 100 Hz. This enables one coil to exert an attractive force on the tube when the other is turned off. Thus, it produces a net sinusoidal force that is measured by Force Transducers F_A and F_B . The total forces exerted on the tube is the summation of two transducer forces. Figure A.2 shows the force-current relationship used to calculate the forces applied to the tube.

Fig. A.1. Force calibration apparatus.

Fig. A.2. Exciting force-current relation of one pair of drive coils.

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