EFFECT OF TUBE/SUPPORT INTERACTION
ON THE VIBRATION OF A TUBE ON MULTIPLE SUPPORTS

by

Y. S. Shin, J. A. Jendrzeczyk, and M. W. Wambsganss

Components Technology Division

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TABLE OF CONTENTS

	<u>Page</u>
NOMENCLATURE	1x
ABSTRACT	1
I. INTRODUCTION	2
II. EXPERIMENTAL APPARATUS AND MEASUREMENT TECHNIQUES	4
A. Experimental Apparatus	4
B. Measurement Techniques	11
B.1 Tube Vibration Characteristics and Response	11
B.2 Exciting Force Magnitude	13
III. ANALYSIS OF IDEALIZED MULTI-SPAN MODEL	21
A. General	21
B. Natural Frequencies and Mode Shapes	23
C. Steady-State Harmonic Analysis	23
IV. ANALYSIS OF EXPERIMENTAL DATA	36
A. Tube Orientation	36
A.1 Resonant Frequencies	36
A.2 Displacement Response Amplitude	38
A.3 Damping	42
B. Tube/Support Misalignment	42
B.1 Resonant Frequencies	42
B.2 Displacement Response Amplitude	42
B.3 Damping	44
C. Fluid Medium	44
C.1 Resonant Frequencies	44
C.2 Displacement Response Amplitude	44
D. Tube/Support-Hole Clearance	47

TABLE OF CONTENTS (Contd.)

	<u>Page</u>
D.1 Resonant Frequencies	51
D.2 Displacement Response Amplitude	51
D.3 Damping	54
E. Support Thickness	54
E.1 Resonant Frequencies	57
E.2 Displacement Response Amplitude	57
E.3 Damping	60
F. Support Spacing	60
F.1 Variation of Double Supports	60
F.1.a Resonant Frequencies	60
F.1.b Displacement Response Amplitude	63
F.2 Equal Span	68
F.2.a Resonant Frequencies	68
F.2.b Displacement Response Amplitude	68
F.3 Damping	73
F.4 Optimum Support Spacing	73
V. SUMMARY OF RESULTS	74
VI. DISCUSSIONS	76
A. Nonlinear Response	76
B. Tube Motion	76
C. Damping	78
D. Repeatability of Test	78
E. Comparison with Other Results	81
E.1 Resonant Frequencies	81
E.2 Damping	82

TABLE OF CONTENTS (Contd.)

	<u>Page</u>
F. Limitations and Application of Test Results	83
F.1 Limitations	83
F.2 Application	83
ACKNOWLEDGMENTS	85
REFERENCES	86

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
1	Tube/Support Configurations for the Thirteen Cases Tested	6
2	Test Model Configuration	7
3	Test Model in Vertical Orientation	8
4	Test Model in Horizontal Orientation	9
5	Details of Drive Coil Assembly	10
6	Details of Transducer Setup	12
7	Data Acquisition and Reduction Scheme	14
8	Force Measurement Apparatus	15
9	Measurement of Preload-Linear-Spring Stiffness	16
10	Coil-Force vs. Coil-to-Tube Clearance for a Constant Coil Current	18
11	Total Preload in Spring vs. Snap-In Distance for Four Different Current Levels	19
12	Total Force Exerted on Tube vs. Drive Current for Two Different Coil-to-Tube Clearances	20
13	Natural Frequencies and Mode Shapes for "Both Supports - A - and - B - Included"	24
14	Natural Frequencies and Mode Shapes for "Support - A - Removed/ In-Water" Condition	25
15	Natural Frequencies and Mode Shapes for "Support - B - Removed/ In-Water" Condition	26
16	Natural Frequencies and Mode Shapes for "Support - A - Moved- to - H / In-Water" Condition	27
17	Natural Frequencies and Mode Shapes for "Equal-Span/In-Water" Condition	28
18	Peak Displacement and Phase Angle for "Both-Supports - A - and - B - Included" and "in-Air/in-Water Condition" (Total Exciting Force Amplitude = 7.5 lbs (3.4 kg))	30
19	Peak Displacement and Phase Angle for "Support - A - Removed" and "In-Water Condition" (Total Exciting Force Amplitude = 7.5 lbs (3.4 kg))	31

LIST OF FIGURES (Contd.)

<u>No.</u>	<u>Title</u>	<u>Page</u>
20	Peak Displacement and Phase Angle for "Support - B - Removed" and "In-Water Condition" (Total Exciting Force Amplitude = 7.5 lbs (3.4 kg))	32
21	Peak Displacement and Phase Angle for "Support - A - Moved-to - H - " and "In-Water Condition" (Total Exciting Force Amplitude = 7.5 lbs (3.4 kg))	33
22	Peak Displacement and Phase Angle for "Seven Equal Spans" and "In-Water Condition" (Total Exciting Force Amplitude = 6.85 lbs (3.1 kg))	34
23	Effect of Tube Orientation on Tube Resonant Frequencies	37
24	Effect of Tube Orientation (Test Cases 1 and 5): RMS Displacement Amplitude at the Center of the Span Experiencing the highest Response for the First Three Resonant Frequencies as a Function of Total Exciting Force Amplitude	40
25	Effect of Tube Orientation (Test Cases 1 and 5): Peak Displacement Amplitude along the Span at 1st, 2nd and 4th mode Excitations for Total Exciting Force Amplitude of 7.5 lbs (3.4 kg)	41
26	Effect of Tube/Support Misalignment on Tube Resonant Frequencies	43
27	Effect of Tube/Support Misalignment (Test Case 6): RMS Displacement at the Center of the Span for the First Four Resonant Frequencies as a Function of Total Exciting Force Amplitude	45
28	Effect of Fluid Medium on Tube Resonant Frequencies	46
29	Effect of Fluid Medium (Test Cases 1 and 2): RMS Displacement Amplitude at the Center of the Span Experiencing the Highest Response for the First Four Resonant Frequencies as a Function of Total Exciting Force Amplitude	48-49
30	Effect of Fluid Medium (Test Cases 1 and 2): Peak Displacement Amplitude along the Span at 1st, 2nd and 4th Mode Excitations for Total Exciting Force Amplitude of 7.5 lbs (3.4 kg)	50
31	Effect of Tube/Support-Hole Clearance on Tube Resonant Frequencies	52
32	Effect of Tube/Support-Hole Clearance (Test Cases 5, 8 and 9): RMS Displacement Amplitude at the Center of the Span Experiencing the Highest Response for the First Three Resonant Frequencies as a Function of Total Exciting Force Amplitude	53

LIST OF FIGURES (Contd.)

<u>No.</u>	<u>Title</u>	<u>Page</u>
33	Effect of Tube/Support-Hole Clearance (Test Cases 5, 8 and 9): RMS Displacement Amplitude at the Center of the Span for the First Three Resonant Frequencies as a Function of Tube/Support-Hole Clearance	55
34	Effect of Tube/Support-Hole Clearance (Test Cases 5, 8 and 9): Peak Displacement Amplitude along the Span at 1st, 2nd and 4th Mode Excitations for Total Exciting Force Amplitude of 7.5 lbs (3.4 kg)	56
35	Effect of Tube-Support Thickness on Tube Resonant Frequencies	58
36	Effect of Tube-Support Thickness (Test Cases 5, 10, 11 and 12): RMS Displacement Amplitude at the Center of the Span Experiencing the Highest Response for the First Three Resonant Frequencies as a Function of Total Exciting Force Amplitude	59
37	Effect of Tube-Support Thickness (Test Cases 5, 10, 11 and 12): RMS Displacement Amplitude at the Center of the Span for the First Resonant Frequency as a Function of Tube-Support Thickness	61
38	Effect of Double-Supports A,B Arrangement on Tube Resonant Frequency	62
39	Effect of Double-Supports (A, B) Arrangement (Test Cases 3, 4, 5 and 7): RMS Displacement Amplitude at the Center of the Span Experiencing the Highest Response for the First Three Resonant Frequencies as a Function of Total Exciting Force Amplitude	64
40	Support - A - Removed (Test Case 3): Peak Displacement Amplitude along the Span at 1st, 2nd and 4th Mode Excitations for Total Exciting Force Amplitude of 7.5 lbs (3.4 kg)	65
41	Support - B - Removed (Test Case 7): Peak Displacement Amplitude along the Span at 1st, 2nd and 4th Mode Excitations for Total Exciting Force Amplitude of 75. lbs (3.4 kg)	66
42	Support - A - Moved-to - H - (Test Case 4): Peak Displacement Amplitude along the Span at 1st, 3rd and 4th Mode Excitations for Total Exciting Force Amplitude of 7.5 lbs (3.4 kg)	67
43	Tube Resonant Frequencies of Seven Equal Span Case	69
44	Seven Equal Span (Test Case 13): RMS Displacement Amplitude at the Center of Each Span for the First Four Resonant Frequencies as a Function of Total Exciting Force Amplitude	70-71
45	Seven Equal Spans (Test Case 13): Peak Displacement Amplitude along the Span at First Four Mode Excitations for Total Exciting Force Amplitude of 6.85 lbs (3.1 kg)	72

LIST OF FIGURES (Contd.)

<u>No.</u>	<u>Title</u>	<u>Page</u>
46	Nonlinear Jump Phenomenon of Response Amplitude: Displacement Response Amplitude Curves for Three Different Exciting Force Amplitudes	77
47	Typical Tube Motion Normal to the Tube Axis at the Center of the Span	79
48	The Range of Measured Viscous Damping Factors for the First Three Resonances	80

LIST OF TABLES

<u>No.</u>	<u>Title</u>	<u>Page</u>
1	Description of Test Cases	5
2	Idealized Multi-Span Model Properties	22
3	Summary of Natural Frequencies and Spans Experiencing Maximum Amplitude for a Given Mode Shape	29

NOMENCLATURE

<u>Symbol</u>	<u>Description</u>
D1- D10	Displacement Transducer Number
d	External tube diameter
F_p	Total preload in preload-linear spring
f	Exciting forcing frequency
X_g	Drive coil-to-tube clearance
X_p	Preload-linear-spring deformation
X_s	Snap-in distance between drive coil and tube
ξ	Viscous damping coefficient

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ABSTRACT

The effects of tube orientation (horizontal/vertical), tube/support misalignment, fluid medium (air/water), tube/support-hole clearance, support thickness, exciting force amplitude, and support spacing on the vibrational characteristics (resonant frequencies, mode shapes, and damping) and displacement response amplitude of a multi-span tube were determined from thirteen different tests. The test model features a 15.875 mm (0.625 in.) dia. x 3.175 mm (0.125 in.) thick x 5.232 m (206 in.) long seamless Croloy (2-1/4% Cr - 1% Mo) tube, and combinations of four support thicknesses (12.70 mm (1/2 in.), 19.05 mm (3/4 in.), 38.10 mm (1-1/2 in.), and 50.80 mm (2 in.)) and three sizes of diametral clearance of tube/support-hole (0.254 mm (10 mils), 0.508 mm (20 mils), and 0.762 mm (30 mils)). The test results were compared with the analytical results based on the multi-span beam with "knife-edge" supports.

I. INTRODUCTION

Flow-induced tube vibrations in heat exchangers have resulted in tube failures in operating heat exchangers [1,2] which, in turn, have resulted in extended and costly plant outages. To avoid detrimental tube vibration in heat exchangers, resonant conditions and instabilities must be avoided and/or peak dynamic response amplitudes must not exceed allowable limits. The determination of tube vibrational characteristics and response is an important first step in any vibration analysis. However, in attempting a theoretical analysis, questions arise as to the effects of tube-support conditions on dynamic response. This report presents an experimental study to evaluate such effects; test results are compared with analytical results based on a multi-span beam with "knife-edge" supports.

The objective of the study is to determine the effects of tube orientation (horizontal/vertical), tube/support misalignment, fluid medium (air/water), tube/support-hole clearance, support thickness, exciting force amplitude and support spacing on the tube vibrational characteristics (resonant frequencies, mode shapes, and damping) and response amplitudes. Orbital motion of the tube and nonlinearity of response amplitude as a function of frequency are also discussed. The experiments were performed using a Croloy (2-1/4% Cr - 1% Mo) tube multiply supported by seven stainless steel plates and a simulated tubesheet. The tube and supports were prototypic, the tube spans nearly prototypic, and the overall length approximately one-third that of a straight tube segment of the AI steam generator designed for Clinch River Breeder Reactor Plant (CRBRP) [1].

Several authors [3-11] have studied, theoretically and experimentally, to determine dynamic behavior of tubes with motion-limiting supports. Sebald and Nobles [3] conducted a test to evaluate the effect of tube/support

hole clearance on the natural frequencies of a tube using an eight-span symmetric tube model. Among other things, they concluded that tube natural frequency decreases as the clearance increases. Moretti and Lowery [4,5] reported experimental results, from 1 to 5-span models, illustrating the effects of tube-support conditions on tube natural frequencies and damping. Not only are natural frequencies and damping important, the response amplitude of a tube is also important especially when there exists a potential for exciting a resonant mode at various operating levels less than full heat exchanger design flow [12]. Several investigators [6-8] have applied the finite element method to predict tube vibration by introducing clearances and assuming linear elastic contact deformation between tube and supports. Chen et al. [9] theoretically studied a force-excited beam supported at the ends, with a motion-limiting stop at the midpoint, and developed a solution method which involves determining the "gross" steady state motion from a single mode approximation in which the system is reduced to a bilinear vibrating model. A numerical/experimental study of a cantilevered beam with a motion-limiting stop at the free end [10,11] showed that a single mode approximation for displacement response may not be valid and, further, that the displacement response is a quasi-narrow band process, whose spectral density exhibits a predominant peak. While these investigations have provided useful insights, tube/support interaction phenomena are not fully understood and, consequently, existing mathematical models are inadequate.

As a part of ANL's Flow-Induced Vibration Program in support of the CRBRP steam generator design activity, tube/support interaction experiments are being performed, not only to gain insight into the dynamic response of the CRBRP steam generator tubes, but also to provide useful design guidance to heat exchanger designers.

II. EXPERIMENTAL APPARATUS AND MEASUREMENT TECHNIQUES

A. Experimental Apparatus

The test model features a 15.875 mm (0.625 in.) dia x 3.175 mm (0.125 in.) thick x 5.232 m (206 in.) long seamless Croloy (2-1/4% Cr - 1% Mo) tube, a 101.60 mm (4 in.) thick brass "tubesheet", and four sets of seven stainless steel tube-support plates (12.70 mm (1/2 in.), 19.05 mm (3/4 in.), 38.10 mm (1-1/2 in.), and 50.80 mm (2 in.) thick), with three different sizes of tube-to-support hole clearances (0.2540 mm (10 mils), 0.5080 mm (20 mils), and 0.7620 mm (30 mils) diametral clearances). A series of 13 different experiments was performed in air/water at room temperature (see Table 1). The tube was securely clamped at one end and supported by seven intermediate support plates (see Fig. 1). The test model was mounted on W 12 x 79 steel beam (304.30 mm (12 in.) deep x 304.30 mm (12 in.) wide x 19.05 mm (3/4 in.) flange thick x 12.70 mm (1/2 in.) web thick) with a 254 mm (10 in.) wide x 228.6 mm (9 in.) deep aluminium cage to enclose the test model and contain the water (see Fig. 2). The test rig was mounted both vertically and horizontally (see Figs. 3-4). For the vertical orientation, the tubesheet end was placed at the bottom. Test case 1 is a nearly prototypic model of the lower one-third of the straight segment of the AI CRBRP steam generator. The test rig was sufficiently rigid that significant interaction between tube and test rig caused by mechanical or fluid coupling was avoided.

A means to excite the tube was provided by an electromagnetic exciter assembly consisting of four pairs of coils (see Fig. 5) mounted along tube span 3 (see Fig. 2) in an attempt to simulate a distributed loading. The excitation span corresponds to the span exposed to the crossflow at the shellside fluid outlet. Ten non-contacting displacement transducers were used to measure the vibration response of the tube; two transducers,

TEST CASE	SUPPORT THICKNESS (INCHES)*				DIAMETRICAL GAP OF TUBE-SUPPORT HOLE (MILS) **			ORIENTATION		TUBE-SUPPORT ALIGNMENT		FLUID MEDIUM		CLOSELY - SPACED DOUBLE SUPPORTS ARRANGEMENT						EQUAL SPAN
	1/2	3/4	1 1/2	2	10	20	30	HORIZ.	VERT.	ALIGN	MIS-ALIGN.	AIR	WATER	(A) (B)	(A) OUT	(B) OUT	(A) → (H)			
1			●		●				●	●			●	●						
2			●		●				●	●		●	●	●						
3			●		●			●		●			●		●					
4			●		●			●		●			●				●			
5			●		●			●		●			●	●						
6			●		●			●			●		●	●						
7			●		●			●		●			●			●				
8			●			●		●		●			●	●						
9			●				●	●		●			●	●						
10	●				●			●		●			●	●						
11		●			●			●		●			●	●						
12				●	●			●		●			●	●						
13			●		●			●		●			●					●		

NOTES: * 1 inch = 25.40 mm
 ** 1 mil = 0.0254 mm

Table 1. Description of Test Cases
 (ANL Neg. No. 113-76-745)

Fig. 1. Tube/Support Configurations for the Thirteen Cases Tested (ANL Neg. No. 113-76-739)

Fig. 2. Test Model Configuration
(ANL Neg. No. 113-76-741)

Fig. 3. Test Model in Vertical Orientation
(ANL Neg. No. 113-75-644)

Fig. 4. Test Model in Horizontal Orientation (ANL Neg. No. 113-76-718)

Fig. 5. Details of Drive Coil Assembly
(ANL Neg. No. 113-76-719)

orthogonal to one another (See Fig. 6), were positioned at the center of each span with the exception of the excitation span (Span 3) and the 88.90 mm (3-1/2 in.) short span (Span 2). Tube-support spacing and a typical displacement transducer setup are shown in Figs. 2 and 6 respectively. At the excitation span, the strong magnetic field generated by the drive coils would disturb the transducer signals and, consequently, transducers were not mounted at that location.

B. Measurement Techniques

B.1 Tube Vibration Characteristics and Response

The tube resonant frequencies were determined by applying different amplitudes of sinusoidal current (force) to the drive coils and sweeping through a forcing frequency range from 0 to 250 Hz. Resonant frequencies up to the sixth mode were determined from the resulting frequency response curve.

Sinusoidal excitation at up to four different current (force) amplitudes was applied to the tube at each of the resonant frequencies from the lowest up to the sixth mode. Response amplitudes in the form of tube displacements were measured at the center of each span using displacement transducers and were recorded on magnetic tape. RMS current in the drive coils, being proportional to the input force level, were simultaneously recorded. For sinusoidal excitation, peak current (or peak force) was obtained by multiplying RMS current (or peak force) by $\sqrt{2}$; peak response amplitude was approximated in the same way. The real time history of displacement response in excitation direction was simultaneously plotted for each span to determine the phasing relation of peak displacement response at each span.

Fig. 6. Details of Transducer Setup
(ANL Neg. No. 113-76-723)

Tube-support interaction model is a piecewise-linear system and energy dissipation cannot be adequately expressed in terms of the linear viscous damping representation. However, to obtain a feel for the degree of energy dissipation, viscous damping factor was calculated from the spectral density curves using the bandwidth (half-power point) method. Two different levels (high/low) of a white noise input force were applied to the tube.

A nonlinear jump phenomenon of response amplitude was measured from the spectral density curves by sweeping the forcing frequency in upward/downward directions. The data acquisition and reduction scheme is shown in Fig. 7.

B.2 Exciting Force Magnitude

A difficult problem in the use of electromagnetic excitation is to correlate the input current and the physical force magnitude. It is a common practice to calculate the force magnitude from the measured response for a given current level. In this experiment, a unique method was developed to find the current-force relationship. The force measurement apparatus (see Fig. 8) featured a 50.80 mm (20 in.) long Croloy tube, the drive coil assembly used for the tube-support interaction experiments, 2-displacement transducers, 2-preload linear springs mounted on the tube-support pins, 2-preload screws, 12.70 mm (1/2 in.) thick brass sliding plates and a support frame. Transducers, preload linear springs, and preload screws were mounted at both ends of the support channel, and placed symmetrically about the center of the tubespan. The tube and the preload linear spring were held in place by the tube-support pin. The drive coil assembly was mounted on the sliding plate parallel to the tube axis.

The preload linear spring stiffness was defined by measuring tube displacement through the transducer reading for different weights placed at the center of tube-span (see Fig. 9). The exciting force magnitude was measured

Fig. 7. Data Acquisition and Reduction Scheme
(ANL Neg. No. 113-76-742)

(a)

(b)

Fig. 8. Force Measurement Apparatus
(ANL Neg. No. 113-76-746)

Fig. 9. Measurement of Preload-Linear-Spring Stiffness
(ANL Neg. No. 113-76-156)

as follows: the tube was initially placed such that there was no initial preload-linear-spring deformation; the tube was preloaded by equally advancing the preload-screws, which compressed the preload-linear-springs mounted on the tube-support pins; the preload-linear-spring deformation was then measured by the displacement transducer and the preload magnitude was calculated; the drive coils were loaded with a constant current and the sliding plate was slowly advanced toward the tube by turning the sliding-plate-advance bolt until the tube snapped into the drive coils; the snap-in distance of the tube was determined from the transducer reading. The input force magnitude is not only a function of the drive current level, but also a function of coil-to-tube clearance. For the given current level, the input force magnitude increases rapidly as coil-to-tube clearance decreases as shown in Fig. 10. For various levels of currents, total preload in the linear springs was plotted in terms of snap-in distance as shown in Fig. 11. In Fig. 12 the total force exerted on the tube is plotted in terms of drive current for the two different coil-to-tube clearances used in the tube/support interaction experiments. From these curves an approximate linear relationship between force and coil current can be obtained for each of the two coil-to-tube clearances employed; the two constants of proportionality are given on Fig. 12.

Fig. 10. Coil-Force vs. Coil-to-Tube Clearance
for a Constant Coil Current
(ANL Neg. No. 113-76-743)

Fig. 11. Total Preload in Spring vs. Snap-In Distance for Four Different Current Levels (ANL Neg. No. 113-76-740)

Fig. 12. Total Force Exerted on Tube vs. Drive Current for Two Different Coil-to-Tube Clearances (ANL Neg. No. 113-76-744)

III. ANALYSIS OF IDEALIZED MULTI-SPAN MODEL

A. General

The vibration of a tube on multiple supports, with clearance between the tube and support holes, is inherently a nonlinear problem because of the impacting which can occur between the tube and support. A question which arises relates to the applicability of linear theory in predicting multi-span tube vibrational characteristics and response. To gain insight, experimental results will be compared with results from a linear analysis of an idealized model. In this section of the report, the idealized model is discussed and results obtained from a steady state harmonic analysis are presented.

The idealized model is, herein, defined as a multi-span continuous beam with one end perfectly clamped and intermediately supported by "knife-edge" supports. The geometrical and material properties of the idealized model, given in Table 2, are uniform and the same as those of the experimental model. The reference responses of the idealized model are undamped natural frequencies, mode shapes and peak displacement response amplitudes with phasing as determined from steady-state harmonic analysis. The reference responses were calculated for six different cases to provide the basis for comparison with the experimental results: (1) supports - A - and - B - included/in-air condition (no added mass), (2) supports - A - and - B - included/in-water condition (added mass), (3) support - A - removed/in-water, (4) support - B - removed/in-water, (5) support - A - moved-to - H /in-water, and (6) equal-spans/in-water. The DRIPS code* was used to calculate the reference responses of the idealized model. In the computer simulation, the 8.89 cm

*DRIPS (Dynamic Responses In Piping System) code is a linear elastic dynamic analysis program for piping system, developed by A. Gelman, Atomics International, under ERDA funding.

Table 2. Idealized Multi-Span Model Properties

Description	Properties	
Modulus of Elasticity	2.0×10^{11} PA	29×10^6 psi
Weight Density	0.007737 kg/cm^3	0.2795 lb/in.^3
Added Mass of Surrounding Water	0.00277 kg/cm	0.0155 lb/in.
Sectional Area	1.1890 cm^2	0.1843 in.^2
Section Moment of Inertia	0.4995 cm^4	0.0120 in.^4
Span Length	See Fig. 2	
Modal Viscous Damping Factor (ξ)	$2\% \sim 8\%$	$2\% \sim 8\%$
Excitation Force	Sinusoidal Force	

(3-1/2 in.) short span (span 2) was assumed to be rigidly linked to realize a numerically stable solution.

B. Natural Frequencies and Mode Shapes

The first six natural frequencies and mode shapes, for each of the six cases studied, are plotted in Figs. 13-17. For each mode, the associated natural frequency and the span(s) experiencing the highest amplitude are summarized in Table 3. Table 3 shows that tube natural frequencies are relatively insensitive to whether both supports A and B are included or either one of the supports is removed. As expected, the equal span case has a higher first mode natural frequency than any of the other cases.

C. Steady-State Harmonic Analysis

Steady-state harmonic analysis was performed by applying uniformly distributed sinusoidal forces at Span 3, the span which corresponds to the excitation span in the experiment. Resonant conditions were excited by selecting the excitation frequencies to be the multi-span tube natural frequencies. Responses were computed for viscous damping factors (ξ) of 0.02, 0.05 and 0.08, assumed to be the same for each mode. Peak displacement amplitudes and the corresponding phase angles along the length of the beam are plotted for the six reference cases in Figs. 18-22. Variation of phase angles is small and, therefore, only the phase angle for the 0.02 viscous damping factor is plotted.

The effect of added mass on the peak displacement amplitude was negligible for "Supports - A - and - B - Included" case (Fig. 18). Exciting the beam at second and third natural frequencies, showed second mode contribution to be dominant with maximum displacement occurring at Span 1, except for the cases, "Support - A - moved - to - H" and "Equal-Span" (Figs. 18-20). By moving support - A - to - H, modal excitation at 40 Hz was eliminated (Fig. 21).

Fig. 13. Natural Frequencies and Mode Shapes for "Both Supports - A - and - B - Included" (ANL Neg. No. 113-76-760)

Fig. 14. Natural Frequencies and Mode Shapes for "Support - A - Removed/In-Water" Condition (ANL Neg. No. 113-76-758)

Fig. 15. Natural Frequencies and Mode Shapes for "Support - B - Removed/In-Water" Condition (ANL Neg. No. 113-76-761)

Fig. 16. Natural Frequencies and Mode Shapes for "Support - A - Moved - to - H / In-Water" Condition (ANL Neg. No. 113-76-759)

Fig. 17. Natural Frequencies and Mode Shapes for "Equal-Span/In-Water" Condition (ANL Neg. No. 113-76-762)

Table 3. Summary of Natural Frequencies and Spans Experiencing Maximum Amplitude for a Given Mode Shape

Description	Natural Frequencies (Hz)						Span No. Experiencing Maximum Amplitude for the Given Mode					
	1st Mode	2nd Mode	3rd Mode	4th Mode	5th Mode	6th Mode	1st Mode	2nd Mode	3rd Mode	4th Mode	5th Mode	6th Mode
Supports - A - and - B - Included/ In-Air (Fig. 13)	33.6	47.1	50.8	116.9	129.9	142.3	7→6	1	6→7	6→5	1→4	4→5
Supports - A - and - B - Included/ In-Water (Fig. 13)	29.4	41.3	44.5	102.5	113.9	124.8	7→6	1	6→7	6→5	1→4	4→5
Support - A - Removed/ In-Water (Fig. 14)	29.4	36.2	44.5	100.0	103.6	123.4	7→6	1	6→7	1→4,5	6→5,1	4→5
Support - B - Removed/ In-Water (Fig. 15)	29.4	40.6	44.5	101.7	110.2	123.5	7→6	1	6→7	6→5	1→3,4	4→5
Support - A - Moved-to - H / In-Water (Fig. 16)	29.4	44.5	89.7	102.7	122.2	137.8	7→6	6→7	1→2	6→5	4→5	7→4,6
Equal-Span/In-Water (Fig. 17)	60.0	66.3	77.8	92.1	107.7	122.3	6→7	3→7→6	6→3→4	all same	6→3→1	5→1→6

Fig. 18. Peak Displacement and Phase Angle for "Both-Supports - A - and - B - Included" and "in-Air/in-Water Condition". (Total Exciting Force Amplitude = 7.5 lbs (3.4 kg)) (ANL Neg. No. 113-76-765)

Fig. 19. Peak Displacement and Phase Angle for "Support - A - Removed" and "In-Water Condition"
(Total Exciting Force Amplitude = 7.5 lbs (3.4 kg)) (ANL Neg. No. 113-76-763)

Fig. 20. Peak Displacement and Phase Angle for "Support - B - Removed" and "In-Water Condition" (Total Exciting Force Amplitude = 7.5 lbs (3.4 kg)) (ANL Neg. No. 113-76-764)

Fig. 21. Peak Displacement and Phase Angle for "Support - A - Moved-to - H " and "In-Water Condition" (Total Exciting Force Amplitude = 7.5 lbs (3.4 kg)) (ANL Neg. No. 113-76-767)

Fig. 22. Peak Displacement and Phase Angle for "Seven Equal Spans" and "In-Water Condition"
(Total Exciting Force Amplitude = 6.85 lbs (3.1 kg)) (ANL Neg. No. 113-76-766)

For the "Equal Span" case, high displacement responses occurred at all spans except at Span 1 for the first two modal excitations (Fig. 22). Closely-spaced double supports did not significantly dampen out the peak displacement response at Span 1 (Figs. 19-21).

IV. ANALYSIS OF EXPERIMENTAL DATA

The effects of tube orientation (horizontal/vertical), tube/support misalignment, surrounding fluid medium (air/water), tube/support-hole clearance, support thickness, tube-support spacing, and exciting force amplitude on tube vibrational frequencies, mode shapes, displacement response amplitudes, and energy dissipation were evaluated. As discussed in Section II.B.1, energy dissipation cannot be adequately expressed in terms of the linear viscous damping representation for tube/support interaction model. However, to obtain a feel for the degree of energy dissipation, viscous damping factors were calculated from measured response for some test cases.

A. Tube Orientation

The experiment was performed in both a horizontal and vertical tube orientation using the reference tube/support-spacing configuration (Fig. 1(a)); 38.10 mm (1-1/2 in.) support thickness, 0.254 mm (10 mils) tube/support-hole diametral clearance, and in-water condition (see Test Cases 1 and 5 in Table 1).

A.1 Resonant Frequencies

The experimental results showed that the tube orientation did not significantly change the tube resonant frequencies. Comparison with analytical results showed that the measured frequencies were close to the natural frequencies of a multi-span beam with "knife-edge" supports (see Fig. 23).

In the vertical orientation, two distinct resonant frequencies were observed in the neighborhood of the first mode frequency: 30.5 Hz and 33.6 Hz. The former is close to the calculated first natural frequency and the latter is close to the measured first resonant frequency (35 Hz) of Test Case 3; Support - A - Removed. This might be caused by the fact that the tube is not resting against the supports when in a vertical orientation

Fig. 23. Effect of Tube Orientation on Tube Resonant Frequencies
(ANL Neg. No. 113-75-772)

and, when vibrating, the tube is not always contacting the closely-spaced double supports simultaneously.

Third mode frequency in the neighborhood of 45 Hz was not observed in the experiment. The steady-state harmonic analysis result (see Fig. 18(c)) shows that the displacement responses are higher in Spans 1, 6 and 7. The high response amplitude in Span 1 has a dominant second mode contribution (see Fig. 13), and the high response amplitudes in Spans 6 and 7 are third mode responses. This indicates that the third mode response in the experiment was masked by a strong second mode response. However, the third mode might show a weak contribution in Spans 6 and 7 which are two spans away from the excitation. An additional reason that the third mode frequency was not observed in the experiment can be attributed to the fact that excitation of that mode by excitation forces applied at Span 3 is not very efficient, as can be seen from Fig. 13, noting the small (non-detectable from the curve) modal amplitude at Span 3 for the third mode. By exciting Span 6 in a subsequent test, third mode response was detected as expected.

The added mass associated with the tube-support gap fluid appears to have a negligible effect on system response; the measured frequencies are close to the analytical results. This is most probably the result of the support location being an effective node point and also the large span-length to support-thickness ratios. Upon increasing the exciting force amplitude, the resonant frequency was slightly increased on the order of 1 Hz.

A.2 Displacement Response Amplitude

Sinusoidal excitation at four different input force amplitudes was applied to Span 3 at each of the first three experimentally-determined resonant frequencies.

RMS displacement amplitude at the center of the span experiencing the maximum response was plotted in terms of the total exciting force

amplitude (see Fig. 24). The analytical results based on the steady-state harmonic analysis were imposed on the same figure for comparison. The measured displacement responses in the horizontal orientation were, in general, higher than those in vertical one. This indicates that the energy dissipation is higher in the vertical orientation. It may be that the tube is exposed to more impacting in the vertical orientation (it is better centered in the support hole and less likely to be resting against a support) and that impact damping results in greater energy dissipation than rubbing/sliding damping which might be dominant in the horizontal orientation. The tube responded primarily in excitation direction. Fig. 24 shows that the measured RMS displacement amplitudes are always lower than those of analytical results with linear viscous damping (ξ) of 2%. It is postulated that energy dissipation due to tube-support impacting, sliding and rubbing, accounts for the analytical results providing an upper bound to the displacement responses.

The measured peak displacements along the spans were plotted at each experimentally-observed resonant frequency from the lowest up to the third for total exciting force amplitude of 7.5 lbs (3.4 kg) and analytical results were also presented in the same figure (see Fig. 25). When the tube is excited at its first natural frequency in analysis, maximum displacement occurs at Span 1 (see Fig. 25(a)), and it is dominated by second mode contribution (see Fig. 13). For second mode excitation, Span 1 again experiences the maximum displacement which in this case is about 15 times higher than that of the first mode with $\xi = 2\%$. The experimental results showed similar characteristics. The measured displacement at Span 1 is about 5 times lower than that of analytical results with $\xi = 2\%$. It indicates that the closely-spaced double supports may dampen the response amplitudes of Span 1. For 4th mode excitation, maximum displacements at Spans 1, 6 and 7

Fig. 24. Effect of Tube Orientation (Test Cases 1 and 5): RMS Displacement Amplitude at the Center of the Span Experiencing the highest Response for the First Three Resonant Frequencies as a Function of Total Exciting Force Amplitude (ANL Neg. No. 113-76-809)

Fig. 25. Effect of Tube Orientation (Test Cases 1 and 5): Peak Displacement Amplitude along the Span at 1st, 2nd and 4th mode Excitations for Total Exciting Force Amplitude of 7.5 lbs (3.4 kg) (ANL Neg. No. 113-76-802)

could not be measured because the transducers were positioned at the centers of each span which approximated node locations. Measured displacement responses along the spans are always lower than the analytical results with $\xi = 2\%$.

A.3 Damping

Equivalent viscous modal damping factor varied from 2% to 7.5% for both tube orientation. Its magnitude decreased in higher mode.

B. Tube/Support Misalignment

This experiment was performed with the tube in a horizontal orientation using the reference tube-support configuration (Fig. 1(a)); 38.10 mm (1-1/2 in.) support thickness, 0.254 mm (10 mils) diametral clearance of tube/support-hole, and in-water condition (see Test Case 6 in Table 1). Tube-to-support misalignment was made by off-setting the supports A, D and F against the tube: 1.59 mm (1/16 in.) shim removed from Support A, 1.59 mm (1/16 in.) shim added to Support D, and 3.18 mm (1/8 in.) shim removed from Support H. The tube was no longer loose within the support hole. It was firmly in contact with the supports.

B.1 Resonant Frequencies

The experimental results were compared with those of the aligned case (Test Case 5) and with analytical results (see Fig. 26). Each resonant frequency of misaligned case was a little higher than the corresponding frequencies of the aligned case and the analytical result. This may be caused by the increased fixidity at the support location due to the firm contact. Third mode frequency was not observed in the experiment when exciting Span 3 for the reasons discussed in Section IV.A.1.

B.2 Displacement Response Amplitude

Sinusoidal excitation at three different input force amplitudes was applied to Span 3 at each measured resonant frequency from the lowest

NOTE: N/M = NOT MEASURED

Fig. 26. Effect of Tube/Support Misalignment on Tube Resonant Frequencies
(ANL Neg. No. 113-76-769)

up to 137 Hz. RMS displacement amplitude at the center of each span was plotted in terms of the total exciting force amplitude (see Fig. 27). It showed that the overall displacement response amplitude was negligibly small and magnitude of tube motion in both excitation and normal to excitation is nearly same.

B.3 Damping

The equivalent viscous modal damping factor varied randomly from 1.5% to 5.9%.

C. Fluid Medium

The experiment was performed with the tube in the vertical orientation using a reference tube-support configuration (Fig. 1(a)): 38.10 mm (1-1/2 in.) support thickness, and 0.254 mm (10 mils) diametral clearance of tube/support-hole (see Test Cases 1 and 2 in Table 1). The fluid media used in the tests were air and water.

C.1 Resonant Frequencies

Measured tube resonant frequencies in water were lower than those in air due to the added mass effect of the surrounding dense fluid. Measured frequencies were close to the calculated natural frequencies of a multi-span beam with "knife-edge" supports (see Fig. 28).

For both fluid media, two distinct resonant frequencies were observed in the neighborhood of first mode frequency. Again, third mode frequency was not observed in the experiment when exciting Span 3, for the same reasons discussed in Section IV.A.1. Resonant frequencies were insensitive to the exciting force magnitude for both fluid media.

C.2 Displacement Response Amplitude

Sinusoidal excitation at three to four different input force amplitudes was applied to Span 3 at each resonant frequency from the lowest up to the fourth.

Fig. 27. Effect of Tube/Support Misalignment (Test Case 6): RMS Displacement at the Center of the Span for the First Four Resonant Frequencies as a Function of Total Exciting Force Amplitude (ANL Neg. No. 113-76-811)

NOTE: N/M = NOT MEASURED

Fig. 28. Effect of Fluid Medium on Tube Resonant Frequencies
(ANL Neg. No. 113-76-775)

RMS displacement amplitude at the center of the span experiencing the maximum response was plotted in terms of the total exciting force amplitude and the analytical results were also presented in the same figure for comparison (see Fig. 29). For the spans near the excitation span, the tube motion was primarily in the excitation direction and the displacement amplitude in water was, in general, higher than that in air. The opposite behavior was observed for the spans away from the excitation span. This may be caused by the different level and type of energy dissipation at the tube-support locations due to various combinations of impacting, rubbing and sliding mechanisms. However, the damping phenomena associated with tube/support interaction are not known yet. Analytical results show that the displacement responses are approximately the same for both fluid media. This implies that the added mass of the surrounding water does not appreciably effect the displacement response. Measured displacement amplitudes are always lower than those of analytical results with 2% linear viscous damping.

Measured peak displacement along the spans was plotted at each resonant frequency from the lowest up to the third for total exciting force amplitude of 7.5 lbs (3.4 kg) and the analytical results were also imposed in the same figure (see Fig. 30). Maximum displacement occurred at Span 1 and in-water condition when exciting the tube at third resonant frequency (see Fig. 30(b)). Measured displacement was slightly lower than the analytical results with 8% damping factor. This implied that there may be significant energy dissipation as a result of the closely-spaced double-support.

D. Tube/Support-Hole Clearance

The experiment was performed with the tube in a horizontal orientation, with a water medium, using a reference tube-support configuration (Fig. 1(a)): 38.10 mm (1-1/2 in.) support thickness and three different diametral clearances

Fig. 29. Effect of Fluid Medium (Test Cases 1 and 2): RMS Displacement Amplitude at the Center of the Span Experiencing the Highest Response for the First Four Resonant Frequencies as a Function of Total Exciting Force Amplitude (ANL Neg. No. 113-76-807) - Continue to Next Page -

Fig. 29. - Continued (ANL Neg. No. 113-76-806)

Fig. 30. Effect of Fluid Medium (Test Cases 1 and 2): Peak Displacement Amplitude along the Span at 1st, 2nd and 4th Mode Excitations for Total Exciting Force Amplitude of 7.5 lbs (3.4 kg) (ANL Neg. No. 113-76-801)

of tube/support-hole; 0.254 mm (10 mils), 0.508 mm (20 mils), and 0.762 mm (30 mils). (See Test Cases 5, 8 and 9 in Table 1.)

D.1 Resonant Frequencies

Experimental results showed that the tube/support-hole clearance did not significantly change the tube resonant frequencies. Also, the measured frequencies were close to the calculated natural frequencies of a multi-span beam with "knife-edge" supports (see Fig. 31).

Third mode frequency in the neighborhood of 45 Hz was not observed in the experiment when exciting Span 3, for the reasons discussed in Section IV.A.1. The effect of the added mass due to gap fluid for various tube/support-hole clearances tested appears to have a negligible influence on system response since the measured frequencies are close to the analytical results. Resonant frequencies, for the gap sizes tested, were insensitive to the exciting force magnitude.

D.2 Displacement Response Amplitude

Sinusoidal excitation at three to four different input force amplitudes was applied to Span 3 at each resonant frequency from the lowest up to the third.

RMS displacement amplitude at the center of the span experiencing the maximum response was plotted in terms of the total exciting force amplitude and analytical results were presented in the same figures (see Fig. 32). When exciting the tube at its first and third resonant frequencies, the measured displacement amplitude generally increased with increasing tube/support-hole clearance. However, when exciting the tube at its second resonant frequency, the measured displacement amplitude for the case of 0.762 mm (30 mils) clearance was the lowest. The tube responded primarily in excitation direction for any clearance tested. The measured displacement amplitude for the case of 0.254 mm (10 mils) clearance is always lower than

NOTE : N/M = NOT MEASURED

Fig. 31. Effect of Tube/Support-Hole Clearance on Tube Resonant Frequencies
(ANL Neg. No. 113-76-770)

Fig. 32. Effect of Tube/Support-Hole Clearance (Test Cases 5, 8 and 9): RMS Displacement Amplitude at the Center of the Span Experiencing the Highest Response for the First Three Resonant Frequencies as a Function of Total Exciting Force Amplitude (ANL Neg. No. 113-76-810)

the corresponding analytical result with 2% viscous damping. Contrary to results from previously reported test runs, this was not true for Test Cases 8 and 9 with 0.508 mm (20 mils) and 0.762 mm (30 mils) clearances respectively.

The measured RMS displacement at the center of each span was plotted in terms of the tube/support-hole clearance (see Fig. 33), and the resultant peak displacement along the span length was plotted for a given existing force amplitude (see Fig. 34). The analytical results were also presented in the same figures. As increasing the size of the tube/support-hole clearance, the displacement amplitude did not consistently increase or decrease. This indicates that the energy dissipation due to tube/support interaction is highly non-linear. Maximum displacement occurred at Span 1 for 0.508 mm (20 mils) clearance, when exciting the tube at its second resonant frequency, implying that it has the least energy dissipation due to tube/support interaction. Fig. 34 showed that the measured displacement amplitude for 0.254 mm (10 mils) clearance was always lower than the analytical results.

D.3 Damping

The equivalent viscous modal damping factor varied arbitrarily from 1.5% to 8.5%. Its magnitude decreased for higher mode.

E. Support Thickness

The experiment was performed with the tube in a horizontal orientation, with water medium, using a reference tube-support configuration (Fig. 1(a)): 0.258 mm (10 mils) diametral clearance of tube/support-hole and four different support thicknesses; 12.70 mm (1/2 in.), 19.05 mm (3/4 in.), 38.10 mm (1-1/2 in.), and 50.80 mm (2 in.) (see Test Cases 10, 11, 5, and 12 in Table 1).

Fig. 33. Effect of Tube/Support-Hole Clearance (Test Cases 5, 8 and 9): RMS Displacement Amplitude at the Center of the Span for the First Three Resonant Frequencies as a Function of Tube/Support-Hole Clearance (ANL Neg. No. 113-76-804)

Fig. 34. Effect of Tube/Support-Hole Clearance (Test Cases 5, 8 and 9): Peak Displacement Amplitude along the Span at 1st, 2nd and 4th Mode Excitations for Total Exciting Force Amplitude of 7.5 lbs (3.4 kg) (ANL Neg. No. 113-76-798)

E.1 Resonant Frequencies

Experimental results showed that the support thickness did not significantly change the tube resonant frequencies. Also, the measured frequencies were close to the calculated natural frequencies of a multi-span beam with "knife-edge" supports (see Fig. 35).

The first resonant frequency in the neighborhood of 30 Hz was not observed in the experiment for the case of 50.80 mm (2 in.) support thickness.

Third mode frequency in the neighborhood of 45 Hz was not observed in the experiment when exciting Span 3, for the same reasons discussed in Section IV.A.1. The effect of the added mass due to gap fluid for various support thicknesses tested appears to have a negligible effect on the system response since the measured frequencies are close to the analytical results. Resonant frequencies, for the support thicknesses tested, were insensitive to the exciting force magnitude.

E.2 Displacement Response Amplitude

Sinusoidal excitation at three to four different input force amplitudes was applied to Span 3 at each resonant frequency from the lowest up to the third.

RMS displacement amplitude at the center of the span experiencing the maximum response was plotted in terms of the total exciting force amplitude and the analytical results were presented in the same figure (see Fig. 36). The measured displacement for 12.70 mm (1/2 in.) thick support was, in general, higher than that for other thicker supports. For excitation at first and second resonant frequencies, the displacement amplitude for 19.05 mm (3/4 in.) thick support was the lowest among others. The tube responded primarily in the excitation direction for the span experiencing maximum displacement for each resonant frequency. The measured

NOTE: N/M = NOT MEASURED

Fig. 35. Effect of Tube-Support Thickness on Tube Resonant Frequencies
(ANL Neg. No. 113-76-771)

Fig. 36. Effect of Tube-Support Thickness (Test Cases 5, 10, 11 and 12): RMS Displacement Amplitude at the Center of the Span Experiencing the Highest Response for the First Three Resonant Frequencies as a Function of Total Exciting Force Amplitude (ANL Neg. No. 113-76-808)

displacement amplitude for 12.70 mm (1/2 in.) thick support was slightly higher than the analytical results with 2%, viscous damping. However, analytical results provide a good approximation.

The RMS displacement at the center of each span was plotted in terms of support thickness for a given exciting force amplitude and excitation at the first resonant frequency (see Fig. 37). For the 19.05 mm (3/4 in.) support thickness, the measured displacements, for the spans away from the excitation, in both excitation and normal to excitation directions were approximately the same, resulting in orbital tube motion.

E.3 Damping

The equivalent viscous modal damping factor varied arbitrarily from 1.4% to 7.5%. Its magnitude decreased for higher mode.

F. Support Spacing

The experiment was performed with the tube in a horizontal orientation, with water medium, using five different tube/support configurations (Fig. 1) with 0.258 mm (10 mil) tube/support-hole diametral clearance and 38.10 mm (1-1/2 in.) thick tube-support. To evaluate the effect of closely-spaced double supports on system responses, four different tube-support configurations were employed: support A removed, support B removed, support A moved to H, and supports A, B included. Also, the case of seven equal spans was tested and compared with the results of the other cases (see Test Cases 3, 4, 5, 7 and 13).

F.1 Variation of Double Supports

F.1.a Resonant Frequencies

For each case, the measured resonant frequencies were close to the calculated natural frequencies of a multi-span beam with "knife-edge" supports (see Fig. 38). By moving support A to H, the resonant frequency in the neighborhood of 40 Hz could be eliminated. However, a resonant

Fig. 37. Effect of Tube-Support Thickness (Test Cases 5, 10, 11 and 12): RMS Displacement Amplitude at the Center of the Span for the First Resonant Frequency as a Function of Tube-Support Thickness (ANL Neg. No. 113-76-805)

NOTE: N/M = NOT MEASURED

Fig. 38. Effect of Double-Supports A,B Arrangement on Tube Resonant Frequency
(ANL Neg. No. 113-76-773)

frequency at 95 Hz appeared; it corresponds to the third mode frequency in the analysis (see Fig. 16). The first resonant frequency in "Supports A,B Included" case was lower than that for any other case. For "Support A Removed," "Support B Removed," and "Supports A,B Included" cases, third mode frequency in the neighborhood of 45 Hz was not observed in the experiment, when exciting Span 3, with the same reason discussed in Section IV.A.1.

F.1.b Displacement Response Amplitude

Sinusoidal excitation at three to four different input force amplitudes was applied to Span 3 at each resonant frequency from the lowest up to the third.

RMS displacement amplitude at the center of the span experiencing the maximum response was plotted in terms of the total exciting force amplitude and the analytical results were also presented in the same figure (see Fig. 39). The highest displacement amplitude occurred at Span 1 for "Support-A-Removed" case when exciting the tube at second resonant frequency. By repositioning Support A to H, the displacement amplitude was significantly lower than for any other case. The tube responded primarily in the excitation direction. The measured displacement was not always lower than the analytical result for 2% damping. However, for the highest displacement response, the analytical result with 2% damping was conservative.

For each resonant frequency, from the lowest up to the third, and for a total exciting force amplitude of 7.5 lbs (3.4 kg), the measured peak displacement along the spans in each case was plotted along with the analytical results (see Figs. 40-42). The highest displacement response amplitude occurred at Span 1, the measured displacements were approximately the same for the following cases: "Support A Removed," "Support B Removed" and "Supports A,B In." This implies that the closely-

Fig. 39. Effect of Double-Supports (A, B) Arrangement (Test Cases 3, 4, 5 and 7): RMS Displacement Amplitude at the Center of the Span Experiencing the Highest Response for the First Three Resonant Frequencies as a Function of Total Exciting Force Amplitude (ANL Neg. No. 113-77-4)

Fig. 40. Support - A - Removed (Test Case 3): Peak Displacement Amplitude along the Span at 1st, 2nd and 4th Mode Excitations for Total Exciting Force Amplitude of 7.5 lbs (3.4 kg) (ANL Neg. No. 113-76-797)

Fig. 41. Support - B - Removed (Test Case 7): Peak Displacement Amplitude along the Span at 1st, 2nd and 4th Mode Excitations for Total Exciting Force Amplitude of 7.5 lbs (3.4 kg) (ANL Neg. No. 113-76-800)

Fig. 42. Support - A - Moved-to - H - (Test Case 4): Peak Displacement Amplitude along the Span at 1st, 3rd and 4th Mode Excitations for Total Exciting Force Amplitude of 7.5 lbs (3.4 kg) (ANL Neg. No. 113-76-796)

spaced, double supports do not effectively dampen on the vibration response amplitude of Span 1. By moving Support A to H, the overall vibration response amplitude was significantly reduced.

F.2 Equal Span

F.2.a Resonant Frequencies

For each case the measured frequencies were close to the calculated natural frequencies of a multi-span beam with "knife-edge" supports (see Fig. 43). The measured first resonant frequency was the highest among the 13 cases tested. However, the first seven modes were more closely spaced in occurring in range between 65 Hz and 150 Hz.

F.2.b Displacement Response Amplitude

Sinusoidal excitation at five different input force amplitudes was applied to Span 3 at each resonant frequency from the lowest up to the fourth.

RMS displacement amplitude at the center of each span (except Spans 1 and 3) was plotted in terms of the total exciting force amplitude (see Fig. 44). The tube responded primarily in the excitation direction. The measured displacement response was high when the tube was excited at its first resonant frequency, and its magnitude was close to that of each of the other spans. The maximum displacement amplitude in the equal span case was, in general, higher than that in any of the other cases.

The measured peak displacement along the spans was plotted at each resonant frequency from the lowest up to the fourth for the total exciting force amplitude of 6.85 lbs (3.1 kg) and the analytical results were also presented in the same figure (see Fig. 45). The measured displacement response amplitude was significantly lower than the analytical results with 2% damping. It implies that the energy dissipation at tube/support location was high.

NOTE : N/M = NOT MEASURED

Fig. 43. Tube Resonant Frequencies of Seven Equal Span Case (ANL Neg. No. 113-76-774)

Fig. 44. Seven-Equal-Span (Test Case 13): RMS Displacement Amplitude at the Center of Each Span for the First Four Resonant Frequencies as a Function of Total Exciting Force Amplitude (ANL Neg. No. 113-77-3) - Continue to Next Page -

Fig. 44. - Continued (ANL Neg. No. 113-77-2)

Fig. 45. Seven Equal Spans (Test Case 13): Peak Displacement Amplitude along the Span at First Four Mode Excitations for Total Exciting Force Amplitude of 6.85 lbs (3.1 kg) (ANL Neg. No. 113-76-799)

F.3 Damping

The equivalent viscous modal damping factor varied arbitrarily from 1.9% to 11%. Its magnitude decreased for higher modes.

F.4 Optimum Support Spacing

To maximize the fundamental natural frequency for a given number of supports, the longest span should be next to the tubesheet [14], or supports should be placed near the natural node points--that is, the end spans are 1.25 times as long as the intermediate span [5]. However, this support spacing may be inadequate to minimize the displacement response amplitude. The test results showed that closely-spaced, double supports placed next to the excitation span did not effectively dampen out the vibration amplitude of the long end span next to the tubesheet and also equally-spaced support spacing was not effective to reduce the vibration amplitude. However, the short spans placed at both sides of the excitation span reduced the overall vibration amplitude significantly (Test Case 4, Fig. 42), even if the fundamental frequency was not maximized.

V. SUMMARY OF RESULTS

1. The experimental results showed a small variation of resonant frequencies due to tube orientation, tube/support-hole clearance, support thickness, tube-support spacing and excitation force amplitude. Measured frequencies were close to the calculated natural frequencies assuming a multi-span beam with "knife-edge" supports.
2. Added mass due to the gap fluid between tube and support-hole appeared to have a negligible effect on system response based on the close agreement between measured frequencies and analytical results.
3. Measured displacements of the tube in a horizontal orientation were, in general, higher than those for the tube in a vertical orientation.
4. Measured displacements in the tube/support misaligned case were negligibly small. However, the potential for increased wear at the tube/support contact locations cannot be evaluated.
5. Measured displacements in water were, in general, higher than those in air for the spans nearest the excitation span. The opposite behavior was observed for the spans away from the excitation span.
6. A significant variation in displacement amplitude was observed in the displacement response for the tube/support-hole clearance over 20 mils. The tube/support-hole clearance was found to be the most sensitive parameter for response amplitude. For a small tube/support-hole clearance (say, 10 mils in this test), the measured displacement was always lower than that from analytical results with 2%, viscous damping.
7. Smaller tube/support-hole clearances and thicker supports have a definite tendency to lower displacement amplitudes; this may be the result of an increase in damping.
8. Closely-spaced, double supports placed next to the excitation span did not act effectively to dampen the vibration amplitude of the adjacent long span, fixed at one end to the tubesheet.

9. Short spans placed at both sides of the excitation span reduced the overall displacement amplitude significantly, despite the fact that the lowest resonant frequency is not maximized.
10. Damping data was widely scattered from one test to another. This wide variation may be caused by the dependence of damping energy on the tube-support impacting/sliding/rubbing phenomena and the importance of local tube/support conditions.

VI. DISCUSSIONS

A. Nonlinear Response

As seen in Section IV, the experimental results did not follow exactly the linear response obtained from the idealized model. This is to be expected since vibration of a multi-span tube with motion-limiting supports has non-linear characteristics due to tube-support impacting with local deformation at the tube-supports.

For example, a nonlinear jump phenomenon of response amplitude was observed by applying, in turn, three different amplitudes of sinusoidal current (force) to the drive coils and sweeping slowly the forcing frequency in first an upward and then a downward direction. The seven-equal-span model was used in this experiment (see Fig. 1(e) and Test Case 13 in Table 1). Displacement frequency response is plotted in Fig. 46 for the three cases tested. For the case of a small exciting force [2.1 kg (4.7 lbs)] the impacting was negligible, as detected from the lack of chattering noise coming from the test article, and response curve generated by sweeping down in frequency closely coincided with that obtained by sweeping up. For the higher force levels, in sweeping the forcing frequency slowly in an upward direction, the displacement response amplitude increased rapidly with severe tube-to-support chattering occurring until a frequency was reached at which a small increase in forcing frequency resulted in the displacement response "dropping out." The opposite phenomenon was observed when slowly sweeping the forcing frequency in a downward direction. The frequency difference at the jump was about 1 Hz.

B. Tube Motion

In addition to the parameters studied in Section IV, initial curvature of tube and initial tube/support-hole misalignment associated with manufacturing and assembling practices also affect the tube motion. Measured

SYMBOL	DIRECTION OF FREQUENCY SWEEP
—	UPWARD (→)
- - -	DOWNWARD (←)

Fig. 46. Nonlinear Jump Phenomenon of Response Amplitude: Displacement Response Amplitude Curves for Three Different Exciting Force Amplitudes (ANL Neg. No. 113-76-803)

tube motion normal to the tube axis were plotted for several typical cases in Fig. 47. In general, the tube responded in the excitation direction at the spans nearest the excitation span. However, for the spans away from the excitation span, the tube has a significant motion in the normal to excitation direction. This kind of orbital motion may be the main cause of the fretting wear often observed at the tube/support locations in operating heat exchangers.

C. Damping

Viscous damping factors were calculated from the spectral density curves using the bandwidth (half-power point) method. The resulting values were widely scattered and varied from one test to another. Measured viscous damping factor ranged from 1.7% to 11% as shown in Fig. 48. In spite of the widely scattered damping data, the following observations are considered to be valid:

- (1) The damping factor decreases with higher modes and the first mode damping has the highest magnitude with the widest variation.
- (2) There is a definite trend toward higher damping with increased excitation force amplitude.

The latter can be observed from Fig. 46; the spectral curves for the higher excitation force are wider than that for the lower excitation force, resulting in higher values for effective viscous damping.

The wide scattering in the damping data may be attributed to the use of a linear viscous damping representation for energy dissipation of non-linear system. The damping mechanism at tube/support location remains to be investigated.

D. Repeatability of Test

In setting up the individual tests tube alignment was accomplished "by hand." Sensitivity to this method of alignment was checked by repeating

EXCITATION DIRECTION

(a) $f = 31.2$ Hz
SPAN 1
TEST CASE 5

(b) $f = 41.5$ Hz
SPAN 1
TEST CASE 5

(c) $f = 31.2$ Hz
SPAN 4
TEST CASE 5

(d) $f = 35.0$ Hz
SPAN 6
TEST CASE 2

(e) $f = 35.0$ Hz
SPAN 7
TEST CASE 2

Fig. 47. Typical Tube Motion Normal to the Tube Axis at the Center of the Span (ANL Neg. No. 113-77-1)

Fig. 48. The Range of Measured Viscous Damping Factors for the First Three Resonances (ANL Neg. No. 113-76-768)

several tests to determine the degree of repeatability of tube vibration response test data. These tests were accomplished by disassembling and reassembling the test model and repeating the previous test. In all cases the resonant frequencies and mode shapes were fairly well repeated. For higher displacement response amplitudes (say, over 10 mils) repeatability was also good. However, for lower displacement response amplitudes (say, less than 10 mils), there was disagreement of a factor of $2 \sim 3$. Since the displacements are on the order of mils (a fraction of mm), only slight difference in the tube/support configuration can account for the observed differences in vibration response amplitude.

E. Comparison with Other Results

Several authors [3-5, 11, 13] have experimentally studied the dynamic behavior of a tube with motion-limiting supports. Test models, techniques for data acquisition and reduction, and workmanship varied widely. However, it is possible to make comparisons of resonant frequencies and damping.

E.1 Resonant Frequencies

Based on an eight-equal-span tube model test, Sebald and Nobles [3] found that increasing tube/support-hole clearance resulted in a decrease in resonant frequency. Moretti and Lowery [4, 5] also reported a slight decrease in frequency based on a three-span test model. Shin, et al. [11] observed a similar decrease based on tests with a cantilevered beam with motion-limiting stop at the free end. The present seven-span tube model test revealed slight, random variation of resonant frequencies on the order of maximum 6 percent.

Moretti and Lowery [4, 5] found that increasing tube-support thickness resulted in a slight decrease in resonant frequency, based on the two-span test. The present seven-span tube model test revealed a

slight, random variation of resonant frequencies with a maximum variation of 3 percent.

With regard to horizontal misalignment of tube and supports, Moretti and Lowery [4, 5] observed a slight increase in resonant frequency about $1/2 \sim 1$ Hz based on their two-span test. The present seven-span test results also showed an increase in resonant frequency of about 1 - 6 Hz.

E.2 Damping

Damping, as reported by previous investigators, was usually represented in terms of the logarithmic decrement, determined by simply cutting the power to the drive coils of the excitation device and recording the decay transient from which a measure of the rate of decay of oscillation can be calculated. To characterize damping resulting from tube/support interaction, this conventional method may be inadequate; relatively large damping is expected due to tube/support interaction and simply cutting the power can be expected to result in an initial rapid decay of oscillation amplitude to the point where significant tube/support impacting is minimized with the result that the measured damping approaches that of "linear damping." In the present study, an equivalent viscous damping factor was calculated from the spectral density curve using the bandwidth (half-power point) method by applying white noise input current (force) to the drive coils.

The range of measured logarithmic decrements by several authors [5, 13] was 0.02 - 0.05, which is equivalent to $0.3 \sim 0.8\%$ of viscous damping factor. However, the present experimental results showed somewhat high damping with a wide variation of viscous damping ranging from 1.7% to 11% of viscous damping. This implies that damping energy cannot be properly expressed in terms of viscous damping or logarithmic decrement

for tube/support interaction model and it is largely dependent on the tube-support geometry and tube/support contact phenomenon.

F. Limitations and Application of Test Results

F.1. Limitations

The tests performed and reported herein were intended, not only to explore the dynamic response of CRBRP steam generator tube, but also to provide the basis for useful design guidance to heat exchanger designers. Tube diameter, tube/support-hole clearances, support thicknesses, and tube span lengths used in the experiments were typical of those encountered in heat exchanger design. The main objective of this investigation was to evaluate the overall effects of these parameters on system response, and not to explore local dynamic behavior at the tube/support locations.

F.2. Application

While appreciable scatter was observed in the overall test data, some basic trends were observed and noted. Several useful implications, having practical significance, are summarized below:

Resonant Frequencies. For the tube support configurations tested, tube orientation, and tube/support conditions did not significantly change the tube resonant frequencies and mode shapes. Also, the measured resonant frequencies were close to the calculated natural frequencies assuming a multi-span beam with "knife-edge" supports.

Displacement Responses. In most cases the measured displacement amplitude was lower than the calculated response assuming a multi-span beam with "knife-edge" supports and a damping factor of 2%; the exceptions were the cases with tube/support-hole diametral clearance over 20 mils (0.51 mm). Consequently, if the excitation force is known, an upper bound on response can be estimated for many cases. Displacement response in the horizontal orientation was generally higher than that in the vertical

orientation. The displacement response was the least for the case of tube/support misalignment.

Closely-Spaced Double Supports. The closely-spaced double supports placed next to the excitation span did not effectively dampen out the vibration amplitude of the adjacent span fixed at one end to the tube sheet. The use of closely-spaced, double supports does not appear to be an effective way to isolate a span from responding to excitation.

Tube-Support Spacing. To maximize the fundamental natural frequency for a given number of supports, the supports should be placed in the natural node points [5]. The test results indicate that, to reduce the displacement response amplitude, short spans should be placed at either side of the excitation span even though the fundamental frequency for a given number of spans is not maximized. The reasonable agreement between analytical and experimental results suggests that analysis can be used to evaluate support locations with the objective of establishing an optimum location with respect to a given excitation and configuration.

Damping. As the highly scattered damping data implies, energy dissipation due to tube/support impacting, sliding, and rubbing has a significant effect on the tube vibration responses and, therefore, a better understanding on damping mechanism due to tube/support interaction is needed.

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