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DYNAMIC CHARACTERISTICS OF HEAT EXCHANGER TUBES

MASTER

VIBRATING IN A TUBE SUPPORT PLATE INACTIVE MODE

by

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ABSTRACT

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Tubes in shell-and-tube heat exchangers, including nuclear plant steam generators, derive their support from longitudinally positioned tube support plates (TSPs). Typically there is a clearance between the tube and TSP hole. Depending on design and fabrication tolerances, the tube may or may not contact all of the TSPs. Noncontact results in an inactive TSP which can lead to detrimental flow induced tube vibrations under certain conditions dependent on the resulting tube-TSP interaction dynamics and the fluid excitation forces. The purpose of this study is to investigate the tube-TSP interaction dynamics. Results of an experimental study of damping and natural frequency as functions of tube-TSP diametral clearance and TSP thickness are reported. Calculated values of damping ratio and frequency of a tube vibrating within an inactive TSP are also presented together with a comparison of calculated and experimental quantities.

I. INTRODUCTION

The tubes comprising the tube bundle of a shell and tube heat exchanger derive their support from tube support plates (also called baffle plates; the designations 'tube support plate' and 'baffle' are used interchangeably in this report) positioned along their length. The tube support plates (TSPs) can also serve the function of inducing crossflow, thereby, increasing turbulence and heat transfer, as they direct the shellside flow through the exchanger. To allow for ease of fabrication and, in many cases, to allow

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differential thermal expansion between the tube bundle and shell, there is a clearance between the tube and TSP hole. In particular, the TEMA (Tubular Exchanger Manufacturers Association) Standards states the following with regard to the drilled holes in baffles and tube support plates [1]:

"Where the maximum unsupported tube length is (36) inches or less, or for tubes larger in diameter than 1-1/4 inches O.D., standard tube holes are drilled 1/32" over the O.D. of the tubes. Where the unsupported tube length exceeds 36" for tubes 1-1/4 inches diameter and smaller, standard tube holes are drilled 1/64" over the O.D. of the tubes."

TEMA recommendations with regard to baffle or support plate thickness are dependent on shell diameter and span lengths as indicated in Table 1.

The TEMA Standards have been prepared for general process plant heat exchangers employing segmental or multi-segmental TSPs. Typically, nuclear plant steam generators must be designed for special, usually more severe, service. In such cases, while TEMA Standards might be consulted for general guidance, the TSPs are generally of special design and often thicker than TEMA Standards specify. As examples, the TSP of a PWR (pressurized water reactor) steam generator is 1.91 cm (0.75 in.) thick while the TSP of a recently designed LMFBR (liquid metal fast breeder reactor) plant steam generator is 3.81 cm (1.5 in.) thick.

In the design of heat exchangers to avoid detrimental tube vibrations, knowledge of the tube's vibrational characteristics is crucial. The vibrational characteristics include natural frequencies, modes, and damping. Traditionally natural frequencies and modes have been computed based on the assumption that the TSPs are effective in providing support for the tubes. In particular, it is typically assumed that the TSPs act as knife edge supports

forcing a vibration node at the TSP location. This is a reasonable assumption in many cases, particularly, those designs involving relatively long spans and small clearances [0.0397 cm (0.0156 in.)]. However, for the case of relatively short spans and large clearances [0.0794 cm (0.03125 in.)], the probability of a tube "floating" in a TSP hole is increased. In such a case, the TSP is effectively inactive. Since a tube will generally respond in its lowest mode, it will tend to vibrate within the TSP hole as illustrated schematically in Fig. 1. A recent tube failure in a pressurized water reactor plant steam generator has been attributed to just such a vibration mode.

Obviously, the particular mode of motion under consideration will be amplitude limited by the size of the radial clearance. However, particularly for the case of dense, viscous fluids, the motion will also be influenced by the damping associated with the fluid in the finite length annulus defined by the TSP hole and the tube O.D. This paper reports the results of an experimental study of the effects of TSP thickness, tube-to-TSP hole clearance, and excitation force level, on the effective modal damping factor and natural frequency. The experimental results are also compared with theoretical predictions.

II. TEST DESCRIPTION

A. Test Hardware

The test fixture consists of a steel tube clamped at each end in what can be considered active tube support plates (TSP). A TSP with clearance is centered between the two end support plates. All three TSPs, and the excitation coil and displacement transducers shown schematically in Fig. 2, are mounted to a 3.81 cm (1.5 in.) thick, 10.2 cm (4 in.) wide, by 132 cm

(52 in.) long steel base which provides rigidity. Figure 3 is a photograph of the complete test fixture.

Pairs of displacement transducers (D_1, D_2) and pressure transducers (P_1, P_2) are orthogonally mounted in the center section of the center TSP to measure tube to tube support plate displacement and fluid pressure within the gap. The center tube support plate width can be varied from 0.318 cm (0.125 in.) to 5.08 cm (2 in.) with sections which are designed to be installed or removed without disrupting the tube-TSP radial alignment. The base of the center TSP is designed with adjustments to accurately locate the plate in the desired position.

Excitation coils provide a sinusoidal force to the 1.6 cm (0.628 in.) O.D., 0.277 cm (0.109 in.) wall, by 122 cm (48 in.) long steel tube. Depending on the phase between each set of coils, the first or second tube bending mode can be excited. First mode would be representative of an inactive TSP and second mode of an active TSP; see Fig. 1. The excitation coils provide tube motion in the horizontal (D_2, P_2) direction. Displacement transducers 3 and 4 are used to determine tube response when the tube is excited in the second bending (active TSP) mode.

B. Test Parameters and Conditions

Two series of tests were conducted in air, water, and oil (Table 2). The dimensions in the A series are typical of those used in steam generators of light water and breeder reactors. The B series test dimensions were obtained from the Standards of Tubular Exchanger Manufacturers Association, Mechanical Standards, TEMA Class R Heat Exchangers, Section 5 of Ref. 1, and are representative of heat exchangers used in the process industries (Table 1).

C. Test Procedures

Testing at a given diametral clearance commences with the installation of a combination of TSP segments to form a TSP of the desired width. Alignment of the tube-TSP is accomplished with careful adjustment of the jackscrews at the support plate base, using displacement transducers D_1 and D_2 to measure the position of the tube within the TSP.

After alignment, a small force is applied to the tube. The forcing frequency is varied to allow measurement of the displacement-frequency response of the tube in air.

After obtaining a satisfactory response in air, the vessel is filled with room temperature water [26.6°C (80°F)] and a set of displacement-frequency response curves are plotted (Fig. 4) for the force levels listed in Table 3. The displacement limiting effect of the TSP is readily seen in the higher force level responses of Fig. 4.

The response curves are used to verify the experimental setup (diametral clearance, angular alignment) and to calculate the modal damping ratio in air. In water, swept sine response was measured for force levels similar to those shown in Table 3. At each force level, the frequency was adjusted to the peak on the displacement-frequency response curve and the displacements (D_1, D_2) and pressures (P_1, P_2) recorded and analyzed. The initial series of tests was performed with a 5.08 cm (2 in.) wide TSP. For subsequent tests, TSP segments are removed, and measurements are taken for each of the TSP widths so obtained. The TSP widths which were tested are shown in Table 2.

A series of tests was conducted in #150 oil to study the effect of viscosity on tube-TSP response. Oil temperature, thus viscosity, was varied by either cooling the oil with cold water, or heating it with steam, circulated through several coils of copper tubing placed in the vessel. A

small propeller-type mixer was used to circulate the oil and assure a homogeneous temperature. The effect of temperature on the kinematic viscosity of the oil is shown in Fig. 5.

D. Data Acquisition

Displacements and pressures are analyzed with a H.P. 5451C FFT (Fast Fourier Transform) analyzer and are also recorded on magnetic tape. The majority of the rms (root-mean-square) displacements and pressures were acquired from on-line data reduction.

Modal damping ratios were calculated using the bandwidth method [4] and averaging the values measured at 0.707 and 0.5 of the maximum displacement except at large values where only the 0.707 value was used. Force level is servo controlled by a swept sine data analysis system which maintains a constant current to the excitation coils as frequency is varied.

E. Analysis

Consider a clamped tube with a baffle plate at midspan (Fig. 6). We wish to calculate the natural frequencies and damping values under different conditions.

The equation of motion for the tube is

$$EI \frac{\partial^4 u}{\partial x^4} + [C_a + C_v + \bar{C}_v \delta(\frac{l}{2})] \frac{\partial u}{\partial t} + [m + C_M M_d + M' \delta(\frac{l}{2})] \frac{\partial^2 u}{\partial t^2} = 0, \quad (1)$$

where EI = flexural rigidity, u = tube displacement, x = axial coordinate, t = time, C_a = viscous damping coefficient in air, C_v = viscous damping of infinite fluid, $\bar{C}_v$ = fluid gap damping, C_M = added mass coefficient in infinite fluid, m = tube mass per unit length, M_d = displaced mass of fluid

per unit length, M' = added mass of fluid gap. Equation (1) can be used for different conditions as follow:

(a) Vibration in Air

For a tube vibrating in air, $C_v \approx \bar{C}_v \approx M_d \approx M' \approx 0$, and the equation of motion can be written as

$$EI \frac{\partial^4 u}{\partial x^4} + C_a \frac{\partial u}{\partial t} + m \frac{\partial^2 u}{\partial t^2} = 0 . \quad (2)$$

For a fixed-fixed tube, the orthonormal mode shape is

$$\phi_n(x) = \cosh \lambda_n x - \cos \lambda_n x - \sigma_n (\sinh \lambda_n x - \sin \lambda_n x) ,$$

$$\lambda_1 l = 4.73004 , \quad \sigma_1 = 0.982502 , \quad (3)$$

$$\lambda_2 l = 7.85320 , \quad \sigma_2 = 1.007773 .$$

The natural frequencies and in-air viscous damping coefficient are

$$\omega_n = \frac{(\lambda_n l)^2}{l^2} \left(\frac{EI}{m} \right)^{0.5} \quad (4)$$

$$C_a = 2m\omega_n \zeta_n . \quad (5)$$

(b) Oscillations in Water with No Tube Support

The equation of motion is

$$EI \frac{\partial^4 u}{\partial x^4} + (C_a + C_v) \frac{\partial u}{\partial t} + (m + C_M M_d) \frac{\partial^2 u}{\partial t^2} = 0 . \quad (6)$$

The modal damping and natural frequency can be analyzed in a straightforward manner [Ref. 2] to obtain

$$\omega_n = \frac{(\lambda_n l)^2}{l^2} \left(\frac{EI}{C_M M_d + m} \right)^{0.5} \quad (7)$$

$$\zeta_n = \frac{C_a}{2(m + C_M M_d) \omega_n} - \frac{1}{2} \left[\frac{M_d}{m + C_M M_d} \right] \text{Im}(H) \quad (8)$$

where

$$C_M = \text{Re}(H)$$

$$M = 1 + \frac{4K_1(a)}{aK_0(a)}$$

$$a = (1 + i) \sqrt{\frac{2 \omega_n^2 r^2}{\nu}}$$

r = tube radius

ν = fluid kinetic viscosity .

Note that there are two terms in the damping: one is associated with structural damping and the other is associated with fluid damping.

(c) Oscillations in Fluid with Tube Support

The full equation given in Eq. (1) has to be used in this case. Let

$$u = q(t) \phi_n(x) . \quad (9)$$

Using Eqs. (1) and (9) yields

$$\frac{d^2 q}{dt^2} + 2 \zeta_n \omega_n \frac{dq}{dt} + \omega_n^2 q = 0 \quad (10)$$

where,

$$\omega_n = \frac{(\lambda_n \ell)^2}{\ell^2} \left\{ \frac{EI}{m + C_M M_d + \frac{M' \phi_n^2(\frac{\ell}{2})}{\ell}} \right\}^{0.5} \quad (11)$$

$$\zeta_n = \frac{1}{m + C_M M_d + \frac{M' \phi_n^2(\frac{\ell}{2})}{\ell}} \left\{ \frac{C_a}{2\omega_n} - \frac{M_d}{2} \text{Im}(h) + \frac{\bar{C}_v}{2\ell\omega_n} \right\}. \quad (12)$$

The added mass associated with the gap M' and damping coefficient $\bar{C}_v$ are given by

$$M' = \int_{-L}^L (C - 1) M_d dz \quad (13)$$

$$\bar{C}_v = \int_{-L}^L C'_v dz \quad (14)$$

where $2L$ is the tube support plate width and C and C'_v are given by [Ref. 3]

$$C = \left(\frac{r}{R - r} \right) \text{Re}(H') \left[1 - \frac{\cosh(z/R)}{\cosh(L/R)} \right] \quad (15)$$

$$C'_v = M_d \left(\frac{r}{R - r} \right) \omega_n [-\text{Im}(H')] \left[1 - \frac{\cosh(z/R)}{\cosh(L/R)} \right] \quad (16)$$

Substituting Eq. (15) into (13) and (16) into (14) and performing the integrations yields

$$M' = 2LM_d \left[K \left(\frac{r}{R - r} \right) \text{Re}(H') - 1 \right] \quad (17)$$

$$\bar{C}_v = 2LM_d K \left(\frac{r}{R - r} \right) \omega_n [-\text{Im}(H')] \quad (18)$$

where,

$$K = 1 - \frac{r}{L} \tanh \left(\frac{L}{r} \right)$$

$$H' = \frac{a \sinh a}{(\alpha + a \sinh a - 2 \cosh a)}$$

$$a = (1 + i) \sqrt{\frac{\omega_n (R - r)^2}{2\nu}}$$

Using Eqs. (11) and (12), together with related equations, the natural frequency and modal damping ratio can be calculated.

The values of $\text{Re}[H]$, $-\text{Im}[H]$ and $\text{Re}[H']$, $-\text{Im}[H']$ can be obtained from Figs. 7 and 8.

III. RESULTS FOR A TUBE WITHOUT A CENTER TSP

Tests of the tube without a center TSP were conducted in air and water to determine tube natural frequency and damping. Using displacement transducer 3, the average measured values of frequency and damping were 55.5 Hz and 0.934% in air and 49.0 Hz and 1.11% in water for input force levels of 0.0356 N (0.008 lb) to 1.09 N (0.245 lb) per coil pair.

The first mode natural frequency of the tube in air may be calculated by using Eq. (4). With $n = 1$,

$$\omega_1 = \frac{(\lambda_1 l)^2}{l^2} \left[\frac{EI}{m} \right]^{0.5} \quad (19)$$

The equivalent viscous damping coefficient for the first mode is calculated with Eq. (5),

$$(C_a)_1 = 2\zeta_1 m \omega_1 . \quad (20)$$

In Eqs. (19) and (20), ω_1 is the first mode circular natural frequency, $(\lambda_1 l)$ is the first modal constant, l is the tube length between the end TSPs, E is the bending modulus, I is the bending moment of inertia, m is the mass per unit length, ζ_1 is the first mode damping ratio, and $(C_a)_1$ is the first mode viscous damping coefficient in air. The following values apply to the test set-up:

$$(\lambda_1 l)^2 = 22.37$$

$$l = 121.9 \text{ cm (48 in.)}$$

$$E = 1.857 \times 10^7 \text{ N/cm}^2 \text{ (26.88} \times 10^6 \text{ lb/in.}^2\text{)}$$

$$I = 2.597 \times 10^{-1} \text{ cm}^4 \text{ (6.246} \times 10^{-3} \text{ in.}^4\text{)}$$

$$m = 8.979 \times 10^{-5} \frac{\text{N sec}^2}{\text{cm}} \text{ (1.302} \times 10^{-4} \frac{\text{lb sec}^2}{\text{in.}^2}\text{)}$$

$$\zeta = 0.00934, \text{ average measured value in air.}$$

Using the above values in Eqs. (19) and (20), the calculated value of ω_1 is 348.72 rad/sec or 55.5 Hz and $(C_a)_1$ is $5.848 \times 10^{-4} \frac{\text{N sec}}{\text{cm}}$ ($8.481 \times 10^{-4} \frac{\text{lb sec}}{\text{in.}^2}$).

The first mode natural frequency and damping in water may be calculated using Eqs. (7) and (8).

$$\omega_1 = \frac{(\lambda_1 l)^2}{l^2} \left(\frac{EI}{C_M M_D + m} \right)^{0.5}, \quad (21)$$

and

$$\zeta_1 = \frac{(C_a)_1}{2\omega_1(C_M M_D + m)} - \frac{1}{2} \left(\frac{M_D}{C_M M_D + m} \right) \text{Im}[H], \quad (22)$$

where C_M is the added mass coefficient ($C_M = \text{Re}[H]$), M_D is the mass per unit length of the displaced fluid. $\text{Im}[H]$ and $\text{Re}[H]$ are functions of ω_1 , tube radius r , and fluid kinematic viscosity, ν (see Fig. 7).

The average first mode frequency in water is 49 Hz, or 308 rad/sec. Using this value with $r = 0.798$ cm (0.314 in.) and $\nu = 1.01 \times 10^{-5}$ m²/s (1.57×10^{-2} in.²/sec), the nondimensional number $S(= \omega r^2/\nu)$ from Fig. 7 is calculated to be 1.934×10^4 . From Fig. 7 we obtain

$$\text{Re}[H] = C_M = 1,$$

and

$$-\text{Im}[H] = 0.027.$$

Using $M_D = 1.99 \times 10^{-5}$ N-sec²/cm² (2.89×10^{-5} lb-sec²/in.²), a value of $\omega_1 = 315.45$ rad/sec or $f_1 = 50.21$ Hz and $\zeta_1 = 1.09\%$ is calculated. These values compare favorably with the measured values.

Equations (21) and (22) were also used to calculate the first mode natural frequency and damping for the tube vibrating in oil at various temperatures. The viscosity-temperature relationship of the oil is derived from Fig. 5. Table 4 is a summary of the values of $\text{Re}[H]$ and $-\text{Im}[H]$, determined from Fig. 7 and used in the calculations.

The calculated values of frequency are 48.73 Hz, 48.83 Hz, and 48.99 Hz; the calculated values of damping ratio are 0.0268, 0.0177, and 0.0154, at oil temperatures of 26.7°C (80°F), 48°C (118°F) and 65.6°C (150°F), respectively.

Measured average values of frequency are 49.5 Hz, 50.0 Hz, and 49.6 Hz; the damping ratio values are 2.62%, 2.26%, and 1.93% at oil temperatures of 26.7°C (80°F), 48°C (118°F) and 65.6°C (150°F) respectively.

Considering experimental accuracy, the calculated and measured frequencies compare well but there is a larger variation between the calculated and measured damping ratios. This variation is attributed to the sensitivity of the function $-Im[H]$ (Table 4), which is used to calculate the damping ratio, to viscosity (oil temperature). It was not possible to maintain a constant temperature distribution along the entire length of the tube during tests.

IV. TSP ACTIVE MODE

A TSP active mode (Fig. 1) is simulated by reversing the phase of one set of driving coils and exciting the tube at its second natural frequency. The experimental air and water data presented are for a 3.81 cm (1-1/4 in.) wide TSP with a 0.061 cm (0.024 in.) diametral clearance. The oil data were acquired using a 1.27 cm (0.5 in.) wide TSP with a 0.0457 cm (0.018 in.) diametral clearance. Since the nodal point is centered within the TSP, tube displacement within the TSP is very small, therefore, the air, water, and oil data can be compared.

Theoretical and average measured values of are presented in Table 5. The calculated and measured frequencies and damping ratios are in good agreement, considering the sensitivity of the experimental values to oil temperature.

V. TSP INACTIVE MODE - A SERIES

When the tube vibrates without contacting the center TSP, the condition is defined as an inactive TSP. A series of tests was conducted (Tests A1 to A5 in Table 2) to study the tube response with TSP widths typical of PWR and breeder reactor steam generators. Measured quantities are compared to calculated data to develop confidence in the accuracy of the governing equations.

A. Representative Results: Test A4 in Water

Space limitations precluded presenting detailed data from all the test cases. Therefore, test A4, with a tube-TSP diametral clearance of 0.061 cm (0.024 in.), was selected as a representative case for presentation in detail.

The nonlinear tube-TSP system can be studied with the aid of Fig. 9. Figure 9 shows the force-displacement response, and displacement and pressure time histories for force levels ranging from 0.285 N (0.064 lb), which is below the level required for tube-TSP contact, to a force level of 3.02 N (0.68 lb), which is considerably higher than that required for contact. Displacement and pressure time histories are recorded at frequencies corresponding to the peak, or what would be the peak, of the respective displacement-frequency response curves.

At an input force level of 0.285 N (0.064 lb) per coil pair, the tube response can be considered linear. The displacement time history is sinusoidal and there is only a very small fluid pressure developed in the tube-TSP gap.

When the force level is increased to 0.796 N (0.179 lb), the resulting tube motion brings it very close to the TSP, but without contact. A peak-to-peak pressure of 1.816×10^4 Pa (2.6 psi) is developed in the tube-TSP gap. The positive pressure is developed as the tube is approaching the pressure

transducer (P_2), which is mounted flush with the wall of the TSP. As the tube comes to a stop and changes direction, the pressure decreases and becomes negative.

Increasing the force to 1.48 N (0.332 lb) per coil pair results in tube-TSP contact. The slight "flattening" of the displacement is a result of the motion limiting effect of the TSP. The slight rise in the flat top part of the frequency-displacement response curve can be attributed to a slight misalignment of the tube within the TSP. Peak-to-peak pressure amplitude is now 4.539×10^4 Pa (6.5 psi) with the same basic shape as occurred with a force level of 0.796 N (0.179 lb), but with higher frequency noise between pulses. This high frequency noise can be attributed to the excitation of vibration modes in the TSP base mounting when impacting takes place.

A force input of 2.4 N (0.54 lb) results in a wider top of the frequency-displacement plot. A dwell time, at which the tube remains in contact with the TSP, can clearly be seen on the displacement time history. This dwell time results in a change in the gap pressure in that there are now two pressure pulses per cycle. The positive pressure pulse is the result of the tube approaching the TSP. The pressure then decreases and remains low until the tube travels away from the TSP; this produces a suction effect which results in a negative pressure pulse.

The frequency-displacement response at 3.02 N (0.68 lb) force per coil pair yields a still wider top. The solid line represents the response when the frequency is swept from a low to a high value while the dashed response is for a high to low frequency sweep. From this, the system nonlinearity can clearly be seen. The pressure pulse amplitude is slightly higher than that at 2.4 N (0.54 lb) but its basic shape remains the same. The displacement-time

history remains unchanged except for a slight addition of tube higher mode response.

Measured tube displacement-force response (Test A4) in water at TSP widths ranging from 1.27 cm (0.5 in.) to 5.08 cm (2 in.) with a constant diametral tube-TSP clearance of 0.061 cm (0.024 in.) is shown in Fig. 10. TSP width effect on tube displacement can be readily seen. This is primarily the result of fluid damping in the tube-TSP clearance. Fluid damping increases with both TSP width and excitation force level as shown in Fig. 11. For the same series of tests, the measured frequencies are shown in Fig. 12. Frequency decreases with increasing TSP width but is nearly independent of excitation force level.

Tube dynamics, such as resonant frequency and damping, play an important role in the stability of heat exchanger and steam generator tube arrays [5]. Figures 10, 11, and 12, from Test A4, clearly show the influence of TSP width on tube dynamics. Test A4 is for a constant diametral clearance of 0.061 cm (0.024 in.). Similar results are obtained from tests with other values of clearance.

B. Calculation of Damping and Frequency

Damping and frequency of a tube with an inactive TSP may be calculated with Eqs. (11) and (12).

Equations (11) and (12) were evaluated for a series of tube-TSP diametral clearances and TSP widths. The effect of TSP width on damping and frequency for tube-TSP diametral clearances of 0.0152 cm (0.006 in.) to 0.0813 cm (0.032 in.) is shown in Figs. 13 and 14. From these figures, it can be seen, as one would intuitively expect, that the TSP width has a greater influence on damping and frequency at small diametral clearances.

C. Comparison of Measured and Calculated Damping and Frequency

The series of tests (A1-A5) described above provide data to test the validity of Eqs. (11) and (12). The tests were conducted in air and water.

Since the measured damping ratio is influenced by excitation force, it is plotted (Fig. 15) as a range of values for a small force level up to a force level at which displacement limiting occurs. The variation between the calculated and measured values diminishes as the diametral clearance increases. This is due, in part, to the sensitivity of the experimental measurements to small dimensional changes at small diametral clearances. This can be seen in Fig. 16.

Comparison of the measured and calculated frequencies is shown in Fig. 17. The variation between the measured and calculated frequencies is much less than that of the damping ratios due to the factors $\text{Re}[H]$ and $\text{Re}[H']$ being almost independent of system parameters, such as diametral clearance (2h) and viscosity.

VI. TSP INACTIVE MODE - B SERIES

A. Theoretical Calculations

In this series of tests, the dimensions of TSP width and tube-TSP diametral clearances were chosen to be similar to those of typical process industry heat exchangers as recommended by TEMA [Ref. 1 and Table 1]. Tests were conducted in water and in oil at four different temperatures (viscosities).

Theoretical values of damping are calculated by use of Eq. (12) for room temperature water and oil at 21°C (70°F), 26.7°C (80°F), 48°C (120°F), and 65.5°C (150°F). Kinematic viscosity, $\text{Re}[H]$, $-\text{Im}[H]$, $\text{Re}[H']$, and $-\text{Im}[H']$, are given, for the various temperatures, in Table 6. The calculated damping

ratios for a 0.0397 cm (0.0156 in.) diametral clearance in liquid at various viscosities are shown in Fig. 18. Figure 19 is for a 0.0794 cm (0.03125 in.) diametral clearance and the same liquid viscosities.

From Figs. 18 and 19 the significant effect of the viscosity of the fluid in the tube-TSP can be seen. The damping ratio for a given TSP width is considerably larger with the 0.0397 cm (0.0156 in.) diametral clearance than with the 0.0794 cm (0.03125 in.) diametral clearance.

First mode tube resonant frequency can also be calculated with Eq. (11). However, it is insensitive to the fluid viscosity within the tube-TSP gap. This is due to the fact that $\text{Re}[H']$ remains essentially constant (see Table 6) for the liquid viscosities tested. The only effect of liquid viscosity is that from $\text{Re}[H]$ which is associated with the tube vibrating in an "infinite" fluid.

B. Measured Quantities

The measured damping ratio values for test B2 (0.0794 cm, 0.0313 in. tube-TSP clearance) are shown in Fig. 20. The variation between the measured and calculated damping ratios is the largest for the in-water test. In this case, the variation is attributed to the narrow baffle widths tested and the smaller force excitation levels used. In Eqs. (11) and (12), no provision is made for the "end" effects of the TSP. Therefore with a very narrow TSP, more fluid can flow out of the tube-TSP annulus which results in a lower value of damping ratio. Also, to avoid damage to the tube or narrow TSP, the force level must be limited to a value lower than that used for the A test series. Lower input force will also reduce damping ratios.

For the in-oil tests, the variations are of a random nature. This variation is the result of temperature differences in the oil combined with the "end" effect previously mentioned.

VII. CONCLUDING REMARKS

Theoretical predictions yield damping ratios and frequencies which are, within experimental accuracy, in good agreement with measured quantities. In cases where there is a possibility of designing a steam generator or heat exchanger with TSP inactive supports, prediction of tube response characteristics (damping ratio and frequency) can be useful in determining if a fluidelastic instability could occur and in predicting response to turbulent buffeting. A method of theoretical prediction would also be useful in the following:

- 1) To determine the dynamic response modification of a tube or tube array due to an increase of tube-TSP clearances when chemical cleaning is performed to remove corrosion products in steam generators. Conceivably, in a steam generator, if some initial tube-TSP wear occurred before significant corrosion product buildup, removal of the corrosion product and/or the effects of the cleaning process itself could result in a larger diametral clearance than when the steam generator was first put into service. This larger clearance could increase the possibility of detrimental tube vibrations, for example, by permitting vibration in a TSP inactive mode.

- 2) In some cases, when a tube vibration problem exists in a heat exchanger or steam generator, individual tubes are expanded to reduce tube-TSP clearances. A method to predict resulting tube dynamics would aid in determining the amount of expansion required.

Although the majority of this report relates to system parameters such as tube natural frequencies and damping, it is expected that the pressure measurements may prove useful in correlating tube motion (when impacting) to tube-TSP wear. A large fluctuation of pressure within the tube-TSP annulus would be helpful in "washing" wear particles from within the annulus.

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REFERENCES

1. Standards of Tubular Exchanger Manufacturers Associates, Sixth Edition, 1978.
2. Chen, S. S., Wambsganss, M. W., and Jendrzejczyk, J. A., "Added Mass and Damping of a Vibrating Rod in Confined Viscous Fluids," J. Appl. Mech. 98(2), 325-329 (1976).
3. Mulcahy, T. M., "Fluid Forces on Rods Vibrating in Finite Length Annular Regions," J. Appl. Mech. 47, 234-240 (1980).
4. Structural Damping, ed. J. E. Ruzicka, ASME Publication (1959).
5. Chen, S. S., and Jendrzejczyk, J. A., "Stability of Tube Arrays in Crossflow," Nucl. Eng. Des., 75(3), 351-374 (June 1983).

Table 1. TEMA recommended TSP thickness [1]

Nominal Shell I.D. (in.)	Tube Support Plate Thickness (in.)				
	Distance between adjacent full diameter baffles, supports or the unsupported tube length between other type baffles (in.)				
	24 and under	Over 24 to 36 Inc.	Over 36 to 48 Inc.	Over 48 to 60 Inc.	Over 60
8-14	1/8	3/16	1/4	3/8	3/8
15-28	3/16	1/4	3/8	3/8	1/2
29-38	1/4	5/16	3/8	1/2	5/8
39-60	1/4	3/8	1/2	5/8	5/8

Table 2. Test matrix

Test	Tube-TSP Diametral Clearance	TSP Width	Fluid
A1	0.0203 cm (0.008 in.)	1.27 cm (1/2 in.) 1.91 cm (3/4 in.) 2.54 cm (1 in.) 3.18 cm (1-1/4 in.) 3.81 cm (1-1/2 in.) 4.45 cm (1-3/4 in.) 5.08 cm (2 in.)	Water, Air
A2	0.033 cm (0.013 in.)	Same as A1	Water, Air
A3	0.0457 cm (0.018 in.)	Same as A1	Water, Air, Oil
A4	0.061 cm (0.024 in.)	Same as A1	Water, Air
A5	0.0813 cm (0.032 in.)	Same as A1	Water, Air
B1	0.0397 cm (0.0156 in.)	0.0635 cm (1/4 in.) 0.953 cm (3/8 in.) 1.27 cm (1/2 in.) 1.59 cm (5/8 in.)	Water, Air, Oil
B2	0.0794 cm (0.0313 in.)	0.318 cm (1/8 in.) 0.476 cm (3/16 in.) 0.635 cm (1/4 in.) 0.794 cm (5/16 in.) 0.953 cm (3/8 in.) 1.27 cm (1/2 in.)	Water, Air, Oil

Table 3. Excitation force levels for in-water tests (Fig. 7)

Coil Current/Coil Pair		Force/Coil Pair
F_1	3 amps	0.133 N (0.030 lb)
F_2	4 amps	0.285 N (0.064 lb)
F_3	5 amps	0.49N N (0.112 lb)
F_4	6 amps	0.796 N (0.179 lb)
F_5	7 amps	1.09 N (0.245 lb)
F_6	8 amps	1.48 N (0.332 lb)
F_7	9 amps	1.92 N (0.432 lb)
F_8	10 amps	2.40 N (0.540 lb)
F_9	11 amps	3.02 N (0.680 lb)
F_{10}	12 amps	3.52 N (0.792 lb)

Table 4. Values of $\text{Re}[H]$ and $-\text{Im}[H]$ for a tube vibrating in oil as determined from Fig. 6

Oil Temperature	ν	$\text{Re}[H]$	$-\text{Im}[H]$
26.7°C (80°F)	$2.48 \times 10^{-4} \text{ m}^2/\text{s}$ (0.3845 in. ² /sec)	1.06	0.220
48°C (118°F)	$9.85 \times 10^{-5} \text{ m}^2/\text{s}$ (0.1526 in. ² /sec)	1.04	0.108
65.5°C (150°F)	$4.45 \times 10^{-5} \text{ m}^2/\text{s}$ (0.069 in. ² /sec)	1.01	0.080

Table 5.

Fluid	Calculated		Average Measured	
	f_2 (Hz)	ζ_2 (%)	f_2 (Hz)	ζ_2 (%)
Air	152.9	-	154.2	0.586
Water	138.2	0.667	139.0	0.813
Oil 21°C (69.8°F) $\gamma, \text{m}^2/\text{s} (\text{in.}^2/\text{s})$ $3.62 \times 10^{-4} (0.5616)$	139.1	3.00	136.4	3.63
Oil 26.0°C (78.8°F) $\gamma, \text{m}^2/\text{s} (\text{in.}^2/\text{s})$ $2.48 \times 10^{-4} (0.3845)$	139.5	2.35	137.3	2.65
Oil 47.7°C (118°F) $\gamma, \text{m}^2/\text{s} (\text{in.}^2/\text{s})$ $9.85 \times 10^{-5} (0.1526)$	139.6	1.44	137.0	1.76

Table 6. $\text{Re}[H]$, $-\text{Im}[H]$, $\text{Re}[H']$, and $-\text{Im}[H']$ values in viscous fluids

Fluid	ν	$\text{Re}[H]$	$-\text{Im}[H]$	$\text{Re}[H']$	$-\text{Im}[H']$
Water	$1.0 \times 10^{-6} \text{ m}^2/\text{s}$ (0.00157 in. ² /sec)	1.0	0.027	1.2	1.4
26.7°C (80°F) Oil	$2.48 \times 10^{-4} \text{ m}^2/\text{s}$ (0.3845 in. ² /sec)	1.06	0.22	1.2	160
48°C (120°F) Oil	$9.85 \times 10^{-5} \text{ m}^2/\text{s}$ (0.1526 in. ² /sec)	1.04	0.108	1.2	72
65.5°C (150°F) Oil	$4.46 \times 10^{-5} \text{ m}^2/\text{s}$ (0.0691 in. ² /sec)	1.01	0.08	1.2	43

Figure Captions

- 1 Tube Vibration in TSP Inactive and Active Modes
- 2 Test Fixture
- 3 Photograph of Test Fixture
- 4 Tube Displacement Response in Water for a 0.061 cm (0.024 in.) Diametral Clearance and a 2.54 cm (1.0 in.) Wide TSP
- 5 Kinematic Viscosity Plotted as a Function of Temperature for #150 Oil
- 6 A Fixed-Fixed Tube with a Baffle Plate Support
- 7 Real and Imaginary Values of H for a Cylinder Vibrating in an Infinite Fluid
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BAFFLE PLATE

Fig. 6

Fig 7

Fig 8

FORCE LEVEL

Fig 10

Fig 11

~~Fig 11~~

Fig 12

Fig 14

Fig 19

Fig 18

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Fig 20