VIBRO-IMPACT RESPONSES OF A TUBE
WITH TUBE-BAFFLE INTERACTION

by

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Base Technology

January 1978

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NOMENCLATURE

<u>Symbol</u>	<u>Description</u>
a	Half of the baffle thickness
B	Contact damping constant
b	Half of the width of the surface of contact
$[C]$	System damping matrix in the global coordinate system
$[C_n]$	Element damping matrix in the global coordinate system
$[\tilde{C}_n]$	Element damping matrix in the local coordinate system
$\{D\}, \{\dot{D}\}, \{\ddot{D}\}$	Global displacement, velocity and acceleration vector respectively
D_1, D_2	Diode
E_i	Modulus of elasticity of i^{th} body
e	Natural base
$\{F^{ext}\}$	Applied nodal force vector in the global coordinate system
$\{F^{damp}\}$	Nodal internal damping force vector in the global coordinate system
$\{F^{stiff}\}$	Nodal internal stiffener force vector in the global coordinate system
$\{F^{cd}\}$	Contact damping force vector in the global coordinate system
$\{F^{cs}\}$	Contact spring force vector in the global coordinate system
$\{F^\Sigma\}$	$\{F^{ext}\} - \{F^{cs}\} - \{F^{cd}\}$
F_A, F_B	Piezoelectric force transducers
F_o	Exciting input force magnitude
$F_1 - F_4$	Drive coil assembly
f_o	Exciting input forcing frequency
G	Diametral clearance between the tube and the baffle-hole
$[K]$	Master stiffness matrix in the global coordinate system

NOMENCLATURE (Contd.)

<u>Symbol</u>	<u>Description</u>
$[K_n]$	Element stiffness matrix in the global coordinate system
$[\hat{K}_n]$	Element stiffness matrix in the local coordinate system
k	Contact spring stiffness
k_1	$(1 - \nu_1^2)/\pi E_1$; $(i = 1,2)$
$[L]$	Connectivity matrix
$[L]^T$	Transpose of $[L]$
ℓ	Element length
$[M]$	Master mass matrix in the global coordinate system
$[M_n]$	Element mass matrix in the global coordinate system
$[\hat{M}_n]$	Element mass matrix in the local coordinate system
$[\bar{M}]$	$[M] + \delta\Delta t[C]$
P_s	Contact spring force between tube and baffle
P_d	Contact damping force between tube and baffle
q	Contact pressure between tube and baffle
q_o	Maximum contact pressure between tube and baffle
R_1, R_2	Radius of baffle-hole and tube respectively
$[T]$	Coordinate transformation matrix
$[T]^T$	Transpose of $[T]$
T_1-T_5	Displacement transducers
t	Time
Δt	Time step size
V_e	Elastic wave speed
w	Relative approach between the centers of the two bodies
$\dot{w}$	Relative velocity between the centers of the two bodies
α	Material constant for contact damping
α_1, α_2	Proportional factors for system damping

NOMENCLATURE (Contd.)

<u>Symbol</u>	<u>Description</u>
β, δ	Time integration constants
λ	Contact damping constant
ν_i	Poisson's ratio of i^{th} body
ξ_i	i^{th} mode viscous damping
τ_{max}	Maximum shear stress

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ABSTRACT

The relatively small, inherent tube-to-baffle hole clearances associated with manufacturing tolerances in heat exchangers affect the vibrational characteristics and the response of the tube. Numerical studies were made to predict the vibro-impact response of a tube with tube-baffle interaction. The finite element method has been employed with a non-linear elastic contact spring-dashpot to model the effect of the relative approach between the tube and the baffle plate. The coupled equations of motion are directly integrated with a proportional system damping represented by a linear combination of mass and stiffness. Lumped mass approach with explicit time integration scheme was found to be a suitable choice for tube-baffle impacting analysis. Fourier analyses indicate that the higher mode contributions to the tube response are significant for strong tube-baffle impacting. The contact damping forces are negligible compared with the contact spring forces. The numerical analysis results are in reasonably good agreement with those of the experiments.

I. INTRODUCTION

Operating experience [1-3]^{*} with nuclear-power-plant heat exchangers indicates that many units have incurred extensive damage due to excessive flow-induced tube vibrations. One of the highly vulnerable locations for damage due to vibration was observed to be the tube-baffle interfaces. There exist relatively small inherent tube-to-baffle hole clearances associated with manufacturing tolerances and to facilitate the thermal expansion of the tube. This clearance affects the vibrational characteristics and response of the tube; for example, interaction with the baffle plates changes tube vibrational frequencies and response amplitudes [4,5]. Also, impacting causes dilation of the hole in the baffle plate. Prototype or scale model testing is often employed to verify that a given design is free from damage due to vibrations. However, prototype testing can be very expensive. Also, scale model testing may not provide satisfactory answers because of nonlinear response behavior which results in scaling laws and similitude requirements not being directly applicable. Therefore, adequate prediction methods for the dynamic response of a beam with motion-limiting stops are sought [6-8].

The objective of this study is to develop a numerical technique capable of predicting tube-baffle contact (spring/damping) forces, and the vibrational response of a tube including the effects of tube-baffle interaction. The numerical results are compared with those of experiments to evaluate the approach.

Several authors [6,7] applied a finite element method to analyze the dynamic behavior associated component/support interaction. Bohm and Nahavandi [6] studied the impact between components in reactor internal

^{*}Numbers in the brackets correspond to literature cited in the References.

plane structures, modeling contact stiffness as a linear elastic spring. They used a direct forward explicit integration of the coupled equations of motion with a lumped mass matrix approach. Rogers and Pick [7] employed modal superposition with a consistent mass matrix approach. They modeled the radial contact stiffness as a linear elastic spring, and used the Coulomb model for the frictional force between tube and support. For tube-support impacting, the contact stiffness assumed affects the vibrational characteristics significantly. For a heat exchanger tube supported by baffle plates with circular holes, relative deformation between the tube and the baffle plate has a non-linear relationship with the contact forces, and a linear spring model for contact stiffness is no longer valid.

In this study, the finite element method has been employed with a non-linear elastic contact spring-dashpot to model the effect of the relative approach between the tube and the baffle plate. It is assumed that the tube strikes the annulus uniformly over the contact area. This assumption is quite reasonable as long as the tube span is much larger than the baffle thickness. The coupled equations of motion are directly integrated using a central difference method with proportional system damping represented by a linear combination of mass and stiffness.

The experiments were performed using a 39-inch long Croloy (2-1/4% Cr - 1% Mo) tube with a baffle plate placed at the center of the tube span. The numerical results are in reasonably good agreement with those of the experiments.

II. NUMERICAL ANALYSIS

A. Formulation of Problem

The governing differential equations of motion for a finite element model can be expressed in the matrix form

$$[M]\{\ddot{D}\} = \{F^{ext}\} - \{F^{damp}\} - \{F^{stiff}\} \quad (1)$$

where $\{F^{ext}\}$ are the applied nodal forces, $\{F^{damp}\}$ the nodal internal damping forces, $\{F^{stiff}\}$ the nodal internal stiffener forces, $[M]$ the mass matrix and $\{D\}$ the nodal displacement vector; superscript dots denote time derivatives. All arrays in Eq. (1) are based on the global coordinate system. $\{F^{damp}\}$ and $\{F^{stiff}\}$ are expressed by

$$\begin{aligned} \{F^{damp}\} &= [C]\{\dot{D}\} + \{F^{cd}\} \\ \{F^{stiff}\} &= [K]\{D\} + \{F^{cs}\} \end{aligned} \quad (2)$$

where $[C]$ is the system damping matrix, $[K]$ the stiffness matrix, $\{F^{cd}\}$ the contact damping forces exerted on the tube when it strikes the baffles and $\{F^{cs}\}$ the contact spring forces exerted on the tube due to the relative approach between the tube and baffle at contact. All arrays in Eq. (2) are based on the global coordinate system.

$[M]$, $[C]$ and $[K]$ are expressed by

$$\begin{aligned} [M] &= \sum_n [L]^T [M_n] [L] \\ [C] &= \sum_n [L]^T [C_n] [L] \\ [K] &= \sum_n [L]^T [K_n] [L] \end{aligned} \quad (3)$$

where $[M_n]$ is the element mass matrix, $[C_n]$ the element damping matrix, $[K_n]$ the element stiffness matrix and these arrays are also based on the global coordinate system. $[L]$ is the connectivity matrix and subscript n denotes element number.

The element matrices $[M_n]$, $[C_n]$ and $[K_n]$ are also expressed by

$$\begin{aligned} [M_n] &= [T]^T [\hat{M}_n] [T] \\ [C_n] &= [T]^T [\hat{C}_n] [T] \\ [K_n] &= [T]^T [\hat{K}_n] [T] \end{aligned} \quad (4)$$

where $[\hat{M}_n]$, $[\hat{C}_n]$ and $[\hat{K}_n]$ are (12 x 12) element mass, element damping and element stiffness matrices in element local coordinates respectively. $[T]$ is the coordinate transformation matrix from the local coordinates to the global coordinates. The beam element has two nodes and each node has three translational and three rotational degrees of freedom. Rotary inertia and shear deformation effects are incorporated in the element mass and stiffness matrices [9].

Two types of mass matrices are used: consistent, non-diagonal mass matrix and lumped, diagonal mass matrix. Consistent mass matrices usually overestimate the frequencies, whereas lumped masses underestimate the frequencies [11]. The errors introduced by the mass matrix may compensate or amplify the errors due to the time integration operator.

The exact form of the damping matrix is not known for most structures. In this study, it is assumed that the system damping is proportional to the mass and stiffness matrix, which is general, and is given by

$$\begin{aligned} [\hat{C}_n] &= \alpha_1 [\hat{M}_n] + \alpha_2 [\hat{K}_n] \\ \text{and } [C_n] &= \alpha_1 [M_n] + \alpha_2 [K_n] \\ \text{and } [C] &= \alpha_1 [M] + \alpha_2 [K] \end{aligned} \quad (5)$$

where α_1 and α_2 are constants. For a linear elastic analysis, α_1 and α_2 can be expressed in terms of the modal damping factor [10],

$$\xi_1 = \frac{\alpha_1 T_1}{4\pi} + \frac{\alpha_2 \pi}{T_1} \quad (6)$$

where ξ_i is the i th mode damping factor and T_i is i th mode period. Hence, α_1 and α_2 can be expressed in terms of modal damping factor at two different periods [10],

$$\alpha_1 = \frac{4\pi(T_j \xi_j - T_i \xi_i)}{T_j^2 - T_i^2} \quad (7)$$

$$\alpha_2 = \frac{T_i T_j (T_j \xi_i - T_i \xi_j)}{\pi(T_j^2 - T_i^2)} .$$

B. Tube-Baffle Contact Behavior

When the tube strikes the baffle, forces are exerted on the tube. In this study, only the normal impact forces between the tube and the baffle are considered (see Fig. 1). The normal contact forces are represented by a nonlinear elastic contact spring and a nonlinear elastic contact damper. It is assumed that the tube contacts the baffle uniformly over the entire area of contact. This assumption is valid as long as the baffle spacing is much larger than the baffle plate thickness. The contact deformation used in this study is based on the classical Hertz contact theory: this implies that the area of contact is small compared to the areas of two bodies; the area of contact is plane; and the material has a linear, stress-strain relationship represented by the static modulus of elasticity [12].

B.1 Contact Spring

Timoshenko [13] showed that the relative approach between the centers of the two bodies is the sum of the local deformations of the bodies, evaluated at the center of the contact area and is expressed by

$$w = (k_1 + k_2) \iint_A \frac{q(x,y)}{r} dA \quad (8)$$

Fig. 1. Tube-Baffle Contact Configuration

$$k_i = \frac{1 - \nu_i^2}{\pi E_i} ; \quad i = 1, 2 \quad (9)$$

where w is the relative approach between two bodies, $q(x,y)$ the contact pressure between two bodies at the point (x,y) , r the center of the contact area to the point (x,y) , ν_i Poisson's ratio and E_i the modulus of elasticity. Integration is performed over the entire contact area. Dubowsky [12] later derived the force-displacement relationship for an elastic pin in a circular hole using equations (8) and (9). This expression is applicable to a tube in a circular baffle plate hole if the effect of tube ovalization is neglected. It is expressed by

$$w = \left(\frac{k_1 + k_2}{2a} \right) P_s \left\{ \ln \left[\frac{8a^3 e(R_1 - R_2)}{R_1 R_2 (k_1 + k_2)} \right] - \ln P_s \right\} \quad (10)$$

where a is half the baffle thickness, P_s the contact force, R_1 and R_2 the radius of the baffle hole and the tube respectively, k_1 and k_2 are the constants shown in equation (9), subscript 1 is for the baffle plate and 2 for the tube, and e the natural base (see Fig. 1).

The surface of contact between the tube and the baffle is a narrow rectangle, and the distribution of pressure q along the width $2b$ of the surface of contact is represented by a semi-ellipse [13]. Half of the width of the surface of contact [13] is given by

$$b = \sqrt{\frac{2P_s (k_1 + k_2) R_1 R_2}{a(R_1 - R_2)}} \quad (11)$$

The maximum pressure q_o is at the center of the ellipse [13],

$$q_o = \frac{P_s}{\pi ab} \quad (12)$$

and the pressure $q(x,y)$ is expressed by [13]

$$q(x,y) = q_o \sqrt{1 - \frac{y^2}{b^2}} \quad (13)$$

The maximum shear stress $\tau_{\max}$ occurs at $0.78 b$ below the surface of contact [13]

$$\tau_{\max} = 0.304 q_0 \quad (14)$$

Equations (10) through (14) are used to calculate the contact spring forces-contact deformations.

B.2 Contact Damping

During contact between the tube and the baffle, some amount of energy is dissipated. The energy loss can be represented by the (hatched) area of a hysteresis loop as shown in Fig. 2. Hunt and Crossley [14] expressed the contact damping force (P_d) for the discrete mass impacting the fixed body as

$$P_d = \lambda w^B \dot{w} \quad (15)$$

where λ and B are constants and the superscript dot denotes a time derivative. Equation (15) satisfies the boundary conditions such that the contact damping force is zero at $w = 0$ and $w = w_0$ ($\dot{w}_0 = 0$) where w_0 is the maximum relative approach. Equating the loss of the kinetic energy of the moving body over a single-impact sequence to the hysteretic energy loss represented by the hatched area of a hysteresis loop (Fig. 2), and interpreting the coefficient of restitution as damping, λ is expressed by

$$\lambda = \frac{3}{2} \alpha k \quad (16)$$

where k is the contact spring stiffness and α the material constant. For steel, bronze or ivory, α has a value between $0.08 \sim 0.32$ sec/m ($0.002 \sim 0.008$ sec/in.). Substituting equation (16) into equation (15),

$$P_d = \frac{3}{2} \alpha k w^B \dot{w} = \left(\frac{3}{2} \alpha P_s \right) \dot{w} \quad (17)$$

where $k w^B$ is the contact spring force P_s .

Fig. 2. Hysteresis Loop for Contact Damping

Equation (17) is used to calculate the contact damping forces.

C. Solution Procedures

To obtain the solution of the governing equations of motion shown in equation (1), a numerical integration technique is used. The step-by-step integration method can be applied to either the coupled equations of motion as is Eq. (1) or the uncoupled equations of motion after modal superposition. For a linear system where the dominant frequencies are the first few modal frequencies, modal superposition is useful to reduce the number of equations to be solved. However, for a tube/baffle impact problem, the frequency content of the response may be broad and the high frequency response at impacting may also be significant. The direct integration of the coupled equations of motion enables to include all of the frequencies and is therefore more appropriate to the tube/baffle impact problem.

There are two basic methods for the direct step-by-step integration of equations of motion: implicit and explicit. Belytschko [11] has summarized these direct integration methods along with their advantages, disadvantages, and stability criteria. In the implicit method, the equations of motion are combined with the temporal integration developed from difference formulas so that displacements are calculated directly. In the explicit method, the accelerations are found from the equations of motion and then integrated to obtain the displacements. The implicit method permits larger time steps while the explicit method is restricted to small time steps by numerical stability requirements.

One of the commonly used implicit methods is Newmark's β -method [15] which uses the following difference formulas for temporal integration,

$$\{\dot{D}(t + \Delta t)\} = \{\dot{D}(t)\} + [(1 - \delta) \{\ddot{D}(t)\} + \delta \{\ddot{D}(t + \Delta t)\}] \Delta t \quad (18a)$$

$$\{D(t + \Delta t)\} = \{D(t)\} + \{\dot{D}(t)\} \Delta t + [(1/2 - \beta)\{\ddot{D}(t)\} + \beta\{\ddot{D}(t + \Delta t)\}] \Delta t^2 \quad (18b)$$

where δ and β are constants, Δt the time step size, and $D(t)$, $\dot{D}(t)$ and $\ddot{D}(t)$ are displacement, velocity, and acceleration at time t respectively.

Substituting equation (2) into equation (1), the equations of motion at time $t + \Delta t$ are expressed by,

$$[M] \{\ddot{D}(t + \Delta t)\} + [C] \{\dot{D}(t + \Delta t)\} + [K] [\{D(t)\} + \{\Delta D\}] = \{\Sigma F(t + \Delta t)\} \quad (19)$$

where

$$\{\Delta D\} = \{D(t + \Delta t)\} - \{D(t)\} \text{ and } \{\Sigma F(t + \Delta t)\} = \{F(t + \Delta t)\}_{ext} - \{F(t + \Delta t)\}_{cs} - \{F(t + \Delta t)\}_{cd}$$

Substituting equation (18a) into equation (19) and rearranging terms,

$$[\bar{M}] \{\ddot{D}(t + \Delta t)\} = \{\Sigma F(t + \Delta t)\} - [C] \{\dot{D}(t)\} - (1 - \delta) \Delta t [C] \{\ddot{D}(t)\} - [K] (\{D(t)\} + \{\Delta D\}) \quad (20)$$

where

$$[\bar{M}] = [M] + \delta \Delta t [C]$$

Premultiplying equation (18b) by $[\bar{M}]$, substituting equation (20) into equation (18b) and using the approximation $\{F(t + \Delta t)\}_{cd} \approx \{F(t)\}_{cd}$ and $\{F(t + \Delta t)\}_{cs} \approx \{F(t)\}_{cs}$, obtains

$$[[\bar{M}] + \beta \Delta t^2 [K]] \{\Delta D\} = \Delta t [\bar{M}] \{\dot{D}(t)\} + \Delta t^2 [(1/2 - \beta) [\bar{M}] - \beta(1 - \delta) \Delta t [C]] \{\ddot{D}(t)\} + \beta \Delta t^2 [\{F(t + \Delta t)\}_{ext} - \{F(t)\}_{damp} - \{F(t)\}_{stiff}] \quad (21)$$

For the tube/baffle impact problem, the duration of the contact between the tube and the baffle plate (assuming both are metals) is very small and the time step size must be smaller than the duration of the contact.

However, if the time step size is too small, Δt^2 terms in equation (20)

becomes very small and the effect of the system stiffness $[K]$ on the

calculation of the incremental displacement ΔD is negligible. This causes

numerical errors and unstable numerical results. Therefore, the implicit method is not used in this study.

There are many explicit methods for time integration. Among these, the central difference formulas are most often used. Their stability and solution properties are described in reference [16]. These difference formulas are,

$$\{\dot{D}(t + \frac{\Delta t}{2})\} = \{\dot{D}(t - \frac{\Delta t}{2})\} + \{\ddot{D}(t)\} \quad (22a)$$

$$\{D(t + \Delta t)\} = \{D(t)\} + \Delta t \{\dot{D}(t + \frac{\Delta t}{2})\} \quad (22b)$$

Newmark's β -formulas (equation 18) with $\delta = 1/2$ and $\beta = 0$ are equivalent to the central difference formulas provided that $\{\dot{D}(t + \frac{\Delta t}{2})\} = \{\dot{D}(t)\} + \frac{\Delta t}{2} \{\ddot{D}(t)\}$. This method is conditionally stable and the time step must be bounded by [11],

$$\Delta t^2 \leq \frac{\ell}{V_e} \quad (23)$$

where ℓ is the element length and V_e the elastic wave speed. However, for the tube/baffle impact problem, the requirement of a time step smaller than the duration of the contact usually satisfies equation (23). If the time step is not small enough, the numerical results grow very rapidly and reach the overflow limits on the computer (usually within 50 time steps). Consequently, numerical instability is easily recognized.

The calculational procedure for the explicit integration method is as follows,

- (a) calculate the acceleration at time t using equation (1);

$$\{\ddot{D}(t)\} = [M]^{-1} \left[\begin{matrix} \text{ext} & \text{damp} & \text{stiff} \\ \{F(t)\} & - \{F(t)\} & - \{F(t)\} \end{matrix} \right]$$

- (b) calculate the velocity $\{\dot{D}(t + \frac{\Delta t}{2})\}$ using equation (22a).

- (c) calculate the displacement $\{D(t + \Delta t)\}$ using equation (22b).

The explicit method requires less calculational effort per time step than the implicit method and it is simple to implement in the computer program.

The global stiffness matrix $[k]$ and the global damping matrix $[C]$ are not assembled, but the internal stiffner and damping forces are calculated for each element at each time step and summed to calculate the nodal internal forces. This approach significantly reduces computer storage requirements. This kind of technique was used by Bohn and Nahavandi [6]. Some numerical studies by Krieg and Key [17] showed that the errors introduced by the lamped mass-explicit method are cummulative. For the tube/baffle impact studies, two different mass representations are combined with the explicit time integration method to study the sensitivity of the numerical solutions. Nodal displacements, velocities, accelerations, $[F]^{stiff}$, $[F]^{damp}$ at each time step and the corresponding Forier transforms are calculated.

III. EXPERIMENTAL METHOD

A. Experimental Apparatus and Measurement Technique

The test model features a 15.875 mm (0.625 in.) outer diameter x 3.175 mm (0.125 in.) thick x 99.06 cm (39 in.) long seamless Corloy (2-1/4% Cr - 1% Mo) tube, two sets of 76.20 mm (3 in.) thick brass tubesheets, a 12.70 mm (2 in.) thick stainless steel plate and two 12.70 mm (1/2 in.) diameter impact transducers that also serve as motion-limiting stops. The experiments were performed in air and water at room temperature. The tube was vertically mounted and securely clamped at both ends (see Figs. 3-5). As motion-limiting stops, either the baffle plate or impact transducers are placed at the midpoint of the tube (see Fig. 6). The diametral clearance between the tube and the baffle plate annulus is 0.254 mm (10 mils.) or one side tube-to-impact transducer gap is 0.127 mm (5 mils.). For the in-water tests, the test model was enclosed by 19.05 mm (3/4 in) thick Epoxy-glass (see Fig. 5). 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 Figs. 3 and 4) mounted along the length of the tube. The responses of the tube was measured by five displacement transducers mounted along the tube span, and the impact transducers mounted at the midpoint of the tube span measured the impact forces (see Figs. 3 and 4). The impact transducer stiffness itself is 1.75×10^{10} N/M (10^8 lbf/in). However, the stiffness of transducer assembly including effect of support frame is found to be approximately 1.75×10^6 N/M (10^4 lbf/in).

Fig. 3. Test Model Configuration

Fig. 4. Test Model without Encasement

Fig. 5. Test Model with Encasement

Fig. 6. Details of Impact Transducers, Displacement Transducers and Drive Coils

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 500 Hz.

B. Force Calibration

The force calibration apparatus features a 20.32 cm (8 in.) long Croloy tube fastened to two rigid supports with two piezoelectric force transducers and a pair of drive coil placed at the center of the tube (see Fig. 7). The coils were then driven at current levels from 1 to 10 Amp. at 80 Hz (see Fig. 8). Diodes D_1 and D_2 were used to control the current supplied to each coil. This enables one coil to exert an attractive force on the tube when the other is turned off. Thus producing a net sine force which is measured by force transducers F_A and F_B . The total forces exerted on the tube is the summation of two transducer forces. Fig. 9 shows the force-current relationship which is used to calculate the forces applied to the tube.

Fig. 7. Force Calibration Apparatus

Fig. 8. Exciting Force Measurement Scheme

Fig. 9. Exciting Force-Current Relation of One Pair of Drive Coils

IV. NUMERICAL RESULTS

A. Description of a Numerical Model

To allow for comparison with experiments, the geometrical and material properties of the numerical model, given in Table 1, are taken to be same as those of the experimental model shown in Fig. 3. The sample model is shown in Fig. 10. The constants α_1 and α_2 for the proportional system damping (see Eqs. 5-7) are calculated based on 0.5% viscous damping of the 0.99 m (39 in.) long Croloy tube with both ends clamped. The motion-limiting stop is either a 12.70 mm (1/2 in.) thick plate or 1.75×10^6 N/M (10^4 lbf/in) stiff linear spring placed symmetrically about the tube axis at the midpoint of the tube span. The tube is excited by the sinusoidal input forces at the first natural frequency of the tube with in-air conditions. For the in-water condition, the added mass due to the surrounding water along the length of the tube is included, but the fluid viscosity and the added mass due to the gap fluid between the tube and the baffle-hole are neglected because, when the baffle thickness is much smaller than the length of the tube span, the gap fluid has a negligible effect on system response [5]. Two different loading conditions are used (see Fig. 10) loading 1 = Forces are applied at four nodes 3, 4, 6, and 7 with in-air condition, and loading 2 = Forces are applied at two nodes, 3 and 4 with in-water condition.

The first four mode shapes and the corresponding natural frequencies of fixed-fixed end tubes are shown in Fig. 11.

Table 1. Numerical Model Properties

Description		Properties	
Tube	Modulus of Elasticity	2.0×10^{11} Pa	29×10^6 psi
	Poisson's Ratio	0.3	0.3
	Weight Density	7.74×10^3 kg/m ³	483 lbs/ft ³
	Outer Diameter	15.875 mm	0.625 in.
	Wall Thickness	2.921 mm	0.115 in.
	Overall Length	0.991 m	39 in.
	Proportional Damping, α_1	4.0	4.0
	α_2	0.5×10^{-5}	0.5×10^{-5}
Baffle	Modulus of Elasticity	7.74×10^3 kg/m ³	483 lbs/ft ³
	Poisson's Ratio	2×10^{11} PA	29×10^6 psi
	Weight Density	0.3	0.3
	Thickness	12.70 mm	0.5 in.
	Material Constant	0.20 sec/m	0.005 sec/in.
	Linear Spring Stiffness if used instead of Baffle Plate	1.75×10^6 N/m	10^4 lbf/in.
Misc.	Diametral Clearance between the Tube and Baffle-Hole (or Overall Clearance between the Tube and the Linear Spring)	0.254 mm	10 mils
	Weight Density of the Surrounding Water	10^3 kg/m ³	62.45 lbs/ft ³

Fig. 10. Numerical Model and Loading Conditions

Fig. 11. The First Four Mode Shapes and Natural Frequencies

B. Tube-Baffle Contact Spring Forces vs. Relative Approaches

The tube-baffle contact spring forces are plotted in terms of the relative approaches between the tube and the baffle plate for different tube-baffle hole clearances and different baffle thickness using equation 10 (see Figs. 12 and 13). It shows that the contact spring stiffness is approximately linear in the logarithmic scale. The increase in tube-baffle hole clearance decreases the contact stiffness and the increase in baffle thickness increases the contact stiffness. Since Hertz contact theory is used, the area of contact must be small compared to the areas of two bodies and the 5% limit of contact area is plotted in the same figures using equation 11. For example, for 5% limit, the maximum allowable contact spring force is 900 N with maximum relative approach, 0.012 mm for 12.70 mm thick baffle and 0.254 mm tube-baffle hole clearance.

C. Consistent Mass vs. Lumped Mass Approaches

Two different mass matrix representations: consistent and lumped, are combined with the central difference time integration to determine whether the errors introduced by the lumped mass-explicit method are compensatory or those of the consistent mass-explicit method are cumulative for tube-baffle impacting problems.

The tube is excited by sinusoidal input forces applied at nodes 3, 4, 6 and 7 along the length of the tube with in-air condition (see Loading 1, in Fig. 10). The forcing amplitude at each node is 2.85 N (0.64 lbf) and the forcing frequency is the fundamental frequency (84 Hz) of a tube with both ends clamped. Linear springs are assumed for motion-limiting stops.

Fig. 12. Tube-Baffle Contact Spring Forces vs. Relative Approaches for Different Tube-Baffle Hole Clearances

Fig. 13. Tube-Baffle Contact Spring Forces vs. Relative Approaches for Different Baffle Plate Thicknesses

Displacement, velocity, acceleration and shear force time histories at node 5 (tube/stop location) and their Fourier transforms are plotted for lumped and consistent mass approaches in Figs. 14-17. The displacement and the velocity time histories and the corresponding Fourier transforms are nearly identical between the two mass representations (see Figs. 14 and 15). For acceleration and shear force response (see Figs. 16 and 17), time histories are similar between the two different mass representations and the Fourier transforms show that the dominant frequency is in the neighborhood of third mode frequency. The consistent mass case shows a broader frequency band in the neighborhood of third mode frequency. However, the lumped mass case shows a predominant peak at 425 Hz. The mean square response at each dominant frequency range is nearly identical between the two mass representations. One of the advantages of using the lumped mass approach is less computational time than the consistent mass approach because the lumped mass matrix is a diagonal matrix and its inverse is simply the inverse of the diagonal element. Therefore, the lumped mass approach is used for all of the following examples.

D. Tube Vibration Response with In-Air Condition

The sinusoidal input forces are applied at four nodes 3, 4, 6 and 7 along the length of the tube (see Loading 1 in Fig. 10). The motion-limiting stop is assumed to be either a baffle plate or a linear elastic spring.

D.1 Linear Elastic Spring Stops

Tube vibration response is plotted in Figs. 18 through 20 for strong ($F_o = 2.85$ N) and mild ($F_o = 1.43$ N) tube/stop impacts. The displacement time histories at Node 5 (tube-stop location) and Node 6 (input force location) are plotted for strong impact in Fig. 18; as the tube strikes the stop, the effective vibration frequency increases and changes the

Fig. 14. Displacement Time Histories at Node 5 and Their Fourier Transforms for Lumped and Consistent Mass Approaches

(a) LUMPED MASS APPROACH

(b) CONSISTENT MASS APPROACH

Fig. 15. Velocity Time Histories at Node 5 and Their Fourier Transforms for Lumped and Consistent Mass Approaches

Fig. 16. Acceleration Time Histories at Node 5 and Their Fourier Transforms for Lumped and Consistent Mass Approaches

Fig. 17. Internal Shear Force Time Histories at Node 5 and Their Fourier Transforms for Lumped and Consistent Mass Approaches

(a) NODE 6

(b) NODE 5

Fig. 18. Displacement Time Histories at Nodes 5 and 6 with Linear Spring Stops and In-Air Condition (Loading 1: $f_0 = 84$ Hz and $F_0 = 2.85$ N)

phase with respect to the input force. The tube vibrates in a fixed-fixed end beam mode until the displacement response regains its phase relationship to the input force, resulting in increased displacement, causing the tube to again strike the stop, and eventually reaching the quasi-state response. Quasi-steady-state displacement time histories at Nodes 5, 6 and 8, and their Fourier transforms are plotted for strong impact in Fig. 19; for nodes 5 and 6, first mode contribution to the displacement response is dominant and, for node 8, which is away from the tube-stop and input force location, the third mode contribution to the displacement response is obvious. The internal shear force time histories the tube/stop impact location (Node 5) are plotted for mild and strong impacts in Fig. 20; The higher mode has a dominant contribution to the response.

D.2 12.70 mm Thick Baffle Plate

Tube vibration responses at Node 5 (tube/support impact location) and Node 8 (near the fixed end) are plotted for strong tube/baffle impact ($F_0 = 2.85 \text{ N}$) in Figs. 21-29.

For quasi-steady-state displacement (see Fig. 21) first mode contribution is dominant. For node 8, a higher mode has a slight contribution. As the tube strikes the baffle plate, it rebounds from the baffle until it is "arrested" by the input force. If the instantaneous forcing amplitude at that moment is sufficiently high to arrest this motion, chattering occurs as can be observed in Fig. 21(a).

For the velocity response (see Fig. 22), first mode contribution is dominant at Node 5. However, for node 8, third mode contribution is dominant. Acceleration responses (see Fig 23) shows that third mode contribution is predominant. The acceleration time history at Node 5 shows the sudden jump when the tube strikes the baffle.

Fig. 19. Displacement Time Histories at Nodes 5, 6, and 8, and Their Fourier Transforms with Linear Spring Stops and In-Air Condition
(Loading 1: $f_o = 84$ Hz and $F_o = 2.85$ N)

Fig. 20. Internal Shear Force Time Histories at Node 5 for Mild ($F_0 = 1.43 \text{ N}$) and Strong ($F_0 = 2.85 \text{ N}$) Impact with Linear Spring Stops and In-Air Condition (Loading 1: $f_0 = 84 \text{ Hz}$)

(a) NODE 5

(b) NODE 8

Fig. 21. Quasi-Steady-State Displacement Time Histories at Nodes 5 and 8 with a 12.70 mm Thick Baffle Plate and In-Air Condition (Loading 1: $f_o = 84$ Hz and $F_o = 2.85$ N)

(a) VELOCITY TIME HISTORIES

(b) FOURIER TRANSFORMS

Fig. 22. Velocity Time Histories at Nodes 5 and 8 and Their Fourier Transforms with a 12.70 mm Thick Baffle Plate and In-Air Condition (Loading 1: $f_o = 84$ Hz and $F_o = 2.85$ N)

(a) ACCELERATION TIME HISTORIES

(b) FOURIER TRANSFORMS

Fig. 23. Acceleration Time Histories at Nodes 5 and 8 and Their Fourier Transforms with a 12.70 mm Thick Baffle Plate and In-Air Condition (Loading 1: $f_o = 84$ Hz and $F_o = 2.85$ N)

Internal shear force time histories are plotted in Fig. 24; The sudden jump in shear force at node 5 (see Fig. 24(a)) is the tube/baffle contact spring force and the occasional high amplitude overloads might have the significant effect on the fatigue damage rate. Damping force time histories are plotted in Fig. 25; The sudden jump in damping force at Node 5 (see Fig. 25(a)) is the tube/baffle contact damping force and its magnitude is much smaller than the tube/baffle contact spring force.

E. Tube Vibration Responses with In-Water Condition

Sinusoidal input forces are applied at two nodes, 3 and 4, along the length of the tube (see Loading 2 in Fig. 10). The reason for exciting the tube at oneside of the stop is to detect possible second mode contributions to the tube vibration response. The stop is assumed to be either a baffle plate or a linear elastic spring.

E.1 Linear Elastic Spring Stops

Tube vibration responses are plotted in Fig. 26 through 28 for strong ($F_o = 2.85$ N) and mild ($F_o = 1.43$ N) tube/stop impacts. Quasi-steady-state displacement time histories at Node 3 (input force location), Node 5 (tube-stop location) and Node 7 are plotted for strong impact in Fig. 26; It shows that the first mode contribution is dominant. Quasi-steady-state velocity time histories at Nodes 3, 5, and 7, and their Fourier transforms (see Fig. 27) show that the first mode contribution is dominant and that the higher mode in the neighborhood of 375 Hz has a significant contribution. The internal shear force time histories at tube/stop impact location for mild and strong impacting (see Fig. 28) show that the higher mode has a dominant contribution to the response.

(a) NODE 5

(b) NODE 8

Fig. 24. Internal Shear Force Time Histories at Nodes 5 and 8 with a 12.70 mm Thick Baffle Plate and In-Air Condition (Loading 1: $f_o = 84$ Hz and $F_o = 2.85$ N)

Fig. 25. Damping Force Time Histories at Nodes 5 and 8 with a 12.70 mm Thick Baffle Plate and In-Air Condition (Loading 1: $f_o = 84$ Hz and $F_o = 2.85$ N)

Fig. 26. Quasi-Steady-State Displacement Time Histories at Nodes 3, 5 and 7 with a Linear Spring Stop and In-Water Condition (Loading 2: $f_o = 74$ Hz and $F_o = 2.85$ N)

(a) VELOCITY TIME HISTORIES

(b) FOURIER TRANSFORMS

Fig. 27. Quasi-Steady-State Velocity Time Histories at Nodes 3, 5 and 7, and the Corresponding Fourier Transforms with a Linear Spring Stop and In-Water Condition (Loading 2: $f_0 = 74$ Hz and $F_0 = 2.85$ N)

(a) $F_0 = 1.43 \text{ N}$

(b) $F_0 = 2.85 \text{ N}$

Fig. 28. Internal Shear Force Time Histories at Node 5 for Two Different Forcing Amplitudes ($F_0 = 1.43 \text{ N}$ and 2.85 N) with a Linear Spring Stop and In-Water Condition (Loading 2: $f_0 = 74 \text{ Hz}$)

E.2 12.70 mm Thick Baffle Plate

Tube vibration responses at Node 3 (input force location), Node 5 (tube-baffle location) and Node 7 are plotted for strong tube/baffle impact ($F_0 = 2.85 \text{ N}$) in Figs. 29 through 32.

For quasi-steady-state displacement responses (see Fig. 29) first mode contribution is predominant. For velocity responses (see Fig 30), first mode contribution is dominant at Node 5. However, for node 7, a higher mode in the neighborhood of 375 Hz has a significant contribution. The acceleration time histories (see Fig. 31) shows the sudden jump when the tube strikes the baffle.

Internal shear force and damping force time histories for node 5 are plotted in Fig. 32. The sudden jump in shear force is the tube/baffle contact spring force and that in damping force is the contact damping force. Again, the contact damping force is much smaller than the contact spring force.

(a) NODE 3

(b) NODE 5

(c) NODE 7

Fig. 29. Quasi-Steady-State Displacement Time Histories at Nodes 3, 5 and 7 with a 12.70 mm Thick Baffle Plate and In-Water Condition (Loading 2: $f_o = 74$ Hz and $F_o = 2.84$ N)

(a) VELOCITY TIME HISTORIES

(b) FOURIER TRANSFORMS

Fig. 30. Quasi-Steady-State Velocity Time Histories at Nodes 5 and 7, and the Corresponding Fourier Transforms with a 12.70 mm Thick Baffle Plate and In-Water Condition (Loading 2: $f_o = 74$ Hz, and $F_o = 2.85$ N)

(a) NODE 5

(b) NODE 7

Fig. 31. Acceleration Time Histories at Nodes 5 and 7 with a 12.70 mm Thick Baffle Plate and In-Water Condition (Loading 2; $f_o = 74$ Hz and $F_o = 2.85$ N)

(a) INTERNAL SHEAR FORCE

(b) DAMPING FORCE

Fig. 32. Quasi-Steady-State Shear Force and Damping Force Time Histories at Node 5 with a 12.70 mm Thick Baffle Plate and In-Water Condition (Loading 2: $f_o = 74$ Hz and $F_o = 2.85$ N)

V. COMPARISON OF NUMERICAL AND EXPERIMENTAL RESULTS

Numerical results are compared with the experimental results to evaluate whether the numerical technique developed is adequate to predict tube vibration with tube/baffle interaction. The experimental results are first described and the numerical results are then compared with those of the experiments. The reference response to be compared include tube-baffle contact force and displacements.

The tube is excited by the sinusoidal input force with a forcing amplitude of 2.85 N applied at each pair of coils and a forcing frequency at the first resonant frequency. For in-air environment, forces are applied through four pairs of coils, $F_1 \sim F_4$ and, for in-water environment, forces are applied through two pairs of coils, F_1 and F_2 .

When the tube vibrates in a fluid, the gap fluid between the tube and the baffle plate follows a complex three dimensional flow pattern. The presence of the gap fluid gives rise to a fluid reaction force being interpreted as an added mass and a damping contribution of the gap fluid to the dynamic response of the tube. The effect of the gap fluid on the vibrational responses has been previously studied for a long cylindrical rod vibrating in a viscous fluid enclosed by a rigid, concentric cylindrical shell [18], which is based on the two-dimensional viscous flow theory. The added mass and the damping calculated based on the ref. [18] were too high, implying that it is not applicable to the case where the baffle thickness is much smaller than the tube span. ANL 7-Span tube/support interaction experiments [2] reveal that the tube support locations act as natural nodal points and the effect of the gap fluid is negligible to the vibrational responses of the tube.

(a) IN-AIR

(b) IN-WATER

Fig. 33. Measured Contact Forces and Calculated Internal Shear in Air and Water Environments with Linear Spring Stops

A. Contact Forces (Fig. 33)

The measured average contact force is approximately 20 N for in-air and 12 N for in-water. For analysis, internal shear force is calculated; that includes the contact force. The contact force is reasonably assumed to be dominant in shear at tube-baffle impact. The calculated contact force is approximately 15 N for in-air and 9 N for in-water. Both results are in reasonably good agreement.

B. Displacements

The motion-limiting stop is either impact transducer or baffle plate. The displacements at tube/stop location are compared.

Impact Transducer Stop (Fig. 34): The calculated displacement with in-air condition Fig. 34(a) shows a good agreement with an experimental result: first mode response is dominant and the tube/stop chattering phenomenon is observed. For in-water environment (Fig. 34(b)), the measured response shows a tube/stop chattering, but the calculated response does not. However, overall response in both results agrees well.

Baffle Plate Stop (Fig. 35): The measured spectral density curve in air showed the significant band limited peak in the frequency range, 83 ~ 120 Hz. The response excited at 120 Hz shows a strong tube/baffle impact and it agrees well with tube/baffle impact and a first mode contribution to the response is dominant. For in-water condition, both calculated and measured responses agree well. The flat response at tube/baffle impact indicates that the tube stays in contact with the baffle at impact.

ANALYSIS

EXPERIMENT

(a) IN-AIR

96

ANALYSIS

EXPERIMENT

(b) IN-WATER

Fig. 34. Measured and Calculated Displacements in Air and Water Environments with Linear Spring Stops

Fig. 35. Measured and Calculated Displacements in Air and In-Water Environment with a Baffle Plate

VI. CONCLUSIONS

A numerical method of analysis is presented for calculating tube vibration response with tube/baffle impacting. The analysis is based on a finite element technique, using a non-linear elastic contact spring-dashpot model to represent tube/baffle interaction, and on the assumption that the tube strikes the baffle-hole annulus uniformly over the contact area. The coupled equations of motion are directly integrated using a central difference method.

A fixed ended tube with motion-limiting stops at the center is tested in both air and water environments; tube/stop impact forces and tube displacement response are measured. The motion-limiting stop is represented by either an impact transducer pair or a stainless steel baffle plate. Results from the numerical analysis are in reasonably good agreement with those of the experiments.

Based on the numerical/experimental results, the following general conclusions can be drawn:

1. Lumped mass approach with an explicit time-integration of coupled equations of motion is found to be a suitable choice for a tube-baffle impacting analysis.
2. The chattering between the tube and the stop was observed for strong tube/stop impacts in both analysis and experiment; As the tube strikes the stop, it rebounds from the stop until it is "arrested" by the input force. If the instantaneous forcing amplitude at that moment is sufficiently high to arrest this motion, chattering occurs. Tube-stop chattering might be a significant factor for a long-term tube/stop impact wear.

3. When exciting the tube at the first mode frequency, the displacement response has a dominant first mode contribution. However, responses to other excitations show significant higher mode contributions.
4. Contact damping forces are negligible compared with the contact spring forces.
5. Tube-baffle contact forces increase with decreasing tube/baffle-hole clearances and increasing baffle thickness.
6. When the baffle location acts as a natural nodal point, the added mass and viscosity of the gap fluid have a negligible effect on tube vibration characteristics.

ACKNOWLEDGMENTS

This work was performed under the sponsorship of the Division of Reactor Research and Technology, U. S. Department of Energy.

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