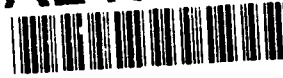


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A Closed Form Solution Of a Longitudinal Beam With A Viscous Boundary Condition

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New London, Connecticut**

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Preface

This report was prepared under the *Acoustic Array Technology Program* as part of the Submarine/Surface Ship ASW Surveillance Program sponsored by the Antisubmarine Warfare/Undersea Technology Directorate of the Office of Naval Technology: Program Element 62314N, ONT Block Program NU3B, Project No. RJ14R13, NUWC Job Order No. V60010, NUWC Principal Investigator D. A. Hurdis (Code 2141), and NUWC Program Director G. C. Connolly (Code 2192). The sponsoring activity's Technology Area Manager for Undersea Target Surveillance is T. G. Goldsberry.

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A CLOSED FORM SOLUTION OF A LONGITUDINAL BEAM WITH A VISCOUS BOUNDARY CONDITION

1. INTRODUCTION

The dynamic response of a beam with a viscous boundary condition is important because the designers of various structures often use viscous dampers to reduce force transmissibility and decrease displacement. The need for reduced force transmissibility is evident since lower force levels permit simpler and lighter structural designs. For this reason, viscous (shock) absorbers are currently critical to many systems, such as buildings, cars, and airplanes.

The closed form response of structures with fixed and free boundary conditions has been previously analyzed [1,2,3]. These analyses form the basis for many classical beam problems. Their self-adjoint operators are discretized using mutually orthogonal modes to produce models of structural dynamic response. These models, however, admit only standing waves into the response and do not consider viscous damping at the boundary. Recently, a number of papers have appeared that model different effects in beams, such as compressive axial loads [4,5], elastic foundations [6], and the coupling between flexural and torsional modes to axial loads [7]. These papers, which use a variety of analytical techniques to solve for the structural response of a beam, do not consider viscous dissipation at the boundary.

Although the structural response of a beam with a viscous damper can be determined using finite element analysis [8,9], this method does have a number of drawbacks. It is computationally intensive, does not provide explicit eigenvalues and eigenvectors, and does not yield a closed form solution. Finite element discretizations are

often too large when real time computations are required, as in the case of active control systems. Additionally, the effects of changing model parameters is not always as apparent in a finite element model as it is in a closed form algebraic solution.

This paper develops a closed form series solution for the axial wave equation with a fixed boundary condition at one end and a viscously damped boundary condition at the other. The addition of a damper to the boundary allows propagating and standing waves to exist in the structure simultaneously. The system model produces a differential operator that is nonself-adjoint and corresponding eigenfunctions that are not mutually orthogonal. By redefinition of the problem on another interval, the space and time modes will decouple and a closed form series solution can be found.

2. SYSTEM MODEL

The system model represents an axial beam fixed at $x = 0$ and a viscous damper at $x = L$ (Figure 1). A force is applied to the beam at location $x = x_f$. The addition of the damper to the beam will admit standing and propagating wave energy simultaneously. The linear second order wave equation modeling particle displacement in the beam is

$$\frac{\partial^2 u(x,t)}{\partial t^2} - \frac{E}{\rho} \frac{\partial^2 u(x,t)}{\partial x^2} = \frac{\delta(x - x_f)F(t)}{\rho A}, \quad (1)$$

where $u(x,t)$ is the displacement (m), E is the modulus of elasticity (N/m^2), ρ is the density of the beam (kg/m^3), x is the spatial location (m), t is time (s), A is the area of the beam (m^2), F is the applied force (N), and δ is the Dirac delta function ($1/\text{m}$). The wave equation assumes uniform area and negligible internal loss in the beam.

The boundary at $x = 0$ is fixed and can be expressed as

$$u(0,t) = 0 \quad (2)$$

The boundary condition at $x = L$ is obtained by matching the strain energy of the beam to the viscous dissipative force in the damper. This can be expressed as

$$AE \frac{\partial u}{\partial x}(L, t) = -c \frac{\partial u}{\partial t}(L, t) \quad , \quad (3)$$

where c is the viscous damping coefficient (Ns/m). When c is equal to zero (or infinity), the boundary at $x = L$ reflects all the wave energy, and the system response is composed only of standing waves. When c is equal to $A\sqrt{\rho E}$, the boundary at $x = L$ absorbs all the wave energy, and the system response is composed only of propagating waves. All other values of c exhibit some combination of standing and propagating wave energy in their response.

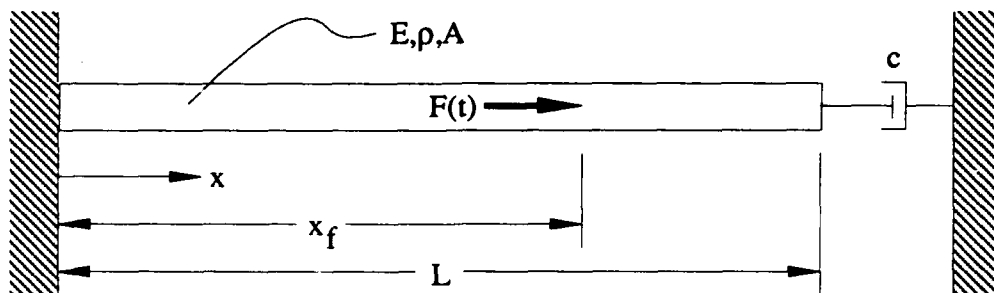


Figure 1. Beam With Viscous Damper

3. SEPARATION OF VARIABLES

A decoupled series of ordinary differential equations that represent the system are now developed from equations (1) - (3) and the initial conditions of the beam. The first step in deriving decoupled differential equations involves finding the eigenvalues and eigenfunctions of the model. This is accomplished by application of separation of variables to the homogeneous version of the wave equation (equation (1)) and the corresponding boundary conditions (equations (2) and (3)). Separation of variables assumes that the independent variable can be written as a product of two functions: one in the time domain and one in the spatial domain. This form is

$$u(x, t) = T(t)X(x) \quad . \quad (4)$$

Equation (4) is now substituted into the homogeneous version of equation (1), which produces the following ordinary differential equations:

$$\frac{d^2 T(t)}{dt^2} - \lambda^2 \frac{E}{\rho} T(t) = 0 \quad (5)$$

and

$$\frac{d^2 X(x)}{dx^2} - \lambda^2 X(x) = 0 \quad , \quad (6)$$

where λ is the complex-valued separation constant.

The general solution to equation (5) is

$$T(t) = Ae^{\lambda \sqrt{\frac{E}{\rho}} t} + Be^{-\lambda \sqrt{\frac{E}{\rho}} t} \quad . \quad (7)$$

The fixed boundary condition (equation (2)), is now applied to the spatial ordinary differential (equation (6)) which gives

$$X(x) = e^{\lambda x} - e^{-\lambda x} \quad . \quad (8)$$

Applying the viscous boundary condition (equation (3)) to equations (7) and (8) yields $B = 0$ and the n -mode-indexed separation constant

$$\lambda_n = \frac{1}{2L} \log_e \left[\frac{AE - c \sqrt{\frac{E}{\rho}}}{AE + c \sqrt{\frac{E}{\rho}}} \right] + \frac{(2n+1)\pi}{2L} i \quad , \quad n = 0, \pm 1, \pm 2, \dots \quad , \quad (9)$$

where i is the square root of -1 . The real part of the separation constant is associated with the propagating wave energy, and the imaginary part of the separation constant is associated with the standing wave energy. The eigenvalues of the system are the separation constants multiplied by the wave speed,

$$\Lambda_n = \sqrt{\frac{E}{\rho}} \lambda_n \quad , \quad (10)$$

where Λ_n has units of rad/s. A plot of the eigenvalue location in the complex plane is shown in Figure 2. The eigenvalues are equally spaced and parallel to the real axis. Once the indexed separation constant is determined, the spatial eigenfunctions can be defined by inserting equation (9) into equation (8), which gives

$$\varphi_n(x) = e^{\lambda_n x} - e^{-\lambda_n x} \quad . \quad (11)$$

A typical eigenfunction ($n = 2$) is shown in Figure 3 for $c = (0.5)A\sqrt{\rho E}$. The eigenfunctions are not mutually orthogonal on $[0,L]$; therefore their inner product with respect to one another on $[0,L]$ is not zero, and traditional boundary value techniques will not decouple the time and space modes. A method is developed below that redefines the problem interval over $[-L,L]$ and decouples the time and space modes of the system. Once the modes have been decoupled, the problem solution can be transformed back to the original interval of $[0,L]$.

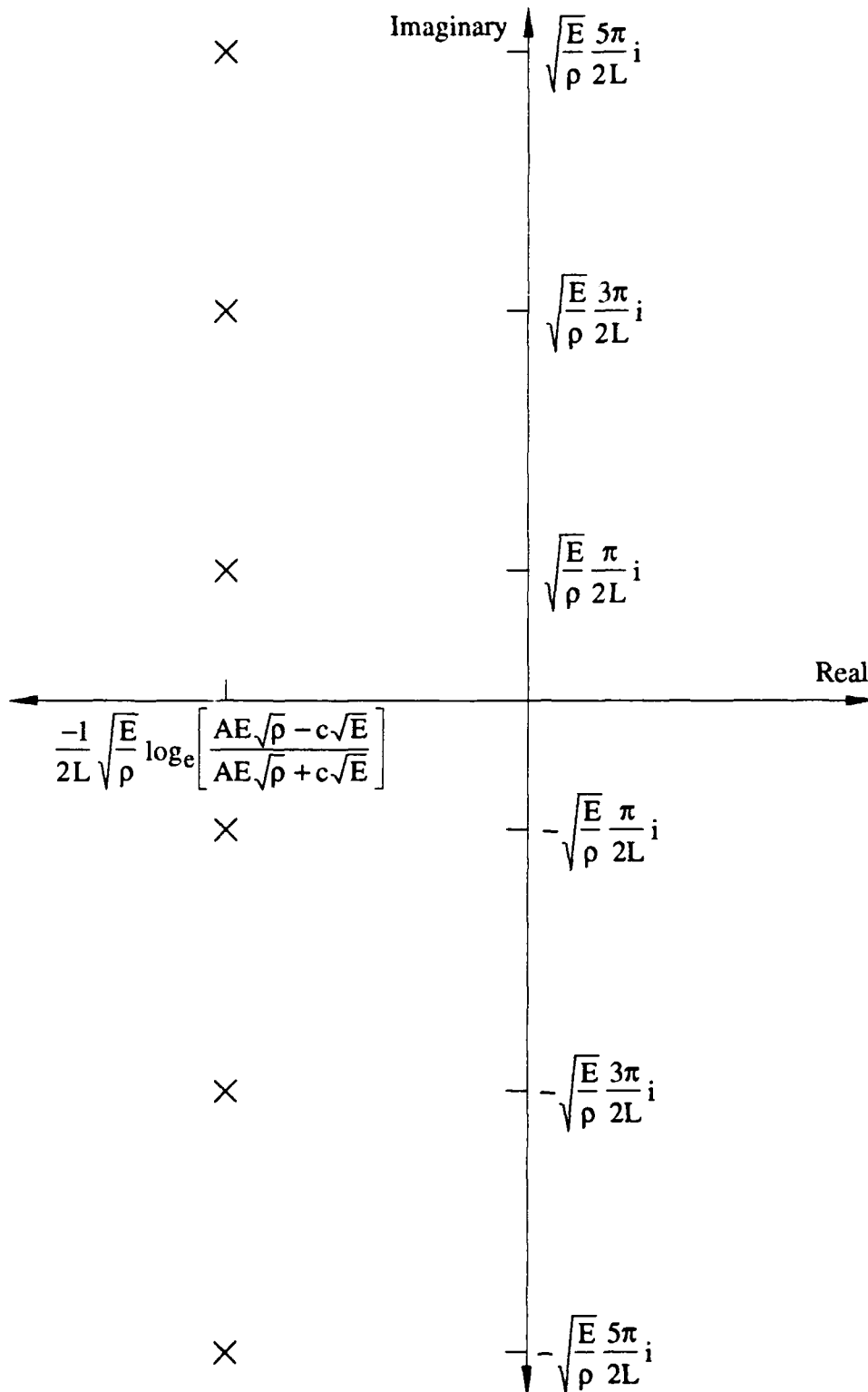
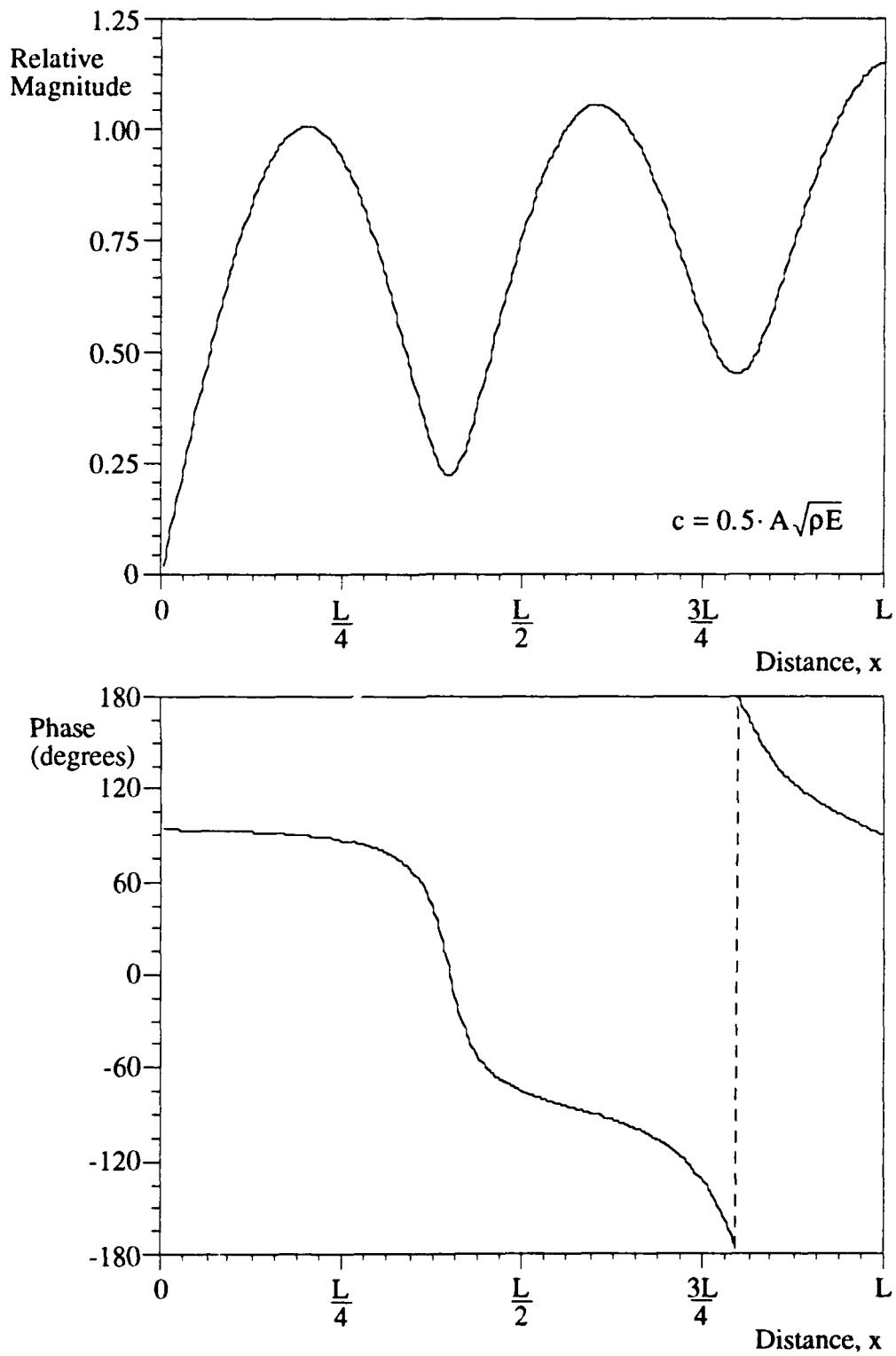


Figure 2. Eigenvalue Location in the Complex Plane

Figure 3. The $n = 2$ Eigenfunction

4. SERIES SOLUTION

The displacement (or solution) to the forced wave equation, written as a series solution, is

$$u(x,t) = \sum_{n=-\infty}^{\infty} b_n(t) \varphi_n(x) \quad , \quad (12)$$

where the $b_n(t)$'s are the generalized coordinates and the $\varphi_n(x)$'s are the spatial eigenfunctions. Derivation of an inner product that decouples the time and space modes requires the time derivative (velocity) of the particle displacement to be written in two different forms. The first form is the derivative of equation (12) and yields

$$\frac{\partial u}{\partial t}(x,t) = \sum_{n=-\infty}^{\infty} \dot{b}_n(t) \varphi_n(x) \quad , \quad (13)$$

where the dot over b denotes the time derivative of the function. The second, developed by using equations (7) and (4), is written as

$$\frac{\partial u}{\partial t}(x,t) = \sum_{n=-\infty}^{\infty} \Lambda_n b_n(t) \varphi_n(x) \quad . \quad (14)$$

Equating equations (13) and (14) produces

$$\sum_{n=-\infty}^{\infty} [\dot{b}_n(t) - \Lambda_n b_n(t)] \varphi_n(x) = 0 \quad . \quad (15)$$

The assumption is now made that differentiation will distribute over the summation. Decoupled space and time modes will validate this assumption. The forced wave equation (1) is rewritten with the above equations. The second partial time derivative is found from the time derivative of equation (14), and the second partial spatial derivative is found from the second spatial derivative of equation (13). Inserting these derivatives into equation (1) yields

$$\sum_{n=-\infty}^{\infty} [\dot{b}_n(t) - \Lambda_n b_n(t)] \Lambda_n \phi_n(x) = \frac{\delta(x - x_f) F(t)}{\rho A} \quad (16)$$

Equation (15) is now differentiated with respect to x and multiplied by the wave speed $\sqrt{\frac{E}{\rho}}$. The result is then added to equation (16) to form

$$\sum_{n=-\infty}^{\infty} [\dot{b}_n(t) - \Lambda_n b_n(t)] 2\Lambda_n e^{\lambda_n x} = \frac{\delta(x - x_f) F(t)}{\rho A} \quad x \in [0, L] \quad (17)$$

and is subtracted from equation (16) to give

$$\sum_{n=-\infty}^{\infty} [\dot{b}_n(t) - \Lambda_n b_n(t)] 2\Lambda_n e^{-\lambda_n x} = \frac{-\delta(x - x_f) F(t)}{\rho A} \quad x \in [0, L] \quad (18)$$

The interval of equation (18), now changed from $[0, L]$ to $[-L, 0]$ by substitution of $-x$ for x , yields

$$\sum_{n=-\infty}^{\infty} [\dot{b}_n(t) - \Lambda_n b_n(t)] 2\Lambda_n e^{\lambda_n x} = \frac{-\delta(-x - x_f) F(t)}{\rho A} \quad x \in [-L, 0] \quad (19)$$

Combining equations (17) and (19) into a single equation and breaking the exponential into terms that contain the index n and terms that do not contain the index n results in

$$\sum_{n=-\infty}^{\infty} [\dot{b}_n(t) - \Lambda_n b_n(t)] 2\Lambda_n e^{\frac{(2n+1)\pi x i}{2L}} = \begin{cases} e^{\frac{-1}{2L} \log_e \left[\frac{AE\sqrt{\rho} - c\sqrt{E}}{AE\sqrt{\rho} + c\sqrt{E}} \right] x} \left[\frac{-\delta(-x - x_f) F(t)}{\rho A} \right] & x \in [-L, 0] \\ e^{\frac{-1}{2L} \log_e \left[\frac{AE\sqrt{\rho} - c\sqrt{E}}{AE\sqrt{\rho} + c\sqrt{E}} \right] x} \left[\frac{\delta(x - x_f) F(t)}{\rho A} \right] & x \in [0, L] \end{cases} \quad (20)$$

The exponential $e^{\frac{-(2m+1)\pi xi}{2L}}$ (where m is an integer) is now multiplied on both sides of equation (20), and the resulting equation is integrated from $-L$ to L . The left-hand side of the equation is orthogonal on the new interval and can be expressed as

$$\int_{-L}^L [\dot{b}_{nm}(t) - \Lambda_n b_{nm}(t)] 2\Lambda_n e^{\frac{(2n+1)\pi xi}{2L}} e^{\frac{-(2m+1)\pi xi}{2L}} dx$$

$$= \begin{cases} [\dot{b}_n(t) - \Lambda_n b_n(t)] 4L\Lambda_n & n = m \\ 0 & n \neq m \end{cases} \quad (21)$$

Use of the reflection property of integrals and the bound of $0 < x_f < L$ results in the right hand side of equation (20) becoming

$$\frac{F(t)}{\rho A} \left[\int_{-L}^0 -\delta(-x - x_f) e^{-\lambda_m x} dx + \int_0^L \delta(x - x_f) e^{-\lambda_m x} dx \right]$$

$$= \frac{-F(t)\phi_m(x_f)}{\rho A} \quad (22)$$

Equations (21) and (22) can be equated (for $n = m$) to form ordinary differential equations for the generalized coordinates b_n as

$$\dot{b}_n(t) - \Lambda_n b_n(t) = \frac{-F(t)\phi_n(x_f)}{4L\Lambda_n\rho A} \quad (23)$$

An explicit solution to equation (23) cannot be found until a time-dependent forcing function has been specified.

The initial conditions of the generalized coordinates can be determined from the initial conditions of the beam. This equation is

$$a_n(0) = \frac{1}{4\lambda_n^2 L} \int_0^L \frac{\partial u}{\partial x}(x,0) \frac{d\phi_n(x)}{dx} dx - \frac{1}{4\lambda_n L} \int_0^L \frac{\partial u}{\partial t}(x,0) \phi_n(x) dx \quad , \quad (24)$$

where $\frac{\partial u}{\partial x}(x,0)$ is the initial strain energy in the beam (dimensionless) and $\frac{\partial u}{\partial t}(x,0)$ is the initial velocity of the beam (m/s). The formulation of equation (24) is presented in the appendix.

5. FREQUENCY RESPONSE

A frequency domain solution to equation (23) can be found by specifying that the forcing function $F(t)$ be equal to a harmonic function $F_0 e^{i\omega t}$, where F_0 has units of newtons. Solving the differential equation (23) results in the following solution to the generalized coordinates:

$$b_n(t) = \frac{-F_0 \phi_n(x_f)}{(i\omega - \Lambda_n) 4L\Lambda_n \rho A} e^{i\omega t} \quad . \quad (25)$$

When equation (25) is inserted into equation (12), the displacement of the beam becomes

$$u(x,t) = \sum_{n=-\infty}^{\infty} \frac{-F_0 \phi_n(x_f) \phi_n(x)}{(i\omega - \Lambda_n) 4L\Lambda_n \rho A} e^{i\omega t} \quad . \quad (26)$$

Equation (26) can be truncated using $2N$ symmetric terms to yield an engineering solution to the problem.

6. A NUMERICAL EXAMPLE

The accuracy of the model and the effects of truncation error were investigated with a numerical example. The following constants were used: $L = 20$ m, $E = 207 \times 10^9$ N/m², $\rho = 7.8 \times 10^3$ kg/m³, $F_0 = 1000$ N, $A = 0.01$ m², $x_f = 3$ m, and $c = 75000$ Ns/m. Figure 4 is the frequency domain response of the structure $u(x_f, \omega)$ viewed at $x_f = 11$ m for a 6-term ($-3 \leq n \leq 2$) and a 50-term ($-25 \leq n \leq 24$) model. Numerical simulations suggest that two first order terms are needed to model each beam resonance. There is only

a 0.45 percent (-46.9 dB) difference between the two truncated models up to the third resonance of the beam. The addition of terms to the model does not change its value at the lower frequencies. This is due to the frequency content of each generalized coordinate: the lower indexed coordinates contain only lower frequency response and the higher indexed coordinates contain only higher frequency response.

Figure 5 shows the 6-term frequency response of the structure compared with finite element analysis results using 5-node (square marker), 8-node (diamond marker) and 21-node (triangle marker) finite element models. The addition of terms to the finite element model produces more accurate results at lower frequencies due to the coupling between the nodes in a finite element formulation. Because the bandwidth of the system matrices are greater than one, the mode shapes are not explicit to the analysis, and the addition of terms (nodes) to the analysis can affect the accuracy of many beam resonance modes. The continuous formulation presented above eliminates the problem of banded system matrices by use of an orthogonal inner product to decouple the mode shapes of the beam.

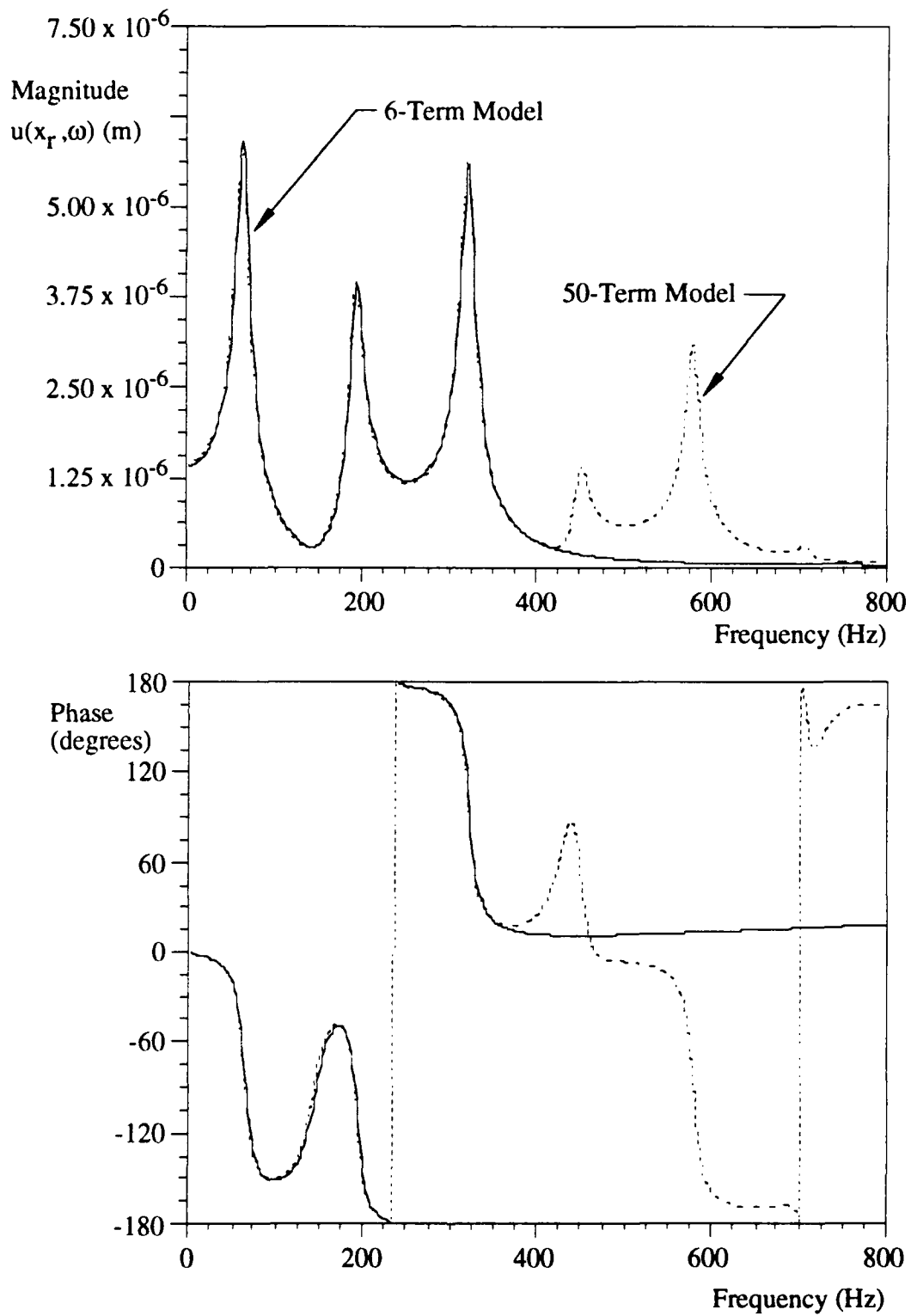


Figure 4. Frequency Response of an Axially Forced Beam: 6- and 50-Term Models

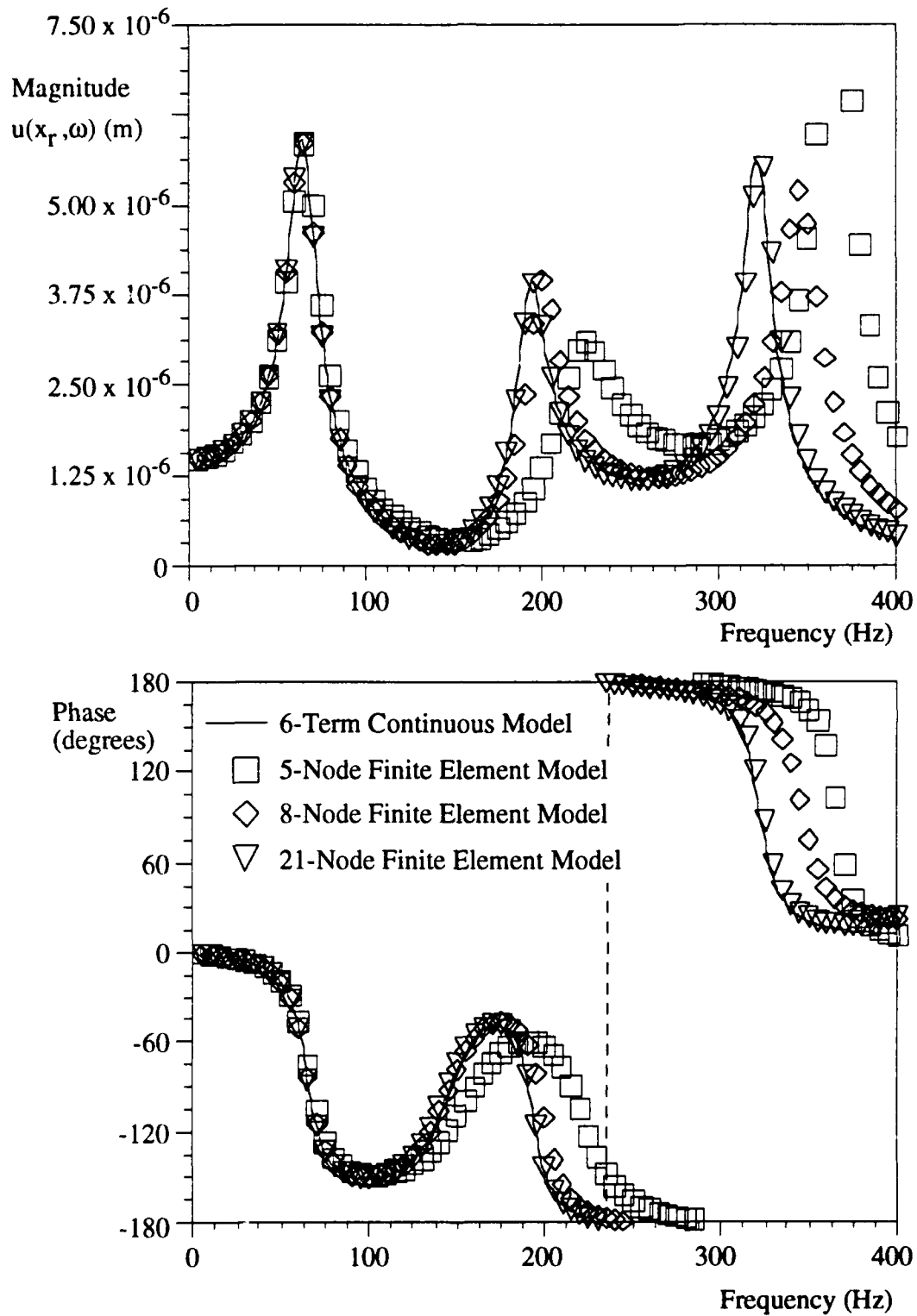


Figure 5. Frequency Response of an Axially Forced Beam: Continuous Solution Compared with Finite Element Solution

7. CONCLUSIONS

The axial response of a beam with a damper at the end can be determined by using separation of variables and by changing the interval of the problem. A truncated series solution can be implemented to approximate the exact dynamic response. Truncation error at lower frequencies is extremely small. This closed form solution is computationally efficient and the eigenvalues and eigenvectors of the system are explicit.

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APPENDIX

INITIAL CONDITIONS OF GENERALIZED COORDINATES

The initial conditions of the generalized coordinates can be determined from the initial conditions of the beam. Rewriting equation (12) at time $t = 0$ yields

$$u(x,0) = \sum_{n=-\infty}^{\infty} b_n(0) \varphi_n(x) \quad . \quad (A.1)$$

Equation (A.1) is now differentiated with respect to the spatial variable x to give

$$\frac{\partial u}{\partial x}(x,0) = \sum_{n=-\infty}^{\infty} b_n(0) \lambda_n (e^{\lambda_n x} + e^{-\lambda_n x}) \quad (A.2)$$

and differentiated with respect to the time variable t to give

$$\frac{\partial u}{\partial t}(x,0) = \sum_{n=-\infty}^{\infty} b_n(0) \lambda_n (e^{\lambda_n x} - e^{-\lambda_n x}) \quad . \quad (A.3)$$

Equation (A.2) is the initial strain energy in the beam and equation (A.3) is the initial velocity of the beam. Equations (A.2) and (A.3) are now added to yield

$$\sum_{n=-\infty}^{\infty} 2b_n(0) \lambda_n e^{\lambda_n x} = \left[\frac{\partial u}{\partial x}(x,0) + \frac{\partial u}{\partial t}(x,0) \right] \quad (A.4)$$

and subtracted to yield

$$\sum_{n=-\infty}^{\infty} 2b_n(0) \lambda_n e^{-\lambda_n x} = \left[\frac{\partial u}{\partial x}(x,0) - \frac{\partial u}{\partial t}(x,0) \right] \quad . \quad (A.5)$$

The interval of equation (A.5) is now changed from $[0,L]$ to $[-L,0]$ by the substitution of $-x$ for x . This equation is

$$\sum_{n=-\infty}^{\infty} 2b_n(0)\lambda_n e^{\lambda_n x} = \left[\frac{\partial u}{\partial x}(-x,0) - \frac{\partial u}{\partial t}(-x,0) \right] \quad x \in [-L,0] \quad . \quad (\text{A.6})$$

Equations (A.4) and (A.6) are now combined and the exponential is broken into two terms: one contains the index n and one does not contain the index n . The resulting equation is

$$\sum_{n=-\infty}^{\infty} 2b_n(0)\lambda_n e^{\frac{(2n+1)\pi x i}{2L}} = \left\{ \begin{array}{l} \left[\frac{\partial u}{\partial x}(-x,0) - \frac{\partial u}{\partial t}(-x,0) \right] e^{\frac{-1}{2L} \log_e \left[\frac{AE\sqrt{\rho} - c\sqrt{E}}{AE\sqrt{\rho} + c\sqrt{E}} \right] x} \quad x \in [-L,0] \\ \left[\frac{\partial u}{\partial x}(x,0) + \frac{\partial u}{\partial t}(x,0) \right] e^{\frac{-1}{2L} \log_e \left[\frac{AE\sqrt{\rho} - c\sqrt{E}}{AE\sqrt{\rho} + c\sqrt{E}} \right] x} \quad x \in [L,0] \end{array} \right. \quad (\text{A.7})$$

The exponential $e^{\frac{-(2m+1)\pi x i}{2L}}$ is now multiplied by both sides and the left-half of equation (A.7) is integrated from $-L$ to L . This yields

$$b_n(0)4\lambda_n L = \int_{-L}^0 \left[\frac{\partial u}{\partial x}(-x,0) - \frac{\partial u}{\partial t}(-x,0) \right] e^{-\lambda_n x} dx + \int_0^L \left[\frac{\partial u}{\partial x}(x,0) + \frac{\partial u}{\partial t}(x,0) \right] e^{-\lambda_n x} dx \quad . \quad (\text{A.8})$$

Using the reflection property of integrals and equation (11), equation (A.8) can be rewritten to give

$$b_n(0) = \frac{1}{4\lambda_n^2 L} \int_0^L \frac{\partial u}{\partial x}(x,0) \frac{d\varphi_n(x)}{dx} dx - \frac{1}{4\lambda_n L} \int_0^L \frac{\partial u}{\partial t}(x,0) \varphi_n(x) dx \quad . \quad (\text{A.9})$$

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