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EGG-EA-5038
October 1979

PIPING SYSTEM STRESS DEPENDENT DAMPING STUDY

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This is an informal report intended for use as a preliminary or working document

PREPARED FOR
U. S. Nuclear Regulatory Commission
Under DOE Contract No. DE-AC07-76ID01570
FIN No. A6156

EG&G Idaho

8007030015

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FORM EG&G-398
(Rev. 05-79)

INTERIM REPORT

Accession No. ____________

Report No. EGG-EA-5038

Contract Program or Project Title:

Code Assessment and Applications Program

Subject of this Document:

Piping System Stress Dependent Damping Study

Type of Document:

Evaluation Report

Author(s):

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Date of Document:

October 1979

Responsible NRC Individual and NRC Office or Division:

J. R. Fair, NRC-DOR

This document was prepared primarily for preliminary or internal use. It has not received full review and approval. Since there may be substantive changes, this document should not be considered final.

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Prepared for the
U.S. Nuclear Regulatory Commission
Washington, D.C.
Under DOE Contract No. DE-AC07-76ID01570
NRC FIN No.
A6156

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ABSTRACT

Experimental results indicate that, as deflections of a piping system increase to a point where the yield stress in the pipe material has been exceeded, an increase in the energy absorption capability of the yielded portion of the system occurs. This increased energy absorption can be thought of as an effective damping of the system modes at these yielded sections even though many different factors contribute to the observed phenomenon. A study using finite element techniques has been performed to assess the effects of this phenomenon on a simple piping system.

SUMMARY

A damping study is described in this report. This study consists of the calculation of stresses and support loads on an idealized piping system using time history analysis techniques incorporating both constant percent critical damping and variable element percent critical damping values.

Two cases have been studied. The CASE A system consists of a pipe fixed at one end, free at the other end with four intermediate supports. CASE B is identical to CASE A with different properties being used between the fixed end and the first intermediate support. The forcing function applied to the system at the free end is a simple sine function. The maximum amplitude of the force is determined such that the maximum stress in the system is slightly greater than the material yield stress.

Three damping conditions were assumed for each case. A base condition of one percent constant damping was analyzed followed by a constant two percent damping condition. The third condition is derived from the base case by assuming one percent damping occurs when the base case stress is less than or equal to one-half the material yield stress and two percent damping occurs at yield.

The response calculations were performed using the full mass, damping, and stiffness matrices. Internal moments were then calculated and used to determine bending stresses at each structural node point.

The final results are in the form of a ratio of variable damping stresses to constant two percent damping stresses. From these results it is concluded that the response of the system analyzed using variable stress dependent damping is essentially the same as the response using constant damping determined by the maximum stress value.

CONTENTS

I. INTRODUCTION

An effective damping approach has been used to assess the local energy absorption effect on a simplified piping system and is documented in this report. This study was initiated at the request of the Nuclear Regulatory Commission's Division of Operating Reactors.

A large amount of energy can be absorbed by a system through local nonlinearities. The absorption of this energy locally may preclude the total failure of an overall system; thus, if a system is close to failure it would be benefi al to know the effects of these local nonlinearities on the overall system response.

Local nonlinearities can be intuitively described by postulating an effective damping mechanism to account for additional energy absorption. The actual energy absorbing mechanism is somewhat complicated by material yielding, fluid flow, local temperature transients, hysteresis and many other factors. The use of effective damping appears to be the most direct approach to analytically assess the experimentally demonstrated energy absorption phenomenon.

The concept of a local change in the percent critical damping of the system first mode has been developed and applied to a simplified piping system. The percent of critical damping of each element of the system is determined based on the percentage of yield stress exhibited in the element due to a given sinusoidal input function. The percent critical damping values applied to the system first mode shape are incorporated in the mechanical response calculations thus allowing an assessment of local damping variations on the system response.

Section II describes the system model used in the present study. This description is followed in Section III by a discussion of the method used to incorporate the variations in element damping.

Section IV describes the response calculation procedures. Results of the study are detailed in Section V followed by some general observations in Section VI. Conclusions and references are presented in Section VII and VIII, respectively.

II. STRUCTURAL REPRESENTATION

A finite element model has been formulated to mathematically describe a simple piping system. The model consist of a series of uniform beam elements containing a total of twenty-one nodes with two degrees-of-freedom at each node (one translation and one in plane rotation). One end of the beam is fixed in both translation and rotation while intermediate translational supports are supplied at equal intervals along the beam (the model is shown in Figure 1).

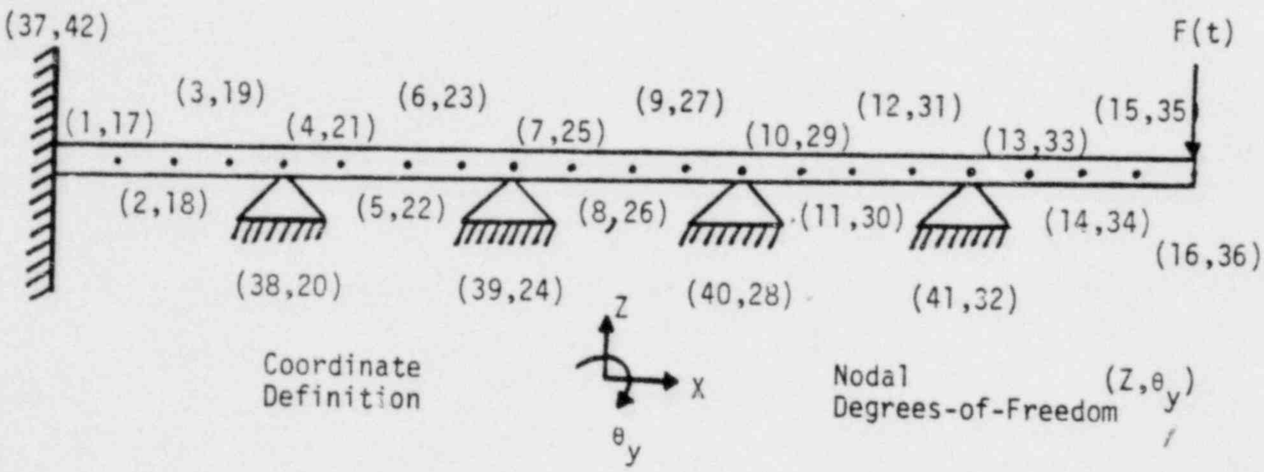


Fig. 1 Piping System Structural Representation

Two independent structures are analyzed. These analytical models are designated CASE A and CASE B. Each case is analyzed for three independent damping conditions.

The beam properties used in the CASE A and CASE B mathematical representations are defined in Table I. The general properties correspond to a .254 m diameter, schedule 40, A376, Type 316 pipe at 288^{O}C, while CASE B element one properties correspond to a .102 m diameter pipe at the same conditions. Translational masses and rotational inertias were lumped at each nodal point. The stiffness and damping matrices were generated using the FORMA subcode package documented in Reference 1. The free-free system was formed on an element by element basis, combined, and then the appropriate boundary conditions were applied. This procedure allows each element damping kernel to have different properties.

The forcing function, F(t), was applied at the free end of the system. The force is a simple sinusoidal input at the first system natural frequency. The input force properties are defined in Table I.

TABLE I

SYSTEM PROPERTIES

STRUCTURAL		
CASE B**	ITEM	CASE A
1.79×10^{11}	Elastic Modulus N/m^2	1.79×10^{11}
3.01×10^{-6}	Moment of inertia m^4	6.69×10^{-5}
2.05×10^{-3}	Area m^2	7.68×10^{-3}
.5	Form Factor	.5
1.14×10^{-1}	Outside Diameter m	2.73×10^{-1}
6.02×10^{-3}	Thickness m	9.27×10^{-3}
3.06×10^{2}	Weight per unit length N/m	1.22×10^{-3}
3.26×10^{2}	Weight Inertia per unit length N	1.65×10^{4}
.3	Poisson's Ratio	.3
1.65×10^{8}	Material Yield Stress N/m^2	1.65×10^{8}

FORCE	
FORCE = A sin wt	
Initial Time s	0.0
Final Time s	9.5
First Circular Frequency CASE A rad/s	30.14
Maximum Amplitude CASE A N	1.0×10^{3}
First Circular Frequency CASE B rad/s	30.13
Maximum Amplitude CASE B N	1.0×10^{3}

** CASE B Properties for the first section from the fixity to the first support. All other CASE B sections are the same as CASE A.

III. VARIABLE DAMPING DESCRIPTION

Experimental results indicate that energy absorption in a piping system varies as the stress (deflection) varies. The present study assumes that 2% critical damping occurs at the material yield stress (Sy) on the first modal shape and decreases to 1% critical at 0.5 Sy. Interpolation on the percent critical damping was performed for points other than those described with 1% being the minimum damping allowed. The damping variations were assumed to occur on an elemental level with an element consisting of all points between adjacent supports.

Variable damping was incorporated into the analysis by assuming that the element damping was proportional to the system stiffness. The initial stress calculations are made based on a system model with 1% critical damping on the first system mode shape. The element damping factors are then calculated using Equation 1.

$$\beta_i = \frac{2\zeta_i}{W} \quad i = 1 \text{ to number of elements} \qquad (1)$$

ζ_i = percent critical damping

W = system first mode circular frequency

The percent value, ζ, for critical damping is obtained from the stress values as outlined above and the first mode circular frequency in based on the first system natural frequency.

The damping matrix kernels are obtained from the stiffness kernels as shown in Equation 2.

$$[C]_i = \beta_i [K]_i \qquad (2)$$

with the $[K]$ matrices being free-free elemental stiffness kernels. These damping kernels are then added together in the appropriate manner as indicated in Equation 3 and the boundary conditions are applied.

$$\begin{bmatrix} [C_1] & 0 \\ & \\ 0 & 0 \end{bmatrix} + \begin{bmatrix} 0 & 0 & 0 \\ 0 & [C_2] & 0 \\ 0 & 0 & 0 \end{bmatrix} + \ldots + \begin{bmatrix} 0 & 0 \\ & \\ 0 & [C_n] \end{bmatrix} = [C] \qquad (3)$$

The damping matrix formed following the procedure outlined above is incorporated directly in the mechanical response equations.

IV. RESPONSE CALCULATIONS

The structural representation described in Section II is coupled with the damping matrix discussed in Section III to form the mechanical response equations. The solution of these equations is found utilizing the TR2 subcode from Reference 1. The algorithim used by the subcode is the Newmark-Chan-Beta technique.

The output responses are in the form of discrete deflections and rotations at each node point retained in the dynamic analysis. The applied forces on the system are calculated from these deflections as shown in Equation 4.

$$\left\{ L \right\} = \left[K_{free\text{-}free} \right] \left\{ \begin{array}{c} X \\ 0 \\ \hline 0 \end{array} \right\} \tag{4}$$

The zero deflections and rotations correspond to the fixed point degrees-of-freedom. Internal moments and selected support loads are calculated by summing the applied loads and moments along the beam as shown in Equation 5. The matrix Σ is of the form shown in Equation 6.

$$\left\{ \begin{array}{c} M \\ V \end{array} \right\} = \left[\Sigma \right] \left\{ L \right\} \tag{5}$$

$$\left[\Sigma \right] = \begin{bmatrix} \Delta p & I_m \\ I_p & 0 \end{bmatrix} \qquad \left\{ \begin{array}{c} p \\ M \end{array} \right\}_{\text{Applied}} \tag{6}$$

where

Δp = Selected moment arms

I_m = Moment selector matrix (not diagonal)

I_p = Applied load selector matrix (diagonal)

The stresses in the pipe are calculated using the section properties of the pipe and the linear relation for stress due to internal moments presented in Equation 7. These stresses along with

$$\sigma = \frac{Mc}{I} \tag{7}$$

the corresponding reaction point loads are tabulated in Section V.

V. RESULTS

The results of the previously described study are presented in this section. Maximum bending stresses and reaction loads are presented for both CASE A and CASE B. Results are tabulated for three separate conditions:

1. Uniform 1 percent damping (base case)
2. Uniform 2 percent damping
3. Variable stress level dependent damping

The results for CASE A are presented in Table II while those for CASE B are shown in Table III.

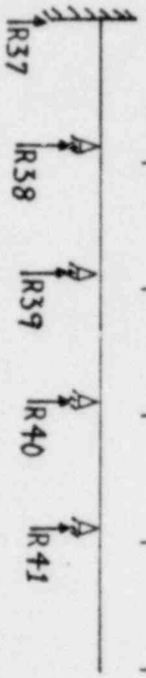
The major value of interest is the ratio of the condition 3 variable damping stresses to the condition 2 constant 2 percent damping stresses. A large value for this ratio would indicate that the practice of using constant damping to account for element yielding may not be justified.

The maximum ratio value for CASE A and CASE B are presented below.

CASE A ratio = 1.02
CASE B ratio = 1.02

TABLE II

LOAD SUMMARY CASE A

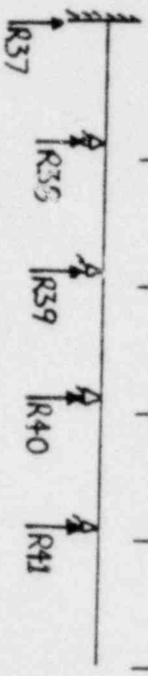


Degree of Freedom	Constant Damping 1% Base Case Stresses	Constant Damping 2% Case Stressing	Variable Damping Case Stresses
M42	5.42	2.86	2.93
M17	1.27	0.67	0.69
M18	1.31	0.69	0.71
M19	1.43	0.76	0.78
M20	1.57	0.81	0.85
M21	17.88	9.47	9.66
M22	17.58	9.31	9.50
M23	17.01	8.99	9.20
M24	16.46	8.71	8.90
M25	39.43	20.90	21.34
M26	38.30	20.30	20.72
M27	36.16	19.15	19.57
M28	34.11	18.07	18.44
M29	169.97	89.88	91.77
M30	165.75	87.65	89.49
M31	157.89	83.50	85.25
M32	150.33	79.51	81.17
M33	165.78	87.81	89.61
M34	149.21	79.05	80.64
M35	109.36	57.94	59.07
M36	42.59	22.63	23.00
R37	1664.	881.	899.
R38	6564.	3479.	3545.
R39	22521.	11877.	12153.
R40	82119.	43464.	44300.
R41	127273.	67159.	68618.

NOTE 1: The M values represent bending stresses in units of (10^6) N/m^2
NOTE 2: The R values are reaction loads in units of N
NOTE 3: The yield stress value is 165. x 10^6 N/m^2

TABLE III

LOAD SUMMARY CASE B



Degree of Freedom	Constant Damping 1% Base Case Stresses	Constant Damping 2% Case Stressing	Variable Damping Case Stresses
M42	5.23	2.76	2.82
M17	1.12	.59	.61
M18	1.35	.71	.73
M19	1.94	1.02	1.04
M20	2.59	1.37	1.40
M21	10.03	5.30	5.42
M22	9.75	5.16	5.26
M23	9.02	4.77	4.87
M24	8.38	4.43	4.52
M25	43.56	23.07	23.56
M26	42.37	22.45	22.92
M27	40.19	21.29	21.74
M28	38.11	20.19	20.60
M29	164.99	87.24	89.05
M30	160.75	85.00	86.76
M31	152.88	80.84	82.51
M32	145.31	76.84	78.44
M33	170.70	90.42	92.22
M34	154.13	81.65	83.27
M35	114.24	60.53	61.72
M36	47.44	25.20	25.63
R37	177.	94.	95.
R38	3922.	2073.	2117.
R39	20874.	11059.	11291.
R40	81685.	43190.	44079.
R41	126897.	67227.	68[illegible]05.

NOTE 1: The M values represent bending stresses in units of (10^6) N/m^2
NOTE 2: The R values are reaction loads in units of N
NOTE 3: The yield stress value is 165. x 10^6 N/m^2

VI. OBSERVATIONS

The study described in this report is scoping in nature. The primary assumption is that a modal form of damping could represent the effects of several internal and external damping mechanisms. This is of course not completely realistic since some of the external effects will not be a direct function of the mode shapes much less a single mode shape. The fact that the first system modal frequency was used to calculate the factors used to vary the damping on each element is also a fairly unrealistic assumption.

The method outlined does yield some insight into the effects of variable damping in spite of the large number of assumptions and limitiations. The conclusions to be drawn from the present study should be generally applicable to systems similar to the one analyzed. Caution should be used, however, in extending these conclusions to other types of systems.

VII. CONCLUSIONS

The conclusion to be derived from the present analysis can be simply stated:

> The response of the system analyzed using variable stress dependent damping is essentially the same as the response using constant damping determined by the maximum stress value.

Intuition tends to support this conclusion. The primary response of the system occurs in the first two bays of the structure from the free end. The response of these two bays is considerably larger (3 to 4 times) than that for the remaining bays. The variable damping and constant damping in these bays are equal and, therefore, the overall response of the system is essentially equivalent. This would be the tendency of any system of similar design. Larger differences would be expected for larger variations in damping throughout the system.

VIII. REFERENCES

1. R. L. Wohlen, "Synthesis of Dynamic Systems Using FORMA-FORTRAN Matrix Analysis," Martin Marietta Corporation, Denver, Colorado, Report MCR-71-75, Contract NAS 8-25922, May 1971.