EVALUATING STRUCTURAL DETERIORATION BY DYNAMIC RESPONSE

By H. L. Chen,[1] C. C. Spyrakos,[2] and G. Venkatesh[3]

ABSTRACT: The dynamic response of two C 3 × 4.1 channel steel beams is evaluated in terms of the reduction in load-carrying capacity of the damaged state versus the undamaged stage. The beam specimens are sized and configured to have similar frequency response observed in actual bridges. Each beam is subjected to different damage scenarios, i.e., various types, locations, and degree of damage. For each scenario, the characteristics of the dynamic response are determined from acceleration measurements made during low-level free-vibration tests. A finite-element model was used to calculate the dynamic response of the simulated damaged structures. The experimental and analytical results are scrutinized to correlate increasing deterioration and changes in the dynamic response. Good correlations are found between the experiments and analysis. For each damage scenario, the yield and ultimate-load capacities of the test beams are defined by a damage index. Both experimental and numerical results indicate that there is less than 10% change in the natural frequencies at critical damage state.

INTRODUCTION

A structure's dynamic behavior is defined by a discrete spectrum of an infinite number of natural frequencies and corresponding mode shapes, which are determined by geometry, distribution of mass, stiffness, and boundary conditions. Within these parameters, changes in stiffness are directly related to changes in the safety condition of the structure. Therefore, the problem in this study concentrates on identifying damage in the structure by comparing measured natural frequencies and mode shapes.

Numerous studies have indicated that increase in damage reflects decreased natural frequency of a structure. Adams et al. (1978) and Cawley and Adams (1979) developed a technique to determine the presence and location of damage in simple beams and plates using changes in modal parameters. Initial efforts (Adams et al. 1978) focused on identifying damage in beams. The location of damage (consisting of a reduction in cross section) was accurately determined in a rectangular aluminum rod from changes in the rod's fundamental frequencies of axial vibration. Cawley and Adams (1979) recognized that the modes of vibration of a damaged beam are affected differently from the location of damage. They specifically utilized the relative change in magnitude of each modal frequency between successive tests to estimate damage location. Tests were conducted on an aluminum beam, a glass-fiber-reinforced plastic tube, a tapered steel bar, and a camshaft. In all cases, the location of transverse damage in the object was successfully determined from the modal frequencies of transverse vibration. The investigators concluded that removal of 1% of the cross-sectional area of the element could be detected using their equipment.

The problem of detecting damage in simple beams was also addressed by Rehm et al. (1987). Damage was inflicted on a simply supported, wide-flange steel beam by notching the center of the specimen. The dynamic response of the beam was characterized for several different lengths and depths. The investigators concluded that a 30% reduction in cross section over 2% of the span was readily detectable by dynamic characterization. At this level of damage, the fundamental frequency of vibration of the beam decreased from 7.041 to 6.998 Hz (less than 1% change). No comment was made with respect to the effect of damage on the other frequencies of vibration or the mode shapes of the beams.

A relatively complex beam was studied by DeWolf et al. (1988). The beam, 4.57 m (15 ft) long, consisted of an I-shaped section assembled from aluminum angles and bars. The section was purposely cracked at some point along its length resulting in a 26% reduction in the moment of inertia at that location. This damage resulted in 10%, 0.6%, and 5.7% decreases in the frequencies of the first three modes of vibration.

OBJECTIVES

One specific objective of the present study is to develop a relationship between changes in the frequencies of vibration of a structure and its level of damage, and thus express the threshold of damage detection in terms of the residual safety of the structure. A second objective is to determine if the damage locations could be identified from changes in the mode shapes of vibration of the structure.

To accomplish the objectives of the program, two simple beams designed to respond dynamically similar to full-size bridges were progressively damaged, and their dynamic response (frequencies and mode shapes of vibration) was determined at several damage levels. The dynamic response of the beams was excited by a drawdown/quick release, and determined from the acceleration measurements of the ensuing free vibration of the beams. The safety of the beams at each damage increment was expressed as a function of the ratio of the elastic design capacity of the undamaged structure to the plastic capacity of the damaged structure. A simple beam vibration equation was used to model the response of the undamaged beams. The response of the damaged beams was studied using the SAP-IV finite-element program.

EXPERIMENTS

In order to choose a specimen with similar dynamic characteristics to those found in actual bridges, two simply supported C 3 × 4.1 steel channel sections 3.66 m (12 ft) long were selected as the experimental specimens. The first flexural frequency of the beams was in the range of 5–10 Hz, similar to that of an actual bridge. The selection of a channel section permitted the simple infliction of damage through symmetrical removal of sections of the flanges. Two identical beams were selected and subjected to common damage scenarios of interest. The first beam, (Fig. 1 and Table 1), was subjected to severe localized damage and investigated as a

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[1] Assoc. Prof., Dept. of Civ. Engrg., West Virginia Univ., Morgantown, WV 26506.

[2] Prof., Dept. of Civ. Engrg., West Virginia Univ., Morgantown, WV.

[3] Civ. Engr., California Dept. of Transp., San Bernardino, CA; formerly, Res. Asst., West Virginia Univ., Morgantown, WV.

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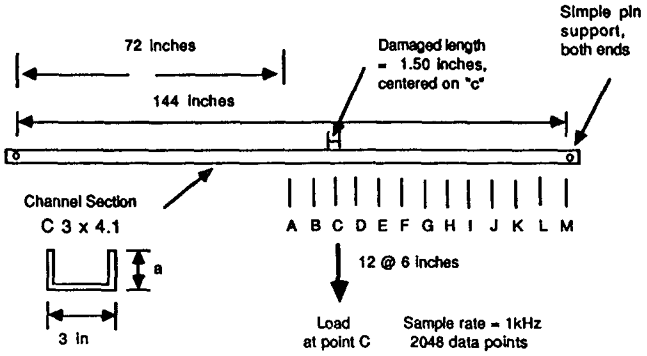

FIG. 1. Configuration for Experimental Investigations Conducted on Beam 1 (1 in. = 2.54 cm)

TABLE 1. Testing Procedures Conducted on Beam 1

Run (1)	Percent of plastic moment capacity (2)	*a* mm (3)	*a* in. (4)
1	100	35.8	1.41
2	80	30.2	1.19
3	60	24.9	0.98
4	50	22.1	0.87
5	40	19.3	0.76
6	30	16.3	0.64
7	20	13.0	0.51
8	10	8.1	0.32

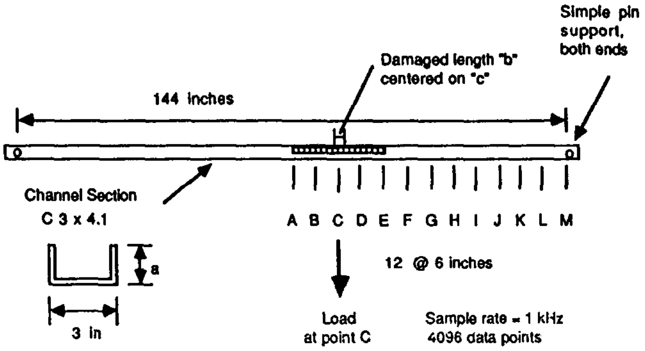

FIG. 2. Configuration for Experimental Investigations Conducted on Beam 2 (1 in. = 2.54 cm)

TABLE 2. Testing Procedures Conducted on Beam 2

Run (1)	Percent of plastic moment capacity (2)	*a* mm (3)	*a* in. (4)	*b* mm (5)	*b* in. (6)
11	100	35.8	1.41	38.1	1.5
12	80	32.8	1.29	38.1	1.5
13	70	30.2	1.19	38.1	1.5
14	60	27.4	1.08	38.1	1.5
15	60	24.6	0.97	38.1	1.5
151	60	24.6	0.97	76.2	3.0
152	60	24.6	0.97	114.3	4.5
153	60	24.6	0.97	152.4	6.0
154	60	24.6	0.97	304.8	12.0
155	60	24.6	0.97	609.6	24.0

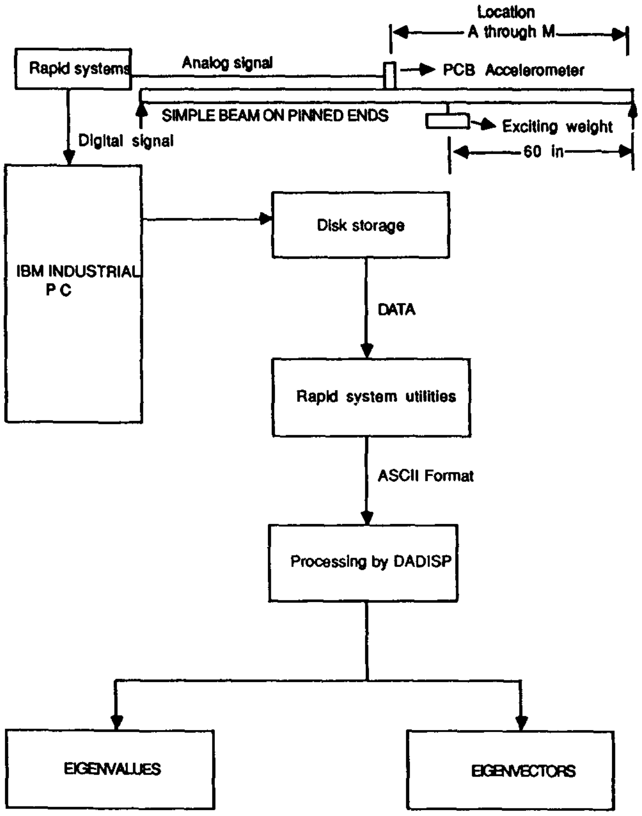

FIG. 3. Representation of Data Acquisition and Processing (1 in. = 2.54 cm)

TABLE 3. Comparison of Theoretical, Experimental, and FEM Frequencies (Hz) for Undamaged Specimen

Modes (1)	I (2)	II (3)	III (4)	IV (5)
Theoretical	6.09	24.34	54.78	97.38
FEM	6.08	23.4	54.69	97.22
Experimental	6.11	23.46	52.79	92.49

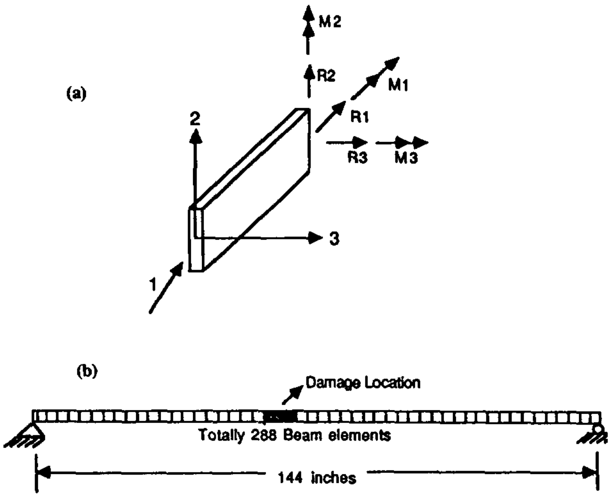

FIG. 4. Finite-Element Analysis (SAP IV) of Damaged Beam: (a) Three-Dimensional Beam Elements with Six Degrees of Freedom; (b) Location of Damaged Elements (1 in. = 2.54 cm)

safety problem. A notch, 38 mm (1.5 in.) long was cut and the depth of notch was increased in successive test runs, with a 10–20% reduction in the plastic moment capacity of the section in each run as indicated. The second beam, (Fig. 2 and Table 2) was subjected to a generalized, but less severe damage and addressed as a serviceability/maintenance problem. Initially a notch of 38 mm (1.5 in.) was inflicted and investigations were conducted with increase in depth of notch to 60% of the original plastic capacity, and later with the depth constant, the length of the notch was increased for

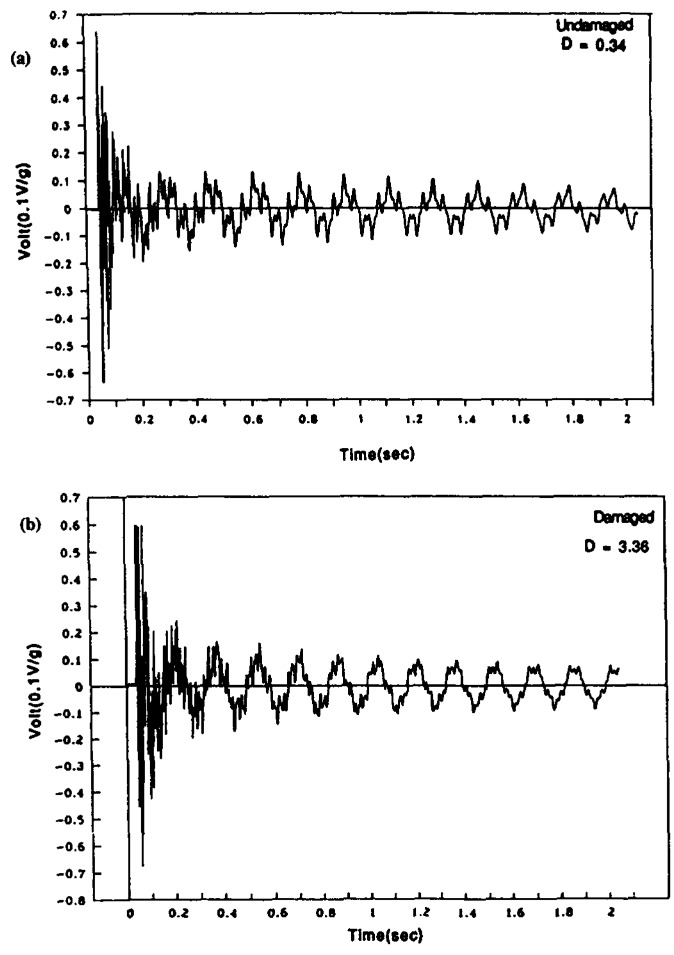

FIG. 5. Acceleration Time Histories for Beam 1 from Undamaged State to Severely Damaged State

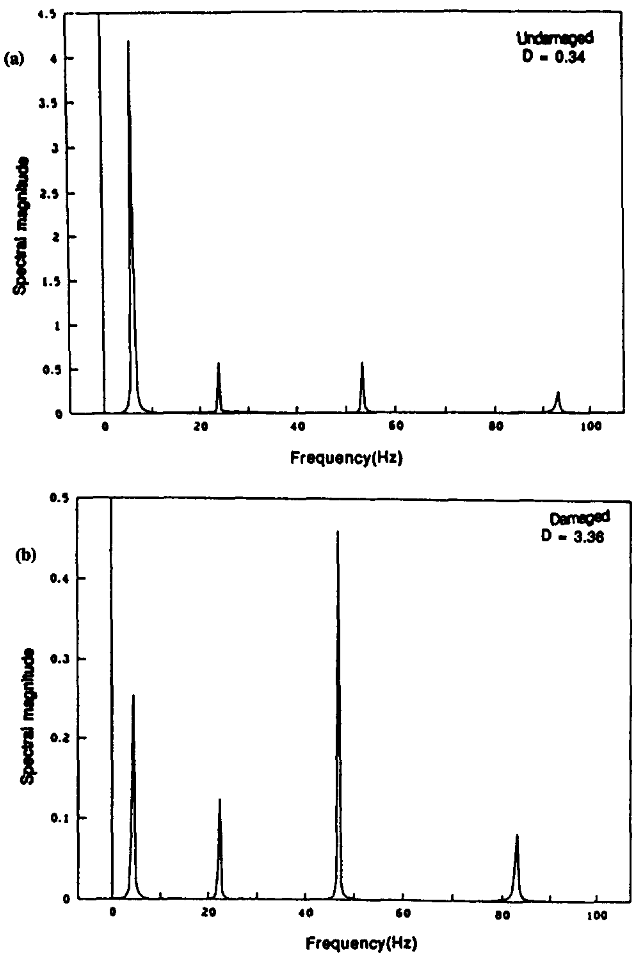

FIG. 6. Spectral Magnitudes for Beam 1 from Undamaged State to Severely Damaged State Showing Dominant Modes

successive test runs.

In both cases, the damage location was selected to potentially influence the dynamic response of the beams at several odd and even lower modes of vibration (that is, the damage location was well removed from the node points for the lower modes of vibration). The reduction in the plastic capacity of the section was evaluated using the plastic section modulus of the beam at the damaged location. The plastic section modulus was first calculated for an undamaged section, and in later stages a percentage of the plastic section modulus was selected to arrive at the exact reduced dimensions, resulting in a reduction of the plastic capacity of the section.

INSTRUMENTATION

An IBM industrial computer and a Rapid System R360, real-time spectrum analyzer were used. The analyzer consists of the R300 digital processing (DSP) interface board, the Rapid System four-by-four digital oscilloscope peripheral and the turnkey R360 Real Time Spectrum Analyzer software. The spectrum analyzer utilizes four simultaneous acquisition channels. The measuring device is a PCB accelerometer (PCB Piezotronics, Inc. 1989), Model 348A09, with a magnetic base and a mountable base. The accelerometer sensitivity is 100 mV/g with a useable frequency range from 1 to 2,000 Hz.

TEST PROCEDURES

The test specimen was excited by quick release of a suspended weight. The standard test procedures, starting with the physical experiment and concluding with the extraction of the dynamic response characteristics of the structure, is shown schematically in Fig. 3. Responses of the specimen were acquired by mounting a PCB accelerometer with a magnetic base at predetermined locations (A through L) along the beam specimens as shown in Figs. 1 and 2. Data in the form of acceleration histories were collected at one half of the span at 152 mm (6 in.) intervals, specimens being excited by releasing a weight hanging at the damage location. The location of the excitation/load point was selected to excite several of the low even and odd vibration modes of the structure. Several investigators have observed that the frequencies of vibration of a structure are related to the level of the excited response (Trifunac 1972; Ellis and Jeary 1980; Foutch and Housner 1977). Thus, the magnitude of the weight in the tests was adjusted as increasing damage was inflicted on the structure. The weight was decreased to ensure that elastic stress levels were maintained throughout the damaged structure under this load, and the magnitude of the excited response did not increase dramatically as the structure softened with increasing damage.

The acceleration response measured during each test was processed, and collective results were scrutinized. The response at each point was viewed as superposition of contributions from the various modes of vibration. The accelerometers were sensitive and capable of acquiring minor disturbances together with the intended vibration response. A linear regression was first performed on the data to remove any drift.

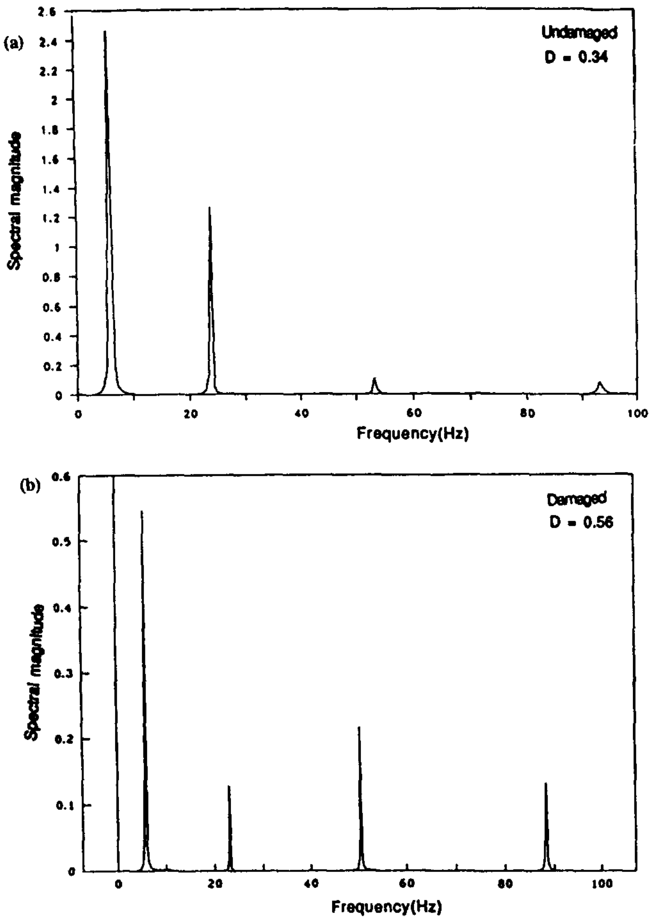

FIG. 7. Spectral Magnitudes for Beam 2 from Undamaged State to Severely Damaged State Showing Dominant Modes

DATA PROCESSING

A detailed format involving the data processing is also shown in Fig. 3. The processing involved reformatting the data by the Rapid System utilities. A signal-analysis software, DADiSP was used for waveforms processing. Processing consisted of obtaining power spectral density (PSD)/magnitude plots in the frequency domain using a fast Fourier transform (FFT) algorithm. The PSD is magnitude squared of the first half of the FFT and the factored-out length of the signal. This aids in comparison of frequency spectra of different signals and considered to be the power of the signal at a particular frequency. The accelerometer, being sensitive, was capable of acquiring minor disturbances. A linear regression was performed on the data to remove any noise, and to eradicate any baseline shift as can be seen in one of the time histories. Power spectral magnitude plots were obtained for various test configurations. The results were in terms of spectral magnitudes and frequencies.

THEORETICAL ANALYSIS

The eigen properties of an intact beam with uniform material properties can be obtained through the closed-form solution of the equation of motion. The beam was assumed to have uniform properties along the length. Considering an elementary case that neglects shear, rotary inertia, and axial-force effects, the solution for free vibration of a prismatic beam is obtained by solving the equation of motion

$$EI\frac{d^4y}{dx^2} + m\frac{d^2y}{dt^2} = 0 \qquad (1)$$

For a simply supported single span beam, the natural radial frequencies, ω_n, are given by

$$\omega_n = n^2\pi^2\sqrt{\frac{EI}{mL^4}} \qquad (2)$$

where E = modulus of elasticity for steel; I = moment of inertia about the weak axis; m = uniform mass per unit length of beam; and L = length of the beam. The vibrating mode shapes are given by

$$\phi_n(x) = A_n \sin\frac{n\pi x}{L} \qquad (3)$$

The theoretical frequencies for n = 1, 2, 3 and 4 are tabulated in Table 3 for an undamaged specimen in terms of cycles per second (Hz) which can be calculated as $\omega_n/2\pi$.

On the damaged specimens, dynamic analysis was performed using the SAP IV finite-element package. Three-dimensional beam elements with translational degrees of freedom in the vertical direction were used to model the beam specimens. The mesh generator in SAP IV allowed the provision of 288 elements uniformly distributed along the span as shown in Fig. 4. The damaged cross-sectional properties were incorporated in the analysis at identical locations in the model.

Comparison of the natural frequencies of the theoretical and finite-element method (FEM) analysis of an undamaged beam is shown in Table 3 and the existence of a good correlation between the two can be seen. Material properties corresponding to a standard steel section provided in the analysis for the beam specimens are as follows:

Young's modulus, E = 200 GPa (29 × 10^6 psi)
Moment of inertia about x-axis, I_x = 69 cm^4 (1.66 in.^4)
Moment of inertia about y-axis, I_y = 8.2 cm^4 (0.197 in.^4)
Torsional constant, J = 0.03 in^4 (1.25 cm^4)
Poisson's ratio, ν = 0.3

TABLE 4. Comparison of FEM and Experimentally Measured Frequencies (Hz) for Beam 1

Depth of Cut *a*		Damage index	Mode 1		Mode 2		Mode 3		Mode 4	
mm (1)	in. (2)	*D* (3)	FEM (4)	Measured (5)	FEM (6)	Measured (7)	FEM (8)	Measured (9)	FEM (10)	Measured (11)
35.8	1.41	0.34	6.076	5.859	24.30	23.93	54.69	53.23	97.22	93.32
30.2	1.19	0.42	6.066	5.859	24.29	23.93	54.64	53.23	97.10	93.24
24.9	0.98	0.56	6.043	5.859	24.27	23.93	54.52	52.73	96.82	91.80
22.1	0.87	0.67	6.035	5.859	24.26	23.93	54.48	52.73	96.73	91.80
19.3	0.76	0.84	6.030	5.859	24.26	23.44	54.45	52.25	96.67	90.82
16.3	0.64	1.11	6.029	5.371	24.26	23.44	54.40	51.27	96.62	89.36
13.0	0.51	1.68	6.028	5.371	24.23	22.95	54.39	49.81	95.10	86.91
8.1	0.32	3.36	4.143	4.395	22.77	22.46	48.51	46.88	86.60	83.01

TABLE 5. Comparison of FEM and Experimentally Measured Frequencies (Hz) for Beam 2

Depth of Cut *a* mm (1)	Depth of Cut *a* in. (2)	Width of Cut *b* mm (3)	Width of Cut *b* in. (4)	*b*/*L*·100 (%) (5)	Mode 1 FEM (Hz) (6)	Mode 1 Measured (Hz) (7)	Mode 2 FEM (Hz) (8)	Mode 2 Measured (Hz) (9)	Mode 3 FEM (Hz) (10)	Mode 3 Measured (Hz) (11)	Mode 4 FEM (Hz) (12)	Mode 4 Measured (Hz) (13)
35.8	1.41	38.1	1.5	1.04	6.076	6.110	24.30	23.46	54.69	52.79	97.22	92.49
32.8	1.29	38.1	1.5	1.04	6.076	6.110	24.30	23.46	54.68	52.79	97.21	92.38
30.2	1.19	38.1	1.5	1.04	6.066	6.110	24.29	23.46	54.64	52.79	97.1	91.89
27.4	1.08	38.1	1.5	1.04	6.054	6.110	24.28	23.46	54.57	52.30	96.95	91.64
24.6	0.97	38.1	1.5	1.04	6.043	5.859	24.27	23.46	54.52	52.05	96.82	91.40
24.6	0.97	76.2	3.0	2.08	6.011	5.859	24.23	23.35	54.40	51.76	96.41	91.07
24.6	0.97	114.3	4.5	3.13	5.970	5.859	24.19	23.35	54.23	51.76	96.02	90.33
24.6	0.97	152.4	6.0	4.17	5.950	5.859	24.16	23.19	54.16	51.27	95.79	89.84
24.6	0.97	304.8	12.0	8.33	5.830	5.859	24.04	23.19	53.76	50.54	94.95	88.62
24.6	0.97	609.6	24.0	16.66	5.662	5.370	23.82	22.46	53.08	49.07	94.16	85.93

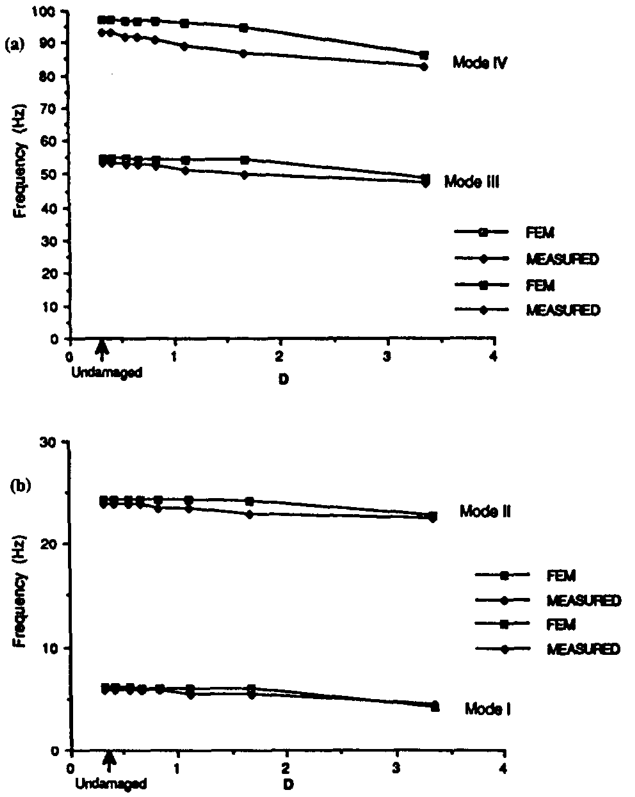

FIG. 8. Comparison of Experimental and FEM Calculated Frequencies for Beam 1

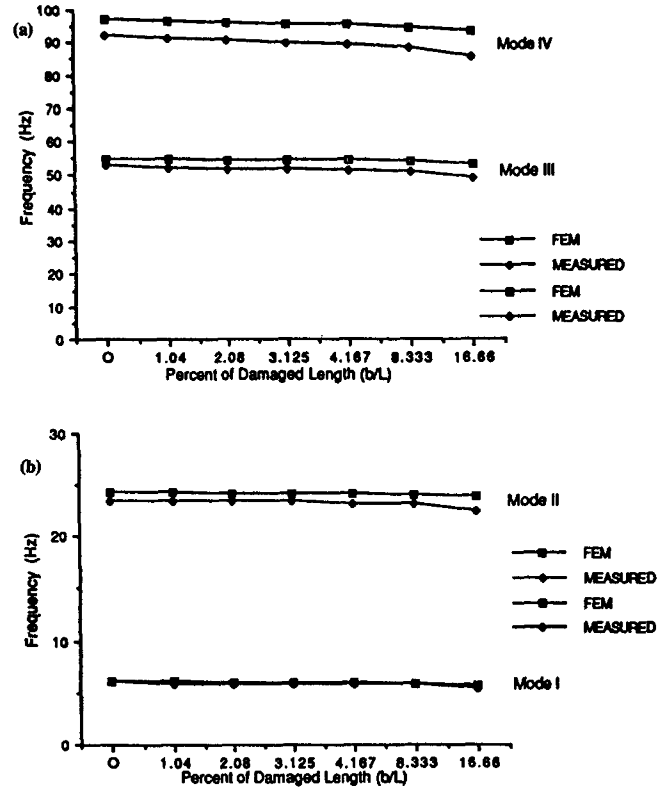

FIG. 9. Comparison of Experimental and FEM Calculated Frequencies for Beam 2

Cross sectional area, $A = 780 \text{ mm}^2$ (1.21 sq in.)
Mass density, $\rho = 7{,}840 \text{ kg/m}^3$ (0.000734 lb·sec^2/in.^{4})

EXPERIMENTAL RESULTS

The acceleration time histories for beam 1 of the undamaged and the severely damaged states are presented in Fig. 5. A sampling rate of 1 kHz, with 2,048 points was used in the acquisition of the data. The dominant spectral magnitudes generated from these time histories are presented in Fig. 6. As shown in Fig. 6, the first four modes of vibration can be clearly identified, and the corresponding magnitude of the natural frequencies for these first four modes are 5.859 Hz, 23.93 Hz, 53.23 Hz, and 93.32 Hz for the undamaged beam. For the severely damaged beam, the first four natural frequencies are 4.395 Hz, 22.46 Hz, 46.88 Hz, and 83.01 Hz. A sampling rate of 1 kHz, with 4,096 points was used in the acquisition of the data for the second beam. Spectral magnitudes showing the dominant modes for beam 2 of the undamaged and the severely damaged states are shown in Fig. 7. As expected, the results indicated a reduction in the natural frequencies with increase in damage.

Efforts were made to relate observed changes in the dynamic response of the beams with increasing damage. To accomplish this objective, a rational measure for structural damage has to be selected and evaluated for each damage condition. There are no generally accepted measures of structural damage. For this study, the ratio of the design capacity of the undamaged structure to the plastic capacity of the damaged structure was selected as a measure of the level of damage in the structure indicated as D, and refer to as damage index

$$D = \frac{\text{Design Strength}}{\text{Plastic Strength}} \tag{4}$$

Using this approach, a damage index of 1.0 indicates that at application of the original design load, formation of a plas-

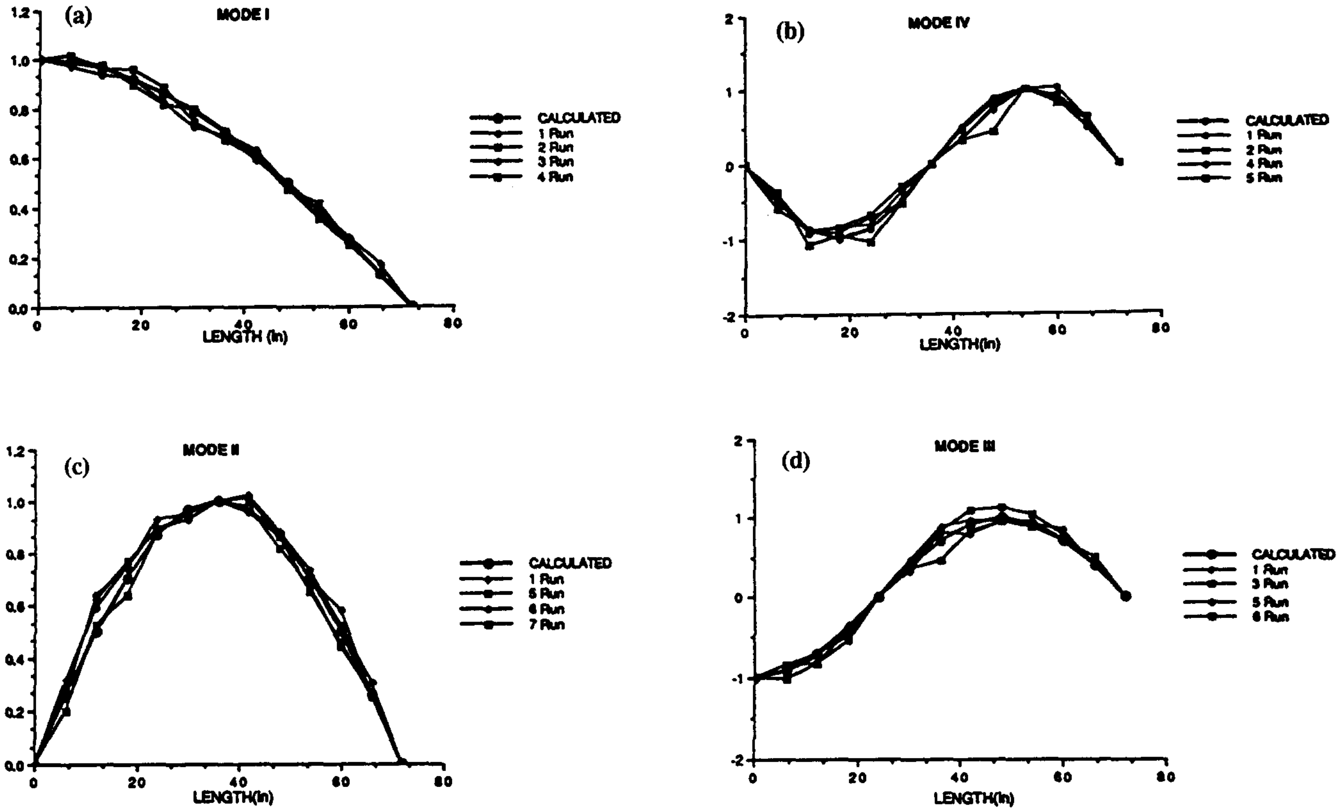

FIG. 10. Calculated and Measured Mode Shapes of First Four Dominant Modes for Beam 1 (1 in. = 2.54 cm): (a) Mode 1; (b) Mode 4; (c) Mode 2; (d) Mode 3

tic hinge in the structure is imminent. Thus, a damage index greater than one indicates that a collapse mechanism will form if the structure is subjected to the original design load.

For the simply supported beams considered herein, the plastic capacity of the damaged structure was simply calculated as the cross-section capacity necessary to cause formation of a plastic hinge at the center of the damaged area. The design capacity was calculated as the load that would generate a maximum flexural stress in the beam equal to 66% of the yield stress (this allowable elastic stress is nominally consistent with the provisions of the 1989 Allowable Stress Design Code for steel published by the American Institute of Steel Construction). Following this approach, the index, D, for the undamaged beam was 0.335.

COMPARISON OF RESULTS AND COMMENTS

Variation of frequency with increasing damage is also shown in Table 4, for the first beam. Frequencies were calculated by modeling the specimen incorporating various levels of inflicted damage at similar locations as those that of the experiments, using the FEM package SAP IV. A comparison between the experimentally measured and FEM-calculated frequencies is presented for the first four modes in Table 4. The difference between the two can be attributed to the fact that a low frequency resolution was used in the experiment of the first beam. A frequency resolution of 0.488 Hz was used in the first beam, whereas a frequency resolution of 0.122 Hz was used in the second beam. As can be seen in Tables 4 and 5, the measured and calculated mode 4 frequencies seem decreasing sensitively responding to the increasing damage.

It should be important to know that, as shown in Tables 4 and 5, although both the FEM analysis and the measurements are sensitive to the inflicted damage, the frequency differences from the undamaged to the severely damaged states are very small. The first four modal frequencies of the first beam at various levels of inflicted damage are plotted as a function of the corresponding damage index in Fig. 8. As expected, the frequencies of vibration decrease as the damage index increases. One case of definite interest is the point at which the structure has a damage index of 1.0. A useful damage detection scheme should be able to detect damage before the structure would fail if subjected to the design load. At this damage state (D = 1.0), the first four modal frequencies of vibration decreased, respectively, by only 4.9%, 2.0%, 2.9%, and 3.6% from the undamaged state. These values have been determined from the measured response of the beam, and the results obtained using the finite-element response are even smaller. These changes of natural frequency appear fairly small in magnitude. Ellis and Jeary (1980), for example, noted that up to a 3% variation might be expected for frequencies estimated for a structure under incidental/ambient vibration conditions. Askegaard and Mossing (1986) further demonstrated that up to a 10% variation could occur in the frequencies of vibration of a bridge over a 1 year period, without any apparent change in the condition of the structure. In light of these variabilities, the less than 5% change in the frequencies of vibration of the test articles, corresponding to a critical reduction in safety condition, may be insufficient as a useful indicator in predicting damage. Note that even in the extreme case of a damage index of 3.33, which corresponds to incipient plastic hinge formation, the first four frequencies of vibration decreased only, 25.0%, 6.1%, 11.9%, and 11.1%, respectively, from the undamaged to this severely damaged state.

Higher frequency resolution shows better agreement between the measured and the finite-element frequencies in case of beam 2 as shown in Table 5. In the last five runs of the tests conducted on beam 2, the damage index had approximately a value of 0.5597 with a width of cut equal 38.1 mm (1.5 in.). After this level of damage, the reduction in section was extended longitudinally along the structure. The changes in the frequencies of vibration of the second beam are plotted as a function of damaged length in Fig. 9. Naturally, as the damaged length increased, the structure softened and the frequencies of vibration decreased. At the maximum extent of damage considered, equal to 17% of the total length of the structure, the frequencies of the first four modes of vibration decreased, respectively, by 12%, 4.3%, 7.0%, and 7.1% from the undamaged to that damaged state.

It is obvious now, as seen from Figs. 8 and 9, the accuracy and the effectiveness of using free-vibration experiments to detect the damage should be questioned. Our results indicate that the reduction in the natural frequencies of a critically damaged structure is too small. Therefore, in a real-world situation while the frequency variation can be as high as 5–10%, the critical damage in structures may not be accurately detected by the measured natural frequency.

In Figs. 8 and 9, the difference between the measured and FEM results for the higher modes will decrease if shear effects are considered in the FEM model. A detailed study on the effects of shear deformation on natural frequencies presented in Spyrakos (1994) shows that shear effects can lead to a substantial reduction of the natural frequencies that are higher than the third flexural mode of vibration.

An effort was made to locate the position of the inflicted damage on the beam by the aid of the vibrating mode shapes. The vibrating mode shapes of the beam at various levels of damage were obtained from processing the data. The mode shapes were generated by normalization and are presented in Fig. 10. For example the amplitude of the first mode was normalized with the amplitude of the first mode at the midspan where the maximum of the vibrating mode shape occurs. Fig. 10 shows the mode shapes extracted from measurement at point A through M (as indicated in Figs. 1 and 2) along the span of the beam. The expectation was that with increasing in damage the beams would vibrate with a higher amplitude at higher damage levels. Though the preceding phenomenon can be seen from the mode shapes in some instances (for example run 2), it was not consistently observed. Our results indicated that the mode shapes of the free vibration are not sensitive enough to identify the localized damage.

It is noted that the length of the artifically inflicted damage on the specimen is short compared to the length of the beam. It is possible that a very high mode could have nodes spaced at a length that is shorter or at least comparable to the length of the damaged section. However, such modes would be very difficult to excite with the quick release method used in this study. Theoretically, a high mode excitation is possible by using impact or ultrasonic pulser to generate high-frequency loading functions in order to identify local damage. Nevertheless, there are numerous practical problems in achieving such tasks, such as noises, excitation locations, structure's complexity, and boundary conditions.

CONCLUSIONS

The experimental and numerical results indicate a reduction in frequencies with increase in damage. The numerical results were obtained from a dynamic analysis using a finite-element modeling of the specimens for various levels of inflicted damage. A comparison between the experimentally measured and calculated frequencies agrees well for the first four modes. Both numerical and experimental results indicate that a critical damage index, namely a damage factor equal to 1.0, shows only less than 10% change in the frequencies of vibration. This leads us to believe that the information of the natural frequency change is insufficient to be a useful indicator of structural safety of actual bridges. However, for a severely damaged structure (damage index greater than 2.0), the dynamic characterization method appears to be a simple technique capable of providing information about the remaining serviceability of the damaged structure. The mode shapes of the damaged structure were investigated and the results indicate that the damage location could not be identified by the measured free-vibration mode shapes. In conclusion, more research is needed to develop nondestructive testing techniques to evaluate and characterize structural damage. Recent research studies of using strain mode shape and total strain energy have indicated to be more sensitive to quantify structural damage than the natural frequencies, and the writers would like to recognize some recent studies that show promise in resolving several of the issues addressed in his paper, e.g., Agabian et al. (1991), Hearn and Testa (1991), Mazurek and DeWolf (1990), Salane and Baldwin (1990), and Yao et al. (1992).

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