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LETTERS TO THE EDITOR

IS DAMAGE IDENTIFICATION USING VIBRATION DATA IN A POPULATION OF CYLINDERS FEASIBLE?

T. MARWALA AND H. E. M. HUNT

Department of Engineering, University of Cambridge, Trumpington Street, Cambridge, CB2 1PZ, England

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1. INTRODUCTION

Fault identification in mechanical and aerospace structures at the manufacturing stage offers substantial economic benefits. Vibration data have been employed with varying degrees of success to identify faults in structures [1–5]. However, in the literature there is little coverage of fault identification using vibration data from a population of structures.

Srinivasan and Kot [6] have studied the feasibility of using vibration data to identify faults in cylinders. These authors tested a cylinder, which had a machined notch, suspended by relatively soft springs to simulate free boundary conditions. The authors examined changes in the natural frequencies and mode shapes as a result of damage. They found that the presence of damage changes the vibration response of the cylinder.

In this study, the feasibility of using vibration data to identify faults in a population of 22 seam-welded cylindrical shells made of steel is assessed. Each cylinder is excited at various locations using an impulse hammer and vibration responses are measured using an accelerometer located at fixed position. Each cylinder is measured three times under different boundary conditions by changing the orientation of a rectangular sponge inserted inside the cylinder which is rested on bubble *wrap*, to simulate a free–free environment (see Figure 1). The number of sets of measurements taken for undamaged population is 66 (22 cylinders $\times$ 3 for different boundary conditions). Each cylinder is divided into three equal substructures and holes of 10–15 mm are introduced at the centres of the substructures. The total number of measurements taken for cylinders with holes is 66. From the measured vibration data frequency response functions (FRFs) are calculated. Modal properties, i.e., natural frequencies and mode shapes, are then extracted from the FRFs using modal analysis. Mode shapes are transformed into the co-ordinate modal assurance criterion (COMAC) [7] by computing the correlation between mode shape matrix from each fault case and the median mode shape matrix of a population of undamaged cylinders. Modal energies are then calculated from the FRFs by determining the integrals of the real and imaginary components of the FRFs over chosen bandwidths that bracket the natural frequencies [4]. Similarly, modal energies are transformed into the co-ordinate modal energy assurance criterion (COMEAC) by calculating the correlation between each measured modal energy matrix and the median of the modal energy matrices from population of undamaged cylinders.

By comparing the modal properties, modal energies, the COMAC and the COMEAC between undamaged and damaged populations the feasibility of using vibration data for fault identification is assessed. The changes of these parameters as a result of faults are

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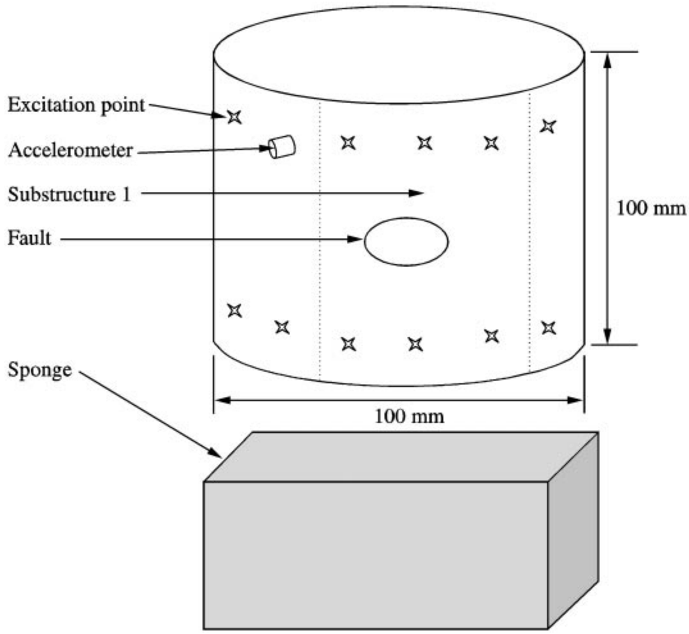

Figure 1. Cylindrical shell of 1·75 mm thickness.

investigated taking into account the following issues: (1) changes in modal properties or modal energies resulting from the presence of faults compared with those resulting from variation in measurements and in physical properties of the population of cylindrical shells; (2) uncertainty in measurement positions; (3) errors introduced during modal analysis; and (4) changes in support conditions and environmental conditions.

2. PARAMETERS TO BE MONITORED

The modal properties are extracted using the Structural Dynamics Toolbox [8], which runs in MATLAB [9] environment. The modal energies are defined as the integrals of the real and imaginary components of the FRFs over frequency ranges that bracket the natural frequencies of the system (in this study these ranges span over $\pm 6\%$ of the natural frequencies). The modal properties and modal energies to be used for the assessment of damage identification are chosen by employing these five steps: (1) find the means and standard deviations of the modal properties and modal energies at each index for data from undamaged and damaged cylinders (e.g., mode 5 co-ordinate 3 will be assigned its own index number); (2) calculate the difference between the means of data from undamaged and damaged cylinders at each index; (3) calculate the average of the standard deviations from undamaged and damaged cylinders while keeping track of the indices; (4) calculate the ratio between the mean-differences in step 2 to the average-standard-deviations in step 3 at each data index; and (5) from these ratios, select 19 indices with the highest ratios and assess their corresponding data.

The COXAC is a criterion that measures the correlation between two sets of data of same dimension. The COXAC for co-ordinate i between the measured data x_m and the median data for undamaged structures x_{MED} is

$$COXAC(i) = \frac{\left(\sum_{j=1}^{n} |x_{MED}(i,j)\, x_m^*(i,j)|\right)^2}{\sum_{j=1}^{n} |x_{MED}(i,j)^2 \sum_{j=1}^{n} |x_m(i,j)|^2}, \qquad (1)$$

where * is a complex conjugate. If x in equation (1) is substituted by mode shape vector then the COXAC is a familiar COMAC [9], whereas if it is substituted by the modal energies it is called the COMEAC [10]. When x_m and x_{MED} are perfectly correlated then the COXAC for all degrees of freedom is 1. Otherwise, when perfectly uncorrelated then the COXAC for all degrees of freedom is 0.

The most reliable and sensitive modal energies and modal properties as well as the COMEAC and the COMAC are compared.

3. EXPERIMENTAL EXAMPLE

In this section, an impulse hammer test is performed on each of the 22 steel seam-welded cylindrical shells ($1.75 \pm 0{\cdot}02$ mm thickness, $101{\cdot}86 \pm 0{\cdot}29$ mm diameter and of height $10{\cdot}150 \pm 0{\cdot}20$ mm). These cylinders are supported by inserting a sponge and rested on bubble *wrap*, to simulate a free–free environment (see Figure 1). The types of faults that are introduced to the structures do not influence damping significantly. The orientation of the sponge (which is not circular) is oriented in three different positions to simulate changes in boundary conditions.

The structure is excited using a modal hammer of sensitivity of 4 pC/N, with the head mass of 6·6 g, and cut-off frequency of 3·64 kHz. The response is measured using an accelerometer with a sensitivity of 2·6 pC/m s^{-2}, which has a mass of 19·8 g. Conventional signal processing procedures are applied to convert the time domain impulse history and response data, into frequency domain. The data in the frequency domain are utilized to calculate the FRFs. From the FRFs modal properties are extracted using modal analysis and modal energies are obtained by finding the integrals under the peaks.

The impulse is applied at 19 different locations (see Figure 1), nine on the upper ring of the cylinder and 10 on the lower ring of the cylinder. For each fault case, measurements are taken by measuring the acceleration at a fixed position and roving the impulse position. Some of the problems that are encountered during impulse testing include difficulty to excite the structure at an exact the position (especially for an ensemble of structures) and that the direction of the hammer cannot be accurately repeated. The same procedure is repeated for all the cylinders. A hole of about 12 mm in diameter is introduced to each cylinder as shown in Figure 1.

4. RESULTS AND DISCUSSION

The ensemble of 20 cylinders and their respective fault cases are measured and the results, showing the frequency response functions are shown in Figure 2. In Figure 2 it may be observed that the repeatability of the measurements of the FRFs is generally good at certain low frequencies and as expected become poor at high frequencies. The presence of an accelerometer and the imperfection of cylinders destroy the axis-symmetry of the structures. The incidence of repeated natural frequencies is destroyed.

The average and the sample standard deviation of the natural frequencies for both damaged and undamaged cylinders are shown in Table 1 and are compared to the results from the finite element (FE) model (which includes the accelerometer) using the ABAQUS

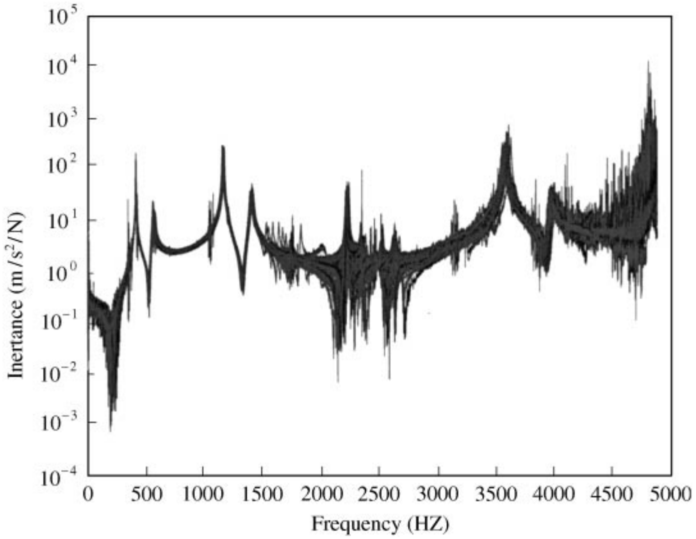

Figure 2. Measured frequency response functions of a population of structures.

TABLE 1

Natural frequencies for both damaged and undamaged cylinders key: FE = finite element

Mode no.	FE results (Hz)	Average (f_n) undamaged (Hz)	Standard deviation (σ) undamaged (Hz)	Average (f_n) damaged (Hz)	Standard deviation (σ) damaged (Hz)
1	433·3	413·7	1·5	412·9	1·9
2	445·5	425·3	3·2	425·1	1·9
3	587·5	561·0	2·4	558·7	2·8
4	599·0	576·6	3·0	576·9	2·8
5	1218·3	1165·0	5·5	1164·6	6·0
6	1262·9	1196·8	6·9	1196·8	7·2
7	1480·0	1408·1	5·7	1404·4	6·3
8	1510·0	1483·4	73·5	1463·4	52·5
9	2273·5	2229·3	11·0	2224·7	11·5
10	2323·6	2346·2	12·6	2360·4	17·5
11	2422·3	2520·1	9·6	2511·4	13·8
12	2657·4	2612·1	39·9	2630·1	10·5
13	2711·3	—	—	—	—
14	2778·4	—	—	—	—
15	3713·7	3330·2	96·5	3239·7	113·3
16	3914·3	3585·8	12·1	3580·7	22·4
17	4138·5	3990·6	16·6	3983·8	13·5
18	4222·8	4309·5	21·2	4316·8	21·0

software [11]. The FE model has 1001 shell elements with 3100 nodes. In Table 1 it is shown that for undamaged cases the average natural frequency for mode 1 is the most repeatable and in order of repeatability followed by 3, 4, 2, 5, 7, 6, 11, 9, 26, 10, 17, 18, 12, 8 and then mode 15. For the damaged cases the average natural frequency for model 1 is the

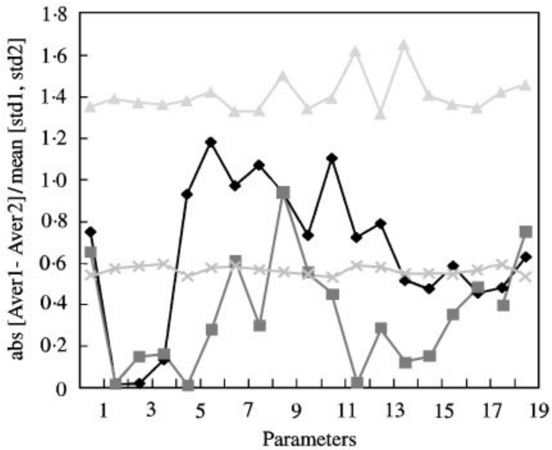

Figure 3. Graph showing the ratio of standard deviations of damaged oparameters to that of undamaged parameters Key: abs = absolute value of; Aver1 = average of undamaged population; Aver2 = average of damaged population; std1 = standard deviations of undamaged population; std2 = standard deviations of damaged population: —◆— COMEAC; —■— COMAC, model energy; -×- modal property.

most repeatable, then 2, 3, 4, 5, 6, 12, 9, 17, 11, 10, 18, 16, 8 and then mode 15. These trends are observed by comparing the magnitudes of the sample standard deviations. In Table 1 the natural frequencies corresponding to modes 13 and 14 could not be identified. When the average natural frequencies for a population of undamaged cylindrical shells are compared to that of damaged cylinders, it is observed that none of the natural frequencies decrease by more than 2% except mode 15.

Modes 1, 2, 3, 4, 5, 6, 7, 9, 10, 11, 12, 15, 16 and 17 were used to calculate the COMAC while the first seven peaks were used to calculate the COMEAC. The ratios between the absolute value of the difference between the mean from undamaged and damaged cylinders, to the mean of the standard deviations of undamaged and damaged cylinders, are shown in Figure 3. This figure shows that, on average faults are mostly observed on the modal energies, followed by the COMEAC, then the modal properties followed by COMAC. The reason why modal properties are less indicators of damage than the modal energies is because modal properties require a significant amount of computation to extract than modal energies which require simple integration. This makes modal properties more susceptible to errors than modal energies. Furthermore, in calculating modal energies, determining the areas under the peaks have effect of smoothing out the noise from the frequency response functions. The reason why using raw data (modal energies and modal properties) are better indicators than the correlation criteria (COMEAC and COMAC) is because the correlation criteria average the changes as a result of faults while the raw data uses the exact changes.

5. CONCLUSION

Vibration data of a population of cylinders were measured and modal analysis was employed to obtain natural frequencies and mode shapes. Mode shapes were transformed into the COMAC. Furthermore, modal energies were extracted by calculating the integrals of the real and imaginary components of the FRFs over frequency ranges that bracket the natural frequencies of the system. Modal energies were transformed into the COMEAC. It is observed that using modal energies and modal properties directly is better than using the correlation criteria such as the COMEAC and COMAC respectively. It is also found that modal energies are better indicators of damage than modal properties.

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