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Artificial Excitation in Operational Modal Analysis

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ABSTRACT: In operational modal analysis, natural or operating loads are used to excite structures and mechanical systems, which represent an important advantage in applications where the use of artificial excitation devices can be considered expensive or impractical. However, when operational modal analysis is applied to structures located in the labs or places where the magnitude of the natural loads is relatively low, artificial devices must be used in order to make sure that the loading is multiple-input and stationary broad banded. In this work, the results of several modal tests performed on a steel cantilever beam and a steel plate located in the lab, using different artificial devices as excitation, are presented and compared.

1 INTRODUCTION

In Operational modal analysis the natural operating loads are used to excite the structures instead of using artificial devices such as shakers or impact hammers. In the case of civil engineering structures the wind or waves are natural forms of excitation which are stochastic in nature. Therefore, it cannot be described by an explicit time-dependent function, but must be characterized by certain statistical parameters (Anderson 1997). The response will also be stochastic and may also be represented by its statistical characteristics.

To clearly identify closely spaced modes and even repeated modes, the loading must be multiple-input so that Operational Modal Analysis is a Multiple Input Multiple Output technique.

When operational modal analysis is applied to structures located in the labs or places where the magnitude of the natural loads is relatively low, artificial devices must be used in order to reach a reasonable load magnitude which, additionally, must be multiple-input and stationary broad banded. Multiple input artificial loading is easy to apply compared to the work required to setup a forced vibration test: shakers have to be installed, forces have to be controlled and measured, etc. To achieve this objective, we have to make sure that at least one of two different types of loads produces a clear multiple-input is present (Brincker 2003):

- A loading that is moving over a large part of the structure. With this type of loading, the structure is excited at many points during the tests and it also help us ensure that all modes that are sensitive to the loading will be excited.
- A distributed loading with a correlation length significantly smaller than the structure. An example of this load is the wind acting along the height of a building or waves loading an offshore structure. To make sure that the wind load on a building is multiple-input, the correlation length of the wind loading must be significantly smaller than the width and height of the building.

The ideal force for operational modal analysis is stationary white noise. In practice, we just need to make sure that the loading is reasonable random in time and space. A procedure to approximate to this loading consists of applying many hits over a large part of the structure. Basically, we want the input spectrum to have sufficient excitation over the frequency range of concern. If the input spectrum were to completely drop off to zero, then the structure would not be excited at that frequency. If impacts are applied to a structure, the input power spectrum is controlled by the length of time of the impact pulse. A long pulse in the time domain, results in a short or narrow frequency spectrum. A short pulse in the time domain, results in a wide frequency spectrum.

The Fourier transform of an infinite unitary pulse train in time domain, with amplitude A and period T_0, see Fig. 1, is also an infinite pulse train in the frequency domain with amplitude $1/T_0$ and separated by $1/T_0$, i.e., with an infinite bandwidth (Newland 1993). However, in order to avoid harmonics and excite all modes in the frequency range of interest, the pulse train must be non periodic, i.e., the time interval between consecutive pulses must be random.

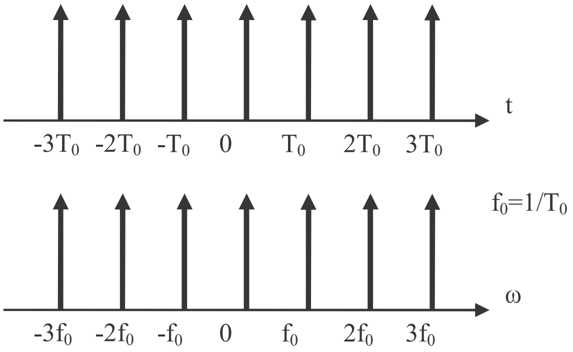

Figure 1: FFT of an infinity pulse train

An impact hammer, the fingers of both hands, a ball dropping repetitively over the structure and a pressure air stream were used in this work. An alternative method consists of moving some rough device across the surface of the structure, such as a hand metal file, a saw blade or a pencil, which can be considered small hits applied continuously to the structure surface.

In this work, the results of several operational modal tests performed on a steel beam structure and a steel plate located in the lab, using different artificial devices as excitation, are presented and compared.

2 EXPERIMENTAL PROGRAM

2.1 The Steel Beam Structure

Several operational modal tests were performed on a steel beam structure which consists of two welded 100x40x4 mm rectangular tubes. The shape and the dimensions are shown in Fig. 2b.

The responses were recorded in 18 DOF's using eight 4508B Brüel & Kjær accelerometers, located as is shown in Fig. 3b. Three reference accelerometers were used (indicated with letter R in Fig. 3b) in each of the 3 data sets carried out. The responses were recorded for a period of approximately 8 minutes at a sampling frequency of 2000 Hz and using a National Instruments data acquisition dynamic board, DSA PCI4472.

The structure was artificially excited moving the following devices along the bars of the structure: a hand metal file, a saw blade, repetitive hits using an impact hammer, idem using the fingers of both hands and repetitive pressure air streams.

2.2 The Steel Plate

Then, operational modal tests were performed on a steel plate which shape and dimensions are shown in Fig. 2a.

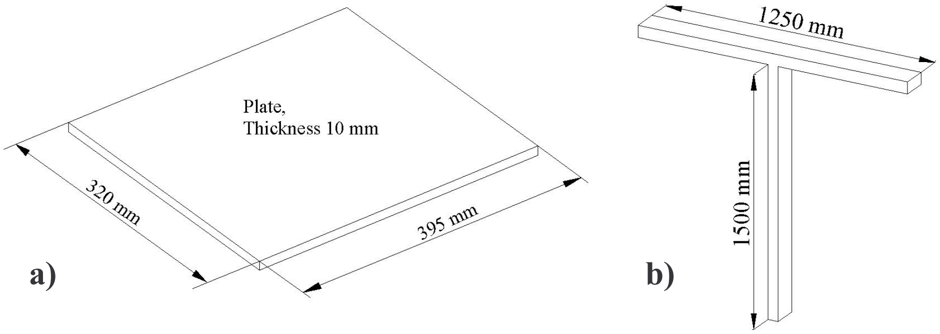

Figure 2: Dimensions of the beam structure and the plate

The responses were recorded in 19 DOF's using seven 4508B Brüel & Kjær accelerometers, located as is shown in Fig. 3a. Three reference accelerometers were used (indicated with letter R in Fig. 3a in each of the 4 data sets carried out. The responses were recorded for a period of approximately 2 minutes at a sampling frequency of 2000 Hz using a National Instruments data acquisition dynamic board, DSA PCI4472.

The structure was artificially excited with the following devices: moving a pencil across the plate, repetitive hits using an impact hammer, idem using the fingers of both hands, idem dropping a rubber ball and repetitive pressure air streams.

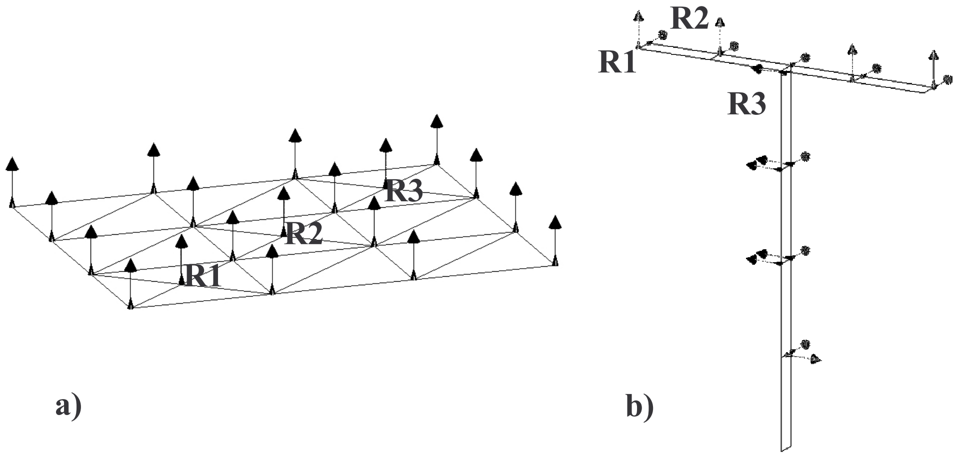

Figure 3: Location of the accelerometer in both structures

3 EXPERIMENTAL RESULTS

3.1 The beam Structure

The modal parameters were identified with both Enhanced Frequency Domain Decomposition (EFDD) (Brincker, 2001) and the Stochastic Subspace Identification (SSI) (Van Overschee, 1996). The first fifteen modes where considered in the investigation. The normalized singular values corresponding to tests using the hand metal file and air streams are presented in Fig. 4, where it can be observed that all modes were excited. The same can be concluded for the rest of the loading devices used to excite the structure.

The operational modal analysis was always performed using 2048 points, with an overlap of 66.6% and 4 projection channels. The first 7 modes were identified using a decimation of order 4. The natural frequencies estimated with EFDD and SSI are shown in Tables 1 and 3, respectively. The results show that all the artificial devices used to excite the structure provide good estimates, being the coefficient of variation (CV) less than 1%.

With regards to the damping ratios, the results are presented in Tables 2 and 4, respectively. The damping ratios are very low for all modes as it is common in steel structures. The results presented in Tables 2 and 4 show a high scatter, being the coefficient of variation (CV) less than 25% (except modes 5 and 11 identified with SSI which show coefficients of variation around 100%) which is considerably high compared with the dispersion obtained in the natural frequency estimates.

However, it is well known that the current operational modal identification techniques provide a good quality in mode shapes and natural frequencies, but a significant high scatter is obtained in the damping ratios (Magalhães 2007). Moreover, it must be emphasized that the quality of the estimates can be improved increasing the length of the time series.

The modal assurance criteria (MAC) between the mode shapes corresponding to air streams and saw blade excitations, is presented in Fig. 5a, whereas Fig. 5b shows the MAC between the mode shapes corresponding to the hand metal file and hands excitation. A good correlation between the data corresponding to the differences devices was obtained.

Table 1: EFDD natural frequencies.

Mode	File	Air	Hands	Hammer	Saw	Mean	CV
			Hz			Hz	%
1	8.67	8.673	8.64	8.66	8.667	8.66	0.173
2	16.29	16.3	16.61	16.29	16.27	16.35	0.885
3	27.08	27.13	27.10	27.11	27.1	27.10	0.067
4	42.16	42.16	42.16	42.19	42.18	42.17	0.034
5	116.40	116.4	116.30	116.4	116.4	116.38	0.038
6	120.70	120.9	120.70	120.8	120.5	120.72	0.123
7	150.40	150.4	150.30	150.4	150.4	150.38	0.03
8	294.80	294.9	294.60	294.80	294.80	294.78	0.037
9	319.60	319.7	319.60	319.60	319.60	319.62	0.014
10	467.90	468	467.60	468.10	467.70	467.83	0.047
11	527.80	527.8	527.70	527.70	527.80	527.76	0.01
12	614.10	613.8	613.70	613.70	613.90	613.84	0.027
13	711.80	712	711.20	711.40	711.90	711.66	0.048
14	783.80	783.7	783.70	783.70	783.80	783.74	0.007
15	932.50	932.5	932.50	932.40	932.40	932.46	0.006

Therefore, it can be concluded that the loading devices used in the operational tests can be used successfully to excite artificially this type of structures in the lab.

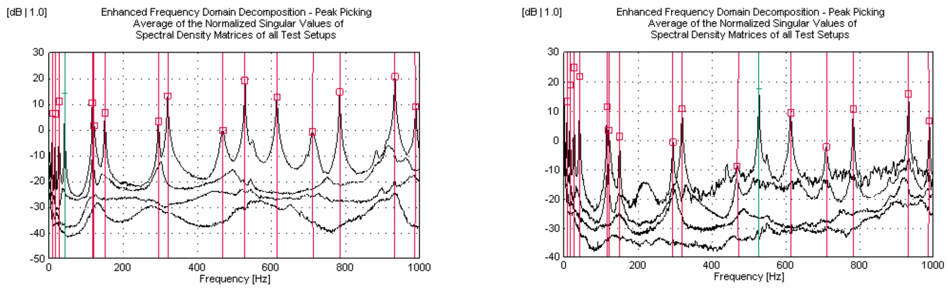

Figure 4. Normalized singular values corresponding to the hand metal file (left) and air streams (right)

Table 2: EFDD damping ratios.

Mode	File	Air	Hands %	Hammer	Saw	Mean %	CV %
1	1.112	1.098	1.294	1.250	1.152	1.181	7.334
2	0.557	0.515	0.646	0.549	0.541	0.562	8.883
3	0.495	0.450	0.572	0.491	0.465	0.495	9.525
4	0.521	0.623	0.500	0.500	0.552	0.539	9.504
5	0.379	0.265	0.333	0.271	0.331	0.316	15.105
6	1.079	1.062	0.983	1.042	0.827	0.999	10.267
7	0.498	0.400	0.462	0.377	0.447	0.437	11.103
8	0.543	0.509	0.515	0.509	0.532	0.522	2.918
9	0.360	0.286	0.324	0.334	0.334	0.328	8.143
10	1.138	1.05	1.047	1.243	1.095	1.131	7.389
11	0.140	0.112	0.137	0.136	0.129	0.131	8.736
12	0.263	0.275	0.266	0.246	0.253	0.261	4.290
13	0.770	0.764	0.805	0.602	0.728	0.734	10.707
14	0.107	0.104	0.102	0.084	0.088	0.097	10.842
15	0.114	0.110	0.107	0.139	0.115	0.117	10.870

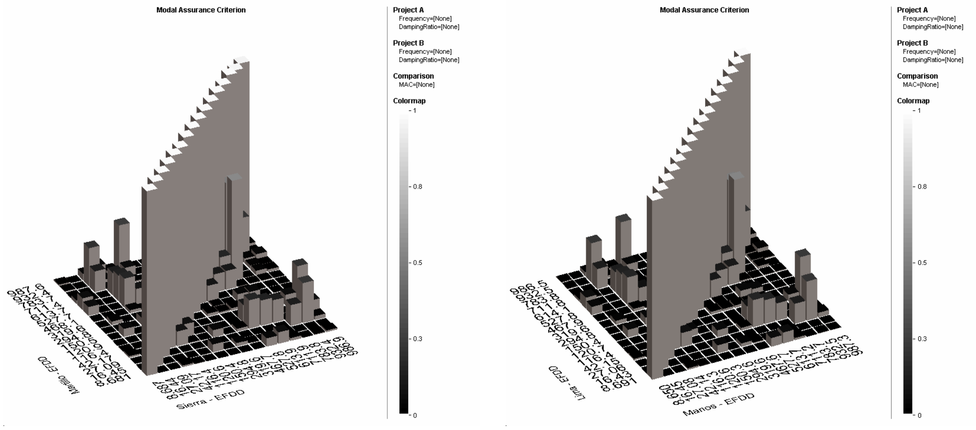

Figure 5. MAC: air streams-saw blade (left) and metal file-hands (right)

Table 3: SSI natural frequencies.

Mode	File	Air	Hands Hz (SSI)	Hammer	Saw	Mean Hz	CV %
1	8.61	8.602	8.597	8.607	8.614	8.61	0.079
2	16.43	16.43	16.42	16.44	16.43	16.43	0.043
3	27.04	27.09	27.07	27.07	27.07	27.07	0.066
4	42.15	42.16	42.15	42.16	42.16	42.16	0.013
5	116.30	116.4	116.3	116.6	116.4	116.40	0.105
6	120.60	120.9	120.8	120.9	120.8	120.80	0.101
7	150.40	150.4	150.2	150.4	150.4	150.36	0.059
8	294.80	294.90	294.00	294.70	294.80	294.64	0.124
9	319.70	319.70	319.50	319.70	319.80	319.68	0.034
10	467.50	468.50	467.00	468.10	467.30	467.68	0.13
11	527.90	527.90	526.70	527.70	528.10	527.66	0.105
12	614.20	613.90	613.20	613.60	614.20	613.82	0.07
13	711.90	712.60	709.40	711.30	710.00	711.04	0.186
14	783.90	783.70	782.90	783.70	784.40	783.72	0.069
15	932.00	931.90	932.60	932.50	930.60	931.92	0.086

Table 4: SSI damping ratios.

Mode	File	Air	Hands	Hammer	Saw	Mean	CV
			% (SSI)			%	%
1	1.034	1.023	1.249	0.9918	0.912	1.042	12.00
2	0.718	0.748	0.924	0.7505	0.827	0.793	10.48
3	0.251	0.260	0.332	0.2643	0.286	0.279	11.71
4	0.283	0.335	0.309	0.2778	0.326	0.306	8.292
5	0.341	0.226	0.275	1.438	0.292	0.514	100.6
6	1.798	1.295	1.483	1.578	1.476	1.526	12.01
7	0.502	0.388	0.427	0.3817	0.440	0.428	11.29
8	0.434	0.513	0.614	0.323	0.415	0.459	23.84
9	0.261	0.274	0.442	0.326	0.304	0.321	22.49
10	1.027	0.900	1.180	1.016	0.961	1.017	10.27
11	0.108	0.078	0.331	0.134	0.084	0.147	71.67
12	0.215	0.215	0.335	0.239	0.187	0.238	23.90
13	0.771	0.778	0.620	0.670	1.089	0.786	23.22
14	0.101	0.101	0.080	0.074	0.062	0.083	20.57
15	0.149	0.149	0.105	0.139	0.181	0.145	18.99

3.2 The Steel Plate

As in the previous case, the modal parameters were identified with EFDD and SSI. The first six modes where considered in the analysis. The normalized singular values and the stabilization SSI diagram corresponding to the test using the rubber ball are shown in Fig. 6. As it can be seen all the modes were excited in the frequency range of interest. The same can be concluded for the rest of the loading devices used to excite the plate. The operational modal analysis was always performed using 2048 points, with an overlap of 66.6% and 3 projection channels. The natural frequencies estimated with EFDD and SSI are presented in Tables 5 and 7, respectively, whereas the damping ratios are shown in Tables 6 and 8.

Tables 5,6,7 and 8 show that the dispersion of the estimates on natural frequencies for the various loading devices is very low (coefficient of variation (CV) less than 0.06 %), whereas the estimates of modal damping ratios present significantly high scatter (coefficient of variation (CV) less than 25 %). Compared with the steel beam structure, a better accuracy has been obtained in the natural frequencies of the plate and similar scatter in the damping ratios.

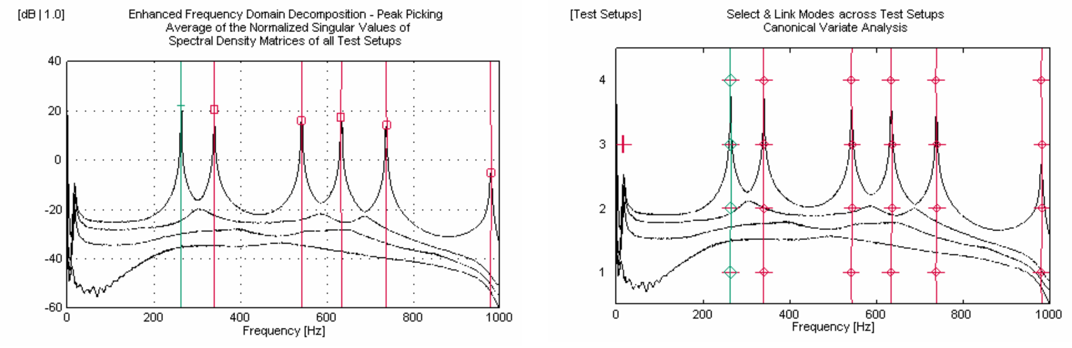

Figure 6. Rubber ball: Modes identified by EFDD (left) and SSI(right).

Table 5: EFDD natural frequencies.

	Air	R. Ball	Pencil	Hands	Hammer	Mean	CV
Mode			Hz			Hz	%
1	263.41	263.38	263.40	263.42	263.40	263.40	0.0056
2	340.13	340.10	340.11	340.21	340.14	340.14	0.0127
3	542.62	542.52	542.46	542.35	542.59	542.51	0.0199
4	634.14	634.06	634.06	634.10	634.09	634.09	0.0052
5	737.50	737.44	737.40	737.35	737.47	737.43	0.0080
6	979.09	978.94	978.90	978.84	979.02	978.96	0.0101

Table 6: EFDD damping ratios.

	Air	R. Ball	Pencil	Hands	Hammer	Mean	CV
Mode			%			Hz	%
1	0.1326	0.1337	0.1502	0.1377	0.1319	0.1372	5.54
2	0.1539	0.1483	0.1657	0.1573	0.1504	0.1551	4.41
3	0.1214	0.1184	0.1555	0.0833	0.1316	0.1220	21.40
4	0.0708	0.0825	0.0914	0.0725	0.0853	0.0805	10.83
5	0.0746	0.0735	0.0886	0.0686	0.0859	0.0782	10.99
6	0.1151	0.1044	0.1095	0.0829	0.1055	0.1035	11.82

Table 7: SSI natural frequencies.

	Air	R. Ball	Pencil	Hands	Hammer	Mean	CV
Mode			Hz			Hz	%
1	263.38	263.36	263.38	263.41	263.41	263.39	0.0082
2	340.18	340.08	340.09	340.18	340.15	340.14	0.0142
3	542.62	542.51	542.44	542.47	543.55	542.72	0.0866
4	634.09	634.06	634.00	634.11	634.45	634.14	0.0279
5	737.51	737.43	737.37	737.47	737.82	737.52	0.0238
6	979.54	979.97	978.65	978.74	979.62	979.30	0.0592

Table 8: SSI damping ratios.

	Air	R. Ball	Pencil	Hands	Hammer	Mean	CV
Mode			%			Hz	%
1	0.0765	0.0759	0.1058	0.0898	0.1063	0.0909	16.44
2	0.1269	0.1168	0.1516	0.1348	0.1461	0.1352	10.43
3	0.1140	0.1048	0.1448	0.1045	0.1288	0.1194	14.50
4	0.0599	0.0657	0.0841	0.0581	0.0907	0.0717	20.59
5	0.0675	0.0622	0.0842	0.0700	0.0820	0.0732	13.01
6	0.1494	0.1219	0.1947	0.1844	0.1319	0.1565	20.44

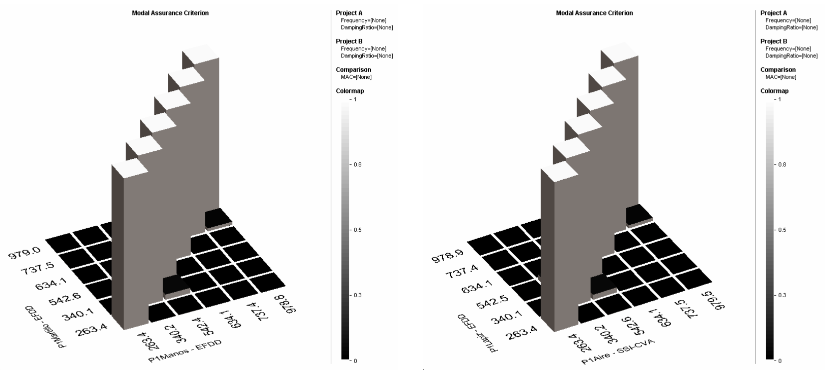

Figure 7. MAC: hands-impact hammer (left) and metal file-air stream (right)

The Modal Assurance Criterion (MAC) corresponding to hands- impact hammer and metal file –air stream excitations are presented in Fig. 7. A good correlation between the data corresponding to the different loading devices was obtained.

4 CONCLUSIONS

- Operational modal analysis has been applied successfully to small structures located in the lab, using different artificial devices as excitation. The loading devices have been tested on a steel beam structure and a steel plate.
- A good accuracy has been achieved in the natural frequencies and mode shapes. A relative high scatter has been obtained in the damping ratio estimates which, on the other hand, is common in operational modal analysis.

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