Modal Extraction on a Diesel Engine In Operation

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Abstract

In this paper an output only modal testing and identification of a diesel engine is presented. The only loading on the engine is the unknown loading from the engine itself. Two test cases were considered: engine run-up, and engine Run-Down. The response data were analysed using two different techniques: a non-parametric technique based on Frequency Domain Decomposition (FDD), and a parametric technique working on the raw data in time domain, a data driven Stochastic Subspace Identification (SSI) algorithm. Validating the results of the two techniques and the results of the two test cases against each other, 6 modes could be identified under 65 Hz. The first three modes were rigid body modes,, and the last two modes were clear bending modes. Frequency uncertainty was typically of the order of 0.5 Hz, damping ratios were uncertain, but relatively high (3-7 %) for the rigid body modes and around 2 % for the bending modes.

Nomenclature

Δt	sampling time step
y_t	response vector
f	natural frequency
ς	damping ratio
Φ, Ψ	mode shape matrices
$MAC(i, j)$	MAC matrix

Introduction

When modal properties are to be identified using output only techniques, the modal identification has to be carried out without using information about the input exciting the structure. In this particular case, the response of a diesel engine was measured for two cases: engine run-up and engine run-down. During engine run-up, the RPM of the engine was slowly increased to make the rotations of the engine and it's over harmonics perform a sweep over a suitable frequency range. During engine run-down the RPM was decreased similarly. In this case the loading is impossible to measure since it consists of a combination of engine explosions and inertial loading from rotational parts.

The response data were analysed using two different techniques: a non-parametric technique based on Frequency Domain Decomposition (FDD), and a parametric technique working on the raw data in time domain, a data driven Stochastic Subspace Identification (SSI) algorithm.

The results from the two test cases and the two techniques were compared and validated against each other.

Test conditions and data pre-processing

The measurements presented here were a part of a series of tests on the diesel engine. Measurements were taken during run-up, coast-down as well as stationary RPM with variable as well as invariable load.

The goal was to investigate vibration properties both against frequency using frequency analysis and order analysis as well as modal analysis by operational data.

The data taken with variable speed gives the possibility of being used for determination of modal data. As variable speed gives a load containing a broad frequency band for the excitation.

Accelerometers were mounted in ten points for detection of response in the horizontal direction perpendicular to the main shaft. Vertical accelerometers were also mounted in the points at the support of the engine. The engine was expected to be rather stiff in the vertical direction in the frequency range of interest, so vertical accelerometers were only mounted at the lower part of the engine.

At total ten points were measured in one or two directions. As Horizontal direction were not measured in the upper part the animation of these points were considered as slave for the lower points. At total 15 signal were acquired by a Brüel & Kjær PULSE™ multi-analysis system.

A 16th channel was used for RPM information. This were not used for determination of the modal parameters but for the order analysis taking the full advantage of the fact that the order-analysis could be done with data from the same measurement as the modal parameter estimation.

For both run-up and run-down test case the response data were sampled at a sampling rate of 2048 data points per second. For each time series 40960 data points were acquired corresponding to a total measurement time of 20 seconds.

The data were decimated by a factor 10 corresponding to a Nyquist frequency of 102.8 Hz and a effective number of points of 4096. The modal identification was carried out in the range 0-65 Hz.

A time-frequency plot obtained using a travelling window FFT analysis on the decimated data for the run-down is shown in figure 1. A large set of what is judged to be over harmonics of the engine RPM is clearly visible in the signal.

Because of the harmonics sweeping over the frequency range of interest, many harmonic peaks were present in all spectral density functions, and thus, also in the decomposed spectrum. This made it difficult to perform the identification using both the FDD. Also the SSI technique had problems because of these harmonic components. For the run-down test case the harmonics were removed by using a time weighting on the response signals. For the run-up test case, in the last part of the response signal, all harmonics were stable, this "no-sweep" part of the signal was omitted from the analyses. This reduced the effective length of the

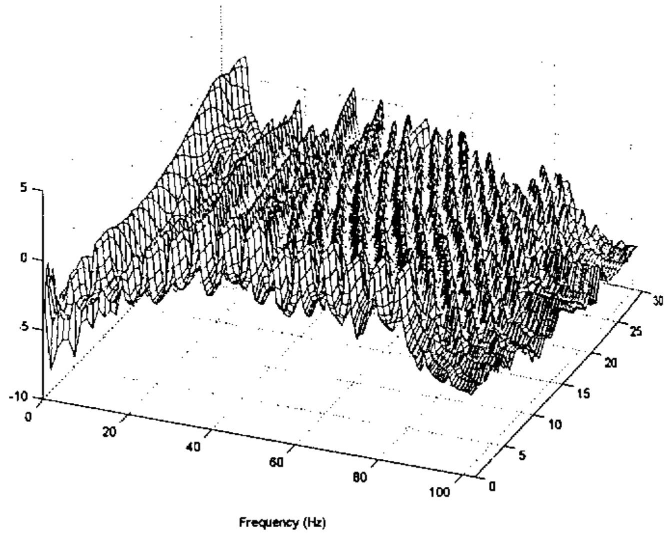


Figure 1. Time-frequency plot using a travelling window FFT analysis.

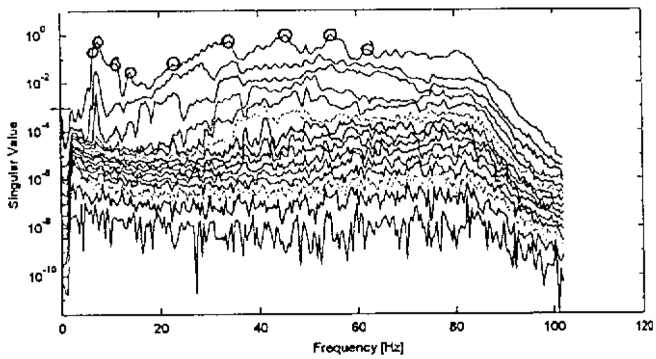
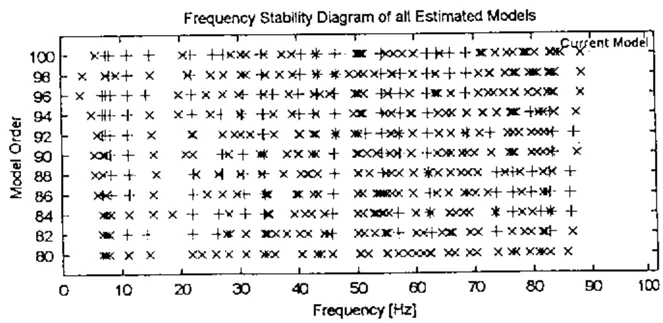


Figure 2. Singular value decomposition of the spectral density matrix of the engine run-down test case.



Figur 3. Stabilisation diagram based on data from the run-down test case.

response data for the run-up test case to 1640 data points per channel.

Principle of Frequency Domain Decomposition (FDD)

The Frequency Domain Decomposition (FDD) technique is an extension of the classical frequency domain approach often referred to as the Basic Frequency Domain (BFD) technique, or the peak picking technique. The classical approach is based on simple signal processing using the Discrete Fourier Transform, and is using the fact, that well separated modes can be estimated directly from the power spectral density matrix at the peak.

I the FDD technique first the spectral matrix is formed from the measured outputs using simple signal processing by discrete Fourier Transform (DFT). However, instead of using the spectral density matrix directly like in the classical approach, the spectral matrix is decomposed at every frequency line using Singular Value Decomposition (SVD) By doing so the spectral matrix is decomposed into a set of auto spectral density functions, each corresponding to a single degree of freedom (SDOF) system. This is exactly true in the case where the loading is white noise, the structure is lightly damped, and where the mode shapes of close modes are geometrically orthogonal. If these ssumptions are not satisfied, the decomposition into SDOF systems is an approximation, but still the results are significantly more accurate than the results of the classical approach.

The singular vectors in the SVD are used as estimates of the mode shape vectors, and the natural frequencies are estimated by taking each individual SDOF auto spectral density function back to time domain by inverse DFT. The frequency and the damping were simply estimated from the crossing times and the logarithmic decrement of the corresponding SDOF auto correlation function.

The theoretical background of the FDD technique is described in Brincker et al [1].

Results of Frequency Domain Decomposition (FDD)

Figure 1 shows the singular value decomposition of the spectral density matrix for the run-down test case.

Because of the limited number of data points and the relatively high damping for most of the modes, the main part of the modes are not easily recognised visually. However, using the FDD technique, the MAC criterion clearly identifies the modes indicated in Figure 1. As it appears, two close modes were identified around 7-8 Hz. The rest of the identified modes were reasonably well separated.

The run-up test case was analysed by the FDD technique in a similar way. The results are shown in tables 1 and 2.

As it appears from the results, nine modes were identified in both cases, however, not all the modes were repeated in the results for both test cases. In the run-up test case, a rigid body mode was detected around 3.4 Hz, this mode was also visible in the run-down case, but the mode was weakly excited and modal estimates were evaluated to be uncertain. Similarly other modes were detected in one test case, but could not be clearly detected in the other case. Only 6 modes were clearly detected in both cases. These six modes include the two close modes around 7-8 Hz.

Principle of Stochastic Subspace Identification (SSI)

Stochastic Subspace Identification (SSI) is a class of techniques that are all formulated and solved using state space formulations of the form

$$\begin{aligned} x_{t+1} &= Ax_t + w_t \\ y_t &= Cx_t + v_t \end{aligned}$$

where x_t is the Kalman sequences that in SSI is found by a so-called orthogonal projection technique, Overschee and De Moor [3]. Next step is to solve the regression problem for the matrices A and C, and for the residual sequences w_t and v_t. Finally, in order to complete a full covariance equivalent model in discrete time, the Kalman gain matrix K is estimated to yield

$$\begin{aligned} \hat{x}_{t+1} &= A\hat{x}_t + Ke_t \\ y_t &= C\hat{x}_t + e_t \end{aligned}$$

It can be shown, Brincker and Andersen [2], that by performing a modal decomposition of the A matrix as $A = V[\mu_i]V^{-1}$ and introducing a new state vector $z_t = V^{-1}\hat{x}_t$ the equation can also be written as

$$\begin{aligned} z_{t+1} &= [\mu_i]z_t + \Psi e_t \\ y_t &= \Phi z_t + e_t \end{aligned}$$

where $[\mu_i]$ is a diagonal matrix holding the discrete poles related to the continuos time poles λ_i by $\mu_i = \exp(\lambda_i \Delta t)$, and where the matrix Φ is holding the left hand mode shapes (physical, scaled mode shapes) and the matrix Ψ is holding the right hand mode shapes (non-physical mode shapes). The right hand mode shapes are also referred to as the initial modal amplitudes, Juang [4].

The specific technique used in this investigation is the Principal Component algorithm, see Overschee and De Moor [3].

Results of Stochastic Subspace Identification (SSI)

For each of the two test cases a set of models with different model orders were identified and the stabilisation diagram was established. Figure 3 shows the stabilisation diagram for the run-down test case.

As it appears, at least nine modes were stabilised in this frequency band from 0-65 Hz. Only the nine modes that was belived to correspond to the modes identified by the FDD tehcnique was used in the further analyses. All modes detected by the FDD technique was also detected by the SSI algorithm. In this case a model order of 100 was needed to stabilise all modes of interest. This means that a relatively large number of noise modes (41 niose modes) were present in addition to the nine modes identified as physical.

A similar analyses was carried out for the run-up test case. Results are given in tables 1 and 2.

Validation of results

First, it was evaluated if the identified modes represent something physical. In this case, since the excitation is non-stationary, and since the sweep of the harmonics created by the engine might stop and start inside the frequency range, computational modes might arise that could look physical, but is in fact just representing the non-stationary loading. Thus, only the modes that were clearly present in the identification results of both test cases were chosen be representing modal properties.

The validity of the modal identification was also evaluated by calculating the Modal Assurance Criterion matrix MAC, where $MAC(i,j)$ denotes the MAC value between mode i in FDD estimation with mode j in SSI estimation. The diagonal elements of the MAC matrix are given in tables 1 and 2. A measure of modal significance was also calculated as

$$MSC = MAC(i,i)^2 / (MAC_{i\max}^{FDD} MAC_{i\max}^{SSI})$$

where $MAC_{i\max}^{FDD}$ is the maximum off-diagonal element in row no i and $MAC_{i\max}^{SSI}$ is the maximum off-diagonal element in column no i. Thus this measure compares the mode shapes in one model with the other mode shapes (other modes) in the other model. If the measure is larger than one, it means that the mode shape compares better with the corresponding mode in the other model than with any other mode shape in the other model.

As it appears from the results, except one of the close modes around 7-8 Hz, all other modes that are repeadedly identified in both test cases show a reasobly MAC and significance value.

Since 4 estimates were available for all the 6 modes so identified (2 ID techniques, 2 test cases), a modal pairing was performed to chose the best identification pair. The modal pairing was based on the MAC values. The pairing is shown in table 3, and the final validated results are shown in table 4. Mode shapes for the validated modes are shown in Figures 4 and 5.

Conclusions

Using less than half a minute of response data from a diesel engine in operation 6 modes have been identified in the frequency range 0-65 Hz using two different output only identification techniques, the FDD technique and the SSI technique. All modes were identified by both techniques, and in both test cases. Thus, it is believed that all 6 modes represent physical modes of the engine and engine support. The first 3 modes so estimated were clear rigid body modes of the engine, whereas the last two modes were clear bending modes.

Using different techniques to validate the mode shape estimates for the two techniques it has been indicated that the modal estimates of the close modes are somewhat uncertain. However, most of the modes were indicated to be reliable modes and the corresponding modal parameters is to be considered as reliable modal parameters.

Frequency deviations between natural frequencies of the estimates obtained by the two techniques were typically of the order of 0.5 Hz or 0.5 % of the Nyquist frequency.

The damping estimates were in the most cases somewhat uncertain. However it can be concluded, that the damping for the rigid body modes were relatively high, in the range from 3-7 %, and that the damping of the bending modes were about 2 %.

References

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Table 1. Modal results – Engine run-up.

Mode No.	Frequency Domain Decomposition (FDD)		Stochastic Subspace Identification (SSI)		Modal Assurance (MAC)	Modal Signifi-Cance
	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)		
1	3.38	16.12	3.56	6.42	0.999	7.27
2	6.99	7.93	6.68	1.02	0.948	1.18
3	6.99	7.89	7.38	1.37	0.806	0.79
4	13.35	6.26	13.29	5.74	0.999	15.76
5	47.52	6.03	47.40	4.62	0.987	1.75
6	50.07	0.67	50.05	0.06	0.997	2.52
7	54.90	1.98	54.89	2.55	0.931	1.42
8	63.15	1.99	62.45	2.20	0.956	1.35
9	83.60	1.14	83.95	1.35	0.829	1.14

Table 2. Modal results – Engine run-down.

Mode No.	Frequency Domain Decomposition (FDD)		Stochastic Subspace Identification (SSI)		Modal Assurance (MAC)	Modal Signifi-Cance
	Frequency (Hz)	Damping (%)	Frequency (Hz)	Damping (%)		
1	7.31	4.24	7.42	1.91	0.939	1.55
2	7.53	6.92	8.04	4.48	0.768	0.92
3	10.88	5.30	11.08	4.65	0.983	2.94
4	14.33	5.48	14.92	5.65	0.908	1.12
5	22.99	5.40	21.86	4.52	0.867	1.34
6	32.55	7.01	34.08	3.37	0.985	1.26
7	44.87	5.73	46.65	3.52	0.985	1.87
8	55.58	1.81	54.39	1.74	0.881	1.13
9	62.68	1.55	62.16	1.28	0.824	0.99

Table 3. MAC values for modes identified on both Run-Up and Run-Down.

Mode No Run-Up	Mode No. Run-Down	UP: FDD DN: FDD	UP: FDD DN: SSI	UP: SSI DN: SSI	UP: SSI DN: FDD
2	1	0.487	0.445	0.373	0.415
2	2	0.525	0.738	0.644	0.491
3	1	0.491	0.449	0.743	0.713
3	2	***0.529***	0.741	***0.932***	0.748
4	4	0.945	0.887	0.876	***0.948***
5	7	0.967	***0.978***	0.964	0.946
7	8	0.757	***0.841***	0.773	0.676
8	9	0.923	0.919	***0.971***	0.863

Table 4. Validated modes.

Approach Up/Down	Frequency (Hz)	Damping Ratio (%)	MAC
FDD/FDD	7.25	7.4	0.53
SSI/SSI	7.71	2.9	0.93
SSI/FDD	13.81	5.6	0.95
FDD/SSI	47.09	4.8	0.98
FDD/SSI	54.65	1.9	0.84
SSI/SSI	62.56	2.0	0.97

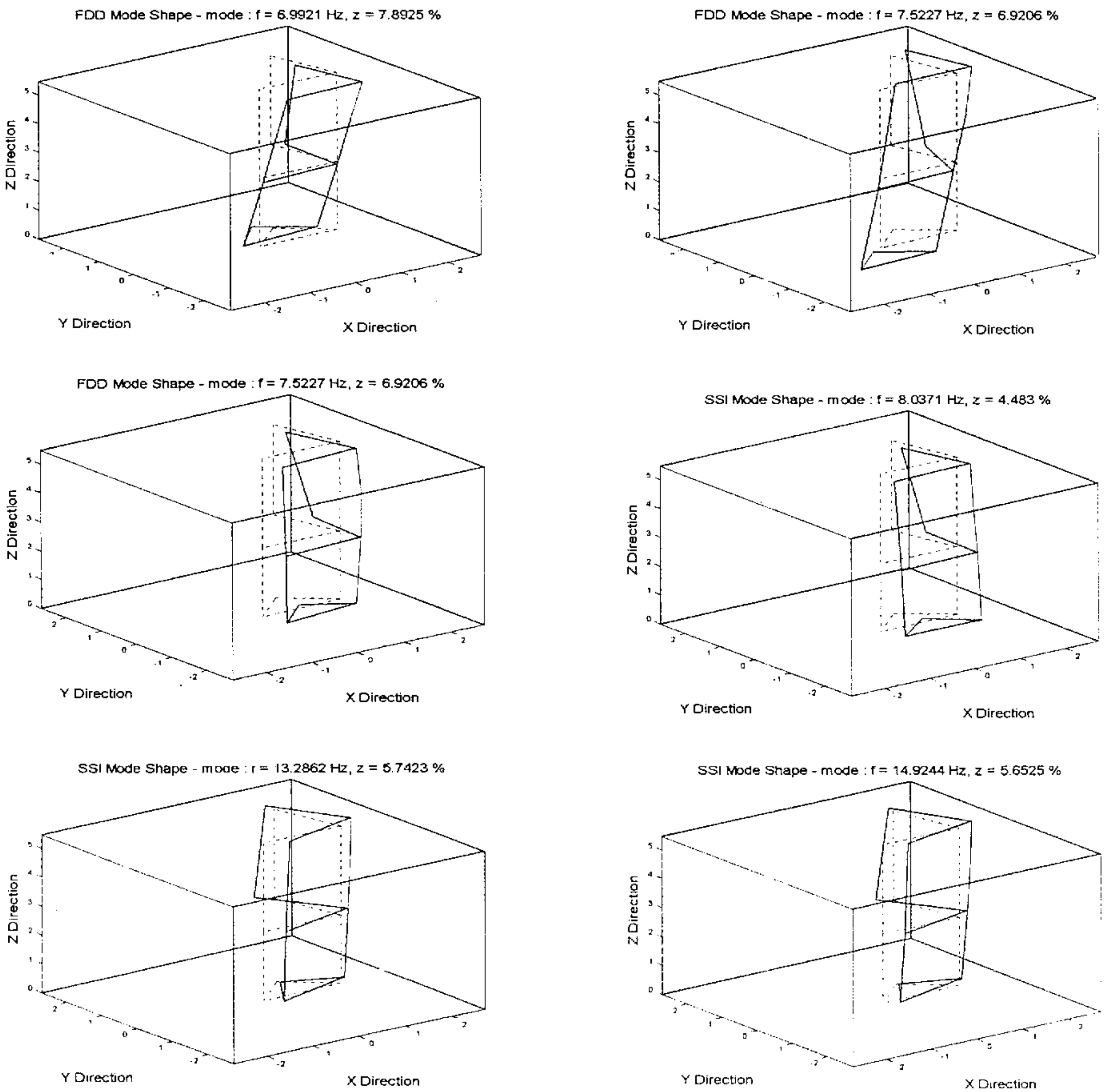

Figure 4. Mode 1-3, left: Engine run-up, right: Engine run-down.

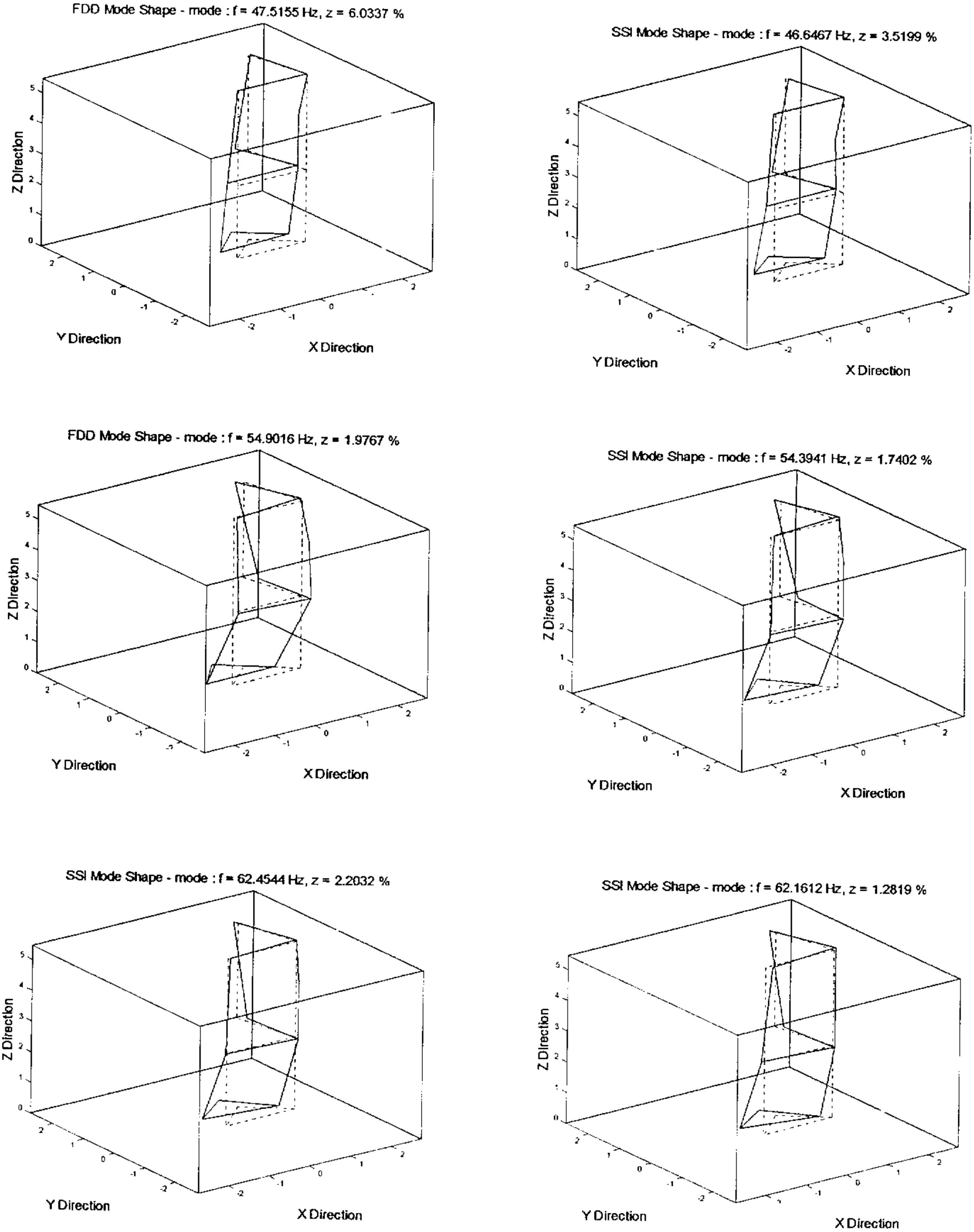

Figure 5. Mode 4-6. left: Engine run-up, right: Engine run-down.