EXPERIMENTAL MODAL ANALYSIS TECHNIQUE, SYSTEM AND APPLICATION FOR MINIATURE STRUCTURES

Q. H. Zeng[1] and D. B. Bogy

Computer Mechanics Laboratory

Department of Mechanical Engineering

University of California

Berkeley, CA 94720

ABSTRACT

This report presents an experimental modal analysis system and some practical techniques that are very suitable to the modal experiments of miniature structures, such as suspensions, and other components of hard disk drives. Two excitation methods, vibration of the base structure and electromagnetic (point) excitation of the measured component, are included in the system. A multiple frequency band measurement is implemented, and a calibration procedure for the base excitation is proposed. The system is first verified by experiments on a small cantilever beam. The results from the two excitation methods are compared with each other, and the measured modal frequencies are found to be very close to the results from a finite element model. The system and techniques are applied to obtain modal parameters of a suspension known as "Type 1650". The lower fourteen modes of the suspension are obtained. The modal frequencies and damping ratios obtained from the two excitation methods are compared. The results show that the system and techniques are robust for the experiments of such miniature structures.

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[1] Visiting scholar, associate professor of vibration engineering, Nanjing University of Aeronuatics & Astronautics, Nanjing, China.

1. INTRODUCTION

Magnetic hard disk drives are made smaller in size, higher in capacity and lower in cost, and are now the most important means of information storage. To further increase the storage density, access speed and reliability, the proper design and simulation of mechanical components in the drives, such as the actuator arm, suspension and flexure, are extremely important. Mathematical models of the components are essential to the design and simulation. Finite element (FE) models can be conveniently created and applied, but the accuracy and reliability of the FE models are strongly related with the complication of the components and the experience of the researcher. More accurate and reliable models can be built by employing modal experiments. The models ares used not only to validate the FE models, but also can be directly applied in the design and analysis of the components in some situations. Therefore, modal experiments play an important role in the research and development of the new generation of drives.

Since the components in the current and future drives are very small, it is difficult to apply conventional experimental modal analysis techniques to obtain the dynamics parameters of the components. Some special excitation and response measurement techniques are required. In the response measurement, it is critical to use non-contact methods. Thus, Laser Doppler Vibrometers (LDV) are widely used to measure the response, as for example in Wilson and Bogy (1994).

There are about five excitation methods used previously in the modal experiments of components. These methods are the conventional mechanical shaker (Radwan and Chokshi, 1985), pulsed air (Mui et al., 1990), servo actuator (Castigna, 1988), base excitation (Beliveanu et al., 1986; Chen et al., 1995), and electromagnetic exciter (Patton and Trethewey, 1987; Wilson and Bogy, 1994). The first three methods are not often used because of their accuracy limitation. A parameter identification method based on a physical parameters model of the structure was developed for the base excitation method about twenty years ago (Link and Vollan, 1978). Before a modal parameter based model was developed (Beliveanu et al., 1986), the use of base excitation to estimate modal parameters had become a practical technique. A more complete derivation of the method was presented by Chen et al., (1995). Base excitation has been successfully applied in modal experiments of the drive's components. However, the base excitation method does not apply a point force directly to the components being investigated, even if the components are coupled with the excitation system, which would result in difficulty in the parameter estimation. Additionally, modal mass cannot be estimated from the base excitation test if the base reaction forces are not measured. The electromagnetic excitation method was first used in modal experiments of engine blades in the aeronautical industry in the early 1980s. Later works were presented in several papers (Patton and Trethewey, 1987; Ebersbach and Irretier, 1988; Wilson and Bogy, 1996). Wilson and Bogy (1996) refined the design of the electromagnetic exciter, improved the techniques, and successfully applied it in modal experiments of the components and system of the drives. However, the effects of the mass of the ferromagnetic target on the test structure could be a potential problem.

Small structures yield a wider frequency response band. In the dynamics analysis of the components, the frequency band of interest is often from 0 to 20 kHz. Modal damping ratios of the components, such as the suspensions, are very small, usually less than 0.5%. To ensure accuracy of the measured results, a high frequency resolution is required. Current data acquisition and analysis equipment have a fixed length, 1024 or 2048 words, of a data block. Therefore, the desired frequency band should be divided into several small frequency bands in the experiment to meet the requirements of a wide frequency band and high resolution. That is equivalent to several modal experiments when an ordinary modal experimental system is used. The experiment becomes very time consuming and tedious work, because it is difficult to locate the measurement points for the miniature structures.

It is clear from the previous discussion that it is necessary to develop an experimental modal analysis system that is suitable for the drives and their components. This report presents an experimental modal analysis system and some practical techniques that are very suitable for the modal experiments of the components and system of the hard disk drives. Two excitation methods, base excitation and electromagnetic excitation, are included in the system. A multiple frequency band measurement is implemented, and a calibration procedure is proposed for the base excitation. The system is verified using a small cantilever plate. The measured modal frequencies are very close to the results from a finite element model. The system and techniques are applied to obtain modal parameters of a Type 1650 suspension. Fourteen modes of the suspension are obtained. The modal frequencies and damping ratios obtained from the two excitation methods are compared. The results show the effectiveness and feasibility of the system and techniques.

2. EXPERIMENTAL MODAL ANALYSIS SYSTEM

Figure 1 shows a schematic representation of the experimental modal analysis system that was developed. Not counting the specimen to be tested, the system includes five principle units: the shaker excitation unit, the electromagnetic excitation unit, the measurement unit, the data acquisition and analysis unit, and a modal analysis software installed in a PC computer.

2.1 Shaker Excitation Unit

A model 203 Ling Dynamics Systems Vibrator and model TPO 25 Power Amplifier are used to generate the base motion. To attach test specimen to the vibrator, a fixture was designed and fabricated as shown in Figure 2. The maximum thrust of the unit is 17.8 N below 3.5 kHz and 16.0 N above 3.5 kHz. Its maximum table acceleration is 90 g (pk). Although its operation frequency range is from 0 to 13 kHz, for miniature and light damping structures, such as suspensions, the upper frequency band would be higher.

2.2 Electromagnetic Excitation Unit

The electromagnetic exciter (Type B) and power amplification circuit (Wilson and Bogy, 1996) are used to generate a non-contact excitation force. A ferromagnetic target is required. Detailed design and fabrication procedures are described in the paper (Wilson and Bogy, 1996). In practical situations, the target should be as small as possible, but its dimension is also relative to the test specimen. For larger specimen, a larger target is preferred for easily positioning the exciter and achieving a high excitation level. The target is attached to the specimen by use of adhesive. The level of the excitation force highly depends on the working gap between the exciter and the target. In the experiment, it is practical to adjust the gap through trial and error, based on the measured excitation and response signals.

2.3 Measurement Unit

In the modal experiments, we should measure the responses (displacement, velocity or acceleration) of the test specimen at each measurement point, and excitation forces or base motions. For the response measurement, non-contact, highly sensitive transducers must be used that have a wide frequency range. For the wide frequency range, velocity responses are preferred. Therefore, a Polytec Laser Doppler Vibrometer is used to measure the absolute velocity responses. Additionally, because the laser spot focused on the surface of the specimen is very small, it is easy to accurately locate the measurement points in the gimbal region of the suspension assembly. An XY stage and microscope are very helpful for locating the laser spot. It would be better to use the tracking and low pass filter in the LDV to improve the signal/noise (S/N) ratio.

The base motion must be measured to determine the base excitation. In the system, a miniature accelerometer (PCB 309A) that is fixed to the base by the fixture is used (Figure 2). A force sensor (PCB model 209A) is used to measure the excitation force in the system. The sensor's resolution is 20 mN, and its sensitivity is 0.5 V/N. Sensors with higher sensitivity are strongly recommended.

2.4 Data Acquisition and Analysis Unit

A HP 3562A Dynamic Signal Analyzer is adopted for data acquisition and analysis to obtain the frequency response functions (FRFs). The analyzer also generates a source signal for the excitation unit. Two types of the source signal are used in the system. The first is a periodic chirp signal, which is used for simple test specimens, such as the suspensions and HGA. When using the chirp signal to excite the specimens, the S/N ratio of excitation and response signals are relatively high, and less averaging is required. However, the chirp signal is not suitable for specimens that exhibit non-linear properties. The second source signal is a burst random signal, which is used in the experiments of the suspensions loaded on stationary and rotating disks when they exhibit small non-linear properties for the friction in the interface between sliders and disks. To reduce the leakage and increase S/N ratio, the percentage of the burst signal should be carefully adjusted. A number of 10 to 50 measurements is required for averaging, depending on the coherence function. The value of the function in the interested frequency region should be larger than 0.8. For these two types of source signals, no window is applied in the data analysis.

The miniature structures have a wide frequency response band. In order to increase the reliability and accuracy of the measured modal parameters, the desired frequency range should be divided into several sub-regions. In the analyzer, there are ten state buffers to store the state of the analyzer. In the experiments we first set up the state for each sub-region, and store the state in the different buffers. Then, a special software is used to automatically control the analyzer, recalling the states to finish the data acquisition and analysis for all sub-regions at each measurement point in the later measurement. As a result, the amount of work is dramatically reduced. The consistency of the results should also be increased.

2.5 Modal Analysis Software

The modal analysis software NAI-MODAL (Zeng and Zhang, 1989) is applied in the system. The function of the software is very similar to other commercial modal analysis software, such as STAR. The function mainly includes geometric modeling of the specimen; controlling the analyzer; reading FRFs from the analyzer and storing the FRFs and measurement parameters in a database; curve fitting; mode sorting; and mode shape animation. In order to meet the requirements of the miniature structures, the software in the system is further enhanced. The following functions have been implemented.

2.5.1 Multiple frequency band measurement. The software can control the analyzer, recalling different states for each frequency band, reading the FRFs from the analyzer when the averaging is finished, and storing the FRFs and measurement parameters in the database. The procedure is as follows.

① Input the number of bands, and set up the analyzer;

② Move the laser spot of the LDV to a measurement point on the test specimen;

③ Enter the measurement point number N_p into the measurement label, press the start key (F3), $N_f=1$;

④ The software controls the analyzer, recalling one state for the corresponding frequency band, and starting a measurement;

⑤ The software automatically reads the FRFs in the PC when the analyzer is finished averaging;

⑥ Save the FRFs and the measurement label into the database by measurement number $100\ N_f + N_p$;

⑦ $N_f = N_f + 1$, repeat step ④ ~ ⑥ until all frequency bands are measured;

⑧ Repeat step ② ~ ⑦ until all measurement points are measured.

As many as 99 measurement points and 10 frequency bands (or 199 points and 5 bands) are allowable in the current system. With this function, the amount of time consuming work can be dramatically reduced.

2.5.2 Calibration for the Base Excitation. In the experiments we observed additional peaks in the measured FRFs using the base excitation. The upper curve in Figure 3 is the measured FRF of the Type 1650 suspension. The first peak represents the first bending mode of the suspension. The second is an additional peak, which could come from the interaction between the suspension and shaker, the fixture, and/or measurement channels. To eliminate the additional peaks, we proposed a calibration procedure. Before and after measurement of the specimen, the FRFs on several selected points on the base are measured. The average number should be large enough such that the curves of the FRFs are smooth. By averaging the FRFs, a measured base FRF H_b is obtained. The FRF H_b is the ratio of the Fourier transformation of the velocity to the Fourier transformation of the acceleration on the base. After calibration, the FRF H_b should be equal to $1/i\omega$. Therefore, we have a calibration FRF H_c

$$H_c = \frac{1}{i\omega H_b} \tag{1}$$

where $i = \sqrt{-1}$, and ω is the radial frequency. The measured FRF H is calibrated by the following equation

$$\tilde{H} = H_c H \tag{2}$$

or

$$\tilde{H} = \frac{H}{i\omega H_b} \tag{3}$$

where $\tilde{H}$ is the calibrated measured FRF. The lower curve in Figure 3 is the calibrated FRF corresponding to the upper curve. It can be seen that the second peak is eliminated. Additionally, an anti-resonance valley appears in the curve, and the calibrated FRF is smoother.

2.5.3 Modifying FRFs for the Base Excitation. When the excitation is base excitation, the measured FRFs are not the originally defined FRFs, and they cannot be directly used to estimate the modal parameters. They should be modified using the following equation

$$\bar{H} = \frac{1}{i\omega}\tilde{H} + \frac{1}{\omega^2} \tag{4}$$

where $\bar{H}$ is the modified FRF. One has

$$\bar{H} = U_c H_o \tag{5}$$

where H_o is an originally defined FRF (ratio of the displacement to the force), U_c is an unknown constant. Therefore, using the modified FRFs, we can estimate the modal frequencies, damping ratios and mode shapes, but we cannot estimate the modal masses.

3. VERIFICATION AND COMPARISON

A small stainless steel cantilever plate was used to verify the system. The plate has a dimension of $25.6 \times 5.22 \times 0.203$ mm, a Young's modulus of 193 GPa, a mass density of 8030 kg/m^3 , and a Poisson's ratio of 0.29. An FE model was created by the FE code ABAQUS. The plate was modeled using 2000 four node shell elements. The cantilevered support was modeled as nodal restraints with zero displacement at one end of the plate. Using the subspace iteration method, the first eight modes in the Z-direction in the frequency band of 0~15 kHz were found. The modal frequencies are shown in Table 1.

In the modal experiments, two excitation methods were used. A periodic chirp signal was inputted into the vibrator and magnetic exciter to excite the plate. One hundred and five measurement points were included in the geometric model which is shown in Figure 4. Points 106-110 are located at the fixture. Points 1, 22, 43, 64 and 85 are located on the cantilevered end of the plate. We made a preliminary measurement in the frequency band of 0~15 kHz. Approximate values of the first eight modal frequencies were obtained. Based on these values, the frequency range of 0~15 kHz was divided into five frequency bands. The first band was from 100 Hz to 260 Hz in which the first mode is included. The second was from 1460 Hz to 2460 Hz in which the second and third modes are included. The third was from 4000 Hz to 4800 Hz in which the fourth mode is included. The fourth band was from 7200 Hz to 8800 Hz in which the fifth and sixth modes are included. The fifth band was from 12500 Hz to 14500 Hz in which the seventh and eighth modes are included. Five averages were used for the first three bands. For the last two bands, twenty averages were used because of the relatively lower S/N ratio in the higher frequency band.

First, the plate was attached on the fixture. Using the shaker to excite the plate, one hundred and five FRFs in each frequency band were measured in the Z-direction. Figure 5 shows the measured FRF and the coherence function of the plate at point 74 in the frequency band of 1460-2460 Hz. The first eight modal parameters were estimated by NAI-MODAL. The modal frequencies and damping ratios are shown in Tables 1 and 2. The mode shapes are shown in Figure 6. Second, a ferromagnetic target, which has a dimension of 0.9×0.7×0.1 mm, and is made of 1010 steel, was attached to the plate at the region of point 68. Also using the base excitation, the modal frequencies and damping ratios were obtained and are shown in Tables 1 and 2. The mode shapes are almost identical with ones shown in Figure 6. Finally, using the electromagnetic exciter to excite the plate with the target attached, the modal frequencies and damping ratios were obtained and are also shown in Tables 1 and 2. The mode shapes are very similar to the ones shown in Figure 6.

The modal frequency differences between experiments and FE analysis are shown in Table 1. The calculated results are always larger than the experimental results. These biased differences could come from errors of the dimensions or material properties of the plate used in the FE analysis. Fortunately, the maximum difference is only -1.64%, and the average difference is about -0.88%. If the biased differences are removed, the maximum difference will be less than 0.8%. The modal frequencies obtained from the three experiments are very close. Table 2 shows that the modal damping ratios from the electromagnetic excitation are slightly larger than the those from the base excitation, except for the first mode.

4. MODAL EXPERIMENTS OF THE TYPE 1650 SUSPENSION

A type 1650 suspension was measured in its unloaded state, that is, it was not deflected as if installed in a drive. A slider was not attached to the suspension. This configuration was chosen in order to detail and accurately model the suspension, especially to model the region of the gimbal of the suspension. If we obtain a good model of the suspension, it will be easy to create the model of the suspension with the slider attached and in the loaded state. The created model will then be more reliable.

The suspension was fixed on its base plate, and excited by the shaker and magnetic exciter. The chirp signal was used in the excitations. The geometric model of the suspension shown in Figure 7 includes 83 points in which there are 69 independent points and 14 dependent points. The responses in the Z-direction were measured. At several selected points, preliminary measurements were made. In the frequency band of 0~20 kHz, fourteen modes were found. The frequency band of 0~20 kHz was divided into five frequency bands: 140-390 Hz, 1500-2750 Hz, 4500-6500 Hz, 8300-10800 Hz, 12000-18250 Hz. The base excitation was used. Sixty nine FRFs in each frequency band were measured. The measured FRF and coherence function at point 31 in the band of 8300-10800 Hz are shown in Figure 8. The coherence function is larger than 0.8 in the frequency region of interest, which indicates random error of the measured FRF is very small. Then, the ferromagnetic target used in the experiment of the plate was glued to the suspension at the region nearby point 65. The shaker and electromagnetic exciter were used to excite the suspension with the target attached. The modal parameters were estimated by NAI-MODAL. The global rational fraction orthogonal polynomial method in the software was chosen to perform the curve fitting. The modal frequencies and damping ratios obtained from the three experiments (base excitation without the target, base excitation with the target, and electromagnetic excitation with the target) are shown in Tables 3 and 4. Measured mode shapes from the three experiments are very similar, and are shown in Figure 9.

From Table 3, we see that the effects of the target on the frequencies are different for each mode. Comparing with the mode shapes shown in Figure 9, the effects can be easily explained. For example, mode 10 has a larger motion at point 65 at which the target is attached. The effects of the mass of the target result in a decrease in the frequency of mode 10. Table 3 shows that the frequencies from the electromagnetic excitation are always slightly smaller than the ones from the base excitation without the target. The maximum difference is about 1.94%, and the average difference is about 0.6%. Table 4 shows that the damping ratios of most modes from the electromagnetic excitation are larger than the ones from the base excitation.

5. SUMMARY

1) A modal experimental system with two excitation methods is established and verified. The experimental results of the small cantilever plate and a Type 1650 suspension show that the system is robust for the miniature structures.
2) Very similar results were obtained from the two excitation methods. In future experiments, we will choose one of the two methods based on which method is more convenient for a specific experiment, or use both methods at the same time to validate the experimental results.
3) A calibration procedure for the base excitation is proposed. The results show it very effectively eliminates the additional peaks of the FRFs under the base excitation.
4) The multiple frequency band measurement is implemented in the system. In the experiments, the use of this function dramatically reduces the time consuming work.
5) Through the modal experiments, fourteen modes of the type 1650 suspension are found. The results will be beneficial for modeling the suspension, especially its gimbal region.

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	Mode Shape	Without the Target			With the Ferromagnetic Target			
		FE Analysis	Base Excitat.	Diff. (%)	Base Excitat.	Diff. (%)	Magnetic Excitat.	Diff. (%)
1	1st Bending	**248.33**	244.26	-1.64	244.57	-1.51	244.97	-1.35
2	2nd Bending	**1554.5**	1539.5	-0.96	1539.2	-0.98	1530.9	-1.52
3	1st Torsion	**2444.3**	2413.3	-1.27	2419.1	-1.03	2412.6	-1.30
4	3rd Bending	**4361.8**	4334.8	-0.62	4333.0	-0.66	4321.8	-0.92
5	2nd Torsion	**7480.3**	7421.0	-0.79	7412.5	-0.91	7390.2	-1.20
6	4th Bending	**8576.1**	8534.9	-0.48	8538.1	-0.44	8509.5	-0.78
7	3rd Torsion	**12942.**	12828.	-0.88	12831.	-0.86	12791.	-1.17
8	5th Bending	**14223.**	14165.	-0.41	14170.	-0.37	14115	-0.76

Table 1 Modal frequencies of the cantilever plate obtained by the FE analysis and experiments using the base and electromagnetic excitations. Modal frequencies obtained by the experiments are compared with those calculated by the FE analysis

	Mode Shape	Without the target	With the ferromagnetic target	
		Base Excitation	Base Excitation	Magnetic Excitation
1	1st Bending	.31	.42	.20
2	2nd Bending	.27	.26	.44
3	1st Torsion	.067	.075	.10
4	3rd Bending	.17	.16	.21
5	2nd Torsion	.054	.058	.082
6	4th Bending	.20	.18	.18
7	3rd Torsion	.065	.063	.081
8	5th Bending	.15	.15	.18

Table 2 Modal damping ratios of the cantilever plate obtained by the experiments using the base and electromagnetic excitations.

	Without Target	With the Ferromagnetic Target			
	Base Excitation	Base Excitation		Magnetic Excitation	
	Frequency (Hz)	Frequency (Hz)	Difference (%)	Frequency (Hz)	Difference (%)
1	**210.85**	209.91	-0.44	209.34	-0.72
2	**1655.9**	1636.8	-1.16	1639.7	-0.98
3	**2406.0**	2403.7	-0.09	2401.2	-0.20
4	**4932.1**	4937.9	0.12	4926.4	-0.12
5	**5266.6**	5267.1	0.01	5251.7	-0.28
6	**5838.1**	5837.7	-0.01	5822.7	-0.26
7	**8359.7**	8326.1	-0.40	8279.7	-0.96
8	**8723.4**	8739.1	0.18	8704.2	-0.22
9	**8966.2**	8980.6	0.16	8944.3	-0.24
10	**10344.**	10210.	-1.30	10144.	-1.94
11	**12501.**	12500.	-0.01	12411.	-0.72
12	**14215.**	14155.	-0.42	14095	-0.84
13	**14460.**	14502.	0.29	14401.	-0.41
14	**17420.**	17409.	-0.07	17303.	-0.67

Table 3 Modal frequencies of the type 1650 suspension obtained by experiments using the base and electromagnetic excitations. Modal frequencies of the suspension with the target attached are compared with those having no target attached.

	Without Target	With the Ferromagnetic Target			
	Base Excitation	Base Excitation		Magnetic Excitation	
	Damping (%)	Damping (%)	Difference (%)	Damping (%)	Difference (%)
1	**0.20**	0.20	0.20	0.16	-18.1
2	**.096**	0.12	20.8	0.11	8.7
3	**0.14**	0.15	7.5	0.20	42.1
4	**0.11**	0.11	5.6	0.11	-3.0
5	**0.064**	0.066	3.1	0.083	30.2
6	**0.049**	0.050	1.1	0.073	47.4
7	**0.18**	0.18	4.6	0.15	-17.3
8	**0.076**	0.088	15.2	0.11	40.4
9	**0.082**	0.095	16.2	0.11	31.6
10	**0.044**	0.049	11.7	0.073	66.2
11	**0.029**	0.030	0.50	0.057	93.5
12	**0.035**	0.051	45.7	0.065	85.7
13	**0.061**	0.076	23.8	0.088	45.0
14	**0.050**	0.052	3.1	0.079	57.8

Table 4 Modal damping ratios of the type 1650 suspension obtained by experiments using the base and electromagnetic excitations. Modal damping ratios of the suspension with the target attached are compared with those having no target attached.

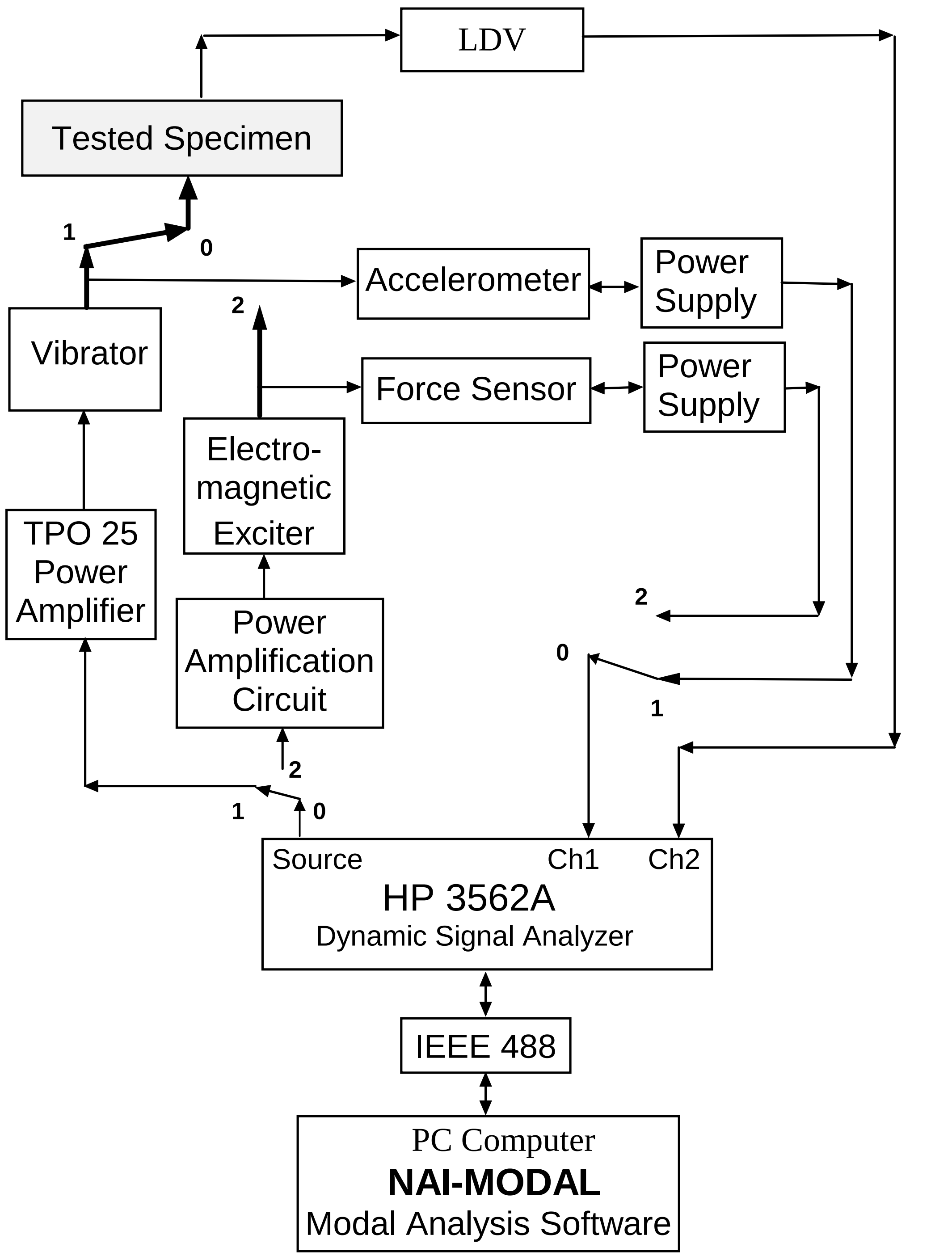

Figure 1 Schematic of the experimental modal analysis system
Base excitation (port 0 connected with port 1)
Electromagnetic excitation (port 0 connected with port 2)

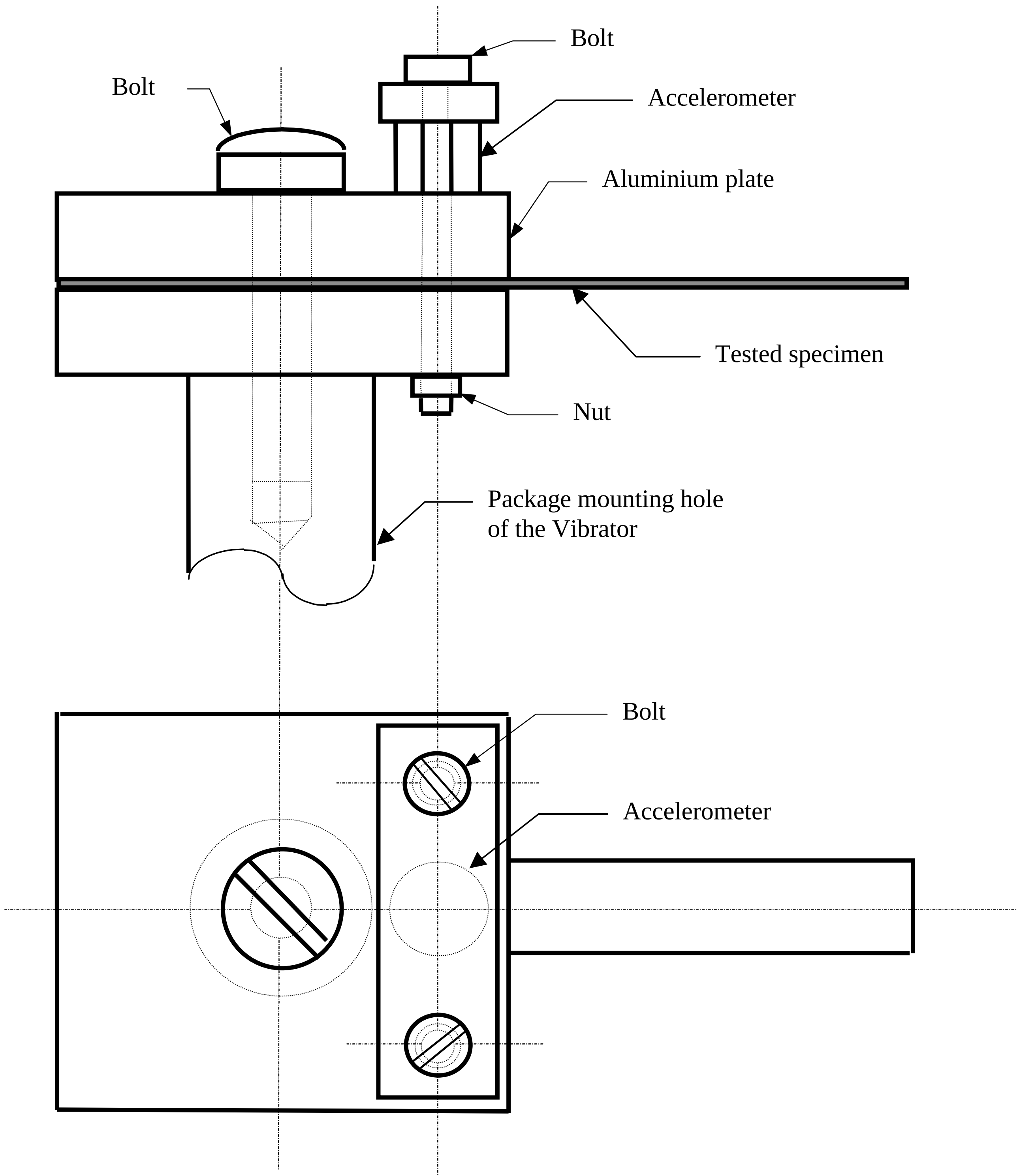

Figure 2 A fixture for the base excitation

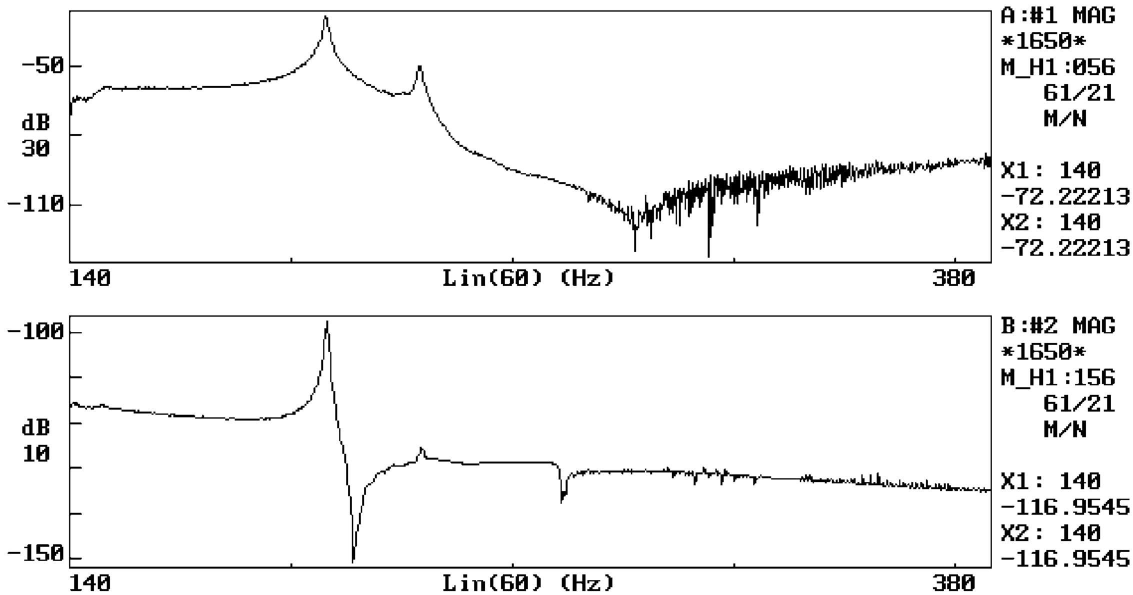

Figure 3 FRF calibration for the base excitation. The upper is the measured FRF, the lower is the calibrated FRF of the upper

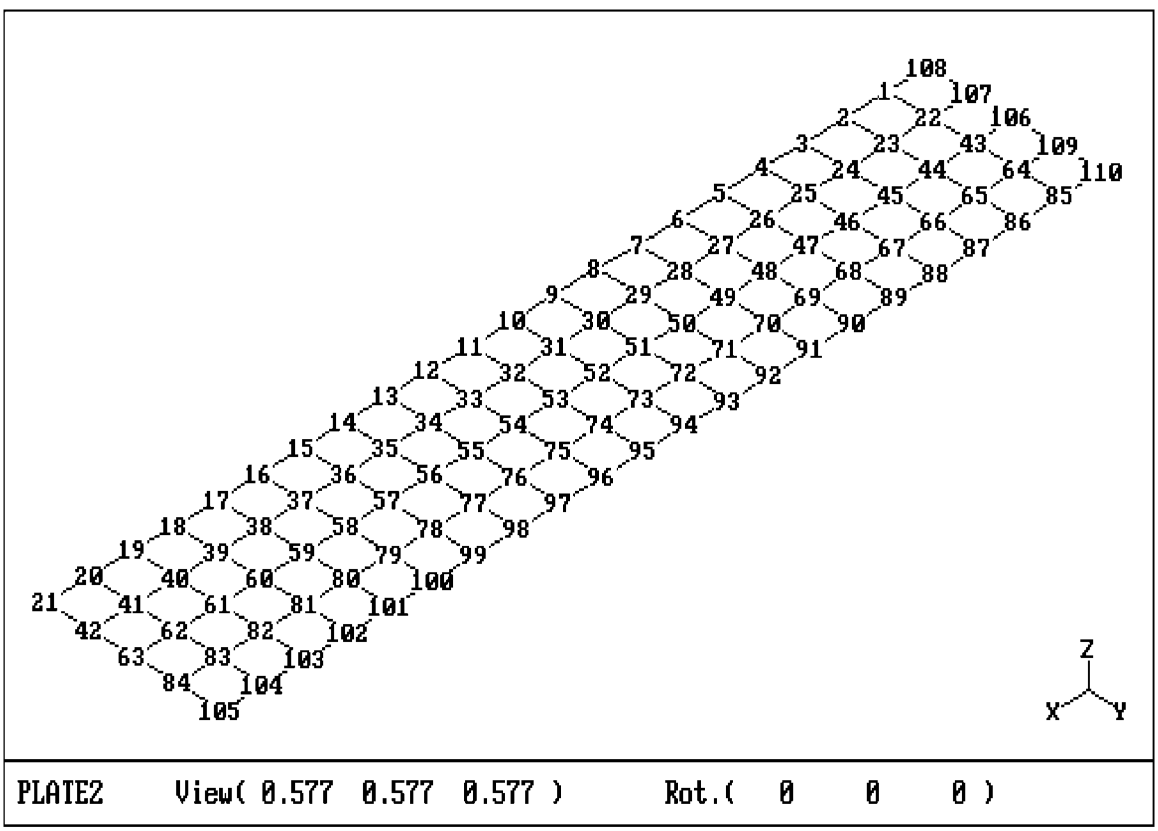

Figure 4 Geometric model of the cantilever plate

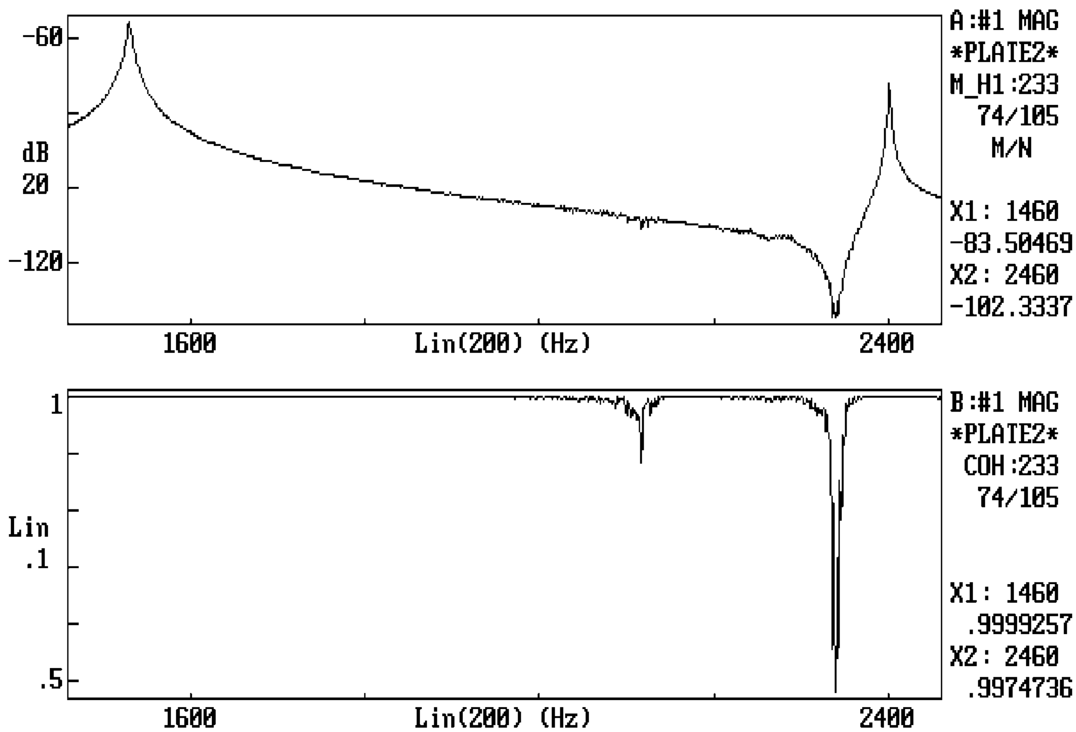

Figure 5 Measured FRF (upper) and coherence function (lower) of the plate at point 74

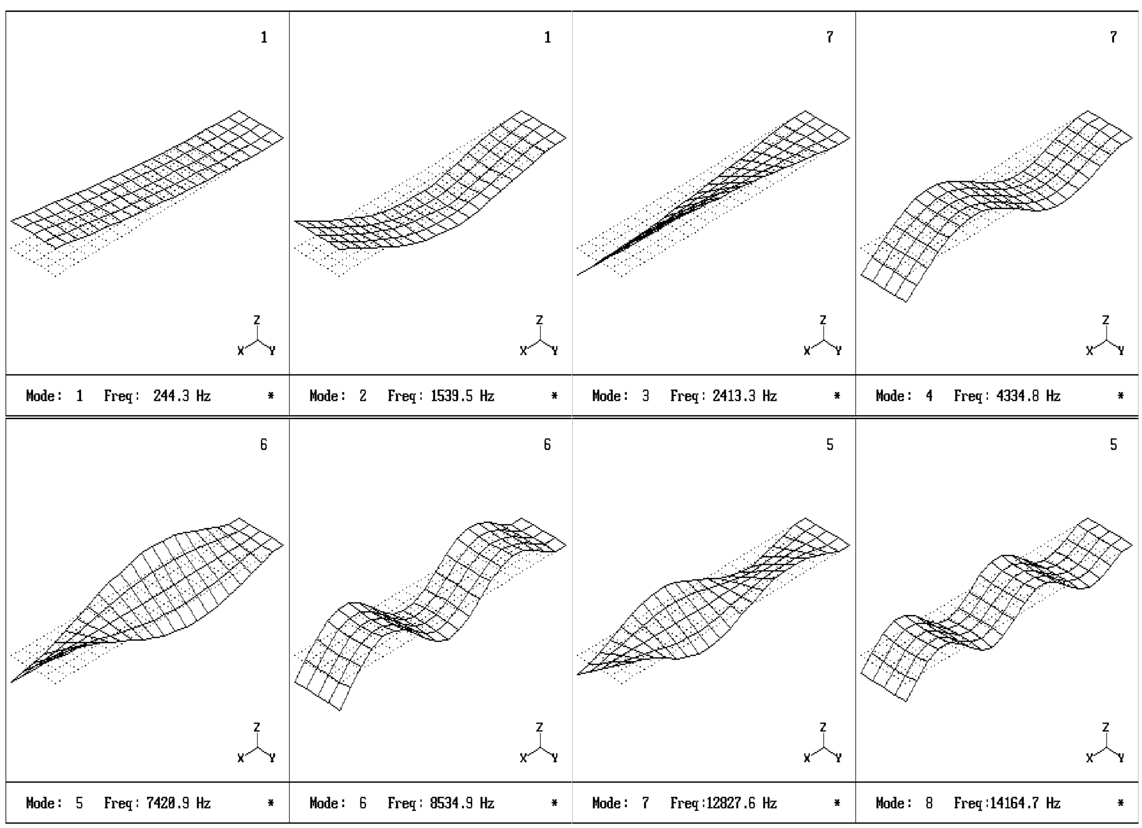

Figure 6 Measured mode shapes of the cantilever plate

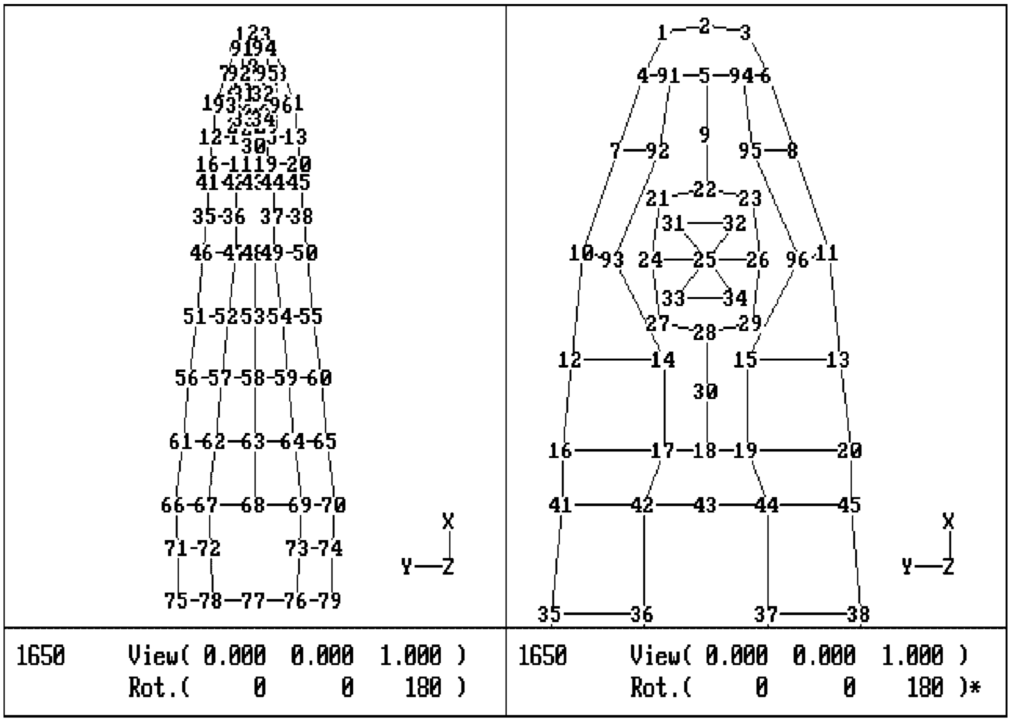

Figure 7 Geometric model of the type 1650 suspension

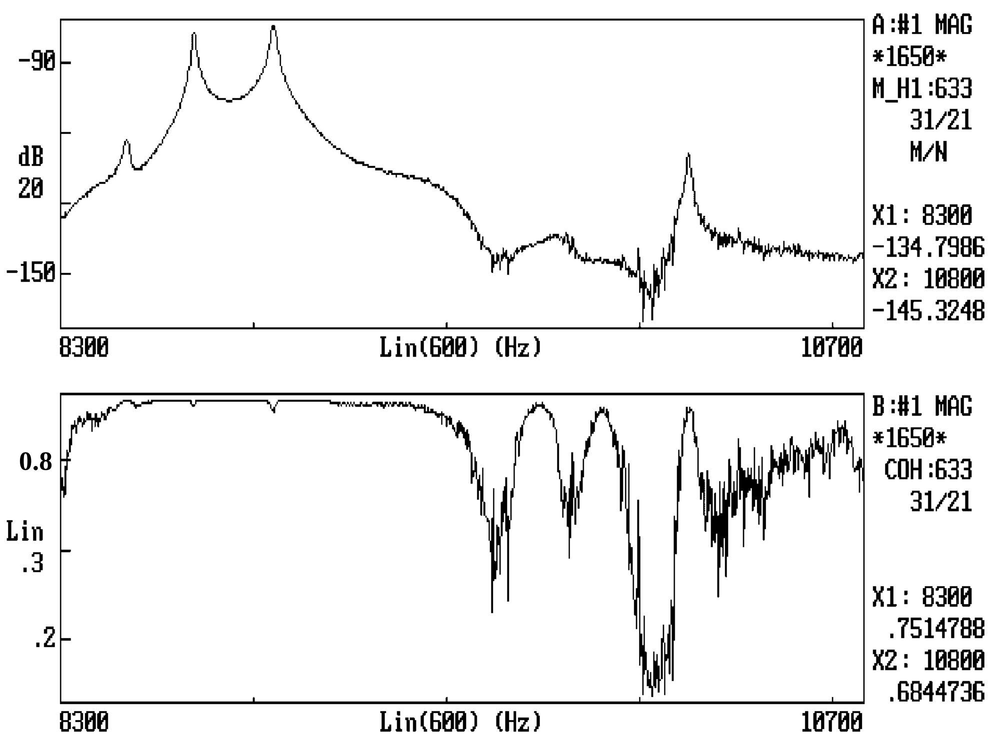

Figure 8 Measured FRF (upper) and coherence function (lower)

of the type 1650 suspension at point 31

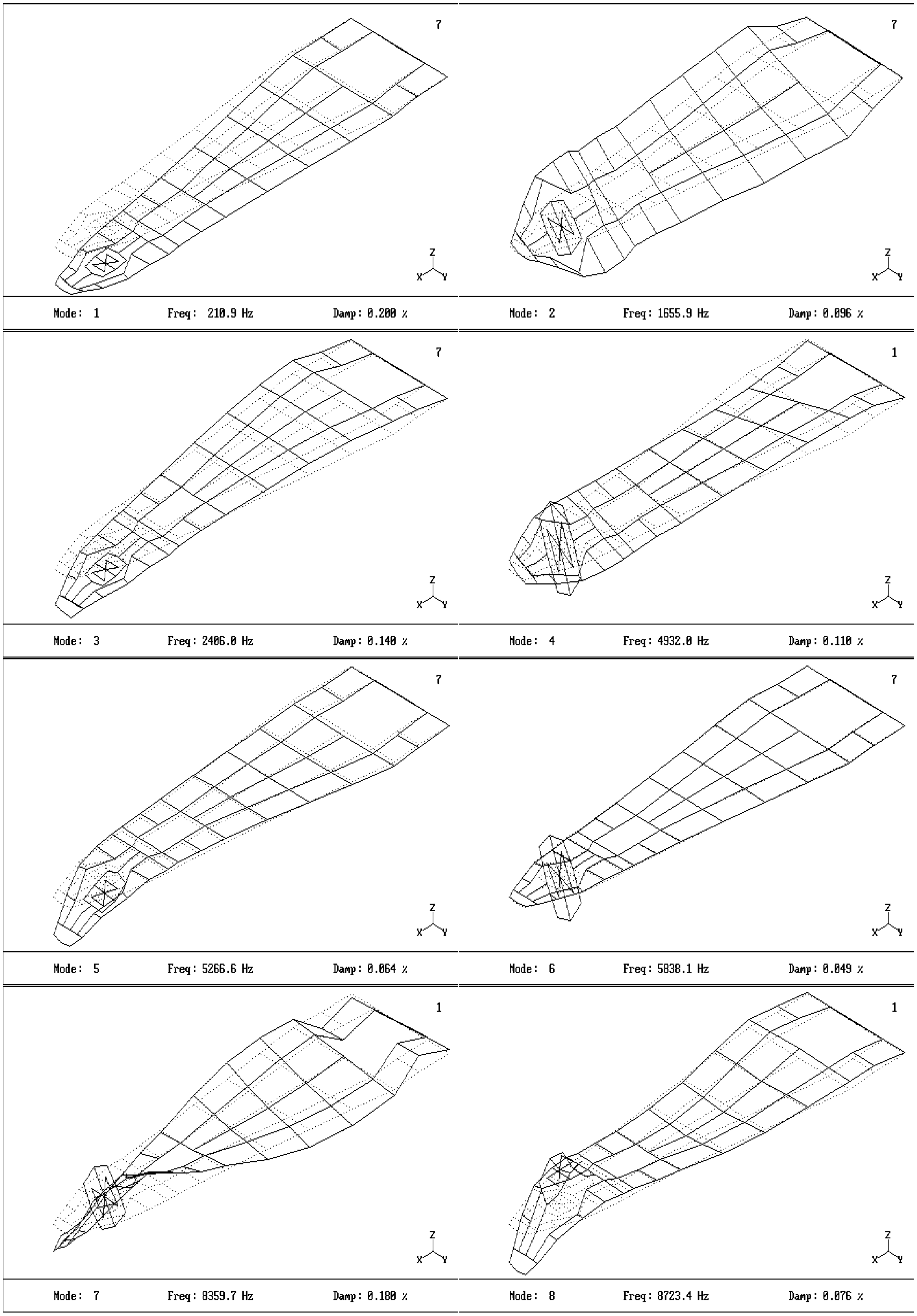

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Figure 9 Measured mode shapes of the Type 1650 suspension

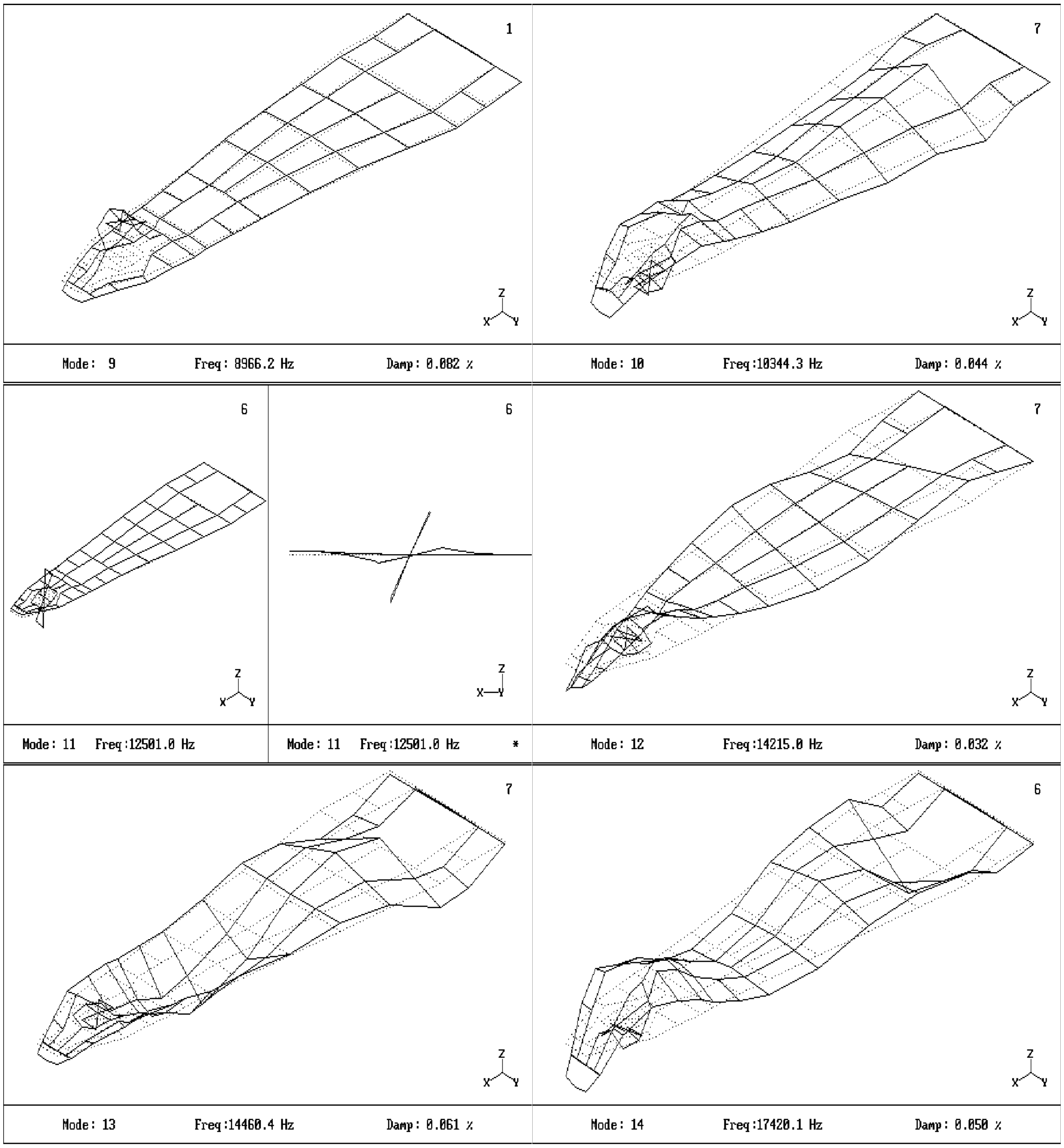

Figure 9 (continue) Measured mode shapes of the Type 1650 suspension