Conference Proceedings of the Society for Experimental Mechanics Series

Gary Foss · Christopher Niezrecki *Editors*

Special Topics in Structural Dynamics, Volume 6

Proceedings of the 32nd IMAC, A Conference and Exposition on Structural Dynamics, 2014

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ISSN 2191-5644 ISSN 2191-5652 (electronic)
ISBN 978-3-319-04728-7 ISBN 978-3-319-04729-4 (eBook)
DOI 10.1007/978-3-319-04729-4
Springer Cham Heidelberg New York Dordrecht London

Library of Congress Control Number: 2013940179

Printed on acid-free paper

Springer is part of Springer Science+Business Media (www.springer.com)

Preface

Special Topics in Structural Dynamics, Volume 6 represents one of the eight volumes of technical papers presented at the 32nd IMAC, A Conference and Exposition on Structural Dynamics, 2014 organized by the Society for Experimental Mechanics, and held in Orlando, Florida, February 3–6, 2014. The full proceedings also include volumes on Dynamics of Coupled Structures; Nonlinear Dynamics; Model Validation and Uncertainty Quantification; Dynamics of Civil Structures; Structural Health Monitoring; Topics in Modal Analysis I; and Topics in Modal Analysis II.

Each collection presents early findings from experimental and computational investigations on an important area within structural dynamics. Special Topics in Structural Dynamics represents papers on enabling technologies for modal analysis measurements such as sensors and instrumentation, and applications of modal analysis in specific application areas. Topics in this volume include:

Aircraft/aerospace
Active control
Analytical methods
System identification
Sensors and instrumentation

The organizers would like to thank the authors, presenters, session organizers, and session chairs for their participation in this track.

Seattle, WA, USA — Gary Foss
Lowell, MA, USA — Christopher Niezrecki

Contents

Chapter 1
Vibration Class at GIST, Korea

Semyung Wang, Jongsuh Lee, Youngeun Cho, Homin Ryu, and Kihwan Park

Abstract Vibration class covers vibration phenomena of mechanical systems due to dynamic load is studied. It covers from single DOF to multi DOF and theory as well as numerical and experimental methods. It deals various subjects: Lagrange equation, Laplace transformation, Fourier transformation, mode superposition, finite element method, experimental modal analysis, random vibration, vibro-acoustics and model validation.

Keywords Laboratory • Structural dynamics • Numerical analysis • Experimental modal analysis • Model updating

1.1 Introduction

Advanced vibration course is intended for graduated students; its goal includes deep understanding and estimating the mechanism of how vibration takes place in a mechanical system as well as understanding the vibration theory [1–4]. The class begins with a detailed description of single degree of freedom (DOF) and two DOF systems for the underlying understanding of vibration. In these parts, the system responses, which are governed by differential equation, are investigated in time and frequency domain. These equations are used to explain the characteristics of response with respect to different damping ratio values (under, over and critical damping) and different kinds of damping (viscous, structural). For the case of multi DOF system which is represented by matrix, the system is analyzed based on the eigenvalue problem.

Especially, mode shape of the system which is represented by eigenvector is introduced, and it is described that the several characteristics (reciprocal theorem, modal matrix) caused by the orthogonal property between this vector and system matrix. In addition, it is introduced that the frequency response function (FRF), which is the response of the applied force in the frequency domain, can be described by mode summation approach. For continuous system, modal parameters of the system are investigated through the governing equation, and the form of FRF is examined in the same way to the previous one by mode summation approach.

In order to carry out experiments for vibration analysis, covers following contents are covered in the class [5].

1. Different signals of force and types of sensors
2. Fundamental understanding of digital signal (leakage, window, power spectra, coherence and etc.)
3. Modal parameter estimation methods (peak picking, circle fitting, etc.)

To relate the theory with the experiments and for students' better understanding of the knowledge delivered in the class, two projects are assigned. One, as a common project to every student, is for verifying the theory, which they have learned in the class, by implementing the simple model (beam, plate). The basic purpose is to obtain the dynamic characteristics (natural frequency, damping, mode shape and FRF) of the system through analytic approach and these obtained results are verified from numerical analysis and experimental results. The other project is chosen as an individual topic that should be related to individual research area.

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G. Foss and C. Niezrecki (eds.), *Special Topics in Structural Dynamics, Volume 6: Proceedings of the 32nd IMAC, A Conference and Exposition on Structural Dynamics, 2014*, Conference Proceedings of the Society for Experimental Mechanics Series,
DOI 10.1007/978-3-319-04729-4_1,

This paper is organized as follows. In Sect. 1.2, a brief description of the contents covered in the class is introduced, and in Sect. 1.3, the conducted projects are introduced as divided into common and individual topics. This paper is concluded in the Sect. 1.4. In addition, the detailed description of the commercialized non-contact sensor, i.e. laser scanning vibrometer (LSV), which is developed by this laboratory and is used in the common project, is attached as an Appendix.

1.2 Class Contents

1.2.1 Single Degree of Freedom

In this chapter, a single DOF system as shown in Fig. 1.1 is introduced. It begins with derivation of the equation of motion shown in Eq. (1.1) for the system using Newton's second law.

$$m\ddot{x} + c\dot{x} + kx = F(t) \tag{1.1}$$

Next, the explanation of the resonance and resonance frequency is introduced from the undamped free vibration case. The solution of the homogeneous differential equation is derived. Next, damping ratio is introduced followed by vibration characteristics according to the damping (underdamped, over damped, critical) and the types of damping (structural, coulomb, viscous damping) are presented to the students.

In the next problem, forced vibration is examined and shown through an example of vibration due to rotating unbalance under the harmonic excitation. The solution of forced vibration equation of motion composed of homogeneous solution and particular solution is shown, and the concept of frequency response function as shown in Eq. (1.2) is introduced.

$$\frac{X}{F} = \frac{1}{\sqrt{(k - m\omega)^2 + (c\omega)^2}} \tag{1.2}$$

Students can understand that the vibration frequency response is dominated by the stiffness term at lower frequency than resonance frequency, and by the mass term at frequencies higher than the resonance frequency, also by the damping term at near resonance frequency as shown in Eqs. (1.3)–(1.5). Then, structure modification using vibration frequency response characteristics and estimation of mass and stiffness based on an experiment are introduced.

$$\omega \ll \omega_n : \frac{X}{F} \approx \frac{1}{k} \tag{1.3}$$

$$\omega \gg \omega_n : \frac{X}{F} \approx -\frac{1}{m\omega^2} \tag{1.4}$$

$$\omega \approx \omega_n : \frac{X}{F} \approx \frac{1}{jc\omega} \tag{1.5}$$

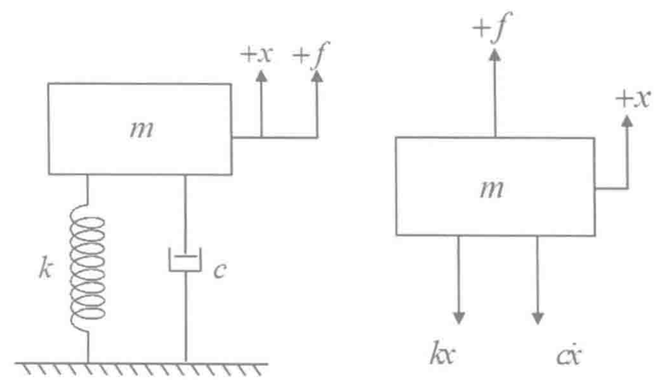


Fig. 1.1 Damped single degree of freedom system

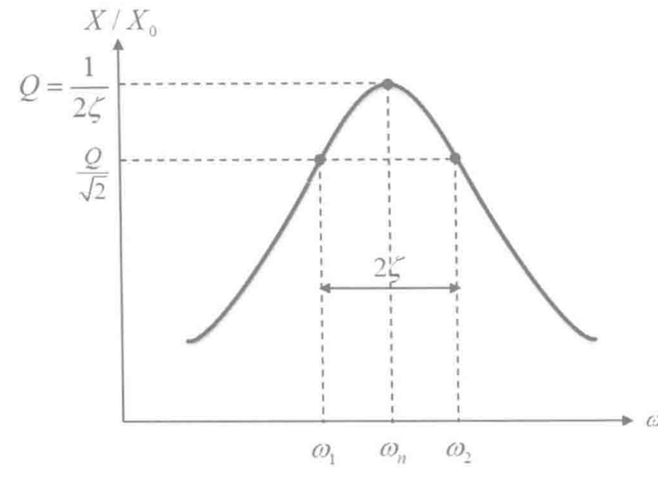


Fig. 1.2 Half power points and Q-factor

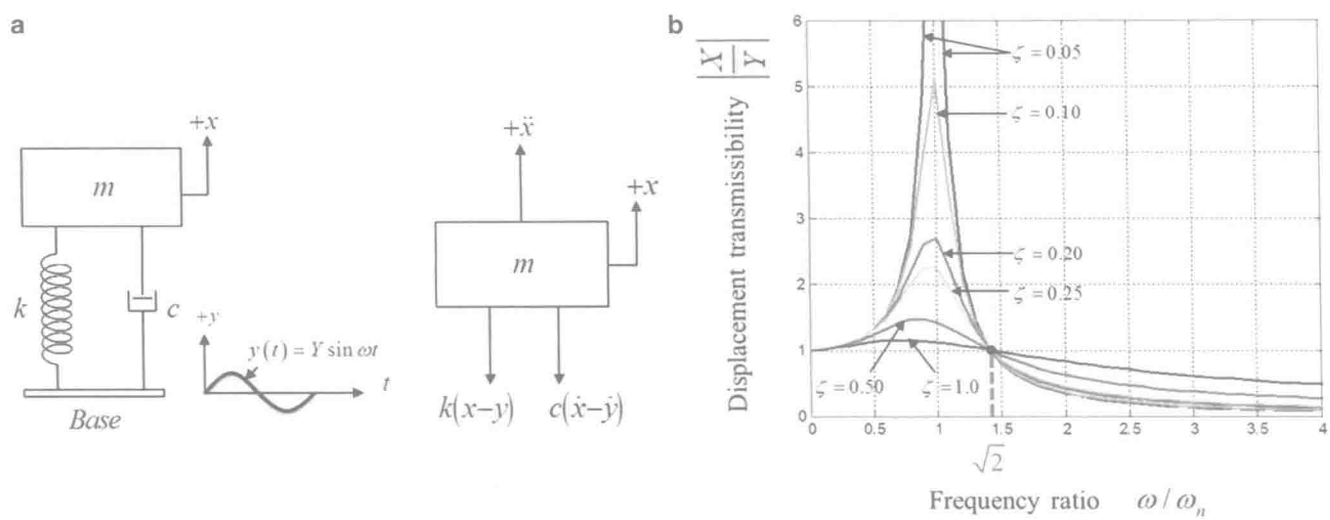


Fig. 1.3 (a) System excited by motion of support point (b) Transmissibility curve

The estimation of the damping ratio using Q-factor in FRF is offered to the students. Specifically, the process of calculating the Q-factor using half power points is explained and the equivalent viscous damping can be estimated from this process as shown in Eq. (1.6) and Fig. 1.2.

$$Q \approx \frac{1}{2\zeta} \approx \frac{\omega_n}{\omega_2 - \omega_1} \tag{1.6}$$

Next, support motion is introduced, and it leads to the explanation of transmissibility as shown in Fig. 1.3. It is shown that the principle of vibration isolation, and the transmissibility becomes smaller than 1 at the frequency range where the frequency ratio is larger than $\sqrt{2}$.

In addition, method of obtaining the transient response under non-periodic excitation is introduced. Transient vibration problem under impulse excitation is firstly introduced in order to convey the content to students successfully. Then, it is shown that vibration response under arbitrary excitation can be obtained using the combination of the impulse responses.

1.2.2 Multi Degree of Freedoms

Even though an actual structure is a continuous system, the continuous system is discretized and modeled as a multi degree of freedoms (MDOF) system (Fig. 1.4) for a practical sense. Finite element method, boundary element method, and transfer

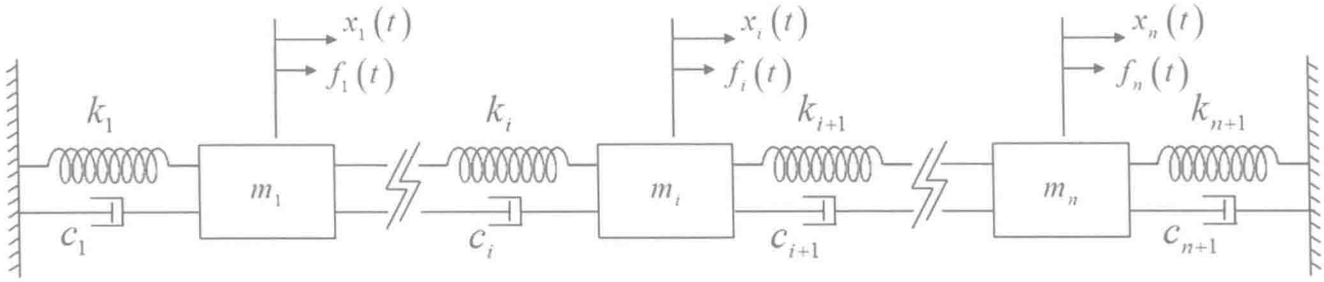


Fig. 1.4 Damped multi degree of freedoms system

matrix method are introduced as general discretization methods, and the discretization process of the vibration system is explained. The equation of motion in matrix form is derived as shown in Eq. (1.7).

$$\mathbf{M}\ddot{\underline{x}} + \mathbf{C}\dot{\underline{x}} + \mathbf{K}\underline{x} = \underline{F}(t) \tag{1.7}$$

The MDOF undamped free-vibration problem is switched to an eigenvalue problem, and the result shows that eigenvalues and eigenvectors can be equally considered as natural frequencies and mode shapes respectively. Also, the relationship between the number of the degree of freedom and the number of the natural frequencies and mode shapes can be explained. The eigenvectors of the system are shown to be orthogonal with respect to both mass and stiffness matrices. Modal matrix which assembles eigenvectors into a square matrix is introduced. By using the modal matrix, decoupling of the forced vibration terms and modal damping concept can be explained.

In forced vibration case, FRF as shown in Eq. (1.8) can be obtained using the orthogonality of eigenvectors. Additionally, Maxwell's reciprocity theorem states that $H_{ik} = H_{ki}$ for the linear system,

$$H_{ik}(\omega) = \sum_{r=1}^{N} \frac{\phi_r^i \phi_r^k}{(k_r - \omega^2 m_r) + j(\omega c_r)} \tag{1.8}$$

When the system becomes larger and more complex, DOF is increased. This leads to the difficulty of calculating the exact solution. To solve this kind of problem, an approximate solution is introduced. Mainly, superposition methods such as mode displacement method (MDM), mode acceleration method (MAM), load dependent Ritz vectors (LDRV) method, Krylov sequence, and Lanczos algorithm are explained, and the advantages and disadvantages of the each method are explained as well.

Newton's second law, energy method, and virtual work method are compared with each other so that the equation of motion is formulated. Consequently, Lagrange's equation as shown in Eq. (1.9) is introduced to formulate the large and complex system.

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_i}\right) - \frac{\partial T}{\partial q_i} + \frac{\partial U}{\partial q_i} = Q_i \tag{1.9}$$

1.2.3 Experimental Modal Analysis

Experimental modal analysis (EMA) contains experimental measurement process for the FRF of the system; signal processing, and extracting the modal parameters (natural frequencies, mode shapes, and damping ratios) from measured FRF. Verifying a numerical model, determining dynamic durability by experiments, and machinery diagnostics for maintenance are possible by using the modal parameters.

The techniques needed to experimentally determine the FRF showing the relationship of response and excitation force are introduced. There are two kinds of methods to excite a structure. First case is supplying excitation force by attaching a vibration exciter as shown in Fig. 1.5a. In this case, in order to reduce the mass loading due to an attached vibration exciter, a stinger should be used. The type of signals that is applied to the vibration exciter is introduced,and the characteristics in the

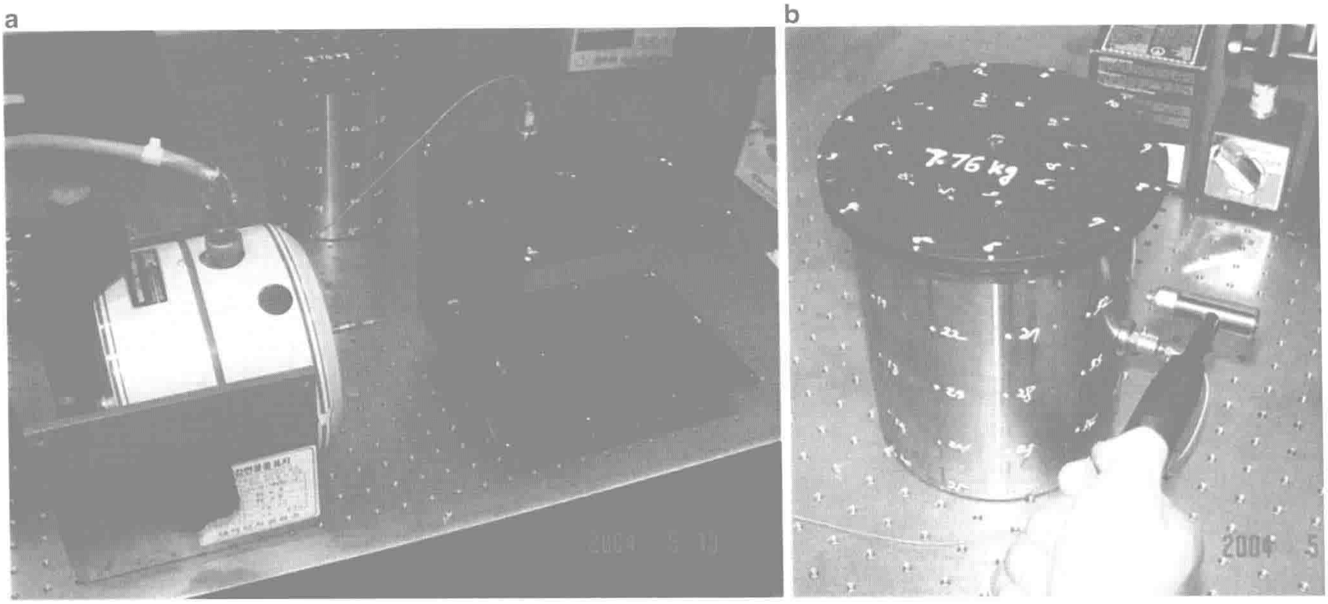


Fig. 1.5 Vibration excitation methods (**a**) Vibration exciter with a stinger (**b**) Impact hammer

frequency domain corresponding to each signal are described. Second case is to use an impact hammer as shown in Fig. 1.5b. In this case, the characteristics of excited force in the frequency domain are explained. Also, effects of the header and tip of the hammer corresponding to the frequency range of interest are discussed. Lastly, the advantages and disadvantages of both methods are compared.

The sensors for detecting a response and excitation force applied to the structure are explained. There are two kinds of sensors to measure vibration signal, which are contact and non-contact. The contact sensors are an accelerometer and a strain gauge. Because they are attached to the structure, there is a mass added effect. The non-contact sensors are a position sensitive detector (PSD) to measure displacement and a LSV to measure velocity. Because LSV can rapidly measure the vibration of several positions, it makes EMA easier. LSV is a device that was developed in this laboratory; a detailed description is attached in the Appendix.

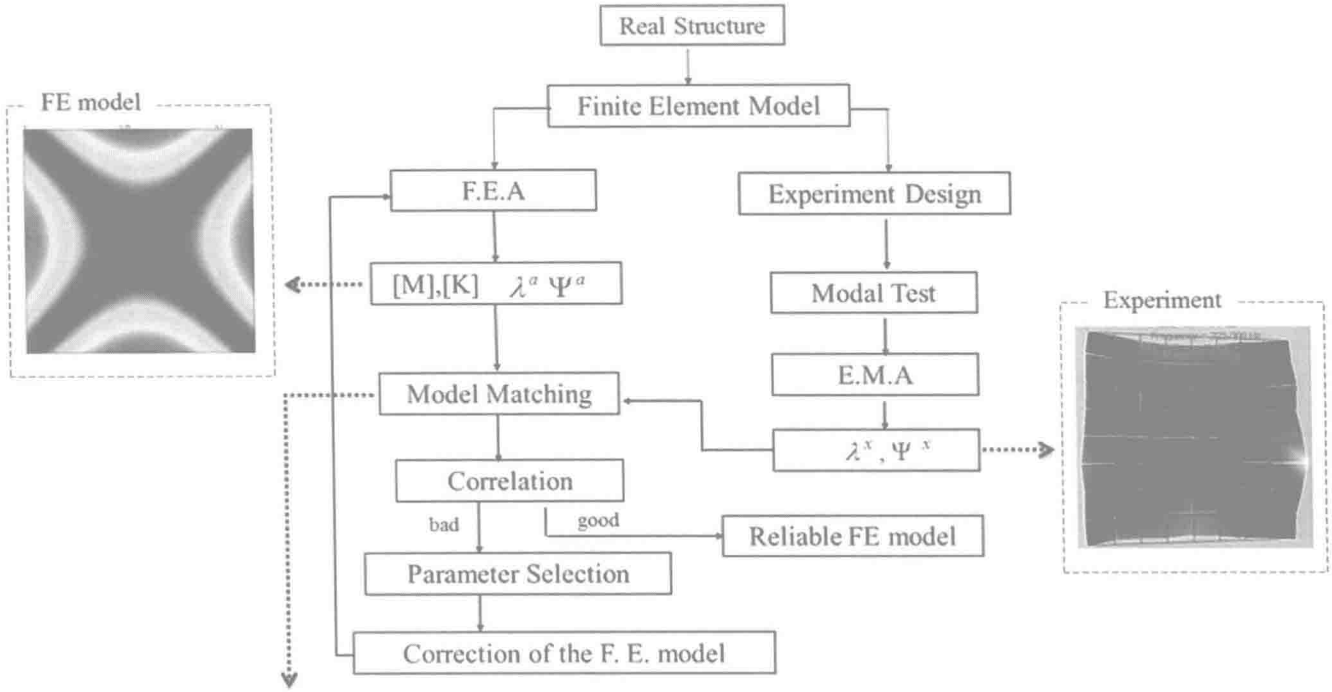
Digital signal processing theory for obtaining FRF from measured data is introduced. Because EMA contains the sampling process which is converting an analog signal into a digital signal, Nyquist sampling theorem as shown in Eq. (1.10) should be described.

$$f_s \geq 2f_0 \tag{1.10}$$

The aliasing phenomenon in which high-frequency component is detected in the low-frequency component occurs when the sampling rate is not enough. Hence, antialiasing filter is used to prevent the aliasing phenomenon. Fourier transform is explained because frequency conversion is needed to obtain FRF from the measured signal. Also, the leakage in which the frequency component power leaks to adjacent frequency component is explained. The window function which can reduce the leakage is shown and, its principle is explained.

In processing techniques of experimental measurement signal, correlation function and spectral density function can obtain the correlation between the two signals in time domain and frequency domain. The linear relationship between the input and output signals (FRF) can be expressed in the correlation function and the spectral density function. And the coherence function which may indicate the degree of noise mixed in the signal via the spectral density function is introduced. Therefore, a coherence function can show whether the characteristic values measured in the experiment are being measured correctly.

The peak picking and circle fitting which extracts the modal parameters from experimentally measured FRF are explained. And it is possible to observe the mode shapes using predicted values at some points, i.e. by the mode analysis method introduced.



Results of MAC analysis

COMSOL \ LSV	$\{\psi\}_1$	$\{\psi\}_2$	$\{\psi\}_3$	$\{\psi\}_4$	$\{\psi\}_5$
$\{\psi\}_1$	0.98273	0.06188	0.041013	0.036478	0.02415
$\{\psi\}_2$	0.02451	0.987455	0.041013	0.032891	0.01541
$\{\psi\}_3$	0.023777	0.08089	0.93237	0.126	0.126
$\{\psi\}_4$	0.0097108	0.019525	0.036478	0.91887	0.057609
$\{\psi\}_5$	0.032891	0.058354	0.10896	0.023777	0.91887

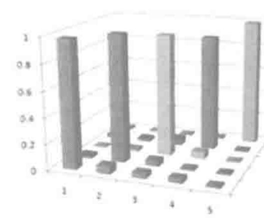


Fig. 1.6 Model updating process

1.2.4 Model Updating Process

It is possible to verify the numerical model using the modal parameters obtained by the EMA. However, since the error exists between FE model and EMA results, the need for reliable FE model should be explained. A model updating for matching between the dynamic characteristics of numerical model and modal parameters obtained via the EMA by changing the parameters of the numerical model is introduced. It is the required process to construct the FE model for the optimal design. Figure 1.6 is a flow chart showing the model updating process of a square steel plate.

1.3 Class Project

1.3.1 Midterm Project

Midterm project is assigned to offer an opportunity to recognize the relationship between vibration theory and experiment based on the class contents introduced in Sect. 1.2. Specifically, student should obtain modal parameters using the governing equation of the system through analytical, numerical and experimental approach. In this way, students can understand each process and the relationship between them. Finally, numerical model updating process based on the experimental results is conducted repeatedly for the numerical model validation. The following two application systems are considered.

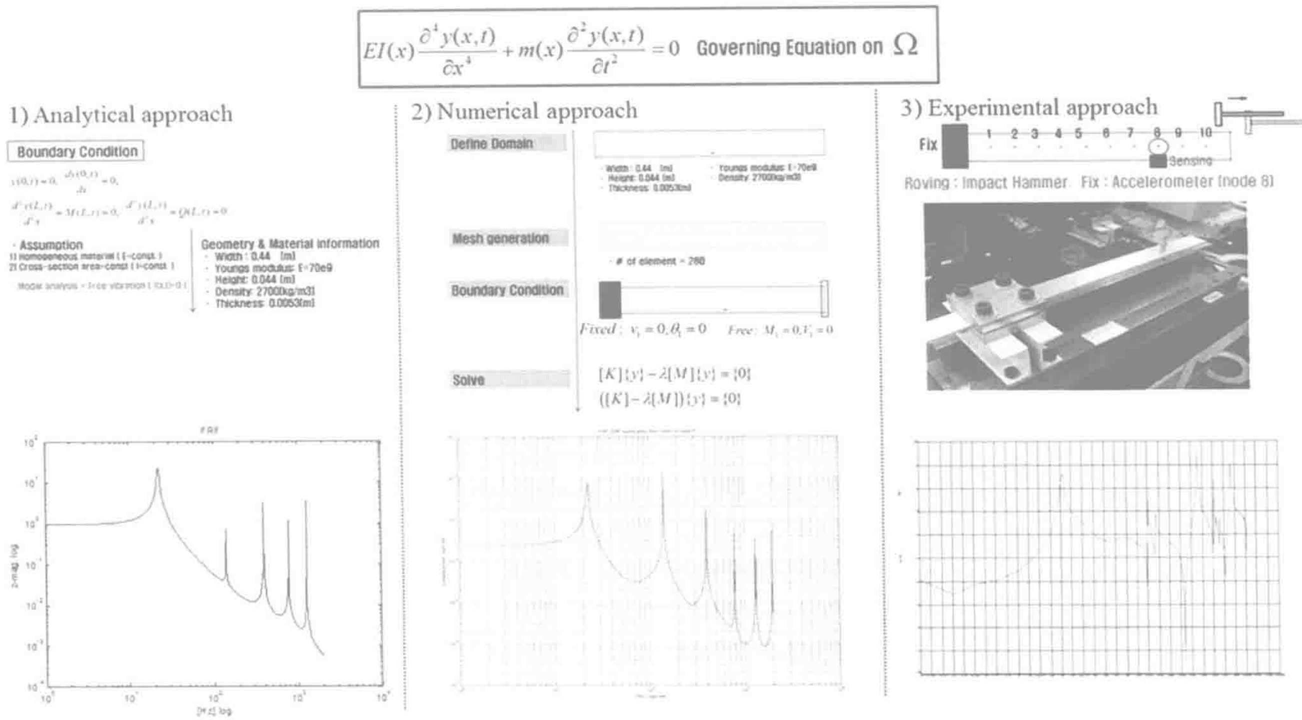


Fig. 1.7 Midterm project of a cantilever beam using analytical, numerical, experimental approach

1. Cantilever beam: Accelerometer and Impact hammer
2. Plate and Brake-disk: LSV and Vibration exciter

1.3.1.1 Cantilever Beam

Cantilever beam is given to the students for the experimental modal test using an impact hammer and an accelerometer. As mentioned previously, modal parameters using analytical and numerical approach are obtained using the governing equation and boundary condition of the system. Meanwhile, modal parameter using experimental approach is obtained through roving impact hammer test as shown in the experimental setup in Fig. 1.7. Student can verify the vibration theory of the cantilever beam by comparing the results from three different approaches.

1.3.1.2 Plate and Brake-Disk

Next, plate and brake-disk are given to the student for the experimental modal test using a laser scanning vibration rather than an impact hammer and an accelerometer. Analytical approach is based on a related reference paper [6]. COMSOL (commercial numerical analysis program) offered to students for finding the modal parameters of the structure numerically; material properties of the structure are given in Table 1.1. A laser scanning vibrometer is offered in experimental approach (non-contact sensor) for measuring the vibration response of the system under the excitation condition using a vibration exciter.

Tables 1.2 and 1.3 represent modal parameters of plate and brake-disk respectively obtained from student's midterm report.

Students can understand different facts from Tables 1.2 and 1.3. Firstly, modal parameter results of the plate using analytical approach and numerical approach are similar. On the other hand, modal parameter results using experimental approach has a little difference as compared with that of other two approaches. This result tells that the numerical model used in numerical approach has a little difference as compared to actual system. Finally, modal validation process is conducted while repeating the numerical model updating in order to reduce this difference.

Table 1.1 Description of plate and brake-disk and experimental setup

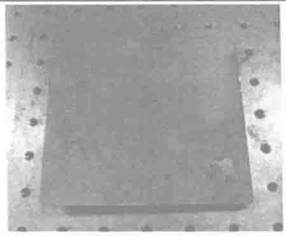
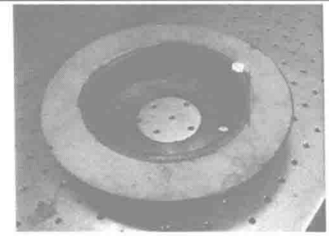
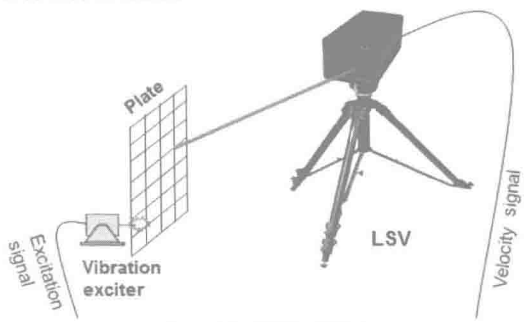
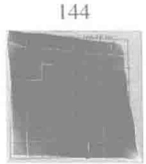
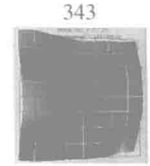
Objectives		
	Plate	Brake-disk
Material properties	1. Dimension: 0.119 × 0.119 × 0.0006 (m^3) 2. Density: 2,700 (kg/m^3) 3. Young's Modulus: 75 (GPa) 4. Poisson's ratio: 0.3	1. Density: 7,450 (kg/m^3) 2. Young's Modulus: 115 (GPa) 3. Poisson's ratio: 0.33
Boundary condition	Free–free condition	
Experimental setup		

Table 1.2 Modal parameter results of a plate

	1	2	4	5	6
Analytical approach (Hz)	145.076	212.833	376.687	376.687	661.719
Numerical approach (Hz)	144.515	210.729	373.430	374.450	656.764
					
Experimental approach (Hz)	144	225	336	343	536
					

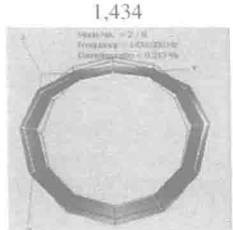
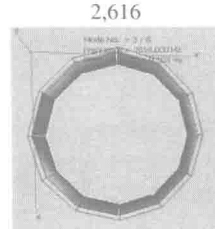
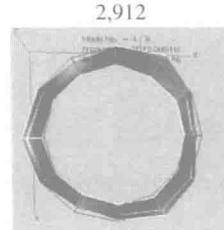
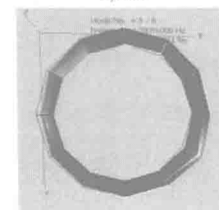
1.3.1.3 Conclusion of Midterm Project

Students have an opportunity to apply the vibration theory into the real system through a modal analysis using analytical, numerical and experimental approach. Moreover, students can understand the features and pros and cons of the each modal analysis approach. Lastly, they can learn the process of numerical model updating and validation based on the experimental result.

1.3.2 Individual Project

Individual project proceeds with proposal and presentation. First of all, the description of the project and the information regarding student final goal are presented in the proposal, the students have to show detail project overflow, as develop through discussion and brainstorming. The evaluation on the conducted individual project is carried out depending on the

Table 1.3 Modal parameter results of a brake-disk

	1	2	3	4
Numerical approach (Hz)	1,358.199	2,555.593	2,578.565	3,050.416
				
Experimental approach (Hz)	1,434	2,616	2,912	3,370
				

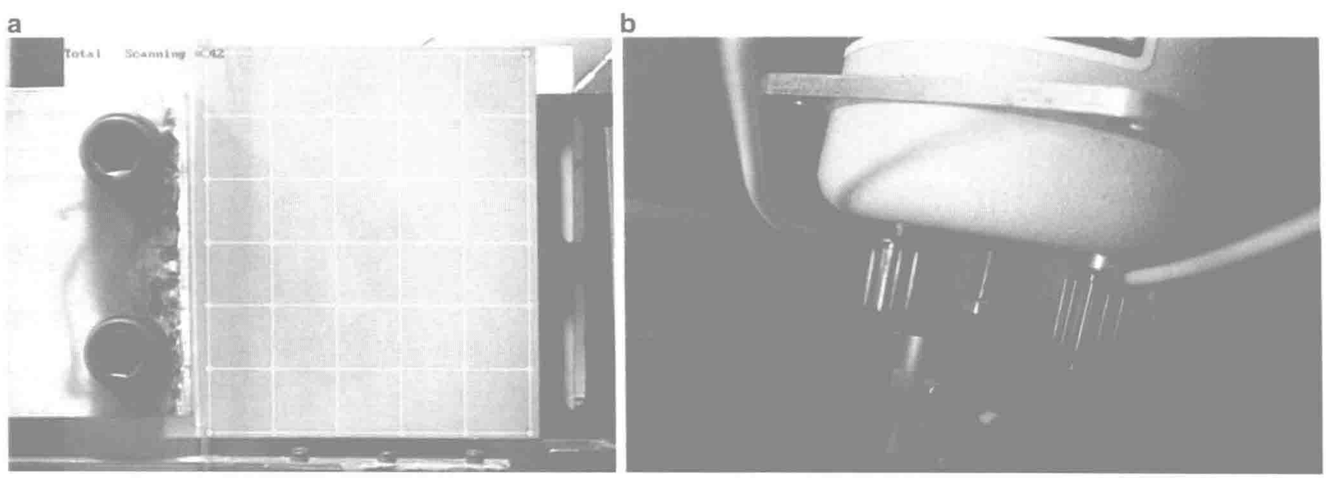


Fig. 1.8 (**a**) Experimental configuration and measured nodes using LSV (**b**) Attached vibration exciter and accelerometer on the backside of the plate as an operator and a reference

novelty, difficulty, degree of performance compared to the proposal, presentation material, etc. In this paper, a research on the damage detection by using operating deflection shape (ODS) is introduced, which is selected from the presented researches. Its purpose is to find out the location of damage or failure of a mechanical system by comparing [through Eq. (1.11)] the deflection shapes between un-damaged and damaged one. This project is briefly introduced in the following Figs. 1.8 and 1.9. This research performed in the class was published in the journal paper [7].

$$\text{Damage detection} = (\varphi_{h_i} - \psi_{u_i})^2 \tag{1.11}$$

1.4 Conclusion

This paper is an introduction given to graduate students in vibration class at GIST, Korea. Vibration class contents and individual projects are introduced. These individual projects are assigned to the students for better understanding of vibration by applying the vibration theory that they learned in the class to the real mechanical system. Midterm project is used to familiarize the students with the vibration system using analytical numerical and experimental approach. Finally, validation of the numerical model is conducted using model updating procedure. Another individual project is assigned to combine the theory with the individual research topic of the respective student. Moreover, Appendix covers the syllabus of the vibration class and the explanation of the commercial non-contact sensor developed and used by this laboratory.

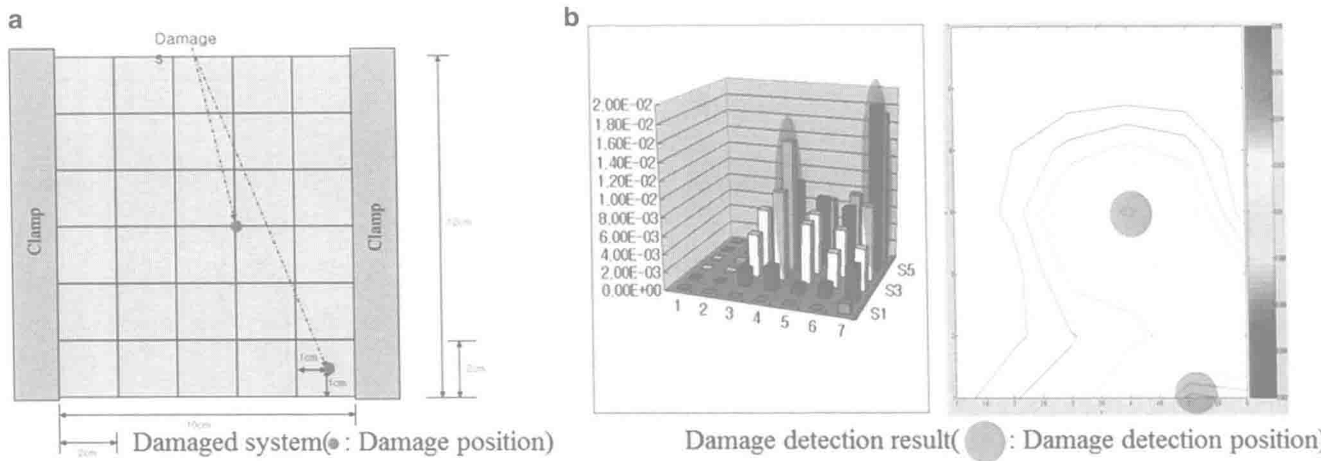


Fig. 1.9 (**a**) Locations of the damages (**b**) Comparison result between damaged ODS and un-damaged ODS bar and contour plot

A.1 Appendix

A.1.1 Syllabus

5603 Advanced Vibration Summary:
In this course, vibration phenomena of electro-mechanical systems due to dynamic load is studied. It covers from single DOF to multi DOF and theory as well as numerical and experimental methods. It deals various subjects: Lagrange equation, Laplace transformation, Fourier transformation, mode superposition, finite element method, experimental modal analysis, random vibration, mode component synthesis, rotor dynamics, vibro-acoustics.

Text:

1. *Theory of Vibration with Applications*, 5th ed., W. T. Thomson and M. D. Dahleh, Prentice Hall, 1998.

References:

2. *Vibration with Control*, D. J. Inman, Wiley, 2006.
3. *Structural Dynamics: An Introduction to Computer Methods*, R. R. Craig, John Wiley & Sons, 1981.
4. *Finite Element Procedures*, K.J.Bathe, Prentice Hall, 1996.

Prerequisites by Topics:
Engineering Mathematics
Fundamental Vibration
Tools Used:
Experimental Modal Analysis: Laser Scanning Vibrometer, Pulse Modal Test, LMS CADA-X
FEA codes: MSC/NASTRAN, ANSYS, COMSOL, SYSNOISE
Math tools: Matlab

Topics:
SDOF [1]
Free Vibration (damp free, damped)
Forced Vibration (unbalance, vibration isolation, damping)
Transient Vibration (impulse, arbitrary, shock)
MDOF [1]

(continued)

(continued)
MDOF (mode, forced harmonic vibration)
Properties of Vibration Systems (flexibility influence, stiffness influence, Castigliano's theorem, modal matrix)
Lagrange's Equation (virtual work, Hamilton's Equation)
Vibration Test [2]
Measurement Hardware
Digital Signal Processing
Random Signal
Mode Shape
Computational Methods [3]
Finite Element Method
Static Problem (Gaussian elimination, Cholesky decomposition)
Eigenvalue Problem (Rayleigh method, Lanczos)
Harmonic Problem (direct frequency, modal frequency)
Acoustics
Transient Problem (direct integration, mode superposition)
Component Mode Synthesis [2]
Static Condensation/Super Element
Component Mode Synthesis
Design Sensitivity Analysis
DSA of Static Problem
DSA of Eigenvalue Problem
DSA of Noise and Vibration
Random Vibration [1]
Vibro-acoustics
Rotor Dynamics

Exams: Midterm (35)
Projects: 2 (15, 30)
Homeworks: (20)

A.1.2 Introduction to the Development of Laser Scanning Vibrometer

A.1.2.1 Motivation of LSV Development

In general, vibration measurement equipment is essential in the development process of the structure system, which is used to identify the vibration characteristics of the structures including electric motors, automobiles, aircraft structures, nuclear reactors, towers, etc.

In the past, the contact sensor is mostly used in identifying vibration characteristics. However, the contact sensor can induce changes in the dynamics characteristics of the structure and has several limitations in vibration measurement due to its own features. For these reasons, the demand for the laser vibrometer is on the increase globally [8].

Currently, the most famous and expensive laser vibrometer has been developed by Polytec. However, as it is too expensive, intelligent system design laboratory (ISD) and venture company "EM4SYS" (www.em4sys.com) are collaborating for the development of the LSV (Fig. 1.10) in the School of Mechatronics of Gwangju Institute of Science and Technology.

A.1.2.2 Specification of LSV

Next is the specification of LSV as shown in Table 1.4 [9].