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Multi-Reference Modal Analysis of a Disc Brake to Identify the Coupled Mode Pair with Similar Eigen Value and Dissimilar Eigen Vectors

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ABSTRACT: Experimental modal analysis is used to achieve structure modal parameter by measuring and analyzing the dynamic response of a structure when excited by a stimulus. Modal view supports a variety of MDOF curve fitting analysis method and MIMO multi-reference experimental method using impact hammer. Multi-reference model testing is used under situations where a single reference DOF is not sufficient to find all modes. In this work, the vibration data of a disc brake made of aluminium was captured using the National Instruments Data Acquisition System (NI-DAQ) with uni-axial accelerometer. The natural frequencies, damping ratios and mode shapes of the disc brake were determined experimentally. Following this, finite element analysis was carried out using the finite element software, ANSYS. The values of dynamic properties determined from both numerical and experimental analysis were compared.

KEYWORDS: Experimental modal analysis, mode shapes, multi reference, symmetric structure, multivariate mode indicator function, mimo.

I. INTRODUCTION

Experimental Modal Analysis (EMA) has developed into a most important technology for the study of structural dynamics in the past several decades. Through Experimental Modal Analysis, complex structurephenomena in structural dynamics can be represented using coupled modesconsisting of natural frequency, damping, with mode shapes. The set of these modal parameters is referred to as Modal Model. Experimental Modal Analysis is normally referred to as Modal Analysis.

Experimental modal analysis is extensively used to express the dynamic characteristics of structures and to confirmAnalytical models. Moreover, these extracted modal parameters from the modal analyses are essential for various algorithms. Some experimental techniques are used to excite structures and to extract their modalparameters.

The modal parameters can be extracted from a set of Frequency Response Function (FRF) measurements among one or more reference positions and a number of measurement positions required in the model. A position is a point and a direction on the structure and is hereafter called a Degree of Freedom (DOF). The resonance frequencies and damping values can be found from any of the FRF measurements on the structure (except those for which the excitation or response DOF is in a nodal position, that is, where the mode shape is zero). To exactly model the associated mode shape, frequency response measurements must be made over a sufficient number of DOFs to ensure enough detailed coverage of the structure under test. The extraction of the modal parameters from the FRFs can be done using a variety of mathematical curve-fitting algorithms. In order to calibrate (scale) the modal model, the driving-point measurement, the measurement where the excitation and the response is in the same DOF, needs to be included.

The FRFs are obtained using multi-channel FFT measurements. To arrive at these FRFs, the excitation force from an impact hammer with a proper signal and responding vibrations are measured. The FRFs can be represented if the vibration response is measured in terms of acceleration; the FRF represents an accelerance function as it gives the complex ratio of acceleration over force in the frequency domain. For impact hammer excitation, each accelerometer response DOF is usually fixed and reflects a reference DOF. The hammer is then moved around the structure and used to excite every DOF needed in the model. Using test data obtained by exciting at two or more locations often resolves this problem. This approach of using more than one excitation and response location is called MIMO - (multiple-input, multiple-output).

In this paper consists of a specimen that is in the form of symmetrical structure. This symmetrical structures are widely used in the industrial machineries, automobiles etc. Modal estimation of these structures cannot be done by common procedure of experimental modal analysis using single reference, as these would miss out the estimation of closely coupled modes of a symmetrical structure. In this work we perform a multi reference modal testing in disc brake plate to estimate the modal parameter.

A Modal Analysis includes both data acquisition and the following parameter identifications. From its inception till now, Modal Analysis hasbeen widely applied in mechanical and structuralengineering for designing, optimizing, and validating purposes. It has been widely accepted for broad applications in industries, automotive, aerospace, power generation, civil engineering, musical instruments etc...

II. MATERIALS AND METHODS

Conducting experimental modal analysis considering the multi-reference model point and to extract the modal parameter for better correlation of the consequences with the FE model of a disc was developed considering the realistic dimensions. Disc brake plate assembly was examined considering the free-free boundary conditions as it allows the structure to vibrate freely without interference of the other parts. It also facilitates better visualization of mode shapes associated with each natural frequency. For validation of FE model one must recognize the dynamic properties as modulus of elasticity, density and Poisson's ratio of the components.

III. EXPERIMENTAL MODAL ANALYSIS (EMA) PROCEDURE

The disc brake plate assembly was tested through the EMA with free-free boundary condition. The observational approach to investigate the way patterns and natural frequencies of the structure through impact hammer test consists of the next steps.

1. Model test setting.
2. Divide the structure inadequate number of points with the appropriate special distribution.
3. Shake up the structure with impact hammer.
4. Taking the measurements.
5. Analysis of measured output data.
6. Establishment with the FEM data.

The test equipment used for the experimentation is the Fast Fourier Transform (FFT) with four channels along with data acquisition system (NI-DAQ 9234). The structure was excited using impact hammer (model number 086C03) at all predefined locations as indicated in fig. 1 and the response was collected using uni-axial accelerometer (PCB 603C01**)** at an identified driving point transfer function (DPTF) location. The type of EMA is known as the Frequency Response Function (FRF) method which evaluates the input excitation and output response simultaneously. The essence of all frequency response functions (FRF's) was resolved to extract natural frequencies, damping and mode shapes. Fig.2 shows the experimental modal analysis set up for disc brake plate.

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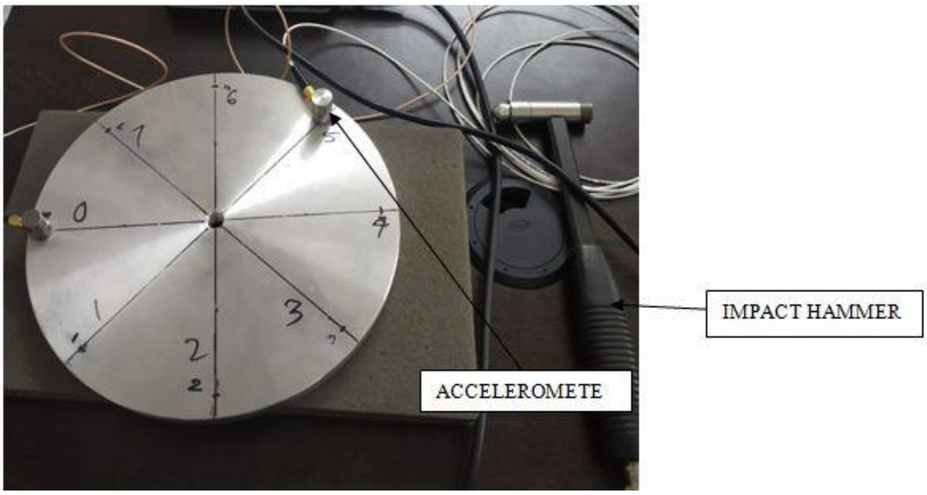

FIG-1 ACCELEROMETER LOCALIZATION

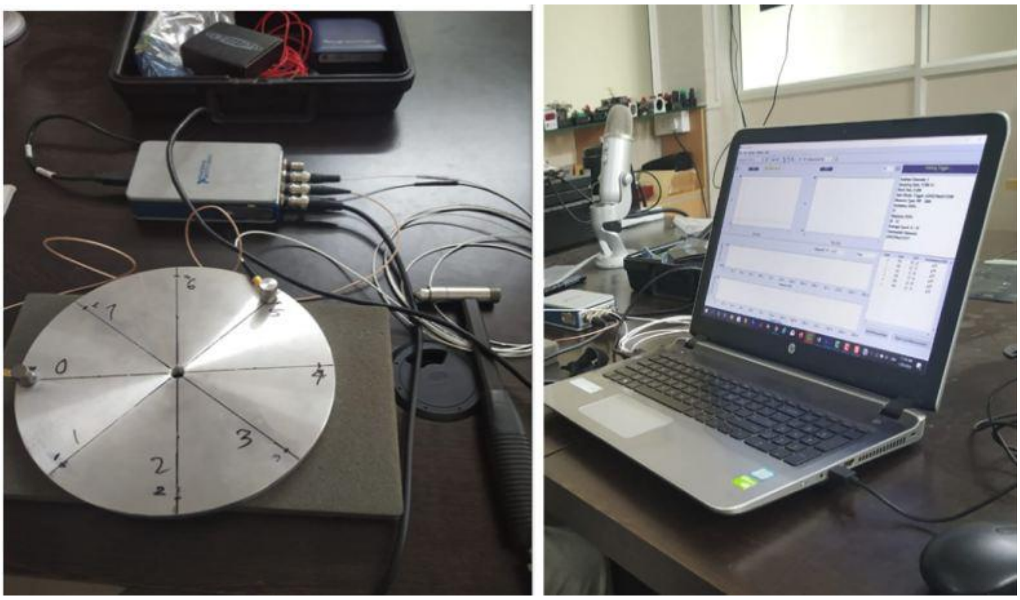

FIG-2 FREE-FREE CONDITION FOR DISC BRAKE PLATE EMA

IV. FE MODAL ANALYSIS OF DISC BRAKE PLATE

Modal analysis of disc brake plate has been carried out for the materials Aluminium (Al) as it is one of the essential parts of the brake system which contributes more in the generation of unbalance in a system. First ten models are extracted for the disc brake plate for predicting the natural frequencies. Solid 45-3D structural solid element is utilized for the three-dimensional modelling of structure. The element is defined by eight nodes having one degree of freedom at each node. Role of element shape and order of interpolation decides the element selection

V. RESULTS AND DISCUSSION

Natural frequencies are obtained for a disc brake plate of aluminium material using EMA and FEM are presented in the table1 and table2. While the table3 shows the value of deformation i.e. displacement of disc at particular mode.

Index	Frequency (Hz)		Damping (%)		Index	Frequency (Hz)		Damping (%)	
	0Z	5Z	0Z	5Z		0z	5z	0z	5z
0	1497	1497	0.8464	0.8464	6	3440	3440	0.6579	0.6579
1	1497	1497	0.8207	0.8207	7	3497	3497	0.60	0.60

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2	1510	1510	0.6769	0.6769	8	3498	3498	0.5827	0.5827
3	2706	2706	0.537	0.537	9	3498	3498	0.6275	0.6275
4	2717	2717	0.6618	0.6618	10	4837	4837	0.619	0.619
5	2719	2719	0.5873	0.5873	11	5082	5082	0.6783	0.6783

Table1. Natural Frequency and Damping Ratio of Aluminium Material of Disc Brake Plate from EMA.

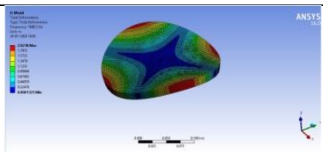
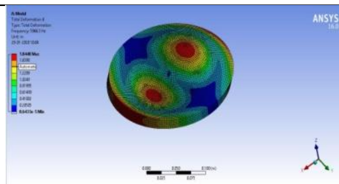
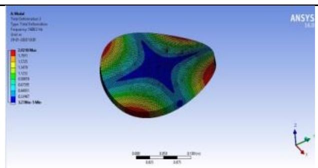
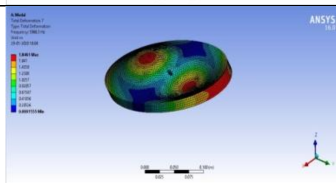
Mode No.	Frequency (Hz)	Mode No.	Frequency (Hz)
01	1608.3	06	5966.3
02	1608.3	07	5966.3
03	2713	08	6245.9
04	3658.2	09	6245.9
05	3658.2	10	9288.5

Table2. Natural Frequencies For First 10 Modes Of Aluminium Materials Of Disc Brake Plate From FE Method.

Sr. no	Aluminium	
	Minimum	Maximum
01	1.325e-4	2.0218
02	3.2786e-5	2.0218
03	3.7147e-4	1.7984
04	3.5236e-5	2.2178
05	3.7316e-5	2.2178
06	8.6433e-5	1.8448
07	1.555e-4	1.8461
08	2.8211e-6	2.3706
09	5.5962e-6	2.3697
10	8.8121e-8	2.4967

Table3. Minimum and Maximum Deformation of disc brake plate for First 10 Modes in m

A disc was tested with the aid of experimental modal analysis. As a consequence, the natural frequencies and mode shapes obtained are really much nearer to the aluminium material.

Mode No.	Natural Frequency (Hz)		Mode Shapes	Mode No.	Natural Frequency (Hz)		Mode Shapes
	EXPT	FEM			EXPT	FEM	
1	1497	1608.3		6	4837	5966.3	
2	1497	1608.3		7	4837	5966.3	

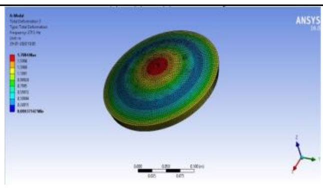
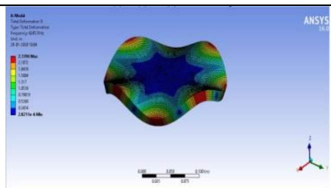
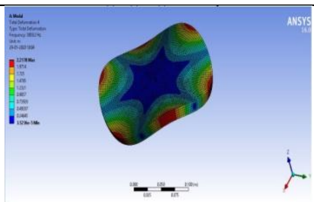
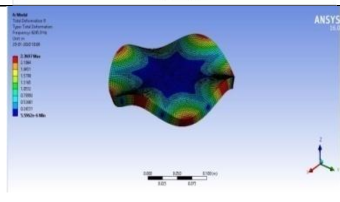
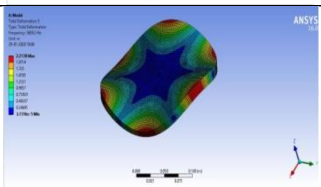
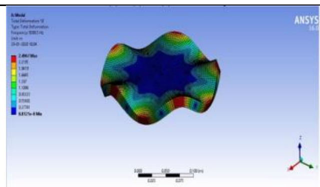
3	2717	2713		8	5082	6245.9	
4	3440	3658.2		9	5082	6245.9	
5	3440	3658.2		10	-	9288.5	

Table 4. Comparison of EMA and FEM Results for a Disc brake plate material Aluminium

The above figure is a display of the mode shapes of a pair of repeated roots of the circular disc brake. In above figures we have notice that the two modes have the same frequency, but their node lines show that their mode shapes are different. The mode shapes are similar looking but are "rotated" by 45^0 degrees from one another.

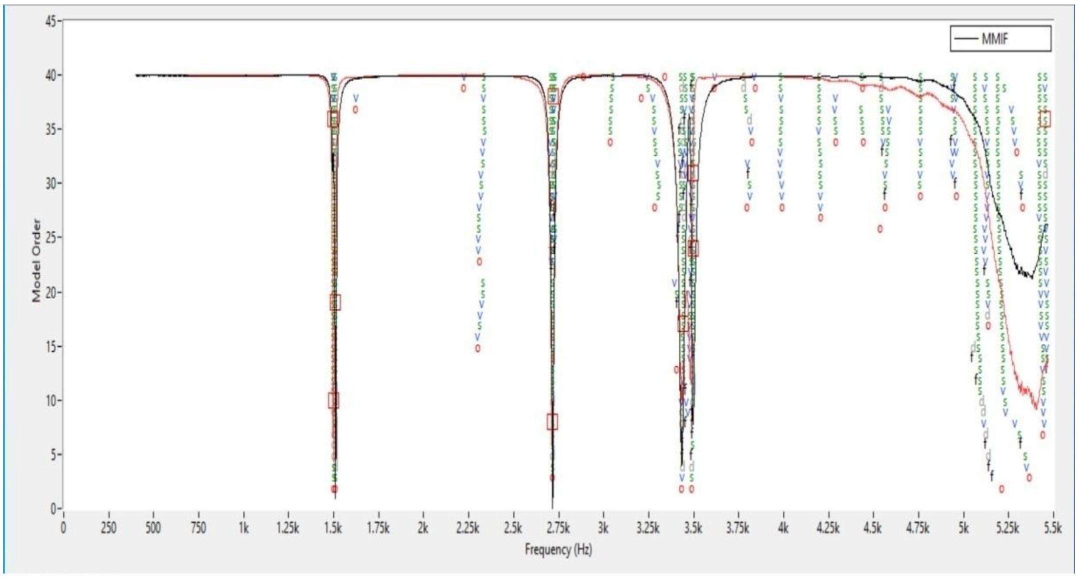


Fig3. Stabilization on disc brake plate

The characteristics of a system that describe its response to excitation as a purpose of frequency is the frequency Response function. Frequency of response for every excitation is measured as shown in fig 3. The phase of response in general case will be dissimilar than that of the excitation. The phase deviation between the response and the excitation will vary with frequency. Fig. 3 shows the resonant frequency for disc brake which is chosen based on phase separation at that frequency. The peaks of amplitude correspond to the natural frequency of every mode. In fact, they correspond to the natural frequencies of the principal modes of the disc brake that were excited, or participated in vibration of the disc brake due to impacts.

VI. CONCLUSION

The aim of the paper was to mention the possibilities of using modern methods of modal analysis by investigation of dynamic characteristics of mechanical systems. Using the multi-reference impact technique, we showed how to measure an augmented data set with multiple references and it is require multiple-input FRF estimation. Multiple reference impact testing more effectively treats local modes, structures with uni-directional modes, and reduces the chances of missed modes. Also the way of identification of coupled modes by using Multivariate Mode Indicator Function (MMIF) based on singular value decomposition of FRF matrix. The accuracy of modal parameters depends on the resolution of frequency spectrum and conditions of an experiment. The benefit of performing a MIMO test is that the number of tests and testing time also reduced. The increased number of actuators would ensure good global excitation of the entire structure, while the increased number on sensors would eliminate any unnecessary disturbance of the experimental setup. MIMO modal test was conducted that was very useful for mode separation, symmetric structures and FE model correlation.

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