Study of Experimental Modal Analysis on Two Wheeler Frame

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Abstract **- The frame is an important part in a Two Wheeler and it carries the load acting on the vehicle. So it must be strong enough to resist the shock, twist, vibration and other stresses. In vehicle frame different types of failure occur due to static and dynamic loading conditions. Structural properties of frame are an important design attribute. Natural frequency, damping and mode shapes are the inherent structural properties and can be found out by experimental modal analysis. Experimental Modal analysis (EMA) is the process of determining the modal parameters of a structure for all modes in the frequency range of interest. The objective of this study is to determine the natural frequencies, damping and mode shapes of the two wheeler frame by using experimental modal analysis**.

Keywords**- Experimental Modal analysis, Modal parameters, modes.**

I. INTRODUCTION

The frame is a skeleton upon which parts like gearbox and engine are mounted. So it is very important that the frame should not buckle on uneven road surface. Also it should not be transmitted distortion to the body. Two wheeler frames can be made of steel, aluminium or an alloy. Mostly the frame is consisting of hollow tube. If the natural frequency of two wheeler frame is coincides with excitation frequency then the resonance will occur. Due to resonance the frame will undergo dangerously large oscillation, which may lead excessive deflection and failure. To solve these problems, experimental modal analysis is very essential. Natural frequency, damping and mode shapes are the inherent structural properties and can be found out by experimental modal analysis. Experimental Modal analysis (EMA) is the process of determining the modal parameters of a structure for all modes in the frequency range of interest.

The main purpose of this paper is to find out natural frequency, damping and mode shape of two wheeler frame using experimental modal analysis.

II. MODAL THEORY

Modal Analysis Theory refers to that portion of classical vibrations that explains, theoretically, the existence of natural frequencies, damping factors, and mode shapes for linear systems.

The basic equations and their various forms will be presented conceptually to give insight into the relationships between the dynamic characteristics of the structure and the corresponding frequency response function measurements. Although practical systems are multiple degrees of freedom (MDOF) and have some degree of nonlinearity, they can generally be represented as a superposition of single degree of freedom (SDOF) linear models and will be developed in this manner.

The Frequency Response Function (FRF) is a fundamental measurement that isolates the inherent dynamic properties of a mechanical structure. Experimental modal parameters (frequency, damping, and mode shape) are also obtained from a set of FRF measurements.

The FRF describes the input-output relationship between two points on a structure as a function of frequency, as shown in Fig. 1. Since both force and motion are vector quantities, they have directions associated with them. Therefore, an FRF is actually defined between a single input DOF (point & direction), and a single output DOF.

An FRF is a measure of how much displacement, velocity, or acceleration response a structure has at an output DOF, per unit of excitation force at an input DOF. FRF is defined as the ratio of the Fourier transform of an output response (X(w)) divided by the Fourier transform of the input force (F(w)) that caused the output.

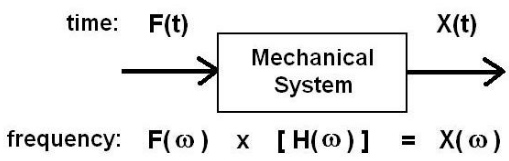

Fig.1 .Block diagram of FRF.

Depending on whether the response motion is measured as displacement, velocity or acceleration, the FRF and its inverse can have a variety of names.

TABLE I. DIFFERENT FORMS OF FREQUENCY RESPONSE FUCTION

Definition	Variable
Compliance	Displacement / Force
Mobility	Velocity / Force
Accelerance	Acceleration / Force
Dynamic Stiffness	Force / Displacement
Impedence	Force / Velocity
Dynamic mass	Force / Acceleration

With the ability to compute FRF measurements in an FFT analyzer, impact testing was developed during the late 1970's,

ISSN: 2278 – 7798

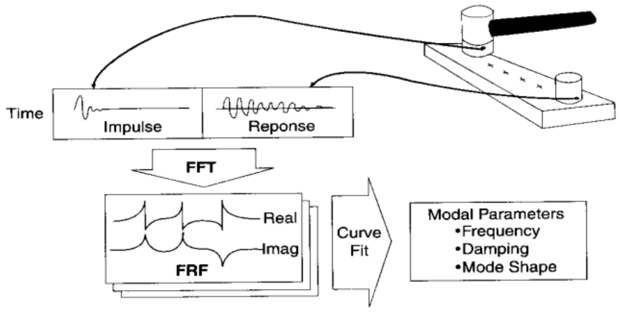

Fig.2. Impact testing

and has become the most popular modal testing method used today. Impact testing is a fast, convenient, and low cost way of finding the modes of machines and structures.

Impact testing is depicted in Figure 2. Equipments required to perform an impact test are an impact hammer with a load cell attached to its head to measure the input force and an accelerometer to measure the response acceleration at a fixed point & direction. A 2 or 4 channel FFT analyzer is also required to compute FFT. High speed data acquisition system acquires the impact and vibration data. Post-processing modal software is required for identifying modal parameters and displaying the mode shapes in animation. With the help of above instruments modal analysis for any structure can be performed. A wide variety of structures and machines can be impact tested. Of course, different sized hammers are required to provide the appropriate impact force, depending on the size of the structure; small hammers for small structures, large hammers for large structures are selected.

III. TEST SET UP

Component whose natural frequencies, damping values and mode shapes are to be obtained must be in free boundary condition. To achieve free- free condition bungee cord or foam is used in testing labs.

Vehicle frame is hanged by bungee cord to achieve a free-free condition. Excitation method used for EMA of frame is Impact Hammer excitation. Plastic tip is used for impact hammer. Plastic tip will excite the lower frequency band with higher amplitude and good flat response. Transducer used for response is accelerometer. Accelerometer is adhesive mounted on the frame. LMS Test lab software is used for modal data acquisition and analysis.

IV. EMA TEST PROCEDURE

Mode shapes needs to be represented by the schematic model. Geometry is created in the post processing software. Number of nodes in the geometry should be exactly equal to the nodes marked on the actual frame. Number of node in the geometry and respective node on frame should be numbered same.

After completion of geometry, each node on the frame must be excited by the impact hammer. Response location for accelerometer needs to be selected on the basis of adequate response amplitude at all nodes. Defining the direction of impact and the direction of response, measurements at all nodes should be completed.

All FRF set is selected for forming a modal matrix. Synthesized curve showing the response for whole frame is calculated by the post processing software. MDOF Curve fitter needs to be selected for the modal parameter estimation and to obtain optimum number of modes.

Frequency bandwidth selected for two wheeler frame is up to 200 Hz. Vehicle frame will be subjected to operational frequencies within 200 Hz. Engine operational frequencies will be up to 120 Hz max. Due to these reasons, frequency bandwidth is selected till 200 Hz.

V. CONCLUSION AND DISCUSSION

Natural frequency and damping value are obtained by experimental modal analysis. First bending mode observed at 119 Hz frequency with 0.12% Damping. As shown in figure 3 Critical areas at 119 Hz are tail end of frame, Front tube joint locations.

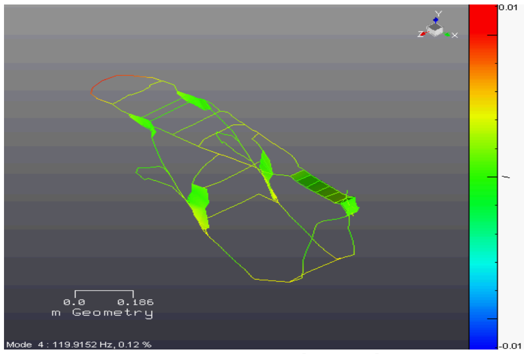

Fig. 3 First Bending Mode
(Frequency: 119 Hz,Damping: 0.12%)

First twisting mode observed at 176 Hz frequency with 0.08% damping. Figure 4 shows the first twisting mode for two wheeler frame. Critical Locations at 176 Hz are Pillion Footrest Holder, and the front Tube.

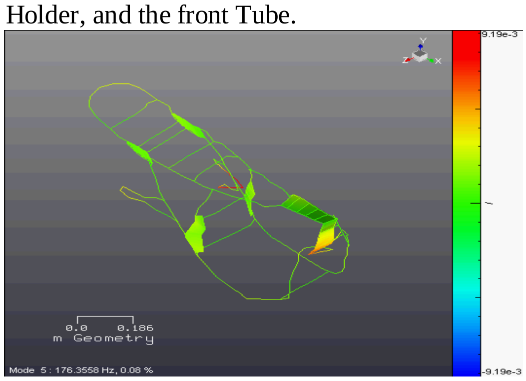

Fig.4. First Twisting Mode
(Frequency: 176 Hz, Damping: 0.08%)

Second bending mode observed at 180 Hz frequency with 0.15% damping. Figure 5 shows the second bending mode at 180 Hz. Critical locations are frame tail end and pillion footrest holder.

ISSN: 2278 – 7798

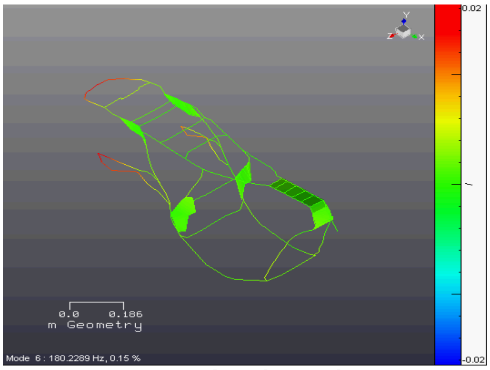

Fig.5 second Bending Mode
(Frequency: 180 Hz, Damping: 0.15%)

Combine bending and twisting mode is observed at 191 Hz frequency with 0.20% damping. Figure 6 shows the combined bending and twisting mode for frame. Critical areas for this mode are rear links of frame, front tube of frame & Swing arm support location.

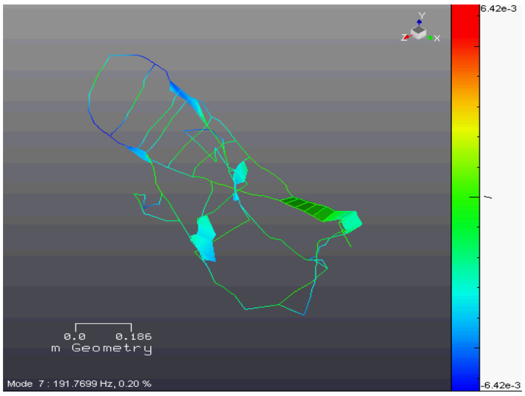

Fig.6. Combine Bending and Twisting Mode
(Frequency: 191 Hz, Damping: 0.20%)

Natural frequencies and damping values are shown in table below At 7000 rpm of Engine, first resonant frequency is likely to resonate which will amplify the vibration at rider interface. Natural frequencies and damping values obtained by EMA helps in various Vibration and Noise control tasks.

TABLE II. NATURAL FREQUENCYS AND DAMPING VALUE

Mode	Frequency	Damping %
1	119.915 Hz	0.12%
2	176.356 Hz	0.08%
3	180.229 Hz	0.15%
4	191.77 Hz	0.20%

VI. CONCLUSION

This paper presents the overview of experimental modal analysis. It also determines the natural frequencies, damping and mode shapes of the two wheeler frame by using experimental modal analysis highlighting the critical areas at resonant frequencies.

ACKNOLEGEMENT

It gives me a great sense of pleasure and satisfaction to present this paper on "Study of Experimental Modal Analysis on Two Wheeler Frame". Most especially to my family, friends and God, who made all things possible. I would like to also thanks to Dr. S. H. Sarje for his valuable contribution.

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ISSN: 2278 – 7798