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Modal Analysis of Go - Kart's Chassis based on FEM and FFT Analyzer

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***Abstract* -Vibrations are those mechanical oscillations which originate about an equilibrium position. Vibrations can be desirable as well as undesirable. In mechanical term, vibration analysis is one of the most important technique. It totally relies on the highly valued content provided by the machine vibration signals that indicates the condition of the machine, used for the analysis of faults. Vibration analysis are fundamentally used in predictive maintenance program being widely used for detection and monitoring severe faults in machinery parts, such as bearings, shafts, couplings, motors etc. Some problems that are generally detected by vibration analysis are: unbalancing, misalignment, bent shaft, rolling bearing faults, resonance, looseness, rotor rub, fluid-film bearing instabilities, belt/sheave problems. In case of automobiles, vibration analysis can be very effective to increase the performance of the vehicle. Proper validation of the design with the aspects of desirable and undesirable vibrations can lead to more optimized, safer design. In this project the same analysis is completed for the case of a Go-kart chassis i.e. a small race car. In relation to safety and the achievement of the best possible achievable ride comfort, it is very important to test and analyze the design under all vibrating conditions.**

1. INTRODUCTION

Vibrations are those mechanical oscillations which can originate about an equilibrium position. There are many cases when vibrations are advantageous, like in certain types of machine tools or production lines. Most of the time, the vibration of mechanical systems is not required as it wastes energy, reduces efficiency and may be harmful or even fatal. For example, passenger ride comfort in automobiles is hugely affected by the vibrations that are caused by outside disturbances, such as aero-elastic effects or rough road conditions. In other cases, eliminating vibrations may save many human lives. One of the best examples is the vibration control of architectural structures in an earthquake scenario.

Almost all machines vibrate due to one or more of these causes:
(a) Repeating forces
(b) Looseness
(c) Resonance

Machines tend to vibrate at specific oscillation rates. The oscillation rate at which a machine tends to vibrate is known as its natural rate of oscillation. The natural oscillation rate of a machine is the rate at which the machine 'prefers' to vibrate. A machine left to vibrate freely will prefer to vibrate at its own natural oscillation rate. Most machines have more than one natural oscillation rate. Like , a machine comprising of two substructures of different natural oscillation rates will exhibit minimum two natural oscillation rates.

A repeating force that causes resonance may be small and may arise from the motion of a good machine component. Such a mild repeating force would not be a problem until it results in resonance. Resonance should always be avoided as it causes rapid and severe damage. In automobiles, while engine is in running condition, repeating forces at particular speed can lead to resonance. To avoid increase in vibration amplitude and severe damage, knowledge of resonance frequencies for all vehicle components is of more important.

2. EXPERIMENTAL SET UP & MEASUREMENT

2.1 Experimental Set up

Component whose natural frequencies, damping factors and mode shapes have to be obtained must be in free-free boundary condition. To achieve free- free condition a bungee cord or foam is used in testing labs. Experimental set up for vehicle frame is as shown in figure 2.1.

Figure 2.1 Experimental Set-up for Vehicle Frame

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Vehicle frame is hanged by a bungee cord to achieve a free-free condition. Excitation method used for FFT of frame is Impact Hammer excitation. The Plastic tip is used for impact hammer. As discussed earlier, plastic tip will excite the lower frequency band with higher amplitude and a good flat response. Transducer used for response is accelerometer. Accelerometer is magnetic, mounted on the frame. Dewesoft X2 SP5 software is used for modal data acquisition and analysis.

2.2 Test Preparation

Before starting the actual measurement the vehicle frame must be clean from grease, oil, dirt & dust. The actual weight of the vehicle frame is measured which is 65 kg. Uni-axial accelerometer is mounted over the vehicle frame by magnetizing force.

2.3 Geometry Creation

Minimum distance to be maintained in each Node is calculated. Each node is marked over the mesh geometry. Each node is labeled for specifying node number and component name. Each node is connected with line and line diagram is completed for the frame which will represent the actual model in FFT Software. Nodes are marked over the Vehicle Frame with marker pen. Each node is labeled as marked on Vehicle Frame.

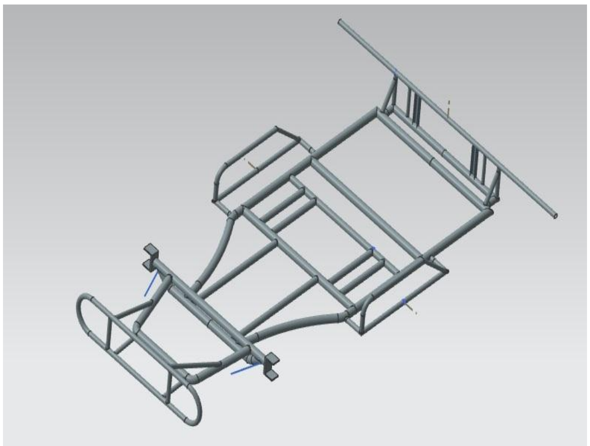

Figure 2.2 CAD Model of Vehicle Frame

2.4 Test Procedure

Impact Hammer is connected at channel 1 and accelerometers X Direction at channel Number 2. Data acquisition system is switched on and it is connected with LAN connection to PC. Geometry Created for Frame is opened in the Dewesoft X2 SP5 Software.

For Rowing Hammer method, Accelerometer Location should be fixed and each point needs to be excited with hammer. To find out the accelerometer mounting Location, the structure was excited at few different locations. Responses at these locations were checked & location where all the resonant frequencies are excited with maximum amplitude is selected for correct data acquisition. Accelerometer location should be selected in such a manner that excitation at any location will result in good response and nearly every frequency should be excited.

Measurement sequence for the test is prepared considering the accelerometer mounting location. The accelerometer was mounted using Magnetization at the selected location for mounting of accelerometer. This location must not be disturbed till all locations are excited and FRF runs are over.

As per the measurement sequence, respective nodes are excited with the specified direction and direction for responses is ensured as per the sequence. Average of five correct hits is considered for each node.

While taking the FRF measurements, Coherence window, Force Autopower, Response Autopower & FRF are checked for each hit. Coherence value should be ranging from 0.8 to 1.0. Coherence below 0.8 throughout the selected bandwidth must always be rejected. Force Autopower should be flat throughout the bandwidth. Wringing appeared in the force Autopower shows that there are multiple hit.

Incorrect hit or double hit should be rejected for the correctness of data. Once all nodes are excited, project is opened for post-processing. Matrix is formed of all runs in terms of Rows & Columns. Modal size is entered of 2^n for post-processing of the data. Bandwidth of 0-500 Hz frequency data is defined required for the analysis of data. Polymax curve fitter enables to select the stable modes over the selected bandwidth.

2.5 FEA Measurements

For FEA Analysis model is considered as a linear Isotropic material. Structural damping of vehicle frame is assumed as 2% for the calculation purpose. The analysis sequence for the Finite Element Analysis is as shown in Figure 2.3.

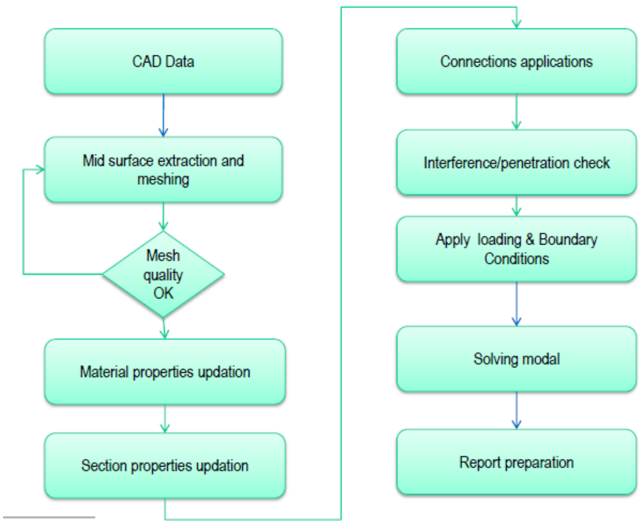

Figure 2.3 Finite Element Analysis Methodology

Material of Vehicle frame is Steel (ASTM grade B 106). Following properties as shown in table 2.1 are assumed for updating the CAD model.

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3. RESULT & VALIDATION

The natural frequency and the damping ratio of the stable poles are listed in Table 3.1.

Table 3.1 Natural Frequencies of FFT

Mode	Frequency
1	39.06 Hz
2	39.06 Hz
3	48.83 Hz
4	29.30 Hz
5	29.30 Hz
6	39.06Hz
7	39.06 Hz
8	29.30 Hz
9	39.06 Hz
10	39.06Hz
11	9.77 Hz
12	29.30 Hz
13	39.06 Hz
14	39.06 Hz

As shown in table 3.1, first natural frequency is at 39.06 Hz. This natural frequency is likely to occur at 5520 rpm of engine. If vehicle is running at this rpm then the operational frequency of vehicle will coincide with the natural frequency. At this condition vibration felt by the rider and the vibration at critical points will be more.

Damping at this frequency needs to be improved to avoid the excess vibration.
Operational frequency can match the resonant frequency so we need to control this frequency. At 39.06 Hz , first bending mode is observed.

Table 3.1 Material Properties Input for Frame

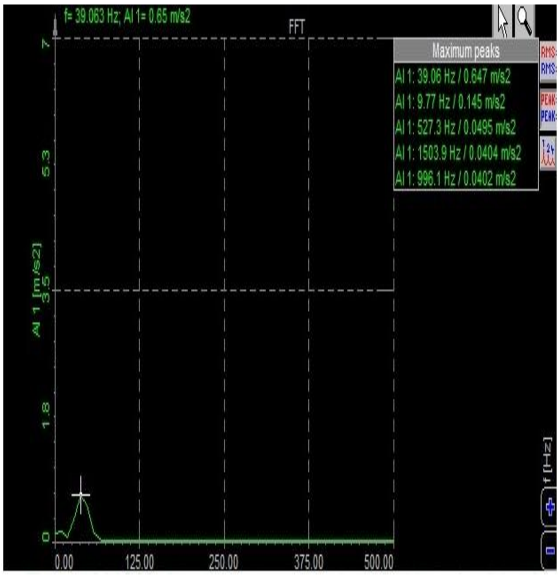

Figure 3.1 FFT Analyser Peak frequency at node 1

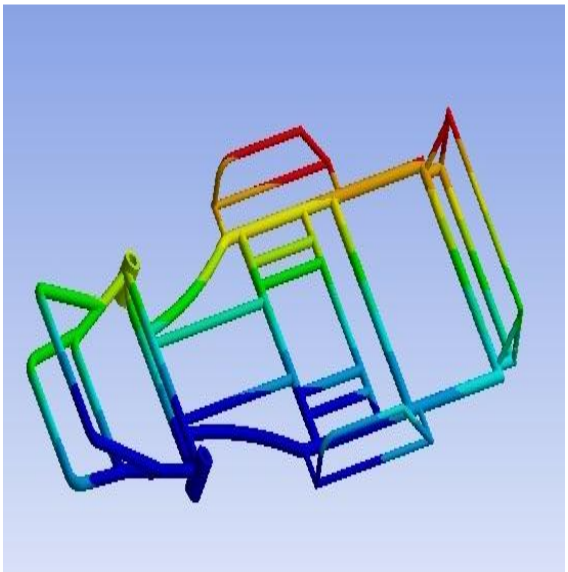

Figure 3.2 Ansys Picture of deformation at node 1

Similarly, the values in Table 3.1 have been obtained and compared with their respective ansys deformation at each nodes (14)

4. CONCLUSION

The failure of the component can be avoided by not allowing it to vibrate at resonance frequency. Modal analysis experimental (FFT analyzer) as well as analytical (FEM) helps to find out natural frequencies of the component and then these frequencies are compared with the operating frequencies.

If any of the natural frequencies coincides with the operating frequencies then the stiffness and mass of the component is varied, either by changing the material or design which helps to avoid the condition of resonance.

Components	Material	Properties			
		Young's Modulus (Mpa)	Poisson's Ratio	Density (kg/m^3)	Yield Strength (Mpa)
Frame tube	ASTM 106 grade B	2 e5	0.2	7840	420

And the design is considered safe.

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