

Design and Analysis of Vibration Test Fixture for 6.2 kW Siemens^[R] Motor

Srivatsa E⁰, Rahul Sanjay Gidaveer⁰, Sachin N⁰, Sujay Komal Patil⁰, Anil Kumar K⁰, Ekanthappa J⁰

U.G. Student, Dept. of Mechanical Engineering, Siddaganga Institute of Technology, Tumkur, Karnataka, India^[1]

U.G. Student, Dept. of Mechanical Engineering, Siddaganga Institute of Technology, Tumkur, Karnataka, India^[2]

U.G. Student, Dept. of Mechanical Engineering, Siddaganga Institute of Technology, Tumkur, Karnataka, India^[3]

U.G. Student, Dept. of Mechanical Engineering, Siddaganga Institute of Technology, Tumkur, Karnataka, India^[4]

Scientist – C, Additive and Special Manufacturing Process, CMTI, Bangalore, Karnataka, India^[5]

Assistant Professor, Dept. of Mechanical Engineering, Siddaganga Institute of Technology, Tumkur, Karnataka, India^[6]

ABSTRACT: Vibrations induced is one of the important parameters to be considered while designing an avionic equipment. The intense sound generated by the propulsion system engenders acoustic pressure which induces vibration. In avionic equipment additional vibrations are generated during engine ignitions, shutdowns and sudden shock generated by motors. The different modes of failure due to vibration in an avionic equipment is to be analysed. Prior to the mounting of the equipment in the aircraft it needs to be tested with its ability to withstand high frequency vibrations occurring during operation of the aircraft so that equipment does not fail. These vibrations can be simulated during the modal testing but the test component cannot be directly subjected to the vibrations, hence it needs an interface to hold it rigidly, restrict orientation and faithfully transfer the vibrations. This can be achieved by a fixture. Vibration test fixtures are generally designed to support electronic components that are mounted in missiles, fighter aircrafts. The fixture is designed by considering various parameters like mounting orientations, fixating design allowances and weight of the Device-Under-Test as well as the stress concentration factor and factor of safety while primarily maintaining the natural frequency of the fixture above the operation frequency. Intension of the work is to fabricate L-type fixture for the aircraft component 6.2 kW Siemens^[R] Motor. Suitable material is selected considering various criteria's such as cost, stiffness and strength. Structural steel sufficed the requirements and hence forms the fixture material. The design of vibration test fixture is done in Solid works and is analysed virtually using finite element analysis in Ansys Workbench thereafter experimentally through Impact Hammer Modal Testing. In this Modal testing the fabricated fixture is subjected to modes of vibrations to compute natural frequency experimentally and then is compared with analytical values.

KEYWORDS: Vibrations, 6.2 kW Siemens^[R] Motor, Test Fixture, Natural Frequency, Solid Works, ANSYS Workbench, Impact Hammer Modal Test.

I. INTRODUCTION

Aircraft equipment undergo rigor environment during lift off and ascent phase of launch. During which an acoustic pressure is exerted on system due to sound generated by aircraft propulsion which leads to the system vibration. Failure due to vibration is most common and the components must be tested for high vibrations. The aircraft component for which the fixture is designed is 6.2 kW Siemens^[R] motor. Siemens^[R] motor is of the Squirrel cage asynchronous type. It is highly efficient in converting electrical energy into mechanical with minimal heat dissipation since it is embedded with forced ventilation for high effective cooling. In the aircrafts it is one of the Line Replacement Units (LRU), used to generate rotatory mechanical energy and transfer it to hydraulic pump through shaft coupling.

The aircraft component is exposed to the real operation conditions by the simulation generated by the integration of the devices like shaker table, closed loop control system, accelerometer, power amplifier and fixture. As the vibration test fixture is used as an interface to mount the test specimen over the shaker table, it is significant prerequisite to analyse the dynamic behaviour of the designed fixture for it to perform as intended. The response of the fixture to the vibration: Impulse in nature, applied at the surface of the fixture interfacing test specimen, is analysed both numerically and

experimentally. For numerical analysis, finite element method is used. For experimental analysis, impulse vibration is applied with the Impact hammer.

Modal analysis is performed with the help of either shaker tables or impact hammer test. Generally, for lower test frequency and lower weight specimens the impact hammer test is recommended because as the weight of the specimen increases the signal to noise ratio becomes poor. In “Impact Hammer Modal Test” the fixture is impacted with the hammer to create a force for infinitesimally small duration with constant amplitude in the provided frequency domain, the outcome of this is reflected in various modes of vibration with same amount of energy. The “Test Setup” to perform this experiment requires a “Structure” (Specimen whose modal analysis is to be performed), support (to hold the fixture), exciter (hammer), load cell, digital data analyser, controller (computer) and software to obtain the data in the form simulation, tables and graph.

The Siemens^[R] motor has its own operation frequency which is calculated as the desired frequency by converting the RPM of the motor to frequency. So, the natural frequency of the fixture must not lie above this desired frequency. Fixture should have mounting capabilities such that it can be affixed to the shaker table in different orientation so that vibration is transmitted from electrodynamic shaker to device under test (DUT) in three mutually perpendicular directions.

Generally, a good fixture has optimum mass, high stiffness, good strength, low damping and high transmissibility. The parameters need to be prioritized during fixture design are test environment conditions and exaggeration factor. The vibrations transmitted through the fixture should be such that all the mounting points undergo vibration which is in phase, uniaxial and having equal amplitude throughout the spectrum of frequency of interest. The dynamic characteristics of the test fixture to be considered while designing are mechanical impedance, transmissibility, cross axis response and fixture specimen resonance. Mechanical impedance is that force which is required to impart desired motion to the specimen. The DUT is exposed to random vibration excitation during launch and it is to be designed such that these random vibrations will not damage the structure. In order to achieve this the fixture must have comparable mechanical impedance to that of mounting structure. The centre of gravity of the fixture should be calculated precisely so that it aligns with the armature centre of electrodynamic shaker. This indirectly reduces the cross-axis stress during vibration.

The material to be selected for fixture must be as low as possible in mass without compromising the stiffness. The material selected must be easily machinable, strong and its dynamic characteristics must be such that over the long usage it should not fail. The material to be selected must have high specific stiffness i.e., ratio of young’s modulus to density.

The objectives of the work are: (1) To design the vibration test fixture whose natural frequency is higher than the maximum test frequency using Solid Works software (2) To minimize the weight of the test fixture without compromising its strength (3) To subject the vibration test fixture to modal analysis using ANSYS software (4) To verify the static stresses on the fixture are within the limits using ANSYS software (5) To produce manufacturing drawings of any one fixture which is designed and analysed showing optimum results.

II. DESIGN CONSIDERATIONS

The various parameters essential for the fixture design are listed below.

1. Specification
 - a. Frequencies
 - b. Axes of excitation
 - c. Condition of excitation
2. Specimen
 - a. Size, shape, weight, centre of gravity location
 - b. Mounting dimensions
 - c. Lowest natural frequency
 - d. Mounting hardware, power and signal lines
3. Exciter
 - a. Force output, frequency range
 - b. Control capability

4. Fixture concept
 - a. Design for use along three mutually perpendicular axes
 - b. Rotating fixture from one axis to another without dismounting the specimen
 - c. Keeping the load lines as straight as possible
 - d. Use of the fixture for more than one type of test
5. Design review
 - a. Fixture layout should be compatible with the latest configuration
 - b. Verification of manufacturing methods
6. Evaluation
 - a. Modal analysis
 - b. Determination of the lowest natural frequency using ANSYS
 - c. Experimental determination of the natural frequency using the accelerometers
 - d. Comparison of experimental data and theoretical data

Specific Siemens^[R] Motor design data required for fixture design such as the dimensions and design features are listed below:

1. Body:
 - a. Weight: 63 kg
 - b. Length: 579.50 mm
 - c. Width: 196 mm
2. Natural frequency: 85.9 Hz
 - a. Desired frequency: $1.3 \times 85.9 = 111.67 \sim 112$ Hz – Equation 1 (Considering a factor of safety of 1.3)
3. Centre of gravity: X = 152.48, Y = -0.37, Z = 15.01 (in mm)
4. Shaft:
 - a. Diameter: 38 mm
 - b. Length overhanging out of the body of the motor: 483.50 mm
5. Face:
 - a. Size of plate: 196×196 mm²
 - b. No of bolt holes: 4
 - c. Diameter of the bolt hole: 14 mm
 - d. Extrusion ring diameter: 48 mm
 - e. Extrusion ring width: 5 mm
6. Base:
 - a. No of bolt holes: 4
 - b. Diameter of bolt: 12 mm

The materials generally considered for fixtures are steel, aluminium and magnesium. Comparing the specific stiffness of different materials structural steel is selected as fixture material. Since the environmental frequency is very low i.e., less than 200 Hz structural steel is recommended for such low frequency. It is strong and stiff as compared to other materials. Even though the weight of the fixture increases cut outs and pockets reduces the weight to an extent. It is cheaper and also easily available in various standard sizes. It can be machined easily to the required shape.

III. DESIGN, ANALYSIS AND FABRICATION

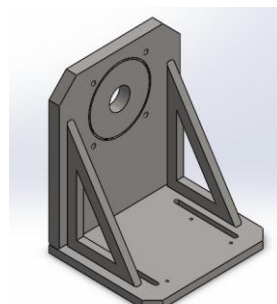


Figure 1: Fixture for 6.2 kW Siemens^[R] Motor

L – Type fixture for the Siemens^[R] motor is designed in SolidWorks as depicted alongside in the figure 1. This fixture is evolved over numerous iterations with the motive to reduce the weight, increase the stiffness so that the desired natural frequency can be attained. The design features are explained as follows:
The weight reduction is attained with the provision of design features such as chamfers at the edges of the slab and triangular pockets in the ribs.

- The stiffness is increased with the provision of ribs by aiming to extend enhanced support to the vertical slab and by increasing the slab thickness to 25 mm by aiming to sustain the weight of the motor.
- The parallel equidistant slots were provided at bottom plate in order to facilitate the fixing of the fixture on shaker Table. These slots are of the width of 12 mm in order to be fixed with M12 bolts. The motor will be fixed to the top plate with the help of the holes provided.
- The material for the fixture is selected as structural steel given to its commercial availability and cost effectiveness. With this material, the weight of the fixture was estimated to be 53.748 Kg using SolidWorks.
- In order to facilitate the mounting of the motor on the specimen, the design features such as: (1) Hole of diameter 60 mm is provided for bearing motor shaft (2) Recess cut of diameter 180 mm and width 4 mm (3) Bolt holes of 8mm diameter are provided to mount the motor on the vertical slab.

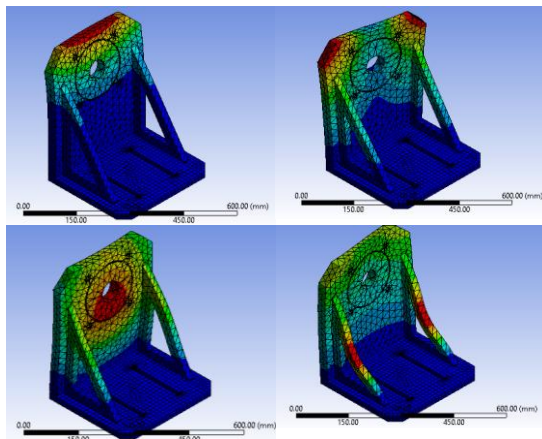


Figure 2: Modal Analysis of 6.2 kW Siemens^[R] Motor

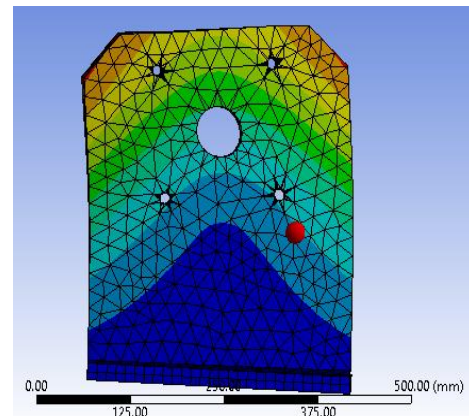


Figure 3: Torque Analysis of 6.2 kW Motor

This fixture was then later subjected to analysis, the modal analysis was performed preliminarily where the natural frequency was obtained to be 487.84 Hz using ANSYS, which is greater than 112 Hz (maximum operating frequency). Since the natural frequency was satisfactory the fixture was subjected to torque analysis. Figure 2 shows the modal analysis and Figure 3 shows the torque analysis. In modal analysis the second modal frequency was obtained to be 826.384 Hz, the third modal frequency was obtained to be 1167.7 Hz and fourth was found to be 1256.78 Hz. All the frequency and the differences between frequencies were satisfactory. In torque analysis the maximum deformation was found to be 0.00106 mm and maximum principal stress was found to be 5.74 MPa which is well within the limits.

To evaluate the effectiveness of FE analysis values, experimental testing must be carried out with the physical model. Testing the physical model will fetch the faithfulness of the model in transferring the applied vibration. The manufacturing drawing of Siemens^[R] model is created to carry out the testing. These drawings are depicted in the following figures (Figure 4):

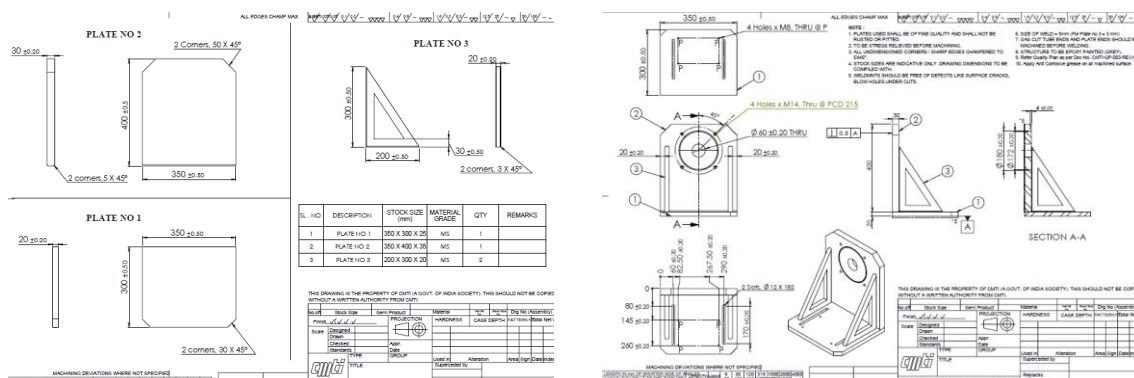


Figure 4: Manufacturing Drawings

The manufacturing drawings were kept in reference in order to fabricate the fixture. The fixture was so designed such that the slabs readily available in the market could be bought, machined and welded as per the drawings. The weldment tolerance was also provided in the manufacturing drawing. The slabs purchased are to be much thicker than the dimensions specified in the drawings in order to facilitate the machining tolerances, finishing and other sort of errors that incur during the fabrication. The concentric tolerances, chamfer tolerances were also taken care while designing and fabricating the fixture. Fabricated fixture is as shown in the below figures (Figure 5):



Figure 5: Fabricated 6.2kW Motor Fixture

IV. EXPERIMENTAL RESULTS

1. Impact Modal Hammer Testing



Figure 6: Experimental Setup

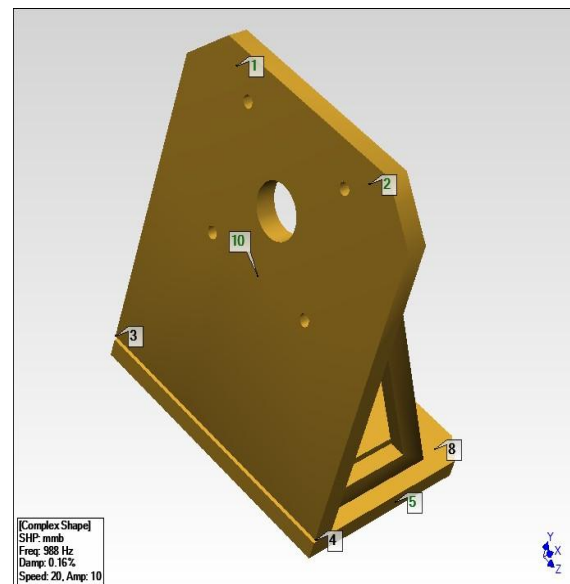


Figure 7: Position of the Accelerometer

The dynamic behaviour of the designed fixture is tested by the Impact hammer modal testing. Through this test various dynamic factors are obtained such as the Force Magnitude, Damping Ratio, Mode Shape and the Natural frequency is derived. Figure 5 shows the experimental setup in which the hammer impact is utilized to generate the impulse vibration. This vibration is picked up by the accelerometer positioned at various points as depicted in the figure 4.

2. Test Result and Implication

This data was exported to the software. The simulations were obtained at three modal frequency of fixture, which are 401 Hz, 556 Hz and 988 Hz for a given amplitude and speed. These results indicate that the natural frequency of the fixture is 401 Hz. These simulations and results also provide the damping factor in percentage. The below figures show the simulation at those frequencies:

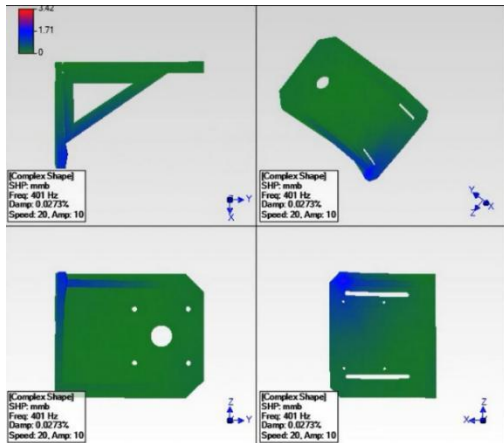


Figure 8: Simulations at 401 Hz

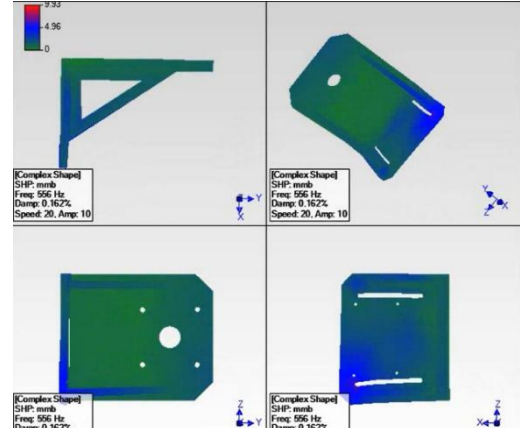


Figure 9: Simulations at 556 Hz

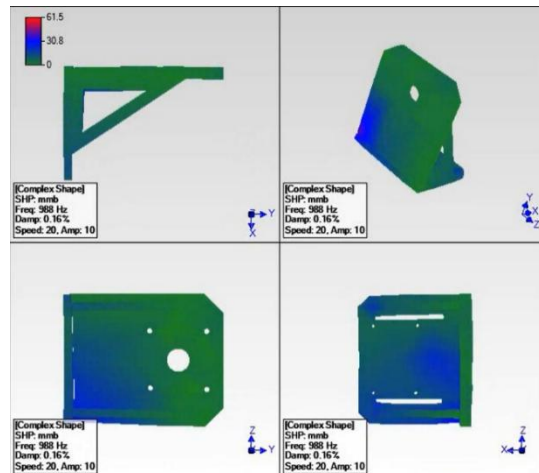


Figure 10: Simulations at 988 Hz

The experimental data for various modes were obtained. The main modes of concern were first mode and third mode. The magnitude (g/N) versus frequency (Hz) were obtained. With the help of these values we could calculate the natural frequency experimentally. The graphs of 1st modal analysis and 3rd modal analysis are shown in the below figure:

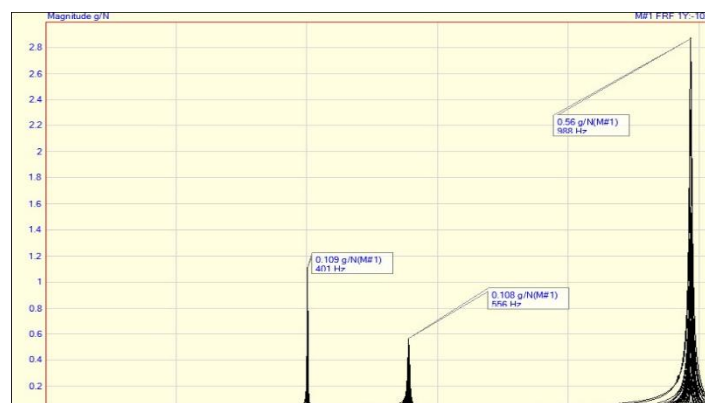


Figure 11: Magnitude vs Frequency for 1st Mode

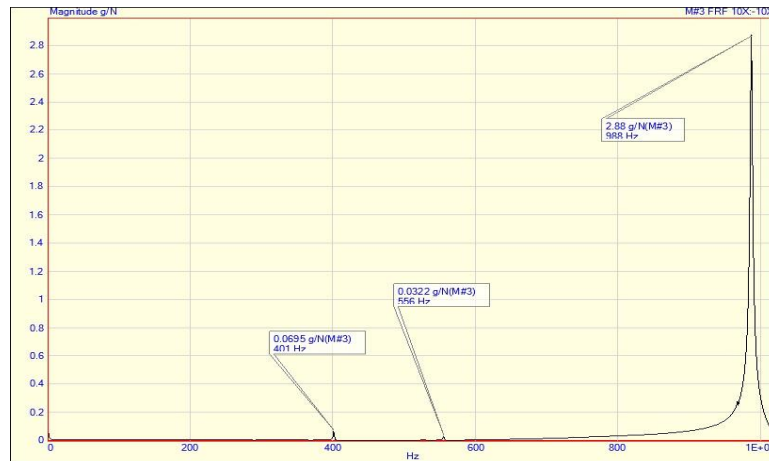


Figure 12: Magnitude vs Frequency for 3rd Mode

Along with these graphs the damping factors for those typical frequencies were obtained. These values of magnitude and damping factor help in finding the natural frequency. The below table depicts the damping factor in percentage obtained at those typical frequencies:

Frequency (Hz)	Damping Factor (%)
401	0.0273
556	0.162
988	0.16

Table1: Damping factor at typical frequencies

To calculate driving frequency for 1st mode of fixture we can take the values from the graph shown in Figure 11 where the 1st natural frequency for the fixture occurs at 401 Hz and the magnitude (H) is 0.109 g/N. From Table 1 we get the value of damping factor (ξ) to be 0.0273. Substituting these values in the below Equation (2) we will get the natural frequency for the first mode when motor is attached experimentally:

$$H = \frac{\frac{1}{m}}{\sqrt{\left[1 - \left(\frac{f}{f_n}\right)^2\right]^2 + \left(2\xi \frac{f}{f_n}\right)^2}} \text{ --- (2)}$$

Where, H – Magnitude, ξ – Damping factor, f – driving frequency, f_n – Natural frequency, m – Fixture mass

$$0.109 = \frac{1.88 \times 10^{-5}}{\sqrt{\left[1 - \left(\frac{f}{f_n}\right)^2\right]^2 + \left(2 \times 0.0273 \times \frac{f}{f_n}\right)^2}}$$

On simplification we get,

$$\left(\frac{f}{f_n}\right)^4 - 1.99 \left(\frac{f}{f_n}\right)^2 - 0.99 = 0$$

On solving quadratic equation and selecting one value we get,

$$\frac{f}{f_n} = 1, \text{ by substituting } f_n = 401 \text{ Hz (Obtained from Figure 11)}$$

we get,

$$f = 401 \text{ Hz}$$

The experimentally obtained value of 1st mode driving frequency is 401 Hz which is more than 112 Hz (Desired frequency from Equation 1). Since the fixture's driving and natural frequency both are higher than the environmental frequency resonance will not occur and faithful transmissibility of vibration takes place which is one of the main objectives of a vibration fixture.

V. CONCLUSION

The models are designed to have natural frequency greater than maximum operating frequency. The natural frequency of the fixture increases with increase in thickness of cross section. Increase in weight of the fixture increases the manufacturing cost, so management of cost and weight is essential to improve the performance of the fixture. Cut-outs and pockets reduce the weight of the fixture without compromising with strength and stiffness of the fixture. Ribs are attached to increase the stiffness of the fixture. The fixture is manufactured using structural steel material which are cheap and also readily available in standard sizes. The machinability of the structural steel is good so it can be machined easily to the required shape. Structural analysis in Ansys is done to get structural details like tensile strength, shear strength, maximum deformation and various stresses acting on the fixture. Modal analysis of the fixture is done to get natural frequency of the fixture. The natural frequency of the fixture is 487.84 Hz according to modal analysis done in Ansys and the natural frequency with motor load obtained from "Impact Hammer Modal Testing" is 401 Hz. Since the fixture's driving and natural frequency both are higher than the environmental frequency resonance will not occur and faithful transmissibility of vibration takes place which is one of the main objectives of a vibration fixture. The method followed and Impact Modal Test has given faithful output we would like to recommend this method and Test.

VI. ACKNOWLEDGEMENT

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