Vibration Analysis of Propellant Actuated Devices

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***Abstract**: Vibration qualification testing of propellant actuated devices or power cartridges used in fighter aircraft pose a challenge for the design and fabrication of the vibration test fixture for multi-direction. The vibration test fixture is made for CDBU type of cartridge among the various cartridges. Fixture for vibration testing is of cylindrical structure having maximum diameter of 235mm and of 36 mm thickness. The weight of the fixture is restricted up to the 5 kg.*

This structure has analyzed and experimentally evaluated. The design of fixture has its certain advantages like; it can accommodate 16 cartridges in multi-directions at a time and can be tested, it uses all the fixing points of the shaker table. Fixture is designed with Aluminum 6061-T6 Alloy adhering to the general principles of vibration fixture design avoiding resonance in the frequency range of test. Theoretical analysis for the natural frequency calculations is based on advanced theory of plate bending. Computer Aided Design software Pro-E Wildfire 4.0 is used for modeling and drafting work of the fixture. Finite element software ANSYS is used for the modal and random response analysis. Experimental trial was conducted for the two axis for 6 hrs. Control and monitoring accelerometers are bonded at critical locations to get the best results.

Experimental investigations suggested good correlation between computed and experimentally determined values and plots.

Index Terms: *Power Cartridges, Vibration Testing, PAD's Test fixture, Governing equilibrium equations, FE modal Analysis, Experimental results.*

I. INTRODUCTION

Today's all the modern fighter aircraft has most essential integral part with it is power cartridges. Power cartridges are low cost explosive actuated devices. They are essential parts of any military aircraft. The importance of the power cartridges can be gauged from the fact that no military aircraft, otherwise flight worthy, is allowed to take off without installing the essential power cartridges. Some of the important functions of power cartridges are to operate auxiliary systems and sub-systems of aircraft, Effective release of weapon with high degree of reliability and for giving distress signaling.

The purpose of installing power cartridges is to save the precious life of pilot from the endangered aircraft in emergency within shortest possible time. Such cartridges are also known as Propellant Actuated Devices (PAD). PAD's are used to give certain specified performance in actual system or subsystem where they are installed in the aircraft. The most important and valuable inventory of air force is its combat aircraft. The time, efforts and cost to groom a pilot to the required efficiency are enormous. Their life is extremely precious for the services and is required to be protected against all conditions and adverse circumstances. All military aircrafts are fitted with ejection seat for escape of the pilot from endangered aircraft in emergency. Various types of power cartridges are provided in the seat ejection system to perform different functions during the ejection process. In addition to the seat ejection, power cartridges are employed for other operational requirements also.

Power cartridges are categorized on the means of initiation are, mechanical types and electrical types.

Consumers expect and demand products of high quality and reliability. To fulfill these requirements the system after design should undergo various qualification tests. In these qualification tests vibration is one of the stringent test in which the product should undergo. However, vibration is very ramified topic. During this vibration tests, poor mechanical design will cause mechanical failure and customer dissatisfaction. This also causes increase in product cost and loss of credibility.

In order to prove the acceptability of the indigenous cartridges the various tests like Impact Test, Sealing Test, Vibration Test, Salt Mist Test, Life Assessment Test etc are carried out as per JSG 0102 Standard. Out of these Vibration Qualification Test is considered here.

Vibration tests are performed for the following reasons, Develop materiel to function in and withstand the vibration exposures of a life cycle including synergistic effects of other environmental factors, materiel duty cycle, and maintenance.

This method is limited to consideration of one mechanical degree-of-freedom at a time.

Verify that materiel will function in and withstand the vibration exposures of a life cycle.

Vibration results in dynamic deflections of and within materiel. These dynamic deflections and associated velocities and accelerations may cause or contribute to structural fatigue and mechanical wear of structures, assemblies, and parts. In addition, dynamic deflections may result in impacting of elements and/or disruption of function.

This paper briefly describes the fixture design methodology, facility augmentation to the testing, analysis carried out and evaluation methods adopted for the fixture experimentally.

II. FIXTURE DESIGN METHODOLOGY

An ideal fixture is a structure having infinite stiffness at all frequencies with zero mass. Since this is not achievable in practice, it is always a trade-off between the stiffness and mass relationships [13]. Hence, the fixture has to weigh less and at the same time not modify the dynamic characteristics of the UUT. It is also recommended that the designer understands the manufacturing process and the time and cost involved in producing a fixture.

A vibration fixture is a device that typically interfaces the vibration shaker / shock machine and the test item. The basic idea with any fixture is that it should convey all possible forces produced by the shaker to the test object. Generally it is not possible to fix a test object directly to the shaker table itself. For that fixture acts as a transition piece between the two.

The objective of making fixture is, to take vibration test of Power Cartridges by fixing it on the fixture. The purpose of vibration test is to check proper functioning of Cartridges under vibrant conditions.

2.1 Basic Fixture Design:

The knowledge of the dynamic behavior of structures is often of primary importance in many applications, particularly in the field of aerospace and mechanical design. Prediction of modal parameters such as resonance frequencies and mode-shapes is an essential step of design. For this purpose, numerical techniques based on finite element model are commonly used.

Unfortunately, uncertainties on mechanical properties, tolerances in fabrication and assembly processes may cause discrepancies between numerical predictions and experimental results. To ensure that the mechanical component will survive the dynamic environment in which it is operating; vibration qualification testing is required according to international standards or manufacturer specifications.

Understanding of the dynamic behavior of the power cartridges is important during the operating condition. Therefore, it is necessary to carry-out vibration qualification testing to international standards (or as per user's requirements) to ensure that the structures survive the service or operating environment. Vibration testing generally requires a fixture which is an intermediary structure to interface with the specimen (often called DUT, Device Under Test / UUT,Unit under test) and the vibration generating equipment (vibration shakers). A fixture is designed to take care of different shapes and sizes of the specimens and the attachment points. The design methodology proposed is based on topological optimization tools.

An ideal fixture is a structure having infinite stiffness at all frequencies with zero mass. Since this is not achievable in practice, it is always a trade-off between the stiffness and mass relationships.

Also the acceleration level possible with the shaker is inversely proportional to the total moving mass it has to drive. Hence, the fixture has to weigh less and at the same time not modify the dynamic characteristics of the UUT. A bad fixture results in isolation at attachment points and a good fixture transmits the input with fidelity. A good fixture has resonances above the frequency range of interest. Location of center of gravity of the shaker, slip table with the UUT has to be as close as possible to mitigate overturning moment concerns.

Rigs and Fixtures are used to primarily to connect a test specimen to a shaker table. The specimen is often referred to as the Unit Under Test (UUT). As the UUT rarely couples directly to the shaker, an interface is required to convert the hole pattern of the attachment points to the hole pattern of the shaker inserts. The main objective of the Fixture is to couple the UUT as directly as possible without adding or subtracting energy from the applied test.

The vibration fixture will require a design that when loaded with the specimen and connected to the shaker, will have no resonances within the specified frequency range.

This is rarely achievable but our design can assist in keeping the fixture resonances low in Q so that our controller can apply some compensation during testing.

Materials generally considered for vibration fixtures like Stainless Steel, Aluminum, Magnesium have similar E/ρ ratio (ratio of Young's modulus to density) not affecting the natural frequency of the fixture. However, when the shakers are operating at their full performance level, like in the present case, weight of the fixture dominates the selection of the material. Composite materials though are ideal for fixture to test large and heavy specimen, fabrication of the same is highly difficult.

Though Magnesium is a lighter metal, fabricability issues and availability of indigenous fabrication techniques have compelled to choose Aluminum alloy as the candidate material.

To design a test fixture that can be used for many different test pieces is extremely difficult and the usual practice is to treat each UUT separately. An interface plate can be designed to support each of these test items and is therefore required to be of a high standard. Individual test fixtures may not require the same build standard as the interface plate and this can affect the choice of material. When choosing the material it needs to be consider the test frequency range, the overall mass and the cost.

A low level vibration test that requires a frequency range from 20 Hz to 100 Hz may find that a wooden or plastic fixture is adequate. Whereas high level high frequencies test (typically above 500Hz) will require a much stiffer material. Often the fixture has a mass many times greater then the UUT and this will impact upon the Force capability of the shaker. This too will influence the material used as certain materials have a very good stiffness-to-mass ratio such as duralumin, magnesium or aluminum. Another characteristic of a material is its damping properties, e.g. the hysteretic damping of aluminum is approximately four times greater than that for steel. It is advisable to avoid steel for the above reasons i.e. it is heavy, it rings, it is difficult to work with, the resonances are of high Q and it is expensive.

Considerations for the fixture material are,

Mass - The material should be as low in mass as possible without detriment to the stiffness. As F = M x A the material mass will have an influence on the shakers thrust ability. Often a fixture will be sculptured and material removed to reduce the mass. The dynamics of the fixture remain unchanged but the stiffness and strength may be compromised and should be considered.

Stiffness - The stiffer the fixture the higher any resonant frequencies. If the first resonant can be kept above the test frequency range the quality of test will be improved. The fixture must be stiff enough not to influence the test or change during a test.

Strength - The Fixture will see high stress levels from the applied vibration or the UUT response and must be strong enough to transmit the forces and survive the tests.

Natural Frequencies- The natural or first resonant frequency should be measured during a rig assessment and will preferably be higher then the test requirements upper frequency.

Damping -. For fixtures with high 'Q' resonances, the control method will require some investigation. If possible it is preferable to use damping at the fixtures points of influence to reduce the 'Q'. For example a hollow tube will be damped by filling it with foam and often clamping or additional supports will prevent a high 'Q' response.

Attachment - The fixture should hold the UUT in multi-direction at a time. The UUT is held in a manner in which it is used in service and the brackets, isolation sup ports or other in service fittings should be attached to the fixture to represent an in service configuration. The UUT is rigidly held in a manner that ensures a known test specification is applied with out loss or compromise from the fixture.

Dynamic response - The UUT has its own response to a vibration test specification. The fixture should not modify the natural dynamics of the UUT or the test will be compromised. A dynamically inert fixture is not possible but its influence can be kept to a minimum by following the step by step guidelines above.

Fatigue - A fixture needs to be significantly stronger then the UUT so that it does not fatigue during use. Avoid thin brackets, small bolts, sharp corners, overhanging areas and weak areas that will fatigue quickly.

This system is designed to have high stiffness at right angles to the vibration axis. If high overturning moments are applied to the armature then the suspension guide will be compromised. This cross axis force applied during vibration needs to be understood and kept low so as not to damage the bearings and armature.

Designing all fixtures to be light and stiff will assist in preventing unwanted cross axial motion. The Centre of Gravity (C of G) should be precisely calculated and each rig or fixture should be rigidly coupled to the armature or slip table. If the C of G of the payload is kept low and aligns with the armature centre then cross axial stress will be minimized. However if the payload C of G is high or offset, then a turning moment is introduced during vibration. he easiest and cheapest fixture for this size of shaker is that mad of welded aluminum plate.

The mating surfaces of the fixture is machined as flat as possible.

Attachment of bolts from fixture to shaker table is of counter- sunk type so that the stressed length of the bolt is twice the diameter.

The objective is to design and evaluate the vibration fixture to qualify the power cartridge to the specifications generated to simulate flight vibration conditions. Prediction of modal parameters such as resonance frequencies and mode-shapes is an essential step of design. A fixture is designed to take care of different shapes and sizes of the specimens and the attachment points. Also the acceleration level possible with the shaker is inversely proportional to the total moving mass it has to drive. Location of center of gravity of the shaker, slip table with the UUT has to be as close as possible to mitigate overturning moment concerns. As the UUT rarely couples directly to the shaker, an interface is required to convert the hole pattern of the attachment points to the hole pattern of the shaker inserts. .

The fixture is well designed and not too large. Overhanging the shaker table is avoided. Its first resonant mode occurs well above the highest test frequency. Consider the mounting method for each axis of motion.

There are many fixture types that can be used to support the UUT. The main considerations that influence a design are dictated by the UUT itself. These are the physical shape, the mass, the size, the test requirement frequency range and severity, the number of axis to be tested, the attachment points, the quantity of units per test, the centre of gravity, off load moments etc.

With the target weight of < 6 kg, number of design configurations with different parameters, shapes are studied. Finally, the fixture is configured as a cylindrical structure and fabricated with AA-6061 T-6 alloy. By considering all these factors and according to the bolt size pattern holes on shaker table, 'M10 standard bolts' are used here to bolt a fixture with shaker table. While in the fixture assembly 'M4 bolts' are used here for bolting. The basic dimensions of the fixture are, maximum diameter of the fixture is ϕ 235 mm with the thickness of plate 36 mm. Centre of gravity of the fixture is maintained with the centre of gravity of shaker machine table. Fig. (1) gives the details of the fixture.

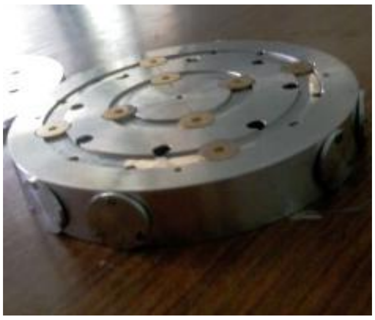


Fig (1). Details of fixture.

2.2 Governing equilibrium equations:

The theoretical work produces very practical benefits because it allows the prediction of the modal response of a structure. By finding and addressing potential problems early in the design process, manufacturers save money and improve product quality.

In the advanced theory of plate bending, theory of circular plate bending coincides with natural frequency calculation of our circular fixture (adapter plate). So equations are derived from the advanced theory of plate bending [8]. It is also observed that formulas mentioned [4] coincides with our derived formula for calculation of first natural frequency.

Derivations:

Equilibrium equations for small-displacement theory of flat plate,

$$\frac{\partial}{\partial x}\left\{\frac{1}{\alpha}\left[\frac{\partial}{\partial x}(\beta M_{xx})+\frac{\partial}{\partial y}(\alpha M_{xy})+(\alpha_y M_{xy})-(\beta_x M_{yy})+\alpha\beta R_y\right]\right\}$$
$$+\frac{\partial}{\partial y}\left\{\frac{1}{\beta}\left[\frac{\partial}{\partial x}(\beta M_{xx})+\frac{\partial}{\partial y}(\alpha M_{yy})-(\alpha_y M_{xx})+(\beta_x M_{xy})-\alpha\beta R_x\right]\right\}$$
$$+h\alpha\beta B_z+\alpha\beta P_z=0 \quad \text{.........(1)}$$

For rectangular axes, $\alpha = \beta = 1$ and equation (1) reduces to,

$$\frac{\partial^2 M_{xx}}{\partial x^2} + 2\frac{\partial^2 M_{xy}}{\partial x \partial y} + \frac{\partial^2 M_{yy}}{\partial y^2} + hB_z + P_z = 0 \quad \text{……(2)}$$

Where, M_{xx} , M_{yy} is bending moment and $M_{xy} = M_{yx}$ is twisting moment . α and β are constants. H = plate thickness.

Stress-Strain-Temperature relations for isotropic elastic plates:

Let σ_{xx}, σ_{yy} & σ_{zz} are plane stresses of plate. Whereas $\varepsilon_{xx}, \varepsilon_{yy}$ & ε_{zz} are strains of plate.

For linearly elastic isotropic materials and plane stress relative to (x, y) plane stress-strain-temperature relations are,

$$\sigma_{xx} = \frac{E}{1-\upsilon^2}\left(\varepsilon_{xx} + \upsilon\varepsilon_{yy}\right) - \frac{EkT}{1-\upsilon}$$

$$\sigma_{yy} = \frac{E}{1-\upsilon^2}\left(\upsilon\varepsilon_{xx} + \varepsilon_{yy}\right) - \frac{EkT}{1-\upsilon}$$

$$\sigma_{xy} = 2G\varepsilon_{xy} = G\gamma_{xy} \quad \text{…………(3)}$$

Where, E = Young's Modulus.
υ = Poission's Ratio.
k = Coefficient of Linear Thermal Expansion.
G = Shear Modulus.
T = Temperature Measured above an arbitrary zero.

By Kirchhoff's theory of Kinematics strain-displacement relations for plates,

Solve for equation (2) and then substitution of it into stress resultant equations of flat plate [4],

Then moment equations we get are,

$$\left.\begin{aligned} M_{xx} &= -D\left(\frac{k_{xx}}{\alpha^2} + \frac{\upsilon k_{yy}}{\beta^2} + T^1\right) \\ M_{yy} &= -D\left(\upsilon\frac{k_{xx}}{\alpha^2} + \frac{k_{yy}}{\beta^2} + T^1\right) \\ M_{xy} &= -D\frac{(1-\upsilon)}{\alpha\beta}k_{xy} \end{aligned}\right\} \text{…….(4)}$$

Where, $D = \dfrac{Eh^3}{12(1-\upsilon^2)}$. ………(5)

The quantity D = Flexural rigidity of the plate.

Considering strain energy and boundary conditions of the plate

On substitution for M_{xx}, M_{xy}, M_{yy} in terms of 'w' in equation (2) with $B_z = 0$ and $P_z = P$

$$\nabla^2\nabla^2 w = \frac{p}{D} \quad \text{………} \quad (6)$$

Where, $\nabla^2\nabla^2 w = w_{xxxx} + 2w_{xxyy} + w_{yyyy}$

$\nabla^2\nabla^2$ = Invariant vector operator.
p = Lateral Pressure.
w = Lateral Displacement.

Solutions of $\nabla^2\nabla^2 w = \dfrac{p}{D}$ for circular plates,

For the circular plates with radius 'a' and thickness 'h' employing polar coordinate with the origin 'a' the center of the plate, equation (6) may be written as,

$$\nabla^2\nabla^2 w = \left(\frac{\partial^2}{\partial r^2} + \frac{1}{r}\frac{\partial}{\partial r} + \frac{1}{r^2}\frac{\partial^2}{\partial\theta^2}\right)\left(\frac{\partial^2 w}{\partial r^2} + \frac{1}{r}\frac{\partial w}{\partial r} + \frac{1}{r^2}\frac{\partial^2 w}{\partial\theta^2}\right) = \frac{p}{D}$$

……………….(7)

Here as per our requirement considering only axial symmetric case, in which plate is loaded and supported symmetrically. Then equation (7) reduces to,

$$\left(\frac{\partial^2}{\partial r^2} + \frac{1}{r}\frac{\partial}{\partial r}\right)\left(\frac{\partial^2 w}{\partial r^2} + \frac{1}{r}\frac{\partial w}{\partial r}\right) = \frac{p}{D} \quad \text{…………} \quad (8)$$

Solution of above equation with p = p0 = constant is,

$$w = \frac{p_0 r^4}{64D} + A_1 + A_2 \ln r + B_1 r^2 + B_2 r^2 \ln r$$

or

$$\delta = \frac{p_0 r^4}{64D} + A_1 + A_2 \ln r + B_1 r^2 + B_2 r^2 \ln r$$

Where, δ = Deflection.
A1, A2, B1, B2 = Constants of integration

These constants of integration are determined by boundary conditions at r = a.

Therefore Circular Plate with Fixed Edges.

$$\delta(a) = A_1 + B_1 r^2 + \frac{p_0 r^4}{64D} = 0$$

After solution for A1 and B1

$$\delta = \frac{p_0 r^4}{64D}\left[1 - \left(\frac{r}{a}\right)^2\right]^2$$

Or

$$\delta = \frac{qa^4}{64D}\left[1 - \left(\frac{r}{a}\right)^2\right]^2 \quad \text{………(9)}$$

Where, q = Intensity of Load …...(N / mm^2)

Therefore as per the data of the fixture, the numerical value of the first natural frequency obtained after solution of equation (7) is **7925.85 Hz**.

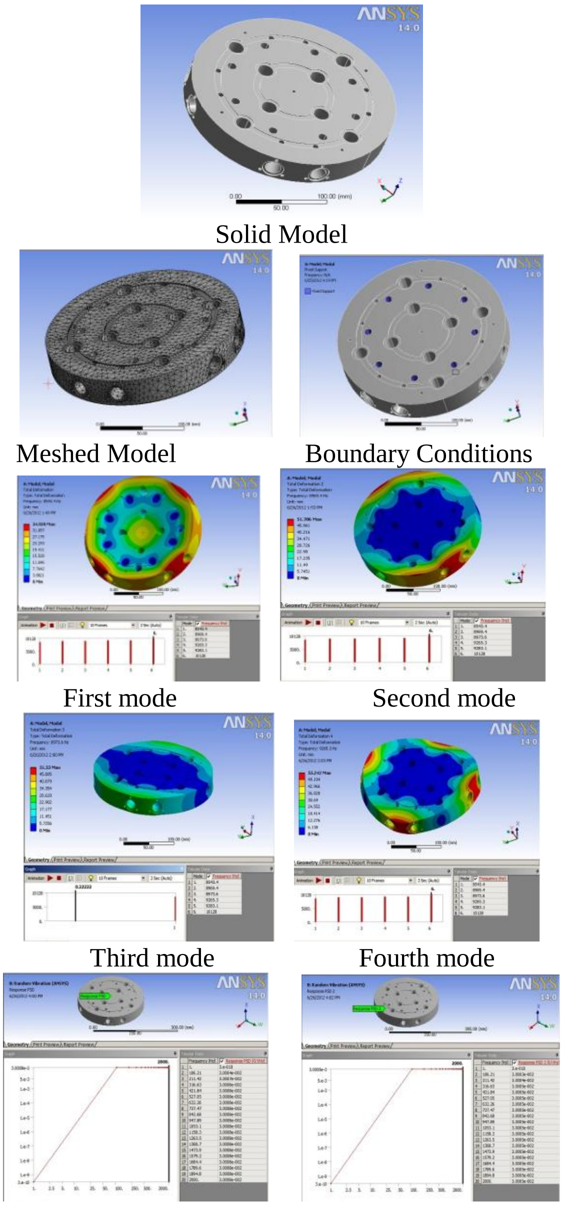
III. ANALYSIS

Modal analysis is a powerful tool for understanding the vibration characteristics of mechanical structures. It simplifies the vibration response of a complex structure by reducing the data to a set of modal parameters that can be analyzed with relative ease. This application note discusses the concept of modal analysis, applications where modal analysis is useful, and techniques for the acquisition and visualization of modal data. In this project modal analysis of the vibration fixture is to be performed and after experimental trials response analysis is to be shown.

Analysis of the fixture is carried out by geometric modeling in Pro/E, meshing in ANSYS and applying boundary conditions. Modal analysis is a process of describing a structure in terms of its natural characteristics which are the frequency, damping and mode shapes its dynamic properties. The modal testing has become an effective means for identifying, understanding, and simulating dynamic behavior and responses of structures.

Analysis yields a 1st natural frequency mode of **8543.4 Hz**. The random vibration analysis is carried out for the response with constant value of 0.03 g^2/Hz. The FE analysis by ANSYS indicates that there is no resonance observed in the required range of frequency. In random analysis controlled value of rms is equal to the demanded value of rms which shows uniformity in the structure. Theoretical estimated weight of the fixture is considered for the analysis. The fixture designed meets the mass and frequency requirements.

Fig (2) gives few representative plots of analysis.



Solid Model

Meshed Model Boundary Conditions

First mode Second mode

Third mode Fourth mode

Response at the centre Response at the edge

Fig (2).Representative plots of analysis

IV. Fixture Evaluation

Certain augmentation to the facility is carried out before evaluating the fixture to facilitate testing of the power cartridges.

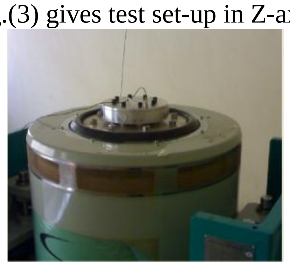
Subsequent to the design, analysis & fabrication the fixture is evaluated. Weight of the fixture after fabrication is 5 kg as compared to die theoretically predicted value of 3.7 kg. Main purpose of evaluating a fixture is to investigate the fixture natural frequencies, transmissibility, cross-axial response. Resonance search, Sine and Random vibration tests are carried out to evaluate the fixture. A low vibration level resonance search followed by sine, random search is carried out.

Sine and Random vibration tests to the test specification are carried out with two control accelerometers on the fixture top as shown in fig (2). Table (1) gives sequence of test conducted.

Table1. Sequence of test conducted.

Sine / Random	Specifications
Sine vibration test	5-11 Hz - 12mm DA. 11-2000 Hz – 3G Sweep rate – 1 Oct / Min.
Random vibration test	70-2000 Hz – 0.03 g2 / Hz

Fig.(3) gives test set-up in Z-axis.



Fig(3) Fixture Evaluation in Z-axis.

2.3 Discussions:

Resonance survey is conducted before and after the vibration tests. Lowest acceptance level based on facility limitation is employed.

The sine resonance survey conducted before and after tests from UUT in between frequency range of 5-2000 Hz. The plot of the sine vibration test is as shown in fig.(4) The machine was not tripped during test which indicates good design of fixture without any potential weakness. This indicates the uniformity in vibration level in the fixture and damping is good in structure.

Fig.(5) shows plotted result of the random vibration test of the fixture. Results shows control rms value is very close to the demanding rms value which indicates uniform structure of the fixture. There was no any peak or resonance observed within the range of vibration i.e. up to 2000 Hz. This indicates that fixture resonance is greater than UUT resonance.

The vibration fixture is ideally designed so that it gives unit transmissibility in the frequency range of test. Cross axial response is not observed within the frequency range is observed.

Control accelerometers provided on the top of the fixture with maximal control options yielded smooth control.

After correlation of the theoretical, simulated and experimental results observation shows that all the results are very close.

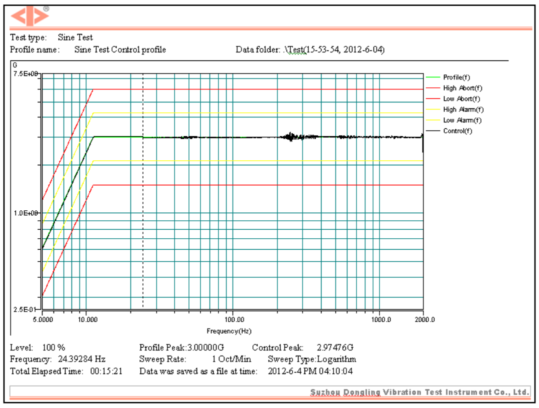


Fig (4) Sine vibration test

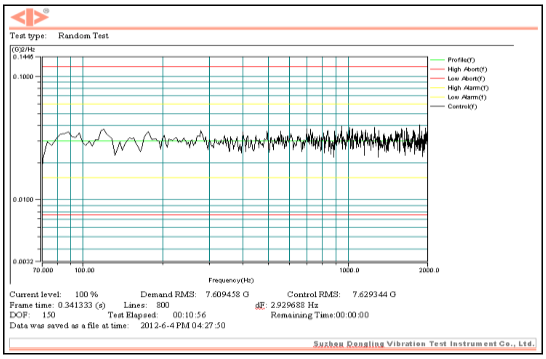


Fig (4) Random vibration test

V. CONCLUSION

An aluminum vibration fixture with AA6061-T6 alloy for power cartridges is designed, analyzed and evaluated. For vibration test of a single small test object, designing a vibration fixture is trivial task. However, it becomes complicated one while vibrating many of such test objects in a single fixture in multiple directions. A good vibration fixture avoids under-test or over-test of the test items. This has its implications on timely completion of test programme and further development activity as a whole.

Various configurations are conceived before arriving at the present configuration. FE analysis is carried out using ANSYS tool. Evaluation is carried out using 4t duel shaker system. The control mechanism of vibration shaker works best for properly designed vibration fixtures.

The first natural frequency of the fixture from FE analysis is 8543.4 Hz. From the test the natural frequency observed is above the frequency limits. This is in close correlation with the prediction. All the accelerometers response at the top of the fixture are similar indicating a good design.

The fixture is suitable for vibration testing of cartridges used here and of future upcoming cartridges of similar types.

VI. ACKNOWLEDGMENT

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