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# Dynamic Behaviour of the Rubber Isolator Under Heavy Static Loads in Aerospace Systems

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***Abstract: Vibration isolators made of rubber, which is highly nonlinear, are extensively used in airborne systems to protect the highly sensitive units like Inertial Navigational Systems (INS) against flight vibrations and stage separation shocks. Vibration isolators are subjected to static load due to forward acceleration and dynamic load due to vibration simultaneously. In this project an attempt has been made to predict the dynamic behavior of the rubber isolator for this kind combined loading. A single degree-of-freedom system has been realized with a rubber vibration isolator, an attached mass and also a provision to apply static load. The isolator system was subjected to a random base excitation at different levels ranging from 0.5grms to 2.5grms using an electrodynamics shaker with any static load. Further vibration testing was carried out at the above excitation levels by applying static load on the isolator at different levels on the rubber isolator. Static load was applied in both compression and tension. It is observed that by increasing the static load the dynamic stiffness of the rubber increases and damping reduces and under the same static load with increase in the dynamic load, dynamic stiffness reduces and damping increases. A mathematical model of the system consisting of linear as well as nonlinear dynamic parameter was developed. From the lowest possible excitation level of 0.05g, linear dynamic parameters of the system were identified. The optimum coefficients of stiffness and damping nonlinearities were found so that the experimental transmissibility values at different frequencies were matched with simulated transmissibility values using mathematical model.***

***Keywords: Rubber Isolator, Heavy Static Load, Electrodynamics Shaker, Random Excitation, Mathematical Model***

## I. INTRODUCTION

Any component or sub-assembly in an airborne system, like missile or aircraft, is to be qualified for the specific vibration environment of that system to ensure proper functioning of the component during the flight. Each component in a missile or aircraft may function well under ideal static conditions; they also have to perform in dynamic environmental conditions during the flight. Failure of any single component will lead to catastrophic failure of the whole flight. Here failure does not mean only structural failure of the component; it can be momentarily malfunctioning of the system. Hence the general practice is to simulate the flight vibrations on the subsystems using electrodynamics or hydraulic shakers, with vibration testing specifications based on MIL-STDs (Military- standards) and simultaneously checking the performance of the subsystem. But it is not always possible to do the test due to safety issues, sensitivity or cost of the system. In such situations one has to predict the behavior or dynamic response of the system for specified excitation levels. This can be done by deriving the mathematical model of the system using its dynamic parameters, including nonlinearities from available test data at lower excitation levels. Very few researchers worked on the various isolators under different loading conditions. Smith et.al [1] presented inertial navigation system represents the ideal in automated navigation. They operate entirely without external assistance other than information as to the starting conditions, which make them of great interest to both military and civil operators. Titterton et.al [2] in this work, strap down inertial navigation technology, they involved the combination of accurate mechanical gyroscopes and sensitive accelerometers to carry out navigation. The early stimulus had been the requirement to develop air navigation systems which were independent of ground beacons, which might suffer jamming or destruction in the time of war, or fading or atmospheric problems in peacetime. Obviously, INS find their principle application in the guidance of airborne, marine and land vehicles. More recently, however, other applications have been developed, such as surveying underground pipelines and bore holes during drilling operations. Storer and Tomlinson [3] higher order frequency response functions form a good basis to describe a class of nonlinear structures which can be represented by polynomial equations. By defining their characteristics in terms of the leading diagonals of the general multi-dimensional FRFs, significant simplifications are made which leads to a practical procedure for experimentally measuring the higher order transfer functions that approximate to the exact higher order FRFs. Atkins and Worden [4] discussed the method of structural identification nonlinear dynamic systems are offered in the conditions of uncertainty. The method of constructing the set containing the data about a nonlinear part of system is developed. The concept of identifiability system for a solution of a problem structural identification is introduced. The method of estimation the class of nonlinear functions on the basis of the analysis sector sets for the offered structure S is described. We offer algorithms of structural identification of single-valued and many-valued nonlinearities. Examples of structural identification of nonlinear systems are considered. Kerschenet al. [5] reviewed the past and recent developments in system identification of nonlinear dynamical structures. The objective is to present some of the popular approaches that have been proposed in the technical literature, to illustrate those using numerical and experimental applications, to highlight their assets and limitations and to identify future directions in this research area. The fundamental differences between linear and nonlinear oscillations are also detailed in a tutorial. Hence, to the author's best knowledge, few literatures is available on the dynamic behaviour of the rubber isolator under heavy static loads in aerospace systems. So, in the current study, following objectives were selected for extensive analysis on dynamic behaviour of the rubber isolator under heavy static loads in aerospace systems because of key applications in many emerging applications.

The objectives of the work reported here can be summarized

*A.* To develop single-degree-of-freedom (SDOF) nonlinear model of a vibration isolator of silicone rubber with an attached mass, from measured data under harmonic excitation.
*B.* To validate the model with test data for lower levels of random vibration input.
*C.* Using this model, predict the response of the system when subjected to higher levels of random excitation.

## II. EXPERIMENTAL METHOD AND PROCEDURE

### *A. Random Vibration Testing*

Random base excitation tests were done on the isolator at various acceleration amplitude levels to generate the transmissibility data. Experimental setup and test procedures are explained in the following sections.

### *B. Experimental Setup*

A single-degree-of-freedom system was made with a conical shape, isoelastic silicone rubber vibration isolator as shown in **Fig.** 1 and an attached cylindrical steel mass of 2.5Kg, which was machined carefully to have its centre of gravity on the central axis. This was mounted on a 2 ton electrodynamic shaker by fixing the base of the isolator to the shaker with a proper fixture as shown in Fig. 2. One ICP (Integrated Circuit Piezoelectric) accelerometer was mounted at the base of the isolator to measure the excitation (control) and one more on the centre of the top of the cylinder to measure the response. ICP accelerometers have an in-built signal conditioner. The accelerometers are connected to closed-loop digital vibration controller using two ultra low noise microdot cables. Instrumentation details are given in Table 1. A torque wrench is used where the tightness of screws and bolts is crucial. It allows the operator to measure the torque applied to the fastener so it can be matched to the specifications for a particular application. This permits proper tension and loading of all parts. A torque wrench measures proxy for bolt tension. The technique suffers from inaccuracy due to inconsistent or uncalibrated friction between the fastener and its mating hole. Measuring bolt tension is actually what is desired, but often torque is the only practical measurement which can be made.

### *C. Modeling of the Structure and Individual Parts in Solid Works*

Experimental was performed considering that extra static load is acting on the isolator while the re-enter of the missile systems. Experiment was done with following and the re-entry continuous loads of missile systems.

*1)* Isolator is in compression state.
*2)* Isolator is in tension state.

Experimental setups used to carry out the experiments under compression state and tension state are shown in Fig. 3 and 4 respectively. Gap pads are used in the experiment for the precise measurement of the gap between the isolator and the plate and also for the precise compression of the isolator. Four sets of gap pads with different dimensions were used. After precise measurement of the height and compression gap pads were removed and will carry out the experiment.

*3)* 12.3mm (plate compressed by 0.3mm)
*4)* 12mm (plate compressed by 0.7mm)
*5)* 11.5mm (plate compressed by 1.2mm)
*6)* 11mm (plate compressed by 1.7mm)

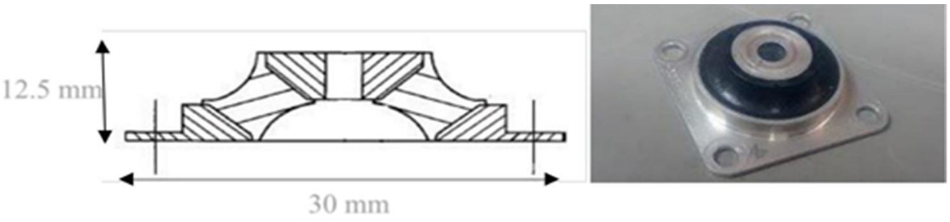

Fig. 1 Silicone rubber vibration isolator, Lord, AM006-15.

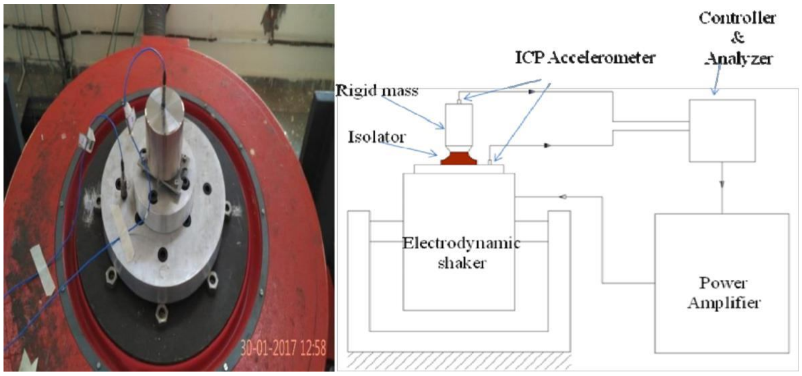

Fig. 2 Electrodynamic shaker by fixing the base of the isolator to the shaker with a proper fixture

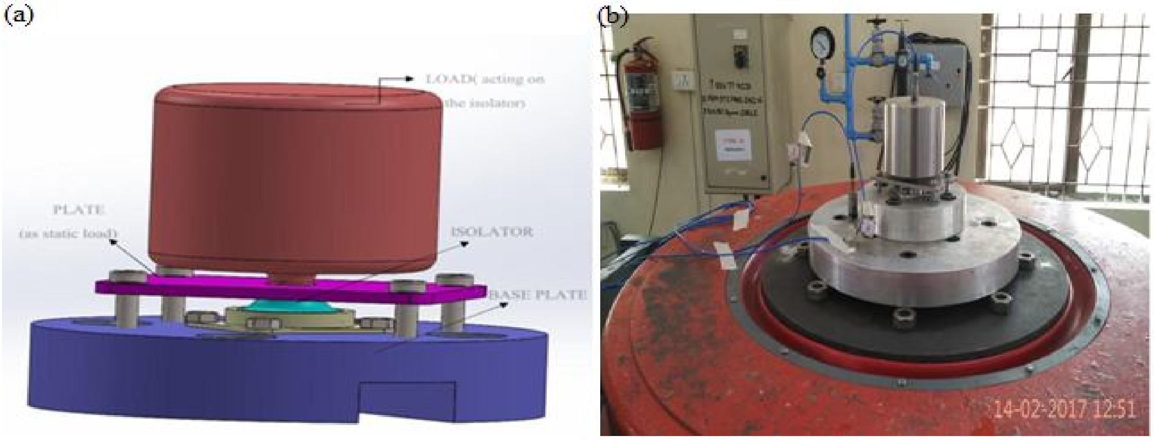

Fig. 3 Assembly setup for compression state (a) CAD model (b) Actual

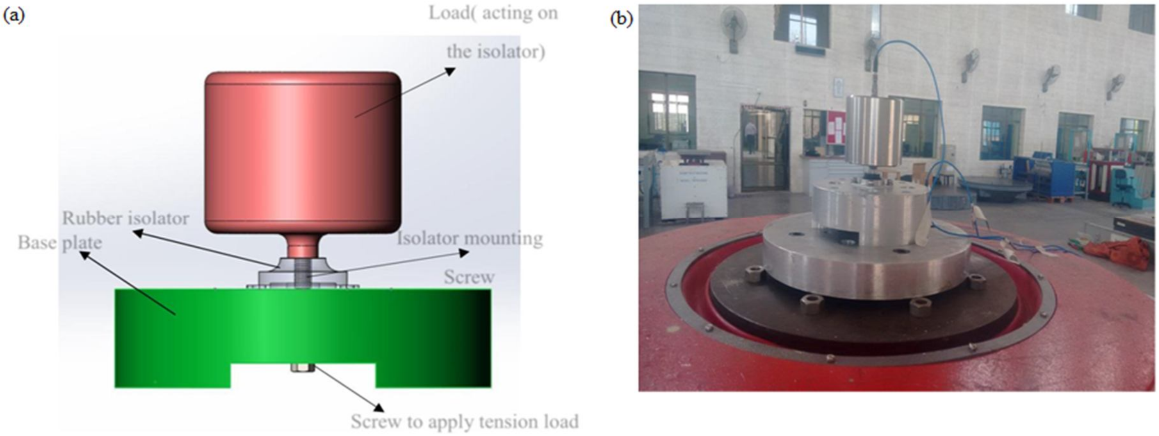

Fig. 4 A typical assembly model with rubber isolator in tension state (a) CAD model (b) Actual setup.

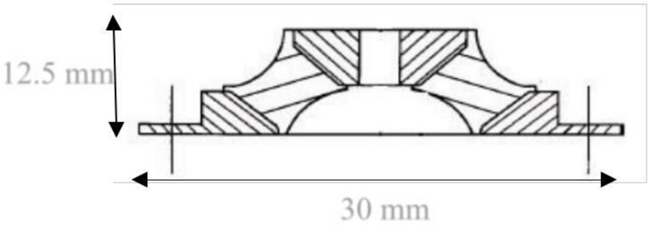

Table 1 Instrumentation Details

| S.No. | Instrument | Make | Details |
|---|---|---|---|
| 1 | Electrodynamic Shaker | Saraswathi Dynamics | Dynamic force capacity: 2Ton |
| 2 | Digital vibration controller | LDS Dactron | 16 Channel, Sine& Random vibrations, Closed loop control |
| 3 | ICP accelerometers | PCB | Model No: 353B51<br>S. No.: SN79538 & SN 79541<br>Sensitivity : 519.93 & 485.6 mV/g |

### *D. Experimental Procedure*

*1)* The vibration rubber isolator setup for which the experiment is to carry out is shown in fig. 3.10.
*2)* Vibration testing is carried out at dynamic test centre.
*3)* The entire setup is placed on the fixture which is made up of aluminum alloy and fixed on electrodynamic shaker.
*4)* The dynamic capacity of the shaker is 2 ton.
*5)* After placing the setup on the shaker accelerometers are attached to the setup.
*6)* These will give the response and control of the isolator.
*7)* Constant frequency, base excitation sinusoidal tests were done with base acceleration levels from 0.05g (g, acceleration due to gravity, 9.81 m/sec2).
*8)* The test was done for 30 seconds at each frequency for a given excitation level, so that steady state is attained. The experiment was done at the lowest possible excitation level of 0.05g with the above equipment, to capture the linear behaviour of the isolator.
*9)* Transmissibility peak value and corresponding frequency value for each excitation amplitude are given in Table.
*10)* While the RMS value is varying from 0.5 to 2.5, the corresponding results i.e. FRF, CONTROL and RESPONSE are observed.
*11)* These are recorded in the CONTROLLER software
*12)* As mentioned that the rubber is in compression state, going to explain about that process.
*13)* The plate that placed on the isolator is to apply static load. This plate is tightened to the base plate by means of screws by applying the torque on the screws, they will get more tightened and compress the rubber isolator. Vibration testing was carried on this rubber isolator which is under compression state.

## III. RESULTS AND DISCUSSIONS

Random base excitation tests were done on vibration electrodynamic shaker. Testing was done at different RMS value varying from 0.5grms to 2.5grms. The experiment was done to capture the linear behavior of the isolator. Testing was done in two steps i.e. without static load and with static load for isolator is in compression and rubber isolator is in tension state.

### *A. Rubber Isolator in Compression*

*1)* Without static loa
*2)* With static loa
*3)* Static load applied on the plate by 0.3mm compression
*4)* Static load applied on the plate by 0.7mm compression
*5)* Static load applied on the plate by 1.2mm compression\
*6)* Static load applied on the plate by 1.7mm compression

a) *Without Static Load (Plate):* The entire experimental setup was fixed to the fixture and was placed on the electrodynamic shaker. By varying the RMS values from 0.5 to 2.5 the corresponding transmissibility peak frequency was recorded. The isolator frequency values for the various grams values are given in the Table 2.

Table 2 Without Static Load

| S. No. | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|
| Excitation (grams) | 0.5 | 1 | 1.5 | 2 | 2.5 |
| Transmissibility peak frequency(Hz) | 80 | 75 | 70 | 65 | 65 |
| Transmissibility peak, ratio | 4.71 | 3.91 | 3.84 | 3.45 | 3.06 |

b) *With Static Load:*

i) *Plate Compressed by 0.3mm:* The experimental setup was fixed to the fixture and is placed on the electrodynamic shaker. The plate was compressed by means of screws on the plate that are fixed to the fixture. By varying the RMS values from 0.5 to 2.5 the corresponding transmissibility peak frequency is recorded. The frequency values for the various grams values are given in the Table 3.

Table 3 Static Load On Plate Of 0.3mm

| S. No. | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|
| Excitation (gms) | 0.5 | 1 | 1.5 | 2 | 2.5 |
| Transmissibility peak frequencyb(Hz) | 110 | 90 | 80 | 75 | 75 |
| Transmissibility peak, ratio | 5.33 | 4.22 | 3.86 | 3.63 | 3.80 |

ii) *Plate Compressed by 0.7mm**:*** The experimental setup was fixed to the fixture and is placed on the electrodynamic shaker. The plate is compressed by means of screws on the plate that are fixed to the fixture. By varying the RMS values from 0.5 to 2.5 the corresponding transmissibility peak frequency is recorded. The frequency values for the various grams values are given in the Table 4.

Table 4 Static Load On Plate Of 0.7mm

| S. No. | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|
| Excitation (grams) | 0.5 | 1 | 1.5 | 2 | 2.5 |
| Transmissibility peak frequency(Hz) | 150 | 125 | 120 | 110 | 105 |
| Transmissibility peak, ratio | 6.78 | 6.13 | 5.66 | 5.11 | 4.85 |

iii) *Plate Compressed by 1.2mm:* The experimental setup was fixed to the fixture and is placed on the electrodynamic shaker. The plate is compressed by means of screws on the plate that are fixed to the fixture. By varying the RMS values from 0.5 to 2.5 the corresponding transmissibility peak frequency is recorded. The frequency values for the various grams values are given in the below Table 5.

Table 5 Static Load On Plate Of 1.2mm

| S. No. | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|
| Excitation (grams) | 0.5 | 1 | 1.5 | 2 | 2.5 |
| Transmissibility peak frequency(Hz) | 205 | 200 | 190 | 180 | 170 |
| Transmissibility peak, ratio | 13.86 | 10.63 | 7.25 | 7.05 | 7.05 |

iv) *Plate Compressed by 1.7mm:* The experimental setup was fixed to the fixture and is placed on the electrodynamic shaker. The plate is compressed by means of screws on the plate that are fixed to the fixture. By varying the RMS values from 0.5 to 2.5 the corresponding transmissibility peak frequency is recorded. The frequency values for the various grams values are given in the Table 6.

Table 6 Static Load On Plate Of 1.7mm

| S. No. | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|
| Excitation (grams) | 0.5 | 1 | 1.5 | 2 | 2.5 |
| Transmissibility peak frequency(Hz) | 215 | 210 | 210 | 200 | 200 |
| Transmissibility peak, ratio | 16.57 | 12.46 | 11.09 | 10.72 | 7.72 |

### B. *Rubber Isolator in Tension*

1) *Rubber is in Elongation by 0.3mm:* The experimental setup was fixed to the fixture and is placed on the electrodynamic shaker, now we have to make the rubber to elongated i.e. in tension state. While the isolator is in tension state, we will vary RMS value from 0.5 to 2.5 the corresponding transmissibility peak frequency is recorded. The frequency values for various grams values are given in the Table 7.

Table 7 Isolator Is In Elongation Of 0.3mm

| S. No. | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|
| Excitation (grams) | 0.5 | 1 | 1.5 | 2 | 2.5 |
| Transmissibility peak frequency(Hz) | 185 | 120 | 115 | 100 | 100 |
| Transmissibility peak, ratio | 3.41 | 3.34 | 3.08 | 2.59 | 2.53 |

2) *Rubber is in Elongation by 0.7mm:* The experimental setup was fixed to the fixture and is placed on the electrodynamic shaker, now we have to make the rubber to elongated i.e. in tension state. While the isolator is in tension state, we will vary RMS value from 0.5 to 2.5 the corresponding transmissibility peak frequency is recorded. The frequency values for various grams values are given in the Table 8.

Table 8 Isolator Is In Elongation Of 0.7mm

| S. No. | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|
| Excitation (grams) | 0.5 | 1 | 1.5 | 2 | 2.5 |
| Transmissibility peak frequency(Hz) | 280 | 260 | 195 | 210 | 170 |
| Transmissibility peak, ratio | 3.58 | 3.13 | 2.91 | 2.44 | 2.23 |

### *C. MATLAB Figures for the Experimentation Results*

#### *1) Comparison of all Loads Acting for Separate Grams Values:*

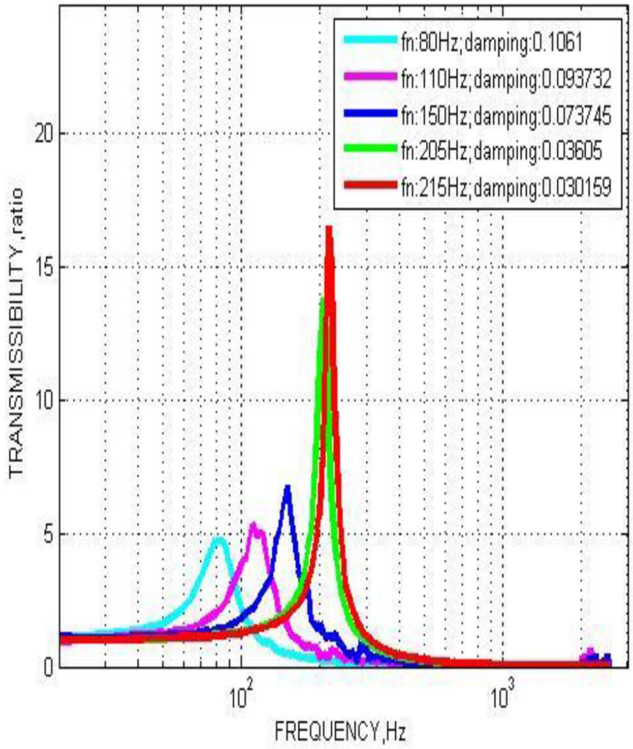

Fig. 5 Transmissibility curve for random excitation of 0.5grams

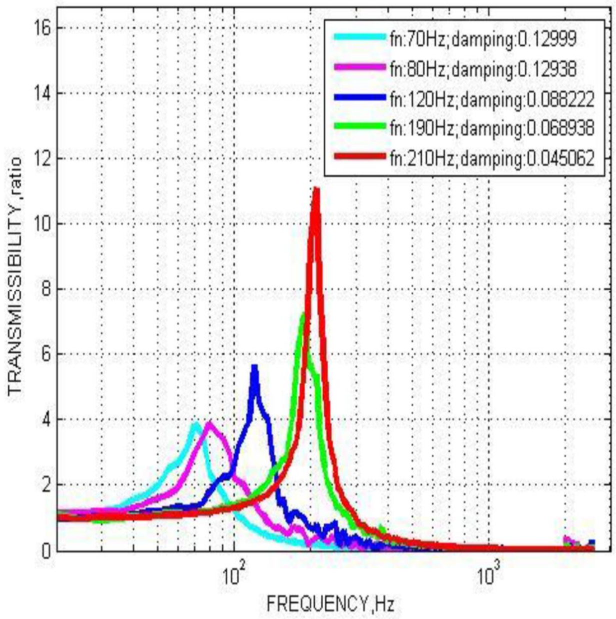

Fig. 6 Transmissibility curve for random excitation of 1 grams

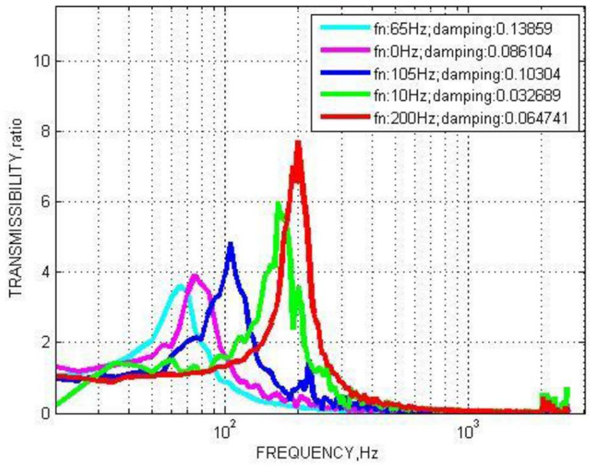

Fig. 7 Transmissibility curve for random excitation of 1.5 grams

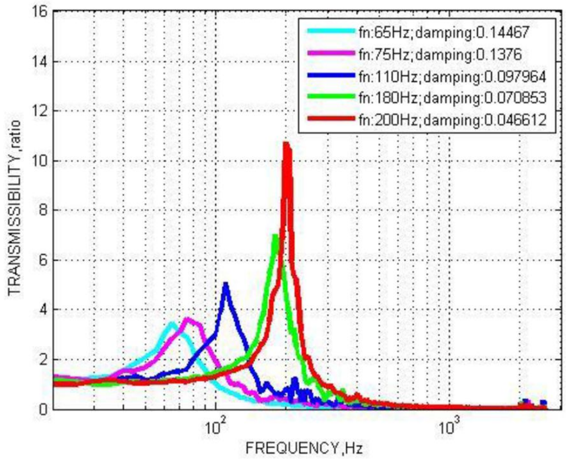

Fig. 8 Transmissibility curve for random excitation of 2 grams

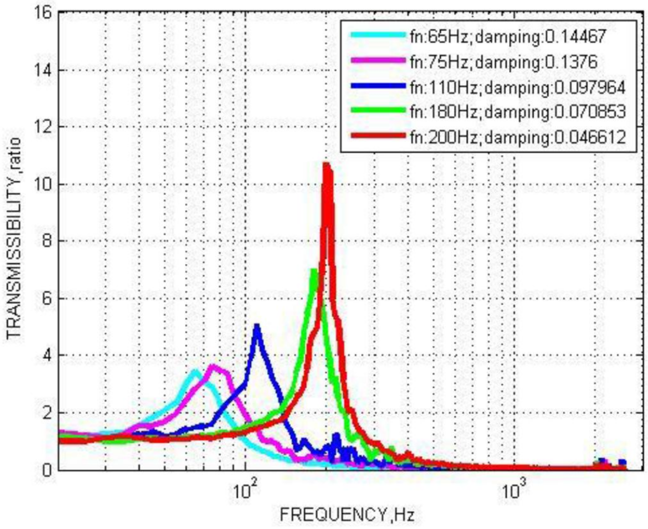

Fig. 9 Transmissibility curve for random excitation of 2.5 grams

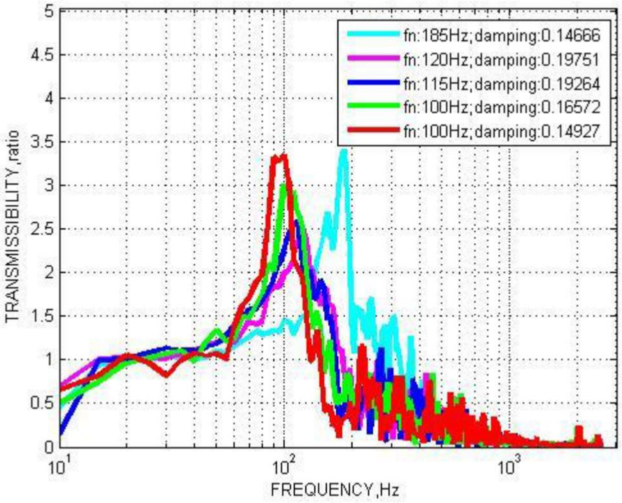

Fig. 10 Transmissibility curve for random excitation of 0.5grms to 2.5grams(With static load) tension of 0.3 mm

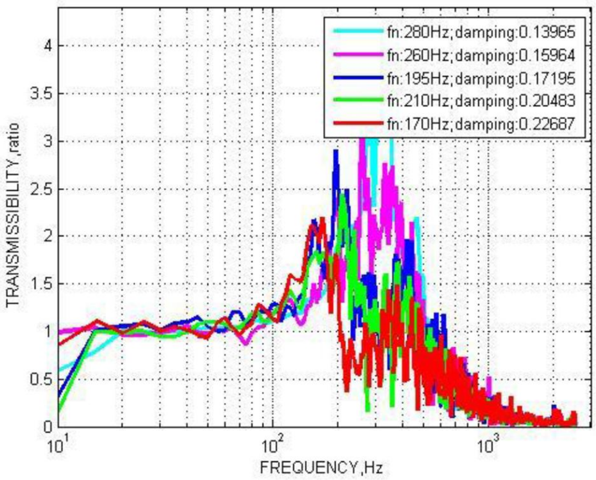

Fig. 11 Transmissibility curve for random excitation of 0.5grms to 2.5grams (With static load) tension of 0.7 mm

From the test data it is observed that by applying the static load on the rubber either compression or tension, it's dynamic stiffness has been increased, i.e. isolators natural frequency is increasing with application of static load compared to no static load condition. It is also observed that more the static load more is the increase in the dynamic stiffness of the isolator. One more important observation is that with increase in the dynamic excitation level, the dynamic stiffness is getting reduced and it is happening in both the cases of with static load and without static load. It is also observed that with by applying the static load on the rubber either compression or tension, damping value has been decreased. And also with increase in the dynamic excitation level, the damping value is getting increased and it is happening in both the cases of with static load and without static load.

### *D. Mathematical Modeling*

Mathematical modeling refers to the use of mathematical language to simulate the behavior of a 'real world' (practical) system. Its role is to provide a better understanding and characterization of the system. Theory is useful for drawing general conclusions from simple models, and computers are useful for drawing specific conclusions from complicated models. In the theory of mechanical vibrations mathematical models- termed structural model are helpful for the analysis of the dynamic behavior of the structure being modeled. From Figures (3-9) and Tables (2-8), it is clearly evident that natural frequency of the system is reducing with increasing excitation level, exhibiting soft spring behavior (Negative nonlinear coefficients), the peak value of the transmissibility is also reducing with increasing excitation level (Positive nonlinear damping coefficients), i.e. system characteristics are function of the input energy. Jump phenomenon is also visible in Fig. 12. Which are all indicates the nonlinear behavior of the system [4]. Hence the rubber isolator with cylindrical mass can be idealized as a SDOF system with nonlinearities in stiffness as well as damping as shown Fig. 12. With the application of the static load either compression or tension, the dynamic stiffness of the isolator is increasing and damping is reducing.

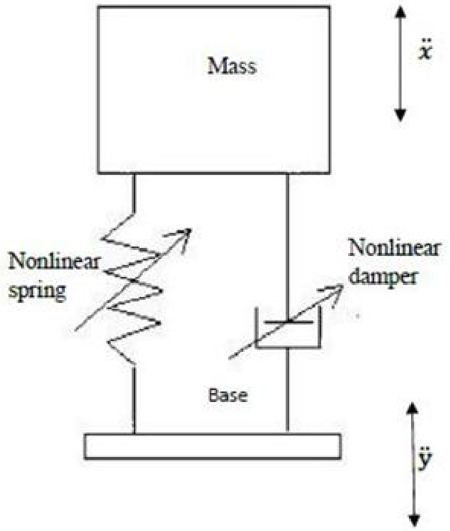

Fig. 12 Idealized Nonlinear SDOF system for isolator

In Fig. 10., $\ddot{y}$ is the base excitation (control) and $\ddot{x}$ is the absolute response of the isolator. We can define relative acceleration, $\ddot{z}=\ddot{x}-\ddot{y}$. The governing equation for motion for the above system can be written as

$$m\ddot{z} + c\dot{z} + kz - k_2|z|z + c_2|\dot{z}|\dot{z} - k_3z^3 + c_3\dot{z}^3 + k_{2\,static}|z|z - c_{2\,static}|\dot{z}|\dot{z} + k_{3\,static}|z|z - c_{3\,static}|\dot{z}|\dot{z} = -m\ddot{y} \quad (1)$$

In the first step, the transmissibility curve with the minimum base excitation i.e., 0.05 g is considered as linear and the nonlinear effects are neglected. Hence the peak frequency (where the transmissibility is maximum) for 0.05g excitation, which was found to be 110 Hz is assumed as the linear natural frequency of the system. With $\omega n$= 2π x 110 rad/s and m = 2.5 kg, the linear stiffness coefficient is obtained as $k$ = 1193.02 kN/m. The viscous damping factor '$\zeta$' is then obtained from the value of the resonance transmissibility ($Tr$), using the standard result, '$\zeta$'=1/2($\sqrt{Tr2-1}$), which comes out as 0.0746. Thus the linear viscous damping coefficient c=2 $\zeta\sqrt{km}$ =162.96 Ns/m. Thus all the linear dynamic parameters are found out.

The next step is to find out nonlinear coefficients. As there is no prior information about the functional form of nonlinearities, trail values of the k2, k3, c2, c3, 'k' static, 'c' static are to be assumed for different excitation levels. A MATLAB based program was developed to find out the optimum coefficients of stiffness and damping nonlinearities to match experimental transmissibility values at different frequencies with simulated transmissibility values using the mathematical model (Eqn.1) in least square sense. But due to lack of time parameter estimation step is continued further.

## IV. CONCLUSIONS AND FUTURE SCOPE

### *A. Conclusion*

1) Isolator shows linear behavior at 0.05g excitation. Beyond 0.05g excitation it exhibits nonlinear system behaviour with soft spring nature.
2) With increase in the static load on the rubber isolator its frequency is increasing. This means that the stiffness of the isolator is increasing both in compression state and in tension state of the rubber, where damping is increasing with increase in the static load. But with increase in the dynamic loading the dynamic stiffness is reducing and damping is increasing irrespective of static load application.
3) This is showing clearly that during the accelerating phase and re-entry phase of the missile, where rubber isolator subjected to compression and tension, its natural frequency increases only. Hence INS sensors band width during these phases does not reduces,infact it increases.

## V. FUTURE SCOPE

*A.* Modeling of the rubber isolator for the rubber in tension state can be improved

*B.* The nonlinear dynamic parameters values are to be found out by using different methods such as the nonlinear resonant decay method, the conditioned reverse path method, the nonlinear identification through feedback of the output method. MATLAB based program to be developed to find out the optimum coefficients to prove that the values obtained from test data and the simulation are equal.

## REFERENCES

[1] Deger, S. and Smith, R., (1983). Military expenditure and growth in less developed countries. Journal of conflict resolution, 27(2), pp.335-353.
[2] Titterton, D. and Weston, J.L. (2004). Strapdown inertial navigation technology (Vol. 17). IET.
[3] D.M. Storer and G.R. Tomlinson, (1993). Recent developments in the measurements and interpretation of higher order functions from nonlinear structures, Mechanical Systems and Signal Processing 7, 173–189
[4] P. Atkins and K. Worden, (1997) Identification of a multi-degree-of-freedom nonlinear system, in: Proceedings of the 15th International Modal Analysis Conference, Orlando, pp. 1023–1028
[5] G. Kerschen, K. Wordenb, A. F. Vakakisc and J.C. Golinvala, (2006). Past, present and future of nonlinear system identification in structural dynamics,Mechanical Systems and Signal Processing 20 ,505–592.