Dynamic Vibration Analysis of an Isolator

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***Abstract*—** This paper discusses the experimentally measured dynamic properties of a vibration isolator. A vibration test rig is developed for measuring the vibration properties of an elastomer. The excitation frequency and amplitude is varied and transmissibility and damping ratio is calculated for provided material. The behaviour of the material at varying load is studied. The dynamic natural frequency is reduced as load increases. This analysis is helpful for proper selection of vibration isolating material for a particular application.

***Index Terms* — Vibration Isolator, Elastomer,Vibration test rig, ExcitationFrequency, Amplitude, Damping ratio, Transmissibility.**

I. INTRODUCTION

The purpose of vibration isolation is to control unwanted vibration so that its adverse effects are kept within acceptable limits.An isolator is a resilient support which decouples an object from steady state or forced vibration. Elastomers are used in the design of isolators due to their unique ability to provide effective control of dynamic forces. As engineering materials, elastomers are able to undergo rapid cyclic deformations at discrete frequencies or over a range of frequencies. This is due to the inherent properties of rubber, mainly flexibility, high elongation and combination of resilience & damping. Natural frequency and damping are the basic properties of an isolator which determine the transmissibility of a system designed to provide vibration and/or shock isolation. Additionally, other important factors must be considered in the selection of an isolator/isolation material. The current work investigates dynamic vibration properties of an isolator. The dynamic characteristics of elastomer depend on static pre-load, temperature and the frequency and amplitude of vibration. The effect of two parameters i.e. excitation frequency and amplitude on behaviour of the vibration isolatorisstudied.

Number of vibration test rigs have been developed for the investigation of the characteristics of isolator. John D. Dickens (1999) developed a novel vibration isolator test facility. The vibration isolator test facility is capable of measuring the four-pole parameters over the frequency range from 5 Hz to 2 kHz, static load range from 1 to 30 kN, and temperature range from 6 to 60 -C. In order to demonstrate the usefulness of the facility, experimental data for three commercial asymmetrical vibration isolators used for maritime applications are presented. Y Du [2002] shows that the internal dynamics of the isolator, which are also known as internal resonances (IRs) or wave effects, can significantly affect the isolator performance at high frequencies. MattiasSjoberg [2002] aims at enhancing the understanding and to provide improved models of the dynamic behavior of rubber vibration isolators which are widely used in mechanical systems. Mustafa E. Levent [2003] presented two approximate methods for the determination of dynamic properties of vibration isolators or similar components exhibiting elastic and damping nonlinearities without the need for dedicated hardware. Eugene I. Rivin [2005] shows that a proposed model for vibration isolation of precision objects together with the use of 'smart' constant natural frequency (CNF) isolators substantially widens the application range of inexpensive and reliable passive isolators. J.A. Forrest [2006] discussed the experimentally measured free–free dynamics of three small-scale vibration isolator models: two single-stage isolators and one two-stage isolator.

YongjunXu et al [2009] carried out the experiments on steel rubber vibration isolator to investigate the compressive properties and fatigue properties in different low temperature conditions.Sivaraman R [2013] overcomes the usual problem of vulcanization of rubber by selecting carbon fibers as the filler material. The material was modeled as a composite beam structure initially using the ANSYS software and dynamic analysis was performed to find out the fundamental mode of vibration. Martin Ostberg et al [2013] worked on the dynamic stiffness of hollowed cylindrical vibration isolators using a wave-guide modeling approach. The isolators consist of rubber and metal elements in series. G.S.Dutta et al [2013] performed experimental studies on elastomeric layered composites to characterize the nonlinearity in dynamic stiffness and specific damping energy, so that their performance can be enhanced as isolators.Lijesh.K.P et al[2014] proposed a hybrid bearing (passive magnet + Elastomer). The effect of Yong's modulus was investigated on butyl rubber isolator.

It can be seen from the literature review that work is done on the rubber and steel isolator. Dynamic properties such loss factor, stiffness, shear stiffness, mode shape, strain, damping factor, etc. are studied for a particular isolator. These properties were studied at different input conditions (temperature, frequency, load). Less work has been reported on the behaviour of the elastomer material at varying amplitude and excitation frequency. Hence for investigating the transmissibility and damping effect of an elastomeric material a vibration test rig is developed. The model of the elastomer thus created is a linear model, which consists of a parallel combination of a linear spring and a viscous damped element. Linearizing these coefficients simplifies the calculational procedure, but the method of obtaining results may be subject to errors in procedure, instrumentation and the sub-jective human element. [14]

Natural Frequency

All mounting systems have a natural frequency (f_n). It isthe frequency at which the system will oscillate if it is displaced from its static position and released.The natural frequency, f_n, is dependent upon the stiffness of the spring, K, and the mass of the load that it is supporting (M), and can be determined by the following equations:

$$f_n = \frac{1}{2\pi}\sqrt{\;}$$

where K is the stiffness in Newton per meter (N/m) and M is the mass in kilograms (Kg),

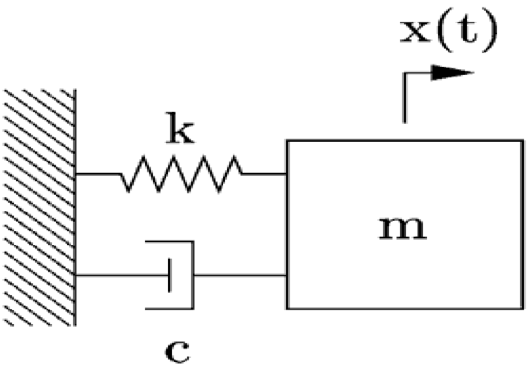

Fig No: 1 Single degree-of-freedom system with viscous damper.

Damping

Controlling the natural frequency provides one means to control vibration. Damping provides another. Damping is the dissipation of energy, usually by releasing it in the form of low grade heat. For example, dry friction, the most common damping mechanism, is the reason an object sliding on a surface will slow down and stop. Some mechanical devices use viscous damping as a means of energy dissipation. In these systems, fluid losses caused by a liquid being forced through a small opening provide the necessary energy loss. The shock absorbers on an automobile are an example of viscous dampers. Mathematical models for viscous damping are well established and provide a means for analysis. Viscous damping capability is characterized by the damping ratio, C/C_c or ζ .

$$\zeta = C / C_c$$

Transmissibility

The performance of an isolation system is determined by the transmissibility of the system. The ratio of the energy going into the system to the energy coming from the system. This can be expressed in terms of acceleration, force or vibration amplitude. Transmissibility (T) is equal to

$$T = \sqrt{\frac{1+(2\zeta\ r^2)}{(1-r^2)^2+(2\zeta\ r}}$$

Where:
T = Transmissibility
ζ = Damping ratio
f_d = Driving frequency
f_n = Natural frequency
r = frequency ratio (fd/fn)

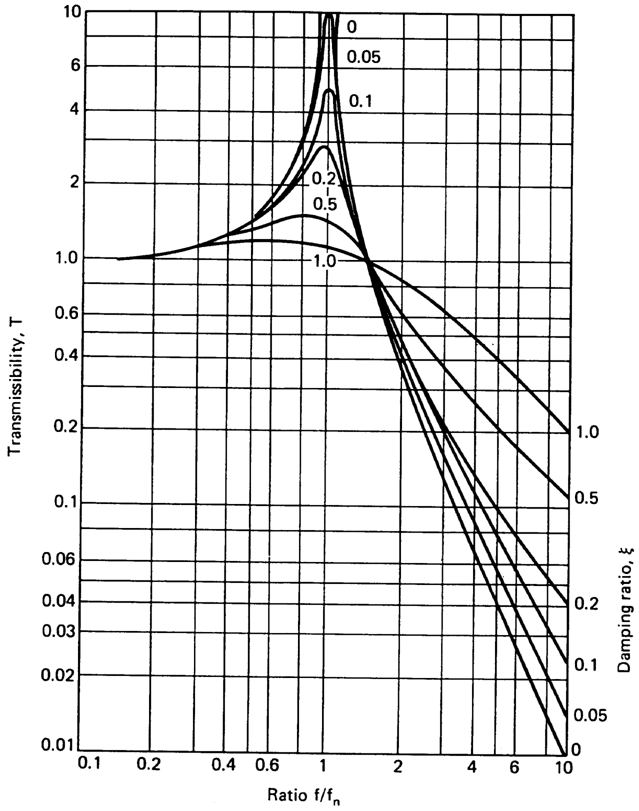

Fig No: 2 Transmissibility of a viscous-damped system[15]

Fig 2 shows typical transmissibility curves, for a highly damped material (ζ= 0.5), and for a material with much lower damping (ζ=0.05).At very low frequencies (fd/fn<< 1), the input vibration virtually equals output (the transmissibility is equal to 1), and input displacement essentially equals that of output. If the driving frequency equals the natural frequency (fd/fn = 1), the system operates at resonance. A system that is operating at resonance will have a transmissibility approaching infinity. As damping increases, the transmissibility at resonance decreases.When the frequency ratio equals the square root of two (fd/fn $=\sqrt{2}$), transmissibility will once again drop to 1. This is known as thecrossover frequency, and the area below this frequency is known as the amplification region. Above this frequency lies the isolation region, where transmissibility is less than 1. As a goal, the isolator designer tries to design a mounting system that puts the primary operating frequencies of the system in the isolation region. Methodology of the work is as follows :

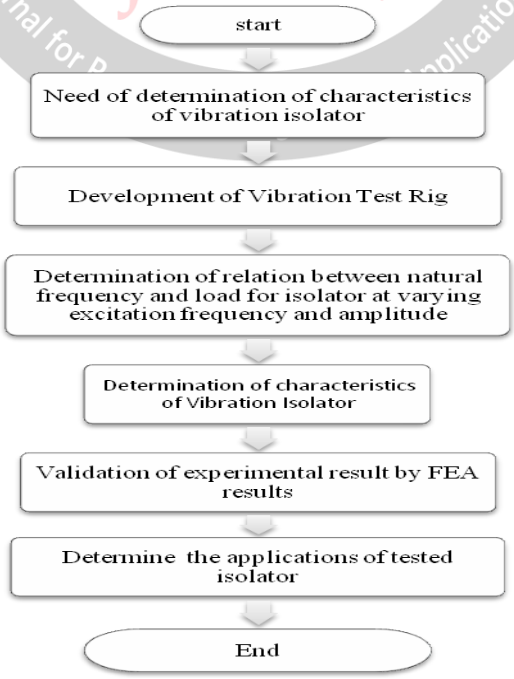

Fig 3: Methodology

Experimental Setup consists of a 1HP, AC motor which drives the slider crank mechanism. A dimmer is provided for speed variation so that excitation frequency can be varied. A slider of mechanism continuously hit on the isolator providing dynamic load of 3.35kg. For getting the variation in the amplitude isolator is clamped on the screw jack. By changing the height of the screw jack the isolator is tested for different loads (static load of 4kg, 5kg & 6kg). FFT analyser is used to measure the natural frequency.

The test is carried out for three conditions –

1) Without isolator,
2) With isolator,
3) With isolator at different loads.

Fig 4: Vibration Test Rig

A compression test had been taken on UTM. Slope of force vs displacement graph was found out, which provided the value of stiffness K of the material. The calculated value for K i.e 3.66 kN/mm is then compared with the standard values (Table No 1) of stiffness of an elastomeric materials. After that the natural frequency of the system is determined considering the maximum displacement of the test specimen. Disturbing frequencies are varied by varying the voltage (speed). At every frequency ratio (fd/fn) transmissibility is calculated and graph is plotted.

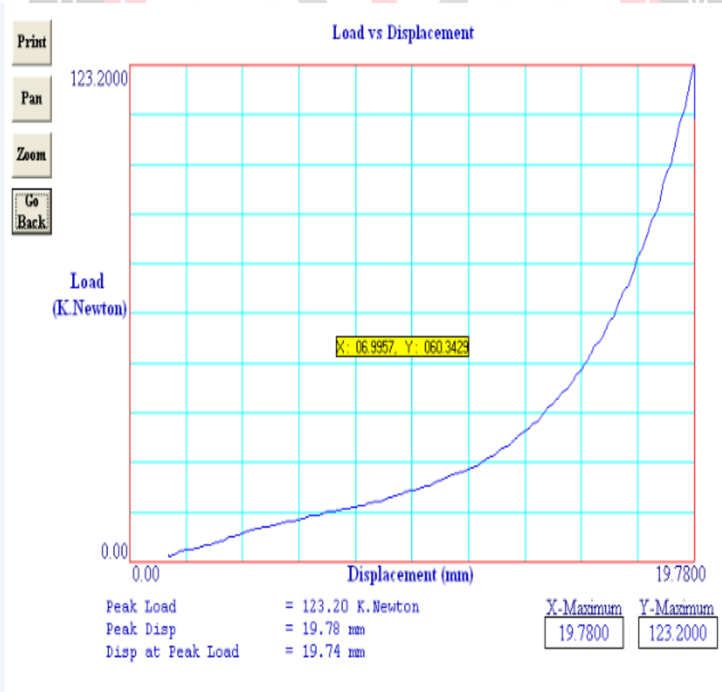

Fig No 5: Graph Force Vs Displacement (from UTM)

The maximum displacement of the test is 19.78mm at 123kN. Considering the static displacement natural frequency fn of the system is 3.566 Hz. Damping ratio is 0.07 which is the damping ratio of Butyl Rubber.

Table 2 - Abbreviations for Rubbers

Abbreviation	Description
SBR	Styrene-butadiene rubber
BR	Polybutadiene
EPDM	Ethylene-propylene-diene terpolymer
IIR	Isobutylene-isoprene rubber
NR	Natural rubber (SMR 5)
IR	Isoprene rubber, synthetic
CR	Chloroprene rubber

Table 3 - K' and C at Different Temperatures for Various Rubbers

Compound No.		10	11	12	13	14	22
Rubber		SBR	SBR/BR	BR	IIR	EPDM	NR
Property	T, F						
K'	0	1650	1560	1440	5600	2695	2505
K'	80	965	1090	1170	2070	1655	1495
K'	160	750	932	1070	1440	1225	1150
K'	240	695	856	1015	1200	1040	950

Table No: 1 Stiffness of different elastomer [13]

Comparing the calculated stffness with valuse in table no:1 the material is identified. Further damping ratio is calculated (0.07) which is same as Butyl rubber as given in table No:2

Material / Structure	Loss factor, *eta*	Viscous damping ratio, *zeta*
steel	0.0004	
aluminium	0.0001	
fiber mats	0.1	
natural rubber		0.01 - 0.08
butyl rubber		0.05 - 0.5
polystyrene	2	
welded structure		0.02
bolted structure		0.06

Table No: 2 Damping Ration of Different Materials [16]

Sr No	Speed	f_d/f_n	Transmissibility
1	100	0.4673	1.2794
2	125	0.5842	1.5181
3	200	0.9344	7.8342
4	230	1.0748	6.5756
5	531	2.4817	0.1932
6	1086	5.0754	0.0402
7	2000	9.3475	0.115

Table No: 3 f_d/f_n and Transmissibility for test specimen

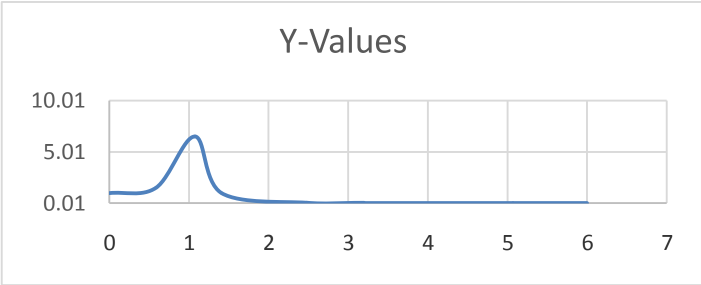

Fig No: 6 Graph of Transmissibility (Y axis) vs. fd/fn Ratio(x axis)

The resonant frequencies at different load are measured. The graph shows Load to Natural Frequency characteristics.

Load (Kg)	Resonance Dynamic Frequency(Hz)
3.5	828
7	822
8	820
9	818

Table No: 4 Resonance frequencies at different load

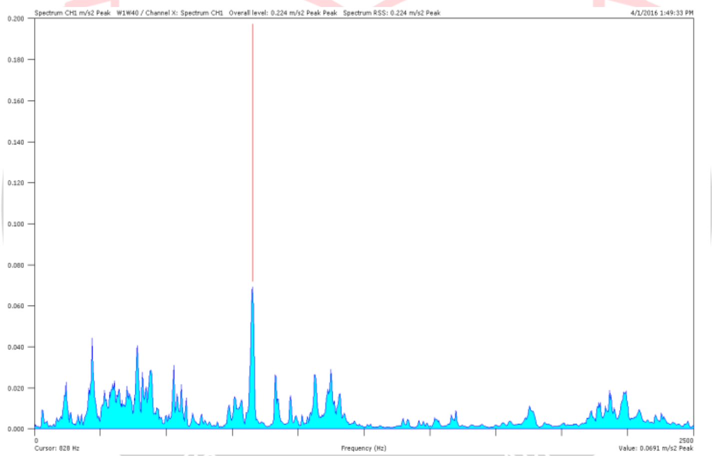

Fig No: 7 FFT analyser Result

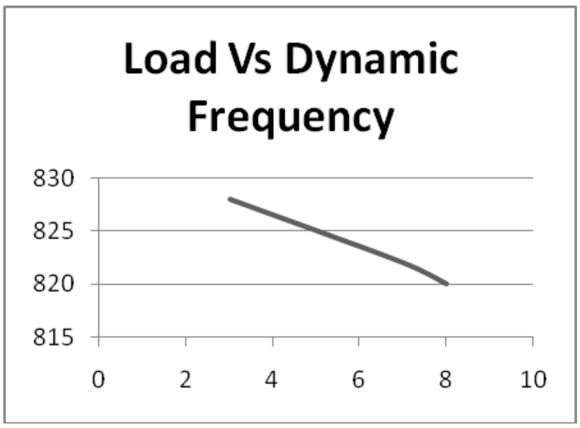

Fig No: 8 Graph Dynamic Frequency (Y axis) Vs Load (X axis)

An Experimental apparatus is configured to investigate the dynamic behaviour of the Elastomeric Isolator. Vibration properties of the test specimen are determined. i.e Damping Ratio is 0.07and stiffness is 3.66 KN/mm. From the readings of FFT at different loads it can be see that the dynamic frequency reduces as load increases. From the graph of frequency ratio Vs transmissibility (Fig No: 6) it can be say that as the excitation frequency increases transmissibility increases from 1 up to 7.83 and again reduce to 0.01. The nature of curve is similar to the standard frequency ratio Vs transmissibility graph with damping ratio of 0.05. Further as amplitude increases dynamic frequency reduces. Present study is done at static load of 4kg, 5kg and 6kg. Considering the load carrying capacity of the test specimen the dynamic response of the isolator can be tested with higher load.

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