

Vibration Analysis of an All-Terrain Vehicle

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Abstract - Good NVH is becoming must feature in recent commercial vehicles. One of the major discomforts caused to driver by Engine vibration during idling. The power train is suspended on the vehicle frame on several flexible mounts, whose purpose is to isolate the vibration between engine and frame. Total 6 different modes of Engine like roll, yaw, pitch and Vertical, lateral and longitudinal need to isolate. Engine mount stiffness and position is critical and need to have methodology to verify in early stage of designing [1].

In this paper, the stiffness and modulus of elasticity value of engine rubber mount is calculated experimentally by carrying out compressive test on UTM. Later, the critical frequency is found by carrying modal analysis on this engine mount through ANSYS. Also by doing calculation, the position of driver's seat is found ensuring the minimum NVH condition.

Index Terms- Engine mount, Stiffness, NVH, Transmissibility.

1. INTRODUCTION

Noise, vibration and harshness (NVH) have become increasingly important factors in vehicle design as a result of the quest for increased refinement. Vibration has always been an important issue closely related to reliability and quality, while noise is of increasing importance to vehicle users and environmentalists. Harshness, which is related to the quality and transient nature of vibration and noise, is also strongly linked to vehicle refinement. Considering the major contribution of the powertrain vibrations to overall vehicle NVH, the use of simulation tools during powertrain design stages is a critical stage of the vehicle NVH development. Power train mounting system and its transmissibility characteristics are also the key governing factors. Engine mounts protect the engine from excessive movement and forces due to low frequency road and high frequency engine excitations. On the other hand, body mounts protect the cabin from vibration forces exerted by the body [1].

1.1. Theory of Vibration

Idle vibrations of any automotive system originate from the power train. The major causes being:

1.1.1 Unbalanced Mechanical and Combustion Forces of the Engine

Fig. 1 shows the typical vibration generating mechanism of an engine. An important parameter influencing engine vibration especially at low speed is combustion. The excitation of the system originates from cylinder pressure, which causes a thrust force against cylinder liner wall and acts on crank train components which results in various modes of the power train. At higher running speeds of the engine mechanical noise is dominant. The factors contributing to it are the inertial forces and the variable torque generation of the engine. This leads to problems like piston slap and bearing impact forces, which gets radiated as noise or as vibration through different engine parts like crank pulley, crank case, oil pan etc.

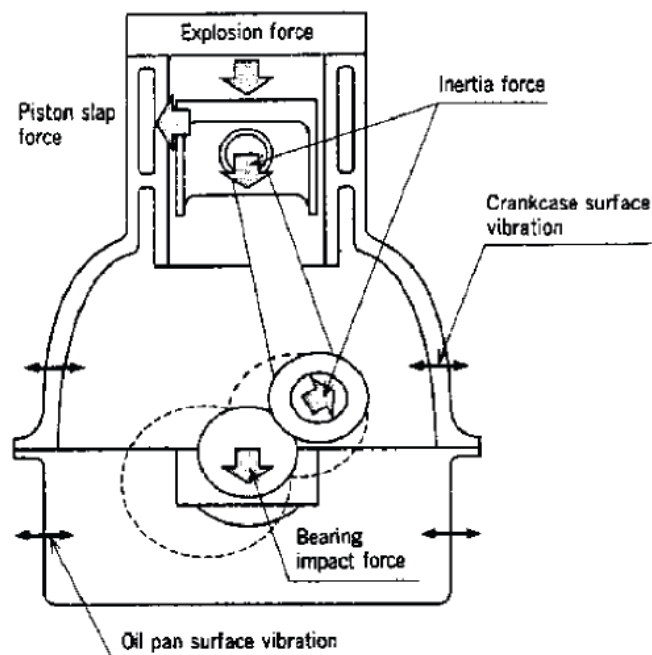


Fig 1: Unbalanced Mechanical and Combustion Forces of the Engine

Idle and low speed comfort can also be influenced due to changes of the engine excitation and the transfer mechanisms. The main transfer paths for the vibrations are the engine mounts, wheel suspensions and components mounted on to the body. The excitation becomes more critical when the main firing orders coincide with the Eigen frequencies of different components. The rigid body modes of the power train normally occur at very low frequencies. Care need to be taken while designing the mounting system so that the highest mode of the power train is at least $\sqrt{2}$ times lower than the first firing frequency of the engine. The presence of power train modes in the operating frequencies leads to higher transmissibility during low speed operation. When an engine is fastened directly to its support frame, it has a direct path for the transmission of vibration and noise. When the engine is attached to its support by means of properly selected resilient isolators, the path of vibration and noise disturbances is broken.

The engine produces two types of disturbances. We have to identify these in order to position the mounts correctly and also to choose the right dynamic characteristics (stiffness etc.) of the isolators.

The first type consists of disturbances originating in torsional dynamic pulses caused by variations in Cylinder gas pressure. This is the firing frequency. The firing frequency can be calculated as follows:

With a 2 stroke engine

$$F_d = \frac{(RPM * No. of Cylinders)}{60}$$

With a 4 stroke engine

$$F_d = \frac{(RPM * No. of Cylinders)}{2 * 60}$$

The second type of disturbance consists of unbalanced forces caused by reciprocating pistons or rotating crankshaft and rod masses within the engine [1].

1.1.2 Fundamentals of vibration isolation

In general, the transmission of vibration can be thought of in terms of a source, which generates an excitation force or displacement, the path, through which the vibratory disturbance is transmitted, and a receiver. The objective of isolation is to minimize the transmission of vibratory disturbances from the source to the receiver. The single degree of freedom model, which defines the uniaxial behavior of a linear system consisting of a lumped mass and spring and dampening elements, is fundamental to the understanding of vibration isolation. The model for the undamped condition is shown in Fig 2.

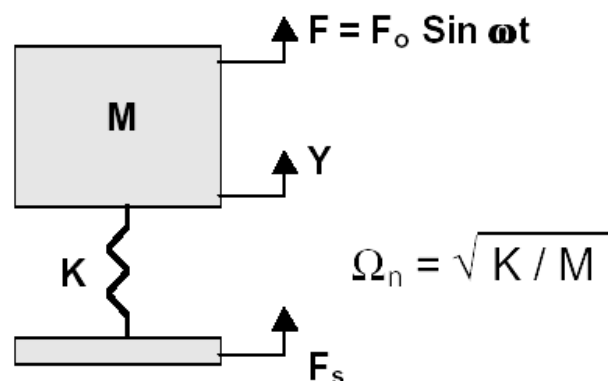


Fig 2: Undamped Single Degree of Freedom Model

The transmissibility through the spring is the ratio between the force applied on the mass and the force acting on the support.

$$T = \frac{1}{1 - \left(\frac{\omega^2}{\Omega_n^2}\right)}$$

When $\omega/\Omega_n = 1$, the transmissibility is infinite. When $\omega/\Omega_n = \sqrt{2}$, the transmissibility is unity, and as the excitation frequency increases, the transmitted force becomes less than the excitation force [1].

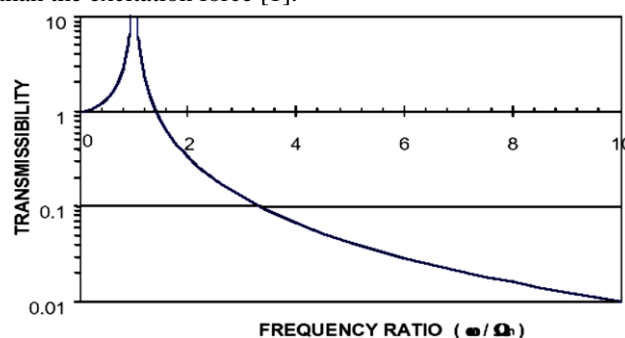


Fig 3: Transmissibility of an Undamped System

2. EXPERIMENTATION

To find out the stiffness of rubber mounts for vertical direction, they are tested on UTM.

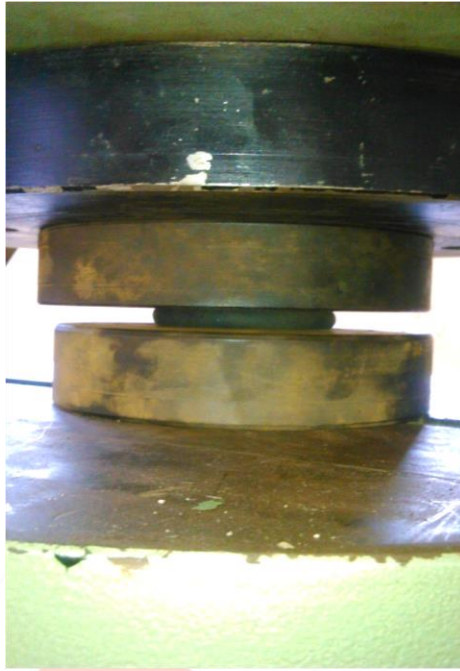


Fig. 4 Experimental Setup



Fig. 5 Rubber mount before testing



Fig. 6 Rubber mount after testing

2.1. Experimental Results

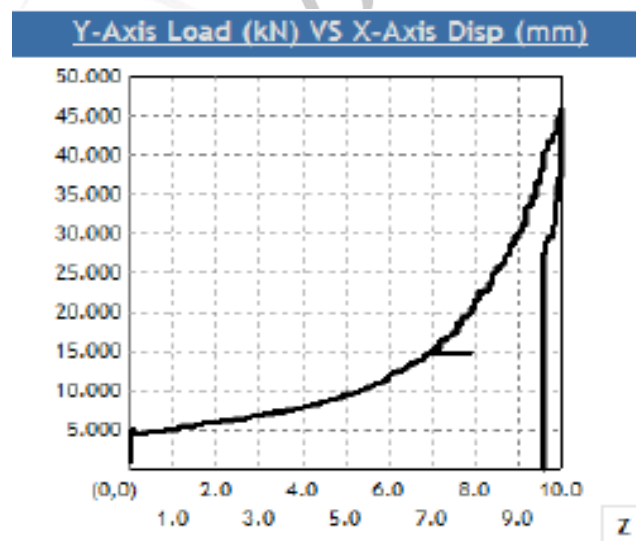


Fig. 7 Load vs. Displacement

TABLE.1 GRAPH RESULT

MAXIMUM LOAD (kN)	MAXIMUM DISPLACEMENT (mm)
45.92	10.0

$$\text{Stiffness} = \frac{\text{Load}}{\text{Deformation}}$$

The stiffness of rubber mount is **4592 N/mm**.

TABLE.2 STRESS-STRAIN DATA

STRESS	STRAIN
3.19	0.074
3.49	0.11
3.74	0.148
3.95	0.185
4.3	0.22
4.85	0.296
5.91	0.37
7.27	0.44
9.35	0.518
13.45	0.592
18.8	0.66
23.82	0.7
28.1	0.74
28.88	0.74

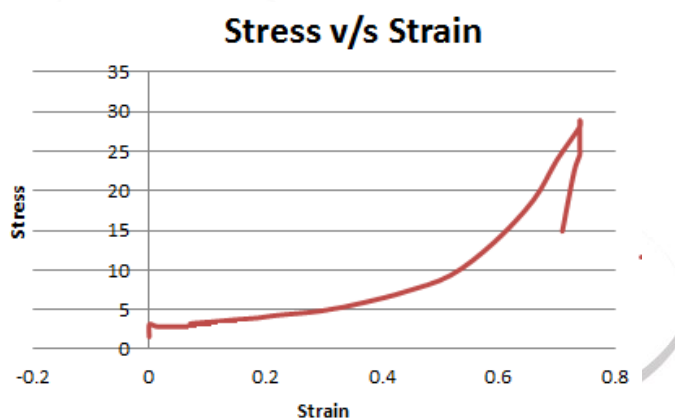


Fig.8 Stress vs. Strain

The Modulus of Elasticity (E) is **24.82 MPa**.

3. MODAL ANALYSIS OF ENGINE- FRAME ASSEMBLY WITH DERIVED RUBBER MOUNTS

For this analysis, the engine mount and frame of transmission compartment of BAJA ATV is considered for analysis. Analysis is done using the designed rubber mounts according to above Modulus of Elasticity.

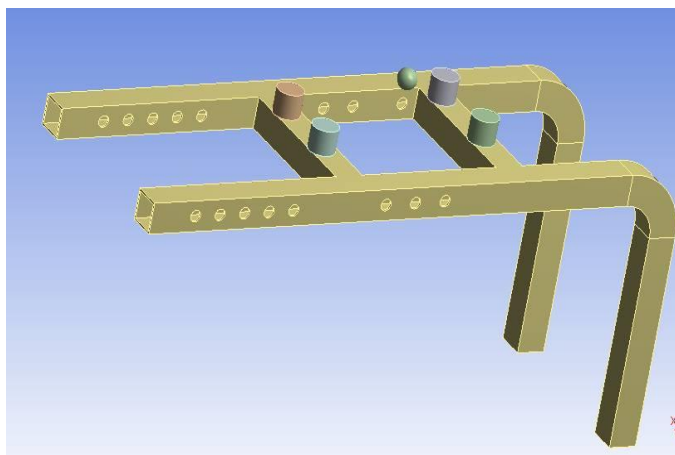


Fig.9 Engine mount frame modal

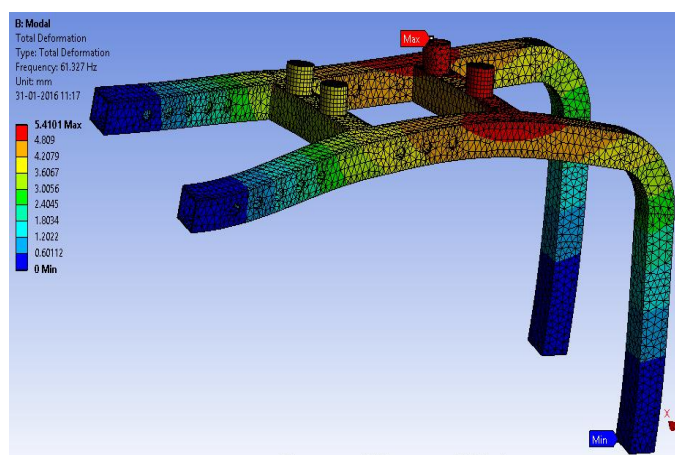


Fig.10 Modal analysis

The excitation frequency of engine mount system is found to be 61.327 Hz. The engine use for this vehicle is single cylinder engine having idling rpm of 1800 rpm.

Therefore, firing frequency of engine = 15 Hz. Hence, Transmissibility= -0.006

4. SEAT LOCATION OF BAJA ATV CONSIDERING NVH



Fig.11 CAD Model of ATV

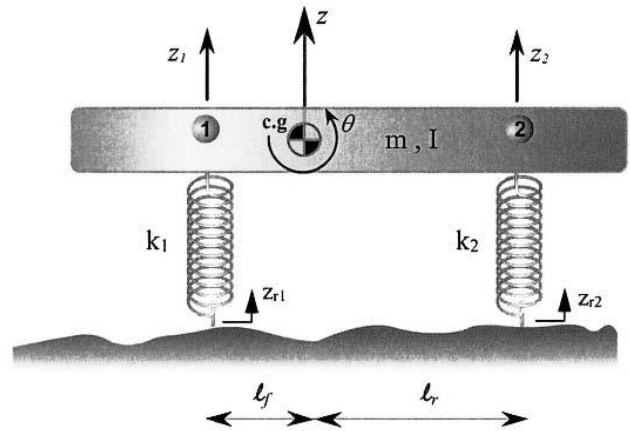


Fig.12 Vehicle Sprung Model

Where,

I = moment of inertia of vehicle about wheel base axis

K_1 = Stiffness of rear spring

K_2 = Stiffness of front spring

L_f = Distance of c.g. from front spring

L_r = Distance of c.g. from rear spring

At a distance of l_{01} from c.g. towards 2, the vehicle vibrates about this point when bump comes on a rear wheel and is termed as a nodal point.

At a distance of l_{02} from c.g. towards 1, the vehicle vibrates about this point when bump comes on front wheels and is termed as second nodal point.

AS TO MAKE THE DRIVER FEEL LESS TORQUE, WE HAVE TO TAKE THESE POINTS AT THE CENTRE OF HINGED POINTS OF SEAT LOCATION.

FOLLOWING FORMULAS ARE USED TO DETERMINE THE SEAT LOCATION:

$$l_{01} = D_2 / w_{n1}^2 - D_1$$

$$l_{02} = D_2 / w_{n2}^2 - D_1$$

$$w_{n1}^2 = \frac{D_1 + D_3}{2} - \sqrt{\frac{(D_1 - D_3)^2}{4} + \frac{D_2^2}{r_y^2}}$$

$$w_{n2}^2 = \frac{D_1 + D_3}{2} + \sqrt{\frac{(D_1 - D_3)^2}{4} + \frac{D_2^2}{r_y^2}}$$

$$D_1 = (K_f + k_r) / m_s$$

$$D_2 = (l_f k_f - l_r k_r) / m_s$$

$$D_3 = (l_f k_f + l_r k_r) / m_s$$

If $l_{01} < 0$; then oscillation centre will be located to the right of the centre of gravity and converse if vice-versa.

Specifications of ATV needed for calculation:

$$M_s = 180 \text{ kg};$$

$$K_f = 12.658 \text{ N/mm};$$

$$K_r = 23.98 \text{ N/mm}$$

$$l_f = 897 \text{ mm}$$

$$l_r = 261.29 \text{ mm}$$

$$D_1 = 203.5 \text{ s}^{-2}$$

$$D_2 = 28.25 \text{ ms}^{-2}$$

$$D_3 = 47.3 \text{ s}^{-2}$$

$$w_{n1}^2 = 47.3$$

Similarly, $W_{n2}^2 = 207.49$

$$L_{01} = -18.1 \text{ cm}$$

i.e. l_{01} is toward right to c.g. which means it is at a distance of 18.1 cm towards the front of the vehicle.

Thus, distance between front suspension mount and seat = $897 - 181 = 716 \text{ mm} = 71.6 \text{ cm}$

$$L_{02} = 708.02 \text{ cm}$$

which gives seat location beyond wheel base axis which means that this vibration cannot be completely damped.

5. CONCLUSION

- As transmissibility is negative, indicates the reaction force is out of phase with the excitation and thus the resonance will never occur.
- The single seat which we have in ATV is recommended to be placed at a distance of 71.6 cm from front suspension mount.
- Practically, we have achieved the seat location at a distance of 72.3 cm from front suspension mount giving an error of 0.9%.

REFERENCES

- [1] S.S.Sane, Vyankatesh Madane, Gaurav Upadhyay, "Engine Mount Analysis Methodology", 2011
 [2] S.S. Rattan, "Theory of Machines", Pg. 644

