Study of Engine Mounting Bracket

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Abstract: ***In a development of a vehicle mounting of an engine within the desired space that can hold engine weight, vehicle safety, vibration while working, bumpy roads, vibration oscillation, and force during acceleration-deceleration of speed make a huge impact on the structure of a vehicle and occupant's safety. This study trying to find the recent work on the mounting bracket of the engine and analytically designing the material used weight reduction analysis of frequency vibration and noise reduction.***

Keywords: Engine mounting bracket, vibration, frequency, weight reduction

1. Introduction

The automobile engine is the most important part of a vehicle, the development of an engine add more forces on the static part of the vehicle an engine working condition or static condition also impact on a different structural body part. Internal combustion induced an ample amount of excitation of engine structure itself, shafting structure and supporting structure. Structural vibration is a result of dynamic properties and the Excitation force. A designer should consider all parameters during the design phase. A natural frequency does not reach to resonance condition. Engine disturbance subjected to the engine itself, road condition and wheel.

Changing material and cross-section of a part to square helps in reducing the weight that can withstand high stresses and modal analysis of engine bracket to determine whether the design has a natural frequency lower than the excitation frequency of the engine bracket. [1][7].vibration and fatigue of engine brackets have been a concern which leads to structural failure [2].modeling through reverse engineering for the optimized design for weight and cost reduction [3][5].Study the reliability of a mounting bracket and stress distribution [4] and the process optimizing the damageability nonlinear effect on the dynamic behavior of the engine mount[6]. The Noise & vibration harness analysis (NVH) is an important consideration of the high-performance vehicle (FSAE) on varying stress and durability calculations to ensure engine safety [8].

2. Characteristics of engine mounting bracket:

The mounting bracket should have some basic characteristics that qualify the working criteria like as follows:

- Weight should be less,
- the cost factor should be in consideration,
- the frequency within a desired range (to prevent the system to bounce)
- Low frequency leads to large amplitude and higher frequency leads to the low amplitude of displacement.

3. Types of Engine Bracket

The mounting bracket may vary according to purpose and type of vehicle some are as follows:

3.1. The mounting bracket of an engine used in car

It made of a cast iron and made by press forging. It joins the engine with a large face and connected to a vehicle structure on a smaller end to take the load to absorb shocks and vibrations. Its life cycle will be more because of fewer vibrations and a low knocking rate of the engine. But after continuous use, there are some problems related to structure that increase the chances of failure of the engine mounting bracket. The failure also occurs due to the cracking ad necking on a point where the stress value will be more and stress generated due to the increase in temperature and main reason due to uneven road conditions and vibration generated by the vehicle itself.

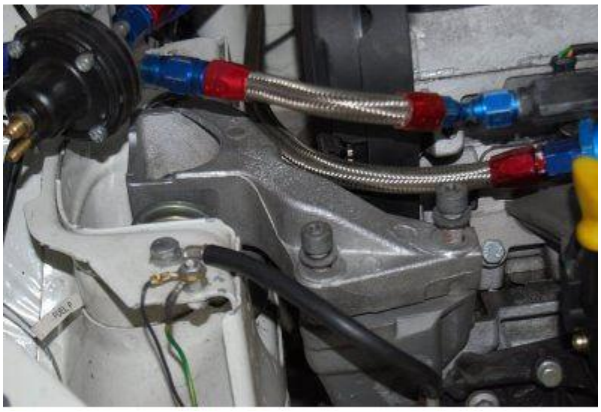

Figure 1: Engine mounting bracket of a car

3.2. The mounting bracket used in powertrain

Powertrain mounting bracket has a great impact on the noise, vibration and hardness characteristics of the vehicle. They are prone to failure since they have to withstand heavy dynamic loads of the powertrain in the operating conditions. To ensure that the design is foolproof, it is necessary to bring down the stress levels to a permissible limit [9].

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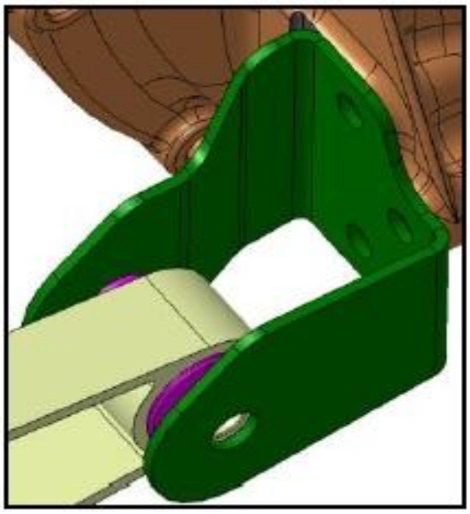

Figure 2: Powertrain Mounting Bracket

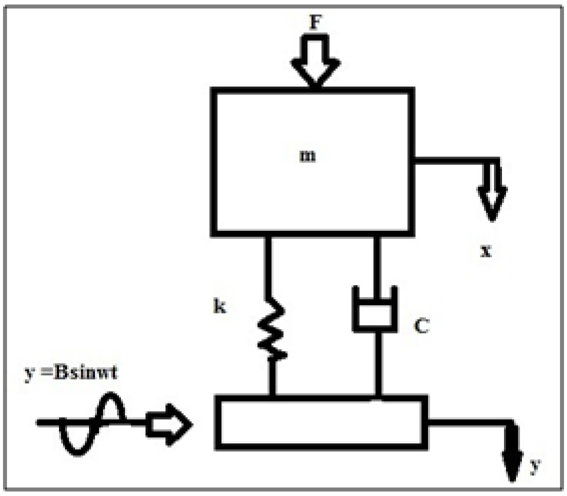

Figure 3: Excitation System

4. Vibrations in Mounting Bracket

4.1. To determine Nonlinear Vibration: (Analytical Method)

Consider the conservation system defined by the equation

$$U + f(x) = 0 \qquad (4.1)$$

Now acceleration,

$$U = V\,dv/dx \qquad (4.2)$$

Substituting equation 2 in equation 1 we get

$$V dv = -f(x)\,dx$$

Where U = V, if x = X when V=0, its integral is

$$|V| = [dx/dt] = \sqrt{\int_x^X f(\lambda)\,d\lambda} \qquad (4.3)$$

The second integral yields

$$t - to = \int_0^x \left(\frac{dU}{\int_x^X f(\lambda)(d\lambda)} \right) \qquad (4.4)$$

Where t_0 is the corresponding to x=0 the equation 4 express time as a function of displacement and its inverse is the displacement time relationship.

If the motion is periodic with period T, the time corresponding to the motion from (x=0) ($t=t_0$) to the x=0 (V=0) represent a quarter period. Hence

$$T = 4\int_0^X \left(\frac{dU}{\int_x^X f(\lambda)\,d\lambda} \right) \qquad (4.5)$$

Thus the period becomes a function of amplitude X [6].

4.2. Support Motion

At the same time, there is relative motion between the wheels and the chassis. So the chassis having motion relative to the wheels and the wheels are having motion relative to the road surface. The amplitude of vibration depends upon the speed of the vehicle and the nature of the road surface. The vibration measuring instruments are designed on the support motion approach. Such that system is subjected to have a single degree of freedom (Shown in fig. 3) for the simplicity of mathematical expression.

In a vibratory system where the support is put to excitation absolute and relative motion become important.

4.2.1. Absolute Motion

Absolute motion of the mass means its motion with respect to the coordinate system attached to the earth. The absolute displacement of support is $y = B\sin\omega t$ and for absolute displacement of the mass m from its equilibrium portion is x. The displacement of mass m relative to the support is z. The net elongation of the spring is (x-y) and the relative motion between the two ends of the damper is$(\dot{x} - \dot{y})$. Then $z = x - y$ and $\dot{z} = \dot{x} - \dot{y}$.

The equation of a motion can be written as

$$m\ddot{x} + c(\dot{x} - \dot{y}) + k(x - y) = 0$$
$$m\ddot{x} + c\dot{x} + kx = c\dot{y} + ky \qquad (2.1.1)$$

The support is subjected to harmonic vibration, $y = B\sin\omega t$ substituting this value of y in equation (2.1.2) we get

$$m\ddot{x} + c\dot{x} + kx = cB\omega\cos\omega t + kB\sin\omega t$$
$$= B[k\sin\omega t + c\omega\cos\omega t]$$
$$=B\sqrt{(k^2+c^2\omega^2)}\left[\frac{k}{\sqrt{k^2+c^2\omega^2}}\sin\omega t+\frac{c\omega}{\sqrt{k^2+c^2\omega^2}}\cos\omega t\right]$$
$$= B\sqrt{k^2 + c^2\omega^2}[\cos\alpha\sin\omega t + \sin\alpha\cos\omega t]$$
$$m\ddot{x} + c\dot{x} + kx = B\sqrt{k^2 + c^2\omega^2}\sin(\omega t + \alpha) \qquad (2.1.3)$$

$$\text{And } \tan\alpha = \frac{c\omega}{k}$$

$$\text{Where } \alpha = \tan^{-1}\frac{c\omega}{k} = \tan^{-1}(2\varepsilon\omega/_{\omega_n}) \qquad (2.1.4)$$

The equation of motion for steady state system behavior at time interval's'.

The steady state amplitude written as

$$\frac{A}{B} = \frac{\sqrt{1 + (2\varepsilon\,\omega/\omega_n)^2}}{\sqrt{[1 - (\omega/\omega_n)]^2 + [2\varepsilon\,\omega/\omega_n]^2}} \qquad (2.1.5)$$

The ratio of A/B is the displacement transmissibility which is the ratio of the body to the amplitude of the support. We conclude from equation the frequency ratio $\omega/\omega_n = \sqrt{2}$ for all values of damping the amplitude ratio (A/B) is unity. For high values of frequency ratio ($\omega/\omega_n \gg \sqrt{2}$), the amplitude ratio is quite small which means that the support displacement is almost negligible.

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4.2.2. Relative motion

We have assumed that z is the relative displacement of the mass with respect to the support. So the equation

$$m(\ddot{z}+\ddot{y})+c\dot{z}+kz=0$$

Let us assume that steady state relative amplitude Z lags the excitation by angle α, so

$$z=Z\sin(\omega t-\alpha)$$

Z/B can be express as

$$\frac{Z}{B}=\frac{(\omega/\omega_n)^2}{\sqrt{[1-(\omega/\omega_n)^2]^2+[2\varepsilon\,\omega/\omega_n]^2}} \qquad (2.2.1)$$

Conclusion are drawn from equation When (ω/ω_n)>3, the amplitude ratio $(\frac{Z}{B})$ is almost unity. It means that the relative amplitude Z and the support amplitude B are equal. For $(\frac{Z}{B})$ ratio being unity, the mass m will be having no displacement. Damping does not have any effect on $\frac{Z}{B}$ ratio for high values of (ω/ω_n) i.e., for $(\omega/\omega_n>3)$.
Hence the same number of the natural frequency. But on practical lower natural frequency important [10].

4.3. Vibration isolation

The high speed engines and machines when mounted on foundations and supports cause vibrations of excessive amplitude because of unbalanced forces setup to damage the foundation on which the machines are mounted. The isolation is expressed in terms of force and motion. Lesser the amount of force or motion transmitted to the foundation greater is set to be isolation.

5. Conclusion

On the study of a mounting bracket, we analysis that the vibration due to engine and its motion (absolute and relative motion) of a vehiclemakes an impact on structural part of a vehicle that results vibration isolation and the changing in a material property of mounting bracket and improving design gives the more feasible result on improving noise reduction and natural frequency of mounting bracket.

6. Notation

f(x)	function of x
z	support
U	acceleration
v	Velocity
ω	Angular frequency
ω_n	Natural frequency
α	Excitation angle
T	Time interval
t_0	Time interval at 0
x, y	Displacement
X	Amplitude of vibration
λ	Change in amplitude
M	Mass
k	stiffness
C	Damping
c	Damping coefficient
c_c	Critical damping
ε	Damping ratio c/c_c
F	Force on a mass m
A, B	Amplitude of vibration
Z	Relative amplitude
A/B	Transmissibility ratio
Z/B	Amplitude ratio

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