Engine Mounting Characteristic for Vibration Isolation and Active Vibration Control Strategies

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Abstract: With the trend of lighter vehicle and more powerful engine (power plant) passengers are subjected to large vibration and harshness coming from the engine through the chassis. This paper discusses the application of active vibration control to reduce this vibration effect. Using a mass-spring-damper model which represents a passive engine mount, dynamic characteristic of the mount is obtained for both lower frequency level (< 20 Hz) and higher frequency level (> 20 Hz.). Based on this characteristic an ideal vibration isolation model is deduced and it has been found that it is frequency dependent. Feedback control system strategy is presented and simulated using MatLab to see the amplitude comparison between passive engine mount and active engine mount. The active vibration control strategies show good improvement in reducing the vibration effect from the power plant to the chassis of the vehicle and thus to the passenger.

Keyword: Engine mounting, active vibration control, feedback control

1) Introduction

Nowadays vehicles are made lighter in terms of structure and higher power output from the engine. These requirements most of the time has a negative effect to the comfort of the passenger since vibration and noise level will increase [1]. The high level of vibration is mainly caused from input disturbances from the road condition which occur at a frequency level less than 20 Hz and secondly the disturbance from the engine which occur at a frequency higher than 20 Hz (i.e. between 20 Hz and 200 Hz). At these two different levels of frequencies the engine mount has to be stiff at the lower range of frequency to hold the engine from bouncing off the chassis and soft at the other end of the frequency level to isolate the vibration produced by the engine [2]. It is therefore necessary to introduce active engine mounting system that can perform the above said characteristic [3].

2) Engine Mounting Characteristic

The force disturbance generated to the chassis of the vehicle can be classified in to two types. The first type is the disturbance created from the road conditions e.g. pot holes, road bumps, road irregularities etc. This type of disturbance normally occurs within the frequency less than 20 Hz. The second type of disturbance is the force transmitted from the engine to the chassis which occur at a frequency higher than 20 Hz. The engine mounting system can be presented by a spring, mass damper model. Fig. 1 shows the system for a single mount.

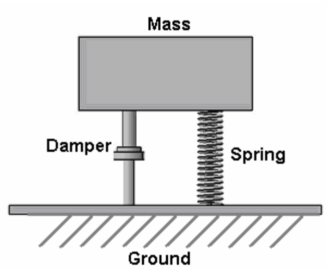

Fig. 1. Representation of an engine mounting system

The mass represents the engine of the vehicle and the ground represents the chassis of the vehicle whereas the spring and damper represent the engine mount.

At the lower frequency range i.e. less than 20 Hz, the system can be represented by a base excited single degree of freedom system as shown in Fig. 2. At this frequency level the engine mount should be as stiff as possible to minimize any relative displacement between the chassis and the engine in order to prevent any damage that can occur to the engine.

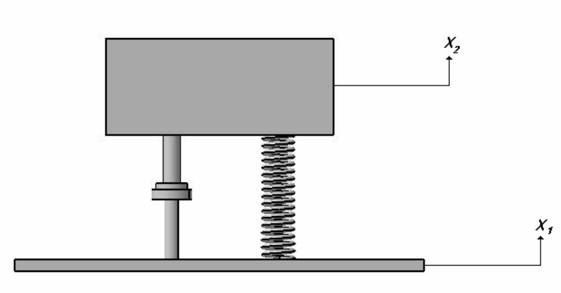

Fig. 2. SDOF base excited representation

Applying Newton's equation of motion to the system results the following Eqn. (1).

$$m\ddot{x}_2 + c(\dot{x}_2 - \dot{x}_1) + k(x_2 - x_1) = 0 \quad (1)$$

where m is the mass of the engine, $\ddot{x}_2$ is the acceleration, $\dot{x}_1$ is the velocity of the chassis, $\dot{x}_2$ is the velocity of the mass, x_1, is the displacement of the chassis which is also the input displacement and x_2, is the displacement of the engine which is the output displacement. Applying Laplace transform to Eqn. (1) and rearranging the equation results in Eqn. (2) which is the transfer function of the system that relates the input displacement, x_1, and the output displacement, x_2.

$$\frac{x_2(s)}{x_1(s)} = \frac{2\varsigma\omega_n s + \omega_n^2}{s^2 + 2\varsigma\omega_n s + \omega_n^2} \quad (2)$$

where $x_1(s)$ is the chassis displacement in the S domain, $x_2(s)$ is the engine displacement in the S domain.

$$\omega_n^2 = k/m \quad (3)$$

Eqn. (3) is also known as the natural frequency of the system and

$$\varsigma = c/2\sqrt{km} \quad (4)$$

Eqn. (4) is the damping ratio. Simulating Eqn. (2) using MatLab gives the following results as in Fig. 3.

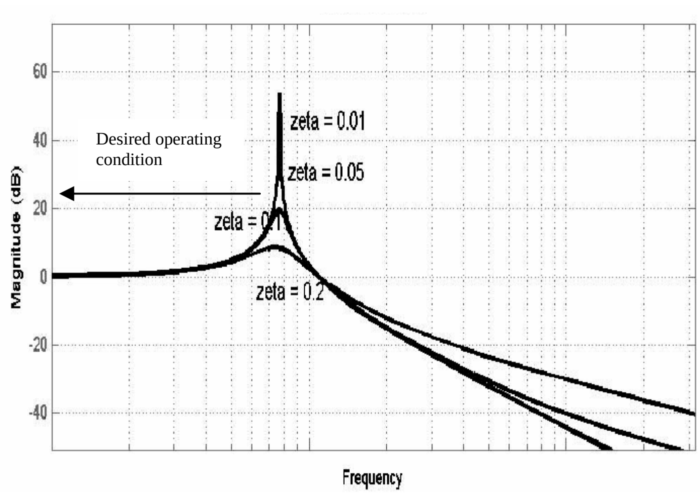

Fig. 3. SDOF response for various values of ζ

The figure shows that at the lower frequency level the relative displacement between the engine and the chassis is zero which is the desired operating condition. By increasing the value of the stiffness, k the following figure shows the results.

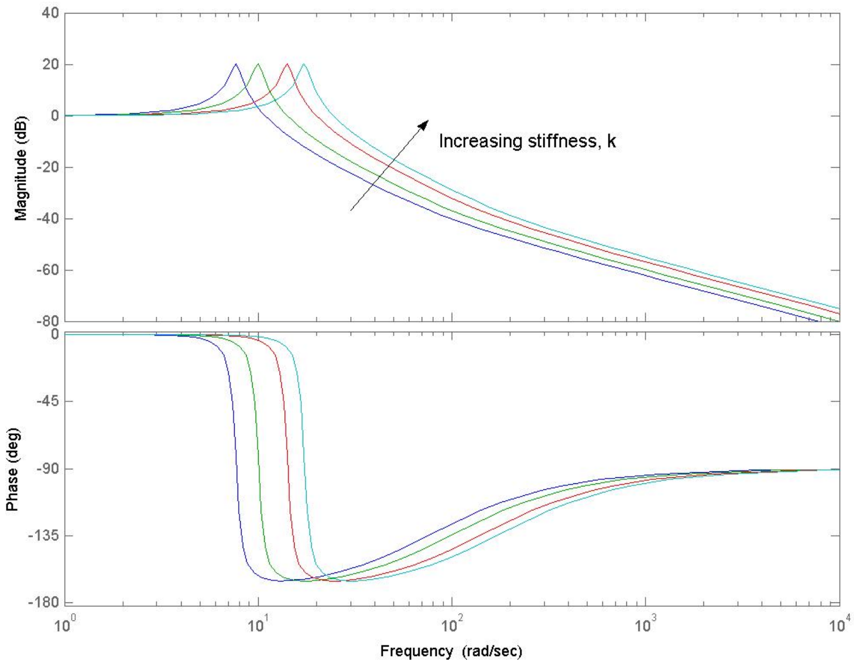

Fig. 4. Bode plot of a SDOF response for an increasing value of stiffness, k

Fig. 4 shows by increasing the stiffness of the mount improves the system relative displacement at higher frequency level and thus preventing the engine from being damaged.

At a frequency larger than 20 Hz vibration isolation is desired. Which means that the force exerted from the engine needs to be isolated from the chassis of the vehicle to prevent discomfort to the passenger. A single degree of freedom representation of the system can be described as in Fig. 5.

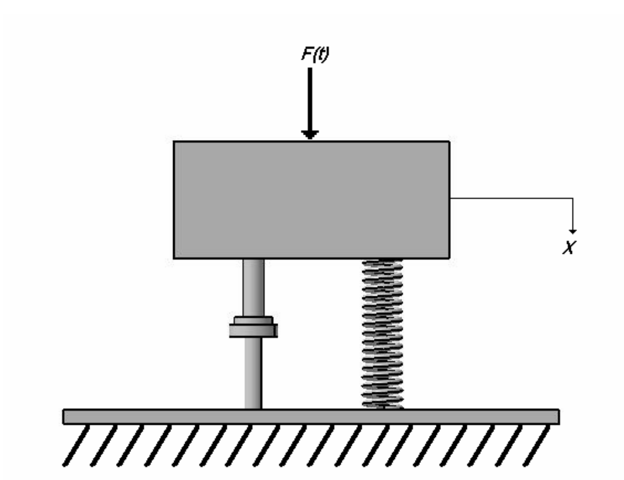

Fig. 5. A SDOF of a Force induced excitation

Applying Newton's law of motion to both the engine and the chassis gives the following two equations.

$$F(t) = m\ddot{x}(t) + c\dot{x}(t) + kx(t) \tag{5}$$

$$F_T(t) = c\dot{x}(t) + kx(t) \tag{6}$$

Where $F(t)$ and $F_T(t)$ are the disturbance force form the engine and the transmitted force respectively. Again applying Laplace transform to Eqn. (3) and (4) and rearrange the equations gives the following Force Transmisibility equation.

$$\frac{F_T(s)}{F(s)} = \frac{2\varsigma\omega_n s + \omega_n^2}{s^2 + 2\varsigma\omega_n s + \omega_n^2} \qquad (7)$$

where in Eqn. (7) $F_T(s)$ is the force transmitted from the engine through the damper and the spring and $F(s)$ is the disturbance from the engine. Simulating Eqn. (7) in Matlab gives the following results.

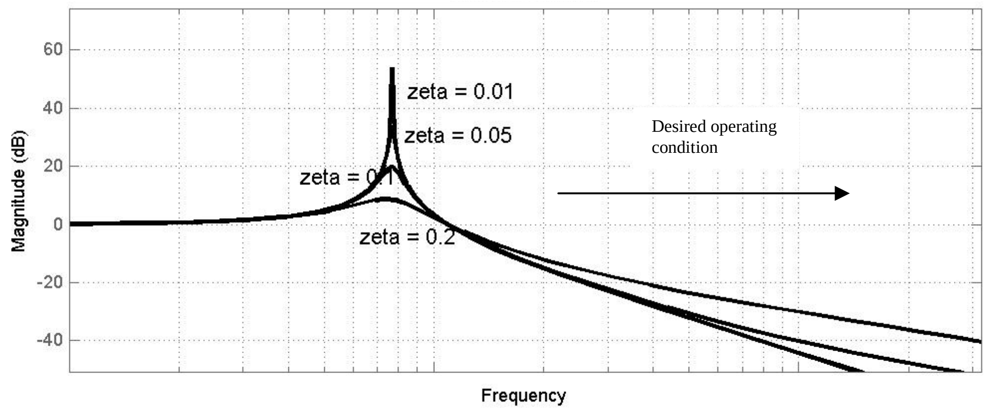

Fig. 6. Response of a SDOF force excited system

From Fig. 6 it shows that as frequency increases, after the resonance frequency, vibration isolation is improved which results to the desired operating condition. Fig. 7 shows the response of the system as the value of the stiffness k is decreased.

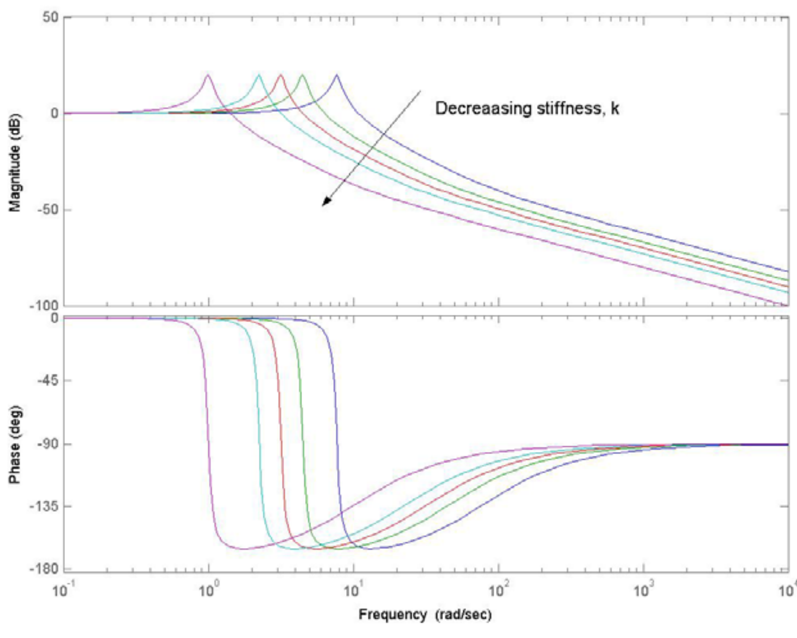

Fig. 7. Bode plot of a SDOF response for a decreasing value of stiffness, k

From Figure 4 and Figure 7 thus it can be concluded that the characteristic of an ideal engine mounting system should be the one depicted in Fig. 8 [4].

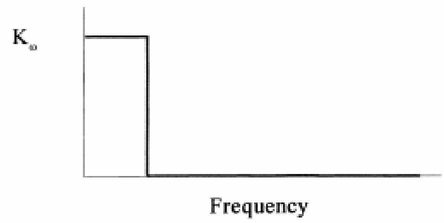

Fig. 8. Dynamic characteristic of an ideal engine mount

Fig. 8 shows the contradictory characteristic of an engine mount (high stiffness and low stiffness) that cannot be achieved by using a passive mount and thus active engine mount has to be introduced.

3) Active Engine Mount

Active engine mount system consists of a passive mount (elastomeric or hydraulic), a control system and a force actuator which can actively apply a force. It is assumed at the lower frequency level (i.e. less than 20 Hz) no active force actuation is needed in order to prevent the engine from bouncing off the chassis since the stiffness of the passive mount is good enough to prevent this from happening. Therefore the force actuator will only be activated at the higher frequency level i.e. larger than 20 Hz to cancel the disturbance force generated from the engine and this is called active vibration isolation.

The main focus of this paper is to present the control system algorithms and strategies to establish the active engine mount system. In principal there are two methods to in creating the control system i.e. feedback control and feedforward control system. Feedback control system is chosen since the disturbance forces from the engine are unpredictable and possibly random [5].

4) Methodology

It is identified that there are three different feedback control strategies to achieve active vibration isolation. The three strategies are obtained by applying the actuated force to the 1) engine, 2) chassis or 3) both engine and chassis [6].

4.1) Applying actuated force to the engine

Fig. 9 shows the schematic diagram when the actuated force is applied only to the engine where *g* is the proportional gain of the controller.

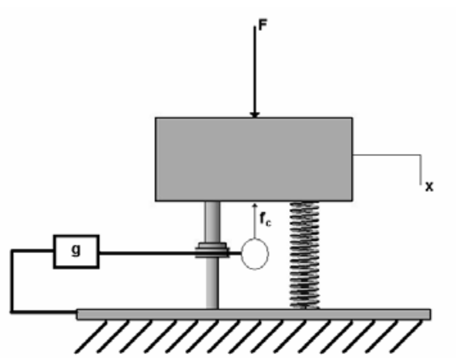

Fig. 9. Schematic diagram showing the application of actuated force to the engine

Applying Newton's second law of motion to both the engine and the chassis gives the following equations.

$$m\ddot{x} + c\dot{x} + kx + F_c = F \qquad (8)$$

$$c\dot{x} + kx = F_t \qquad (9)$$

where F is the disturbance force, F_t is the transmitted force to the chassis through the damper and spring and $F_c = gF_t$. Apply Laplace transform to Eqn. (8) and Eqn. (9) and by rearranging the equations will obtain the transfer function as in Eqn. (10).

$$\frac{F_T(s)}{F(s)} = \frac{2\varsigma\omega_n s + \omega_n^{\ 2}}{s^2 + \left(2\varsigma\omega_n s + \omega_n^{\ 2}\right)\left(1+g\right)} \tag{10}$$

which is the relationship between the transmitted force and the disturbance force.

4.2) Applying actuated force to the chassis

Fig. 10 shows the schematic diagram when the actuated force is applied only to the chassis.

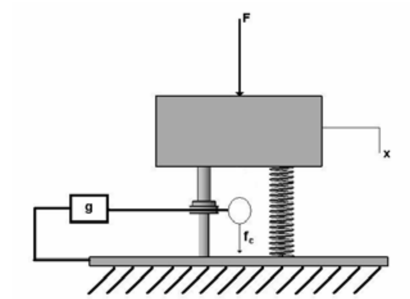

Fig. 10. Schematic diagram showing the application of actuated force to the chassis

Again applying equation of motion to both the engine and the chassis and finally apply Laplace transform gives the following Eqn. (11):-

$$\frac{F_T(s)}{F(s)} = \frac{2\varsigma\omega_n s + \omega_n^{\ 2}}{\left(s^2 + 2\varsigma\omega_n s + \omega_n^{\ 2}\right)\left(1+g\right)} \tag{11}$$

4.3) Applying actuated force to the engine and chassis

When applying the actuated force to isolate vibration from the engine to the chassis the following strategy as in Fig. 11 is applied.

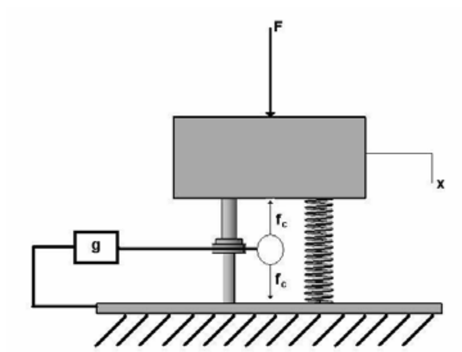

Fig. 11. Schematic diagram showing the application of actuated force to the engine and chassis

Applying Newton's equation of motion to the engine and chassis and apply Laplace transform gives the following transfer function:-

$$\frac{F_T(s)}{F(s)} = \frac{2\varsigma\omega_n s + \omega_n^{\ 2}}{\left(1-g\right)s^2 + 2\varsigma\omega_n s + \omega_n^{\ 2}} \tag{12}$$

5) Results and Discussions

Simulating Eqn. (10) gives the following results:-

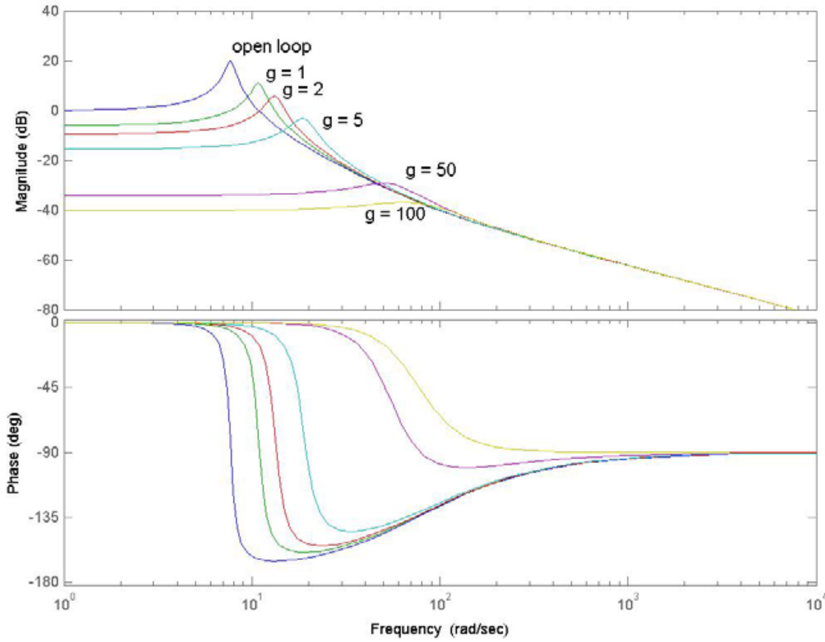

Fig. 12. Bode plot of the frequency response when actuated force is applied to the engine

Fig. 12 shows the frequency response of the system when the actuated force is applied only to the engine. The open loop is the response of the passive system i.e. no active force actuation is applied. From the figure it shows that as the proportional gain g is increased there is an improvement towards the vibration isolation at the lower and higher frequency range with a shift in the natural frequency and damping ratio.

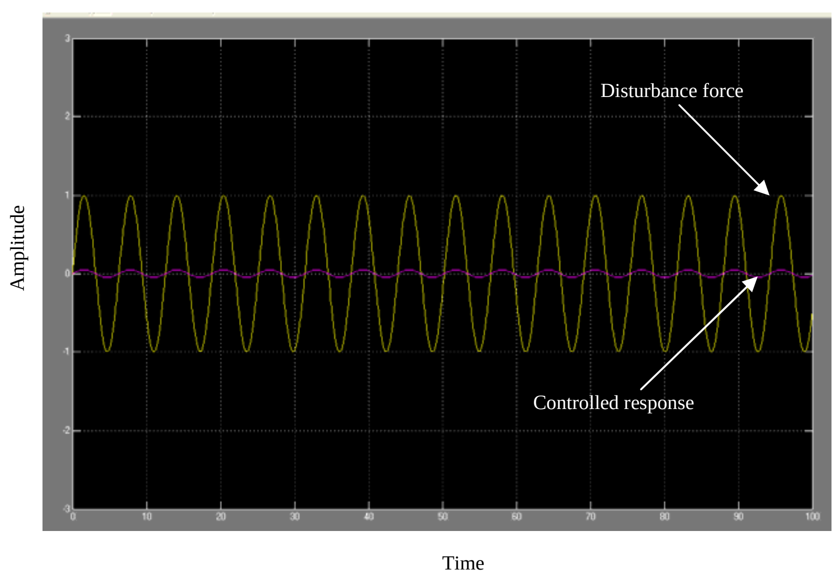

Fig. 13. Time response of an active system when actuated force is applied to the engine

Fig. 13 shows the time response of the system when the proportional gain is equal to 20. It shows that the when the actuated force is applied to the engine the amplitude of the transmitted force is reduced as compared to the disturbance force.

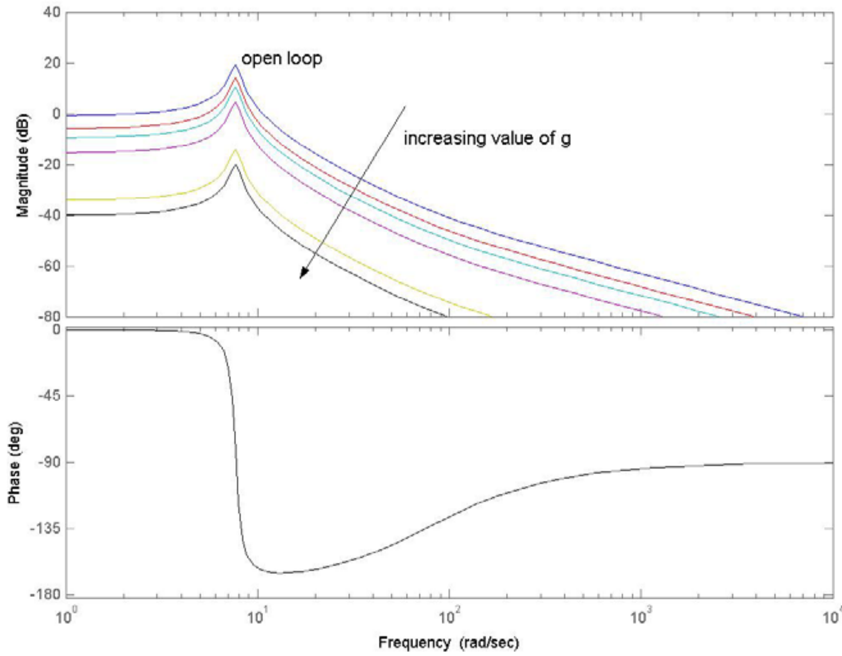

Fig. 14. Bode plot of the frequency response when actuated force is applied to the chassis

Fig. 14. shows the frequency response of the system when the actuated force is applied to the chassis. It shows from the figure that as the proportional gain of the control system *g* increased vibration isolation improves before and after resonance frequency without any shift on the natural frequency and an increased in the damping ratio. Fig. 15 shows the time response of the system when g = 20 which shows the reduction of the transmitted force to the chassis and thus improving vibration isolation.

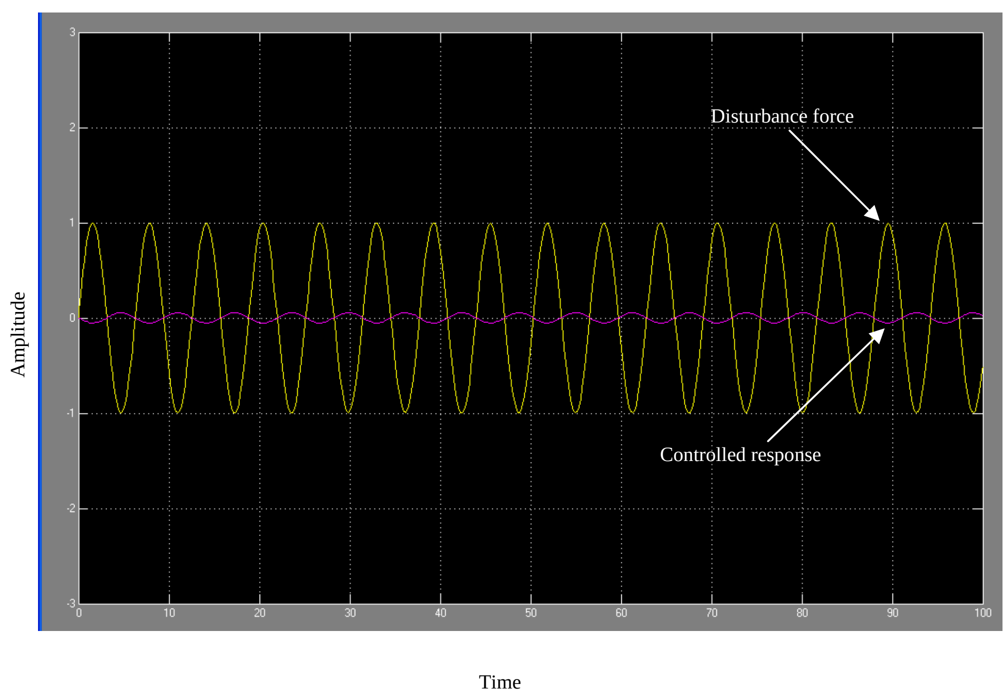

Fig. 15. Time response of an active system when actuated force is applied to the chassis

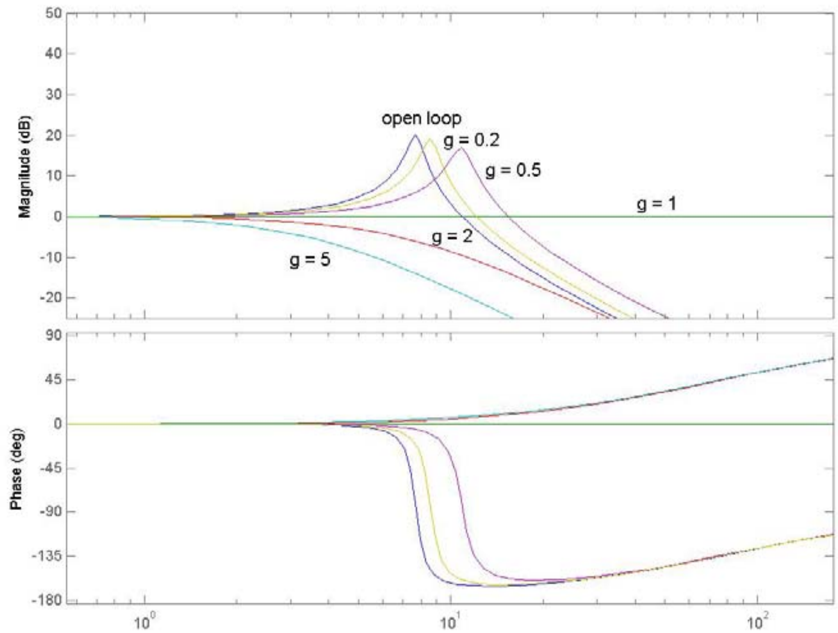

Fig. 16. Bode plot of the frequency response when actuated force is applied to the engine and chassis

Fig. 16 shows the frequency response of the system when the actuated force is applied to both the engine and the chassis. It interesting to note that the system will become unstable when the proportional gain *g* is larger than 1. This is due to the first term in the denominator of Eqn. (12) that will cause instability when the term $(1- g) < 0$.

6) Conclusion

It can be concluded that an engine mount will need to serve two purposes that is to prevent the engine from bouncing off the chassis at low frequencies and to isolate the vibration developed by the engine to the chassis at high frequencies. This requires the mount to have a larger stiffness at frequency lower than 20 Hz and lower stiffness at frequency larger than 20 Hz. Due to this contradictory characteristic active engine mounting system is necessary as the new generation engine mounting system. Feedback control strategy is suitable for active vibration isolation in the automotive field since disturbance forces are unpredictable and random. From the discussion it can also be concluded that from the three feedback control strategies applying the actuated force to the engine or the chassis will improve vibration isolation substantially when proportional gain of the control system is increased. However, when the actuated force is applied to the engine and chassis instability might occur in the control system.

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