1417. A numerical investigation on active engine mounting systems and its optimization

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Abstract. In this paper, based on the previous research experiences in the lumped parameter modeling and study of active control mounts (ACM) model, an analytical model of active ACM in powertrain is developed and implemented in MATLAB. In order to validate this newly developed model in this work, a finite element analysis (FEA) method is conducted in ANSYS and the results of FEA is compared with analytical model for validation. After the validation, the control strategy is integrated into the analytical model by using the linear quadratic regulator (LQR) method. Numerical results show a good control performance. Furthermore, this work examines the application of genetic algorithms (GA) in optimizing the weight matrices of LQR. An optimal configuration is obtained and thus this approach could help the practical design of ACM systems.

Keywords: test bench model, finite element analysis, linear quadratic regulator, genetic algorithm, optimization.

1. Introduction

During the last decade, the noise, vibration and harshness (NVH) has received attention in several publications. As a result, engine mounts are becoming more important as being not only engine vibration isolation but also a part of engine support. One of the main functions of the automotive vehicle engine mounting system is to support the engine body and provide comfort ride for passengers by reducing vibration caused by engine excitation. There is much literature in which extensive investigations have been conducted on different kinds of engine mounting systems, from elastomeric to hydraulic mounts, from passive to semi-active and active mounts.

Although elastomeric mounts have been successfully used for automotive industry for many years, but the conventional elastomeric mounts do not meet all the requirements, it just can only provide a small dampingand a solution between static deflection and vibration isolation [1]. Hydraulic engine mounts (HEM) can offer a better performance than traditional elastomeric mount in the low frequency. Hydraulic mounts have been promising alternatives to conventional elastomeric mounts because of their ability to create frequency-dependent damping.

Hydraulic mounts have been used in the automotive industry since 1985 in General Motors (GM). Today, almost all passenger cars are installed hydraulic mounts. The basic idea of the hydraulic mounts is to use highly elastic rubber for vibration isolation and to use a hydraulic device (inertia track and decoupler) to generate the large damping at a constant frequency for vibration control. The lumped parameter of a floating-decoupler type hydraulic mount (the most common) is as illustrated in Fig. 1, and consists of two fluid-filled chambers separated by a metallic plate containing the decoupler and inertia track. K_r and B_r are the equivalent stiffness and damping of the rubber spring. The volumetric compliance of the upper chamber and lower chamber are modeled as C_1 and C_2. Also, the pressures in the upper and the lower chambers are captured by P_1 and P_2. A_p is the effective piston area, and the flow through the inertia track Q_i and the decoupler Q_d. I_i, R_i representing the effective inertia and resistance of inertial track. Similarly, effective inertia and resistance of decoupler are I_d and R_d.

During low-frequency high-amplitudes vibrations, the ideal mount should exhibit large stiffness and damping characteristics to reduce relative displacement transmissibility whereas for high-frequency low-amplitude vibrations the ideal mount should have low stiffness and damping [2]. For that the characteristics of passive elastomeric mount or semi-active mount can hardly meet the requirement of broad frequency band of engine vibration and noise reduction. One of the effective methods to reach ideal vibration isolation is using active control engine mounts (ACM). A typical active control engine mount (ACM) consists of a passive hydraulic mount, an active actuator, a vibration sensor, and electronic controller. The structure of the ACM is illustrated in Fig. 2 (K_r and C_r are the equivalent stiffness and damping of the rubber spring. I_i, R_i representing the effective inertia and resistance of inertial track. Similarly, effective inertia and resistance of decoupler are I_d and R_d).

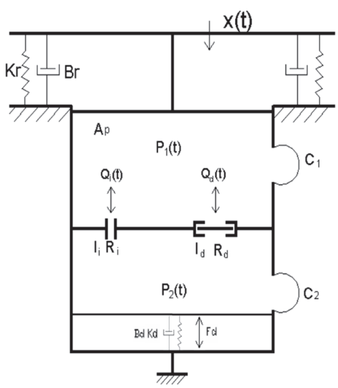

Fig. 1. Lumped parameter model of a floating-decoupler type hydraulic mount

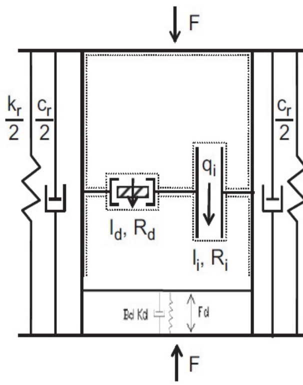

Fig. 2. Structure model of an active control mount

Fig. 2 shows thestructure model of the ACM, which is an amalgamation of the structure of the hydraulic mount and an actuator system. But the role of the decoupler is changed to a piston because of the actuator, so that it transmits the force from the actuator to the engine and chassis through the upper chamber. The role of the inertia track is also changed: in hydraulic engine mounts, the inertia track generates frequency dependent stiffness and damping. However, in the active control engine mount, it just relieves the static pressure in the upper chamber.

Because passive hydraulic mount has superior isolation ability in the low frequency range, and active actuator can provide highly efficient vibration control performance in relative high frequency range, so the active engine mount can isolate the vibration of engine in much wider frequency range. At the same time, the numerical simulation shows that the active control engine mount is capable of significantly reducing the vibration transmission.

2. Analytical modeling

The modeling of the active engine system is restricted to three degrees of freedom. However, note that the assumptions are made for this system [3].

1) The displacement is small compared to system dimensions.
2) The spring force is linear around the working point.
3) The upper plate (engine body) on the vibration isolation system is a rigid body.
4) Fluid is incompressible, and fluid density in chambers and fluid track is the same.
5) Mass of rubber spring, upper and lowerconnectors are negligible.
6) Damping of both lower diaphragm and decoupler membrane are negligible.
7) Leakage path tending to short-circuit fluid track is ignored.

These assumptions are used to model the vibration system in this work and Fig. 3 shows the coordinate system [3].

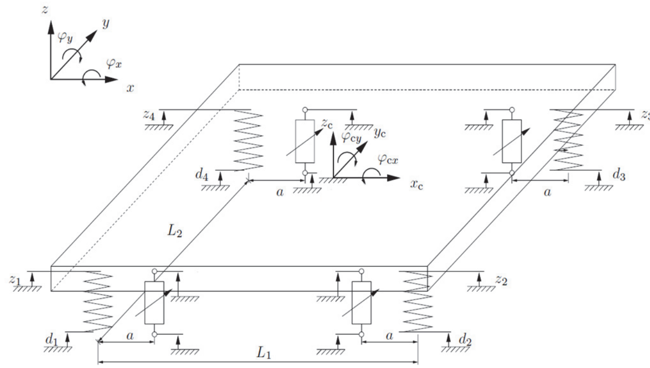

Fig. 3. Coordinate system [3]

The state space format for the equations of motion of vibration system is:

$$\dot{x} = Ax + Bx, \tag{1}$$
$$y = Cx + Du, \tag{2}$$

where matrix A, B, C and D are defined as:

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ -\dfrac{4(k+k_a)}{m} & -\dfrac{4D_a}{m} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & -\dfrac{L_2^2(k+k_a)}{J_{xx}} & -\dfrac{D_a L_2^2}{J_{xx}} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & 0 & \alpha & \beta \end{bmatrix},$$

$$B = \begin{bmatrix} 0 & 0 & 0 & 0 \\ \dfrac{1}{m} & \dfrac{1}{m} & \dfrac{1}{m} & \dfrac{1}{m} \\ 0 & 0 & 0 & 0 \\ -\dfrac{L_2}{2} & -\dfrac{L_2}{2} & \dfrac{L_2}{2} & \dfrac{L_2}{2} \\ 0 & 0 & 0 & 0 \\ -a+\dfrac{L_1}{2} & a-\dfrac{L_1}{2} & a-\dfrac{L_1}{2} & -a+\dfrac{L_1}{2} \end{bmatrix},$$

$$C = \begin{bmatrix} -\dfrac{4(k+k_a)}{m} & -\dfrac{4D_a}{m} & 0 & 0 & 0 & 0 \\ 0 & 0 & -\dfrac{L_2^2(k+k_a)}{J_{xx}} & -\dfrac{D_a L_2^2}{J_{xx}} & 0 & 0 \\ 0 & 0 & 0 & 0 & \alpha & \beta \end{bmatrix},$$

$$D = \begin{bmatrix} \dfrac{1}{m} & \dfrac{1}{m} & \dfrac{1}{m} & \dfrac{1}{m} \\ -\dfrac{L_2}{2} & -\dfrac{L_2}{2} & \dfrac{L_2}{2} & \dfrac{L_2}{2} \\ -a+\dfrac{L_1}{2} & a-\dfrac{L_1}{2} & a-\dfrac{L_1}{2} & -a+\dfrac{L_1}{2} \end{bmatrix},$$

$$\alpha = -\left(L_1^2 k + (-4aL_1 + 4a^2 + L_1^2)\frac{ka}{J_{yy}}\right),$$

$$\beta = -\frac{D_a(L_1^2 - 4aL_1 + 4a^2)}{J_{yy}},$$

where k_a is the actuator spring constant, D_a is the damping parameter.

3. Validation of the mathematical model

In order to validate this newly analytical model in this work, in this section, the analytical model is validated using a finite element model set up in ANSYS which is a widely accepted commercial FEM software [4]. A finite element analysis (FEA) method is conducted in ANSYS and compared with analytical model for validation.

Table 1 shows the modal frequency of the vibration system from both the finite element model in Fig. 4 and the mathematical model by state space method.

Table 1. Validation of the mathematical model by comparing to the finite element model in MATLAB and ANSYS

Modes	Frequency (Hz) Finite element model	Frequency (Hz) Mathematical model	Difference (%)
The first modal	6.84	6.84	0
The second modal	11.69	11.84	1.3
The third modal	11.73	12.76	8.0

It can be seen that both modal frequency are very close to each other, so that it can said the model implemented in this work is reliable [5]. With the validated active mounting system model, it is possible to conduct the control strategy modeling and the design optimization using GA.

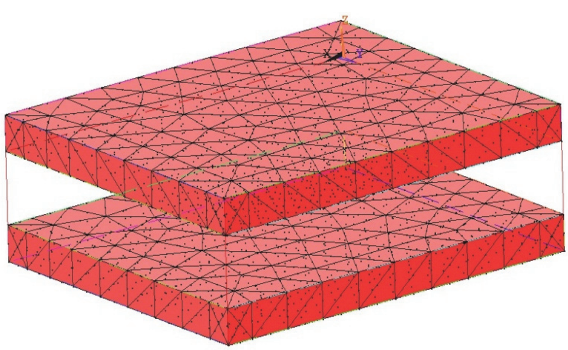

Fig. 4. The finite element model

Table 2. Model parameters

Parameter	Value
D_a damping	20 Ns/m
k_a stiffness	8902 N/m
m upper mass	19.274 kg
L_1	0.348 m
L_2	0.3 m
a	0 m
J_{xx} inertia of x-axis	0.1446 Kg/m^{4}s
J_{yy} inertia of y-axis	0.1676 Kg/m^{4}s

4. Control strategy

In this section, the control strategy will be analyzed by using the linear quadratic regulator (LQR) method, which is a well-known design technique that provides practical feedback gains [6]. The LQR controller in Simulink/MATLAB for this feedback active vibration control is shown in Fig. 5, where step is disturbance input, u is the control vector. By implementation of the LQR controller, we find that the vibration isolation takes place attenuating the disturbance coming form engine.

Transmitted acceleration with and without control are presented in Fig. 6 and Fig. 7 in time domain.

Figs. 6 and 7 show the response of system under the disturbance in time domain. It can be seen form Fig. 6 and Fig. 7 that the active engine mounting vibration system takes place right from the very beginning when subjected to step disturbance signal input. As input of the step disturbance activates, the transmitted acceleration has been increase to a greater range. This increase in the transmitted acceleration at engine mount decreases passenger comfort and smooth ride, working condition such as a vehicle through a rough road in a very short time.

The second line and the third line in Fig. 6 and Fig. 7 show that the response of the longitudinal acceleration is much faster and the lateral acceleration weakens a lot with LQR controller. It can be noticed that at the time interval of 0-0.5 sec, transmitted accelerationat active engine is reduced from 0.05 to 0.01 m/s^2 (approximately). The first line in Fig. 6 and Fig. 7 shows that the attenuation of the vertical acceleration is more outstanding for LQR controller. It can be seen that at time interval of 0-0.3 sec, the transmitted acceleration has been attenuated to a value of 0.02 m/s^2. The magnitude of the transmitted acceleration using LQR controller is reduced from 0.04 to 0.02 m/s^2 from time interval 0.3 sec. At the same time, the LQR controller can also weaken the heel and pitching direction of rotation.

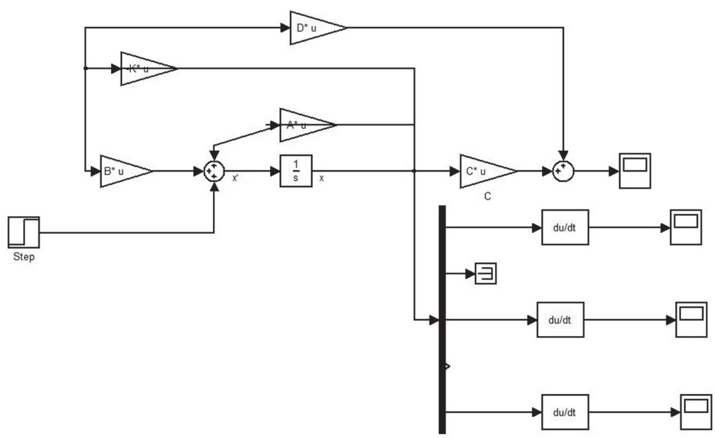

Fig. 5. Implementation of LQR controller in Simulink/MATLAB

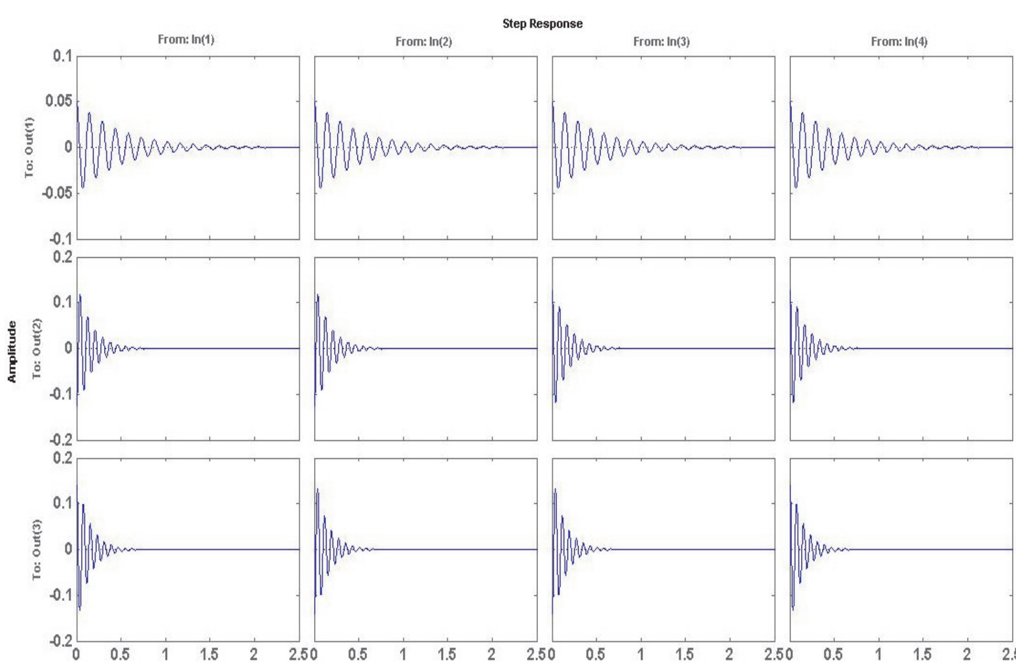

Fig. 6. Control response without LQR controller

By comparing the simulation results, it is quite clear that LQR controller is effective in restricting the transmitted force to the chassis. It is observed that the vibration attenuation is obtained for a period less than 0.3 second. The result confirmed that the model with LQR control algorithm is able to reduce significantly the vibration transmission.

By implementing LQR controller in Fig. 7, the value of gain K is calculated by adjusting the Q and R weights matrix. By doing various iterations the values for R and Q are set, so that optimal results are obtained. There are various methods of selecting values of the weights which will lead to the optimal control of an associated system. These methods are cheap control, expensive control and terminal control. In this paper, each of the methods was used to obtain our weights and the one leading to optimal control of our system was chosen.

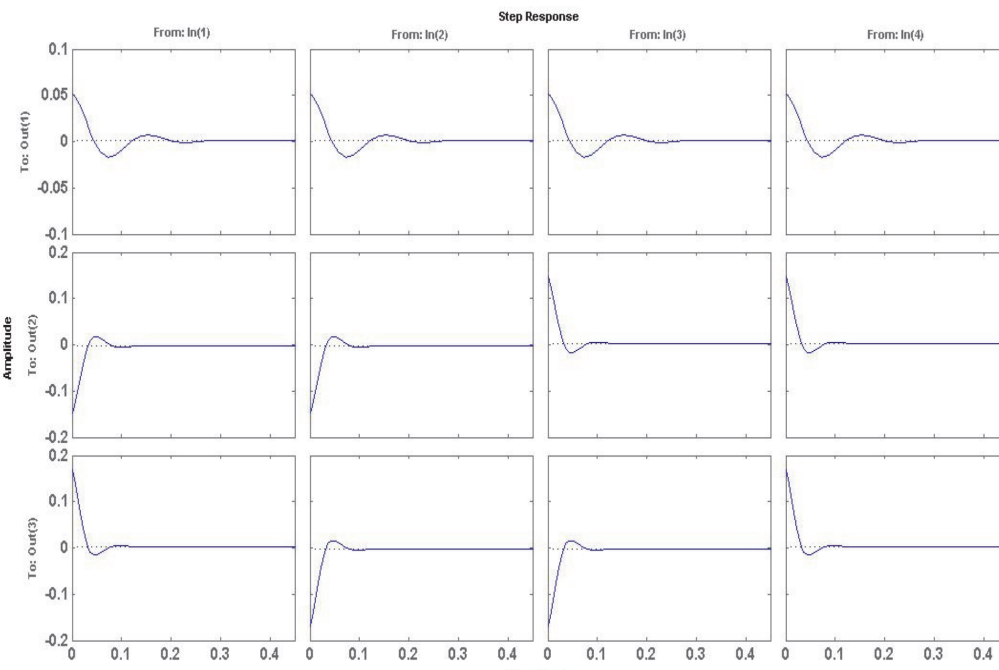

Fig. 7. Control response with LQR controller

5. Optimization via GA

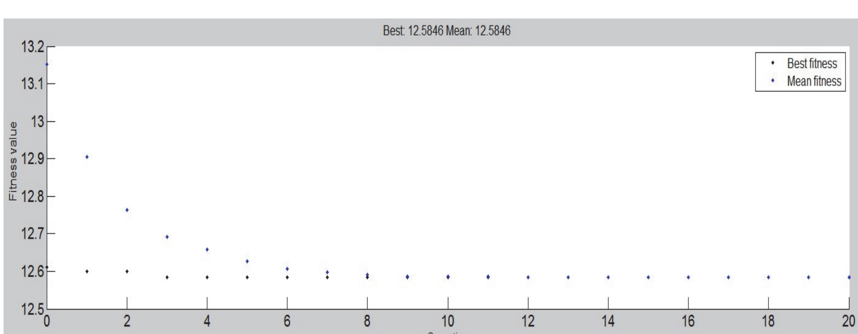

Fig. 8. Fitness function value in GADST

The selection of weight matrices in LQR is very importance and it straight affects the control performance. But the weight matrices are usually set by experience of designer and so the optimal control performance could not be obtained. The genetic algorithm (GA) is an optimization and search technique based on evolution, and it has been applied to many optimization problems [7, 8]. This work examines the application of genetic algorithms in optimizing the weight matrices of linear quadratic regulator [9]. The objective function for GA is:

$$\text{Minimize } L = \frac{EA(x)}{EApas} + \frac{PA(x)}{PApas} + \frac{RA(x)}{RApas}, \tag{3}$$

where EA is the engine body vertical acceleration, PA is the pitching acceleration, RA is the roll acceleration, pas is the passive acceleration. The result of the GA objective function in MATLAB/GADST GA toolbox is shown in Fig. 8.

6. Conclusions

In this paper, the vibration control performance of a selected active control engine mount system with LQR controller is evaluated. Then, an analytical model which includes the hydraulic engine mount and active control technique are implemented and analyzed [10]. This newly proposed model is validated by compared with FEA. In addition, the LQR controller is integrated into the model in order to achieve better vibration reduction as shown in numerical examples. Also, the application of GA for optimization design of LQR weight matrices is studied as well to have the optimal configuration of control strategy. For future work, different control techniques can be examined for the disturbance rejection such as LQG and robust control methods, which probably could provide more effective ways of dealing with disturbance reduction.

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