ICSV13 - Vienna
The Thirteenth International Congress on Sound and Vibration
Vienna, Austria, July 2-6, 2006

CONTROL OF AN ER ENGINE MOUNT FOR VIBRATION SUPPRESSION

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Abstract

In this paper, an ER engine mount is studied for vibration suppression of an engine subject to base excitations. Recently, electro-rheological (ER) fluid has become a popular material for actuator use. There are some good properties associated with ER fluid such as the reversible, controllable, and continuous change of rheological characteristics upon application of electric field. However, the dynamic equation of ER mount is highly nonlinear, hence making the controller design an extremely difficult task. This paper aims to develop a semi-active control technique for suppressing vibration of the engine mounted on such an ER mount subject to its base disturbances. The adaptive control scheme is employed for vibration attenuation. Function approximation technique is used here to represent the unknown disturbance and time varying system parameters in some finite linear combination of the orthogonal basis. The dynamics of ER engine mount system can thus be proved to be a stable first order filter driven by function approximation errors. Moreover, the adaptive update law can be obtained by using the Lyapunov stability theory. The well-known skyhook control scheme and a controller with constant applied maximum voltage are to be compared with the proposed adaptive controller for semi-active vibration control of the ER engine mount.

INTRODUCTION

Nowadays, more and more consumers pay much attention to the quality of a vehicle when purchasing. The quality means not only the driving performance but also the riding comfort, such as low vibration and noise. The riding qualities become an important evaluation index of a vehicle. In this paper, we will focus on the vibration problem. There are many reasons for vibration of a vehicle. One primary reason is the vibration coming from the operating engine. The impulse due to the exploding of fuels inside the engine is not avoidable. Therefore, engine mounts are constructed and used for isolating this vibration source.

The designs of engine mount have been developed for many years. The traditional engine mount is constructed by passive elements, such as rubber mount (elastomeric mount) and hydraulically-damped rubber mount [1]. The required properties of a good engine mount are high rigidity for bearing the weight of engine in static state and low rigidity for vibration isolation in dynamic state. Nevertheless, the increasing of rigidity is in direct proportion with vibration frequency for traditional rubber mount. Therefore, rubber mount can not satisfy these properties at the same time for broad band case; it can only deal with narrow band disturbance. To conquer the broad band vibration problem, the fluid engine mount is developed [1,2]. By appropriate designing, the fluid mount has better static rigidity and dynamic vibration isolation in the designed frequency range. However, adaptive abilities for passive mounts are generally poor. The variation of system parameters or non-designed vibration frequency applied can seriously affect the performance of vibration suppression. The adaptive ability of fluid mount can be improved by adding a hydraulic actuator. But, the complexity of the system grows while incorporating this device. Moreover, the closed loop stability is hard to guarantee.

Recently, smart materials are developed rapidly. There are some well known smart materials appeared in the last decade, such as piezoelectric ceramics [3], magnetostrictive fluids [4], and electro-rheological fluids [5], etc. Actuators incorporating these materials have prominent potentials for vibration suppression, under complex and varied environment. The ER fluid is used here for constructing the engine mount. There are three kinds of ER mounts appeared in the literatures: flow mode, mixed mode and squeezed-flow mode [6,7]. This paper adopts a squeezed-flow mode ER engine mount for vibration suppression study. Mathematical model of ER engine mount can be found in [8]. The dynamic behavior of ER fluid is highly nonlinear; therefore, controller design is extremely difficult. Some control algorithms have been discussed in previous literatures neural network [9], H_∞ control [10], skyhook control [11] and optimal control [12,13] are examples. Nevertheless, the ER mount systems contain time varying parameters. Thus, closed loop stability is hard to guarantee by linear controller. Using robust control techniques may work for time varying parameters or disturbances, yet requiring that all uncertainties be defined in some compact sets. The neural network based controllers could deal with those time varying uncertainties effectively. However, the stability of overall system is difficult to obtain.

In this paper, an adaptive sliding controller and function approximation technique are proposed to deal with modeling uncertainty and unknown disturbance [13-15]. Since the uncertainties are assumed to be time varying, function approximation technique may be used here to represent the unknown disturbances or uncertainties in some finite linear combinations of the orthogonal basis functions. By selecting appropriate update law, the time derivative of some Lyapunov function candidate can be proved to be negative semi-definite. Furthermore, not only the convergence of tracking error but also boundedness of all signals can be obtained. The paper is organized as follows. Section II gives a brief formulation of ER mount model and its properties. Section III derives the adaptive sliding control in detail. Section IV shows the simulation results of the proposed control law for vibration suppression of the one dimensional ER mount. Moreover, the well known skyhook control scheme and fixed maximum applied voltage control are to be compared with the proposed adaptive controller. Finally in Section V conclusions are briefly made.

PROBLEM FORMULATION

Consider the dynamic equations of a one-dimension ER mount in Fig. 1 [8]:

$$M\ddot{x} = k(y-x) + c(t)(\dot{y}-\dot{x}) + f_{er}(t) \tag{1}$$

where

$$c(t) = \frac{3}{2}\frac{\pi \upsilon R^4}{[x_0-(y-x)]} \tag{2}$$

$$f_{er}(t) = \frac{4}{3}\frac{\pi R^3}{[x_0-(y-x)]}\alpha\left(\frac{V}{[x_0-(y-x)]}\right)^{\beta}\operatorname{sgn}(\dot{y}-\dot{x}) \tag{3}$$

It is noticed that M is the mass of the bearing load, k is the stiffness constant of spring, c (t) is the damping coefficient of the ER fluid, f_{er} represents controllable damping force, υ is viscosity constant of ER fluid, R is the radius of circular electrode, α and β are known parameters determined by experiment, V is the applied voltage, x_0 is the initial gap between upper and lower electrodes, x and y represent the response displacement and base disturbance, respectively. We may represent Eqs. (1)-(3) in state space form

$$\begin{cases} \dot{z}_1 = z_2 \\ \dot{z}_2 = \dfrac{k}{M}(y-z_1) + \dfrac{c(t)}{M}(\dot{y}-z_2) + b(t)u \end{cases} \tag{4}$$

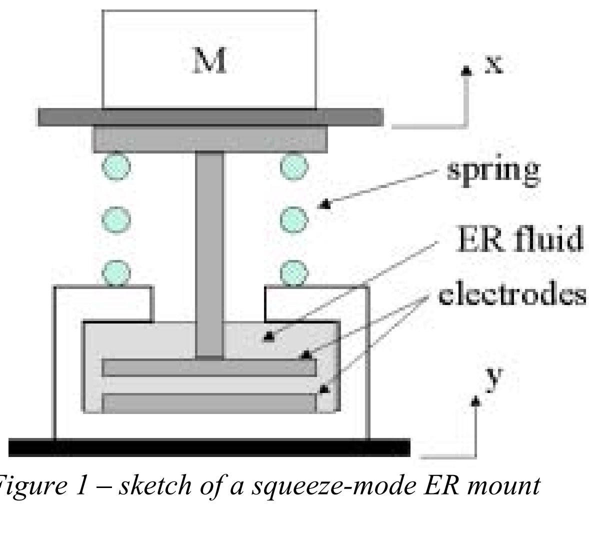


Figure 1 – sketch of a squeeze-mode ER mount

where $z_1 = x$ and $z_2 = \dot{x}$ are state variables, $b(t)$ and u represent the unknown input gain and control input, respectively

$$b(t) = \frac{4}{3}\frac{\pi R^3}{[x_0 - (y - z_1)]^{\beta+1}}$$

$$u = \alpha V^{\beta} \operatorname{sgn}(\dot{y} - z_2)$$

Eq. (4) can also be rewritten as

$$\begin{cases} \dot{z}_1 = z_2 \\ \dot{z}_2 = f(\mathrm{z}, t) + b(t)u \end{cases} \tag{5}$$

where $\boldsymbol{z} = [z_1 \quad z_2]^{\mathrm{T}}$ is the state vector and

$$f(\mathrm{z}, t) = \frac{k}{M}(y - z_1) + \frac{c(t)}{M}(\dot{y} - z_2)$$

Assumption 1: $f(\boldsymbol{z}, t)$ is an unknown function with unknown variation bound, but it remains continuous and bounded for all admissible $\boldsymbol{z}$ and for all $t \in [t_0, \infty)$.

Assumption 2: There exists some known continuous function $b_{\max}(z,t) > 0$ and $b_{\min}(z,t) > 0$ such that $b_{\min} < b < b_{\max}$ for all admissible state and for all $t \in [t_0, \infty)$.

By assumption 2, we may apply the robust control technique [16] to deal with the uncertainty of input gain. Thus, $b(t)$ can be represented as $b = b_m \Delta b$, where $b_m = \sqrt{b_{\max} b_{\min}}$ is the nominal function and Δb is the multiplicative uncertainty satisfying

$$0 < \beta_{\min} \equiv \frac{b_{\min}}{b_m} \le \Delta b \le \frac{b_{\max}}{b_m} \equiv \beta_{\max} \tag{6}$$

The objective is to design a controller such that $z_1 \to 0$ as $t \to \infty$. However, the bound of unknown time varying function $f(\boldsymbol{z}, t)$ is unavailable, traditional robust or adaptive control could not be applied in this case. In next section, an adaptive sliding controller with function approximation is used to deal with this given problem.

CONTROLLER DESIGN

In this section, we will give a brief description for controller design including the proof for stability analysis. The controller is designed in following steps.

Step 1: Define the sliding surface $s = \dot{e} + \lambda e$, where $e = z_1 - z_{1d}$, $\dot{e} = z_2 - z_{2d}$, z_{id} represents the desired value of state z_i, $i = 1,2$, λ is a parameter to be arbitrarily selected, and the time derivative of s can be derived as

$$\dot{s} = f + b_m \Delta b u + \lambda \dot{e} \tag{7}$$

Step 2: Eq. (7) can be stabilized by selecting u as

$$u = \frac{1}{b_m}[-\hat{f} - \lambda \dot{e} - \eta_{robust} \operatorname{sgn}(s)] \tag{8}$$

where $\hat{f}$ is an estimation of f, η_{robust} is a robust term to cover the effect induced by input channel uncertainty Δb. Substituting Eq. (8) into Eq. (7), we can obtain

$$\dot{s} = \tilde{f} + (1-\Delta b)(\hat{f} + \lambda\dot{e}) - \Delta b\,\eta_{robust}\,\mathrm{sgn}(s) \tag{9}$$

where $\tilde{f} = f - \hat{f}$. Since f is a time varying uncertainty, the function approximation technique [13-15] can be applied here to transform the uncertainty into a finite linear combination of the orthogonal basis. Specifically, f and $\hat{f}$ can be represented as

$$f = \boldsymbol{w}^{\boldsymbol{T}}\boldsymbol{\varphi} + \varepsilon \tag{10}$$

$$\hat{f} = \hat{\boldsymbol{w}}^{\boldsymbol{T}}\boldsymbol{\varphi} \tag{11}$$

where $\boldsymbol{w}, \hat{\boldsymbol{w}} \in \Re^n$ are weighting vector, $\boldsymbol{\varphi} \in \Re^n$ is the vector of basis function, the positive constant n is the number of basis functions used in the approximation, ε is the truncation error. Substituting Eqs. (10) and (11) into Eq. (9) yields

$$\dot{s} = \tilde{\boldsymbol{w}}^{\boldsymbol{T}}\boldsymbol{\varphi} + (1-\Delta b)\rho - \Delta b\,\eta_{robust}\,\mathrm{sgn}(s) + \varepsilon \tag{12}$$

where $\rho = \hat{\boldsymbol{w}}^{\boldsymbol{T}}\boldsymbol{\varphi} + \lambda\dot{e}$.

Step 3: We may select the Lyapunov function candidate as

$$V_1 = \frac{1}{2}s^2 + \frac{1}{2}\tilde{\boldsymbol{w}}^{\boldsymbol{T}}\boldsymbol{Q}\tilde{\boldsymbol{w}} \tag{13}$$

where $\boldsymbol{Q} \in \Re^{\boldsymbol{n}\times\boldsymbol{n}}$ is a positive definite matrix. Taking time derivative of Eq. (13), we have

$$\dot{V}_1 = \tilde{\boldsymbol{w}}^{\boldsymbol{T}}\boldsymbol{\varphi}s + s[(1-\Delta b)\rho - \Delta b\,\eta_{robust}\,\mathrm{sgn}(s)] - \tilde{\boldsymbol{w}}\boldsymbol{Q}\dot{\hat{\boldsymbol{w}}} + s\varepsilon \tag{14}$$

We may select the update law as

$$\dot{\hat{\boldsymbol{w}}} = \boldsymbol{Q}^{-\boldsymbol{1}}\boldsymbol{\varphi}s \tag{15}$$

then Eq. (14) becomes

$$\begin{aligned}\dot{V}_1 &= (1-\Delta b)\rho s - \Delta b\,\eta_{robust}\,|s| + s\varepsilon \\ &\le (1-\beta_{\min})\,|\rho|\,|s| - \beta_{\min}\eta_{robust}\,|s| + |s|\,|\varepsilon|\end{aligned} \tag{16}$$

Now choosing η_{robust} as

$$\eta_{robust} = \frac{1}{\beta_{\min}}[(1-\beta_{\min})\,|\rho| + \eta] \tag{17}$$

where η is a positive constant to be arbitrarily selected. Substituting Eq. (17) into (16), we obtain that

$$\dot{V}_1 \le -\eta\,|s| + |s|\,|\varepsilon| \tag{18}$$

If sufficient basis are used such that the function approximation error $\varepsilon \approx 0$, then $\dot{V}_1 \le 0$. Furthermore, we can also prove that

$$\int_0^\infty s^2 dt = -\frac{1}{\eta}\int_0^\infty \dot{V}_1 dt = -\frac{1}{\eta}V_1\,\big|_0^\infty < \infty \tag{19}$$

From Eqs. (12), (18) and (19), we may conclude that $\tilde{\boldsymbol{w}} \in L_\infty$, $s \in L_\infty \cap L_2$, $\dot{s} \in L_\infty$. Therefore, the system is asymptotical stable by using Barbalat's lemma.

Remark 1: If approximation error can not be neglected, and there exists a positive constant $\delta > 0$ such that $|\varepsilon| \leq \delta$. Then to cover the effect of this bounded approximation error, Eq. (8) is modified to be

$$u = \frac{1}{b_m}[-\hat{f} - \lambda\dot{e} - \eta_{robust}\,\mathrm{sgn}(s)] + u_{robust} \tag{20}$$

In this case, we have

$$\dot{V}_1 \leq -\eta\,|s| + |s||\varepsilon| + su_{robust} \tag{21}$$

By selecting $u_{robust} = -\mathrm{sgn}(s)\delta$, we may also conclude the asymptotical stability of system.

Remark 2: To avoid parameter drift, the σ -modification can be used in Eq. (15)

$$\dot{\hat{\boldsymbol{w}}} = \boldsymbol{Q}^{-1}\boldsymbol{\varphi}s - \sigma\hat{\mathrm{w}} \tag{22}$$

where σ is a small positive constant.

Remark 3: The ER mount control is a semi-active one; therefore control action should follow the actuating condition [7].

$$u = \begin{cases} u & for \quad z_2(\dot{y} - z_2) > 0 \\ 0 & for \quad z_2(\dot{y} - z_2) \leq 0 \end{cases} \tag{23}$$

SIMULATION RESULTS

The system parameters used in the simulation are shown in Table 1 and n=21 terms of Fourier orthogonal basis are used for function approximation. The sinusoidal disturbances *y(t)* with frequency between 10Hz~30Hz and amplitude at $1\times10^{-3}m$ are employed as the base excitation. The vibration control results are shown in Fig. 2-5. The comparison of the proposed control law and other control schemes is shown in Fig. 2. Fig.3 shows the displacements of the bearing mass by each controller. From Figs. 2-3, we can find that the proposed controller has much better performance for vibration attenuation than other controllers, even we do not know exactly the parameter values. The parameter estimation is shown in Fig. 4. Since reference input do not satisfy the persistent excitation condition; therefore, the estimation do not converge to actual value but it is bounded. Fig. 5 is the control voltage input to the electrodes.

Table 1. System parameters

R	$25\times10^{-3}m$
x_0	$2\times10^{-3}m$
M	$0.75kg$
υ	$20cSt$
k	$13.7\times10^{3}N/m$
α	0.00247
β	1.2

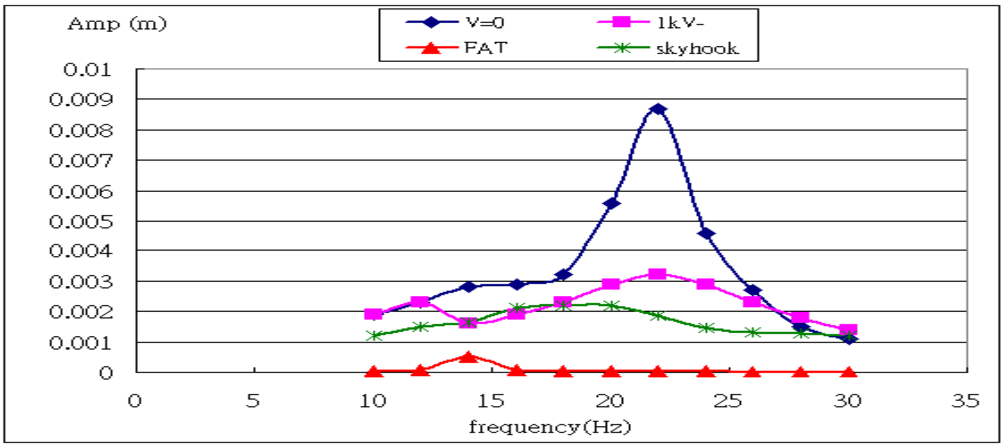
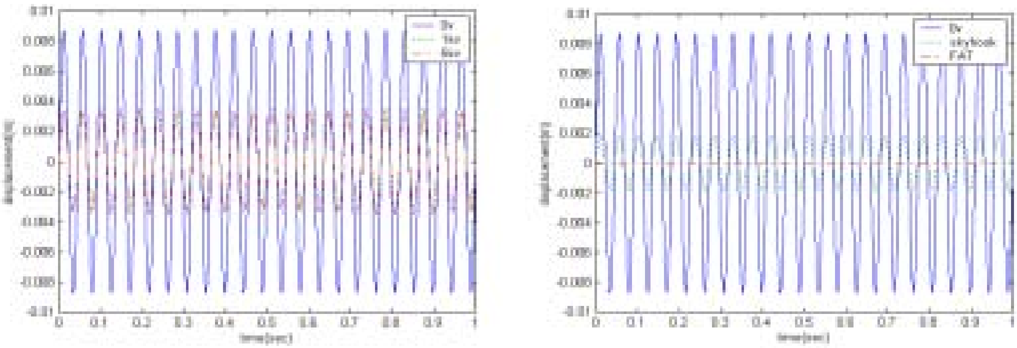


Figure 2 – Vibration attenuation for 10Hz~30Hz disturbance



(a) (b)

Figure 3 – Vibration attenuation for resonant frequency excitation (22Hz)

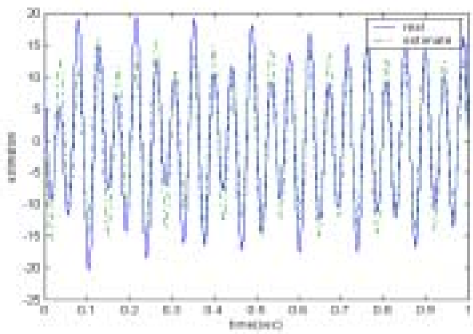


Figure 4 – Approximation of $f(z,t)$

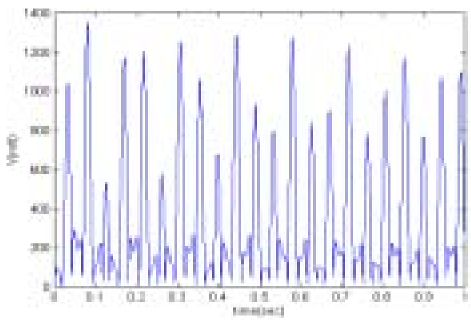


Figure 5 – Applied voltage

CONCLUSION

In this paper, we have proposed an adaptive sliding control with function approximation technique for ER engine mount. The convergence of errors has been proved. However, the reference input does not satisfy the persistent excitation condition. Therefore, the estimates do not converge to actual values, but all remain bounded. With the proposed controller, the control strategy does not require much information about the system model. Results of the computer simulation justify the performance of the proposed controller.

ACKNOWLEDGEMENT

This work was partially supported by National Science Council, Taiwan, under grant number NSC 94–2212–E–027–008.

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