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Active Damper Control System Based on LMFC

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Abstract

All electric-damper is now a fashionable method for vehicle balance as the development of the high power linear and motor drive system allow the fast and efficient solution to dampers. The conventional damper is then replaced by active motor drive that provides faster response than the conventional system. In this paper, an LMFC method is proposed to provide a better control method for the four dampers that is located in the vehicle. The 4-coordinate control is difficult as they are closely related to each other. The heave position, pitch and roll angle are the main concerns to keep consistent for the passenger's comfort and ride performance. To achieve this goal, an effective LMCF is investigated and applied to the active damper system, with its simplicity and reliable for implementation. For the active damper system, the inner loop is with PD control to eliminate the disturbances and nonlinearities, while the outer loop is with linear model following control to track the required dynamic performance. The computed results verify the proposed control methods, and the dynamic response of the active damper systems is studied by optimization.

Keywords: Active damper system, vehicle suspension, LMFC

1 Introduction

Active damper system, or active suspension system, has been studied for years and equipped in some luxury models. The main difference between the passive suspension and the active damper system is the active implementation unit, such as hydraulic actuator (Citroën Hydractive 3 suspension system) or electromagnetic actuator (Bose suspension system). Regardless of the cost and complexity of the actuator, the active damper system enables control flexibility for riding and achieves high performance of safety and comfort.
According to ISO 2631 standard, the human body could suffer from vibrations within specified frequencies ranged from 0.5 to 80 Hz. Meanwhile, the basic ride frequency for a passenger car is typically about 1.4 Hz [1]. Hence, passengers of ground vehicle have a high risk of injury to the vertebrae in the lumber region and the nerves connected to these segments.
In order to eliminate or minimize vibrations, suspension system is equipped to isolate the vehicle body from road irregularities. In addition,

vehicle suspension system maximizes passenger ride comfort and improves the vehicle handling quality [2].

Comparing with passive suspension system, active damper system employs hydraulic or electromagnetic actuators to shorten response time greatly and improve dynamic ride performance. Martins et al. (2006) proposed the potential of applying linear actuators in active suspension system, and specified the "reduced comfort boundary" with about 1050 N [3].

Various control strategies of active damper system with electromagnetic actuator have been investigated. Sliding mode controller (SMC) had been developed for a nonlinear vehicle model with full knowledge of the system states, which its robustness was highlighted [4]; a modified SMC was designed for a linear full vehicle active suspension system with partly knowledge of the system states [5].

Intelligent control methods, such as fuzzy inference systems, neural networks, genetic algorithms and their combination, have been investigated for suspension system [6]. In addition, interest of combination between SMC and intelligent methodologies is raised to achieve high performance against conventional controls [7] [8].

A practical problem, however, is the capability of implementation of these control methods for its comprehensive calculation. In this paper, linear model following control (LMFC) is introduced for its simplicity and easy implementation. Morse (1973) proposed the solving method against the model following problem [9]. LMFC is developed to design a compensating control for a linear multivariable system, and its main focus is to minimize the differences of the states of the reference model and the plant [10] [11].

Since there is a multivariable vehicle suspension model, a PD controller is used here to decouple the vehicle system into three linear subsystems. For the proposed double-looped control system, PD control is used as the inner loop control to compensate the uncertain variables and LMFC is employed as the outer loop to improve the dynamic performance of the vehicle.

This paper is organized as follows. In Section 2, the vehicle active damper system is described and a seven degree-of-freedom (DOF) model is proposed. The control strategy of LMFC and its application on the active damper system are presented in Section 3. Simulation results against road irregularities are shown in Section 4 to verify the proposed controller. In last section, the conclusions are given, and the dynamics of the proposed actuators [13-15] will be considered in future.

2 Model of Full-car Active Damper system

The model of the full-car active damper system is shown in fig.1. The linearized seven DOF system [12] is proposed by Ikenaga (2000) without passive dampers, which consist of a single sprung mass (rigid car body) connected to four unsprung mass through springs and electromagnetic actuators. Each wheel is represented by a linear spring.

Quarter-vehicle model is more extensively used to analyze and understand the influence of suspension parameters. It has simple structure with two degree-of-freedom in the vertical direction, which can be easily applied for the design and control of suspension systems. Although roll and pitch behaviors are eliminated in this kind of model, they can be simulated as external disturbance acting on the vehicle body.

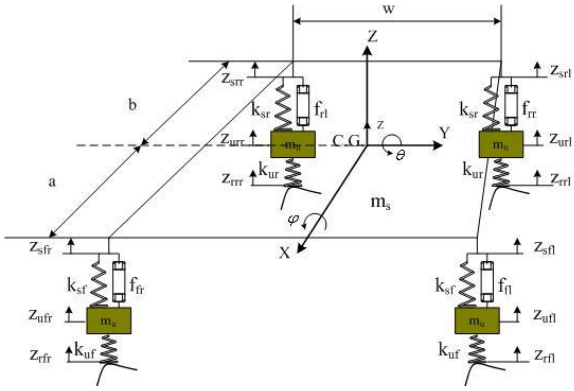


Figure 1: Linearized model of a full-car active damper system

The equations of the full-car active damper system model are given as follow:

$$
\begin{cases}
m_s\ddot{z} = -m_s g - \left(2K_{sf} + 2K_{sr}\right)z + \left(2aK_{sf} - 2bK_{sr}\right)\theta \\
\quad + K_{sf}z_{ufl} + K_{sf}z_{url} + K_{sr}z_{url} + K_{sr}z_{urr} \\
\quad + f_{fl} + f_{fr} + + f_{rl} + f_{rr} \\
I_{yy}\ddot{\theta} = \left(2aK_{sf} - 2bK_{sr}\right)z - \left(2a^2K_{sf} + 2b^2K_{sr}\right)\theta \\
\quad - aK_{sf}z_{ufl} - aK_{sf}z_{ufr} + bK_{sr}z_{url} + bK_{sr}z_{urr} \\
\quad - af_{fl} - af_{fr} + bf_{rl} + bf_{rr} \\
I_{xx}\ddot{\varphi} = -\frac{1}{4}w^2\left(2K_{sf} + 2K_{sr}\right)\varphi + \frac{1}{2}wK_{sf}z_{ufl} \\
\quad - \frac{1}{2}wK_{sf}z_{ufr} + \frac{1}{2}wK_{sr}z_{url} - \frac{1}{2}wK_{sr}z_{urr} \\
\quad + \frac{1}{2}wf_{fl} - \frac{1}{2}wf_{fr} + \frac{1}{2}wf_{rl} - \frac{1}{2}wf_{rr} \\
m_u\ddot{z}_{ufl} = -m_u g + K_{sf}z - aK_{sf}\theta + \frac{1}{2}wK_{sf}\varphi \\
\quad - \left(K_{sf} + K_u\right)z_{ufl} + K_u z_{rfl} - f_{fl} \\
m_u\ddot{z}_{ufr} = -m_u g + K_{sf}z - aK_{sf}\theta - \frac{1}{2}wK_{sf}\varphi \\
\quad - \left(K_{sf} + K_u\right)z_{ufr} + K_u z_{rfr} - f_{fr} \\
m_u\ddot{z}_{url} = -m_u g + K_{sr}z + bK_{sr}\theta + \frac{1}{2}wK_{sr}\varphi \\
\quad - \left(K_{sr} + K_u\right)z_{url} + K_u z_{rrl} - f_{rl} \\
m_u\ddot{z}_{urr} = -m_u g + K_{sr}z + bK_{sr}\theta - \frac{1}{2}wK_{sr}\varphi \\
\quad - \left(K_{sr} + K_u\right)z_{urr} + K_u z_{rrr} - f_{rr}
\end{cases} \tag{1}
$$

where z is the heave position of the sprung mass, θ is the pitch angle of the sprung mass, φ is the roll angle of the sprung mass. z_{ufl} , z_{ufr} , z_{url} and z_{urr} are the unsprung mass heights of four corners; z_{rfl} , z_{rfr} , z_{rrl} and z_{rrr} are the terrain disturbance heights of four corners. m_s and m_u are the sprung mass and unsprung mass respectively. K_{sf} , K_{sr} and K_u are the stiffness of front springs, rear springs and tires, respectively. I_{xx} and I_{yy} are the roll axis and pitch axis moment of inertia. a is the distance from C.G. to the front axle, b is the distance from C.G. to the rear axle, w is the width of the sprung mass. g is the acceleration due to gravity. f_{fl} , f_{fr} , f_{rl} and f_{rr} are the active forces generated by the electromagnetic actuators.

3 LMFC and its Application on the Active Damper System

In this section, a brief introduction of LMFC is presented first, and the control strategy is specified as follows.

3.1 Theory of Linear Model-Following Control (LMFC)

Suppose the controllable plant can be described as

$$\dot{x}_p = A_p x_p + B_p u_p \tag{2}$$

$$y_p = C_p x_p \tag{3}$$

where $x_p \in R^n$, $u_p \in R^m$ and $y_p \in R^l$; A_p , B_p and C_p are constant matrices of appropriate dimensions.
The referenced model has the following form:

$$\dot{x}_m = A_m x_m + B_m r \tag{4}$$

$$y_m = C_m x_m \tag{5}$$

where $x_m \in R^n$, $r \in R^m$ and $y_m \in R^l$; A_m , B_m and C_m are constant matrices with the same dimensions of A_p , B_p and C_p .
The goal of LMFC is to minimize the difference between the states of x_p and x_m , i.e. minimize the errors

$$e = x_m - x_p \tag{6}$$

$$\varepsilon = y_m - y_p \tag{7}$$

where e is the error of the states and ε is the output error.
Assume $C_p = C_m$, e and ε have the same variation and the output error will be eliminated when e approaches zero.

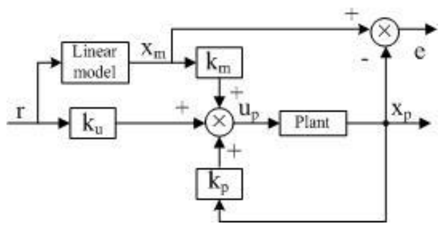


Figure 2: Schematic diagram of LMFC

The control law of the system is chosen as

$$u_p = K_u r + K_m x_m + K_p x_p \tag{8}$$

where K_u , K_m and K_p are constant gain matrices of appropriate dimensions, as shown in Fig.2.
From (2) - (8), the equation of states error can be written as:

$$\dot{e} = \left(A_m - B_p K_m\right)e + \left(A_m - A_p - B_p(K_m + K_p)\right)x_p + \left(B_m - B_p K_u\right)r \tag{9}$$

From (9), the state error e will decay to zero asymptotically if the constant matrices are appropriately chosen to meet the following requirements:

(i) $\left(A_m - B_p K_m\right)$ be a Hurwitz matrix;

(ii) $\left(A_m - A_p - B_p(K_m + K_p)\right) = 0$; (10)

(iii) $\left(B_m - B_p K_u\right) = 0$. (11)

Once the above three conditions are satisfied, the output of the controlled plant will follow the output of the referenced model.

To solve eqns. 18 and 19, the necessary and sufficient conditions are shown below:

$$K_m + K_p = B_p^{+}\left(A_m - A_p\right) \tag{12}$$

$$K_u = B_p^{+} B_m \tag{13}$$

where $B_p^{+} = \left(B_p^T B_p\right)^{-1} B_p^T$ is the left Penrose pseudo inverse of B_p.

Here, the constant gain matrices K_u, K_m and K_p are selected to minimize the output error. K_u can be calculated from (13); K_m and K_p are determined by (12), with the constraint of Hurwitz matrix.

3.2 Control Strategy of the Active Damper

The main goal of the active damper system is to isolate the vehicle body from road irregularities. Hence, for the state equations of the system, the control goal is to keep the height, pitch and roll angle of the car body constant.

From (1), the inputs of car body can be rewritten as

$$u_z = f_{fl} + f_{fr} + + f_{rl} + f_{rr} \tag{14}$$

$$u_\theta = -af_{fl} - af_{fr} + bf_{rl} + bf_{rr} \tag{15}$$

$$u_\varphi = \frac{1}{2}wf_{fl} - \frac{1}{2}wf_{fr} + \frac{1}{2}wf_{rl} - \frac{1}{2}wf_{rr} \tag{16}$$

Define the state vector of the system model as $x = [x_1\ x_2\ \cdots\ x_{14}]^T$, which $x_1 = z$, $x_2 = \dot{z}$, $x_3 = \theta$, $x_4 = \dot{\theta}$, $x_5 = \varphi$, $x_6 = \dot{\varphi}$, $x_7 = z_{ufl}$, $x_8 = \dot{z}_{ufl}$, $x_9 = z_{ufr}$, $x_{10} = \dot{z}_{ufr}$, $x_{11} = z_{url}$, $x_{12} = \dot{z}_{url}$, $x_{13} = z_{urr}$, $x_{14} = \dot{z}_{urr}$.

The equations of the sprung mass can be written as

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = f_2(x) + b_2 u_z \end{cases} \tag{17}$$

$$\begin{cases} \dot{x}_3 = x_4 \\ \dot{x}_4 = f_4(x) + b_4 u_\theta \end{cases} \tag{18}$$

$$\begin{cases} \dot{x}_5 = x_6 \\ \dot{x}_6 = f_6(x) + b_6 u_\varphi \end{cases} \tag{19}$$

where

$$f_2(x) = \left[\begin{matrix} -\left(2K_{sf} + 2K_{sr}\right)x_1 + \left(2aK_{sf} - 2bK_{sr}\right)x_3 \\ +K_{sf}x_7 + K_{sf}x_9 + K_{sr}x_{11} + K_{sr}x_{13} \end{matrix}\right] \Big/ m_s$$

$b_2 = 1/m_s$

$$f_4(x) = \left[\begin{matrix} \left(2aK_{sf} - 2bK_{sr}\right)x_1 - \left(2a^2K_{sf} + 2b^2K_{sr}\right)x_3 \\ -aK_{sf}x_7 - aK_{sf}x_9 + bK_{sr}x_{11} + bK_{sr}x_{13} \end{matrix}\right] \Big/ I_{yy}$$

$b_4 = 1/I_{xx}$

$$f_6(x) = \left[\begin{matrix} -\frac{1}{2}w^2\left(K_{sf} + K_{sr}\right)x_5 + \frac{1}{2}wK_{sf}x_7 \\ -\frac{1}{2}wK_{sf}x_9 + \frac{1}{2}wK_{sr}x_{11} - \frac{1}{2}wK_{sr}x_{13} \end{matrix}\right] \Big/ I_{xx}$$

$b_6 = 1/I_{yy}$

Equations (17)-(19) contain 14 variables; the first step of applying linear model-following control is to compensate the rest of the variables. Here, PD controllers are used to compensate and decoupling the active damper system into three first order subsystems.

Define the required heave position, pitch angle and roll angle of the car body as z_m, θ_m and φ_m. The output errors and their deviations are $e_2 = z_m - z$, $\dot{e}_2 = \dot{z}_m - \dot{z}$, $e_4 = \theta_m - \theta$, $\dot{e}_4 = \dot{\theta}_m - \dot{\theta}$, $e_6 = \varphi_m - \varphi$ and $\dot{e}_6 = \dot{\varphi}_m - \dot{\varphi}$.

The control input of the PD controllers as follow:

$$u_z = \frac{1}{b_2}\left(k_{p2}e_2 + k_{d2}\dot{e}_2\right) \tag{20}$$

$$u_\theta = \frac{1}{b_4}\left(k_{p4}e_4 + k_{d4}\dot{e}_4\right) \tag{21}$$

$$u_\varphi = \frac{1}{b_6}\left(k_{p6}e_6 + k_{d6}\dot{e}_6\right) \tag{22}$$

The obtained decoupling equations are

$$\dot{x}_1 = a_1x_1 + b_1u_1 \tag{23}$$

$$\dot{x}_3 = a_3x_3 + b_3u_3 \tag{24}$$

$$\dot{x}_5 = a_5x_5 + b_5u_5 \tag{25}$$

where $u_1 = z_m$, $u_3 = \theta_m$, $u_5 = \varphi_m$; $a_1 = -b_1$, $a_3 = -b_3$, $a_5 = -b_5$.

The following step is embedded the linear model-following controllers into the subsystems. Here, let us take the heave tracking problem for example to illustrate the control strategy.

When road irregularities occur, the referenced dynamic characteristic of the heave position of car body is modeled as follow:

$$\dot{x}_{1m} = a_{1m}x_{1m} + b_{1m}r_1 \tag{26}$$

where r_1 is the required heave position, $-a_{1m} = b_{1m} > 0$, i.e. x_{1m} is the tracking of r_1.

For (23) and (26), the output matrices $c_m = c = 1$, the control law is obtained as

$$u_1 = k_{1u} r_1 + k_{1m} x_{1m} + k_{1p} x_1 \qquad (27)$$

where $k_{1u} = b_{1m}/b_1$, $k_{1p} = -k_{1m} + (a_{1m} - a_1)/b_1$ and $k_{1m} > a_{1m}/b_1$. The control laws of pitch angle and roll angle are similar to (27). The control strategy of the active damper system is shown as Fig.3.

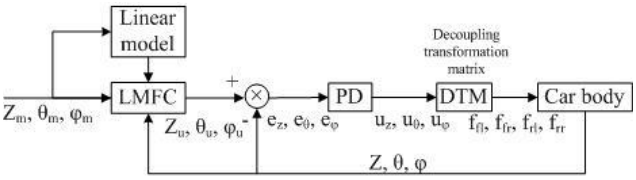


Figure 3: LMFC-PD control of the full-car active damper system

In addition, the transformation of inputs u_z , u_θ and u_φ into the actuators output forces is represented here:

$$\begin{pmatrix} u_z \\ u_\theta \\ u_\varphi \end{pmatrix} = \begin{pmatrix} 1 & 1 & 1 & 1 \\ -a & -a & b & b \\ w/2 & -w/2 & w/2 & -w/2 \end{pmatrix} \begin{pmatrix} f_{fl} \\ f_{fr} \\ f_{rl} \\ f_{rr} \end{pmatrix} \qquad (28)$$

The required generated forces of electromagnetic actuators are:

$$\begin{pmatrix} f_{fl} \\ f_{fr} \\ f_{rl} \\ f_{rr} \end{pmatrix} = \begin{pmatrix} 1 & 1 & 1 & 1 \\ -a & -a & b & b \\ w/2 & -w/2 & w/2 & -w/2 \end{pmatrix}^{+} \begin{pmatrix} u_z \\ u_\theta \\ u_\varphi \end{pmatrix} \qquad (29)$$

where $(\cdot)^+$ is the pseudo-inverse of the matrix $(\cdot)$.

4 Simulation results

The parameters of the active damper system of the vehicle are shown in Table 1. The controller is verified by comparison between PD control and LMFC control; the simulation results against road irregularities are shown as follows. Without generality, the gains of the subsystems are selected to be identical for simplicity, i.e. $k_p = k_{p1} = k_{p3} = k_{p5}$, $k_d = k_{d1} = k_{d3} = k_{d5}$, $a_m = a_{1m} = a_{3m} = a_{5m}$, $b_m = b_{1m} = b_{3m} = b_{5m}$, $k_u = k_{1u} = k_{3u} = k_{5u}$, $k_m = k_{1m} = k_{3m} = k_{5m}$, $k_p = k_{1p} = k_{3p} = k_{5p}$. The optimal control gains are shown in Table 2.

The road irregularities contain an isolated trapezoidal bump and sinusoidal two-track roads, as illustrated in Fig. 4 and 7. The time delay τ must be considered due to the track length between the front and rear axles which is $L = a + b = 3.1m$. Once the longitudinal velocity of the vehicle is assumed $v = 20 m/s$, the time delay can be calculated as $\tau = L/v = 0.155 s$.

The response output of car body with LMFC under isolated bump road is shown in Fig. 5, with a comparison of PD controller. The oscillation of the car body is reduced with LMFC control, which illustrates that LMFC-PD control achieves higher performance than PD control. The required forces generated by four actuators are shown in Fig. 6. The output forces are low due to the low PD gain, which reduces the burden of actuators at the price of extending the settling time.

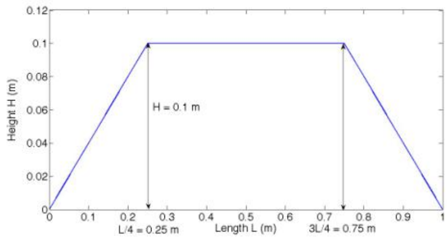


Figure 4: Isolated trapezoidal bump road

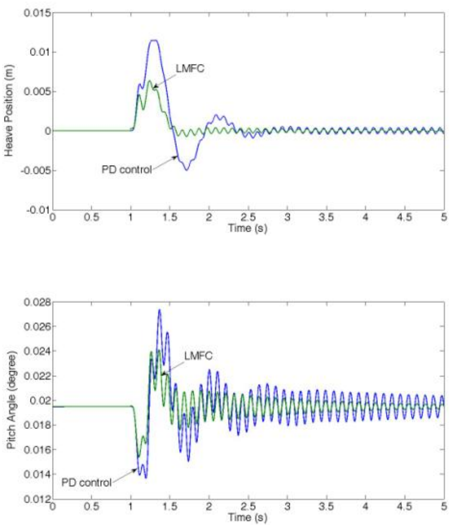
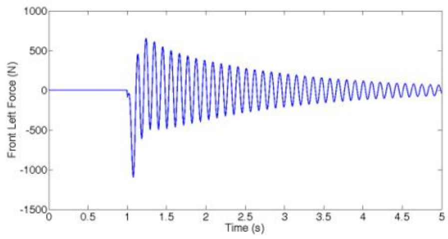


Figure 5: Response between LMFC and PD control under isolated trapezoidal bump road



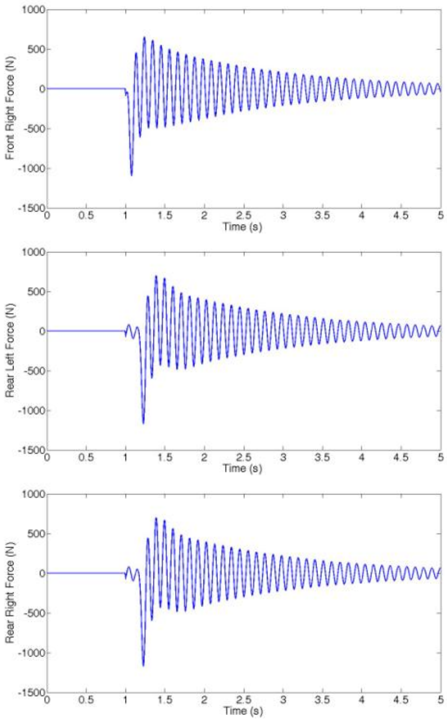


Figure 6: Corresponding output actuators forces under bump road

Sinusoidal two-track roads are general form of the practical road profile. With changing the wavelength and phase shift, all the road profiles can be obtained. When the phase shift equals zero, two tracks are symmetrical and the response is similar to the continuous bump; when the phase shift equals 180°, two tracks are anti-symmetrical and the roll angle is the main issue to compensate. Fig. 8 shows the comparison between LMFC and PD controller for the car body under sinusoidal road which the phase shift angle equals 90 °. The results show that LMFC greatly reduces the oscillations due to the road irregularity, almost half of that of PD controller. It is convinced that the active damper system with LMFC improves the ride comfort. The corresponding generated forces are shown in Fig. 9. Since the required maximum forces are about 2000N, it is feasible to test the experimental platform in the next step.

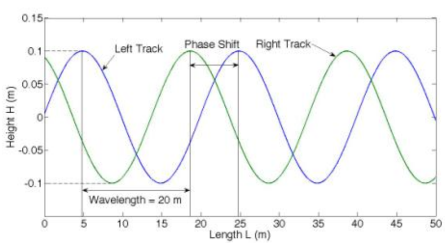


Figure 7: Sinusoidal road

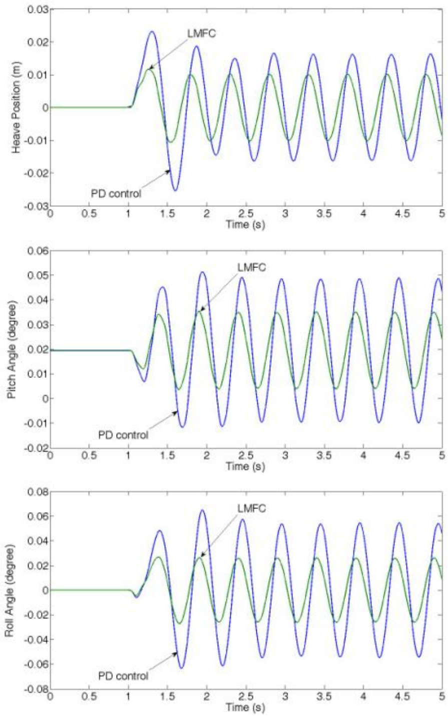


Figure 8: Response between LMFC and PD control under sinusoidal road

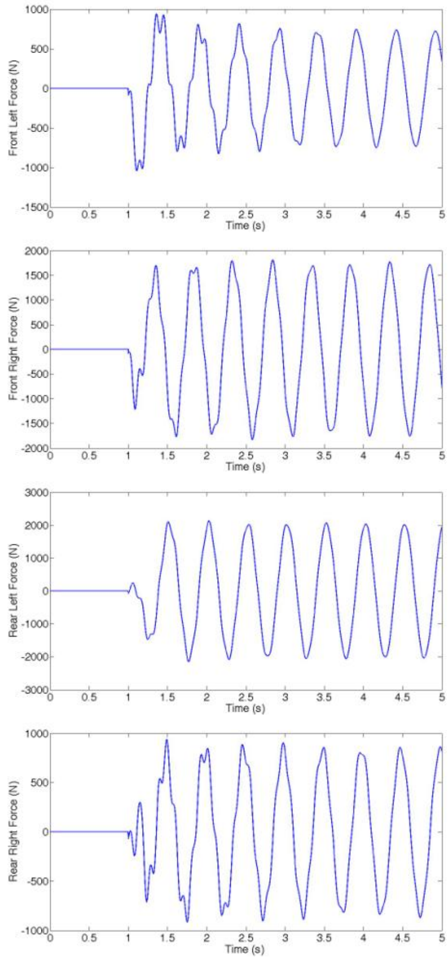


Figure 9: Corresponding output actuators forces under sinusoidal road

Table 1: System parameters

Parameter	Value
m_s	1500 (kg)
m_u	59 (kg)
K_{sf}	16000 (N/m)
K_{sr}	17000 (N/m)
K_u	190000 (N/m)
I_{xx}	460 (kg · m2)
I_{yy}	2160 (kg · m2)
a	1.4 (m)
b	1.7 (m)
w	1.59 (m)

Table 2: Controller parameters

Parameter	Value
k_p	20
k_d	5
a_m	-80
b_m	80
k_u	20
k_m	-17.1
k_p	-1.9

5 Conclusion

Linear model-following control is introduced in this paper, which allows assessing the design of pole assignment conveniently and achieves the preferred system dynamics. Comparing with the PD control, the simulation results show that LMFC responses faster and suppresses the car body oscillation greatly. Since the road profiles can be expressed by a combination of sinusoidal roads with different wavelengths, amplitudes and phase shift angles, it indicates that the active damper system with LMFC will achieves high dynamic performance and improves the ride safety and comfort at road test which is similar to the simulation results with low frequencies. The control strategy against high frequencies will be investigated in following.

The proposed LMFC-PD controller is significant for its simplicity by reducing the calculation burden greatly while maintain high performance, which is critical in practical implementation. The dynamics of the electromagnetic actuators is another issue to be considerate. The control of active damper system and the operation of force actuators will be implemented simultaneously. The experimental validation of the active damper system will be investigated in future.

Acknowledgments

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