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Semi-active control of engine mount system improving vehicle ride comfort

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ABSTRACT: The main purpose of this study is to propose a comparison of the performance of semi-active hydraulic engine mounting system (SHEMs) with passive hydraulic engine mounting system (PHEMs) via vehicle ride comfort. A full-vehicle dynamic modelis established under the combination of two excitation sources such as internal combustion engine and road surface excitations.A fuzzy logic controller is designed for control of the damping coefficient of a semi-active hydraulic engine mounting system.The time domain acceleration responses of the vertical motion, pitch and roll angles of vehicle body are chosen as objective functions to compare the vehicle ride comfort performance of the SHEMs and PHEMs. The obtained results indicate that the peak amplitude values of the time domain acceleration responses with SHEMs respectively reduce in comparison with PHEMs under different survey conditions.

KEYWORDS: Internal combustion engine (ICE), Semi-active hydraulic engine mount (SHEM), Hydraulic engine mount (HEM), Full-vehicle dynamic model, Ride comfort.

I. INTRODUCTION

Improved designfor the engine mounting systemnot only improves vehicle ride comfort, but also reduces vehicle noise.A full-vehicle vibration model with 10 degrees of freedom was established under the combination of road surface roughness and ICE excitations to evaluate the effect of the adding damping coefficient values into the rubber mounting system on vehicle ride comfort[1]. A13–DOF dynamic model of vehicle was proposedto evaluate the effect ofthe hydraulic engine mounts (HEMs)on the engine shake performance [2].A full-vehicle dynamic model under the combination of two excitation sources such asinternal combustion engine and road surface excitationswas proposedto evaluate the vehicle ride comfort performance between the hydraulic engine mount system (HEMs) and rubber engine mount system (REMs) [3]. In orderto improve vehicle comfort, designers and manufacturers are constantly improving the technology of the engine mounting system. MagnetoRheological (MR) engine mount is a semi-active engine mount which uses MR Fluid (MRF) asits working fluid. A mathematical model of the system was expressed. Next, the stability analysis was studied for the system and stability condition was determined. Then, the controller methods were used to design appropriate controllers, and finally the results of the controller methods are compared with that of HEM [4]. A hierarchical fuzzy control (HFC) system for a magnetorheological fluids (MRF) mount was proposed to decrease the vertical vibration force and roll moment transmitted from an engine to a foundation [5]. A study presented the modelling, simulation and design of a semi-active engine mount that is designed specifically to address the complicated vibration pattern of variable displacement engines (VDE). The ideal isolation for VDE was required the stiffness to be switchable upon cylinder activation/deactivation operating modes [6].The vibration control of a passenger vehicle using an electronically controllable electro-rheological (ER) engine mount wasproposed and analyzedthrough a hardware-in-the-loop simulation (HILS), and control responses [7].Based on the 6-DOF vibration coupling model of the powertrain mounting system, an optimization algorithm was used to extract the best design parameters of each mount, thus rendering the mounting system fully decoupled and the natural frequency well configured

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and the optimal parameters were used to design the mounting system. Subsequently, vibration simulation analysis was applied to the mounting system, considering both transmission and road excitations [8]. An overview of recent advances in semi active engine mounts were presented, in term of working operation of Magnetorheological (MR) Fluid namely flow mode, shear mode, squeeze mode and mix mode. The issues were discussed with regard to the design and performance as vibration isolator device [9].

The main purpose of this study is to propose a comparison of the performance of semi-active hydraulic engine mounting system (SHEMs) with passivehydraulic engine mounting system (PHEMs)via vehicle ride comfort. A full-vehicle dynamic modelis establishedunder the combination of two excitation sources such as internal combustion engine and road surface excitations and a fuzzy logic controller is designed for control of the damping coefficient of a semi-active hydraulic engine mounting system. The ride performance of semi-active hydraulic engine mounting system is analyzed and compared with passive hydraulic engine mounting system (PHEMs) via vehicle ride comfort.

II. FULL-VEHICLE DYNAMIC MODEL

In order to evaluate the vehicle ride comfort performance of the semi-active hydraulic engine mount system (SHEMs) compared with passive hydraulic engine mounting system (PHEMs), a full-vehicle dynamic model is established under the combination of two excitation sources such as the internal combustion engine and road surface excitations, as shown in Fig.1.

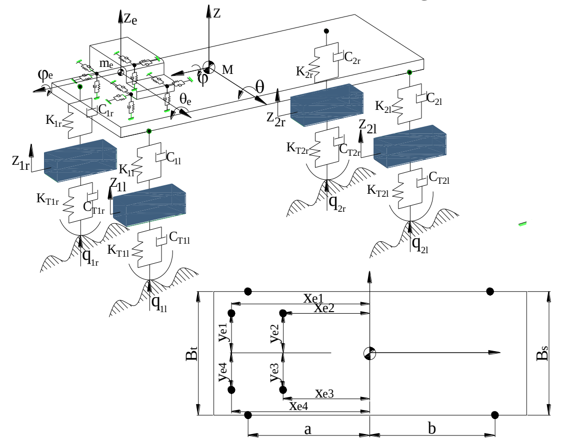

Figure 1. Full-vehicle dynamic model

In Fig. 1,K_{ie}arethe stiffness**c**oefficientsofthepassivehydraulic engine mounting system, C_{ie}aredamping coefficients ofthe passivehydraulic engine mounting system; K_{ij} are the stiffnesscoefficientsofvehicle suspension system; C_{ij} are the damping coefficients ofvehicle suspension system; K_{Tij} are the stiffnesscoefficientsoftires; $C_{Tij}$are the damping coefficients oftires; Z, φ, θ arethe vertical and angular displacements of vehicle body; Z_e, φ_e, $\theta_e$arethe vertical and angular displacements of engine body;$q_{ij}$are the road surface excitations; a, b, L, B_t and B_s are the distances; m_{ij} và M are the mass of axles and vehicle body; $m_e$is mass of engine (i=1,2,3,4 và j=r,l) (x_1,y_1); (x_2,y_2); (x_3,y_3) ; (x_4,y_4) are the coordinates of the force points of the four engine supports in the coordinate system via XYZ; (x_{1e},y_{1e}); (x_{2e},y_{2e}); (x_{3e},y_{3e}) ; (x_{4e},y_{4e}) are the coordinates of the force points of the four engine supports in the coordinate system via$X_eY_eZ_e$

The equations of motion of the bodies in Fig.1could be written by using a combined method of the multi-body system theory and D'Alembert's principle as follows.For example, the equations of motion for the vertical, pitch and roll motions of engine body are written by Eq.(1).

$$
\begin{aligned}
& m_e\ddot{z}_e = P_{ez} \\
& -\left[K_{e1}\left(Z_e + x_{2e}\varphi_e + y_{2e}\theta_e - Z - x_2\varphi + y_2\theta\right) + C_{e1}\left(\dot{Z}_e - x_{1e}\dot{\varphi}_e + y_{1e}\dot{\theta}_e - \dot{Z} - x_1\dot{\varphi} + y_1\dot{\theta}\right)\right] \\
& -\left[K_{e2}\left(Z_e - x_{2e}\varphi_e + y_{1e}\theta_e - Z - x_1\varphi + y_1\theta\right) + C_{e2}\left(\dot{Z}_e + x_{2e}\dot{\varphi}_e + y_{2e}\dot{\theta}_e - \dot{Z} - x_2\dot{\varphi} + y_2\dot{\theta}\right)\right] \\
& -\left[K_{e3}\left(Z_e + x_{3e}\varphi_e - y_{3e}\theta - Z + x_3\varphi + y_3\theta\right) + C_{e3}\left(\dot{Z}_e + x_{3e}\dot{\varphi}_e - y_{3e}\dot{\theta} - \dot{Z} + x_3\dot{\varphi} + y_3\dot{\theta}\right)\right] \\
& -\left[K_{e4}\left(Z_e + x_{4e}\varphi_e - y_{4e}\theta_e - Z + x_4\varphi + y_4\theta\right) + C_{e4}\left(\dot{Z}_e + x_{4e}\dot{\varphi}_e - y_{4e}\dot{\theta}_e - \dot{Z} + x_4\dot{\varphi} + y_4\dot{\theta}\right)\right]
\end{aligned} \tag{1}
$$

$$
\begin{aligned}
& I_{ye}\ddot{\varphi}_e = M_{ex} \\
& +\left[K_{e1}\left(Z_e + x_{2e}\varphi_e + y_{2e}\theta_e - Z - x_2\varphi + y_2\theta\right) + C_{e1}\left(\dot{Z}_e - x_{1e}\dot{\varphi}_e + y_{1e}\dot{\theta}_e - \dot{Z} - x_1\dot{\varphi} + y_1\dot{\theta}\right)\right].x_{1e} \\
& +\left[K_{e4}\left(Z_e + x_{4e}\varphi_e - y_{4e}\theta_e - Z + x_4\varphi + y_4\theta\right) + C_{e4}\left(\dot{Z}_e + x_{4e}\dot{\varphi}_e - y_{4e}\dot{\theta}_e - \dot{Z} + x_4\dot{\varphi} + y_4\dot{\theta}\right)\right].x_{4e} \\
& -\left[K_{e3}\left(Z_e + x_{3e}\varphi_e - y_{3e}\theta - Z + x_3\varphi + y_3\theta\right) + C_{e3}\left(\dot{Z}_e + x_{3e}\dot{\varphi}_e - y_{3e}\dot{\theta} - \dot{Z} + x_3\dot{\varphi} + y_3\dot{\theta}\right)\right].x_{3e} \\
& -\left[K_{e2}\left(Z_e - x_{2e}\varphi_e + y_{1e}\theta_e - Z - x_1\varphi + y_1\theta\right) + C_{e2}\left(\dot{Z}_e + x_{2e}\dot{\varphi}_e + y_{2e}\dot{\theta}_e - \dot{Z} - x_2\dot{\varphi} + y_2\dot{\theta}\right)\right].x_{2e}
\end{aligned}
$$

$$
\begin{aligned}
& I_{ye}\ddot{\varphi}_e = M_{ey} \\
& -\left[K_{e1}\left(Z_e + x_{2e}\varphi_e + y_{2e}\theta_e - Z - x_2\varphi + y_2\theta\right) + C_{e1}\left(\dot{Z}_e - x_{1e}\dot{\varphi}_e + y_{1e}\dot{\theta}_e - \dot{Z} - x_1\dot{\varphi} + y_1\dot{\theta}\right)\right].y_{1e} \\
& +\left[K_{e4}\left(Z_e + x_{4e}\varphi_e - y_{4e}\theta_e - Z + x_4\varphi + y_4\theta\right) + C_{e4}\left(\dot{Z}_e + x_{4e}\dot{\varphi}_e - y_{4e}\dot{\theta}_e - \dot{Z} + x_4\dot{\varphi} + y_4\dot{\theta}\right)\right].y_4 \\
& +\left[K_{e3}\left(Z_e + x_{3e}\varphi_e - y_{3e}\theta - Z + x_3\varphi + y_3\theta\right) + C_{e3}\left(\dot{Z}_e + x_{3e}\dot{\varphi}_e - y_{3e}\dot{\theta} - \dot{Z} + x_3\dot{\varphi} + y_3\dot{\theta}\right)\right].y_{3e} \\
& -\left[K_{e2}\left(Z_e - x_{2e}\varphi_e + y_{1e}\theta_e - Z - x_1\varphi + y_1\theta\right) + C_{e2}\left(\dot{Z}_e + x_{2e}\dot{\varphi}_e + y_{2e}\dot{\theta}_e - \dot{Z} - x_2\dot{\varphi} + y_2\dot{\theta}\right)\right].y_{2e}
\end{aligned}
$$

To determine the vertical forces ofthe passive hydraulic engine mount (PHEM), the dynamic model of PHEM is shown in Figure 2.

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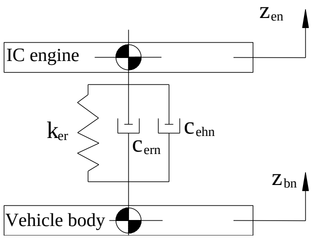


Figure 2. The dynamic model of hydraulic engine mounting system (PHEMs)

From Figure 2, the vertical forces of PHEM transmitting to engine and vehicle bodies[3] are defined as

$$F_{epn} = k_{ehn}\left(z_{en} - z_{bn}\right) + c_{ern}\left(\dot{z}_{en} - \dot{z}_{bn}\right) + c_{ehn}\left(\dot{z}_{en} - \dot{z}_{bn}\right) \qquad (2)$$

where, k_{ehn} and c_{ern} are the stiffness and damping coefficients of PHEM, c_{ehn} are the hydraulic damping coefficients of PHEM.

To determine the vertical forces ofthesemi-active hydraulic engine mount (PHEM), the dynamic model of PHEM is shown in Figure 3.

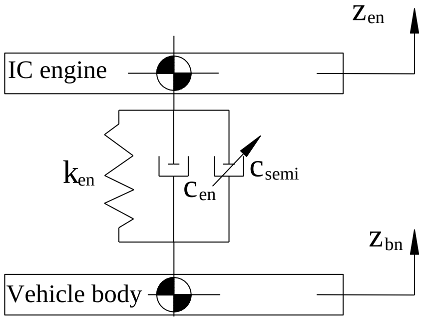


Figure 3. The dynamic model of semi-active hydraulic engine mounting system (SHEMs)

From Figure 3, the vertical forces of SHEM transmitting to engine and vehicle bodies [3] are defined as

$$F_{epn} = k_{ehn}\left(z_{en} - z_{bn}\right) + c_{ern}\left(\dot{z}_{en} - \dot{z}_{bn}\right) + c_{semi}\left(\dot{z}_{en} - \dot{z}_{bn}\right) \qquad (3)$$

where, k_{ehn} and c_{ern} are the stiffness and damping coefficients of PHEM, c_{semi} are the controldamping coefficients of SHEM.

Road surface excitation[3]:In this study, the filtering white noise method is used to describe the time domain excitation of the road surface based on reference [3] and time domain representation of the road surface can be given

$$\dot{q}(t) + 2\pi f_0 q(t) = 2\pi n_0 \sqrt{G_q(n_0)}\, v w(t) \qquad (4)$$

where, $G_q(n_0)$ is the road roughness coefficient which is defined for typical road classes from A (very good) to H (very poor) according to ISO 8068(1995) [7], v=f/n is the speed of vehicle from 10 m/s to 30 m/s, n is the road space frequency from 0.013 m^{-1} to 3.33 m^{-1}, and it can guarantee the temporal frequency of road surface f ranges from 0.33 Hz to 28.3 Hz which is the low excitation frequencies of road surfacetransmitted to vehicle body; f_0 is a minimal boundary frequency with a value of 0.0628 Hz;n_0 isareference spatial frequency which is equal to 0.1 m; w(t) is a whitenoise signal.

Internal combustion engine excitations [3]: In this study, the vertical inertia excitation force due to the reciprocating mass of engine, the roll and pitch excitation moments of engine with a 4-stroke in-line engine are defined as

$$F_{ez} = 4m_p r\lambda\omega^2\cos(2\omega t) = 4m_p r\lambda\omega^2\cos(2\pi ft) \qquad (5)$$

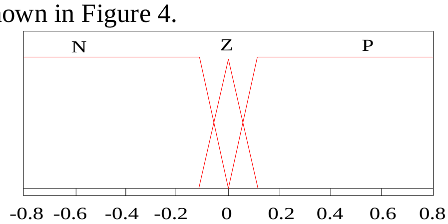
$$M_{ex} = M_e\left[1+1.3\sin(2\omega t)\right] = M_e\left[1+1.3\sin(2\pi ft)\right] \qquad (6)$$

$$M_{ey} = 4m_p r\lambda\omega^2 l_r\cos(2\omega t) = 4m_p r\lambda\omega^2 l_r\cos(2\pi ft) \qquad (7)$$

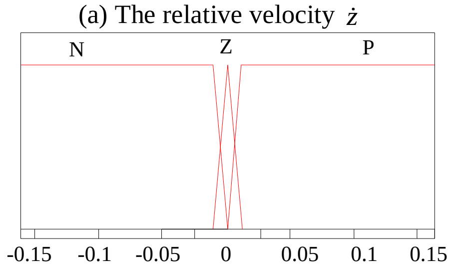
where, $\omega=2\pi f$ is the angular velocity of crank shaft, $f= n_e/60$ is the excitation engine frequency, n_e is the engine speed, m_p is the pistonmass, M_e is mean value of ICE torque M_e= -6.810$^{-6}n_e^2$ + 0.059n_e + 112.5 N.m, r is the rotational radius of crank arm, λ is the ratio of r to the length of the shaft, l_r isthe distance between the CG and the centre-line of the second and third cylinders.

III. FUZZY LOGIC CONTROLLER DESIGN FOR SHEM

Fuzzy logic-based control for semi-active hydraulic engine mounting system (SHEMs) is suggested and the capabilities for the improvement of ride comfort are studied through the software simulation. The relative displacement z and the relative velocity $\dot{z}$ are considered as two input variables while the damping coefficient of SHI, c_{csemi} is the output of the fuzzy control.The membership function form of fuzzy sets is selected as shown in Figure 4.



(a) The relative velocity $\dot{z}$



(b) The relative displacement z

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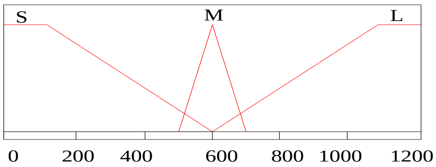
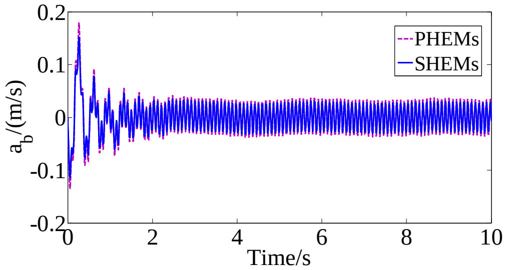


(c) The output of the fuzzy control, c_{semi}

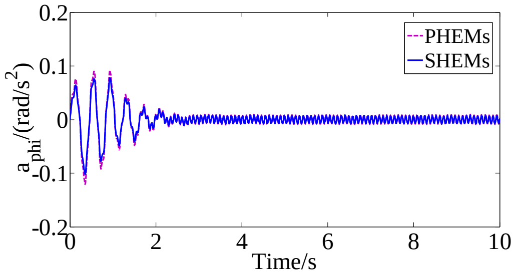
Figure 4. The membership function

VI. RESULTS AND ANALYSIS

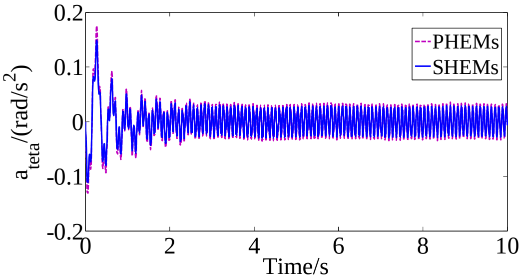
In order to compare the ride comfort performance of the semi-active hydraulic engine mounting system (SHEMs) with that of passivehydraulicengine mounting system (PHEMs) and design the fuzzy logic controller (FLC) for control SHEMs, Matlab/simulink software is used to solve the equations of motion in the above section with vehicle and engine parameters in references [11] and when the vehicle and engine operate under different road conditions. The simulation results of the time domain acceleration responses of the vertical motion (a_b), pitch and roll angles (a_{phi} and a_{teta}) of vehicle body with SHEMs compared to PHEMs are shown in Figure 5when ICE engine operates at the speed of 680prm (vehicle speed of 0 km/h).



(a) Vertical motion of vehicle body



(b) Pitch angle of vehicle body

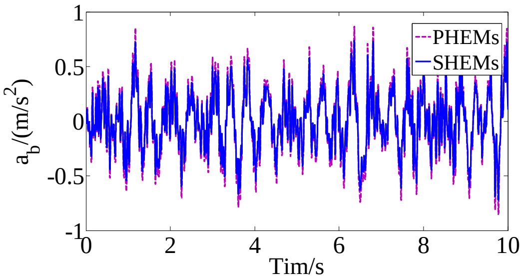


(c) Roll angle of vehicle body

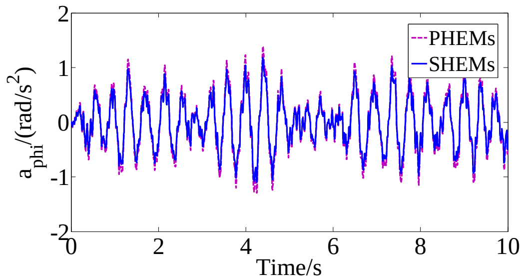
Figure 5. The simulation results of the time domain acceleration responses of the vertical motion (a_b), pitch and roll angles (a_{phi} and a_{teta}) of vehicle bodywhen ICE engine operates at the speed of 680prm (vehicle speed of 0 km/h).

From the achieved results in Fig.5, we show that the peak amplitude values of a_b, $a_{phi}$and a_{teta} with SHEMs respectively significantly reduce compared to PHEMs which indicates that the efficiency of the fuzzy logic controller has greatly improved the ride comfort.

The simulation results of the time domain acceleration responses of the vertical motion (a_b), pitch and roll angles (a_{phi} and a_{teta}) of vehicle body are shown in Figure 6when the vehicle moves on ISO class Bsurfaces road condition and ICE engine operates at the speed of 1680prm (vehicle speed of 72 km/h). Similarly,the obtained results of Figure 6 show that the peak amplitude values of the time domain acceleration responses of the vertical motion (a_b), pitch and roll angles (a_{phi} and a_{teta}) of vehicle body with SHEMsare respectively reduce in comparison with PHEMs.However, the peak amplitude values of the time domain acceleration responses of the vertical motion (a_b), pitch and roll angles (a_{phi} and a_{teta}) of vehicle bodyincrease rapidly under the road surface conditions.



(a) Vertical motion of vehicle body



(b) Pitch angle of vehicle body

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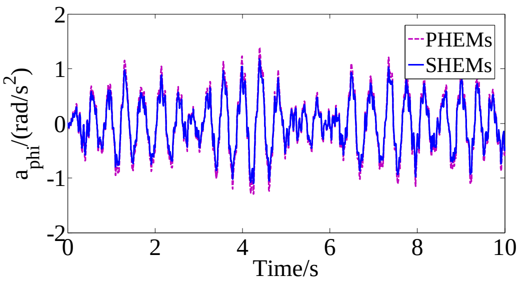

(c) Roll angle of vehicle body

Figure 6. The simulation results of the time domain acceleration responses of the vertical motion (a_b), pitch and roll angles (a_{phi} and a_{teta}) of vehicle bodywhen the vehicle moves on ISO class B surfaces road condition and ICE engine operates at the speed of 1680prm (vehicle speed of 72 km/h).

IV.CONCLUSION

In this study, a fuzzy logic controller is designed for control of the damping coefficient of a semi-active hydraulic engine mounting systemusing a full-vehicle dynamic modelunder the combination of two excitation sources such as the internal combustion engine and road surface excitationsto analyzethe ride comfort performance of SHEMscompared with that of PHEMsunder the different operating conditions. The majorconclusions drawn from the analysis can be summarizedas follows:(1) The peak amplitude values of the time domain acceleration responses of the vertical motion (a_b), pitch and roll angles (a_{phi} and a_{teta}) of vehicle body with SHEMsare respectively reduce in comparison with PHEMs and (2)The peak amplitude values of the time domain acceleration responses of the vertical motion (a_b), pitch and roll angles (a_{phi} and a_{teta}) of vehicle bodyincrease rapidly under the road surface conditions.

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