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Fuzzy control for a semi-active hydraulic engine mounting system of a passenger car

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ABSTRACT: *To improve the ride comfort of a passenger car, an improved semi-active hydraulic engine mounting system (improved SHEMs) is proposed to evaluate the ride comfort performance of its compared with semi-active hydraulic engine mounting system recommended by reference (SHEMs). A fuzzy logic controller with an improved control law is designed for control of the damping coefficient of a semi-active hydraulic engine mounting system. The obtained results indicate that the peak amplitude values of the time domain acceleration responses of vehicle body with improved SHEMs respectively reduce in comparison with SHEMs under different survey conditions. Especially, the values of the root mean square (r.m.s) acceleration responses of vehicle body (a_{wb}, a_{wphi} and a_{wteta}) with improved SHEMs reduce by 8.68%, 7.52% and 7.54% respectively in comparison with SHEMs.*

KEYWORDS: *Passenger car, Engine mounts, hydraulic engine mounts, semi-active hydraulic mounts*

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I. INTRODUCTION

Internal combustion engine (ICE) vibration not only affects the vehicle noises but also affects the vehicle ride comfort, especially, the low-frequency vibration excitation sources greatly affects the vehicle ride comfort. Engine vibration can also be generated by road pattern and transmitted from chassis. Engine mounts and bushings are devices that are used to isolate engine and chassis/body from each other. Engine hydraulic mount are designed to address the low frequency part of the excitation, which is usually the most important part. The higher frequency region is automatically filtered by soft materials inside the mount such as the main rubber. A method to improve vehicle ride comfort using additional damping coefficient values for an internal combustion engine (ICE) rubber mounting system was proposed to analyze the effect of the adding damping coefficient values into the rubber mounting system on vehicle ride comfort using a full-vehicle vibration model with 10 degrees of freedom under the combination of road surface roughness and ICE excitations [1]. A new semi-active magnetorheological (MR) engine mount in half car model was proposed for improving ride comfort. A dynamic sliding mode controller was developed for controlling the magnetic field strength of the engine mount coil [2] 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) was described to control the pressure regulator [3]. A new type of active engine mount system based around the use of the bellows, a voice coil motor and an accumulator was developed and The final design has been shown to reduce the transmitted forces significantly while it has a relatively small electrical power consumption [4]. The magnetorheological fluids (MRF) mount in squeeze mode for engine isolation was proposed and its damping characteristics were analyzed. A hierarchical fuzzy control (HFC) system was proposed to decrease the vertical vibration force and roll moment transmitted from an engine to a foundation based on an engine isolation dynamic model with three degrees of freedom [5].

For whole vehicle ride dynamic model, a 3D dynamic model with 14 degrees of freedom was developed with the dynamic models of the traditional and new air suspension systems to compare the performance of the air suspension systems for reducing the negative impacts on the road surface when vehicle moves on the different road conditions [6]. A full vehicle model with 15 degrees of freedom was established to analyze the performance of the hydro-pneumatic suspension system of heavy truck on the ride quality of road surfaces [7]. A 3D nonlinear dynamic model of a typical heavy truck with 14 degrees of freedom was established to analyze the effects of different road surface conditions on the safety of vehicle movement and the durability of parts of a vehicle [8]. A three-dimensional vibration model of bus with 10 DOF (degree of freedom) based on Dragan Sekulić model was proposed to analyze the suspension parameters directly influenced ride comfort [9]. However, these dynamic models do not take into account the internal combustion engine model and the influence of ICE mounting system. A full-vehicle dynamic model with 10 degree of freedoms was proposed to investigate the effect of internal combustion engine vibrations on vehicle ride comfort based on the value of the root mean square (RMS) of acceleration responses of the vertical, pitch, and roll vibrations of vehicle body according to the international standard ISO 2631-1 [10]. A full-vehicle dynamic model under the combination of two excitation sources such as internal combustion engine and road surface excitations was proposed to analyze the comparison of the performance of semi-active hydraulic engine mounting system (SHEMs) with passive hydraulic engine mounting system (PHEMs) via vehicle ride comfort.

The main purpose of this study is to establish a full-vehicle dynamic model with 10 degree of freedoms based on reference [10, 11] to evaluate a comparison of the performance of improved semi-active hydraulic engine mounting system (Improved SHEMs) with semi-active hydraulic engine mounting system (PHEMs) [11] via vehicle ride comfort. A fuzzy logic controller with an improved control law is designed for control of the damping coefficient of a semi-active hydraulic engine mounting system, Finally, the efficiency of the proposed controller is evaluated based on the value of the root mean square (RMS) of acceleration responses of the vertical, pitch, and roll vibrations of vehicle body according to the international standard ISO 2631-1 [14].

II. FULL-VEHICLE DYNAMIC MODEL

A vehicle structure consisting of two axles, four suspension systems, four engine mounts is chosen to set up vehicle dynamic model and a full-vehicle dynamic model of a passenger car is established based on model reference [11], as shown in Fig.1.

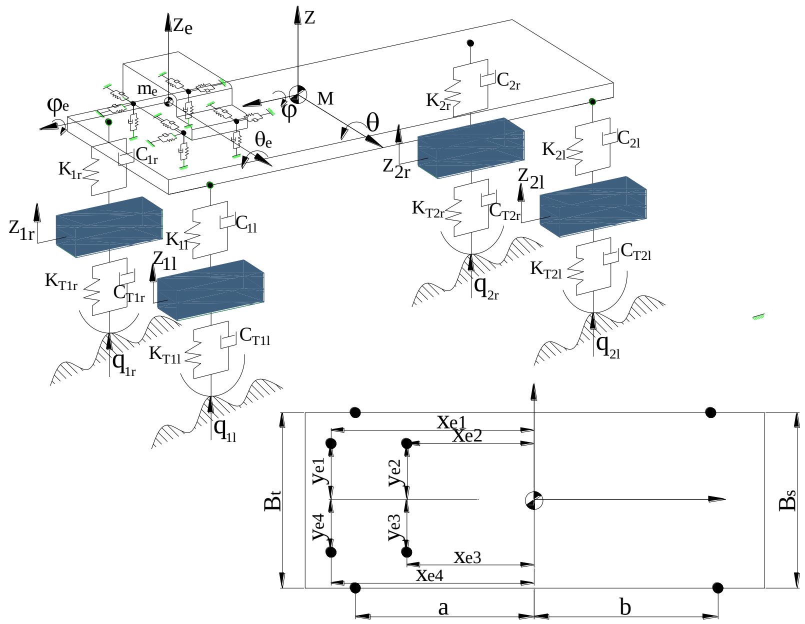


Figure 1. Full-vehicle dynamic model of a passenger car

The equations of motion of the bodies in Fig.1 could be written by using a combined method of the multi-body system theory and D'Alembert's principle as follows.

The vertical balance force of ICE is determined below

$$m_e \ddot{z}_e = F_{ez} - \left(F_{e1} + F_{e2} + F_{e3} + F_{e4}\right) \tag{1}$$

The moment balance equations ICE are determined below

$$I_{ey} \ddot{\varphi}_e = M_{ey} + F_{e1}x_{e1} + F_{e4}x_{e4} - F_{e2}x_{e2} - F_{e3}.x_{e3} \tag{2}$$

$$I_{ex} \ddot{\theta}_e = M_{ex} + F_{e3}y_{e3} + F_{e4}.y_{e4} - F_{e1}y_{e1} - F_{e2}y_{e2} \tag{3}$$

The vertical balance force of vehicle body is determined below

$$m_b \ddot{z}_b = (F_{e1} + F_{e2} + F_{e3} + F_{e4}) - (F_{1r} + F_{1l} + F_{2l} + F_{2r}) \tag{4}$$

The moment balance equations vehicle body are determined below

$$I_{by} \ddot{\varphi} = (F_{1r} + F_{1l})a - (F_{2r} + F_{2l})b - (F_{e1}x_1 + F_{e2}x_2 + F_{e3}x_3 + F_{e4}x_4) \tag{5}$$

$$I_{bx} \ddot{\theta} = F_{1l}\frac{B_t}{2} + F_{2r}\frac{B_s}{2} - F_{1r}\frac{B_t}{2} - F_{2l}\frac{B_s}{2} + (F_{e1}y_1 + F_{e2}y_2) - (F_{e3}y_3 + F_{e4}y_4) \tag{6}$$

The vertical balance forces of vehicle axles are determined below

$$m_{1r}\ddot{z}_{1r} = F_{1r} - F_{T1r} \quad (7)$$

$$m_{1l}\ddot{z}_{1l} = F_{1l} - F_{T1l} \quad (8)$$

$$m_{2r}\ddot{z}_{2r} = F_{2r} - F_{T2r} \quad (9)$$

$$m_{2l}\ddot{z}_{2l} = F_{2l} - F_{T2l} \quad (10)$$

where, F_{e1}, F_{e2}, F_{e3}, and F_{e4} are the vertical forces of engine mounts; F_{1r}, F_{1l}, F_{2r}, and F_{2l} are the vertical forces of vehicle suspension systems; F_{T1r}, F_{T1l}, F_{T2r}, and F_{T2l} are the vertical forces of tires.

To determine the vertical forces of the semi-active hydraulic engine mount system (PHEM)s, the dynamic model of PHEM is shown in Fig. 2.

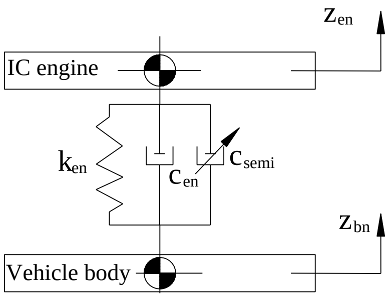


Figure 2. The dynamic model of semi-active hydraulic engine mounting system (SHEMs)[11].

From Fig.2, the vertical forces of SHEM transmitting to engine and vehicle bodies are defined as

$$F_{ensemi} = k_{en}\left(z_{en} - z_n\right) + c_{en}\left(\dot{z}_{en} - \dot{z}_n\right) + c_{nsemi}\left(\dot{z}_{en} - \dot{z}_{bn}\right) \quad (11)$$

where, k_{en} and c_{en} are the stiffness and damping coefficients of passive hydraulic engine mounting system (PHEMs), c_{nsemi} are the control damping coefficients of SHEMs (n=1÷4).

Road Surface excitation [10, 11]

To evaluate the effect the internal combustion engine vibrations on vehicle ride comfort, the road surface roughness is road excitation which is simulated in space domain and acts as an input to the vehicle-road model. In this study, the road surface roughness is simulated according to the International Standards Organization (ISO) 8608 [13]. A road surface roughness is usually assumed to be a zero-mean stationary Gaussian random process and can be generated through an inverse Fourier transformation based on a power spectral density (PSD) function [10, 11]. The road surface roughness is generated as the sum of a series of harmonics:

$$q(t) = \sum_{k=1}^{N} \sqrt{2G_q\left(n_{mid-k}\right)\Delta n_k} \cos\left(2\pi n_{\mathrm{mid}-k} t + \phi_k\right). \quad (12)$$

where, the spatial frequency range, $n_1 < n < n_2$, is divided into several uniform intervals which have a width of Δn_k; $G_q(n)$ is PSD function (m^3/cycle/m) for the road surface elevation, the power density $G_q(n)$ in every small interval is substituted by $G_q(n_{mid-k})$, where n_{mid-k} (k=1, 2, ... , n) is center frequency among its intervals; n_k is the wave number (cycle/m); ϕ_k is the random phase uniformly distributed from 0 to 2π.

ICE excitations [10,11]

The engine is supported by three mounts arranged vertically and both the foundation and the engine are assumed to be rigid, the foundation has a large mass and the mount mass is ignored, as shown in Fig 1. The vertical inertia force due to the reciprocating mass of the engine and the roll excitation moment of the engine, and it is herein called the torque, is a result of the torque from the inertia force and the gas explosion pressure and pitch excitation moment of the engine [5] is defined as

$$F_{ez} = 4m_c r\lambda\omega_0^2\cos(2\omega_0 t). \quad (13)$$

$$M_{ex} = M_e\left[1 + 1.3\sin(2\omega_0 t)\right]. \quad (14)$$

$$M_{ey} = 4m_c r\lambda\omega_0^2\, l\cos(2\omega_0 t). \quad (15)$$

where, m_c is the reciprocating mass of a piston, r is the radius of a crank, λ is the ratio of r to the length of the shaft, ω_0 is the rotational frequency of the crank, l is the distance between the CG and the centre-line of the second and third cylinders, M_e is mean value of torque.

III. FUZZY LOGIC CONTROLLER DESIGN

In this study, Fuzzy logic-based control for semi-active hydraulic engine mounting system (SHEMs) of aa passenger car is suggested and the capabilities for the improvement of ride comfort are studied through the software simulation. The relative displacements $z_n = (z_{en} - z_n)$ and the relative velocities $\dot{z}_n = (\dot{z}_{en} - \dot{z}_n)$ are considered as two input variables while the damping coefficient of SHEM, c_{nsemi} is the output of the fuzzy control. The three values of the linguistic variables of input/output signals are defined by the positive (P), zero (Z), negative small (N). The rules of "if-then" are used to define the relationship of z_n, $\dot{z}_n$ and c_{nsemi} according to the designers' knowledge and experience, the fuzzy controller has all 9 possible rules listed in Tab. 1. Mamdani's fuzzy inference system is used to control SHEMs of a passenger car.

Table 1. Rules for fuzzy control

$\dot{z}_n$	z_n	**c_{nsemi}**	$\dot{z}_n$	z_n	**c_{nsemi}**
N	N	S	Z	P	M
N	Z	M	P	N	L
N	P	L	P	Z	M
Z	N	M	P	P	S
Z	Z	M			

Improved control law: The nine value of the linguistic variables of input/output signals are defined by the positive very big (PVB), positive big (PB), positive medium (PM), positive small (PS), zero (Z), negative small (NS), negative medium (NM), negative big (NB), negative very big (NVB). The rules of "if-then" are used to define the relationship of z_n, $\dot{z}_n$ and c_{nsemi} according to the designers' knowledge and experience, the fuzzy controller has all 81 possible rules listed in Tab.2. Mamdani's fuzzy inference system is used to control SHEMs of a passenger car.

Table 2. Rules for fuzzy control

c_{nsemi}		z_n								
		NVB	**NB**	**NM**	**NS**	**ZE**	**PS**	**PM**	**PB**	**PVB**
$\dot{z}_n$	**NVB**	ZE	PVB	PM	PS	ZE	NS	NS	NM	NM
	NB	NM	ZE	PB	PS	ZE	NS	NM	NM	NM
	NM	NVB	NB	ZE	PM	ZE	NM	NM	NM	NB
	NS	NVB	NVB	NVB	ZE	ZE	NM	NB	NB	NB
	ZE	NVB	NVB	NVB	NVB	ZE	NVB	NVB	NVB	NVB
	PS	NB	NB	NB	NM	ZE	ZE	NVB	NVB	NVB
	PM	NB	NM	NM	NM	ZE	PM	ZE	NB	NVB
	PB	NM	NM	NM	NS	ZE	PS	PB	ZE	NM
	PVB	NM	NM	NS	NS	ZE	PS	PM	NVB	ZE

IV. RESULTS AND DISCUSSION

Matlab/simulink software is used to solve the equations of motion in the above section with vehicle and engine parameters in references [12] 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 Fig. 3 under the road condition and ICE engine operates at the speed of 1780 prm (vehicle speed of 76.6 km/h). The obtained results of Fig. 3 show that the peak amplitude values of the time domain acceleration responses of the vertical motion of vehicle body (a_b), the pitch and roll angles of vehicle body (a_{phi} and a_{teta}) of vehicle body with improved SHEMs are respectively reduce in comparison with SHEMs[11].

The values of the root mean square (r.m.s) acceleration responses of motion, pitch and roll angles of vehicle body (a_{wb}, a_{wphi} and a_{wteta}) are determined by Eq.(16) according to ISO 2631-1: Mechanical vibration and shock- Evaluation of human exposure to whole-body vibration to compare with improved SHEMs and with

SHEMs[11]. The a_{wb}, a_{wphi} and a_{wteta} values in comparison with ICE vibrations and without vibrations are shown in Fig.3. The root-mean-square (r.m.s.) acceleration [14] is defined as

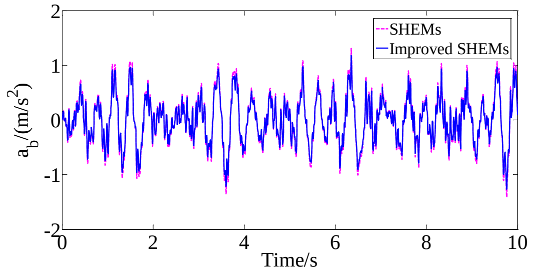
$$a_w = \left[\frac{1}{T} \int_0^T a^2(t)dt \right]^{1/2} . \tag{16}$$

where, a(t) is the weighted acceleration as a function of time, m/s^2; T is the duration of the measurement or simulation time, s.

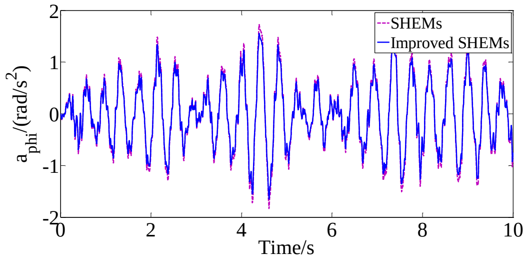
Table 3. a_{wb}, a_{wphi} and a_{wteta} values with SHEMs[11] and improved SHEMs

Change control law	a_{wb} (m/s^2)	a_{wphi} (rad/s^2)	a_{wteta}(rad/s^2)
SHEMs[11]	0.4525	0.6816	0.6013
Improved SHEMs	0.4163	0.6339	0.5592
Decrease (%)	8.69	7.52	7.54

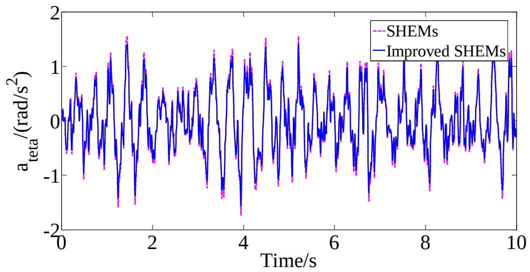
The results in Tab.3 indicate that the values of the root mean square (r.m.s) acceleration responses of vehicle body *(a_{wb}, a_{wphi} and a_{wteta})* with improved SHEMs reduce by 8.68%, 7.52% and 7.54% respectively in comparison with SHEMs[11].



(a) Vehicle body vertical motion



(b) Vehicle body pitch angle



(c) Vehicle body roll angle

Figure 3. The time domain acceleration responses of vehicle body in comparison with improved SHEMs and with SHEMs[11]

V. CONCLUSION

In this study, a fuzzy logic controller with an improved control law is designed for control of the damping coefficient of a semi-active hydraulic engine mounting system using a full-vehicle dynamic model under the combination of two excitation sources such as the internal combustion engine and road surface excitations[10, 11] to evaluate a comparison of the performance of improved semi-active hydraulic engine mounting system (Improved SHEMs) with semi-active hydraulic engine mounting system (PHEMs) [11] via vehicle ride comfort. The obtained results indicate 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 improved SHEMs are respectively reduce in comparison with SHEMs [11]. That indicates that the optimization of the control laws has significantly improved the controller efficiency.

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