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Optimal design parameters of a combustion engine mounting system using a genetic algorithm (GA)

Nguyen Dinh Tan[1], Vu Thi Hien[2], Hoang Anh Tan[3*]
[1]Faculty of Mechanical Engineering, East Asia University of Technology, Bac Ninh, Vietnam
[2,3]Faculty of Vehicle and Energy Engineering, Thai Nguyen University of Technology, Thai Nguyen, Vietnam
Email: hoanganhtan@tnut.edu.vn

ABSTRACT:
The proposal of this paper is to use Genetic Algorithm (GA) to find out a set of the optimal design parameters of engine passive mounting system. A full-vehicle dynamic model with 10 degrees of freedom is established under the combination of two excitation sources such as internal combustion engine and road surface excitations. The GA genetic algorithm and the weighted sum method are combined to find a set of the optimal design parameters. The study results indicate that the root mean square (r.m.s) values of acceleration responses of the vertical vehicle body (a_{wz}), vehicle body pitch angle (a_{wphi}) and vehicle body roll angle (a_{wphi}) with GA optimal parameters greatly reduce by 10.2%, 8.2%, and 7.9% in comparison with the original parameters of engine mounting system which means that the performance optimization of engine mounting system is better than the original engine mounting system improving the ride comfort.

KEYWORDS: *Mounting system, Genetic Algorithm, Design parameter, Ride comfort.*

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I. INTRODUCTION

The different kinds of engine mounting systems, from elastomeric to hydraulic, and from passive to active, have been developed to improve the engine mount performance such as vehicle ride comfort and noise. The performance requirements of engine mounting systems, the different kinds of engine mounts and mounting optimum tuning were reviewed and discussed by Yunhe Yu and associates [1]. The application of neural network as a controller to isolate engine vibration in an active engine mounting system was proposed by Fadly Jashi Darsivan and associates [2]. The performance of the neural network controller was compared with conventional PD and PID controllers tuned using Ziegler-Nichols. The active mount based on the smart material, i.e., Terfenol-D rod was proposed by Zhiyuan Si and associates [3] which mainly includes three parts: rubber spring, magnetostrictive actuator (MA), and hydraulic amplification mechanism (HAM). The x-LMS state feedback controller with the system state as the reference signal was constructed by employing Sage-Husa Kalman filter to realize the state estimation of the active mounting system with the consideration of the unmeasurable state parameters in the active mounting system. The analytical model of active ACM in powertrain was developed and implemented in MATLAB by Zhengchao Xie and associates [4]. The control strategy was integrated into the analytical model by using the linear quadratic regulator (LQR) method. the behavior of MR damper was studied and used in implementing vibration control by Banna Kasemi and and associates [5]. The methodology was adopted to get a control structure was based on the experimental results. The methodology was adopted to get a control structure was based on the experimental results. A method to improve vehicle ride comfort using additional damping coefficient values for an internal combustion engine (ICE) rubber mounting system was proposed by Hoang Anh Tan and associates 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 [6]. A full-vehicle dynamic model under the combination of two excitation sources such as internal combustion engine and road surface excitations was proposed by Ta Tuan Hung and associates to assess the vehicle ride comfort performance between the hydraulic engine mount system (HEMs) and rubber engine mount system (REMs) [7]. The main objective of this paper is to find out a set of the optimal design parameters of engine passive mounting system using a full-vehicle dynamic model with 10 degrees of freedom under the combination of two excitation sources such as internal combustion engine and road surface excitations and genetic algorithm (GA).

II. VEHICLE DYNAMIC MODEL

In order to find out the optimal parameters of the internal combustion engine mounting system, a full-vehicle dynamic model with 10 degrees of freedom under the combination of two excitation sources such as internal combustion engine and road surface excitations is established, as shown in Fig.1.

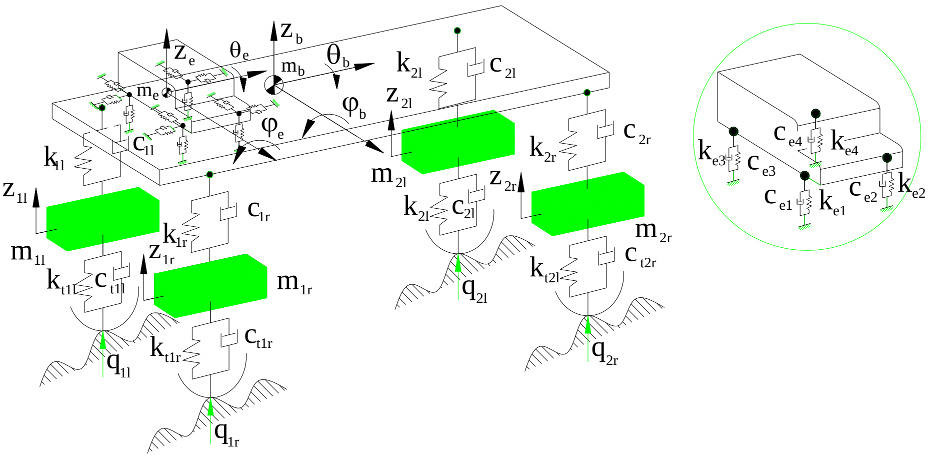


Figure 1. Full- vehicle dynamic model

Interpretation of symbols on figure 1, m_b and m_1, m_2 are the sprung of vehicle body and unsprung axle masses, respectively; k_{tij} and c_{tij} are the stiffness and damping coefficients of tires; k_{ij} and c_{ij} are the stiffness and damping coefficients of vehicle suspension system, respectively; k_{en} and c_{en} are the stiffness and damping coefficients of vehicle suspension system, respectively $z_{ij,}$ z_b and z_e are the vertical displacements of the sprung, unsprung, and engine masses; φ_b, φ_e and θ_b, θ_e are the pitch and roll angle displacements of vehicle body and engine, respectively (i=1÷2, j=r,l, n=1÷4).

From Fig.1, the motion equations of vehicle dynamic using Newton's second law are written as follows

$$\begin{cases} m\ddot{z}_{1l} = F_{1l} - F_{t1l} \\ m\ddot{z}_{1r} = F_{1r} - F_{t1r} \\ m\ddot{z}_{2l} = F_{2l} - F_{t2l} \\ m\ddot{z}_{2r} = F_{2r} - F_{t2r} \\ m_b\ddot{z}_b = (F_{e1} + F_{e2} + F_{e3} + F_{e4}) - (F_{1r} + F_{1l} + F_{2l} + F_{2r}) \\ I_{by}\ddot{\varphi}_b = (F_{1r} + F_{1l}).a - (F_{2r} + F_{2l}).b - (F_{e1}.x_1 + F_{e2}.x_2 + F_{e3}.x_3 + F_{e4}.x_4) \\ I_{bx}\ddot{\theta}_b = F_{1l}\dfrac{B_t}{2} + F_{2r}\dfrac{B_s}{2} - F_{1r}\dfrac{B_t}{2} - F_{2l}\dfrac{B_s}{2} + (F_{e1}y_1 + F_{e2}y_2) - (F_{e3}y_3 + F_{e4}y_4) \\ m_e\ddot{z}_e = F_z - \left(F_{e1} + F_{e2} + F_{e3} + F_{e4}\right) \\ I_{ey}\ddot{\varphi}_e = M_y + F_{e1}x_{e1} + F_{e4}x_{e4} - F_{e2}x_{e2} - F_{e3}x_{e2} \\ I_{ex}\ddot{\theta}_e = M_x + F_{e3}y_{ee} + F_{e4}y_{e4} - F_{e1}y_{e1} - F_{e2}y_{e2} \end{cases} \quad (1)$$

where, 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; F_{e1}, F_{e2}, F_{e3}, and F_{e4} are the vertical forces of engine mounts.

Road Surface excitation [6,10] 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:

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

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 [6,11]: 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 is defined as

$$F_{ez} = 4m_c r\lambda\omega_0^2 \cos(2\omega_0 t). \tag{3}$$

$$M_{ex} = M_e\left[1+1.3\sin(2\omega_0 t)\right]. \tag{4}$$

$$M_{ey} = 4m_c r\lambda\omega_0^2\, l\cos(2\omega_0 t). \tag{5}$$

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. OPTIMAL FUNCTION VIA GENETIC ALGORITHM (GA)

The individuals or chromosomes in the current generation are used to produce the next population through the subsequent steps: (1) Score each number of the current population by computing its fitness value and scales the raw scores to convert them into a more practical range of values; (2) Choose parents based on their fitness values; (3) Perform elitist selection, in which some of the better individuals in the current population are allowed to carry over to the next population, unchanged; (4) Produce children from the parents either by mutation or crossover; (5) Replace the current population with the children to form next generation and (6) stop- when one of the stopping criteria is met [10].

Objective function: The process of optimal design parameters of engine mounting system is carried out using a multi-objective optimization function formed by the root mean square (r.m.s) values of acceleration responses of the vertical vehicle body (a_{wz}), vehicle body pitch angle (a_{wphi}) and vehicle body roll angle (a_{wphi}) according to the ISO 2631:1997(E) standard [13]. To obtain the optimal design variables values, the objective is minimized a multi-objective function shown below

$$F(x) = \mathrm{w}_1\left\{a_{wz}(x)\right\} + \mathrm{w}_2\left\{a_{wphi}(x)\right\} + \mathrm{w}_3\left\{a_{wteta}(x)\right\} \to \min \tag{6}$$

$$s.t\begin{cases} x = \left[k_{ei}, c_{ei}\right] \\ \Delta z = \left|\left(z_{ei} - z_i\right)\right| \le 0.006 \\ k_{ei}^{\mathrm{low}} \le k_{ei} \le k_{ei}^{up} \\ c_{ei}^{\mathrm{low}} \le k_{ei} \le c_{ei}^{up} \end{cases} \Leftrightarrow \quad s.t\begin{cases} x = \left[k_{ei}, c_{ei}\right] \\ \Delta z = \left|\left(z_{ei} - z_i\right)\right| \le 0.006 \\ 1.0 \le \dfrac{1}{2\pi}\sqrt{\dfrac{k_{ei}}{m_e}} \le 2.0 \\ 0.2 \le \dfrac{c_{ei}}{2\sqrt{m_e k_{ei}}} \le 0.4 \end{cases}$$

where: the weight of the objective functions is chosen such as w_1=0.4, w_2=0.3 and w_2=0.3.

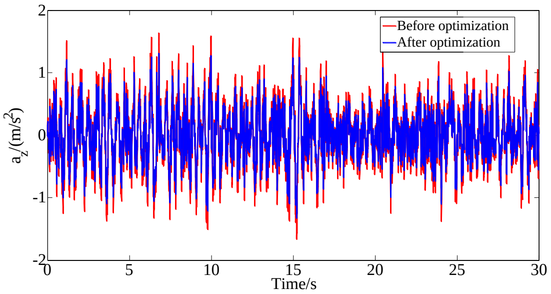
The root mean square (r.m.s) values of acceleration responses of vehicle body based on ISO 2631-1 (1997) [9] is defined by

$$a_w = \left[\frac{1}{T}\int_0^T a^2(\mathrm{t})\,\mathrm{dt}\right]^{\frac{1}{2}} \tag{7}$$

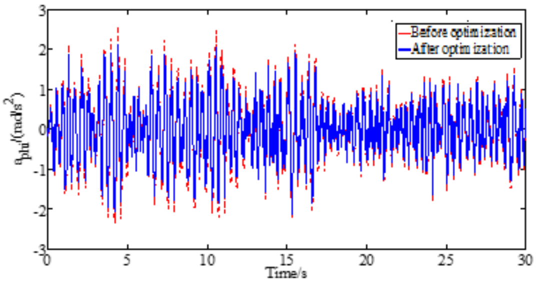
where, a(t) is the weighted acceleration (translational and rotational) as a function of time, m/s^2; T is the duration of the measurements.

IV. RESULTS AND DISCUSSION

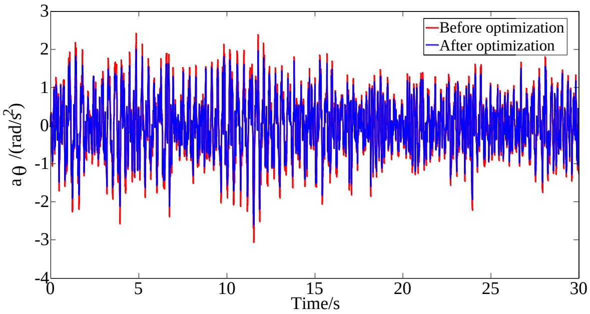
In order to find out the optimal design parameters of engine mounting system for a passenger car. Matlab/Simulink environment with the initial vehicle design parameters based on reference [10]. A program of genetic algorithm is written in Matlab to declare input parameters such as objective function and boundary conditions Eq.(6) with w_1=0.4, w_2=0.3 and w_2=0.3 and GA parameters such as population size as 100 and generation as 150, which called by Simulink module function using the sim function. The acceleration responses of the vertical vehicle (a_z) and vehicle body pitch angle (a_{phi}) with GA optimal parameters in comparison with the original parameters of engine mounting system are shown Fig.2 when vehicle moves on ISO class B road condition at vehicle speed of 20 m/s. The obtained results of Fig.2 show that the peak amplitude values of the time domain acceleration responses of the acceleration responses of the vertical vehicle (a_z) and vehicle body pitch angle (a_{phi}) with GA optimal parameters of engine mounting system are respectively reduce in comparison with original design parameters of engine mounting system.



(a) Vertical vehicle body



(b) Vehicle body pitch angle



(c) Vehicle body roll angle

Figure 2. Acceleration responses of the vertical vehicle (a_z) and vehicle body pitch angle (a_{phi}) with GA optimal parameters in comparison with the original parameters of engine mounting system [10].

From the results of Fig.2, we could determine the a_{wb}, a_{wphi}, and a_{wteta} values by Eq.(7) which the a_{wb}, a_{wphi}, and a_{wteta} are 0.2724 m/s^2, 0.4529 rad/s^2, and 0.4391 rad/s^2 with the original parameters and 0.2473 m/s^2, 0.4187 rad/s^2, and 0.4068 rad/s^2 with GA optimal parameters of engine mounting system. The results show that the a_{wb}, a_{wphi}, and a_{wteta} values with GA optimal parameters greatly reduce by 10.2%, 8.2%, and 7.9% in comparison with the original parameters of engine mounting system, which means that the performance optimization of engine mounting system is better than the original engine mounting system improving the ride comfort.

V. CONCLUSION

In this study, a full-vehicle dynamic model with 10 degrees of freedom under the combination of two excitation sources such as internal combustion engine and road surface excitations is established to find out the optimal parameters of engine mounting system using GA to improve the ride comfort when the vehicle moves on the ISO class B road surface at the vehicle speed of 20 m/s^2. The major conclusions that can be drawn from the analysis results as follows: The peak amplitude values of the time domain acceleration responses of the acceleration responses of the vertical vehicle (a_z) and vehicle body pitch angle (a_{phi}) with GA optimal parameters of engine mounting system are respectively reduce in comparison with original design parameters of engine mounting system. The a_{wb}, a_{wphi}, and a_{wteta} values with GA optimal parameters greatly reduce by 10.2%, 8.2%, and 7.9% in comparison with the original parameters of engine mounting system, which means that the performance optimization of engine mounting system is better than the original engine mounting system improving the ride comfort.

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