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Effects of Internal Combustion Engine Vibrations on Vehicle Ride Comfort

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***Abstract*— The vibration sources of the internal combustion Engine (ICE) are widely known to affect not only the noise but also the vehicle ride comfort. In order to evaluate their effects on vehicle ride comfort, a full-vehicle dynamic model with 10 degree of freedoms combining from road surface roughness and ICE excitations is established. The vehicle ride comfort with ICE vibrations is compared to that without ICE vibration based on the values of the root mean square (r.m.s) acceleration responses of motion, pitch and roll angles of vehicle body according to the international standard ISO 2631-1 (1997). The results indicate that the ICE vibrations have a significant effect on vehicle ride comfort. The study results also provide a theoretical basis for designers and researchers in the field of dynamics and controlling of vehicle and ICE systems**

***Index Terms*—ICE, Vibration source, ICE noise, Vehicle ride comfort**

I. INTRODUCTION

Internal combustion engine vibrations have impact on vehicle noise, so do they on vehicle ride comfort. The unbalanced forces produced from the engine are transferred to the engine supporting parts, causing the structure borne vibrations. Vibration behavior of an internal combustion engine depends on unbalanced reciprocating and rotating parts, cyclic variation in gas pressure, shaking forces due to the reciprocating parts and structural characteristics of the mounts [1]. In order to reduce the engine vibrations to vehicle body as well as prevent the vehicle ones to the engine body, the engine mount system is equipped with an elastic part connecting the vehicle body to the engine body. A study of the crankshaft torsional vibration phenomenon in internal combustion engines was proposed and analyzed via the model of the six cylinders diesel engine manufactured by MWM International Motores [2]. A dynamic model of an internal combustion engine OM-355 was proposed to survey the vibrations of the main components of the engine due to combustion on engine noise [3]. The different vibration sources of internal combustion engine vibration signals from valve cover, cylinder head and cylinder block were analyzed by using the short-term Fourier-transform (STFT) [4].

For the study of the characteristics of the engine mount systems, a proposed method to improve vehicle ride comfort was using additional damping coefficient values for an internal combustion engine (ICE) rubber mounting system. These values were examined in term of their effects on vehicle ride comfort when using a full-vehicle vibration model with 10 degrees of freedom [5]. The characteristics of rubber mountings of the engine system were proposed to perform a vibration analysis of a vehicle using a measurement methodology [6]. The hydraulic engine mounts (HEMs) was offered to analyze the characteristics of the vehicle engine shake performance using a full vehicle model with 14 degree of freedoms (DOFs) [7,14]. Three different types of engine mounting systems such as rubber, hydraulic and semi-active engine mounting systems were suggested to consider the effects on vehicle ride comfort using a full vehicle ride dynamics model under the combination of two excitation sources such as Internal Combustion Engine (ICE) and road surface excitations [8]. However, most of the research results have not mentioned the effect of ICE vibrations on vehicle ride comfort. An engine isolation system via tuneable damping mounts was proposed and controlled to improve their effectiveness via a coordination strategy for tuning the damping of each MRF mount in order to minimize the transmitted torque. An electronically controllable electrorheological (ER) engine mount was offered to investigate its effects on vehicle ride comfort via a full-vehicle dynamic model [12].

The main purpose of this study is to propose a full-vehicle dynamic model with 10 degree of freedoms to investigate the effect of internal combustion engine vibrations on vehicle ride comfort which is analyzed 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[13]

II. FULL-VEHICLE DYNAMIC MODEL

A. Dynamic model of vehicle

An automotive 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 is established based on model reference [5], as shown in Fig.1. Symbols on Fig.1 include: m_e, m_b, and m_{ij} - the masses of engine body, vehicle body, and vehicle axles; k_{tij} and c_{tij} - the stiffness and damping coefficients of tires; k_{ij} and c_{ij} - the stiffness and damping coefficients of vehicle suspension systems; k_{en} and c_{en} are the stiffness and damping coefficients of engine mounts; I_x, I_y and I_{ex}, I_{ey} are the mass moment of inertia of vehicle body and of IC engine; z_{ij}, z_{ij}, z và z_e are the vertical displacements of the axles, vehicle body and of IC engine, respectively; φ, θ and ; φ_e, θ_e are the pitch and roll angle displacements of the vehicle body and of IC

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engine, respectively; L, a, b, B_f, B_r are the distances (i=1÷2; j=left, right, n=1÷3), q_{ij} is road Surface excitations.

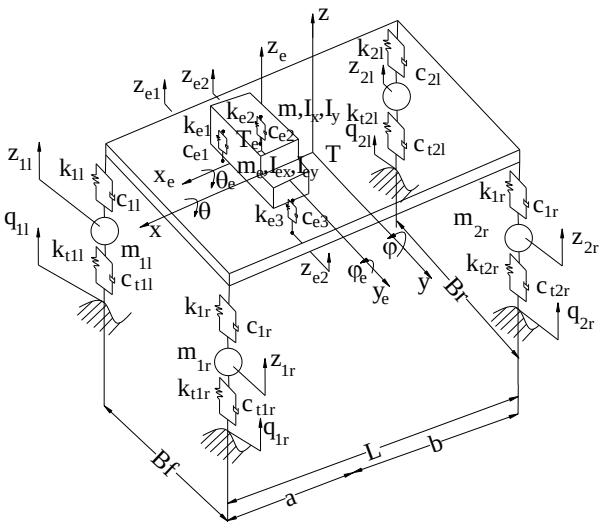


Figure1: Full-vehicle dynamic model with 10 DOFs:

B. *Equations of motion*

The equations of motion are equations that describe by Lagrange's equation given below.

The equations of motion for vehicle axles are written as follows:

$$m_{1l}\ddot{z}_{1l}=\left[k_{1l}\left(z_{1l0}-z_{1l}\right)+c_{1l}\left(\dot{z}_{1l0}-\dot{z}_{1l}\right)\right]-\left[k_{t1l}\left(q_{1l}-z_{1l}\right)+c_{t1l}\left(\dot{q}_{1l}-\dot{z}_{1l}\right)\right]. \quad (1)$$

$$m_{1r}\ddot{z}_{1r}=\left[k_{1r}\left(z_{1r0}-z_{1r}\right)+c_{1r}\left(\dot{z}_{1r0}-\dot{z}_{1r}\right)\right]-\left[k_{t1r}\left(q_{1r}-z_{1r}\right)+c_{t1r}\left(\dot{q}_{1r}-\dot{z}_{1r}\right)\right]. \quad (2)$$

$$m_{2l}\ddot{z}_{2l}=\left[k_{2l}\left(z_{2l0}-z_{2l}\right)+c_{2l}\left(\dot{z}_{2l0}-\dot{z}_{2l}\right)\right]-\left[k_{t2l}\left(q_{2l}-z_{2l}\right)+c_{t2l}\left(\dot{q}_{2l}-\dot{z}_{2l}\right)\right]. \quad (3)$$

$$m_{2r}\ddot{z}_{2r}=\left[k_{2r}\left(z_{2r0}-z_{2r}\right)+c_{2r}\left(\dot{z}_{2r0}-\dot{z}_{2r}\right)\right]-\left[k_{t2r}\left(q_{2r}-z_{2r}\right)+c_{t2r}\left(\dot{q}_{2r}-\dot{z}_{2r}\right)\right]. \quad (4)$$

where,

$$z_{1l0}=z-a\varphi+\frac{B_f}{2}\theta,\ z_{1r0}=z-a\varphi-\frac{B_f}{2}\theta.$$

$$z_{2l0}=z+b\varphi+\frac{B_s}{2}\theta,\ z_{2r0}=z-b\varphi+\frac{B_s}{2}\theta.$$

The equations of motion for vehicle body are written as follows:

$$m\ddot{z}=\sum_{n=1}^{n=3}\left[k_{en}\left(z_{en0}-z_{en}\right)+c_{en}\left(\dot{z}_{en0}-\dot{z}_{en}\right)\right]-\sum_{i=1,j=l}^{i=2,j=r}\left[k_{ij}\left(z_{ij0}-z_{ij}\right)+c_{ij}\left(\dot{z}_{ij0}-\dot{z}_{ij}\right)\right]. \quad (5)$$

$$I_y\ddot{\varphi}=\sum_{j=l}^{r}\left[k_{1j}\left(z_{1j0}-z_{1j}\right)+c_{1j}\left(\dot{z}_{1j0}-\dot{z}_{1j}\right)\right]a-\sum_{n=1}^{n=3}\left[k_{en}\left(z_{en0}-z_{en}\right)+c_{en}\left(\dot{z}_{en0}-\dot{z}_{en}\right)\right]x_n-\sum_{j=l}^{r}\left[k_{2j}\left(z_{2j0}-z_{2j}\right)+c_{2j}\left(\dot{z}_{2j0}-\dot{z}_{2j}\right)\right]b. \quad (6)$$

$$I_x\ddot{\theta}=\sum_{i=1}^{i=2}\left[k_{ei}\left(z_{ei0}-z_{ei}\right)+c_{ei}\left(\dot{z}_{ei0}-\dot{z}_{ei}\right)\right]y_i-\left[k_{e3}\left(z_{e30}-z_{e3}\right)+c_{e3}\left(\dot{z}_{e30}-\dot{z}_{e3}\right)\right]y_3+\left[k_{1r}\left(z_{1r0}-z_{1r}\right)+c_{1r}\left(\dot{z}_{1r0}-\dot{z}_{1r}\right)\right]\frac{B_f}{2}+\left[k_{2r}\left(z_{2r0}-z_{2r}\right)+c_{2r}\left(\dot{z}_{2r0}-\dot{z}_{2r}\right)\right]\frac{B_r}{2}-\left[k_{1l}\left(z_{1l0}-z_{1l}\right)+c_{1l}\left(\dot{z}_{1l0}-\dot{z}_{1l}\right)\right]\frac{B_f}{2}-\left[k_{2l}\left(z_{2l0}-z_{2l}\right)+c_{2l}\left(\dot{z}_{2l0}-\dot{z}_{2l}\right)\right]\frac{B_r}{2}. \quad (7)$$

where,

$z_{e10}=z_e-x_{e1}\varphi_e+y_{e1}\theta_e, z_{e20}=z_e+x_{e2}\varphi_e+y_{e2}\theta,$

$z_{e30}=z_e+x_{e2}\varphi_e-y_{e3}\theta_e.$

$z_{e1}=z-x_1\varphi+y_1\theta, z_{e2}=z-x_2\varphi+y_2\theta, z_{e3}=z-x_3\varphi-y_3\theta.$

The equations of motion for IC engine are written as follows:

$$m_e\ddot{z}_e=F_{ze}-\sum_{n=1}^{n=3}\left[k_{en}\left(z_{en0}-z_{en}\right)+c_{en}\left(\dot{z}_{en0}-\dot{z}_{en}\right)\right]. \quad (8)$$

$$I_{ey}\ddot{\varphi}_e=M_{ey}+\left[k_{e1}\left(z_{e10}-z_{e1}\right)+c_{e1}\left(\dot{z}_{e10}-\dot{z}_{e1}\right)\right]x_{1e}-\left[k_{e2}\left(z_{e20}-z_{e2}\right)+c_{e2}\left(\dot{z}_{e20}-\dot{z}_{e2}\right)\right]x_{2e}. \quad (9)$$

$$I_{ex}\ddot{\theta}_e=M_{ex}-\left[k_{e1}\left(z_{e10}-z_{e1}\right)+c_{e1}\left(\dot{z}_{e10}-\dot{z}_{e1}\right)\right]y_{1e}-\left[k_{e2}\left(z_{e20}-z_{e2}\right)+c_{e2}\left(\dot{z}_{e20}-\dot{z}_{e2}\right)\right]y_{2e}+\left[k_{e3}\left(z_{e30}-z_{e3}\right)+c_{e3}\left(\dot{z}_{e30}-\dot{z}_{e3}\right)\right]y_{3e}. \quad (10)$$

III. Analysis of Vibration Excitation Sources

A. *Road Surface excitation*

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[9]. 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]. The road surface roughness is generated as the sum of a series of harmonics:

$$q(t)=\sum_{k=1}^{N}\sqrt{2G_q\left(n_{mid-k}\right)\Delta n_k}\cos\left(2\pi n_{\text{mid}-k}t+\phi_k\right). \quad (11)$$

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 $G_q(n_{mid\text{-}k})$, where $n_{mid\text{-}k}$ (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.

B. *ICE excitations*

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, 8,11,12] defined as

$$F_{ez} = 4m_c r\lambda\omega_0^2 \cos(2\omega_0 t). \quad (12)$$

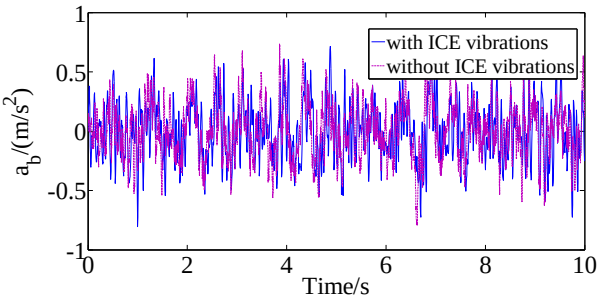
$$M_{ex} = M_e\left[1+1.3\sin(2\omega_0 t)\right]. \quad (13)$$

$$M_{ey} = 4m_c r\lambda\omega_0^2 \,l\cos(2\omega_0 t). \quad (14)$$

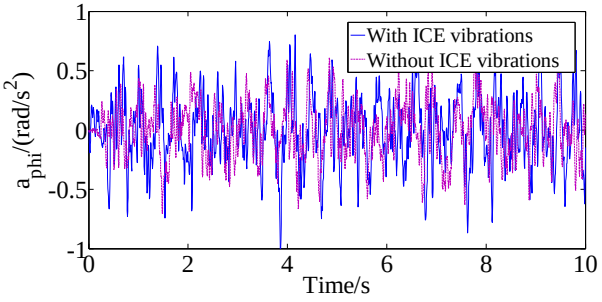
where, m_c is the reciprocating mass of a piston, r is the radius of a crank, λ is the ratio of r to the length of the shaft, ω_0 is the rotational frequency of the crank, l is the distance between the CG and the centre-line of the second and third cylinders, M_e is mean value of torque.

IV. RESULTS AND DISCUSSION

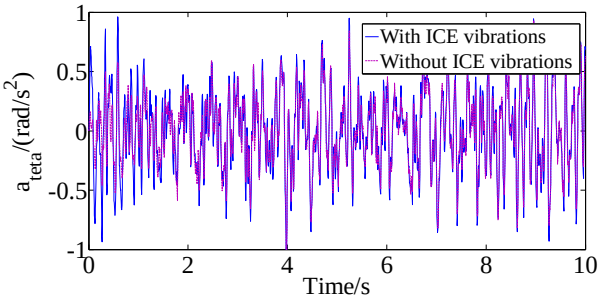
Differential equations of motion of vehicle in the above are simulated by Matlab/Simulink software under (with/without) ICE speed of 1200prm (vehicle speed of 14 m/s) in order to compare and evaluate the effect of ICE vibrations on vehicle ride comfort when vehicle moves on ISO class B road surface.The time domain acceleration responses of the vertical motion (a_b), pitch and roll angles (a_{phi} and a_{teta}) of vehicle body in comparison with ICE vibrations and without ICE vibration are shown Fig.2.



(a) Vehicle body vertical motion



(b) Vehicle body pitch angle



(c) Vehicle body roll angle

Figure 2: The time domain acceleration responses of vehicle body in comparison with ICE vibrations and without ICE vibration.

From the results of Figure 2, we can see that the peak values of the time domain acceleration responses of the vertical motion, pitch and roll angles of vehicle body with ICE vibrations respectively increase in comparison without ICE vibration.

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.(15) according to ISO 2631-1: Mechanical vibration and shock- Evaluation of human exposure to whole-body vibration to compare with ICE vibrations and without ICE vibrations. 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 is defined as

$$a_w = \left[\frac{1}{T}\int_0^T a^2(t)dt\right]^{1/2}. \quad (15)$$

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.

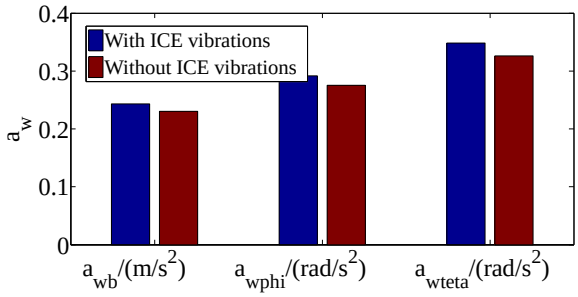


Figure 3: The a_{wb}, a_{wphi} and a_{wteta} values in comparison with ICE vibrations and without vibrations

By comparing the bar chart in Fig. 11, it can be concluded that the a_{wb} value with ICE vibrations raises by 5.8% in comparison without ICE vibrations, furthermore, the a_{wphi} and a_{wteta} also rises by 5.9% and 6.8% respectively in comparison without ICE vibrations. ICE vibrations has a significant effect on vehicle ride comfort. Therefore, we cannot underestimate the ICE vibrations in the problems of dynamic analysis of the whole vehicle.

V. CONCLUSION

The problem of the effect of the internal combustion engine vibrations is presented and investigated in the study. A full-vehicle dynamic model with 10 DOFs combining road surface roughness and ICE excitations was proposed to investigate the effect of ICE vibrations on vehicle ride comfort. The study results indicate that the values of the root mean square (r.m.s) acceleration responses of vehicle body with ICE vibrations rises by 5.8%, 5.9% and 6.8% respectively in comparison without ICE vibrations. In addition, the study results also provide a theoretical basis for designers and researchers in the field of dynamics and controlling of vehicle and ICE systems.

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