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ELASTIC, VISCOUS AND FRICTION PHENOMENA BASED COMPUTATIONAL MODEL OF ENGINE MOUNT DYNAMICS

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ABSTRACT

Vehicle engine mounting system is a vibration control system that can be passive, semi-active or active. It isolates the noise and vibration, and also the transmitted forces to the vehicle structure. At present conventional rubber-metal mounts are widely used for engine mounting systems. Usually in vehicle models a linear parallel spring and viscous damper represent an engine mount. This model does not take care of the nonlinear behavior of the mount dynamics. In the present study, a new computational model is developed suitable for prediction and analysis the nonlinear behavior of dynamic stiffness and damping for front and rear engine mounts of commercial vehicles under harmonic external excitations. To model the conventional mount, the functional components representing elastic, $F_e = k_e x$, viscous, F_v, and friction, F_f, forces are put together in parallel. For viscous part, two sets of series linear spring and viscous damper with the stiffness and damping coefficients k_1, c_1, k_2, and c_2, respectively, are used. Nonlinear friction of the mount is represented by a friction model with parameters k_f, k_u and β:

$$F_f(i) = F_1(i) + (F_f(i-1) - F_1(i))\, e^{\frac{-|x(i)-x(i-1)|}{\beta}}, \quad where \tag{1}$$
$$F_1(i) = k_f x(i) + k_u \quad if\ x(i) \geq x(i-1), \quad and \quad F_1(i) = k_f x(i) - k_u \quad if\ x(i) < x(i-1)$$

The plots of friction forces F_f versus displacement for different displacement amplitudes and two different sets of model parameters are depicted in Fig. 1.
The developed mathematical model of the mount has 8 parameters: k_e, k_1, c_1, k_2, c_2, k_f, k_u and β. Optimization with LMS algorithm is done to identify model parameters using measurements for real elastomeric engine mounts for commercial vehicles. With known model parameters nonlinear stiffness, S, and

non-dimensional damping, D , of the mount are calculated as functions of frequency and amplitude of excitation in Eq. 2. Here F_0 is the total steady state force amplitude, x_0 is the displacement amplitude and E is the total energy loss per cycle.

$$S = F_0 / x_0 \,, \quad D = E / (F_0 x_0) \qquad (2)$$

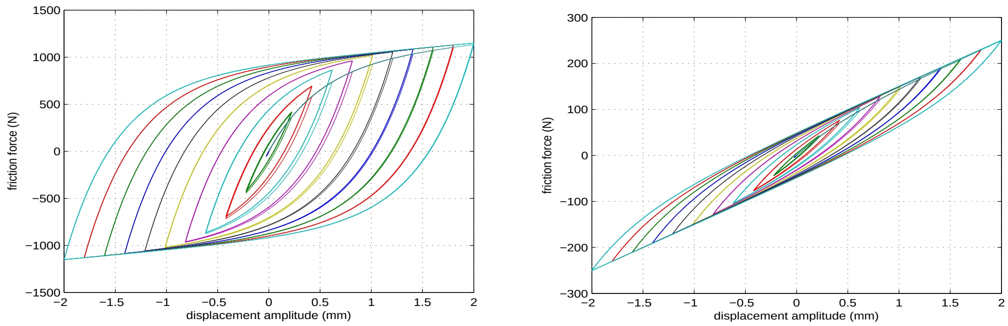

Fig. 1. Friction force as a function of displacement amplitude for two different sets of model parameters

Using the developed model the stiffness and damping of two different engine mounts have been estimated for the frequency range of 5-100 (Hz) and amplitude range of 0.025-2 (mm). It was found that the changes of stiffness of the mount are up to %32 with respect to amplitude and up to %29 with respect to frequency of excitation. Non-dimensional damping of the mount changes up to %88 with the change in amplitude and frequency of excitation. As an example, the plots of the stiffness and the damping of one engine mount in one direction are shown in Fig. 2.

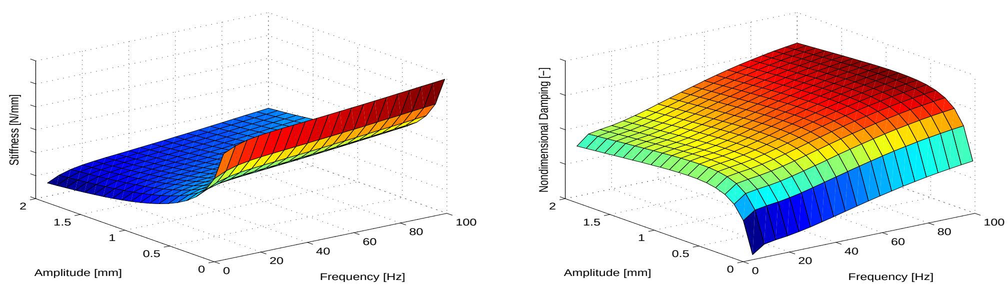

Fig. 2. Stiffness and damping of one engine mount in one direction

The computational model has been validated and verified against measurement data for harmonic excitations of conventional mounts. For different inputs of amplitude and frequency, the model shows admissible agreement with the measurement data. The tolerances of estimation of stiffness and damping regarding the measurement data are about %10. The developed computational mount model and obtained results can be used in complete vehicle dynamics analysis and in the design of semi-active and active engine mounts for commercial vehicles.

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