Ride comfort analysis and optimization of heavy vehicles based on entire dynamic model

Jizhi Zhu, Youjie Wang*, Liwei Yang and Yue Liu
Tianjin aerospace reliability technology Co., Ltd. Tianjin, China
*Corresponding author’s e-mail: wangyj@relialab.com

Abstract. Ride comfort improvement based on dynamic characteristic analysis is investigated. An entire dynamic model of a typical heavy truck is developed in order to obtain further investigation. The effects of sensitive mode shapes pitch around rear axle, bounce, pitch around front axle and cab resonance on vibration transmissibility from wheel to cab base are observed. The influence of leaf spring stiffness at front and rear axles on important mode shapes has been presented. Appropriate stiffness adjustment is suggested to improve ride comfort level, and the research flow can be also used for any other vehicle design.

1. Introduction
Ride comfort plays an important role in the design and reliability of various vehicles. The attenuation in vibration that transmitted from wheels to the driver is of considerable importance. Much attention has been paid to the control of vibration by seat design because it has a direct effect on vibration transmitted to the upper body [1]. According to these studies, improvement can be achieved by designing the natural frequency and damping of the seat. Moreover, a large number of model updating and optimization methods have been developed [2,3].

However, extremely low natural frequency may lead to large relative displacement and fatigue failure. Suspension displacement without an allowable range will result in larger stroke and a considerable deterioration of ride comfort [4]. Moreover, it is worth mentioning that the existing results ignore the fact that vibrations, imported into the seat suspension system, are mainly concentrated in low frequency band. According to the investigations in Ref. [5], humans are sensitive to random vibration at frequencies below 1Hz and least sensitive at frequencies above 10Hz, which can be explained by physiological factors. Thus, it may be difficult to obtain very high ride comfort level through mechanical design of seat.

When entire structure of vehicle is considered, cab is usually mounted on the front edge of the longitudinal beam. Obviously, vibration response on joint interface of the cab and the beam to road input works as the base excitation source for the cab. Suppose that dynamic response the joint interface is reduced to some extent, the vibration of cab will decrease directly. However, investigations are relatively limited in the literature, although this seems to be a meaningful work. In this study, some dynamic characteristics of vehicle under road excitation inputs are introduced. The ride comfort evaluation and cab isolation performance are further discussed. A whole vehicle model based on modal test and stiffness experiments is developed. The effects of possible change in stiffness on mode shape and ride comfort are explored.
2. Dynamic model analysis
The literature reviewed [6-8] has shown that most papers consider humans are more sensitive to random multi-directional vibration than to one-directional random vibration (in the fore-and-aft (x), lateral (y) and vertical (z)). The test vehicle used in this study is a heavy mining dump truck. The truck weighs 23,500 kg and the legal limit on the gross vehicle weight for this vehicle is 50,000 kg. Leaf spring suspension system is located in order to inhibit the vibration of vehicle body from the tires. Two rubber isolators are mounted at cab front, and rear suspension of the cab consists of spring shock absorber and rubber isolator. Fig.1 shows the entire model. In low frequency band, the dynamic effects of seat and cab suspension can only reach insignificant values [9]. Therefore, the cab (from cab base to seat) is assumed to be single degree of freedom system. The interaction between tires and road is complicated. Also, it is difficult to obtain an accurate nonlinear model for leaf spring, because of its tapered structure and interleaf friction. Here, two dimensional spring stiffness and damping coefficient values are used in order to exploit the simple theoretical formulation.

![Dynamic vehicle model](image)

The most important mode shapes are shown in Fig.2, explanation of the mode shapes pitch around rear axle, bounce, pitch around front axle and cab resonance can also be found. Usually, cab resonance frequency is higher than that of pitch and bounce frequencies for heavy vehicle. However, at resonance frequencies, vibration level on the joint interface of cab and beam will be very high, although the amplitude of the excitation is very low, especially for mode shapes of pitch around rear axle and bounce. Obviously, mode shape of pitch around front axle has a negligible effect on cab vibration. According to basic dynamic theory, mode shapes pitch and bounce are primarily determined by stiffness and damping of front and rear tires and leaf springs, it is necessary to utilize a suitable stiffness adjustment to improve ride comfort level [10].

![Mode shape](image)
3. Development of the dynamic model based on parameter identification

Eqs.(1) shows the dynamic motion equations of the whole vehicle system, where \( M \), \( C \) and \( K \) are mass, damping and stiffness matrixes, respectively. In order to obtain the system parameters, modal test and stiffness test of leaf springs have been carried out.

\[
M \{ \ddot{u} \} + C \{ \dot{u} \} + K \{ u \} = C_0 \{ \ddot{u}_0 \} + K_0 \{ u_0 \}
\]  

(1)

The single-degree-of-freedom system motion equations in modal coordinates can be seen in Eq.(2). Here, Laplace transformation is performed to yield equations of motion in the s-domain, and transfer function for each model is shown in Eq.(3). The transfer function from road excitation to the cab can be obtained as shown in Eq.(4).

\[
\ddot{\eta}_i + 2\xi_i\omega_i\dot{\eta}_i + \omega_i^2\eta_i = c_{0i}\ddot{u}_{0i} + k_{0i}u_{0i}
\]  

(2)

\[
\frac{\eta_i(s)}{u_{0i}(s)} = \frac{c_{0i}s + k_{0i}}{s^2 + 2\xi_is + \omega_i^2}
\]  

(3)

\[
H_i(s) = \frac{u_i(s)}{u_{0i}(s)} = \sum_{j=1}^{10} \phi(i,j) \frac{\eta_j(s)}{u_{0i}(s)}
\]  

(4)

Therefore, response PSD \( (G_{yy}(\omega)) \) can be achieved by PSD of road excitation \( (G_{xx}(\omega)) \) and FRF, as shown in Eq.(5).

\[
G_{yy}(\omega) = |H_i(\omega)|^2 G_{xx}(\omega)
\]  

(5)

4. Results

Upon substituting the actual parameters into Eq. (1), the matrix containing mode shapes and natural frequencies can be solved. The calculated natural frequencies are 3.29Hz, 5.27Hz and 6.42Hz, which are slightly lower than the measured value and the errors are 2.67%, 1.13% and 1.68%. This acceptable error may be attributed to assumptions and simplifications during mathematical modal analysis, which is related to the inaccuracy in computing, uncertainties in geometry and boundary conditions. Fig.3 shows the mode shapes based on mathematical model. Compared with experimental results, the same results could be achieved.
Following above validating procedures, an accurate updated model can be obtained. Consequently, if appropriate mechanical parameter is adjusted for optimization, output response PSD can be conveniently obtained. The weighted root mean square (RMS) acceleration in all directions and total RMS can be given by square root of the statistical power. Two groups of measured response accelerations at the cab under full-loaded and unloaded conditions are selected for the study, and PSD of each acceleration signal is computed by Welch method. Based on the FRFs, PSD of each base excitation \((x_{01}, x_{02}, z_{01} \text{ and } z_{02})\) input can be inverse generated. Such methodology to obtain the road excitation has significant advantages because it is practically difficult to mount accelerometers directly on the wheel to measure actual road excitation.

Total RMS with suitable changes in stiffness of leaf spring and tire at front axle under full-loaded and unloaded conditions is shown in Fig.4. Moreover, the effect of stiffness adjustments on mode shape (mode 1) is also presented. Obviously, these two curves show similar trends in the shape, which indicates that acceleration RMS decrease faster as the stiffness increase and, thereby, showing a significant improvement on ride comfort. For the change factor 2.5, acceleration RMS values under unloaded and full-loaded conditions are 30% and 35% less than RMS with no adjustment (change factor is 1). In high stiffness region (higher than 2.5), the effect on RMS is nearly insignificant, which is reasonable to expect an optimized stiffness for higher ride comfort level. Theoretical models reveal that the first mode shape pitch around rear axle will give significant contributions to vibration of the cab. Slope of the first mode shape decreases with the increase in vertical stiffness at front axle. Vibration in the vertical direction with pitch motion at the joint interface of the cab and beam will notable decrease, this may be the main reason for this phenomenon. The first three natural frequencies are 3.61Hz, 5.27Hz and 6.42Hz when the change factor is of 2.5, and the changes in natural frequencies are 9.75%, 0.01% and 0.01%. As expected, this indicates that the stiffness adjustment leads to a small change in dynamic characteristics of the entire system.
Fig.4. Total RMS with suitable changes in vertical stiffness of leaf spring and tire at front axle under full-loaded and unloaded conditions.

A set of change factors of appropriate stiffness are chosen, and the effects of stiffness adjustment at front axle \( (k_{1z}, k_{3z}, k_{1x}, \text{and} k_{3x}) \) on total weighted RMS are shown in Fig.5. Obviously, total RMS decreases as stiffness of front tire and leaf spring increases under both unloaded and full-loaded conditions. However, the results also reveal insignificant effects of stiffness in longitudinal direction on the ride comfort level. This may be attributed to FRF from vertical excitation on front tire to the cab. Moreover, the stiffness adjustment gives coupled effects on cab responses, which may be the main reason for the roughness showed on the described surface.

Fig.5. Effect of stiffness adjustment at front axle \( (k_{1z}, k_{3z}, k_{1x}, \text{and} k_{3x}) \) on total weighted RMS under unloaded and full-loaded conditions.

Fig.6 shows the effect of stiffness adjustment at rear axle \( (k_{2z}, k_{4z}, k_{2x}, \text{and} k_{4x}) \) on total weighted RMS under unloaded and full-loaded conditions. RMS curves show decreased trend with the increase of stiffness in longitudinal direction at rear axle. Moreover, it is indicated that the effects of vertical stiffness adjustment at rear axle on ride comfort are limited. Obvious roughness is seen, indicating that all vehicle parts are coupled together, and the stiffness gives a coupled effect on the dynamic responses in three directions.
Fig. 6. Effect of stiffness adjustment at rear axle ($k_{2z}$, $k_{4z}$, $k_{2x}$ and $k_{4x}$) on total weighted RMS under unloaded and full-loaded conditions.

From Fig. 5 and Fig. 6, it seemed reasonable to combine the effects of vertical stiffness at front axle and longitudinal stiffness at rear axle, and the results are shown in Fig. 7. Trends observed for the combined adjustment indicate that increasing the stiffness at designed position will result in improved ride comfort for this vehicle. Usually, larger tire pressure and contact area may provide higher tire stiffness, but the non-linear properties should be experimentally investigated. Recently, leaf spring design and performance analysis are focusing mainly on the static and dynamic characteristics in vertical direction (bending stiffness) [11, 12]. For this truck, leaf springs are simply hanged on the designed fixer, and longitudinal stiffness is mainly determined by the contact friction. The results presented in this paper also indicate that the increase in longitudinal stiffness of leaf spring will affect the model shape, and the ride comfort can be improved to some extent. Therefore, optimized stiffness of the connection of leaf spring and beam should be considered in future investigations.

Geometric and mechanical certain parameters of this vehicle give specific FRFs, different transfer characteristics and optimization may be obtained for other vehicles. However, the optimization flow from the dynamic model and validation to the ride comfort optimization seems practically feasible.

Fig. 7. Combined effect of stiffness adjustment.

5. Conclusions
An entire dynamic model of a typical heavy truck is developed, multi-directional vibration (vertical, longitudinal (fore-and-aft), and rotational) is considered. Based on preliminary analysis, mode shapes of pitch around rear axle and bounce give detrimental effect on vibration of the cab base. However, mode shape of pitch around front axle has a negligible effect.

The slope of first mode shape decreases with the increase in vertical stiffness at front axle, which leads to lower magnitude acceleration RMS. For this truck, improvement in ride comfort can be obtained by appropriate increase in vertical stiffness at front axle and longitudinal stiffness at rear axle.

Acknowledgements
The authors would like to thank Taian Spaceflight Special Vehicle Co. Ltd. for their contributions to the detailed discussions and test assistance of the vehicle (the grant number is DR707-23).
References
[1] Stein, G.J., Múcka, P. (2011) Study of simultaneous shock and vibration control by a fore-and-aft suspension system of a driver’s seat. Int. J. Ind. Ergonom., 41(5): 520-529.
[2] Maciejewski, I., Meyer, L., Krzyzynski, T. (2009) Modelling and multi-criteria optimisation of passive seat suspension vibro-isolating properties. J. Sound Vib., 324(3-5):520-538.
[3] Bouazara, M., Richard, M. J. (2001) An optimization method designed to improve 3-D vehicle comfort and road holding capability through the use of active and semi-active suspensions. Eur. J. Mech. A-Solids, 20(3): 509-520.
[4] Sun, W.C., Li, J. F., Zhao, Y., Gao, H. J. (2011) Vibration control for active seat suspension systems via dynamic output feedback with limited frequency characteristic. Mechatronics, 21:250-260.
[5] Demic, M., Lukic, J., Milic, Z. (2002) Some aspects of the investigation of random vibration influence on ride comfort. J. Sound Vib., 253(1): 109-129.
[6] Mansfield, N.J., Maeda, S. (2007) The apparent mass of the seated human exposed to single-axle and multi-axle whole-body vibration. J. Biomech., 40(11): 2543-2551.
[7] Patrik, H., Ronnie, L. (2001) Mechanical impedance of the sitting human body in single-axle compared to multi-axle whole-body vibration exposure, Clin. Biomech., 16(1):S101-S110.
[8] Mandapuram, S., Rakheja, S., Boileau, P-É., Maeda, S. (2012) Apparent mass and head vibration transmission responses of seated body to three translational axle vibration. Int. J. Ind. Ergonom., 42:268-277.
[9] Anthonis, J., Kennes, P., Ramon, H. (2000) Design and evaluation of a low-power mobile shaker for vibration tests on heavy wheeled vehicles. J. Terramechanics, 37: 191-205.
[10] Harris, C.M., Piersol, A.G. (2002) Shock and Vibration Handbook. 5th ed. New York: McGraw Hill.
[11] Osipenko, M.A., Nyashin, Y. I., Rudakov, R. N. (2003) A contact Problem in the theory of leaf spring bending. Int. J. Solids Struct., 40(12): 3129-3136.
[12] Sugiyama, H., Shabana, A. A., Omar, M. A., Loh, W. Y. (2006) Development of nonlinear elastic leaf spring model for multibody vehicle systems, Comput. Math. Appl. Mech. Eng. 195(50-51): 6925-6941.