Investigation of dynamic characteristics for road vehicle based on multiple-input multiple-output vibration test

Liwei Yang *, Youjie Wang, Dejun Liu and Zhongping Liu

Tianjin aerospace reliability technology Co., Ltd. Tianjin, China

*Corresponding author’s e-mail: yanglw@relialab.com

Abstract. This study is intended to investigate dynamic characteristics of a heavy truck. An experimental setup is designed for full vehicle vibration test consisting of four electro-mechanical shakers with high force rating (35000kgf). Transfer matrix is taken into account for multiple-input multiple-output (MIMO) vibration control. Vibrat ion inputs under the tires can simulate both vertical and rotational excitations of the road. Furthermore, transfer function estimate (TFE) can be obtained based on the known road inputs and measured dynamic responses. Operational modal analysis has been carried out to identify the rigid mode shapes pitch, bounce, roll in low frequency range (1-10Hz). Based on experimental results, vibration response is higher on seat, which results in an uncomfortable level.

1. Introduction

Vibration test in laboratory is of considerable interest to engineers because expected signals can be generated on shaking table and any dynamic behaviour of vibration transmission component can be clearly given [1]. It is usually conducted as an effective means to identify faults in machineries and in prevention of failures leading to the mechanism of fault diagnostics[2-4]. Usually, vibration shakers are used to excite the selected input points on the workpiece. Simultaneously, many sensors are fixed on the shaker table to control the vibrations. Unfortunately, due to large-scale and heavy mass, vibration tests in laboratory always suffer from many difficulties, such as creation of a suitable excitation force and physically large mechanisms.

This study attempts to conduct a multi-dimension vibration test of a heavy truck in order to investigate dynamic characteristics. Four electrodynamic shakers (35000kgf) have been used at frequencies between 1 and 32Hz. Operational modal analysis has been carried out using vibration test data under base excitation.

2. Test procedure

Please Vibration test is performed in the lab using four electro-mechanical shakers, as shown in Fig1 and Fig.2. The system force rating is 35000 kgf for both sine and random in frequency range from 2 Hz to 2000 Hz, capable of acceleration up to 100 g. The maximum velocity and displacement are respectively 2 m/s and 76 mm. Hydrostatic supports are employed under fixture table in order to constraint the movement to axial direction only. Air springs are used to support the weight of external load. Multiple-input multiple-output (MIMO) random vibration test control technology is used to implement the multiple shaker vibration tests [5]. Reference power spectrum density (PSD) is obtained based on real road test.

As mentioned in many works, human is more sensitive to vibration in low frequency band [6-7]. Therefore, frequency range of 1-32Hz has been used during the test. Random vibration tests have been
carried out under unloaded and full-loaded conditions. More than 20 three-axle piezoelectric accelerometers are attached to the seat, cab suspension, beam, compartment, tire, and leaf springs. Laser displacement sensors (OPTO NCDT 1401; Germany) are used to detect deformation of leaf springs. Obviously, three-input random vibration (z, rotational (x-axis), rotational (y-axis)) can be provided by the four shakers. The input transformation to define the control degrees of freedom in terms of measured accelerations on the four vibration tables is given by Eq. (1).
Where $z$ is vibration in vertical direction, $R_x$ and $R_y$ are rotational vibration around x-axis and y-axis, $Y_1$, $Y_2$, $Y_3$ and $Y_4$ are vibration on shaker tables, $L_x$ is the distance from front axle to rear axle, $L_y$ is distance between right and left tires.

\[
\begin{bmatrix}
  z \\
  R_x \\
  R_y
\end{bmatrix} = \begin{bmatrix}
  1 & 1 & 1 & 1 \\
  4 & 4 & 4 & 4 \\
  2L_y & 2L_y & 2L_y & 2L_y \\
  -1 & -1 & -1 & -1 \\
  2L_y & 2L_y & 2L_y & 2L_y \\
  1 & 1 & 1 & 1 \\
  2L_y & 2L_y & 2L_y & 2L_y
\end{bmatrix} \begin{bmatrix}
  Y_1 \\
  Y_2 \\
  Y_3 \\
  Y_4
\end{bmatrix}
\] (1)

Based on MIMO test control technology, the transformation for the computed drives to the four shakers is given by Eq. (2)[8]. Before starting the test, frequency response function matrix $H$ can be obtained by outputting low level random excitation signal to the shaker. During the experiment, updated transfer function equalization is used to generate the control signal $X$ so as to compensate the estimated process dynamics.

3. Results
Reference spectrum of auto-power spectrum and control results on shakers are shown in Fig.3. The present research work is focused on dynamic analysis of full vehicle, thus -6dB of the measured spectrum is used for ensuring operational safety. Coherence between right and left control points is 0.7 and coherence between rear and front axle is set to zero. From Fig.3, the spectrum lines keep well within ±3 dB alarm boundary of reference spectrum.
Quantitatively, natural characteristics of the vehicle have been obtained based on operational modal analysis [9-10]. Resonant frequencies and mode shapes of the first four mode shapes, pitch around rear axle, bounce, roll and pitch around front axle are shown in Fig.4. It can be seen that vehicle components such as compartment, beam and cabin are modeled as rigid bodies. Natural frequencies are 2.88Hz, 5.03Hz, 5.71Hz and 6.23Hz under unloaded condition, and 1.21 Hz, 3.83 Hz, 4.34 Hz, 5.63 Hz under full-loaded condition.

When entire vehicle structure is considered, cab is usually mounted on front edge of the longitudinal beam. Vibration response on joint section of the cab and the beam to road input works as base excitation source for the cab. Usually, cab resonance frequency is higher than that of rigid mode, the effects on these vibrations isolation in lower frequency band are nearly insignificant [11]. Consequently, vibration from the road will be amplified in low frequency range especially at resonant frequencies of mode shape pitch around rear axle and bounce.
Because of the roll and pitch motion, in addition to vertical response to vertical excitation, there is also response in x and y directions [12]. Vibration on the seat is measured in order to evaluate the ride comfort level. Acceleration PSD values are obtained using frequency resolution 0.3125 Hz, Hanning window with 50% overlap and averaging 96 blocks. Fig.5, as an example, illustrates power spectral density (PSD) on seat under unloaded and full-loaded conditions. Clearly, vibration energy is transmitted mainly in low frequency range (below 10Hz). The acceleration PSD values are distinctly higher than that of the high frequency range (above 10Hz). This might be attributed to the unsatisfied vibration isolation performance of cab and seat suspension in low frequency range. Moreover, curves in Fig.5 (b) show an evident peak around 3.44Hz.

Acceleration root mean square (RMS) is shown in Table 1. Calculated results show that the total acceleration RMSs on the seat (0dB) are 1.54m/s$^2$ and 1.32m/s$^2$ under unloaded condition and full-loaded condition, respectively. This indicates “uncomfortable” according to the standard ISO2631. Therefore, for this vehicle, there is a need for further research on ride comfort improvement. Furthermore, x-direction and y-direction vibrations are of the same magnitude as in the vertical direction, thus much attention should be paid to control of vibration in all directions [13-14].

![Fig.5 PSD on seat under unloaded and full-loaded conditions](image)

Fig.5 PSD on seat under unloaded and full-loaded conditions (a) unloaded condition (b) full-loaded
### Table 1 condition.

|                | X direction | Y direction | Z direction | Total RMS | 0dB | Subjective comment |
|----------------|-------------|-------------|-------------|-----------|-----|-------------------|
| Unloaded       | 0.1554      | 0.2204      | 0.2777      | 0.3871    | 1.5411 | Uncomfortable     |
| Full-loaded    | 0.1395      | 0.1417      | 0.2654      | 0.3317    | 1.3203 | Uncomfortable     |

4. Conclusions

In this study, MIMO vibration test of a heavy truck is conducted in order to investigate the dynamic characteristics. The following conclusions can be made:

From power spectrums, the test can be control by MIMO method. Based on operational modal analysis, four rigid modes below 10 Hz are identified. Natural frequencies are 2.88 Hz, 5.03 Hz, 5.71 Hz and 6.23 Hz under unloaded condition, and 1.21 Hz, 3.83 Hz, 4.34 Hz, 5.63 Hz under full-loaded condition. Mode shapes, pitch around rear axle, bounce, roll and pitch around front axle are shown. Under recommended road excitations, acceleration RMS on the seat for full-loaded condition is lower than that of unloaded condition. However, total acceleration RMS under all conditions are higher and "uncomfortable" results are evaluated according to the standard.

Acknowledgments

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] P.A. Donati, Procedure for developing a vibration test method for specific categories of industrial trucks, Journal of Sound and Vibration 215 (1998) 947-957.

[2] F.X. Che, John H.L. Pang, Vibration reliability test and finite element analysis for flip chip solder joints. Microelectronics Reliability 49 (2009) 754–760.

[3] M. Pearce, J. Lund, M. Lundin, J. Lundquist, Random vibration tests of the anticoincidence system of the PAMELA satellite experiment, Nuclear Instruments and Methods in Physics Research A 488 (2002) 536–542.

[4] Z.H. Nie, H. Hao, H.W. Ma, Structural damage detection based on the reconstructed phase space for reinforced concrete slab: Experimental study, Journal of Sound and Vibration 332 (2013) 1061-1078.

[5] A. Wildschek, Z. Bartosiewicz, D. Mozyrska, A multi-input multi-output adaptive feed-forward controller for vibration alleviation on a large blended wing body airliner, Journal of Sound and Vibration 333 (2014) 3859-3880.

[6] O. Thuong, M.J. Griffin, The vibration discomfort of standing persons: 0.5–16-Hz fore-and-aft, lateral, and vertical vibration, Journal of Sound and Vibration 330 (2011) 816-826.

[7] M. Demic, J. Lukic, Z. Milic, Some aspects of the investigation of random vibration influence on ride comfort, Journal of Sound and Vibration 253 (2002) 109-129.

[8] E.J. Ruggiero, G. Park, D.J. Inman, Multi-input multi-output vibration testing of an inflatable torus, Mechanical Systems and Signal Processing 18 (2004) 1187–1201.

[9] P. Mohanty, D.J. Rixen, Operational modal analysis in the presence of harmonic excitation, Journal of Sound and Vibration 270 (2004) 93-109.

[10] P. Kindt, P. Sas, W. Desmet, Measurement and analysis of rolling tire vibrations, Optics and Lasers in Engineering 47 (2009) 443-453.

[11] J. Anthonis, P. Kennes, H. Ramon, Design and evaluation of a low-power mobile shaker for vibration tests on heavy wheeled vehicles, Journal of Terramechanics 37 (2000) 191-205.

[12] N. Nawayehe, M.J. Griffin, Tri-axial forces at the seat and backrest during whole-body vertical vibration, Journal of Sound and Vibration 277 (2004) 309–326.
[13] S. Rakheja, R.G. Dong, S. Patra, P.-É. Boileau, P. Marcotte, C. Warren, Biodynamics of the human body under whole-body vibration: synthesis of the reported data, International Journal of Industrial Ergonomics 40 (2010) 710-732.

[14] G.J. Stein, P. Múcka, Study of simultaneous shock and vibration control by a fore-and-aft suspension system of a driver’s seat, International Journal of Industrial Ergonomics 41 (2011) 520-529.