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Comparative Research on Optimal Damping Matching of Seat System for an off-Highway Dump Truck

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ABSTRACT

To protect the driver of off-highway dump trucks from the harmful vibration, this paper presents the comparison results to determine the optimal damping of the seat system by different optimization design plans. Three optimization schemes are considered including individually optimizing the damping of the cushion, individually optimizing the damping of the seat suspension, and integrately optimizing both of them. To compare the three optimization schemes, the seat system for an off-highway dump truck is taken as the baseline model. Initially, the parameters of the cushion, the seat suspension, and the air spring for the seat system were determined by corresponding test and the nonlinear dynaimic model of the seat system was created. Then, the model was validated by the test data from the field measurement. Subsequently, on the basis of the measured seat base acceleration, the corresponding damping coefficients under the three schemes were optimized. Finally, the road tests were conducted to verify and compare the degree of the comfort improvement. The results show that there is a relative smaller room for the comfort improvement by individually optimizing the cushion or by individually optimizing the seat suspension. The integrated optimization is the best to improve the comfort.

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NOMENCLATURE

F_s	Damping force of the shock absorber (N)	a_d	Driver vertical vibration acceleration (m/s^2)
C_1	Rebound damping coefficient (Ns/m)	m_d	Driver mass (kg)
C_2	Compression damping coefficient (Ns/m)	a_s	Seat frame vertical vibration acceleration (m/s^2)
V	Relative motion velocity of the shock absorber (m/s)	m_s	Seat frame mass (kg)
k_1	Linear stiffness coefficient (kN/m)	σ_{a_d}	Driver vertical frequency weighted RMS acceleration (m/s^2)
k_3	Nonlinear stiffness coefficient (kN/m^3)	f	Vibration frequency (Hz)
Δz	Vertical deformation of the air spring (m)	$G_a(f)$	Power spectral density of the driver vertical acceleration (m^2/s^3)
C_0	Damping coefficient of the cushion (Ns/m)	$w(f)$	Frequency weighted coefficient
F_c	Force generated by the cushion (N)	f_d	Seat suspension dynamic travel (m)
k_c	Stiffness of the cushion (kN/m)	$[f_d]$	Limited travel of f_d (m)
x	Vertical deformation of the cushion (m)	RMS	Root mean square

1. INTRODUCTION

Off highway dump truck is a type of off-road vehicle. The truck is often used for transportation tasks of the rock and ore in the open air [1]. All the year round, it runs on poor road conditions such as gravel roads which often causes the driver suffering the low frequency vibration from the seat system. The vibration is easy to cause fatigue for the driver, reduce the work efficiency and affect the work quality. If the driver is always exposed to such low frequency and high intensity vibrations, this will lead to a number of diseases and endanger the cardiovascular, nervous system, and musculoskeletal systems of the driver [2, 3]. It is an important and practical problem to take feasible

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technical measures to isolate the harmful vibration for the improvement of the ride comfort.

The seat system is an important part of off highway dump trucks for isolating vehicle vibration to protect the driver [4]. The system attenuates the vibration from the cab floor mainly through the seat suspension and the cushion. Due to the high cost, the application of semi-active and active suspension is restricted [5, 6]. At present, the feasible technical measures are to optimize the passive suspension of the seat system or the cushion in order to reduce the vibration [7]. The damping coefficients of the seat suspension and the cushion have important influence on the vibration isolation performance of the seat system. In recent years, many research efforts on the modeling and design of the seat and/or cushion have been made. For example, Raiendiran, et al. investigated the PID and fuzzy controller for seat suspension [8]. Ji, et al. [9] analyzed the vibration attenuation characteristics of an air-inflated cushion. However, there are not many researches on the integrated damping matching of the seat suspension and the cushion for off highway dump trucks.

This paper presents the comparison results to determine the optimal damping of the seat system by three optimization schemes, namely individually optimizing the damping of the cushion, individually optimizing the damping of the seat suspension, and integrately optimizing both of them. The seat for a off-highway dump trucks was taken as the baseline model. The seat system model was created and validated by the test data. Base on the measured seat base acceleration and the seat system model, the corresponding damping coefficients under the three schemes were optimized. Finally, the road tests were conducted to verify and to compare the effectiveness of the comfort improvement.

2. SEAT SYSTEM MODEL

The analyzed off-highway dump truck is used for transportation tasks of the ore, whose drive type is 8×4. Its gross vehicle weight and load capacity are 43.0 and 65.0 ton, respectively. The seat system for the off-highway dump truck was taken as the baseline model, as shown in Figure 1. The seat employs a typical dual shock pendulum scissor mechanism. It includes a double-tube hydraulic shock absorber and an air spring.

2. 1. Modelling of the Shock Absorber The double-tube hydraulic shock absorber is a complex hydraulic component. It generates the damping force, when the oil flows through the throttle orifice or the aperture of the throttle slice.

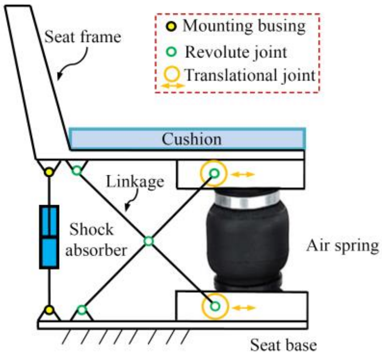

Figure 1. The seat system for the off-highway dump truck

At present, for the double-tube hydraulic shock absorber, many fine models including very complex internal structure parameters exist [10]. However, this study mainly investigates the characteristics of the seat system to improve the ride comfort. On the basis of the phenomenological modeling method [11], according to the relationship between the measured relative velocity and the damping force of the shock absorber, the mathematical model of the shock absorber can be built. Therefore, we can only focus on the external characteristic curve of the shock absorber. According to the bench test method of automobile shock absorber [12], the damping characteristics test of the shock absorber was conducted by the multi-function hydraulic vibrating test bench, as shown in Figure 2. The velocity-force curve clearly depicts the asymmetric damping characteristics of the shock absorber, as shown in Figure 3. Based on the curve, the damping force F_s between the seat frame and the seat base can be expressed as:

$$F_s = \begin{cases} C_1 V & V \geq 0 \\ C_2 V & V < 0 \end{cases} \tag{1}$$

where, the rebound damping coefficient C_1 equals 1124.6 Ns/m, the compression damping coefficient C_2 equals 410.3 Ns/m, V is the relative motion velocity of the piston rod and the cylinder of the shock absorber.

Figure 2. The damping characteristics test of the shock absorber

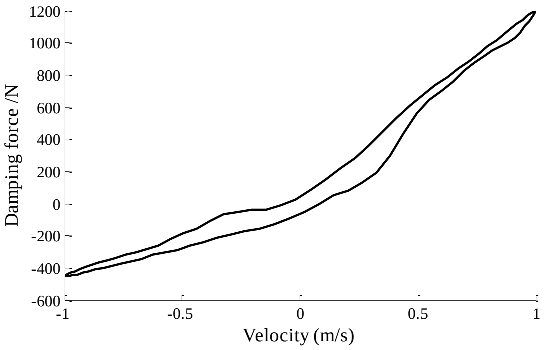

Figure 3. The velocity-force curve of the shock absorber

2. 2. Modelling of the Air Spring In order to determine the relationship between the displacement and the elastic force of the air spring in the working status, the stiffness characteristics testing of the air spring was conducted, as shown in Figure 4. According to the data sheets from the seat manufacturer, the inflation pressure of the air spring under the design load of 65 kg is 0.5 MPa. Thus, before the test, the inflation pressure of the air spring was set at 0.5 MPa, and the test rig applied a preload force 650 N. Then, the test rig exerted a displacement of ±20.0 mm on the spring at a constant speed of 5.0 mm/min. The elastic force and displacement data were collected using the force displacement sensors, respectively. According to the these data, the elastic force F_k between the seat frame and the seat base can be expressed as:

$$F_k = k_1 \Delta z + k_3 \Delta z^3 \quad (2)$$

where, the linear stiffness coefficient k_1 equals 16.9 Ns/m, the nonlinear stiffness coefficient k_3 equals 9746.3 kN/m^3, and Δz is the vertical deformation of the air spring.

2. 3. Modelling of the Cushion For the comfort analysis of the seat under stationary random excitations, it is acceptable to regard the cushion as a linear damper and a linear stiffness spring in parallel corresponding to a range of known driver weights [13].

Figure 4. The stiffness characteristics testing of the air spring

On the basis of the measurement method for the single DOF cushion model [14], the cushion parameters was measured by the School's seat indenter test rig at School of Transportation and Vehicle Engineering, Shandong University of Technology. The cushion was placed under the indenter plate and driven from below by a hydraulic vibrator. The indenter was used to exert a preload force of 650 N on the cushion. The acceleration on the vibrator platform, the axial force at the indenter plate and the displacement of the indenter plate were measured. Based on the measured data, the force F_c generated by the cushion can be expressed as

$$F_c = C_0 \dot{x} + k_c x \quad (3)$$

where, the equivalent damping coefficient of the cushion C_0 equals 217.8 Ns/m, the equivalent stiffness of the cushion k_c equals 58.4 kN/m, and x is the vertical deformation of the cushion.

2. 4. Modelling of the Seat Including the Driver To analyze the ride comfort, the seat system should include the driver model. The driver is not a simple mass. The driver body has a certain quality, damping, and stiffness. Many models of the human body were presented for investigating complex human biomechanics [15, 16]. The appropriate choice of the human body model was analyzed in detail by Stein et al. [17] and their investigation shows that the human body can be regarded as a single mass in simulations with sufficient accuracy for the comfort analysis and more refined models are not needed. Thus, in this study, a mass m_d located on the cushion was used. According to Newton's second law, the vibration equation of the driver body from the static equilibrium position can be expressed as:

$$m_d a_d = -F_c \quad (4)$$

where, a_d is the driver vertical vibration acceleration.

The vibration equation of the seat frame from the static equilibrium position can be expressed as:

$$m_s a_s = -F_k - F_s + F_c \quad (5)$$

where, a_s is the seat frame vertical vibration acceleration and the weighted seat frame mass m_s equals 14.9 kg.

3. MODEL VERIFICATION

3. 1. Field Measurements The field measurements were conducted using the off highway dump truck, on the gravel road. For the measurements, an Entran EGCSY-240D-10 accelerometer (range ±10 g) was installed on the floor at the installation position of the driver seat. A "Sit-pad" conforming to ISO 10326-1 with built-in accelerometer (range ±10 g) was installed on the cushion. The driver mass m_d is 65 kg. The driver loosely restrained with a seat-belt and his back in touch with the seat back in upright position. The driver's hands loosely placed on the steering wheel. Using the data acquisition and analysis system "M+P Smart Office", with the off highway dump truck at 40, 50, 60 km/h, under the fully laden condition, the acceleration signals were collected for 112 s. The sampling frequency and the frequency resolution are set as 500 Hz and 0.1 Hz, respectively. The seat base vertical acceleration at 40 km/h is shown in Figure 5. The measured driver vertical acceleration a_d is shown in Figure 6.

3. 2. Verification of the Seat System Model Using the measured seat base vertical acceleration as the seat system model input, the driver vertical acceleration a_d was simulated. A comparsion between the simulated (a_d) and the measured driver vertical accelerations (a_d) are depicted in Figures 7 and 8, respectively.

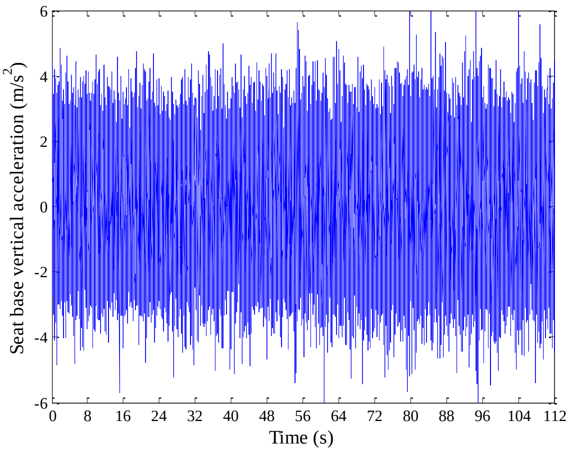

Figure 5. The measured seat base vertical vacceleration

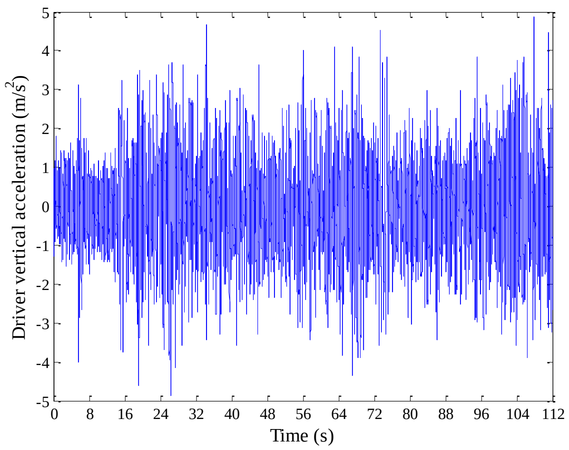

Figure 6. The measured driver vertical acceleration a_d

In order to more clearly show the identical points and the differences between the measured acceleration a_d and the simulated acceleration a_d, the time ranges were set as 35~40 s and 102~107 s, respectively.

To give insight into the accuracy of the seat system model and to quantify the quality of the whole time course of the simulation, the relative error δ_{ti} for each time point between the simulated acceleration and the measured acceleration can be used, which can be expressed as

$$\delta_{ti} = \left| \frac{a_{\mathrm{d\_sim}}(i) - a_{\mathrm{d\_mea}}(i)}{a_{\mathrm{d\_mea}}(i)} \right| \times 100\% \tag{6}$$

According to the simulated driver vertical acceleration a_d and the measured driver vertical acceleration a_d, the the relative error δ_{ti} for each time point was calculted in the whole time range 0~130 s. The maximum value of the relative error δ_{ti} equals 7.13% for t =105.8 s. In addition, the relative error between the RMS (root mean square) of the simulated acceleration a_d and the RMS of the measured acceleration a_d is 1.63%.

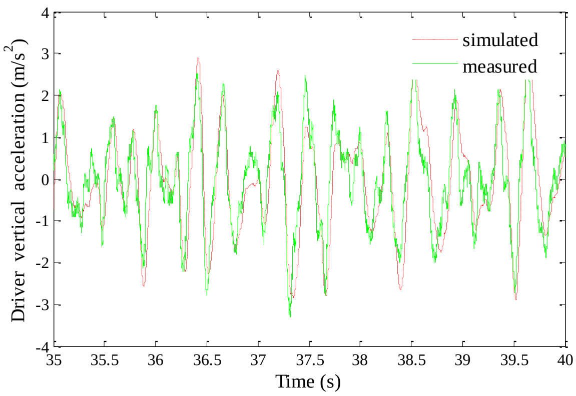

Figure 7. Comparsion of the driver vertical accelerations for 35~40 s

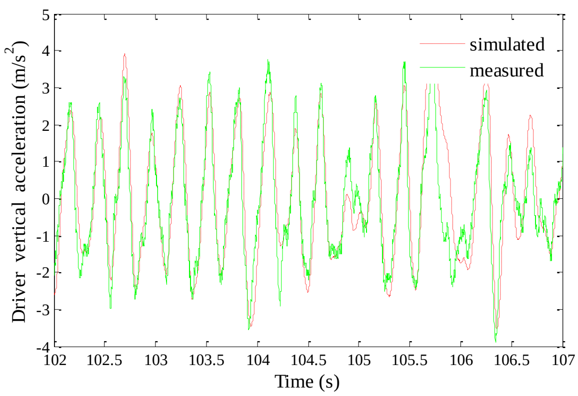

Figure 8. Comparsion of the driver vertical accelerations for 102~107 s

From Figures 7 and 8, it can be seen that the simulated driver vertical acceleration a_d is very close to the measured driver vertical acceleration a_d. The seat dynamic model can well reproduce the dynamic response of the real seat system, although some small differences exist. The small differences may be related to the friction forces between the kinematic pairs. The results show that the seat system model has sufficient accuracy for the comfort analysis of the off highway dump truck.

4. OPTIMAL DAMPING MATCHING

4. 1. Damping Optimization The seat system is critical for ride comfort. For the comfort analysis, the commonly used index is the weighted RMS acceleration recommended by ISO2631-1:1997 [18]. Because the ride comfort of the seat system is strongly correlated with the vertical vibration, the driver vertical frequency weighted RMS acceleration σ_{a_d} is used to estimate the ride comfort in this study, which can be calculated as:

$$\sigma_{a_d} = [\int_{0.5}^{80} w^2(f) G_a(f) \mathrm{d}f]^{1/2} \tag{7}$$

Where, $G_a(f)$ is the power spectral density of the driver vertical acceleration.

The frequency weighted coefficient $w(f)$ is expressed as follows:

$$w(f) = \begin{cases} 0.5 & (0.5 \le f < 2.0) \\ f/4.0 & (2.0 \le f < 4.0) \\ 1.0 & (4.0 \le f < 12.5) \\ 12.5/f & (12.5 \le f < 80.0) \end{cases} \tag{8}$$

From previous literature, it can be seen that the damping coefficients of the seat suspension and the cushion have important effects on the ride comfort. The damping coefficients of the seat suspension and the cushion can be individually optimized. They can also be integrately optimized. In order to compare the comfort improvements of the seat with the corresponding optimal damping under the three schemes, three objective functions can be created, respectively. In this study, the three objective functions in a unified manner can be expressed:

$$\min J_i(X_i) = \min\{\sigma_{a_d}\} \tag{9}$$

where, for i=1, X_1=C_0 ; i=2, X_2=[C_1, C_2] ; i=3, X_3=[C_0, C_1, C_2].

To ensure that the collision probability between the bump stop and the seat frame is within 0.135% [19], the constraint can expressed as:

$$\frac{\sigma(f_d)}{[f_d]} \le \frac{1}{3} \tag{10}$$

where, f_d is the the seat suspension dynamic travel, [f_d] the limited travel of f_d, and $\sigma(f_d)$ the standard deviation of f_d. The desired value of [f_d] equals 0.056 m.

The mathematical formulation of the single-objective optimization problem can be expressed as:

$$\begin{cases} \text{Minimize} & J_i(X_i) \\ \text{Subject to} & \dfrac{\sigma(f_d)}{[f_d]} - \dfrac{1}{3} \le 0 \end{cases} \tag{11}$$

This study adopts Multi-Island Genetic Algorithm (MIGA) to solve the optimization problem. The MIGA algorithm is on the basis of the traditional genetic algorithm (TGA), which can simulate the biological evolutionary processes in the natural environment. This algorithm has some advantages. It is an adaptive global optimization method and can solve some large scale and multi-variable nonlinear problems. The flow chart of the optimization process is shown in Figure 9. In the MIGA, like other genetic algorithms, each design point is perceived as an individual with a certain value of fitness based on the value of the objective function and constraint penalty. An individual with a better value of the objective function and penalty has a higher fitness value. Each individual is represented by a chromosome in which the values of design variables are converted into a binary string of 0 and 1 characters [20].

On the basis of the measured seat base acceleration and the seat system model, using the optimization program, the corresponding damping coefficients under the three schemes were optimized. The optimization results were concluded in Table 1.

4. 2. Test Verification and Discussion To verify the effectiveness of the optimized damping parameters and to compare the three optimization schemes, the road tests were conducted on the previous gravel road.

The shock absorber was redesigned by its manufacturer and the cushion was replaced based on the optimal damping parameters. The tests were conducted for the seat system with the optimal damping parameters obtained by the three optimization schemes.

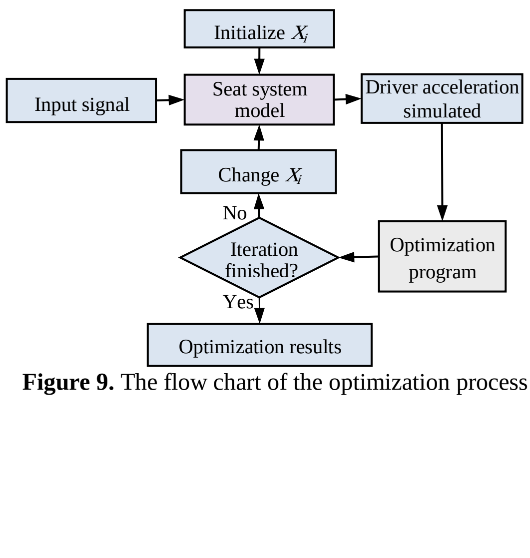

Figure 9. The flow chart of the optimization process

TABLE 1. The optimization results for three optimization schemes

Optimization scheme	Optimization variable X_i	Optimization result (Ns/m)	Convergence values of J_i
Individually optimize cushion	C_0	740.5	0.000012
Individually optimize suspension	C_1, C_2	900.6, 859.7	0.000008
Integrated optimization	C_0, C_1, C_2	676.7, 890.2, 886.3	0.000014

For each road test, the speed and the load of the truck were the same as the previous field measurements and the accelerometers were installed at the same locations as before. The driver was also the same one. For each test condition, the driver vertical acceleration signal was collected for 112 s.

The driver vertical frequency weighted RMS acceleration was calculated and the maximum value of the driver vertical acceleration was extracted for each condtion. The calculated results and comparisons of the optimization schemes were concluded in Table 2. In order to clearly illustrate the degree of the comfort improvement for each optimization scheme, the relative deviations of the driver vertical frequency weighted RMS acceleration were calculated between the original seat system and the optimized seat system using the three optimization schemes, as shown in Figure 10. In order to more comprehensively evaluate the ride improvement, the relative deviations of the maximum acceleration were also calculated, as shown in Figure 11.

TABLE 2. The results and comparisons of the optimization schemes

Optimization scheme	Speed (km/h)	Weighted RMS acceleration (m/s^2)	Maximum acceleration (m/s^2)
	40	1.22	4.99
Original seat system	50	1.36	5.43
	60	1.45	5.98
	40	1.13	4.58
Individually optimize cushion	50	1.26	4.96
	60	1.34	5.48
	40	1.09	4.44
Individually optimize suspension	50	1.22	4.82
	60	1.30	5.32
	40	1.03	4.25
Integrated optimization	50	1.14	4.65
	60	1.22	5.11

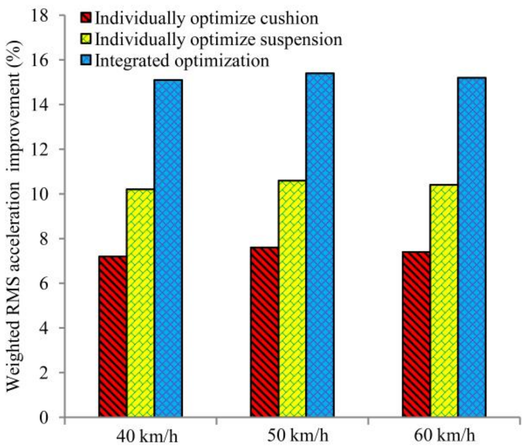

Figure 10. Comparsion of the Weighted RMS acceleration improvements.

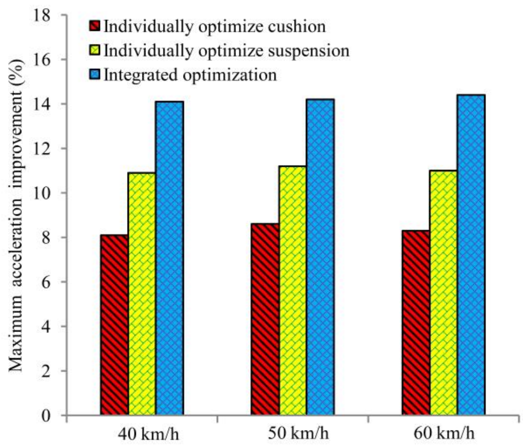

Figure 11. Comparsion of the maximum acceleration improvements.

From Table 2, it can be seen that for the seat system under the three optimization schemes, the ride comfort was improved in varying degrees compared with the original seat system. For the three optimization methods, under the 40 km/h condition, the driver vertical frequency weighted RMS acceleration is decreased by 7.2%, 10.2%, and 15.6%, respectively; the maximum acceleration is decreased by 8.1%, 10.9%, and 14.5%, respectively. Similar improvements are also observed to occur under other conditions in Figure 10 and Figure 11. There is a relative smaller room for the comfort improvement by individually optimizing the cushion or by individually optimizing the seat suspension. The integrated optimization is the best to improve the comfort. The results prove that all the three optimization schemes are effective and the integrated optimization method has the best effectiveness.

5. CONCLUSIONS

In this paper, three optimization schemes were investigated for determining an appropriate candidate design plan for the seat system of off-highway dump trucks. The three optimization schemes include individually optimizing the damping of the cushion, individually optimizing the damping of the seat suspension, and integrately optimizing both of them.

A standard commercially available seat for an off-highway dump truck was taken as the baseline model for the comparative analysis. The cushion model, the seat suspension model, and the air spring model for the seat system were created on the basis of the phenomenological modeling method, respectively. The nonlinear dynamic model of the seat system was built and validated by the test data from the field measurement. The results show that the dynamic model with sufficient accuracy can represent the real seat system. Based on the measured seat base acceleration, the corresponding damping coefficients under the three schemes were optimized. The road tests were conducted with the seat system under the three optimization schemes. The results show that all the three optimization schemes are effective. There is a relative smaller room for the comfort improvement by individually optimizing the cushion or by individually optimizing the seat suspension. The integrated optimization method has the best effectiveness. The mutual coupling between the cushion and the seat suspension can not been ignored for the damping optimization of the seat system. This study presents a valuable reference to protect the driver of off-highway dump trucks from the harmful vibration as much as possible.

6. ACKNOWLEDGMENT

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Damping Matching

چکیده

برای محافظت رانندەهای کامیون‌های تخلیه خارج از بزرگراه از ارتعاشات مضر، این پژوهش نتایج مقایسه‌ای بین بهینه‌ی سازی میرا کردن ارتعاشات را با طرح‌های مختلف طراحی بهینه‌ی سیستم صندلی ارائه می‌دهد. سه طرح بهینه سازی در نظر گرفته شده‌اند: بهینه سازی بالشتک به طور جداگانه، بهینه سازی سیستم تعلیق صندلی به طور جداگانه و بهینه سازی هر دو به طور یک‌پارچه. برای مقایسه‌ی سه طرح بهینه سازی، سیستم صندلی برای کامیون تخلیه خارج از بزرگراه به عنوان مدل پایه در نظر گرفته شده است. ابتدا پارامترهای بالشتک، تعلیق صندلی و هواپیما برای سیستم صندلی توسط آزمون مربوطه تعیین شد و مدل دینامیکی غیرخطی سیستم صندلی ایجاد شد. سپس مدل با داده‌های آزمون از اندازه گیری میدانی تایید شد. پس از آن، بر اساس شتاب پایه اندازه گیری شده، ضرایب مربوط به ماندگاری تحت سه طرح به ترتیب بهینه شدند. در نهایت، آزمایش‌های جاده‌ای برای بررسی و مقایسه میزان بهبود راحتی انجام شد. نتایج نشان می‌دهد که یک اتاق نسبتاً کوچک برای بهبود راحتی با یک دست بهینه سازی بالشتک یا به طور جداگانه بهینه سازی تعلیق صندلی وجود دارد. بهترین بهینه سازی برای بهبود راحتی، بهینه سازی یک‌پارچه ست.

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