Simulation Analysis and Optimization of a Passenger Car Ride Comfort

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\textbf{Abstract.} The vehicle ride comfort research is an important part of the vehicle development process. This paper takes a passenger car as the research object, establishes the vehicle rigid-flexible coupling ride comfort simulation model based on ADAMS, and simulates the vertical vibration acceleration time domain curve at the driver's position. Through the derivation of the theoretical formula, the corresponding weighted vibration level is obtained. Next, the weighted vibration level of the front seat is used as the measurement index. The stiffness of the front suspension torsion bar spring and the damping of the front and rear suspension dampers are selected as the optimization variables to optimize ride comfort. The optimization of the performance makes the peak of vertical acceleration vibration response significantly reduced, which has certain practical value.

1. Introduction

With the advancement of China's industry and the development of new automotive technologies, automobiles have provided more and more convenience to human beings in the daily life, so that people are increasingly coming up with new demands on automobiles when using them. In the case of ensuring its safety and economy, it will consider its ride comfort more [1]. So when developing and designing a car, it requires designers to consider not only their power and fuel economy, but also take into account the ride comfort indicators. In the process of driving a car, it is inevitable to be affected by shocks and vibrations from various factors such as road surface. The research on ride comfort of the car is to reduce the impact of such shocks and vibrations on the passengers and drivers, and control them within a reasonable range to make people feel comfortable by some technical measure [4]. Therefore, the ride comfort of the car is mainly evaluated by subjective driving experience of the members. In addition, the qualities of the car's ride comfort will directly affect the reliability of the car. The dynamic load, which generated by the vibrations and shocks of the road and the rotating parts such as the engine, the drive train and the wheels, will cause wear and tear on some parts of the car [5]. If conditions continue this way, the fatigue life of these part will be gradually reduced, which will affect the safety and reliability of the car.

Based on the theory of automobile system dynamics, this paper uses ADAMS/Car to establish the dynamic model of the vehicle and the torsion beam suspension model of the flexible body structure [2], and the simulation analysis of ride comfort under random road conditions is carried out [3]. At the same time, the RMS value of the vertical weighted acceleration at the driver's seat is taken as the optimization target, and the accuracy of the model is verified by simulation experiments. Whereafter, the stiffness and damping of the front and rear suspensions are optimized in order to improve the ride comfort [6].
2. Theoretical Analysis

2.1. Evaluation Method of Ride Comfort

When the car driving at high speed on the road, the shocks and vibrations are mainly generated by the driving speed, the unevenness of the random road, etc., passing by a complete set of vehicle vibration system which are composed of the elastic and damping elements such as tires, suspensions and cushions, and deriving from the acceleration from the seat to the human being’s body. Therefore, the ride comfort of the car is judged by the response of the human body to different vibration accelerations, that is, the comfort of the ride. So the main purpose of studying ride comfort of the car is to control the dynamic characteristics of the vibration system during driving. The weighted acceleration RMS value is usually used to evaluate the impact of vibration on comfort and health of people.

When using the most basic method to determine the vibration size, firstly we should calculate the value of the RMS of the weighted acceleration of each axial vibration. In the actual research, the commonly used method is spectrum analysis method, which converts the vibration acceleration time collected by the acceleration sensor into the power spectral density function, and then transforms the data through the filter grid for the weighted acceleration time, the formula is as follows:

\[
a_w = \left[ \int_{t_0}^{t_2} W^2(f)G_a(f)dt \right]^{\frac{1}{2}}
\]

Due to the convenience of experimental research, the three axial vibrations of the seat support surfaces, , , are usually considered together, and the total weighted acceleration value of the three axial directions is calculated as follows: Where: , , represent the weighted acceleration rms values of the x, y, and z axes, respectively.

\[
a_w = \left[ (1.4a_{xw})^2 + (1.4a_{yw})^2 + a_{zw}^2 \right]^{\frac{1}{2}}
\]

In order to more intuitively display the frequency of the vibration, the weighted vibration level often appears in the "human vibration meter", and the conversion formula between it and is as follows: Where represents the root mean square value of the reference acceleration, taking \( a_0 = 10^{-6} \) m/s²

\[
L_{aw} = 20\log\left(\frac{a_w}{a_0}\right) \text{ (dB)}
\]

Table 1 below shows the correspondence between the subjective perception of vibration of the human body and the weighted acceleration root mean square value and the weighted vibration level.

| \( a_w (m/s^2) \) | \( L_{aw} / dB \) | People's subjective feelings |
|-------------------|-----------------|-----------------------------|
| < 0.315           | 110             | No discomfort               |
| 0.315~0.63        | 110~116         | Some discomfort             |
| 0.5~1.0           | 114~120         | Rather uncomfortable        |
| 0.8~1.6           | 118~124         | Uncomfortable               |
| 1.25~2.5          | 122~128         | Very uncomfortable          |
| > 2.0             | 126             | Very uncomfortable          |

2.2. Ride Comfort Optimization Model

Design variables \( X=[K_s, K_{cr}]^T \) generally refer to independent parameters that need to be selected preferentially during the design process. In this paper, since the average damping coefficient of the suspension, the vertical stiffness of the seat, and the average damping coefficient are all fixed values,
the vertical stiffness value of the front and rear suspensions are selected as the design variable. Through the different matching of the vertical stiffness of the front and rear suspensions, the purpose of optimal analysis of ride comfort is achieved.

2.2.1. Determination of Optimization Objective Function \( \min_\sigma \). The purpose of optimizing the design is to optimize one or a few indicators. In this paper, the RMS of the vertical acceleration experienced by the driver when the car is driving at a speed of 60m/s on the B-class road surface is used as a single optimization goal, which optimizes ride comfort of the car to the greatest extent possible. To this end, this paper selects an eight-degree-of-freedom optimization model, the objective function \( \min_\sigma \) is defined as follows:

\[
\min_\sigma = \left[ \int_{0}^{B_0} w^2 (f) G_a (f) df \right]^{1/2}
\]  

(4)

2.2.2. Restrictions. In the process of optimizing the design, constraints are usually used to ensure the accuracy, safety, reliability, functionality and practicability of the system. Therefore, the design variables and the state variables of the system are often used as constraints. The following five aspects will be analyzed:

Vertical stiffness of the front and rear suspension: Usually, in order to ensure good ride comfort, the suspension of the car will be designed soft, so its static deflection will be relatively large. When \( 15 \leq f_s \leq 30 \text{(cm)} \), The limit value of the vertical stiffness of the front and rear suspensions should meet the following conditions:

\[
\frac{G \times l_r}{2(l_r+l_t) \times 0.30} \leq K_{st} \leq \frac{G \times l_f}{2(l_f+l_t) \times 0.15}
\]

(5)

Suspension deflection: The relationship between the limit stroke \( f_d \) of the suspension and the static stroke \( f_s \) is \([f_d] = 1.8 - 2.0 \sqrt{f_s} \), and \([f_d] = 1.9 \sqrt{f_s} \) is used in this paper. It is analyzed that when the \( f_0 \) is reduced, the limit stroke \([f_d] \) of the suspension is also reduced, and ride comfort of the car becomes better, but as the \( f_0 \) decreases, the dynamic deflection \([f_d] \) of the suspension increases. The probability that the suspension may hit the limit block will increase, resulting in deteriorating the ride comfort. Therefore, it is necessary to make certain restrictions on the dynamic deflection of the car suspension during design to ensure ride comfort of the car. After calculation and verification, we found that when the RMS value of the suspension deflection \( \sigma_d \leq [f_d]/3 \), the probability of the suspension impact limit stroke can be controlled within 0.3%, then:

\[
\begin{align*}
\sigma_{\text{def}} & \leq \frac{1}{3} [f_d] \\
\sigma_{\text{drt}} & \leq \frac{1}{3} [f_d] \\
\sigma_{\text{defr}} & \leq \frac{1}{3} [f_d] \\
\sigma_{\text{dtrh}} & \leq \frac{1}{3} [f_d]
\end{align*}
\]

(6)

Natural frequency of the suspension system: In the actual design process, reducing the natural frequency of the suspension system can significantly reduce the vibration acceleration of the vehicle body. Therefore, the natural frequency of the suspension system is generally required to be between 1.1 and 1.2 Hz, so that:

\[
f_0 = \frac{1}{2\pi} \sqrt{\frac{2(K_{st}+K_{sr})}{m_{cb}}}
\]

(7)

Relative dynamic load between wheel and road: The related experiments prove that the relative dynamic load \( F_d/G \) between the wheel and the road surface is closely related to the safety of the car. If the relative dynamic load value of the wheel is greater than 1, the car may appear to be dragged off the
road and lose road attachment. In the case of force, this may lead to dangerous situations. Therefore, we
must consider the influence of the relative dynamic load of the wheel and the road. After calculation, it
is found that the greater the relative dynamic load, the greater the damage to the road surface. When the
mean square value of the relative dynamic load \( R_{f} / G \leq 1/3 \), the probability of occurrence of the wheel
leaving the ground is 0.15\%, so there is:

\[
\begin{align*}
\sigma_{F_{df}} & \leq \frac{1}{3} \\
\sigma_{F_{dfr}} & \leq \frac{1}{3} \\
\sigma_{F_{dfl}} & \leq \frac{1}{3} \\
\sigma_{F_{dfk}} & \leq \frac{1}{3}
\end{align*}
\]

(8)

Suspension system damping ratio: In general, the damping of the suspension system is a small
damping, its value \( \zeta \in [0.2,0.4] \), and \( \zeta = \frac{C_{s}}{2\sqrt{\text{mass}}} \), so another related constraint can be got:

\[
\begin{align*}
0.2 & \leq \frac{C_{s}}{2\sqrt{k_{f}m_{cb}x_{l}(2l_{f}+l_{r})}} \leq 0.4 \\
0.2 & \leq \frac{C_{s}}{2\sqrt{k_{r}m_{cb}x_{l}(2l_{f}+l_{r})}} \leq 0.4
\end{align*}
\]

(9)

Through the analysis above, we combine the objective function \( \min \sigma \) with five constraints to form a
mathematical model for the optimization of the car. The model can visually reflect the various indicators
of ride comfort and provide a theoretical basis for the subsequent optimization process.

3. Model Establishment

3.1. Establish an ADAMS Model
The model of the passenger car is established in ADAMS/Car, and is optimized based on the
mathematical model of the optimization and related theories. The front suspension of the car studied in
this paper is the McPherson independent front suspension, and the rear one is the torsion beam semi-
independent rear suspension. The main parameters of the vehicle are as follows:

| Table 2. The main parameters of the vehicle. |
|--------------------------------------------|
| Basic parameters value Basic parameters value |
| Dimensions (mm) 7005×2040×2645 | Wheelbase (mm) 3935 |
| Front/rear Track (mm) 1690/1490 | Full load center of mass (mm) 1008 |
| Curb quality (kg) 3550 | Full load total mass (kg) 6150 |
| No-load front/rear axle load distribution (kg) 1676/1874 | Full load front/rear axle load distribution (kg) 2200/3950 |
| Front/rear suspension unsprung mass (kg) 80/200 | Centroid distance from the front axle (mm) 2024 |

The torsion beam suspension model of the flexible body structure is established according to the
given parameters in order to obtain the ride comfort data on the driver's seat, and then the finite element
model shown in figure 1 is established. The finite element mesh of the torsion beam suspension is
divided, then constrained according to the condition proposed above. And finally assigning parameters
to the constraints and performing analytical calculations.
Performing modal analysis in HyperMesh, calculate the modal neutral file (.MNF file) of the flexible body of the torsion beam suspension, and then import the generated neutral file into ADAMS/Car, combined with the specific hard point coordinates of the suspension, the load and constraints applied before. Finally, a simulation model of the rigid-flexible coupling of the whole vehicle is generated, as shown in figure 2 below.

**Figure 1.** Cross-sectional view of the beam.  
**Figure 2.** The vehicle model.

### 4. Ride Comfort Analysis

#### 4.1. Model Analysis

After the vehicle rigid-flexible coupling simulation model is established, the simulation analysis of ride comfort of the random input can be performed. The simulation was carried out at a speed of 60-100 km/h on the B-level road surface, each speed interval was 10 km/h, then test the vibration acceleration that was carried from the driver's seat position to the human body in three directions. Using the ADAMS/Post-Processor function, we can obtain the longitudinal, lateral and vertical vibration acceleration time domain curves at the driver's position. To acquire the axial RMS value, the FFT (Fast Fourier Transform) function in the ADAMS post-processing module can be used to transform each axial acceleration curve to obtain each axial acceleration power spectral density function $G_a(f)$. In figure 3 and 4, the vertical acceleration simulation curve and the vertical acceleration power spectral density curve at the driver's position when the virtual vehicle is driving on the B-level road at a speed of 80 km/h is shown, respectively.

**Figure 3.** Vertical acceleration simulation curve.
From this, the RMS value of the weighted acceleration at the driver’s position in each direction can be obtained, so we can calculate the weighted vibration level of the RMS and sum of the total weighted acceleration. According to the analysis method above, the above two parameter values of the vehicle driving at speeds of 60 km/h, 70 km/h, 80 km/h, 90 km/h, and 100 km/h can be calculated as shown in Table 3 below.

**Table 3.** Weighted acceleration RMS and weighted vibration levels of vehicles at different speeds.

| v (km/h) | a_w (m/s^2) | L_{aw} (dB) | People's subjective feelings |
|----------|-------------|-------------|-----------------------------|
| 60       | 0.3088      | 109.8       | No discomfort               |
| 70       | 0.3485      | 110.8       | Some discomfort             |
| 80       | 0.4728      | 113.5       | Some discomfort             |
| 90       | 0.3948      | 111.9       | Some discomfort             |
| 100      | 0.4368      | 112.8       | Some discomfort             |

As can be seen from the table above, the passenger may have an uncomfortable feeling when the driving speed is greater than 60 km/h, so it can be concluded that the car will be brought discomfortableness to the passenger due to vibration and other factors at high speed. Therefore, the ride comfort without optimization is not particularly desirable, and the ride comfort can be improved by changing the optimization of certain parameters.

**4.2. Real Vehicle Test.**

In order to explain whether the simulation model can accurately simulate the actual situation when the vehicle is running, actual test verification is required. The random input test of the real vehicle is carried out on the B-class road surface with similar simulation conditions. After comparing the simulation data with the experimental data which obtained by sensors and other experimental instruments, the feasibility of the simulation experiment and the correctness of the simulation model can be verified. This experiment uses three-way accelerometer, dynamic signal test system and data acquisition system for data acquisition and analysis, shown in figure 5.
Figure 5. Three-way accelerometer and data acquisition system.

The test method is as follows: The three-way accelerometer is mounted on the seat support bracket directly below the driver’s seat to measure the acceleration time in three directions. When the vehicle is started, accelerating to a speed of about 60 km/h and then drive at a constant speed. At this time, the acceleration signals of each axes can be acquired by the sensor and recorded by the data acquisition system. Similarly, recording the acceleration time of the vehicle in three directions when driving at 70km/h, 80km/h, 90km/h, and 100km/h. After analysis and calculation by dynamic signal test system and related equipment, the weighted vibration level of the front seat can be obtained as shown in table 4.

Table 4. Weighted vibration level of the front seat.

| v (km/h) | 60 | 70 | 80 | 90 | 100 |
|----------|----|----|----|----|-----|
| L_{aw} (dB) | 108.6 | 110.3 | 112.4 | 110.7 | 112.0 |

So it can be seen clearly from the table 4 above: although other factors such as the vibration caused by the sliding friction between the tire and the ground are not considered in the simulation process, but the simulation data is basically consistent with the experimental data. It shows that the vehicle model is basically consistent with the actual vehicle situation, and its correctness and accuracy can be guaranteed, which lays a foundation for the following ride comfort optimization.

5. Ride Comfort Optimization

In the optimization, considering that the human body is relatively sensitive to the vertical action, the weighted acceleration root mean square value $a_{aw}$ at the driver’s position is selected as the design target optimization function. Although many factors must be considered in the optimization of ride comfort of the vehicle, such as: structural parameters and stiffness of the suspension, damping parameters, shock absorber characteristics, cab parameters, tire stiffness and damping parameters, sprung mass and unsprung mass Parameters, etc., combined with the characteristics of the optimized platform and the size of the vehicle structure, this paper choose torsional spring stiffness of the front suspension and the damping of the front and rear suspension dampers as the optimization variables.

In the ADAMS insight module, the design parameters are arranged and combined, and select the parameters which are closest to the optimization target from them. Table 5 shows the design parameters of the front suspension stiffness and the front and rear suspension damping before and after optimization.

Table 5. Design parameters before and after optimization.

| | Front suspension stiffness (N·mm/deg) | Front suspension damping (N·s/mm) | Rear suspension damping (N·s/mm) |
|-----------------|-------------------------------------|---------------------------------|-------------------------------|
| Before          | 175000                              | 7.9                             | 4.3                           |
| After           | 174000                              | 8.7                             | 3.8                           |
The optimization result is brought into the previously established vehicle model for re-simulation, and the acceleration curve of the vertical direction at the driver's position before and after optimization at 80 km/h as shown in figure 6 is obtained.

![Comparison of vertical vibration acceleration curves before and after optimization.](image)

**Figure 6.** Comparison of vertical vibration acceleration curves before and after optimization.

After optimization, the vertical acceleration vibration response peak at the driver's position decreased from 0.1094 m/s² to 0.0954 m/s², which decreased by 12.8%, so the damping effect was better. After further processing, the comparison table of the acceleration root mean square value and the weighted vibration level at different speeds are obtained as shown in table 6.

| v (km/h) | \( a_w \) (m/s²) | \( L_{aw} \) (dB) |
|---------|----------------|------------|
|         | Before | After | Before | After |
| 60      | 0.3088 | 0.1048 | 109.8  | 100.4 |
| 70      | 0.3485 | 0.1376 | 110.8  | 102.8 |
| 80      | 0.4728 | 0.2968 | 113.5  | 109.4 |
| 90      | 0.3948 | 0.1972 | 111.9  | 105.9 |
| 100     | 0.4368 | 0.2514 | 112.8  | 108.0 |

It can be seen clearly from the above table 6 that after optimization, the weighted vibration level of the vertical direction of the driver is reduced at each speed, so that the vibration intensity transmitted to the human body becomes smaller, and the human body feel become well. At the same time, with the improvement of the speed of the vehicle, the ride comfort and stability of the vehicle are greatly improved, so the operational reliability is.

6. Conclusion
In this paper, the RMS of the vertical direction of the driver is taken as the optimization targe. After analyzing various factors affecting ride comfort of the car, the author establishes the eight-degree-of-freedom vehicle to optimize. The accuracy of the model is verified from the perspective of theoretical calculation and actual experiment, which lays a foundation for the later optimization analysis.

The ride comfort simulation model was established by ADAMS/Car. The front suspension stiffness, front suspension damping and rear suspension damping parameters were selected for optimization design. Then comparing and analyzing the related indicators before and after optimization, the result proves the correctness of the optimization scheme.
The parameterized optimization method is used to optimize its structure. By analyzing and optimizing the structural coordinate values, the target performance can be provided under the condition of small variation, and the guiding direction for future structural optimization is provided.

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