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Research on Dynamic Optimization for Road-friendly Vehicle Suspension

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Abstract: The heavy vehicle brings large dynamic loads to the road surface, which would reduce vehicle ride comfort and shorten road service life. The structure characteristic of heavy vehicle suspension has a significant impact on vehicle performance. Based on the D'Alembert principle, the dynamics models of independent and integral balanced suspension are proposed considering mass and inertia of balancing rod. The sprung mass acceleration and the tire dynamic force for two kinds of balanced suspension and the traditional quarter vehicle model are compared in frequency-domain and time-domain respectively. It is concluded that a quarter vehicle model simplified for balanced suspension could be used to evaluate the ride comfort of vehicle well, but it has some limitations in assessing the vehicle road-friendliness. Then, the sprung mass acceleration and the road damage coefficients are also analyzed under different vehicle design and running parameters at detail. Some conclusions are obtained: low suspension stiffness, high suspension damping and low tire stiffness are all favorable to improve vehicle performance; there is a saturation range of suspension damping enhancing vehicle performance; improving the road surface roughness and avoiding the no-load running are two effective methods to accomplish the better ride comfort and road-friendliness. The suspension stiffness and damping parameters are chosen for optimal parameters matching of road friendliness based on the approximation optimization method. *Copyright © 2014 IFSA Publishing, S. L.*

Keywords: Heavy vehicle dynamics, Balanced suspension, Ride comfort, Road-friendliness, Optimization.

1. Introductions

It is a tendency to improve the efficiency of heavy vehicle transport through enhancing cargo capacity. The maximum axle load of the vehicle is limited by the regulations, so the multi-axle vehicle suspension for a tandem or a tri-axle vehicle comes into being to provide for an even distribution of the vehicle load across the axles. However the heavy vehicle brings large dynamic loads to the road surface, which would reduce vehicle ride comfort and shorten road service life [1]. The DIVINE (Dynamic Interaction between Vehicles and Infrastructure Experiment) project report shows that 50 % of the road maintenance and rehabilitation are caused by the heavy vehicles [2]. Researchers have also proved that the structure characteristic of vehicle suspension has a significant impact on vehicle performance including ride comfort and road-friendliness [3, 4]. The detailed studies on ride comfort and handling stability for the two-axle vehicle have been done by previous researchers. The vehicle road-friendliness is first defined by EU legislation [5], which is used to describe the vehicle damage effects on road surface. The interaction study between vehicle and road has attracted wide attentions. At present the traditional two-axle [6] or quarter-vehicle [7] model is still widely used to study tandem axles or tri-axle heavy vehicle dynamic behaviors. These heavy vehicle dynamics models cannot consider the structure characteristic of the multi-axle vehicle, such as the balanced suspension. The simplified for the balanced suspension may lead to some calculation errors during the pavement design. Huang pointed out that it is insecurity to regard the tandem axles or tri-axles loads as a group of loads acting on road surface once, but it is too conservative to make every axle load calculated to the road surface once during the pavement life design [8]. Cebon proposed that for equal damage to flexible or rigid pavements, tandem and tri-axle groups can carry more weight than the same number of widely spaced single axles, because the primary response fields of nearby axles overlap [9]. So it is necessary to establish the reasonable balanced suspension model of heavy vehicle, which could be used to accurately analyze the effect of structural features on ride comfort and road-friendliness. The best advantages of balanced suspension is all the wheels of vehicle are able to ensure good contact with the ground and both wheel bear the same load avoid overloading [10]. Leaf spring suspension systems are widely used on both single-axle and tandem axles and tri-axles vehicle. This kind of heavy vehicle suspension could bring more dynamic loading shock to the road surface than airbag suspension, which is restricted by the EU legislation [5]. However, more than 95 % of heavy vehicle suspensions are equipped with the leaf spring balanced suspension system in China according to statistics. There are few literatures about the balanced suspension modeling and dynamics analysis. Kuai established a tri-axle heavy vehicle model equipped with the balanced suspension system and the only ride quality is discussed [11]. Yang presented a parametric model of tractor with tandem suspension in multi-body dynamics software ADAMS not considering the road-friendliness [12]. A nine degrees-of-freedom tractor and semitrailer suspension model is established, the balancing rod mass and rod moment is not considered [13]. These researches on the balanced suspension are mainly built to analyze the vehicle ride comfort and the relationship between the balanced suspension structure feature and the road damage caused by dynamic loading is less discussed.

In this paper, according to the vehicle suspension working principle and structural features, the dynamics models of independent and integral balanced suspension are firstly established. In order to analyze the difference in evaluating ride comfort and road-friendliness for different suspension types, the sprung mass acceleration and the tire dynamic force are compared in frequency-domain and time-domain for two kinds of balanced suspension and traditional quarter vehicle model. The influence of vehicle design and running parameters on ride comfort and road-friendliness are analyzed at detail, which could be a scientific and rational basis for improving the vehicle performance. Finally, the suspension stiffness and damping parameters are chosen for optimal parameters matching of road friendliness based on the approximation optimization method.

2. Modeling of Balanced Suspension

In order to discuss structure features of balanced suspension effects on vehicle ride comfort and road-friendliness, two kinds of balanced suspension models are proposed in this paper and the specific structure of balanced suspension is according to [14].

2.1. Balanced Suspension System

The balanced suspension structure is designed to achieve load distribution of vehicle axles. It utilizes the ‘level’ principle to distribute the axle load equally and to reduce the shock from the road irregularities. The suspension structure includes leaf springs, intermediate axle, rear axle, spindle, reaction rod, U-bolt, buffer stopper and so on, as shown in Fig. 2. Both ends of leaf springs are supported on intermediate and rear half shaft bushings, respectively. Then the center of leaf springs is connected to the vehicle frame by a pin joint.

2.2. Independent Balanced Suspension Model

The leaf spring is a main component of balanced suspension which could only transfer vertical force and lateral force, but not deliver traction, braking force and reaction torque. For the first kind of balanced suspension model: the leaf spring is simplified as spring and balanced rod. Thus a leaf spring is equivalent to two same springs linked with a rigid balancing rod. The front and the rear axle of balanced suspension vibrate independently and don’t interfere with each other. The balancing rod weight and the moment of inertia are all taken into account in this model. The model is named as “Independent Balanced Suspension’ in this paper, as shown in Fig. 1.

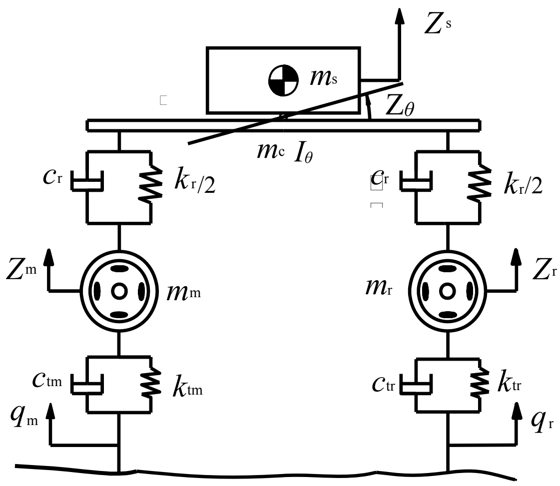


Fig. 1. Independent Balanced Suspension model considering rod moment of inertia.

Some main parameters of the vehicle model are listed as follows: c_r, k_r – damping and stiffness of leaf spring. c_{tm}, k_{tm} – damping and stiffness of front tire. c_{tr}, k_{tr} – damping and stiffness of rear tire. m_m, m_r – unsprung mass. m_s – sprung mass. m_c – mass of balancing rod. I_θ – moment of inertia of balancing rod. q_m, q_r – vertical displacement of front and rear tire. Z_s, Z_θ, Z_m, Z_r – vertical displacement of sprung mass, pitch angle of balancing rod, vertical displacement of front and rear tire. According to D' Alembert principle, the differential equation of motion for the Independent Balancing Suspension (4 degrees-of-freedom).

$$\begin{cases} (m_s+m_c)\ddot{Z}_s+k_rZ_s-\frac{1}{2}k_rZ_m-\frac{1}{2}k_rZ_r+c_r\dot{Z}_s-\frac{1}{2}c_r\dot{Z}_m-\frac{1}{2}c_r\dot{Z}_r=0 \\ I_\theta\ddot{Z}_\theta+\frac{1}{4}k_rb^2Z_\theta+\frac{1}{4}k_rbZ_m-\frac{1}{4}k_rbZ_r+\frac{1}{4}c_rb^2\dot{Z}_\theta+\frac{1}{4}c_rb\dot{Z}_m-\frac{1}{4}c_rb\dot{Z}_r=0 \\ m_m\ddot{Z}_m-\frac{1}{2}k_rZ_s+\frac{1}{4}k_rbZ_\theta+(\frac{1}{2}k_r+k_{tm})Z_m-\frac{1}{2}c_r\dot{Z}_s+\frac{1}{4}c_rb\dot{Z}_\theta+(\frac{1}{2}c_r+c_{tm})\dot{Z}_m=0 \\ m_r\ddot{Z}_r-\frac{1}{2}k_rZ_s-\frac{1}{4}k_rbZ_\theta+(\frac{1}{2}k_r+k_{tr})Z_r-\frac{1}{2}c_r\dot{Z}_s-\frac{1}{4}c_rb\dot{Z}_\theta+(\frac{1}{2}c_r+c_{tr})\dot{Z}_r=0 \end{cases} \tag{1}$$

2.3. Integral Balanced Suspension Model

For the second class model, the leaf spring of balanced suspension is regarded as a whole. The middle part of the leaf springs is connected with sprung mass. Both ends of the leaf springs are separately linked with a front and a rear axle through a rigid balancing rod.

The front and the rear axle's vibration are coupled together with the rod. The rod weight and the moment of inertia are also taken into account in this model. This model is named as "Integral Balanced Suspension', as shown in Fig. 2.

According to D' Alembert principle, the differential equation of motion for the integral balanced suspension (3 degrees-of-freedom) is:

$$\begin{cases} m_s\ddot{Z}_s+k_rZ_s-k_rZ_c+c_r\dot{Z}_s-c_r\dot{Z}_c=0 \\ (m_c+m_m+m_r)\ddot{Z}_c-k_rZ_s+(k_r+k_{tm}+k_{tr})Z_c+(-\frac{b}{2}k_{tm}+\frac{b}{2}k_{tr})Z_\theta-c_r\dot{Z}_s+(c_r+c_{tm}+c_{tr})\dot{Z}_c \\ +(-\frac{b}{2}c_{tm}+\frac{b}{2}c_{tr})\dot{Z}_\theta=k_{tm}q_m+k_{tr}q_r+c_{tm}\dot{q}_m+c_{tr}\dot{q}_r \\ (I_\theta+m_mb^2/4+m_rb^2/4)\ddot{Z}_\theta+b/2(-k_{tm}+k_{tr})\ Z_c+b^2/4(k_{tm}+k_{tr})\ Z_\theta+b/2(-c_{tm}+c_{tr})\ \dot{Z}_c+ \\ b^2/4(c_{tm}+c_{tr})\ \dot{Z}_\theta=-k_{tm}q_mb/2+k_{tr}q_rb/2-c_{tm}\dot{q}_mb/2+c_{tr}\dot{q}_rb/2 \end{cases} \tag{2}$$

As previously mentioned a quarter suspension model is usually adopted to study the balanced suspension dynamics behaviors. In order to compare the differences between the traditional quarter suspension and proposed balanced suspension, the differential equation of motion for a quarter suspension is also given as follows.

$$\begin{cases} m_s\ddot{Z}_s+c_s(\dot{Z}_s-\dot{Z}_t)+k_s(Z_s-Z_t)=0 \\ m_t\ddot{Z}_t+c_s(\dot{Z}_t-\dot{Z}_s)+k_s(Z_t-Z_s)+c_t(\dot{Z}_t-\dot{q})+k_t(Z_t-q)=0 \end{cases} \tag{3}$$

where m_s and m_t are the sprung mass and unsprung mass, c_s and k_s are the damping and stiffness of suspension, c_t and k_t are the damping and stiffness of tire.

3. Comparison Analysis for Three Kinds of Suspension Model

The vehicle ride comfort and the road-friendliness are compared for two kinds of balanced suspensions and quarter suspension in frequency-domain and time-domain, respectively. In order to illustrate conveniently, the Independent Balanced Suspension mode is called as Model 1 and the Integral Balanced Suspension is called as Model 2 for short. The parameters of a heavy vehicle suspension are chosen as follows:

m_s=11523 kg, $m_m=m_r$=676 kg, I_θ=351 kg.m^2, m_c=177 kg, k_r=2064000 N/m, c_r=25320 N.s/m, k_{tm}=2200000 N/m, c_{tm}=7000 N.s/m.

And the parameters of a quarter suspension are m_s=5761.5 kg, m_t=676 kg, k_s=1032000 N/m, c_s=12 660 N.s/m, k_t =2200000 N/m, c_t =7000 N.s/m, u=60 km/h. The front and rear wheel inputs of balanced suspension are obtained considering the delay times τ , according to the literature [15, 16].

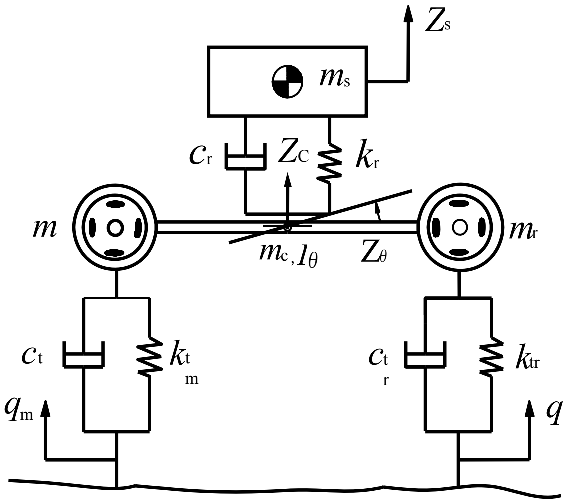


Fig. 2. Integral balanced suspension model considering rod moment of inertia.

3.1. Dynamics Response Analysis in the Frequency Domain

Because the proposed suspensions model are all linear system, the PSD (Power Spectral Density) of sprung mass acceleration and the tire dynamic force could be obtained by TF (Transfer Function) method. The results are shown in Fig. 3 and Fig. 4.

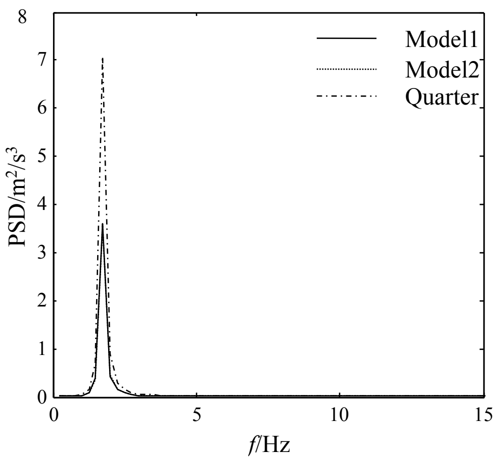


Fig. 3. PSD of sprung mass acceleration.

It can be observed from Fig. 3 and Fig. 4 that one main peak frequency concentrates at 2.25 Hz for three kinds of suspension models, which corresponds to bounce modal of vehicle sprung mass. As shown in Fig. 4, in addition to the same low peak frequency of 2.25 Hz, there is another peak at a frequency of approximately 8.5 Hz and 9.5 Hz for the balanced model, which reflects a pitching modal of the balancing rod. And the quarter suspension model does not include such high frequency component.

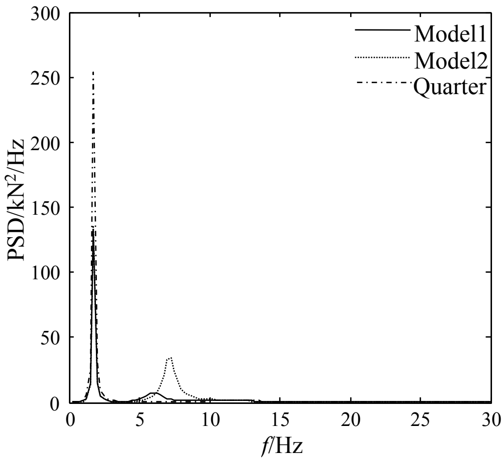


Fig. 4. PSD of tire dynamic force.

3.2. Dynamics Response Analysis in the Time Domain

Based on a numerical method of Newmark-β [17], the transient dynamics responses in time domain are computed, as shown in Fig. 5 and Fig. 6.

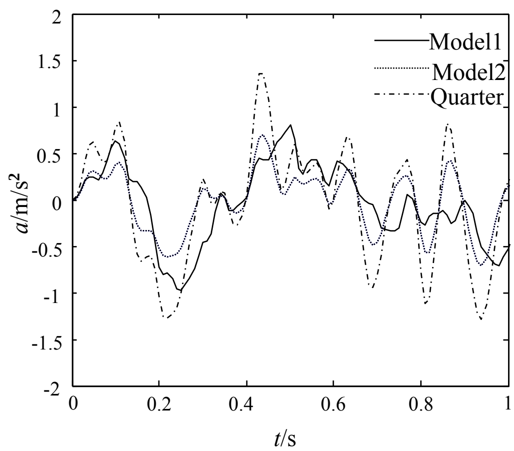


Fig. 5. Sprung mass acceleration response in time domain.

It can be seen from Fig. 5 and Fig. 6 that there has the similar distribution in vehicle body acceleration (mass acceleration) and tire dynamic force for three kinds of models, which is decided by the bounce modal of sprung mass at 2.25 Hz. The tire force peak amplitudes of the balanced suspension are bigger than the quarter vehicle model's due to the pitch motion of balancing rod in high frequency.

In short, a quarter vehicle model simplified for balanced suspension could be used to evaluate the vehicle ride comfort well. However when the model is used to assess vehicle road-friendliness, there exist some limitations (the really shock to the road surface could not reflect).

To obtain the accurate response results, the proposed balanced suspension model provide a better model. In addition, the balanced suspension is initially designed to equilibrate the vehicle heavy axle load, but it is found that the balanced suspension tends to increase tire dynamic loads, causing bigger road damage.

Therefore, it is necessary to analyze the difference in evaluating ride comfort and road-friendliness for different suspension types.

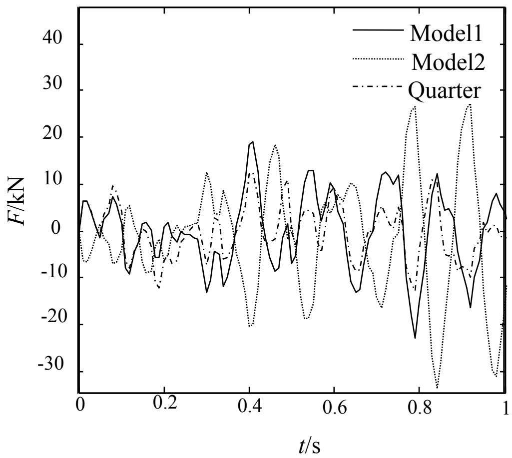


Fig. 6. Tire force response in time domain.

4. Effect of Vehicle Design and Running Parameters

Based on subjectivity and objectivity factors, the parameters affecting vehicle performances can be divided into two groups: design parameters and running parameters. The design parameters include suspension stiffness, suspension damping, tire stiffness and tire damping, balancing rod moment of inertia. And the running parameters include road roughness, vehicle speed and sprung mass. It is important to realize that the actual forces on the road surface are not equal to the static axle loads, but vary because of vehicle dynamic, which is called dynamic loads. It is well known that DLC (Dynamic Load Coefficient) is widely used to evaluate the vehicle dynamic load on road damage (firstly proposed by Sweatman [18]). But DLC is an oversimplified criterion without considering the tire force distribution on road surface. The dynamic tire forces are found to be repeatable in space because heavy vehicles often travel at similar speeds with similar payloads, dimensions, suspensions, and tire [13], Cole proposed 95th percentile fourth force by combining the fourth power theorem and spatial repeatability characteristics [19, 20], which is given by:

$$J = \frac{1.65\sigma_{A^4}}{m_{A^4}} + 1, \tag{4}$$

where A^4 is the aggregate fourth power force, σ_{A^4} and m_{A^4} are the standard deviation and the mean value of aggregate fourth power force. The aggregate fourth power force of tires is expressed by:

$$A_k^n = \sum_{j=1}^{N_a} F_{jk}^n, \quad k=1,2,...N_s, \tag{5}$$

where F_{jk} is the force applied by tire j to point k along the wheel path, N_a is the number of vehicle axles, N_s is the number of points along the wheel path. Based on the 'fourth power law', n=4 (bituminous pavement), the aggregate fourth power force, expressed in Eq. 4, could be written as a continuous function of time t:

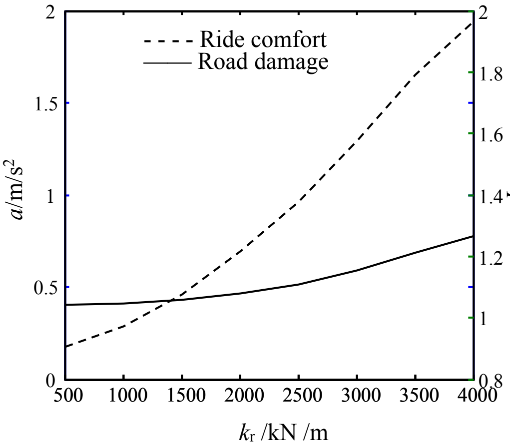
$$A^4(t) = F_1^4(t) + F_2^4(t+T_{12}) + \cdots + F_n^4(t+T_{1n}), \tag{6}$$

where $F_n(t)$ is the transient force history of axle j and T_{1n} is the time delay between the first and j^{th} axle following the same wheel path.

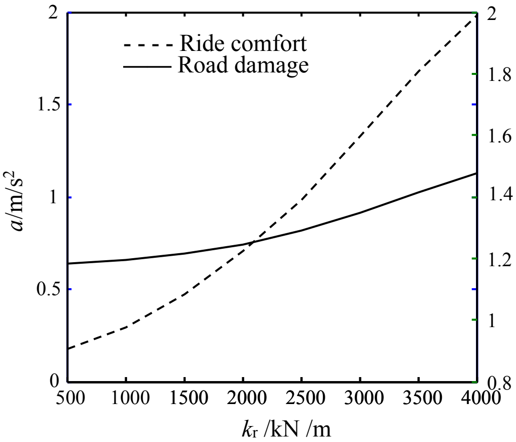
This kind of assessment criterion is more accurate to describe the destructive effects of dynamic tire force on road surface. Thus the vehicle body acceleration and the 95th percentile fourth coefficient are separately chosen to weigh vehicle ride comfort and road-friendliness in the following analysis.

4.1. Effect of Suspension Stiffness

The ride comfort and the road-friendliness are compared under different stiffness of balanced suspension, respectively, as shown in Fig. 7 (a) and Fig. 7 (b). In the following figures, the left longitudinal coordinates represent vehicle body acceleration and the right longitudinal coordinates represent road damage coefficient, which can well display the ride comfort and road friendliness in the same picture.



(a) Model



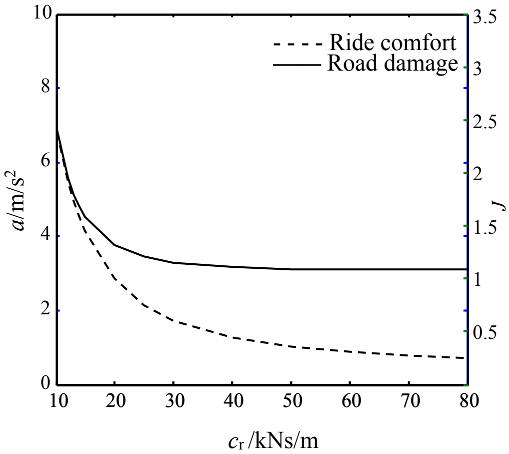
(b) Model 2

Fig. 7. Ride comfort and road-friendliness under different suspension stiffness.

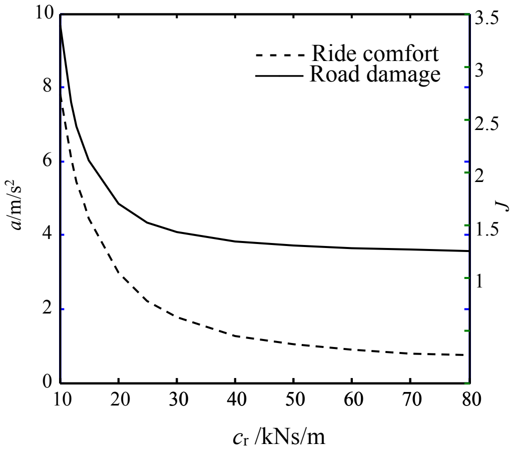
It can be observed from Fig. 7 that: both of vehicle body acceleration and road damage coefficient increase with the increase of leaf spring stiffness. The influence of leaf spring stiffness on ride comfort is larger than that on road-friendliness. So it is very unfavorable for the ride comfort when the vehicle suspension has relatively large stiffness. With the increase of leaf spring stiffness, both models have the same change of ride comfort. Road damage coefficient for the Model 1 is smaller than that for the Model 2 under the same parameter of leaf spring stiffness.

4.2. Effect of Suspension Damping

The ride comfort and the road-friendliness are compared under different damping of balanced suspension, respectively, as shown in Fig. 8 (a) and Fig. 8 (b).



(a) Model 1



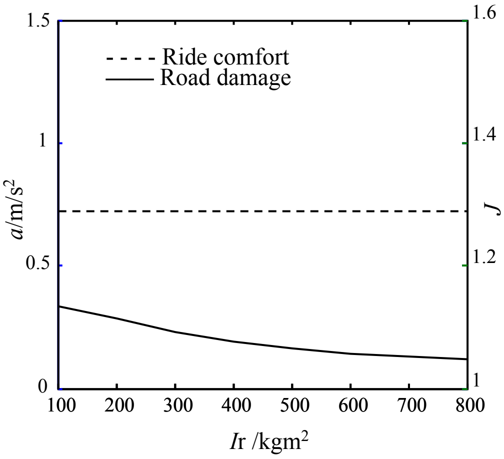
(b) Model 2

Fig. 8. Ride comfort and road-friendliness under different suspension damping.

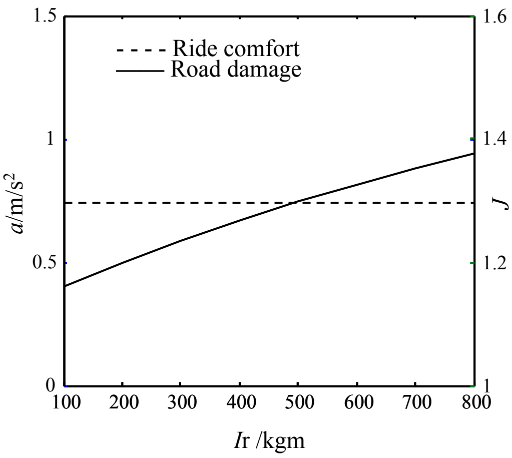
It can be observed from Fig. 8 that: the road damage coefficient of both models shows the same downtrend while the suspension damping increases. The Model 1's coefficient falls nearly 55 % and the model 2's falls 63.1 %. So the high suspension damping is benefit to improve vehicle ride comfort and road-friendliness. The ride comfort and the road-friendliness are particularly poor with a small suspension damping, especially for Model 2. Furthermore, the road damage coefficient tends to a stable value when the damping exceed 25 kN·s/m, which could be called "performance hardening". So as to achieve the better vehicle performance, it is not only need to restrict the lowest suspension damping, but also find out an effective damping range.

4.3. Effect of Balanced Suspension Rod Moment of Inertia

The ride comfort and road-friendliness are compared under different rod moment of inertia of balanced suspension, respectively, as shown in Fig. 9 (a) and Fig. 9 (b).



(a) Model 1



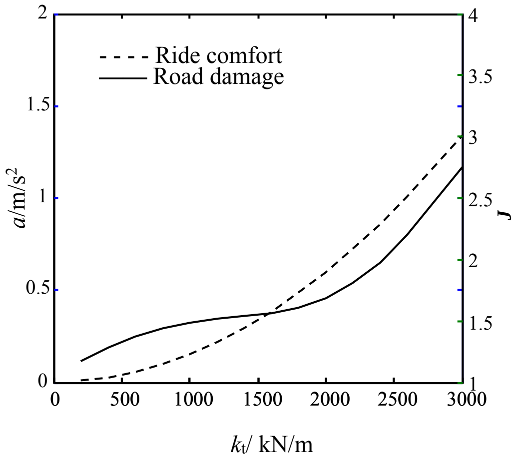
(b) Model 2

Fig. 9. Ride comfort and road-friendliness under different rod moment of inertia.

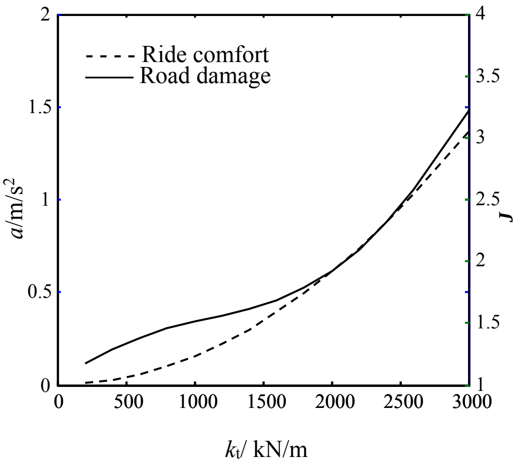
Fig. 9 (a) and Fig. 9 (b) show that: the rod inertia of balanced suspension has none influence on ride comfort, which could also be derived from Eq. 1 and Eq. 2. With the increase of rod inertia, the road damage coefficient of two models have contrary tendency: Model 1's decreases and Model 2's increase. As we know the balancing rod is simplified by leaf spring and its inertia is decided by leaf spring weight, section area and assembly height. Thus the vehicle performance could be improved by optimizing these parameters of leaf spring.

4.4. Effect of Tire Stiffness

The ride comfort and road-friendliness are compared under different tire stiffness of balanced suspension, respectively, as shown in Fig. 10 (a) and Fig. 10 (b). It can be observed from Fig. 10 that: with the increase of tire stiffness (where the tire stiffness represent the dual-tire stiffness), the vehicle body acceleration and the road damage coefficient tend to increase.



(a) Model 1



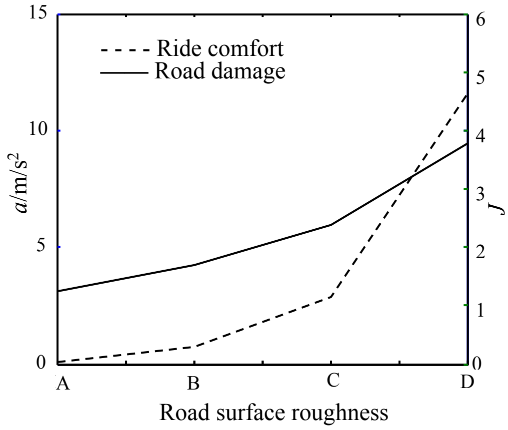
(b) Model 2

Fig. 10. Ride comfort and road-friendliness under different tire stiffness.

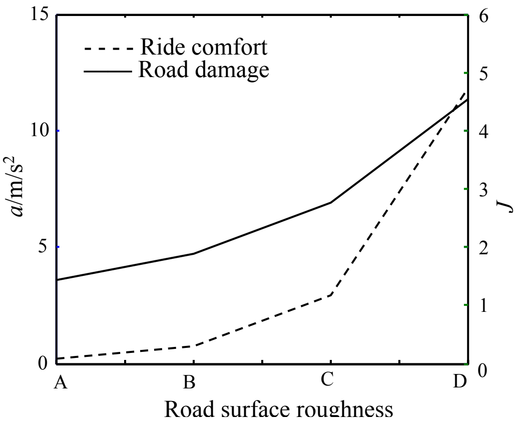
Research shows that there is a linear relationship between tire pressure and tire stiffness [21]. In order to improve vehicle performance, it is an effective method to limit the maximum tire pressure. In the range of about 1000~2000 kN/m, both models change very slightly. When the tire stiffness is larger than 2000 kN/m, the road damage coefficient suddenly increases.

4.5. Effect of Road Roughness

The ride comfort and the road-friendliness are compared under different road roughness, respectively, as shown in Fig. 11 (a) and Fig. 11 (b).



(a) Model 1



(b) Model 2

Fig. 11. Ride comfort and road-friendliness under different road roughness.

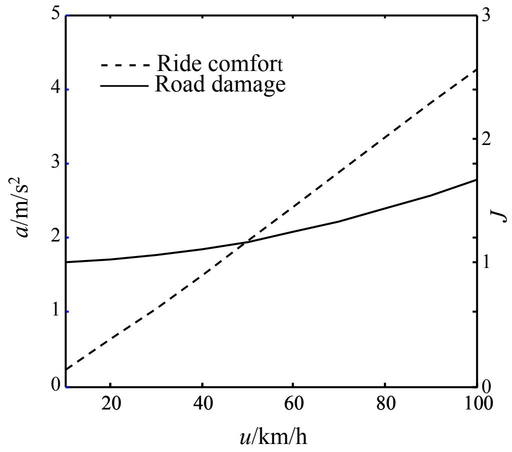
It can be observed from Fig. 11 that: the ride comfort and the road-friendliness of both vehicle models worsen gradually with the road level rising. When the road level is national standard C-class, vehicle ride comfort and road- friendliness is deteriorating rapidly, especially for ride comfort. Therefore, improving the road surface roughness is also an effective way to improve vehicle ride comfort and road-friendly.

4.6. Effect of Vehicle Speed

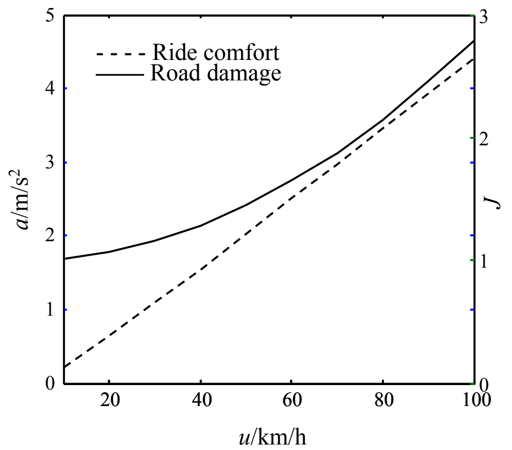
The ride comfort and the road-friendliness are compared under different vehicle speed, respectively, as shown in Fig. 12 (a) and Fig. 12 (b).

It can be observed from Fig. 12 that:

With the vehicle speed increasing, body acceleration and road damage coefficient also show a rise tendency. In addition, the vehicle body acceleration of the increases linearly with the increase of vehicle speed. Thus the high-speed trend for vehicle traffic has put forward higher requirement of ride comfort and road-friendliness. The increase rate of road damage coefficient for Model 2 is faster than that for the Model 1.



(a) Model 1



(b) Model 2

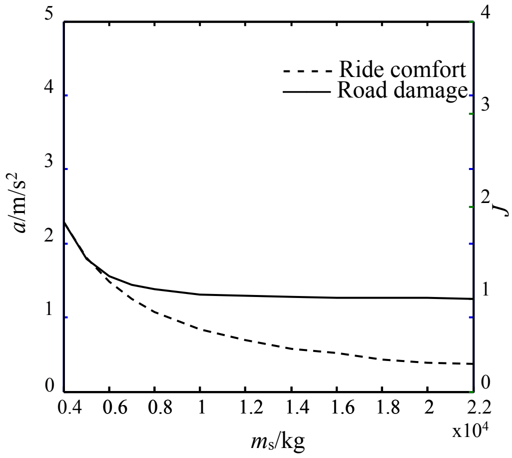
Fig. 12. Ride comfort and road-friendliness under different vehicle speed.

4.7 Effect of Sprung Mass

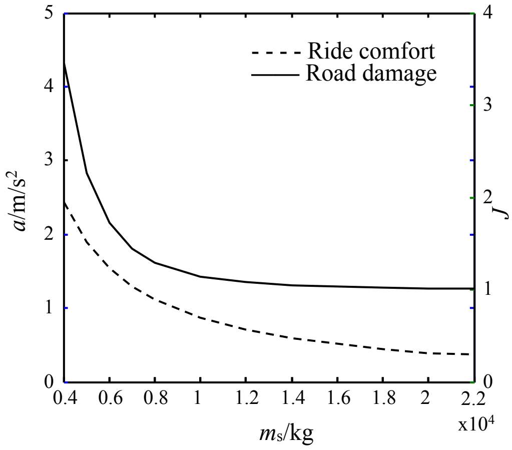
The ride comfort and the road-friendliness are compared under different sprung mass, respectively, as shown in Fig. 13 (a) and Fig. 13 (b).

It can be observed from Fig. 13 that:

When the sprung mass is greater than 8000 kg, the body acceleration decreases slowly and road damage coefficient decreases gradually to a stable value. For the Model 1, the road damage coefficient under the non-loading is 1.3 times the coefficient under the full-loading. The road-friendliness performance of Model 2 is even worse than that of Model 1's. Thus avoiding vehicle no-load running is favorable for achieving the better ride comfort and road-friendliness.



(a) Model 1



(b) Model 2

Fig. 13. Ride comfort and road-friendliness under different sprung mass.

4.8. Optimal Parameters Matching for Road Friendliness

Based on the proposed balanced suspension models, the vehicle ride comfort and road damage trends are analyzed followed by the vehicle parameters changing. Some useful conclusions are obtained for improving the road friendliness. And the aforesaid analysis also shows that the suspension stiffness and damping are the uppermost parameters for the vehicle performance. But it is not enough to achieve a better vehicle performance only through the analysis of one parameter changing. The suspension stiffness and damping parameters are chosen to study an optimal parameter matching of road friendliness based on the approximation optimization method. Fig. 14 show the contour pictures of an integrated display of road damage between suspension stiffness and damping parameters. The correspondent conditions for these pictures are the standard road roughness in national standard of A-class, B-class, C-class and D-class.

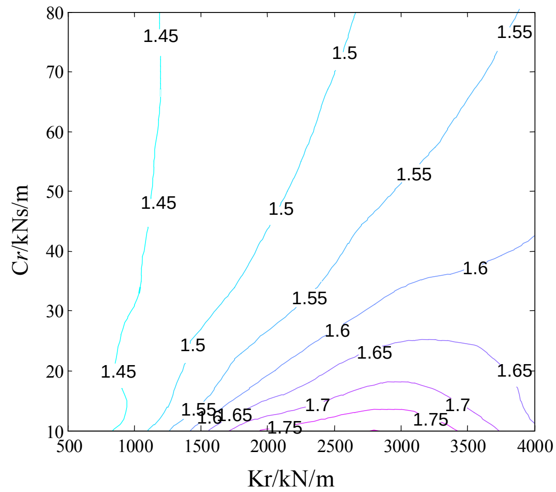


Fig. 14 (a). Contour pictures of road damage between suspension stiffness and damping parameters: A-class Road surface roughness.

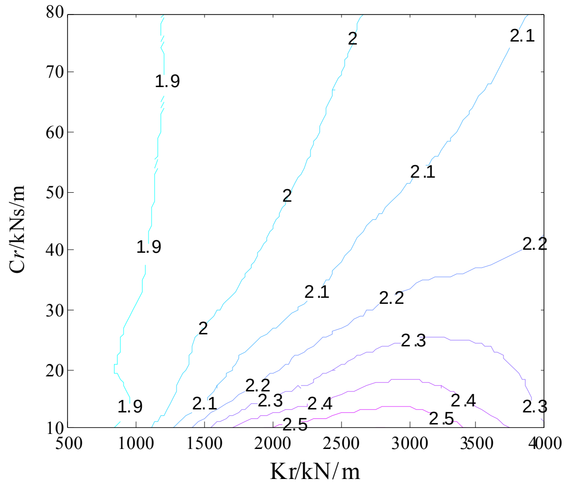


Fig. 14 (b). Contour pictures of road damage between suspension stiffness and damping parameters: B-class Road surface roughness.

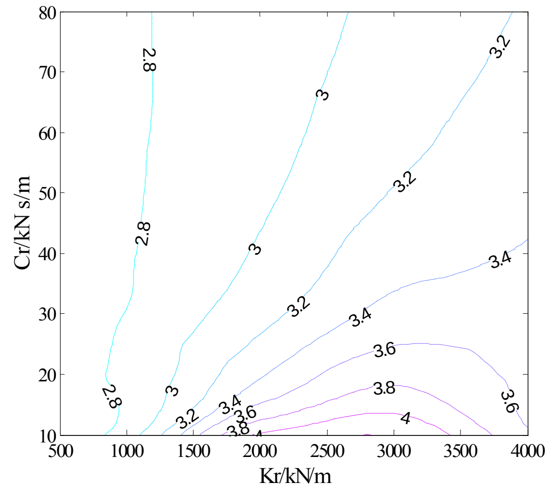


Fig. 14 (c). Contour pictures of road damage between suspension stiffness and damping parameters: C-class Road surface roughness.

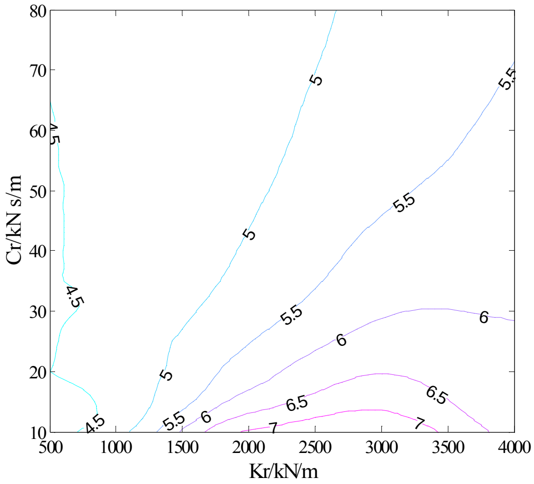


Fig. 14 (d). Contour pictures of road damage between suspension stiffness and damping parameters: D-class Road surface roughness

It can be observed from Fig. 14: the road damage coefficients suddenly increased with the road condition worsen (same as the prior conclusion). Under every road surface condition, a set of optimal parameter matching between the stiffness and the damping parameter can be obtained for the least damage coefficient. And the suspension stiffness parameter plays a major role for the road damage than the damping parameter. When the suspension stiffness is less than 1000 kN/m, the road damage keeps a lower level. But this optimum value of stiffness is to low not to match the practice condition (the normal leaf spring stiffness is about 2000 kN/m) except for the air spring. Therefore for the leaf spring suspension adopted widely in our country, it is an effective method to minimize the heavy loading to the road surface through designing an optimal matching relationship between the stiffness and the damping parameter.

5. Conclusions

According to working principle and structural features of balanced suspension, the independent and the integral balanced suspension are established. Several conclusions may be drawn as follow:

1) The quarter suspension model could be adopted to assess vehicle ride comfort well, but it has limitation in evaluating vehicle road friendliness. Furthermore, the balanced suspension is designed to equilibrate the heavy vehicle axle load, but it is found that the balanced suspension tends to increase the tire dynamic loads, causing bigger road damage.

2) The same law for both balanced suspension models affecting vehicle ride comfort and road-friendliness is: ① Low suspension stiffness, high suspension damping and low tire stiffness are all favorable to improve vehicle performance. ② There is a saturation range of suspension damping enhancing vehicle performance. ③ Improving the road surface roughness and avoiding the no-load running are two effective methods to accomplish the better ride comfort and road-friendliness.

3) The suspension stiffness plays a major role for the road damage than the damping parameter. For the leaf spring suspension adopted widely in our country, it is an effective method to minimize the heavy loading to the road surface through designing an optimal matching relationship between the stiffness and the damping parameter.

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