Dynamic performance analysis and parameters perturbation study of inerter–spring–damper suspension for heavy vehicle

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Abstract
Inerter, a new type of mass element, can increase the inertia of motion between two endpoints. In order to study the dynamic inertia effect of inerter–spring–damper suspension for heavy vehicle on ride comfort and road friendliness, the inerter–spring–damper suspension is applied and its mechanism is studied. This paper establishes a half vehicle model of inerter–spring–damper suspension for heavy vehicle. The parameters of inerter–spring–damper suspension for heavy vehicle are optimized by multi-objective genetic algorithm and system simulations are carried out. The parametric influence of different spring stiffness, damping coefficient, inertance, and load on suspension performance is also studied. The simulation results demonstrate that the centroid acceleration and pitch angular acceleration are improved by 24.90% and 23.54%, respectively, and the comprehensive road damage coefficient is reduced by 4.05%. The results illustrate that the inerter–spring–damper suspension can decrease the vertical vibration of vehicle suspension especially in low frequency and reduce the road damage. The analyses of suspension parameters perturbation reveal their different effect laws of the different wheels on vehicle ride comfort and road friendliness, which provide a theoretical basis for setting parameters of inerter–spring–damper suspension.

Keywords
Heavy vehicle, inerter–spring–damper suspension, parameter matching, ride comfort, road friendliness

Introduction
In recent decades, with the rapid development of global economy and highway transportation, heavy vehicles as the main equipment for freight transport, their production scale, and quantity are also growing rapidly. Heavy vehicles have become the main factors of road damage due to their heavy load and dynamic tire load during driving.¹ Road friendliness is the performance of reducing road damage caused by tire load as much as possible when vehicles are driving on uneven roads.² The research³ shows that the damage of vehicle to road is mainly related to the axle load and the tire ground contact, and the dynamic tire load is the decisive factor to cause road fatigue damage and improve road friendliness. In the case that road conditions cannot be greatly improved, the damage to the road caused by dynamic tire load can be reduced by improving the suspension system for heavy vehicle. In this way, the road friendliness will be improved while improving the ride comfort.

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Aiming at the coordination of road friendliness and ride comfort of vehicle suspension, the vehicle–road interaction of heavy vehicle was studied in the 1990s. They concluded that suspension parameters, driving speed and road grade are the main factors affecting dynamic tire load. Kyongsu and Hedrick of the United States found that the dynamic tire load is much larger than the static tire load. The deduction and test verified that the semi-active suspension controlled by bilinear disturbance observer can effectively reduce the dynamic tire load, thereby reducing the damage of heavy vehicle to the road. Sun of the University of Texas took the peak value of dynamic tire load as the optimization objective to design vehicle suspension. By changing the weighting coefficient, the active suspension system designed effectively improves the comprehensive performance of suspension. Xie et al. of the University of Macau proposed a semi-active suspension based on integrated fuzzy-wheelbase preview control, which improves the road friendliness of heavy vehicle while taking into account ride comfort.

The traditional passive suspension and semi-active suspension mainly adopt the optimization method of damping coefficient and spring stiffness to achieve performance coordination. The research shows that the heavy vehicle needs to reduce the spring stiffness and improve the damping coefficient to achieve it. However, a significant reduction in stiffness may lead to deterioration of the suspension working space. Obviously, heavy vehicles are limited by the inherent structure of “spring–damper” in the efficient coordination of road friendliness and ride comfort, so the further improvement of performance meets the theoretical bottleneck. Active suspension force generator can take both performance into account, but it has high energy consumption when used in heavy vehicle, which is not in line with the development direction of low-carbon vehicle.

In 2002, Professor Smith invented the two-terminal mass element “inerter,” which effectively introduced mass impedance into the vibration isolation system and laid the foundation theory of vehicle inerter–spring-damper (ISD) suspension. Then, the performance benefits in passive vehicle suspensions employing inerter were studied by Smith and Wang. Afterward, Hu et al. used inerter to improve multiple performance requirements especially suspension deflection performance. Shen et al. investigated an improved design of dynamic vibration absorber by using the inerter, and the ISD suspension can effectively improve the damping performance. At present, ISD suspension has been widely employed in vehicle engineering, civil engineering, and train suspension, which shows that ISD suspension system has good vibration isolation performance.

Thoresson et al. using a gradient-based approximation method optimized a vehicle’s suspension parameters for ride comfort and handling. The undesirable effects associated with noise in the gradient information are effectively reduced. Javanshir et al. optimized geometric parameters of suspension system using integrated anti-roll bar and coiling spring, and effects of optimization of suspension system during various parameters are compared. Silveira et al. compared symmetrical (linear) shock absorbers and asymmetrical (nonlinear) shock absorbers. The comparisons show that the nonlinear system has a smoother and more progressive performance, both for vertical and angular movements. In addition, Seifi et al. presented a new multi-objective optimization method to improve ride comfort, vehicle handling, and workspace using nonlinear asymmetrical dampers.

In general, the effect of reducing heavy vehicles on road surface damage can be achieved by adjusting or changing the suspension structure of heavy vehicles to reduce the dynamic tire load. However, traditional passive suspension and semi-active suspension mainly achieve road friendliness and ride comfort performance coordination by optimizing suspension damping coefficient and spring stiffness, which is limited by the inherent structure of traditional spring–damper structure, therefore the lifting effect is limited.

In addition, the research of vehicle ISD suspension is a multivariable and multi-objective problem. The design of component parameters should not only meet the requirements of ride comfort but also have better road friendliness. For heavy vehicle suspension system parameters, the current research mainly focuses on the parameter performance optimization based on vehicle ride comfort or road friendliness, while there are few studies on the combination of ride comfort and road friendliness. Moreover, the parameters of heavy vehicle suspension system are mostly compared and analyzed on the basis of traditional passive suspension, which takes less suspension parameters into account. There is also a lack of discussion on inerter component in the research reports of the simulation analysis and the influence of parameters changes on heavy vehicle suspension. Therefore, the study on the main parameters of heavy vehicle ISD suspension needs to be expanded, which requires a lot of model data simulation work in ISD suspension for heavy vehicle, evaluation of simulation results, and analysis of parameter selection.

In this paper, the inerter is creatively applied to the field of heavy vehicle, and the principle of ISD suspension with the inerter for heavy vehicle is analyzed on the basis of the dynamic inertia effect of inerter. The action mechanism of ISD suspension for heavy vehicle on road friendliness and ride comfort is also studied, so as to realize the coordinated optimization of road friendliness and ride comfort for heavy vehicles. In order to improve the comprehensive performance of an ISD suspension for heavy vehicle, the multi-objective genetic algorithm is
adopted as the optimization method of ISD suspension structural parameters based on establishing a half vehicle model. The perturbation effects of spring stiffness, damping coefficient, inertance and load on road friendliness, and ride comfort for heavy vehicle are studied, which provides a theoretical basis for further research and coordinated control of ISD suspension for heavy vehicle.

In order to improve the comprehensive performance of heavy vehicle ISD suspension, based on the establishment of half car model, multi-objective genetic algorithm is used to optimize the main parameters.

The organization of this paper is as follows. The half vehicle model of heavy vehicle ISD suspension is established in Establishment of ISD suspension model for heavy vehicle section. The parameters of ISD suspension for heavy vehicle are optimized in Optimization of ISD suspension parameters for heavy vehicle section. And System simulation analysis section carries out the system simulation and obtains the simulation result of heavy vehicle. Then, the influence of parameters perturbation on ISD suspension performance of heavy vehicle is analyzed in Performance analysis of ISD suspension with parameters perturbation section. Finally, conclusions are drawn in Conclusion section.

Establishment of ISD suspension model for heavy vehicle

According to Shen et al., in the three-element structure, the inerter is connected in series with the damper first, and then in parallel with the spring. On this basis, considering the body structural characteristics of heavy vehicle, the half vehicle structure model of three-axle heavy vehicle ISD suspension is established, as shown in Figure 1.

It should be pointed out that the vehicle is a complex nonlinear mechanical network system in motion. In order to facilitate the study of ISD suspension for heavy vehicle, the following assumptions are made for the established model:

The stiffness of the body and frame is much higher than that of the suspension, that is to say, assuming it is rigid body. Vehicles are symmetrical in left and right, and the road surface roughness conditions on both sides are the same. Besides, vertical and pitching motions in the plane are considered only in the analysis. The tire always keeps contact with the road surface without jumping when the vehicle is moving at a constant speed on the road. Assuming that the road surface roughness corresponding to the tire on the same side of the axle is different and there is a response hysteresis caused by the wheelbase. The vibration caused by the engine, transmission system, and so on is neglected.

Based on the above assumptions, a six-free-degree vibration differential equation is established according to Newton’s second law.

The vertical vibration equation at the centroid of the car body is

\[ M_b \ddot{z}_b = F_c + F_f \]  

The pitching motion equation of the car body around the centroid is

\[ I_b \ddot{\theta}_b = eF_c - aF_f \]  

The vertical vibration equation of the front axle is

\[ M_f \ddot{z}_1 = k_f(z_f - z_1) + c_f(\dot{z}_f - \dot{z}_1) - F_f \]  

Figure 1. Half vehicle model of heavy vehicle ISD suspension.
The vertical vibration equation of balanced suspension is

$$\Sigma M \ddot{z}_c = k_{im}(z_m - z_c) + c_{im}(\dot{z}_m - \dot{z}_c) + k_{ir}(z_r - z_c) + c_{ir}(\dot{z}_r - \dot{z}_c) - (k_{ir} - k_{im})\theta_c d - (c_{ir} - c_{im})\dot{\theta}_c d - F_c$$  (4)

The pitching motion equation of the balanced suspension is

$$I_c \ddot{\theta}_c + (M_m + M_r)\ddot{\theta}_c = (k_{ir}z_r - k_{im}z_m)d + (c_{ir}\dot{z}_r - c_{im}\dot{z}_m)d - (k_{im} + k_{ir})\theta_c d^2 - (c_{im} - c_{ir})\dot{\theta}_c d^2 - (k_{ir} - k_{im})z_c d - (c_{ir} - c_{im})\dot{z}_c d$$  (5)

The suspension force is

$$\begin{cases} F_f = k_f(z_1 - z_2) + u_f \\ F_c = k_r(z_c - z_5) + u_r \end{cases}$$  (6)

Among equations (4) and (6)

$$\Sigma M = M_m + M_c + M_r$$  (7)

$$\begin{cases} u_f = b_f(\ddot{z}_1 - \ddot{z}_{11}) = c_f(\dot{z}_{11} - \dot{z}_2) \\ u_r = b_r(\ddot{z}_c - \ddot{z}_{22}) = c_r(\dot{z}_{22} - \dot{z}_5) \end{cases}$$  (8)

When the pitch angle $\theta_b$ is small, the following approximation of equation (8) can be obtained

$$\begin{cases} z_2 = z_b - a\theta_b \\ z_5 = z_b + e\theta_b \end{cases}$$  (9)

In the above formulas, $M_b$ is the body mass, $z_b$ is the vertical displacement of the body mass center, $F_f$ and $F_c$ are the force of front suspension and balance suspension, $\dot{\theta}_b$ is the centroid angular acceleration and $\theta_b$ is the balance suspension pitch angle, $M_f$ is the unsprung mass of the front suspension, $M_m$, $M_r$ are the unsprung mass of the balanced suspension respectively, $M_r$ is the mass of the balanced rod, $I_m$, $I_c$ are the rotational inertia coefficients of the body and the balanced suspension respectively, $k_f$, $k_r$, $k_{ir}$, $k_{im}$, $k_{ir}$ are the spring stiffness coefficients of the front suspension, spring stiffness of balanced suspension, tire stiffness coefficients of front, middle, and rear wheels, respectively, $c_f$, $c_r$, $c_{ir}$, $c_{im}$, $c_{ir}$ are the damping coefficients of front suspension and balanced suspension, tire damping coefficients of front wheel, middle wheel, and rear wheel, respectively, $z_j$, $z_m$, $z_r$, $z_1$, $z_r$, $z_{11}$, $z_{22}$ are the displacement inputs of the front wheel, the middle wheel, and the rear wheel, the vertical displacements of the front suspension and the balanced suspension, the inerter displacements of the front suspension and the balance suspension, respectively, $b_f$, $b_r$ are the inerter of the front suspension and the balanced suspension, respectively. $a$, $e$, $d$ are the distance between the front axle and the centroid, the distance between the center of balance rod and the centroid, and the length of the balance rod, respectively.

**Optimization of ISD suspension parameters for heavy vehicle**

In order to determine the parameters of ISD suspension for heavy vehicle, the traditional passive suspension with mature parameters is taken as the contrast object. On the basis of DFL1250A9, a three-axle heavy vehicle, the related parameters of heavy vehicle are shown in Table 1.

Considering the multi-parameters and multi-objective of suspension system, the multi-objective genetic algorithm is adopted as the optimization method of ISD suspension structural parameters for heavy vehicle, and the system parameters are determined by numerical simulation.

There are six key parameters to be optimized

$$X = [k_f \ c_f \ b_f \ k_r \ c_r \ b_r]$$  (10)
Then, three vehicle ride comfort indexes and one road friendliness index are selected as optimization objectives. As the above four performance indicators have different units and orders of magnitude, it is necessary to establish a unified objective function. The four performance indexes of ISD suspension for heavy vehicle are divided by traditional passive suspension indexes, and their quotient sum is taken as the objective unified function. Therefore, the optimization of evaluation indexes of ride comfort and road friendliness is transformed into the minimum value problem of unified objective function. The expression of the unified objective function is

\[
\min T = w_1 \frac{BA(X)}{BA_{\text{pas}}} + w_2 \frac{SWS(X)}{SWS_{\text{pas}}} + w_3 \frac{DTL(X)}{DTL_{\text{pas}}} + w_4 \frac{J(X)}{J_{\text{pas}}}
\]  

(11)

In the equation, \( T \) is the fitness function (unified objective function), \( BA(X) \), \( SWS(X) \), are the root mean square (RMS) values of body acceleration, suspension working space, and dynamic tire load of ISD suspension for heavy vehicle, respectively. \( J(X) \) is the road damage coefficient of ISD suspension for heavy vehicle. \( BA_{\text{pas}} \), \( SWS_{\text{pas}} \), \( DTL_{\text{pas}} \) are the RMS values of body acceleration, suspension working space, and dynamic tire load of traditional passive suspension, respectively. \( J_{\text{pas}} \) is road damage coefficient of traditional passive suspension. Among them, the \( BA(X) \) and \( BA_{\text{pas}} \) include four specific values respectively: body acceleration, front suspension acceleration, balance suspension acceleration, and centroid angular acceleration. The \( SWS(X) \) and \( SWS_{\text{pas}} \) include two specific values respectively: working space of front suspension and working space of balanced suspension. The \( DTL(X) \) and \( DTL_{\text{pas}} \) include three specific values respectively: dynamic tire load of front wheel, dynamic tire load of middle wheel, and dynamic tire load of rear wheel. \( w_1, w_2, w_3, w_4 \) are the weighted coefficients of different performance indicators and their values are equal.

According to Cole and Cebon, the road damage coefficient \( J \) is calculated by the 95th percentile aggregate fourth power force, and its formula is

\[
J = 1 + \frac{1.65\sigma_{A^4}}{m_{A^4}}
\]

(12)

In the formula, \( A^4 \) is the power sum of dynamic load and static load, \( \sigma_{A^4} \) is the standard deviation of \( A^4 \), and \( m_{A^4} \) is the mean value of \( A^4 \).

The constraints of the unified objective function are

\[
UB < X_i < LB \quad i = 1, 2, 3 \ldots, 6
\]

(13)

\[
\begin{align*}
&UB < X_i < LB \quad i = 1, 2, 3 \ldots, 6 \\
&\begin{aligned}
&BA(X) < BA_{\text{pas}} \\
&SWS(X) < SWS_{\text{pas}} \\
&DTL(X) < DTL_{\text{pas}} \\
&J(X) < J_{\text{pas}}
\end{aligned}
\end{align*}
\]

(14)

Table 1. Relevant parameters of inerter–spring–damper (ISD) suspension for three-axle heavy vehicle.

| Name                              | Value   |
|-----------------------------------|---------|
| Vehicle mass \( M_b/\text{kg} \)  | 11,523  |
| Unsprung mass of front suspension \( M_{f}/\text{kg} \) | 412     |
| Unsprung mass of balanced suspension \( M_{m} \), \( M_{r}/\text{kg} \) | 676     |
| Mass of balanced rod \( M_c/\text{kg} \) | 177     |
| Rotational inertia of body \( l_{b}/(\text{kg} \cdot \text{m}^2) \) | 55,502  |
| Rotational inertia of balanced rod \( l_{b}/(\text{kg} \cdot \text{m}^2) \) | 351     |
| Tire stiffness of front wheels \( k_{pf}/(\text{N}/\text{m}) \) | 11,00,000 |
| Tire stiffness of middle wheels \( k_{pm}/(\text{N}/\text{m}) \) | 22,00,000 |
| Tire stiffness of rear wheels \( k_{pr}/(\text{N}/\text{m}) \) | 22,00,000 |
| Tire damping coefficient of front wheel \( c_{pf}/(\text{N} \cdot \text{s}/\text{m}) \) | 3500    |
| Tire damping coefficient of middle wheel \( c_{pm}/(\text{N} \cdot \text{s}/\text{m}) \) | 7000    |
| Tire damping coefficient of rear wheel \( c_{pr}/(\text{N} \cdot \text{s}/\text{m}) \) | 7000    |
| Distance between the front axle and the centroid \( a/\text{m} \) | 3.64    |
| Distance between the center of balance rod and the centroid \( e/\text{m} \) | 2.71    |
| Length of the balance rod \( d/\text{m} \) | 1.3     |
Among them, $X$ is the parameters collection to be optimized. $UB$ and $LB$ are the upper and lower limits of parameters to be optimized, respectively.

In addition, the mathematical characteristics of the optimization algorithm in this paper are as follows: the optimal individual coefficient is 0.3, the population size is 40, the maximum evolution algebra is 50, the stop algebra is 200, and the fitness function deviation is 0.01.

It is assumed that the vehicle runs at a uniform speed of 20 m/s on the B grade road surface with a driving time of 20 s and a sampling time interval of 0.005 s. The Gauss white noise with the intensity of 20 dB and the mean value of zero is taken as the road input. The time domain diagram of random road input is shown in Figure 2.

In addition, the time when the front wheel, middle wheel, and rear wheel of heavy vehicle receive the road input is different in the optimization simulation process. Therefore, the value of wheelbase divided by speed is considered as delay in the simulation.

The optimization results of ISD suspension parameters are shown in Table 2.

### Table 2. Parameter optimization results.

| Name                                      | Value   |
|-------------------------------------------|---------|
| Spring stiffness of front suspension $k_f$ | 2,92,000|
| Spring stiffness of balanced suspension $k_r$ | 15,43,000|
| Damping coefficient of front suspension $c_f$ | 40,420  |
| Damping coefficient of balanced suspension $c_r$ | 29,520  |
| Inertance of front suspension $b_f$        | 1500    |
| Inertance of balanced suspension $b_r$      | 2000    |

Among them, $X$ is the parameters collection to be optimized. $UB$ and $LB$ are the upper and lower limits of parameters to be optimized, respectively.

In addition, the mathematical characteristics of the optimization algorithm in this paper are as follows: the optimal individual coefficient is 0.3, the population size is 40, the maximum evolution algebra is 50, the stop algebra is 200, and the fitness function deviation is 0.01.

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The optimization results of ISD suspension parameters are shown in Table 2.

### System simulation analysis

It is assumed that the vehicle runs at a uniform speed of 20 m/s on B grade road. Compared with traditional heavy vehicle suspension, the influence of ISD suspension on ride comfort and road friendliness is analyzed. Under random input conditions, the results of time domain simulation and frequency domain simulation are shown in Figure 3, Tables 3 and 4.

As can be seen from Figure 2, when heavy vehicle travel at a uniform speed of 20 m/s on B grade road surface, the centroid acceleration, suspension working space and dynamic tire load of ISD suspension are reduced to varying degrees than those of traditional passive suspension in time domain. In frequency domain, the centroid acceleration, suspension working space, and dynamic tire load of ISD suspension are significantly lower than those of traditional suspension in the range of 0–5 Hz, while they are almost the same as those of traditional suspension in the range of 5–15 Hz.

Combining Figure 3 and Table 4, we can see that the RMS values of centroid acceleration and pitch angular acceleration of ISD suspension, compared with traditional passive suspension, are improved by 24.90% and
23.54% respectively in time domain analysis, and the working spaces of front suspension and balanced suspension are improved by 21.82% and 8.11% respectively, and the RMS values of dynamic tire load of front, middle, and rear wheel decrease by 7.80%, 3.50%, and 3.87%, respectively. In frequency domain analysis, the ISD suspension of heavy vehicle effectively reduces the peak resonance of suspension in 0–5 Hz band compared with traditional
Table 3. The 95th percentile aggregate contrast.

| Road damage coefficient | Traditional passive suspension | ISD suspension | Improvement (%) |
|-------------------------|--------------------------------|----------------|-----------------|
| Front wheel             | $1.9074 \times 10^{18}$       | $1.8099 \times 10^{18}$ | 5.11            |
| Middle wheel            | $8.2489 \times 10^{18}$       | $7.9479 \times 10^{18}$ | 3.65            |
| Rear wheel              | $9.1089 \times 10^{18}$       | $8.7177 \times 10^{18}$ | 4.29            |
| Vehicle integration J   | $7.3246 \times 10^{18}$       | $7.0282 \times 10^{18}$ | 4.05            |

ISD: inerter–spring–damper.
passive suspension. In addition, the low-frequency resonance frequency decreases and the vibration isolation performance improves significantly, which effectively improves the ride comfort of heavy vehicle.

Table 3 shows that the 95th percentile aggregate values of front, middle, and rear wheels of ISD suspension for heavy vehicle decrease by 5.11%, 3.65%, and 4.29% respectively, and the comprehensive road damage coefficient decreases by 4.05% compared with traditional passive suspension, which effectively improves the road friendliness of heavy vehicle.

Compared with traditional passive suspension, the low-frequency resonance frequency and peak value of resonance peak of ISD suspension of heavy vehicle are reduced, and the vertical motion characteristics of the system are improved. Therefore, ISD suspension of heavy vehicle can effectively improve ride comfort and road friendliness of heavy vehicle.

Performance analysis of ISD suspension with parameters perturbation

For heavy vehicle, the performance of suspension not only affects the ride comfort but also affects the road friendliness of vehicle. Therefore, the influence of spring stiffness, damping coefficient, inertance, and load on suspension performance is analyzed in this paper.

The ranges of spring stiffness, damping coefficient, inertance, and sprung mass of ISD suspension are shown in Table 5.

| Table 5. Range of inerter–spring–damper (ISD) suspension parameters. |
|-----------------------------|-----------------------------|
| Parameter                   | Range of values             |
| Spring stiffness of front suspension/(N/m) | 1,46,000–4,38,000 |
| Spring stiffness of balanced suspension/(N/m) | 7,71,500–23,14,500 |
| Damping coefficient of front suspension/(N s/m) | 20,210–60,630 |
| Damping coefficient of balanced suspension/(N s/m) | 14,760–44,280 |
| Inertance of front suspension/(kg) | 750–2250 |
| Inertance of balanced suspension/(kg) | 1000–3000 |
| Sprung mass | No load–full load |

Table 4. Root mean square (RMS) comparison of suspension performance.

| Suspension performance index | RMS of traditional passive suspension | RMS of ISD Suspension | Improvement (%) |
|------------------------------|----------------------------------------|------------------------|-----------------|
| Centroid acceleration/(m/s²) | 1.3013                                  | 1.0448                 | 24.90           |
| Pitch angular acceleration/(rad/s²) | 0.3573                                             | 0.2732                 | 23.54           |
| Working space of front suspension/m | 0.0055                                               | 0.0043                 | 21.82           |
| Working space of balanced suspension/m | 0.0037                                               | 0.0034                 | 8.11            |
| Dynamic tire load of front wheel/N | 3435                                                  | 3167                   | 7.80            |
| Dynamic tire load of middle wheel/N | 8678                                                  | 8374                   | 3.50            |
| Dynamic tire load of rear wheel/N | 9504                                                  | 9136                   | 3.87            |

ISD: inerter–spring–damper; RMS: root mean square.
with the increase of spring stiffness of balanced suspension and front suspension, respectively. It is not difficult to find that the influence of front suspension spring stiffness on ride comfort and road friendliness is higher than that of balanced suspension spring stiffness. The reason is that the middle and rear axles of heavy vehicle need to bear most of the body weight. The value of balanced suspension spring stiffness is much larger than that of front suspension, so the influence of RMS and road damage coefficient is different.

**Figure 4.** Influence of spring stiffness combination on suspension performance.

**Figure 5.** Influence of damping coefficient combination on suspension performance.
Influence of damping coefficient combination on suspension performance

According to the values of the damping coefficients of the front suspension and the balanced suspension in Table 4, the effects of the variation of the damping coefficients on the ride comfort and road friendliness of heavy vehicle are analyzed in the time domain, as shown in Figure 5.

From Figure 5, it can be seen that the RMS values of centroid acceleration, pitch angular acceleration, working space of balanced suspension, dynamic tire load of middle wheel, dynamic tire load of rear wheel, and road damage coefficient decrease with the increase of front suspension damping coefficient, while the RMS values of...
front suspension working space and front wheel dynamic tire load increase with the increase of spring stiffness of balanced suspension. Comparing and analyzing the influence of front suspension damping coefficient and balanced suspension damping coefficient on ride comfort and road friendliness, the influence of front suspension damping coefficient on ride comfort and road friendliness is greater under the same change rate. In the parameters design, in order to make the vehicle have good ride comfort and road friendliness, the larger front suspension damping coefficient and the smaller balanced suspension damping coefficient should be selected appropriately.

Figure 6. Influence of inertance combination on suspension performance.

Figure 7. Influence of load change on suspension performance.
Influence of inertance combination on suspension performance

According to the inertance of front suspension and balanced suspension in Table 4, the effects of inertance on ride comfort and road friendliness of heavy vehicle are analyzed in time domain, as shown in Figure 6.

Figure 6. Influence of load change on suspension performance.
From Figure 6, similar to the effect of damping coefficient, it can be seen that the influence of centroid acceleration, pitch angular acceleration, balanced suspension working space, middle wheel dynamic tire load, rear wheel dynamic tire load, and road damage coefficient decrease with the increase of front suspension iner-tance, and when the front suspension iner-tance is larger than a certain value, the change of the above indexes tends to be flat. The RMS values of the front suspension working space and the dynamic tire load of front wheel decrease with the increase of the iner-tance of the balanced suspension. The application of iner-tor in suspension system, combined with other components, effectively improves vehicle ride comfort and road friendliness. Comparatively speaking, the change of the front suspension iner-tance has a greater impact on ride comfort and road friendliness than the balanced suspension iner-tance within the allowable range of suspension working space. The larger values of the front suspension iner-tance and the balanced suspension iner-tance should be selected appropriately when designing suspension parameters.

**Influence of load change on suspension performance**

The function of multi-axle heavy vehicle is mainly to carry goods. The sprung mass of the vehicle varies greatly under the condition of no load and full load. Therefore, when the sprung mass is in the state of no load, half load, and full load, the regularity of ride comfort and road friendliness for heavy vehicle is analyzed compared with the traditional passive suspension, as shown in Figure 7.

As shown in Figure 7, the RMS values of centroid acceleration, pitch angular acceleration, balanced suspension working space, front, middle, and rear dynamic tire load decrease with the increase of load, while the RMS of front suspension working space and road damage coefficient increase with the increase of load. Compared with the traditional passive suspension, the evaluation indexes of ISD suspension have improved its ride comfort and road friendliness under no load, half load, and full load conditions. When the load of heavy vehicle is large, the ride comfort is better than that of no load vehicle, but the road damage is aggravated. Thus it can be seen that overloading of heavy vehicle will cause serious damage to the road and reduce the service life of the road, so overloading behavior should be resolutely eliminated.

**Conclusion**

This paper applies ISD suspension with iner-tor to heavy vehicle suspension and establishes a half vehicle model of heavy vehicle ISD suspension. Taking both ride comfort and road friendliness as optimization objectives, multi-objective genetic algorithm is used to optimize the components parameters of ISD suspension for heavy vehicle. By analyzing the dynamic inertia mechanism and vehicle performance coordination of the optimized ISD suspension, the dynamic inertia mechanism on road friendliness and ride comfort is summarized. Compared with the traditional passive suspension, the ride comfort and road friendliness of ISD suspension for heavy vehicle have been significantly improved under the condition of 20 m/s driving speed on B grade road. The results show that, compared with traditional passive suspension, the RMS values of centroid acceleration and pitch angular acceleration of ISD suspension are improved by 24.90% and 23.54%, respectively, the working spaces of front suspension and balanced suspension are improved by 21.82% and 8.11% respectively, the comprehensive road damage coefficient is reduced by 4.05%, and the ride comfort and road friendliness of heavy vehicle are improved significantly. The influence of parameters perturbation on ISD suspension for heavy vehicle is analyzed. The different effects of spring stiffness, damping coefficient, iner-tance, and load on ride comfort and road friendliness of the front, middle, and rear wheels of ISD suspension for heavy vehicle are summarized. The above results provide a theoretical basis for further research and improvement of parameters setting of ISD suspension for heavy vehicle.

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