Research Article

Optimal Design and Dynamic Performance Analysis Based on the Asymmetric-Damping Vehicle ISD Suspension

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This paper concerns the optimal problem of the vehicle ISD (inerter-spring-damper) suspension based on the asymmetric-damping effect. In order to explore the benefits of the asymmetric damping, a quarter car model of the four-element ISD suspension is built by considering the symmetric and asymmetric reciprocating damping factors. The parameters of the proposed vehicle ISD suspension with symmetric-damping and asymmetric-damping features are optimized by means of the genetic algorithm in single-objective scenario and multiobjective scenario, respectively. The dynamic performances are analyzed through simulations in time and frequency domains, and the impacts of the compression and tensile damping on the body acceleration, the suspension working space, and the dynamic tire load are discussed. Results indicate that, compared with the conventional passive suspension, the proposed ISD suspensions manifest excellent vibration isolation performance, and the asymmetric reciprocating damping ISD suspension even showcases extra improving space of the dynamic performances except for the dynamic tire load in the impulse input condition. It seems that the dynamic performance of the vehicle ISD suspension will be much superior when considering the asymmetric reciprocating damping factors.

1. Introduction

As one of the linchpins of the vehicle chassis system, the suspension system can exert significant influences on ride comfort and vehicle handling performance. Until now, semiactive suspension [1–5] and active suspension [6–11] systems are widely known for their versatility. For the semiactive suspension system, it can adjust the damping force by changing the command current, which generates advantages such as lower consumption, rapid response, and simplicity. Apart from all the merits the semiactive suspension system can possess, the active suspension system can better coordinate the contradiction between the ride comfort and the vehicle handling stability. However, the complicated structures and high cost requirements limit its widespread application.

Because of the aspiration for both the excellent dynamic performances and structural simplicity, a novel device, inerter, was firstly introduced in [12]. Generally, the inerter element has been successfully applied in interdisciplinary fields, such as automotive engineering [13–18], civil engineering [19, 20], and aerospace engineering [21]. In [22], the influences of the ball-screw inerter nonlinearities triggered by frictional ball-screw assembly and elastic effect of screw on the suspension performance were studied. The nonlinearities of the fluid inerter were considered in the modeling of the full car model, and the parameters were optimized in [23]. More complex suspension structures, such as HIISDS (hydraulically interconnected inerter-spring-damper suspension) [24] and HEI (hydraulic electric inerter) suspension, comprised of a hydraulic piston inerter and a linear motor, which considers a bicubic impedance function in the optimal design of a vehicle suspension system employing both mechanical and electrical elements [25], were used to counterbalance the conflict between vehicle ride comfort and handling stability. In [26], the trade-offs in designing the ball-screw inerter and
the permanent magnet electric machinery were further studied in detail. Nevertheless, research studies above are all concentrated on the structural design and nonlinearity analysis without considering the asymmetry of reciprocating damping. Given the assumption that asymmetrical damping may interact with inerter and spring, thus bringing up unprecedented changes to vehicles’ dynamic performances, which is not discussed yet, it seems necessary to unveil the potential correlations. This paper aims at analyzing the dynamic performances of both symmetric and asymmetric reciprocating damping ISD suspension and exploring the impacts of the asymmetric damping on the dynamic performances. The paper is arranged as follows.

In Section 2, a quarter car model of the four-element ISD suspension is built when taking asymmetric damping into consideration. In Section 3, single-objective and multi-objective optimizations are finished in the condition of random road input by means of the genetic algorithm. In Section 4, simulations are carried out to analyze the dynamic performances of the proposed vehicle ISD suspensions. The impacts of the asymmetric damping on the body acceleration, the suspension working space, and the dynamic tire load are discussed in Section 5. Finally, some conclusions are drawn in Section 6.

2. Establishment of Suspension Model

2.1. The Conventional Passive Suspension Model. A quarter car model can effectively show the relative motion between the sprung mass and the unsprung mass, with the spring and damper in parallel connection; the structure of the conventional passive suspension model is shown in Figure 1.

In Figure 1, \( m_s \) is the sprung mass, \( m_u \) is the unsprung mass, \( k \) is the stiffness of spring, \( c \) is the damper, \( k_t \) is the stiffness of tire spring, and \( z_s, z_u, \) and \( z_r \) are the vertical displacements of the sprung mass, unsprung mass, and road input, respectively.

2.2. The Vehicle ISD Suspension Model. There are many structures of the vehicle suspension system employing the inerter, spring, and damper elements. In this paper, a four-element vehicle ISD suspension is considered, with the integration of the deputy spring and inerter in series connection and the main spring and damper in parallel connection; the dynamic model structure of the vehicle ISD suspension is shown in Figure 2.

In Figure 2, \( k_1 \) is the stiffness of main spring, \( k_2 \) is the stiffness of deputy spring, \( b \) is the inerter with the unit kg, \( c_1 \) is the coefficient of tensile damping, \( c_2 \) is the coefficient of compression damping, \( c_0 \) is the coefficient of symmetric reciprocating damping, \( z_0 \) is the vertical displacement of the inerter, and \( F \) is the force between inerter and deputy spring. In Figure 2, when \( c_0 = c_1 = c_2 \), it represents the proposed vehicle ISD suspension with symmetric reciprocating damping. The corresponding equations of the suspension force are shown in the following equation:

\[
\begin{align*}
    m_s \ddot{z}_s + k_1 (z_s - z_u) + c_0 (\dot{z}_s - \dot{z}_u) - F &= 0, \\
    m_u \ddot{z}_u + k_1 (z_u - z_s) + k_2 (z_u - z_r) + c_0 (\dot{z}_u - \dot{z}_r) - F &= 0, \\
    F &= k_2 (z_u - z_b) = b(\ddot{z}_b - \ddot{z}_r),
\end{align*}
\]

whereas when \( c_1 \neq c_2 \), it represents the proposed suspension with asymmetric reciprocating damping. The corresponding equations of the suspension force are shown in the following equation:
3. Optimization of the Suspension Parameters

3.1. Single-Objective Optimization Based on GA. In order to explore the benefits of both the symmetric- and asymmetric-damping vehicle ISD suspension, the root mean square (RMS) values of the vehicle body acceleration, suspension working space, and dynamic tire load under the random road input are taken into consideration in the optimization process. The vehicle ISD suspension systems are optimally designed by designating the performance indices as single-objective and multiobjective separately. There are certain constraints in the process of optimization, which are when one of the three performance indices is set as the objective, the other two cannot be beyond those of the conventional passive suspension, in which case the punishment number will be set as 0 or that will be set as 100. The above algorithms can be shown in equations (3)–(6) in the single-objective optimization:

\[
F = k_2(z_u - z_b) = b(\ddot{z}_b - \ddot{z}_u),
\]
\[
\begin{align*}
if & \quad (\ddot{z}_s - \dot{z}_w) > 0: \quad \begin{cases} m_z\ddot{z}_s + k_1(z_s - z_u) + c_1(\dot{z}_s - \dot{z}_w) - F = 0, \\
m_w\ddot{z}_w + k_1(z_u - z_s) + k_2(\dot{z}_u - \dot{z}_w) + F = 0, \end{cases} \quad (2) \\
if & \quad (\ddot{z}_s - \dot{z}_w) < 0: \quad \begin{cases} m_z\ddot{z}_s + k_1(z_s - z_u) + c_2(\dot{z}_s - \dot{z}_w) + F = 0, \\
m_w\ddot{z}_w + k_1(z_u - z_s) + k_2(\dot{z}_u - \dot{z}_w) - F = 0, \end{cases}
\end{align*}
\]

\[ F = \frac{BA}{BA_{BD}} + \text{punishment}, \]
\[ F = \frac{SWS}{SWS_{BD}} + \text{punishment}, \]
\[ F = \frac{DTL}{DTL_{BD}} + \text{punishment}, \]
\[ \begin{align*}
lb_1 & \leq k_1 \leq ub_1, \\
lb_2 & \leq k_2 \leq ub_2, \\
lb_3 & \leq b \leq ub_3, \\
lb_4 & \leq c_0 \leq ub_4, \\
lb_5 & \leq c_1 \leq ub_5, \\
lb_6 & \leq c_2 \leq ub_6. 
\end{align*} \]

For the record, the RMS values of the vehicle body acceleration, the suspension working space, and the dynamic tire load of conventional passive suspension under random road input, shown in Figure 3, are 1.1948 m·s⁻², 0.0119 m, and 839.2 N, respectively, and the random road input model can be shown in the following equation:

\[ \ddot{z}_r(t) = -2\pi f_0 z_r(t) + 2\pi \sqrt{G_0} \nu w(t), \]

where BA_{BD}, SWS_{BD}, and DTL_{BD} are the RMS values of the vehicle body acceleration, the suspension working space, and the dynamic tire load of the conventional passive suspension, respectively. BA, SWS, and DTL are the RMS values of the vehicle body acceleration, the suspension working space, and the dynamic tire load of the proposed ISD suspension, respectively. \( lb \) is the lower boundary of the parameter, \( ub \) is the upper boundary of the parameter, \( z_r(t) \) is the displacement, \( G_0 \) is the roughness coefficient, \( f_0 \) is the cut-off frequency, \( v \) is the speed, and \( w(t) \) is the Gaussian white noise with mean value as 0. In this study, \( G_0 \) is \( 5 \times 10^{-3} \text{m}^3 \cdot \text{cycle}^{-1} \) and \( f_0 \) is 0.1 Hz.

For the symmetric-damping ISD suspension, the stiffness of main spring \( k_1 \), the stiffness of deputy spring \( k_2 \), the inertance \( b \), and the symmetric reciprocating damping \( c_0 \) are set as the variables to be optimized; however, for the asymmetric-damping ISD suspension, the stiffness of main spring \( k_1 \), the stiffness of deputy spring \( k_2 \), the inertance \( b \), the tensile damping \( c_1 \), and the compression damping \( c_2 \) are set as the variables to be optimized. The model parameters are selected from a mature passenger car and listed in Tables 1 and 2:

For the sake of gaining all the satisfied parameters, the genetic algorithm (GA) is used to optimize the relative parameters of the proposed vehicle ISD suspensions. The population is set as 100, iteration value is set as 20, and other parameters remain unchanged. The optimal parameters of suspensions are listed in Tables 3–5.

3.2. Multiobjective Optimization Based on GA. With the intention of exploring overall dynamic performances, ride comfort and handling stability, of the proposed vehicle ISD suspension with symmetric and asymmetric damping, the comprehensive objective with the punishment number is established, described as equation (8), and the punishment rule is decided that, among the RMS of the body acceleration, the suspension working space, and the dynamic tire load, as long as one of them is inferior to the corresponding value of conventional passive suspension, the value of the punishment number is set as 100 or it is set as 0. The other relative parameters are consistent with those in Table 1. The optimization model can be described as the following equation:

\[ F = \frac{BA}{BA_{BD}} + \frac{SWS}{SWS_{BD}} + \frac{DTL}{DTL_{BD}} + \text{punishment}, \]

\[ \text{min } F = (\text{BA, SWS, DTL}), \]
\[
\begin{align*}
\begin{cases}
lb_1 & \leq k_1 \leq ub_1, \\
lb_2 & \leq k_2 \leq ub_2, \\
lb_3 & \leq b \leq ub_3, \\
lb_4 & \leq c_0 \leq ub_4, \\
lb_5 & \leq c_1 \leq ub_5, \\
lb_6 & \leq c_2 \leq ub_6.
\end{cases}
\end{align*}
\]

(10)

\(lb_i\) and \(ub_i\) are the same as those of equation (6). In order to search the global optimal solution, the toolbox of GA is used and its parameters are the same as those in Table 2, and results are listed in Tables 6 and 7.

Furthermore, the dynamic performance indices are all disproportionately decreased in comparison with the conventional suspension. For the ISD suspension with symmetric damping, the RMS values of the body acceleration, the suspension working space, and the dynamic tire load are decreased by 4.0%, 28.5%, and 13.3%. In the aspect of those three indicators of the ISD suspension with asymmetric damping, they are decreased by 4.2%, 31.1%, and 13.7%. The conclusion can be made that, despite the absence of drastic boosting of the sole indicator, which can be realized in single-objective optimization, all the dynamic indicators can be promoted simultaneously, meaning overall dynamic performances can still expect noteworthy promotions.

4. Performance Analysis

4.1. Random Road Input. In order to show dynamic performances of the optimized ISD suspension under random road input, this paper reuses the concerning single-optimized parameters to realize simulations in time domain; results are shown in Figures 4(a), 5(a), and 6(a).

![Figure 3: Random road input at the speed of 20 m/s in time domain.](image)

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### Table 1: Parameters of the suspension model.

| Parameter name                        | Value   |
|---------------------------------------|---------|
| Sprung mass \( m_s \) (kg)            | 320     |
| Unsprung mass \( m_u \) (kg)          | 45      |
| Stiffness of conventional spring \( k \) (N·m\(^{-1}\)) | 22000   |
| Stiffness of tire \( k_t \) (N·m\(^{-1}\)) | 200000  |
| Coefficient of conventional damper \( c_0 \) (N·s·m\(^{-1}\)) | 1000    |
| Speed \( v \) (m·s\(^{-1}\))         | 20      |
| Cut-off frequency \( f_0 \) (Hz)      | 0.1     |
| Roughness coefficient \( G_0 \) (m\(^3\)·cycle\(^{-1}\)) | \(5 \times 10^{-6}\) |

### Table 2: Parameters of GA toolbox.

| Optimized parameters (symmetric damping) | lb\(_i\) | ub\(_i\) |
|-----------------------------------------|---------|---------|
| Stiffness of main spring \( k_1 \) (N·m\(^{-1}\)) | 0       | 30000   |
| Stiffness of deputy spring \( k_2 \) (N·m\(^{-1}\)) | 0       | 20000   |
| Coefficient of symmetric damping \( c_0 \) (N·s·m\(^{-1}\)) | 0       | 4000    |
| Inertance \( b \) (kg)                  | 0       | 8000    |

| Optimized parameters (asymmetric damping) | lb\(_i\) | ub\(_i\) |
|------------------------------------------|---------|---------|
| Stiffness of main spring \( k_1 \) (N·m\(^{-1}\)) | 0       | 30000   |
| Stiffness of deputy spring \( k_2 \) (N·m\(^{-1}\)) | 0       | 20000   |
| Coefficient of tensile damping \( c_1 \) (N·s·m\(^{-1}\)) | 0       | 4000    |
| Coefficient of compression damping \( c_2 \) (N·s·m\(^{-1}\)) | 0       | 4000    |
| Inertance \( b \) (kg)                  | 0       | 8000    |
It is noted that, compared with the conventional passive suspension, the ISD suspension, regardless of the presence of asymmetric reciprocating damping, shows notable improvements in terms of the RMS of the body acceleration, the suspension working space, and the dynamic tire load. For the proposed ISD suspension involving symmetric damping, the three dynamic indicators discussed above have decreased by 21.8%, 31.1%, and 14.8% in sequence. For the proposed ISD suspension involving asymmetric damping, those indicators have decreased by 23.2%, 31.9%, and 16.0% in sequence, implying that when the asymmetry of the damper is involved, the dynamic performances of the ISD suspension could get boosted to a greater level. Furthermore, the tensile damping is greater than the compression damping when the RMS of the body acceleration is optimal, the tensile damping is less than the compression damping when the RMS of the suspension working space is optimal, and the tensile damping is less than the compression damping when the RMS of the dynamic tire load is optimal. So, it can be inferred that the tensile damping may have more impacts on ride comfort, whereas the compression damping may have more impacts on vehicle handling stability.

Similarly, simulations in frequency domain are also completed with frequency varying from 0 Hz to 15 Hz based on the same parameters as in the time domain. The

### Table 3: Optimal parameters with body acceleration as the single objective.

| Optimized parameters (symmetric damping) | Value | Optimized objective |
|-----------------------------------------|-------|---------------------|
| Stiffness of main spring $k_1$ (N-m$^{-1}$) | 7465 | 0.9334 m-s$^{-2}$ |
| Stiffness of deputy spring $k_2$ (N-m$^{-1}$) | 1184 |
| Coefficient of symmetric damping $c_0$ (N-s-m$^{-1}$) | 1056 |
| Inertance $b$ (kg) | 7797 |

| Optimized parameters (asymmetric damping) | Value | Optimized objective |
|------------------------------------------|-------|---------------------|
| Stiffness of main spring $k_1$ (N-m$^{-1}$) | 2860 |
| Stiffness of deputy spring $k_2$ (N-m$^{-1}$) | 847 |
| Coefficient of tensile damping $c_1$ (N-s-m$^{-1}$) | 1149 | 0.9180 m-s$^{-2}$ |
| Coefficient of compression damping $c_2$ (N-s-m$^{-1}$) | 1057 |
| Inertance $b$ (kg) | 5351 |

### Table 4: Optimal parameters with suspension working space as the single objective.

| Optimized parameters (symmetric damping) | Value | Optimized objective |
|-----------------------------------------|-------|---------------------|
| Stiffness of main spring $k_1$ (N-m$^{-1}$) | 6437 |
| Stiffness of deputy spring $k_2$ (N-m$^{-1}$) | 2294 |
| Coefficient of symmetric damping $c_0$ (N-s-m$^{-1}$) | 1816 |
| Inertance $b$ (kg) | 903 |

| Optimized parameters (asymmetric damping) | Value | Optimized objective |
|------------------------------------------|-------|---------------------|
| Stiffness of main spring $k_1$ (N-m$^{-1}$) | 7351 |
| Stiffness of deputy spring $k_2$ (N-m$^{-1}$) | 3146 |
| Coefficient of tensile damping $c_1$ (N-s-m$^{-1}$) | 1797 |
| Coefficient of compression damping $c_2$ (N-s-m$^{-1}$) | 1836 |
| Inertance $b$ (kg) | 315 |

### Table 5: Optimal parameters with dynamic tire load as the single objective.

| Optimized parameters (symmetric damping) | Value | Optimized objective |
|-----------------------------------------|-------|---------------------|
| Stiffness of main spring $k_1$ (N-m$^{-1}$) | 11864 |
| Stiffness of deputy spring $k_2$ (N-m$^{-1}$) | 1151 |
| Coefficient of symmetric damping $c_0$ (N-s-m$^{-1}$) | 1757 |
| Inertance $b$ (kg) | 5049 |

| Optimized parameters (asymmetric damping) | Value | Optimized objective |
|------------------------------------------|-------|---------------------|
| Stiffness of main spring $k_1$ (N-m$^{-1}$) | 11519 |
| Stiffness of deputy spring $k_2$ (N-m$^{-1}$) | 4072 |
| Coefficient of tensile damping $c_1$ (N-s-m$^{-1}$) | 1733 |
| Coefficient of compression damping $c_2$ (N-s-m$^{-1}$) | 1822 |
| Inertance $b$ (kg) | 24 |

### Table 6: Optimal parameters of multioptimization.

| Optimized parameters (symmetric damping) | Value | Optimized objective |
|-----------------------------------------|-------|---------------------|
| Stiffness of main spring $k_1$ (N-m$^{-1}$) | 2811 |
| Stiffness of deputy spring $k_2$ (N-m$^{-1}$) | 5612 |
| Coefficient of symmetric damping $c_0$ (N-s-m$^{-1}$) | 1702 |
| Inertance $b$ (kg) | 2778 |

| Optimized parameters (asymmetric damping) | Value | Optimized objective |
|------------------------------------------|-------|---------------------|
| Stiffness of main spring $k_1$ (N-m$^{-1}$) | 4446 |
| Stiffness of deputy spring $k_2$ (N-m$^{-1}$) | 3948 |
| Coefficient of tensile damping $c_1$ (N-s-m$^{-1}$) | 1702 |
| Coefficient of compression damping $c_2$ (N-s-m$^{-1}$) | 1686 |
| Inertance $b$ (kg) | 1562 |

### Table 7: Dynamic performance indices.

| | Conventional suspension | Symmetric-damping ISD | Asymmetric-damping ISD |
|----------------|-------------------------|------------------------|------------------------|
| RMS of body acceleration (m-s$^{-2}$) | 1.1948 | 1.1469 | 1.1446 |
| RMS of suspension working space (m) | 0.0119 | 0.0085 | 0.0082 |
| RMS of dynamic tire load (N) | 839.2 | 727.0 | 724.3 |
PSD (power spectrum density) of the body acceleration, the suspension working space, and the dynamic tire load is shown in Figures 4(b), 5(b), and 6(b). It can be inferred that, all of the amplitudes, in low-frequency bands, of PSD of the body acceleration, the suspension working space, and the dynamic tire load are plummeted in the frequency spectrum from 1 Hz to 3 Hz without performance degradation being seen in other bands, showing superior vibration isolation performance among all the studied spectrum. The exact values are listed in Table 8. In comparison with the conventional suspension, amplitudes of each PSD of the proposed ISD suspension with symmetric damping have dramatically decreased by 91.3%, 79.7%, and 80.8% in sequence, and the asymmetric-damping ISD suspension outperforms the symmetric-damping ISD suspension to an extent that the corresponding figures of the asymmetric-damping ISD suspension have been promoted to 94.5% and 87.8% in terms of amplitudes of PSD of the body acceleration and dynamic tire load, though the amplitude of PSD of the
Table 8: The maximum PSD indices of low-frequency bands.

| Name                                           | Conventional suspension | Symmetric-damping ISD | Asymmetric-damping ISD |
|------------------------------------------------|-------------------------|-----------------------|------------------------|
| Amplitude of PSD of body acceleration (m·s⁻²·Hz⁻¹) | 1.84                    | 0.16                  | 0.10                   |
| Amplitude of PSD of suspension working space (m²·Hz⁻¹) | 3.40 × 10⁻⁴             | 6.90 × 10⁻⁵           | 7.44 × 10⁻⁵            |
| Amplitude of PSD of dynamic tire load (N²·Hz⁻¹)    | 2.03 × 10⁵              | 3.89 × 10⁴            | 2.47 × 10⁴             |

Figure 6: Comparison of dynamic tire load in time (a) and frequency (b) domain.

Figure 7: The outputs under impulse input.

Table 9: Peak-to-peak indices.

| Name                        | Conventional suspension | Symmetric-damping ISD | Asymmetric-damping ISD |
|-----------------------------|-------------------------|-----------------------|------------------------|
| PTP of body acceleration (m·s⁻²) | 15.0                    | 11.2                  | 10.3                   |
| PTP of suspension working space (m) | 0.1552                 | 0.1343                | 0.1432                 |
| PTP of dynamic tire load (N)   | 5206                    | 5071                  | 5250                   |
4.2. Impulse Input. The impulse input model with the long-slope lump as the trigger can be set as follows:

$$z_r(t) = \frac{\pi u A_m}{L} \sin \frac{2\pi u}{L} t, \quad 0 \leq t \leq \frac{L}{u},$$  \hspace{1cm} (11)

where $u$ is the speed with the value of 10 m/s, $A_m$ is the amplitude with the value of 0.1 m, and $L$ is the effective length with the value of 5 m. The suspension system’s outputs under this input are shown in Figure 7.

From Figure 7, the peak-to-peak (PTP) value of each dynamic indicator can be gained, and results are listed in Table 9.

Obviously, the PTP of almost all dynamic indicators of the optimized ISD suspension, compared with the conventional suspension, can witness notable improvements in the way that the PTP of body acceleration has decreased by 31.3% for asymmetric-damping ISD suspension and 25.3% for symmetric-damping ISD suspension and the PTP of suspension working space has decreased by 7.7% for asymmetric-damping ISD suspension and 13.4% for symmetric-damping ISD suspension. Though only the PTP of the dynamic tire load shows little change, the time consumption needed for all curves to be flatten is much less than that of the conventional suspension, and the overshoot shows the same feature.

4.3. Sinusoidal Input. The model of sinusoidal input can be defined as follows:

$$z_r(t) = h \sin \frac{\pi u}{L} t,$$ \hspace{1cm} (12)

where $h$ is the height of uneven road, the value of it is 0.05 m, $u$ is the speed with the value of 25 m/s, and $L$ is the half wavelength with the value of 10 m. Results of simulations under sinusoidal input are shown in Figure 8.

Table 10: Amplitude indices.

| Name                               | Conventional suspension | Symmetric-damping ISD | Asymmetric-damping ISD |
|------------------------------------|-------------------------|-----------------------|------------------------|
| Amplitude of body acceleration (m·s⁻²) | 10.3                    | 2.6                   | 1.6                    |
| Amplitude of suspension working space (m) | 0.141                  | 0.0768                | 0.0566                 |
| Amplitude of dynamic tire load (N)             | 3450                    | 840                   | 557                    |
The amplitudes of the above curves can be gained, and results are listed in Table 10.

Surprisingly, in comparison with the conventional suspension, phenomenal improvements of the three indices of both asymmetric-damping suspension and symmetric-damping suspension can be perceived. For the former suspension, amplitudes of the body acceleration, the suspension working space, and the dynamic tire load have extraordinarily decreased by 84.4%, 59.8%, and 83.8%; for the latter suspension, the corresponding values are 74.7%, 45.5%, and 75.6%. So, it can be inferred that dynamic performances of the asymmetric-damping suspension are comprehensively superior to those of the symmetric-damping suspension under sinusoidal input, implying that the asymmetry of the damper may render more potentials for improvements of the proposed vehicle ISD suspension.

5. Impacts Analysis of the Asymmetric Damping

In order to unravel the secret of how the asymmetry of damper may act on the body acceleration, the suspension working space, and the dynamic tire load, those three dynamic characteristics are regarded as response variables while the coefficients of tensile damping and compression damping are regarded as independent variables. This paper makes the two independent variables change in the region of [500, 3500] N·s·m⁻¹ with a certain step size; results are shown in Figures 9 and 10.

Generally speaking, the influencing features of the coefficients of tensile damping $c_1$ and compression damping $c_2$ are symmetric. When one of them increases with the other remaining unchanged, some performance degradation can be expected for body acceleration whereas some improvements can be noticed before the suspension working space is degraded. To be precise, the RMS of suspension working space tends to decrease a bit before increase, meaning that performance may turn better initially and fall down subsequently. When those two coefficients change within the region of [2000, 3000] N·s·m⁻¹, the RMS of dynamic tire load shows negligible fluctuations, differing from the situation that the dynamic tire load has increased in acceleration when one coefficient drops in the region of [500, 2000] N·s·m⁻¹ while the other remains unchanged, meaning that there may be some performance degradation when the two coefficients are lower than 2000 N·s·m⁻¹, which should be averted while designing the ISD suspension.

6. Conclusion

In this paper, the dynamic model of the proposed vehicle ISD suspension involving the symmetric and asymmetric damping is built and analyzed. Based on that single-objective and multiobjective optimizations are carried out by means of GA under random road input. The impacts of the asymmetry of the damper acting on the body acceleration, the suspension working space, and the dynamic tire load are unveiled. Furthermore, the dynamic performances of the optimal ISD suspension considering the asymmetry of the damper are analyzed in time and frequency domains under random input, in time domain solely under impulse and sinusoidal input. Results show that all of the dynamic performance indices of the optimal ISD suspension, regardless of the presence of asymmetric damping, have decreased except for the dynamic tire load in the impulse input.
condition. The tensile and compression damping will have symmetric impacts on all dynamic indicators; when the two coefficients increase, the body acceleration performance will be degraded, the suspension working space performance will show fluctuations, but the dynamic tire load performance will only be degraded in acceleration while two coefficients decrease in the region of [500, 2000] N·s·m⁻¹. Judging from phenomenal improvements in all kinds of indices under different inputs, the design of the asymmetric damper should be an important factor when optimizing the vehicle ISD suspension.

Data Availability

The data used to support the findings of the study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this manuscript.

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