Research Article

Allocation Optimization of Multi-Axis Suspension Dynamic Parameter for Tracked Vehicle

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1.Introduction

Tracked vehicles are specifically designed to move on rugged off-road terrain, such as military combat tanks [1]. These vehicles have strong off-road maneuverability and ensure the vehicle’s ability to pass under extreme road conditions [2, 3]. The suspension is an important part of tracked vehicles, which improves the operator’s endurance to withstand the transmitted shock and vibration. It bears the load from the vehicle body, and more importantly, it reduces the impact and vibration caused by the unevenness of the road surface and improves the adaptability of the occupants [4, 5]. The pros and cons of the suspension system directly determine the performance level of the tracked vehicle, especially the maneuverability and ride comfort [6]. With the development of modern military equipment, the requirements for the mobility of tracked vehicles are increasing [7, 8]. Therefore, improving the suspension performance is practical in the industry.

Passive suspensions are widely used because of their simplicity and efficiency. These suspensions mean that the characteristics of the components (springs and dampers) are fixed and determined by the designer according to the expected application. Ideally, a suspension should adjust its characteristics to meet the needs of different road surfaces [9, 10]. For the suspensions to achieve the ability to cope with different road surfaces, it has appeared a variety of active and semiactive suspensions [11–14], such as electrohydraulic active suspension [15] and electromagnetic semiactive suspension [16]. Zhang et al. [17] design a composite electromagnetic suspension (CES) in parallel with an electromagnetic actuator (EA) and a magnetorheological damper (MRD). Compared with passive suspension, semiactive and active suspensions have complex devices, high
technical requirements, and poor stability, so they are not widely used in tracked vehicles.

Moreover, most of the active and semiactive suspensions mentioned before are based on the quarter vehicle model. Although the suspensions can well attenuate from road-induced vibration and shocks, due to the influence of vehicle body structure, mass distribution, and other factors, the position of each road wheel is generally nonuniform. When the tracked vehicle is traveling, each load-bearing wheel is affected by the excitation of random road roughness. The distribution of the suspension dynamics parameters (stiffness and damping) will affect the stability of the vehicle, and it is difficult to achieve the optimal stabilization effect simply by employing equal parameter allocation, or in other words, the optimal damping effect cannot be achieved.

However, the existing studies are not enough to consider the allocation optimization of multi-axis suspension dynamic parameter for a tracked vehicle, and there is no general method to realize the optimal allocation of suspension system dynamics parameters. To overcome the influence from the distribution of the suspension dynamics parameters (stiffness and damping), it is necessary to design a dynamic parameter allocation optimization method of tracked vehicle multi-axle suspension.

There are some studies focusing on the optimization of the stiffness, damping, and other parameters of the suspension system. Dong et al. [18] propose a statistical linearization analysis method to find the optimal damping coefficient for the damping matching problem of 2-DOF nonlinear suspension, to minimize the root mean square (RMS) of vehicle acceleration. When the stiffness and damping parameters of the suspension system are optimized, it is necessary to establish a vehicle dynamics model, which can be a quarter vehicle model, a half vehicle plane model, or the whole vehicle model [19–21]. At the same time, the objective function of stiffness and damping parameter optimization should be set. Since the objective function will directly affect the optimization results, different objective functions will obtain different matching of the suspension system’s stiffness and damping parameters. Therefore, the optimal suspension performance is usually considered when the objective function is designed [22]. It can be concluded that it is particularly important to establish a reasonable evaluation index for suspension performance.

It is necessary to consider the ride comfort of the vehicle for automobiles, where the vibration acceleration is mainly used to evaluate the suspension performance [23]. By considering the influence of nonlinear damping characteristics of the automobile shock absorber on ride comfort, Feng et al. [24] take the RMS of body vibration acceleration as the objective function of optimal design of damping parameters of shock absorber in the suspension system and optimize the nonlinear damping of the suspension system. But, for tracked vehicles, such as military battle tanks, the traveling surface is usually off-road [25]. When traveling under bad road conditions, to evaluate the suspension performance, it should be considered to reduce the number of times that the suspension balance elbow hits the limiter, reduce the vibration and pitching vibration of the vehicle body, and maintain the heat of the suspension system damper in a reasonable range [26]. Due to its complexity, various factors have different effects on the suspension performance of the tracked vehicle. While considering the safety and the ride comfort of the vehicle, the evaluation of the suspension performance of the tracked vehicle should include more aspects, such as considering the impact of the suspension performance on the launch of onboard weapons. A reasonable objective function should be designed to optimize the stiffness and damping parameters of the suspension system. Based on the tracked vehicle road roughness input model and parameterized dynamics model, the evaluation indexes reflecting the multiple aspects of performance of the tracked vehicle suspension system and their quantization algorithm are analyzed and expounded. Then, the objective function of parameter allocation based on multi-index information fusion is designed to realize the optimal allocation of dynamics parameters of the tracked vehicle suspension system.

This study makes several theoretical and knowledge contributions. Firstly, the five evaluation indexes and quantization algorithm of vehicle suspension system performance are analyzed and proposed. The evaluation indexes are not only considered safety and ride comfort but also considered other factors, such as the impact of vibration on the launch of weapons, dissipation performance of dampers, and impact of suspension balance elbow on limiter. Secondly, according to different target needs, the objective function of parameter allocation for multi-index information fusion is designed, and two methods of equal weight allocation optimization and expert knowledge-based weight allocation optimization are used to solve the parameter allocation optimization. For tracked vehicles, the distribution of suspension stiffness and damping parameters of each axle is nonuniform. However, their specific allocation value is often determined based on experience and lack of theoretical basis. The method presented in this paper provides a way to solve this problem. Thirdly, based on the results of parameter allocation optimization, the suspension performance of the tracked vehicle traveling on different roads at different speeds is compared and analyzed, and an effective method for the allocation optimization of suspension system stiffness and damping parameters is verified. The method can provide knowledge reference for multi-shaft suspension system design.

The remainder of this paper is organized as follows. Section 2 presents the parametric dynamics model of the tracked vehicle. Section 3 describes the evaluation index and optimization method. The suspension performance of the tracked vehicle is analyzed and discussed under the condition that the vehicle travels on different road surfaces at different speeds in Section 4. Finally, the conclusions are summarized.

2. Parameterized Dynamics Model of Tracked Vehicle

2.1. Road Roughness Input (Excitation). The uneven road surface is the main factor to produce the vertical vibration of the vehicle, which has an important influence on the
ride comfort and stability of the vehicle. Its measurement index is usually expressed by the road roughness, that is, the change of the height of the road surface relative to the base plane along the length of the road. At present, according to the upper and lower limits of the power density value, the road surface is divided into A–H altogether eight grades in the International Standards Association. In the Chinese military standard, four typical road surfaces for field vibration tests of armored vehicles are stipulated: paved road, rough road, gobi road, and gravel road. In this study, four typical road roughness inputs specified in Chinese military standard GJB 59.15–88 are introduced, and a time-domain road excitation mathematical model is established by reference to Dong and Liang’s typical road roughness input method, namely, harmonic superposition method of power spectral density of rational function [27], as shown in the following equation:

\[ q(t) = \sum_{k=1}^{N} a_k \sin(2\pi f_k t + \varphi_k), \]  

where \(a_k\) – amplitude coefficient of road roughness input, \(f_k\) – the input frequency of the road roughness input, and \(\varphi_k\) – the initial phase angle of the road roughness input, which obeys uniformly distributed in the interval \((0, 2\pi)\), and the \(N\) phase angles are independent of each other.

If the tracked vehicle travels at the speed \(v\), \(f_k\) and \(\varphi_k\) can be solved in turn by the method introduced by Dong and Liang [27], which will not be described in this study.

If the influence of road deformation is not considered, when different weight-bearing wheels on the same side pass a certain point on the road in turn, the input function of road roughness can be regarded as only having the effect of time lag. For example, if the road roughness input functions of the first and second load-bearing wheels of the left rut are \(q_{l_1}(t)\) and \(q_{l_2}(t)\), respectively, the road roughness input functions of the adjacent load-bearing wheels have the following relationship:

\[ q_{l_2}(t) = q_{l_1}\left(t - \frac{\Delta L_{1,2}}{v}\right), \]  

where \(\Delta L_{1,2}\) is the distance between the first and second load-bearing wheels.

For any two adjacent loading wheels, it can be expressed as follows:

\[ q_{l(i,i+1)}(t) = q_{li}\left(t - \frac{\Delta L_{i,i+1}}{v}\right). \]  

Then, considering the road surface on both sides, the input function of random road roughness corresponding to the left and right load-bearing wheels can be described through phase correlation. As shown in equation (4), at low frequency, the phase angle of the left and right weight-bearing wheels is strongly correlated; on the contrary, the correlation is weak [28].

\[ \varphi_r = \frac{e^{-2nd,n^5} \varphi_l + \sqrt{1 - e^{-2nd,n^5}} \varphi_{new}}{\sqrt{1 - e^{-2nd,n^5}} + e^{-2nd,n^5}}, \]  

where \(\varphi_r\) is the initial phase angle input for the road roughness of the right rut, \(\varphi_l\) is the initial phase angle input for the left rut road roughness, \(\varphi_{new}\) is a newly generated random sequence of \((0, 2\pi)\), and \(d_n\) is the wheelbase.

The input function expression of road roughness of each load-bearing wheel can be obtained as follows by combining equations (1)–(4):

\[
\begin{aligned}
q_{l_1} &= \sum_{k=1}^{N} a_k \sin(2\pi f_k t + \varphi_l) \\
q_{l_2}(t) &= \sum_{k=1}^{N} a_k \sin\left[2\pi f_k\left(t - \frac{\Delta L_{1,2}}{v}\right) + \varphi_l\right] \\
&\vdots \\
q_{l_6}(t) &= \sum_{k=1}^{N} a_k \sin\left[2\pi f_k\left(t - \frac{\Delta L_{5,6}}{v}\right) + \varphi_l\right] \\
q_{r_1} &= \sum_{k=1}^{N} a_k \sin(2\pi f_k t + \varphi_r) \\
q_{r_2}(t) &= \sum_{k=1}^{N} a_k \sin\left[2\pi f_k\left(t - \frac{\Delta L_{1,2}}{v}\right) + \varphi_r\right] \\
&\vdots \\
q_{r_6}(t) &= \sum_{k=1}^{N} a_k \sin\left[2\pi f_k\left(t - \frac{\Delta L_{5,6}}{v}\right) + \varphi_r\right]
\end{aligned}
\]

where \(q_{li}\) is the road roughness input of the \(i\)-th load-bearing wheel on the left and \(q_{ri}\) is the road roughness input of the \(i\)-th load-bearing wheel on the right.

As shown in Figure 1, the surface roughness input curve of Gobi road is obtained by taking the vehicle speed as \(v = 40\) km/h. Among them, Figure 1(a) is the road roughness input curve of the 1st and 4th load-bearing wheels on the left, and Figure 1(b) is the road roughness input curve of the 1st and 4th load-bearing wheels on the left.

2.2 Parametric Dynamic Model. Reasonably simplifying the dynamics model of tracked vehicles [29] is the premise for the dynamic parameter allocation optimization of the suspension system, which can greatly improve the optimization efficiency. When establishing the dynamic model, the following assumptions are first presented.
(1) The road on which the tracked vehicle travels is a rigid road, and the surface morphology will not be changed due to the rolling of the traveling system. There is only a time delay in the input of each load-bearing wheel.

(2) The track structure is regarded as an infinite track that is similar to the ground excitation, and the effect of the track on the vehicle is ignored.

(3) Since the focus of this study is how to optimize the dynamic parameter allocation of the suspension system, during the parameter allocation optimization process, the road surface should have the same excitation, which does not affect the effectiveness of the method proposed in this study, so the filter effect of the track cannot be considered.

(4) The mass center of the vehicle body is symmetrical about the longitudinal axis, and there are six load-bearing wheels on each side. When only the roll, pitch, and vertical vibration of the vehicle body are considered, 15 degrees of freedom vehicle model can be employed to establish a parameterized dynamics model of tracked vehicles.

(5) The vehicle adopts independent linear suspension, which decomposes the suspension mass to the sprung mass and unsprung mass, and the suspension can be simplified into stiffness and damping [30, 31].

(6) The damping force, elastic force, and gravity of the load-bearing wheel all act on the mass center of the load-bearing wheel.

Based on the above assumptions, the parameterized dynamics model of tracked vehicles is established.

As shown in Figure 2, the complete suspension model on the left is drawn. In order to make the picture clearer, the model on the right is not completely described. The road wheels and suspension of nos. 8, 9, 10, and 11 shafts are omitted, and only nos. 7 and 12 are displayed. The direction of speed $v$ is the forward direction of the vehicle; $x$ is the vertical direction of the mass center of the vehicle; $z$ is the direction of the horizontal axis; $y$ is the traveling direction of the tracked vehicle; $m_s$ is the body mass; $I_z$ is the pitching moment of inertia of the vehicle; $\alpha_z$ is the pitch angle of the vehicle; $I_y$ is the roll moment of inertia of the vehicle; $\alpha_y$ is the roll angle of the vehicle; $m_{ti}$ is the mass of the $i$-th load-bearing wheel of the vehicle; $k_{si}$ is the suspension stiffness of the $i$-th load-bearing wheel; $c_{si}$ is the suspension damping of the $i$-th load-bearing wheel; $c_{ti}$ is the tire damping of the $i$-th load-bearing wheel; $k_{ri}$ is the tire stiffness of the $i$-th load-bearing wheel; $l_{yi}$ is the horizontal distance between the centroid of the $i$-th load-bearing wheel and the centroid of the vehicle in the $y$ direction; $l_{zi}$ is the vertical distance of the centroid of the vehicle body with the $i$-th load-bearing wheel in the $z$ direction; $x_{gi}$ is the vertical displacement of the $i$-th load-bearing wheel; $q_i$ is the road input under the $i$-th load-bearing wheel; $i = 1, 2, \ldots, 12$, where 1 to 6 denote the first to the sixth load-bearing wheel on the left; and 7 to 12 denote the 7th to the 12th load-bearing wheel on the right.

According to the above physical analysis model, the dynamic differential equations of the whole tracked vehicle can be deduced as follows:

\[
\begin{align*}
    m \ddot{x}_i + \left[ K_s \left( X_i - X_d \right) + C_s \left( \dot{X}_i - \dot{X}_d \right) \right] A^T &= 0, \\
    I \ddot{\alpha}_z + \left( K_s \left( X_z + X_d \right) \right) L_y^T &= 0, \\
    I \ddot{\alpha}_y + \left( K_s \left( X_y + X_d \right) \right) L_z^T &= 0, \\
    M \ddot{X}_i - K_s \left( X_i - X_d \right) - C_s \left( \dot{X}_i - \dot{X}_d \right) + K_t \left( X_i - X_r \right) + C_t \left( \dot{X}_i - \dot{X}_r \right) &= 0,
\end{align*}
\]
where $A = [1, \ldots, 1]_{1 \times 12}$ is the unit row vector,

$K_s = [k_{s1}, \ldots, k_{s12}]$,  
$K_t = [k_{t1}, \ldots, k_{t12}]$,  
$C_s = [c_{s1}, \ldots, c_{s12}]$,  
$C_t = [c_{t1}, \ldots, c_{t12}]$,  
$L_y = \left[ l_{y1} \cos \alpha_x, \ldots, l_{y12} \cos \alpha_x \right]$,  
$L_z = \left[ l_{z1} \cos \alpha_x, \ldots, l_{z12} \cos \alpha_x \right]$,  
$M_t = [m_{t1}, \ldots, m_{t12}]$,  

$X_s = \text{diag}(x, \ldots, x, x)$,

$X_r = \text{diag}(q_1, \ldots, q_1, \ldots, q_1)$,

$X_t = \text{diag}(x_{t1}, \ldots, x_{t12})$,

(7)

where $\text{diag}$ stands for diagonal matrix.

2.3. Bench Test. The validity of the model is that the established model can reflect the characteristics of the system to be studied comprehensively. Therefore, it is necessary to verify the simulation model. Through inspection and verification, this gap can be reduced as far as possible to get the most reasonable description of the real system. To confirm the validity of the model, the test data can be obtained by sample car test on the premise of ensuring the similarity principle. According to the modeling goal, the validation parameters and the recognized range of parameter values are selected. By comparing the calculated data with the experimental data, if the error range of the two is within the allowed range of the evaluation parameters, it can be considered that the parameterized dynamic model has achieved the modeling goal; otherwise, the model is unreliable and needs to be continuously modified.

In order to verify the rationality of the established parameterized dynamics model of the tracked vehicle, a road simulation vibration test is carried out based on the road simulation test rig as shown in Figure 3. The schematic diagram of the road simulation test rig is shown in Figure 4, which mainly includes: actuator and its platform, test vehicle, and vibration test system. During the vibration test, the measured road roughness is properly processed; the actuator is the simulated road roughness information; and the vibration test system can obtain the dynamic response signal of the vehicle body. In this study, the Gobi road of a test site in China was simulated. The measured geometric elevation signal of Gobi road is used as input, and the speed is simulated at 25 km/h. The sampling frequency of the acceleration sensor is 2,048 Hz. The acceleration signal and its power spectrum signal are used to describe and verify the model. The main frequency of acceleration power spectrum signal is the resonance frequency of vertical vibration of the vehicle body.

The response signal of the acceleration of the vehicle body's center of mass is tested and analyzed, and the experiment and numerical calculation results are shown in Figure 5. Figure 5(a) is the acceleration signal curve of the vehicle body's centroid, and Figure 5(b) is the acceleration power spectrum signal of the vehicle body's centroid. Combined with the acceleration signals shown in Figure 5, the RMS of the acceleration center of the vehicle body obtained through numerical calculation is...
3.49 m/s², and the RMS value of the acceleration center of the vehicle body in the bench test is 3.04 m/s², with only a 12.89% difference between them. The body resonance frequency obtained by numerical calculation is 1.97 Hz, and that obtained by bench test is 1.91 Hz, which are very close to each other. In addition, the other roads of the test site are simulated, including paved road, rough road, and gravel road. The test conditions are the same as the Gobi road. The experiment and numerical calculation of the acceleration RMS is shown in Table 1. In the road simulation test, the difference of RMS between the experiment and numerical calculation is less than 16%, and the established parameterized dynamics model of tracked vehicle is in good agreement with the test results.

3. Dynamic Parameter Allocation Design

3.1. Design of Evaluation Indexes. The tracked vehicle is affected by the random excitation of the road surface in the process of traveling, and the resulting vibration will affect the occupant's adaptability, attack accuracy, reliability of onboard devices, and walking system [26]. In order to improve the global performance, stability, and reliability of the suspension system, dynamic attribute parameters of the suspension system are optimized and assigned. However, how to effectively evaluate the suspension performance of the tracked vehicle is the first problem that needs to be solved. Based on different perspectives, the indexes for evaluating its performance will be different. As shown in Figure 6. The quantitative value of each performance evaluation index of the suspension system is expressed by \( \text{PI}_1, \text{PI}_2, \text{PI}_3, \text{PI}_4, \) and \( \text{PI}_5 \), respectively, and they have a similar meaning in Tables 2 and 3 and Figures 7–11.

In Figure 6, performance evaluation indexes of the suspension system are proposed in this study from five aspects, namely: (1) the vertical acceleration of the vehicle body above the first load-bearing wheel, which reflects the adaptability of the driver; (2) the dynamic displacement of the first load-bearing wheel relative to the vehicle body is smaller, which can avoid the impact of the weight-bearing wheel on the limiter and reduce the alternating load of suspension with large deformation; (3) the vertical velocity of the first load-bearing wheel relative to the vehicle body, which reflects the damping power (calorific value) of the suspension shock absorber; (4) the vehicle pitch angle acceleration, which will affect the shooting accuracy of the fire control system; and (5) the dynamic load coefficient of the first load-bearing wheel relative to the ground, which reflects the ground stickability of the load-bearing wheel.

It should be noted that indexes 1, 2, 3, and 5 all choose the corresponding indexes on the first load-bearing wheel because the tracked vehicle is mainly pitching when traveling normally, and the position of the driver is generally on the first load-bearing wheel. In fact, it is also meaningful to observe these indexes on other load-bearing wheels. However, in order to reduce the additional difficulty of subsequent optimization solutions caused by excessive evaluation indexes, the corresponding indexes on other load-bearing wheels are taken as constraint conditions. For example, it is set that the maximum dynamic displacement of the second load-bearing wheel relative to the vehicle body should be less than the maximum dynamic displacement of the first load-bearing wheel relative to the vehicle body. In the optimization process, these constraints are judged, and if the set constraints are not satisfied, the optimization solution needs to be reoptimized.

In order to better evaluate and analyze the suspension performance, it is necessary to quantitatively describe the proposed evaluation indexes, which are described as follows:

1. The vertical acceleration of the vehicle body above the first load-bearing wheel

The vertical acceleration of the vehicle body above the first load-bearing wheel on the left is expressed by \( \text{Acc}_1 \), and its quantitative expression is shown in the following equation:

\[
\text{Acc}_1 = (x_1 + l_{1x} \sin \alpha_y + l_{1y} \sin \alpha_z)^n |\rho.
\]  

(8)

2. Dynamic displacement of the load-bearing wheel relative to the vehicle body

The dynamic displacement of the load-bearing wheel relative to the vehicle body [29] is shown in the following equation:
Vertical speed of the weight-bearing wheel relative to
the vehicle body can be obtained by taking the derivative of the corresponding dynamic dis-
placement, as shown in the following equation:

\[ v_i = \dot{x}_i - \dot{x}_s - \alpha_y \dot{y}_i \cos \alpha_y - \alpha_z \dot{z}_i \cos \alpha_z, \quad (10) \]

where \( \dot{x}_i \) is the velocity of the mass center of the load-
bearing wheel, \( \dot{x}_s \) is the velocity of the mass center of
the vehicle body, \( \alpha_y \) is the pitch angle angular ve-
locity of the vehicle body, and \( \alpha_z \) is the roll angle angular velocity of the vehicle body.

(4) Pitch angular acceleration of the vehicle body
The pitch angular acceleration of the vehicle body can be expressed as follows:

$$PA = \ddot{\alpha}_y,$$

(11)

where $$\ddot{\alpha}_y$$ is the pitch angular acceleration of the vehicle body.

(5) Dynamic load coefficient of the load-bearing wheel relative to the ground.
The instantaneous dynamic load of the load-bearing wheel relative to the ground is shown in the following equation:

\[ F_d(t) = k_i (x_i - q_i) + c_i (\dot{x}_i - \dot{q}_i) \quad (12) \]

where \( k_i \) and \( c_i \) are the stiffness and damping coefficients, respectively, \( x_i \) and \( \dot{x}_i \) are the vertical displacement and velocity of the load-bearing wheel, and \( q_i \) and \( \dot{q}_i \) are the vertical displacement and velocity of the ground.

The ground stickability of the load-bearing wheel is usually expressed by the dynamic load coefficient \([32]\), as shown in the following equation:

\[ D_i = \frac{\sigma_{F_d} + F_i}{F_i} \quad (13) \]

where \( \sigma_{F_d} \) is the RMS value of the dynamic load for the load-bearing wheel and \( F_i \) is the static load for the load-bearing wheel.

3.2 Parameter Allocation Objective Function. Whether the objective function design is reasonable directly affects the
optimization result and the suspension performance. In this study, five performance evaluation indexes of the suspension system are put forward, and five objective functions are needed to be designed. Since one objective function is optimal, the other objective function is not necessarily optimal, and the optimal solutions of some objective functions are contradictory to each other, the optimization results do not necessarily have a globally optimal solution \[33, 34\].

Therefore, the final compromise solution should be selected according to the designer’s preference \[35, 36\]. In order to make each objective function as small as possible, the minimum value of each objective function, namely the optimal solution, can be obtained first. Then, multiple indexes were fused to form the objective function of comprehensive optimization of suspension dynamics parameter allocation. The specific steps are shown in Figure 7.

\[\text{Figure 9: Comparison for the suspension system performance index (on gravel road).}\]
**Step 1.** Design the objective function of a single index

According to vehicle operating characteristics, the smaller index 1 to index 5 is, the better its performance will be. The optimization objective functions can be designed as minimum functions, which are represented by $e_1$, $e_2$, $e_3$, $e_4$, and $e_5$, respectively, as shown in equations (14)–(18), where $n$ is the number of data volumes. Moreover, RMS values are taken for index 1, 3, and 4, which is mainly to reflect the condition of tracked vehicles in a period. If the RMS value is small, it indicates that the vertical vibration of the vehicle body above the first load-bearing wheel is small, the caloric value of the vibration damping system is low, and the elevation angle acceleration of the vehicle body has little influence on the fire control system. Index 2 mainly reflects the
possibility of suspension impacting the limiter, so its objective function is expressed by the single maximum dynamic displacement. Index 5 mainly reflects the ground attachment of the loading wheel, and the objective function can be directly described by the dynamic load coefficient of the first load-bearing wheel relative to the ground.

$$e_1 = \min \left( \frac{1}{n} \sum_{k=1}^{n} \text{Acc}_k^2 \right),$$  \hspace{1cm} (14)

$$e_2 = \min \left( \max |\text{Disp}_1| \right),$$  \hspace{1cm} (15)
$$e_3 = \min \left( \frac{1}{n} \sum_{k=1}^{n} v_k^2 \right),$$  \hspace{1cm} \text{(16)}$$

$$e_4 = \min \left( \frac{1}{n} \sum_{k=1}^{n} PA_k \right),$$  \hspace{1cm} \text{(17)}$$

$$e_5 = \min D_{ij}.$$

\text{Step 2. Solve the optimal solution of the objective function of each index.}

According to the objective function designed in step 1, the value range of parameters such as stiffness and damping of the suspension system is determined, and a reasonable optimization algorithm is introduced for the iterative solution to obtain the optimal solution of the objective function of each index.

\text{Step 3. Dimensionless processing of the objective function.}

Because of the different dimensions of each objective function, it is not convenient to conduct information fusion directly. Therefore, before constructing the objective function of the comprehensive optimization of the dynamic parameter allocation of the suspension system, the dimensionless processing of each objective function should be carried out. That is, the minimum value of each objective function is obtained based on step 2, and the dimensionless processing is carried out for each objective function, and the nonquantitative expression is designed as shown in the following equation:

$$D_j^* = \frac{(e_j - e_j^0)}{e_j^0},$$ \hspace{1cm} \text{(19)}$$

where $D_j^*$ is the dimensionless objective function and $e_j^0$ is the extreme value of each objective design criterion.

\text{Step 4. Build the objective function of allocation optimization with multi-index fusion.}

According to equation (19), dimensionless objective functions of five evaluation indexes are obtained. Then, a weight is assigned to each objective function, and multiple objective functions are fused into one comprehensive objective function, whose expression is shown in the following equation.

$$U = \left\{ \frac{1}{\sqrt{\sum_{j=1}^{g} w_j D_j^{*2}}} \right\},$$ \hspace{1cm} \text{(20)}$$

where $U$ is the distance between the objective function and the ideal solution, $w_j$ is the weight coefficient of each objective function, and $j$ is the subscript of the performance index, take $1 \sim 5$.

As for equation (20), the key to the problem is how to determine the weight coefficients of each objective function $(D_j^*)$. Because each performance evaluation index has a different influence on the suspension performance, it may not get the optimal effect by simply using equal weight allocation. However, there is no effective method to determine the weight allocation at present. In view of this, expert scoring may be a solution and can make full use of the experience of experts. Experts with rich knowledge and experience will be invited from different roles of tracked vehicle design, production, and traveling to evaluate the weight coefficients of each target, and the design of the evaluation table is shown in Table 2. It can be seen from Table 2 that the vehicle body’s pitch angle acceleration index accounts for the largest weight, followed by the vehicle body’s vertical acceleration index above the first load-bearing wheel, while the dynamic load coefficient index is accounted for the least weight. After the average value method is used to process the evaluation results of each expert, the final weight coefficient allocation of each objective is obtained.

3.3 Parameter Allocation and Solution Process. Based on the above analysis, the optimization process of dynamic parameter allocation of the suspension system is designed in Figure 12.

Firstly, the input model of road roughness and the parameterized dynamics model of tracked vehicles are established. Then, according to experience, the variation range of parameters such as stiffness and damping of the suspension system is determined, as shown in Table 4. For the studied tracked vehicle, the natural frequency of vertical linear vibration of the vehicle body is allowed to be $1\sim2$ Hz [26], based on which the search range of suspension system stiffness can be set. In addition, by referring to the determination method of the critical damping coefficient of the suspension system given by Liu et al. [26], the maximum damping coefficient and minimum damping coefficient allowed can be set. That is, $C_c = 2\pi\sqrt{mk}$, where $m$ is the sprung mass equivalent to each suspension system and $k$ is the suspension stiffness. On this basis, the objective function of each index of the suspension system and the allocation optimization objective function of multi-index fusion are determined according to the method described in Section 3.2. Finally, the optimal allocation of dynamic parameters of the suspension system is obtained by using the iterative solution of the multi-index fusion objective function. At the same time, because the optimization involves multiple suspensions and many parameters, it is difficult to obtain the optimal allocation result by means of matching. Therefore, artificial intelligence algorithm-particle swarm optimization is introduced in the iterative solution process [37–39].

4. Case Analysis of Allocation Optimization

According to the optimization method of suspension dynamics parameter allocation mentioned above, a tracked vehicle is taken as an example for analysis. The main parameters are shown in Table 5. The related algorithm programming involved in this study is completed under the
When optimizing on the Gobi road, and the speed is set as 50 km/h. In order to compare the difference between equal weight allocation optimization and expert knowledge-based weight allocation optimization, the two weight allocation methods are used for optimization, and fixed stiffness and damping were used as the benchmark, where the suspension stiffness and damping are not changed. The optimized fitness curve is shown in Figure 13; both converge after iterating to about 120 steps, and there was very little difference in the optimized fitness value. Therefore, the optimized fitness value does not indicate which weight allocation method is more effective.

The performance index of the suspension system before and after optimization is shown in Table 3, and the corresponding response curve is shown in Figure 14. Figure 14(a) is the acceleration of the vehicle body above the first load-bearing wheel; Figure 14(b) is the dynamic displacement of the first load-bearing wheel relative to the vehicle body; Figure 14(c) is the dynamic velocity of the first load-bearing wheel relative to the vehicle body; Figure 14(d) is the pitch angular acceleration of the vehicle body; and Figure 14(e) is the dynamic load of the first load-bearing wheel relative to the ground. In Table 3, “reduction ratio” represents the improvement degree of the suspension

![Vibration damping performance evaluation](image)

**(Figure 12:** The process of dynamic parameter allocation optimization of the suspension system.)

**Table 4:** Range of dynamic parameters of the tracked vehicle suspension.

| Parameter | Unit       | Ranges            |
|-----------|------------|-------------------|
| $k_{ij}$  | N.m$^{-1}$ | $8.16 \times 10^4$ - $3.39 \times 10^5$ |
| $c_{ij}$  | N.s.m$^{-1}$ | $4.08 \times 10^3$ - $1.65 \times 10^5$ |

MATLAB environment. When optimizing on the Gobi road, and the speed is set as 50 km/h. In order to compare the difference between equal weight allocation optimization and expert knowledge-based weight allocation optimization, the two weight allocation methods are used for optimization, and fixed stiffness and damping were used as the benchmark, where the suspension stiffness and damping are not changed. The optimized fitness curve is shown in Figure 13; both converge after iterating to about 120 steps, and there was very little difference in the optimized fitness value. Therefore, the optimized fitness value does not indicate which weight allocation method is more effective.

The performance index of the suspension system before and after optimization is shown in Table 3, and the corresponding response curve is shown in Figure 14. Figure 14(a) is the acceleration of the vehicle body above the first load-bearing wheel; Figure 14(b) is the dynamic displacement of the first load-bearing wheel relative to the
Table 5: Main parameters of the tracked vehicle.

| Parameter | Unit     | Values  |
|-----------|----------|---------|
| $m_1$     | kg       | 24,500  |
| $I_1$     | kg.m$^2$ | 75,564  |
| $I_2$     | kg.m$^2$ | 216,531 |
| $m_0$     | kg       | 332     |
| $l_{21}/k_{7}$ | m | 2.462   |
| $l_{23}/k_{8}$ | m | 1.486   |
| $l_{23}/k_{9}$ | m | 0.556   |
| $l_{24}/l_{210}$ | m | -0.324  |
| $l_{25}/l_{111}$ | m | -1.024  |
| $l_{26}/l_{122}$ | m | -1.764  |
| $l_{y1-6}$ | m | 1.30    |
| $l_{y7-12}$ | m | -1.30   |
| $k_0$     | N.m$^{-1}$ | 6,597,421 |

Figure 14: Comparison for the response signals (on Gobi road).
system performance index. The greater the value, the better the improvement effect. The calculation expression of the reduction ratio is: \( D = d_i / P_d \), where \( d_i \) is the difference between the corresponding index quantization value of fixed stiffness and damping and the corresponding index quantization value after allocation optimization and \( P_d \) is the corresponding index quantization value of fixed stiffness and damping.

Table 3 and Figure 14 show that when each performance index is optimized by equal weight allocation, except that performance index 4 is not significantly improved, which has a decrease rate of 1.65% and no obvious improvement, but other performance indexes of the suspension system have been improved to varying degrees, among which performance index 3 has the most obvious improvement with a decrease rate of 23.51%. When the expert knowledge-based weight allocation optimization is adopted, the weight coefficients of performance indexes 1 and 4 are high, and the improvement effect of these two indexes is more obvious than that of the equal weight allocation optimization. Among them, the reduction ratio of performance index 1 increased by 4.73%. The decrease rate of performance index 4 changed the most obviously, increasing by 13.53%. However, the improvement effect of performance indexes 2, 3, and 5 is weakened, and the reduction ratio of these three indexes is about 4%. By using the expert knowledge-based weight allocation optimization, the performance indexes 1 and 4 with high weight have more obvious improvement, but the improvement effect of other performance indexes will be weakened. The weight can be set according to the suspension design requirements, and the weight coefficient of the index that is focused on will be increased, and the improvement effect will be more obvious.

Considering that the above allocation optimization is carried out when the vehicle speed is 50 km/h and the road surface is Gobi Road, further analysis of the performance characteristics under other typical working conditions is needed to evaluate the optimization effect more fully. For this reason, different speeds and different road surfaces (Gobi road, gravel road, paved road, and rough road) are selected for analysis, and the results are shown in Figures 8–11. The faster the vehicle speed is, the greater the quantified value of the corresponding performance index of the suspension system is, indicating the worse the vehicle traveling state is. The quantitative value of the suspension system performance index is larger than that of Gobi road and paved road when the tracked vehicle is traveling on gravel road and rough road, indicating that the vehicle is in a worse traveling state on gravel road and rough road.

As shown in Figure 8, when using equal weight allocation optimization on Gobi road, performance indexes 2, 3, and 5 are significantly improved, while performance indexes 1 and 4 are not significantly improved. Especially, when the traveling speed is 60 km/h or higher, the performance index 4 is slightly larger 5.26% than that of fixed stiffness and damping, indicating that performance index 4 is getting worse. Meanwhile, with the expert knowledge-based weight allocation optimization, the performance indexes 1 and 4 with high weight are significantly improved when the tracked vehicle travels at different speeds. When the traveling speed is 60 km/h, index 1 decreased by 24.83%, and index 4 decreased by 13.14%. However, the improvement effect of performance less concerned index 2 is obviously weakened. As shown in Figure 9, by using equal weight allocation optimization on gravel roads, when the traveling speed is 50 km/h, the performance index 4 begins to be larger than that of fixed stiffness and damping, and the higher the traveling speed, the more obvious it is; it shows that the use of equal weight allocation under this driving condition cannot obtain the ideal allocation optimization. At the same time, when the expert knowledge-based weight allocation optimization is adopted, the performance indexes 1 and 4 with high weight is significantly improved and reduced 26.63% and 15.91%, respectively, while the performance index 4 with high weight is not significantly improved compared with the performance index 1.

As shown in Figure 10, when the equal weight allocation optimization is adopted on paved roads, the improvement effect of performance index 2 is not obvious, but other performance indexes are improved to varying degrees. However, when the traveling speed reaches 60 km/h or higher, the performance index 4 is not significantly improved. Meanwhile, by using the expert knowledge-based weight allocation optimization, the performance indexes 1 and 4 with high weight are significantly improved, which is the same as traveling on the Gobi road surface, and the improvement effect of performance index 2 is obviously weakened. As shown in Figure 11, when the equal weight allocation optimization is adopted in the rough road surface, all performance indexes are improved to varying degrees. Among them, performance indexes 2 and 5 are the most significantly improved, followed by indexes 1 and 3. However, when the traveling speed reaches 40 km/h or higher, the improvement of performance index 4 is less and less obvious. When it reaches 70 km/h, the performance index 4 does not improve basically. At the same time, by using the expert knowledge-based weight allocation optimization, the performance indexes 1 and 4 with high weight have significant improvement, and the faster the traveling speed, the more obvious the improvement. But it weakens other performance indexes, especially indexes 3 and 5.

Through the comparative analysis of the tracked vehicle traveling on different road surfaces, and at different speeds, it is shown that the dynamic parameter allocation optimization method proposed in this study has a good optimization effect on the comprehensive performance of the suspension system. In this paper, the allocation optimization of multi-axis suspension dynamic parameter for tracked vehicles is carried out on Gobi road with a speed of 50 km.h\(^{-1}\). The suspension system of the tracked vehicle still has good damping performance when it is running on other roads and at different speeds. However, this parameter allocation result may not be the optimal one for other road surfaces and different vehicle speeds. For the suspension performance of tracked vehicles, such as military combat tanks, the adaptability of the driver and the firing accuracy of the fire control system should be important considerations. Therefore, indexes 1 and 4 need to be paid more attention,
and the corresponding weight is higher, which is more in line with the actual circumstances. The allocation optimization method is proposed that can also be applied to wheeled vehicles and even optimization problems in other professional fields.

5. Conclusions

The suspension performance of the tracked vehicle has an important influence on the comfort of passengers, the safety of traveling, and the reliability of onboard equipment. In this study, the optimization design of dynamic parameter allocation of the tracked vehicle multi-axle suspension is carried out for multiple performance indexes. Considering the vertical vibration excitation caused by uneven road surface, the input (excitation) model of road roughness is embedded to establish the parametric dynamic model of the tracked vehicle in this paper. Then, the evaluation index and its quantification algorithm are proposed to reflect the suspension performance of the tracked vehicle in many aspects, and the objective function of parameter allocation for multi-index information fusion is designed.

Taking the tracked vehicle traveling at a speed of 50 km/h on Gobi road as a typical working condition, the equal weight allocation optimization and the expert knowledge-based weight allocation optimization are implemented. The results show that when equal weight allocation is adopted, except for the performance index 4, other performance indexes of the suspension system are improved to varying degrees, and the performance index 3 is the most obvious improvement. When the expert knowledge-based weight allocation optimization is adopted, the improvement effect of performance indexes with a high weight coefficient is more obvious, but the improvement effect of other performance indexes is weakened. Through the comparative analysis of different road surfaces and different speeds, it is shown that the proposed optimization method of dynamics parameter allocation of the tracked vehicle suspension system has a good effect on the comprehensive performance of the suspension system under a variety of typical working conditions. As the speed of tracked vehicles increases, the effect of allocation optimization is better. However, when driving at low speeds on different roads, optimizing the dynamic parameters of the tracked vehicle’s suspension system has no difference in damping performance.

Due to the influence of different types and different vehicle speeds, the incentives for tracked vehicles are significantly different. Therefore, in the future, the optimization of the stiffness and damping parameter allocation of the suspension system under each working condition will be further studied, which has more practical significance. It is very interesting to decide which stiffness and damping allocation should be based on different road surfaces and different vehicle speeds.

Data Availability

The data used to support the findings of this study are included within the article.

Conflicts of Interest

The authors declare that there are no conflicts of interest.

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