

Research Article

Multiobjective Optimization of Vehicle Handling and Stability Based on ADAMS

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In order to improve the steering characteristics, roll characteristics, and lateral characteristics of heavy commercial vehicles, a multiobjective handling stability comprehensive score optimization model (MHSCS optimization model) was proposed in this study. In this study, the main factors affecting the steering characteristics, roll characteristics, and lateral characteristics are determined by combining the road test method and ADAMS software simulation analysis method. Based on the above road tests and ADAMS software simulation analysis, the MHSCS optimization model was proposed with “understeering degree,” “vehicle roll angle,” and “rearward amplification (RWA)” as evaluation indexes, and the response surface method combined with genetic algorithm was used to carry out multiobjective optimization of heavy commercial vehicle. ADAMS simulation results show that the comprehensive improvement degree of steering characteristics, roll characteristics, and lateral characteristics of a heavy commercial vehicle after optimization is 15.26%. Finally, the field road test results show that the scoring error between the comprehensive scoring optimization model and the real vehicle test was controlled at 0.4%, which proves the accuracy of the optimization model established in this study and effectively improves the handling stability of heavy commercial vehicles.

1. Introduction

Handling and stability is very important for vehicle safety, and many vehicle structures and parameters will affect the operating stability [1]. In order to improve the vehicle handling stability, the vehicle handling stability optimization method based on d-Optimal experimental design, which established a response surface model based on D-Optimal experimental design was proposed by Li et al. [2]. This method is only better effective in improving the steering performance of the vehicle, but the ability to improve the roll characteristics and lateral characteristics of the vehicle is not too obvious effect. An improved genetic particle swarm optimization algorithm was used to optimize the vehicle handling stability. In view of the requirements of a four-wheel steering system on vehicle stability control and the existence of uncertainties, the hybrid H₂/H_∞ robust control method of a four-wheel steering system improves the handling stability of 4WS vehicles, which was proposed by

Xu et al. [3]. This method has a good effect in improving the stability and robustness of four-wheel steering, but this method also has no way to improve the handling stability of the vehicle with multiple objectives. To solve the problem of poor handling stability of a car, Zhang et al. used the response surface method combined with a unified objective method to carry out multiobjective optimization of the handling stability of the car, so as to improve the handling stability of the car [4]. However, this method only provides multiobjective optimization in the roll characteristics and steering performance of the vehicle and does not take into account the roll characteristics and ability optimization of the vehicle. Deng et al. conducted simulation optimization on handling stability of FSAE racing Car based on ADAMS/Car and carried out multi-objective optimization design for the optimization objectives of front wheel positioning parameters and roll center height, which improved the steady-state response characteristics of the vehicle [5].

Because heavy commercial vehicles have the characteristics of high centroid and large inertia, the steering characteristics, roll characteristics, and lateral characteristics are particularly important to the steering performance and rollover resistance of heavy commercial vehicles. Therefore, it is necessary to optimize the steering, roll, and lateral characteristics of heavy commercial vehicles.

The purpose of this study is to establish a multiobjective handling stability comprehensive score optimization model. This model is an optimization model of the multiobjective manipulation stability comprehensive score with “understeering degree,” “vehicle roll angle,” and “rearward amplification (RWA)” as the evaluation indicators. Meanwhile, using the optimization model established, ADAMS simulation and road real vehicle test were used to optimize the steering characteristics, roll characteristics, and lateral characteristics of heavy commercial vehicles, which has high engineering practical value for improving the control stability of heavy commercial vehicles.

2. Validation of the Multibody Dynamics Model

2.1. Vehicle Multibody Dynamic Model Construction. Using ADAMS/Car software and combined with the dimension parameters and mechanical parameters of a heavy commercial vehicle, each vehicle subsystem was established and completed the assembly of the vehicle [6, 7]. The front suspension of the heavy commercial vehicle adopts a leaf-spring rigid axle suspension, the steering bridge adopts an integral structure, and the steering system adopts a recirculating ball type steering system. In order to make the multibody dynamic model established in this study more close to the real vehicle state and make the calculation result more accurate, the flexible body is adopted in this study [8]. Hyperwork software was used to generate MNF neutral files. The component modal synthesis method was used to deal with the problem that the finite element model has too many degrees of freedom.

Figure 1 shows the overall modal diagram of the frame, and its modal value is close to that of the real frame. Finally, the generated flexible body frame was imported into Adams/Car module to build the vehicle model, as shown in Figure 2.

2.2. Load Verification. To ensure that the built vehicle model can truly reflect the real vehicle system, the virtual prototype simulation model should be verified before the vehicle simulation analysis [4, 5]. The accuracy of each axle load is the basic of the vehicle multibody dynamic model, so it is necessary to verify the load. The axle load test platform of the real vehicle and model simulation is shown in Figure 3.

It can be seen from Table 1 that the multibody dynamic model established in this study is similar to the test results of the real vehicle. The loading error of tractor 1 axle is 0.02%, and that of tractor 2 + 3 axle is 0.1%. The 1-axis load error of a full load with a trailer is 0.3%, 2 + 3 axle load error is 0.2%, and 4 + 5 + 6 axle load error is 0.1%. The error of the vehicle model is controlled within 0.3%, which proves that the multibody dynamic model established in this study is very accurate in loading.

2.3. Handling Stability Working Condition Verification. In order to prove the multiobjective comprehensive score optimization model (MHSCS optimization model) proposed in this study has an optimization effect on the handling stability of heavy commercial vehicles, it is necessary to verify the multibody dynamics model established in this study. In this study, real vehicle and multibody dynamic models are compared under the same conditions of steady-state rotation test and single-lane transformation test to verify the accuracy of the model. Installation position of control stability test equipment is shown in Figure 4.

In order to more truly reflect the performance of steering characteristics, roll characteristics, and transverse characteristics of heavy commercial vehicles, this study selects classic understeering degree, vehicle roll angle, and rearward amplification (RWA) as evaluation indexes according to the standard GB/T 623–2014 [9]. Therefore, the steady-state rotation test and single-lane change test were carried out according to the current national standard vehicle handling and stability test method, and the experimental results were compared with the simulation results of the multibody dynamics model under the same working conditions. The comparative analysis results are shown in Figure 5.

Considering the complexity of the vehicle system and the uncertainty of actual test conditions, the consistency of the two outputs was defined as: the fluctuation trend of output curves is roughly the same, and the magnitude of peak values is the same. Local relative error according to different working conditions, the absolute error of simulation value, and corresponding test value should be less than 10% [10]. The error values were calculated by the following formula:

$$E = \frac{100 \times |(SV - TV)|}{TV} < 10\%, \quad (1)$$

where E is the error rate (%). SV is the simulated value. TV is the test result value.

According to formula (1) and the calculation of simulation and test results, the error value B of real vehicle test and multibody dynamics model simulation is sorted out as shown in Table 2.

As can be seen from Table 2 and Figure 5, the multibody dynamics model established in this study can truly reflect the state of the real vehicle under steady-state rotation and single-lane transformation test conditions. In the evaluation index of steering characteristics, the error of understeering degree U is 4.8%, the error of roll characteristics evaluation index of body roll degree K_ϕ is 0.8%, and the error of RWA of lateral characteristics evaluation index is 2.61%. The error of each evaluation index is controlled within 10%. It is proved that the multibody dynamic model established in this study is very accurate in handling stability.

3. Validation of the Multibody Dynamics Model

3.1. Optimization Goal. The evaluation method of handling stability is the main method to study the performance of vehicle handling stability [11]. In this study, an optimization model of a multiobjective handling stability comprehensive score optimization model (MHSCS optimization model) is established according to the handling stability evaluation

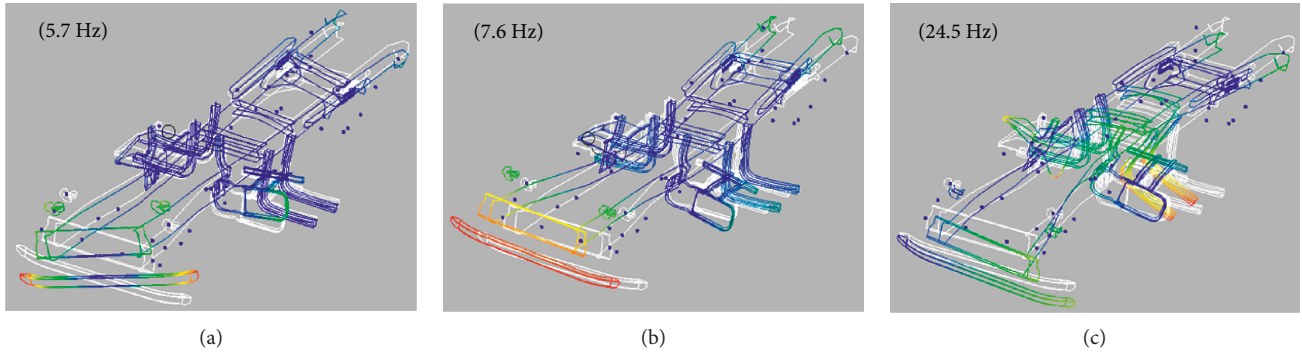


FIGURE 1: (a) Order 1 torsion mode of the flexible body frame. (b) Order 1 transverse bending mode of the flexible body frame. (c) Order 1 vertical bending mode of the flexible body frame.

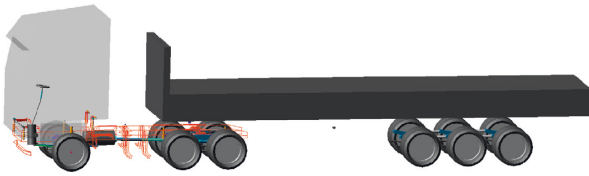


FIGURE 2: Vehicle multibody dynamics model.

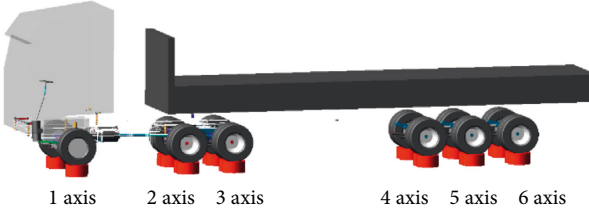


FIGURE 3: Load test and verification of each axle of the vehicle.

method given by standard QC/T 480–1999 [12]. According to the handling stability evaluation method given by QC/T 480–1999, the scoring model of the item of vehicle roll angle K_ϕ is as follows:

$$N_\phi = 60 + \frac{40}{K_{\phi 60} - K_{\phi 100}} \times (K_{\phi 60} - K_\phi), \quad (2)$$

where N_ϕ is the score value of vehicle roll angle. $K_{\phi 60}$ is the upper limit of the vehicle roll angle ($K_{\phi 60} = 1.2^\circ/(\text{m/s}^2)$). $K_{\phi 100}$ is the lower limit of the vehicle roll angle ($K_{\phi 100} = 0.7^\circ/(\text{m/s}^2)$).

Due to QC/T 480–1999, there is no evaluation method for rear amplification factor. Therefore, based on the single score model of rearward amplification H_δ proposed in the literature [13], this study determines:

$$N_\delta = 60 + \frac{40}{H_{\delta \max} - H_{\delta \min}} \times (H_{\delta \max} - H_\delta), \quad (3)$$

where N_δ is the score value of rearward amplification (RWA). $H_{\delta \max}$ is the maximum value of the index in the test design sample ($H_{\delta \max} = 1.932$). $H_{\delta \min}$ is the minimum value of the index in the test design sample ($H_{\delta \min} = 0.812$).

According to the handling stability evaluation method given by QC/T 480–1999, the scoring model of the item of understeering degree U is as follows:

$$N_u = 60 + \frac{40}{(U_{100} - U_{60})} \times (U_{100} - U), \quad (4)$$

where N_U is the score value of understeering degree. U_{60} is the maximum value of the index in the test design sample ($U_{60} = 0.731^\circ/(\text{m/s}^2)$). U_{100} is the minimum value of the index in the test design sample ($U_{100} = 0.072^\circ/(\text{m/s}^2)$).

Each index has a different influence on vehicle handling stability, that is, different contribution rates. In this study, the contribution rate of each objective evaluation index calculated in reference to [14] was used to determine the weight of each index. Finally, the comprehensive evaluation score model of heavy commercial vehicle handling stability optimization is as follows:

$$\begin{aligned} N_E &= 0.429N_U + 0.232N_\phi + 0.339N_\delta \\ &= 124.69 - 26.04U - 18.56K_\phi - 12.12H_\delta, \end{aligned} \quad (5)$$

where N_E is the comprehensive score of heavy commercial vehicle handling stability optimization.

The objective of this study is to maximize the comprehensive optimization score of heavy commercial vehicle handling stability.

3.2. Optimize Variables and Constraints. The optimization variables of the optimization model of heavy commercial vehicles refer to the variable parameters that affect the optimization design results, and the general principle of selection should be the parameters that affect the handling stability of heavy commercial vehicles [15]. According to the above test and simulation results, front suspension

TABLE 1: Comparison of real vehicle test axle load and simulated axle load.

	1 axis of the tractor (kg)		2 + 3 axis of the tractor (kg)		4 + 5 + 6 axis of the trailer (kg)	
	Simulation	Test	Simulation	Test	Simulation	Test
Tractor	4889	4890	3786	3790		
Full load with trailer	5621	5640	14751	14720	24928	24960



FIGURE 4: The installation position of test equipment.

spring stiffness K_f , tire lateral stiffness K_t , and front wheel toe angle β_T were selected as the design variables of this optimization.

According to the optimization mathematical model, the design variables in this study can be expressed as:

$$x = (K_f, K_t, \beta_T)^T. \quad (6)$$

In order to ensure the accuracy of the optimization result, the value range of the optimization variable should not be set too large, because too large or too small range will cause the distortion of the fitting model, and the accuracy of the optimization result cannot be guaranteed. This study makes constraints on optimization variables as shown in Table 3. At the same time, in order to facilitate the calculation and processing in the optimization design, the tire lateral stiffness K_t and the front suspension spring stiffness K_f were multiplied by the scale factor λ , and the initial value of λ was set as 1.

3.3. Optimize Variables and Constraints. The advantage of the response surface model is that the experimental random error is taken into account and the calculation is relatively simple [1]. Therefore, the optimization process in this study adopts the response surface method, and the quadratic polynomial mathematical model of the response surface is as follows:

$$y = \beta_0 + \sum_{i=1}^n \beta_i x_i + \sum_{i=1}^n \sum_{j=1}^n \beta_{ij} x_i x_j + \sum_{j=1}^n \beta_j x_j^2, \quad (7)$$

where y is the response value. x_1, x_2, \dots, x_n is the variable factor. β_0 is the constant term. β_i are polynomial coefficients respectively.

It is a key step to use the response surface method to select test design in the optimization process, and sample points and response values need to be established in the process of test design. In this study, response values are

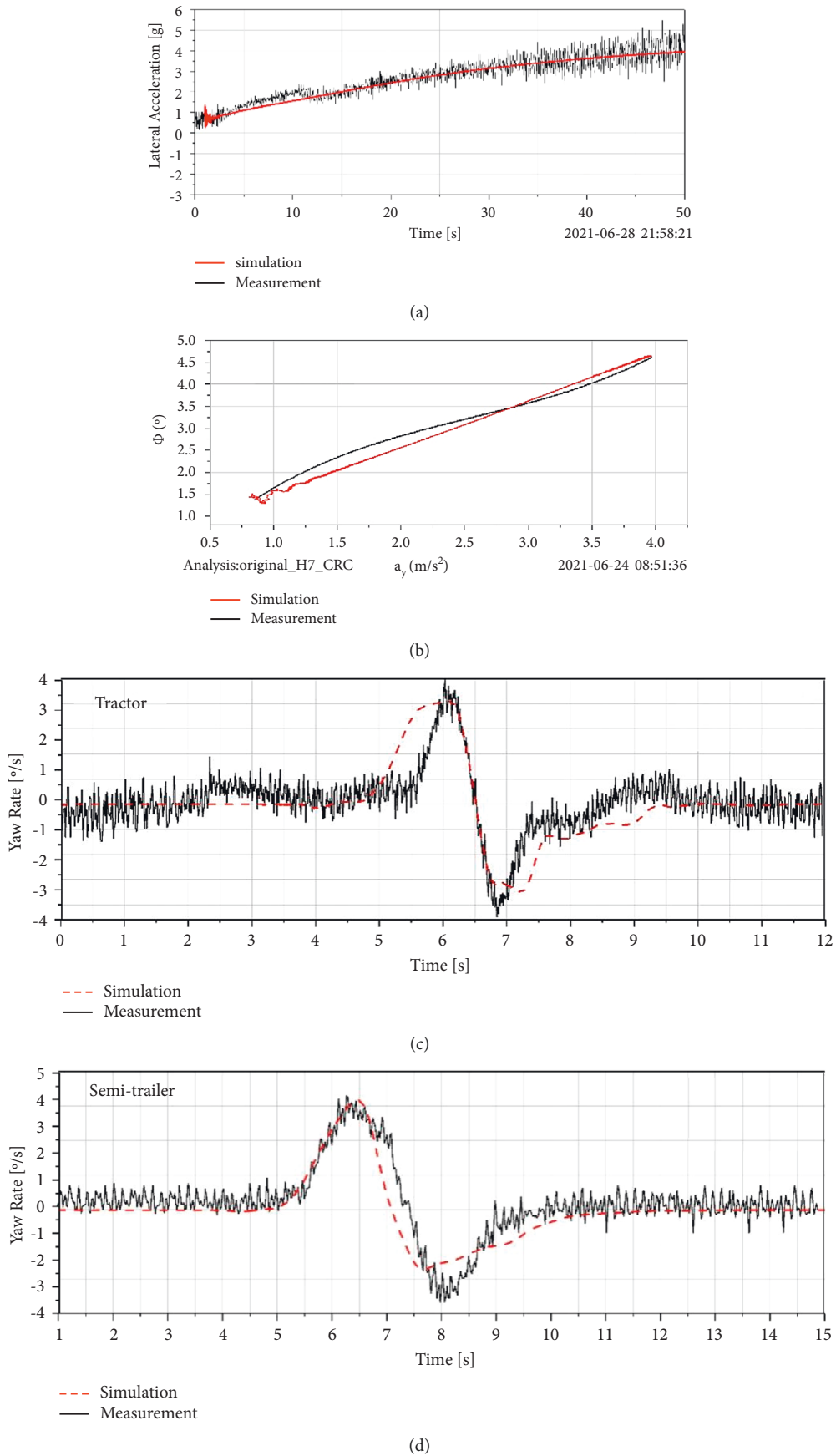


FIGURE 5: (a) Comparison of lateral acceleration test and simulation results. (b) Comparison of roll Angle test and simulation results. (c) Comparison between yaw of tractor test and simulation results. (d) Comparison between yaw of semi-trailer test and simulation results.

TABLE 2: Error values of real vehicle test and multibody dynamics model simulation.

Evaluation index	Test value	The simulation value	Error rate (%)
Understeer U	0.42	0.40	4.80
Vehicle roll angle K_φ	1.26	1.25	0.80
RWA	1.15	1.12	2.61

TABLE 3: Range of values of the variable factors.

Variable factors	Variable name	Initial value	Value range
λ_{K_f}	Front suspension spring stiffness	1	0.8~1.2
λ_{K_t}	Tire lateral stiffness	1	0.8~1.2
λ_{β_T}	Front wheel toe angle	0	-1.0~1.0

obtained through the multibody dynamics model established above based on ADAMS simulation. For experimental design, Latin Hypercube Design (LHD), which has better uniformity, was used for each level of design variables and has the same number of tests for each design variable, is used in this study. Then use ISIGHT software to extract 30 sample points, and conduct simulation based on ADAMS in order, and then export the simulation data in the simulation analysis post-processing module. The corresponding objective function values of each sample point are shown in Table 4.

Finally, MATLAB program is used to further process the simulation test data. Call up the Rstool toolbox in MATLAB, and use the Quadratic Full model to fit the scale coefficients and target values of variable factors in Table 4. The fitting agent model is as follows:

$$U = 1.13588 + 1.01807\lambda_{K_f} + 1.566884\lambda_{K_t} - 0.05679\beta_T - 0.95761\lambda_{K_f}^2 - 0.22346\lambda_{K_t}^2 - 0.05608\beta_T^2 - 0.54364\lambda_{K_f}\lambda_{K_t} - 0.32057\lambda_{K_f}\beta_T + 0.02659\lambda_{K_t}\beta_T, \quad (8)$$

$$K_\varphi = 0.81347 + 0.32165\lambda_{K_f} - 0.26756\lambda_{K_t} - 0.05679\beta_T + 0.03954\lambda_{K_f}^2 + 0.05689\lambda_{K_t}^2 - 0.02341\beta_T^2 - 0.26541\lambda_{K_f}\lambda_{K_t} - 0.32057\lambda_{K_f}\beta_T - 0.00569\lambda_{K_t}\beta_T, \quad (9)$$

$$H_\delta = 0.79567 + 0.39865\lambda_{K_f} - 0.76531\lambda_{K_t} - 0.06895\beta_T + 0.02637\lambda_{K_f}^2 + 0.08943\lambda_{K_t}^2 - 0.03016\beta_T^2 - 0.41026\lambda_{K_f}\lambda_{K_t} - 0.26734\lambda_{K_f}\beta_T - 0.00801\lambda_{K_t}\beta_T. \quad (10)$$

The residual errors of the above fitting equations were 0.0234, 0.0452, and 0.0185, respectively. Therefore, the fitted G_U , G_K , and G_β are reliable. Formulas (8)–(10) above were substituted into formula (5) to obtain the final optimization model of multiobjective operational stability comprehensive score:

$$N_E = 70.37016 - 37.312\lambda_{K_f} - 26.56019\lambda_{K_t} + 3.36851\beta_T + 23.8827\lambda_{K_f}^2 + 3.67913\lambda_{K_t}^2 + 2.26035\beta_T^2 + 24.05475\lambda_{K_f}\lambda_{K_t} + 17.53758\lambda_{K_f}\beta_T - 0.48972\lambda_{K_t}\beta_T. \quad (11)$$

3.4. Optimal Solution. Genetic Algorithm (GA) has the advantages of strong global search ability and is widely used in complex problems such as planning. Therefore, this study uses the genetic algorithm GA to optimize the solution. Table 3 is taken as the value range of the design variables; formula (11) is taken as the optimization objective function. MATLAB is used to write the corresponding genetic algorithm optimization program to solve the optimal value of NE, that is, the maximum value, and obtain the optimal value of K_f , K_t , and β_T values. Finally, the optimal value is solved as shown in Table 5.

The optimized variable factor was converted into the actual value to obtain the front suspension spring stiffness, tire sideslip Angle stiffness, and front wheel beam Angle, as shown in Table 6.

4. Simulation Comparison before and after Optimization

According to the values in Table 6, the multibody dynamic model established in this study is used for steady-state rotary test simulation and single-lane transformation test simulation. The results of comparison between the difference of front and rear axle sideshow angles (δ_1 - δ_2) and the lateral acceleration characteristic curves before and after optimization under steady-state rotary test conditions were obtained, as shown in Figure 6, and the vehicle roll Angle and the lateral acceleration characteristic curves, as shown in Figure 7. Comparison results of yaw velocity time domain curves of tractor and semi-trailer before and after optimization under single-lane changing conditions are shown in Figure 8.

According to the standard QC/T 480-1999, the curves in Figures 6-8 were processed and the scores of evaluation indexes before and after optimization were obtained, as shown in Table 7.

As can be seen from Tables 5 and 7, the optimization results of the multiobjective comprehensive stability scoring optimization model established in this study are very consistent with the results of the multibody dynamics simulation analysis, and the error was controlled within 0.97%, which proves the accuracy of the multiobjective comprehensive stability scoring optimization model established in this study. It is of practical engineering significance to improve the steering characteristics, roll characteristics, and transverse characteristics of heavy commercial vehicles. However, a large number of test samples are needed in the process of establishing the optimization model, which is also the limitation of the MHSCS optimization model.

TABLE 4: Target function values corresponding to each sample point.

Simulation serial number	λ_{K_f}	λ_{K_t}	λ_{β_T}	RAW H_δ	Vehicle roll angle K_ϕ	Understeer degree U
1	0.997	1.180	0.116	1.071	1.289	0.463
2	0.990	0.827	-0.748	1.082	1.234	0.18
3	1.010	1.112	0.510	1.162	1.298	0.418
4	1.064	0.963	0.538	1.043	1.266	0.413
5	0.868	1.159	-0.775	1.16	1.281	0.243
6	1.186	0.990	0.749	1.106	1.295	0.553
7	1.044	0.936	0.478	1.158	1.276	0.322
8	0.983	1.092	0.914	1.063	1.273	0.388
9	1.017	1.017	-0.429	1.067	1.265	0.381
•	•	•	•	•	•	•
•	•	•	•	•	•	•
•	•	•	•	•	•	•
26	1.125	0.997	0.653	1.108	1.289	0.479
27	0.807	1.071	0.302	1.101	1.245	0.075
28	0.814	1.105	-0.691	1.137	1.257	0.109
29	1.089	0.942	-0.649	1.071	1.269	0.425
30	0.888	0.895	0.478	1.09	1.224	0.072

TABLE 5: Optimized design variable and target values.

Variable/target function	Initial value	Optimize value
Front suspension spring stiffness K_f	1	1.16
Tire lateral stiffness K_t	1	0.83
Front wheel toe angle β_T	0	1
Composite score N_E	77.50	88.47

TABLE 6: Actual value of the optimized design variables.

Variable/target function	Initial value	Optimize value
Front suspension spring stiffness K_f (N/mm)	220	255.2
Tire lateral stiffness K_t (N/mm)	4000	3340
Front wheel toe angle β_T (deg)	0	1

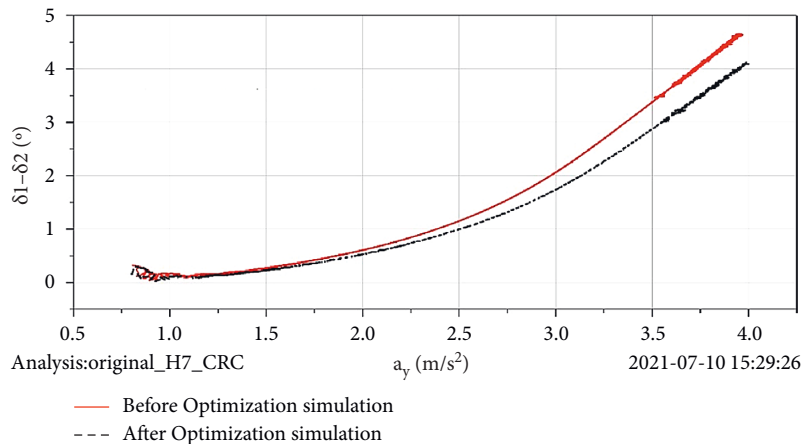


FIGURE 6: Comparison of the relationship curves between a_y and $(\delta_1 - \delta_2)$ before and after optimization simulation.

5. Real Vehicle Experiment Comparison before and after Optimization

In order to verify the optimization effect of the established multiobjective operation stability comprehensive scoring optimization model, a real vehicle verification was carried

out in this study. According to Table 6, replace the front suspension spring of a heavy commercial vehicle with a leaf spring with 255.2 N/mm stiffness, replace the tire with tire with 3340 N/mm lateral stiffness, and adjust the front wheel beam Angle to 1. Steady-state rotation and single lane change test items were carried out on the adjusted real

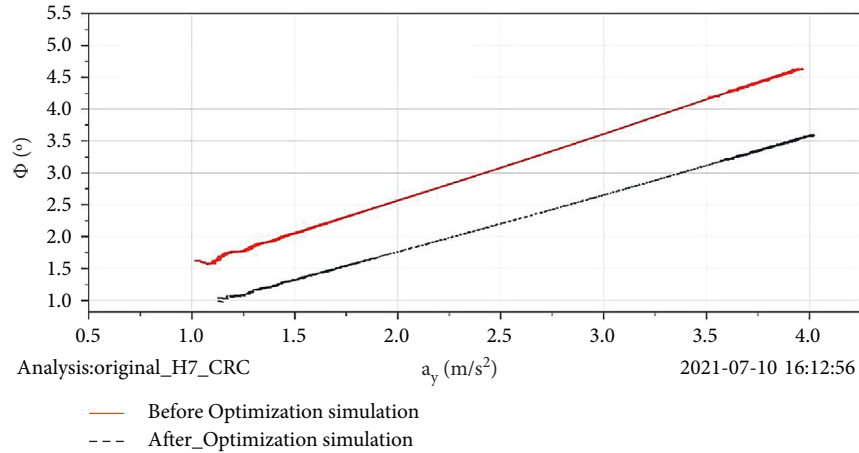


FIGURE 7: Comparison of the relationship curves between a_y and ϕ before and after optimization simulation.

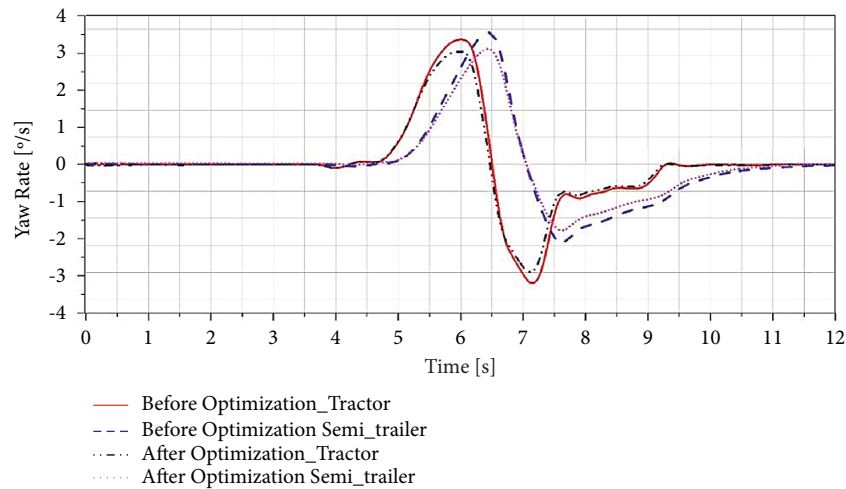


FIGURE 8: Comparison of the time domain curve of yaw velocity of tractor and semi-trailer before and after optimization simulation.

TABLE 7: Score values of the evaluation indicators before and after the optimization.

Evaluation index	After optimization (points)	Before optimization (points)	Improvement rate (%)
Understeer score N_U	84.34	87.85	4.16
Body roll angle score N_ϕ	56.00	88.51	58.05
RWA score N_δ	88.99	91.75	3.10
Composite score N_E	79.34	89.33	15.26



FIGURE 9: (a) Steady turn test site. (b) Single lane change test site.

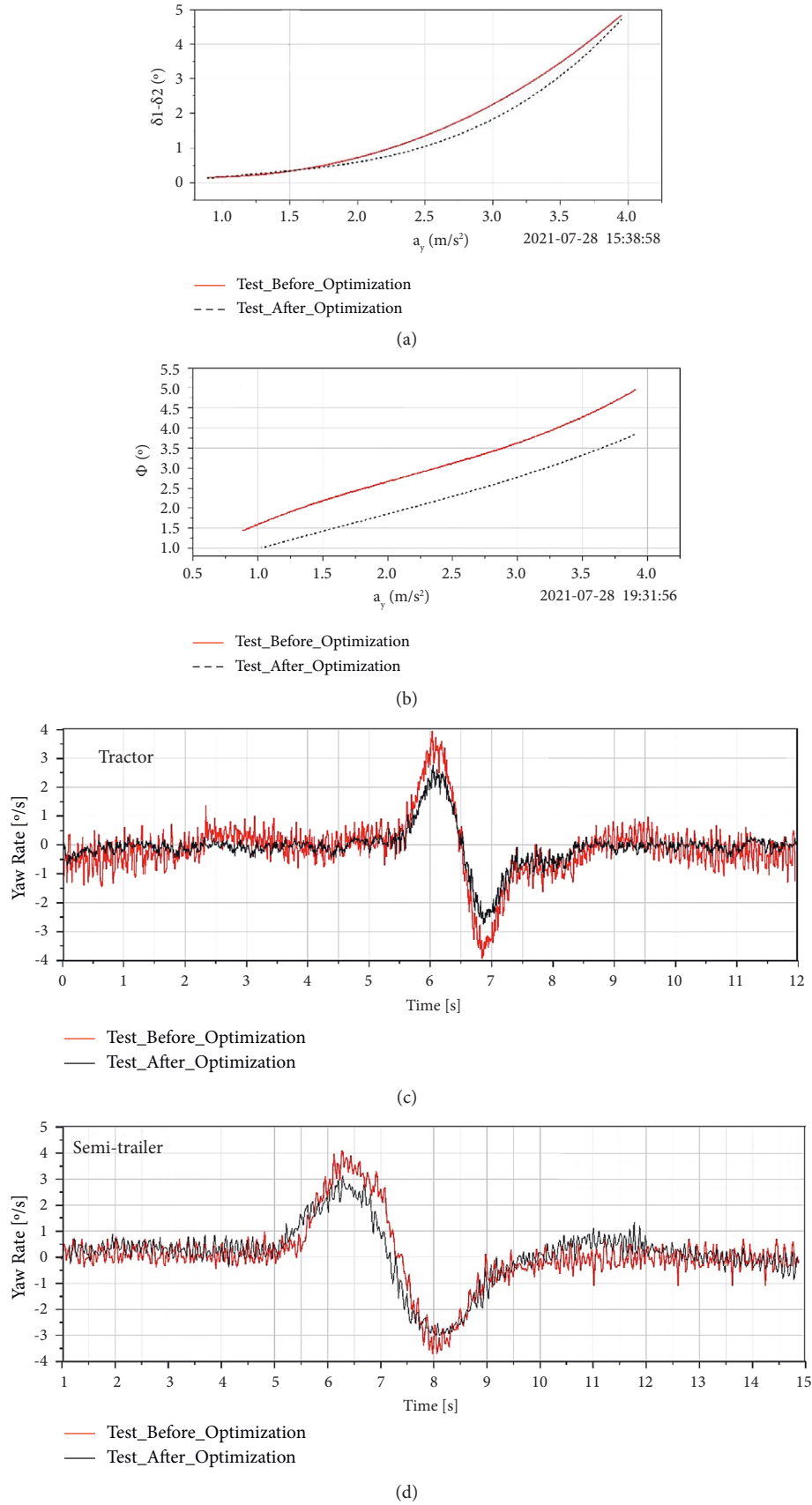


FIGURE 10: (a) Comparison of the relationship curves between a_y and $(\delta_1 - \delta_2)$ before and after optimization test. (b) Comparison of the relationship curves between a_y and ϕ before and after optimization test. (c) Comparison of the time domain curve of yaw velocity of tractor before and after optimization test. (d) Comparison of the time domain curve of yaw velocity of semi-trailer before and after optimization test.

TABLE 8: Optimize the evaluation index score value before and after the real vehicle test.

Evaluation index	MHSCS optimization model	Multi-body kinetic model	Real vehicle tests
Understeer score N_U (points)	87.85	87.85	86.32
Body roll angle score N_φ (points)	86.97	88.51	87.72
RWA score N_δ (points)	90.30	91.75	90.54
Composite score N_E (points)	88.48	89.33	88.10

vehicle to verify the optimization effect. The experimental site is shown in Figure 9.

The comparison of real vehicle test data before and after optimization is shown in Figure 10, and the comprehensive score comparison results of each model after sorting are shown in Table 8.

It can be seen from Table 8 that the comprehensive scoring error between the MHSCS optimization model established in this study and the real vehicle after optimization is only 0.4%, which verifies the accuracy of the optimization effect of the MHSCS optimization model established in this study. At the same time, after the optimization of the MHSCS optimization model established in this study, the improvement degree of steering characteristics of the heavy commercial vehicle is 2.3%, the improvement degree of roll characteristics is 36.2%, and the improvement degree of lateral characteristics is 1.7%. The comprehensive score of heavy commercial vehicle operation stability was improved by 12.0%, and the vehicle handling stability of the heavy commercial vehicle is effectively improved.

6. Conclusions

This study summarizes the current methods of commercial heavy duty handling stability optimization, and proposes a multiobjective handling stability comprehensive score optimization model (MHSCS optimization model), which takes “understeering degree,” “vehicle roll angle”, and “rearward amplification (RWA)” as evaluation indexes.

- (1) The accuracy of the multibody dynamic model was verified by comparing the simulation data with the experimental data of various axial load tests, steady-state turning tests, and single lane changing tests.
- (2) Latin Hypercube Design (LHD) and ADAMS multibody dynamics model were used to obtain the response values. Finally, the Quadratic Full module in MATLAB was used to fit the proportional coefficient of the variable factor and the target value, and the MHSCS optimization model was obtained.
- (3) The MHSCS optimization model was used to optimize the handling stability of heavy commercial vehicles. After optimization, the improvement degree of steering characteristics, roll characteristics, and lateral characteristics of heavy commercial vehicle is 2.3%, 36.2%, and 1.7%, respectively, and the comprehensive score improvement degree is 12.0%. Finally, the MHSCS optimization model established in this study is verified to have a comprehensive

scoring error of only 0.4% by a real vehicle experiment, which proves the accuracy of the MHSCS optimization model established in this study and the effectiveness of the optimization results.

- (4) The MHSCS optimization model proposed in this study is of practical engineering significance in improving the steering characteristics, roll characteristics, and lateral characteristics of heavy commercial vehicles. However, a large number of test samples are needed in the process of establishing the optimization model, and the initial test cost is relatively high. This is also the limitation of the optimization model.

Data Availability

The data used to support the findings of this 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 paper.

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