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Multi-objective Optimization of the Steering Linkages Considering Transmitting Ratios

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Abstract

Trapezoidal mechanisms and swing arm mechanisms in the steering linkages were usually designed separately, and only the steered wheel deflection errors were fixed as the optimization objective. To deal with these design faults, multi-objective optimization model of the steering linkages were developed in this paper. The model considered not only the steered wheel deflection errors but also the unevenness of the steering linkages transmitting ratios. In the model, the optimization variables were the link dimensions of the steering linkages and the optimization was processed in Adams software using SPQ algorithm. The optimization results show that the unevenness of the first axle steering mechanism transmitting ratio reduced from 9.7% *to* 4.9%*, the unevenness of the second axle steering mechanism transmitting ratio reduced from* 3.8% *to* 2.1%*, the maximum deflection error of the first axle right steered wheel reduced* 39.3%*, the maximum deflection error of the second axle left steered wheel reduced* 21.4%*, the maximum deflection error of the second axle right steered wheel reduced* 48.4%*. Benefitting from the unevenness improvement of the steering linkages transmitting ratios, the maximum difference between the bilateral deflection errors of the first axle right steered wheel reduced from* 1° *to* 0.5°*, the maximum difference between the bilateral deflection errors of the second axle left steered wheel reduced from* 4° *to* 0.8°*, the homologous maximum difference of the second axle right steered wheel reduced from* 6° *to* 0.3°*. For the handiness, the maximum difference of the bilateral steering forces reduced from* $1.44N\cdot m$ *to* $1.1N\cdot m$*, the maximum difference of the bilateral number of the steering wheel turns reduced from 0.15 to 0.11. For the lemniscate simulations, the difference between the positive and negative amplitude of the yaw angular velocity reduced from* $1.67^\circ/s$ *to* $0.58^\circ/s$*, the difference between the positive and negative amplitude of the lateral accelerometer reduced from* $83.3mm/s^2$ *to* $49.6mm/s^2$*. Because the unevenness of the steering linkages transmitting ratios was optimized, the steered wheel deflection errors tended to be bilateral symmetry and the vehicle handiness and control stability were both improved.*

Keywords*: Heavy duty truck, Steering linkages, Multi-objective optimization, Handiness, Lemniscate simulations*

1. Introduction

For well-designed steering linkages, the steering forces corresponding to the steered wheel left and right deflections should be approximately equal and that was the evenness

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of the vehicle steering force which was an important factor influencing the steering handiness [1]. Heavy duty trucks had big curb mass and hard steering characteristics. The unevenness of the steering force would greatly influence the vehicle handiness and control stability, and then influence the vehicle safety.

The main factor influencing the unevenness of the steering force was the design quality of the steering linkages, including the trapezoidal mechanisms and swing arm mechanisms, which were more important for the multi-axle steering linkages. The steering force was transferred to the steered wheels through the steering linkages, thus the unevenness of the steering force could be characterized by the unevenness of the steering linkages transmitting ratios. The optimization of the steering linkages transmitting ratios unevenness amounted to the optimization of the steering force unevenness. Traditional design method for the steering linkages only optimized the deflection errors of the steered wheels which did not consider the unevenness of the steering linkages transmitting ratios [2]. As a result, the steering force must be unevenness. Li Yumin [3] optimized the unevenness of the steering linkages transmitting ratios, but the model did not conclude the trapezoidal mechanisms and swing arm mechanisms at the same time. At present, there mainly existed two optimization model developing methods. One was the multi-body dynamic model (for example [4-5]) built in Adams software which could be easily built and be widely used for the vehicle simulations. The second was the strict mathematical model (for example [6]), which was complex and awkward.

From the above analysis, the transmitting ratios were important design parameters of the steering linkages. To start with in this paper, the multi-objective optimization model of the steering linkages considering the unevenness of the transmitting ratios was built based on the vehicle multi-dynamic model, by which the steering linkages were optimized. Secondly, the improvement effectiveness of the steering linkages transmitting ratios was verified though comparing the simulation results of the vehicle handiness and control stability before and after the optimization.

2. Research Subject and Model Coordinates

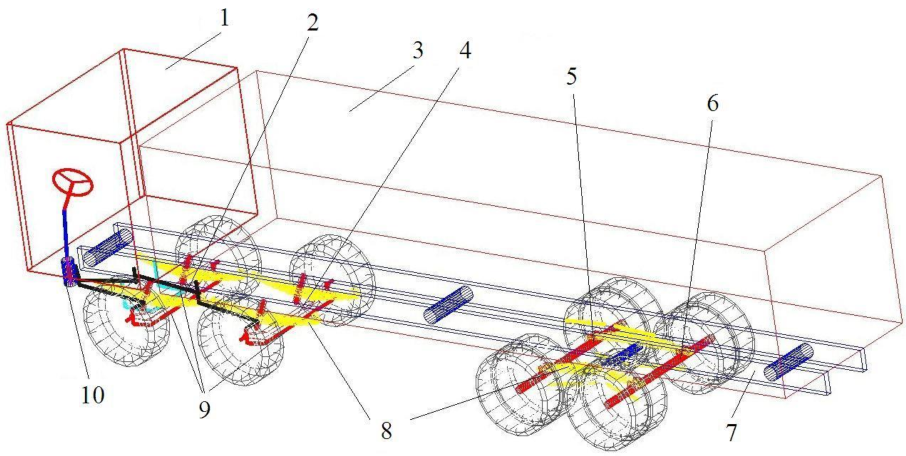

Figure 1. Adams Model of an 8×4 Type Heavy Duty Truck: 1-Cab; 2-First Axle; 3-Cargo; 4- Second Axle; 5-Third Axle; 6-Fourth Axle; 7-Frame; 8-Leaf Spring; 9-Steering Linkage; 10-Steering Gear

The parameterized Adams model of an 8×4 type heavy duty truck was shown in Figure 1, in which the links in the steering linkages were substituted using rods, the leaf spring models were built based on the beam element model [7] in Adams chassis, the propelling rods in equalizing suspension were also modelled using straight rods, the wheels were modelled using pac2002 model [8], the frame, cab and cargo models were created from the Pro/E models which were imported into Adams software, the engine and gearbox were substituted using mass points. In this paper, the optimization model of the double front axle steering linkages shown in Figure 1 was researched based on the vehicle model.

In general, the coordinates of the steering linkages optimization model should be the same as the vehicle multi-dynamic model which should have the same coordinates as the vehicle Pro/E model for convenience of gaining the model parameters. Therefore, the positive direction of axis x was the vehicle forward direction, the positive direction of axis y was the vehicle left direction, the positive direction of axis z was the straight up direction, and the origin of the coordinates o was located at the middle point of the first axle. The signs of the angular displacements of the links were ascertained according to the positive directions of each axis. Thus, anticlockwise direction was the positive direction and clockwise direction was the negative direction for the links.

3. Steering Linkages Optimization

3.1. Optimization Variables

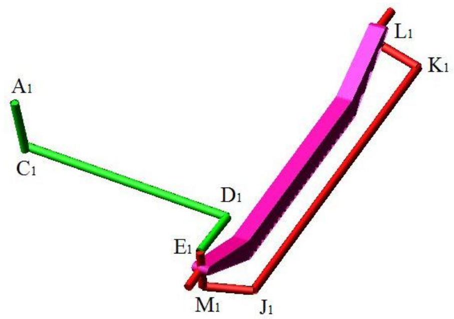

Figure 2. First Axle Steering Mechanisms

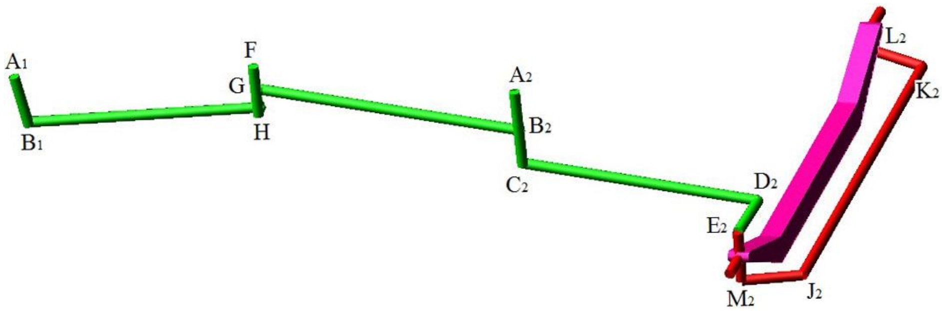

Figure 3. Second Axle Steering Mechanisms

In general, the design of the steering wheel, steering column and the steering gear are not included in the design process of the steering mechanisms which are designed in the vehicle design stage. Therefore the double front axle steering linkages included in Figure 1 consisted of the first axle steering mechanisms shown in Figure 2 and the second axle steering mechanisms shown in Figure 3. A_1, F and A_2 were kinematic pair stations connected the swing arms and frame. As shown in Figure 2, $A_1C_1D_1E_1$ was the first axle swing arm mechanisms, $D_1E_1M_1J_1$ was the first axle left steering knuckle, $M_1J_1K_1L_1$ was the first axle trapezoidal mechanisms. As shown in Figure 3, the transmission mechanisms from A_1B_1 to D_2E_2 were the second axle swing arm mechanisms, B_1 lied on A_1C_1, FH was middle swing arm, A_2C_2 was the second axle swing arm, $D_2E_2M_2J_2$ was the second axle left steering knuckle, $M_2J_2K_2L_2$ was the second axle trapezoidal mechanisms. The rotational displacement loaded on the steering wheel was transmitted to the first axle swing arm A_1C_1 through the steering gear, and then the steering linkages were driven.

The parameters of the steering linkages were listed in Table 1. After the vehicle overall design, the locations of A_1, F and A_2 were all ascertained. Therefore, the design variables included the lengths and initial angles of the swing arms, the lengths and initial angles of the steering knuckle arms, the trapezoidal arm lengths and trapezoidal bottom angles. The optimization parameters were expressed as equation (1).

$$P_x = [l_{A1C1}, \gamma_{10}, l_{A1B1}, \gamma_{10}', l_{FH}, l_{FG}, \gamma_{m0}, l_{A2B2}, l_{A2C2}, \gamma_{20}, l_{E1D1}, \varepsilon_{10}, l_{E2D2}, \varepsilon_{20}, t_1, \psi_1, t_2, \psi_2]' \quad (1)$$

Table 1. Parameters Definition in the Steering Linkages

Parameters	*Parameter denotations*	*Initial values*	*Optimization results*
l_{A1C1}	Length of A_1C_1	278*mm*	279.4*mm*
γ_{10}	Initial angle of A_1C_1	-4.5°	0°
l_{A1B1}	Length of A_1B_1	208*mm*	207.2*mm*
γ_{10}'	Initial angle of A_1B_1	-9°	-7.7°
l_{FH}	Length of FH	208*mm*	212.8*mm*
l_{FG}	Length of FG	137*mm*	131.1*mm*
γ_{m0}	Initial angle of FH and FG	0°	-2.3°
l_{A2B2}	Length of A_2B_2	197*mm*	192.7*mm*
l_{A2C2}	Length of A_2C_2	316*mm*	315.4*mm*
γ_{20}	Initial angle of A_2B_2 and A_2C_2	-1°	-0.8°
l_{E1D1}	Length of D_1E_1	257*mm*	258*mm*
ε_{10}	Initial angle of D_1E_1	0°	-2°
l_{E2D2}	Length of D_2E_2	257*mm*	262.6*mm*
ε_{20}	Initial angle of D_2E_2	0°	-2.2°
t_1	Length of M_1J_1	228*mm*	230.7*mm*
ψ_1	First axle trapezoidal bottom angle	75°	77.8°
t_2	Length of M_2J_2	226*mm*	225.8*mm*
ψ_2	Second axle trapezoidal bottom angle	72°	67.9°

3.2. Objective Function

Based on Figure 2 and Table 1, the first axle swing arm mechanism transmitting ratio could be calculated as:

$$T_{RS1} = \frac{d\gamma_1}{d\varepsilon_1} \tag{2}$$

Where γ_1 and ε_1 were the rotational angles of A_1C_1 and D_1E_1 respectively.

The first axle trapezoidal mechanism transmitting ratio could be calculated as:

$$T_{RT1} = \frac{d\varepsilon_{11}}{d\varepsilon_1} \tag{3}$$

Where ε_{11} was the rotational angle of the first axle right trapezoidal arm L_1K_1.

Based on Figure 3 and Table 1, the second axle swing arm mechanism transmitting ratio could be calculated as:

$$T_{RS2} = \frac{d\gamma_1}{d\varepsilon_2} \tag{4}$$

Where γ_1 and ε_2 were the rotational angles of A_1B_1 and D_2E_2 respectively.

The second axle trapezoidal mechanism transmitting ratio could be calculated as:

$$T_{RT2} = \frac{d\varepsilon_{22}}{d\varepsilon_2} \tag{5}$$

Where ε_{22} was the rotational angle of the second axle right trapezoidal arm L_2K_2.

Transmitting ratios of the first and second axle steering mechanisms shown in equation (6) and equation (7) could be figured out based on equation (2) to equation (5).

$$T_{R1} = \frac{2T_{RT1} \cdot T_{RS1}}{1 + T_{RT1}} \tag{6}$$

$$T_{R2} = \frac{2T_{RT2} \cdot T_{RS2}}{1 + T_{RT2}} \tag{7}$$

Supposed that $T_{R1\max}$, $T_{R1\min}$, $T_{R2\max}$ and $T_{R2\min}$ were the maximum and minimum values of the first and second axle steering mechanism transmitting ratios respectively, which could be calculated according to equation (6) and equation (7). Thus, the minimum unevenness model of the steering linkages transmitting ratios could be expressed as equation (8).

$$OF_1' = \min\left\{\frac{T_{R1\max} - T_{R1\min}}{T_{R1\max} + T_{R1\min}} + \frac{T_{R2\max} - T_{R2\min}}{T_{R2\max} + T_{R2\min}}\right\} \tag{8}$$

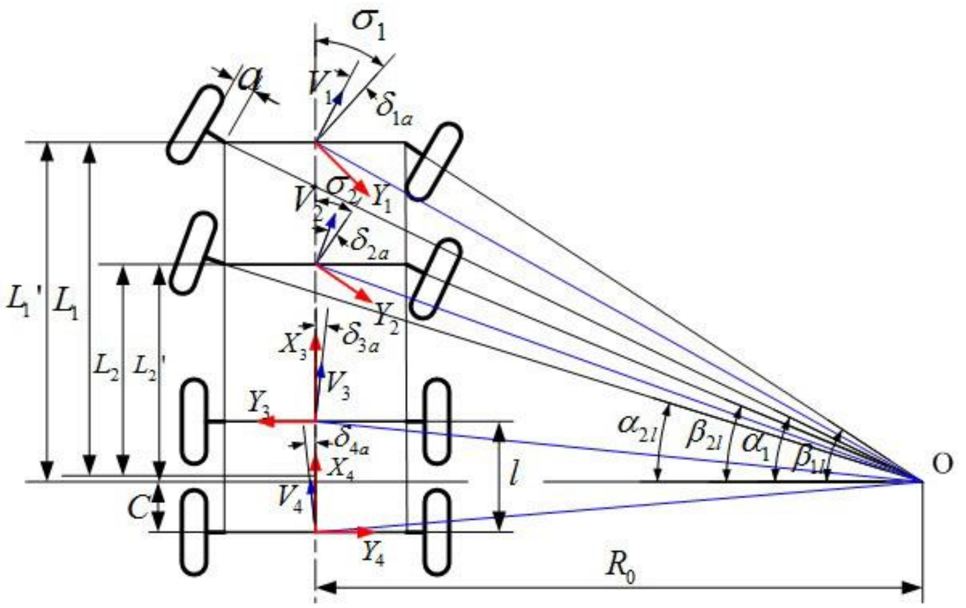

Figure 4. Theoretical Deflections of the Steered Wheels

Figure 4 was a schematic drawing that illustrated the theoretical deflections of the steered wheels, in which O was the instantaneous steering centre in the process of turning, δ_{1a}, δ_{2a}, δ_{3a} and δ_{4a} were the sideslip angles of each axle respectively, σ_1 and σ_2 were the average deflections of the left and right steered wheels in the first and second axles respectively, V_1, V_2, V_3 and V_4 were the velocities of the midpoints of each axle respectively, Y_1, Y_2, Y_3 and Y_4 were the lateral forces loaded on each axle respectively, X_3 and X_4 were the longitudinal forces loaded on the third and fourth axles, a was the wheel steering arm, L_1 and L_2 were the distances between the centre line of the double rear axles and the first and second axles respectively, l was the distance between the double rear axles. Firstly, R_0 and C could be obtained based on Figure 4, and then L_1' and L_2' could be derived. Secondly, by reference to the first axle left steered wheel deflection α_1, which was one of the parameters determined in the vehicle design stage, the theoretical deflections of the steered wheels shown in equation (9) were obtained.

$$\begin{cases} \beta_{1l} = arcctg(ctg\alpha_1 \pm \dfrac{K}{L_1'}) \\ \alpha_{2l} = arctg(\dfrac{L_2'}{L_1'} tg\alpha_1) \\ \beta_{2l} = arctg(\dfrac{L_2'}{L_1'} \dfrac{1}{ctg\alpha_1 \pm \dfrac{K}{L_1'}}) \end{cases} \tag{9}$$

Where β_{1l}, α_{2l} and β_{2l} were the theoretical deflections of the first axle left steered wheel, the second axle left steered wheel and the second axle right steered wheel respectively, K was the distance between the intersection points of the ground and the kingpin centre lines of the first and second axles. Equation (9) took "+" when the steered wheels deflected left, conversely, equation (9) took "-" when the steered wheels deflected right.

Loaded angular displacement driving on the steering wheel in the vehicle model shown in Figure 1, the actual deflections of each steered wheel *i.e.* α_1, β_1, α_2 and β_2 could be acquired. Substituted α_1 in equation (8), and β_{1l}, α_{2l} and β_{2l} could be obtained. Thus, the minimum deflection errors model of the steered wheels could be expressed as equation (10).

$$OF_2{}' = \min\left\{\int_{\alpha_{1R}}^{\alpha_{1L}} w(\alpha_1)\cdot[(\beta_1-\beta_{1l})^2+(\alpha_2-\alpha_{2l})^2+(\beta_2-\beta_{2l})^2]d\alpha_1\right\} \tag{10}$$

Where α_{1L} and α_{1R} were the maximum deflections of the first axle left steered wheel deflecting left and right, $w(\alpha_1)$ was the weight function and $w(\alpha_1)=\begin{cases}1.5 & 0\le|\alpha_1|\le 10^\circ\\ 1.0 & 10^\circ\le|\alpha_1|\le 15^\circ\\ 0.5 & 15^\circ\le|\alpha_1|\le\alpha_{1\max}\end{cases}$.

Based on the unified objective method, $OF_1{}'$ and $OF_2{}'$ could be converted to the values in an interval of $[0,1]$. Firstly, converted $OF_1{}'$ and $OF_2{}'$ to the values in an interval of $[0,2\pi]$, shown as equation (11).

$$\begin{cases}OF_1{}'' = \dfrac{OF_1{}'}{OF_{1\max}{}'}\cdot(2\pi)\\[2ex] OF_2{}'' = \dfrac{OF_2{}'}{OF_{2\max}{}'}\cdot(2\pi)\end{cases} \tag{11}$$

Secondly, converted $OF_1{}''$ and $OF_2{}''$ to the values in an interval of $[0,1]$, shown as equation (12).

$$\begin{cases}OF_1 = \dfrac{OF_1{}''}{2\pi} - \sin(OF_1{}'')\\[2ex] OF_2 = \dfrac{OF_2{}''}{2\pi} - \sin(OF_2{}'')\end{cases} \tag{12}$$

The unified objective function in the optimization was expressed as:

$$OF = w_1\cdot OF_1 + w_2\cdot OF_2 \tag{13}$$

Where w_1 and w_2 were the weight functions, and $w_1 = w_2 = 0.5$.

3.3. Constrains

The parameters in equation (1) were all limited to the range of $90\%\sim110\%$ of the initial parameter values.

The unevenness of the first and second axle steering mechanism transmitting ratios was limited to below 5%.

The minimum transmitting angles in the steering linkages were all greater than 40°.

3.4. Optimization Results

The steering linkages were optimized in Adams software using SPQ algorithm, and the optimization results of the parameters were also listed in Table 1. Set the maximum rotational angles of the steering wheel from the middle position to the left and right limited positions to be $\pm 860^{\circ}$, and the optimization results of the steering linkages transmitting ratios compared with the initial ratios were shown in Figure 5 and Figure 6. As shown in Figure 5, the unevenness of the first axle steering mechanism transmitting ratio before and after the optimization was 9.7% and 4.9% respectively according to equation (8). As shown in Figure 6, the unevenness of the second axle steering mechanism transmitting ratio before and after the optimization was 3.8% and 2.1% respectively. The unevenness was improved after the optimization of the steering linkages, and the unevenness of the steering force was improved consequently.

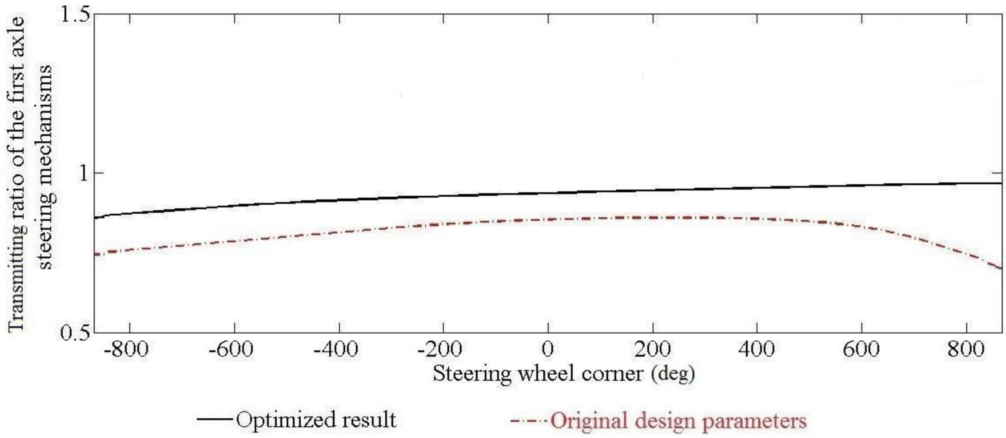

Figure 5. First Axle Steering Mechanism Transmitting Ratio

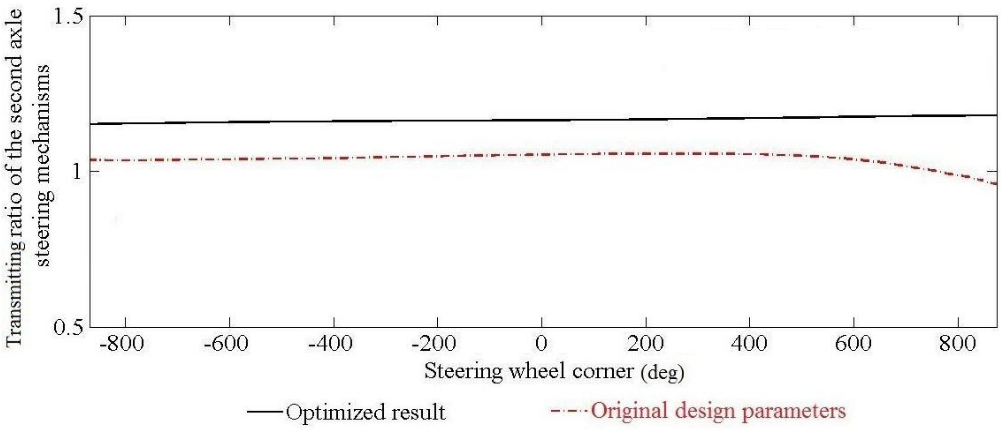

Figure 6. Second Axle Steering Mechanism Transmitting Ratio

By reference to the deflection of the first axle left steered wheel, the deflections of the other steered wheels before and after the optimization were shown from Figure 7 to Figure 9. As shown in Figure 7, the maximum deflection error of the first axle right steered wheel decreased by 39.3% after the optimization, and the maximum difference between the bilateral deflections errors reduced from 1° to 0.5°. As shown in Figure 8, the maximum deflection error of the second axle left steered wheel decreased by 21.4% after the optimization, and the maximum difference between the bilateral deflections errors reduced from 4° to 0.8°. As shown in Figure 9, the maximum deflection error of the second axle right steered wheel decreased by 48.4% after the optimization, and the maximum difference between the bilateral deflections errors reduced from 6° to 0.3°. In general, the actual deflections of the steered wheels were closer to the theoretical deflections after the optimization, and the errors decreased and tended to be bilateral evenness. The actual deflections of the steered wheels at the bilateral corresponding positions tended to be equal, because the steering linkages transmitting ratios were considered in the optimization.

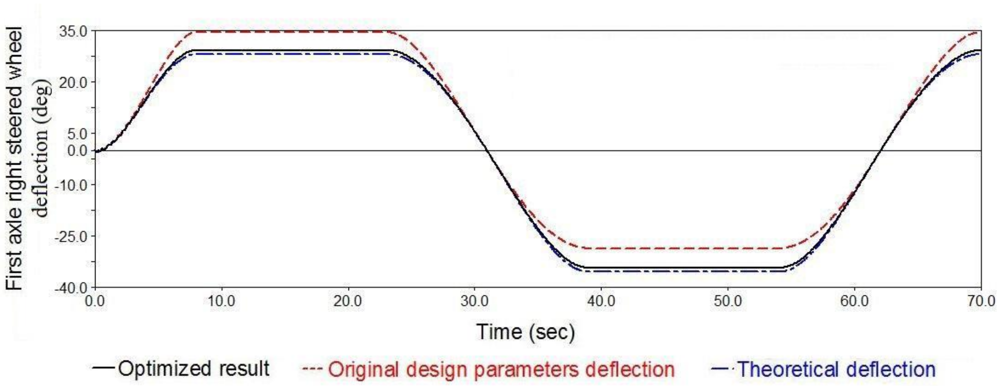

Figure 7. First Axle Right Steered Wheel Deflection

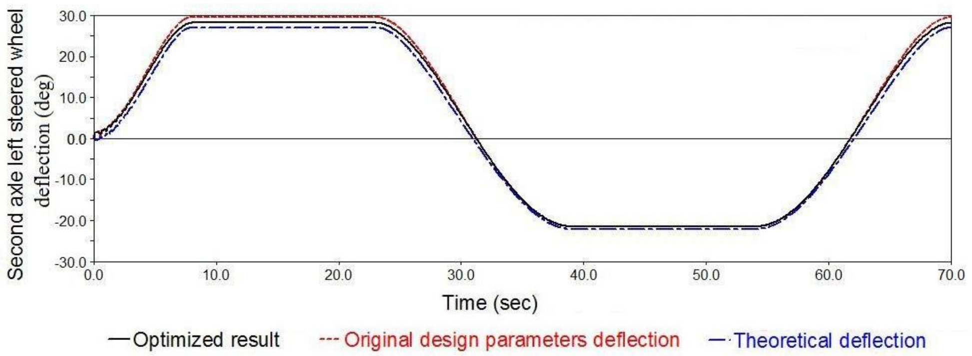

Figure 8. Second Axle Left Steered Wheel Deflection

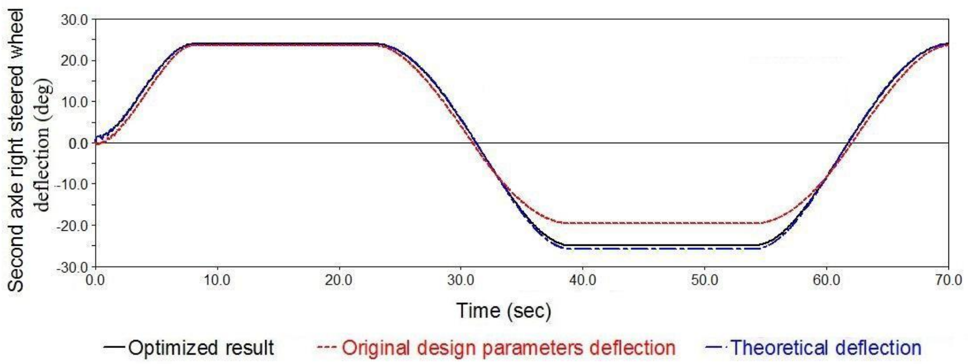

Figure 9. Second Axle Right Steered Wheel Deflection

4. Influences of the Steering Linkages Transmitting Ratios on the Vehicle Handiness

The vehicle handiness was usually measured by two targets, *i.e*. the steering force and the number of the steering wheel turns.

The steering force was expressed as equation (14):

$$T_H = M_I + \zeta \cdot \dot{\theta}_{sw} + \text{sgn}(\theta_{sw}) \cdot M_f + M_{HZ} \tag{14}$$

Where θ_{sw} was the rotational angle of the steering wheel, M_I was the inertial moment of the steering system, ζ was the damping of the steering system, M_f was the friction moment of the steering system, M_{HZ} was the equivalent aligning torque of the steered wheels loaded on the steering wheel. The values of M_I, ζ, M_f and M_{HZ} were obtained through tests.

The number of the steering wheel turns N_{sw} was given by:

$$N_{sw} = \alpha_1 \cdot i_{sg} \cdot T_{RS1} \tag{15}$$

Where i_{sg} was the steering gear ratio. When α_1 reached the maximum value, the number of the steering wheel turns would also reach the maximum value.

Set the design range of α_1 to be $-30^\circ \sim 42^\circ$. The steering force along with the rotational angle of the steering wheel corresponding to α_1 was shown in Figure 10, from which the maximum difference between the bilateral steering forces before and after the optimization were $1.44 N \cdot m$ and $1.1 N \cdot m$ respectively, and the unevenness of the steering force was improved. The number of the steering wheel turns along with α_1 was shown in Figure 11, from which the number of turns turning left and right before the optimization was 2.26 and 2.41, and the number of turns turning left and right after the optimization was 2.29 and 2.4 because of the improvement of the unevenness of the steering linkages transmitting ratios.

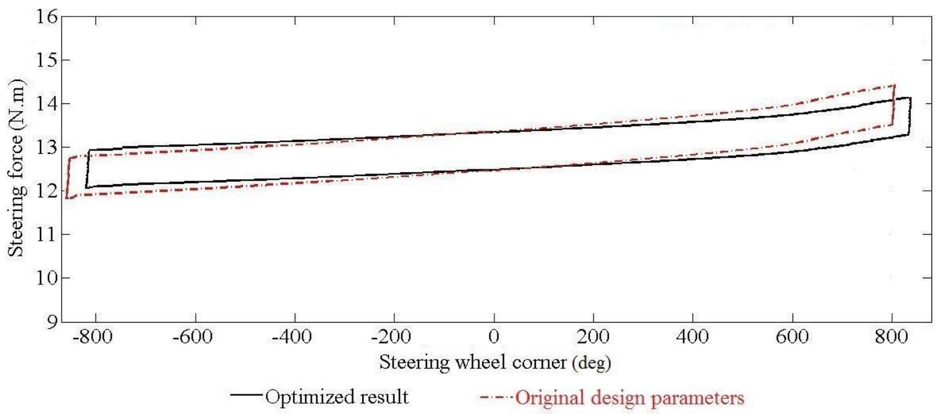

Figure 10. Steering Force

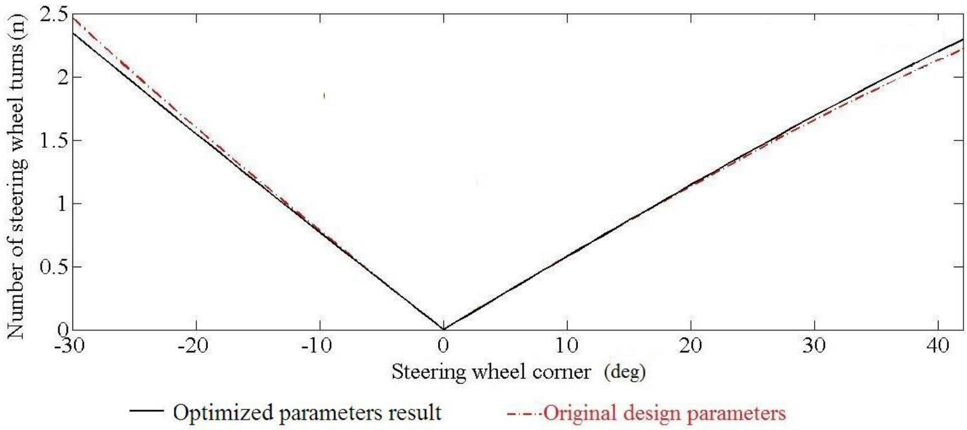

Figure 11. Number of Steering Wheel Turns

5. Influences of the Steering Linkages Transmitting Ratios on the Vehicle Control Stability

Limited by the length of thesis, only lemniscate simulations were taken to research the influences of the steering linkages transmitting ratios on the vehicle control stability. Setting the vehicle velocity to be $28Km/h$ on grade-B road, the vehicle lemniscate simulations were processed in Adams software and the yaw angular velocity and lateral acceleration of the cargo centre mass were tested during the simulations. As shown in Figure 12, the difference between the positive and negative amplitude of the yaw angular velocity before and after the optimization were $1.67^{\circ}/s$ and $0.58^{\circ}/s$ respectively. As shown in Figure 13, the difference between the positive and negative amplitude of the lateral acceleration before and after the optimization were $83.3mm/s^2$ and $49.6mm/s^2$ respectively. Because of the improvement of the unevenness of the steering linkages transmitting ratios, the positive and negative amplitude of the yaw angular velocity and lateral acceleration of the cargo centre mass tended to be symmetrical. Thus the vehicle control stability was improved.

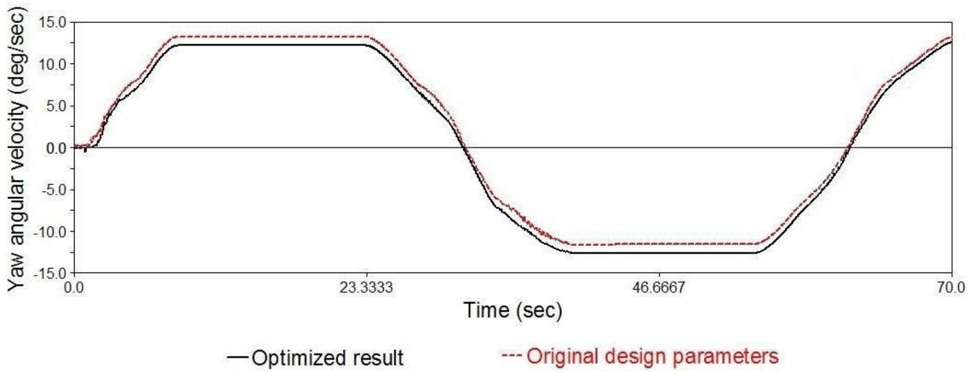

Figure 12. Yaw Angular Velocity Simulations

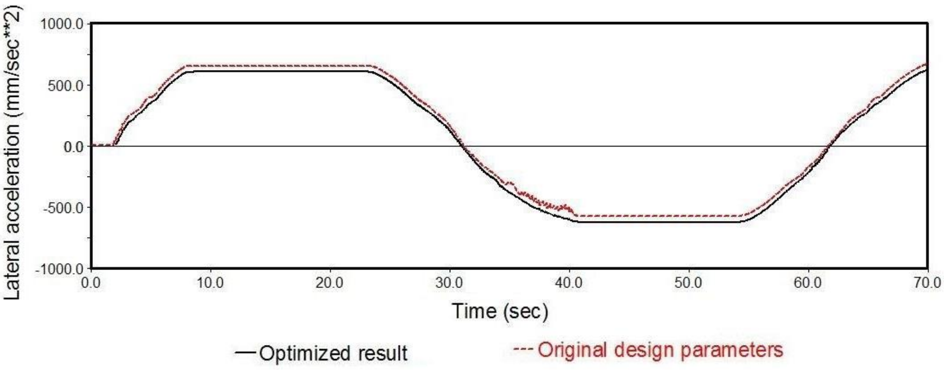

Figure 13. Lateral Acceleration Simulations

6. Conclusions

In this paper, the optimization method of the double front axle steering linkages in an 8×4 type heavy duty truck was researched. Multi-objective optimization model was developed with the objectives of the minimum unevenness of the steering linkages transmitting ratios and the minimum deflection errors of the steered wheels. After the optimization, the unevenness of the steering mechanism transmitting ratio of the first and second axle reduced by 4.8% and 1.7% respectively. By reference to the deflection of the first axle left steered wheel, the other deflection errors of the steered wheels in order reduced by 39.3%, 21.4% and 48.4% respectively. Benefitting from the improvement of the unevenness of the steering linkages transmitting ratios after the optimization, the maximum differences between the bilateral deflection errors of the steered wheels in order reduced by 0.5°, 3.2° and 5.7° respectively, the maximum difference between the bilateral steering forces reduced by $0.34N \cdot m$, the maximum difference of the number of turns of the steering wheel between left and right limited position reduced by 0.04, the differences between the positive and negative amplitude of the yaw angular velocity and lateral acceleration of the cargo centre mass reduced by $1.09^\circ / s$ and $33.7mm/s^2$ respectively.

The advantages of the optimization method used in the paper were that the swing arm mechanisms and the trapezoidal mechanisms were optimized in the same optimization model at the same time; the comprehensive optimization results could reflect the actual situations of steering linkages; the deflection errors of the steered wheels tended to be symmetrical, the maximum difference between the bilateral steering forces decreased, and the differences between the positive and negative amplitude of the yaw angular velocity and lateral acceleration of the cargo centre mass also decreased, because the unevenness of the steering linkages transmitting ratios was led into the objective function. Namely the vehicle control stability was improved.

7. Appendices

Table 2. Main Parameters of the 8×4 Type Heavy Duty Truck

Parameters	*Values*
Curb mass	31 tons
Axle distance	1800mm+4495 mm +1360 mm
Leaf spring stiffness of steering axle suspension	416 N/mm

Leaf spring stiffness of equalizing suspension	3240N/mm
Wheel mass	110Kg×12
Frame mass	1950Kg
Engine mass	880 Kg
Cargo mass	21810Kg
First axle mass	1000 Kg
Second axle mass	905 Kg
Third axle mass	1000 Kg
Fourth axle mass	920 Kg
Cab mass	900 Kg
Gear box mass	315Kg

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