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# Deformation Effect Simulation and Optimization for Double Front Axle Steering Mechanism

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**Abstract.** This paper research on tire wear problem of heavy vehicles with Double Front Axle Steering Mechanism from the flexible effect of Steering Mechanism, and proposes a structural optimization method which use both traditional static structural theory and dynamic structure theory – Equivalent Static Load (ESL) method to optimize key parts. The good simulated and test results show this method has high engineering practice and reference value for tire wear problem of Double Front Axle Steering Mechanism design.

**Keywords;** Double front axle steering mechanism, Deformation effect, Simulation, ESL

## 1. Introduction

According to the limit of domestic road conditions and traffic regulations, the double front axle steering system is used in heavy vehicles to increase carrying capacity and improve handling stability. But this steering system often gives rise to vehicle tire wear problem [1] [2] [3]. Furthermore, the load of double front axle steering mechanism has more than ten thousands Newton under normal working conditions, and the heavy truck is usually drive with large deadweight and poor road conditions. For this reason, a large flexible deformation appears in double front axle steering mechanism in the process of steering which is an important factor for tires wear problem. But few literatures analyzed flexible deformation on influence of wheels steering angle error completely, let alone complete structure optimization method.

For the above problems, this paper takes double front axle steering mechanism of Hualing truck as research object to carry out flexible deformation analysis and optimization. A new structure dynamic optimization algorithm—Equivalent Static Load (ESL) method is applied to optimize steering systems structure. It made a beneficial exploration for ESL dynamic optimization algorithm using in double front axle steering system optimization design.

## 2. Flexible effect Simulation

An 8 x 4 heavy truck with double front axle steering mechanism multi-body dynamics model is show as Fig.1. In order to study the effect of steering mechanism deformation for wheels steering angle error, we established rigid multi-body dynamics model and flexible multi-body dynamics model for the double front axle steering mechanism, and compared the simulation results [4]. In flexible multi-body model, the key parts of steering mechanism are treated as flexible body (As shown in Tab.1), so flexible deformation must be considered. However the flexible deformation in rigid body multi-body model is ignored.

1.Steering wheel
2.First axle swing arm
3. First axle steer rod
4.Steer tire
5.First axle steering knuckle
6. First axle steering trapezium
7.Steer drag rod
8. Middle swing arm bearing
9. Middle swing arm
10.Power steering cylinder
11. Second axle steer rod
12.Second axle steering trapezium

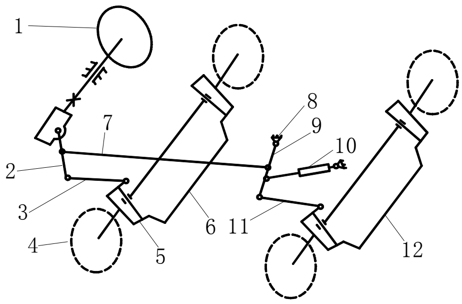


Fig.1 Double front axle steering mechanism

Steering mechanism would bear much more resistance than other conditions when vehicle is steering. The main resistance for steering mechanism is friction resistance between road and wheels whose maximum is observed at the condition of zero-radius turning (ZRT) with fully loaded. This resistance is difficult to calculate accurately, so engineers usually use semi-empirical equation (1) to calculate approximately [5].

$$M_r = \frac{\mu}{3}\sqrt{\frac{F^3}{p}} \quad (1)$$

Where μ is road friction factor, F is axle load, P is the tire pressure.

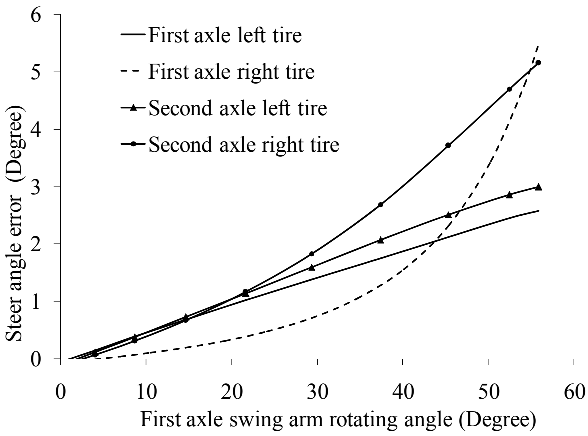


Fig.2 Steer angle error vs. swing arm rotation angle

Simulation results of tire steer angle error for rigid multi-body model and flexible multi-body model in condition of zero-radius turning (ZRT) with fully loaded is showed in Fig.2. The result shows that the flexible deformation of double front axle mechanism makes a tire steer angle error maximum at 5.4 degrees between the first axle and the second axle, which is 35% higher than target value 4.0 degrees. This error will aggravate tire wear and lower vehicle steering performance and traffic safety. Considering conditions of overload and the worse road, the influence of error will be larger. The steering problem of Hualing heavy truck those had been sold also confirms this conclusion.

In order to further research contribution degree of various components flexible deformation on the overall flexible deformation, Tab.1 lists the maximum deformation of various parts of vehicle steering mechanism and deformation rate. It is obvious that Parts ID 3, 7, 11 make the main contribution to flexible deformation. Therefore, these parts need to be optimized in order to reduce the effect of flexible deformation and make them lightweight.

Tab.1 Flexbody deformation results of double front axle steer systems

| Parts ID | Parts name | Max Deformation(mm) | Percentage |
|---|---|---|---|
| 2 | First axle swing arm | 0.9 | 3.7% |
| 3 | First axle steer rod | 6.9 | 28.2% |
| 6 | First axle Steering trapezium | 1.21 | 4.9% |
| 7 | Steer drag rod | 8.2 | 33.5% |
| 8 | Middle swing arm bearing | 0.12 | 0.5% |

| | | | |
|---|---|---|---|
| 9 | Middle swing arm | 0.9 | 3.7% |
| 11 | Second axle steer rod | 4.9 | 20.0% |
| 12 | Second axle steering trapezium | 1.32 | 5.4% |

# 3. Results of structural optimization

The structure optimization algorithms are mainly topology optimization, shape optimization, size optimization and shape optimization. There are plenty of references about these optimization algorithms, and the first three optimization methods are applied to the structure optimization of double front axle steering mechanism.

All the forces in the real world act dynamically on structures. Since dynamic loads are extremely difficult to handle in analysis and design, static loads are usually utilized with dynamic factors. Generally, the dynamic factors are determined from design codes or experience. Therefore, static loads may not give accurate solutions in analysis and design and structural engineers often come up with unreliable solutions. An analytical method based on modal analysis is proposed for the transformation of dynamic loads into Equivalent Static Loads (ESLs). The ESLs are calculated to generate an identical displacement field with that from dynamic loads at a certain time [6] [7].

## 3.1. ESL optimization method

Using the vibration theory with the finite element method (FEM), the dynamic behavior of a structure is expressed by the following differential equations:

$$M(b)\ddot{d}(t) + K(b)d(t) = f(t) = \{0 \cdots 0 f_i \cdots f_{i+l-1} 0 \cdots 0\}^T \quad (2)$$

Where M is the mass matrix, K is the stiffness matrix, f is the vector of external dynamic loads, d is the vector of dynamic displacements, and l is the number of nonzero components of the dynamic load vector. The static analysis with FEM formulation is expressed as:

$$K(b)x = s \quad (3)$$

Where x is the static displacement vector and s is the external static loads vector. An ESL set, which generates the same displacement field as that of the dynamic loads at an arbitrary time $t_a$ is defined as:

$$s = Kd(t_a) \quad (4)$$

Therefore, according to equation (2) (3) (4), the relation between dynamic load and static load at an arbitrary time is expressed as:

$$s = f(t) - M(b)\ddot{d}(t) \quad (5)$$

Equation (5) shows that the object in the function of an equivalent static load can produce the same displacement field as in the function of a dynamic load. According to the finite element theory, stress is getting through the node displacement calculation, so the same displacement field will lead to the same stress field.

If $x_p = d_p(t_a)(p = 1,2,\cdots,N)$, then the two fields from the dynamic and the static load sets are identical. Therefore, the following equations are obtained:

$$\mathrm{d_p(t_a) = x_p = \sum_{k=1}^{N} \frac{1}{w_k^2}\left(\sum_{j=1}^{N} u_{pk} u_{jk} s_j\right)(P = 1,\cdots,N)} \quad (6)$$

Where $w_k$ is the k order natural frequency, Equation (6) is a system of linear simultaneous equations which have N variables of s, and needs a modal analysis and a lot of calculations. Since equations (4) and (6) can be extremely large, the external static force vector s is approximated for engineering applications as follows:

$$s = [0 \ldots 0 s_i \ldots s_{i'+l'-1} 0 \ldots 0]^T \quad (7)$$

The non-zero components s can be made arbitrarily. In engineering sense, the nonzero components can be imposed on the important places where the dynamic loads act. Equation (6) is transformed into equations as follows:

$$d_p(t_a) = x_p = \sum_{k=1}^{N} \frac{1}{w_k^2}\left(\sum_{j=i'}^{i'+l'-1} u_{pk}u_{jk}s_j\right) (P = 1,\cdots,l') \tag{8}$$

$$d_p(t_a) \approx x_p = \sum_{k=1}^{N} \frac{1}{w_k^2}\left(\sum_{j=i'}^{i'+l'-1} u_{pk}u_{jk}s_j\right) (P = l'+1,\cdots,N) \tag{9}$$

Equation (8) is a system of linear simultaneous equations with l` variables for $s_j$. Therefore, vector s is determined from Equation (8). As l` becomes larger, the number of approximated displacements is reduced. If a static load set is calculated by solving equation (8) directly to make the same displacement field as that from the dynamic load set, it can be awkward in the engineering sense. That is, the magnitude of the loads can be extremely large in order to satisfy all the conditions at many nodes. Therefore, the equations are modified to inequality equations as follows:

$$d_p(t_a) \le x_p = \sum_{k=1}^{n} \frac{1}{w_k^2}\left(\sum_{j=i'}^{i'+l'-1} u_{pk}u_{jk}s_j\right) \quad (P = 1,\cdots,h) \text{ when } d_p \text{ is positive} \tag{10}$$

$$d_p(t_a) \ge x_p = \sum_{k=1}^{n} \frac{1}{w_k^2}\left(\sum_{j=i'}^{i'+l'-1} u_{pk}u_{jk}s_j\right) \quad (P = 1,\cdots,h) \text{ when } d_p \text{ is negative} \tag{11}$$

The process of transformation of dynamic load to Equivalent Static Load is shows in Fig.3.

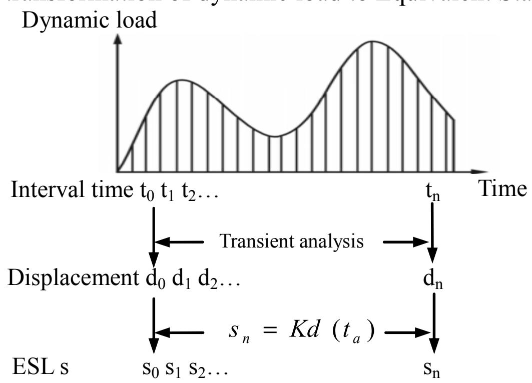


Fig.3 Transformation of dynamic load to ESL

The mathematical model of ESL optimization process is expressed by the following expression:

Minimize $F(b)= sum((F(a)- F(a`))^2)$ (12)

Subject to $K(b)x=s_i(b)$ (i=1, …,m) (13)

$\varphi(b,x)_j \le 0$ (j=1, …,n) (14)

Where F(b) is target function, F(a)and F(a`) are steer angle simulation and target value function respectively, b is design variable vector, K is stiffness matrix, m is interval time, n is constraints number, $s_i(b)$ is the number of i equivalent static load vector, $\varphi(b,x)_j$ is constraint function.

## 3.2. Results of structural optimization

Tab.2 The comparison results of before and after optimization

| Items \ Part Name | | Middle swing arm bearing | First axle steer rod | Second axle steer rod | Steer drag rod | Steer drag rod -ESL |
|---|---|---|---|---|---|---|
| Max Deformation (mm) | Before | 0.12 | 6.9 | 4.9 | 8.2 | 8.2 |
| | After | 0.11 | 3.4 | 2.1 | 2.3 | 3.2 |
| | Change | ↓ 8.3% | ↓ 51% | ↓ 57% | ↓ 71% | ↓ 61% |
| Max Stress (MPa) | Before | 340 | 568 | 534 | 764 | 764 |
| | After | 256 | 410 | 320 | 364 | 382 |
| | Change | ↓ 25% | ↓ 28% | ↓ 40% | ↓ 52% | ↓ 50% |
| Weight (kg) | Before | 8.38 | 5.1 | 6.5 | 7.63 | 7.63 |
| | After | 7.47 | 7.76 | 8.15 | 11.9 | 9.1 |
| | Change | ↓ 11% | ↑52% | ↑25% | ↑56% | ↑19% |

The comparison result of before optimization and after optimization of various steer components is shown in Tab.2. Since the original design of double front axle steer mechanism appears large flexible deformation, even occurs plastic deformation problem. Therefore, some components would be improved the stiffness performance and increased weight at the same time, such as First axle steer rod. Some components had achieved better stress performance and lighter weight, such as Middle swing arm bearing. If the optimization method is applied to early product development stage, the results will be more obvious. The ESL method is applied to optimize Steer drag rod, and discrete step is 100 which meets the accuracy requirements of ESL.

### 3.3. Test verification

Tab.3 is the mean and variance value of steer angle test results of before and after optimization comparison with the theoretical values. The results show:

1)The optimization design can reduce front and rear tire steer angle error.

2) The ESL method gives more accurate solutions than static structural optimization. That means the ESL optimization design meets the design target and achieves the more light weight.

Tab.3 Test results of before and after optimization

| Items \ Test results | First axle right tire steer angle | | | Second axle right tire steer angle | | |
|---|---|---|---|---|---|---|
| | Before | After | Target | Before | After | Target |
| Mean value (°) | 1.48 | 1.06 | 1.0 | 2.21 | 1.34 | 1.2 |
| Variance value $(°)^2$ | 2.61 | 1.87 | - | 2.83 | 1.61 | - |

## 4. Conclusion

In allusion to the tire wear problem caused by large deformation of double front axle steering mechanism of hualing heavy truck, this paper creates the flexible multi-body dynamics simulation model of steering mechanism, analyse the flexible deformation influence on wheels steer angle error completely and systematically, puts forward a optimization method which contains static and dynamic optimization theory, and obtains a good result.This optimization design method for double front axle steering mechanism provides an overall structural solution, and has good value of engineering application for tire wear problem of double front axle steering mechanism.

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