

Robust Analysis and Optimization Design of Double Front Axle Steering System

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ABSTRACT

Existing multi-axle steering system designs generally use the deterministic optimization method without considering the uncertainties during the design process; therefore an actual steering movement may deviate from the ideal movement calculated by some mathematical models. In order to make design results have less sensitive to the uncertainties in the design process, some uncertainties need be taken into account at the early design stage. This paper proposes a robust optimization design method for a double front axle steering system (DFASS) of heavy trucks based on Monte Carlo method. The DFASS consists of two trapezoidal steering mechanisms (TSM) and one rocker system, and the optimization objectives of DFASS include the minimum mean value and variance of the maximum turning angle error of the TSM and rocker system. In addition, the robust optimization model includes 13 design variables which are all geometry parameters of DFASS and represented by normal distribution. Through the orthogonal experiment, we obtain the important factors affecting optimization objectives and build the response surface models of optimization objective. Based on the response surfaces, robustness design of DFASS is performed. By simulating analysis, the improved robustness of DFASS based on robust design method is approved.

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INTRODUCTION

The existing vehicles exhibit a heavy steering and abnormal wear tire which might be induced by the imperfect steering mechanism design. On the one hand, it is owing to the synthesis design method, which is an approximate method without an accurate mathematical model; on the other hand, it is the uncertainties in the process of manufacturing and assembling steering mechanism, etc. which change the ideal movement of steering wheels. In traditional designs, the steering system generally used the deterministic optimization method, and the uncertainties rarely had been considered^[1, 2, 3, 4]. However, the steering system is a multi-linkage system, which is easy to appear movement deviation due to the existing uncertainties. Especially for a multi-axle steering system, because of its complex multi-linkage structure, it is easier to induce movement deviation by a tiny variation of design variables. These movement deviations would add the turning angle error of steering mechanism and accelerate tire wearing. In order to solve the problem, the uncertainties need be considered at the early design stage^[5, 6, 7]. Monte Carlo

method applied in many mechanism designs is an effective method to decrease interference from the uncertainties^[8, 9, 10]. In addition, the relationships between robustness and range value of design parameters also need to be considered^[11]. Based on the above analysis, a design method of double front axle steering system (DFASS) is proposed which is different from the method of tradition steering mechanism^[12, 13, 14]. The DFASS consists of two trapezoidal steering mechanisms (TSM) and one rocker system, therefore, the optimization objective of DFASS in this paper is to minimize the mean value and variance of the maximum turning angle error of TSM and rocker system. Furthermore, considering the uncertainties in manufacturing and assembling process of DFASS, design variables have been defined as random variables of normal distribution, moreover the robustness of DFASS has been analyzed applying Monte Carlo method.

1. DOUBLE FRONT AXLE STEERING SYSTEM

1.1. Double Front Axle Steering System Model

When a car goes round a corner, it turns around a point along the line of its rear axle, which means that the two front wheels will have to turn through slightly different angles so that they are also guiding the vehicle round this point, and not fighting the turn by scrubbing. For a vehicle of DFASS, the turning angle of inside and outside wheels at the same axle need ensure a kinematic condition also called Ackerman condition which determines the correct geometry of the turning angle of each steering wheel^[14, 15]. In addition, the perpendicular lines to wheels at the same side of different axles need meet at one point called the instantaneous turning centre as shown in Fig.1. Vehicles will achieve a good turning performance and reduce tire wear if the DFASS met the above requirements.

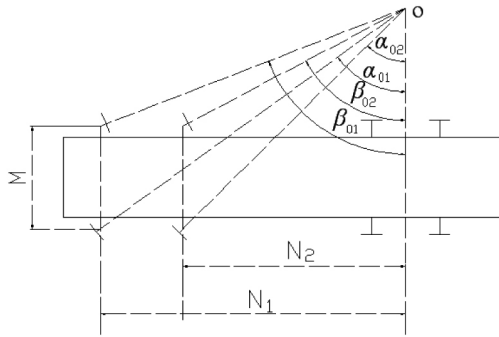


Fig. 1. Geometry of turning angle for DFASS

Figure 1 shows the turning angle of different wheels when a vehicle turns right. The two front axles are called the first axle and the second axle. The inside and outside wheels are distinguished from turning centre O. The inside wheels are right wheels which are closer to the turning centre. The distance between kingpins at the same axle is denoted M. The distance between two front axles and the center line of two rear axles is called wheelbase denoted N_i , $i = 1, 2$. The ideal turning angles of outside wheel and inside wheel respective are expressed as α_{0i} and β_{0i} , $i = 1, 2$.

1.2. Optimum Objective

DFASS design is different from the traditional single axle steering system design which includes a trapezoidal steering system (TSM) design at each axle (the first axle and the second axle) and a rocker system design correlating the two front axles.

Optimization objective of the TSM is expressed as equation (1)

$$\min F_i(x) = \varepsilon_i = \max |\alpha_{ij} - \alpha_{0ij}| \quad (1)$$

where, ε_i is the maximum error between the actual turning angles of α_{ij} and the ideal turning angles of α_{0ij} of outside wheel. $i = 1, 2$; j is the real-time turning angle. α_{0ij} are calculated by kinematic condition of $\alpha_{0ij} = \arctan(M/N_i + \tan\beta_{0ij})$ and α_{ij} are measured by simulation model. Here, β_{0ij} are the ideal turning angle at inside wheels.

For the rocker system, the turning angles of the front and rear pitman arm of rocker system at different axles need meet equation (2).

$$\tan\beta_{01} / \tan\beta_{02} = N_1 / N_2 \quad (2)$$

where, β_{01} is the ideal angle of front pitman arm which is also the turning angle of inside wheel at the first axle. β_{02} is the ideal turning angle of rear pitman arm which is also the turning angle of inside wheel at the second axle. The ratio of tangent value of the two turning angle needs to equal the ratio of wheelbase value.

Therefore, the optimization objective of rocker system is expressed as equation (3)

$$\min F_3(x) = \varepsilon_3 = \max |\beta_{02j} - \beta_{2j}| \quad (3)$$

where, ε_3 is the turning angle error between the ideal turning angle and actual turning angle of β_{2j} at rear pitman arm when the front pitman arm is turning.

Based on the above analysis, optimization objective of the DFASS is expressed as following.

$$\min F(x) = F_1(x) + F_2(x) + F_3(x) \quad (4)$$

1.3. Design Variables

The simulation model of DFASS is shown in Fig.2a which includes 2 front steering axles, 2 rear axles, 8 suspensions, 8 wheels, etc. The steering mechanism is a divided steering mechanism. Fig.2b shows 13 geometry dimensions which are defined as design variables in the DFASS design. Design variables of the two trapezoidal steering systems include the steering arm angle θ_1 and θ_2 , the steering arm length L_{10} and L_{20} , the steering linkage length L_{11} and L_{21} , and the steering linkage angle φ_1 and φ_2 . The variables of rocker system include the pitman arm length L_{31} and L_{33} , the straight link length L_{32} and L_{34} , and the knuckle arm length L_{35} . The design variables are expressed as follows:

$$X = [x_1, x_2, x_3, x_4, x_5, x_6, x_7, x_8, x_9, x_{10}, x_{11}, x_{12}, x_{13}]^T \\ = [\theta_1, L_{10}, L_{11}, \varphi_1, \theta_2, L_{20}, L_{21}, \varphi_2, L_{31}, L_{32}, L_{33}, L_{34}, L_{35}]^T$$

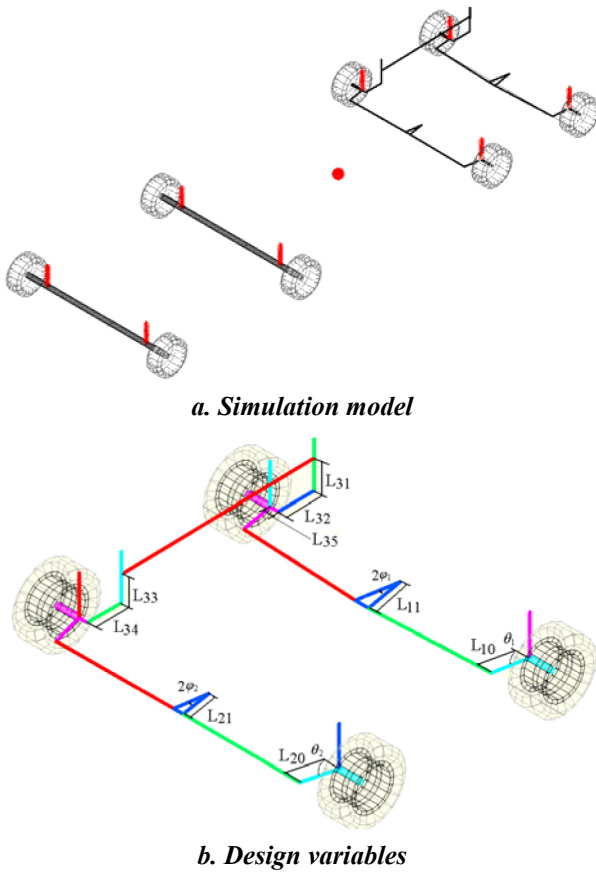


Fig. 2. Double front axle steering system (DFASS)

1.4. Constraints

To maintain turning stability, the maximum turning angle error and the range of variables of DFASS have been constrained. The range of ε_1 and ε_2 is $[0, 3]^\circ$ and ε_3 is $[0, 1]^\circ$. The range of the steering arm length (L_{10} , L_{20}), the steering linkage length (L_{11} , L_{21}), the pitman arm length (L_{31} , L_{33}), the straight link length (L_{32} , L_{34}) and the knuckle arm length (L_{35}) is $[220, 320]$ mm, $[220, 350]$ mm, $[200, 400]$ mm, $[250, 400]$ mm and $[100, 200]$ mm respectively. The range of the steering linkage angle (φ_1 , φ_2) is $[5, 15]^\circ$.

In addition, the range of the arm angle meets [equation \(5\)](#).

$$\theta_i = \arctan 4N_i / 3M \pm 5^\circ. \quad (5)$$

where, θ_i is the steering arm angle of two trapezoidal steering mechanism respectively. $i = 1, 2$.

2. OPTIMIZATION DESIGN BASED ON THE DETERMINISTIC OPTIMIZATION METHOD

The above DFASS model is firstly solved based on the deterministic optimization method in which design variable value is nominal value without considering the uncertainties.

The values of related geometry parameter of DFASS shown in [Fig.1](#) are as follows. Kingpins' distance M is 2300mm, axle distances N_1 and N_2 respective are 6700mm and 5000mm. In addition, the maximum turning angle β_{01} is 40° .

We use the generalized reduced gradient method (GRG) to solve the DFASS model. The GRG is one of the most effective ways to solve optimization problem with nonlinear constrain.

In the GRG method, variable space is simple because some variables are expressed with other variables. The search direction in space is determined according to objective function in the contracted gradient. When the reduced gradient method is proceeding to solve the optimization problem with nonlinear constrain, the GRG is developed.

The GRG method can be summed up as follows.

$$\min F(X) \quad X \in R^n$$

$$\text{s. t.} \quad \begin{aligned} T(X) &= 0 \\ P &\leq X \leq Q \end{aligned} \quad Q, P \in R^n$$

(6)

where, $T(X) = [t_1(X), t_2(X), \dots, t_m(X)]^T$

$$P = [p_1, p_2, \dots, p_n]^T$$

$$Q = [q_1, q_2, \dots, q_n]^T$$

Firstly, design variable X is decomposed into two parts,

$$X = \begin{bmatrix} X_D \\ X_N \end{bmatrix}$$

where, X_D is the basic variable, X_N is the nonbasic variable.

Respectively

$$P = \begin{bmatrix} P_D \\ P_N \end{bmatrix}, \quad Q = \begin{bmatrix} Q_D \\ Q_N \end{bmatrix}$$

Based on the implicit function theorem, the continuous mapping exists.

$$X_D = V(X_N)$$

Therefore objective function $F(x)$ might be converted by [equation \(7\)](#).

$$F(X) = F(X_D, X_N) = f(X_N) \quad (7)$$

By the above transformation, the original objective function $F(X)$ of n variables is changed into $F(X_N)$ of $n-m$ variable. The gradient of X^k in design point X_N is just the reduced gradient of $F(X)$. The reduced gradient $F(X)$ is as follows.