MULTI-OBJECTIVE LIGHTWEIGHT DESIGN OF STEERING ARM BASED ON GENETIC ALGORITHM

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ABSTRACT: In order to improve the steering performance of steering arm, the middle steering arm of the double front axle heavy commercial vehicle was set as the optimization object. According to the function and connection relationship of the steering arm, the left and right deflection limits of the steering wheel were selected as the limited load conditions of the steering arm. Then it was based on the results of finite element analysis, with the minimum mass of the steering arm as the objective function, a lightweight multi-objective optimization design model of the control arm was established, and the topology optimization design of the steering arm was used a variable density method. The model was reconfigured based on the topological optimization results in performing the deformation process of the mesh and the optimized parameters were used as the design parameters. The second generation of non-dominated sorting genetic algorithm (NSGA-II) is used to optimization the multi-objective mathematical model of steering arm, the results of this study indicate that the mass of steering arm is reduced from 8.0620[kg] to 6.6217[kg] with the strength and stiffness of steering arm remain unchanged. Obvious lightweight effects are achieved.

KEYWORDS: Steering arm, Multi-objective optimization, NSGA-II

1. INTRODUCTION

The steering arm is used as the main force-receiving part of the steering transmission mechanism of the automobile. Its role is to transmit the force and movement away the steering gear to the vertical pull rod or the tie rod, which in turn promotes the deflection of the steering wheel (M. M. Chen, J. 2016). As a preservation part of the automobile, the traditional steering droop and arm design generally have a high safety factor set to ensure structural safety, which leads to structural redundancy and material waste to some extent. Due to the need of energy conservation and environmental protection, the automobile lightweight has become a trend of automobile development, and the weight reduction of zero automobile parts is particularly important, it can easily and effectively reduce the quality of parts based on existing materials and processing technology, so it is more meaningful (G. N. Sun, J. 2017; C. F. Ma, L. H. Zhao, Z. Y. Yang, J. Z. Feng and S. L. Zheng, J. 2018), and on the basis of maintaining or improving its performance, it has great significance to consider the lightweight design ideas in the lightweight design of the steering arm.

In recent years, relying on the finite element analysis software platform, many scholars have made lightweight design of the steering arm, then according to the establishment of the finite element analysis method, the topology optimization design of the crane steering rocker reduces the mass of 2.56[kg], meet its safety design requirements (D. H. Wang, G.G. Cao, J. Li and Y. D. Deng, J. 2015). Other researchers use ANSYS to simulate and analyze the car knuckle arm is subjected to non-linear load conditions, and the equivalent stress distribution law of knuckle arm is obtained by using parametric commands. By modifying the arc radius of the region where the stress value is larger, the knuckle arm is optimized with a lightweight design, which provides a theoretical reference for the structural design of the knuckle arm (Y. Ma, H. W. Wang and L.L. Wei, J. 2014; Y. Ma, J. 2011). Researchers use ANSYS' topology optimization function to optimize the steering arm of a mining truck, the topology optimization structures under 18 typical operating conditions and the frequency of use of each operating condition is comprehensively selected to remove some materials of the steering swing arm. After a comparative analysis, the maximum stress under various steering angles of the optimized swing arm are greatly reduced whether it is thrust or tension, and the stiffness of the steering arm is not reduced to meet the optimal design requirements (L. Kong, Y. H. Shen. 2009).

Based on the previous research, this paper selects the top-to-bottom steering limit conditions of the steering arm to optimize the topology, and performs parametric modeling based on the
optimization results. The genetic algorithm is used to optimize the solution considering the mass, maximum stress and deformation of the steering arm. Through the finite element analysis of the pendulum arm model before and after optimization, it is demonstrated that the optimization results meet the safety design requirements and achieves the goal of lightweight design.

2. CALCULATING THE LOAD OF THE STEERING ARM

The research object of this paper is the middle steering arm of a certain type of double front axle commercial vehicle, the main role of which are force transmission, direction change and connection. Since the pressure loaded by the steering arm during operation is mainly derived from the tapered hole for assisting the oil cylinder and the tapered hole connected with the tie rod ball head, and in order to facilitate the calculation of the load and the loading method, the following assumptions are made in this design: it was supposed that the movement of the steering arm was a uniform rotation around the steering pin mounted on the frame, the acceleration of the steering arm was zero. Moreover, in analysis and topology optimization, the entire steering mechanism was balanced due to the transient conditions in each operating condition (J. H. Yang, J. 2018).

Therefore, the steering pinhole portion of the connection frame is fixed, and a fixed constraint is applied to the center of the steering pinhole to eliminate the tendency of the model rotation. In actual work, the main source of the moment in the middle steering arm is the booster cylinder, and a small part is due to the surplus torque of the power steering gear. However, the surplus torque of the power steering gear is difficult to calculate so that it is usually ignored.

The torque applied by the tire resistance moment to the middle boom is calculated by the transmission ratio of the transmission linkage. When turning in place, the tire resistance torque is calculated according to the empirical formula recommended by V•E•GOUGH as follows (Y. C. Wang, X. H. Gao, X. J. Zhang, J. 2010):

\[ M_s = \frac{0.7}{3} \sqrt[3]{\frac{34300^7}{7.4 \times 10^4}} = 1723.1 \text{[N-m]} \]

Calculated based on empirical values, the transmission ratios of the pull rod mechanism (from the steering arm to the upper arm) and the trapezoidal mechanism (from the upper arm to the trapezoidal arm) are: when the steering wheel turns left to the extreme position (left 35°), the gear ratio of the mechanism is \( i_b = 1.231 \), and the gear ratio of the trapezoidal mechanism is \( i_r = 0.694 \); when the steering wheel turns right to the extreme position (right 44°), the gear ratio of the pull mechanism is \( i_b = 0.835 \), and the gear ratio of the trapezoidal mechanism is \( i_r = 1.495 \). The moment acting on the steering arm as follows:

\[ M_p = \frac{1}{\eta r} \frac{\eta b}{\eta D} \cdot M_s \]  \tag{2}

Here, \( \eta_r \) refers to the trapezoidal efficiency, takes 0.8; \( \eta_d \) refers to the lever mechanism efficiency, takes 0.8.

Bring into the data: when the steering wheel turns left: \( M_p = 4736.3\text{[N-m]} \); when the steering wheel turns right: \( M_p = 4901.2\text{[N-m]} \).

The torque of the booster cylinder acting on the middle steering arm is calculated as follows: since the working cylinder of the commercial vehicle matched by the middle steering arm is 50mm, the connecting rod diameter is 25[mm], and the effective length of the middle steering arm connecting the cylinder is 250[mm]. Therefore, when the oil pressure of the booster cylinder is \( p = 13\text{MPa} \), and the efficiency \( \eta = 90\% \), the maximum output force of the cylinder is given as:

\[ F_s = P \times \frac{\pi D^2}{4} \times 10^{-4} \times \eta \]  \tag{3}

Bring into the data: \( F_s = 22972.5\text{[N]} \).

The output torque of the cylinder is:

\[ M_s = F_s \times H_i \]  \tag{4}

Bring into the data: \( M_s = 5901.6\text{[N-m]} \).

When the oil pressure of the booster cylinder is \( p = 13\text{[MPa]} \) and the efficiency is \( \eta = 90\% \), the compression output force of the cylinder is:

\[ F_y = P \times \frac{\pi (D^2-d^2)}{4} \times 10^{-4} \times \eta \]  \tag{5}
Bring into the data: \( F_y = 22972.5 \text{[N]} \).

Similarly, according to formula (4), the torque of the cylinder compression output can be calculated as: \( M_y = 5901.6 \text{[N·m]} \).

3. TOPOLOGICAL OPTIMIZATION OF STEERING ARM

The three-dimensional model of the intermediate steering arm is established and the topology optimization is performed to divide the designable area and the non-designable area. Since the relative positions of the steering knuckle and the steering trapezoidal part does not change, the connection part of the steering arm taper hole cannot be used as an optimization area, and the stress dispersion of the connection part between the pin holes are selected as the design area. So, the design area and the non-design area results are divided (Fig.1), the blue part of the figure is the design_prop, the yellow part is the nondesign_prop, and the white part is the rigid unit (reb2), its main node is the ball pin or the center point of the connecting rod. The tetrahedral mesh is selected as the mesh type, and the model are divided 117,340 nodes and 25,774 tetrahedral elements.

Fig.1 Topological optimization model for steering arm

Since the material of the steering arm is isotropic and it is continuum structure, therefore the topological optimization is performed with variable density method. The design variable is the relative density \( Xe \) of each element of the optimize region in the model (Fig.1), and the constraint defines the ratio of the volume of the steering arm to the designable structure to be 0.3 (M. F. Chen, W. Liu, J. 2013). In the Optistruk module, the minimum mass of the steering arm is selected as the objective function, and the optimization is calculated after convergence in step 17. The topology optimization results are viewed through the HyperView, and selecting the post-processing panel to read the topological results of step 17, as shown in Fig.3.

Fig.2 Topological optimization results density map of steering arm

The study results can be seen from the Fig.2 that the optimization results mainly retains the two sides of the connecting arm body, while the middle part are deleted in different degrees, especially the area connected to the side of the trapezoid, a large area is removed, and even hollowed out. The part of the structures forming void can be selected for cavity removal in the reconstruction of the model, and the materials removed from the upper and lower sides of the connecting steering cylinder are different and are not removed much.

4. PARAMETRIC MODELING OF STEERING ARM

The shape of the mesh is changed based on the deformation of the control block and the free deformation of the steering arm to obtain a parametric model of the steering arm. The parameter size of the steering arm model is selected as a variable to describe the special part of the model in detail (mainly based on topology optimization results for optimization), and by adjusting the variable parameters to obtain the parametric model of the steering arm, as shown in Fig.3.

Fig.3 Steering arm parametric model
In order to avoid the deformation of the whole configuration of the model caused by excessive deformation of a certain part, the size of the selected parameter variable is limited. According to the topology optimization results and the deformation mode of the grid deformation, the relevant parameter variation range is set as Table 1:

<table>
<thead>
<tr>
<th>Variable</th>
<th>Variable Description</th>
<th>Grid deformation method</th>
<th>Initial value</th>
<th>Lower limit</th>
<th>Upper limit</th>
</tr>
</thead>
<tbody>
<tr>
<td>DV1</td>
<td>1# Hole diameter scaling factor</td>
<td>Free deformation</td>
<td>1.0</td>
<td>0.8</td>
<td>1.2</td>
</tr>
<tr>
<td>DV2</td>
<td>1# Slot diameter scaling factor</td>
<td>Free deformation</td>
<td>1.0</td>
<td>0.9</td>
<td>1.1</td>
</tr>
<tr>
<td>DV3</td>
<td>2# Hole diameter scaling factor</td>
<td>Free deformation</td>
<td>1.0</td>
<td>0.8</td>
<td>1.2</td>
</tr>
<tr>
<td>DV4</td>
<td>2# Slot width scaling factor</td>
<td>Based on control block</td>
<td>1.0</td>
<td>0.9</td>
<td>1.1</td>
</tr>
<tr>
<td>DV5</td>
<td>3# Hole diameter scaling factor</td>
<td>Free deformation</td>
<td>1.0</td>
<td>0.8</td>
<td>1.2</td>
</tr>
<tr>
<td>DV6</td>
<td>1# Groove depth/mm</td>
<td>Free deformation</td>
<td>0</td>
<td>0</td>
<td>15</td>
</tr>
<tr>
<td>DV7</td>
<td>2# Groove depth/mm</td>
<td>Free deformation</td>
<td>0</td>
<td>0</td>
<td>15</td>
</tr>
</tbody>
</table>

According to the parametric modeling parameters and the variation range table of the steering arm of Table 1, it can be seen that there are many target parameters for size optimization. Moreover, the change of different parameters has diverse effects on the performance of the overall structure, and the optimization goal cannot be the optimal value of a single parameter. In order to obtain structural parameters that satisfy all objective functions, a multi-objective optimization function has to be established to solve the relevant parameter.

5. MULTI-OBJECTIVE OPTIMIZATION OF STEERING ARM BASED ON NSGA-II

Through the established parametric model of the steering arm, the shape parameters of some deformation slots or holes in the pendulum structure are selected as variables. And defined the range of variable parameters in the parametric model, this range of variation is used as a constraint function. Then considering that the steering arm structure is a safety component in the steering system and has higher requirements for safety performance, the maximum stress is selected to be less than 327.08[MPa] (the allowable stress when the safety factor is 2.4) and the maximum deformation of the joint is less than 1.133[mm] as a constraint (the original steering arm structure maximum deformation). In order to realize the lightweight design of the steering arm model, and the mathematical model with multi-objective.

The optimization of steering arm is obtained by selecting the minimum mass as the objective function as follows (G. C. Wang, N. Ma, X. C. GU, J. 2018):

\[
\begin{align*}
\min y &= (m(x), \sigma(x)) \\
\text{s.t.} & \sigma(x) \leq [\sigma] \\
& d(x) \leq D \\
& DV_{\text{min}} \leq DV \leq DV_{\text{max}}
\end{align*}
\]

Here, \(m(x)\) refers to the steering arm quality; \(\sigma(x)\) refers to the maximum stress under extreme conditions; \(DV_{\text{min}} \cdot DV_{\text{max}}\) refers to the upper-lower extreme values of each design variable of parametric model.

The NGSA-II is used to optimize the multi-objective problem, and the population size is set to 80, the evolution generation is set to 100, the crossover probability is 0.8, and optimizes it in Matlab with using the built-in gamultiobj function, then after 100 iterations, it can be seen from Fig.4 that the optimization iteration tends to converge around 70 times to obtain the optimal solution. Finally, the multi-objective optimization Pareto solution set for the steering arm is obtained, as shown in Fig.5.

By reference to the Pareto solution set, the value of each variable is selected as a specific parameter.
for optimizing the size (Andrew Robison and Andrew Vacca, J. 2018). Due to the characteristics of the steering arm machining process, there are certain requirements for the selection of the parameters in the designs of the forging die. According to the actual requirements of the project, rounding the relevant design parameter results as shown in Table 2:

![Fig. 4](image)

**Table 2. Optimization Results of Design Variables**

<table>
<thead>
<tr>
<th>design variable</th>
<th>DV1</th>
<th>DV2</th>
<th>DV3</th>
<th>DV4</th>
<th>DV5</th>
<th>DV6/mm</th>
<th>DV7/mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Optimization Results</td>
<td>1.0732</td>
<td>1.0310</td>
<td>0.9942</td>
<td>1.0587</td>
<td>1.0619</td>
<td>9.975</td>
<td>13.146</td>
</tr>
<tr>
<td>Rounding Results</td>
<td>1.07</td>
<td>1.03</td>
<td>0.99</td>
<td>1.06</td>
<td>1.06</td>
<td>10.00</td>
<td>13.00</td>
</tr>
</tbody>
</table>

6. **COMPARISON OF THE PERFORMANCE OF STEERING ARM BEFORE AND AFTER OPTIMIZATION**

Since the steering arm working condition is instantaneous, the vicinity of the middle turning pin hole connecting the frame and the vicinity of the two ends of the connecting tie rod are the main stress places during the working process. In order to make the optimized steering arm can work normally under the original conditions. The stress and displacement of the parts are mainly considered.

In addition, according to the data in Table 2, the model of the steering arm is remodeled. The finite element analysis of the steering arm before and after optimization is performed, and the stress and displacement of the steering arm under left and right steering conditions are studied. The displacement and stress distribution of the steering arm before optimization in the case of left turn and right turn (Fig.6) and the displacement and stress distribution of the steering arm after optimization, as shown in Fig.7.
The results are shown in Table 3:

<table>
<thead>
<tr>
<th>Structural model</th>
<th>Mass/kg</th>
<th>Maximum deformation /mm</th>
<th>Maximum stress /MPa</th>
</tr>
</thead>
<tbody>
<tr>
<td>Original structure</td>
<td>8.0620</td>
<td>1.133</td>
<td>237.62</td>
</tr>
<tr>
<td>Optimized structure</td>
<td>6.6217</td>
<td>1.155</td>
<td>288.80</td>
</tr>
</tbody>
</table>

From the above table, we can see that after the topology optimization and multi-objective size optimization design, the mass of the steering arm model has dropped significantly, and the weight loss ratio reaches 17.87%. At the same time, the maximum deformation of the optimized structure is changed from 1.133[mm] to 1.155[mm], the maximum stress is increased from 237.62[MPa] to 288.80[MPa], and the maximum stress concentration site is slightly shifted toward the connecting arm of the steering arm. According to the calculation method of the safety design coefficient, the optimized intermediate steering arm safety factor is 2.7 (X. J. Zhang, H. G. Ding, Y. F. Zhu, J. G. Zhao, J. K. Chen, J. 2013), which meets the safety design requirements.

7. CONCLUSION

This study aims to achieve lightweight design of the steering arm. The parametric design of the
steering arm is performed by using the topology optimization technique and the size optimization collaborative design based on multi-objective genetic algorithm. The results show that the mass of the steering arm optimized is reduced from 8.0620[kg] to 6.6217[kg]. At the same time, the safety is accord with design requirements, the overall optimization is better. Therefore, the research method can efficiently and accurately design the model and provide some reference for the structural optimization of other automotive components.

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9. REFERENCES