Optimization design of torsion beam structure of car based on fatigue life analysis

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Abstract: Taking the torsional beam of the car as the research object, the fatigue life of the torsional beam is analyzed through the simulation experiment, and the optimization scheme is selected according to the life of the torsional beam. Firstly, eight component thicknesses are selected as design variables by Plackett-Burman experimental design method, and the radial basis function neural network approximate model of the beam is established. Taking the maximum equivalent stress of the beam as the optimization objective, the beam mass, and the maximum deformation as the constraint conditions, the multi-objective optimization is carried out, and then two new structural design schemes are carried out for the parts with a rich life. The optimization results show that the maximum deformation is reduced by 15.9%, the maximum equivalent stress is reduced by 10.1%, and the overall life of the beam is extended by 40.6%.

1. Introduction
As one of the key parts in the torsional beam suspension system, the torsional beam not only needs to bear most of the mass of the body but also is impacted by irregular loads from different roads. In the long run, the fatigue failure of the beam will occur, seriously endangering the safety of life and property, so it is particularly important to study and optimize the fatigue life of the torsional beam Feng Yajie [1] predicted the fatigue life of a torsion beam and found the structure most likely to suffer fatigue damage, providing a target for the optimal design of beams for future generations. Bai XD [2] and Krishna PJ [3], in order to make the fatigue life prediction more accurate, ACE technology is used as an auxiliary tool. Not only that, in recent years, a large number of scholars [4.5] have also established approximate models to replace the complex multi-degree of freedom models in Engineering practice, which greatly improves the research efficiency and accuracy. Wang Qiang [6] Set up a response surface model to optimize the fatigue life of automobile rear axle housing to improve the fatigue life of automobile rear axle housing. After optimization, the life of automobile rear axle housing is extended by 17%, and the optimization effect is significant. In this paper, the torsional beam of a car is taken as the research object, and its fatigue life is predicted by software tools. According to the life situation, the the radial basis function neural network model is established by the finite element method to optimize the structural design of the torsional beam.

2. Finite element model of the torsion beam
Considering a large number of torsional beam parts, in order to get the accurate analysis results, it is necessary to simplify the beam reasonably. In the process of simplification, it is mainly to conform to the mechanical properties, so that the actual structure and each element remain the same, and the mechanical properties of element transfer are consistent with the actual[7]. After simplification, the finite
element model of the beam is shown in Figure 1

![Figure 1 finite element model of the torsion beam](image1)

3. Fatigue life analysis of torsion beam
A key step for fatigue life prediction of the torsion beam is to obtain the load spectrum. According to the actual working conditions and national standards of the vehicle, the relevant load spectrum is defined by referring to the relevant literature [8]. Limited to space, only the fatigue test load curve of the bending working condition is given, as shown in Figure 2.

![Figure 2 loading curve under bending condition](image2)

By loading the fatigue curves of two working conditions and adding the S-N curves of materials, the fatigue life of the torsion beam is predicted by using the fatigue analysis software ncode DesignLife. Because of using this software, we only need to select the materials used, the system will automatically generate the fatigue characteristic curve of the corresponding materials from its database, and can also automatically generate the fatigue characteristic curve according to the material properties and the internal empirical formula. Q345 is selected as the main material of the beam, and the S-N curve is shown in Figure 3.

![Figure 3 S-N curve](image3)

The Goodman average stress correction method [9] is used to simulate its fatigue life, and the simulation results are shown in Figure 6.7
It can be seen from figure 4 and 5 that the service life of the torsional beam under the bending condition is 290000 times, less than 300000 times in the standard experiment, so it needs to be optimized. The service life under torsion condition is 510000 times, which meets the requirements of fatigue life. For the torsion beam structure, there is not much room for change. If the structure is changed rashly, it will affect the comfort of the suspension and even the production process. Therefore, it is not recommended to change the structure. But it can reduce the stress, reduce the quality, and improve the anti-fatigue ability by changing the wall thickness of the beam.

4. Structural optimization of the torsion beam

There are many parts of the torsion beam. If all parts are designed as variables, the calculation is complex and it is easy to cause the situation of no solution. The Plackett Burman test design method can effectively screen out the sensitivity of variables to respond due to a large number of variables. Therefore, this method is used to screen 8 structural members including beams in this study. The design variables are the thickness of each member. See column 1 in Table 2 for the members.

The response surface approximation model uses a higher-order function polynomial to approximate the complex model, which has a more intuitive user experience than other optimization models. It can intuitively observe the impact of input parameters on the target parameters, and the accuracy of the results is good. In general, multivariate polynomials are used to fit the input variables and response values. The lower order polynomials are:

\[ y_i = f_i(x) = \sum_{k=1}^{M} w_{ik} \phi_k \left( \| x - c_k \| \right) \]  \hspace{1cm} (1)

\[ \phi_k \left( \| x - c_k \| \right) = \exp \left[ \frac{(x - c_k)^T (x - c_k)}{2\sigma_k^2} \right] \]  \hspace{1cm} (2)

X is the input vector; \( \phi \) is the base function; \( w \) is the weight coefficient; \( c_k \) is the center of the KTH node; \( \sigma_k \) is the base width parameter of the KTH node.

Through formula (1) and (2), eight component variables are collected by using the Hammersley design method, and the approximate relationship between eight design variables and three responses of the torsional beam is fitted by the program as its radial basis function neural network model. The three responses are beam mass, maximum deformation, and maximum equivalent stress. Limited to space, only part of the fitted response surface model of the torsional beam is listed here, as shown in Figure 6.
Whether the later optimization results are correct depends on whether the approximate model is accurate, so it is necessary to verify the established approximate model. In this paper, complex correlation coefficient, root mean square error (RMSE) and average relative error (RAAE) are used as evaluation criteria. The approximate model test table is obtained by extracting sample points of test matrix, as shown in Table 1.

Table 1 Sample point response error

| response                        | $R^2$   | RAAE  | RMSE  |
|---------------------------------|---------|-------|-------|
| The quality of the response     | 0.9996  | 0.0376| 0.0093|
| Maximum total deformation response| 0.9623  | 0.0256| 0.0138|
| Maximum equivalent stress response| 0.9949  | 0.0283| 0.0017|

According to the evaluation standard, the closer $R^2$ is to 0, the greater the value of RAAE and RMSE is, the worse the accuracy of the fitted approximate model is. It is generally believed that $R^2$ greater than 0.9 indicates that the actual model is close enough to the approximate model. As we can see from the table, the approximation model created is accurate enough.

From fatigue, life cloud see, trailing arm is the torsion beam fatigue risk, but here the thickness affects the biggest deformation of beam size, quality, and maximum stress, by increasing the size of its thickness can reduce the maximum stress, the quality of the beam may increase, however, in order to prevent this kind of situation, will be about the quality of the beam, the maximum deformation of constraints. The optimized mathematical model is shown in Equation (3).

\[
\begin{align*}
\text{obj. } & \min (\text{Stress}) \\
& \min (\text{Mass}) \\
& \min (\text{Deformation}) \\
\text{s.t. } & \text{Stress} \leq 345 \text{MPa} \\
& \text{Mass} \leq 24.5 \text{kg} \\
& \text{Deformation} \leq 1.98 \text{mm} \\
& t_{ia} \leq t \leq t_{ib} \quad (i = 1, 2, \ldots, 8)
\end{align*}
\]

$t_i$ —— design variable; $t_{ia}$, $t_{ib}$ —— Upper and lower limits of design variables.

According to Equation (3), the beam is constrained and multi-objective optimization is carried out on the beam in Ansys. The simulation results are shown in Figure 7. Taking quality, maximum deformation, and maximum equivalent stress as evaluation indexes, the three groups of data were simulated and verified respectively. After a comprehensive comparison, the third group was determined as the optimal solution. The optimized design variables are shown in Table 2.
Table 2 design variable value after optimization

| Variable name       | Variable code | Initial Value (mm) | optimization value (mm) | Rounding value (mm) |
|---------------------|---------------|--------------------|-------------------------|---------------------|
| beam P1             | 4             | 3.7285             | 3.7                     |
| Torsion bar P2      | 3             | 2.7917             | 2.8                     |
| liner tube P3       | 4             | 3.6816             | 3.7                     |
| Longitudinal arm P4 | 4.5           | 5.1341             | 5.1                     |
| Hub seat P5         | 5             | 5.1652             | 5.2                     |
| Sidewall of spring seat P6 | 4 | 4.3029 | 4.3 |
| Spring seat bottom P7  | 5.5          | 5.0631             | 5.1                     |
| Beam reinforcement P8 | 4             | 4.4145             | 4.4                     |

It can be seen from the life cloud figure 4 and 5 that some structures still have rich strength, such as the beam, so the beam can be further optimized to achieve the purpose of lightweight. In this paper, two kinds of weight reduction holes are designed, which are special-shapred holes and elliptical holes respectively. The schema diagram, shown in Figure 8.

![Figure 8 design drawing of weight reduction hole](image)

The three schemes are analyzed and calculated respectively. When each scheme is optimized, the full load bending condition is the boundary condition. The comparison of the optimal results of the three layout schemes is shown in Table 3. According to the data in the table, the ellipse hole is selected as the best design.

Table 3 Comparison of three layout optimization results

| program | optimization value | rounding value |
|---------|--------------------|----------------|
| Mass(kg) | Deformation (mm) | Stress (MPa) | Mass(kg) | Deformation (mm) | Stress (MPa) |
| a       | 23.907             | 1.6701         | 299.43   | 23.908             | 1.6701         | 299.43   |
| b       | 23.842             | 1.6712         | 299.81   | 23.843             | 1.6712         | 299.81   |

5. Analysis of optimized fatigue life

Based on the optimized experimental data, the finite element model was rebuilt and the fatigue life prediction was performed again under the same experimental simulation parameters. The simulation results are shown in Figure 9 and Figure 10.

![Figure 9 cloud chart of life under bending condition](image)

![Figure 10 cloud chart of life under torsion condition](image)

As can be seen from Figure 9 and Figure 10, under two different working conditions, the fatigue life of the beam is more than 400,000 times and more than 880,000 times respectively, both of which meet the service life requirements. In order to observe the optimization effect, the optimized results of the
torsional beam are compared with the original model analysis results under bending and torsion conditions. See Table 4 for comparison.

|                         | Before optimization | After optimization | Variation          |
|-------------------------|---------------------|-------------------|--------------------|
| (bending) maximum stress| 333.12Mpa           | 299.82Mpa         | Reduce by 10.1%    |
| (torsion) maximum stress| 204.35Mpa           | 181.6Mpa          | Reduce by 11.25%   |
| (bending) maximum deformation | 1.9803 mm          | 1.6713mm          | Reduce by 15.9%    |
| (bending) fatigue life  | 5.1x10^4            | 8.8x10^4          | Increase by 72.4%  |
| (torsion) fatigue life  | 29.02x10^4          | 40.8x10^4         | Increase by 40.6%  |

6. Conclusion

By predicting the fatigue life of a torsion beam of a car, it is found that the fatigue life of the beam does not meet the service life requirements under bending condition, and then the optimization design is carried out. First of all, through the establishment of RBF neural network approximation model of the beam to beam with the optimization target of the maximum equivalent stress of the beam quality and maximum deformation as constraint conditions of multi-objective optimization, the second for the part of the life rich hole design scheme of two kinds of new weight loss through simulation analysis, the elliptical hole is the best design solution. The optimization results show that the maximum equivalent stress is reduced by 10.1%, the maximum deformation is reduced by 15.9%, the overall life of the beam is extended by 40.6%, and the optimization effect is significant.

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