Performance analysis and optimized design of a certain frame

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Abstract. In recent years, energy shortages and environmental pollution have become increasingly prominent, and automobile lightweight has become an important field in the development of the automobile industry. This paper analyses the frame performance of an all-terrain single-seat off-road vehicle and optimizes the structure design. The weight of the new structure has been reduced by 4.084 kg, and the weight reduction rate has reached 12.83%. The statics analysis and modal analysis of the new frame structure show that the new structure still meets the requirements of usage.

1. Introduction
Truss frames are mostly used in racing and off-road vehicles. The lightweight of the frame is an important means to improve the performance of the vehicle and realize energy saving and emission reduction. Under normal circumstances, there are three ways to reduce the mass of the truss frame. One is to change the materials used, such as high-strength steel or high-quality alloy steel. At present, composite materials such as carbon fiber materials and plastic materials are also widely used in lightweight cars. Another method is to improve the processing technology and use new manufacturing methods to replace traditional processing methods, such as using TWB technology, hot stamping technology, etc. The last method is to optimize its structure, using optimization methods such as finite element analysis and mathematical models to improve the structure.

Many experts and scholars at home and abroad have done a lot of research on the lightweight of truss structures. In order to study the lightweight of the escalator truss structure, Hongbing Zhang et al. [1] first used the finite element method to calculate the performance of the original structure, and then analyzed the sensitivity of the structural parameters to further optimize the design. In order to reduce the weight of the steel pipe frame, Qi Zhou et al. [2] adopted the form of carbon fiber wound lining steel pipe to achieve partial replacement of the material, and the final mass was reduced by 23%. Yali Ma et al. [3] adopted the base structure method to design the truss structure, and realized the lightweight structure while meeting the performance requirements.

This paper takes a small off-road vehicle frame as the research object to establish a three-dimensional model, and uses ANSYS to carry out static and modal analysis on the frame according to actual typical working conditions. Then optimize the design based on the variable density method, and check the performance of the new frame structure to verify its rationality.
2. Finite element analysis

2.1. Build a finite element model
The frame of this single-seater, all-terrain off-road vehicle is composed of three parts: the front cabin, the cockpit and the rear cabin. The selected materials is 30CrMo, material properties: density $7.85 \times 10^3 \, \text{kg/m}^3$, elastic modulus $2.11 \times 10^{11} \, \text{N/m}^2$, Poisson’s ratio 0.279, yield strength $7.85 \times 10^8 \, \text{N/m}^2$, and the initial mass is 31.837 kg. The rods that make up the frame are thin-walled parts, so quadrilateral plate and shell elements are used when meshing. The mesh tool of ANSYS is used to mesh the frame, and finally 119498 elements and 234446 element nodes are obtained. The finite element model is shown in Figure 1.

![Figure 1. Frame finite element model](image)

2.2. Static analysis
The statics analysis mainly analyzes the structure from the three aspects, Hooke's law, the coordinated conditions of displacement and the conditions of static balance. And the performance of the structure is checked by calculating the displacement, stress and strain.

The load applied in this paper is mainly based on the actual load of the vehicle, which is applied to the corresponding rod in a combination of uniform load or concentrated load, and also some small mass loads are ignored. The load of each component of the vehicle is shown in Table 1.

| Part Name            | Mass/kg |
|----------------------|---------|
| Frame                | 38      |
| Driver and Seat      | 80      |
| Motor Controller     | 5       |
| Motor                | 15      |
| Battery Box          | 50      |
| Decelerator          | 10      |

When a car is running, it is mainly subjected to dynamic load, which is measured by a certain dynamic load coefficient. The driving speed, the structural parameters of the car and the road conditions determine the magnitude of the dynamic load factor. The calculation formula of the dynamic load factor is:

$$ n = 1 + \frac{K}{g} \times \frac{C_1}{1 + \frac{C_2}{v^2}} $$

(1)

Where: $K$ is the stiffness of the suspension system; $C_1$ is the road constant; $C_2$ is the experience coefficient; $v$ is the driving speed.

Substituting the parameters for calculation, this article takes $n = 1.5$. 

2
This paper selects two typical working conditions to analyze, which are steering and bending and torsion. In the steering condition, the maximum steering acceleration is taken as 0.4g. In addition to the vertical load, the frame also bears the equivalent dynamic load of the inertia force on the frame during turning. In bending and torsion conditions, the impact of inertial load on the frame is very small and can be ignored.

The calculation result is shown in the figure below. The maximum equivalent stress in steering condition is 104.78MPa, and the maximum equivalent elastic strain is $5.1964 \times 10^{-4} \text{mm}$. The maximum equivalent stress in bending and torsion conditions is 92.466MPa, and the maximum equivalent elastic strain is $4.8929 \times 10^{-4} \text{mm}$. The maximum stress is far below the yield limit of the material, so the material will not undergo plastic deformation, and the structural performance meets the requirements of usage.

![Figure 2. Equivalent stress cloud diagram for steering condition](image)

![Figure 3. Equivalent elastic strain cloud diagram for steering condition](image)

![Figure 4. Equivalent stress cloud diagram under bending and torsion conditions](image)
Figure 5. Equivalent elastic strain cloud diagram under bending and torsion conditions

2.3. Modal analysis

2.3.1. Theoretical basis of modal analysis. Through the static analysis of the frame, it is found that the strength and rigidity meet the requirements of usage. In order to better understand the performance of the frame, a modal analysis is necessary. Modal analysis is an approximate method to study the dynamic characteristics of the structure. By analyzing the dynamic characteristics of the frame, it is helpful to judge the rationality of the finite element model and provide a basis for optimal design.

The frame is simplified to a linear structure with multiple degrees of freedom, and its dynamic differential equation [4, 5] is

\[
[M]\ddot{x}(t) + [C]\dot{x}(t) + [K]x(t) = F(t)
\]  

(2)

Where:
- \([M]\) is quality matrix;
- \([C]\) is damping matrix;
- \([K]\) is stiffness matrix;
- \(\ddot{x}(t)\) is acceleration vector;
- \(\dot{x}(t)\) is velocity vector;
- \(x(t)\) is displacement vector;
- \(F(t)\) is force vector;
- \(t\) is time variable.

This article analyzes the free mode of the frame, so the external load on the frame is not considered. The effect of damping on the structure is relatively small, so it can be assumed that the damping matrix is zero, and the dynamic differential equation can be rewritten as

\[
[M]\ddot{x}(t) + [K]x(t) = \{0\}
\]  

(3)

The solution can be expressed as

\[
x(t) = A \sin(\omega t + \varphi)
\]  

(4)

Where: \(A\) is amplitude; \(\omega\) is natural frequency; \(\varphi\) is initial phase.

Substituting (4) into (3) to get the frequency equation

\[|\left[K\right] - \omega^{2}\left[M\right]| = 0\]  

(5)

The free vibration characteristics of the structure can be obtained by solving the eigenvalues and eigenvectors of the matrix.

2.3.2. Modal result analysis. The excitation of electric vehicles mainly comes from the unevenness of the motor and the ground. The excitation of the road to the vehicle is mainly vertical excitation, and the frequency range is generally between 0 and 20 Hz. The excitation of the motor is mainly caused by the mechanical movement of the rotor inside the motor. The frequency is mainly determined by the number of poles and the speed of the motor. The calculation formula is as follows:

\[
f = \frac{na}{60}
\]  

(6)
Where: \( n \) is working speed of motor; \( a \) is the number of pole pairs of the motor.

The working speed range of the motor is 3000~6000r/min. The number of poles of the motor used in the vehicle studied in this paper is 3, and the excitation frequency of the motor can be calculated as 150~300Hz through (6).

In the free modal analysis of the frame, the frequency should avoid the excitation frequency of the road surface and the motor, that is, the frequency range should be between 20Hz and 150Hz. The free modal analysis results are shown in Table 2. Since the first 6-order frequency of the frame is close to 0, it can be regarded as a rigid body mode, so it is not described one by one. The minimum free mode frequency of the last 6th order is 72.928Hz and the maximum is 121.78Hz, which are all within the ideal frequency range, and it will not cause resonance of the frame. Its dynamic performance is relatively good, which can meet the requirements of vibration performance.

### Table 2. Free mode simulation results

| Order | Modal Frequency /Hz | Mode                | Maximum Deformation /mm |
|-------|----------------------|---------------------|-------------------------|
| 1~6   | 0                    | Rigid body mode     | 0                       |
| 7     | 72.928               | Local bending mode  | 14.303                  |
| 8     | 80.758               | Local torsion mode  | 14.721                  |
| 9     | 82.325               | Overall bending mode| 9.2203                  |
| 10    | 115.23               | Overall torsion mode| 12.275                  |
| 11    | 118.64               | Bending and torsion mode | 14.101             |
| 12    | 121.78               | Bending and torsion mode | 11.219             |

3. Structure optimization

3.1. Modelling

Optimal design refers to transforming a physical model into a mathematical model, rationally using mathematical theories, making computers as tools, and fully considering various constraints to obtain an optimal design that can meet the design goals. Optimal design needs to grasp three elements, namely objective function, state variables and constraints. Structural optimization is divided into shape optimization, size optimization and topology optimization. Shape optimization refers to adjusting the shape and inner boundary size of the structure design domain while keeping the topological relationship of the structure unchanged, and seeking the most ideal geometric shape of the structure. The size optimization makes the shape of the structural parts or the shape of the holes as the optimization object to find the most suitable overall layout. Topology optimization is a mathematical method for optimizing material distribution in a given area according to the given load conditions, constraints and performance indicators. Common topological optimization methods include variable density method, progressive structure optimization method, and homogenization methods etc.

In this paper, the variable density method is used to optimize the structure design. The variable density method discretizes the structure into multiple units. Assuming that each unit is homogeneous, the density \( \rho_i \) varies from 0 to 1. If \( \rho_i \) is closer to 0, it means that the part needs to be discarded. Close to 1, it means that the part should be kept. This paper takes the material density \( \rho_i \) as a design variable, takes allowable stress and allowable deflection as constraints, and minimizes the volume of the structure (that is, the minimization of mass) as the optimization goal. The mathematical model is established as follows:

\[
0 \leq \rho_i \leq 1 (i = 1, 2, 3, ..., n) \\
Min \quad f_0(x) = f(x_1, x_2, x_3, x_4, ..., x_n) \\
\sigma_{max} \leq [\sigma] \\
\omega_{max} \leq [\omega] 
\] (7)
Where: $\rho_i$ is density of structural units; $f_i(x)$ is structure volume; $\sigma_{\text{max}}$ is maximum stress after structural optimization; $\omega_{\text{max}}$ is maximum deflection after structural optimization.

3.2. Result analysis
The optimal iterative curve of the turning condition is shown in Figure 6, the curve converges at the 12th iteration. And the optimal iterative curve of the bending-torsion condition is shown in Figure 7, the curve converges at the 10th iteration. The optimization results of the two working conditions are independent of each other. It is necessary to combine the optimization results of the two working conditions to design a frame structure that can meet various working conditions at the same time.

![Figure 6. Iterative curve for bending and torsion conditions](image)

![Figure 7. Steering condition iteration curve](image)
4. New structure design
This paper analyzes two working conditions of the frame. It is necessary to combine the optimization results of the two working conditions and calculate the weighting coefficient to design a structure that meets the conditions of multiple working conditions.

Considering the manufacturability in the actual machining process and the existing 30CrMo steel pipe specifications on the market, combined with the optimized design results, the frame model is re-modeled in CATIA software. The new model is shown in Figure 8.

![Figure 8. Model after optimization](image)

Import the new model into the workbench software and use the same method to analyze its performance, then compare the results of the analysis with the previous data. The data comparison table is shown in Table 3.

|                         | Before Optimization | Optimized | Rate of change |
|-------------------------|---------------------|-----------|----------------|
| Mass/kg                 | 31.837              | 27.753    | 12.83%         |
| Steering conditions     |                     |           |                |
| Maximum equivalent stress /MPa | 104.78            | 147.9     | 41.15%         |
| Maximum equivalent elastic strain /× 10^-4 mm | 5.1964           | 7.9024    | 52.07%         |
| Bending and torsion conditions |                     |           |                |
| Maximum equivalent stress /MPa | 92.466            | 102.75    | 11.12%         |
| Maximum equivalent elastic strain /× 10^-4 mm | 4.8929           | 5.5932    | 14.31%         |
| 1st~6th mode /Hz        | 0                   | 0         | 0              |
| 7th mode /Hz            | 72.928              | 52.152    | 28.49%         |
| 8th mode /Hz            | 80.758              | 53.565    | 33.67%         |
| 9th mode /Hz            | 82.325              | 81.631    | 0.84%          |
| 10th mode /Hz           | 115.23              | 89.902    | 21.98%         |
| 11th mode /Hz           | 118.64              | 93.923    | 20.83%         |
| 12th mode /Hz           | 121.78              | 104.16    | 14.47%         |

The optimized structure is 27.753 kg, which is 12.83% lighter than the previous structure, and achieves a good lightweight effect. The maximum equivalent stress of the optimized frame is 147.9MPa. Although it is greater than the maximum equivalent stress of the previous frame structure, it is still within the yield limit of the material, and the structure is safe and reliable. In addition, the 7th to 12th
order modes of the frame are lower than the previous structure, but they are still between the ground excitation frequency and the motor excitation frequency. The optimized structure is safe.

5. Conclusion

Through the finite element analysis of the frame structure to check the performance, then optimize the structure and re-verify its performance. The quality of the new structure is reduced by 12.83%, the performance can also meet the requirements of usage, and the goal of lightweight is well achieved. The optimized design process can provide a certain reference for other structural optimization designs.

References

[1] ZHANG Hong-bing, WAN Chang-dong, SHANG Guang-qing, DU Jian-hong. Lightweight analysis of automatic escalator truss structure with FEM [J]. Mechanical and Electrical Engineering, 2012, 29 (10), pp.1139-1142.

[2] Zhou Qi, Ma Qihua, Zhou Tianjun. Lightweight design of load-bearing frame with CFRP winding steel tube [J]. Modern manufacturing engineering, 2020(04), pp.57-63.

[3] Ma Yali, Niu Jinyue, Liang Chen. The complex structural part design method based on truss structure optimization [J]. Mechanical Design and Manufacturing Engineering, 2020, 49 (11), pp.21-25.

[4] JIANG LiHong, WU QingJie. Performance analysis and lightweight design of a front subframe [J]. Mechanical Strength, 2020, 42 (06), pp.1503-1508.

[5] YANG WeiPing, HOU Liang, CAI HuiKun, LI ShengYu, ZHANG EnLai, WANG WenWu. Structural optimization of the excavator engine hood based on modal analysis [J]. Mechanical Strength, 2016, 38 (03), pp.537-542.