The Mechanical Modal Analysis of the Main Motor Supporting Plate and the Optimal Design on the Penalty Function with the Interior Point Method

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Keywords: Parametric three-dimensional model, Strength constraint, Penalty function interior point, Supporting plate, Optimization design.

Abstract. The main motor supporting plate is the connector of super power wire sawing machine between main motion mechanism and the main motor with force concentrated and complex forces, high level of requirements in stress and rigidity, having a great influence on the reliability of the total machine. Based on the physics and mechanics modeling, the current study is carried out on the mechanics and modal analyses. The theory of optimization is adopted, on the premise of meeting the requirements of reliability under the condition of lightness as the goal with the strength and stiffness constraints, basic size of the plate as decision variables, using the penalty function interior point method, for analysis and design optimization. The actual application shows that the design not only guarantees the reliability of the overall working condition of the superpower rope saw, but also reduces the quality of the plate by 9.5%.

Introduction

Diamond-bead rope is the high hard and flexible tool of the sawing machine for quarry and stone processes with diamond and steel wire [1]. High power and large power wire saw as one of raw stone mining equipment development direction, the main motor plate stress concentration and complex, high strength, stiffness requirements, is the key-module system, has a great influence on the whole machine function and reliability, the need for reliability and optimization design. Osman et al. [2-4] studied the concrete ways on increasing the performance of chain saw machines for mechanical excavation of marbles and natural stones. Cao Hanqing et al. [5] by using finite element analysis developed a optimization design of the structure and fatigue analysis for improvement of the safety and reliability of the transmission shaft. W. Penger. Et al. [6] analyzed and comprehensively assessed the stability, loads, and processing quality of the stone cutting with a view to achieve the best parameters and cutting processes. Daniel Meissner et al. [7,8] optimized the structure, function, and technological parameters of the various working conditions of the rope saw machine and achieved desired results. Song Jinling et al. [9] used the penalty function method to optimize the feed system of the rope sawing machine with safety and reliability.

Physics-mechanics Models and Static Analysis

Physics and Mechanics Models

Known from the mechanical analysis that the driving wheel position adjusting mechanism is mainly for the main motor of gravity and quarry mining with the rope tension, the super power mine rope saw machine driving wheel force is larger and more complex. The physical model in Fig. 1 is transformed into a mechanical model. The o-x-z, o-y-z plane projection is shown in the graphical coordinate system, as shown in Fig. 2. G indicates the weight of the simplified plate with the positive direction of Y-axis. Fc is the tension of the string of beads in the working process, with the negative direction of Z axis and through the center of gravity \( G = (715,0, -280) \). \( G \) represents the gravity of the main motor and the active wheel, its direction is the positive direction of Y, through...
gravity center $N(125,0,-280)$. $G_{zz}$ is the gravity of stepping motor and supporting seat in the translation mechanism with direction of the Y-axis and through the center of gravity $W(-600,0,0)$.

Figure 1. Views of the movement mechanism. Figure 2. Mechanics model of the plate.

Put the above forces onto the plate, then the force diagram may be given in Fig. 2.

**Static Analysis**

The commonly dynamic equation is

$$[M][\ddot{x}]+[C][\dot{x}]+[K][x]=\{F(t)\}$$  \hspace{1cm} (1)

where $[M]$ is the mass matrix, $[C]$ the damping matrix, $[K]$ the coefficient stiffness matrix, $\{x\}$ the displacement vector, and $\{F(x)\}$ force vector. In the static structural force analysis, the force has nothing to do with time, so the displacement may be solved from

$$[K][x]=\{F\}$$  \hspace{1cm} (2)

The plate is structure steel with Yang’s modulus $2 \times 10^{11}$ Pa, poisson ratio 0.3, density 7850 kg/m$^3$. The comprehensive analysis is with Hex Dominant Method, and local analysis is with Face Sizing or Edge Sizing for the grid. The node number is 1823732 and element number is 1283314.

The maximum deformation of the plate in X direction is 0.003437 mm and it occurs at the top of the plate. The maximum deformation of in Y direction is 0.0091407 mm and it locates the bottom of the plate. Some of the Maximum deformations are shown in Fig 3. The greatest stress is at the connection of the plate and the rod, and the maximum stress is the shear stress with the value of 75.262 MPa. The allowable stress is $[\sigma]=175$ MPa, so the plate meets the strength requirement.

Figure 3. The clouds of maximum deformation in all directions.

**Modal Analysis of the Supporting Plate**

**Free Vibration Equation**

The free vibration without damping for a structure or a machine as the discreted system may be expressed in terms of

$$[M][\ddot{x}]+[K][x]=\{0\}$$  \hspace{1cm} (3)
where $[M]$ is the mass matrix, $[K]$ the coefficient stiffness matrix, $\{x\}$ the displacement vector, and $\{\ddot{u}\}$ the acceleration vector.

we suppose

$$\{x\} = \{\phi\} \sin \omega t$$ (4)

where $\{\phi\}$ is the amplitude of displacement, $\omega$ is the natural frequency. Substitute equation (4) into equation (3), we have

$$([K] - \omega^2 [M])\{\phi\} = 0$$ (5)

This equation is an eigenequation with non-zero vector $\{x\}$. The necessary condition of the non-zero solution of it is the coefficient determinant

$$\text{det}([K] - \omega^2 [M]) = 0$$ (6)

Equation (6) is extended and a polynomial with respect to $\omega^2$ of N-th power is obtained. Solutions $\omega_i$ ($i = 1, 2, ..., n$) are substituted into equation (5), then the response of system modal vector or modal vector is reached.

**Modal analysis**

The low-order vibration natural frequency has greater influence on structural vibration than that of the high-order frequency. Therefore, only the first six order natural frequencies and vibration modes are taken into consideration in the current modal analysis of plate. The natural frequencies for the first six ranks are 186.83, 227.71, 358.73, 367.17, 435.44, and 452.16Hz. and the modes for the fifth and sixth rank vibration are in Fig. 4.

![Modal analysis](image)

Figure 4. The fifth and sixth rank vibration styles of the plate.

**Optimal Analysis and Design with the Penalty Function Interior Point Method**

**Objective Function and Constraints**

Optimization design is based on the principle of determination of the design variable, objective function and constraint function for optimization mathematical model and solution. The general form of optimization of mathematical model is as follows:

$$
\begin{align*}
\text{min } F(X) &= F(x_1, x_2, \cdots, x_n) \\
g_i(X) &= g_i(x_1, x_2, \cdots, x_n), (i = 1, 2, \cdots, M) \\
X &= (x_1, x_2, \cdots, x_n)^T
\end{align*}
$$

(7)

where $X$ is the design variables, $F(X)$ the objective function, $g_i(X)$ the state variable.

From the original problem, we have the following expression
\[
P(X,r^k) = \hat{f}(X) + \frac{1}{r^k} \sum_{i=1}^{i} \frac{1}{g_i(X)}
\]

where \( r^k \) is the penalty factor, \( g_i(X) \) is the constraint function.

### Determination of the Decision Variables and Target

The optimization design of plate runs by topology optimization for removal of three holes with the length and width as variables \( X_1, X_2, X_3 \), thickness \( X_4 \), and rib dimension \( X_5 \). The selected design variables are continuous and change of these variables is set within default value 10% of the Design Exploration in the Workbench platform.

### Optimal Results and Data Analysis

Prior to optimization, the be 3D software UG should be connected to simulation software Workbench, and Variables \( X_1, X_2, X_3, X_4, X_5 \) ought to be parameterized, and set respectively as \( DS.Length, DS.Length1, DS.Width, DS.Width1, \) and \( DS.Width2 \). Workbench is used to calculate the three optimized candidate design schemes based on the experimental calculation procedure, as shown in Table 1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Candidate A</th>
<th>Candidate B</th>
<th>Candidate C</th>
</tr>
</thead>
<tbody>
<tr>
<td>( X_1 )</td>
<td>153.95</td>
<td>153.81</td>
<td>153.89</td>
</tr>
<tr>
<td>( X_2 )</td>
<td>560.21</td>
<td>560.45</td>
<td>560.18</td>
</tr>
<tr>
<td>( X_3 )</td>
<td>13.607</td>
<td>13.635</td>
<td>13.61</td>
</tr>
<tr>
<td>( X_4 )</td>
<td>16.792</td>
<td>16.13</td>
<td>14.062</td>
</tr>
<tr>
<td>( X_5 )</td>
<td>384.8</td>
<td>384.9</td>
<td>384.91</td>
</tr>
<tr>
<td>Mass</td>
<td>190.47</td>
<td>190.51</td>
<td>190.52</td>
</tr>
<tr>
<td>Total Deformation Maximum</td>
<td>0.0510</td>
<td>0.0512</td>
<td>0.0527</td>
</tr>
<tr>
<td>Equivalent Stress Maximum</td>
<td>10.579</td>
<td>10.656</td>
<td>11.792</td>
</tr>
</tbody>
</table>

Select Candidate A, and analyze the stress and strain before and after optimization. As shown in Table 2, the mass of the plate is reduced by 9.5% compared with the initial plan.

### Table 2. Comparison of the parameters before and after the optimization.

<table>
<thead>
<tr>
<th>Stage</th>
<th>Design variable</th>
<th>State</th>
<th>Target function</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( X_1 ) (mm)</td>
<td>( X_2 ) (mm)</td>
<td>( X_3 ) (mm)</td>
</tr>
<tr>
<td>Initial design</td>
<td>140</td>
<td>510</td>
<td>15</td>
</tr>
<tr>
<td>Optimal design</td>
<td>153.95</td>
<td>560.21</td>
<td>13.61</td>
</tr>
<tr>
<td>Change rate</td>
<td>9.9%</td>
<td>9.8%</td>
<td>10.2%</td>
</tr>
</tbody>
</table>

### Conclusions

The physical model and mechanical model has been established of the main motor supporting plate based on the analysis of the shift mechanism of the active wheel position of the overpower mine rope sawing machine. In the UG environment, a three-dimensional parametric model was then developed and the static analysis and modal analysis were performed with the Workbench.

Based on the theory of optimization, with the premise of the plate reliability the lightweight as the goal, strength and stiffness requirements as constraints, length, width of three holes on the plate, \( X_1, X_2, X_3 \), wall thickness \( X_4 \), and the rib dimension \( X_5 \) as decision variables, the penalty function point method was adopted for optimal analysis and design. As a result, the mass of the plate was reduced by 9.5%.
Acknowledgement

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References


