Structure optimization of DTH hammer piston based on RBF neural network and WSO algorithm

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Abstract: To address the issue of piston fracture due to insufficient strength in the Down-The-Hole (DTH) hammer. The effect of the pressure state of the gas chamber on the piston motion is considered. A piston motion model is established based on the steady flow energy equation and the working principle of the piston. The finite element method is used to analyze the dynamic response of the piston during the impact of the drill bit. RBF neural network is employed to construct a surrogate model for the relationship between piston structural parameters (front-end diameter, front-end length, and center bore) and piston strength and performance (impact energy, frequency, and gas consumption). By combining this model with the white shark optimization search algorithm, the design of the DTH hammer's piston structure parameters is optimized. The minimum piston mass is the objective function. Impact energy, frequency, air consumption and stress are constraint factors. The effect of piston life is considered while meeting rock drilling efficiency. The optimization results show that the stresses in the optimized piston structure are reduced by 13%. The proposed RBF performance prediction model combined with the optimal search algorithm can significantly improve the optimization efficiency.

0 Introduce

The Down-the-Hole (DTH) hammer is a new type percussion drilling tool driven by air. Compared with traditional cutting and grinding drilling tools, the DTH Hammer has stronger drilling efficiency as well as adaptability to complex formations, and it has become an efficient drilling tool that countries are competing to develop[1]. The piston is driven by high-pressure gas to impact the drill bit to achieve drilling. However, the piston is subjected to a strong shock at each impact. The piston often occur fracture in engineering applications. At present, for the problem of piston fracture of DTH hammer, some scholars studied mostly focused on the transient dynamic response of the simplified piston in the cyclic motion and the study of the metallographic material of the piston fracture [2-6], for the research of the transient impact characteristics of the piston collision instant. Bu[7] studied the transient impact ring of the hammer rock drilling system with finite elements method. Sun[8] simplified the piston collision problem to a one-dimensional flexible rod structure to study the magnitude of impact force of piston collision. Ding[9] transformed the collision process of hammer piston and bit into a binary flexible rod co-axial impact process. Previous studies on the piston structure of the DTH hammer and the dynamic response of the piston during impact are less relevant, while the structural discontinuity of the piston is also the key to its stress concentration, so it is necessary to study the effect of the piston structure of the submersible hammer on the dynamic response of the piston at the moment of collision.
In this paper, the piston motion mathematical model is established to determine the boundary conditions of the piston impact the bit. The process of the piston impacting the brazier head is simulated by finite element software. The effect of the instantaneous dynamic response of the collision of the piston structure of the valveless submersible hammer is investigated. The position of the piston fracture in the field is verified to be due to an unreasonable structural design. Performance prediction model are created for piston structural parameters versus hammer rock drilling efficiency and piston stress. The optimal structural parameters of the piston are searched for through the White Shark optimization algorithm to optimize the piston size on the basis of satisfying the working performance. The method improves the optimization efficiency, and the research method and results can provide a reference for the piston structure optimization design of the DTH hammer.

1 Working principle and piston failure description

As shown in Fig. 1, the DTH hammer mainly consists of air distribution device, a piston, a drill bit, a guide sleeve, etc. The piston is driven into the return phase by the high pressure gas that enters the front chamber through the annular air chamber. The intake passage of the front air chamber is switched off when the annular air chamber between the piston's long air hole and the outer casing is closed. The piston continues to move under inertia. The gas in the rear chamber is compressed, the air pressure increases, and the piston moves in a decelerating motion until the velocity is zero. At this point, it reaches the impact stage. The piston is pushed by the pressure in the rear chamber and moves towards the drill bit.

The type of piston under the actual working load, during 6 to 11 cycles (single cycle for 7h) the piston appeared several times front-end fracture, the cracking areas are shown in Fig. 2. Piston fracture location was microscopically observed. The piston material is of good casting quality and there are no visible pores, gaps or inclusions on the fracture surface. Structural failure due to unreasonable material or workmanship is excluded.

2 Piston simulation analysis

2.1 Mathematical model
The working principle of the piston is composed of its front, rear and annular air chamber inlet and exhaust processes. The state of the gas chamber gas, the gas flow in and out of the chamber and the
movement of the piston are closely related. Fig. 3 depicts the forces on the piston.

At this stage, it is difficult to establish a mathematical model that conforms to the actual working conditions using theoretical methods, so in order to obtain a reasonable model of the piston movement, the following assumptions are made\textsuperscript{[10-13]}: (1) A quasi-equilibrium process in which the gas chamber is artificially equilibrated with an ideal gas. (2) The gas change process is an adiabatic process. (3) Ignore energy loss during impact.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{fig3.png}
\caption{Force analysis of piston}
\end{figure}

Piston acceleration in the return stage:

\[ \frac{d^2 s}{dt^2} = \frac{P_1 A_1 - P_2 A_2}{M} - g \cos \alpha \]  

(1)

Piston acceleration in the impact stage:

\[ \frac{d^2 s}{dt^2} = \frac{P_2 A_2 - P_1 A_1}{M} + g \cos \alpha \]  

(2)

Where: \( P_1, P_2 \) are the pressure of the front and rear chambers respectively. \( A_1, A_2 \) are the effective working area of the front and rear chambers respectively. \( s \) is the displacement of the piston movement process, \( t \) is the time; \( M \) is the mass of the piston; \( g \) is the acceleration of gravity; \( \alpha \) is the angle between the acceleration of gravity and the tool.

Pressure changes in the gas chamber:

\[ \frac{dp_i}{dt} = 1.41 \mu p_i \left[ \frac{A_i}{V_i + A_i (X_i \pm s)} \frac{ds}{dt} - \frac{dm}{dt} \right] m^{-1} \]  

(3)

\[ 0.1 \text{MPa} \leq p_i \leq p_n \]  

(4)

Where: \( \mu \) the adiabatic coefficient, \( s \) is the piston displacement. \( V_i \) is the volume of each chamber. \( dV/\text{dr} \) is the initial gas distribution length of each chamber, \( \mu \) is the mass flow rate of gas per unit time; is the air pressure supplied by the air compressor.

For the calculation of the mass flow rate of the gas in the chamber before and after inflow or discharge, the standard orifice flow equation for the speed of sound and subsonic velocity is used, as judged by the ratio of \( p_i/p_j \).

When \( p_i/p_j > 0.5283 \):

\[ m = 0.15478 \eta A_p p_i \left[ \frac{1}{T_0} \left( \frac{p_i}{p_j} \right)^{k+1} - \left( \frac{p_i}{p_j} \right)^{2} \right] \]  

(5)

\[ \frac{dm}{dt} = 0.15478 \eta A_p x \frac{p_i}{\sqrt{T_0}} \frac{dt}{dt} \]  

(6)

When \( 0.5283 < (p_i/p_j) \leq 1 \):

\[ \frac{dm}{dt} = 0.405 \eta A_p \frac{p_i}{\sqrt{T_0}} \frac{dt}{dt} \]  

(7)

Where: \( p_i, p_j \) is the absolute front air chamber and rear air chamber gas pressure. \( \eta \) is the flow coefficient. \( T_0 \) is the absolute temperature of air chamber. \( R \) is the air constant. \( K \) is the adiabatic index.

The time-varying mathematical motion model of the piston is established, based on a 5-inch hammer
as an example. There is 1.8 MPa pressure provided by the air compressor, and the 5-inch hammer is taken as an example. The instantaneous velocity of the piston before collision and other performance parameters (Impact energy, Frequency and Gas consumption) are calculated based on the operating conditions. The calculation results are shown in Tab. 1:

<table>
<thead>
<tr>
<th>Working performance</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>End of impact velocity (m/s)</td>
<td>9.6</td>
</tr>
<tr>
<td>Impact energy (J)</td>
<td>986.3</td>
</tr>
<tr>
<td>Frequency (Hz)</td>
<td>25.17</td>
</tr>
<tr>
<td>Gas consumption (m³/min)</td>
<td>22.65</td>
</tr>
</tbody>
</table>

2.2 Finite element calculation model

Using the explicit dynamics module of the finite element method, the piston strength response is analyzed throughout the impact. The density of the material is 7800 kg/m³, the modulus of elasticity of \(2.11 \times 10^{11}\), Poisson's ratio of 0.3 and Yield strength of 1292 MPa.

The dynamic response of a piston in collision with a drill bit is investigated. In order to improve the computational efficiency, the drill bit model is simplified without affecting the accuracy of the piston stress calculation. The distance between the piston and the drill bit was set to 0.001 m. The effect of friction on the contact surface is neglected. Air pressure: 1.8 MPa. The drill bit is set to a fixed constraint. The piston is set to move at a speed of 9.6 m/s and in a direction that is vertical to the drill bit. The finite element model of the piston and drill bit impact is shown in Fig. 4.

2.4 Piston stress calculation results

The stress cloud of the hammer piston is calculated by the finite element model, as shown in Fig. 5. Maximum stress value of the piston is 951.47 MPa, the safety factor \(n=1.36<1.5\), which does not meet the design safety requirements. The location of the maximum stress is at the variable section of the piston, which coincides with the location of the piston fracture in the field. The significant increase in stress values at the variable section of the piston is due to the structural discontinuity, away from which the stress decreases rapidly.

Fig. 6 shows the curve of stress versus time increment for the piston as a whole and at the variable section respectively. The curves at the piston as a whole and at the variable section are consistent. When the piston does not hit the front end of the drill bit, the piston force is basically negligible. At the moment when the piston collided with the drill bit, the stress wave spread from the front face, and the piston stress rose sharply.
The piston is subjected to such alternating stresses due to cyclic shocks throughout its operation, which leads to mechanical fracture of the piston.

3 Analysis of piston structure parameter

The piston is fracture-prone at the variable cross-section, which has been verified by field fracture and simulation analyzes. The structural parameters in the vicinity of the fracture section are chosen without affecting the overall dimensions of the hammer. As shown in Fig. 7. The effect of front-end diameter $D$, center bore diameter $d$ and front-end length $L$ on the strength of the piston and on the performance of the hammer is investigated.

Fig. 8 shows the piston stress and performance at $D=75$ mm to $D=90$ mm front-end diameter of the DTH piston. Due to the reduction of the volume of the front air chamber (the diameter of the front face increases), the area of the piston affected by the gas is reduced. A larger diameter of the front face will lead to a lower frequency, impact energy, and air consumption, and the effect is approximately linear. The influence of the front-end diameter on the frequency is weakened. It is worth attention that the stress decrease is due to the increasing contact area of the piston with the drill bit. Piston performance and stress are significantly affected by the diameter of the front face.
Fig. 9 depicts the piston stress and performance at d=28 mm to d=38 mm center bore diameter of the DTH piston. The volume of the front and rear gas chambers is independent of the diameter of the center bore, but the contact area when the pistons collide is related to the diameter of the center bore. The gas flow through the center bore is determined by the regulating plug. The effect of the center bore diameter on frequency and gas consumption is negligible. Due to the increase in center bore diameter, the front face collision area is smaller, which leads to increased piston stress. Larger center bores will result in lower piston weights, which in effect cause lower piston impact energy.

![Fig. 9 Stress and performance under different center bore diameter](image)

Fig. 10 shows the piston stress and performance at $L=71.89$ mm to $d=81.89$ mm center bore diameter of the DTH piston. The volume of the front air chamber is affected the front-end length of piston, and the change in volume causes a consequent change in velocity. With the increase of the length, while the increase in the length of the front face leads to an increase in the volume of the front chamber, which leads to the increase of the impact energy and the decrease of frequency. The transmission of stress waves during a collision is related to the length. The increase in piston velocity leads to micro flexure of the piston during the collision, the maximum stress of the piston decreases with the length of the front face first, then increases, and then decreases.

![Fig. 10 Stress and performance under different front end length](image)

4 Piston performance prediction based on RBF neural network

4.1 RBF Neural Networks

RBF neural network is a non-linear fitted forward neural network through local exponential decay, consisting of input, implicit, and output layer. The Euclidean distance between the input value and the center value of the sample point is calculated by the implicit layer as the independent variable of the function. Combined with the radial basis function as the activation equation of the function, linear weighted combinations are applied to the output layer to achieve the prediction of the sample data.

The Gauss function is used for the radial basis function, and i function in the implicit layer is:
\[ G_j(x_i, c_j) = \exp \left( -\frac{\|x_i - c_j\|^2}{2\sigma_j^2} \right) \]  

where: \( i = 1, 2, \ldots, m; j = 1, 2, \ldots, n \); is the \( i \) input sample point; \( m \) is the number of sample points; \( n \) is the number of implicit layer neurons; \( c_j \) is the center of the sample point; \( \sigma_j \) is the width of the sample point, and \( \|x_i - c_j\| \) is the Euclidean distance from the sample data to the center.

The implicit layer and the output layer are linearly fitted to the output by a linear weighting method:

\[ y_k = \sum_{j=1}^{n} w_{jk} G_j(x_i, c_j) \]  

where: \( k = 1, 2, \ldots, l \); \( y_k \) is the \( k \) neuron of the output layer; \( w_{jk} \) is the connection weighting of the \( j \)-implicit layer to the output layer \( k \).

4.2 Dataset construction

In the construction of the piston performance prediction model dataset, a sample point design method is used by combination of central composite test design and Latin hypercube sampling. A total of 150 sample points are selected for the design variables. The data set is constructed by calculating the relationship between structural parameters and output variables through finite element simulation and piston motion model. The random method is adopted to divide 120 training samples and 32 test samples, and the range of values of the design variables is shown in Tab. 2:

<table>
<thead>
<tr>
<th>Variables</th>
<th>Symbol</th>
<th>Initial</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Front-end diameter</td>
<td>( D )</td>
<td>80</td>
<td>70-90</td>
</tr>
<tr>
<td>Centre bore</td>
<td>( d )</td>
<td>43</td>
<td>25-45</td>
</tr>
<tr>
<td>Front-end length</td>
<td>( L )</td>
<td>90</td>
<td>70-110</td>
</tr>
</tbody>
</table>

The structural parameters of front-end diameter, center bore diameter and front-end length are selected as input variables, with frequency Hz, impact energy J, air consumption m³/min and stress MPa as output prediction data.

4.3 Results analysis

The accuracy of the RBF performance prediction model has a large impact on the accuracy of subsequent optimization results. It needs to be subjected to error analysis. In this paper, the test set divided is tested in the prediction model. The error between the predicted results of the model and the true value is calculated. The decision coefficient \( R^2 \) and RMSE are adopted to evaluate the performance prediction model, and the calculation formulae are as follows:

\[ R^2 = 1 - \frac{\sum_{t=1}^{n} (y_t - \hat{y}_t)^2}{\sum_{t=1}^{n} (y_t - \bar{y}_t)^2} \]  

\[ \text{RMSE} = \sqrt{\frac{\sum_{t=1}^{n} (y_t - \hat{y}_t)^2}{n}} \]

where \( y_t, \hat{y}_t, \) and \( \bar{y}_t \) are the test set sample data, the performance prediction model response values and the test set data average.

The value of \( R^2 \) is taken as \([0, 1]\), and the closer the value is to 1 the higher the performance prediction accuracy of the model. The models are verified using 32 test samples. As shown in Fig. 11 (a), the \( R^2 \) of each performance prediction model is close to 1. This is very high than the engineering
requirement of 0.9.

The RMSE means that the prediction model fits well for the structural geometry parameters design variables (front-end diameter, center bore diameter and front-end length) in relation to the response of the hammer frequency, impact energy, air consumption and stress performance parameters and can be used to predict piston performance for different design variables. The results are shown in Fig.11 (b).

5 Structure parameters optimized design

First proposed by Malik Braik and inspired by the hunting style of one of the ocean’s most dangerous predator groups, the White Shark Optimization Algorithm (WSO) is based on the natural behavior of white sharks in hunting (ultra-sensitive hearing and smell, prey localization). White sharks are unique and efficient in avoiding local minima to speed up global optimality in solving complex objective functions due to their constantly changing biometric position. Therefore, this paper uses the white shark algorithm
combined with the RBF piston performance prediction model for simultaneous multi-parameter optimization of the hammer piston, the process of which is shown in Fig. 12.

![Fig. 12 Flow chart of optimal design of monopile foundation](image)

The minimum volume of the piston is taken as the objective function. The front-end diameter, the center bore diameter and front-end length are the optimization design variables. Based on safeguarding the original performance. The working performance and strength design requirements of the hammer are the constraint functions. The optimization function of the piston structure parameters can be described as:

$$\text{Min } F(X) = (\pi \times (d_1^2 + d_2^2)) \times L$$

subject to:

- $J_{\text{min}} \geq 986.13$
- $H_{\text{min}} \geq 25$
- $Q_{\text{max}} \leq 25.01$
- $\sigma \leq 861.3$
- $70 \leq d_1 \leq 90$
- $25 \leq d_2 \leq 45$
- $70 \leq L \leq 110$

$$\text{(12)}$$

<table>
<thead>
<tr>
<th></th>
<th>Before optimization</th>
<th>After optimization</th>
</tr>
</thead>
<tbody>
<tr>
<td>Front-end diameter</td>
<td>80</td>
<td>82</td>
</tr>
<tr>
<td>Center bore diameter</td>
<td>35</td>
<td>26</td>
</tr>
<tr>
<td>Front-end length</td>
<td>90</td>
<td>86</td>
</tr>
<tr>
<td>Piston stress MPa</td>
<td>951.47</td>
<td>841.93</td>
</tr>
<tr>
<td>Impact energy J</td>
<td>986.13</td>
<td>987.01</td>
</tr>
<tr>
<td>Frequency Hz</td>
<td>25.934</td>
<td>25.52</td>
</tr>
<tr>
<td>Gas consumption m³/min</td>
<td>25.01</td>
<td>23.62</td>
</tr>
</tbody>
</table>

If all the finite element analysis is used in the optimization process, the finite element software would be called upon at least 12,000 times, far more than the number of calculations required to construct the surrogate model, and the finite element analysis would consume a large amount of computing time and computer resources.
Fig. 13 Diagram of piston structural parameters

Fig. 13 and Tab. 3 depict the stresses and operating performance of the optimized piston. The piston structural parameters are optimized by the proposed method, on the basis of meeting the requirements of the hammer performance (impact energy, frequency and air consumption).

The performance of the piston before optimization is compared and the stresses constrain the optimization of the structural parameters more significantly. Fig. 14 depicts that the stress at the variable section of the piston is 841.93 MPa. The stress reduction of 109.54 MPa is a change of 13.01%.

6 Conclusion

The law of conservation of energy, the ideal gas state, and the constant flow energy equation are used to develop a mathematical model of piston motion. The velocity of the piston when it impacts the drill bit is calculated. The dynamic response of the piston colliding with the bit is analyzed and it is verified that the field test fracture location is the maximum stress location of the piston. The RBF performance prediction model is combined with the WSO white shark optimal search algorithm to propose an optimized design method for piston structures that takes into account the effects of machining accuracy. An integer optimization of the front-end length, front-end diameter and center bore diameter is achieved. The method is applied to the structural optimization design of an actual submerged hammer and the following conclusions are obtained:

(1) The failure of this type of piston is a mechanical fracture caused by the stress concentration at the variable section of the piston under the impact. Concentrations of stress at variable section is considered in the design of pistons.

(2) The DTH piston optimization design is carried out. Compared to traditional optimization methods, the use of RBF proxy models greatly reduces the number of simulation analyses. Improved optimization efficiency, shorter development cycles and reduced development costs.

(3) The optimization method proposed in this paper is based on not changing the overall structure of DTH. The piston performance requirements are achieved and its service life is extended.

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Data availability

The authors confirm that the data supporting the findings of this study are available within the article.

Conflicts of interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Reference

Structures, 2000, 76(5).


