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Analysis and Optimization of Mud Pump Crankshaft Structure based on Nonlinear Regression Model

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Abstract: The mud pump is an important power component of petroleum drilling. The crankshaft, as an important part of the mud pump, requires optimization of the crankshaft structure in order to improve the efficiency of the mud pump. First analyze the workload of the crankshaft, and obtain the stress point and load size of the crankshaft work. Through the finite element simulation, the corner of the crank rotation and the maximum deformation of the crankshaft and the maximum stress is obtained. Combining the production process to open the hole, use the non -linear regression model to establish the relationship between the crankshaft structure optimization target and the opening diameter of the opening, and use multiple target optimization methods to find the optimal variable. The maximum stress, maximum variable volume, and the total mass optimization effect of the crankshaft is obvious, reducing product costs.

Keywords: Mud Pump; Crankshaft Load; Structural Optimization; Multi-objective Optimization.

1. Introduction

As one of the eight components of oil drilling, the mud pump is called the heart of the oil rig and is an indispensable key component of the oil drilling platform. China's mud pump structure borrowed from the U.S. structure technology to meet the needs of domestic drilling operations, the current large-scale use of high-pressure triplex pumps in China, triplex pump structure technology has been very mature [1], as shown in Figure 1 for the working principle of triplex mud pump.

With the development of the oil industry and the continuous progress of drilling technology, drilling mud pump indicators are also rising, especially for the structure of the mud pump itself, the need for more narrow mud pump equipment to complete the requirements of more power, displacement and pressure. Therefore, it is important to optimize the structure of the mud pump. The crankshaft is the main component of the mud pump, and the three-dimensional structure of the crankshaft is shown in Figure 2. The forged crankshaft used in the three-cylinder mud pump, in which the arbor, eccentric wheel, flange and other parts are assembled using the inlay structure. In recent years, the inlay structure or assembled structure of the crankshaft design is gradually adopted new structure, which is widely used abroad, most of the parts are forgings, and the flange is undercut using the profile. Forged crankshafts have good reliability because of their lighter material weight and symmetrical structure, which avoids additional shocks caused by eccentricity, and forged crankshafts also have the characteristics of other types of crankshafts at the same time, with higher mechanical properties, impact properties, and arbor strength [2]. The part size and fatigue life of the crankshaft have a huge impact on the working volume and reliability of the mud pump [3]. During the drilling process, the crankshaft is subjected to cyclic loads that generate alternating stresses and torques, and once the crankshaft is damaged, the mud pump cannot continue to work. The crankshaft is the internal commutation mechanism of the mud pump to convert circular motion into linear motion, and as a key component of the mud pump, the structure of the crankshaft directly determines the efficiency of the mud pump.

In this paper, the crankshaft structure of the mud pump is studied and analyzed with the existing mud pump as the research object, and the nonlinear regression model [4] is used in combination with the simulation results of the finite element model to analyze the crankshaft structure of the mud pump and optimize its structural parameters to reduce the mass, maximum deformation and maximum bearing stress of the crankshaft on the basis of the original performance of the mud pump, so as to improve the work efficiency and reduce the economic cost of drilling operation.

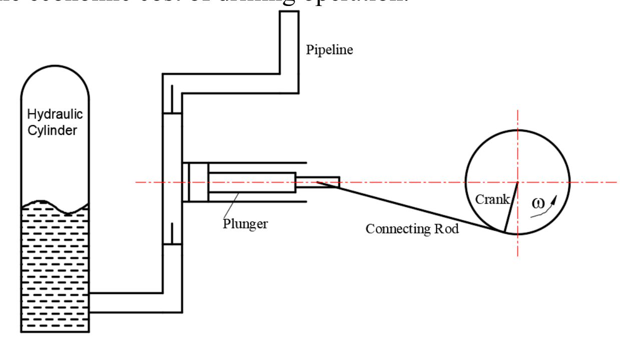

Fig 1. Working principle of three-cylinder mud pump

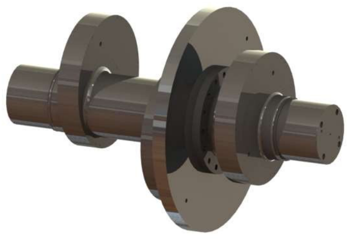

Fig 2. 3D model of the crankshaft

2. Crankshaft Loading Analysis

The load of the crankshaft is associated with the crank angle. Using commercial CAE software, a finite element model is established to analyze the load of the crankshaft at different crank angles, so as to determine the maximum stress and torque the crankshaft is subjected to, and the crank angle is read out at the maximum stress and torque respectively, and all simulation data are collected to provide a data base for subsequent structural optimization calculations.

ANSYS Workbench platform is used to firstly establish the crankshaft 3D model through SolidWorks, then import the 3D model into ANSYS, perform parameter interaction settings in ANSYS Workbench, and finally perform static analysis on the crankshaft. The crankshaft is divided by systematic free meshing.

The force analysis of the crankshaft is shown in Fig. 3. A set of crankshafts connecting rod is used as the object of study, and the direction shown in Fig. 3 is the positive direction, and the connecting rod is set as a plane motion, and the connecting rod is regarded as a rigid body, and the force decomposition of the crank connecting rod mechanism is carried out. The differential equations of rigid body plane motion are used to derive the loads of the piston and crankshaft on the connecting rod, respectively, expressed by the following equation [5]:

$$F_{1x} = \frac{m_2 a + m_2 g f_2 J_2(\varphi) - F}{\sqrt{1-\lambda^2\sin^2\varphi+\lambda\sin\varphi f_2 J_2(\varphi)}} \times \sqrt{1-\lambda^2\sin^2\varphi} \quad (1)$$

$$F_{1y} = -\frac{m_2 a + m_2 g f_2 J_2(\varphi) - F}{\sqrt{1-\lambda^2\sin^2\varphi+\lambda\sin\varphi f_2 J_2(\varphi)}} \times \lambda\sin\varphi \quad (2)$$

$$F_{3x} = -F_{1x} - m_2 a_{cx} \quad (3)$$

$$F_{3y} = -F_{1y} + m_3(\mathrm{g} - a_{cy}) \quad (4)$$

$$F = -pA + (0.06pA)J_1(\varphi) \quad (5)$$

$$J_1(\varphi) = \begin{cases} 1, & 0 < \varphi < \pi \\ -1, & \pi < \varphi \le 2\pi \end{cases} \quad (6)$$

$$J_1(\varphi) = \begin{cases} 1, & 0 < \varphi < \varphi_0,\ \pi < \varphi \le 2\pi \\ -1, & \varphi_0 < \varphi \le 2\pi \end{cases} \quad (7)$$

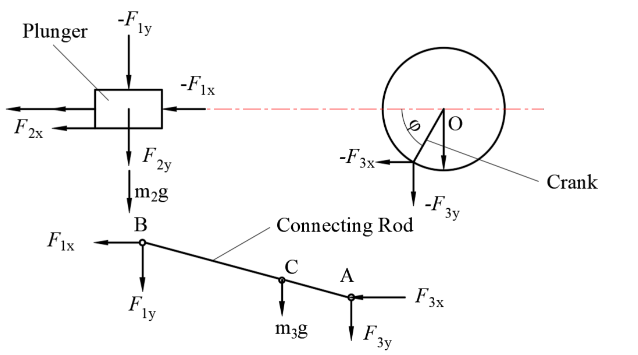

Fig 3. Force analysis of the crank-link mechanism

Table 1. Meaning of symbolic parameters

Symbol	Meaning	Symbol	Meaning
F_{1x}, F_{1y}	Piston and crosshead forces on connecting rod	F_{3x}, F_{3y}	The force of the crank to connecting rod
φ	Crank angle of rotation $\varphi = \omega t$	λ	Stranging link ratio
m_2	The total mass of piston and cross components	m_3	Linked rod mass
a	Pistons and cross - headed components accelerate	a_{cx}, a_{cy}	Acceleration of the connecting rod
f_2	Pistons and external dynamic friction coefficients $f_2 = 0.1 - 0.12$	F	Pistons thrust pA and Pistons' friction F_f joint force
p	Hydraulic cylinder pressure	A	Piston cross-sectional area
$J_1(\varphi), J_2(\varphi)$	Crankshaft rotation angle function	φ_0	Pump valve hysteresis angle

The variables involved in the formula are shown in Table 1, and after the above formula calculation, the size of the load on the crankshaft can be analyzed, and the calculated data are shown in Table 2.

Table 2. Crank force data

Crank angle (°)	Piston Combined Force (KN)	F_{3x}(KN)	F_{3y}(KN)
0	690.207	37.100	6.727
30	690.207	31.422	10.709
60	690.207	16.513	14.688
90	976.100	-1.55	17.313
120	976.100	-16.768	16.829
150	690.207	-26.030	12.847
180	690.207	-762.398	6.762
210	976.100	-765.683	-50.977
240	976.100	-761.080	-93.779
270	690.207	-747.602	-108.763
300	690.207	-727.799	-91.618
330	976.100	-708.232	-48.840
360	690.207	36.123	6.762

3. Finite Element Simulation

3.1. Formation of Finite Element Models

According to the existing three-cylinder mud pump in the structure of the crankshaft for three-dimensional modeling, in Solidworks to draw a three-dimensional model, and import the model into ANSYS, using ANSYS Workbench platform, in ANSYS Workbench for parameter settings, and finally the crankshaft for static analysis, the crankshaft using the system free mesh division, as shown in Figure 4 for the finite element model to divide the mesh.

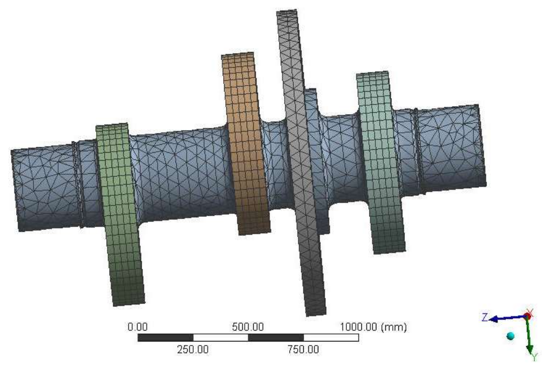

Fig 4. Finite element mesh model

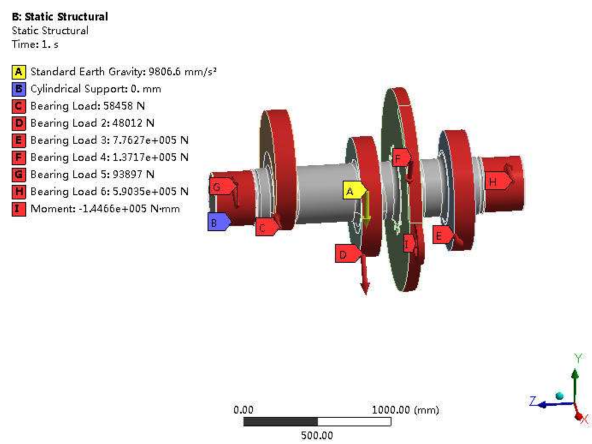

Fig 5. Finite element model boundary condition setting

It can be seen from the force analysis that the load of the crankshaft is composed of supporting force and gear sets of the left and right bearing branches, the meshing force of the gear set, the force of the rod to the crank, and the eccentric mass inertia caused by the eccentric structure of the crankshaft itself. In setting the boundary condition, fix the left and right ends of the crankshaft, the constraint shaft displacement, and the radial is not constrained; the meshing force of the gear set is decomposed into the X direction, Y direction, and Z direction. The position of meshing, so the meshing force is directly set on the surface of the crankshaft; the inertial force is reflected in the self -weight of the crankshaft, and the setting of the boundary condition is shown in Figure 5. Use the data of Table 2 to apply loads. Under the corners of different crank, the maximum stress and maximum deformation of the crankshaft.

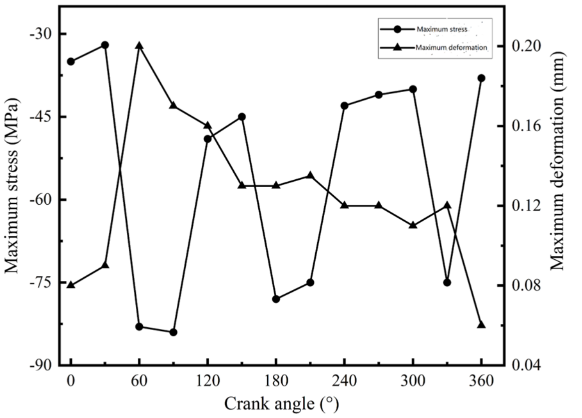

Fig 6. Maximum deformation and crankshaft maximum bearing stress with crank rotation angle

3.2. Finite Element Simulation Results Analysis

As shown in the figure, when the crank is turned from 0° to 30°, the maximum bearing stress of the crankshaft does not change much and the maximum deformation increases slightly. when it is turned from 30° to 60°, the maximum bearing stress of the crankshaft increases steeply to 80 MPa, while the maximum deformation rises sharply to 0.2 mm. from the crank turning angle of 60° to 360°, the maximum deformation decreases gradually and the maximum stress value fluctuates up and down. It is difficult to read the relationship between the crank angle and the other two variables in the graph. In this paper, the data are fitted by mathematical software, with the crank angle as the independent variable and the dependent variables as the maximum stress and maximum deformation, respectively. However, the curve fitting result is not satisfactory, and the maximum R-squared is 0.624, which is too much different from the original data. Therefore, in the existing structure, the maximum stress and maximum deformation of the crankshaft cannot change regularly with the crank angle and cannot be expressed by the functional relationship, so the structure needs to be optimized to make the maximum stress and maximum deformation of the crankshaft structure rise or fall regularly.

4. Structure Optimization of Crankshaft

A nonlinear type regression model was used to establish the relationship between the input and output variables of the crankshaft, and the regression model main effect model is shown in Eq. [6]

$$y = \beta_0 + \sum_{i=1}^{k} \beta_i x_i + \varepsilon \tag{8}$$

where β is the coefficient of each variable in the regression model, k is the number of input variables, and ε is the residual term. When the interaction between the input variables in the main effects model has a direct effect on the output variables, the interaction term of the input variables needs to be added to the main effects model to express the relationship between the input variables and the output variables.

$$y = \beta_0 + \sum_{i=1}^{k} \beta_i x_i + \sum_{i<j} \sum \beta_{ij} x_i x_j + \sum_{i=1}^{k} \beta_i {x_i}^2 + \varepsilon \tag{9}$$

In this paper, the optimized design of the structure is carried out by opening a hole in the crankshaft, as shown in Figure 7, with the three variables of the diameter of the opening circle of the eccentric wheel No. 1, the diameter of the opening circle of the eccentric wheel No. 2 and the diameter of the opening circle of the eccentric wheel No. 3 as the output variables.

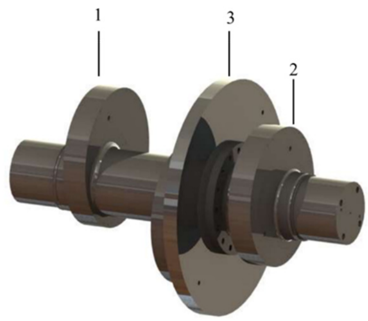

Fig 7. Input variables on the crankshaft

According to the laser processing working conditions and combined with the original dimensions of the crankshaft, the setting range of the input variables is shown in Table 3, and the range is divided into 10 intervals, and the midpoint of each interval is taken for finite element simulation to calculate the maximum deformation and maximum stress of the crankshaft. Using ANOVA, the variables with the greatest influence on the maximum deformation and maximum stress of the crankshaft among the input variables are calculated.

Table 3. Input Variable Value Ranges

Variables	Median values	Upper limits	Lower Limits
No.1 eccentric wheel opening circle diameter	90	100	80
No.2 eccentric wheel opening circle diameter	80	90	70
No.3 eccentric wheel opening circle diameter	300	350	250

In a three-cylinder mud pump, the three crankshaft variations are not synchronized and the three input variables have a high and low degree of influence on the three crankshafts. In order to optimize the structure and thus reduce the maximum crankshaft deformation, the maximum stress and the total crankshaft mass, the input variables of the crankshafts need to be optimized.

Table 4. Input Variable Value Ranges

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No.1 eccentric wheel opening circle diameter	90	100	80
No.2 eccentric wheel opening circle diameter	80	90	70
No.3 eccentric wheel opening circle diameter	300	350	250

Table 5. Comparison of Optimized Solutions

Options	Option 1	Option 2	Option 3
Maximum stress change	2.9%	2.3%	2.0%
Maximum deformation change	5.13%	6.92%	7.2%
Total mass	6.44%	4.12%	5.21%

As shown in Table 4 for the structural optimization of the crankshaft, the three structural optimization schemes were analyzed for the total crankshaft mass, maximum crankshaft deformation, and maximum stresses sustained by the crankshaft. The difference between the three optimization schemes is not obvious from Table 5, so the multi-objective optimization method is needed to further investigate the optimization problem and find the optimal solution after weighing the three optimization objectives of total crankshaft mass, maximum deformation, and maximum stress.

Table 6. Multi-objective optimization weights

Optimization Objectives	Maximum stress	Maximum deformation	Total mass
Weights	33.67%	20.97%	45.36%

Table 7. Multi-objective optimization weights

Optimization Objectives	Maximum stress	Maximum deformation	Total mass
Weights	33.67%	20.97%	45.36%

Table 8. Optimize data comparison before and after

Optimization Objectives	Maximum stress (MPa)	Maximum deformation(mm)	Total mass(kg)
Before Optimization	83.63	0.19651	4393
After Optimization	79.91	0.19449	4032
Change after optimization	4.65%	1.03%	6.95%

The entropy weight method [7] was used to explore the three optimization objectives and the objective weights of the optimization objectives were obtained as shown in Table 6, so after the multi-objective optimization method was carried out, the comparison of the three optimization objectives before and after optimization was worth as shown in Table 7. From Table 7, it can be seen that the maximum deformation of the crankshaft, the total mass and the maximum stresses sustained after optimization are reduced, and the maximum deformation of the crankshaft is still within the strength tolerance range, so this optimized structure can be accepted.

5. Summary

(1) This paper takes the three-cylinder mud pump as the research object, establishes the mechanism of crankshaft stress, carries out the static analysis, finds out the distribution of the main crankshaft load, establishes the finite element simulation model according to the working characteristics of the three-cylinder mud pump, finds that the maximum stress of the crankshaft reaches the peak at 60° of the crank rotation angle, and the maximum deformation is also here, but there is a large safety margin to meet the engineering requirements.

(2) The crankshaft structure was optimized by using nonlinear regression equation and multi-objective optimization method, and the optimization improvement of three input variables of eccentric wheel opening circle diameter of No. 1, No. 2 and No. 3 was completed. The optimization results showed that the maximum stress, maximum deformation and total mass were reduced compared with the original ones, which satisfied the optimization target requirements.

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