

OPTIMIZATION DESIGN AND ANALYSIS OF THE DOWNHOLE HYDRAULIC RELEASING TOOL

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The releasing tool is one of the key components used with the expanded liner hanger. Aiming at the problem that the existing downhole hydraulic releasing tool can't meet the working requirements of gas wells in Sichuan and Chongqing area, the key part of the releasing tool - elastic claw is optimized in this paper. The optimization results show that when the load-bearing bevel angle of the elastic claw is 30°, the number of claw pieces is 10 and the percentage of gaps on the circumference is 25%, the structure of the elastic claw is optimal.

Keywords: Downhole Releasing Tool; Elastic Claw; Finite Element Simulation; Optimization Design

1. Introduction

The releasing tool is one of the key components used with the expansion liner hanger. The releasing tool directly affects the success of the releasing operation. For the expansion liner hanger, the reliability of the releasing tool mainly includes the connection during the well entry process and the smooth releasing of the hanger after hanging [1-4]. According to the way of releasing, it can be divided into the earliest mechanical releasing tool and the later developed hydraulic releasing tool, which are very different in structure and performance. Early drilling mechanical releasing tool is the most commonly used releasing tool, but as most of the assignments section drilling depth increasing, the deep well completion operations increased year by year, mechanical releasing tool is not suitable for deep well

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homework problems are also increasingly prominent, hydraulic releasing tool is developed to solve the lack of the former, so the actual application relatively late [5-7]. In order to overcome the special requirement of continuous operation of a drill string that cannot be completed by traditional releasing tools at the time, China National Petroleum Corporation has developed a releasing tool that can be used for continuous operation and applied for utility model patents which has got an authorization. The Engineering Technology Branch of China National Offshore Oil Corporation Energy Development Co, Ltd, in order to solve the problem of the difficulty of losing hands with the mechanical antifouling tools of offshore oilfields in highly deviated wells, and the inaccurate opening pressure of the drilling fluid channel, combining with the unique characteristics of offshore oilfields, developed a releasing tool specially used for mechanical antifouling systems. In the 1970s and 1980s, some foreign oil companies began to study the cementing technology of rotating liners in order to improve the quality of cementing. The releasing tool is one of its core technologies. The main products of mechanical releasing tools are: Cardium's retractable key block mechanical releasing tools and TIW's SJ-T type rotatable mechanical releasing tools, et cetera [8-13]. The main products of hydraulic releasing tools are Weatherford the company's elastic claw-type hydraulic releasing tool and an incomplete threaded hydraulic releasing tool of a patent product in the United States [14-15].

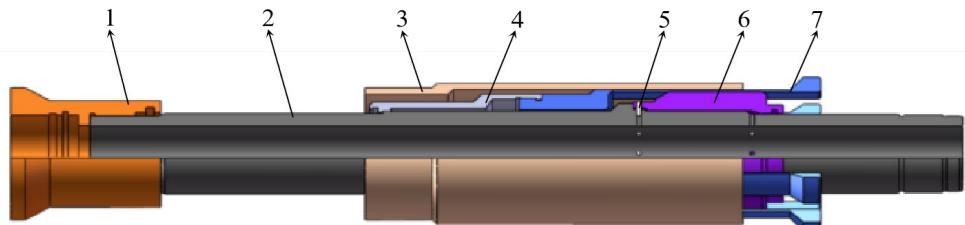
There are multi-set pressure systems in the three-high gas wells (gas wells in high pressure, high sulfur, and high-risk areas), buried deep in Sichuan and Chongqing. The process of casing is complex, the lower tube of liner hanger is large, and axial force is unbalanced during the process of releasing and tied back, which may result in having some problems during this process, such as complex treatment, longer consumption and the risk of casing injury. Therefore, there is an urgent need for a more reliable liner hanger to meet the construction needs of three-high deep gas wells in Sichuan and Chongqing. Analysis of the existing liner hanger, we found that the elastic claw has problem on releasing.

The three-high gas wells in Sichuan and Chongqing area has complex working conditions, and the releasing tool bears a large weight of the lower string during the releasing operation. Referring to the above research, in view of the particularity of the releasing operation, in order to ensure the reliability of the releasing operation, this paper adopts the finite element simulation analysis method to optimize the key part — “elastic claw”.

2. Structure and Working principle of the elastic claw type hydraulic releasing tool

2.1 Structure and technical parameters of hydraulic releasing tool

The elastic claw type hydraulic releasing tool researched in this paper mainly refers to the releasing tool of China National Offshore Oil Corporation (CNOOC) in structural. The lower part of the releasing tool needs to be placed in the well and the upper expansion tool needs to be recovered at the same time. On the basis of inheriting the advantages of the releasing tool of CNOOC, this elastic claw type hydraulic releasing tool cancels the components such as ball seat, limit ring, anti-loose pin and Ball drop construction to make the releasing tool has simple structure and works more reliably. The main parts of the elastic claw type hydraulic releasing tool in this paper include: press sleeve, mandrel, pressure transmission part, sleeve of elastic claw, pin, hydraulic releasing part, elastic claw and so on. The structure diagram of the elastic claw type hydraulic releasing tool is shown in Fig. 1. The mandrel is hollow design, its upper end is connected with the pressure sleeve, and the lower end is connected with the reflux device through threads. The hydraulic releasing part is mounted on the mandrel by shear pins. After installation, there is a hydraulic cavity between the hydraulic releasing part and the mandrel. The hydraulic cavity is connected with the inside of mandrel through a small hole, and there are sealing rings between the hydraulic releasing part and the mandrel for sealing. The purpose of this design is to cut the shear pins by hydraulic pressure holding. Sealing rings of different specifications are used to seal each part to ensure no leakage of liquid inside the release tool. At the same time, each part can slide relative to each other, so it is a dynamic seal between them. The parameters of the main parts of the releasing tool are shown in Table 1.



1—Press sleeve; 2—Mandrel; 3—Pressure transmission part; 4—Sleeve of elastic claw; 5—Shear pin;
6—Hydraulic releasing part; 7—Elastic claw;

Fig. 1. Structure diagram of hydraulic releasing tool

Table 1

The parameters of the main parts of the hydraulic releasing tool

	Mandrel	Pressure transmission parts	Hydraulic releasing parts	Elastic claw	Press sleeve	Sleeve of elastic claw
Material	35CrMo	35CrMo	35CrMo	35CrMo	35CrMo	35CrMo
Total length (mm)	1012.5	432	172	279	172	190
Material yield strength (MPa)	835	835	835	835	835	835
Elastic modulus (GPa)	206	206	206	206	206	206
Poisson's ratio	0.29	0.29	0.29	0.29	0.29	0.29

2.2 Working principle of hydraulic releasing tool

The basic working principle of the releasing tool is as follows: As shown in Fig. 2, after the expanded liner hanger is set successfully, pressure is applied to the releasing tool through the drilling fluid. The red arrow represents the flow direction of the drilling fluid. As can be seen from the local magnification, the drilling fluid enters the hydraulic chamber through the hole in the mandrel, which increases the pressure in the hydraulic chamber. Under the action of liquid pressure, the hydraulic releasing part moves downward relative to the elastic claw and cuts the Shear pins. Finally, driven by the liquid pressure, the hydraulic releasing part moves out of the inside of the elastic claw, causing the inside of the elastic claw to become empty. On the other hand, by pressing down on the press sleeve, the axial pressing force is applied axially to the pressure transmission part, and the pressure transmission part moves down relative to the elastic claw and forces the claw pieces to contract inward. After the claw pieces are contracted inward, the elastic claw is separated from the lower part of the releasing tool to realize the releasing operation. Finally, the upper part of the releasing tool is lifted up to the surface along with the drill string.

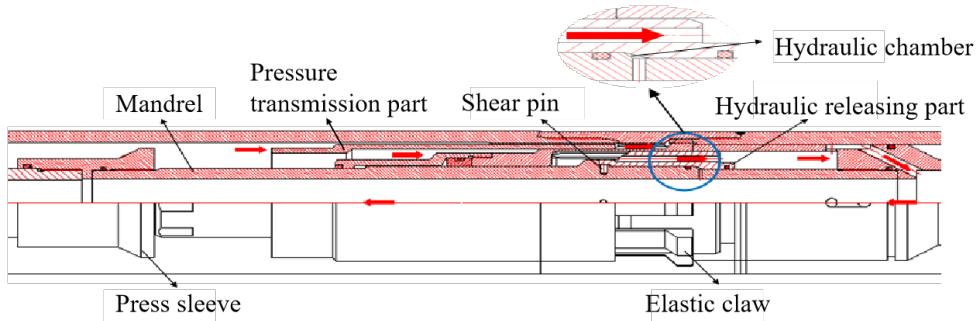


Fig. 2. Schematic diagram of the hydraulic releasing process

3. Finite element analysis and structure optimization of the elastic claw

The elastic claw is one of the key parts of the releasing tool assembly. The elastic claw plays an important role in the releasing operation and recycling. In the process of releasing the hydraulic releasing part from the elastic claw, the pressing sleeve is required to axial pressing force on the pressure transmission part. The greater the required axial pressing force, the more difficult the unloading operation is. Therefore, it is necessary to carry out the following: (1) The load capacity of elastic claw is analyzed by finite element method. (2) The structure of the elastic claw is optimized to reduce the required axial pressing force.

3.1 Establishment of the numerical model of the elastic claw

When the elastic claw works, it is in a high stress state, requiring high elasticity which is not easy to yield. The elastic claw material in this paper is 35CrMo. The elastic modulus of 35CrMo is 206GPa, the Poisson's ratio is 0.29, and the yield strength is 835MPa. According to the structure of the existing elastic claw releasing tool, the basic structure of the elastic claw is designed in this paper shown in Fig. 3. The number of claw pieces of elastic claw is initially set as 8. Take advantage of the Computer Aided Design (CAD) software to establish the solid model of elastic claw and the pressure part and import the finite element software for setting and analysis. The established finite element mesh model of the elastic claw is shown in Fig. 4.

The numerical simulation of the elastic claw includes two processes: the quiescently loaded analysis of the elastic claw and the analysis of the elastic claw in the process of pressing into the pressure transmission part. In the quiescently loaded analysis, the elastic claw is not contracted, and there is a hydraulic releasing part supporting the elastic claw inside. The upper end face of the elastic claw and

the hydraulic release part are set as fixed constraints, and the hydraulic releasing part is regarded as a rigid body. Set the weight of the liner to not exceed 30 tons, and apply a corresponding pulling force to the elastic claw load-bearing inclined surface. In the dynamic released analysis, the end face of the elastic claw is fixed, and displacement is applied to the pressure transmission part, and the pressure transmission part (shrinkage) of the pressure transmission part is simulated by the claw head of the elastic claw, so as to realize the process of releasing.



Fig. 3. Three-dimensional model structure of the elastic claw

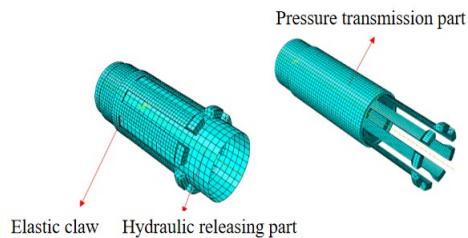


Fig. 4. Finite element mesh model of the elastic claw

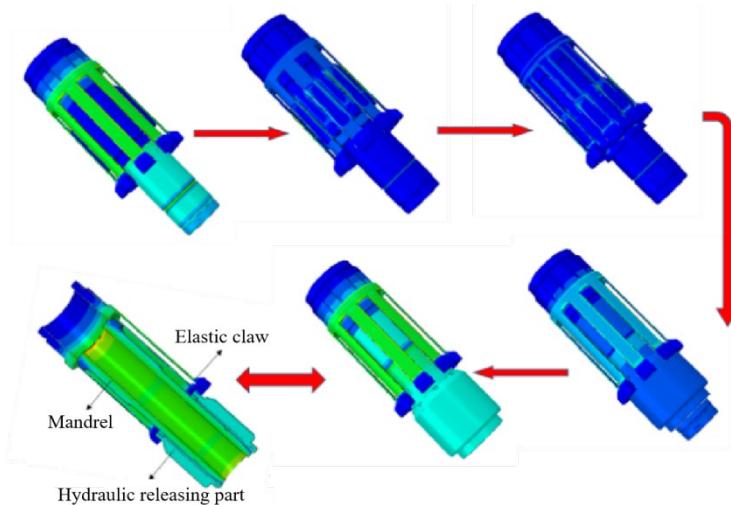


Fig. 5. Equivalent stress diagram of the elastic claw in releasing process

Fig. 5 is a stress diagram of the elastic claw releasing process. It can be seen that the elastic claw is gradually released as the hydraulic pressure pushes the hydraulic releasing part downward.

3.2 Quiescently loaded analysis results of elastic claw

Before the elastic claw is contracted, the load-bearing slope bears the tensile force, and the equivalent stress distribution is shown in Fig.6.

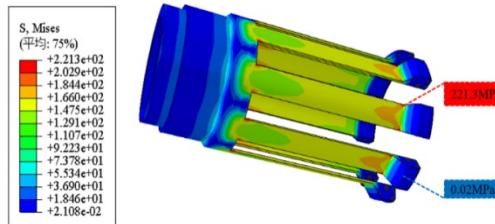


Fig. 6. Quiescently loaded stress diagram of elastic claw

It can be seen from the Fig. 6 that the maximum stress distribution at the transition between the claw head and the claw piece is 221.3MPa, which is less than the yield strength of 35CrMo, indicating that the elastic claw is in a safe working state under static load. In addition, processing the arc transition between the claw head and the claw piece can reduce stress concentration.

3.3 Dynamic released analysis results of elastic claw

In the simulation of the elastic claw release process, the entire analysis step time is set to 1s. Through the numerical analysis of the simplified elastic claw pressing process, the analysis step time of the elastic claw is 0.3s, 0.5s, and 1.0s from left to right. Equivalent stress cloud diagram, as shown in Fig. 7.

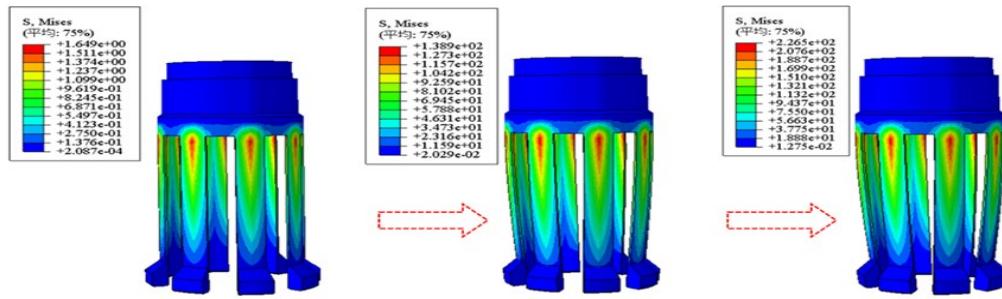


Fig. 7. Equivalent stress diagram of elastic claw in contraction process

It can be seen from the stress cloud diagram of the elastic claw that as the elastic claw is squeezed into the pressure transmission part, the claw pieces gather

radially, and the maximum stress value is much lower than the yield strength of the material, and the elastic claw is in a safe stress state. The part with greater stress concentrates on the root of the elastic claw piece, so the thickness of the claw root can be appropriately thickened and its bearing capacity can be improved.

In the process of releasing the hydraulic releasing part from the elastic claw, the pressing sleeve is required to axial pressing force on the pressure transmission part. In order to accurately analyze the change of the axial pressing force of the elastic claw during the release process, a single piece of the elastic claw is taken for analysis, and the curve of the axial pressing force with the release process is shown in Fig. 8. When the elastic claw is pressed into the pressure transmission part, the required axial pressing force gradually increases; when the claw head is fully pressed (the tool is released), the axial pressing force increases to the maximum, and the maximum load value is 139.7N, Therefore, the total pressing force required for the 8 claw pieces is 1117.6N.

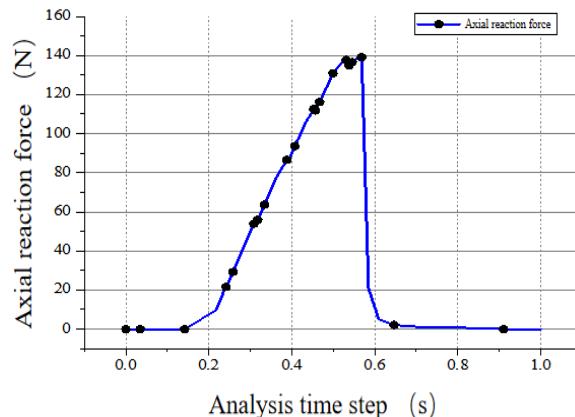


Fig. 8. Axial load change curve of single claw

3.4 Optimal analysis of the load-bearing bevel angle of the elastic claw

Through the quiescently loaded analysis and releasing analysis of the elastic claw base structure, it can be seen that the mechanical parameters of the elastic claw meet the requirements of field use. The basic structure of the elastic claw is shown in Fig. 9. The main influencing factors that affect the action of pressing the elastic claw into the pressure transmission member to realize the separation of the upper and lower parts of the releasing tool include: the load-bearing bevel angle of the elastic claw— α , the number of claw pieces— n , and the width of the gap between the claw pieces— δ . Therefore, the quantitative analysis of the changing law of the

axial pressing force with these influencing factors is of great significance for the rational design of the elastic claw structure. Through the establishment of elastic claw pieces and a simplified finite element model of the pressure transmission part, as shown in Fig. 10. Set different load-bearing bevel angles ($15^\circ, 22.5^\circ, 30^\circ, 45^\circ$), and analyze the relationship between the inclined plane angle down pressure and load-bearing bevel angles.

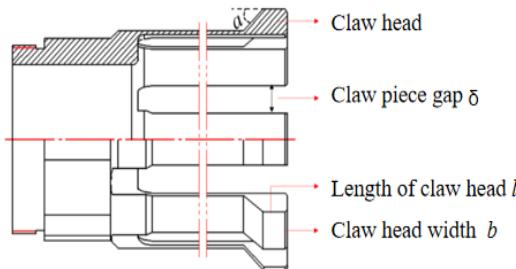


Fig. 9. The basic structure of the elastic claw

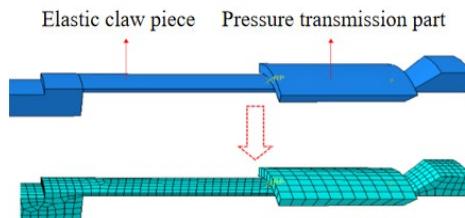


Fig. 10. Finite element mesh model of the elastic claw piece

Through calculation and analysis, the variation law of elastic claw pieces with different α and the maximum pressing force is shown in Fig. 11. When α is less than 25° , the axial pressing force increases with the increase of the bevel angle, but the increasing trend gradually becomes slower; when α is between 30° and 50° , the axial pressing force will not change with α . In addition, it was found from the analysis process that when α is greater than 55° , if the pressure transmission member is forced to move down, it will cause very large plastic strain at the root of the elastic claw piece and interrupt the analysis. Therefore, α cannot exceed 55° , and the specific load-bearing bevel angle value can be flexibly designed according to the size of the elastic claw fitting. In this paper, the load-bearing bevel angle is set to 30° .

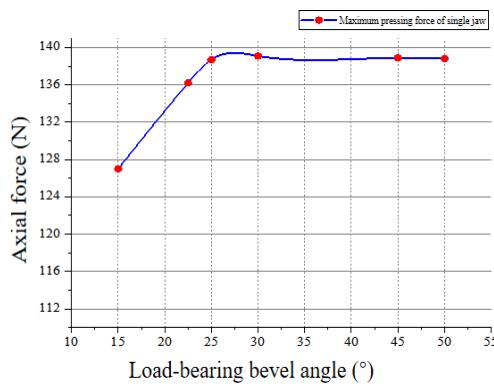


Fig. 11. The relationship between the axial pressing force and the load-bearing bevel angle

3.5 Optimization analysis of the number of claw pieces

In order to analyze the relationship between the axial pressing force of the elastic claws and the number of elastic claws and the width of the claws, 4 groups of elastic claws with different number of claw pieces— n (6, 8, 10, 12). The width of the elastic claw piece and the gap width of the claw piece in the 4 elastic claw models are all equal, and the load-bearing bevel angle is guaranteed to be 30° , as shown in Fig. 12. Perform dynamic releasing analysis on them respectively.



Fig. 12. The models of elastic claw with different number of claw pieces

The maximum equivalent stress and axial pressing force of elastic claw with different number of claw pieces in the process of releasing are obtained by numerical analysis, and the results are shown in Table 2. According to the analysis results, all the maximum equivalent stress values in the process of elastic claw releasing are less than the yield strength of the material. However, when the number of claw pieces is 10, the required axial pressing force is reduced by about 28% compared with the number of claw pieces is 6, by about 4% compared with the number of claw pieces is 8, and by about 18% compared with the number of claw pieces is 12. Therefore, the optimal number of claw pieces for elastic claw is 10.

Table 2

Simulation results of elastic claw with different number of claw pieces

Number of elastic claw pieces (n)	6	8	10	12
Maximum equivalent stress (MPa)	317.5	211	177.5	160
AXIAL PRESSING FORCE (N)	1494	1118	1074	1272

3.6 Optimization analysis of the width of the gap between claw pieces

Obviously, the larger the width of the gap, the smaller axial pressing force required when releasing, but at the same time, the lower the load capacity of elastic claw is. In this paper, according to the previous analysis results, the elastic claw model with different width of the gap is established under the condition that the load-bearing bevel angle $\alpha=30^\circ$ and the number of claw pieces $n=10$. Let “ μ ” be the percentage of gaps on the circumference. Elastic claw models with $\mu=15\%$, 25% , 35% , 45% , 55% , 65% , 75% and 85% were established respectively, as shown in Fig. 13.

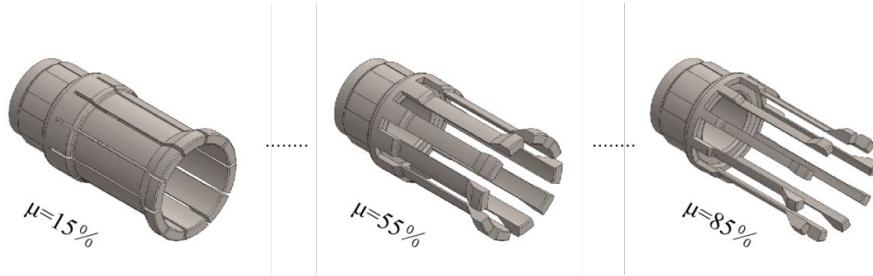


Fig. 13. The models of elastic claw with different claw width

Through finite element analysis, it is found that when $\mu=15\%$, serious material interference occurs when the claw pieces gather in the radial direction, as shown in Fig. 14. Therefore, the percentage of gaps on the circumference should be at least greater than 15%. After analyzing each model, the load capacity and required axial pressing force of the elastic claw under different gap conditions are obtained. The results are shown in Table 3. According to the analysis results, the maximum tensile stress of the elastic claw under static load increases with the increase of the gap width. When $\mu \geq 65\%$, the load capacity of elastic claw is obviously weakened. In order to ensure the safety and reliability of the tool, $\mu < 65\%$ should be ensured.

The analysis results show that the required axial pressing force does not change much when the value of μ is in the range of 25%~55%. In order to ensure the load capacity of elastic claw, $\mu=25\%$ is chosen in this paper.

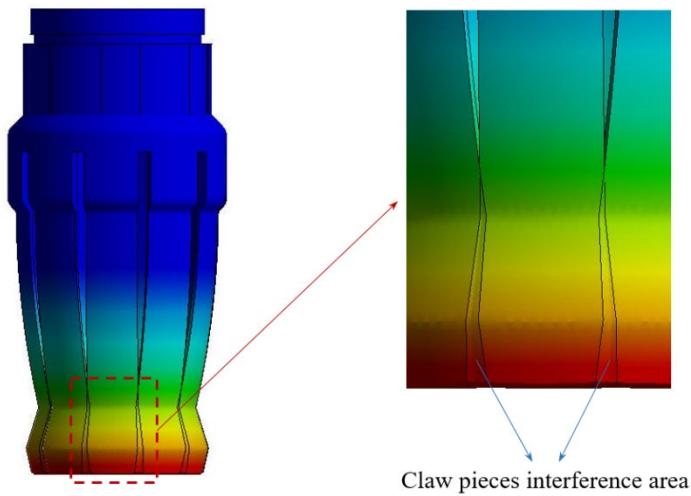


Fig. 14. Interference cloud diagram of elastic claw pieces retracted

Table 3

Analysis results of elastic claw model with different gap

μ	25%	35%	45%	55%	65%	75%	85%
Maximum tensile stress (MPa)	184.3	198.2	204.5	236.2	321.5	483.7	612.4
Axial pressing force (N)	1274	1116	1052	1015	970	732	648

4. Results and Discussion

In summary, through the finite element analysis and structure optimization of the elastic claw, it is believed that when the load-bearing bevel angle of the elastic claw is 30° , the number of claw pieces is 10 and the width of the gap such that the percentage of gaps on the circumference is 25%, the structure of the elastic claw is optimal. In order to reduce the stress concentration, the transition between the load-bearing bevel and the claw piece is processed into an arc shape. The chamfer is processed at the edge of the claw end to improve the reliability of field assembly. The three-dimensional structure of the optimized elastic claw is shown in Fig. 15.



Fig. 15. Physical model diagram of the elastic claw

The optimized downhole hydraulic releasing tool has been successfully applied 7 times in the three-high gas wells in Sichuan and Chongqing area. In field application, the elastic claw type hydraulic releasing tool can be successfully released at 70MPa downhole, and the maximum suspension weight of the elastic claw can be carried 2500kN. The elastic claw structure designed in this paper can effectively avoid the accidents in the well such as elastic claw fracture and tool stuck in the working process. The optimized elastic claw is easier to shrink, and the average setting force applied in the field is 80 ± 5 kN, which reduces 20kN compared with the original one.

5. Conclusions

(1) On the basis of the previous research, this paper carried out the structural improvement of a downhole hydraulic release tool, and optimized its key part — “elastic claw”.

(2) In this paper, the structure of elastic claw is optimized from three aspects: the load-bearing bevel angle of the elastic claw, the number of claw pieces and the width of the gap between claw pieces. The optimization results show that when the load-bearing bevel angle of the elastic claw is 30° , the number of claw pieces is 10 and the percentage of gaps on the circumference is 25%, the structure of the elastic claw is optimal.

(3) The elastic claw structure designed in this paper can effectively avoid the accidents in the well such as elastic claw fracture and tool stuck in the working process. The load capacity of the optimized elastic claw is greatly improved, and the required axial pressing force is reduced. The optimized elastic claw is easier to shrink, and the average setting force applied in the field is 80 ± 5 kN, which reduces 20kN compared with the original one.

R E F E R E N C E S

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