Design on non-stop transportation plugging device for subsea pipeline

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Abstract. The process of plugging with pressure outside the subsea pipeline is cumbersome. The common technology cannot transport oil and gas during the maintenance process, and a new type of subsea pipeline non-stop transportation plugging technology is proposed. In this paper, an oil and gas pipeline with a diameter of 324mm and a pressure of 3MPa is taken as the research object for overall design and feasibility analysis. The anchoring and sealing performance of the packer is calculated, checked and analyzed using the element method. The results indicated that the plugging device can meet the plugging requirements of non-stop transportation of subsea oil and gas pipelines in maintenance.

1. Introduction

Oil and gas pipeline transportation has become the main way of transporting oil and gas over long distances because of economy, safety and environmental protection [1]. According to statistics from the Pipeline Bureau, the total mileage of oil and gas pipelines in China has reached 139,000 kilometers by the end of 2019 [2]. During the laying process of oil and gas pipelines, it will have a significant impact on the participating life of the pipeline in the range of high amplitude. This preliminary deformation process can activate the accumulation of fatigue defects and strain aging [3]. At present, the common technologies include non-stop pressure-opening plugging technology and high-pressure intelligent plugging technology [4, 5].

The non-stop pressure-opening plugging technology needs to weld the split tee outside the pipe under pressure, open the hole with pressure, block and establish a bypass pipeline to isolate the pipeline section to be repaired from the original pipeline system [6], but this technology needs to leave permanent welding scars on the pipeline, which lays safety hazards for the future use of the pipeline.

The high-pressure plugging technology in the pipe is to send two plugging devices into the pipeline successively from the launcher. The two packers are anchored before and after the fault section, and the control center sends instructions to unseal the pipe after the construction is completed [7]. Then, the packers continue to be pushed out by the oil and gas to the receiver to be taken out [8]. This technology avoids damage to the pipeline, but the pipeline needs to be shut down during the process.

To address the shortcomings of the above two technologies, this paper proposes a new type of non-stop transmission pipeline plugging device, which can isolate damaged or leaking pipeline sections without stopping the pipeline and will not leave any additional devices on the pipe to improve the safety performance.
2. Overall scheme design of the non-stop plugging device

The non-stop subsea pipeline plugging system is shown in Figure 1. When the pipeline needs to be maintained, a communication positioning device should be placed outside the pipeline upstream of the fault point in advance, and then the plugging device is fed into pipe through the pig launcher and pushed by the medium to reach the section to be isolated. The ground control center sends commands to the packer. After that, it starts the power system to implement the occluding process. Figure 2 shows the schematic diagram of the overall structure of the plugging device. A non-stop pipeline packer consists of a transition pipe and two plugging heads, each of which includes a plugging module and a power module. The plugging module mainly depends on the slip and the rubber cylinder to achieve the blocking function. The power module is composed of four motors and their drivers. The transition tube is a flexible tube with a certain elasticity to achieve bend. The PVC steel wire pipe can meet the design requirements of the transition pipe in this paper. The PVC steel wire pipe cannot be compressed in the axial direction and can withstand negative pressure. It has good flexibility, small bending radius. Figure 3 shows the sample picture of PVC steel wire pipe. If the length of the transition pipe is long, a support wheel can be installed in the middle of the transition pipe.

3. Mechanical structure design of plugging head

3.1. Introduction to working principle

The internal structure of the plugging head is shown in Figure 4. It is mainly composed of rubber cylinder, pressure-bearing block, slip, push actuator, passive gear plate, driving gear, motor, supporting wheel, box, pressure relief valve and other components. Among them, 8 slips, 4 stepping motors and driving gears are arranged in the circumferential direction. The slip clamp the inner wall of the pipeline to ensure the anchoring performance of the plugging device and the rubber cylinder ensures the sealing performance of the packer. The pressure relief valve is used to eliminate the pressure difference when unsealing.

There are three processes of the plugging operation:

Anchoring process: The packer uses four motors to provide the initial power required for anchoring, and controls the motor to drive the passive gear plate to rotate. The passive gear plate drives the push actuator to move axially through the screw drive to compress the rubber cylinder and the pressure-
bearing block. As the depth increases, the friction continues to increase. When the pressure that can withstand the oil and gas in the pipe is reached, the packer will be anchored in the pipeline.

![Diagram of plugging module](image)

Figure 4. Schematic diagram of the plugging module.

Sealing process: After the anchoring process, a certain pressure difference is generated between the media before and after the packer. The push actuator continues to compress the rubber cylinder axially under the joint action of the motor and the pressure difference, causing the rubber cylinder to expand radially until the contact stress between rubber cylinder and inner wall is greater than the delivery pressure of the pipeline, then, sealing is achieved.

Unsealing process: After the pipeline maintenance operation is completed, open the pressure relief valve to wait for the annular cavity outside the bellows to be filled with medium, and then the packer returns to the initial position under the action of motor and rubber cylinder to achieve unsealing.

3.2. Slip

3.2.1. Theoretical calculation of slip anchoring capacity. The structure diagram of the slip is shown in Figure 5. The slip is made of hard alloy steel. The shape parameters of the clamping teeth determine the locking ability of the slip, and as a result, selecting the optimal values of key slip parameters plays a significant role in improving anchoring performance. The final geometric parameter selection of slip is shown in Table 1 [9].

| Thread angle $\gamma$(°) | Tooth spacing $d$/mm | Connection thickness $t$/mm | Overall tilt angle $\alpha$(°) | Radius of curvature $R$/mm |
|--------------------------|----------------------|-----------------------------|-----------------------------|--------------------------|
| 60°                      | 4                    | 29                          | 15                          | 154                      |

![Diagram of slip structure](image)

Figure 5. Schematic diagram of slip structure.

![Diagram of force analysis](image)

Figure 6. Schematic diagram of force analysis of slip and pressure bearing inclined block.

The force analysis of the slip and pressure-bearing block is shown in Figure 6. The equilibrium condition of the axial force of the pressure-bearing block is:

$$F = N \sin \alpha + fN \cos \alpha$$

(1)

$$N \cos \alpha = F_f \sin \alpha + Q$$

(2)

After the slip is in contact with the pipeline, it can be considered that the slip and the pressure-bearing block no longer move relative to each other. For the whole, the axial force balance condition is:
The method of designing the slip teeth on the surface of the slip is used to increase the equivalent friction coefficient. The equivalent friction coefficient is related to the rake angle of the slip teeth and the friction factor between different metals. The structure when the slip teeth are embedded in the inner wall is shown in Figure 7.

\[ P = f_iQ = f_iF \frac{1 - f \tan \alpha}{f + \tan \alpha} \]  

(3)

Figure 7. Schematic diagram of the structure of slip teeth embedded in the inner wall of the pipeline.

The equivalent friction coefficient between the slip teeth and the inner wall of the pipeline is [10]:

\[ f_i = \frac{f \tan \theta + 1}{\tan \theta - f} \]  

(4)

In the formula: \( f_i \) is the equivalent friction coefficient between slip and pipe wall; \( \theta \) is the rake angle of the tooth. In this paper, the rake angle is 10º. \( f \) is the friction coefficient between the slip material and the pipe material. The value of \( f \) is 0.15.

To enable the slip to anchor the packer, the friction between the slip and the pipe wall and the medium pressure should have the following relationship:

\[ f_i = \frac{f \tan \theta + 1}{\tan \theta - f} \geq \frac{f + \tan \alpha}{1 - f \tan \alpha} \]  

(5)

Bring the related parameters into the above formula and the result is always established, indicating that the slip can achieve the anchoring function under these parameters.

3.2.2. Slip finite element simulation analysis. After calculation, when the pipeline transportation pressure is 3MPa, the pressure difference of the medium will eventually generate a pressure of 178646.66N on the end face of the packer. According to Equations (1) and (2), the radial force between a single slip and the inner wall of the pipe is 51282.06N. Because the eight slips are symmetrically arranged, only a single slip and 1/8 pipe are selected for analysis. The circumferential and axial degrees of freedom of the slip are set to 0, and the radial degrees of freedom are not constrained. The pipeline is fully constrained. A uniformly distributed load of 52000N is applied to the inclined surface of the slip along the radial direction.

The von-Mises stress distribution cloud diagrams are drawn as shown in Figure 8.

![Figure 8](image)

(a) Slip. (b) Pipeline.

Figure 8. Von-Mises stress distribution cloud diagram of slip and the inner wall of the pipeline.

It can be seen from the diagram that the maximum stress of slip occurs in the most prominent part of the slip teeth. The maximum stress is 73.613MPa, which is much less than the yield strength of slip (835MPa), and the maximum stress of the pipeline is 27.116Mpa, which is much smaller than the yield strength of the pipeline (510MPa). Therefore, the design of slip is safe and reliable.
3.3. Sealing performance analysis of the rubber cylinder

The research shows that the packer just needs to provide a certain squeezing force to the rubber cylinder to meet the self-sealing conditions, and then rubber cylinder can play a sealing role. With the increase of the hardness of the rubber cylinder, when it forms a self-sealing condition, the greater force is required, so the higher requirement for the power system. On the other hand, if the hardness of the rubber cylinder is too low, the shoulder of the rubber cylinder will be protruding after the sealing, resulting in a decrease in contact stress, which will lead to seal failure [11].

Polyurethane rubber is a super-elastic material, so the Mooney-Rivlin model is used for calculation. It is the most economical and effective method to determine the material coefficient of the Mooney-Rivlin constitutive model by empirical formula fitting. There is the following relationship between the material constant of the rubber cylinder and the elastic modulus:

\[
6C_{10} \left(1 + \frac{C_{01}}{C_{10}}\right) = \frac{15.75 + 2.15\text{HA}}{100 - \text{HA}}
\]  

(6)

In the formula: \(E_0\) is the elastic modulus of the rubber material, MPa; \(G\) is the shear modulus of the material, MPa; \(C_{10}\) and \(C_{01}\) are the material constants of the rubber material.

It can be seen from Equation (6) that the hardness of rubber material (HA) and the ratio of \(C_{10}\) to \(C_{01}\) should be determined in order to determine the values of \(C_{10}\) and \(C_{01}\). Generally, polyurethane materials with different hardness fit best at that time when the ratio is 0.05. According to the above formula, the mechanical properties of rubber material constitutive model with different hardness are calculated, as shown in Table 2.

| Rubber hardness (HA) | 35   | 45   | 55   | 65   |
|----------------------|------|------|------|------|
| Rubber material mechanical property constant | \(C_{10}\) | 0.222 | 0.325 | 0.473 | 0.705 |
|                      | \(C_{01}\) | 0.011 | 0.016 | 0.024 | 0.035 |

![Figure 9. Finite element model of rubber cylinder.](image1)

![Figure 10. Mesh generation of the finite element model of rubber cylinder.](image2)

The geometric model and size parameters of the rubber cylinder are shown in Figure 9.

When using ABAQUS software for finite element analysis, the self-contact friction coefficient of the rubber cylinder is set to 0.25, and the friction coefficient between the rubber cylinder and the metal is set to 0.15. The grid division result is shown in Figure 10.

The process of applying the load includes two analysis steps. The first step is applying the axial force generated by the motor to the lower end of the push actuator in order to simulate the initial sealing process. The second step is applying the pressure difference generated by the pipeline itself after sealing the pipeline to the upper end surface of the push actuator to simulate the sealing process.

The distribution of the surface contact stress along the axial distance between the rubber cylinder materials with different hardness and the pipe wall is drawn, as shown in Figure 11.
Figure 11. The distribution of the contact stress along the axial distance between the tube material of different hardness and the pipe.

It can be seen from the figure that the contact stress between the rubber tube and the tube wall decays along the axial direction in the effective sealing area. The reason for this attenuation is due to the unidirectional pressurization method.

In order to observe the internal stress of the rubber cylinder more conveniently, the cloud image is set to display only the range of 0–4MPa in the post-processing Stress, as shown in Figure 12.

(a) 35HA. (b) 45HA. (c) 55HA. (d) 65HA.

Figure 12. Mises equivalent stress cloud diagram of different hardness rubber cylinder.

According to the internal stress cloud diagram of different hardness rubber cylinders, it can be observed that as the hardness of the rubber cylinder decreases, the shoulder protrusion phenomenon after the rubber cylinder sealing is more obvious, and the shoulder protrusion phenomenon will have an adverse effect on the sealing process. According to surface contact stress and the deformation cloud diagram of the rubber cylinder, the polyurethane rubber material with a hardness of 45HA is selected as the material of the rubber cylinder.

4. Driving force module design

The power output is divided into two stages. In the first stage, the motor needs to rotate at a high speed to facilitate the contact between the slip and the inner wall quickly, but the torque is not required. In the second stage, after the slip is in contact with the inner wall, the slip is embedded in the wall to lock the position of the packer. The power of the motor is used to squeeze the rubber cylinder together with the pressure difference to achieve the sealing. The motor speed decreases and a large torque is required.

Four 57 stepper motors with maximum torque of 2.3N·m are selected to provide power for the packer. The transmission system enlarges the power output by the high transmission ratio of the gear drive and spiral transmission. The gear transmission ratio is 13:130 and the helix rise angle is 4°. The load formula of spiral transmission is:

$$M_t = \frac{1}{2}d_2F \tan(\lambda + \rho')$$  \hspace{1cm} (7)

In the formula: $d_2$ is the thread diameter(mm), and $d_2$ takes 162mm here; $F$ is the axial load of spiral transmission(N); $\lambda$ is the helix angle (°), 4° is taken here; $\rho'$ is the equivalent friction angle.

According to the gear transmission ratio and the load formula of the screw drive, the maximum thrust is 5513.6N. Since the packer can form a pressure differential self-locking function, once the packer slips open and contact with the inner wall, the pressure difference between the front end of the packer push actuator and the annular cavity will act to seal.
5. Conclusions

(1) Through independent innovation, a new type of oil and gas subsea pipeline packer is proposed and the overall design of its mechanical structure is given;

(2) The design calculation and strength check of the key parts of the packer (slips)are carried out, the sealing performance of the sealing rubber cylinder is studied, and the material of the sealing rubber cylinder is optimized according to the research results;

(3) The power system is designed and calculated, and it is verified that the power provided by the selected motor is enough to trigger the pressure difference self-locking of the packer.

Nomenclature

| Symbol | Description |
|--------|-------------|
| $\gamma$ | Thread angle |
| $d$ | Tooth spacing |
| $t$ | Connection thickness |
| $\alpha$ | Overall tilt angle |
| $R$ | Radius of curvature |
| $F$ | Thrust acting on the end face of the packer |
| $f_1$ | the equivalent friction coefficient between slip and pipe wall |
| $\theta$ | the rake angle of the tooth. In this paper |
| $f$ | the friction coefficient between the slip material and the pipe material |
| $E_0$ | the elastic modulus of the rubber material, MPa; |
| $G$ | the shear modulus of the material, MPa; |
| $C_{10}$, $C_{01}$ | the material constants of the rubber material. |

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