Design and Experimental Research of Expansion-Anchorage Device in Deepwater Pipeline

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Abstract. Anchoring device is one of the key technologies of deep water pipeline internal packer. In this paper, the structural design of the anchorage device is completed on the basis of the principle of the inclined plane reinforced structure and elastoplastic mechanics, and its three-dimensional model and mechanical model are established. Theoretical research and simulation analysis of the strength of key parts of the anchoring device are carried out, with emphasis on the design of the anchoring block, material selection, anchoring depth between the anchoring block and the inner wall of the pipe, and theoretical and simulation research of the contact stress. The correctness of theory and simulation analysis is verified by anchoring experiment. The research content of this paper can lay a foundation for the design and research of anchoring device.

1. Introduction

In recent decades, with the continuous decrease of onshore and offshore oil and gas resources, the exploration and production of deep-sea oil and gas resources have become an important way to meet the global energy demand [1]. Although pipeline oil transfer has always been considered as a safe and efficient way of oil transfer, due to the complex deep-sea environment, pipeline aging and other factors, the submarine pipeline is prone to damage [2-3]. The importance of submarine oil pipelines is self-evident, which requires the development of pipeline repairment equipment for regular repairment [4]. Closure in deep water pipeline plays an important role in deepwater pipeline maintenance, which mainly includes the guiding device, sealing device, anchor device, power plant, switching device and hydraulic control system. Anchor device is the core part of the pipeline closure, and anchor block is an important part of anchoring device. Its performance fit or unfit quality directly affects the reliability of the anchor device, which affects the performance of the pipe inner stopper [5-6]. This paper mainly introduces the design and research of anchoring device.

However, the existing anchoring device has such problems as unstable force applied in the process of anchoring, greater damage to pipelines, inconvenient replacement of anchoring block, and easy damage of anchoring device parts. The design of the anchor device in this paper is based on the theoretical research of the inclined plane force increasing structure and the principle of elastic-plastic mechanics. The theoretical design model of the main parts of the anchor device is established. The anchor block can be disassembled to facilitate the maintenance and replacement of the anchor block. The design strength is checked by simulation analysis, and the theoretical and simulation results are verified by experiments.

2. Design theory of anchoring device

The anchoring device is mainly composed of 1 octahedral cone, 2 t-shaped guide rail, 3 semicircular block, 4 anchoring block, 5 arc block, 6 external expansion block and 7 cylinder connecting sleeve. The anchor device model is shown in figure 1.
2.1 Stress analysis of anchoring process

In the process of the sealing device in the deep sea pipeline reaching the final anchoring, the power device is required to provide the power to ensure that the teeth of the anchoring block are embedded in the pipe wall. Figure 2 is to convert the spatial confluence force system into the plane confluence force system. In the figure 2, $F$ is the total force provided by the fluid pressure in the pipe and the hydraulic cylinder, and $F_n$ is the positive pressure of the pipe to the anchor block group.

![Figure 1. Model anchoring device](image)

![Figure 2. Schematic diagram of anchoring stroke force](image)

1. Octahedral cone 2. External expansion block 3. Anchor pressure block 4. Pipe 5. Hydraulic cylinder connecting frame

In figure 2, the triangle of force is obtained by positive mystery theorem:

$$\frac{F}{2n} = \frac{R_{21}}{\sin(\theta + \beta)} = \frac{F_n}{\sin(90 - (\theta + \beta))}$$

$$\frac{F_n}{\sin(90 - 2\beta)} = \frac{R_{32}}{\sin(90 - \theta + \beta)}$$

Where, $F$ is axial force on the packer; $F_n$ is the positive pressure of the inner wall of the pipe on an anchor block group; $n$ is the number of anchor block groups; $\beta$ is the angle of friction between contact surfaces; $\theta$ is octahedral cone inclination angle; $R_{12}$, $R_{21}$, $R_{13}$, $R_{31}$ are the interaction between the external expansion block and the octahedral cone; $R_{32}$, $R_{25}$, $R_{53}$, $R_{35}$ are the interaction between the external expansion block and the cylinder connecting sleeve.

Because of $|R_{12}| = |R_{21}|$, formula (3) can be obtained:

$$F = F_n \cdot \frac{\sin(90 - \theta + \beta) \sin(2(\theta + \beta))}{\sin(90 - (\theta + \beta)) \sin(90 - 2\beta)}$$

According to the mechanical manual to know that the expansion block and octahedral cone friction coefficient $f = 0.12$, The friction Angle is calculated from formula (4), $\beta = 6.84^\circ$.

According to the graph of positive pressure $F_N$ and inclination Angle $\theta$ in figure 3, When the inclination Angle is $\theta = 4.5^\circ$-$6^\circ$, the positive pressure $F_N$ increases; When the inclination Angle is $\theta = 6^\circ$-$7.5^\circ$, the positive pressure $F_N$ decreases. But when $F_N$ is too large, result in anchor block embedded conduit wall exceeds the yield limit of pipe, in order to meet the expanding tightly embedded pipe when the pressure is $F_N$ requirement, anchorage axial and radial motion displacement...
and the choice of the hydraulic cylinder stroke, and to satisfy the requirement of the anchor device self-locking, take structure of slope angle to $\theta = 6.5^\circ$.

![Figure 3. Curve of positive pressure and inclination](image)

The friction Angle and inclination Angle are substituted into formula (1-2) to get $F_n = 2.11 F/2n$, from which it is known that the inclined plane structure makes the output force 2.11 times of the input force. Substitute in the data to obtain $F = 2.67 \times 10^5 N$, $F_n = 7.04 \times 10^5 N$, $R_{21} = 1.41 \times 10^6 N$.

3. Design and strength analysis of key parts

3.1 Simulation analysis of external expansion block

Figure 4 shows that the maximum deformation of the external expansion block is 0.06mm, the deformation is very small, and the maximum stress is 459MPa. The external expansion block selects no. 45 steel tempered treatment, and the yield strength meets the requirements.

![Figure 4. Deformation cloud chart and stress cloud chart of outer bulge block.](image)

3.2 Simulation analysis of octahedral cone

Formula (2) calculates the vertical pressure $R_{21} = 7.04 \times 10^5 N$ on the contact surface of the conical body. ANSYS analysis results of the octahedral cone are shown in figure 5. The maximum deformation of the octahedral cone is 0.185mm and the maximum stress is 340MPa. They are produced at the end of the cone, so the material of the octahedral cone is also 45 steel that has been tempered and chromed.

![Figure 5. Deformation cloud diagram and stress cloud diagram of octahedral cone](image)

4. Design calculation and simulation analysis of anchor block

After the gate valve is completely closed, the positive pressure of each external expansion block on the pipe wall is $F_n = 7.04 \times 10^5 N$, each external expansion block is arranged with 3 columns of 5 anchor blocks (the Angle between each column is), the positive pressure of each anchor block is as follows:

$$P_i = \frac{F_n}{5 \times (2 \cos 15^\circ + 1)} = 4.8 \times 10^4 N \quad (5)$$

The tooth shape of anchor block is designed as triangle. The deeper the anchor block is embedded into the pipe wall, the greater the contact area between the pipe wall and the anchor block will be. The
anchor block is made of high carbon alloy steel. The tensile strength is \( \delta_t = 1320 \text{MPa} \), the yield strength is \( \delta_y = 600 \text{MPa} \), hardness is HRC60. After calculation, the embedding width of the anchor block can be obtained, as shown in equation (6).

\[
B = \frac{P_1}{\pi d \sigma_p}
\]  

(6)

Where, \( P_1 \) is positive pressure on an anchor block, \( d \) is the circular tooth tip diameter, \( B \) is the width of ring belt, \( \sigma_p \) is stress of anchor block.

The steel used in API standard submarine oil and gas pipelines is X42~X70 (divided by yield limit), and the commonly used material X65 (L450) is taken. When \( \sigma_p = \sigma_s \) is taken, the annular bandwidth is \( B = 1.46 \text{mm} \), and the depth of anchor block embedded in the pipe wall is about \( h = 0.87 \text{mm} \) on the triangular section of tooth shape.

4.1 Strength analysis of anchorage block teeth

Under the action of 4.8t positive pressure, the depth of the anchor block embedded in the inner wall of the pipe is 0.87mm. As can be seen from figure 6, the side length \( L \) and positive pressure \( P_1 \) of the tooth shape can be obtained:

\[
L = \frac{h}{\cos 40^\circ} = 1.136 \text{mm}
\]

\[
P_1 = P \sin 40^\circ = 3.10 \text{T}
\]

Then the compression area \( S \) of the anchor block is:

\[
S = 2\pi \times L \times d = 167.65 \text{mm}
\]

From formula (15), it can be known that the stress on the anchor block is:

\[
\sigma_p = \frac{P_1}{S} = \frac{3.1 \times 10^4}{167.65} = 184.9 \text{MPa}
\]

It can be seen from figure 6 that the dangerous section of annular tooth of anchorage block is a-a section, the height of a-a section is \( b = 6 \text{mm} \), and the distance between the tooth tip and a-a section is \( H = 3.5 \text{mm} \). The bending strength of the dangerous section is shown in formula (7), and the shear strength is shown in formula (8):

\[
\sigma_b = \frac{M}{W} = \frac{6FH}{8ab^2}
\]  

(7)

\[
\tau = \frac{F}{8\pi dB}
\]  

(8)

Where, \( F \) is Axial pressure of a single anchor block \( F = 2.23 \times 10^4 \text{N} \), \( d \) is the circular tooth tip diameter, \( M \) is bending moment of anchor block, \( W \) is a-a section approximates the flexural modulus of the beam.

![Figure 6](image)

Figure 6. The positive pressure on the anchorage block teeth and the force on the cross section of the teeth

By substituting the data, formula (7) and formula (8) can obtain:

\[
\sigma_b = \frac{M}{W} = \frac{6FH}{8ab^2} = 196.4 \text{MPa}
\]
Allowable stress of anchor block:

\[ \left[ \sigma_p \right] = \frac{\sigma_p}{n} = \frac{600}{1.25} = 480 \text{MPa} \]

\[ [\tau] = 0.6 [\sigma] = 0.6 \times \frac{\sigma_p}{1.25} = 288 \text{MPa} \]

To sum up, the extrusion strength, flexural strength and shear strength of the anchor compression block all meet the design requirements.

4.2 Analysis of contact stress and deformation of anchor block

It is known that the yield limit of pipeline material is 450MPa, and the yield limit of anchor block material is 600Mpa. It is known from figure 7 that the maximum plastic deformation of the pipeline is 0.82mm, and from figure 8 and 9 that the maximum stress of the anchor block is 512Mpa. The maximum strain is 0.004mm, so the strength of the anchor block meets the requirements.

Figure 7. Ductal plastic deformation cloud pattern

Figure 8. Stress and deformation nephogram of anchor block

5. Experimental Study

5.1 Experimental purpose and equipment

The test equipment mainly includes the test plugging device prototype as shown in figure 9. Figure 10 shows the pipeline to be blocked; Syl-40/1.6 manual hydraulic pump is shown in figure 2.6.
5.2 Experimental steps  
To simulate the above process in the land environment, the following experiments are designed:  
(1) Take a section of oil pipeline as the damaged pipeline to be sealed, one end of which is open for installing the packer, complete the anchoring steps of the packer.  
(2) Successively open the pipe and the globe valve of the packer to relieve pressure, press the rod cavity of the hydraulic cylinder of the packer, remove the sealing state of the packer by anchoring, remove the equipment, and observe the anchoring traces on the inner wall of the pipeline.

5.3 Experimental results  
During the test, in the process of pressure rising in the oil pipeline, there is no relative displacement between the locating sleeve of the packer and the pipe mouth, and there is no obvious slip in the anchoring trace of the inner wall of the pipeline after removal, indicating that the anchoring of the packer can be reliable. Anchor marks on inner wall of pipe and sealing device test equipment installation as shown in figure 11.

6. Conclusion  
This paper focuses on the force analysis, design theory and design analysis of anchoring device of deep-sea pipeline internal closure device, as well as the key parts of anchoring device. The depth of anchorage block embedded in pipe and the contact stress between anchorage block and pipe are analyzed theoretically and simulated. The theoretical and simulation results of anchor design are verified to meet the practical engineering requirements by simulating the plugging process of a deep sea packer. The research content of this paper can provide reference basis for the stress, design of anchoring device and the research of anchoring on pipeline damage.

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