Research on Large-diameter Solid Expandable Tubular Subsidy Technology and Suspension Seal Device

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Abstract—In view of the issue that the diameter is small and the suspension force is uncertain after solid expandable tubular (SET) subsidy. The uniaxial tensile test of SA106B metal is carried out, and the parameters of metal elastoplastic constitutive model are fitted accordingly. Then, a solid expandable tubular laboratory test is designed, and a finite element model of SET is established based on this test. Finally, the effects of the expansion cone radius and the thickness of the rubber ring on the expansion stress and contact pressure are researched. The results show that the established finite element model has small errors and reliable results. The optimal expansion cone radius is selected through the maximum stress and the residual stress of expanded tubular to improve the expansion quality of the tube. The thickness of the rubber ring has a great influence on the contact pressure and the suspension force, so it must be reasonably optimized under the premise of ensuring the sealing ability. The optimal solution of the expansion cone radius and the thickness of the rubber ring is given by comparison. This study provides a theoretical basis for the optimization of solid expandable tubular technology.

1. INTRODUCTION

Solid expandable tubular (SET) technology originated at the end of the 20th century, and it has emerged in the 21st century and developed rapidly. It has been successfully applied to drilling and completion,
2. METAL TENSILE TEST AND PLASTIC MODEL THEORY

To study the effects of different expansion conditions on the mechanical properties of SET by numerical simulation, a reliable SET metal elastoplastic constitutive model must be established. Therefore, this paper designed the uniaxial tensile test of SA106b metal. The yield strength, tensile strength, elongation and other parameters of SA106b carbon steel are determined by collecting the engineering stress-strain results of the experiment.

2.1. Metal plasticity theory

As mentioned above, the test result of steel properties is the stress-strain relationship. The stress-strain results obtained from the tensile test of the metal specimen are essentially engineering stress $\sigma_{\text{nom}}$ and engineering strain $\varepsilon_{\text{nom}}$, while the plastic property parameters used by the finite element software are the true stress $\sigma_{\text{true}}$ and the true strain $\varepsilon_{\text{true}}$. The relationship between the two parameters is as follows:

$$
\left\{ \begin{array}{l}
\sigma_{\text{true}} = \int_{\varepsilon_{\text{nom}}}^{\varepsilon_{\text{true}}} \frac{d\sigma}{d\varepsilon} = \ln \left( \frac{\varepsilon_{\text{true}}}{\varepsilon_{\text{nom}}} \right) = \sigma_{\text{nom}} (1 + \varepsilon_{\text{true}}) \\
\varepsilon_{\text{true}} = \frac{F}{A} = \frac{F}{\sigma_{\text{nom}} L} = \ln (1 + \varepsilon_{\text{true}})
\end{array} \right.
$$

(1)
Where $A$ is current cross-sectional area, $A_0$ is initial cross-sectional area, $F$ is load, $l$ is current length and $l_0$ is initial length.

The true strain $\varepsilon_{\text{true}}$ is composed of plastic strain $\varepsilon_{\text{pl}}$ and elastic strain $\varepsilon_{\text{el}}$. To define plastic material in finite element software, plastic strain $\varepsilon_{\text{pl}}$ is required:

$$\varepsilon_{\text{pl}} = |\varepsilon_{\text{true}}| - |\varepsilon_{\text{el}}| = |\varepsilon_{\text{true}}| - \frac{|\varepsilon_{\text{true}}|}{e}$$  \hspace{1cm} (2)

Where $E$ is elastic modulus.

Because traditional length measuring systems (clip gauges) have some limits, the stress and strain data do not rise continuously until specimen failure. The data of true stress and strain must be obtained from the engineering stress–strain data which before the start of necking [17].

The tool structure of SET determines that the external force is symmetrical to the axis of rotation during the expansion process, which is an axisymmetric stress state. It can use cylindrical coordinates to represent the stress cell body.

The general form of equilibrium differential equations in cylindrical coordinates is [18]:

$$\begin{align}
\frac{\partial \sigma_{\rho}}{\partial \rho} + \frac{1}{\rho} \frac{\partial \tau_{\rho \theta}}{\partial \theta} + \frac{\partial \sigma_{\theta}}{\partial \theta} + \frac{\partial \tau_{\rho z}}{\partial z} + \frac{\sigma_{\rho} - \tau_{\rho \rho}}{\rho} &= 0 \\
\frac{\partial \tau_{\rho \theta}}{\partial \rho} + \frac{1}{\rho} \frac{\partial \sigma_{\rho}}{\partial \theta} + \frac{\partial \tau_{\theta \theta}}{\partial \theta} + \frac{\partial \tau_{\rho z}}{\partial z} - \frac{\tau_{\rho \rho}}{\rho} &= 0 \\
\frac{\partial \sigma_{\theta}}{\partial \theta} + \frac{1}{\rho} \frac{\partial \tau_{\rho \theta}}{\partial \rho} + \frac{\partial \tau_{\theta \theta}}{\partial \theta} + \frac{\partial \tau_{\theta z}}{\partial z} + \frac{\tau_{\rho \rho}}{\rho} &= 0 \\
\frac{\partial \tau_{\rho z}}{\partial \rho} + \frac{1}{\rho} \frac{\partial \tau_{\theta \rho}}{\partial \theta} + \frac{\partial \tau_{\rho z}}{\partial z} + \frac{\tau_{\rho \rho}}{\rho} &= 0
\end{align}$$  \hspace{1cm} (3)

Since the meridian plane (the plane passing through the axis of the rotating body, the $\theta$ plane) of the expandable tubular is not twisted during the expansion process, the stress state is characterized by $\tau_{\theta \rho} = \tau_{\theta z} = 0$.

Therefore, there are only four independent stress components in the stress tensor, namely $\sigma_{\rho}$, $\sigma_{\theta}$, $\sigma_z$ and $\tau_{\rho z}$, as shown in Fig. 2. Combined with equation (3), the stress differential equation under axisymmetric conditions can be obtained as:

Similarly, the geometric equations under axisymmetric conditions are:

Based on the above theoretical formula, the SET axisymmetric model can be established by finite element software.

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$$\begin{align}
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\frac{\partial \tau_{\rho \theta}}{\partial \rho} + \frac{1}{\rho} \frac{\partial \sigma_{\rho}}{\partial \theta} + \frac{\partial \tau_{\theta \theta}}{\partial \theta} + \frac{\partial \tau_{\rho z}}{\partial z} - \frac{\tau_{\rho \rho}}{\rho} &= 0 \\
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\frac{\partial \tau_{\rho z}}{\partial \rho} + \frac{1}{\rho} \frac{\partial \tau_{\theta \rho}}{\partial \theta} + \frac{\partial \tau_{\rho z}}{\partial z} + \frac{\tau_{\rho \rho}}{\rho} &= 0
\end{align}$$  \hspace{1cm} (4)

Similarly, the geometric equations under axisymmetric conditions are:

Based on the above theoretical formula, the SET axisymmetric model can be established by finite element software.
2.2. Metal tensile test

In order to obtain reliable numerical simulation results, the metal tensile test is carried out according to the standard GB/T 228.1-2010 "Metal material tensile test part 1: room temperature test method". Cut along the longitudinal direction of the tube to obtain a dumbbell-shaped tensile specimen, as shown in Fig. 3.

![Metal tensile specimen](image)

The cut SA106b dumbbell-shaped sample is subjected to a tensile test by a professional material testing mechanism, and the tensile experimental device is shown in Fig. 4(a). The mechanical properties of the plastic elongation force, plastic elongation strength, tensile strength and elongation after fracture of SA1206b are obtained, as shown in Table 1. At the same time, the tensile force-strain results of the material are obtained as shown in Fig. 4(b).

![Schematic diagram of tensile test](image)

2.3. Establishment of constitutive model

The force characteristics and tool characteristics of the expandable tubular expansion process determine the need to establish a constitutive model of metal elastoplastic. The stress-strain data obtained from the metal tensile test is processed. Get tabular data available for finite element software.

Obtaining the engineering tensile force-strain data which before the start of necking by uniaxial tensile test as shown in Fig. 5(a). Zone A-B is the elastic phase of the metal and zone B-C is the plastic phase of the metal.

Zone A-B can be used to derive the tensile modulus in the elastic properties of the finite element software. Zone B-C can derive the true stress and plastic strain in the plastic properties of the finite element software. According to equation (1) and equation (2), the engineering stress-strain data can be converted into real stress-plastic strain data as shown in Fig. 5(b).

![Data processing](image)
TABLE 1. METAL MECHANICS PROPERTIES

| Material | Sample thickness (mm) | Sample width (mm) | Plastic extension force (kN) | Plastic extension strength (MPa) | Maximum tensile force (kN) |
|----------|-----------------------|-------------------|----------------------------|---------------------------------|---------------------------|
| SA160b   | 10.1                  | 38.352            | 114.4                      | 295.42                          | 182.8                     |
| Tensile strength (MPa) | Original gauge length (mm) | Post-break gauge length (mm) | Elongation after break (%) | Test temperature (℃) |
| 471.83 | 110 | 143.66 | 30.6 | 24 |

3. SET FINITE ELEMENT MODELING

A two-dimensional axisymmetric model is established for the expandable tubular, rubber ring, expansion cone and casing, as shown in Fig. 6 [19]. A model is constructed for a 5-1/2in casing with outside diameter of $\Phi 139.7$mm. Its wall thickness is 7.72mm. Its inner diameter is $\phi 124.3$mm. Casing defines material properties for N80 grade. Solid expandable tubular model is $\Phi 114$*4.5mm. Rubber ring models are $\Phi 116.4$*1.2*50mm (diameter*thickness*length), which are made of nitrile rubber and adopt the Mooney-Rivlin model which uses to describe the strain energy function of rubber materials. The maximum outer diameter of the expansion cone is $\Phi 114$mm, which is set to the analytical rigid.

Constraints are imposed on the different structures of the model. According to the downhole environment, the outer wall of the casing is completely restrained, the upper end of the expandable tubular is axially constrained, and the upper end of the expansion cone has displacement conditions. Three sets of models with friction coefficients of 0.05, 0.10 and 0.15 are established. Study the effect of friction coefficient on model results. As a result, for example the coefficient of friction as 0.1. The maximum stress during SET expansion is 534 MPa, which is less than the stress limit of 598 MPa, as shown in Fig. 7(a). The residual stress after expansion of the expandable tubular is at most 473MPa, as shown in Fig. 7(b). The expansion driving force curves of different friction coefficients are shown in Fig. 7(c). The contact pressure curve of single rubber ring is as shown in Fig. 7(d), and the suspension force of the single ring is calculated to be 5-7ton.
Figure 7. SET finite element model diagram. (a) Maximum stress; (b) Residual stress; (c) Driving force; (d) Contact stress

4. SET LABORATORY TEST

Based on earlier research and the above numerical simulation basis, a complete set of experimental equipment for the expandable tubular is designed and processed. The experimental principle is shown in Fig. 8. The experiment uses a pressure test pump to inject a high-pressure liquid into the piston chamber, and pushes the expansion cone to the left to complete the expansion of a piston length. And then, move the piston rod to the left. Unpack the slip element and reinstall it on the left side of the casing against the SET. This cycle completes the expansion of the SET as a whole.

Figure 8. Laboratory equipment diagram

Internal pressure of piston cylinder can be tracked by pressure gauge of pressure test pump. The real-time driving force of the expansion cone can be calculated from the relationship between the internal pressure and the area under pressure. The experimental site is shown in Fig. 9. Through the change of the pump pressure in the experiment, the average driving force required to calculate the expansion cone at the rubber ring position is 17ton, and the maximum driving force is 20ton. The results show that the experimental device should be capable of providing a driving force of more than 20ton. The difference between the laboratory experimental results of the driving force and the simulation results with a friction coefficient of 0.1 is small. The accuracy of the finite element model is proved and the subsequent simulation can be guided.

Figure 9. Expansion process photograph
5. SET FINITE ELEMENT OPTIMIZATION ANALYSIS

According to the chapter 2 analysis conclusion charts, the solid expandable tubular does not fit the expansion cone after expansion, and a certain degree of non-fitting also appeared in the laboratory experiment. This indicates that the cone surface contact of the expansion cone is insufficient, and the expansion zone does not function properly, which reduces the expansion quality. At the same time, because the expandable tubular exceeds the expected expansion ratio, the compression ratio of the rubber ring is increased, which is liable to cause damage to the rubber. In view of the above problems, this chapter is based on finite element simulation, starting from the expansion cone and rubber ring to optimize the expansion stability and enhance the rubber sealing ability of the solid expandable tubular.

5.1. Optimization design of expansion cone round

The optimized design of the expansion cone is achieved by structural improvement of the expansion cone. On the one hand, the expansion driving force is reduced. And on the other hand, the axial stress generated when the sleeve is expanded is reduced. According to the results of earlier research and Section 2, this paper studies the transition section of the sizing zone and the expansion zone of the expansion cone. Pour the transition cone into rounded corners, as shown in Fig. 10.

For a Φ114mm expansion cone, the round has a maximum radius of 200mm. In this paper, the rounded corners are divided into four groups of 50mm, 100mm, 150mm and 200mm. The influence of the radius of the expansion cone on the mechanical properties of the expandable tubular is studied. In order to speed up the simulation efficiency, an expandable tubular model with three rubber rings with a total length of 500mm is used. After numerical calculation, the result is shown in Fig. 11. When the round radius of the expansion cone is greater than 50mm, the sizing method has better sizing effect. This indicates that the optimized expansion cone is uniformly rubbed at the site where the structure is abrupt, the pressing force applied to the solid expandable tubular is relatively uniform, and the expansion process is more stable. This can increase the expansion quality of the expandable tubular.

The maximum stress, residual stress and maximum expansion driving force of the expansion process of different round radius expansion cones and ordinary expansion cones are obtained by simulation, as shown in Fig. 12. As the round radius of the expansion cone increases, the maximum stress decreases during the expansion process. The maximum stress in the expansion process occurs in the sizing zone. The expansion of the expansion cone improves the expansion quality of the expansion zone of the expansion pipe, which makes the expansion rate stable, effectively reduces the maximum stress generated during the expansion process, and reduces the fracture possibility of the expansion pipe. As the round radius of the expansion cone increases, the residual stress first decreases and then increases.
And excessive residual stress causes the solid expandable tubular to weaken its internal pressure resistance. As the round radius of the expansion cone increases, the maximum expansion driving force first increases and then decreases. Expansion driving force is mainly supplied by hydraulic and mechanical means, and the maximum 18.2 ton driving force obtained by the analysis is relatively easy to realize.

| Expansion cone type | Stress (Mpa) | Driving force (t) |
|---------------------|--------------|------------------|
| Tradition           | 400          | 14               |
| R50                 | 450          | 16               |
| R100                | 500          | 18               |
| R150                | 400          | 14               |
| R200                | 450          | 16               |
| R250                | 500          | 18               |

In summary, the optimized expansion cone has obvious mechanical performance advantages over the traditional expansion cone. The optimized expansion cone has the maximum stress, residual stress and maximum expansion driving force. This can effectively improve the expansion quality of the expandable tubular and provide stable expandable tubular performance. Simulation results of the expansion cones with different round radius are combined to determine the best expansion performance for the Φ114 mm expansion cone with a radius of 150 mm.

5.2. Optimization design of expansion cone round

For the large-diameter expandable tubular discussed in this paper, it is necessary to ensure that the rubber ring is small in thickness and provides sufficient suspension force. Here, by analyzing the influence of the shape change of the rubber ring on the maximum contact pressure and the suspension force, it provides a theoretical basis for the design of the large-diameter solid expansion pipe tool.

According to the analysis results, the mechanical properties of the rubber ring are studied by using an expansion cone with a radius of 150 mm. The solid expandable tubular, the expansion cone and the inner and outer diameters of the sleeve remain unchanged. The model has a total length of 500 mm and evenly distributes three rubber rings with a length of 50 mm. The stress results take the average of the three rings.

In order to ensure interference between the rubber and the casing, the thickness of the rubber ring must be greater than 0.65 mm. Therefore, this paper selects the thickness of the five groups of 0.8 mm, 1.2 mm, 1.6 mm, 2.0 mm and 2.4 mm for comparative analysis. The contact stress result of rubber rings of different thicknesses is shown in Fig. 13. The contact pressure increases as the thickness of the rubber
ring increases. And it concentrates on the upper part of the single rubber ring, the outlet section of the expansion cone. This may be due to the accumulation of rubber during the expansion process. The contact pressure of Fig. 13(a) is zero because the wall thickness of the expandable tubular is reduced and the rubber ring cannot reach the desired sealing capacity.

![Contact pressure curve](image)

Figure 14. Average contact pressure curve

Fig. 14 shows the average contact pressure of rubber rings of different thicknesses. The rubber ring contact pressure distribution law of different spatial positions is similar, that is, the concave shape with a high right side. When the thickness of the rubber ring reaches 2.0mm or more, the contact pressure of the rubber ring can reach a maximum of 30MPa or more. The solid expandable tubular rubber ring is made of vulcanized nitrile rubber, which may cause permanent deformation and damage to its sealing ability under excessive pressure. Moreover, the single-ring suspension force of the rubber ring with a thickness of 1.2mm is 2.5ton, and the single-ring suspension force of the rubber ring with a thickness of 1.6mm is 5.0ton. The expansion of the expandable tubular can be achieved inside the interval. In summary, for the environment used in this paper, the optimal rubber ring thickness range is 1.2mm - 1.6mm.

6. CONCLUSIONS

(1) Based on the uniaxial tensile test of SA106b and the laboratory test of ST, the elastoplastic constitutive model of SA106b is fitted, and an accurate finite element model of SET is established. Compare the driving force results of simulation and the laboratory test, taking the friction coefficient as 0.1;

(2) The expansion cone round between the sizing zone and the expansion zone is studied, which provides a theoretical basis for improving the expansion quality of the expansion pipe. As the round radius of the expansion cone increases, the maximum stress decreases, the residual stress first decreases then increases, and the maximum expansion driving force first increases then decreases. For the Φ114mm expansion cone, the round radius of 150mm can be used to achieve the best expansion performance;

(3) The effect of rubber rings thickness on contact pressure and suspension force is studied. Both the contact pressure and the suspension force increase with the thickness of the rubber ring, and the contact pressures distribution exhibit a concave shape with a high right side. To ensure that the solid expandable tubular has sufficient suspension force and does not cause damage to the sealing performance, the optimal rubber ring thickness interval is 1.2mm-1.6mm;

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