Parameter optimization of rubber cylinder of expansion liner hanger based on numerical simulation

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Abstract
The expansion liner hanger has great advantages over the traditional slip liner hanger. Therefore, scholars at home and abroad have carried out extensive research on the expansion liner hanger, but there are few researches on the optimization design of the rubber cylinder of the expansion liner hanger. How to select the size and material of the rubber cylinder is very important to the suspension performance of the expansion liner hanger. Based on the Ø245 mm × Ø178 mm expansion liner hanger as the research object, the condition of the changes under the wall thickness and friction coefficient, using the finite element analysis software of rubber cylinder elongation rate, stress conditions are numerically simulated. The results show that when the wall thickness of the rubber cylinder is \( t = 1.5 \text{ mm} \), the elongation rate of the rubber cylinder is 2.14 mm, and then the elongation rate of the rubber cylinder increases rapidly with the increase of the wall thickness, and the growth of elongation rate slows down when the wall thickness is \( t \geq 3.0 \text{ mm} \). The contact stress of the inner and outer walls of the rubber cylinder increases with the increase of the wall thickness, and the contact stress of the inner wall of the rubber cylinder fluctuates, and the fluctuation tends to be gentle when \( t \geq 2.5 \text{ mm} \), and the residual stress of the rubber cylinder is uniformly distributed when \( t \geq 2.5 \text{ mm} \). When different friction coefficients (0.05, 0.1, 0.15, 0.2, 0.25, 0.3) were taken between the rubber cylinder and the outer casing, the elongation rate of the rubber cylinder was negatively correlated with the friction coefficient, and the average contact stress on the wall was positively correlated with the friction coefficient. The results of this paper provide guidance for the design and material optimization of the rubber cylinder in the expansion liner hanger.

Keywords: Expansion liner hanger, Rubber cylinder, Finite element, Elongation, Stress, The friction coefficient

Introduction
In recent years, with the rise and development of horizontal well drilling technology in deep, ultra-deep and highly deviated Wells, drilling well conditions are more complex, and the traditional slip hanger is gradually exposed to its shortcomings [1–5]: (1) the sealing performance of the overlapping section of liner is poor and hydraulic leakage easily occur; (2) in the process of running, the liner is easy to sit in advance or easy to
hang and fail after being lowered to the position; (3) the small flowing area is easy to
cause leakage, cement return height is not enough, annular air invasion, etc. In order to
solve these problems, scholars at home and abroad have carried out extensive research
on the expansion liner hanger. Zhang Renyong [6], Wei Songbo [7], and Z S Qamar [8]
et al. studied the influence of the material, processing method, and structure size of the
expansion cone on the friction force and the expansion rebound during the expansion
of the expansion tube. B R Shi study confirmed that the bionic non-smooth surface of
the expansion tube has the properties of reducing friction and wear [9]. Liu Feng [10],
Zhang Jianbing [11], and Filippov A [12] respectively established mathematical models
of expansion force and outer diameter reduction of expansion tube, gave the calculation
formula of anti-extrusion strength of high-precision casing after expansion, and tested
the influence of expansion of API class expandable steel pipe on mechanical properties.
Gu Lei and Ma Kaihua et al. calculated and tested the expansion force and suspension
force of the expansion body based on the actual working state, and determined the relation-
ship between them and the expansion tube structure [13]. Colin G Ruan established
a mathematical model including stress and strain, expansion cone diameter and angle,
friction between expansion cone and expandable tube, expansion tube length shrinkage
rate, and expansion rate to accurately calculate the expansion force [14]. Chen Jingjing
and Li Dejun et al. used the computational model to show that the expansive force would
be affected by the strain-hardening behavior and improved the computational model of
the expansive force of the expansion tube based on the Hollomon relation [15]. A C Seibi
identified the influence of vibration effect on the structural response of expansion tube
during expansion process, and extended the quasi-static tubular expansion theory to
describe the dynamic friction effect caused by stick-slip phenomenon [16]. Liang Kun
[17], Omar S Al Abri [18], Zhang Jian [19], Jiang Xiangdong [20], Zhang Yu [21], and
T Pervez [22] et al., based on the finite element theory, established a numerical model
to analyze and study the relationship among factors such as expansion force, expansion
rate, friction coefficient, thinning amount of wall thickness, and cone angle and obtained
relevant mathematical expressions.

At present, the research on the expansion liner hanger mainly focuses on the expan-
sion force, suspension force and structural parameters’ optimization of expansion tube
and expansion cone. However, there is little research on the influence of the wall thick-
ness of rubber cylinder on the contact stress, equivalent stress, and elongation rate and
the influence of the friction coefficient of rubber cylinder on the contact stress and
elongation rate. In this paper, a Ø245 mm × Ø178 mm expansion liner hanger model
is established by using finite element software, and the variation of elongation rate and
stress under different wall thickness and friction coefficient of rubber cylinder is studied,
which provides a basis for the design of the rubber cylinder and material optimization.

Methods
The main design principle of the expansion liner hanger is to push the expansion cone
down to make the expansion tube deform, extruded the high performance rubber of the
vulcanization in the outer wall of the expansion tube to sufficiently fill the annular space
between the expansion tube and the outer casing, and utilize the friction between the
rubber cylinder and the casing to achieve the purpose of suspension and sealing. As the
research focus of this paper is on the rubber cylinder, according to the principle of the expansion liner hanger mentioned above, it is necessary to adopt the appropriate rubber constitutive model and set the corresponding contact properties and friction coefficient for the rubber cylinder according to the actual situation, so as to obtain the elongation rate and mechanical properties of the rubber cylinder.

This study evolved through a set of steps, these are the following:

1) Firstly, the constitutive model of the material of the rubber cylinder is determined;
2) Secondly, the true stress-strain curve of the expansion liner is obtained according to the experimental data processing;
3) Then, the finite element model is established according to the actual size and boundary conditions are set up;
4) Finally, descriptive statistical analysis was performed based on the obtained data.

Constitutive model of rubber cylinder

Among the various NBR constitutive models (stress-strain model of nitrile rubber), Mooney-Rivlin model has been widely used due to its simple model, fewer calculation parameters, and high analysis efficiency [23]. After determining the rubber materials parameters $C_{01}$ and $C_{10}$ of two-parameter Mooney-Rivlin model, the stress-strain curves obtained by simulating different rubber materials through nonlinear finite element analysis have good fit with the measured engineering stress curves [24], so the two-parameter Mooney-Rivlin model is adopted in this paper.

The general form of Mooney-Rivlin model is as follows:

$$W = \sum_{i+j}^N C_{ij}(I_1 - 3)^i(I_2 - 3)^j + \sum_{i}^N \frac{1}{D_i}(I_3 - 1 - R)^2i$$

Where, $W$ strain energy density; $I_1 = \lambda_1^2 + \lambda_2^2 + \lambda_3^2$ first strain tensor invariant; $I_2 = (\lambda_1 \lambda_2)^2 + (\lambda_2 \lambda_3)^2 + (\lambda_3 \lambda_1)^2$ second strain tensor invariant; $I_3$ third strain tensor invariant; $C_{ij}$ elastic material constant; $\lambda_1, \lambda_2, \lambda_3$ three principal elongations respectively; $D_i$ Define the compressibility of the material; $R$ define volume expansion with temperature.

Assuming that the material is incompressible, we get $I_3 = (\lambda_1 \lambda_2 \lambda_3) = 1$ and is at a constant temperature $R = 0$, the two-parameter Mooney-Rivlin model can be obtained as follows:

$$W = C_{10}(I_1 - 3) + C_{01}(I_2 - 3).$$

In the special case of uniaxial tension, the stress-strain equation of the Mooney-Rivlin model is as follows:

$$\sigma/2(\lambda - \lambda^{-2}) = C_{10} + C_{01}/\lambda$$

Where $\sigma$ is the engineering stress; $\lambda$ is the principal elongation.
Considering the working temperature and aging time of rubber, it is considered that under certain conditions, the rubber materials parameters $C_{01}/C_{10}$ only has a great relationship with the temperature change \[25\]. The specific values of material parameters $C_{10}$ and $C_{01}$ at 150 °C are shown in Table 1.

In this paper, the rubber cylinder outside the expansion tube is simulated and analyzed. According to its working conditions, the material parameters under 150 °C condition are selected $C_{10} = 2.573$, $C_{01} = 0.904$.

**Material parameters of the expansion liner**

The essence of expansion liner hanger is the extended application of expansion liner technology in oil and gas field wells, which belongs to the category of metal plastic forming. The expansion forming of the expansion tube is a large plastic deformation problem, which includes both elastic deformation and plastic deformation. The plastic parameters of the material are very important for the simulation of the expansion process when the downhole temperature is considered.

Uniaxial tensile tests of twin-induced plasticity steel (TWIP) at different temperatures have been carried out in some literature, and the engineering stress-strain curves of TWIP at different temperatures have been obtained \[26\]. The typical metal plasticity model in ABAQUS defines the post-yield characteristics of most metals, and the description of plastic behavior requires the input of the real stress-strain data of the material. The real stress-strain curve can be obtained by processing the experimental data, as shown in Fig. 1. At 150 °C, the main material parameters of the expansion tube during elastoplastic deformation are shown in Table 2.

**Establishment of the finite element model with rubber cylinder**

In this paper, Ø245 mm × Ø178 mm expansion liner hanger as the research object. The total length of the expansion liner hanger is designed to be 2.0 m, the size of the expansion tube before expansion is Ø200 × 10 mm, the designed expansion range of the expansion tube is 10%, the diameter of the gauge section of the expansion cone is 198 mm, the angle of expansion cone is 10°, and the initial length of the rubber cylinder is 200 mm. To avoid stacking of rubber cylinders, set the length of rubber cylinder vulcanized on the expansion tube to 280 mm. Based on the working principle and component structure of the expansion liner hanger, it can be determined that in the process of axial expansion, the installation error and ellipticity of its structural components have little influence on the analysis to be carried out, which can be ignored. Therefore, it is established as an axial symmetrical model for analysis \[27, 28\]. The expansion liner hanger has multiple identical rubber cylinder along the axial direction, but the differences between

| Table 1 | Material parameters of NBR at 150 °C |
|---------|-------------------------------------|
| M—R model parameters | The temperature/°C |
| | 150 |
| $C_{10}$ | 2.573 |
| $C_{01}$ | 0.904 |
the rubber cylinders during expansion are slight; therefore, one of the rubber cylinders was taken for analysis, and the finite element model is shown in Figs. 2 and 3.

In the finite element model shown in Fig. 3, a fixed constraint was applied to the outer casing, the upper shaft end of the expansion tube was fixed, and a running speed of 3 m/min to the expansion cone was applied. In order to better get the deformation of the rubber cylinders in the analysis process, the mesh of model contact area is refined [29]. The expansion cone is set as a discrete rigid body, the expansion tube as a deformable body (the plastic strain and real stress data are generated by the real stress-strain curve), and the rubber cylinder material is an isotropic hyper-elastomer [30, 31]. For expansion cone and expansion tube, expansion tube and rubber cylinder, rubber cylinder, and outer casing are respectively provided with face to face contact. The penalty function of tangential behavior between expansion cone and expansion tube is 0.15, and that of expansion tube and rubber cylinder, and rubber cylinder and outer casing is 0.2.

Table 2 Physical performance parameters of the expansion liner at 150 °C

| Temperature/°C | Elasticity modulus/Pa | Poisson's ratio | Yield strength/Pa | Tensile strength/Pa |
|---------------|------------------------|----------------|--------------------|--------------------|
| 150           | $2.06 \times 10^{11}$  | 0.26           | $3.67 \times 10^8$ | $1.139 \times 10^9$ |

Fig. 1 Material characteristic curve of expansion liner
**Fig. 2** Model of expansion liner hanger with rubber cylinder

**Fig. 3** Finite element mesh model of expanded liner with rubber cylinder
Results

Elongation rate analysis of rubber cylinder with different thickness

The thickness parameters of rubber cylinder were simulated by finite element method. The wall thicknesses of several groups of rubber cylinder (1.0 mm, 1.5 mm, 1.8 mm, 2.0 mm, 2.5 mm, 3.0 mm, 3.5 mm, 4.0 mm, 4.5 mm, 5.0 mm) were selected for analysis and optimization. The internal equivalent stress distribution and the contact stress of the inner and outer walls of the rubber cylinder with different wall thicknesses after expansion is studied. The size parameters of the rubber cylinder were optimized based on the analysis results.

The numerical simulation shows the rubber cylinder on the outer wall of the expansion tube expands constantly with the expansion tube during the operation of the expansion cone. When the outer wall of the rubber cylinder contacts with the outer casing, the rubber cylinder is constantly extruded and deformed by the expansion tube and the outer casing, and the axial length is stretched to varying degrees.

As shown in Fig. 4A, when $t = 1.0$ mm, the rubber cylinder does not contact with the inner wall of the outer casing when the expansion tube is driving the rubber cylinder to expand radially, so the rubber cylinder exhibits axial shortening. When the expansion is completed, the axial shortening of the rubber cylinder reaches 7.8 mm, and the wall thickness of the rubber cylinder is reduced to $t = 0.957$ mm. As can be seen from Fig. 4B, there is a difference of the rubber cylinder in the amount of shortening of the upper and lower sections. Because the surface contact stress between the rubber cylinder and the expansion tube is small at the beginning of expansion, the friction force between them cannot restrict the motion of the rubber cylinder. With the continuous expansion, the surface contact stress between the rubber cylinder and the expansion tube increases, so that the friction force can inhibit the movement of the rubber cylinder. So after that, the amount of shortening on the top of the rubber cylinder stays the same.

As shown in Fig. 5, when the rubber cylinder is taken at $t = 1.5$ mm, the rubber cylinder expands radially and contacts with the inner wall of the outer casing, then the rubber cylinder is compressed and elongated axially, the elongation reaching 2.14 mm.

![Axial shortening displacement diagram of rubber cylinder (t = 1.0 mm)](image-url)
When the wall thickness of the rubber cylinder is $t \geq 1.5$ mm, the rubber cylinder shows overall elongation under the extrusion of expansion tube and outer casing. In order to better understand the relationship between the wall thickness and the elongation rate of rubber cylinder, the scatter points of the elongation rate under different wall thicknesses of rubber cylinder is fitted to get the curve shown in Fig. 5. Wherein, the relationship between the elongation rate $\eta$ and its initial length $l$ of the rubber cylinder. The formula of elongation rate is as follows:

$$\eta = \frac{l_1 - l}{l} \times 100\%$$

Where $l$ length of the rubber cylinder before compression; $l_1$ length of the rubber cylinder after compression.

It can be seen from the curve of the relationship between the axial elongation rate and the wall thickness of the rubber cylinder that when the wall thickness of the rubber cylinder is $t < 1.5$ mm, the rubber cylinder shows axial shortening; when the wall thickness of the rubber cylinder is $t \geq 1.5$ mm, the rubber cylinder elongates axially. When the wall thickness of rubber cylinder is $t = 3.0$ mm, the slope of the
axial elongation rate curve of the rubber cylinder is larger and the axial elongation increases faster. When the wall thickness of the rubber cylinder is \( t \geq 3.0 \) mm, the slope of the axial elongation rate curve of the rubber cylinder gradually decreases, and the growth rate gradually slows down.

In order to better explain such changes, the average annular clearance between the expansion tube and the outer casing is given in Fig. 7 according to the compressed thickness of the rubber cylinder shown in Fig. 6. It can be seen that the average annular clearance increases approximately uniformly with the increase of the wall thickness. Assuming that the average annular clearance increases in equal proportion, since the rubber model is incompressible, the relation curve between the axial elongation rate of the rubber cylinder and the wall thickness can be obtained as shown in Fig. 8. Figure 8 shows that the increase of axial elongation rate of rubber cylinder decreases with the increase of wall thickness in a parabolic potential. Combined with Fig. 7, when \( t = 2.0\text{–}2.5 \) mm, the average annular clearance value is slightly larger than the black slant line in Fig. 7, so the axial elongation rate of the rubber cylinder shown in Fig. 7 is smaller than that shown in Fig. 8, and the curve in Fig. 7 is close to a straight line. Similarly, when \( t = 3.0\text{–}4.5 \) mm, the average annular clearance value is larger than the slant line, so the curve tilt rate in Fig. 7 is smaller. Therefore, the relation curve between the axial elongation rate of rubber cylinder and wall thickness in Fig. 7 forms an obvious turning point when \( t = 3.0 \) mm.

![Fig. 6 Compressed thickness of the rubber cylinder along the axial length](image)
Fig. 7 Relation curves of axial elongation rate, average annular clearance, and wall thickness of rubber cylinder.

Fig. 8 The relation curve of axial elongation rate and wall thickness of rubber cylinder with uniform annular clearance growth.
Analysis of contact stress of rubber cylinder with different wall thickness

The rubber cylinder is squeezed by the expansion tube and outer casing to form an effective annular seal and suspend the liner. Therefore, the contact stress on both sides of the rubber cylinder and the equivalent stress inside the rubber cylinder determine the sealing and suspension performance of the rubber cylinder in the hole. Therefore, it is necessary to study the two parameters of the rubber cylinder.

When the wall thickness of the rubber cylinder is $t = 1$ mm, the contact stress of the outer wall of the rubber cylinder is 0, indicating that this wall thickness is not enough to fill the annular clearance between the outer wall of the expansion tube and the inner wall of the outer casing. When the wall thickness of the rubber cylinder is $t \geq 1.5$ mm, the wall thickness of the rubber cylinder fills the annular clearance between the outer wall of the expansion tube and the inner wall of the outer casing and is squeezed. Contact stress distributions on both sides of rubber cylinders of different wall thicknesses (1.5 mm, 2.5 mm, 3.5 mm, 4.5 mm, and 5.0 mm) when are extruded

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**Fig. 9** Contact stress distribution of the inner and outer walls of the rubber cylinder
is shown in Fig. 9A–F. In order to more clearly reflect the overall distribution of contact stress on the inner wall of the rubber cylinder along the length direction, the contact stress on each unit node in Fig. 9A, B was fitted with high-order data.

According to Fig. 9, when the wall thickness of the rubber cylinder is \( t < 2.0 \) mm, the contact stress of the inner and outer walls of the rubber cylinder is small, while when \( t \geq 2.0 \) mm, the contact stress of the inner and outer walls of the rubber cylinder is large. Therefore, to hold liner suspension, the wall thickness of the rubber cylinder should be \( t \geq 2.0 \) mm.

In Fig. 9, the contact stress of the inner wall of the rubber cylinder fluctuates slightly on the contact stress line of the outer wall of the rubber cylinder, and the trend between the two is the same. It can be considered that the contact stress of the inner and outer walls of the rubber cylinder is approximately equal.

In addition to the contact stress, there is also residual stress inside the rubber cylinder when it is squeezed by expansion tube and casing. Residual stress refers to the equivalent stress in the rubber cylinder when it is squeezed after the expansion. The equivalent stress distribution at each point along the axial direction of the rubber cylinder can be determined by Fig. 10. The distribution of residual stress inside the material determines the initial stress state of the rubber cylinder. A better residual stress state of the rubber cylinder can increase its service life and reliability. The equivalent stress distribution and average equivalent stress of rubber cylinders with different wall thicknesses are shown in Fig. 10 and Table 3 respectively.

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**Table 3** Average equivalent stress of rubber cylinder with different wall thickness

| Initial wall thickness (mm) | 1.0 | 1.5 | 1.8 | 2.0 | 2.5 | 3.0 | 3.5 | 4.0 | 4.5 | 5.0 |
|-----------------------------|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|
| Average equivalent stress (MPa) | 0.02 | 1.98 | 2.75 | 3.11 | 5.25 | 7.51 | 8.12 | 8.9 | 9.68 | 10.83 |
Combined with the contact stress distribution diagram of the inner and outer walls of the rubber cylinder shown in Fig. 9 and the equivalent stress diagram after compression shown in Fig. 10, it can be seen that at $t < 2.5$ mm, the contact stress fluctuated greatly up and down along the axis of the rubber cylinder shown in Fig. 9A, B, and C. The equivalent stress distribution along the axis of the rubber cylinder was uneven in Fig. 10 ($t = 1.5$ mm, $t = 2.0$ mm). When the wall thickness of the rubber cylinder is $t \geq 2.5$ mm, the contact stress in Fig. 9C–F fluctuated slightly along the axis of the rubber cylinder with a gentle curve, and the equivalent stress in Fig. 10 ($t = 2.5$–$5.0$ mm) was evenly distributed along the axis.

According to the above, when selecting the thickness of the rubber cylinder, the rubber cylinder of $t \geq 2.5$ mm should be selected.

Analysis of elongation rate and average contact stress of rubber cylinder under different friction coefficients

Due to the complex downhole environment, it is difficult to determine the friction coefficient of the rubber cylinder because of the smoothness of the casing wall, the debris, mud, burr and cracks on the casing wall. Under the influence of its material composition, the friction coefficient of nitrile rubber is generally between 0.05 and 1.0 [32, 33]. Six sets of data (friction coefficients: 0.05, 0.1, 0.15, 0.2, 0.25, 0.3) were selected for analysis based on the requirements of liner suspension.

According to the results of the above-mentioned wall thickness of the rubber cylinder, the optimal wall thickness of the rubber cylinder for the expansion liner hanger is $t \geq 2.5$ mm, and the initial wall thickness of the rubber cylinder is determined to be 4.0 mm.

Through post-processing of the analyzed data, the elongation rate and average contact pressure of the rubber cylinder under different friction coefficients were obtained as shown in Table 4, and the obtained data were curved to obtain curves as shown in Fig. 11. According to the curve in the figure, with the change of the friction coefficient between the rubber cylinder and the outer casing, the elongation rate of the rubber cylinder decreases with the increase of the friction coefficient, and the reduction trend of the elongation rate gradually slows down with the increase of the friction coefficient. The average contact stress increases with the increase of the friction coefficient, and its increasing trend is also slowed down with the increase of the friction coefficient. In order to better represent the relationship between friction coefficient, elongation rate,

| Friction coefficient | Elongation rate (%) | Average contact stress (MPa) |
|----------------------|---------------------|-----------------------------|
| 0.05                 | 53.6                | 15.7                        |
| 0.10                 | 37.8                | 18.5                        |
| 0.15                 | 30.3                | 20.3                        |
| 0.20                 | 25.1                | 22.1                        |
| 0.25                 | 20.9                | 23.2                        |
| 0.30                 | 17.8                | 24.1                        |
and average contact stress, the curves of friction coefficient—elongation rate and friction coefficient—average contact stress in Fig. 11 are numerically fitted, and the following relation was obtained:

The relationship between axial elongation rate of rubber cylinder and friction coefficient:

\[ y = 547x^2 - 325x + 67; \]

The relationship between average contact stress and friction coefficient of rubber cylinder wall:

\[ y = 87.8x^2 + 63.8x + 12.8. \]

Based on determining the allowable elongation rate of the rubber cylinder of the expansion liner hanger and the average contact stress required for the expansion liner hanger to be held in place, it is easy to select the appropriate friction coefficient according to Fig. 11.

In order to verify the reliability of the fitting formula, error analysis was performed on the values of the two formulas at different friction coefficients and the data in Table 4, and the residual distribution diagram was obtained as shown in Figs. 12 and 13. Figures 12 and 13 show the distribution of residual values corresponding to average contact stress and elongation rate when different friction coefficients are taken. The base line 0.0 in the figure represents the value shown in Table 4, and the red dot
Fig. 12  Residual distribution diagram of the relation average contact stress and friction coefficient

Fig. 13  Residual distribution diagram of the relation between elongation rate and friction coefficient
represents the residual values between the fitting formula and the value in Table 4. As can be seen from the figure, the residual value of the two sets of formulas is smaller, and the fluctuation of the red point on the baseline is also smaller.

After calculation, the $R^2$ (COD) of the two formulas can be obtained as 0.99875 and 0.98823, respectively. The calculation formula of $R^2$ (COD) is

$$R^2 = \frac{TSS - RSS}{TSS} = 1 - \frac{RSS}{TSS}$$

where TSS is the total sum of square, and RSS is the residual sum of square.

$R^2$, which is also known as the coefficient of determination (COD), is a statistical measure to qualify the linear regression. It is a percentage of the response variable variation that explained by the fitted regression line. Hence, $R^2$ is always between 0 and 1. If $R^2$ is 0, it indicates that fitted line explains none of the variability of the response data around its mean; while if $R^2$ is 1, it indicates that the fitted line explains all the variability of the response data around its mean. It is generally considered that the goodness of fit is higher when the $R^2$ (COD) is greater than 0.8, and the closer the $R^2$ (COD) is to 1, the better the best fit. Therefore, it can be proved that the fitting formula has high fitting accuracy and small calculation error.

Discussion

This article focuses on the research on the rubber cylinder on the expandable liner hanger. The selection of the size of the rubber cylinder is simulated, and the stress and deformation of the rubber cylinder under different sizes and friction coefficients are analyzed. Based on stress analysis, the hanging weight of liner hanger, sealing performance, and service life of rubber cylinder can be determined. Through the deformation analysis, it is beneficial to design the structural size of the expansion liner hanger; application of different friction coefficients for analysis is conducive to the selection of rubber cylinder materials. In general, the analysis of the rubber cylinder in this article can provide guidance and help for the design of the expansion liner hanger.

However, the downhole temperature selected in this paper is 150 °C. There is no simulation under different temperature fields, which has certain limitations. At the same time, the experiments planned for the next stage have not been carried out, and the simulation results have not been compared with the actual situation, so there may be some errors.

Conclusions

This paper studied the Ø245 mm × Ø178 mm expansion liner hanger, established the finite element model according to the actual engineering size, simulated the rubber cylinder with different wall thickness and friction coefficient, and obtained the change of elongation rate and the various stresses of the rubber cylinder. The specific conclusions are as follows:

1. When the wall thickness of the cylinder $t \geq 1.5$ mm, the elongation rate of the cylinder increases with the increase of the wall thickness of the cylinder; when $t \geq 3.0$ mm, the growth rate of the elongation rate of the cylinder slows down.
2. When the contact stress of the inner and outer walls of the rubber cylinder is greater when \( t \geq 2.0 \) mm, the liner can be suspended effectively. The fluctuation of contact stress and the distribution of residual stress are improved when \( t \geq 2.5 \) mm.

3. The formula fitting among the elongation rate, the average contact stress and the friction coefficient of the rubber cylinder is carried out, and the relationship of high precision is obtained, which provides the basis for the reasonable selection of the rubber cylinder material.

Abbreviations
\( t \): The wall thickness; NBR: Stress-strain model of nitrile rubber; \( \lambda_1, \lambda_2, \lambda_3 \): Three principal elongations respectively; \( D_i \): Define the compressibility of the material; \( C_p \): The compressibility of the material; \( W \): Strain energy density; \( i_1 \): First strain tensor invariant; \( i_2 \): Second strain tensor invariant; \( i_3 \): Third strain tensor invariant; \( R \): Volume expansion with temperature; \( \sigma \): Engineering stress; \( \lambda \): Principal elongation; \( M—R \): Mooney-Rivlin; TWIP: Twin-induced plasticity steel; \( \eta \): Elongation rate; \( l \): Length of the rubber cylinder before compression; \( l_1 \): Length of the rubber cylinder after compression; COD: \( R_1^2; R_2^2; R_3^2 \); TSS: The total sum of square; RSS: The residual sum of square.

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Authors’ contributions
YC helped with the writing of the manuscript, provided technical guidance, and was a major contributor in writing the manuscript. XL collected and analyzed the data and wrote the manuscript. HY provided funding support. CH provided ideas and guidance on the manuscript. GPX edited the manuscript and provided some references. JT, HCW, and AQS provided language modification and polishing for this article. All authors read and approved the final manuscript.

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Declarations
Ethics approval and consent to participate
Not applicable.

Consent for publication
All the authors have agreed to publish this article.

Competing interests
The authors declare that they have no competing interests.

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