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Thermal failure of rubber bushing of a Positive Displacement Motor: A study based on thermo-mechanical coupling

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HIGHLIGHTS

- Material testing of a kind of nitrile-butadiene rubber was carried out to get Mooney–Rivlin parameters for calculation.
- Finite element method was used to calculate the thermo-mechanical coupling effect on PDM rubber bushing.
- The mud pressure, rotor rotation speed and formation temperature effects on bushing temperature rise were studied.
- The rubber bushing's heat failure modes in engineering practice are consistent with the calculation results.

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ABSTRACT

Viscoelastic hysteresis of the rubber material often causes thermal failure of PDM (Positive Displacement Motor) stator bushing. Combining heat transfer modeling with material testing, finite element method was used to calculate the temperature field and thermal stress of the bushing, and their sensitivity to Poisson ratio, mud pressure, rotor speed and formation temperature. The results show that the maximum thermal stress occurs at the bottom of the arc of the bushing, and the maximum deformation occurs at the arc top. The distribution of temperature field appears oval-shaped and the temperature gradient is large. Bushing's temperature decreases with the increasing of formation temperature, but increases with the increasing static pressure and the rotor speed. The rubber bushing's thermal failure modes observations in practice are consistent with the calculation results.

1. Introduction

Positive displacement motor (PDM) drill has become the most widely used downhole tool in oil drilling engineering and well workover since its inception in the 1950s. Its structure is shown in Fig. 1. The motor assembly consists of a stator and a rotor. The stator is a pipe lined with a vulcanized rubber bushing, usually nitrile-butadiene. The steel rotor has a hypocycloidal surface, and is driven by the high-pressure drilling fluid [1,2]. During drilling, the deformation of bushing will directly influence the PDM performance in terms of work efficiency and drilling speed.

Because of the higher formation temperature, the friction between the rotor and the stator, and the hysteresis heat coming from the rubber bushing, the bushing's temperature will rise gradually, and then the working torque will increase too, resulting in the performance decline of the tool [3–5].

High temperature will accelerate the aging and fatigue of rubber bushing, leading to premature failure, and shorten tool life. At present, most scholars have been focusing on the rubber bushing's stress-strain law without considering the temperature and its influence on the properties of bushing. Therefore, it is necessary to research thermal-stress coupling and temperature field of rubber bushing. In this paper, the finite element method was used to establish the model of rubber bushing based on the material testing and PDM's structure. The stator temperature rise mechanism was discussed, the influence of the temperature on the stress and strain was analyzed, and effects of different parameters on the temperature field of the bushing were studied.

2. The rubber's constitutive models and material testing

2.1. The constitutive models for rubber materials

Rubber material is similar to super elastic body which cannot be compressed, with geometric nonlinearity, material nonlinearity

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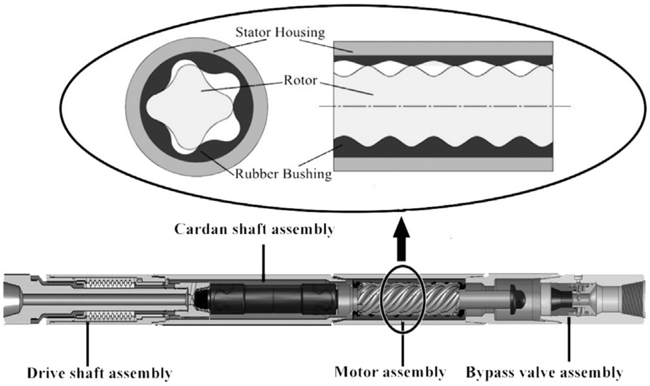

Fig. 1. Structure of Positive Displacement Motor.

and contact nonlinearity. Its Poisson's ratio generally ranges from 0.49 to 0.4999.

Constitutive models of rubber generally include Neo-Hookean strain energy function, exponential-hyperbolic algorithm, Mooney–Rivlin, Klosner–Segal model and Ogden–Tschoegl model. In the strain range of 150%, Mooney–Rivlin constitutive model is of satisfactory accuracy [6], and requires only two experimentally determined parameters. Therefore, in this paper the two-parameter Mooney–Rivlin constitutive model was used to describe the mechanical properties of rubber material under large deformation, using the equation [7]:

$$W = C_{10}(I_1 - 3) + C_{01}(I_2 - 3) \tag{1}$$

Where W is strain energy density, C_{10} and C_{01} are Mooney–Rivlin coefficients of material, I_1 and I_2 are the first and the second invariant of strain tensor, respectively. The relationship between E, G and material constants is:

$$G = \frac{E}{2^*(1+\mu)} \tag{2}$$

Where μ is Poisson's ratio.

For incompressible rubber material, the values of Young modulus E and shear modulus G are calculated as follows:

$$E = 6(C_{10} + C_{01}) \tag{3}$$

$$G = 2(C_{10} + C_{01}) \tag{4}$$

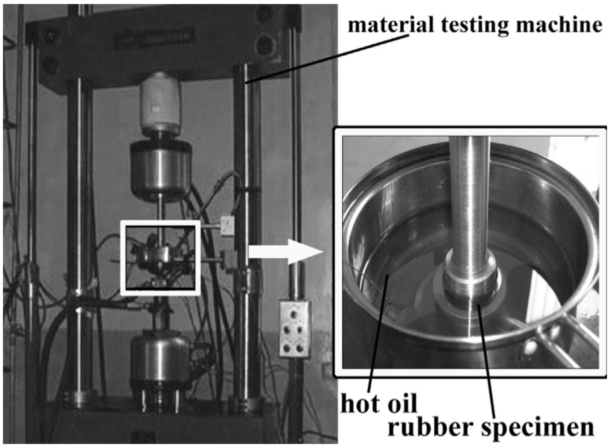

Fig. 2. The experimental apparatus for the testing of the bushing rubber.

2.2. The material experiment

Experiments were conducted to determine parameters of nitrile-butadiene rubber. The specimen tested was a cylinder with the diameter of 29 mm and height of 13 mm, in accordance with Chinese standard GB/T 7757-93. The oil bath temperature was 65 °C, the original ambient formation temperature for drill. The experimental apparatus is shown in Fig. 2, and the obtained data are summarized in Tables 1 and 2. The values of parameters identified in these experiments are $E = 11.49$ MPa, $C_{10} = 1.879$ and $C_{01} = 0.038$ within 10% strain, which corresponds to the actual working conditions.

3. Theoretical analysis of temperature field and calculation model

3.1. Mathematical model of heat transfer for the rubber bushing

There are four sources of heat transferred to the bushings, namely (a) the downhole formation temperature (the main source of the temperature field), (b) frictional heat at the rotor/bushing interface, (c) heat generated by friction between high-pressure drilling fluid and the inner wall bushing, and (d) the lag loss heat [8,9].

For the rubber used in the process, its stress σ and strain ε are given by [10]:

$$\sigma = \sigma_{\max} \sin(\omega t + \delta) \tag{5}$$

$$\varepsilon = \varepsilon_{\max} \sin \omega t \tag{6}$$

Where ω is angular velocity, δ is hysteretic angle, t is the working time. According to the heating mechanism of bushing rubber, rotor in the bushing rotates within a cycle T_c, The heat loss Q from rotor to the bushing per unit volume and over a rotation cycle is given by:

$$Q = 2\int_0^{\pi/\omega} \sigma \frac{d\varepsilon}{dt} dt \tag{7}$$

$$Q = \pi E' \varepsilon_{\max}^2 = \pi E \varepsilon_{\max}^2 \tan \delta \tag{8}$$

Where E' is loss modulus and $\tan\delta$ is loss factor.

At the rotation cycle T_c, at which the rotor rotational speed is n, the heat loss from rotor to the bushing per unit time is given by:

$$q = \frac{Q}{T_c} = \pi n E \varepsilon_{\max}^2 \tan \delta \tag{9}$$

3.2. The heat conduction differential equation

When the screw rotor in the bushing is driven by the mud to rotate continuously, each node of the bushing has to undergo periodic stress and strain. The constant tension and pressure will cause large deformation, and hysteresis loss will generate heat. According to the first law of thermodynamics and the Fourier law [10–12], unsteady temperature field and heat balance equation in polar coordinates of stator screw drill bushing is given by:

$$\frac{1}{r}\frac{\partial}{\partial r}\left(\lambda r \frac{\partial T}{\partial r}\right) + \frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2} + \frac{q}{\rho c} = \frac{\partial T}{\partial t} \tag{10}$$

Where T is temperature (°C), q is the rate of heat generated per unit volume (W/m^3), λ is thermal conductivity (W/(m °C)), ρ is density