Automatic packer reliability prediction under pulsed transient flooding of hydrocarbon reservoirs

M Ya Khabibullin and R I Suleimanov

Ufa State Petroleum Technological University, Branch of the University in the City of Oktyabrsky, 54a, Devonskaya St., Oktyabrsky, Republic of Bashkortostan, 452607, Russian Federation

E-mail: m-hab@mail.ru

Abstract. For pulsed transient flooding of hydrocarbon deposits (installed on the tubing string below the packer) of the injection well, hydraulic impulse devices are used. When they work (generation of pulses in the injected fluid) in the tubing string, hydraulic shocks occur at a frequency equal to the output frequency of pulses generated by the devices. The result is a reciprocating movement of the tubing string and the packer. Hence, for effective injection of fluid, it is necessary to provide automated long-term reliability of the sealing elements of the packer and long-term sealing of the annular space. Joint movement of the packer and pressure pulsation intensify mechnochemical aging processes. To improve durability of packer seals during long-term pulsed injection of fluid into the well, it is necessary to replace rubber materials with rubber-fabric materials and use damping equipment along with impulse devices. Their use will significantly reduce the amplitude of packer oscillation (stroke length of sealing element S) and maximum accumulation time $W_{\text{max}}$.

1. Introduction

The main component of the packers is a sealing element whose important characteristic is a sealing ability. Tightness requirements determine the choice of seals and affect the design scheme of the entire underground unit, including devices that inject fluid into the formation.

When applying pulse transient flooding of hydrocarbon deposits (installed on the tubing string below the packer), hydraulic impulse devices are used [1, 2]. During their operation (generation of pulses in the injected fluid) in the tubing string, hydraulic shocks occur at a frequency equal to the frequency of the pulses generated at the output of the devices [3]. The result is a reciprocating movement of the tubing string and the packer. Hence, it is necessary to provide automated long-term reliability of the sealing elements of the packer and long-term sealing of the annular space for effective injection of fluid.

When designing sealings of dynamic couplings, it is necessary to analyze a set of problems of tightness, friction and wear.

The mechanism of friction and leakage of seals under fluid lubrication conditions was described on the basis of the elastohydrodynamic theory [4] which can be applied to other modes of operation of packer seals by introducing special functions $\Psi_1$ and $\Psi_2$ (dimensionless shape factors), taking into account the mode of friction during up and down strokes. At a high pressure of the medium ($p > 5$ MPa) and a large contact pressure ($p_k > 1$ MPa), the seals work under semi-liquid lubrication conditions. In this case, tightness is ensured due to a large loading coefficient ($b > 0.7$), and the resource – due to the best anti-friction sealing materials. For each double stroke of the packer, the volume of leakage through
the seals is equal to the difference between the film volume during the forward and reverse strokes. As a result, we have \[ V = 0.5\pi DS(\Psi_1\delta_1 - \Psi_2\delta_2) = 0.5\pi DS\Delta h, \] (1) where \( D \) is the outer diameter of the packer seal, mm; \( S \) is the length of the seal stroke, mm; \( \Delta h \) is equivalent film thickness, mm.

The film thickness can be represented as a function of the mode criterion: \( \delta_1 = F(G_1) \) and \( \delta_2 = F(G_2) \). When the packer seals work, combinations of different lubrication modes are possible when the tubing moves up and down. The correspondence of the values of functions \( \Psi_1 \) and \( \Psi_2 \) to the lubrication modes is shown in Figure 1 [5].

2. Results and discussion
Seals can operate without leakage under boundary lubrication and friction lubrication without lubricant \( \Psi = 0 \) (I, II, III); semi-fluid lubrication \( 0 < \Psi < 1 \) (IV); liquid lubrication \( \Psi = 1 \) (V). In the liquid lubrication mode with a certain combination of parameters \( v \) (packer speed up and down), \( \mu \) (dynamic viscosity of the well fluid), \( p_c \) (contact pressure in the packer seal), seals can work without leakage. When the parker moves up and down, velocities \( v_1 \) and \( v_2 \) can be different, therefore, when \( v_2 > v_1 \), there can be \( \delta_1 < \delta_2 \) and \( V = 0 \). To ensure \( V = 0 \) due to \( \delta_1 = \delta_2 \), it is necessary to assign a packer sealing profile which ensures optimal distribution of contact pressure along the sealing surface. The semi-fluid lubrication mode is the most probable when using packer seals, since loading coefficient \( b \) is determined from the sealing condition of the annular space so as to ensure guaranteed axial load. The values of special functions will be \( \psi \approx 0.3 \ldots 1 \).

If the seal makes \( \omega \) double strokes per \( 1 \) s, at \( v_1 = v_2 = v \) (where \( v_1 \) is the speed of the seal moving down; \( v_2 \) is the speed of the seal moving up), the specific leakage volume is equal to the equivalent film thickness:

\[ \bar{V} = \frac{2V}{\pi DL} = \Delta h, \] (2)

where \( V \) is the leakage volume, mm\(^3\)/s; \( D \) is the outer diameter of the seal, mm; \( L \) is the length of the sealing element, mm.

To move the packer in one direction, one can use criterion \( \bar{n} \) (respectively \( \bar{n}_1 = \delta_1/R_c, \bar{n}_2 = \delta_2/R_c \), where \( R_c \) is roughness of the inner surface of the production string) which determines the lubrication mode in the seal and the value of function \( \Psi \). There may be various combinations, but the most common one is as follows: when moving downward, \( \delta_1 = F(p, v, \mu, ...) \) is strong function \( p \); when moving upward,
\( \delta_2 = F(E,k_\varepsilon,\mu, v\ldots) \) is weak function \( p \) (where \( E \) is the modulus of elasticity of the sealing material; \( k_\varepsilon \) is the coefficient determining contact pressure \( p_0 \)). The fluid lubrication mode (\( V \)) corresponds to \( \bar{h} \geq 0.8 \ldots 1 \), the semi-fluid lubrication (\( IV \)) mode corresponds to \( 0.5 < h < 0.8 \) (see Figure 1). The influence of the friction mode parameters on the specific volume of leaks is identified when writing leakage equations (1) in the following form:

\[
\bar{V} = \frac{\mu d}{E k_\varepsilon} \left( \Psi_1 A \sqrt{e a p} + \frac{\Psi_2 B}{\sqrt{E k_\varepsilon + c p^n}} \right),
\]

where \( \mu \) is dynamic fluid viscosity, MPa \( \cdot \) s; \( v \) is speed of seal movement, mm\(^2\)/s; \( d \) determines the size of the seal, mm; \( E \) is the modulus of elasticity of the seal material, MPa; \( a \) is piezoelectric viscosity; \( p \) is pressure of working medium, MPa; \( A, B, c, n, k_\varepsilon \) are coefficients.

Figures 2, 3, 4, and 5 show graphical dependences of film thickness \( \Delta h \), leakage volume \( \bar{V} \), and friction force \( P_f \) on parameters \( p \) and \( v \) [6].

**Figure 2.** Dependence of the leakage volume on the outlet pressure of the pulse device: 1 – at \( v = 0.05 \) m/s; 2 – at \( v = 0.1 \) m/s; 3 – at \( v = 0.2 \) m/s.

**Figure 3.** Dependence of the film thickness on packer speed: 1 - at \( p = 10 \) MPa; 2 - at \( p = 20 \) MPa; 3 - at \( p = 30 \) MPa; 4 - at \( p = 40 \) MPa.
Figure 4. Dependence of the film thickness on the output pressure of the pulse device: 1 - at \( v = 0.01 \) m/s; 2 - at \( v = 1 \) m/s.

Figure 5. Dependence of the friction force on the packer movement speed: 1 - at \( p = 0 \) MPa (when the packer moves up and down); 2 - at \( p = 24 \) MPa (when the packer moves up); 3 - at \( p = 24 \) MPa (when the packer moves down).

In calculating leaks, it is advisable to proceed from practical values as a result of the use of packers for pulse injection established for a certain sealing size and parameters \( \mu, v, p \). Leakage in the packer seal can determine for \( \Psi \approx 0.3...1 \) using a simplified formula obtained from expression (3) by replacing the term in brackets by \( \sqrt{p} \) (analysis of the curves in Figure 2 and Figure 4 shows that \( \bar{V}(p) \) can be approximated by dependence \( \bar{V} \sim \sqrt{p} \)):

\[
V = V_0 \sqrt{\frac{\mu p d}{\mu_0 v_0 p_0 d_0}},
\]  
(4)

where \( V_0, \mu_0, p_0, d_0 \) are practical values.

This equation compares the results of individual tests with average data. For packer seals, basic conditions are as follows: \( p = 10.0...11.0 \) MPa; \( \mu = 10.0...10.5 \) MPa \( \cdot \) s; \( v = 0.1...0.15 \) m/s.
Friction of packer seals depends on the lubrication and its parameters, the nature of physicomechanical or physicochemical interaction of the contacting surfaces. When parameters \( p, v, \mu, \delta \) vary to certain limits, transition from one friction mechanism to another occurs. The following equation was obtained for friction under boundary and semi-fluid lubrication (especially typical for packers operating under high fluid injection pressures). The equation determines the friction coefficient (for engineering works) \([7]\):

\[
F_{lube} = A p^{2/3} v^{0.4} \mu^{1/6},
\]

where \( A \) is the coefficient depending on the material of the production string and composition of the surface-active components of the injected fluid, surface roughness, sealing material.

This equation makes it possible to predict the dependence of the friction coefficient on pressure, sliding speed (frequency and amplitude of liquid impulses), sealing materials and casing string, polarity and density of the spatial rubber mesh.

Taking into account expression \((4)\), the average coefficient of friction \( f \) with the number of cuffs in the packer seal 3 ... 6 has the following values at pressure \( p \), MPa (f): 1 (0.2), 2 (0.125), 5 (0.07), 10 (0.05), 15 (0.03). By combining design parameters, \( f_{max} \) to 1.5 \( f \) can be obtained.

As for seal wear, it should be noted that the requirement for the greatest sealing time is in conflict with the requirement for high tightness. The life of leak-tight seals is usually lower due to more intensive wear and friction \([8]\).

To compare operational benefits of various seals, criteria for evaluation of a set of quality indicators are needed. The most important criteria are tightness \((\bar{V})\), operating time \((W=2Sn)\), and operating temperature range \((\Delta t = t_1-t_2)\). By analogy to pairs of friction of movable joints, the following criteria can be applied to the packer seals:

- nominal specific capacity of the sealing element \( W/m^2 \)
  \[
  \bar{N} = pv,
  \]
  (6)

- nominal specific work of the seal to the maximum wear, \( J/m^2 \)
  \[
  \bar{A} = pL,
  \]
  (7)

- specific power losses, \( W/m^2 \) and work, \( J/m^2 \)
  \[
  N_f = f\bar{N}, \quad A_f = f\bar{A}.
  \]
  (8)

3. Experiment

As a result of the comparison of practical values and calculations, it was found that the average statistical values are within 0.001 ... 0.5 \( cm^2/m^2 \) with a predominance of probable leaks of 0.01 \( cm^2/m^2 \). The usual packers operating time during pulsed injection of liquids before severe leakage in the seals was is 300 ... 500 km (the average value under normal conditions is 700 ... 900 km \([9]\)). It is necessary to take into account that with an increase in the frequency and amplitude of the pulses in the injected fluid, efficiency and durability of the packer seal decreases. During the long-term pulsed injection of fluid into the well, it is necessary to use damping equipment with impulse devices. Their use will significantly reduce the amplitude of packer oscillation (stroke length of the sealing element \( S \)) and maximum accumulation time \( W_{max} \).

The increase in leakage from the initial value \( \bar{V}_0 \) to the maximum \( \bar{V}_{max} \) as the relative operating time \( m \) for a quality manufactured seal is subject to a power dependence:

\[
\bar{V} = \bar{V}_0 + \Delta \bar{V}_m^n,
\]

(9)

Based on expression \((9)\), dependences of specific leaks through packer seals for a different amount of cuffs in the seal were obtained. They are presented in Figure 6 (values are given in nominal units) \([10]\).

Two areas (two seal operation stages) can be identified on the curves of Figure 6:

I - normal low-intensity wear and a slight increase in leakage \( \bar{V}_1 \approx \bar{V}_0 \);

II – an intensive increase in leakages up to \( \bar{V}_{max} \), at which the seal is to be replaced (the values of \( \bar{V}_{max} \) for packer seals are regulated by the standards for sealing elements).

Under adverse operating conditions (violation of the lubrication modes) in expression \((9)\), \( n \approx 3 \) and \( m \approx 0.5 \); under favorable conditions (liquid lubricant) \( n \geq 6 \) and \( m \approx 0.7...0.8 \).
Figure 6. Dependence of the change in leakage during seal operations on the relative operating time: 1 - the number of cuffs in the packer seal $n = 3$; 2 - the number of cuffs in the packer seal $n = 6$.

4. Conclusion
   1. The operation time of packer seals must be predicted based on the results of accelerated tests at increased temperatures. The results should be extrapolated to the working conditions using equations of chemical reactions and diffusion.
   2. The main parameter determining seal leakage is contact pressure which rapidly decreases due to a reversible physical process of stress relaxation in the seal material (at normal temperature for several tens of hours), and then slowly decreases due to aging of the material.
   3. When determining the life time of the packer, it is necessary to take into account changes in contact pressure when the operating temperature changes and acceleration of the aging process under dynamic conditions due to vibration of the seats (reciprocating movement of the sealed packer during prolonged pulse injection of fluid into the well). Combination of the packer movement and pressure pulsation in the seal intensify mechanochemical aging processes.
   4. To increase durability of packer seals during long-term pulse injection of fluid into the well, it is necessary to use damping equipment along with pulse devices. Their use will significantly reduce the amplitude of the packer oscillation (stroke length of the sealing element $S$) and maximum accumulation time $W_{\text{max}}$.

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