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Research Article

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Design and Analysis of the Packer Seals

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ABSTRACT

Packers are considered to be one of the main equipment included in the class of oilfield equipment. They are mainly used to disconnect layers from each other and to isolate the production pipeline from the environment during preventive maintenance work in wells.

Rubber is used as the main material to ensure compaction in packers. Since these equipment operate in complex operating conditions, their compaction criteria should be studied depending on the parameters of the operating conditions. Since compaction is obtained at the expense of rubber, the physical and mechanical characteristics of rubber should be taken into account when calculating the compactor.

In the article were analyzed construction and main design parameters of the hydraulic compression packer seals. The calculation methodology has been given and worked out graphical relationships between some physical and mechanical parameters.

Keywords: Oilfield Equipments, Packer, Sealing, Pressure, Deformation

Introduction

Packers are considered one of the main equipment used in oilfield equipment. They are mainly used to disconnect layers from each other, to isolate the production pipeline from the environment during preventive maintenance work in wells. Packers are widely used in wells when performing technological operations such as hydraulic fracturing of layers, thermal treatment with acid, insulation work, and hydrosand jet perforation [1,2]. The material selection of packer elements, friction characteristics, friction and wear calculation methodology, and methods for increasing wear resistance are given in [3]. In contains detailed information about the construction, purpose and application areas of sealing nodes, and its operational characteristics [4]. Detailed information on the operational characteristics, repair and restoration issues of oilfield equipment, including packers, is also provided in [5].

Equipment used in the oil and gas fields, including packers, should be periodically inspected and, if necessary, carried out urgent technical repairs and restoration work [6].

The design of packer seals is crucial for ensuring the integrity and reliability of downhole tools in the oil and gas industry. In modern times, automated design and interactive calculation of machine and mechanism assemblies and details through computer programs allows for more accurate and important results to be obtained in a very short time. Although discusses topics on the automatic design of many parts of oilfield equipment, unfortunately, packer assemblies are left out of these topics [7].

Although provide extensive information about the construction, technical parameters and characteristics, installation, maintenance, and repair and restoration technologies of oilfield and drilling equipment, the methodology for calculating packers is not disclosed [8,9].

Rubber is used as the main material to ensure compaction in packers [4]. In the last few decades, rubber seal elements have been extensively used in the oil and gas industry. Since these equipment operate in complex operating conditions, their compaction criteria should be studied depending on the parameters of the operating conditions. Since compaction is taken at the expense of rubber, the physical and mechanical

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characteristics of rubber should be taken into account when calculating the seals.

Zheng and Li carried out sample tension test of hydrogenated nitrile butadiene rubber (HNBR) at different temperature (20◦C and 120◦C) and the stress relaxation tests at 120◦C [10]. The constitutive model of rubber was fitted by the test data. At the setting pressure of 14 MPa, the influence of temperature and stress relaxation on the packer rubber seal system was investigated by finite element analysis (FEA). The results indicated that temperature and stress relaxation had an appreciable impact on the sealing performance of packer rubber seal system. The average contact stress at 120 ◦C between the rubber and the borehole wall, compared with the calculation results at 20 ◦C, decreased by more than 15% and the maximum contact stress decreased by 20%. .

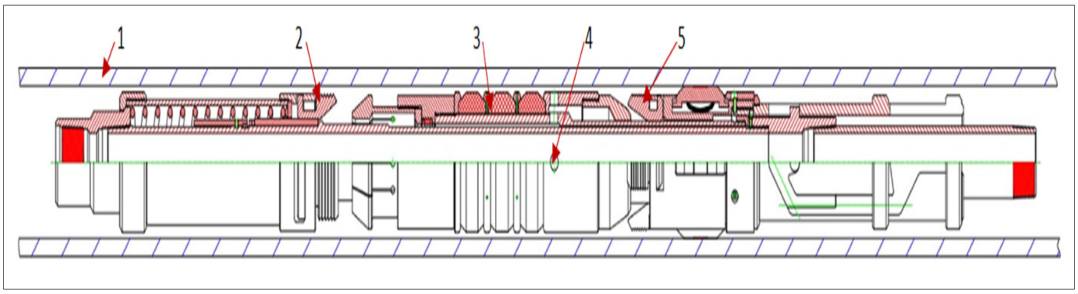
Material and Model

Center pipe of the packer made from the steel 35CrMo and the materials of casing and calking rings are the steel 45. For this mark steel materials, the elastic moduli can be take 2.1 × 105 MPa, and their Poisson's ratios are 0.3. The packer rubber material may be select as HNBR (Hydrogenated nitrile rubber) and its hardness varies approximately between from 80 to 90 H [11,12]. Finite element analysis has been shown that maximum contact pressure (28 MPa) are the areas when rubber barrel contacted with calcking rings [13].

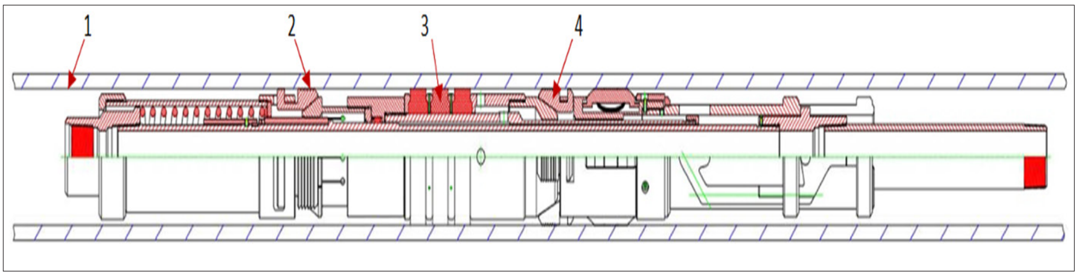
Packer seals basically consist of three rubber barrels. During compression, first the pressure affects the first rubber barrel, in the first 3 seconds it is compressed, inflated, and in the 5th second it touches the wall of the casing. At 7 seconds, the second rubber barrel, and at 9 seconds, the third rubber barrel creates sealing by fully touching the inner wall of the casing [14].

In Figure 1 has been shown the assemble drawing of the racker unit in two positions: a) before compressionb) after compression. All details of the packer placed in the casing, which made from steel. Main elements of the packer are the central tube, locking sleeves, rubber sleeve, slipping rings, support rings and etc. Figure 2.

The mechanical properties of the rubber, such as tensile strength, elongation, and hardness, play a critical role in seal performance.



a)



b)

Figure 1: Working diagram of a compression packer
a) before compression: 1– casing, 2 – upper slip, 3 – rubber cylinder assembly, 4 – bypass hole, 5 – bottom slip
b) after compression: 1 – casing, 2 – upper slip fixing, 3 – extruded rubber cylinder, 4 bottom slip fixing

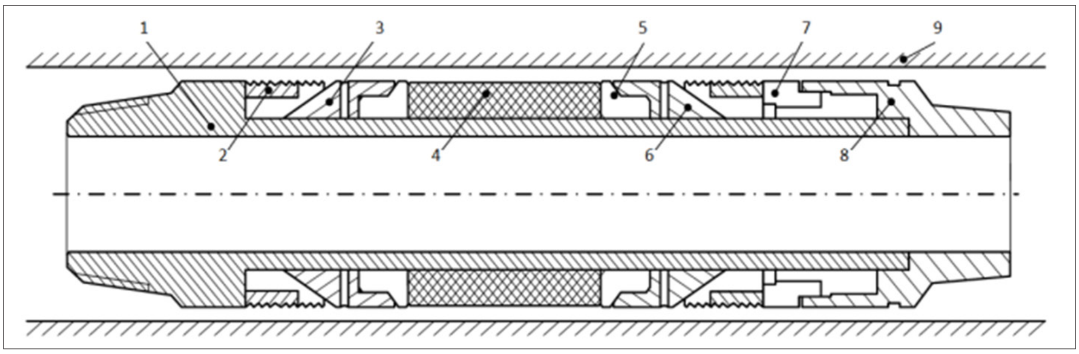


Figure 2: The Compression Packer Structure: 1 – central tube2 – locking sleeve3 – up centrum4 – rubber sleeve5 – support ring6 – down centrum7 – lock tube8 – connection9 – casing tube

Before compression there is an annular gap between the internal wall of the casing and external surface of the packer. When applied compression force rubber inflates, which causes its external diameter to increase and its length to decrease. At the result gap decreases and at the end reaches to zero.

The seal must withstand high pressures, temperatures, and chemical exposure typical of downhole environments.

Design Considerations

Key factors influencing the design of packer seals are the material properties (elastic modulus shear modulus, hardness, strain), geometrical dimensions (height, inner and outer diameter of the rubber) and environmental conditions. The seal design must accommodate the specific operational conditions, such as differential pressure, axial load, and thermal cycling.

The deformation and stress that provide self-sealing of the seals are determined as the following sequence

The strain is determined by the following expression in the compression deformation:

$$\varepsilon = \frac{h_0 - h}{h_0} = \frac{\Delta h}{h_0} \quad (1)$$

where h_0 and h are the initial and final heights of the seal, respectively.

The deformation of the seal according to its current and initial height can be characterized as follows:

$$\lambda_s = \frac{h}{h_0} = 1 - \varepsilon \quad (2)$$

Since the cross-section of a rubber seal changes due to the compressive force acting on it, its stressed state is characterized by conditional and actual stresses:

$$\sigma_0 = \frac{P}{S_0} \sigma = \frac{P}{S} \quad (3)$$

where σ_0 is a conditional stress, σ is an actual stress, P is actting force, S and $S_0$are the corss-sectional area of the seal before and after deformation, respectively.

The relationship between conditional and actual stresses is determined by the following expression:

$$\sigma = \frac{P}{S} = \frac{P}{S_0} = \sigma \quad (4)$$

We can write the expression for the compressive stress dependent on relative deformation as follows.

$$\sigma_C = \sigma(1 - \varepsilon) \quad (5)$$

The effective pressure difference between the top and bottom of the rubber cylinder can be maintained by increasing the height of rubber cylinder and the effective sealing length of the rubber cylinder and the well bore [15].

For engineering calculations, the elastic modulus of a rubber seals, which depends on its modulus of elasticity in compression and its bulk modulus, can be determined as follows:

$$E = \frac{9E_V G}{3E_V + G^{,}} \quad (6)$$

where Ev is a volume compression modulus and G is a shear modulus. If Ev > G and Poission's ratio of an incompressible rubber materials is μ=0.5 then we can take as E=3G.

Poisson's ratio can be determined for rubber seals as follows.

$$\mu = \frac{\Delta a h_0}{a_0 \Delta h} \quad (7)$$

Where h_0 and a_0 are the length and width of the seals, respectively; Δa and Δh are the transverse and longitudinal deformation of the seals.

From expression (7) it can be seen that Poisson's ratio does not remain constant, as the deformation increases (in compression), its value also increases. In practice, rubber seals are considered incompressible (injectable). However, the "vulcanizer" - reagents introduced into rubbers change its volume deformation:

$$\varepsilon_M = \varepsilon(1 - 2\mu) \quad (8)$$

where ε_M=ΔV/V the relative change in volume is the relative deformation coefficient of the seals. For vulcanized rubber sealants, Poisson's ratio (μ) varies in the range from 0.46 to 0.487.

There are following relationship between the elastic modulus and the shear modulus.

$$E = 2G(\mu + 1) \quad (9)$$

E and G practically do not depend on the type of rubber, but on the type and amount of vulcanizers, as well as the technological mode.

One of the important characteristics of rubber seals is their hardness (H).
Relative hardness of the rubber seals in compression is:

$$ES_0 = \frac{Ph}{\Delta h}; C = \frac{P}{\Delta h} \quad (10)$$

And relative hardness of the rubber seals in shear deformation is:

$$GS_0 = \frac{Pa}{\Delta h}; C = \frac{P}{\Delta h} \quad (11)$$

Relative hardness of the sealing ring in bending deformation is:

$$EJ = \frac{PR^3}{0.149\delta}\,; C = \frac{P}{\delta} = \frac{6.7EJ}{R^3} \tag{12}$$

where J is the section module, R is the radii of the seals.

Example: Derermine the compressive stress of a hydraulic packer seal according to folloüing data.
Given:

- Inner / outer diameter of rubber cylinder: $d_i = 80\,mm$, $d_o = 150\,mm$
- Initial height: $d_o = 150\,mm$
- Compressed height: $h = 75\,mm$
- Axial setting force: $Q = 20\,ton$
- Rubber hardness: $H = 85$ Shore A (typical for HNBR 80–90)
- Tensile strength: S_t=24 MPa
- Poisson's ratio (rubber, nearly incompressible): μ=0,49.

Solution:
Relative Strain (1)

$$\varepsilon = \frac{h_0 - h}{h_0} = \frac{90-75}{90} = 0.1617$$

Initial Cross-Sectional Area

$$S_0 = \frac{\pi}{4}\left(d_o^2 - d_i^2\right) = \frac{\pi}{4}\left(0.150^2 - 0.080^2\right) = 0.01265\,m^2$$

Axial Force

$$P = Q \cdot g = 20000 \cdot 9.81 = 196200\,N$$

Compressive Stress of the Seal Material

$$\sigma_c = (0.5\ldots0.8)S_t \approx 0.7S_t = 0.7*24 \approx 17\,MPa$$

Factors ma be take as Folloüs:

- safety factor for short-term peak load is SF= 1.2
- stress pelaxtion factor for static loading is SR=1.0
- temperature factor the operation temperature for (70 …100) °C is TF=1.0

Allowable Compressive Stress

$$\sigma_{c.all} = \frac{\sigma_c}{SF \cdot SR \cdot TR} = \frac{17}{1.2 \cdot 1.0 \cdot 1.0} \approx 14\,MPa$$

Nominal and actual compressive stresses, equations (3), (4)

$$\sigma_0 = \frac{P}{S_0} = \frac{196200}{0.01265} = 15.5\,MPa$$

$$\sigma = \frac{P\lambda}{S_0} = \sigma_0 \cdot \frac{h}{h_0} = 15.5 \cdot \frac{75}{90} = 12.9\,MPa$$

Because the rubber expands laterally under compression, the actual stress is slightly lower than the nominal one, consistent with the stress definitions in the methodology.

The elastic modulus can be determined from the hardness (H) of material. Typical correlation for HNBR Shore A 80–90: E≈7- 12 MPa
Here we take E≈9 MPa

$$G = \frac{E}{2(1+\mu)} = \frac{9}{2(1+0.49)} = 3.02\,MPa$$

Effective modulus onder constraint
Because the rubber is laterally confined, the apparent stiffness is much higher than the free uniaxial modulus:

$$E_{eff} = \frac{\sigma_0}{\varepsilon} = \frac{15.5}{0.1617} \approx 93\,MPa$$

Result of calculation
The calculation results show that under the applied load, the seal undergoes a relative deformation of 16%. In this case, the nominal compressive stress increases up to 15.5 MPa. However, taking into account the elastic deformation of the rubber, the actual stress is approximately 13 MPa, which is lower than the allowable stress value (14 MPa). The results of calculation show that due to 16% of strain effective modulus of the sealing element more than approximately 100% from its nominal value.

This means that the compressed seal will operate reliably under the given loading conditions.

This example demonstrates how **geometric dimensions, axial load, and rubber hardness** are combined to determine the compressive stress of a hydraulic packer seal. It also highlights the difference between nominal, true, and effective stress depending on lateral constraint.

Analysis and Programming
In Figure 3 has been shown the graphical relationship between the initial initial length of the packer rubber and applying compression force for three different hardness. From this diagram it is very easy determine that initial length of the rubber barrel of the packer should be decreased with increasing its hardness and compression force.

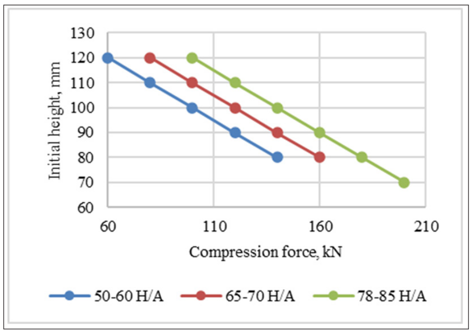


Figure 3: Initial height and compression force diagram

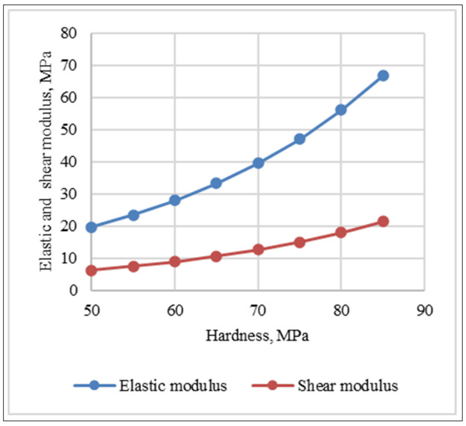


Figure 4: The relationship between the elastic and shear modulus and hardness of rubber material

Elastic modulus and shear modulus are the most important physical parameters for the design from subject of strength calculation. These parameters should be selected depending on types materials and its hardness. For rubber material this relationship has been shown in Figure 4 for different values of hardness. According to this diagram we can say that elastic and shear modulus increases with increasing the hardness.

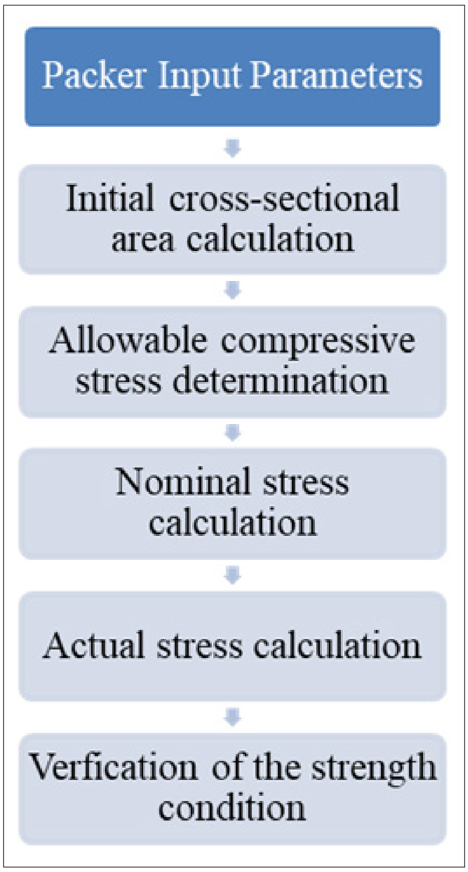


Figure 5: Block diagram for verfication of the strength condition of a packer seal

In Figure 6, the calculation algorithm of the hydraulic packer with a sealing element is presented. This calculation algorithm corresponds to the methodology described above. First, based on the value of the applied force, a standard packer is selected, after which its main overall dimensions, as well as its geometrical, physical, and mechanical characteristics, are determined. Using these parameters, the actual compressive stress is calculated. If its value does not exceed the allowable compressive stress, then the selected packer is considered suitable for the given operating and loading conditions and can be applied. Otherwise, it will be necessary to select a packer with higher performance parameters, or to change the material of the sealing element to one with higher mechanical characteristics, and then repeat the calculations.

In Figure 7, the Matlab program for calculating the hydraulic packer is presented. By using this program, each user can easily determine the actual stress of the hydraulic packer's sealing element and compare it with the allowable value to make the appropriate decision.

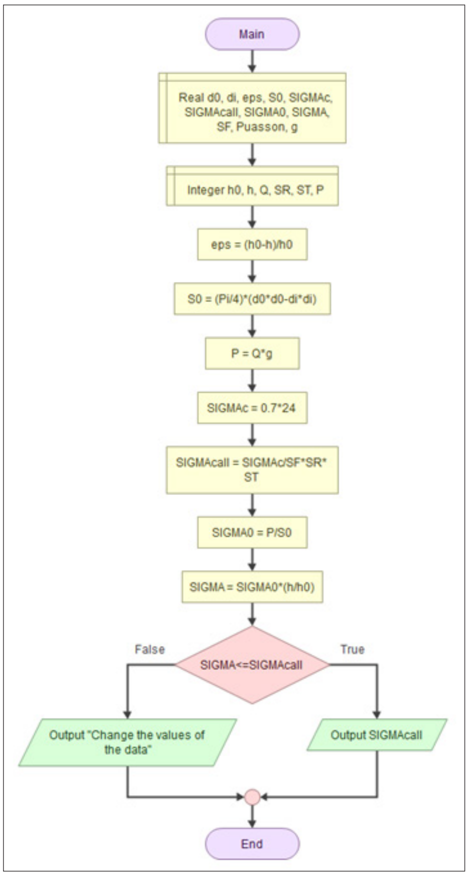


Figure 6: The algoritm of design of the hydraulic packer

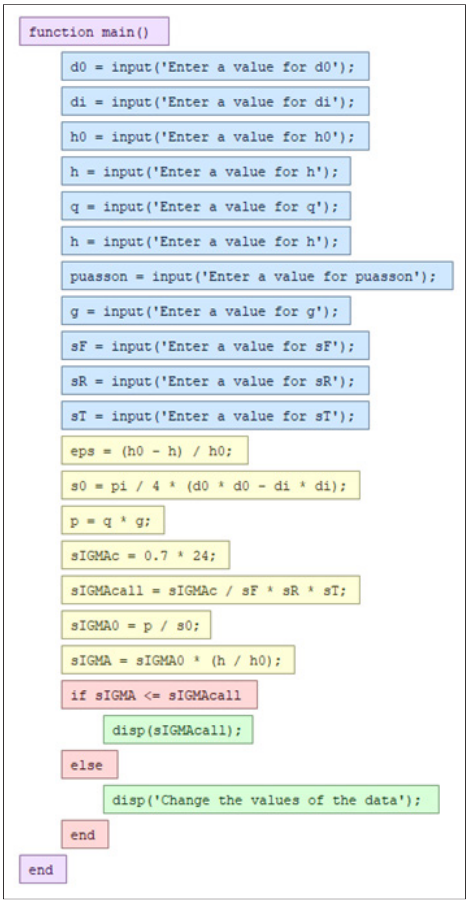


Figure 7: The Matlab program of the hydraulic packer's sealing element

Conclusion

The results of the computational modeling demonstrate the effectiveness of the proposed design methodology. The optimized seal design exhibits improved performance in terms of leakage resistance, mechanical integrity, and durability under various operational conditions. The proposed methodology provides a systematic framework for designing packer seals that can be adapted to different applications and operational requirements.
Considering the critical role of rubber sealing elements in maintaining zonal isolation and operational integrity, special attention was placed on the physical–mechanical behavior of the rubber material, deformation characteristics under axial compression, and stress distribution within the sealing system. The analytical relations, supported by numerical examples and graphical results, confirmed that rubber hardness, elastic modulus, initial geometry, and applied setting force are key parameters that govern the effectiveness and reliability of sealing performance.

The results indicate that a rubber strain level of approximately 16% ensures sufficient sealing contact stress while remaining within allowable stress limits, confirming the packer's stability under expected working conditions. Furthermore, the relationship between rubber hardness and elastic/shear modulus emphasizes the importance of proper material selection to match operational pressure and temperature environments. The developed algorithm and MATLAB-based calculation tool enable engineers to rapidly determine appropriate packer dimensions and evaluate stress states, ensuring reliable and efficient design decisions.

Overall, this work provides a practical foundation for optimizing packer seal design by combining theoretical calculations, material–property relationships, and computational support tools.

Future studies may focus on ploring advanced materials and further refining the optimization algorithms to enhance seal performance. Also attention could be paid to focus on advanced nonlinear finite-element simulations, long-term stress relaxation behavior, and the influence of extreme wellbore conditions to further improve the reliability and service life of hydraulic packer systems.

Nomenclatures

Symbols / Parameters

σ_0: The conditional stress;
σ : The actual stress;
P: The actting force;
S: The corss-sectional area of the seal after deformation;
S_0:The corss-sectional area of the seal before deformation;
E: The elastic module;
E_v: The compression modulus;
G: The shear modulus;
H: The hardness of a rubber material;
h_0: The initial height of the seal;
h: The final height of the seal;
Δh: The difference between the initial and final heights of the seal;
a_0: The width of the seal;
Δa: The transverse deformation of the seals;
Δh: The longitudinal deformation of the seals;
J: The section module;
R: The radii of the seals;
μ: The Poisson's ratio;
ε: The strain;
λ_s : The deformation of the seal.

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