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IMPROVEMENT OF THE OPERATIONAL EFFICIENCY OF DOUBLE SHOULDER TOOL JOINT OF DRILL COLLARS

Summary. The need to design new and improve existing drilling equipment is one of the important conditions for ensuring the efficiency of well construction. During drilling operations, the elements of the drill string, including their tool joints, are exposed to the highest load. Double-shoulder tool joints that have several advantages over standard (single-shoulder) ones, are becoming widespread in the world.

However, such joints require a secure contact and a specified load along three surfaces (the threaded surface, main and auxiliary end faces) simultaneously, which must be secured by a tight tolerance of the distance between the main and auxiliary end faces and the ends of the pin and the tool joint box. Failure to follow stringent requirements can lead to accelerated fatigue failure of the thread and, consequently, to drill string failures. Therefore, it is necessary to design a tool joint that will work effectively throughout the entire period of operation.

The analyzed existing options for solving the problem solve it to some extent, but have several disadvantages. To eliminate them, it is proposed to use elastic elements (ring springs) in the construction of double-shoulder tool joints.

The use of such elements makes it possible to create the necessary axial loads on the internal, and as a consequence, on the outer support joints and maintain their ratio throughout the operating lifetime without affecting the stress distribution in the danger areas of the connection, which was confirmed by simulation modeling.

In addition, the use of elastic elements reduces the impact of fabrication precision of the tool joint on its stress-strain state.

Keywords: drill string, double-shoulder tool joint, elastic elements, drill collars, tension, simulation modeling.

Introduction

The construction of directional and horizontal wells at oil and gas fields has led to the need to create and use qualitatively new sophisticated drilling equipment. First of all, it concerns the elements of the drill string assembly, which in the most difficult operating conditions must withstand high torsional, bending and tensile loads. Existing pipes made in accordance with API specifications cannot withstand such loads.

These loads on the drill string are multiplied several times under increased pressures and temperatures, thus requiring additional equipment strength and maximum reliability of the tool joints of drill pipes and drill collars.

Moreover, in view of the high cost of drilling, operating companies and drilling contractors place utmost importance on reducing operational and other risks in well construction and maximizing production costs. The latter is impossible without the use of highly reliable equipment along with optimization of well construction programs.

Well construction efficiency is mainly related to increased drilling rate and reduced non-productive time. This goal can be achieved by a competent approach to the drill string construction and assembly at the design stage, taking into account the following factors:

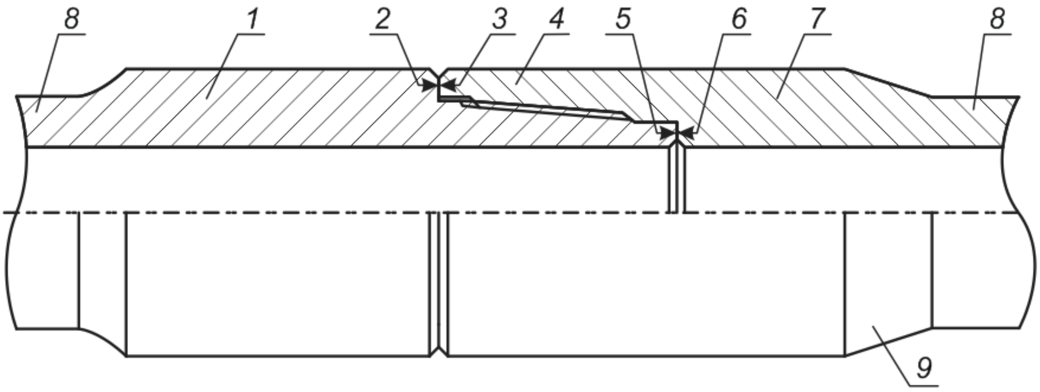
- the external and internal shape of the drill pipes and tool joints should not create much hydraulic resistance when drilling fluid moves;

– the mechanical characteristics of the pipe body, welds and threaded joints must be high enough to withstand the makeup torque, bending, tensile and torsional loads;

– other drill pipe design features (internal upset, turnkey distance, etc.) that affect the fatigue strength of the metal, the possibility of additional repair [1].

Analysis of previous studies and publications.

Joint (threaded connection) is a weak point in the standard design of drill pipes and drill collars. It is known that the efficiency of the joint (the ratio of the torsional load that the drill pipe can withstand to the torsional load that the tool joint withstands) is about 80 - 90% [2]. In order to increase this figure, as well as to increase the efficiency of work with simultaneous torsion, tension and bending (that is, under the most typical loads arising in case of drilling emergencies), double-shoulder tool joints are used in world practice today (Fig. 1) [3-5].



1 – pin; 2 – stop shoulder of the pin; 3 – stop face of the box; 4 – tool-joint thread; 5 – stop face of the pin; 6 – stop shoulder of the box; 7 – box;
8 – pin and box necks; 9 – tapered elevator shoulder
Figure 1 – Scheme of double-shoulder tool joint

Double-shoulder joint design contains the main external support (consists of **stop shoulder of the pin** 2 and **stop face of the box** 3), which serves as the surface of the joint tightening, and additional internal support (consists of **stop shoulder of the box** 6 and **stop face of the pin** 5), which is a mechanical limiter and a friction surface that provides additional resistance to the applied torsional and bending moments. The designs of such connections have threads as well as standard connections according to the American Petroleum Institute (API).

For the design of double-shoulder tool joints of drill collars, the following advantages can be distinguished compared to standard API tool joints:

- higher torque transmission (even 1.5 times) and higher bending resistance;
- preventing jamming of tool joints under maximum torsional moments;
- higher fatigue strength;
- in combination with high-strength pipes allow the construction of complex profile wells with a high intensity of deviation;
- allow to significantly increase the hydraulic characteristics (having an equal internal diameter, which causes a smoother flow of drilling fluid, reducing turbulence, eliminating the possibility of sticking of solid parts of the mud inside the tool joint);
- interchangeability with standard tool joints;
- greater tolerance for wear on the outer surface of the tool joint.

The design and study of double-shoulder tool joints of drill collars are considered in the publications [6-8].

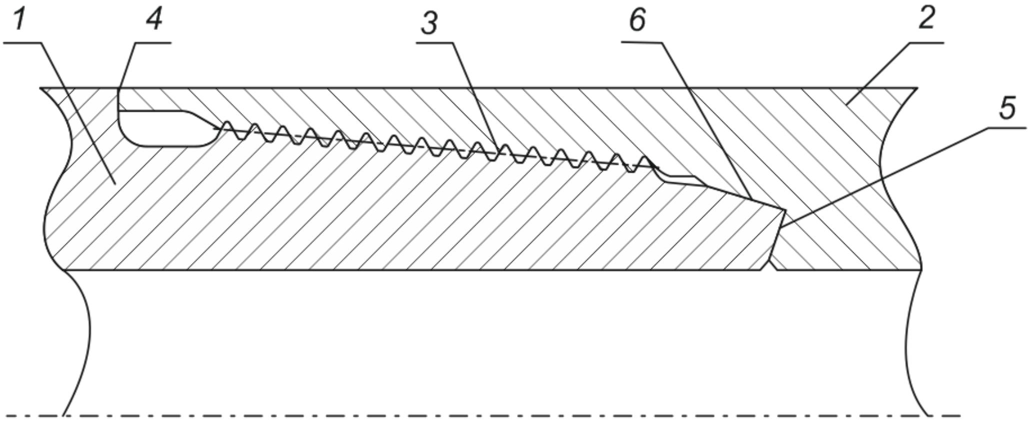
However, such joints require a secure contact and a specified load along three surfaces (the threaded surface, main and auxiliary end faces) simultaneously. It is difficult to do this, despite the tight tolerance of the distance between the main and auxiliary end faces and the ends of the pin and the tool joint box.

As a result, one of the supporting ends may be unloaded and the other end may be overloaded. Particularly dangerous is the case where, due to minor deviations from the specified tolerances, the support end remains unsupported or even open. As a result, it can accelerate the fatigue failure of the thread, and as a consequence, breakage and consequently damage of the drill string.

Laboratory fatigue studies with alternating cantilever bending of site-collected samples of CCK-59 drill pipes with such tool joints show that, depending on the accuracy of their manufacture, the cyclic durability of the tool joint may vary substantially [9].

Therefore, it is necessary to create such a tool joint in which the specified load on the additional support end is provided during manufacture and maintained throughout the entire period of its operation and is not conditioned by a tight tolerance of the distance between the support ends and the shoulders of the connecting parts, as well as their wear.

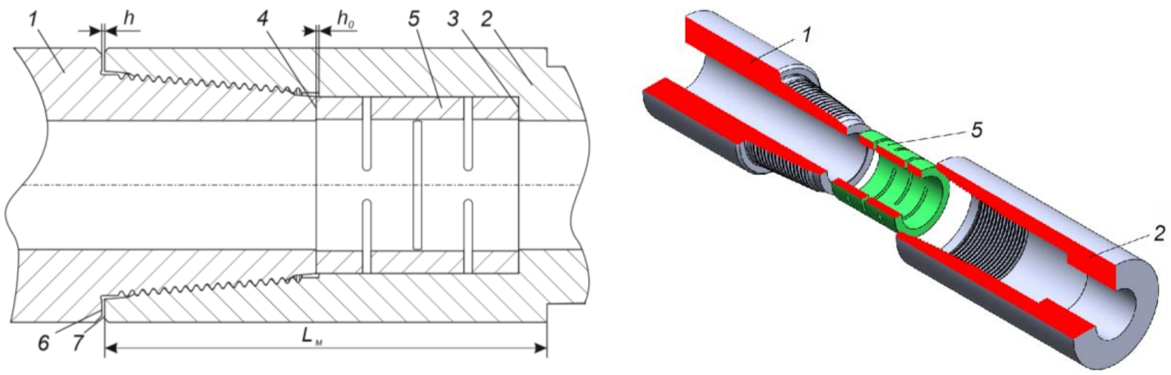
To solve this problem we can use tool joint of pipes [10] (Fig. 2) with additional support of the pin and box end, the feature of which is its execution from two contact planes 5 and 6 at an angle.



1 – pin; 2 – box; 3 –tool joint; 4 – main support; 5 – the first plane of additional support;
6 – the second plane of additional support
Figure 2 – Tool joint for steel pipes with additional pin and box end support

In addition to the fact that, according to [10], this tool joint is more leak-proof than single-shoulder tool joint and, depending on the angles forming the first and second planes of the additional support, we can compensate the tool joint manufacturing errors to some extent by the elastic deformations. But there is a drawback – a significant concentration of stresses in the area of the additional support end, which, in turn, will reduce its service life.

Also, to solve this problem, [9] describes the design of the tool joint of the tool joint (Fig. 3), consisting of a pin 1, box 2 and a procarved spring 5, which is installed between the stop shoulder of the box 3 and the stop face of the pin 4. In this case, a gap *h* is formed between the stop shoulder of the pin 6 and the stop face of the box 7, which is eliminated in the process of tightening the connection and compressing the spring. The gap value *h* is equal to the spring travel h_0 and according to its parameters, strictly corresponds to the specified axial load on the inner support joint. After tightening the joint, this also provides a strictly specified axial load on the outer support joint, as the difference between the total tightening force of the tool joint and the force created by the spring on the inner support joint.



1 – pin, 2 – box, 3 – stop shoulder of the box, 4 – stop face of the pin,
5 – procarved spring; 6 – stop shoulder of the pin, 7 – stop face of the box
а – scheme; б – three-dimensional model
Figure 3 – Threaded connection of the tool joint with a procarved spring inside

The provarved spring installed in the tool joint of the drill pipe between the stop shoulder of the box and the stop face of the pin, has parameters that provide its compression by the amount of travel h = h_0, given the axial force on the inner support joint.

In the process of multiple assembling-disassembling of joints during the round trip, the specified distance between the stop faces is disturbed due to the wear of their surfaces, which leads to significant changes in the given axial force ratios on both support joints and the reduction of threaded joint functionality.

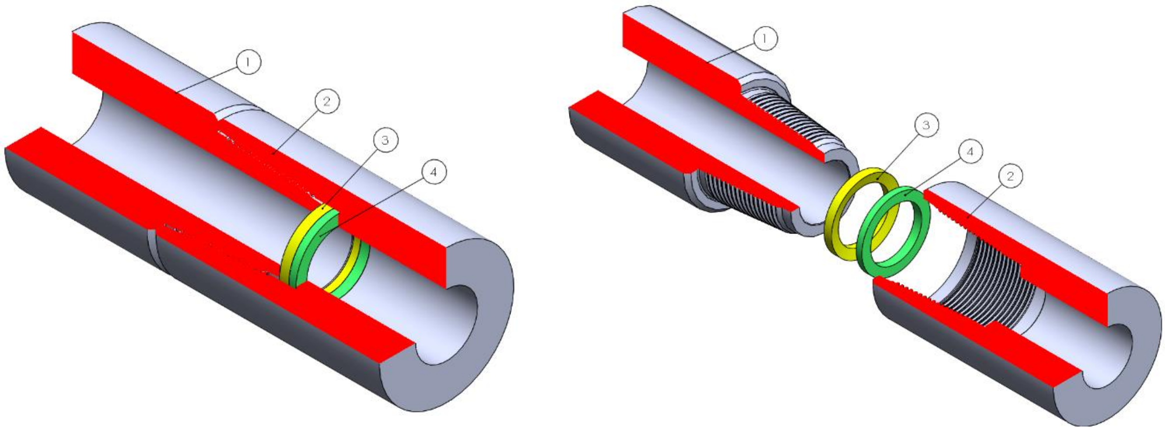
In the tool joint of the drill pipe when applying a procarved spring, the deformation of which is much higher than that of the material of the connection details, the requirements for the accuracy of the distance between the support joints are sharply reduced, and the wear of the support surfaces does not significantly affect the specified axial force of connection tightening on the inner joint, since its value is also much smaller than the travel of the spring.

Therefore, according to [9], a procarved spring in the design of the tool joint of the drill pipes ensures the creation of strictly specified axial loads on the inner and outer support joints. It also provides for maintaining their ratio for the entire period of operation, since it is practically independent of the tolerance for the distance between them and the wear of their support surfaces.

However, in addition to the advantages of using a procarved spring in the design of the tool joint, there are also disadvantages:

- large overall dimensions;
- a significant concentration of stresses in the elements of the spring;
- getting of the drilled rock into the spring openings.

To obtain the same effect, as with the application of a procarved spring and eliminate its disadvantages listed above, it is proposed to use in the design of double-shoulder tool joints of the drill collars elastic elements (ring springs), Fig. 4.



а) *б)*

1 – pin; 2 – box; 3, 4 – elastic elements

Figure 4 – Three-dimensional model of double-shoulder tool joint with installed elastic element

The purpose of the work and justification of the need for its implementation

The purpose of the work is to compensate for the manufacturing errors and the impact of the wear of the structural elements of the double-shoulder tool joints of the drill collars through the use of ring springs, whose parameters are determined by simulation modeling, ensuring strictly specified axial loads on the inner and outer support joints and maintaining their interellation for the entire operation period.

Tasks of the work:

1. To analyze ways to compensate for the manufacturing errors and the impact of wear on the structural elements of the double-shoulder tool joints of the drill collars.

2. To develop a three-dimensional model of structures of the double-shoulder tool joints of the drill collars with an ring spring installed in it and to investigate its stress-strain state using the finite element method.

3. To provide recommendations for geometric parameters of elastic elements (ring springs).

Presentation of basic material of the research

From sources [6-8] it is known that the stress distribution across the roots of a double-shoulder tool joint occurs more uniformly than in a single-shoulder one. Figure 5 summarizes the graphical dependences of the distribution of equivalent stresses on the pin roots of the single-shoulder (curve 1) and double-shoulder (curve 2) toll joints of the drill collars.

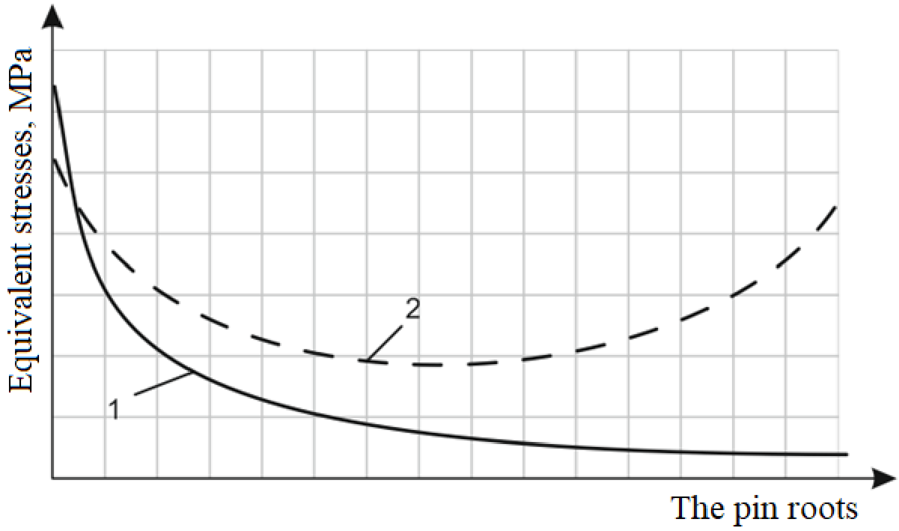


Figure 5 – Generalized graphical dependences of the distribution of equivalent stresses on the pin roots of the single-shoulder (curve 1) and double-shoulder (curve 2) toll joints of the drill collars

In Fig. 6, the distribution of equivalent stresses in the pin roots of double-shoulder toll joint SIFDS50.

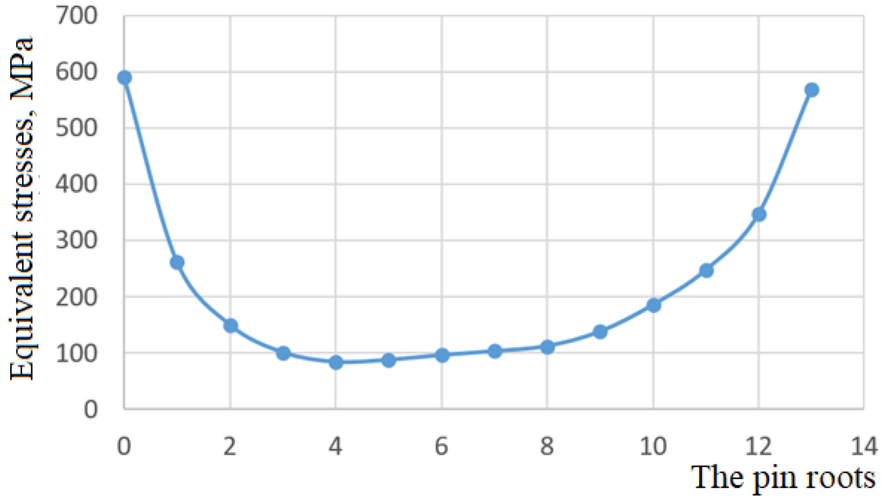


Figure 6 – The distribution of equivalent stresses in the pin roots of double-shoulder tool joint SIFDS50

This distribution of stresses is observed at the values of effort at the joint faces, shown in Fig.7.

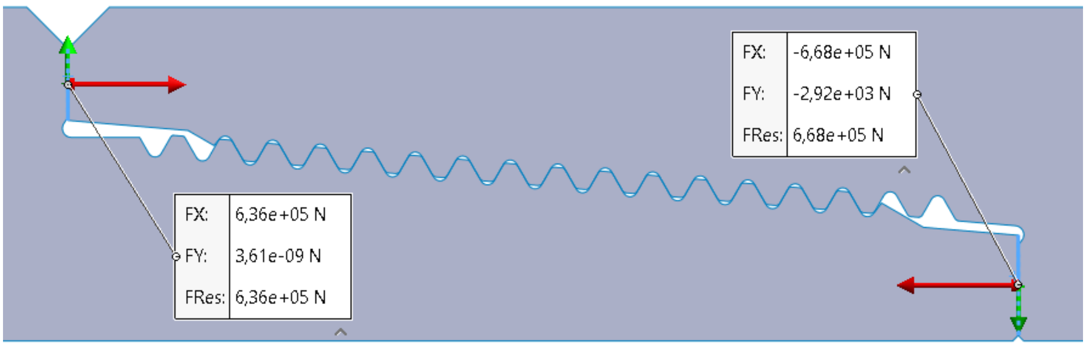
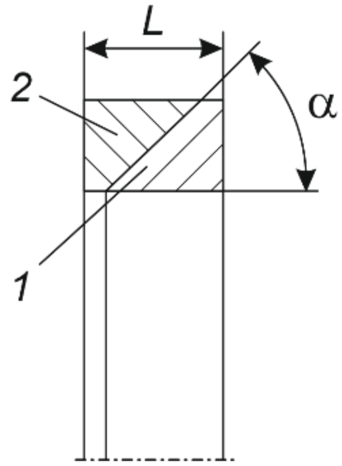


Figure 7 – The distribution of stresses at the faces of joint SIFDS50

Therefore, to investigate the tool joint with the elastic elements installed therein, it is first necessary to determine the dependences of the displacement on the load applied to them at different values of the angle α. The design scheme of the elastic elements is shown in Fig. 8.



1, 2 – elastic elements; α – angle; L – length
Figure 8 – Calculation scheme of elastic elements

It is clear that the load on the elastic elements is applied in the axial direction. The roller / slider [11] was used as the ring support during simulation, which allowed the ring to freely move radially and tangentially.

When applying the axial load, the elastic element 2 will be stretched and the element 1 will be compressed radially.

The length of elastic elements equal to L = 20 mm and different angles α = 45, 56, 68 and 800 were also selected for the study.

The length of the elastic elements is limited for technological reasons and the fact that at given angles α this value is maximum.

Spring steel of grade 60C2A with a yield strength of 1372 MPa was selected as the material for elastic elements.

The obtained results are shown in Fig. 9.

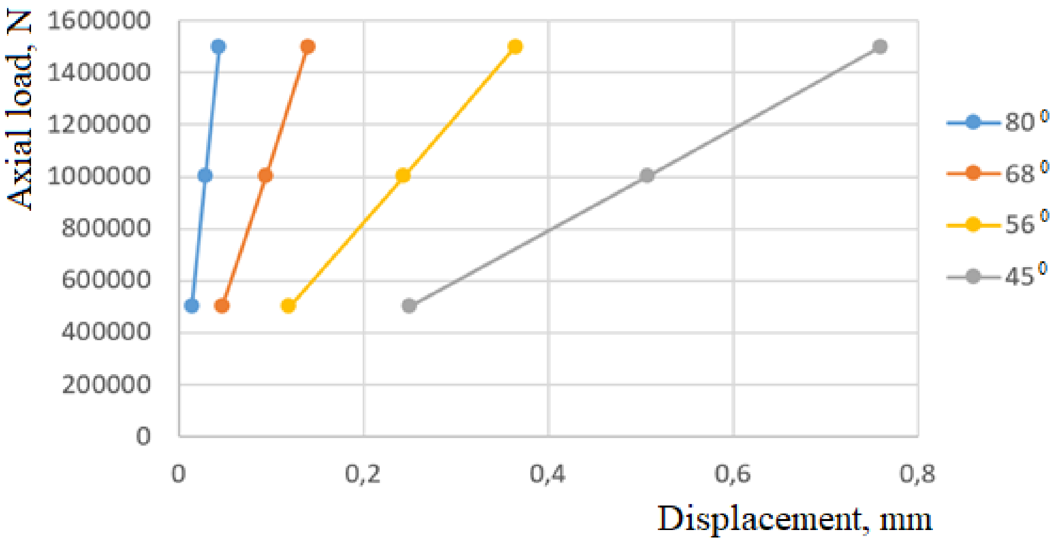
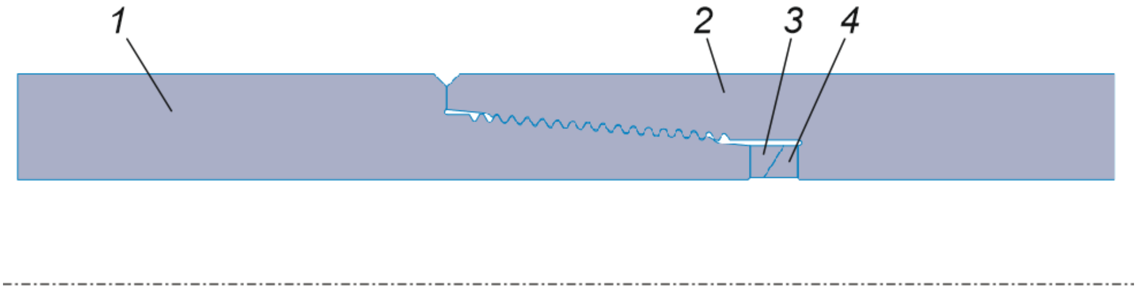


Figure 9 - Relationship between the displacement of elastic elements 3 and 4 and the magnitude of their axial load

Based on the relationship between the displacement of the elastic elements 3 and 4 and the magnitude of their axial load (Fig. 9), it is clear that the approach of the angle to 450 allows to support the load with slight deviations at a large range of displacements of the elastic elements. The use of the elastic properties of the elements reduces the impact of precision of the tool joint and advances to the accuracy of tenths of a millimeter, not hundredths of a millimeter as it was earlier.

However, for a more accurate study of such elastic elements, their simulation was performed as part of the design of a double-shoulder tool joint.

It should be noted that 2D simplification of the model, namely an axisymmetric type of study (Fig. 10), was applied to study the stress-strain state of a double-shoulder tool joint with its elastic elements (Fig. 10), since the design scheme provides for the application of loads acting only in axial direction.



1 – pin; 2 – box; 3, 4 – elastic elements
Figure 10 – 2D model of tool joint for the study

In the study the boundary conditions are accepted as in [6, 8], namely the face end of the pin is limited in displacements in any direction. A friction coefficient of 0.12 is established between the contacting surfaces.

The load from the action of the makeup torque is modeled by the overlapping of the ends of the pin and the box (use of the so-called "shrink fit") (Fig. 11) [11].

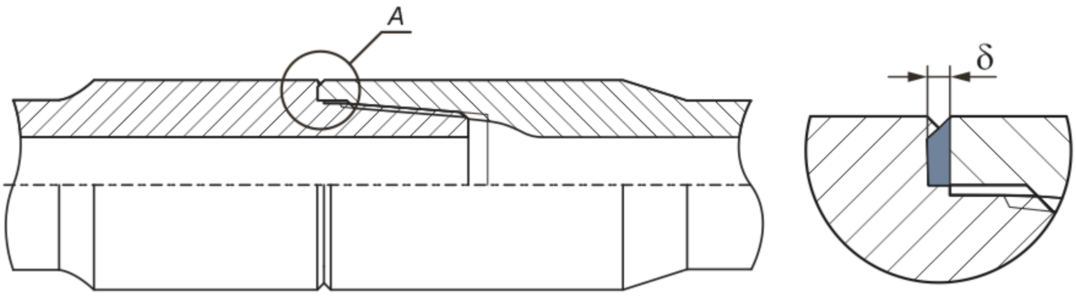


Figure 11 – Setting the makeup torque by overlapping the thrust faces of the pin and the box

The essence of the method of "shrink fit" is that previously at the stage of creating a three-dimensional model of the tool joint, the stop shoulder of the pin 2 and the stop face of the box 3 (the main support of the pin and the box) overlapped to the desired predetermined value. This value, in turn, can be determined by the known angle of rotation of the pin relative to the box and the makeup torque [12].

Similarly, the "shrink fit" method is used for a double-shoulder tool joint.

Using the same axisymmetric model of tool joint to determine its stress-strain state compared to the three-dimensional model has several advantages, including the ability to create a small finite element grid (higher accuracy of the obtained results), a higher speed of calculations on the computer (the ability to process more options for calculations).

To select the angle α, we conditionally assume that for any of its magnitudes, the stress distribution across the roots of the nipple turns remains constant, based on the guaranteed constant efforts at the ends of the joint (Fig. 7).

The choice of the angle α is also influenced by the material properties of the elastic elements. Therefore, stress with a magnitude of 914 MPa (to ensure a minimum factor of safety margin 914 = 1372 / 1.5, where 1.5 is the factor of safety) will serve an additional criterion in determining the magnitude of the angle of elastic elements.

The results of the study are given in table. 1.

Table 1

Results of the study

Angle, ^{0}	45	50	55	58	65
Contact force, кН	625	630	693	665	731
The maximum stresses that arise in elastic elements, МПа	1131	940	846	720	635

Therefore, based on the results (Table 1) obtained by the finite element method and the criteria described above, the angle of magnitude 55^0 is for the elastic elements installed in the structure of the double-shoulder tool joints SIFDS50.

Fig. 12 shows the displacements in the investigated model (at an angle of elastic elements 55^0), and in Fig. 13 – distribution of equivalent stresses in it.

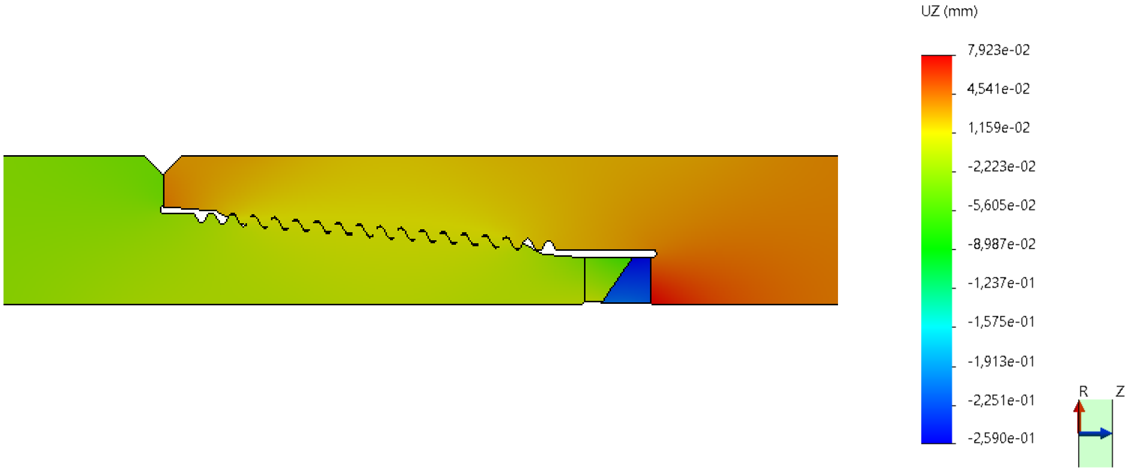


Figure 12 – Displacement in the axial direction

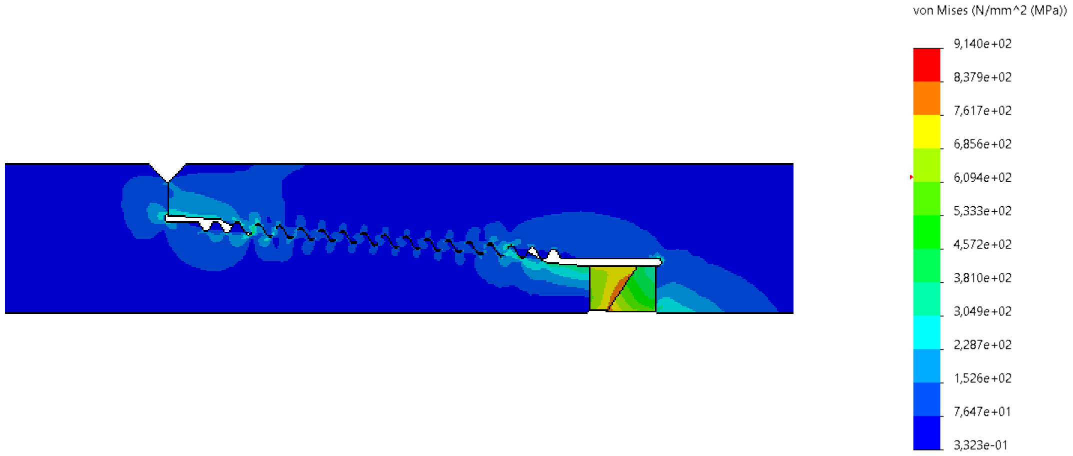


Figure 13 – Equivalent stress distribution

Therefore, the placement of the elastic elements into the tool joint makes it possible to create the necessary axial loads on the internal, and as a consequence, on the outer support joints and maintain their ratio throughout the period of operation. For this tool joint design, the impact of the stop faces is negligible, since their wear will be offset by the displacement of the elastic elements.

Conclusions

Although, the proposed design of double-shoulder tool joints of drill collars contains additional elements in its structure, it has several advantages in comparison with other similar in principle designs: smaller overall dimensions, lower stress concentration in additional elastic elements, no penetration of the drilled rock into the slits of the procarved spring.

The placement of the elastic elements into the tool joint makes it possible to create the necessary axial loads on the internal, and as a consequence, on the outer support joints and maintain their ratio throughout the period of operation. For this tool joint design, the impact of the stop faces is negligible, since their wear will be offset by the displacement of the elastic elements.

Based on the relationship between the displacement of the elastic elements 3 and 4 and the magnitude of their axial load (Fig. 9), it is clear that the approach of the angle to 450 allows to support the load with slight deviations at a large range of displacements of the elastic elements. The use of the elastic properties of the elements reduces the impact of precision of the tool joint and advances to the accuracy of tenths of a millimeter, not hundredths of a millimeter as it was earlier.

However, based on the limitations imposed in the design of elastic elements (technological, material strength) using simulation modeling for the SIFDS50 tool joint, the optimal angle was set to 55°.

It should also be noted that the use of elastic elements during the repair of double-shoulder tool joints of drill collars (their installation in the structure at first repair) would allow the extension of the drill collar operational lifetime with such tool joints.

It is clear that, from a practical point of view, the application of these elastic elements entails a number of additional requirements for the assembly and disassembly of tool joints of this type, but the advantages of using them are much greater than the inconvenience caused.

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ВОЛОКОННО-ОПТИЧЕСКИЕ ДАТЧИКИ

Волоконно-оптические датчики позволяют измерять многие характеристики лабораторных и промышленных объектов, в частности температуру. Не смотря на то, что их использование достаточно трудоемко, оно дает ряд преимуществ, использования подобных датчиков на практике: безиндукционность (т.е. неподверженность влиянию электромагнитной индукции); малые размеры датчиков, эластичность, механическая прочность, высокая коррозийная стойкость и т.д.

Для измерения температуры с помощью световодов, изготовленных из кварцевого стекла, особенно подходит так называемый эффект Рамана. Свет в стеклянном волокне рассеивается на микроскопически малых колебаниях плотности, размер которых меньше длины волны. В обратном рассеивании можно найти, наряду с эластичной долей рассеивания (излучаемое рассеивание) на одинаковой длине волны, как проникший свет, так и дополнительные компоненты на других длинах волны, которые связаны с колебанием молекул и, тем самым с локальной температурой (комбинационное Рамановское рассеяние).

Волоконно-оптические датчики (так же часто именующиеся оптические волоконные датчики) это оптоволоконные устройства для детектирования некоторых величин, обычно температуры или механического напряжения, но иногда так же смещения, вибраций, давления, ускорения, вращения (измеряется с помощью оптических гироскопов на основе эффекте Саньяка), и концентрации химических веществ. Общий принцип таких устройств в том, что свет от лазера (чаще всего одномодового волоконного лазера) или суперлюминесцентного оптического источника передается через оптическое волокно, испытывая слабое изменение своих параметров в волокне или в одной или нескольких брэгговских решетках, и затем достигает схемы детектирования, которая оценивает эти изменения.

В сравнении с другими типами датчиков, волокно-оптические датчики обладают следующими преимуществами:

Они состоят из электрически непроводящих материалов (не требуют электрических кабелей), что позволяет использовать их, например, в местах с высоким напряжением.

Их можно безопасно использовать во взрывоопасной среде, потому, что нет риска возникновения электрической искры, даже в случае поломки. Они не подвержены электромагнитным помехам (EMI), даже вблизи разряда молнии, и сами по себе не электризуют другие устройства. Их материалы могут быть химически инертны, то есть не загрязняют окружающую среду, и не подвержены коррозии. Они имеют очень широкий диапазон рабочих температур (гораздо больше, чем у электронных устройств). Они имеют возможность мультиплексирования; несколько датчиков в одиночной волоконной линии может быть интегрировано с одним оптическим источником.

Сенсоры на основе брэгговских решеток

Волоконно-оптические датчики зачастую основаны на волоконных брэгговских решетках. Основной принцип многих волоконно-оптических датчиков в том, что брэгговская длина волны (т.е. длина волны максимального отражения) в решетке зависит не только от периода брэгговской решетки, но также от температуры и механических напряжений. Для кварцевых волокон изменение брэгговской длины волны на единицу деформации примерно на 20% меньше, чем растяжение, так как