https://doi.org/10.33271/nvngu/2023-3/060

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DESIGN OF DOUBLE-SHOULDER THREADED JOINTS OF DRILLING STRING ELEMENTS

Purpose. Development of a method for simulating the screwing torque/moment and researching the design of double-shoulder threaded joints while preserving their geometric parameters. Determination of the optimal length of the additional shoulder pin part for the developed drill collar (DC) of NC50, NC55 type of various standard sizes. Establishing the dependence between the pin part length of the additional shoulder and other geometric parameters of the double-shoulder joint of the DC for further use when developing other standard sizes of similar design threads.

Methodology. The development of design and the method for simulating the screwing torque of double-shoulder threaded joints was carried out using the finite element method and parametric modeling.

Finding. A method for simulating the screwing torque of threaded joints was proposed and developed, which made it possible to improve the model of their automated design. The optimal lengths of the pin part of the additional shoulder of the developed design of double-shoulder threads of the NC50, NC55 types were determined. For planning threaded joints of similar design with the above-mentioned types of threads, a dimensionless coefficient was derived and its value was calculated. A nomogram was built to determine the length of the joint additional shoulder pin part due to the value of this coefficient.

Originality. The model of automated design for double-shoulder threaded joints of drill string parts has been improved. This model preserves their geometric parameters and enables modeling a wide range of standard sizes of similar joints.

Practical value. The optimal geometric parameters of the developed structures of double-shoulder threaded joints of the drill string elements were determined. A nomogram was built to determine the length of the additional shoulder pin part of the joint by the value of the dimensionless coefficient.

Keywords: *drill string, stress-strain state, threaded joint, pin, coupling*

Introduction. The development of the oil and gas industry is characterized by an increase in the scope and depth of drilling, which facilitates the growth of well construction costs [1]. Therefore, the task arises to pay back investments by ensuring high technical and economic indicators of drilling and production in the shortest possible time. For this, scientists and practitioners solve a number of important tasks to enable the trajectory of the well, high technical and economic indicators [2], reliability of deep drilling equipment and drill string elements in particular [3, 4]. Thus, in the work [5], the authors developed designs of a controlled deflector and an elastic coupling, which differ in a number of advantages, including the possibility of non-discrete changes in the skew angle and bending stiffness, respectively. A methodology for calculating the stress-strain state of the bottom of the drill string is proposed.

As a significant number of accidents occur with tool threaded joints, being an integral part of the drill string elements, the issue concerning the improvement in the drill string reliability by strengthening the threaded joints using vibration-centrifugal processing was considered by the authors [6]. Also, for a more accurate study on the durability of the threaded joints of the drill string elements, the authors investigated the stress-strain state of the threaded joints of the drill pipes by the method of finite element analysis. The distribution of normal stresses in the depression of the nipple thread from the applied tightening torque is shown. It was established that the numerical values of the maximum stresses increase from the last turns to the first and can be described by a polynomial of the third degree. A technique for determining the relative stress gradient in the zone of its concentration using finite element analysis has been developed. It was established that the numerical values of the relative maximum stress gradient, obtained by the method of finite element analysis and analytically, are close to each other. The proposed method for determining the relative maximum stress gradient for concentrators of arbitrary shape allows determining the fatigue failure similarity criterion and using it to determine the fatigue life of structural elements [7].

Another way to increase the reliability of threaded joints is to optimize their geometric parameters under the condition of full-strength and optimal tightening torque [8]. The optimal tightening torque makes it possible to evenly distribute the load on the turns of the threaded joint, to decrease the range of stresses in dangerous sections, especially under cyclic loads [9].

Tool threaded joints are important parts of a drill string. Due to the increase in the depth of the wells and their complex profiles, the existing designs of tool joints today do not fully meet all the requirements applied to them. As practice shows, the number of failures associated with such joints remain quite large [10]. This makes it necessary to find new ways to solve the problem of ensuring the reliability of the elements of tool threaded joints (TTJ).

Literature review. To improve the efficiency of work with simultaneous twisting, stretching, and bending, double-shoulder tool joints are used in the world practice today (Fig. 1). Thus, long-range, deep, and highly deviated wells require the use of a drill string that has a higher torsional yield strength of the drill pipe casing and greater tool coupling torque than standard American Petroleum Institute (API) couplings. The VAM Express two-pillar connection was developed to solve the drilling challenges associated with the development of such complex wells. This signature high-torque coupling incorporates four key design features to push the limits of drilling even further: quick rig feed, high torque, convenience and durability [11].

The design of the joint (Fig. 1) includes a ledge of pin *4* and the end of coupling *5* of the main shoulder, which serve as the sealing surface of the joint. The ledge of coupling *7* and the end of pin *6* of the additional shoulder used in the design ensure the tightness of thread *2* and provide additional resistance to the applied torque and bending moment. Designs of such joints are interchangeable with joints according to the API standard [12].

Fatigue study of a two-support threaded connection with a standard connection according to API was carried out using the Simulia Abaqus fe-safe software. Such joint has better tensile strength, torsional strength, flexural strength and compressive strength, which are increased by 10.65, 62.5, 2.75 and 52 %, respectively. Its fatigue life in compression, bending, and fatigue life in torsion increase by 1.19, 1.74 and 550 times, respectively [13].

Double-shoulder threaded joints have a number of advantages compared to single-shoulder ones:

- transmission of greater torque and greater stability under the action of bending moment;
- prevention of tool joints jamming at extreme moments of torsion;
- higher fatigue strength;
- in combination with high-strength pipes, they make it possible to construct wells with a complex profile and a high intensity of gaining curvature;
- interchangeability with standard tool joints;
- greater wear tolerance of the outer surface of the tool joint.

In work [14], a new thread structure of drill pipe joints "Double shoulder joint" (DSJ) is investigated. Since the mechanical behavior of DSJs was not well studied, the authors created a finite element model of DSJs. The model calculation results were verified using experimental results. The authors analyzed the stress and strain distribution with the results of calculations of the 3D model of the finite elements of the entire structure under torque, compression force, tensile load, and bending moment.

In world practice, a variety of software is used to develop new and improve existing equipment. At the same time, the method of finite elements is implemented, which makes it possible to minimize time and material costs, facilitate the performance of the necessary design works (developing technical documentation, conducting various studies on equipment elements, etc.). The most common software complexes are: ANSYS, CATIA, ProEngeener, SolidWorks, etc.

In works [15, 16], a three-dimensional finite-element model of a threaded connection and a two-support lock connection was created and the mechanical behavior of this connection under the action of screwing torque, axial tension, and bending moment was analyzed using the dynamic method of finite elements.

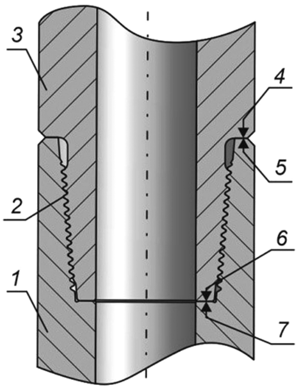

Fig. 1. Design scheme of a double-shoulder joint of drill string elements:

1 – coupling; 2 – thread; 3 – pin; 4 – ledge of the pin of the main shoulder; 5 – end of the coupling of the main shoulder; 6 – end of the pin of the additional shoulder; 7 – ledge of the coupling of additional shoulder

At the same axial tension, the ultimate torsional resistance with only the additional support is 12 % higher than when it has only one primary support, while the connection with a double-supported connection is 69 % higher than when it has only the main support [17].

There are several options for two-dimensional calculation: when modeling thin-walled structures, a plane stress state should be used. The cross-section of sufficiently long structures is usually modeled in a plane-strain state, and the bodies of rotation are modeled in an axisymmetric setting. All three modeling options use the same flat mesh of finite elements, but the stiffness matrices of the elements are different from each other [18].

Threaded joints with a small angle of the thread profile can be approximately considered axisymmetric. However, the application of boundary conditions (screwing moment) for a double-shoulder tool joint causes certain difficulties. Of course, you can apply known approaches to such modeling, but in this case, the geometric parameters of the connection elements are violated, which is unacceptable for determining the necessary unknown quantities. In the case of a double-shoulder connection, a drill collar is the length of the conical part of the pin.

Performing a static linear analysis requires the application of boundary conditions, which include fasteners and load.

Fig. 2 shows the calculation scheme of the tool joint of drill pipes.

However, based on the capabilities of the program for simulation modeling, setting all boundary conditions except for the screwing torque (in an axisymmetric setting) does not cause difficulties.

Three methods are possible for applying the screwing torque to the elements of the tool joint of drill pipes:

1) applying a tightening torque to the elements of the tool joint, which fully simulates real conditions. It is only possible when using three-dimensional models (not suitable for an axisymmetric problem);

2) introduction into the design of the tool joint of the section, the coefficient of the material expansion which is significantly greater than the main material (Fig. 3) [19];

3) the use of overlapping ends of the pin and the coupling (use of the so-called "shrinking fit" (most suitable for researching the connection in an axisymmetric setting) (Fig. 4) [19].

The essence of "shrinking fit" is that, in advance, at the stage of creating a three-dimensional model of the tool joint, the shoulder of pin *2* and the supporting end of coupling *3* (the main shoulder of the pin and the coupling) are overlapped by the necessary previously known value, which in turn can be determined from the known angle of the pin rotation relatively to the coupling and the screwing torque of the joint.

Unsolved aspects of problem. Based on the analysis of possible methods for simulating the screwing moment of a threaded joint, it follows that their use for two-shoulder threaded

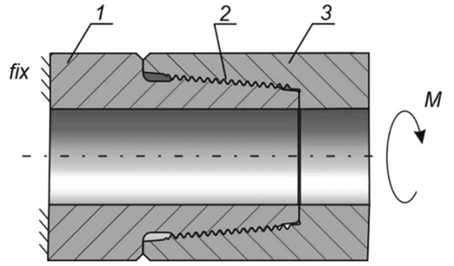

Fig. 2. Calculation scheme of the tool joint of drill pipes:

1 – coupling; 2 – threaded part; 3 – pin; fix – place of fixation; M – screwing torque

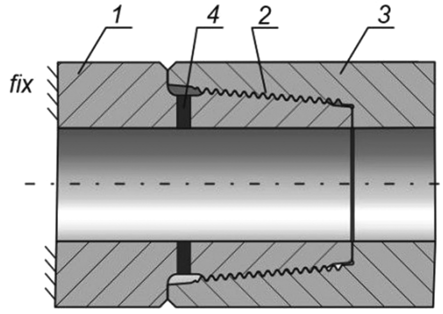

Fig. 3. Scheme of joint using a section of material with a different coefficient of thermal expansion:

1 – coupling; 2 – threaded part; 3 – pin; 4 – section of material; fix – place of fixation

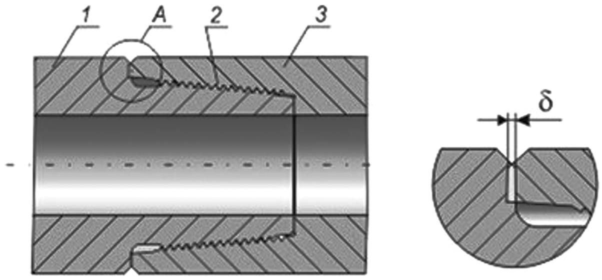

Fig. 4. Scheme of joint with overlapping shoulder ends

joint will not provide the necessary accuracy of the obtained results, since the above-described methods change the geometric parameters of the coupling.

The objective of the article:

- development of a method of simulating the screwing moment and researching the constructions of double-shoulder threaded joint while preserving their geometric parameters;
- determination of the optimal lengths of the pin part of the additional support for the developed the thread of the drill collar of NC50, NC55 types of various standard sizes;
- establishment of the dependence between the length of the pin part of the additional support and other geometric parameters of the double-shoulder joint of the drill collar for further use in the development of other standard sizes of threads of a similar design.

Presentation of the main material. To study the threaded joint, its three-dimensional parametric model was developed (Fig. 5).

Parameterization or parametric modeling is modeling or designing with the help of the use of created parameters of model elements and dependencies between these parameters. In fact, a mathematical model of objects is created with the parameters that change of wich leads to the change in the configuration of the display, the element model, the reciprocal displacement of elements in the assembly, etc. With the help of parameterization, it is possible to quickly and efficiently, using the method of changing some geometric ratios or parameter values, analyze various constructive solutions and choose the most optimal one from them.

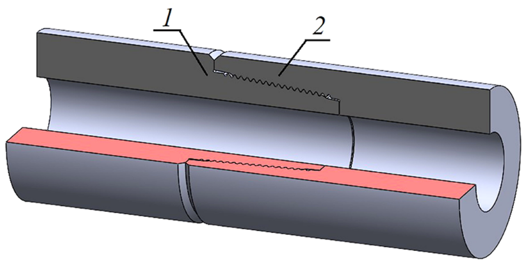

Fig. 5. Three-dimensional parametric model of a threaded joint:

1 – pin; 2 – coupling

During parametric modeling of threaded joints uses geometric parameterization, in which the geometry of each parametric object is recalculated depending on the position of parent objects, its parameters and variables. A parametric model in the case of geometric parameterization consists of construction and display elements. Construction elements specify parametric relationships. Display elements include display lines and design elements. Some construction elements can depend on others. Construction elements can also contain parameters. When one of the elements of the model is changed, all the elements dependent on it are rebuilt according to their own parameters and methods of setting them.

The process of geometric parameterization for the threaded joint is as follows:

- at the first stage, the models of joints were built in accordance with the technical specifications [20]. For convenience, the building of these parts was performed in one SolidWorks program assembly file;
- at the second stage, the sketches of the parts were connected with each other using conjugations, which made it possible to automatically change the geometric parameters of the coupling when changing the geometric parameters of the pin;
- at the third stage, a table of parameters was created, in which data on other standard sizes of the tool joints were entered.

At the next stage of work, in order to further simulate the torque of the tool joint, the control dimensions L_R and L_{PIN} are set (Fig. 6).

Considering the fact that the length L_{BOX} should remain constant, it is suggested to simulate the screwing moment using the control size L_R, namely, when the size is reduced L_R the entire part of the coupling thread will be shifted to the left and the ends of the pin and coupling overlap by the amount δ_{BOX}, which, accordingly, will determine the previous tension of ends.

It should be noted that the tool joint model will automatically change due to parameterization, but this will not affect the accuracy of the obtained results, since the position of the thread in the model will remain unchanged.

To develop an algorithm for conducting research, it is proposed to first determine the stress-strain state of the tool joint when only the main supporting ends are in contact, and then, changing the size L_{PIN} by size δ_{PIN} determine its required value. The criterion for determining the value of the size L_{PIN} was the value of the stresses on the turns of the tool joint [19].

Fig. 7 shows a scheme of a tool joint with thread type NC50 and geometric parameters that are variable. Parame-

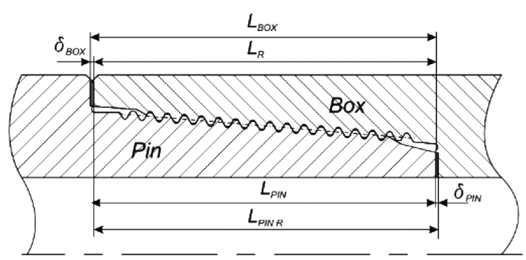

Fig. 6. Scheme of the double-shoulder tool joint with the indicated control dimensions:

δ_{BOX} – overlapping of the ends of the pin and the coupling of the main shoulder; L_{BOX} – coupling length; L_R, $L_{PIN\,R}$ – controls size; L_{PIN} – length of the pin; δ_{PIN} – overlapping of the ends of the pin and the coupling of the additional shoulder

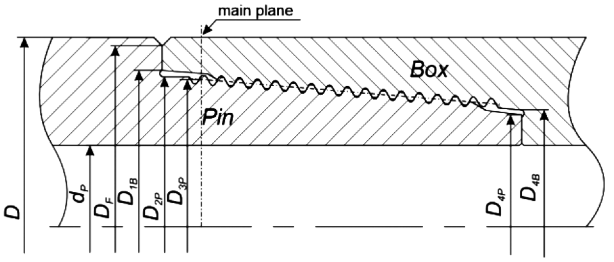

Fig. 7. Scheme of a tool joint with NC50 thread type:

D — outer diameter of the joint; d_p — inner diameter of the joint; D_F — outer diameter of the ledge of the pin of the main shoulder; D_{2p} — diameter of the cylindrical groove of the pin; D_{3p} — average thread diameter in the main plane; D_{4p} — external diameter of the coupling ledge of the additional shoulder; D_{1B} — inner diameter of the ledge of the pin of the main shoulder; D_{4B} — diameter of the cylindrical groove of the coupling

ters for different standard sizes of this connection are given in Table 1.

As a result of the simulated modeling of the DC double-shoulder tool joint, the optimal lengths of the pin part of the additional support for the developed thread NC50, NC55 were obtained (Table 2).

On the basis of the obtained results, a dimensionless coefficient K_F was proposed and a nomogram of the correspondence of its values to the length of the pin part of the additional support δ_{PIN} of the connection was build (Fig. 8).

The coefficient K_F is determined by the formula

$$K_F = \frac{S_{op}}{S_{do}},$$

where S_{op} is the contact area of the end of the pin and the coupling of the main shoulder/support of the joint; S_{do} is the contact area of the end of the pin and the coupling of the additional shoulder support of the joint.

So, as the value of the coefficient K_F, increases, as it can be seen from the nomogram, there is an increase in the value δ_{PIN} for both types of threads. The obtained values of the dimensionless coefficient K_F can be used later in the development of other standard sizes of threads of a similar design.

Conclusions and suggestions:

1. After analyzing the possible methods for simulating the screwing moment of a threaded joint during the study on the stress-strain state and the development of new joint designs, it

Table 1

Parameters for different standard sizes of type connection NC50, NC55

No.	Type of thread	D	d_p	D_F	D_{2p}	D_{3p}	D_{4p}	L	D_{1B}	D_{4B}
1	NC50_161_76	80.95	38.1	77	65.215	64.0296	52.19	114.31	67.475	53.451
2	NC50_165_69	80.95	34.95	77	65.215	64.0296	52.19	114.31	67.475	53.451
3	NC50_165_71	82.55	35.7	77	65.215	64.0296	52.19	114.31	67.475	53.451
4	NC50_165_76	82.55	38.1	77	65.215	64.0296	52.19	114.31	67.475	53.451
5	NC50_168_76	84.15	38.1	77	65.215	64.0296	52.19	114.31	67.475	53.451
6	NC50_168_88	84.15	44.45	77	65.215	64.0296	52.19	114.31	67.475	53.451
7	NC 50_168_71	84.15	35.7	77	65.215	64.0296	52.19	114.31	67.475	53.451
8	NC 55_177_76	88.9	38.1	85.35	72.58	71.0057	57.7865	127.01	75.005	59.0395
9	NC 55_177_88	88.9	44.45	85.35	72.58	71.0057	57.7865	127.01	75.005	59.0395
10	NC 55_177_92	88.9	46.05	85.35	72.58	71.0057	57.7865	127.01	75.005	59.0395
11	NC 55_184_82	92.1	41.3	85.35	72.58	71.0057	57.7865	127.01	75.005	59.0395
12	NC 55_184_92	92.1	46.05	85.35	72.58	71.0057	57.7865	127.01	75.005	59.0395
13	NC55_184_101	92.1	50.8	85.35	72.58	71.0057	57.7865	127.01	75.005	59.0395
14	NC 55_190_82	95.25	41.3	90.1	72.58	71.0057	57.7865	127.01	75.005	59.0395

Table 2

Result of the simulated modeling

No.	Type of thread	D	d_p	L_{BOX}	δ_{BOX}	L_{PIN}	δ_{PIN}	K_F
1	NC50	161.9	76.2	126.93	0.08	127.04	0.03	0.92
2	NC50	165.1	69.9	126.93	0.08	127.05	0.04	1.09
3	NC50	165.1	71.4	126.93	0.08	127.05	0.04	1.05
4	NC50	165.1	76.2	126.93	0.08	127.04	0.03	0.92
5	NC50	168.3	76.2	126.93	0.08	127.04	0.03	0.92
6	NC50	168.3	88.9	126.93	0.08	127.01	0	0.54
7	NC50	168.3	71.4	126.93	0.08	127.05	0.04	1.05
8	NC55	177.8	76.2	139.64	0.07	139.77	0.06	1.14
9	NC55	177.8	88.9	139.64	0.07	139.74	0.03	0.82
10	NC55	177.8	92.1	139.64	0.07	139.74	0.02	0.73
11	NC55	184.2	82.6	139.64	0.07	139.75	0.04	0.98
12	NC55	184.2	92.1	139.64	0.07	139.73	0.02	0.73
13	NC55	184.2	101.6	139.64	0.07	139.72	0.01	0.46
14	NC55	190.5	82.6	139.64	0.07	139.76	0.05	0.67

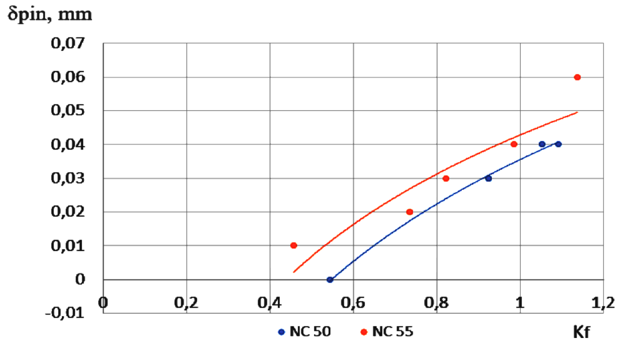

Fig. 8. Nomogram for determining the length of the pin part of the additional shoulder joint (δ_{PIN}) *by the value of the coefficient* K_F *for threads of types NC50, NC55*

was established that they do not meet the requirements for research in an axisymmetric setup. With this in mind, a method of simulating the screwing moment by using the control dimensions L_R and L_{PIN} was proposed and developed. It enables preserving all the geometric parameters of the threaded joint and, if necessary, to connect the double-shoulder design of the tool joint to the traditional one-shoulder one.

2. With the help of the developed method and the built three-dimensional parametric model of the threaded tool joint, the optimal length values of the additional shoulder pin part of two types threaded joints for their different standard sizes were determined. For the investigated threads of type NC50, the maximum value of the length of the pin is constant and is 0.08 mm, and the values δ_{PIN} – from 0 to 0.04 mm. For the studied threads of type NC55 the maximum value of the pin length is also constant (0.07 mm), the values δ_{PIN} – varies from 0.01 to 0.06 mm.

3. According to the results of the research, the values of dimensionless coefficients K_F were determined. For threads of the NC50 their values range from 0.54 to 1.09 mm, and for threads of the NC55 – from 0.46 to 1.14 mm. Based on the obtained values, a nomogram was built to determine the length of the pin part of the additional shoulder/support of the joint δ_{PIN} by the value of the coefficient K_F.

4. It should be noted that within the same type of thread, the overlap values of the pin ends and the coupling have different values due to the change in the geometrical parameters of the location of the thread relative to the inner and outer diameter of the pipes. Taking this into account, further research will be aimed at establishing the values of the coefficients K_F and the corresponding values of the length of the pin part length of the joint additional shoulder δ_{PIN} for other standard sizes of threaded joints.

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Проектування двоопорних замкових різьбових з'єднань елементів бурильних колон

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Мета. Розроблення способу імітування моменту згвинчування й дослідження конструкцій двоопорних замкових різьбових з'єднань із збереженням їх геометричних параметрів. Визначення оптимальних довжин ніпельної частини додаткової опори для розробленої різьби ОБТ типу NC50, NC55 різних типорозмірів. Встановлення залежності між довжиною ніпельної частини додаткової опори та іншими геометричними параметрами двоопорного з'єднання ОБТ для подальшого застосування при розробленні інших типорозмірів різьб аналогічної конструкції.

Методика. Розроблення конструкцій і способу імітування моменту згвинчування для двоопорних різьбових з'єднань здійснено методом скінченних елементів і застосуванням параметричного моделювання.

Результати. Запропоновано й розроблено спосіб імітування моменту згвинчування, що дало змогу вдосконалити модель їх автоматизованого проектування. Визначені оптимальні довжини ніпельної частини додаткової опори розроблених конструкцій двоопорних різьб типу NC50, NC55. Для проектування різьбових з'єднань аналогічної конструкції вищезгаданих типів різьб виведено безрозмірний коефіцієнт і визначена його величини. Побудована номограма для визначення довжини ніпельної частини додаткової опори з'єднання за величиною коефіцієнта.

Наукова новизна. Удосконалена модель автоматизованого проектування двоопорних різьбових з'єднань елементів бурильної колони, що зберігає їх геометричні параметри й надає можливість моделювати широкий спектр типорозмірів аналогічних з'єднань.

Практична значимість. Визначені оптимальні геометричні параметри розроблених конструкцій двоопорних різьбових з'єднань елементів бурильної колони. Побудована номограма для визначення довжини ніпельної частини додаткової опори з'єднання за величиною безрозмірного коефіцієнта.

Ключові слова: *бурильна колона, напружено-деформований стан, різьбове з'єднання, ніпель, муфта*

The manuscript was submitted 20.06.22.