Theoretical investigation of the tapered thread joint surface contact pressure in the dependence on the profile and the geometric parameters of the threading turning tool

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Abstract. The tapered screw joints are widely used in the formation of the drill string. They consist of two parts - a box and a pin. During the process of screwing box and a pin the contact pressure between their thread surfaces arises, the magnitude of which affects the intensity of wear on these surfaces. For the first time, the authors offer in this work to change the contact area of the thread face by the change in the shape of the cutting edge of the threading turning tool. Previously, in the works of these authors themselves, it was offered to change the shape of the cutting edge exclusively to increase the tightness of the joint. In other works, the justification for changing the geometric parameters of such cutters was offered in order to obtain the possibility of the thread manufacturing from materials with a strength limit of more than 1300 MPa. In this paper an analysis of the influence of the geometric parameters size of the cutter on the value of the contact pressure between the thread surfaces of the box and the pin is carried out. Among above geometric parameters the work height is the most important. As a result of theoretical studies, a certain functional dependence of the contact pressure in the tapered thread joint from the shape of the cutter edge and the magnitude of its work height is obtained.

1. Relevance of the problem
Nowadays tapered threads are very widely used for connection of pipes of oil-and-gas variety in general and for connection of drill pipes in particular, as well as pump-compressor pipes. The quality of these joints is determined by the following parameters: screw-in, tightness and mechanical strength. The last parameter largely depends on the contact pressures that arise between the thread surfaces of the two parts of the joint - a box and a pin. The frictional force between the thread surfaces increases in the screwing up process according to the pressure increasing on these surfaces. The magnitude of the deformation of these surfaces at the moment of their complete screwing up increases too. One and the other leads to the loss of the initial shape of the profile and as a consequence to a decrease in the functional suitability of the joint i.e., reduction of mechanical strength, screwing-ability and tightness. Therefore, the reduction of the contact pressures in the process of screwing those types of pipes between themselves and after their full screwing-up is a very topical issue. Therefore, in the opinion of the authors, it is necessary to investigate the functional effect of certain parameters of the tapered thread on the value of contact pressures. Since the manufacture of the tapered thread of oil and gas pipe assortment occurs with the use of turning tools, an important part of the research should be to
obtain functional relationships between the shape of the cutting edge profile of the tool and the contact pressures value on the thread surfaces.

2. Theoretical studies of contact pressures in tapered thread joint in the oil and gas pipes

The contact pressure between the drill string tool joint thread surfaces of the box and the pin is deeply studied at the theoretical level in the work [1]. In this paper, the value of contact pressure is determined for two fundamentally different periods of the joint process. The first period - just when the pin is installed into the box, and the second period when there is a process of screwing the pin into the box already. Figure 1 shows the moment of the installation of the pin in the box and Figure 2 illustrates the layout of the connection parts in the process of the screwing up the pin into the box. On both schemes signs, which correspond to the standard [2], are used:

- $d_1$, the largest outer diameter of the pin thread on the large base of the taper;
- $d_3$, the largest outer diameter of the pin thread on a small base of taper.

In addition, in Fig. 1, 2 the following signs are applied:
- $d_0$ - the largest outer diameter of the pin in the place of its contact with the box at the moment before start to screw;
- $L$ - the length of the tapered thread of the pin.

![Figure 1](image1.png)

**Figure 1.** Scheme of the location of the parts of the threaded connection at the time of the pin installation into the box. The numbers indicate: 1-pin, 2-box.

![Figure 2](image2.png)

**Figure 2.** Scheme of the location of the parts of the threaded connection during the screwdriving of the pin into the box. The numbers indicate: 1-pin, 2-box.

Figure 3 shows the scheme of the mutual placement of the surfaces of the pin(blue) and the box (black) at the moment the pin is set on the box (position I) and in the process of the screwing the pin into the box (position II). Signs used in Figure 3 are conformed to standard [2]:

![Figure 3](image3.png)
h₁ - thread height truncated, P - pitch, ϕ - taper angle, a - crest flat width, α/2 - half profile angle.
In addition, the following notations are applied in Figure 3: x - the value of the mutual displacement of the pin's thread surface onto the box thread surface at the certain moment of screwing, P₁ - the pitch is defined between the larger flanks.

Formula 1 according to the data [1] shows the functional dependence of the area of the mutual contact of the thread surface of the pin and the box in the process of screwing them. This screwing has a certain number of revolutions m corresponding to the displacement of the point C (see figure 3) along the larger flank of the pin in the direction from point D. So distance X depends on number of revolution m.

\[
F_k = \frac{nm}{4 \cos \alpha} \left[ (d_1 - 2h + KP_1 \cdot m)^2 - (d_1 - KL)^2 \right] (\text{mm}^2)
\] (1)

where:  
Fk - the area of mutual contact between the pin and the box in the process of screwing them on the value of X, corresponding to a certain number of revolutions of the pin m;  
h - thread height truncated (mm);  
d₁ - the largest outer diameter of the pin thread on the large base of the taper;  
P₁ - the pitch is defined between the larger flanks  
K = 2tgϕ - taper of thread.  
L - the length of the tapered thread.

According to standard [2], the length of the tapered thread of the pin is determined by the equation:

\[
L = l_n - 12.7 \text{ (mm)}
\]

where \( l_n \) - the length of the tapered part of the pin.
Formula 2 according to the data [1] shows the functional dependence of the area of mutual contact between the crests of the box and the pin on: parameter \( a \) - the crest flat width, the tape angle \( f \) and the length of the screw line on the outer surface of the tapered thread \( L_b \):

\[
F_b = \frac{aL_b}{\cos \phi} \text{ (mm}^2\text{)}
\]  \( \text{(2)} \)

where:
- \( a \) - crest flat width;
- \( \phi \) - taper angle;
- \( L_b \) - length of the screw line on the outer surface of the tapered thread \( L_b \). This parameter is determined by formula 3:

\[
L_b = \frac{\pi}{2PK} \left[ (d_1 - 2h)^2 - (d_1 - KL)^2 \right] \text{ (mm)}
\]  \( \text{(3)} \)

Formula 2 shows the calculation of the area of full contact between crests of the pin and the box, that is, the maximum possible area. In the case of point contact, that is, in the vicinity of points \( B \) or \( A \) - the area will in theory approach zero.

Figure 4 shows the scheme of the moment of setting the pin into the box (the pin is highlighted in dark gray), but the contact is not on their crests, but on their longer flanks. For a better illustration of this moment, the diagram shows the points \( A \) and \( F \) located on a line parallel to the axis of the thread 1. This line is the direction which the pin is setting down into the box along. Unlike the variant of contact I (see figure 3), in this case point \( B \) on the crest of the pin corresponds to the point \( F \), that is, the crest of the pin moved down past the crest of the box and touch to the flank of the box 5. This corresponds to the initial distance \( X_0 \) between the crests of the pin and the box.

![Figure 4](image)

**Figure 4.** Scheme of the moment of installing pin into the box by use the surface of the thread flanks. Numbers mean: 1- axis of the tapered thread joint, 2- profile of the pin thread, 3- pitch diameter of the pin thread, 4- pitch diameter of the box thread, 5 - profile of the box thread, 6 - profile of the box thread at the moment of the full screwing

Formula 4 in accordance to [1] describes the area of contact between the flank of the pin and the box at the moment of the installation of the pin into the box (see figure 4):

\[
F_c = \frac{\pi(P - 2a)}{4P \cos \alpha} \left[ (d_1 - 2h + K \left( \frac{1}{2} \tan \alpha \right)(P - 2a))^2 - (d_1 - KL)^2 \right] \text{ (mm}^2\text{)}
\]  \( \text{(4)} \)
The screw-in process begins immediately after the moment the pin is installed into the box and ends when the pitch diameter of the pin coincides with the pitch diameter 4 of the box, and the crest of the pin moves to the value of \((h-X_0)\). This position of the pin in Figure 4 is shown in light gray and is indicated by the figure 6.

3. Theoretical studies of the functional dependence of the tightness of the tapered thread joint on the value of the working height

The paper [3] presents theoretical and virtual experimental studies aimed at obtaining the dependence of the velocity of the washing liquid moving through the screw technological duct between the thread surfaces of the pin and the box from the geometric parameters of this duct- its height and width. In figure 5 for clarity the cross section of this duct is painted black. Actually it is a gap between the pin and the box thread surfaces and its parameters are: \(a\) - crest flat width and \((h_1-h)\) - the difference between the work height and the thread height truncated of the thread profile. All three parameters according to [2] are optional for the thread and are intended for the design of a cutting tool.

![Figure 5. Scheme of the tool joint tapered thread profile according to [2]: P - thread pitch, H - height of fundamental triangle, h1 - work height, h - thread height truncated, a - crest flat width, H/2 - half height of fundamental triangle, r - root flat corner radius, a/2 - half profile angle](image)

Investigations [3] show that with a decrease in the height of the gap \((h_1-h)\) in the screw channel between the box and the pin from 0.450 mm to 0.134 mm at an input pressure of 10 MPa, the flow rate of the drilling mud decreases from 30 - 55 m / s to 0.56 -1.48 m / s.

4. Theoretical researches of technological possibility of the manufacture of the tapered thread especially designed for low permeability of the clay solution inside the drill pipe tool joint

Studies [4, 5] have been carried out with the aim of identifying the possibility of using the threading cutter tool with negative values of the back rake angle in the range of -3 ° to -5 °. Such a range of the back rake angle value is necessary in case of necessity to ensure an increase in the tool life by the conditions:

- when the requirement for the strength of the material from which the drill pipe thread connection is manufactured increases;
- when a significant reduction of the gap between the contact screw surfaces of the pin and the box is proposed, and this gap is necessary for the design of the cutting tool.

In the paper [4] it is offered:

- according to the technological requirements, to achieve a clearance gap \((h_1-h)\) by increasing the size \(h\) of the profile of the cutting edge of the cutter;
- according to the economic and technological requirements to apply the back rake angle to -5 ° without adjusting the profile angle of the cutting edge of the threading tool since the deviation of this profile angle do not exceed 20% of the tolerance of the value \(\alpha / 2\).
Such offers are positive:

- for use in tools for thread cutting in steel work piece, the ultimate tensile strength of which will exceed 1300 MPa;
- to achieve the technological gap \((h_1-h)\), which is reduced by 2-3 times compared with the standard.

The analysis of primary sources show that the value of the contact pressure between the parts of the pipe connection and the level of its tightness depend on the thread height truncated \(h\), which can be adjusted by correction of the profile of the tool cutting edge in its corresponding part, as shown in Figure 6.

Among the parameters that affect the contact pressure are also the crest flat width \(a\) and the profile angle \(\alpha\) (equation 1, 2, 3, 4).

Figure 6. Illustration of changing the profile of the cutting edge of a full-profile threading lathe tool which ensures a reduction of the technological gap between the box and the pin by 0.3 mm

But studies [1] do not show the dependence of the contact pressure of the collected connection between oil and gas pipes or sucker rods on the value of \(h\) as the most influential parameter. According to studies [1,2,3,4] it cannot to conclude about direct effect of the change in the shape of the cutter edge of the tool on the values of contact pressures during the installation and screwdriving of the pin into the box and during the operation of the pipe joints.

**The purpose** of the paper is to determine the effect of changing the profile of the cutting edge and the geometric parameters of the threading lathe cutter tool on the values of the contact pressure that arises at the moments of installation of the pin into the box, during of the process of their screw-in and during operation of the their connection and of the sucker rod connections.

**Problem setting:** To achieve this goal should

- to conduct a virtual experiment to find out the changes in the contact pressures, stresses and fatigue safety factor (FOS) values that arise during the operation of the pin-box connection of drill pipes and sucker rods, depending on the change in the thread height truncated \(h\);
- to obtain functional dependences of the change of the contact pressures that arise during the installation of the pin into the box and their screw-on by changes in the profile of the cutting edge and certain geometric parameters of the threading lathe tool.

5. Presenting main material. Investigation of the contact pressure in the connection of ZN-80 dependence on the value of the thread height truncated \(h\) by means of finite element simulation

Consider the drilling tool joint ZN-80 GOST 5286 with Z-66 GOST 28487 thread (2 3/8 REG API Spec. 7 equivalent). With the help of a finite element simulation, the effect of the major diameter of the pin thread on contact pressures at profile points 1 and 2 was studied (See figure 7). The value of the working height of the profile \(h\) is changed by reducing the major diameter of the pin thread, which is possible as a result of the wear during frequent make-up operations. The major diameter of the pin thread in the end face plane of the nipple \(d\) varied from 47.674 mm \((h = 2.22\) mm\) to 46.074 mm \((h = 1.47\) mm\). The minor diameter of the box thread is not change.

The material of the parts is steel. Material plasticity and friction are simulated. Two steps of the external axial tensile load were created: \(F_{\text{min}} = 0\) N and \(F_{\text{max}} = 1\) MN. The joint make-up was simulated...
by axial deformation $\Delta = 0.1$ mm of the box shoulder. For the approximate calculation of the fatigue safety factor $D$, the dependence of Sines [6] with a material endurance limit of 207 MPa was used.

![Figure 7](image)

**Figure 7.** Points 1 and 2 for the calculation of the contact pressure

It is noticeable that a decrease in $d$ affects the distribution of equivalent stresses in the connection (see figure 8).

![Figure 8](image)

**Figure 8.** Von Mises stress distribution (MPa) in first threads of the drilling tool joint: $d = 47.674$ mm (a), $d = 46.074$ mm (b)

It is also noticeable the reduction in the fatigue strength of the pin with a decrease in the diameter $d$ (figure 9).

![Figure 9](image)

**Figure 9.** Fatigue safety factor $D$ in the first root of the pin thread as a function of diameter $d$

Based on the results of the finite element analysis, a regression model of contact pressures $CP$ at points 1 and 2 as a function of the thread turn number $N$ and the major diameter $d$ is found (5). Here $a_i$ - coefficients, $p_i$ - exponents.

$$CP = a_0 + a_1N^{p_0} + a_2N^{p_1} + a_3N^{p_2} + a_4d^{p_3} + a_5d^{p_4} + a_6d^{p_5} + a_7Nd$$

(5)
Values of coefficients and exponents at point 1 (Figure 10): $a = -3.77473564e+07, -4.91142995e+03, -2.55477895e+01, 1.00608216e+00, 2.80965184e+07, -4.07911283e+03, 3.66910355e+01, 5.55870891e+00; p = 0.1, 1.9, 2.9$. Coefficient of determination $R^2 = 0.71$.

Figure 10. Values of contact pressures at points 1 as a function of $N$ and $d$ for the drilling tool joint

Values of coefficients and exponents at point 2 (see figure 11): $a = -3.41589228e+07, -1.03089704e+03, 2.37829334e+01, -1.66910424e+00, 2.94843546e+06, -5.95458890e+04, 4.15737420e+02, 1.48764378e+01; p = 0.9, 1.9, 2.9; R^2 = 0.97$.

From dependences it can be seen that below $d = 47.2$ mm contact pressures begin to rise sharply.

Consider the axisymmetric FE models of 19 mm GOST 13877 (3/4 in. API Spec 11B equivalent) sucker rod couplings. Material plasticity and friction are simulated. Two steps of the external axial tensile load were created, which form in the rod body the stress $P_{\text{min}} = 0$ MPa and $P_{\text{max}} = 276$ MPa. The joint make-up was simulated by axial deformation $\Delta = 0.1$ mm of the box shoulder.

Figure 11. Values of contact pressures at points 2 as a function of $N$ and $d$ for the drilling tool joint

The major diameter $d$ of the nipple thread varies from 26.624 mm (maximum allowable) to 25.624 mm. The minor diameter of the coupling thread is constant - 24.79 mm (maximum allowable). It is
noticeable that a decrease in $d$ affects the distribution of equivalent stresses in the connection (see figure 12).

![Figure 12. Von Mises stress distribution (MPa) in first threads of the sucker rod coupling: $d = 26,624$ mm (a), $d = 25,624$ mm (b)](image)

In the case of a decrease in diameter $d$, the most noticeable decrease in the fatigue strength of the coupling in the zone of the last root of the coupling thread (see figure 13).

![Figure 13. Fatigue safety factor D in the last root of the coupling thread as a function of diameter d](image)

Based on the results of the finite element analysis, the coefficients of the regression model (1) are found. Values of coefficients and exponents at point 1 (see figure 14): $a = 1.67302056e+07, 6.12225399e+02, 1.59368037e+01, 2.78069341e-01, -2.43145831e+06, 8.76366121e+04, -1.09226421e+03, -2.32280329e+01; p = 0.9, 1.9, 2.9; $R^2 = 0.97$.

![Figure 14. Values of contact pressures at points 1 as a function of N and d for the sucker rod connection](image)
Values of coefficients and exponents at point 2 (see figure 15): 
\[ a = -8.52515266 \times 10^7, -2.15706639 \times 10^3, -1.36373263 \times 10^2, 5.62253835 \times 10^1, 6.80490010 \times 10^7, -7.08061914 \times 10^4, 1.97004893 \times 10^4, 1.19405460 \times 10^1; \]
\[ p = 0.1, 1.9, 2.2; R^2 = 0.98. \]

**Figure 15.** Values of contact pressures at points 2 as a function of \( N \) and \( d \) for the sucker rod connection.

From dependences it can be seen that below \( d = 26.2 \) mm contact pressures begin to rise sharply. Functions \( CP = f(N, d) \) can be used to justify the allowable value of the major diameter \( d \) of the pin thread of the drilling tool and sucker rod joints.

6. **Investigation of the functional dependence of the contact pressure on the tool cutter edge profile in the drill string tool joint of the 2 3/8 Reg at the time of the pin installation into the box**

Figure 16 shows a schematic on which thick black lines the profile of the tapered thread tool joint 2 3/8 Reg of the drill string are displayed in accordance to standard [2]. It corresponds to Form I [2]. The names of the parameters and their values are shown in table 1. The input parameters and their values for this study are reflected in blue:
- increase of 0.05 mm of the value of \( h \) (thread height truncated). This increase is executed five times starting from the nominal \( h = 2.262 \) and completing of the size \( h^* = 2.876 \). The drawing helps to understand that \( h^* = h + 5 \times 0.05 \) (mm). Because the absence of space, all intermediate dimensions, such as \( h + 0.05 \) (mm), are not shown in the figure, but are presented in Table 1;
- variable width of the crest, which corresponds to the magnitude of the increase in the value of the thread height truncated \( h \); The width \( a = 1.016 \) mm according to [2] corresponds to the nominal value \( h = 2.626 \) mm. All other correspondences between changes \( h, a \) are received by means of geometric construction in the graphic editor. The functional relationship between these variables is shown in Table 2. The red color in Figure 16 shows the limit sizes according to the tolerances [2].

| Table 1. Parameters of the profile of the tool joint tapered thread (Form I) according to the data [2] |
|---------------------------------|-----------------|--------------|
| Legend and parameter name        | Parameter size, mm | Tolerance, mm |
| \( P \) pitch                    | 1.143            | 0.285        |
| \( H \) height of the fundamental triangle | 1.143            | 0.285        |
| \( h \) thread height truncated * | 4.571            | 0.033        |
| \( h_1 \) work height *          | 0.286            | 0.593        |
| \( h/2 \) half of thread height truncated * | 1.143            | 0.285        |

* the sizes and deviations are output for the design of the tool and have an optional dimension for the thread.
Table 2. Functional dependence of the value $a$ (variable crest flat width) on the value $h$ (variable thread height truncated)

| variable $h$ mm | 2.626 | 2.676 | 2.726 | 2.776 | 2.826 | 2.876 |
|-----------------|-------|-------|-------|-------|-------|-------|
| variable $a$ mm | 1.016 | 0.905 | 0.841 | 0.770 | 0.666 | 0.537 |

The results of research on the contact pressure that occurs at the moment of the pin installation into the box is shown in Table 3. Applying the formulas 2, 3, 4 and substituting them with variables $a$ (column 7) and $h$ (column 6), the corresponding values of the contact area were obtained for the installation of the pin in the flank side $F_b$ (column 2) and crests $F_v$ (column 3) according to figure 3. In order to obtain the values of the contact pressure, the value of the drill string weight is selected - 10 tons. The calculation scheme of pressure from the load drill string is shown in figure 17.

Figure 16. The profile of the tool joint thread profile of the 2 3/8 Reg according to the standard [2]

Figure 17. Scheme to calculate the pressure transmitted from the load drill string 1 to the surface of the thread at the time of installation of a pin into the box by the scheme I (crests) or in by the scheme II (flanks)
Using this scheme, the pressure \( p_b \) from the weight of the drill string, which is marked by vector \( I \) is transmitted to the platform of the crest with \( BC \) width can be calculated by the formula:

\[
p_b = \frac{M \cdot 10000 \sin(\varphi)}{F_b} \text{ (MPa)}
\]

where:  
- \( M \) - weight of the drill sting, t;  
- \( F_b \) - area, which is calculated by the formula 2, \( \text{mm}^2 \).

According to the same scheme, the pressure \( p_c \) from the weight of the drill string, which is indicated by the vector \( I \), is transmitted to the side of the flank the width of which is marked \( HF \) corresponds to the moment of installation of the pin into the box according to scheme II. This pressure can be calculated by the formula:

\[
p_c = \frac{M \cdot 10000 \cos\left(\frac{\alpha}{2}\right)}{F_c} \text{ (MPa)}
\]

where:  
- \( F_c \) - area \( \text{mm}^2 \), which is calculated by the formula 4;  
- \( \alpha/2 \) - half profile angle.

The values of contact pressures at the installation of the flank sides are shown in column 4, and when mounted with crests - in column 5.

**Table 3.** Results of theoretical studies of contact pressure between the tapered thread surfaces of the tool joint 2 3/8 Reg at the time of the installation of the pin into the box. Load 10 tons

| No. | contact area, \( \text{mm}^2 \) | contact pressure, MPa | \( h \), mm | \( a \), mm |
|-----|-----------------------------|----------------------|----------|----------|
| 1   | \( F_c \) | \( F_b \) | \( p_c \) between flanks | \( p_b \) between crests |
| 2   | 649,25 | 1400,40 | 133,39 | 8,86 | 2,626 | 1,016 |
| 2   | 693,02 | 1233,60 | 124,96 | 10,05 | 2,676 | 0,905 |
| 3   | 714,85 | 1133,50 | 121,15 | 10,94 | 2,726 | 0,841 |
| 4   | 739,47 | 1026,00 | 117,11 | 12,09 | 2,776 | 0,770 |
| 5   | 778,63 | 877,32 | 111,22 | 14,14 | 2,826 | 0,666 |
| 6   | 828,68 | 699,23 | 104,51 | 17,74 | 2,876 | 0,537 |
| 7   | 665,22 | 1512,50 | 132,17 | 8,20 | 2,446 | 1,055 |
| 8   | -28,88 | -9,54 | -21,7% | 116% |

Row 1 of Table 3 corresponds to the standard values of \( h = 2,626 \text{ mm} \) and \( a = 1,016 \text{ mm} \). Lines 2-6 correspond to the variable \( h \) and \( a \) according to the data in Table 2. Row 8 shows the difference between pressure values in row 6 (lowest value) and row 1 (highest value) in column 4, as well as between pressure values in row 6 (highest value) and row 7 (lowest value) in column 5. Row 9 shows the relative increase / decrease of pressure in the corresponding columns.

Figure 18 shows the graph where the extreme left point corresponds to the lower limit of tolerance to the size \( h \). Judging by the geometric construction in Figure 16, Table 3 and the graph, the pressure remains virtually unchanged at \( h = 2,626 \text{ mm} \). The functional dependence of the rest of the array of points is close to the straight linear one, which can be expressed using the following equation:

\[
p_c = 132 - 0.1(h - 2600), \text{ (MPa)}
\]

where \( h \) - thread height truncated, mm.
Figure 18. The diagram of the dependence the flank contact pressure of the box on thread height truncated $h$ at the moment of the installation of a pin into the box. Tool joint 2 3/8 Reg. Weight of the drill string is 10 tons.

Figure 19 shows the contact pressure dependence on the value of the thread height truncated $h$ onto the crest of the box at the time when the pin is installed in it. The extreme left point $A$ corresponds to the lower tolerance of the size $h$. Judging by the geometric drawing in Figure 16, Table 3 and the graph, the pressure will remain practically unchanged to the point with coordinate $h = 2,626$ mm. The functional dependence of the rest of the array of points besides the extreme right point $B$ is close to the rectilinear. This dependence can be expressed using the following equation:

$$ p_b = 8 + 0.0264(h - 2600) \text{ (MPa)} $$

where $h$ - thread height truncated ($10^{-3}$ mm).

Figure 19. The diagram of the dependence the crest contact pressure of the box on thread height truncated $h$ at the moment of the installation of a pin into the box. Tool joint 2 3/8 Reg. Weight of the drill string is 10 tons.
Table 4. Dependence of the values of the contact area $F_k$ and the contact pressure $p_k$ of the thread flank surface on the value of the thread height truncated $h$ under the weight of the drill string of 10 tons at the start of the screw-in process ($m = 0.61$). Tool joint 2 3/8 Reg. The thread height truncated $h$ varies from the standard $h = 2.626$ mm (row 1) to the maximum (row 7) in increments of 0.05 mm. The separate row 0 (boundary) shows the value of area and pressure for the minimum permissible value $h = 2.446$ mm in accordance with standard [2].

| No. | $F_k$, mm$^2$ | $p_k$, MPa | $h$  |
|-----|--------------|------------|------|
| 0   | 6066.32      | 14.28      | 2.446|
| 1   | 5899.58      | 14.68      | 2.626|
| 2   | 5853.42      | 14.80      | 2.676|
| 3   | 5807.33      | 14.91      | 2.726|
| 4   | 5761.32      | 15.03      | 2.776|
| 5   | 5715.37      | 15.15      | 2.826|
| 6   | 5669.49      | 15.28      | 2.876|
| 7   | 5623.69      | 15.40      | 2.926|

Table 5. Dependence of the values of the contact area $F_k$ and the contact pressure $p_k$ of the thread flank surface on the value of the thread height truncated $h$ under the weight of the drill string of 10 tons at the finish of the screw-in process ($m = 3.86$). Tool joint 2 3/8 Reg. The thread height truncated $h$ varies from the standard $h = 2.626$ mm (row 1) to the maximum (row 7) in increments of 0.05 mm. The separate row 0 (boundary) shows the value of area and pressure for the minimum permissible value $h = 2.446$ mm in accordance with standard [2].

| No. | $F_k$, mm$^2$ | $p_k$, MPa | $h$  |
|-----|--------------|------------|------|
| 0   | 6066.32      | 14.28      | 2.446|
| 1   | 5899.58      | 14.68      | 2.626|
| 2   | 5853.42      | 14.80      | 2.676|
| 3   | 5807.33      | 14.91      | 2.726|
| 4   | 5761.32      | 15.03      | 2.776|
| 5   | 5715.37      | 15.15      | 2.826|
| 6   | 5669.49      | 15.28      | 2.876|
| 7   | 5623.69      | 15.40      | 2.926|

7. Conclusions
1. The contact pressure between the crests of the drill string tool joint thread surfaces at the time of installation of the pin into the box directly proportional depends on the value of $h$ (thread height truncated), which is formed during the process of cutting the thread.
2. The contact pressure between the larger flanks of the drill string tool joint thread surfaces at the time of installation of the pin into the box inversely proportional depends on the value of the $h$ (thread height truncated), which is formed in the process of cutting the thread.
3. The contact pressure between the larger flanks of the drill string tool joint thread surfaces the time of the process of screwing the pin into the box directly proportional depends on the value of the working height, which is formed in the process of threading.

4. In the screwed state of the drill tool joint and the sucker rod connection, contact pressures are in certain regression dependencies $CP = f (N, d)$, where $d$ is major diameter of the nipple and $N$ is a thread turn number. Dependencies can be used to justify the permissible value of the major thread diameter of the nipple $d$. From the dependencies it is noticeable that below $d = 47.2$ mm for ZN-80 connection ($d = 26.2$ mm for 19 mm sucker rod connection) contact pressures sharply increase and FOS values sharply decrease.

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