Improving the fatigue life of the tool-joint of drill pipes by optimizing the variable pitch of the box thread

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Abstract. The reliability of oil drill-strings largely depends on the reliability of tool-joints. It is known that the load along the tool-joint thread is distributed unevenly and it is possible to equalize it by applying a variable thread pitch of a pin or a box. The article describes the optimization method of the pitch change function along the box thread by maximizing of the minimum value of fatigue life in the pin roots. Additionally, the value of the contact pressure on the shoulder, which determines the screw make-up force, is taken into account. To implement the method, we used the Abaqus/CAE and fe-safe software, the parametric axisymmetric finite element model of the tool-joint developed by the authors and the procedures for local (BFGS method) and global (Differential Evolution method) optimization from the SciPy package.

Keywords: tool-joint, fatigue, thread, variable pitch, optimization, finite element model

1. Introduction
It is known that the load in the threaded connections is mainly distributed unevenly – the first turns of the connection are loaded more than the last. This can lead to plastic deformation and fatigue failure in the area of the first turns. In [1] it is analytically proved that the load in the threaded connection of the bolt-nut is distributed according to the function of the hyperbolic cosine. The loads in the tapered thread tool-joint of drill pipes (API and double shoulder) are also uneven [2, 3, 4, 5]. The fatigue fractures (chipping and breakages) of threaded tool-joint are often appeared in the area of the first turns of the pin, and sometimes in the area of the last turns of the box [6]. They are caused by thread deviations (primarily operational wear) and insufficient make-up torque [7, 8].

2. Literature review
There are many design methods to increase the fatigue strength of thread shouldered connections: stress relief grooves [6, 9, 10], modification of the thread profile [11], chamfers on the first turns of the thread, plastic pre-deformation of the connection with a high screw torque, variable pitch diameter and thread pitch [1], double shoulder connections [3], make-up torque optimization [6], the use of coatings [6, 12, 13] and others.

In the work [14] the simulation of a double shoulder tool-joint is performed. It is shown that optimization of its parameters (including a constant pitch) allows to increase fatigue strength. In [15] the influence of the pitch on the stress distribution in the API casing connection is shown. But in these studies the influence of a variable thread pitch is not investigated. In [16] the contact stresses on the surfaces of the pin and box are determined depending on the size of the technological gap but not of...
the pitch. An analytical version of the theory of threaded joint, which is equipped with a spring-loaded collet, is developed in the paper [17] but not relatively tapered thread pitch. The study [18] discusses the problem of deformation simulation of thread joint parts and proposed easy measurement scheme but it does not includes fatigue strength. The increasing of the plasticity of the nut material causes the equalized loads in the joints with an increased nut pitch [1]. In [19] the parameters of the variable pitch thread, in particular the box thread pitch, are optimized using the three-dimensional finite element method. But such a thread has a variable profile and therefore it is less technological. In addition, the strength estimation is performed only by the distribution of equivalent stresses along the thread, and not by fatigue indicators.

The advantages of the method of variable pitch in comparison with other design methods of increase of fatigue strength are:
1. Any variable pitch thread is easily obtained on a machine with numerical program control by turning, i.e. it is technologically easy.
2. The developing of the new profile of the threading tool is not required [20].
3. Thread modification can be made only on the pin or only on the box [21].
4. Can be quite effective, as it directly addresses the problem of uneven loading along the thread [1, 19].
5. It is suitable for different types of threaded connections. For example, for threaded connections of sucker rods [21] and for casing joints [19].
6. Additionally it can improve the leak-tightness [22] and prevent self-loosening of the connection [23, 24]. Choosing a non-linear bolt pitch change function allows preventing the self-loosening of the connection more effectively [25].

The problem with this method is finding the optimal function for changing the pitch along the thread. For optimization it is necessary to develop a parametric finite-element model of the threaded connection with the possibility of automated calculation of fatigue life in unsafe areas [26].

3. Research methodology
The finite element method was used to study the stress-strain state. The three-dimensional FEM-model of the connection is more realistic, for example, it allows simulating asymmetric loads and geometry [19]. But the axisymmetric model requires fewer finite elements and allows you to create a fine mesh, which significantly increases the accuracy without large increasing the computational complexity. This is important for computationally complex optimization problems.

Abaqus/CAE 6.14, fe-safe 6.5 and Python-macro [27], developed by the authors, were used for modelling. The macro was used to automate the iterative optimization process. The parameters are changed, the model is built, the simulation is performed, and the results are obtained at the each iteration. Plastic deformations are not uncommon during normal operation of tool-joints [7], therefore, for adequate simulation, the models of the materials of the threaded joint must be elastic-plastic. Realistic models of the contact surfaces of the threaded connection are also needed. These features of the model can be implemented in Abaqus/CAE.

The parametric axisymmetric finite-element model of the drilling tool joint ZN-80 GOST 5286 with Z-66 GOST 28487 thread (2 3/8 REG API Spec. 7 equivalent) has been developed. The material of the parts is SAE-4140 steel with a Young’s modulus $E = 201$ GPa, a Poisson’s ratio $\nu = 0.33$, a yield strength $\sigma_y = 965$ MPa, a ultimate tensile strength $\sigma_t = 1076$ MPa. Material plasticity and friction are simulated. The nonlinear section of the stress-strain diagram ($\sigma$-$\varepsilon$) is approximated by lines with points: $(\sigma = 965$ MPa, $\varepsilon = 0.0), (985, 0.002), (1005, 0.004), (1024, 0.007), (1044, 0.011), (1064, 0.016), (1084, 0.023), (1103, 0.032), (1123, 0.043), (1435, 0.511).

To simulate the make-up of an axisymmetric model of a threaded connection, the axial elongation (“bolt load”) of the box shoulder [11] by the value $\Delta$ is performed. When changing the value of $\Delta$ in the model, the value of the contact pressure $P_c$ on the shoulder should be monitored. Changing any connection parameters can also change the $P_c$ value even for the same $\Delta$ value. For no-failure operation, you need $P_c >> 0$.

The Brown-Miller strain-life equation [28] is used to calculate the fatigue life $N$. It gives the most realistic life estimates for ductile metals, suitable for multiaxial, proportional and non-proportional...
loading. Here \( N \) is the number of cycles until the fatigue crack appears. The calculations are performed using the fe-safe 6.5 software. The fatigue characteristics of SAE-4140 steel (with a material endurance limit of 572 MPa) from the fe-safe material database are used. At the left end of the pin (Figure 1) the pressure \( L \) acts. It simulates the external tensile load. We consider the fatigue loading cycle \( L_{max} = 0 \text{ MPa}, \ L_{max} = 350 \text{ MPa} \) (roughly corresponds to a force of 1.6 MN).

\[ z(i) = -P \cdot i + x(i), \]  

\[ y = \min (\log N(i)), \]

Figure 1. \( \log N \) distribution: a – standard, \( \Delta=0.25 \text{ mm} \); b – optimized on step 3, \( \Delta=0.27 \text{ mm} \)

Figure 1(a) shows a standard ZN80 tool-joint with numbered thread roots of the pin and box. The axial coordinate of the root with number \( i \) of the box thread can be determined as follows:

\[ z(i) = -P \cdot i + x(i), \]  

where \( i \) is the number of the thread root, 
\( P \) is the thread pitch, 
\( x(i) \) is the function of deviation of the axial coordinate \( z \) of the root \( i \). For the standard connections there is \( x(i) = 0 \).

Let’s formulate the optimization problem:

\[ f(x_1, x_2, ..., x_{13}) \to \max, \]  

where \( f \) is the objective function, the value of which \( y \) is the minima of the function \( \log N(i) \) – the logarithm of the fatigue life \( N \) in the root \( i \) of the pin thread (Figure 1):

\[ y = \min (\log N(i)), \]  

the arguments \( x_i \) are the values of the function \( x(i) \) for the turn \( i \) of the thread.

To optimize \( x(i) \), we use the local (BFGS method) and global (Differential Evolution method) multivariate optimization procedures from the SciPy [29] package. Method BFGS uses the quasi-Newton method of Broyden, Fletcher, Goldfarb, and Shanno (BFGS) [30]. BFGS has proven good performance even for non-smooth optimizations [29]. But the objective function can have many minima, therefore, methods of global multivariate optimization should be applied additionally. For example, Differential Evolution is a stochastic population based method that is useful for global optimization problems. At each pass through the population the algorithm mutates each candidate solution by mixing with other candidate solutions to create a trial candidate [31].

Additionally, you should warn about the possible choice by the optimization algorithm of unacceptable values \( x_i \) (from the point of view of building a model in Abaqus). Differential Evolution has a \textit{bounds} parameter to limit the values of variables. In the case of using BFGS, you can check the \( x_i \) values in \textit{f} routine or apply the L-BFGS-B algorithm [29] for bound constrained optimization.

4. Results

Figure 2(a, b) shows the equivalent von Mises stresses \( \sigma \) in roots of the pin thread. An increase in \( \Delta \) leads to an increase in stresses \( \sigma \). However, for high make-up moments, the stresses increase insignificantly with increasing \( \Delta \). The highest stresses are observed in zones 1, 2, 3, 4. Dependences
σ(i) change their shape during the load cycle. Note the appearance of a minimum in the zone \( i = 9 \) for \( L = 350 \) MPa (Figure 2(b)). This change in shape creates the problem of choosing an optimality criterion. We suggest that it should be a \( \min(\lg N(i)) \) function (Figure 3(a)), not a \( \max(\sigma(i, L=0)) \) or \( \max(\sigma(i, L=350)) \).

Figure 2. Values of von Mises stresses \( \sigma \) in roots \( i \) of pin thread for different values of \( \Delta \): standard (a, b), optimized on step 3 (c, d), \( L=0 \) MPa (a, c), \( L=350 \) MPa (b, d)

It is seen that the value of the decimal logarithm of the fatigue life \( \lg N \) is small in the roots 1, 2, 3 of the pin thread (Figure 1(a)). For \( \Delta = 0.25 \) mm the zone with low values of \( \lg N \) (Figure 1(a)) has the largest area at the root 3 (\( \lg N = 3.64 \)). The smallest values in the box (\( \lg N = 4.32 \)) are in the zone of the root 12 (Figure 1(a), Figure 4(a)). The \( \lg N \) values in roots of the pin thread are sharply different (Figure 3(a)). The values are small in zones 1, 2, 3, 4, and in zones 5, 9, 13 the values are large. An increase in \( \Delta \) usually decreases \( \lg N \) (Figure 4(a)). However, not all dependencies are monotonic. Note the minimum \( \lg N = 3.19 \) in the zone \( i = 2 \) for \( \Delta = 0.3 \) mm. The value in the root 12 of the box thread remains almost unchanged (\( \lg N = 4.3 \ ... 4.2 \)). However, you should not choose the value \( \Delta = 0.2 \) mm, since for it the contact pressure \( P_c \) on the box shoulder is equal to zero (Figure 5). Operation of the tool-joint with \( P_c = 0 \) is not permitted [6]. For example, let us choose the value \( \Delta = 0.25 \) mm, which provides \( P_c = 81 \) MPa (Figure 5).
Figure 3. Values of \( \log N \) in roots \( i \) of pin thread for different values of \( \Delta \): standard (a), optimized on step 3 (b)

Figure 4. Dependence of \( \log N \) in thread roots of pin (roots 1-4) and box (root 12) on \( \Delta \): standard (a), optimized on step 3 (b)

Figure 5. Dependence of \( P_c \) in the box shoulder on \( \Delta \): standard (□), optimized on step 3 (■)

In order to increase the load on the last turns, at the first step of optimization, an approximate (rough) function \( x(i) \) was chosen, which is indicated in Figure 6(a) by number 1 (the standard thread is indicated by number 0). The results (Figure 6(b)) show a significant alignment of the \( \log N \) values, but in roots 1, 2, 3, 4 there are almost no values have changed. In addition, a minimum appeared in the zone of pin root 11 (\( \log N = 3.86 \)).
At the second step, we approximate the first dependence by the line $x(i) = ai + b$ and search for the optimal values of $a$, and $b$ using (2). We use the local optimization algorithm BFGS with initial values $a = -0.0193$ and $b = 0.0338$. As a result, the dependence $x(i) = -0.0212i + 0.1202$ is found and indicated in Figure 6(a) as number 2. It allows to increase the values in the first roots of the pin thread ($\log(N) = 4.16$), but the value in the root 1 remains almost unchanged (Figure 6(b)).

At the third step, we applied the Differential Evolution algorithm and directly searched for the optimal values of thirteen variables $x_1 \cdots x_{13}$. Algorithm parameters: the initial population is the points located near $X_0$ ($X_0$ equal to the values of the optimal function $x(i)$ found in the previous step), $bounds=X_0 \pm 0.1$, $popsze=15$ (population size), $polish=False$ (do not use local optimization algorithm at the end). The result shows an increase in fatigue life in zone 11 (Figure 6(b)). In general, the results are satisfactory for the pin, but the $\log(N)$ value in the zone 12 of the box decreased to 3.8 (Figure 4(b)). However, this is not so unsafe. In addition, the value $\Delta = 0.25 \text{ mm}$ in this case is insufficient, since the contact pressure $P_c$ is equal to zero (Figure 5). The value of $\Delta$ should be increased to 0.27 mm and thus ensure $P_c = 83 \text{ MPa}$ (Figure 5). This will slightly decrease the $\log(N)$ values. The distribution of values is shown in Figure 1(b). The danger zone near the pin root 3 has significantly reduced its area, however, dangerous zones with insignificant areas have appeared at the pin and box roots 11, 12, 13. An increase in $\Delta$ decreases the values of $\log(N)$, but not as significantly as in the original version (Figures 3(b), 4(b)). The stress values at the pin thread roots are shown in Figure 2(c, d). Significant stress equalization is noticeable.

The shape of dependence 3 (Figure 6(a)) is similar to the shape of the logarithmic dependence. Therefore, at the fourth optimization step, the last dependence (found at step 3) is approximated by the function $x(i) = a \cdot \ln(i) + b$ and the optimal values of the coefficients $a$ and $b$ are sought by the BFGS method (initial values: $a = -0.097$, $b=0.1447$). As a result, the dependence $x(i) = -0.081 \cdot \ln(i) + 0.1478$ is found (Figure 6(a)), which make it possible to increase the value of $\log(N)$ in zone 12 of the box to 7 ($\Delta = 0.25 \text{ mm}$), but to slightly reduce it in zones 3 and 4 of the pin (Figure 6(b)).

5. Conclusions
1. By choosing the optimal function of the coordinates deviations of the box (or pin) thread root $x(i)$, it is possible to achieve a significant equalization of the loads along the thread of the tool-joint and to increase the value of the fatigue life $N$ in unsafe zones. Under these conditions – from $N=4200$ to $N=11000$.

2. When optimizing this function $x(i)$, you should also control the value of the contact pressure $P_c$ on the shoulder, since changing the pitch can significantly reduce it, and also control the value of the fatigue life in the last roots of the box thread. It is recommended that the objective function takes these values into account.

3. Due to the complexity of the objective function, one should use combined optimization methods and be careful with the choice of the parameters of their algorithms. Due to the computational
complexity of the optimization problem, it is not possible to completely reduce the irregularity of $\log N$ values in the roots. The reader can try to refine the result using the above method.

4. In order to achieve the highest load equalization effect and increase the fatigue life, an attempt should be made to complement the variable pitch method with other methods of durability increasing. For example, to increase the value of the fatigue life in the area of free roots, you can try to apply a stress relief groove and increase its length.

5. The technological possibilities of reproducing the accuracy of the variable thread pitch require some further research.

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