

## Research of stress-deformation state of worn sucker rod thread connection

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### Abstract

Threaded connections are responsible assemblies, defining sucker rod string reliability. Despite the variety of their design, industrial materials analysis shows that breakages occur in rod body and threaded connections.

In the process of lowering and lifting operations, aimed at well pump replacement, screwing and unscrewing of sucker rods threaded joints is repeated, so pin threads and couplings are worn.

After analyzing the method of pin threads wear determining while screwing, unscrewing, connection and breakout sucker rods tightening, proposed by S. A. Babayan, it has been found out that the amount of thread wear during the makeup is 20–30 times greater than thread wear while fastening. According to this method the last three threads are practically unworn and deterioration of the first coil is 10–12 times greater than the wear of the last turns. The results of calculation correspond with the wear data in the course of sucker rods operation in an industrial environment.

To identify and compare the distribution of stresses in the sucker rods threaded connections using the developed computer ANSYS, finite element axisymmetric models of new and worn standard threaded connections with a diameter of 19 mm have been created. Screw connections were loaded by axial force and torque make-up.

Having analyzed the simulation results we have made a conclusion that with the increase of sucker rod wear, stress distribution in pin thread becomes uniform, respectively, it has positive effect on the threaded connection durability under cyclic dynamic loads. With further deterioration, thread tensions are rising in the middle turns, and in extreme turns – are reduced, so the impact of further deterioration on the connection durability is negative.

Also it has been found that at the limit thread wear maximum stresses are concentrated in the thread coupling, resulting in the fracture of its body.

Key Words: *coil, FEM, stress, sucker rod, threaded connection, wear.*

Threaded connections are responsible assemblies that define the structural reliability of sucker rod string. Despite the diversity of their structural designs, industrial materials analysis shows that the sucker rod failures occur in body rods as well as in threaded joints [1].

The surface integrity influences sucker rod durability. The focus in the manufacturing process of rods is made on cleaning treatment of the surface. Improper transportation and storage conditions lead to rod bending and subsequent straightening of them before lowering into the well – to the formation of surface microcracks, which concentrate stresses.

To prevent the breaks and accidents the strict control of rods surface is required. One of the simplest and most common ways is the direct examination of rods before lowering into the well. In addition, in oilfield practice, a control method based on registration of the nature and measurement of the magnetic flux in the sucker rod is applied. This method of control allows you to separate the rods into categories depending on the capabilities of their work in specific conditions.

Sucker rod breakage during operation occurs sometimes through jamming of the plunger in the barrel

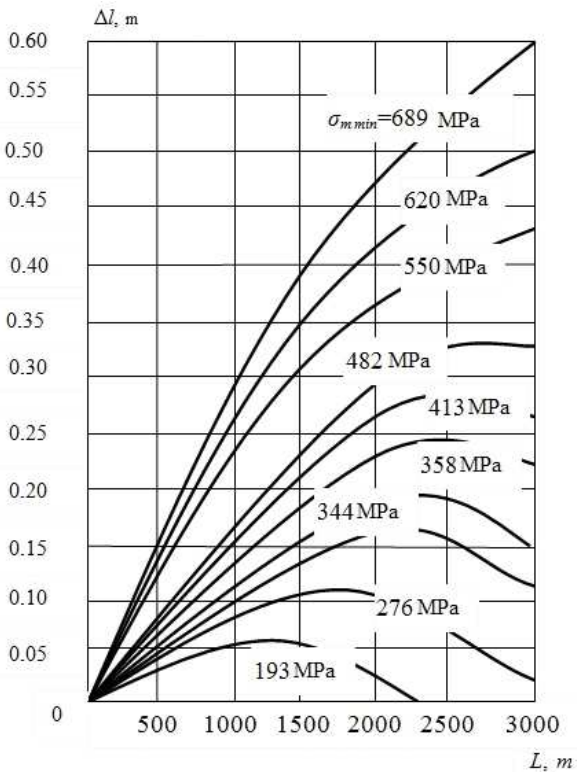
of the downhole pump. In this case, the tension arising in the column of sucker rods exceeds the tensile strength of the rod material. There is a way of column control that is based on the dependence of the applied load on the column rods elongation. During elongation one can draw the conclusions about the size of the load and yield strength. Graph of the maximum allowable safe elongation of the rod string  $\Delta l$  depending on the length of rod string  $L$  with a given minimum yield strength of the material  $\sigma_{m.min}$  is presented in Fig. 1.

At the operation of sucker rod the breakage of heads rods occur arising from fatigue of metal out of which they are made. In this case, the characteristic is a fracture, which is always a line at right angles to the axis of the rod. It's been found out that fatigue breaks occur not only due to high stress, but mostly due to alternating loads, creating low stresses [2]. Analysis of fracture mechanism indicates that the stresses arising from alternating loads, excess the tension from the previous load, and the thrust surface of the shoulder head end of the rod is opened, which eventually leads to breakdown.

As it was proposed by Dagerti, Karter and Mills to prevent the breakdowns of head rod one should pay attention that the end separation of the contacting surfaces of pin and coupling did not occur at the time of workload application [2]. This is possible if the specified torque is applied to the connection.

Torque can be defined by the equation of Dagerti–Carter:

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**Figure 1 – Dependence of the maximum allowable elongation of the sucker rod column with the different yield strength on their length**

$$M_t = k \frac{P}{2} \left( \frac{l}{\pi} + \frac{df_p}{\cos \frac{\alpha}{2}} + d_m f_s \right), \quad (1)$$

where  $M_t$  is the required torque;  $k$  is the coefficient of proportionality;  $P$  is the axial load tension of rod heads;  $l$  is the pitch;  $d$  is the thread diameter;  $d_m$  is the average diameter of contacting surfaces;  $f_p$  is the coefficient of friction between thread coils;  $f_s$  is the coefficient of friction between contacting surfaces;  $\alpha$  is the thread angle, equal to  $60^\circ$ .

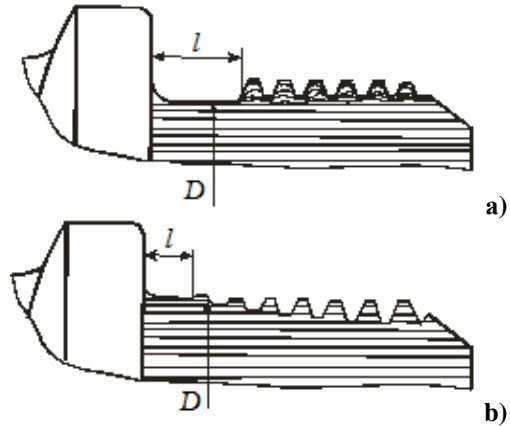
Analysis of heads rods failures [2] made it possible to detect the following regularity: greatest number of accidents is associated with the rods with a diameter of 25 mm, the smallest – with the rods with a diameter of 19 mm and almost no damage – to rods with a diameter of 16 mm. This can be explained by the fact that for screwing of sixteen millimetres rod, small torque  $M_t$  is applied which provides regular hand wrench. For rod with a diameter of 22 mm and 25 mm, screwing torque is necessary, that is difficult to be created using manual key, that’s why the couplings are nontightened, which is the cause of breakdowns.

While analyzing the construction of existing sucker rod threaded connections, it is interesting that at the beginning of their creation they are not supplied with thread groove [2]. The essence of use of thread grooves is that after cutting of the last coil, the head is turned to a smaller diameter than the inner diameter of the thread.

Thread groove provides the following benefits:  
 reduces the concentration of stresses in the root of the last complete coil;

it creates a spring effect, i.e. its elongation from the preload is used to compensate for minor relaxation of stresses arising from the application of workload.

Fig. 2 shows threaded rods profiles.

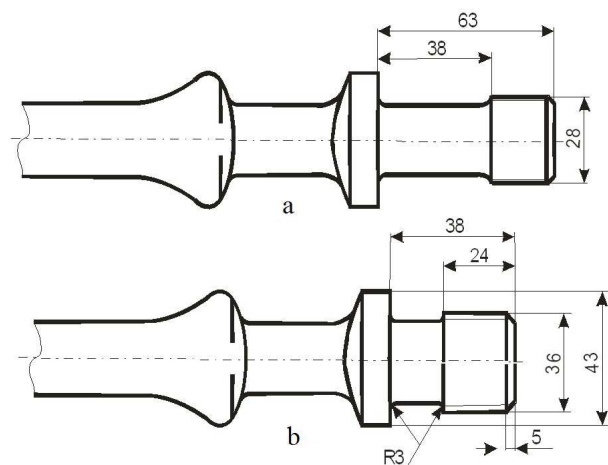


a – the head with a thread groove;  
 b – the head without thread grooves

**Figure 2 – The profile of threaded rod heads**

For example, for the sucker rod of 22 mm in diameter without a groove (b) the length of not turned part  $l = 14$  mm, diameter  $D = 29.1$  mm, and for the rods with the head with a thread groove (a)  $l = 18.2$  mm and  $D = 26.3$  mm. The stresses with the value 275.5 MPa cause in the head an elastic elongation of pin without threaded grooves equal to 0.016864 mm, and in the head with a groove – elongation equal to 0.026416 mm.

Fig. 3 shows the head sucker rod, which differ in the length of thread grooves. For the same magnitude of stress in these heads, which is equal to 275.5 MPa, elastic elongation of pin, shown in Fig. 3a is equal to 0.023622 mm, and elastic elongation of heads in Fig. 3b is equal to 0.051816 mm. For a standard head, this value is equal to 0.01905 mm.



**Figure 3 – The head rods of 22 mm in diameter by API standard**

Thus, the elongation of sucker rod head, shown in Figure 3b is equal to 25%, and for the head shown in Fig. 3a the elongation is up to 172% higher than as usual.

It is well known that the failure of sucker rod couplings is initiated from the inside in the place of first contact of thread with the threaded head of rod. This type of damage has fatigue character. Sometimes the breaks of couplings occur at the site of square section of the head [1].

After the strikes of couplings that are possible when performing careless transactions of screwing – unscrewing of sucker rod threaded connections, there are places of stress concentration. In addition, the bumps can also cause the deformation of the contacting end surfaces of pin and coupling that facilitates access of fluid in thread even when connection is screwed with the required torque [1].

The most common measures and tools that are used to prevent accidents due to fatigue fracture of couplings are:

- use of couplings of a larger size (if the size allows tubing);
- use of safety factor of corrosive working environment in calculating. Sometimes the coupling before work is placed in special inhibitors, but mostly it has little effect;
- change of thread profile;
- limit of stress amplitude (especially for the columns of a large diameter).

During operation of the sucker rod column in the tubing the abrasion wear of rods and tubes is observed. Previously it was believed that this erasure is associated with distortion of the well, and with bending of rods while moving down. Now it has been found that in addition to these reasons, there is bending of the column tubing associated with double stroke (so-called Lubinsky effect).

The main factors affecting the durability of any rod column (regardless of specific conditions) are:

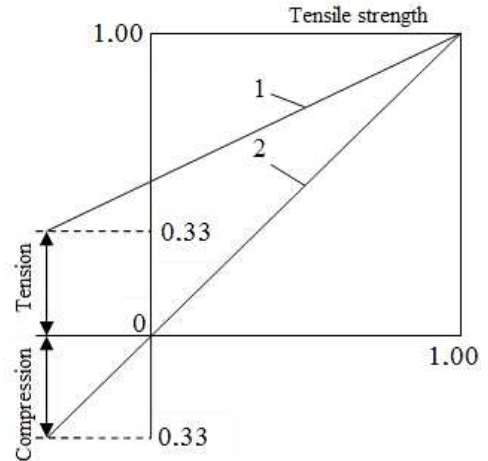
- maximum load  $P_{max}$  ;
- amplitude of the stress;
- frequency of loads that are applied.

So, the increasing of the maximum load  $P_{max}$  on a column sucker rod increases the stresses in the rod, and accordingly, leads to their rapid destruction.

If the working stresses in the column of sucker rods are close to the tensile strength of the material of the rods and couplings, their fatigue failure is possible. Tensile strength of the material rods and joints may be increased by heat treatment of the material. It is also advisable to modify the design of the rods column, rod diameter is increased to a value for which the working stress is much lower then tensile strength of the material. Of course, this option is not always possible because the diameter of the rods is limited to the internal diameter of the tubing column.

Change of the frequency of varying loads that are applied to sucker rod string, not only leads to increase or decrease of dynamic loads, but also to changes in the natural (free) fluctuations of the sucker rod column.

Let's consider another factor – amplitude of stress changes. The basic requirement is the consistency of the amplitude of the stress loads on rods while loads changing. If you modify the maximum stress, minimum should vary to such an amount that the amplitude remains preliminary. Such dependence is expressed graphically in the form of diagrams of Goodman [3]. As shown in Figure 4, the closer the maximum and minimum stress in absolute value are to the value of the tensile strength, the lower is the amplitude of the stress change (at the point of tensile strength it goes to zero).

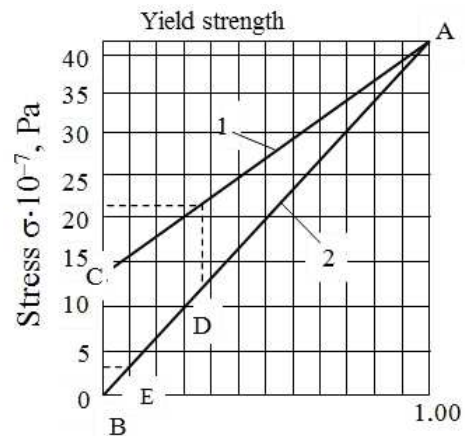


1 – the maximum stress, 2 – the minimum stress

Figure 4 – Normal Goodman diagram

During the operation we are trying to reduce the amplitude of stress, defined by the depth of pump lowering into the well and by the design of the sucker rod column. In deep well the pump rod weight is the main workload, supported by polished rod. Under these conditions, the amplitude of the stress changes would have little value, aided by a small difference between the load while moving the column up and down.

It is especially difficult to maintain appropriate amplitude stress changes in the operation of boom downhole pump at medium depth wells at high speed pumping of production. For these conditions, A.A. Hardy with regard to certain assumptions and propositions changed Goodman diagram (Fig. 5) [3].



1 – the maximum stress, 2 – the minimum stress

Figure 5 – Modified Goodman diagram

As it is shown in Figure 5, in the base of this diagram there is not the tensile strength of the rod material, but yield strength. Diagram has been constructed for a hypothetical steel. In addition, the line of maximum stress starts from zero stress line (point B). This compressive stress is discarded that have high destructive ability for metals. Point C – the value of maximum stress, which is equal to half the value of fatigue limit for this steel as defined in the air in the laboratory conditions.

Due to the arbitrariness of the choice of the individual points for Goodman diagram their configuration can be varied. These diagrams are typically used to control the allowable stress amplitude changes for different grades of steel making up the rod, and for the different working conditions of these rods.

It is known that the failure of threaded joints is about 30% of the rod breaks. They can be classified by origin [1]:

- wear (abrasion) of coupling to the inner surface of the tubing column;
- unscrewing of connection;
- damage due to negligence of staff;
- defects in manufacture;
- metal fatigue.

In addition, the failure of threaded joints occurs due to wear of surfaces of threaded couplings and rods which occurs when they are being repeatedly screwed and unscrewed during round trips to replace downhole pump (Fig. 6).

Thread wear reduces its strength and leads to thread breakage. Reducing of thread wear is an urgent problem, the solution of which is connected with a detailed study of the conditions of threaded connections wear during operation, study of design parameters impact upon threaded connection, physical and mechanical properties of the contacting surfaces, the type of grease used, coatings etc.

Wear of sucker rod thread in one cycle operation takes place in four stages:

- screwing of connection;
- tightening of thread;
- unfastening;
- unscrewing of connection.

Mathematical modeling of wear of sucker rod thread for screwing- unscrewing was made by Babayan S.A. [4]. Fig. 7 shows the scheme of connection, which in this case has only external force  $Q$ , which is equal to the weight of rod.

The size of wear when screwing-unscrewing of sucker rod threaded connection is defined as:

$$W_z = CS_0 f^{-1} Q \sqrt{1 + \pi^2 d_2^2 / S_0^2} (\ln th(mz/2) - \ln th(mH/2)) ch(mz), \quad (2)$$

where  $f$  is the coefficient of friction;  $C$  is the coefficient of proportionality between work and wear;  $S_0$  is the thread pitch;  $d_2$  is the average thread diameter;  $H$  is the couplings thread length;  $Q$  is the tightening force.

Thus, the formula, obtained by S. Babayan (2), describes the wear of pin thread contact surfaces while screwing and unscrewing.



Figure 6 – Worn sucker rod thread

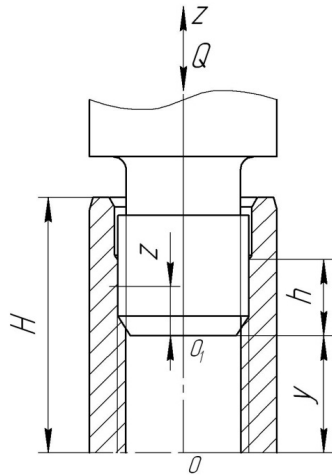


Figure 7 – Diagram to determine the wear of sucker rod thread

In the process of threaded joints tightening the opposed sides of profile are worn out, and the value of wear is determined by the same procedure as for screwing. Distribution of normal pressure under the loading by tightening force corresponds with the scheme when the pin is stretched and coupling is compressed [5]. The tightening by the torque of screwing for tight connection is determined by the axial movement on the value of deformation of pin part H and coupling from the end of the thread to thrust end of shoulder and by the deformations of last coils.

The value of wear while tightening and unfastening of sucker rod threaded connections is defined as:

$$W_z^1 = \frac{\beta C m S_0}{2f} \sqrt{1 + \frac{\pi^2 d_2^2}{S_0^2} Q_t^2} \left( H_1 \beta_1 + \frac{\beta c t h(mH)}{m} \right) \frac{ch(mz)}{sh(mH)}, \quad (3)$$

where  $Q_t$  is the tightening force;  $\beta$  is the coefficient depending on the material properties and geometrical parameters of thread.

The calculations obtained by the formulas (2) and (3) to determine the wear of the thread while screwing and tightening consider the contact malleability of turns in the connection.

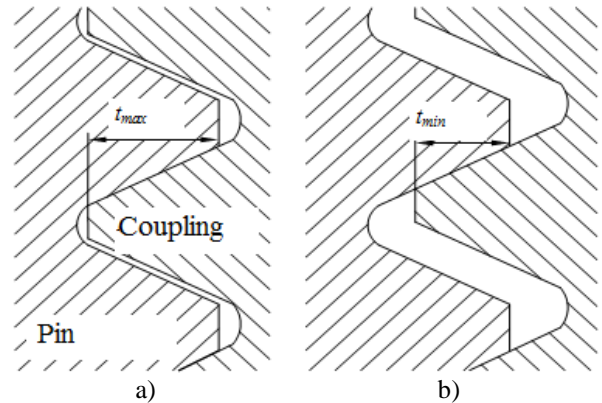
According to the comparative calculation, the value of wear while screwing is 20–30 times higher than the thread wear while tightening, so only the side profile of pin and coupling turns are worn, which are in contact while connections screwing and unscrewing [4].

According to the results of calculation method, the last three turns of thread are practically not worn and the wear of the first coil is 10–12 times higher than the wear of the last coils. The results of calculation correspond with the deterioration data in the course of operation of sucker rod threaded connections in industrial environments.

Equation (2) takes into account the impact of fit in thread connection on wear because the values of  $m$ ,  $\lambda_1$  and  $\lambda_2$  depend on the diametrical clearance.

According to calculations with the increase of the gap in the threaded connection the wear of thread increases. For example, wear of coils surface in a connection with a maximum gap is 20% higher than the wear of coils in a connection without a gap [4].

Thus, analyzing the method of determining of pin thread wear while screwing-unscrewing and tightening-unfastening of the sucker rod connections offered by S.A. Babayan, it is advisable to determine the distribution of stresses arising in a very worn connection (according to GOST 13877–96) with regard to values of thread coils intermeshing (Fig. 8).



a) – the maximum overlap of coils, b) – the minimum overlap of coils

Figure 8 – Extreme case of contact of pin and coupling thread coils

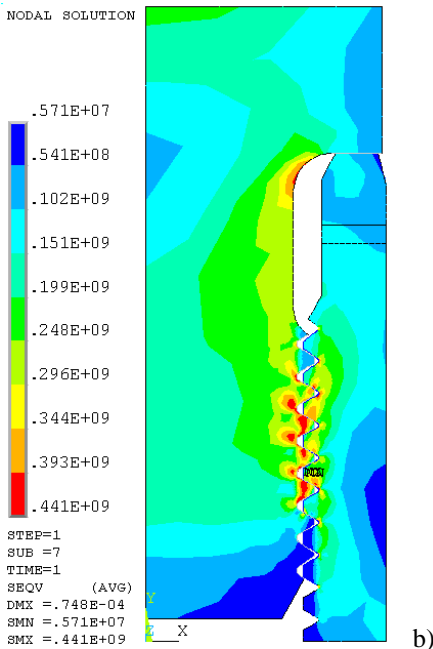
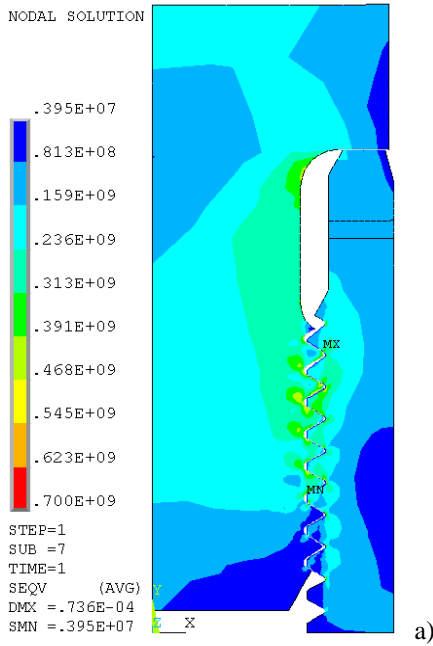
To identify the distribution of stresses in the sucker rod threaded connections the ANSYS finite-element computer axisymmetric model of worn standard connection for sucker rods with a diameter of 19 mm (GOST 13877–96) has been developed. Material of the connection was selected – 20N2M steel (GOST 4543–71) after tempering at a temperature of 860°C in oil and tempering at 180°C in air ( $E = 2.1 \cdot 10^{11}$  Pa,  $\nu = 0.28$ ,  $\sigma_y = 685$  MPa,  $\sigma_b = 880$  MPa). Threaded connection has been loaded by axial force and screwing torque.

The distribution of equivalent stresses (by Mises) in a worn connection of sucker rod with different values of coils overlapping in thread is shown in Fig. 9 and graphical relationship between the equivalent stresses (by Mises) in the root of pin thread in Fig. 10.

Fig. 9 shows that in the threaded joints with a maximum overlapping of coils arise limit stresses of 700 MPa. These stresses appear at the top of the first root of pin thread (Fig. 11), but do not affect the durability of the overall connection.

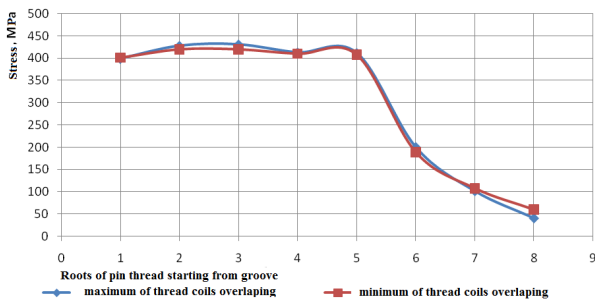
Graphical dependences (Fig. 10) imply that the distribution of equivalent stresses (by Mises) along the coils thread in threaded pin for the maximum and minimum overlapping coils is similar, the main difference is that values of stresses in connection with maximum overlapping coils are bigger.

Let's consider the distribution of axial stresses (Fig. 12).

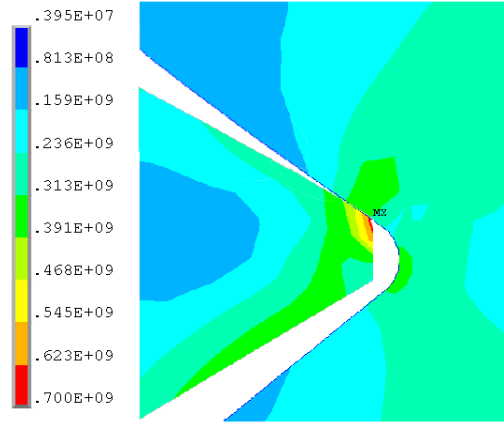


a - with a maximum overlap of turns,  
b - with a minimum overlap of turns

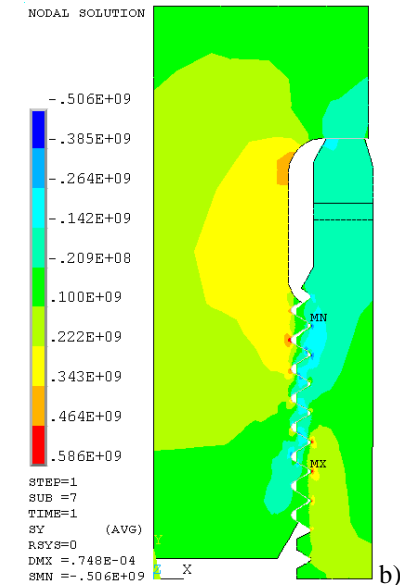
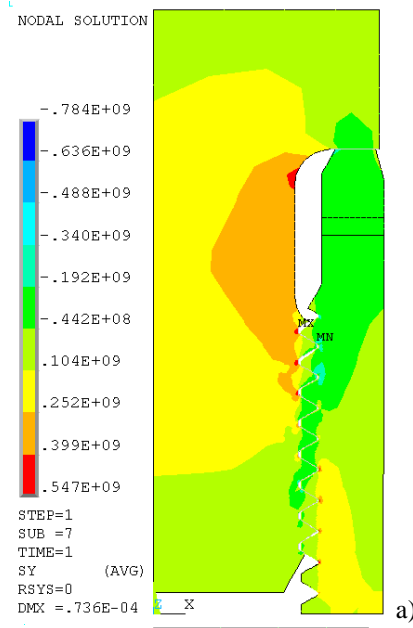
**Figure 9 – Equivalent stress (by Mises) in a connection of a worn sucker rod and a 19 mm coupling**



**Figure 10 – Distribution of equivalent stress (by Mises) in the roots of worn thread pin of sucker rod with a diameter of 19 mm**



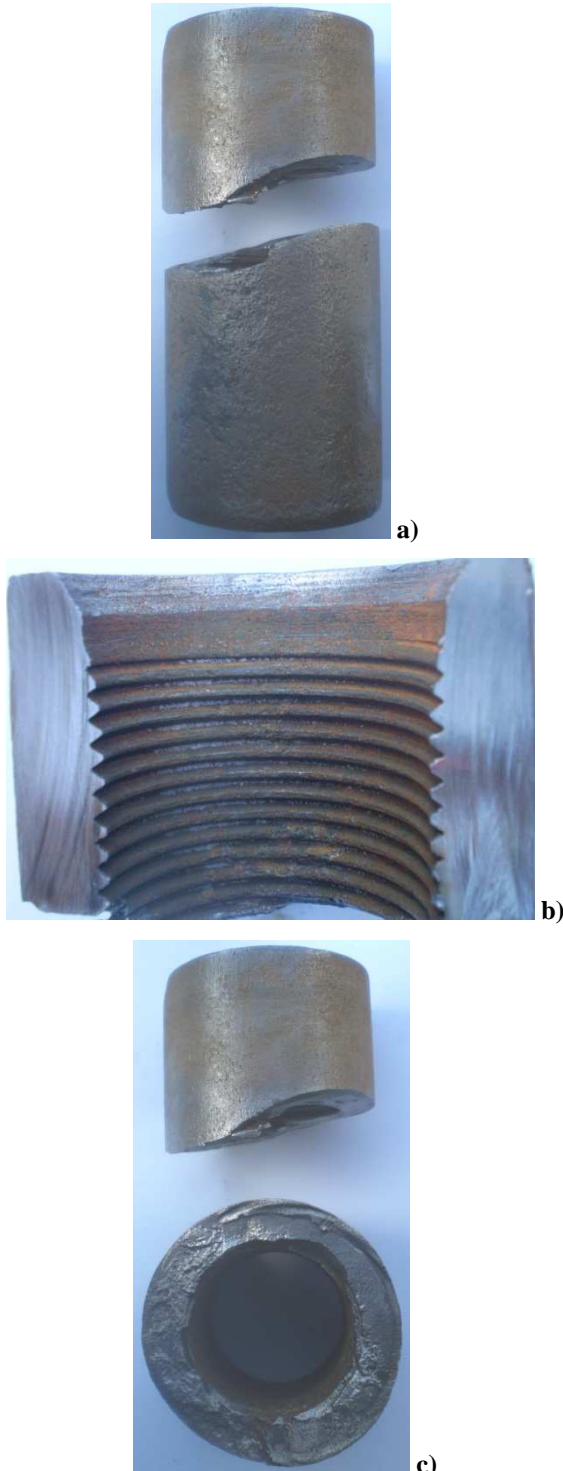
**Figure 11 – Equivalent stresses (by Mises) in the first coil of pin thread of worn sucker rod with a diameter of 19 mm**



a – with a maximum overlapping of coils;  
b – with a minimal overlapping of coils

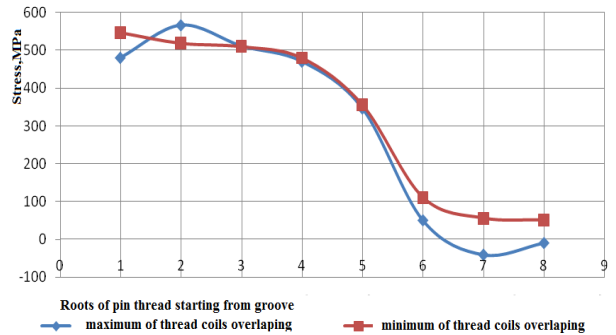
**Figure 12 – Axial stresses in the connection of rod and the coupling for a diameter of 19 mm**

Larger values of stresses are arising in a connection with a minimum overlap of coils and are concentrated in the seventh root of coupling coil thread. We can assume that this is the stress distribution that leads to the destruction of the couplings body while operation (Fig. 13).



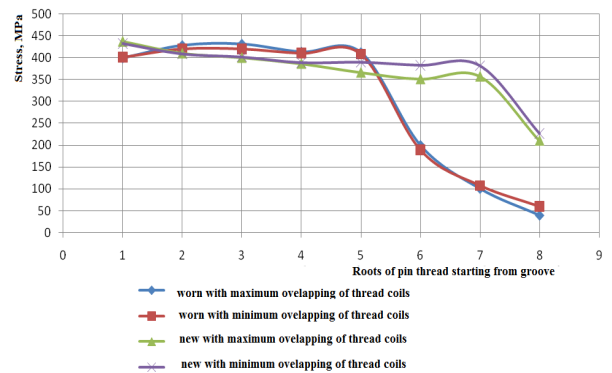
a – coupling (side view),  
 b – coupling (kind of fracture),  
 c – longitudinal section of the coupling  
**Figure 13 – Destruction of worn coupling thread**

According to Fig. 14 more uniform distribution of axial stress in the worn pin thread occurs in connection with a minimum overlap of coils, although in the last three roots the stress values are higher than in a connection with a maximum coils overlapping.

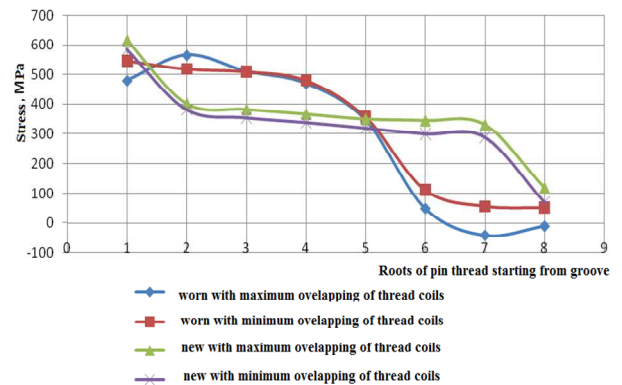


**Figure 14 – Distribution of axial stresses on the worn pin thread of sucker rod with a diameter of 19 mm taking into account the size of the overlap of coils thread**

A comparative analysis of the stress state of threaded connections with a worn and a new thread is made. Graphical dependencies of stresses distribution are shown in Fig. 15 and Fig. 16.



**Figure 15 – Distribution of equivalent stress (per Mises) along the root of coils threading of new and worn sucker rod of 19 mm in diameter**



**Figure 16 – Distribution of axial stress (per Mises) along the root of coils threading of new and worn sucker rod of 19 mm in diameter**

**Conclusion**

Having analyzed the results shown in Fig. 15 and 16 we made a conclusion that with the increase of sucker rod wear, stress distribution in pin thread becomes uniform, respectively, it has positive effect on the threaded connection durability under cyclic dynamic loads. With further wear of thread coils the stresses are rising in middle coils, and in the extreme – coils they are reduced, and therefore the impact of further deterioration on the durability of the connection is negative.

Also it has been found that in the worn threaded coupling the maximum stresses are concentrated in the root of coupling thread, which results in the destruction of its body.

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## **Дослідження напружено-деформованого стану різьбового з'єднання насосних штанг**

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Різьбові з'єднання є відповідальними вузлами, що визначають надійність конструкції колони насосних штанг. Незважаючи на різноманітність їх конструктивних виконань, аналіз промислових матеріалів свідчить, що поломки насосних штанг відбуваються як по тілу штанг, так і різьбових з'єднаннях.

У процесі спуско-підйомних робіт для заміни свердловинного насоса відбувається багаторазове згинчування і розгвинчування різьбових з'єднань насосних штанг, внаслідок чого зношуються різьби ніпелів та муфт.

На основі аналізу методики визначення величини зношування різьби ніпеля під час згинчування-розгвинчування та затягування і розкріплення з'єднання насосних штанг, запропонованої С.А. Бабаяном, встановлено, що величина зношування різьби під час згинчування в 20–30 разів перевищує зношування різьби під час затягування, а також, згідно з цією методикою, три останні витки різьби практично не зношуються, а зношування першого витка в 10–12 разів перевищує зношування останніх витків. Одержані результати розрахунку відповідають даним зношування в процесі експлуатації різьбових з'єднань насосних штанг у промислових умовах.

Для виявлення та порівняння розподілу напружень у різьбових з'єднаннях насосних штанг у середовищі ANSYS розроблено осесиметричні моделі нових і зношених стандартних різьбових з'єднань діаметром 19 мм. Різьбові з'єднання навантажувалися осью силою та моментом згинчування.

На основі результатів моделювання зроблено висновок, що зі збільшенням величини зношування різьбового з'єднання насосної штанги розподіл напружень по впадинах різьби ніпеля стає рівномірнішим, відповідно це позитивно впливає на довговічність роботи різьбового з'єднання за циклічних динамічних навантажень. За подальшого зношування різьби напруження зростають у середніх витках, а в крайніх – зменшуються, і вплив подальшого зношування на довговічність роботи з'єднання є негативним.

Також встановлено, що за граничного зношування різьбового з'єднання максимальні напруження концентруються у впадині витка різьби муфти, в результаті чого виникає руйнування її тіла.

Ключові слова: виток, зношування, метод скінченних елементів, напруження, насосна штанга, різьбове з'єднання.