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STUDIES AND RESEARCH ON THE DESIGN OF WORK SHAFTS FOR MUD PUMPS AND DRILLING DRAW WORKS

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Abstract: *This paper examines the subject of static analysis using modern computing software with the help of the finite element of important components of the equipment that make up any fixed or mobile drilling rig such as the drive shaft of a pump drilling rig (mud pumps), respectively of a drilling winch. The role of these shafts, also called main shafts, is to transfer/transform mechanical energy into hydraulic energy in the case of pumps or to lift using the operating drum and the hook the technological loads of the drilling rig, in the case of the drilling winch. Thus, two distinct cases were analyzed, but sensitive in terms of technical and economic performance.*

Keywords: *mud pump, draw works, finite element method, stress, strains.*

1. GENERAL CHARACTERISTICS OF MUD PUMPS

According to [2], mud pumps are the generator of hydraulic energy without which efficient, optimized drilling can no longer be conceived today. The aim is to develop a new drilling technology based on research conducted in rock dislocation laboratories, which led to the following practical conclusions (Table 1):

- The flow of drilling fluid for screeds with a diameter between 190 – 270 mm, is 30 – 35 l/s;
- The drilling fluid pressure at the nozzle outlet is 1000 – 1400 bar;
- The minimum hydraulic power required at the bottom of the well is 3700 kW.

The research concluded that the mud pumps must be able to achieve a pressure of 1400 bar, within the individual power limit of 1830 kW.

NATIONAL OILWELL VARCO features a full range of triplex mud pumps, called the P series, that meet all market requirements. This series includes mud pumps with drive powers from 373 kW to 1491 kW.

P – series mud pumps have the following characteristics:

- The structures are made of high strength steel;
-The arrangement of the shareholding is variable, either on one side or on both sides;
- Spray lubrication system (dual for pump 14 P 220);

The range of high power pumps that were manufactured in Romania had the following constructive and technical characteristics [1]:

- drive shaft speed 700 rpm;
- crankshaft speed between 130 – 150 rpm;
- V-toothed gear, improved with 10 mm front module.

The metal consumption index is 80 – 85 %, and the specific mass index is 13 – 15 kg/1HP.

Table 1 – Triple drilling pumps with simple effect

Characteristics	Mud pump type				
	3PN 320	3PN 700	3PN 1000	3PN 1300	3PN 1600
Power (CP)	320	700	1000	1300	1600
Pressure (bar)	200	250	350	350	550
Flow (l/min)	10,8	18,8	19	25	27
Maxim flow	32	42	32	32	52
Pressure	68	101	130	168	208

2. PUMP SHAFT MODELING

The modeling of the mud pump drive shaft is performed using the 3D graphic modeling and calculation program by the finite element method NX 1847 (Figure 1). This shaft is made

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of 42CrMo4 in SR EN 10269 being a low carbon alloy, low alloy. According to the material catalog library, the material yield strength is σ_c = 750 MPa. The pump shaft is machined in steps and its diameter varies in different sections. Step 1, identical for both ends, is for mounting the drive sprockets of the shaft, this being the section with the smallest shaft diameter Ø 172 mm. The wedge channel (2) was machined in this section, with the role of securing the pinion to the shaft during operation. Step 3, (Ø184 mm) in the area where the mounting sleeve for the inner ring of the shaft support bearing is mounted which is mounted on step 4 of the shaft with a diameter of Ø 192 mm. Step 5, with the diameter representing most of the body of the shaft, in this area is finding the "V" toothed pinion (6), which is part of the shaft body, acting as the transmission of torque and rotational motion to the crankshaft of the mud pump used to make the sections.

The method is a discretization process: the geometric shape and the fields of displacements, specific deformations, and stresses are described by discrete quantities (eg. coordinates) distributed throughout the structure.

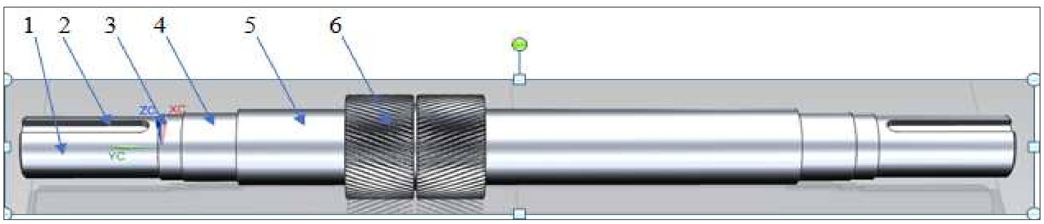

Figure 1 – Pump drive shaft model

3. DETERMINATION OF STRESS AND DEFORMATION

3.1 Estimated analytical calculation

The following product technical data (Table 2) [3] are known:

Table 2 – Drilling pump technical data 3PN – 1300

Size	Value
Drive power P [kW]	969
Drive shaft speed n [rpm]	700
The frontal module of the teeth m	4
Gear ratio u	4.5
Number of drivewheel teeth z1	60
The inclination angle of the teeth β°	30
Reference profile angle α n °	20
The wheelbase of [mm]	740

According to [2], in the theory T_τ (theory of maximum tangential stresses, Tresca) it is assumed that the destruction occurs by sliding on the planes on which τ_{max} acts. The shaft is twisted along its entire length with the forces in the xOy plane and the diagram M_z can be drawn.

It is a required torsion bending compound, the most required section of the shaft is when the M_{ech} moment has the highest value [4]:

$$M_{ech} = \sqrt{M_y^2 + M_z^2 + M_t^2} \quad (1)$$

The yield strength of the steel material (42CrMo4 in SR EN 10269) is σ_c = 750 MPa. Assuming a safety factor of 2.5 and the permissible material stress σ_a = 300 MPa, the corresponding value of the minimum diameter required for the drive shaft dmin = 84.36 mm.

It adopts d = 85 mm.

3.2 3PN 1300 pump drive shaft resizing

The total size of the drive shaft shall be resized, taking into account its initial proportions as follows:

Dimension 1, identical for both ends, is for mounting the drive shafts of the shaft, this being the section with the smallest shaft diameter Ø 85 mm.

Dimension 2, (Ø 97 mm) is the area where the mounting sleeve is mounted for the inner ring of the shaft support bearing which is mounted on step 3 of the shaft with a diameter of Ø 105 mm.

Dimension 4, with a diameter of Ø 212 mm representing most of the shaft body, in this area is found the pinion with "V" teeth (Figure 2).

3.3 Determining the state of stresses and strains. Estimated analytical calculation of gears

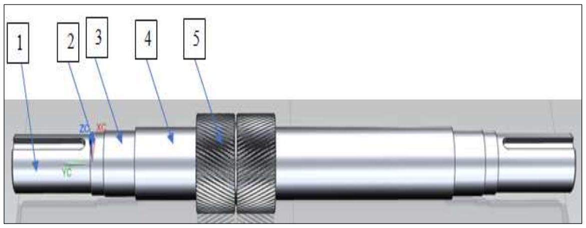

Figure 2 – Pump drive shaft recalculated

The following data is known:

Drive power $P = 1300$ hp $= 969$ kW;

Drive shaft speed $n = 700$ rpm;

The frontal modulus of the teeth $m = 4$;

Gear ratio $u = 4.5$;

Number of drivewheel teeth $z_1 = 60$ teeth;

Number of teeth driven wheel $z_2 = 270$ teeth;

Tooth tilt angle $\beta = 30°$;

The angle of the reference profile $\alpha = 20°$;

Wheelbase $a = 740$ mm.

According to in theory T_τ (theory of maximum tangential stresses) it is assumed that the destruction occurs by sliding on the planes on which τ_{max} acts.

It was originally formulated by Coulomb (1773) and in its current form by Tresca in 1865. Confirmed by the precise experiments performed by Guest in 1903, the statement of the theory is: „from that point it reaches a limit value τ_k, regardless of the type of request".

Knowing the maximum tangential stresses results:

$$\tau_1 = \frac{\sigma_2 - \sigma_3}{2};\ \tau_2 = \frac{\sigma_1 - \sigma_3}{2};\ \tau_3 = \frac{\sigma_1 - \sigma_2}{2} \quad (2)$$

And the limit value from the extent τ_k:

$$\tau_k = \frac{\sigma_k}{2} \quad (3)$$

In order not to exceed the limit state, the conditions must be met:

$$\begin{aligned} -\tau_k &\le \tau_3 \le \tau_k; \\ -\tau_k &\le \tau_2 \le \tau_k; \\ -\tau_k &\le \tau_1 \le \tau_k \end{aligned} \quad (4)$$

For actual situations where the value σa must not be exceeded, the resistance conditions after T_τ are expressed by the relation:

$$\sigma_{ech} = \max\ [|\ \sigma_1\text{-}\sigma_2\ |, |\ \sigma_1\text{-}\sigma_3|, |\ \sigma_2\text{-}\sigma_3\ |] \le \sigma_a \quad (5)$$

For a flat state of stress, the resistance condition becomes:

$$\sigma_{ech} = \max\ [|\ \sigma_1\text{-}\sigma_2\ |, |\ \sigma_1\ |, |\ \sigma_2\ |] \le \sigma_a \quad (6)$$

$$\sigma_{ech} = \sqrt{}\ (\sigma^2 + 4\sigma^2) \quad (7)$$

$$\sigma = \sigma_{max} = |M_i\,| / W_i \quad (8)$$

$$\tau = \tau_{max} = |M_t| / W_p \quad (9)$$

where:

W_i – bending strength module;

W_p – polar resistance module.

$$W_i = [\![\pi \cdot D]\!]^{3/32} \quad (10)$$

$$W_t = [\![\pi \cdot D]\!]^{3/16} \quad (11)$$

The estimation of the forces acting perpendicular to the shaft can be done by following the following steps:

Calculate the torque Mt acting on the drive shaft, using the relation:

$$M_t = 9.55 \cdot [\![10]\!]^6\ P/n \quad (12)$$

$$M_t = 9.55 \cdot [\![10]\!]^6\ 969/700 = 132.1992 \cdot [\![10]\!]^5\ \text{[Nmm]} \quad (13)$$

Cylindrical wheels have "V" teeth, they can be considered as coming from the juxtaposition of two identical cylindrical wheels with inclined teeth, but with different inclination directions of the teeth, resulting in equal axial forces and opposite direction, their result being zero, $F_a = 0$.

According to the gears are the machine parts that have the role of taking over and transmitting the rotational movement using the teeth arranged on their active surface.

The gear is the mechanism consisting of two gears, which transmits – through the teeth that are successively and continuously in contact (gearing) – the rotational motion and the torque between the two shafts. Gears are forces used in mechanical transmissions, due to their advantages: constant transmission ratio; safety in exploitation; high durability, high efficiency, small size, usability for a wide range of powers, speeds,d transmission ratios.

The disadvantages they present are high precision of execution and assembly.

Complicated manufacturing technology, noise, and vibration in operation.

Driving wheel gear – driven wheel consists of gears characterized by the numbers of teeth z_1 and z_2, the coefficients of displacement of profiles x_1 and x_2, respectively, and the same mode m on the dividing circles.

The materials used for the construction of the gears must take into account several factors: the way the semi-finished product is obtained, the quality – price ratio, the operating conditions, and the mechanical characteristics of the material. In general, the most used materials are steels, among which are quality carbon steels for cementation and improvement, alloy steels for machine building, carbon steel cast in parts, and alloy steel cast in parts. Depending on their mechanical properties and machinability, they are divided into mild steels (surface hardness < 350 HB) and hard steels (surface hardness> 350 HB).

Calculate the splitting diameters of the gears in gear:

$$d_1 = [\![m \cdot z]\!]\ 1/\cos\beta \quad (14)$$

$$d_2 = [\![m \cdot z]\!]\ 2/\cos\beta \quad (15)$$

$$d_1 = (4 \cdot 60)/(\cos 30°) = 277.1281\ [mm] \quad (16)$$

$$d_2 = (4 \cdot 270)/(\cos 30°) = 1247.0765\ [mm] \quad (17)$$

The tangential force Ft is given by the relation:

$$F_{t1} = (2 \cdot M_{t1})/d_1 \quad (18)$$

where:

M_{t1} – maximum torque at the pinion;

$$M_{ech} = \sqrt{([\![11744.78]\!]^2 + [\![13219.92]\!]^2)} = 1768\ [Nm] \quad (19)$$

Resistance condition:

$$\sigma_{ech} = M_{ech} / W_i = \sigma_a \quad (20)$$

The yield strength of the material is σc = 750 MPa.

Assuming a safety factor of 2.5, the allowable stress of the material is σ_a = 300 MPa.

From the condition of resistance:

$$W_{i\ nec} = M_{ech} / \sigma_a = (17683.5 \cdot [\![10]\!]^3)/300 = 58945\ [\![mm]\!]^3 \quad (21)$$

$$W_{i\ nec} = Wi_{\ dim} = [\![\pi \cdot d]\!]^3/3 \quad (22)$$

$$\Rightarrow d = \sqrt[3]{((32 \cdot 58945)/\pi)} = 84.36\ mm \quad (23)$$

It is adopted d = 85 mm, the minimum diameter required for the drive shaft. The total size of the drive shaft is resized, taking into account its initial proportions as follows: Step 1, identical for both ends, is for mounting the drive sprockets of the shaft, this being the section with the smallest diameter of the shaft Ø85 mm. Step 2, (Ø 97 mm) is the area where the mounting sleeve is mounted for the inner ring of the shaft support bearing which is mounted on step 3 of the shaft with a diameter of Ø 105 mm. Step 4, with a diameter of Ø 212 mm, represents most of the shaft body, on this area is the pinion with "V" teeth (5) (Figure 2).

$$F_{t1} = (2 \cdot 132{,}1992 \cdot [\![10]\!]^5)/277{,}1281 = 95406{,}5646\ [N] = 95{,}4065\ [kN] \quad (24)$$

Calculate the radial force given by the relation:

$$F_{r1} = (F_{t1} \cdot tg\alpha n)/\cos\beta \quad (25)$$

$$F_{r1} = (95.4065 \cdot tg20°)/(\cos 30°) = 40.0971\ [kN] \quad (26)$$

According to the principle of action and reaction we obtain:

$$F_{t1} = F_{t2},\ F_{r1} = F_{r2} \quad (27)$$

Sizing of the mud pump drive shaft:

According to the calculation relation (25), the torque applied to the ends of the drive shaft Mt = $132{,}1992 \cdot [\![10]\!]^5$ [Nmm]. The shaft is twisted along its entire length with the forces in the xOy plane (Figure 3).

3.4 Finite element analysis (A.E.F)

The method is a discretization process: the geometric shape and the fields of displacements specific deformations and stresses are described by discrete quantities (eg. coordinates) distributed throughout the structure. This requires a matrix notation. Tools are numerical computers, capable of storing long lists of numbers and processing them [5].

The object of the A.E.F. is the replacement of the system with infinite degrees of freedom encountered in continuum applications by a finite system that has the same basis in a discrete analysis (Figure 3a,b).

Computers are able not only to solve discretized equilibrium equations, but also to help formulate equations, by making decisions about refining the discretization network, and assembling stiffness matrices. But most importantly, the finite element method can be applied to systems with complex geometry and complicated parameter distributions, (Figure 4).

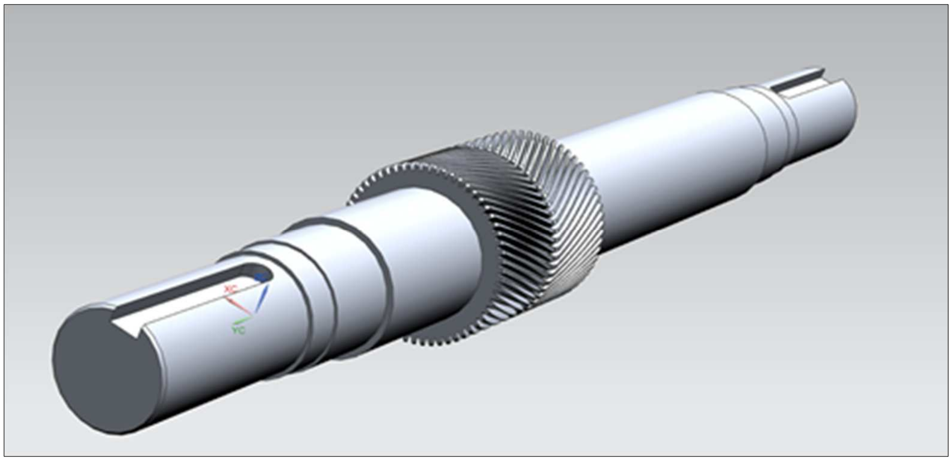

Figure 3a – Pump drive shaft model recalculated

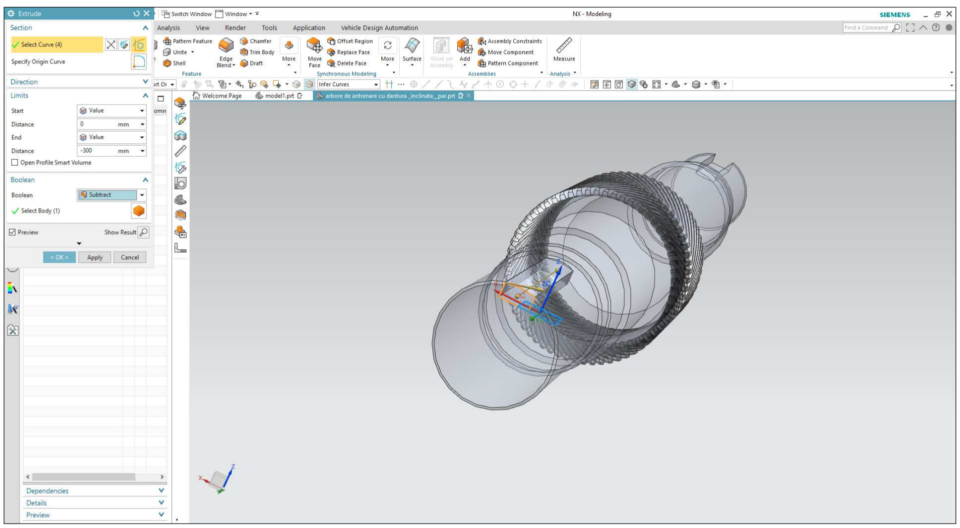

Figure 3b – Pump axis drive shaft model

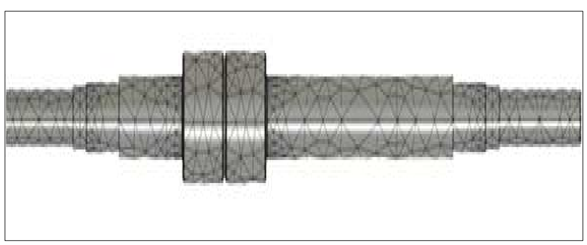

Figure 4 – Automatically discretized pump drive shaft model, recalculated

4. INTERPRETATION OF RESULTS

As stated in paragraph 3.2, the shaft has been completely resized to reduce costs and production time. After resizing we performed a simulation of the in-operation requests for the new size tree to have a more concrete picture of how it will behave under the effect of in-service requests. Figure 5 shows the stress map obtained from the simulation where it can be seen that the maximum stress reached as a result of operating stress is σ_{max} = 180.2 MPa. The resulting conclusion is that the sizing was done correctly from the point of view of the requests.

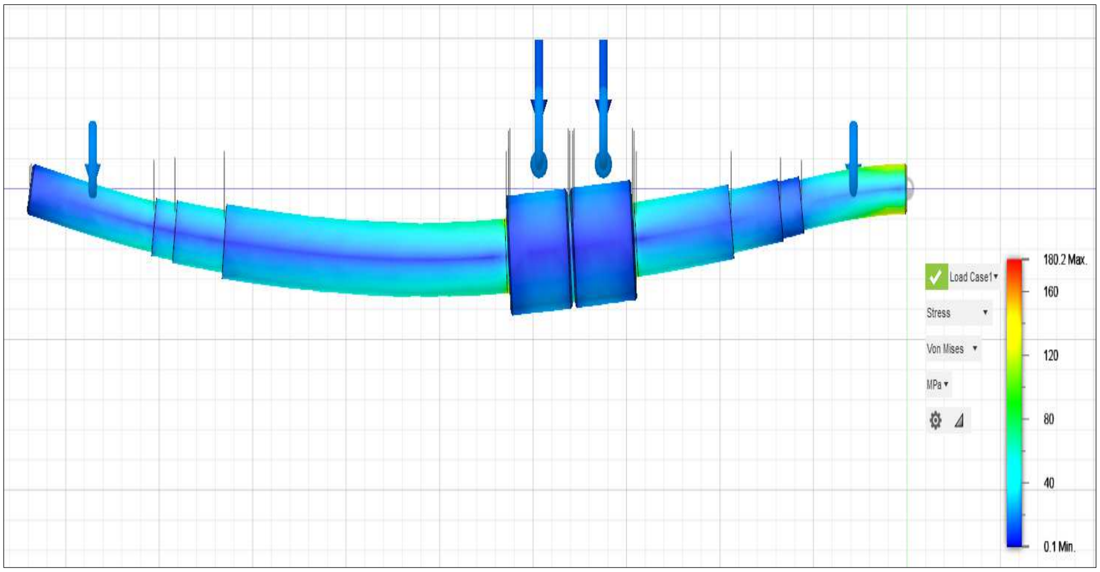

Figure 5 – Resized drive shaft tension map subjected to test load

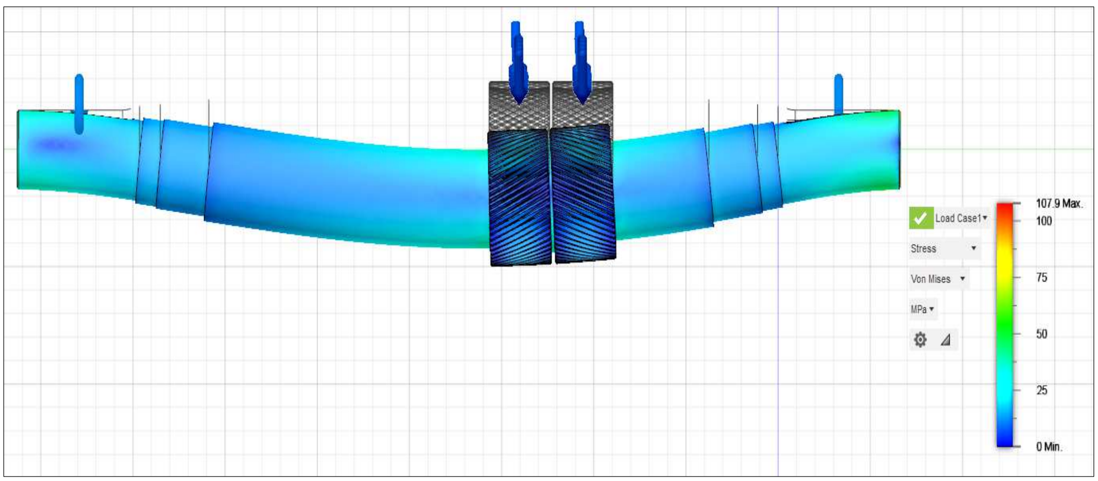

Figure 6 – Tension map of the drive shaft Subjected to the test load

Comparing the stress maps in (Figures 5 and Figure 6), it is observed that the maximum allowable stresses on the resized shaft reach a value almost double compared to the stresses that appear in the drive shaft with the initial dimensions respectively σmax = 107.9 MPa (Figure 7 and Figure 8).

5. DRAW WORKS

The drilling DRAW WORKS (TF) ensures the proper winding and placement of the cable, with the help of the operating drum (TM) with the secure and accessible fixing of the active end of the cable and the brake drums shaft assembly (Figure 9 and Figure 10).

1 – Water and air supply and exhaust head;
2 – Heated water drain hole from the drum chambers
3 – Bellows AVB;
4 – AVB driven half–coupling connection flange;
6 – Ring box;
7 – Chain wheel driven by "slow" and "fast" transmission, respectively;
8 – Wall;
9 – Oscillating roller bearing barrels on two rows;
10 – Drum cooling chamber;
11 – Belt brake drums (left and right);
12 – The connecting pipe between the chambers of the two drums;
13 – Bearing box;
14 – Spiral sleeve mounted on TM.

5.1 Maneuvering drum

The operating drum will be made of welded, cast, or combined construction. It is considered a combined construction. The actuating drum shall be made of a material of type OT 50 checked at bending = 240 kN/m.

5.2 Maneuvering drum shaft calculation

The shaft will be dimensioned for the maximum stresses (maximum effort from the active end of the cable (FM) so the transmission is done on the "slow" side) (Figure 11).

It is recommended that the position of the cable on the drum for which the calculations are performed by the opposite end of the active transmission ("slow" transmission).

6. CONCLUSIONS AND PROPOSALS

The companies producing oil equipment and spare parts aim at implementing advanced technologies for the permanent development of these products. The aim is to make machines with a compact design that will work at high pressures for as long as possible. Another trend is the interchangeability between the mud pumps, respectively the winches made by different manufacturers, which consists of the use of common spare parts and the performance of the service by a single company at several types of pumps. The fierce competition in the market makes established companies such as NOV, Weatherford, and EWECO manufacturers eager to assert themselves in this field.

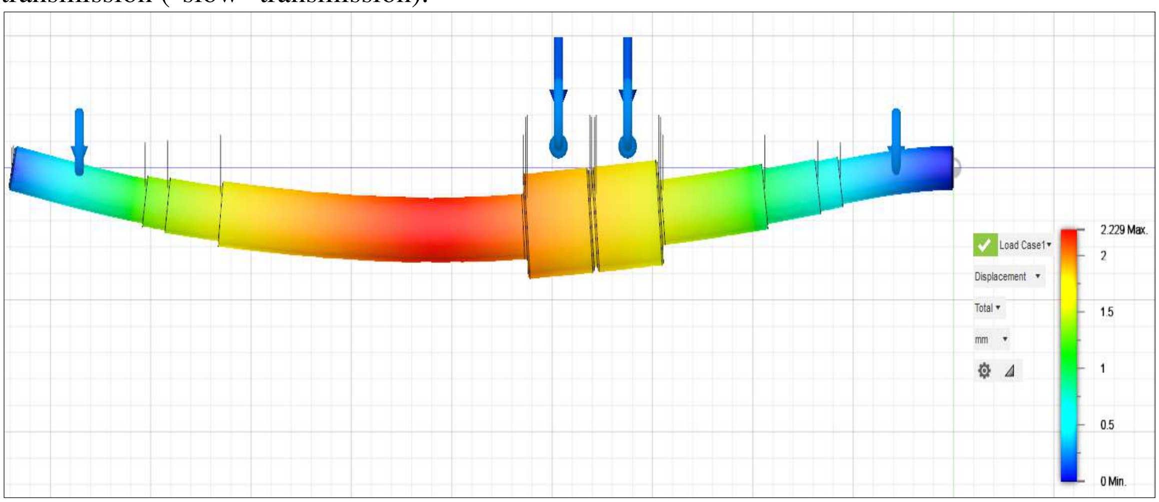

Figure 7 – Deformed drive shaft deformation map subjected to test load

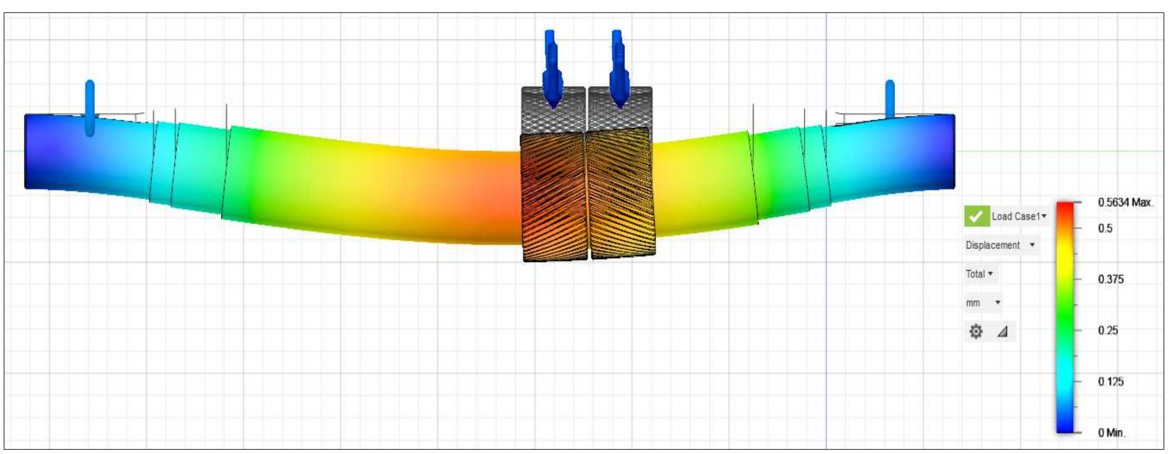

Figure 8 – Deformation map of the drive shaft subjected to the test load

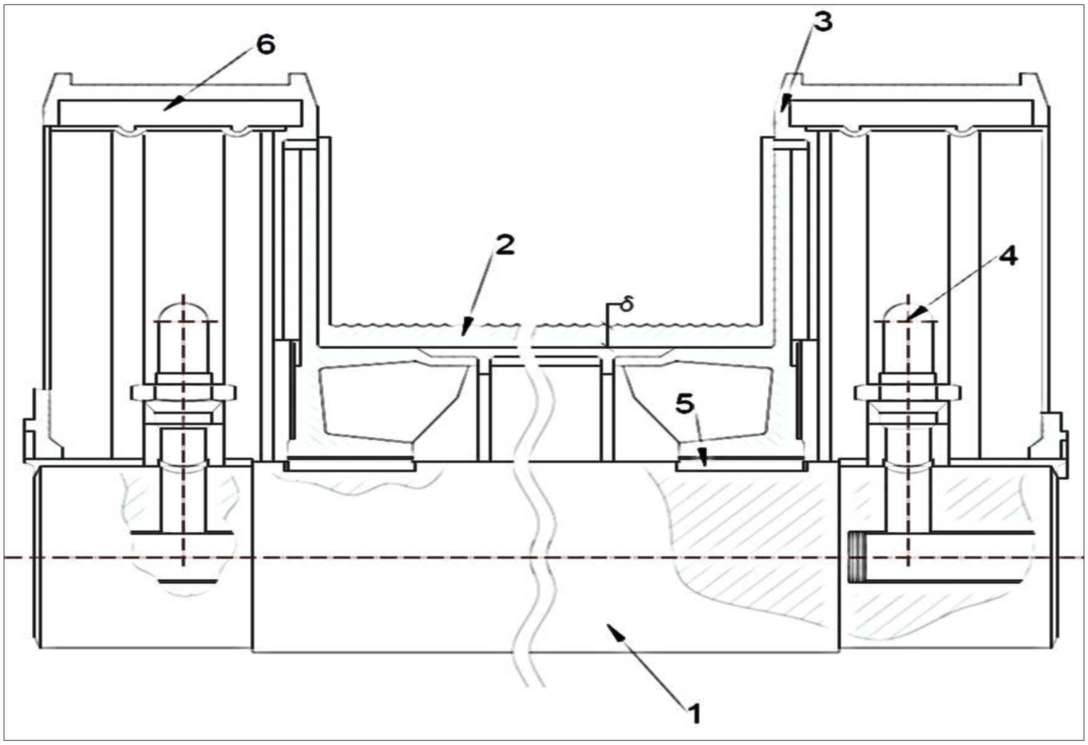

Figure 9 – Maneuvering drum
1 – The drum shaft; 2 – The actual drum; 3 – Brake drum; 4 – Sealing rings;
5 – Parallel assembly wedges; 6 – Drum cooling chamber.

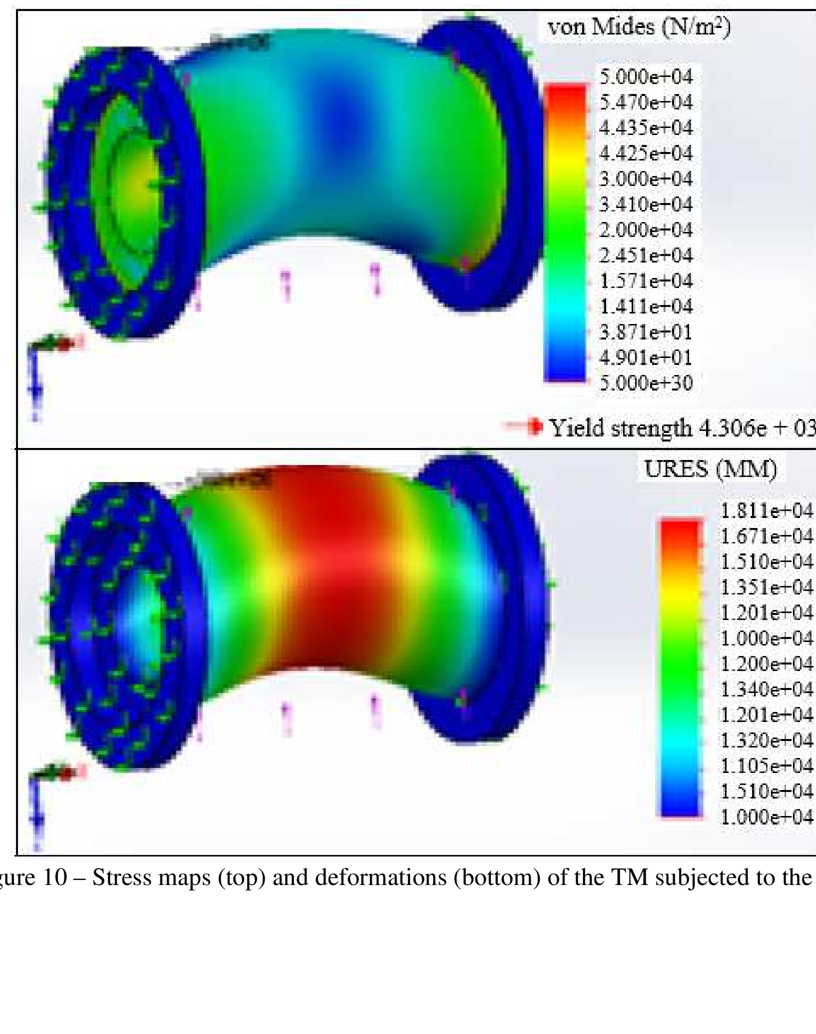

Figure 10 – Stress maps (top) and deformations (bottom) of the TM subjected to the test load

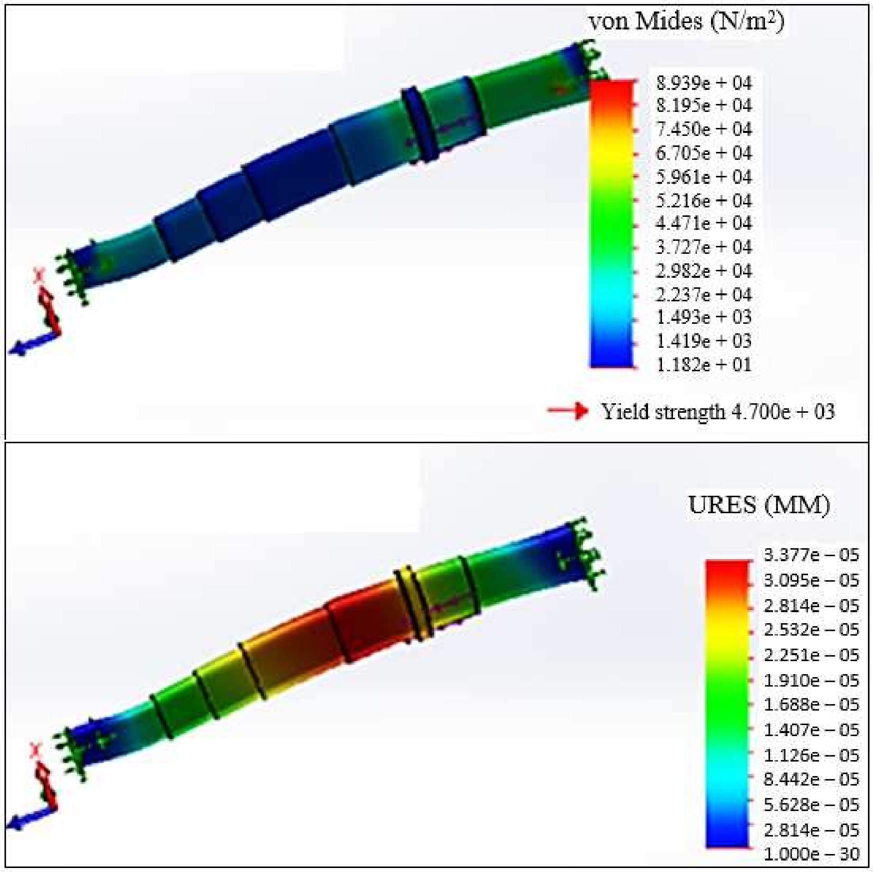

Figure 11 – Stress maps (top) and deformations (bottom) of the AT subjected to the test load

The first strong point of this paper proposes the redesign of both the drive shaft of a 3PN – 1300 single–acting triplex mud pump and the drive shaft of the TF 40 drilling winch, which is part of the structure of transportable drilling rigs.

The second strong point is that the sizing calculation of the two drive shafts was performed taking into account the specific deformations of rotations and arrows (maximum displacements) with an important role in maintaining the durability of the bearings, depending on their type and working regimes. the two distinct cases. In this sense, the mud pump drive shaft was resized by calculation using MEF, following these calculations it was concluded that the size of the drive shaft can be considerably reduced, reducing the material consumption required without affecting its behavior under operating conditions. After performing the analytical calculation, a calculation was performed by the finite element method, and the results obtained on the tree with the new dimensions were compared with the results obtained on the initial tree. The differences in stresses and strains in the body of the shaft are relatively small and within acceptable limits.

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Studii și cercetări privind proiectarea arborelor de lucru pentru pompe de noroi și trolii de foraj

Abstract: În prezenta lucrare este supusă spre studiu o temă ce vizează analiza din punct de vedere static folosind programe moderne de calcul cu ajutorul elementului finit a unor componente importante ale utilajelor care alcatuiesc oricre instalatie de foraj fixa sau mobila cum ar fi arborele de antrenare al unei pompe de foraj (pompe de noroi), respective al unui troliu de foraj. Rolul acestor arbori, numiti si arbori principali, este Acela de a transfera/transforma energia mecanică în energie hidraulică in cazul pompelor, sau acela de a ridica prin intermediul tobei de manevra si al carligului sarcinile tehnologice ale instalatiei de foraj, in cazul troliului de foraj. Astfel au fost analizate doua cazuri distincte , dar sensibile din punct de vedere al performantelor tehnico-economice.

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