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Cumulative Effects of the Phenomena Accompanying the Big Diameter Drilling Process on the Damage over Time of the Drill Pipes

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Abstract

Drill pipes used for large diameter drilling are damaged over time, occurring cracking and breaking, due to the cumulative effects of certain physical phenomena. These phenomena are specifically for the drilling process of big diameter, with reverse circulation of the drilling mud, with transitory phases, with big mass in rotary motion, with big weight-on-bit, and with low rotative speed of the drill bit. As a result, appear important dynamic loadings, a state of complex and variable stress during a complete rotation, and torsion and flexural vibrations of the drill string, which may have pulsations in the field of resonance. These phenomena are due to the existence of stress concentrators, which have constructive and metallurgical nature. The stress concentrators are determined by the construction and manufacture of the drill pipes. In this paper, are studied cumulative effects resulting from these phenomena, on the basis of theoretical and experimental researches.

Key words: *drill pipes, dynamic and stationary variable loadings, oscillations, cracking and breaking*

Introduction

During operation of the drill strings of 10¾" and 13⅜" or 14⅜", used to drilling of the mine shafts with a diameter of 3.620 m and 4.978 m, by means of drilling rigs F320-3DH-M and, respectively, FM400-4DH, cracks and fractures of drill pipes have been ascertained. This has led to the development of theoretical and experimental studies, in order to clarify the causes that have led to the degradation of the drill pipes. In this framework lies the currently paper, where are studied phenomena accompanying the process of large diameter drilling and their effects on durability of the drill pipes.

Rotary systems of the drilling rigs mentioned above are composed of groups of electro-hydrostatic driving with hydrostatic motors (HsM), rotary table (RT) (with cylindrical gear reducer), master bushing, kelly drive, kelly (K), swivel (Sw), drill-pipe string and bottom assembly, consisting of special drill collar (DC), stabilizer with rollers (StRo) and drill bit (B). The drill-pipe string consists of drill pipes with air tubes mounted on the top, inside or outside the drill pipes, and drill pipes without air tubes. Air tubes are necessary for the introduction of air compressed to the inside of the drill-pipe string, to a specific depth, for lifting the drilling mud, because reverse circulation is used.

Large-diameter Drilling Process and Phenomena Accompanying Him

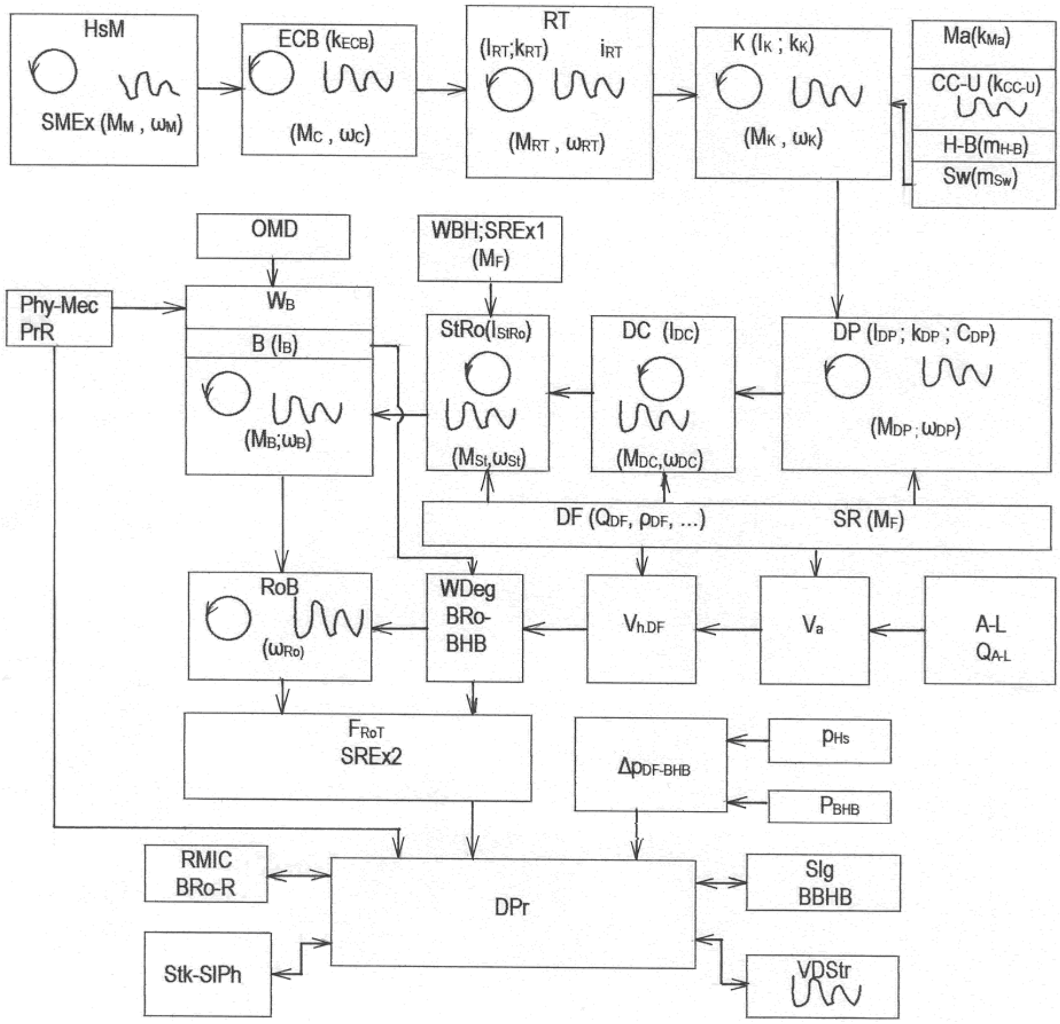


Fig. 1. The block-scheme of the determinant factors in case of large-diameter drilling process and phenomena which accompanying and influencing him: HsM – hydrostatic motors; SMEx – source of motive excitation; M – rotation moment; ω – angular speed; ECB – elastic coupling with bolts; k – elasticity constant; RT – rotary table; I – mass moment of inertia; i_{RT} – transmission ratio of the rotary table; K – kelly; Ma – mast; CC-U – cable coiling-up between crown-block and hook-block; H-B – hook-block; m – mass; Sw – swivel; DP – drill pipes; C – viscous damping coefficient; DC – drill collars; StRo – stabilizer with rollers; WBH – wall of the borehole; SREx – source of reaction excitation; DF – drilling fluid; SR – soft reaction; M_F – friction moment; B – drill bit; W_B – weight-on-bit; Phy-MecPrR – physical-mechanical properties of the rocks; OMD – operating mode of the driller; RoB – rollers of the bit; WDegRo-BHB – washing degree of the bit rollers and the borehole bottom; A-L – airlift; $Q_{A\text{-}L}$ – flow of the airlift; v_a – ascension velocity; $v_{h.DF}$ – horizontal velocity of the drilling fluid; F_{RoT} – impact forces of the bit roller teeth on the rock; $\Delta p_{DF\text{-}BHB}$ – differential pressure between the drilling fluid and the borehole bottom; DPr – drilling process; RMICBRo-R – rotary movement with intermittent contact between the bit rollers and rock; Stk-SlPh – stick-slip phenomenon; SlgBBHB – sloughing of the bit and the borehole bottom; VDStr – vibrations of the drill string

The rotary system is a dynamic system, having a certain structure, with fundamental properties – inertia (I), elastic (k) and dissipative properties –, on which are manifest motive actions and reactions determining its non-steady-state motion, characterized by a variable angular speed (ω) (see fig. 1). The motive action is carried out by hydrostatic motors (HsM), the main reaction proceeds from the bottom hole rock contacting the drill bit, but the other reactions proceed from the wall of the borehole (WBH), which comes in contact with the stabilizer rollers, and from drilling mud running round the drill string. Therefore, three sources of dynamic excitation are noticeable producing and influencing the motion of the rotary system: the source of motive excitation (SMEx), represented by hydrostatic motors, and two sources of reaction excitation (SREx) or „hard” reactions due to the contact between the stabilizer rollers and the shaft wall (SREx1), and between the drill bit rollers and the rock (SREx2). SREx2 has a random character determined by the variation of the rock physical-mechanical properties. The reaction of the drilling mud on the drill string, which is subjected to a rotary motion, is characterized as a „soft” reaction (SR), with a dissipative effect but not exciting one.

Being agreed with this conception of entirety „rotary system-rock-drilling mud-wall of the borehole” („RS-R-DF-WBH”), the determining factors of the large-diameter drilling process and the phenomena which accompanying him and influencing him are shown in Figure 1.

Due to the large diameter of drilling pipes and, especially, of the component elements of depth ensemble, the moment of inertia is far greater than in the case of normal drilling for oil and gas wells. It is also necessary to a bigger weight-on-bit, of 20 ÷ 250 kN, but a small drive speed of the drill bit, of 5 ÷ 20 rpm. Existence of reverse circulation and of large-diameter drill bit creates problems regarding dislocation and crumbling process of rocks and washing process of bit rollers and borehole bottom, with negative effects on the progress of the drilling process and on the dynamics of drill string and the entire system of rotation.

Due to the fundamental dynamic properties of the entire system of rotation, to the elasticity (k_{CC-U}) of the cable coiling-up (CC-U) between the crown-block and hook-block assembly (H-B), to the existence of the dynamic excitation sources, to the technical and technological peculiarities of the drilling of large diameter, compared to the normal size, and due to changes in mechanical parameters of drilling (W_B, M_B and ω_B), the drilling process of mine-shafts is, by excellence, a dynamic and vibrator process. He is accompanied, and its effectiveness is affected by phenomena such as: sloughing of the drill bit and the borehole bottom (SlBBHB), momentary blockings of the drill bit, shocks and torsion, axial and bending vibrations (overlaid to dynamic actions of the same type) of the drill string (VDStr), the motion of rotation with intermittent contact between the drill bit rollers and the rock (RMICBRo-R) and stick-slip phenomenon (Stk-SlPh) of the drill bit. Their harmful effects on the bit rollers, drilling pipes and hydrostatic transmission were recorded in time and have imposed performing some theoretical and experimental studies [1].

Stress Concentrators in the Drill Pipes and the Cracking and Breaking Zones

The drill pipes, of 10¾", 13⅜" and 14⅜", used for drilling of large diameter, have a construction adapted to reverse circulation of the drilling mud and to big torsion moment. They are made of casing pipes, with nominal diameter specified, that joins, from both ends, through taper threads, of 1:8, with sleeves in the body and flanges with pins (see fig. 2), with threaded flanges on the inside and with additional weld between flange and casing pipe body (see fig. 3), or with threaded flanges on the inside and with crenellated collar, fixed outer flange neck through the butt weld, and assembled with the casing pipe body through the corner welds on the inside of the crenels (according to figs. 4 and 5).

It is found that both the threaded areas of the flanges with pin, and areas with additional welds and terminal areas of the side weld between the supporting collar and the casing pipe body are areas of stress concentrators of constructive and/or metallurgical nature, where early cracks and breakings of the drill pipes have occurred, due to the phenomenon of fatigue produced by axial, torsion and, especially, bending variable loadings, and by shocks/dynamic torsion loadings during the transitional phases from starting the rotary table and during drilling, in case of momentary blockings of the drill bit and of stick-slip motion of it.

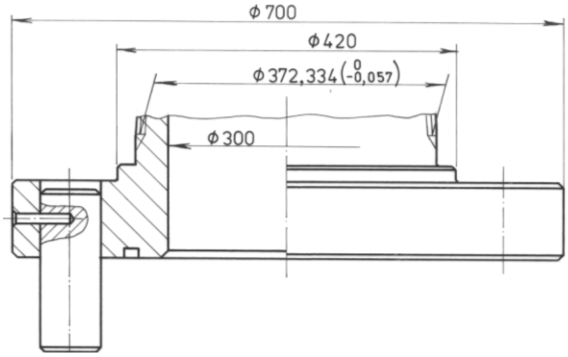


Fig. 2. Flange with pin broken in the sleeves of the Y-26 and Y- 48 drilll pipes, of 14⅜ in, in the area of the last two threads (Baraolt, FM400-4DH)

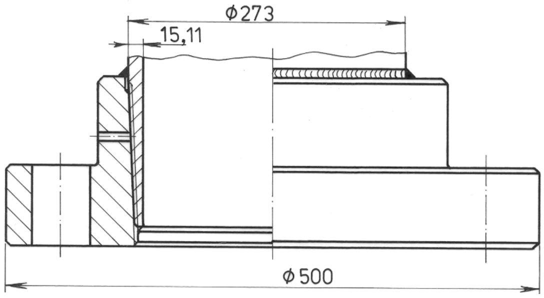


Fig. 3. Flange with broken pin of the drill pipe 4-85, of 10¾ in, during the first mine shaft drilling in Buştenari, by means of the drilling rig F320-3DH-M

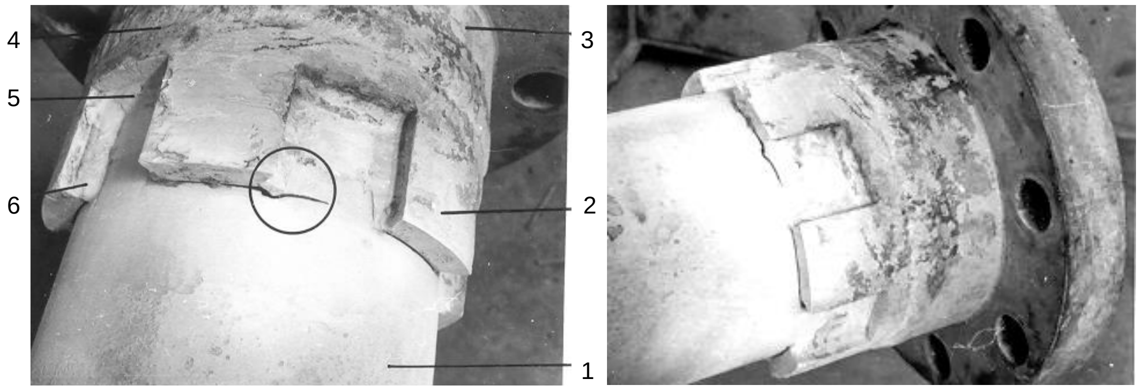


Fig. 4. Drill pipe of 10¾ in, of series 122: sight of the zone with penetrated macro-crack; 1− drill pipe body; 2 − crenelated collar; 3 − flange; 4 − frontal weld between the flange throat and the crenelated collar; 5 − additional weld from the internal base of the crenel; 6 − lateral corner weld from the crenel base

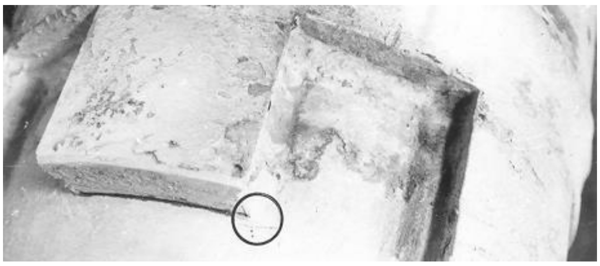


Fig. 5. Drill pipe of 10¾ in, of series 122: sight of the area with surface fissure

Theoretical Studies

There were studied the free torsion, flexural and longitudinal vibrations of the drill string of 10¾", considering the different models of this drill string. It was shown that the physical model simplification results in larger measures of natural pulsation, therefore disadvantageous pulsation in terms of the true dynamic behavior of the system studied.

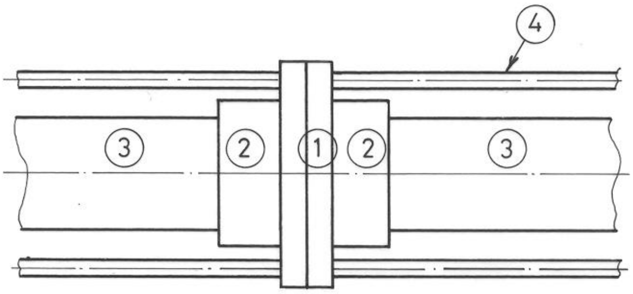


Fig. 6. Connecting zone between two drill pipes, with the four elements with distinct geometrical characteristics: 1 – flange disk; 2 – flange neck and collar; 3 – drill pipe body; 4 – air tube

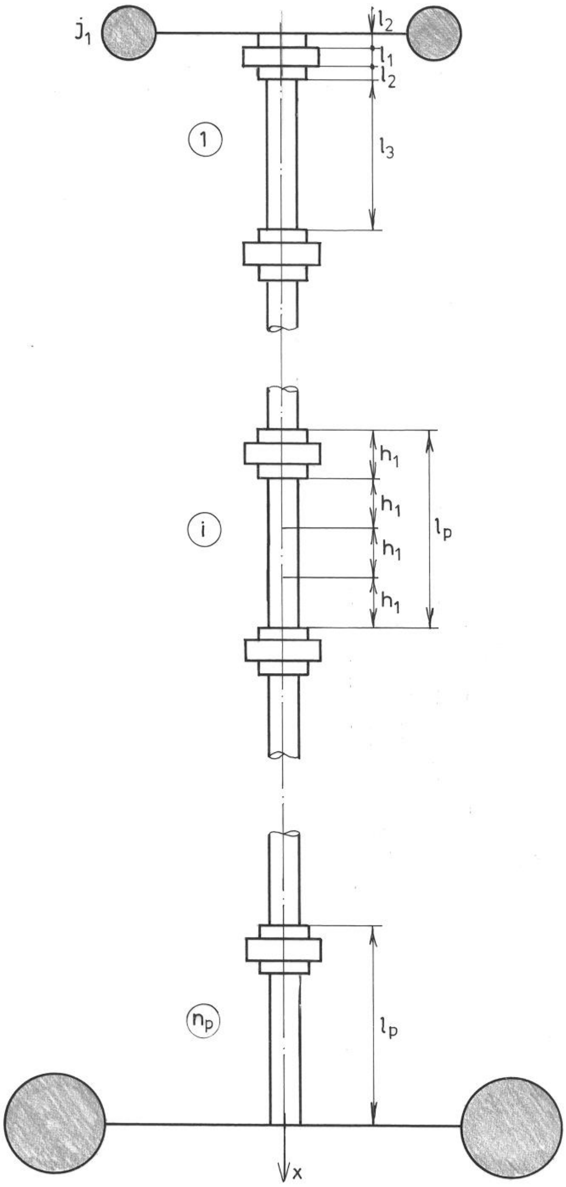


Fig. 7. Graphical model of the 10¾" drill string with non-uniformly distributed mass and mass moments of inertia concentrated at the ends

Therefore, it is necessary to take into account the structural discontinuity of the drill pipe (see figs. 4 and 6), represented by joint of the drill pipes with flanges, centering bolts and fastening screws, by computing a equivalent polar moment of inertia of cross-section and corresponding unitary torsion elastic constant. In the case of torsional vibrations, we must take into consideration the hydraulic medium elasticity of hydrostatic transmission.

For free torsion vibrations of the drill string, we considered the following physical models [2]: model with uniformly distributed mass along the length of the drill string, mass moments of inertia concentrated at the ends and free ends, model with non-uniformly distributed mass and mass moments of inertia concentrated at the ends and free ends (fig. 7), and drill string model (mass evenly distributed along the length of the drill string, mass moments of inertia concentrated at the ends and free ends) integrated into the rotation system to take into account the elasticity of the hydraulic medium in hydrostatic transmission and the concentrated mass of the drive electric motors and hydrostatic transmission reduced at the kelly (fig. 8).

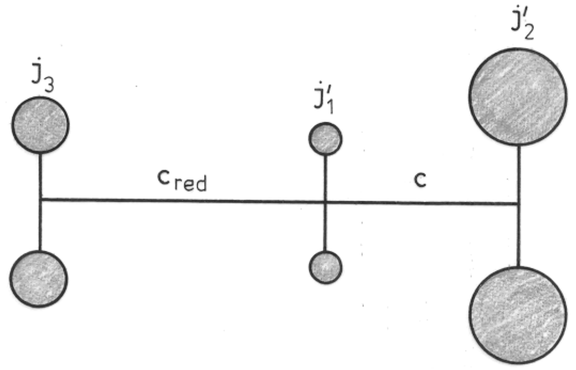


Fig. 8. Rotary system model of the F320-3DH-M drilling rig having three concentrated masses and two elastic bonds

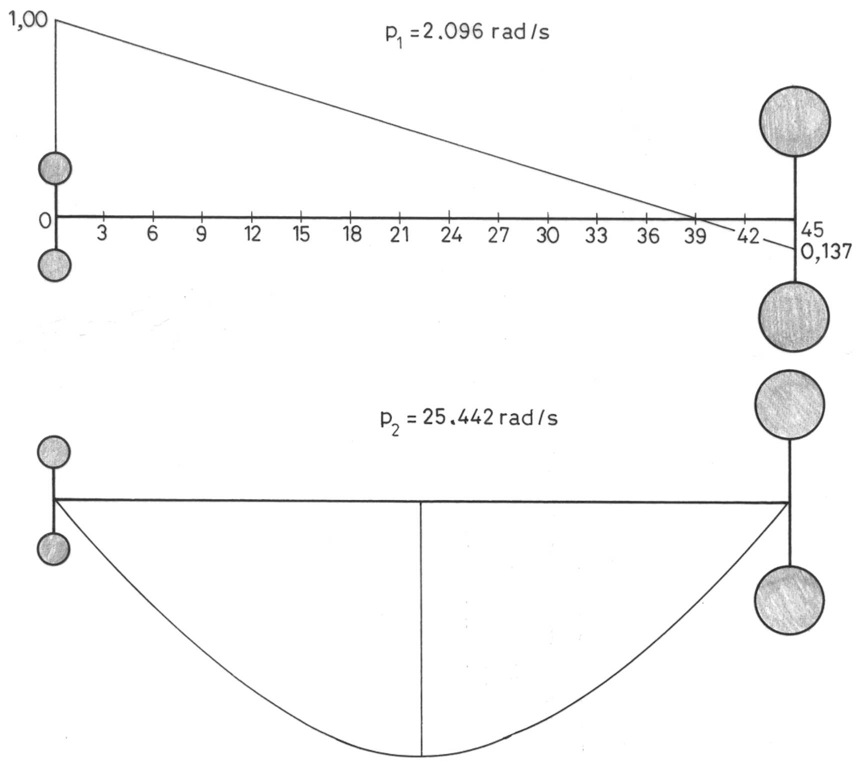
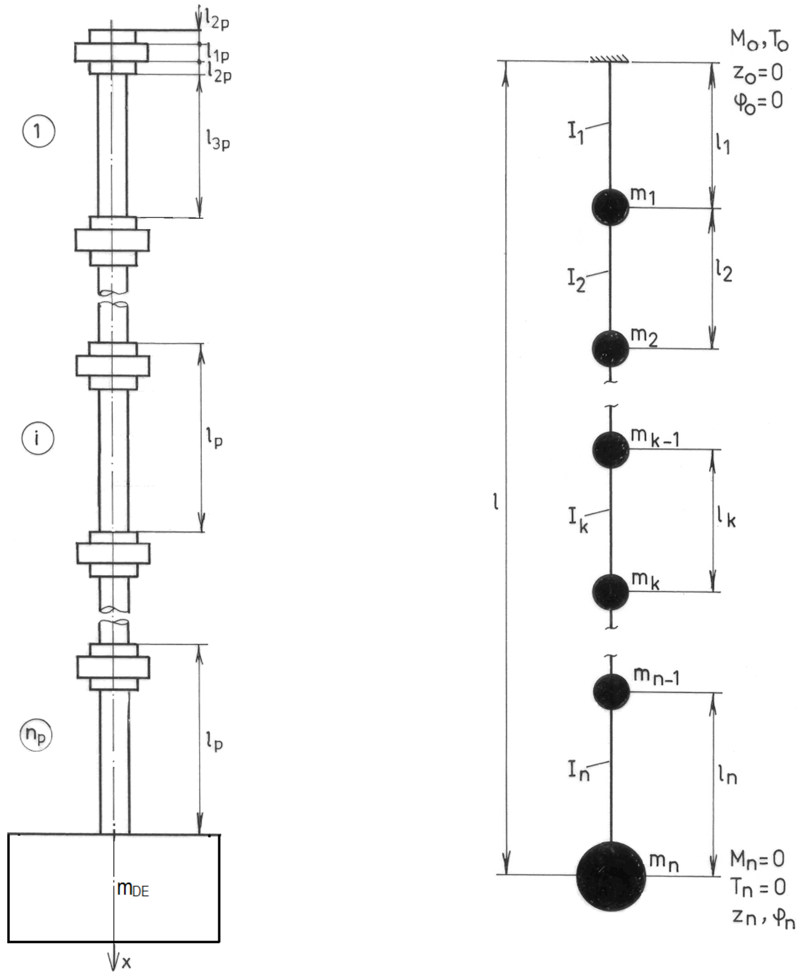


Fig. 9. First two natural modes of torsion vibration of the 10¾" drill string, with non-uniformly distributed mass and mass moments of inertia concentrated at the ends

For model with non-uniformly distributed mass, mass moments of inertia concentrated at the ends and loose ends were obtained following first six natural pulsations: p_1 = 2.096 s^{-1};

p_2 = 25.422 s^{-1}; p_3 = 50.505 s^{-1}; p_4 = 75.487 s^{-1}; p_5 = 100.257 s^{-1} and p_6 = 124.727 s^{-1}, finding that these measures are lower than in the case of the drill string model with uniformly distributed mass. The natural modes of torsion vibration for the first two natural angular frequencies are represented in Figure 9.

For model with three concentrated masses, it is found that the elasticity of the hydrostatic transmission is about 8.3 times higher than that of the drill string, this means that the hydrostatic transmission, by the hydraulic medium compressibility and the line elasticity, carries out the function of a dynamic absorber. The following measures of the first two natural angular frequencies will result: p_1 = 0.48673 s^{-1}; p_2 = 2.19732 s^{-1}.



a b

Fig. 10. Physical model of the drill string, considering the drill pipe mass as uniformly distributed on the proper body and concentrated directly near the union flange: $l_{j.p}$ – length of the drill pipe part j, j = 1, 2, 3; l_3 – drill pipe body length; l_p – drill pipe length; n_p – number of drill pipe; l – drill pipe ultimate length (upper ensemble length of the drill string); m_k – mass concentrated directly near the union flanges between a drill pipe of order k and the drill pipe of order k+1, k = 1, 2, ..., n–1, $m_1 = ...= m_k = ...= m_{n-1}$; l_k – length of the drill pipe of order k, $l_k = l_p$; I_k – axial geometrical moment of the drill pipe body, $I_1 = I_2 = ...= I_k = ...= I_n$; m_n – concentrated mass of the depth ensemble, $m_n = m_{DE}$; M – bending moment; T – shearing force; z – transverse displacement (arrow); φ – rotation

To study the free flexural vibrations were considered two physical models [3]: 1) drill string model with lengthwise uniformly distributed mass, with constant stiffness, with the depth ensemble mass concentrated in the lower end and the upper end recessed; 2) model with non-uniformly distributed mass, with the depth ensemble mass concentrated in the lower end and the upper end recessed (fig. 10 b).

The calculation of the natural frequencies and natural modes of oscillation with the model two is made by using Myklestad-Prohl's method. By means of a program written in MATLAB language and developed on the basis of this algorithm, the following values of the natural pulsation have been obtained [1]:

$$p_1 = 0.06875\,\text{rad/s};\ p_2 = 0.45625\,\text{rad/s};\ p_3 = 0.77813\,\text{rad/s};\ p_4 = 1.14062\,\text{rad/s}.$$

The natural modes of vibrations corresponding of these pulsation values are presented in Figure 11.

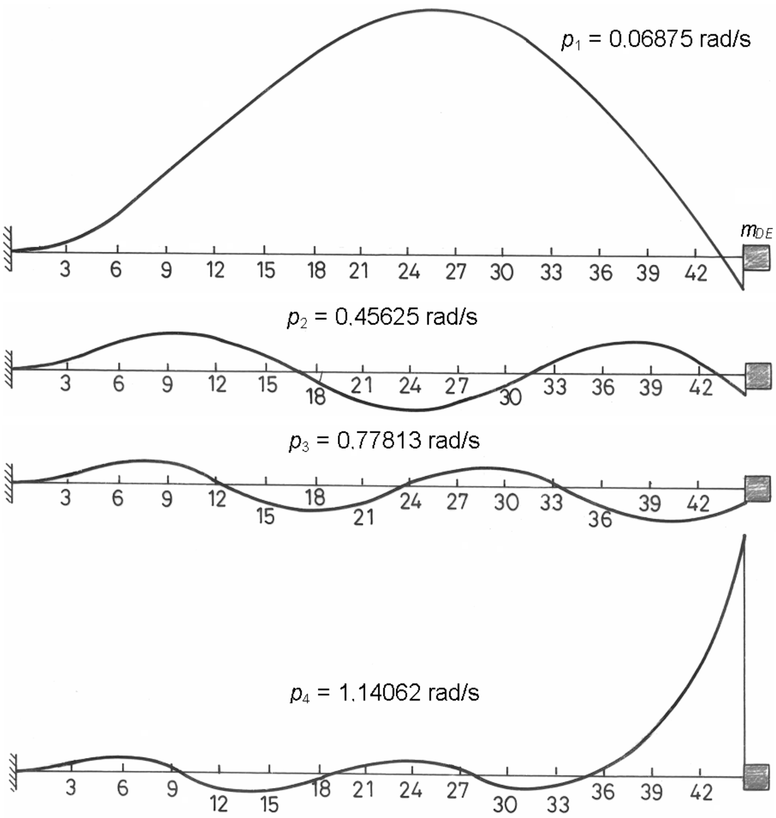


Fig. 11. The first four natural modes of flexural vibrations of the 10¾" drill string model having the drill pipe mass non-uniformly distributed and the depth ensemble mass concentrated at the lower end

It may be found that the basic pulsation, obtained in this way is higher than that obtained for the model with uniformly distributed mass, contradictory to the results obtained for longitudinal and torsion vibrations (see [1] and [5]). This may be explained by the fact that natural flexural pulsations are influenced by axial forces – in our case, the proper weight of the drill string by concentrated masses – and rotational inertia of the masses concentrated, influences which were neglected in case of the first model.

Longitudinal vibrations of the drill string were studied on two physical models [1]: 1) model with mass evenly distributed along the length of the drill string, constant stiffness and depth assembly mass concentrated at the lower end; 2) model with drill pipe mass unevenly distributed and depth assembly mass concentrated at the lower end (fig. 10 a).

For the second model, the following measures have been obtained in the first six natural pulsations:

$$p_1 = 29.473\,\text{s}^{-1};\ p_2 = 73.192\,\text{s}^{-1};\ p_3 = 119.127\,\text{s}^{-1};\ p_4 = 165.070\,\text{s}^{-1};\ p_5 = 210{,}971\,\text{s}^{-1};$$
$$p_6 = 256.932\,\text{s}^{-1}.$$

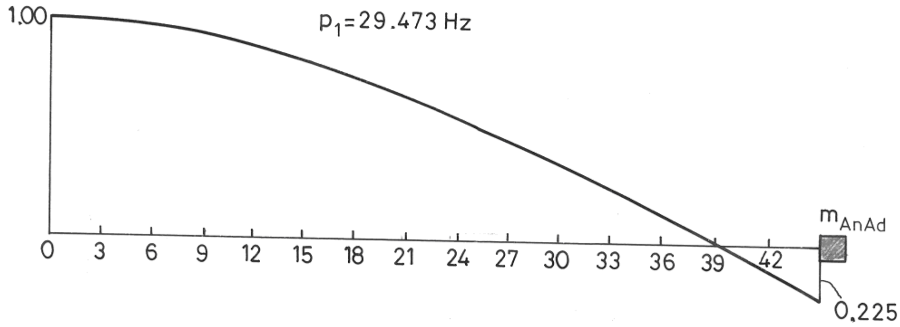


Fig. 12. Fundamental mode of longitudinal vibration of drill string model with unevenly distributed mass

The natural mode of vibration corresponding fundamental pulsation was represented in Fig. 12.

Experimental Research

Dynamic loadings of the drill pipe during the start-up phase and stationary loadings have been studied on the basis of experimental recordings made while drilling the well used in the mining exploitation of oil from Buștenari with the rig type F320-3DH-M [1].

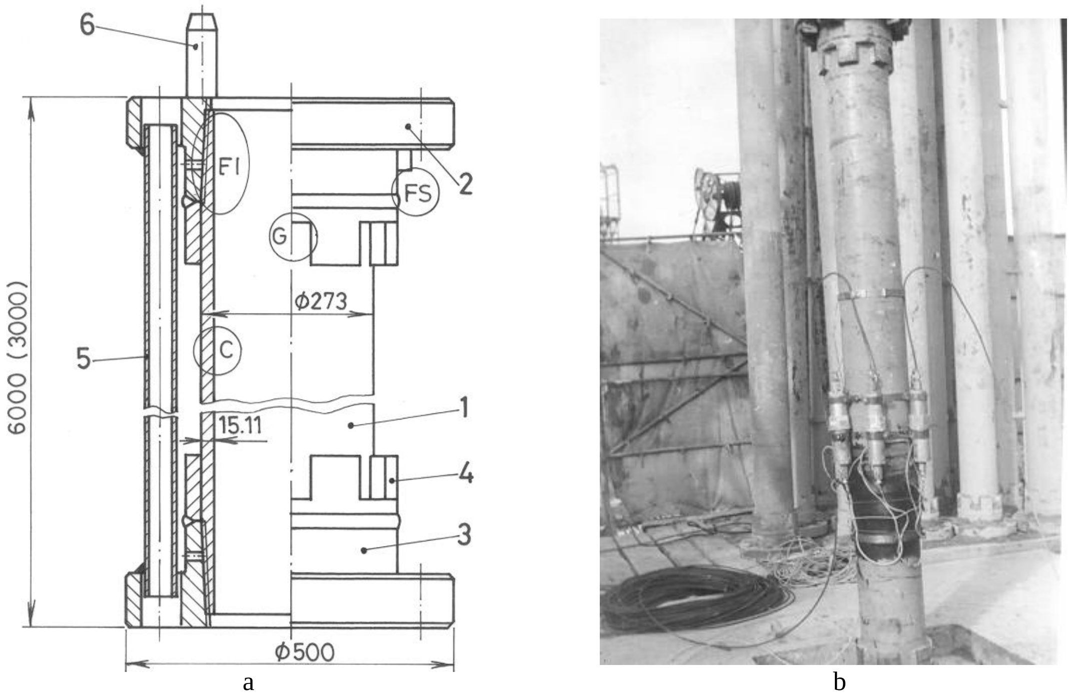


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Fig. 13. 10¾" drill pipe/Loading captor, with areas of location/positioning of the strain gauges (FS, G, C and FI): 1 – body, 2– upper flange, 3 – lower flange, 4 – crenellated support collar, 5 – air tube, 6 – centering and torsion moment transmission bolt (a); loading captor, assembled in the drill string before its introduction in drill shaft (b)

For measurements, three loading captors were made (see, for example, fig. 13), represented by three drill pipes, equipped with strain gauges, for the measurement of the specific strain due to axial force, torsion moment and bending moment, of the complex/total specific strain (fig. 13 a, area „C", and fig. 13 b) and of the specific strain of zones with stress concentrators: flange neck, in the vicinity of the welding seam between the flange neck and the collar (fig. 13 a, area „FS", and fig. 14 a), the body of the drill pipe, from the cut-out part of the support collar, in the vicinity of the weld between the collar and the drill pipe (fig. 13 a, area „G”, and fig. 14 a), and zone of joining through the thread (fig. 13 a, area „FI”, and fig. 14 b).

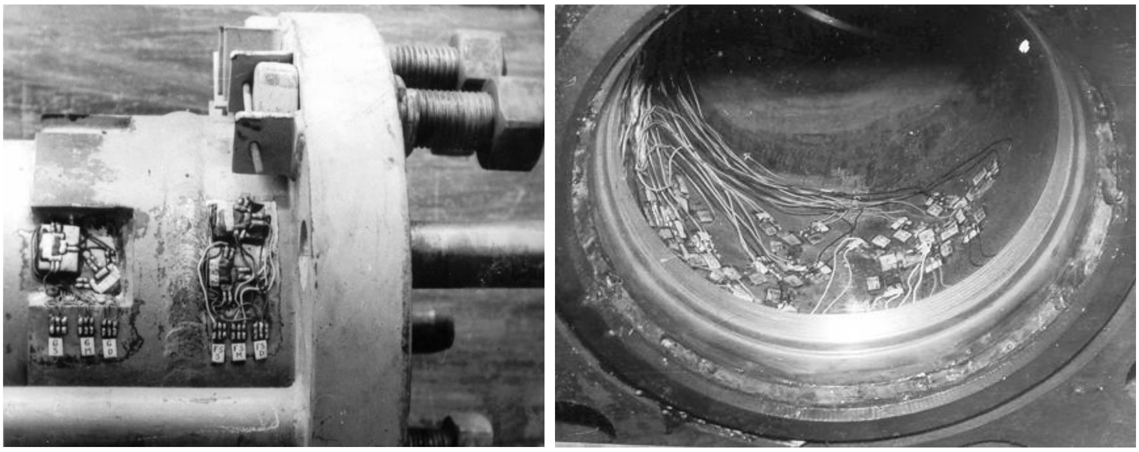

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Fig. 14. Areas of application of strain gauges: a) zone of the flange neck („FS”) and cut-out zone of the support collar („G”); b) zone of joining through the thread („FI”)

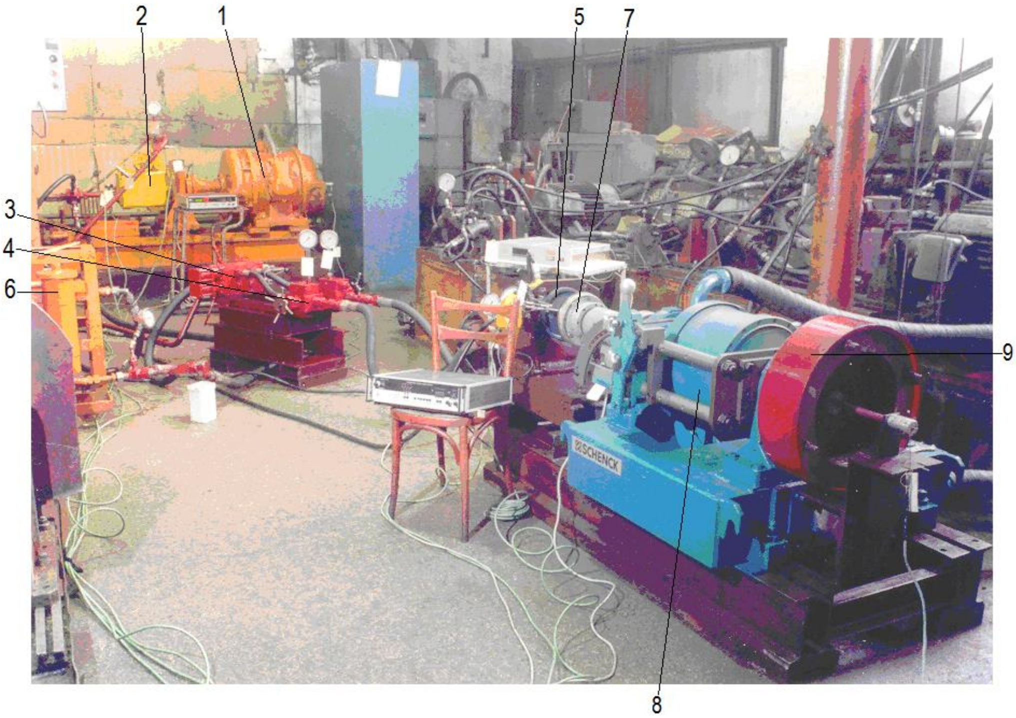


Fig. 15. Stand for study of the non-stationary hydro-mechanical processes in the frame of the electro-hydrostatic driving groups in case of large diameter drilling rigs: 1 – asynchronous electric motor of type MIB2X 315s 80-6; 2 – pump with axial pistons of type F232 K.1.V1100M.P.; 3 – directional control valve of type DN32×4/3; 4 – pressure control valves; 5 – hydrostatic motor with axial pistons of type F132-25.I.P.; 6 – refrigerator/cooler; 7 – torque transducer TM500 (with nominal torque of 500 Nm); 8 – hydraulic brake of type Schenck U2-30; 9 – flywheel with variable moment of inertia

These experimental researches „in situ” were completed in the laboratory, by building a stand which simulates, on a real scale, the rotary system of the F320-3DH-M drilling rig, reduced to the hydrostatic transmission (fig. 15). The stand was built with the purpose of investigation of the non-stationary hydro-mechanical processes in the frame of electro-hydrostatic driving groups of the rotary system belonging to the large diameter drilling rigs [1]. Therefore, the stand was called „Stand for Research of the Non-Stationary Hydro-Mechanical Processes".

The stand was equipped with measuring instruments (with analogical and digital display) for reading the average measures of the different mechanical, hydraulic and thermal sizes (the rotative speed, torque, pressure, and hydraulic medium temperature), and also, with strain gauge transducers for recording of the stated mechanical and hydraulic sizes in real time.

With hydraulic brake type Schenck U2-30 (see fig. 15) can be done a variable braking moment corresponding to the friction moment between the drill string and drilling mud, and between the rollers of the stabilizer and wall of the borehole (determined on experimental path during measurements of the site).

By using the flywheel V500 (see fig. 15) you can get five different measures for the moment of inertia, in accordance with the moment of inertia reduced to hydrostatic motor shaft of all elements which are rotating and are between this shaft and drill bit, for different measures of the length of the drill string, according to the drilling depth.

Dynamic Loadings during the Start-up Phase

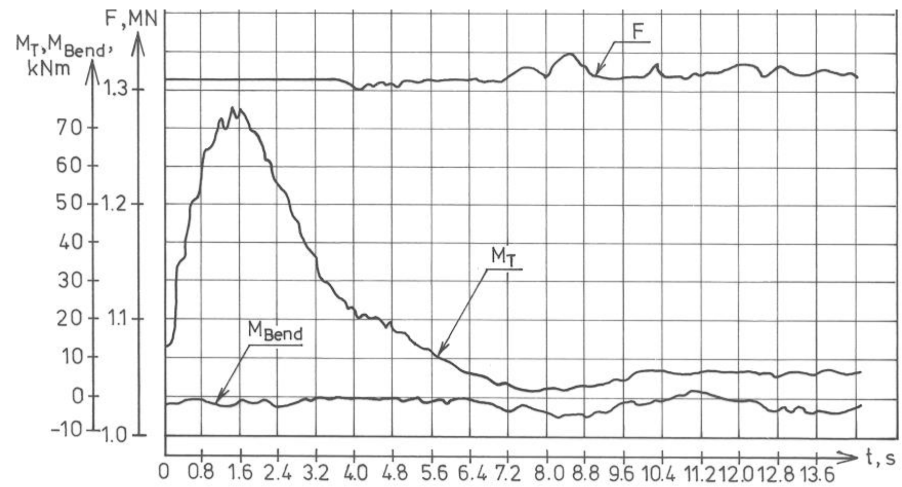


Fig. 16. Time variation of the axial/tensile force (F), torsion moment (M_T) and bending moment (M_{Bend}) which loads the cross section of the drill pipe body CS1, during the drill string starting by using both of the generator groups

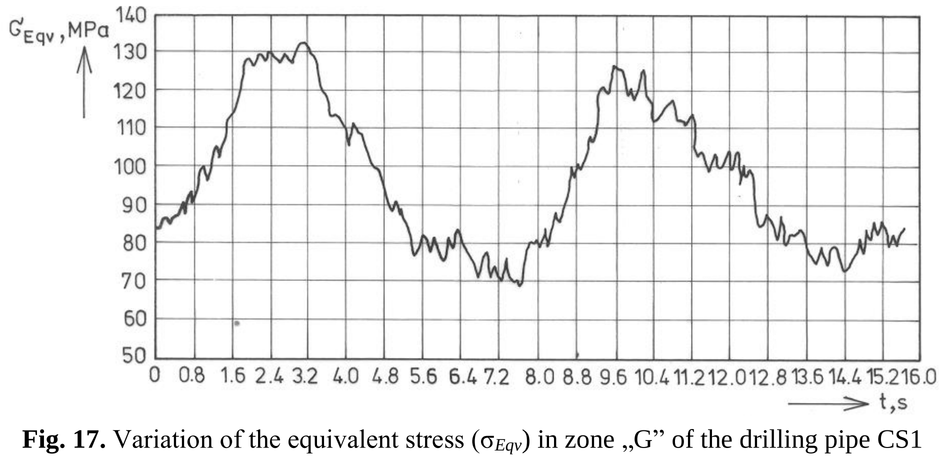


Fig. 17. Variation of the equivalent stress (σ_{Eqv}) in zone „G" of the drilling pipe CS1 in the case of starting with a single generator group being initially in stationary regime of running and the subsequent starting of the second group

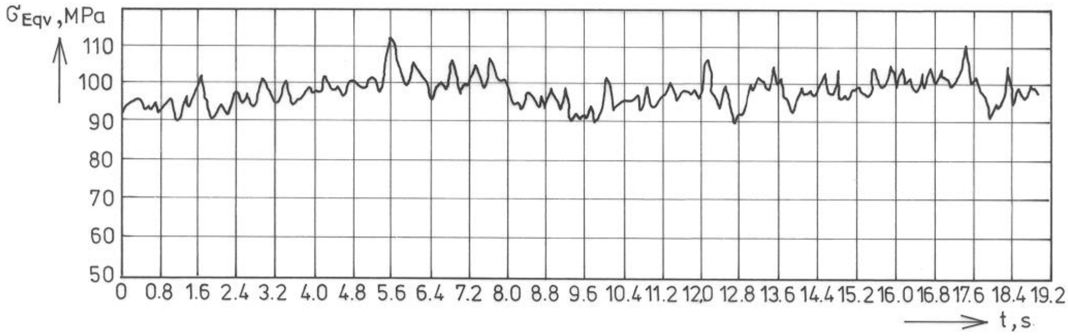


Fig. 18. Time variation of the equivalent stress (σ_{Eqv}) in zone „FS" of the drill pipe CS1 in the case of starting of the drill string with a single generator group running in stationary regime and the subsequent starting of the second group

From the processing of experimental data recorded in the oil field during the startup of the rotary table were obtained diagrams represented in Figures 16 ÷ 18 [1].

Table 1. Size measures characterizing the loading variations in the cross section of the drill pipe body CS1, during the drill string starting, with two generator groups: f − physical size; f_n − mean value; f_M − maximal value; f_m − minimal value; d_f − quadratic mean deviation

f	F, kN	M_T, kNm	M_{Bend}, kNm	σ_F, MPa	τ_T, MPa	σ_{Bend}, MPa	σ_1, MPa	σ_2, MPa	τ_1, MPa	φ_{01}, grade	σ_{Eqv}, MPa
f_n	1314.1	23.43	-1.08	107.3	15.66	-1.44	109.9	-4.03	56.98	0.133	112.1
f_M	1337.3	75.20	3.07	109.2	50.26	2.84	125.4	-0.03	72.52	0.383	136.3
f_m	1301.9	2.57	-4.95	106.3	1.72	-6.31	102.3	-20.2	51.17	-0.016	102.4
d_f	6.73	22.75	1.42	0.55	15.20	1.90	5.88	5.98	5.87	0.120	9.07

Mathematical processing of registration of Figure 16 led to the results presented in Table 1, where f represents the physical size, $f \in \{F, M_T, M_{Bend}, \sigma_F, \sigma_{Bend}, \tau_T, \sigma_x, \sigma_1, \sigma_2, \tau_1, \varphi_{01}, \sigma_{Eqv}\}$.

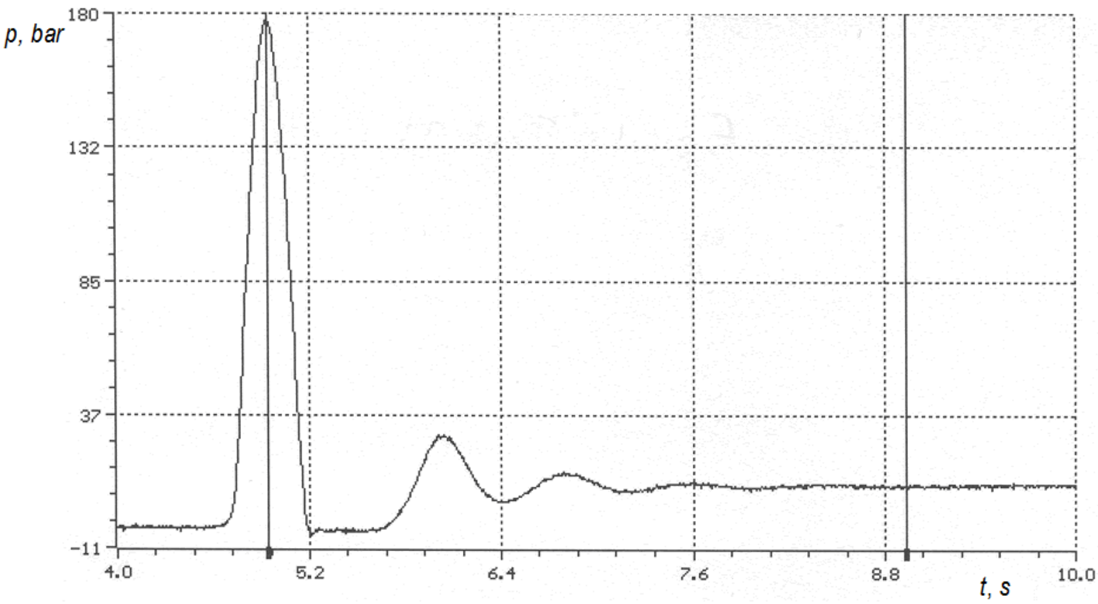


Fig. 19. Variation of the supply pressure of the hydrostatic motor to start the whole system without flywheel and hydraulic brake without water

In Figures 19 and 20 are two records made under the „Stand for Research of the Non-Stationary Hydro-Mechanical Processes", which shows the change in supply pressure of the hydrostatic motor and change in the torque of the hydrostatic motor shaft during the starting, by driving the distributor of type DN32×4/3 (see fig. 15).

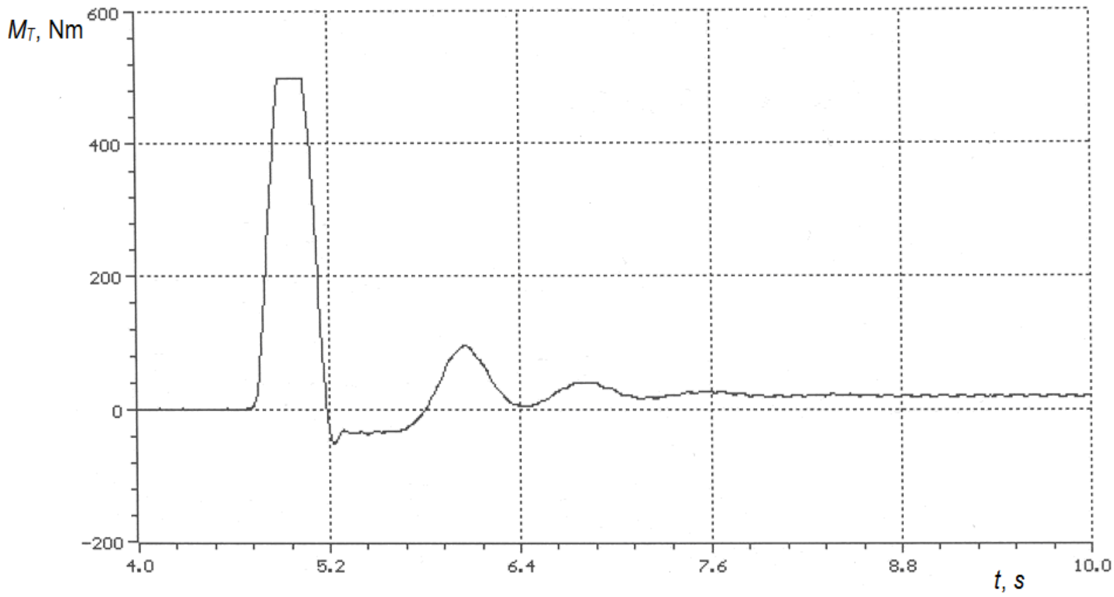


Fig. 20. Variation of the hydrostatic motor torque during the starting of the entire hydrostatic system, without flywheel and with hydraulic brake without water

Stationary Loadings and Their Variability

During the drilling, there is a variation of the weight-on-bit (W_B), the angular speed of the kelly (ω_K) and the torque of this kelly (M_K), according to Figure 21.

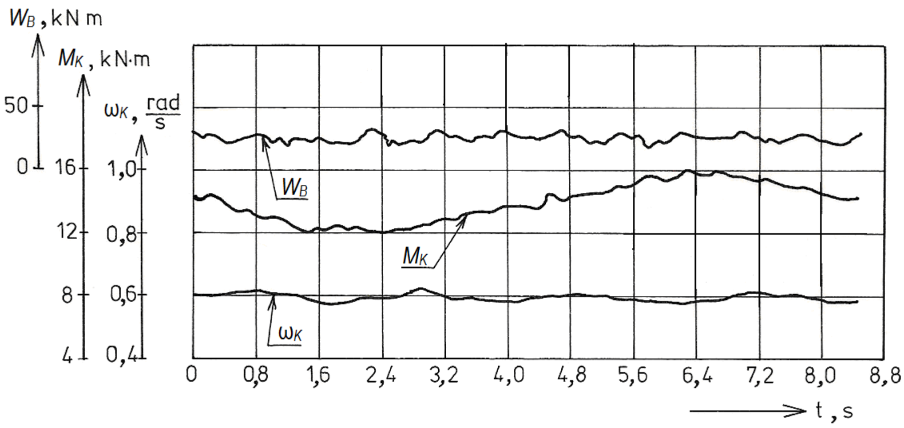


Fig. 21. Variation of the weight-on-bit (W_B), the angular speed of the kelly (ω_K) and the torque of this kelly (M_K)

Table 2. Measures of the physical sizes characterizing the loading variation in the cross section of the drill pipe body CS1 during the drilling by using both of the generator groups (fig. 22)

f	F, kN	M_T, kNm	M_{Bend}, kNm	σ_F, MPa	τ_T, MPa	σ_{Bend}, MPa	σ_1, MPa	σ_2, MPa	τ_1, MPa	φ_1, degrees	σ_{Eqv}, MPa
f_n	1176.8	49.6	-1.34	96.10	33.12	-1.79	104.9	-10.5	57.68	0,310	110.5
f_M	1204.8	53.89	7.79	98.41	36.02	10.41	117.3	-8.53	36.02	0.345	122.8
f_m	1130.8	44.14	-9.91	92,37	29.50	-13.30	94.0	-12.8	29.50	0.277	100.1
d_f	15.0	2.80	4,47	1.22	1.87	5.97	5.83	1.21	2.82	0.020	5.67

As a result, the axial, torsion and bending loadings of the drilling pipes are also variables (see [4]), as shown in Figure 22. Their variability is quite important, as shown in Figure 22 and Table 2.

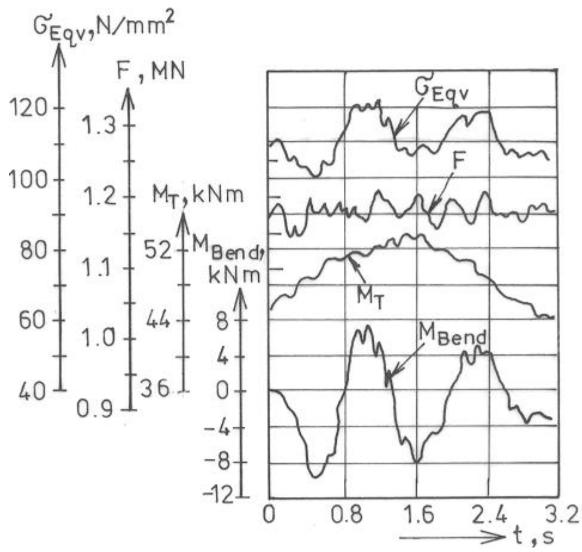
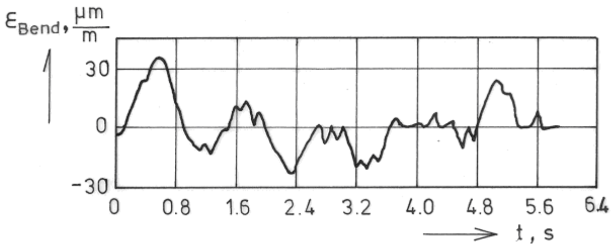
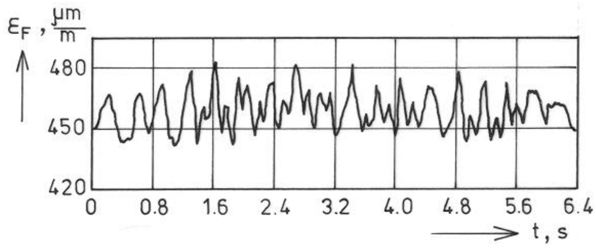


Fig. 22. Changes in axial force (F), torque/moment of torsion (M_T), bending moment (M_{Bend}) and equivalent stress (σ_{Eqv}), one for each half period, for W_B = 130 kN



a



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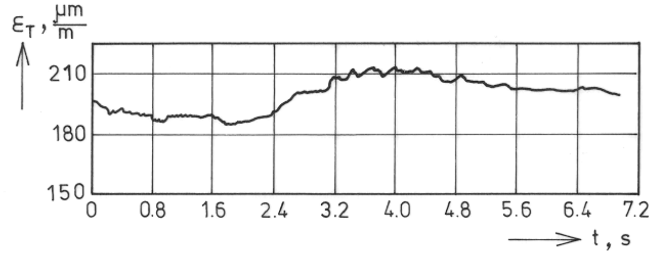


c

Fig. 23. Strain variation due to bending moment (for W_B =120 kN and $n_K \cong$ 10.2 rpm) (a), axial force (for W_B =160 kN and $n_K \cong$ 9.4 rpm) (b) and torsion moment (for W_B =200 kN and $n_K \cong$ 8.7 rpm) (c)

Forms of variation over time of axial force, torsion moment and bending moment are shown in Figure 23, through changes in specific strains concerned.

Conclusions

The drilling process of large diameter mining shafts presents peculiarities in comparison with the drilling process of wells for oil and gas. These features are due to problems of insufficient washing, as well as the structure of the drill string, the greater moment of inertia and the drill pipe with areas of stress concentrations of design and metallurgical nature. These characteristics are determined by using the method of reverse circulation of the drilling fluid, by problems of insufficient washing of the drill bit rollers and bottom of the shaft, by drilling regime, with great weight-on-bit and low speed of the drill bit, as well as the structure of the drill string, with great moment of inertia and drill pipes with areas of stress concentrations of design and metallurgical nature.

This paper focuses on phenomena that occur during the large diameter drilling and have a cumulative effect of reducing the lastingness of the drill pipes.

Cracking and breaking the drill pipes by 10¾", 13⅜" și 14⅜", found during operation, imposed the development of theoretical and experimental studies to highlight the causes of these degradations and failures.

Thus, based on the ideas presented in the previous chapters, we can formulate the conclusions below.

- Analysis of the large diameter drilling process and drilling practice have shown that phenomena appear as momentary drill bit blockings, rotation motion with intermittent contact between the bit rollers and rock, and stick-slip phenomenon of the drill bit, accompanied by shocks and vibrations of the drill string. These phenomena are caused by uneven contact of the bit rollers with rock and insufficient washing of the rollers and bottom of the shaft with drilling fluid.
- The existence of dynamic shocks during drilling led to hasten development of cracks, as happened while performing measurements on site when the drill string was subjected to repeated shocks in a short time, by stops and starts, by changing the weight-on-bit and by changing the drive speed of the drill bit. Thus, at the end of measurements, after being extracted drill string was a macro-crack penetrated (see Fig. 4), although visual inspection of drill pipes when they were introduced into the well did not confirm the existence of any cracks in the stage of development.
- Analysis of cracking and breaking areas of drill pipes showed that these areas coincide with areas of stress concentrations of constructive and metallurgical nature, represented by a threaded joint, the additional welds or weld end side areas of the support collar and the body of the drill pipe (in the case of drill pipes of 10¾") [5].
- Theoretical study of the free vibration of drill string with different physical models highlighted the importance of using weight unevenly distributed model, which better approximates structural unevenness at the ends of the drill pipes and determines lower natural pulsations, according to the actual dynamic behavior of the rotation system.
- In the case of torsion vibrations, using physical models with the integrated use of drill string within the whole rotation system, taking into account the elasticity of the hydraulic medium in hydrostatic transmission, had the same effect of reducing natural pulsations of the drill string oscillations.
- Thus, the model with integrated use of drill string within the whole rotation system, which takes into account the elasticity of the hydraulic medium in hydrostatic transmission and concentrated mass of electric motors and hydrostatic transmission reduced to kelly, showed that the own fundamental pulsation of torsion oscillations is found in the measures of working angular velocity of the kelly for operation with a single electro-hydraulic generator group. This means that it is possible manifestation of the phenomenon of resonance in torsion oscillations [2].
- The emergence of torsion resonance phenomenon in [0.389; 0.584]·rad/s is evidenced both by experimental research conducted in the „Stand for Research of the Non-Stationary Hydro-Mechanical Processes" of Hydraulic Laboratory of the S.C. Plopeni S.A. (see fig. 24) and experiments with rig F320-3DH-M in the Buștenari site. It states that the hydrostatic motor speed of 540 rpm, which were found the knocking phenomenon, corresponding to an angular

speed of the kelly of 0.568 rad/s, taking into account the reduction ratio of the gear train of the rotary table.

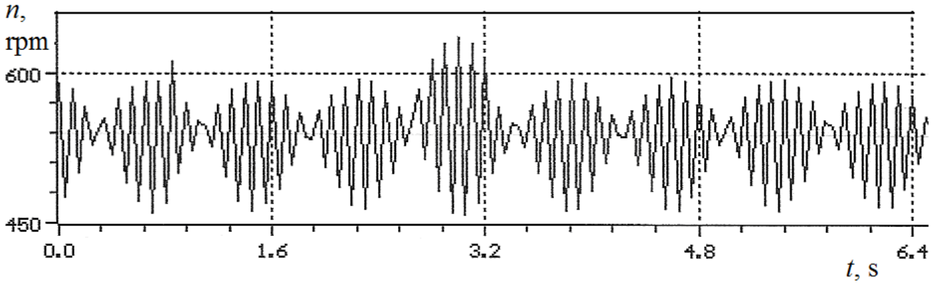


Fig. 24. Variation of the hydrostatic motor rotation speed (*n*), with appearance of the knocking phenomenon, for the average rotation speed of 540 rpm, in the frame of the „Stand of Research of the Non-Stationary Hydro-Mechanical Processes” [1]

- By studying the free flexural vibrations of drill string of 10¾" using model with drill pipe weight unevenly distributed and bottom assembly mass concentrated at the lower end and with the top end recessed, it was concluded that natural pulsations of orders 2, 3 and even 4 are in measure field of working angular speed of the drill string, for operation with a single electro-hydraulic generator group and two such drive groups. It is therefore very likely to occur resonance phenomenon of flexural oscillations. The first natural modes of transverse vibration have vibration nodes in which the slope of the deformation curve (dφ/dx) is highest in the lower first third of the drill string (see Fig. 11). These cross sections of the drill string are subject to a maximum bending loading, according to

$$M_{Bend} = E \cdot I \cdot \frac{\mathrm{d}\varphi}{\mathrm{d}x}. \tag{1}$$

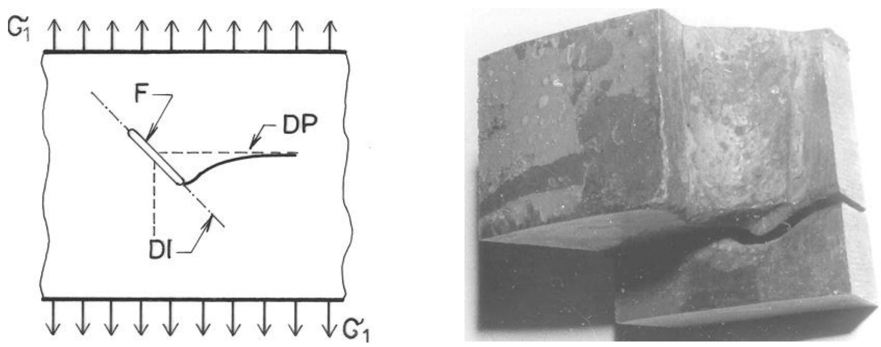
The study of the conditions under which it occurred the cracking and breaking of the drill pipes has indicated that they have affected the pipes located in the aforementioned area of maximum bending stress and research of the areas of breakage has shown the manifestation of the phenomenon of fatigue.

- The longitudinal vibrations do not pose a danger to degradation of the drill pipes, because the fundamental natural pulsation of these longitudinal oscillations of the drill string is higher than the working angular velocity, as evidenced by the theoretical and experimental records.

- Experimental recordings made in the conditions of the drilling site and laboratory have contributed much to elucidate the causes which have led to a deterioration in the time of the drilling pipes. Research of the site focused on dynamic loadings of the drilling pipes, during the start-up phase, and stationary loadings during drilling, in the body area and in areas with stress concentrators. The laboratory research had as object non-stationary hydrodynamic processes taking place in the hydrostatic transmission. For carrying out of researches in the drilling site, loading collectors were built. These collectors are represented by drilling pipes equipped with resistive stress transducers, for measuring and recording in real time of the specific strains due to axial force, torsion moment and bending moment, and with stress rosettes for the measurement and recording of the strains after three directions in the body area of the pipe and in areas with stress concentrators [6]. For research in the laboratory, has built a stand (fig. 15) in the Hydraulic Laboratory of the S.C. Plopeni S.A. [7].

- During the start-up phase of the drill string, develops a maximum torque of 75.2 kNm, corresponding to a dynamic coefficient of 10.65, which means that dynamic torque of 9.65 times greater than the torque of the stabilized movement. The maximum torque obtained is very close to the maximum torque required by the designer, of 78.5 kNm, while, however, the recorded maximum force from the hook was 136.3 tf, to the maximum working load specified by the designer which is 150 tf. Maximum dynamic torque due to higher moment of inertia of the drill string, especially of the assembly depth, represented by the drill bit of 3.62 m, the stabilizer with rollers and special heavy drill pipe.

- The same form of variation have the supplying hydrostatic pressure of the hydrostatic motor and the torque applying this motor shaft, as recorded in the stand remembered (see fig. 19 and fig. 20). The maximum pressure and, hence, the maximum torque to the shaft of the hydrostatic motor and the maximum torque applying drill pipes depends on the mode of operation of the directional control valve, more abruptly or slowly.
- The state of stress in the area „G" with the stress concentrations, represented by the transition from the flange collar to the body of the drill pipe and by the joint with side and front weld of the two elements, is characterized by a equivalent stress to at least 10 higher than the equivalent stress of the body section of the drill pipe and also by more pronounced oscillations. Thus, it is shown that this area is dangerous in terms of equivalent stress measure and its variability; is an area conducive to the emergence of fatigue degradation, especially as it is, with high probability, and crack nucleation site.
- Based on the analysis of stress state in point „FS" can be considered that this area with stress concentrator of the drill pipe, represented by butt welding between the neck flange and the crenellated collar, is less dangerous to degradation by fatigue appearance even than the drill pipe body, where it complies with the technology to achieve the weld. This is due to the larger section of the flange neck and to the better technological conditions to achieve the weld. There is a much smaller influence of torsion and bending stresses in relation to tensile stress within the complex loading that characterize this area.
- The character of the variation in case of axial force, bending moment and torque loadings during drilling is highlighted in Fig. 23: axial force has a pulsating and random character, with high frequency compared to the other two loadings, as determined by the variation of the weight-on-bit (Fig. 21); torque has a pulsating character, with a sinusoidal variation, a period equal to that of a complete revolution; bending moment variation has alternating positive and negative values with maximum amplitude sinusoidal five loops.
- Bending stress variation character is printed equivalent stress (fig. 22).
- From the point of view of the average value, the stress caused by the axial force is the greatest, followed by the torque tangential stress, and then the bending stress (see Table 2).



a b

Fig. 25. Crack propagation: a) in case of poly-axial state of stress which induces the modes of propagation/displacement I (displacement by extension) and II (displacement by sliding in plane): σ_1 – normal main maximal stress; F – crack; DI – crack initial direction; DP – direction of propagation; b) in case of the drill pipe, series 122

- Angle φ_{01} formed by the direction of the maximum principal normal stress (σ_1) with the axial direction of the drill pipe is very low, with average values of [0.06 °; 0.31 °] (see Table 2), that σ_1 is practically perpendicular to the axial direction of the drill pipe. These values of φ_{01} and direction of propagation of cracks found in drill pipes (see Ffig. 25) confirm the validity of phenomenological criterion of maximum normal stress, that the crack propagates perpendicular to the direction of the maximum normal stress, so that the tangential stress is zero on the line of crack propagation.
- Analyzing the maximum and minimum deviations and standard deviations, it is estimated that stresses applying body drill pipe has a fairly pronounced variability in relation to the average measure so that the drill pipe assembly is a structure sensitive to degradation by repeated loadings. Cracking process is accelerated by dynamic loading to start and during momentary stalls and stick-slip phenomenon and also is influenced by changing drilling regime imposed by technological needs.

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Efectele cumulate ale fenomenelor ce însoțesc procesul de foraj de diametru mare asupra degradării în timp a prăjinilor de foraj

Rezumat

Prăjinile de foraj utilizate pentru forajul de diametru mare se degradează în timp, producându-se fisurări și ruperi, datorită efectelor cumulate ale unor fenomene. Aceste fenomene sunt caracteristice procesului de foraj de diametru mare, cu circulație inversă a fluidului de foraj, cu faze tranzitorii, masă mare în mișcare de rotație, forță mare de apăsare pe sapă și turație mică a sapei. Ca urmare, apar solicitări dinamice importante, o stare de tensiune complexă și variabilă pe durata unei rotații complete și vibrații torsionale și flexionale ale garniturii de foraj, care pot avea pulsații în domeniul de rezonanță. Aceste fenomene se desfășoară pe fondul existenței unor concentratori de tensiune, de natură constructivă și metalurgică, determinați de construcția și tehnologia de fabricare a prăjinilor de foraj. În lucrare, sunt studiate efectele cumulate ale acestor fenomene, pe baza unor cercetări teoretice și experimentale.