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Reasons of drilling assembly vibration during the operation of rotary drilling rigs

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Abstract
Sources of dynamic loads and nature of their course during the operation of roller-bit drilling rigs are considered. The assumption that dynamic loads when drilling arise because of change of parameters of the rotational moment on roller bit is accepted. It leads to joint shifting and axial vibrations of a drilling assembly. The mathematical model of determination of critical force of drilling assembly is given. Qualitative assessment when calculating a dynamic rotational moment on a rotator and dynamic squeezing force for drilling assembly is given.
Key words: DRILLING ASSEMBLY, ROLLER BIT, ROTATOR, FEEDING MECHANISM, BAR, VIBRATION RESISTANCE, CRITICAL FORCE

Introduction

For successful design and modernization of modern high-production and reliable roller-bit drilling rigs and drilling device, data on the source of dynamic loads and the character of their operation when drilling are necessary.

Literature review

Up to the present day, engineering techniques are not developed for calculation of parameters of drilling rigs taking into account the dynamic modes during their operation [1, 2].

Initial data and methods

The technique directed to determination of the optimum modes of roller-bit drilling (from the point of view of minimization of torsional and longitudinal fluctuations) were selected on the basis of the analysis of technical and economic indicators during the operation of rigs. At the same time, with increase in axial pressure P_p and speed of drilling assembly (ω) (Fig. 1), and also with improvement of bottom-hole cleaning of rock failure products, the speed of drilling and productivity of rig increases. At the same time, reliability of operation of machines and roller bit decreases under forced operation conditions. At that, there can emerge such values of axial pressure and angular speed (rotation number) of drilling assembly which will lead to loss of drilling assembly stability, emergence of the increased vibration of assembly and elements of a metal construction of the rig, as a result, further operation of the rig (on these modes) will be impossible.

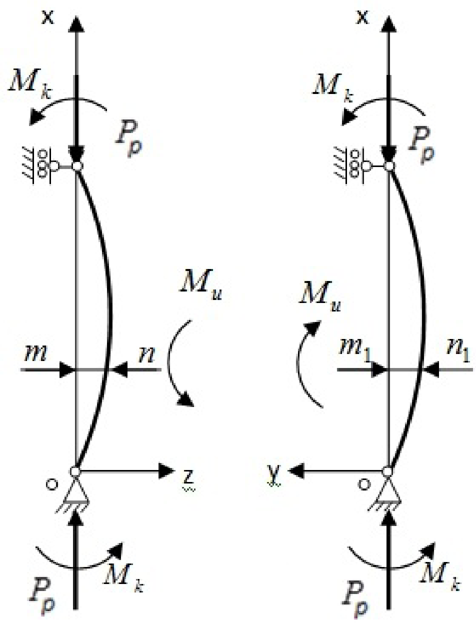
Owing to the set before, intensification boundary of the drilling modes are determined by resistance to vibration and reliability of drilling assembly operation.

The following basic factors affect the drilling assembly resistance to vibration:

1. The drilling modes (angular speed of rotation – ω, axial force - P_p and degree of bottom-hole cleaning of the destroyed rock particles).

2. Parameters of drilling assembly (length, outer diameter, thickness of walls of bars, rigidity of joints at connection of bars).

3. Features of fixing of the ends of drilling assembly (on a rotary head and in a zone of interaction of roller bit with rock).



(bottom reaction) (bottom reaction)

a) b)

Figure 1. Loading diagram of drilling assembly by power factors

Moreover, the jointing of rocks, drilling bit cutting structure type, condition of bearing assembly of roller bits, mass of the machine and other factors can exert a certain impact on drilling assembly resistance to vibration.

Let us note that these factors in most cases are of random nature, badly undergo mathematical description and often have insignificant influence on the nature of the proceeding processes; therefore, they can be neglected at this stage of researches without errors.

Features of drilling assembly operation of real machines SBSh (Atlas Copco and others), the analysis of publications [1, 2] and more modern papers make it possible to assume that drilling assembly undergoes complex deformations during drilling under the action of compressive, twisting and bending force factors. Moreover, these power factors can have significant dynamic components during rig operation.

Let us note that when drilling wells with "long" assembly (when the total length of assembly approaches the height of ledge), the drilling assembly may lose elastic stability several times (that is, the elastic line of the assembly will take the form of a wave line with several periods). Such a wave line is caused by a significant compressive force of the P_p and a radial clearance between the bars and walls of the well. This clearance can vary significantly along the depth of the well depending on the fracture and strength of the rock mass to be drilled. In this case, intensive destruction of the cross layers of the rock can occur and the gap between the walls of the borehole and the assembly may be significantly reduced, which will lead to a change in the conditions of assembly support in the face, and also to a certain increase in the critical (compressive) force. It is also necessary to consider a significant circumstance that drilling assembly is under influence of both the compressive force factors (P_p), and the twisting force factors (M_{tw}); therefore, critical force calculated by Euler formula [3] P_{cr} will give significant error.

Besides, during evaluation of dynamic loading of drilling assembly and the whole machine in general, there appears the following question: how do considerable fluctuations emerge in the drilling assembly, bearing elements and the whole machine in the system "rotary drive - drilling assembly - drill tower" at constant rig pulldown (P_p) and the constant moment in rotator (M_{tw}).

Let us assume that considerable dynamic loads in elastic system of the rig are caused by interaction of drill bit with the rock massif through resilient system of drilling assembly, feeder and mechanism of rotation (rotator). For the purpose of obtaining the calculation model for drilling assembly with mechanisms of rotation and giving of the machine and development of the mathematical model describing emergence of difficult longitudinally tortional fluctuations (taking into account joint cross fluctuations) of drilling assembly, let us proceed from the fact that the roller bit introduced into the rock faces resistance to its rotation. At the same time, the amount of potential energy of drilling assembly is insufficient for overcoming this resistance. At such moment, the roller bi either significantly reduces the angular speed of rotation, or completely stops; at the same time, the top end of a drilling assembly continues to rotate and potential energy from rotation is accumulated in drilling assembly. When potential energy of deformation is enough for bit rotation, its intermittent break in a face takes place and the speed of its rotation increases significantly (ω). Thus, there appear tortional fluctuations along the length of drilling assembly. At the moments of assembly twisting, its length decreases significantly, and when assembly untwisting, its length increases in proportion to the angle of its untwisting, that is there are periodic (oscillatory) changes of length of assembly. At interaction of roller bit with a face, these changes lead to longitudinal oscillatory processes in a drill stem, as well as in the bearing elements, and also to considerable vibrations of the whole machine in general.

Let us describe behavior of a drill stem (assembly) proceeding from the considered mechanism of emergence of tortional and, as a result, longitudinal fluctuations of drilling assembly (see Fig. 1). Such oscillatory process is called self-oscillatory and it usually takes place in the systems with friction ("dry" friction). At such vision of the mechanism of fluctuations emergence, it is necessary to build a system of joint tortional and longitudinal self-oscillations in system: drilling device – drilling assembly – rotator – feeder. Considering mathematical complexity of the formulated task, at the first stage of researches we will limit ourselves to quasistatic problem statement without considering forces of inertia, and considering previously that movements of z (x), y(x) conditionally do not depend on time *t*.

Based on the accepted assumptions, we consider that the cylindrical stem, which schematically represents drilling assembly (see Fig. 1) is subjected to influence of squeezing force P_p and twisting couples of M_t. Let us consider that under the influence of P_p and M_t (in the course of building of stem) drilling assembly can lose longitudinal stability, that is lateral buckling of assembly will take place (in two planes). Usually power factors at buckling of stem are determined from a condition of maintenance of stem in "slightly" curved position. The elastic line of assembly will not be a flat curve, but an analog of helical curve. According to the accepted orientation of system of coordinates, the moments formed by the longitudinal force P_p concerning axes y, z will be respectively equal to P_pZ and P_pY, and the moments of the twisting couple M_t presented by vectors will be equal (concerning the same axes) to

$$M_t\frac{dy}{dx};-M_t\frac{dz}{dx}\ .$$

At the first stage, considering P_p and M_t as constant values, it is possible to present the system of the differential equations of an elastic curve in each plane as follows:

$$\begin{cases} EI\dfrac{d^2z}{dx^2}=-P_pZ+M_t\dfrac{dy}{dx} \\ EI\dfrac{d^2y}{dx^2}=-P_pY-M_t\dfrac{d\dot{z}}{dx} \end{cases} \quad (1)$$

In the given system of equations:

P_p – the axial effort squeezing a drill stem; generally the variable value in time connected to impact of rock on roller bit (through resistance moment M_r); M_t – the torsional moment attached to rotator and transferred to drilling assembly (the variable depends significantly on M_r); l – length of drilling assembly (the value fixed for this timepoint in the course of drilling); E – elastic modulus of material of drilling assembly; I – inertia moment of a transverse section of drilling assembly (the value is constant for this drill stem); x, y, z – coordinates of points of assembly.

At the ends of assembly in case of its hinge support, we will have the following boundary conditions:

$$y\big|_{x=0}=0;\ y\big|_{x=l}=0;\ z\big|_{x=0}=0;\ z\big|_{x=l}=0. \quad (2)$$

Let us consider some functions which satisfy to initial system of the equations (1) and boundary conditions (2) under condition of hinge support of the assembly ends

$$\begin{cases} y=A\sin(m_1x+\alpha_1)+B\sin(m_2x+\alpha_2) \\ z=A\cos(m_1x+\alpha_1)+B\cos(m_2x+\alpha_2) \end{cases}, \quad (3)$$

Where A, B, α_1, α_2 – the constant integrated systems (3), and A, B – the maximum values of the functions.

Expressions (3) take place at a hinged support; if conditions of fixing of the ends differ, it is necessary to choose other functions.

In expression (3), m_1, m_2 are roots of the characteristic equation

$$E\,\mathrm{I}m^2+M_tm-P=0\ . \quad (4)$$

After substitution of expressions $y(x)$, $z(x)$ into boundary conditions (2), it is possible to obtain ratios between integration constants α_1, α_2 and proper numbers of system (3); in fact, these ratios do not depend on amplitudes A, B in a form:

$$\begin{cases} A\left[\sin(m_1l+\alpha_1)-\sin(m_2l+\alpha_1)\right]=0 \\ A\left[\cos(m_1l+\alpha_1)-\cos(m_2l+\alpha_1)\right]=0 \end{cases}. \quad (5)$$

Considering the same initial phase α_1 for the different additive components of first and second ratios (5), it is possible to conclude that the smallest value of a difference (m_1l-m_2l) will be equal to ±2π, that is

$$m_1l-m_2l=\pm 2\pi\ . \quad (6)$$

As m_1, m_2 are roots of a quadratic equation (4), we will obtain

$$m_{1,2}=-\frac{M_t}{2EI}\pm\sqrt{(\frac{M_t}{2EI})^2+\frac{P}{EI}}\ . \quad (7)$$

Then the equation with consideration of (7) will be

$$-\frac{M_t}{2EI}l+\sqrt{\frac{M_t^2}{4(EI)^2}+\frac{P}{EI}}l+\frac{M_t}{2EI}l+\sqrt{\frac{M_t^2}{4(EL)^2}+\frac{P}{EI}}=\pm 2\pi\ . \quad (8)$$

Then expression for determination of critically unstable condition of drilling assembly under the influence of squeezing force P_p and torsional moment M_t will be of the form

$$\frac{M_t^2}{4(EI)^2}+\frac{P}{EI}=\frac{\pi^2}{l^2}\ . \quad (9)$$

The expressions (1) – (8) allowed us to receive expression (9) for determination of critical condition of drilling assembly. For the description of drilling assembly behavior in case of a variable torsional moment M_t, it is necessary to consider in addition to system (1) a differential equation of tortional fluctuations of system "drive – rotator – drilling assembly – bit" under the influence of the variable moment of M_C that is a subject of further research.

Results, discussion, analysis

Thus, when calculating stability of drilling assembly for the known Euler formula, we find a significant error in results. Therefore, at the choice of optimum values of parameters of drilling modes (at use of devices on automatic control of drilling modes), it is necessary to recognize that automatic fine adjustment of the rig must be conducted at the same time by tortional moment M_t in a rotator and by the longitudinal effort arising in the feeder P_p.

When calculating of drilling assembly stability, the problem of the dynamic loads arising in the course of drilling is important. For an assessment of these loadings, we consider the fact that due to the variable moment of resistance in roller bit and "insignificant" rotating and longitudinal rigidness of drilling assembly together with a bit, there is a periodic braking and twisting of drill stem (which with essential lengths of drilling assembly reaches the considerable value). The pulsing nature of change of the moment of resistance to rotation of bit is the reason and source of emergence at first tortional, and then longitudinal fluctuations of a drilling assembly.

We determine the dynamic tortional moment for drilling assembly proceeding from balance of kinetic and potential energy of system "drilling assembly – roller bit (the moment of inertia of mass of bit and about a half of the moment of inertia of mass of drilling assembly are introduced into calculation).

$$\frac{I_m(\omega^2-\omega_1^2)}{2}=\frac{GI_p\varphi_d^2}{2l}, \tag{10}$$

where ω – rated angular frequency of rotation of drilling assembly; ω_1 – angular frequency (smallest) of rotations of drilling assembly in case of overcoming resistance forces when destruction of rocks (in the worst case situation ω_1=0); I_m – bit inertia moment taking into account association of inertia moments of a half of mass of assembly; G – rigidity modulus; I_p – an inertia moment of cross section of drill rod; φ_d – a dynamic angle of twisting of drilling assembly (value is variable or sign-variable); l – length of drilling assembly (value is divisible by a number of the rods in the assembly).

Let us calculate the dynamic tortional moment arising at torsion of drilling assembly in a zone of roller bit or the drive of rotator.

$$M_d=\frac{GI_p}{l}\varphi_d . \tag{11}$$

From expression (10) for balance of kinetic and potential energy we obtain

$$\varphi_d=\sqrt{\frac{I_m l(\omega^2-\omega_1^2)}{GI_p}} . \tag{12}$$

Then from (11) taking into account (12) we obtain

$$M_d=\frac{GI_p}{l}\sqrt{\frac{I_m l}{(GI_p)}(\omega^2-\omega_1^2)}=\sqrt{\frac{GI_pI_m}{l}(\omega^2-\omega_1^2)} . \tag{13}$$

In the most loaded case, we obtain expression for the moment M_d (the greatest dynamic moment)

$$M_d=\sqrt{\frac{GI_pI_m}{l}}\cdot\omega. \tag{14}$$

The greatest dynamic moment develops in assembly at a partial or full stop of bit; at that, the assembly is completely twisted and longitudinal deformations in it are minimum while the angular speed of rotation of assembly after a stop begins to increase, the assembly begins to untwist and longitudinal blow of bit to a face is very probable. Thus, the greatest value of M_d and the largest longitudinal force R_d cannot occur in drilling assembly simultaneously.

Let us take into consideration a case of longitudinal blow of assembly (bit) to a face. In this case, let us assume that kinetic energy at interaction of bit with the T face completely turns into the potential energy of deformation of assembly of U_d and kinetic energy of its movement T_1 caused by deformations (flexibility) of the feeder of rig (in this case we do not consider residual deformations of the massif at blow of bit to a face).

$$T=U_d+T_1 \tag{15}$$

Let us note that energy of deformation is generally used for squeezing of drilling assembly, at the same time the dynamic squeezing load exceeds much more the static one that often leads to loss of drilling assembly stability.

Let us equate the work of squeezing effort to potential energy of drilling assembly considering him it as centrally squeezed stem.

$$P(h+\delta_д)=\frac{EF\delta_d^2}{2l} , \tag{16}$$

where P – the effort (static) developed by the feeder; h – movement of drilling assembly, as continuous body, caused by a feeder flexibility (for example, cable polyspast system) and a gap between a face and roller bit; δ_d - dynamic shortening (lengthening) under action of dynamic effort of R_d on stem; l – the current length of assembly taking into account lengthening when assembly supply and its addition; EF – longitudinal rigidity of drilling assembly; E – module of longitudinal elasticity of drill rods; F – cross-sectional area of drill rod.

$$P(h+\delta_d)-\frac{EF}{2l}\delta_d^2=0\,. \qquad (17)$$

Shortening of stem $\delta_{st}=\frac{Pl}{EF}$ caused by static influence of feed force P allows us to write down the formula (17) in the following form

$$\delta_{st}\frac{EF}{l}(h+\delta_d)-\frac{EF}{2l}\delta_d=0\,. \qquad (18)$$

After transformations, the expression for determination of δ_d will be of the form

$$\delta_d^2-2\delta_d\delta_{st}-2\delta_{st}h=0\,. \qquad (19)$$

Then shortening of stem as a result of action of dynamic forces takes place

$$\delta_d=\delta_{st}+\sqrt{\delta_{st}^2+2h\delta_{st}} \qquad (20)$$

Let us introduce designation $k_d=\frac{\delta_d}{\delta_{st}}$ for dynamism coefficient at squizing of drill stem. Then from (20) we obtain dependence for calculation k_d in the form

$$k_d=1+\sqrt{1+\frac{2h}{\delta_{st}}}\,. \qquad (21)$$

Then the dynamic force in the drilling assembly can be represented in the form

$$P_d=k_dP\,, \qquad (22)$$

Thus, definition of k_d for the set parameters of certain modification of the drilling rig of the considered type anf dynamic loading can be found out by a formula (22). As appears from the conducted researches, dynamic loads in assembly R_d exceed static ones by two times.

Conclusions

From the stated material, it follows that the dynamic loads laid at design of drilling rigs are determined with considerable errors. They need to be eliminated at a stage of modernization of drilling rigs.

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