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The physical basis of the vibration of the upper part of the pipe string when drilling wells

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Abstract. The article discusses the fundamental issues related to the assessment of the dynamics of the top drive system as one of the sources of vibration when drilling wells. The operational parameters of the drive load at which a resonance phenomenon is observed are determined. The optimal parameters for reducing the amplitudes of forced oscillations, their resonance phenomena are revealed. Methods of protecting the derrick and equipment for protecting unbalanced masses of the top drive from vibration are considered. The prospect of using the layout of the columns, including alloy pipes, is to protect the bit.

1. Introduction

The casing hole drilling technique is currently undergoing its active stage of development. Like any innovative technology, in practice casing hole drilling technique faces a number of problems. A necessary casing hole drilling technique attribute is a top drive system (TDS) - a wellhead movable rotator that combines the functions of a rotor and a swivel [1].

One of the significant drawbacks of using the top drive is the vibration of the system "top drive (TDS) - drilling rig – casing" [2]. Vibration leads to failure of the drive mounts to the rig, a decrease in the strength of the casing, thread of the casing and the hoist rope due to the accumulation of fatigue damage. Premature wear of the diamond bit which is an emergency for casing hole drilling is also possible, as well as other harmful consequences that adversely affect the reliability of the system as a whole. The worn out drilling rig can aggravate the situation [3].

The forced vibrations of the top drive system were recorded by a full-scale experiment using a diagnostic complex designed to measure vibration on rotating mechanisms and stationary structures [4]. Amplitude of forced oscillations ranged from 5 mm and more. It is noted, in particular, that the oscillations had such a high amplitude that they were striking the monkey board.

2. The physical basis of the vibration of the upper part of the pipe string

The vibrations of the upper part of the pipe string are primarily due to the centrifugal forces from the unbalanced masses of the shaft and rotor impeller, as well as due to kinematic inaccuracies in the manufacture of pipes and rotator, rapid wear - due to severe loading conditions and other reasons [5].

We describe a model characterizing the forced nature of the oscillations of the top drive system – drilling rig - casing system, the mechanical model is presented in Figure 1.

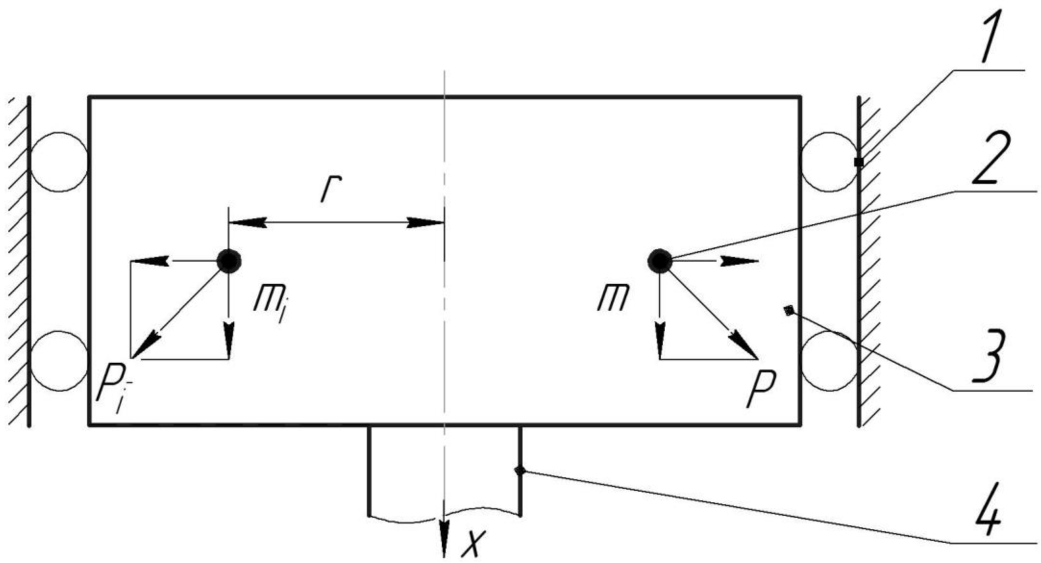

Figure 1. Mechanical model. Unbalanced rotary shaft mass of a drilling rig: 1 – guide rails; 2 - unbalance; 3 – top drive system drive box; 4 - output shaft (pipe string); mi are eccentric masses; r - eccentricity.

The equation describing the model can be represented as:

$$\frac{d^2x}{dt^2} + 2n\frac{dx}{dt} + k_d{}^2 x = p_{\max}\sin(wt+\varphi) \tag{1}$$

where: $k_d{}^2 = \dfrac{k}{M}$ is the dynamic stiffness of the system; $n = \dfrac{\beta}{2M}$ is the coefficient of friction of the medium; $p_{\max} = \dfrac{P_{\max}}{M}$ is the inertia force.

The oscillations have a localized source, they extend to the entire of "top drive system – drilling rig - casing": on the one hand, through the guide and hook to the derrick and drilling equipment, on the other – through the output shaft of the rotator to the pipe string.

Equation (1) is a second-order heterogeneous differential equation. The general calculation of x of such an equation is found in the form:

$$x = X + \tilde{x} \tag{2}$$

where: X is the general solution of the homogeneous equation; $\tilde{x}$ is a particular solution of the inhomogeneous equation.

A particular solution $\tilde{x}$ (2) is obtained in the form:

$$\tilde{x} = H\sin(wt+\varphi_0) \tag{3}$$

where: $H = \dfrac{p_{\max}}{\sqrt{(k_d{}^2 - w^2)^2 + (2nw)^2}}$ is an amplitude of forced oscillations; $\varphi_0 = \dfrac{2nw}{\sqrt{(k_d{}^2 - w^2)^2 + (2nw)^2}}$ is an initial phase of forced oscillations.

The general calculation of x can be written as:

$$x = e^{-nt}(C_1 \cos k_1 t + C_2 \sin k_1 t) + H \sin(wt + \varphi_0) \tag{4}$$

where: C_1, C_2 are constants determined by the initial and boundary conditions of motion; $k_1 = \sqrt{k_d^{\,2} - n^2}$ is the free vibrations frequency (natural frequency).

Dynamic stiffness is a function of the stiffness characteristics of the top drive system and the guide. The value of n reflects the influence on the frequency of natural oscillations of the resistance to movement (friction, medium, and wind loads). The values for n are determined on the base of field studies of the natural vibrations of the top drive system device and the guide.

We consider the value of the amplitude of forced vibrations H. For this we transform expression (3) by dividing by k_d^2 / k_d^2, substituting the value for p_{max} and designating: $U=w/k_d$ is the dimensionless rotation frequency (forced oscillation frequency), $Y = n / k_d$ is the damping coefficient. As a result:

$$H = \frac{U^2}{\sqrt{(1-U^2)^2 + 4Y^2U^2}} \cdot \frac{\sum_{i=1}^{N} m_i r}{M} = H(U) \cdot H^* . \tag{5}$$

We show the logarithmic decrement of attenuation:

$$\delta = \ln e^{nT} = nT = n\frac{2\pi}{k_1} = \frac{2\pi n}{\sqrt{k_d^{\,2} - n^2}} . \tag{6}$$

We show the change in amplitude, since the value Y <1. From the formula associated with the logarithmic damping decrement (6), we obtain the dependence:

$$Y^2(\delta) = \frac{\delta^2}{4\pi^2 + \delta^2} . \tag{7}$$

3. Results

To simplify the calculations, we accept steel grades for casing and drill pipes to be the same. According to experimental studies [6] for a steel pipe with a diameter of 114 mm, the range of the logarithmic damping decrement δ ranges from 0.16 to 0.43, and for the ENAW-2024 alloy (duralumin) of a light alloy drill pipe with a diameter of 98 mm from 0.27 to 0.67.

In this case, for (7) we obtain numerically: the maximum, minimum, and average values of the damping coefficient. For ordinary steel, they will be equal respectively: $Y^2(0.43)\cdot 4 = 0.019$; $Y^2(0.16)\cdot 4 = 2.6\cdot 10^{-3}$; $Y^2(0.30)\cdot 4 = 9.1\cdot 10^{-3}$. For duralumin $Y^2(0.67)\cdot 4 = 0.045$; $Y^2(0.27)\cdot 4 = 7.4\cdot 10^{-3}$; $Y^2(0.47)\cdot 4 = 0.022$.

Thus, based on these values, we can construct the corresponding *H(U)* curves (Figure 2 and Figure 3).

According to Figure 2 and Figure 3, the extremum of the amplitude value *H(U)*, corresponding to the resonance phenomenon occurs at $U = 1$ ($w = k_d$), and the optimal amplitude values are found at values $U \geq 3$ ($w \geq 3k_d$), while $H(U) \to 1$, and $H \to H^*$.

We consider the resonant amplitude. Taking in the expression for *H(U)* *U* equal to one, we obtain:

$$H = \frac{1}{2Y} H^* = H_{rez} H^* \tag{8}$$

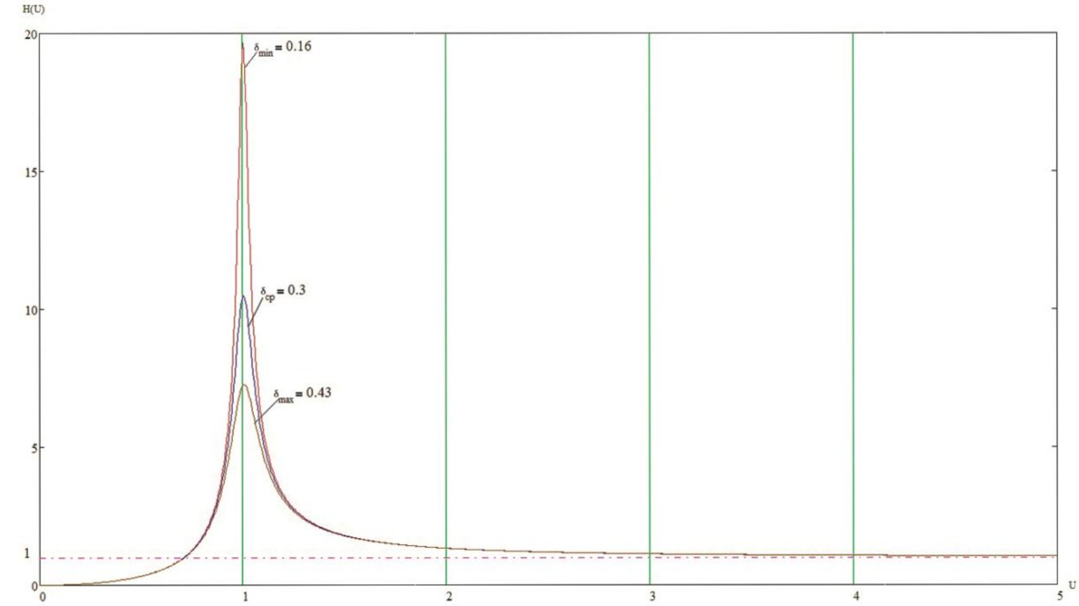

Figure 2. Frequency response curves depending on the value of the logarithmic attenuation decrement δ for a steel pipe.

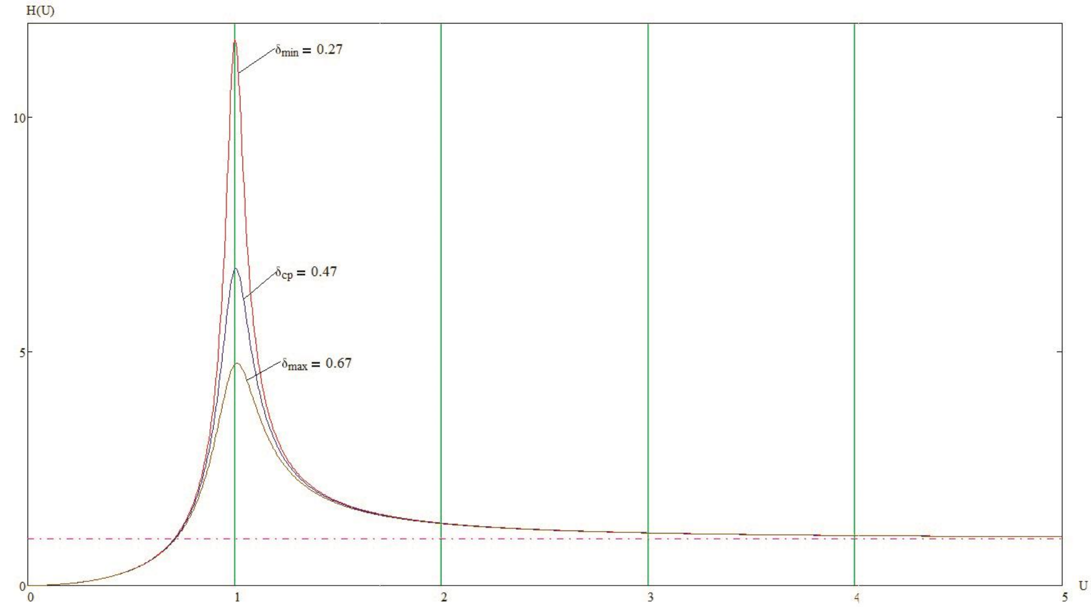

Figure 3. Frequency response curves depending on the value of the logarithmic attenuation decrement δ for an alloy pipe.

According to the data of (8), it can be seen that, for $Y < 0.5$ ($n < 0.5 \cdot k_d$) a sharp increase in the resonance amplitude occurs according to the hyperbolic law, with $H_{rez} > H^*$.

Thus, the optimal parameters to increase the reliability of the system can be written in the form:

$$\begin{cases} w \geq 3k_d; \\ n > 0.5k_d. \end{cases} \tag{9}$$

In addition, according to the graphs (figures 2 and 3) it is shown that the logarithmic damping decrement plays a significant role: as it increases from minimum to maximum, the amplitude at resonance (U = 1) for steel decreases 3 times, for ENAW-2024 alloy – 2.5 times.

We compare the average values of the logarithmic attenuation decrements for steel and alloy pipes. The damping ability of the ENAW-2024 alloy is 50% more than ordinary steel, while the amplitude at resonance is proportionally less by 50. Moreover, the larger the pipe diameter is, and, consequently, its thickness, the greater the damping ability will be demonstrated by pipes based on ENAW-2024 alloy.

4. Discussion

In view of the foregoing, it makes sense to consider the possibility of using a vibration damper of torsional and longitudinal vibrations in a layout with a top drive [7], as well as pipes with an alloy based on aluminum, in particular ENAW-2024, which, unlike steel, has a lower coefficient of propagation of the elastic wave.

As a fact, one of the most dangerous situations for casing hole drilling is not bringing the string to the design value. In this case, it is not possible to open a drill as when drilling on a drill string and changing the bit. One of the reasons for this emergency situation is the premature wear of the casing thread and / or diamond teeth of the shoe bits due to vibration from the top drive system acting on them.

Vibrations are detrimental to PDC bits, they cause teeth to fall out and the extra work to grind them, resulting in the remaining teeth wear out faster. Particularly dangerous in this case are directional drilling and the conductor casing because the oscillation amplitudes will be large enough [8].

It is also known that when drilling on "heavy" casing pipes, the required margin of safety of the drilling rig increases, that is, in the general case, its carrying capacity. Thus, in conjunction with vibration of the top drive, the system “top drive system – drilling rig - casing” will have rather low reliability indicators.

An alternative in this situation can be the use of the layouts of conventional and alloy casing pipes, in particular, based on the ENAW-2024 alloy, at least for directional drilling and conductor casing, by installing them in the lower part of the pipe string [9]. As already shown, ENAW-2024 alloys have fairly high damping characteristics, linearly increasing with the diameter of the pipes, and also have the ability to reflect part of the vibration energy.

In addition, light-alloy pipes have a much lower weight, and their relative high cost is compensated by a decrease in the number of equipment failures and high rates of commercial drilling speed when casing hole drilling [10].

5. Conclusion

Based on studies of the dynamics of the rotator of the drilling rig, optimal system parameters have been obtained that characterize the nature of the forced vibrations of the "top drive system – drilling rig – casing" system. The essential role of the logarithmic damping decrement in damping the forced oscillations of the top drive system is shown. Possible solutions to increase the reliability of the system are proposed, consisting in using the layout of conventional and alloy casing pipes based on ENAW-2024, as well as incorporating a vibration damper into the top drive.

References

[1] Abrahamsen E andr Reid D 2007 *SPE/IADC Drilling Conf.* (20-22 February, Amsterdam, The Netherlands) 105822
[2] Ashcheulov A V et al 2015 *Chem. Petrol. Eng.* **51(1-2)** 94–9
[3] Jansen J D and Van den Steen L 1995 *J. Sound Vib.* **179(4)** 647–68
[4] Minihanov R F 2008 *Mining equipment and electromechanics* [in Russian – Gornoye oborudovaniye i elektromekhanika] **11** 14–8
[5] Petrovskii E A, Bashmur K A and Nashivanov I S 2019 *Chem. Petrol. Eng.* **54(9-10)** 711–6

[6] Luk'yanov E Ye, Molchanov A A and Rapin V A 2001 *Geophysical Exploration of Horizontal Oil and Gas Wells* [in Russian – Geofizicheskiye issledovaniya gorizontal'nykh neftegazovykh skvazhin] (International Academy of Ecological Sciences)
[7] Bashmur K A et al 2019 *J. Phys.: Conf. Ser.* **1353** 012039
[8] Warren T M and Oster J H 1998 *SPE Annual Technical Conf. and Exhibition* (27-30 September, New Orleans, Louisiana) 49204
[9] Yanturin R A and Yanturin A Sh 2008 *Construction of Oil and Gas Wells on Land and Sea* [in Russian – Stroitel'stvo neftyanykh i gazovykh skvazhin na sushe i na more] **1** 5–9
[10] Patel D et al 2019 *Petroleum* **5(1)** 1–12