VIBRATION PROTECTION OF ROLLER-BIT DRILL RIGS
FOR OPEN-CUT MINING

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Owing to the increased drilling speeds and powers of drills for open-cut mining, the dynamic loads on the drill rigs are increasing, and vibration is having an increasing effect on their operation.

The main sources of vibration are the compressors on the drill-rig platform and the dynamic interaction between the bit and the rock during its deformation and fracture.

A group of measures is now being developed to protect the rigs and their components from vibration, namely, suspension of the operator's cabin, damping of the compressor vibrations, use of elastic couplings between the rotor and the rods, etc. The destructive action of vibrations on the bit can often be eliminated by fitting a vibration guard above the bit, thus improving the efficiency of the whole system by boosting the drilling rate.

The aim of this article is to determine the influence of the vibration-guard characteristics on the efficiency with which it damps the vibrations.

If we neglect linear losses, we can represent the longitudinal vibrations of a drill rod by the following equations:

\[
\begin{align*}
\frac{d}{dx} u(x, p) &= L p v(x, p), \\
\frac{d}{dx} v(x, p) &= C p u(x, p),
\end{align*}
\]

where \(u(x, p)\) and \(v(x, p)\) are the Laplace transforms of the forces and velocities, respectively, in sections \(x\); \(L = \rho S\) is the mass of the drill rod per unit length, \(\rho\) is the density of the material, \(S\) is the cross-sectional area, \(C = 1/ES\) is the flexibility of the drill rod per unit length, and \(E\) is the modulus of elasticity of the material.

The boundary condition at the top end \(x = l\) is

\[u(l, p) = Z_l v(l, p)\]

If we take account only of inertial and elastic properties of the bit feed system,

\[Z_l = p L + \frac{1}{p C}\]

Assuming that at any time the mechanical drilling speed (the speed of advance of the bit) is equal to the speed of advance of the instrument, i.e., eliminating from consideration the oscillations of the feed speed, the frequency spectrum of which lies well below the spectrum of vibrations of the bit-face system, we can write

\[u(0, p) = f(p) - Z_0 v(0, p),\]

where \(f(p)\) is the Laplace transform of the disturbances applied to the bit, and
where \( L_g, C_g, \) and \( R_g \) are the parameters of the vibration guard above the bit and \( Z_z \) is the input operator resistance of the face.

Assuming that the stress waves arising at the face as a result of impact by the roller-bit teeth are completely absorbed on propagation through the rock, we can model the face by a semi-infinite cylinder with linear parameters \( L_z, C_z, \) and \( R_z \). In this case the input resistance of the face is equal to its wave impedance,

\[
Z_z = \sqrt{\frac{L_z}{C_z}} \sqrt{1 + \frac{R_z}{\rho L_z}}
\]

or, expanding the expression as a power series and discarding terms after the first two,

\[
Z_z = \sqrt{\frac{L_z}{C_z}} \left( 1 + \frac{R_z}{\rho L_z} \right).
\]

The solution to system (1)-(3) can be written in the form

\[
\begin{align*}
&u(0, p) = \frac{1}{1 + Z_0 V_{in}} f(p) = \frac{g_0 Z_{in}}{1 + g_0 Z_{in}} f(p), \\
&v(0, p) = \frac{Y_{in}}{1 + Z_0 V_{in}} f(p) = \frac{g_0}{1 + g_0 Z_{in}} f(p),
\end{align*}
\]

where

\[
\begin{align*}
g_0 &= \frac{1}{Z_0}; \quad Y_{in} = \frac{1}{Z_{in}} = \frac{Z}{Z_f + th \frac{L}{a} p}; \\
Z &= \sqrt{\frac{L}{C}}; \quad a = \frac{1}{\sqrt{LC}}.
\end{align*}
\]

It is difficult to find the inverse transforms of (4) owing to the transcendental expressions in it. Even if we had the inverse transforms, we would have a complicated multivariate problem of choosing the characteristics of the vibration guard. Thus it is convenient to solve Eq. (4) approximately in rational fractions, which corresponds to replacing the original distributed system by a system with a finite number of degrees of freedom and then simulating it on an analog computer.

We will use the expansion of the function \( \tanh \left( \frac{L}{a} p \right) \), which corresponds in the time domain to the series

\[
\tanh \frac{L}{a} p \approx \frac{2}{\pi} \sum_{n=1}^{\infty} \frac{1}{n - \frac{1}{2}} \sin \left[ \left( n - \frac{1}{2} \right) \frac{\pi a}{L} p \right].
\]

Limiting ourselves to \( m \) harmonics (i.e., putting the upper limit of the sum equal to \( m \)), we get

\[
\tanh \frac{L}{a} p \approx \frac{2a}{L} \sum_{n=1}^{m} \frac{p}{p^2 + \left[ \left( n - \frac{1}{2} \right) \frac{\pi a}{L} \right]^2},
\]

The root-mean-square error of approximation (5) is

\[
\delta \approx 0.4 \sqrt{\sum_{m+1}^{\infty} \frac{1}{(2n-1)^2}}.
\]