STUDY OF THE DYNAMIC STABILITY OF THE TRANSMISSION SHAFTS OF LARGE MINE-VENTILATION FANS

N. N. Petrov, A. M. Krasyuk, and N. A. Popov

UDC 622.44

Reliable operation of the main ventilation unit (MVU) is one of the most important prerequisites to the stable and safe operation of mines. The most important parameter characterizing the reliability of such units is the accrued operating time of the main components and parts, i.e., those components and parts whose malfunction could result in a major accident. A statistical analysis of MVS malfunctions made in [1] showed that up to 75% of the failures of the axial fans could be attributed to failures of the transmission shafts and the toothed half-couplings.

In [2], MVSs in which the axial fans have a vertical axis of rotation (Fig. 1a) were shown to have several advantages over MVSs with a horizontal axis of rotation.

Estimates made in [3] of the residual static strength of the shafts of VOD-30 horizontal axial fans and VO-30VK fans (with a vertical axis of revolution) showed that their numerical values were 4.0-4.2 times greater than the norms [4]. Thus, that parameter is not the reason for the unreliable performance of the transmission shafts of existing units. It is proposed here that the equipment failures are due to dynamic instability of the transmission shafts of large fans during flexural and torsional vibrations.

The goal of this investigation is to analyze dynamic instability in the transmission shafts of large axial fans during flexural and torsional vibrations.

ANALYSIS OF FLEXURAL VIBRATIONS OF TRANSMISSION SHAFTS

The flexural vibrations of transmission shafts are calculated to determine the frequencies of natural vibration. The forces that cause such vibrations in the transmission shafts of mine fans are due to transients which develop during operation of the ventilation unit (such as during startup, speed adjustments, and shutdown of the fan) and to various types of perturbations that arise. Among the latter are: 1) structural perturbations connected with an imbalance between the rotating blades and parts of the rotor and with inadequate stiffness in the shaft bearings; 2) process perturbations dependent on slag inclusions, voids, pores, and changes in the crystalline structure of the material of the shaft; 3) manufacturing perturbations due to deviations from the dimensions in the drawings during the fabrication of the parts, imprecise dynamic balancing, and poor-quality assembly; 4) service perturbations connected with wear of the toothed half-couplings and elastic deformation of the bearings of the fan-rotor and transmission-shaft systems.

All of the above perturbations and may other types of disturbances help create periodic mechanical and aerodynamic forces that are excited within a broad range of frequencies.

Transmission shafts are usually regarded as nonrotating multispan beams. In design calculations for vertical shafts, it is customary to ignore the effect of longitudinal compressive forces, gaps in the bearings and toothed half-couplings, and the compliance of the bearings [5].

The natural frequencies of flexural vibration are determined from the solution of differential equations.

For a beam whose cross section is small compared to its length, the differential equation that describes its flexural vibration has the form [5, 6]:

\[
EJ \frac{\partial^4 y}{\partial x^4} + \rho h \frac{\partial^2 y}{\partial x^2} = q,
\]

(1)
Fig. 1. Diagrams used to design transmission shafts of the main ventilation units of mines for dynamic stability: a) vertical unit with a VO-30VK fan; b) design diagram of ventilation unit; 1) fan; 2) transmission shaft; 3) electric motor of drive; 4) compartment door; 5) diffuser with revolving knee; 6) rotor of electric motor; 7) shaft of rotor; 8) toothed coupling; 9) radial bearing (elastic-damping support); 10) toothed coupling with hinge pivot; 11) radial bearing (intermediate support); 12) shaft of fan rotor; 13) fan rotor; 14) radial-thrust bearing; S_i) i-th cross section; L_i) length of a section of the shaft between cross sections.

where EJ is bending stiffness, N·m²; E is the elastic modulus, N/m²; J is the moment of inertia of area, m⁴; y is the deflection of the beam, m; ρ is the density of the material of the beam, kg/m³; F is the cross-sectional area of the beam, m²; q is the intensity of the load acting on the beam; x is the length of a section of the beam, m.

If there are no external forces, then the right side of Eq. (1) vanishes. This situation is typical of vertical transmission shafts. Then the natural flexural vibrations of transmission shafts on two bearings will be found from the homogeneous differential equation

$$EJ \frac{\partial^4 y}{\partial x^4} + \rho F \frac{\partial^2 y}{\partial x^2} = 0.$$  (2)

The solution of Eq. (2) for a beam lying on two bearings shows that, without allowance for the effect of the gyroscopic couple, the critical natural frequencies of vibration for rotating circular transmission shafts with a constant cross section are determined from the formula [4, 5]

$$\omega_n = \frac{n^2 \pi^2 L_2^2}{m},$$  (3)

where n = 1, 2, 3, ... is the number of the mode of vibration; L_2 is the length of the shaft, m; m is the mass of a unit length of the shaft, kg/m.

The transmission shafts of mine fans may be made either solid or hollow (from pipes). The latter allows a significant reduction in weight, assuming that the safety factor is large enough.

If we take Eq. (3) and insert expressions for the moment of area of a solid shaft J = πD⁴/64 and the mass of that shaft m = πρD²/4, we can represent Eq. (3) in the form

$$\omega_n = \frac{n^2 \pi^2 D}{4L_2^2} \sqrt{\frac{E}{\rho}},$$  (4)