Analysis of dynamic parameters of mine fans

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Abstract. The design of the rotor of an axial fan and its main units, namely double leaf blades impeller and the main shaft are discussed. The parameters of a disturbed mine air flow under sudden outbursts are determined and the influence of disturbances on frequencies of axial fan units is assessed. The scope of the assessment embraces the disturbance effect on the blades and on the torsional vibrations of the main shafts. The dependences of the stresses in the elements of the rotor versus the disturbed air flow parameters are derived.

Service reliability as the prime specification of the main mine fan facilities in most depends on the ultimate safety factor and vibration level of fan modules, which are under permanent influence of airflow disturbances, induced by unsteady aerodynamic forces arising as a consequence of irregular flows in interaction with ribs of the body, blades of guiders and straighteners [1], as well as disturbances induced by sudden outbursts and blasting.

The main module of the mine axial fan is rotor. Its reliability determines fan performance as a whole. Rotor parts are main shaft and an impeller consisting of a body and working blades [2]. Let consider the main mine fan facility of VO–series [2], which impeller has 8 double leaf welded–design blades. The main fan shaft is connected to transmission shaft through coupling; the transmission shaft is coupled kinematically with electromotor shaft. The fan schematic is shown in Figure 1.

Figure 1. VO fan schematic: 1—synchronous electromotor of 630 kW, 750 rpm, 6.0 kV; 2—electromagnetic brake; 3—impeller with double leaf turning-in-motion blades; 4—unturning blades of a straightening unit; 5—mechanism for concurrent turning of impeller blades; 6—diffuser.
Under consideration is interaction of a disturbed air flow with a double blade of an axial fan.

Figure 2 presents the scheme of a double leaf blade of VO-36K axial fan [2]. Specifications of VO-36K fan are diameter within blade ends $D = 3.6$ m, static pressure $P_{sv} = 2550$ Pa, electromotor capacity $N = 2000$ kW.

![Figure 2. Double leaf blade (schematic): 1–large blade, 2–small blade, 3–rotatable base, 4–partition.](image)

Eigen frequencies of a double leaf blade for VO-36K are calculated in Ansys package (Figure 3; (a, b)–bending-type vibrations, (c)–torsion vibrations).

![Figure 3. Forms of eigen vibrations of RK double leaf blade: (a) the first vibration form (66.5 Hz frequency), (b) the second vibration form (101.5 Hz frequency), (c) the third vibration form (165.1 Hz frequency).](image)

Eigen frequencies of vibrations are 66.5 Hz in the first vibration form, 101.5 Hz in the second, and 165.1 Hz in the third (Figure 3). The principal exciting frequency is the rotor rotation frequency equal to 10 Hz. Appreciable detuning of eigen frequencies from the principal exciting ones is observed.

Along with the common loads: centrifugal forces, air flow resistance, thrust and inertia of an electromotor the fan blades experience cyclic-in-time disturbing loads which are capable to induce additional dynamic stresses in blades [3]. Coincidence of a disturbing force frequency with one of eigen blade frequencies generates resonance phenomenon, characterized with a substantial increase in vibration amplitudes. Resonance phenomena are also observed when the eigen frequency is not equal, but multiple to the exciting force frequency.
Resonance frequency values for air flow oscillations from ribs of a guider, straightener blades and loss of rotating self-oscillations can be written as [5]:

\[
\omega_n^{(P)} = n N_{p} \omega, \quad \omega_n^{(CA)} = n N_{CA} \omega, \quad \omega_n^{(VO)} = n N_{VO}(1-\alpha) \omega,
\]

where \( n = 1, 2, 3, \ldots \) is number of exciting force harmonics; \( \alpha \) is angle between incoming air flow and the blade profile chord, \( 0 < \alpha < 1, \text{ rad} \); \( \omega_n^{(P)} \), \( \omega_n^{(CA)} \), \( \omega_n^{(VO)} \) are resonance frequencies of airflow oscillations from ribs of guider, straightener blades, and loss of rotating self-oscillations, \( \text{rad/s} \); \( \omega \) is angular frequency of rotor rotation, \( \text{rad/s} \).

We have to plot a vibration diagram in order to determine the influence of disturbing frequencies on blade vibrations (Figure 4).

![Figure 4. Relationship of eigen frequencies of double leaf blade vibrations and disturbing frequencies vs. the electromotor rotation number: 1—first eigen frequency; 2—second eigen frequency; 3—third eigen frequency; 4—disturbing frequency of guider’s blades; 5—disturbing frequency of straightener’s blades; 6—disturbing frequency of a vortex separation.](image)

It follows from the analysis of the curves, when a fan is actuated, the blades pass through several resonance zones (Figure 4). This situation does not lead to a rise of dangerous stresses and strains in the blade section structure because of a short time of being in these zones and a small-scale energy of disturbed air flow on the part of ribs of a guider, straightener blades, and vortex separation disturbances. In the case of run-down operation of the fan a brake for the fan electromotor should be provided to cut down time of being in resonance zones.

Calculation and analysis of torsion vibrations of transmission shafts of fan facilities is practiced to evaluate the maximum stresses and torsion angles (amplitudes) of the transmission shaft cross-sections versus angular velocity of electromotor rotor in the periods of run-up and run-down of the fan and also when aerodynamic disturbances, strong in rotation moment, be precise, an explosion or a sudden methane outburst affect a fan facility operation. Structural schematic of a fan for evaluation of torsion vibrations is shown in Figure 5.

![Figure 5. Structural schematic of a fan for evaluation of torsion vibrations.](image)

The mathematical model of fan rotation is described by a differential equation set for the mechanical system with four degrees of freedom [5].

Calculation results on torsion deformation of VO-36K transmission shaft are reported in Figure 6.

![Figure 6. Graph of relationship of tangential stresses in a material of the transmission shaft versus time in the launch of VO-36K fan facility.](image)
The excess of the maximum tangential stresses ($\tau_{\text{max}}$) over stresses at nominal frequency ($\tau_n$) is:

$$\frac{\tau_{\text{max}}}{\tau_n} = \frac{30 \times 10^6 \text{ MPa}}{6.05 \times 10^6 \text{ MPa}} \approx 5.$$  

![Diagram](image)

**Figure 5.** Structural scheme of a fan facility: 1–radial bearing; 2–actuating electromotor; 3–gear couplings; 4–transmission shaft; 5–main rotor shaft; 6–impeller; 7–angular-contract bearing; $M_d$–torsion moment of electromotor; $\varphi_i$–angular coordinate if the $i$-th cross-section; $c_{i,i+1}$–torsion rigidity of a shaft section between the $i$-th and $(i+1)$-th cross-section; $J_1$–inertia moment for electromotor rotor; $J_4$–inertia moment for fan rotor; $J_2, J_3$–inertia moments for gear couplings.

![Graph](image)

**Figure 6.** Relationship of torsion deformations of transmission shaft of the rotor (curve 1) and frequency of rotor rotation (curve 2) versus time in start of VO-36K fan facility.

![Graph](image)

**Figure 7.** Relationship of tangential stresses in torsion of the transmission shaft of the rotor versus time in the start of VO-36K fan facility.

In a fan run down the resistance moment in an impeller against an air flow is much less than the electromotor moment in start, that is why vibration amplitudes are not significant $3.3 \times 10^{-13}$ rad.
Therefore, it is possible not to use a brake for cutting down the run-up time, viz., a time of being in a resonance zone. The brake will be helpful for technological needs, for example, to repair a fan facility.

The experimental evidence of the present research enables to infer:

– service resource of transmission shaft and rotor shaft mainly depends on a number of fan starts and a number of sudden outbursts;

– it is possible not to use the brake to cut down the time of running down, viz., to cut time of being in a resonance zone. The brake is helpful for technological needs, let, fan repair;

– to reduce probability of rotation loss it is preferable to exploit a fan with impeller blade mounting at angles not exceeding the rated parameters.

Considering a short-term start mode (less than 15 s) and a small probability of high-power outbursts in the vicinity of a fan facility, the service life of transmission shaft should be determined in terms of a number of the fan starts and expectant sudden outbursts.

References

[1] Russky EYu 2015 Vibrational reliability of rotors of main mine fans GIAB No 7 pp 168–174
[2] Klepakov IV and Rudenko VA 1986 Design of a new line of axial main mine fans, in: Theoretical and Operational Problems of Stationary Mine Installations Donetsk: Fedorov VNIIGM pp 110–121 (in Russian)
[3] Manushin EA and Surovtsev IG 1990 Design and Strength Analysis of Turbomachines of Gas-Turbine and Combined Installations Moscow: Mashinostroenie (in Russian)
[4] Krasyuk AM, Lugin IV, Popov NA and Russky EYu 2015 Substantiation of parameters and estimation of strength for basic structural units of axial tunnel fan Journal of Mining Science Vol 51 No 6 pp 1139–1149
[5] Kovchin SA and Sabinin YuA 2000 Theory of Electric Drive: university textbook Saint-Petersburg: Energoatomizdat 2000 (in Russian)
[6] PanovkoYaG 1960 Internal Friction under Vibrations in Elastic Systems Moscow: Fizmatgiz (in Russian)
[7] Russky EYu 2015 Strength Analysis of Axial Mine Fans Rotors Vestn. KuzGTU No 2 pp 31–34
[8] Krasyuk AM and Russky EYu 2013 Analysis of stress state and frequency properties of impellers of axial main mine fans GIAB No 8 pp 152–156