DEVELOPING AND TESTING A SERVOMECHANISM FOR VARYING THE PITCH OF THE BLADES OF AN AXIAL FAN

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The deepening of existing horizons in mines and pits and the intensification and concentration of mining operations are raising the requirement of further improvement in the ventilation of mine workings.

Automatic control of ventilation presupposes automatic regulation of the operation of the main fans in a pit system. In systems with axial fans, deep and economic regulation of the process conditions is achieved by feathering the blades of the fan rotor. To secure automatic control of the operations and reversal of the air flow, the fan must be fitted with a servomechanism for simultaneous rotation of the fan blades in action.

The main requirements for the construction of a servomechanism are as follows: I) smooth rotation of the blades through 120°; II) simultaneous rotation of the blades through equal angles; III) possibility of using the system as the effector mechanism in an automatic control system for the fan; IV) automatic control of the blade positions; V) sufficient reserve of thrust and rate of action; VI) possibility of simultaneous rotation of the blades by hand when the rotor is stationary; VII) remote control of the blade positions; VIII) automatic disengagement of the servomechanism with the blades in a given position; and IX) reliability, i.e., serviceability and ease of repair.

There are several types of servomechanism, differing in the type of drive motor (with rotating or reciprocating force elements) and in the method of converting displacements of the force element of the motor to angular displacement of the fan blades. The simplest servomechanism to construct and the most reliable in action makes use of hydraulic cylinders as motors, and cranks and connecting rods to turn the blades. This was the principle on which was based the blade feathering of an experimental fan built at the Institute of Mining of the Siberian Branch of the Academy of Sciences of the USSR (Fig. 1).

The fan blades 1 are fixed on bushes 2 in the fan hub 3. Dismountable hydraulic cylinders 4 are hinged to hub 3 and adjustable crosspiece 5, which is moved by them along a slot in the fan shaft. Crosspiece 5 is connected via cranks and links to the tails of the blades. When the crosspiece moves along the fan shaft, the blades are simultaneously turned through equal angles up to 120°. The servomechanism is fitted with three stops 7 which, via intermediate elements, disconnect the hydraulic system from the pressure source when the blades are at 15, 45, or 135°. The working fluid is fed to the rotating part of the fan via head 8 and grooves and holes 9 in the shaft.

The servomechanism is fitted with a transducer (of which the moving element is linked to the tail of one of the blades) which yields feedback from the fan blades when the servomechanism is used as the effector of an automatic blade control system or for remote control of the blade positions.

The electrical circuits are connected to the rotating part of the fan via contact system 10.

Since this is the first time that such a servomechanism for fan blade control has been discussed, we have no method of determining its operation characteristics. The main characteristic determining the operation of a servomechanism and the construction of its components from the standpoint of strength is the load on its motor. This is the maximum force required to be applied to the crosspiece to rotate the blades at a given rate throughout their range of angles.

The motor power of the servomechanism will be found from known relations [1] and the calculated maximum load for the given type of fan and the required speed of operation.

The necessary actuating force of the hydraulic cylinders is found from the appropriate recommendation [2] as

$$T = k_l P \text{ newtons,} \tag{1}$$

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The moment M_A is found as the friction in the radial-thrust bearings due to the action of M_a in the yx plane, due to the action of the resultant aerodynamic force A which can be calculated by means of a formula given by I. M. Khumakhov:

$$M_A = \frac{a_1 f d}{2} + \frac{a_2 f d}{2}, \tag{4}$$

where a_1, a_2 are the reactions in the radial-thrust bearings of the blades.

The moment M_Q is found as the friction in the radial-thrust bearings of the blades under the action of a moment, found as

$$M_Q = Q\, l, \tag{5}$$

where l is the deviation of the center of gravity of a blade from its axis of rotation, in meters.

When the blades are inclined at less than 90°, the aerodynamic moment M_{ae} acts to reduce the angle, whereas when they are inclined at more than 90° it acts to increase it, and becomes a maximum at angles close to 45°:

$$M_{ae} = h\, A, \tag{6}$$

where h is the deviation of the line of action of the resultant aerodynamic force A from the axis of rotation of the blades, in meters.

From the aerodynamic profile we know that $h_{max} = 0.25\, b_1$, where b_1 is the length of the average chord of the blade profile in meters.

The inertial load P_{in} depends on the mass and moments of inertia of the moving parts of the servomechanism and also on the accelerations of its moving elements.

The acceleration of the servomechanism depends on the required rapidity of action. In this case, by "rapidity of action" we mean the time necessary to rotate the fan blades from one extreme position to the other. Using the published recommendations [2] on the choice of servomechanism times and the characteristics of the fan-network system as an object of control, we can recommend that the time should be $T_{dis} = 15$ sec.

The inertial load arising when the servomechanism is started up can be found by means of the theorem on the momentum and impulse of a force:

$$P_{in} = \sum m \frac{\Delta v}{\Delta t} + \sum L \rho F \frac{\Delta v}{\Delta t}, \tag{7}$$

where L is the distance of change of velocity in meters, m is the reduced mass of the servomechanism in kilograms, Δv is the drop in the velocity of the crosspiece in m/sec, Δt is the time of change of the speed of the moving parts of the servomechanism in seconds, ρ is the density of the working fluid in kg/m^3, and F is the cross-sectional area of the cylinders in m^2.

The acceleration $\Delta v/\Delta t$ of the servomechanism is governed by the duration of a transient process in the hydraulic blade-swivelling system which varies between $\Delta t = 0.01$ and 0.5 sec.

For the velocity drop we can substitute its average value,

$$v_{av} = S / T_{dis}, \tag{8}$$

where S is the travel of the crosspiece along the fan shaft when the blades turn through 120°.

Regarding the transient process of starting up the servomechanism as a uniformly accelerated motion, we can find L as

$$L = \frac{1}{2} v_{av} \Delta t. \tag{9}$$

The mass of the servomechanism, reduced to the actuating disk by means of the usual reduction formulas [3], can be represented as follows:

$$m = m_1 + n_1 m_2 + n_2 m_3 \left(\frac{v_c}{v_A}\right)^2 + n_2 m_4 \left(\frac{v_{c_1}}{v_A}\right)^2 + n_2 j_3 \left(\frac{\omega_c}{v_A}\right)^2 + n_2 j_4 \left(\frac{\omega_c}{v_A}\right)^2, \tag{10}$$

where m_1 is the mass of the actuating crosspiece in kilograms, m_2 is the mass of the pistons with coupling rods and the oil in the coupling-rod cavities of the hydraulic cylinders in kilograms, m_3 is the mass of the connecting rod in kilograms, m_4 is the mass of the crankshaft in kilograms, n_1 is the number of operating hydraulic cylinders, n_2 is the number of fan blades, v_{C_1} is the velocity of the center of gravity of the crankshaft in m/sec, j_3 is the moment of inertia of the connecting rod at $\theta = 15°$ in $kg \cdot m^2$, j_4 is the moment of inertia of the blades and crankshaft at $\theta = 15°$ in $kg \cdot m^2$, ω_C is the angular velocity of the center of gravity of the connecting rod in sec^{-1}, ω_{C_1} is the angular velocity of swivel of the blades in sec^{-1}, and v_A is the velocity of the crosspiece in m/sec.

The moments of inertia of the connecting rod and blades with crankshaft can be found by the method of pendulum vibrations.

Experimental and theoretical investigations revealed that the sum of the forces of friction in the bearings of the crankshaft-connecting-rod system N do not exceed 1% of the maximum load on the servomechanism, and consequently they can be allowed for by introducing a coefficient K = 1.01 on the right-hand side of Eq. (2) and discarding N.

The frictional load arising from the motion of the crosspiece along the fanshaft, without allowance for misalignment, is given by

$$T_{fr} = \mu M, \tag{11}$$

where μ is the coefficient of friction, M the weight of the crosspiece and other components attached to it in newtons.

The accuracy with which the load on the servomechanism can be found from the above formulas depends largely on correct choice of the coefficients of friction and allowance for the action of aerodynamic forces.

The actual values of the loads on a servomechanism in an experimental fan with a 500 mm diameter rotor (see Fig. 1) were found by taking oscillograms of the pressures in the hydraulic cylinders when the blades are feathered during operation of the fan, and by processing the data thus obtained by means of the known formulas [1]. For all positions of the blades, the calculated loads agreed with the experimental values.

LITERATURE CITED

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