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SELF-ALIGNING CENTRIFUGAL SUPPORTS

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A brief review of the basic types of rotation-type supports has been given. The structures of self-aligning gas-dynamic and hydrodynamic centrifugal supports have been described. A comparative computational analysis of their efficiency has been made; the advantage of the hydraulic support in realizing large-tonnage processes has been shown on the basis of the analysis.

Keywords: *high-speed rotary systems, centrifugal supports, hydraulic suspensions, gasdynamic bearings, self-balancing of centrifugal supports, damping of vibrations.*

Introduction. One challenge of mechanical engineering is to create rotary machines with a high rotational velocity, large weight-carrying capacity and permissible imbalance, a low resistance to rotation, and a long service life. Also, it is important to develop reliable methods of calculation, when self-aligning centrifugal apparatuses and stands are designed.

Formulation of the Problem. 1. Determination of the field of efficient application of one type of self-balancing centrifugal support or another from the viewpoint of energy consumption by rotation, manufacture costs, structural reliability, and the capability for perceiving imbalances occurring during both the manufacture and the technological process.

2. Ensuring the maximum weight-carrying capacity and permissible imbalance and the minimum power expenditure of an engine on realizing the technological or testing process.

3. Substantiation of calculated dependences enabling one to design and determine numerical values of the basic parameters characterizing the capability of self-balancing centrifugal supports for tackling the posed problems with the lowest expenditure of energy and materials, in so doing preserving the high reliability of the structure.

Structures and Principles of Operation of Rotation-Type Supporting Devices. The results of investigations of block diagrams of centrifugal supports and test stands have been presented in [1]. Domestic and foreign literature devoted to the problems of the dynamics and balancing of rotary systems has been reviewed and analyzed in [2]. The substantiation and methods of designing and calculation of high-speed resilient-support rotors have been given in [3]. The issues of the dynamics of high-speed rotors and suspensions on liquid- and gas-lubricated sliding supports have been set forth in [4]. The principles of design and the methods of calculation of gas-dynamic-bearing rotors have been given in [5]. The hydrodynamic theory of plain bearings has been reviewed in [6] The influence of the structure of an air feeder on the stability of a spherical suspension as applied to large-scale telescopic supports has been considered in [7]. The structures of magnetic-suspension rotary systems have been described in patent applications [8–11].

Rotary apparatuses and stands in which the traditional rigid supports were replaced by resilient ones have found wide commercial use [3]. The rigidity of supports was determined from dynamic calculation of the entire rotary machine. The optimum selection of the rigidity of resilient supports made it possible to pass critical velocities with small vibration amplitudes and a considerable reduction in the pressure between the rotor and the bearings. The technology of manufacture of rotary machines was simplified because of this and their reliability was improved. It has turned out that the rigid shaft in two resilient supports possesses the property of self-centering with finite growth in the rotational velocity, and the shaft's flexural vibrations diminish throughout the range of rotational velocities. For supports fairly compliant compared to the rigid shaft, the latter passes the first and second critical velocities without

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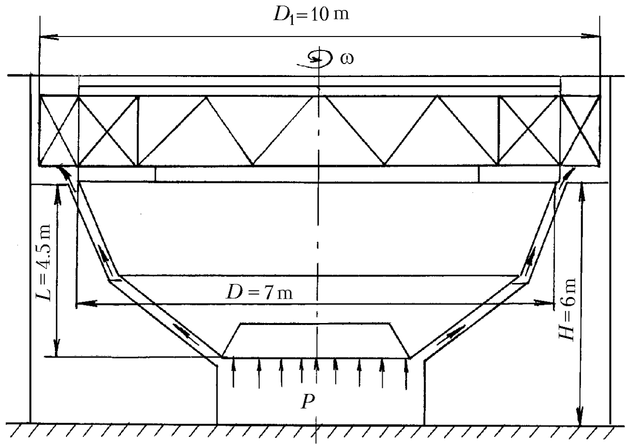

Fig. 1. Gasdynamic centrifuge. Q = 22 t, ω = 25 sec^{-1} (250 rpm), and δ = 800 kg/cm^2 (AmG-6).

flexure and with a reduction in the vibration amplitude. Decreasing the rigidity of the dynamic system incorporating the shaft together with the supports has become the general method of reducing and contracting the zone of increased vibration and of diminishing the vibration amplitude.

As the rotating masses increased, designers of rotary equipment frequently addressed the idea of realization of systems possessing self-centering (self-balancing) properties. The apparatus for deposition of a metal coating by spraying became one of the first apparatuses in structural realization of this idea. A rotor in the form of a spherical segment, which revolved inside a spherical cup of the same radius of curvature, was used in this apparatus; air was fed under pressure between them. The resulting clearance amounted to 4% of the radius of the rotor's end plane. Revolution was ensured by the airfoils made on the rotor's lateral surface. However, this was a small-scale apparatus. Subsequently an extended and thorough development of the structure of a gasdynamic support was carried out at the Heat and Mass Transfer Institute of the National Academy of Sciences of Belarus; in 1980, it was transferred to the Belarusian Republic Science and Production Association "Tsentr" newly set up to deal with this subject. Self-balancing centrifugal equipment for many industries has been created on the basis of the performed investigations and development works in this association, and it continues to be improved.

Figure 1 shows the block diagram of a self-balancing gasdynamic centrifugal support of large weight-carrying capacity with a rotational velocity constrained only by the permissible strength of the structure. The absence of rigid ties enables the rotor, after its emersion under the action of a compressed-air flow produced by centrifugal fans, to rotate about its principal central axis of inertia, which is not coincident with the vertical geometric axis of the support because of the structural and technological deviations and the imbalance of the load. The possibility of self-balancing in this support is constrained by the size of the radial clearance at the end plate of the stator. If the static imbalance exceeds the permissible one, the rotor will come in rigid contact with the stator and the stable operation of the centrifuge will be disturbed. With the additional safeguarding devices and mechanisms missing, sudden failure of the fans producing the rotor's lift and a sudden supercritical mass redistribution of the object of processing may lead to an emergency. The positive quality of a gasdynamic centrifugal support is that it can compensate not only for the static imbalance but for the dynamic imbalance as well.

The quest to diminish or to eliminate the dependence of the functioning of a gasdynamic support on devices for ensuring the gas emersion of the rotor over the stator has led to the idea of creating a self-aligning centrifugal support with its hydraulic hang-up. A block diagram of such a support is shown in Fig. 2. The rotor in the form of a cylinder is kept from emerging freely by a freely rotating tie between its bottom and the bottom of a tank. The support is damped, in rotation, by a slightly tapered wall equidistant to the bearing sleeve and tied horizontally to the lateral surface of the stator. In rotation, the axial component of the liquid flow is enhanced by a wire spiral fixed on the bearing-sleeve surface. A vertically flanged annular plate is installed on the sleeve's body at the level of the upper end of the damping wall to enhance damping. The size of the formed clearance is established from the calculation of the dynamics of revolution of the rotor with allowance for a possible unbalance. We have theoretically and experimentally

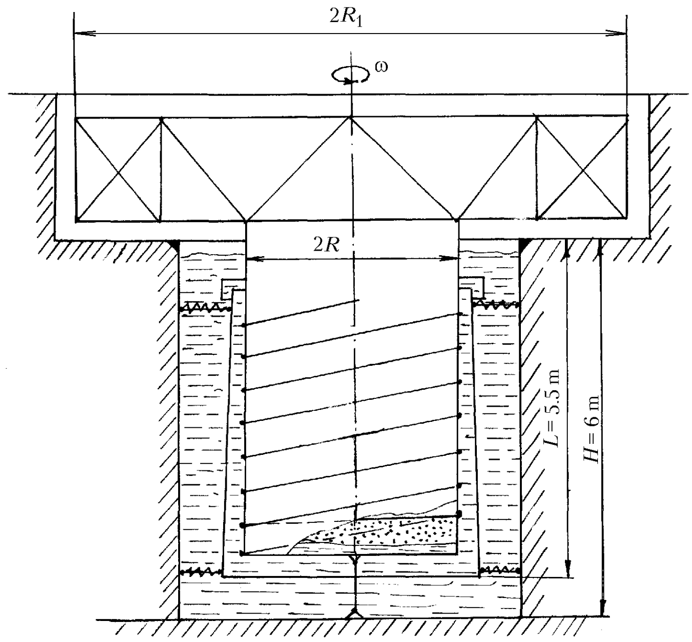

Fig. 2. Hydrodynamic centrifuge. The weight-carrying capacity is $Q = 22$ t; $\omega = 100\ \mathrm{sec}^{-1}$.

confirmed that damping grows with decrease in the clearance between the exterior lateral surface of the sleeve and the interior surface of the damping wall and in the clearance between the annular plate on the rotor and the upper end of the damping wall.

Experimental investigations were carried out on self-balancing hydrodynamic supports with float diameters of 800 and 1500 mm and lengths of 1200 and 2500 mm respectively. To avoid rigid contact between the rotating floats and the damping wall we installed tracking systems of radially movable kinematically constrained damping wheels on the lateral tank surface. The clearance between them and the lateral float surface in the initial state was made smaller than the permissible eccentricities for each experimental setup.

Calculated Formulas and Comparison to Experiment. In this work, we present results of the comparative efficiency of two types of self-aligning centrifugal supports — an air-cushion spherical support and a hydraulic-cushion cylindrical support. The calculations have been performed on the basis of boundary-layer theory developed by G. Schlichting [12]. Formulas describing the power expenditure on overcoming friction in rotation of a cylinder inside a cylinder with a clearance between them not exceeding the boundary-layer thickness and a disk — the bottom of the rotating cylinder — in a compressed-air flow were taken as a basis. The formulas allowed for liquid and air flow in the laminar and turbulent regimes. The dependences describing the resistance of a sphere to rotation in the free air stream were used to determine the energy expenditure on overcoming the friction of a spherical segment in the air layer between it and the equidistant spherical base. The calculations were performed for both the spherical segment rotating with high velocities and for the support rotating with low velocities, i.e., creeping motion where frictional forces significantly exceed inertial forces. The influence of the surface roughness of the rotating bodies of spherical and cylindrical shapes on the rotation resistance in air and in the liquid was taken into account. The degree of accuracy of the dependences employed was checked by comparing to the results of tests of experimental centrifugal stands with hydraulic bearing cylinders with diameters of 0.8 and 1.5 m and lengths of 1.5 and 2.5 m respectively which rotated in coaxial cylinders equipped with the damping walls at distances of 5 and 8 mm from them (clearances between the rotating cylinders and the damping walls). The disagreement between the calculated powers for overcoming friction in rotation of the float cylinders inside the bearing ones and the values measured in the experiments amounted to 12–21% depending on the rotation regimes. The high degree of agreement was also observed for the values of the first resonant rotational velocities. The efficiency of devices for stabilization of the revolution of rotors in a bounded volume of a liquid was confirmed in the experiments.

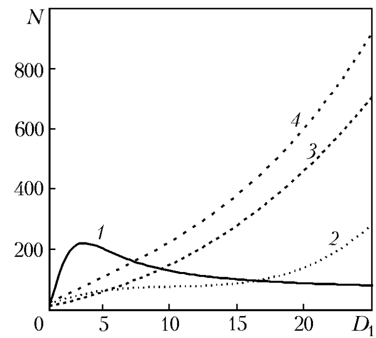

Fig. 3. Power expenditure on overcoming friction vs. diameter of the setup: 1) hydrodynamic support; 2) gasdynamic support; 3) power expenditure on supercharging in the gasdynamic setup; 4) total power expenditure in the gasdynamic setup. N, kW; D_1, m.

The experiments performed on hydrodynamic centrifugal stands enabled us to determine the efficiency of drives on the air and hydraulic supports. Figure 3 gives the power expenditure on overcoming friction in the hydrodynamic and gasdynamic centrifugal supports as a function of the rotor diameter for equal weight-carrying capacities and rotational velocities. The power expenditure on overcoming friction in rotation of the beating cylinder inside the damping wall is determined according to [12] from the dependences

$$M_{\text{fr}} = C_{\text{m}} \frac{\pi}{2} \rho\omega^2 R^4 H \,, \tag{1}$$

where we have $C_{\text{m}} = \frac{\omega R d}{\nu} \sqrt{d/R}$ in the turbulent regime of flow and $C_{\text{m}} = 4\left(\frac{\omega R d}{\nu}\right)^{-1}$ in the laminar regime. For the case where $d/R = 0.028$, introducing the notation $\text{Ta} = \frac{\omega R d}{\nu} \sqrt{d/R}$ for the Taylor number, we obtain $C_{\text{m}} = 0.67\ \text{Ta}^{-1}$. For Ta > 400, when flow is turbulent, we have

$$M_{\text{fr}} = \text{Ta}^{-0.2} \frac{\pi}{2} \rho\omega^2 R^4 H \,. \tag{2}$$

The power for overcoming the friction between the plane bottom of the float cylinder, bitten deeper into the cylindrical wall, and the liquid surface contacting it via compressed air in the tank is determined from the formula

$$M_{\text{fr}} = 0.0365\rho\omega^2 R^5 \left(\frac{\nu}{R^2\omega}\right)^{1/5} . \tag{3}$$

In the case of the turbulent regime of interaction of the bottom surface and the compressed air where $\text{Re} > 3 \cdot 10^5$, formula (3) yields good agreement of the calculated values of the moment of friction torque) with the experimental data. The values required for calculation of the resistance coefficient $C_{\text{m}} = 3.87/\text{Re}^{1/2}$ for other values of Re were taken from [12, p. 105].

The boundary-layer thickness is determined from the dependence

$$\delta = 0.526R \left(\frac{\nu}{\omega R^2}\right)^{1/5} . \tag{4}$$

For a spherical thrust bearing (analog of a gasdynamic support), the energy expended on overcoming friction in the clearance with a spherical segmental thrust journal is determined from the formula [7]

$$M_{\mathrm{fr}} = \frac{\omega^2 \nu R^4}{32.87}\left[\cos\theta_1\,(\sin^2\theta_1 + 2) - \cos\theta_2\,(\sin^2\theta_2 + 2)\right] C_{\mathrm{m}}\,, \tag{5}$$

where θ_1 is the angle of arrangement of the side channels for feeding the air flow and θ_2 is the central angle of the end surface of the bearing thrust journal. When the compressed air is fed through the channel along the central vertical axis of the thrust journal, we have $\theta_2 = 0$.

For air flow past a sphere, its friction resistance to the flow is

$$W = 6\pi\mu U_\infty R_{\mathrm{s}} C_{\mathrm{w}}\,, \tag{6}$$

where $C_{\mathrm{w}} = \frac{24}{\mathrm{Re}}\left(1 + \frac{3}{16}\mathrm{Re}\right)$, and $\mathrm{Re} = U_\infty D_{\mathrm{s}}/\nu$. The value of C_{w} corresponds to the real one up to the values $\mathrm{Re} \approx 5$. For high values of the Reynolds number, the moments of resistance to air flow past the rotating sphere were determined from the formula

$$M_{\mathrm{fr}} = C_{\mathrm{w}}\pi R_{\mathrm{s}}^3 \rho / 2U_\infty^2\,. \tag{7}$$

The C_{w} values required for calculations of other values of Re were taken from [12, p. 236].

Discussion of Results. The given formulas (1)–(7) make it possible to determine the comparative efficiency of self-aligning air-cushion and hydraulic-suspension supports. Figure 3 gives the power expenditure on overcoming friction in rotation of the above supports as a function of their dimensions for equal values of the permissible unbalance and the rotational velocity. The power expenditure of the self-aligning gasdynamic support includes the power for overcoming friction resistance and the power for ensuring the emersion of the rotor by a prescribed value equal to that of the permissible imbalance corresponding to the clearance between the stator and the rotor in the end's horizontal plane. In the hydraulic support, the power is expended on overcoming friction in the clearance between the rotor and the damping wall and in the clearance formed between the flanged edge and the end of the damping wall. Both types of these supports are assumed to be manufactured from the same material, and the highest rotational velocity is assumed to be constrained by the permissible safety factor of the structure of the end of the gasdynamic-support's rotor. It follows from the data of Fig. 3 that gasdynamic supports are efficient up to a weight-carrying capacity not exceeding 7.5 tons, as far as the power for overcoming rotation friction is concerned. For a larger weight-carrying capacity, more efficient are hydraulic supports whose weight is much smaller, the technology of manufacture is more simple, and the reliability of operation is incomparably higher.

Conclusions. Based on the hydrodynamic type of support, one can create centrifuges for such large-tonnage processes as: a) impregnation and drying of wood lumber of length to 6 m with a single-feed volume of to 10 m^3; b) treatment of industrial and domestic sewage with a capacity of thousands of cubic meters an hour; c) drying of coal on industrial scale; d) separation of sand and water from petroleum.

NOTATION

C_{m}, coefficient of moment of resistance; d, clearance between the float cylinder and the damping wall, mm; D, diameter of the cylindrical rotor, m; D_1, diameter of the platform, m; D_{s}, diameter of the spherical rotor, m; H, height of the bearing cylinder, m; L, height of the submerged part of the float cylinder, m; M_{fr}, moment of friction resistance. kgf·m; N, power expenditure, kW; Q, weight-carrying capacity, t; P, excess pressure from the fan, Pa; Re, Reynolds number; R, radius of the bearing cylinder, m; R_1, radius of the platform, m; R_{s}, radius of the spherical rotor, m; Ta, Taylor number; U_∞, flow velocity, m/sec; δ, boundary-layer thickness, m; μ, dynamic viscosity of the liquid, kg/(m·sec); ν, kinematic viscosity of the liquid, m^2/sec; ρ, density of the liquid, $\mathrm{kg/m}^3$; σ, ultimate strength, $\mathrm{kg/cm}^2$; ω, angular rotational velocity, sec^{-1}. Subscripts: m, moment, torque; fr, friction; w, wall; ∞, unperturbed flow; s, spherical.

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