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RESEARCH, DESIGN, CALCULATIONS, AND OPERATING EXPERIENCE

PROCESSES AND EQUIPMENT FOR CHEMICAL AND OIL-GAS PRODUCTION

ANALYSIS OF THE POWER OF A ROTARY-PRESS DRIVE FOR THE MOLDING OF MULTIPLE-COMPONENT COMBINED ARTICLES

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A procedure is presented for analysis of the power of rotary-press drives used for the molding of multiple-component combined articles. The effect of inertial forces, frictional forces in the supports of the rotor, and the resistance due to molding forces is defined more precisely for calculation of the power of a rotary-press drive. The influence exerted by the structural parameters of these articles on selection of the power of the press is demonstrated, proceeding from versatility requirements for production equipment employed to manufacture multiple-component combined articles.

A principal drawback of rotary machines with a mechanical or hydromechanical drive is the necessity of a rigid link between transport and production operations [1, 2], rendering adjustment more expensive, and the operations less effective. Moreover, safety requirements for the molding process dictate restrictions on the rate, and, consequently, productivity of the molding. The cyclical output of a rotary press in pieces/min is [2]:

$$Q_c = u_r n_r = 30 u_r \omega_r / \pi = u_r / T_c, \tag{1}$$

where u_r is the number of positions assumed by the rotor; n_r is the rotational speed of the rotor, rpm; ω_r is the angular velocity of the rotor, rps; which is linked to the rotational speed by the relationship $\omega_r = \pi n_r/30$; $T_c = t_b + t_m + t_i$ is the time required for the molding of a combined article, min; t_b is the time for batching of the components; t_m is the molding time; and t_i is the idle time. The molding time for a combined article [3] can be determined from the formula

$$t_m = \frac{1}{2\bar{v}_p} \sum_{i=1}^{n} \left(H_{i0} - H_i\right), \tag{2}$$

where $\bar{v}_p$ is the average displacement rate of the plunger, i is the number of layers in the combined article, $H_{i0} = M_i/(\pi r^2 \rho_{i0})$ is the initial height of the ith layer, ρ_{i0} is the bulk density of the ith layer, r is the radius of the combined article, M_i is

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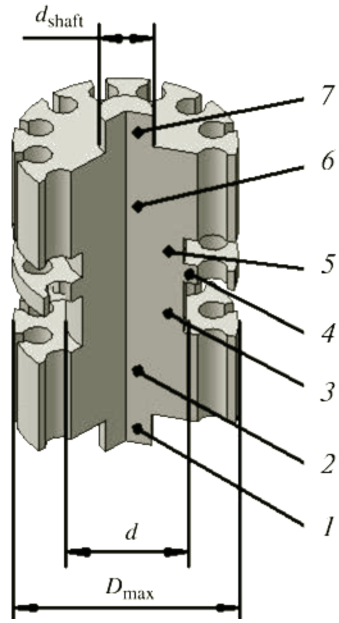

Fig. 1. Schematic of typical molding rotor: *1*) lower support; *2*) lower drum; *3*) intermediate shaft; *4*) die plate; *5*) intermediate shaft; *6*) upper drum; *7*) upper support.

the mass of the material in the *i*th layer, and H_i is the final thickness of the *i*th layer assigned by technical specifications for the article.

The average displacement rate $\bar{v}_p$ of the plunger is determined by the procedure proposed in [3], and is optimal for a specific type of article. It will depend on the design of the combined article and the properties of the compositions that enter into the article. A broad list of articles manufactured by series, or low-volume series production is currently being developed. The equipment employed for the production of these articles should therefore be universal [1]. In order that the press meet versatility requirements, it is necessary to design it such that the power required for the drive of the press is sufficient for the molding of a broad assortment of combined multiple-component articles.

The power of the drive of the rotary press [2]:

$$N = M\omega_r/\eta, \tag{3}$$

where M is the twisting moment in the shaft of the molding rotor, and η is the efficiency of its drive.

The drive of the production molding rotor should overcome the moment of resistance M_{in} due to inertial forces, the moment of resistance M_{fr} that develops due to frictional forces in the supports, and the moment of resistance M_m due to the molding forces:

$$M = M_{in} + M_{fr} + M_m. \tag{4}$$

The moment of resistance due to inertial forces can be calculated from the formula [4]

$$M_{in} = J_z\varepsilon,$$

where J_z is the moment of inertia about the vertical axis, and ε is the angular acceleration of the molding rotor during its acceleration period. A typical molding rotor consists of seven components (Fig. 1).

The moment of inertia of the molding rotor about the vertical axis can be represented as

$$J_z = \sum_{i=1}^{7} m_i r_{iz}^2 ,$$

where m_i is the mass of the ith component of the molding rotor, and r_{iz} is the radius of gyration of the ith component of the molding rotor.

Representing the rotor as a set of cylindrical bodies with a radius R_i, we have

$$J_z = \frac{1}{2}\sum_{i=1}^{7} m_i R_i^2 . \quad (5)$$

Considering that the diameters of the support shafts in modern rotor designs are many times smaller than the diameters of the drums, and the intermediate shafts are hollow [1], the moments of inertia of these rotor components, like the shafts of the lower and upper supports and the intermediate shafts (see Fig. 1, positions *1*, *3*, *5*, and *7*), can be disregarded. Formula (5) will then take on the form

$$J_z = \frac{(D_{\max}/2)^2}{2}(m_2 + m_4 + m_6) \approx \frac{m_r D_{\max}^2}{8}, \quad (6)$$

where m_2 is the mass of the lower drum, m_4 is the mass of the die plate, m_6 is the mass of the upper drum, and $D_{\max}$ is the maximum rotor diameter (as a rule, equal to the diameter of the drum).

Assuming an acceleration period with a constant acceleration, the angular acceleration can be defined as

$$\varepsilon = \omega_r/\Delta t, \quad (7)$$

where Δt is the acceleration time of the rotor.

Substituting (6) and (7) in (5), we obtain

$$M_{in} = \frac{m_r D_{\max}^2 \omega_r}{8\Delta t}. \quad (8)$$

Frictional forces develop in the supports due to the action of the force of gravity of the rotor in the lower support, and due to inertial forces in the lower and upper supports.

The load due to forces acting in the plane perpendicular to the axis of rotation is taken up by the two supports, while the load due to the weight of the rotor is taken up only by the lower support. If the rotor is balanced, and its center of gravity is on the axis of rotation, the pressure exerted by the rotor on the bearing will be independent of the angular velocity and angular acceleration, i.e., the reactions that develop in the bearings due to the inertial forces of the rotor are equal to zero [4]; it is therefore necessary to consider only the moment of the initial forces, which develops due to the force of gravity.

$$M_{fr} = f_k m_r g, \quad (9)$$

where f_k is the coefficient of rolling friction in the rotor support.

Figure 2 shows the typical design of a support for a rotary press [5]. The force of gravity is taken up by the thrust bearing, while radial bearings take up the forces acting in the plane perpendicular to the axis of rotation.

The moment of resistance due to the molding forces will depend on the magnitude of the external load, which is determined by the properties of the material being molded (molding curve), and by the profile of the thrust template. Creation of the template profile is the most labor-intensive and complex task in designing rotary presses with a mechanical drive.

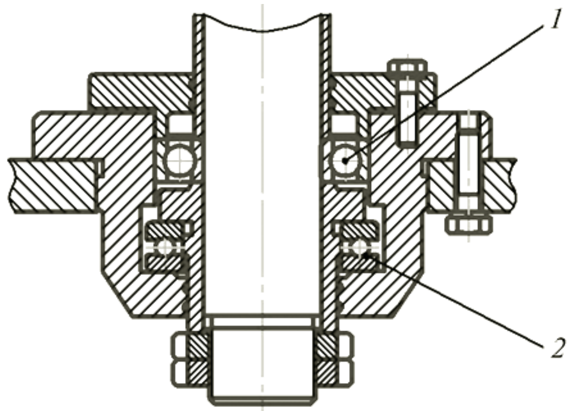

Fig. 2. Typical design of rotary-press supports: *1*) radial bearing; *2*) thrust bearing.

To initiate the design of a template, it is necessary to assign the relationship between the displacement of the slide block and the turn angle (timing diagram) of the rotor [6]. It is usually assigned as a linear function, partitioning the molding sections into zones of preliminary accelerated molding and final molding. Since all operations on the rotary press are carried out concurrently [2], the moment of resistance due to the molding forces will be

$$M_{\mathrm{m}} = \frac{D_0}{2} \sum_{j=1}^{k} F_{\mathrm{cir}j}, \tag{10}$$

where D_0 is the diameter of the initial circumference of the molding rotor, $F_{\mathrm{cir}j}$ is the circumferential force required to overcome the force of resistance of the material during molding in the *j*th section of the profile of the thrust template, and *k* is the number of linear sections on the profile of the thrust template.

To determine the magnitude of the circumferential force, let us examine a rectilinear section of the force template (Fig. 3).

Normal reaction *N* develops when the roll of the slide block acts on the template with a force equal to the molding force P_{m}. To displace the slide block along the template, it is necessary to apply a force *F* equal to the absolute value of the sum of the absolute values of the frictional forces F_{fr} and the components of the molding force $P_{\mathrm{m}x}$. Let us write the equation of equilibrium of the system of forces in question:

$$\begin{cases} \sum F_x = F - P_{\mathrm{m}} \sin\gamma - F_{\mathrm{fr}} = 0; \\ \sum F_y = P_{\mathrm{m}} \cos\gamma - N = 0, \end{cases} \tag{11}$$

where γ is the angle of elevation of the profile of the template.

The magnitude of the frictional forces is determined by one of the following two equations [4]: $F_{\mathrm{fr}} = f_k N/r$ (rolling regime), or $F_{\mathrm{fr}} = fN$ (sliding regime), where *r* is the radius of the axis of the roll, and *f* is the coefficient of sliding friction of the roll against the surface of the template. If it is assumed that a sliding regime is realized during molding, the solution of system (11) will be

$$F = P_{\mathrm{m}}\left(\sin\gamma + \frac{f_k}{r}\cos\gamma\right).$$

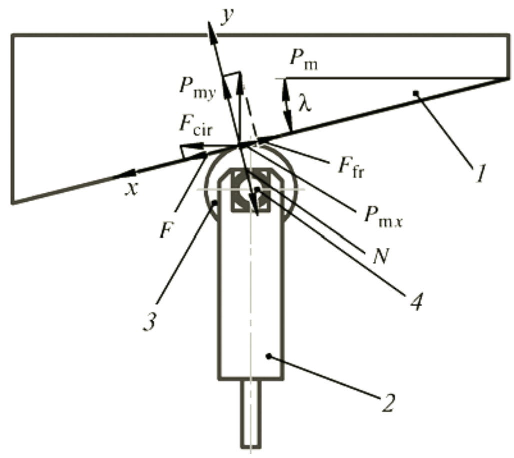

Fig. 3. Forces developing during molding of articles on rotary press with mechanical drive: *1*) thrust template; *2*) slide block; *3*) roll; *4*) axis of roll.

The moment of resistance due to the molding force on a given section will be equal to the product of the absolute value of the circumferential force F_{cir} and the arm equal to half the initial diameter of the rotor. Here, the projection of the circumferential force onto the *x* axis should be no lower than the force *F* to overcome the resistance forces:

$$F_{cir}\cos\gamma \geq P_m\left(\sin\gamma + \frac{f_k}{r}\cos\gamma\right).$$

Considering (10), consequently, the moment of resistance due to the molding forces as the rotor turns in the *j*th section of the template will be

$$M_{mj} = \frac{D_0}{2}P_{mj}\frac{\sin\gamma_j + \frac{f_k}{r}\cos\gamma_j}{\cos\gamma_j} = \frac{D_0}{2}P_{mj}\left(\tan\gamma_j + \frac{f_k}{r}\right).$$

The total moment of resistance due to the molding forces will be

$$M_m = \frac{D_0}{2}\sum_{j=1}^{k}P_{mj}\left(\tan\gamma_j + \frac{f_k}{r}\right). \quad (12)$$

Substituting (8), (9), and (12) in (4), we obtain

$$M = \frac{m_r D_{max}^2 \omega_r}{8\Delta t} + f_{k1}m_r g + \frac{D_0}{2}\sum_{j=1}^{k}P_{mj}\left(\tan\gamma_j + \frac{f_k}{r}\right). \quad (13)$$

Knowing the maximum twisting moment on the rotor, it is possible to calculate the power required for the drive of the molding rotor by substituting the value of the twisting moment on the shaft of the rotor (13) in (3):

$$N = \left[\frac{m_r D_{max}^2 \omega_r}{8\Delta t} + f_{k1}m_r g + \frac{D_0}{2}\sum_{j=1}^{k}P_{mj}\left(\tan\gamma_j + \frac{f_k}{r}\right)\right]\frac{\omega_r}{\eta}. \quad (14)$$

In order that the press meet versatility requirements for a group of articles, it is necessary to calculate its power with respect to that article for the pressing of which the maximum force is required. As a rule, this article will have the smallest diameter of those within the group.

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