

More Efficient Rotation of Excavator Platforms

V. N. Kuznetsova^a, * and V. V. Savinkin^b

^aSiberian State Automobile Academy, Omsk, Russia

^bKozybaev North Kazakhstan State University, Petropavlovsk, Kazakhstan

*e-mail: dissogetsibadi@bk.ru

Abstract—In the rotation of an excavator platform, energy is mostly consumed in acceleration and deceleration. The energy consumed in platform rotation may be reduced by uniform redistribution of the load over the whole contact surface, with equalization of the time of load action.

Keywords: inertial loads, stress concentrations, rotation, positional angle, energy consumption

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In the course of operation, the rotary platform of an excavator experiences large alternating loads, which determine the life of the transfer mechanisms and the operational efficiency of the drive as a whole [1, 2]. The range of the cyclic load depends on the operating conditions of the excavator and the total forces and torques in different periods of platform rotation [3, 4]. If we study the many factors that influence the energy consumption in excavation, we may more accurately model the operating systems in roadbuilding equipment. Despite extensive research, the load variation in different periods of platform rotation remains to be accurately described. Likewise, we still need to explain the considerable energy losses in the platform's hydraulic drive and in the mechanism that generates the power in the gear system responsible for platform rotation.

Such research is complicated by the nonuniform speed of the platform (acceleration and deceleration). Accordingly, in the present work, attention focuses on the dynamic and energy characteristics of platform rotation in the excavator. In Fig. 1, we show the algorithm for calculating the inertial forces of the rotary platform and the excavator's operational equipment.

The time T_{ro} of platform rotation is expressed as the sum of all the periods within a single cycle

$$T_{ro} = t_1 + t_2 + t_3,$$

where t_1 is the time of platform acceleration; t_2 is the time of uniform rotation at steady velocity; t_3 is the time of platform deceleration to a complete stop.

The time t_2 of uniform rotation at steady velocity is not considered here, since there is no tangential acceleration in this period [5]. The other dynamic parameters must be investigated at the end of deceleration, which is the extreme point of time period dt .

The moment of inertia M_{in} acting on the drive gear with angular acceleration ε_1 takes the form

$$M_{in} = \varepsilon_1 J_s = \frac{2\varphi m r^2}{t_1^2},$$

where J_s is the intrinsic torque of the platform's supporting and rotary gear, N m²; m is the gear mass, kg; r is the radius of its dividing circle, m; φ is the platform's angle of rotation.

The power N at the driven gear is determined by taking account of the azimuthal velocity v and the force F applied to the gear. Then the torque on the gear is

$$M_g = \frac{Fv}{2\varphi t_1/t_1^2} = \frac{N}{\omega},$$

where ω is the angular velocity of the platform, rad/s.

On the basis of the calculated forces and moments of inertia in each period t_1 , t_2 , and t_3 of platform rotation, its critical angles of rotation may be determined. Analysis of these angles yields the distribution of the dynamic load and the energy consumption in the rotary cycle within interval dt . The critical time intervals are the acceleration time t_1 and deceleration time t_3 . The time for rotation of the excavator platform is $T_{ro} \approx 8.4$ s. Subdivision of this time results in very small intervals. We find that two thirds of the cycle is consumed in overcoming the drag forces. The useful work of the hydraulic motor is associated with no more than 1.8–2 s of the total operating time (8 s). The remaining time is associated with consumption of the hydraulic drive's power in acceleration and deceleration of the platform. Energy saturation is observed at the moment of deceleration of the platform, when the negative influence of the inertial forces and angular velocities of the operational equipment and the mass of the material being excavated is added to the tangential inertial

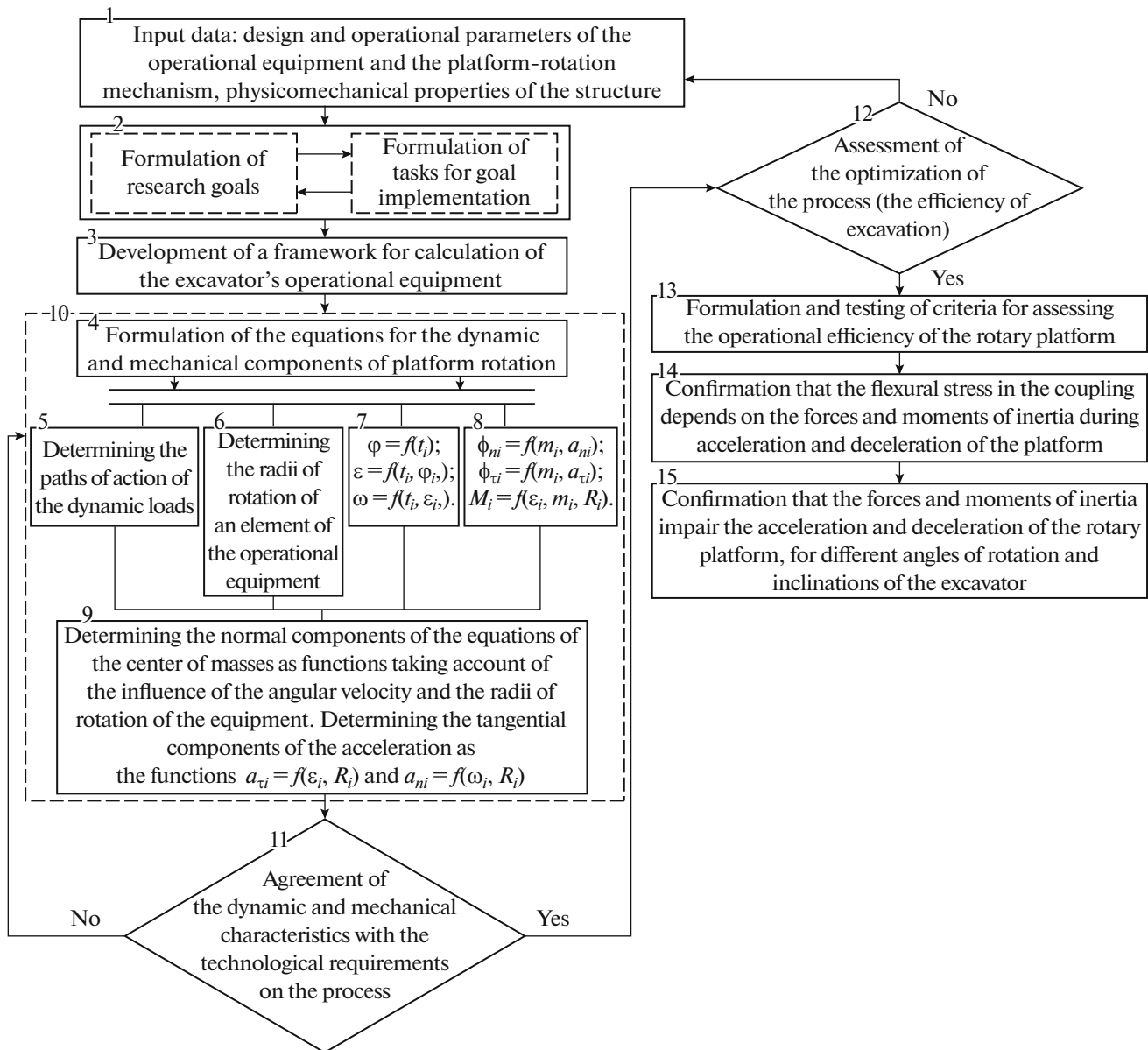


Fig. 1. Algorithm for calculating the inertial forces of the rotary platform and the excavator's operational equipment.

force of the rotary platform. During the very brief deceleration period, the gravitational forces at the centers of mass tend to prolong the motion at specified velocity over a specified trajectory.

When the angle of platform rotation is $\varphi = 70^\circ - 180^\circ$, the mechanisms ensure optimal parameter values. When $\varphi < 60^\circ$, only 0.6–0.7 of the power is used in acceleration and, with the specified parameters, the maximum rotary velocity is not attained, on account of the need for deceleration of the platform. Table 1 presents the acceleration and deceleration times. If design measures are adopted to reduce those times,

the overall time T_{ro} of platform rotation may be reduced.

One possible approach is to increase the pump and motor power [6, 7]. In order to reduce the time for each operating cycle by 0.3–0.5 s, the motor power must be increased by 25–30%. To improve excavator performance, different inclinations of the working arm are employed. On account of the change in the inertial masses, the time for platform rotation in discharge will also be changed. According to calculations and practical confirmation, increase in inclination of the working arm changes the duration of rotation and the cycle

length (Table 2). If the inclination of the working arm is increased from 35° to 47°, the time for platform rotation in discharge is reduced by 3%, while the cycle is shortened by 1%. Obviously, the best way to improve the efficiency, reduce the rotation cycle, and increase excavator productivity is to reduce the angle of rotation.

However, that sharply increases the energy consumption and entails increase in drive power by 25–30%. At the same time, the energy consumption in each cycle increases, and the drag forces have more influence on the load of the kinematic pairs in the rotary mechanism. If cyclic loads operate continuously, the fatigue stress at the contact surfaces of the gear tooth increases [8]. As a result, the tooth shatters or breaks off. The additional motion of the drive gear due to the tooth damage must then be regarded as an idling path, since there is no engagement of the corresponding tooth pair. The gear's idling path results in worse values of the deceleration time t_3 and the specified angle φ of platform rotation.

We assume that the increase $\Delta\varphi$ in the angle of platform rotation depends on the number z_d of damaged teeth. Then

$$\Delta\varphi = \varphi'_f - \varphi_f,$$

where φ_f is the final planned angle of platform rotation in insertion. $\varphi'_f = \varphi_f + \sum \varphi_i(z_{di})$ is the angle of forced rotation of the platform with damaged gear teeth; $\sum \varphi_i(z_{di})$ is the sum of the angles of platform rotation corresponding to the distances traveled at each subsequent damaged tooth.

Calculation of the geometric and dynamic characteristics shows that the distance traveled by the platform, which is equal to the width of the damaged tooth, may be assumed to correspond to the angle $\varphi = 4^\circ 1'$. Increase in the free path of the drive gear under the action of the inertial forces during platform deceleration considerably increases the moment of inertia M'_{in} in position φ'_f , which impairs the working life and energy efficiency of the mechanism

$$M'_{in} = \varepsilon_1 J_s = \frac{2\varphi + (\varphi_f + \sum \varphi_i(z_{di}))mr^2}{t_1^2}. \quad (1)$$

By numerical approximation, we may write the moment of inertia M'_{in} as a function of the sharply varying angular acceleration ε_1 as follows

$$M'_{in} = 136.8\varepsilon_1^3 - 113.9\varepsilon_1^2 + 43.1\varepsilon_1 + 4.2.$$

Variation in the forces and torques that act significantly redistributes the flexural stress σ_{fl} that accompanies the engagement of the drive gear and the driven gear in the platform-rotation mechanism with varia-

Table 1. Acceleration and deceleration times of platform, s

Time	Mechanisms		
	frictional couplings and brakes	pumps with	
		constant productivity	variable productivity
Acceleration:			
when loaded	2.00	2.20	2.34
when empty	1.43	1.59	1.54
Deceleration			
when loaded	1.40	1.54	1.66
when empty	1.00	1.11	1.06

Table 2. Time of platform rotation in discharge, s

φ , deg	Inclination of the working arm	
	30°	45°
90	1.00	0.95
135	1.36	1.31
180	1.72	1.67

tion in the operating time t_d during the platform-rotation cycle at discharge (Fig. 2).

In the first seconds of platform rotation when the shovel is loaded, the hydraulic motor responsible for platform rotation transmits the torque through the

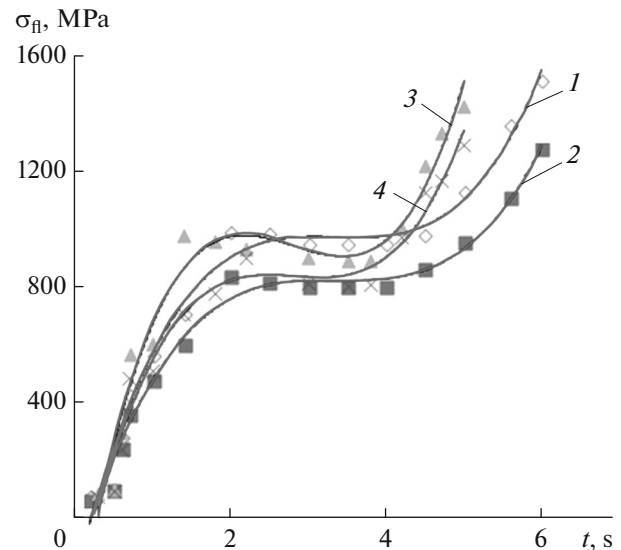


Fig. 2. Dependence of the flexural stress σ_{fl} in the drive gear ($\diamond$, $\blacktriangle$) and driven gear ($\blacksquare$, $\times$) in platform rotation on the operating time t and corresponding correlation curves (1–4).

drive gear to the driven gear (the crown of the supporting and rotary gear). In acceleration, the stress tends to rise to the following final values at $t = t_1 = 2.3$ s: $\sigma_{fi} = 900\text{--}1000$ MPa at the drive gear and $\sigma_{fi} = 800$ MPa at the driven gear. The period $t = t_2 = 2.3\text{--}3.1$ s corresponds to platform rotation at steady velocity. This short period corresponds to gear operation without pump pressure. In this period, the flexural stress σ_{fi} is practically constant: it declines slightly to 960 MPa. The constant σ_{fi} value may be attributed to the inertial force acquired during platform acceleration. The pressure of the working fluid on the hydraulic motor has practically no effect, and the steady inertial forces ensure rolling in tooth engagement. At the end of the deceleration period ($t = t_3 = 0.86$ s), the flexural stress increases sharply to $\sigma_{fi} = 1700\text{--}1800$ MPa at the contact surface bounded by the drive-gear tooth.

Taking account of the large flexural stress σ_{fi} during $t = t_1 + t_3$, which is two thirds of the cycle, we conclude that most of the work A_{ro} in rotation is associated with overcoming the drag forces at engagement. This accounts for the low efficiency of the platform rotation $\eta = 0.6\text{--}0.65$, in comparison with the value $\eta = 0.85\text{--}0.9$ in the manufacturer's documentation.

In Fig. 2, correlation curves 1–4 describe the variation in the flexural stress σ_{fi} as a function of the time t and the corresponding time of action of the drag forces. On account of the complex distribution within the operating period, we obtain third-order polynomials.

1. Platform rotation with a loaded shovel:
for the drive gear (curve 1)

$$\sigma_{in} = 32.185t_i^3 - 324.45t_i^2 + 1082.8t_i - 226.09;$$

for the driven gear (curve 2)

$$\sigma_{in} = 26.086t_i^3 - 264.75t_i^2 + 893.63t_i - 182.17.$$

2. Platform rotation with an empty shovel:
for the drive gear (curve 3)

$$\sigma_{in} = 72.307t_i^3 - 609.75t_i^2 + 1627.4t_i - 422.69;$$

for the driven gear (curve 4)

$$\sigma_{in} = 53.134t_i^3 - 451.09t_i^2 + 1258.8t_i - 318.59.$$

The cyclic action of the stress in dynamic loading significantly affects the life of the mechanism. A relation between the stress σ_{HN} and cyclic life was established in research on planetary gears [9, 10]. The hardness of the working surfaces is employed in determining the contact strength and the basic number of cycles to failure. On the basis of operational experience with excavators, metallographic data, and analysis of gear repair and restoration, it appears that the surface hardness is not the main source of failure in most cases. In

studying tooth shear, integrity of the coating at the contact surface is observed, while the grain size and color range vary from the point of load application to the base of the tooth. Research indicates that the cyclic life largely depends on the applied load per second.

The high stress (curves 1–4 in Fig. 2) is associated with the considerable tangential inertial forces of the working equipment during acceleration of the rotary platform. In platform rotation with an empty shovel, anomalous surges are observed in the flexural stress σ_{fi} . However, the curve is steeper. In deceleration of the platform, the flexural stress σ_{fi} is still small but higher than in the previous case. This may be explained in that the acceleration time t_1 and deceleration time t_3 of the excavator's rotary platform are reduced to 1.06–1.5 s when the shovel is empty. That overloads the gear teeth and consequently high dynamic loads appear at engagement of the kinematic pair, with accompanying decrease in working life.

CONCLUSIONS

(1) We have demonstrated that the existing design of the excavator's rotary platform is ineffective, on the basis of calculations of the moments of inertia and force balance during a specified operating time.

(2) We have developed a method for determining the inertial forces and torques on the rotary platform with a nonuniform distribution of the dynamic loads over each tooth of the drive gear. The correlation of the flexural stress σ_{fi} in the kinematic pair on engagement with the gear's operating time is established.

(3) One method of reducing the energy consumption in platform rotation is to redistribute the forces over the kinematic pairs in accordance with the gravitational and inertial torques on the whole platform. Equalization of the times of load application also proves effective.

(4) The results obtained regarding the forces at the contact surface of the teeth in transient acceleration and deceleration of the rotary platform improve our understanding of the efficiency and energy consumption in the dynamic process.

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