Relationship of dynamic properties of mine excavator hoisting mechanism versus design parameters of operating equipment

S U Kuvshinkin¹, I E Zvonarev¹, P V Ivanova¹

¹ Saint-Petersburg Mining University, 2, 21st line, Saint-Petersburg, 199106, Russia

E-mail: ZVano@mail.ru

Abstract. In this paper the effect of design parameters of excavators operating equipment on the dynamic load of drives is demonstrated using the example of the hoisting mechanism as the most energy-loaded one (the total power of the drive motors of the hoisting mechanism is comparable to the power of all other excavator drives combined).

1. Introduction
The high level of excavators downtimes results in significant material costs caused both by mining operations decline and by high cost repair of large-size units and metal structures.

The analysis of contribution of excavators’ mechanical and electrical subsystems to the total number of failures shows that mostly mechanical equipment failures take place and amount to 50-70 % of the total amount.

The rate of failures occurred for both excavators mechanical and electrical subsystems is essentially affected by the high dynamic component of the external load that occurs when the bucket touches the bottom-hole. The dynamic component of the external load when passing through the operating equipment dynamic system is transformed as a result of oscillations excitation occurred within the latter. The main parameters of the operating equipment of the dipper power shovel that determines its mass and rigidity and therefore its dynamic properties are the jib length and the bucket capacity.

2. Theory research
The purpose of developing and analyzing the dynamic model is to determine the load on the output shaft of the reducing gear of the hoisting mechanism – on the dynamic system ‘output’ provided certain load on the bucket – ‘input’.

The following assumptions are made for the dynamic simulated circuit.

1. We regard the jib mounting as absolutely rigid.
2. Dissipative losses in ropes and blocks are neglected.
3. We consider connections to be weightless.

When developing the dynamic simulated circuit, the circuit of hoisting installation used in EKG-8I excavators (the dynamic simulated circuit is given in figures 1) was taken as a basis.
Figure 1. The dynamic simulated circuit.

Figure 1: m – the total mass of the bucket including rocks and the bucket arm mass specific for the bucket mass centre; $c_1$ – the rigidity of the line part within the area from the bull block to the bucket; $c_2$ – the rigidity of the line part within the area from the bull block to the hoisting drums; $c_3$ – the rigidity of the line part within the area from the bull block to the balancing half-block; $Q(t)$ – the external load; $S(t)$ – the hoisting ropes force to be determined at the point of reeling on the drum spool.

The hoisting mechanism dynamic circuit can be simplified by determining the specified rigidity for all line parts of the hoisting rope:

$$c_{de} = \frac{2c_2 \cdot 2(c_1 + \frac{c_1c_3}{c_1 + c_3})}{2c_2 + 2(c_1 + \frac{c_1c_3}{c_1 + c_3})} = \frac{2c_2(c_1^2 + 2c_1c_2)}{c_1^2 + c_1c_2 + 2c_1c_3 + c_2c_3}.$$ (1)

According to the method for rope rigidity calculation, the rope rigidity ratio can be determined as follows:

$$c_r = \frac{nk_r E_r F_r}{l_r}$$ (2)

where $n$ – the number of line parts, $E_r$ – modulus of rope elasticity; $k_r = 0.8$ – flexibility coefficient of the drive elements and excavator metal structures; $F_r$ – aggregate area of wires; $l_r$ – the effective length of the line part.

The equivalent mass of the system is composed of the bucket mass including rock mass and the bucket arm mass specific for the bucket mass centre:

$$m = \frac{1}{g}(G_b + G_{rm} + G_{arm})$$ (3)

where $g$ – gravitational acceleration; $G_b$, $G_{rm}$, $G_{arm}$ – accordingly the gravity force of the bucket and of rock mass and specific gravity force of the bucket arm.

The load of the bucket in the determinate form can be expressed as follows:

$$Q(t) = S_{bf} + \sum_{i=1}^{n} A_i \sin(\omega_i t + \alpha_i)$$ (4)

where $S_{bf}$ – the average hoisting force; second term – load dynamic component; $A_i$, $\omega_i$, $\alpha_i$ – accordingly the amplitude, circular frequency and the initial phase of $i$-harmonic of load dynamic component.
The force of hoisting ropes at the point of their reeling on the drum spool can be determined as a sum of static and dynamic components:

$$S(t) = S_{st} + S_d.$$ (5)

According to the d'Alembert's principle, the effect of the external load static component and the dynamic component harmonic on the dynamic system can be considered independently. At the same time the load at the system 'output' is expressed as the sum of results of the external load components effect.

The dynamic component of the external load when passing through the operating equipment dynamic system is transformed as a result of oscillations excitation occurred within the latter. In order to determine the dynamic component of hoisting ropes force, we compose and solve a differential equation of the system motion affected by the external load dynamic component harmonics.

The differential equation of the system motion is written as:

$$m \ddot{x} + c_{x0} \dot{x} + A \sin \omega t = 0.$$ (6)

The general solution to this non-homogenous linear differential equation of the second kind is composed of the complete integral and complementary function:

$$x = x_0 + x_1$$ (7)

where $x_0, x_1$ – accordingly free and forced oscillations of the system.

$$x_0 = c_1 \cos kt + c_2 \sin kt$$ (8)

where $k$ – circular frequency of characteristic oscillations; $c_1, c_2$ – integration constants.

Free oscillations will be damped within the mining excavator hoisting mechanism due to the friction forces that is why the basic value in the expression for $x$ will be forced oscillations, i.e. the partial solution of $x_1$:

$$x_1 = c_3 \sin \omega t.$$ (9)

The solution to the equation (6) is written as:

$$x = \frac{A}{c_{x0} - m\omega^2} \sin \omega t.$$ (10)

The hoisting ropes force at the point of their reeling on the drum spool is determined as follows:

$$S(t) = \frac{S_{hf}}{i_r} + A \frac{k^2}{k^2 - \omega^2} \sin \omega t.$$ (11)

The expression for calculation of the amplitude of hoisting ropes force at the point of their reeling on the drum spool is written as:

$$A_t = A\frac{k^2}{k^2 - \omega^2}.$$ (12)

The torque of the output shaft of the hoisting mechanism reducing gear will be determined for the following formula:

$$M(t) = \frac{d_{ho}}{2} \left( \frac{S_{hf}}{i_r} + A_t \sin \omega t \right).$$ (13)
where $d_d$ – hoisting drum diameter.

3. Experimental research

Figure 2 shows amplitude-frequency characteristics of the system as well as graphs on oscillations characteristic frequency versus the jib length ($L_c$) relationship at three values of the bucket capacity. The jib length mainly determines the system rigidity while the bucket capacity at the constant value of bucket fill factor determines the value of the specific mass. As it can be seen from the graphs, the system characteristic frequency decreases if the length of the operating equipment and the bucket capacity are increased, while the resonance area is shifted to the low-frequency area. The external load low-frequency random component is determined by variations of the cut depth during excavating i.e. by the driver input. Therefore, the amplitude of this component can be decreased, for example, by means of the excavating process automation, at the same time the increase of operating equipment design parameters will allow avoiding the significant rise of load under resonance.

![Figure 2](image_url)

**Figure 2.** Dynamic properties of the hoisting ropes system: a – amplitude-frequency characteristics, b – operating equipment characteristic frequency versus design parameters relationship.

Figure 3 shows the gain factor versus the jib length and the bucket capacity relationship for two values of the external load frequency ($f$), one of which (0.8 Hz) corresponds to the low-frequency (sub-resonance) portion of the spectrum, while the other (4.8 Hz) one corresponds to the high-frequency (super-resonance) portion. The frequency selection is based on the studies of Olenich V.I. and Bagin B.P. who established that maximums can be observed at these particular frequencies within the spectrum of the excavator external load. The calculated value of the system characteristic frequency varies within 2.6-3.4 Hz depending on the changes in the operating equipment parameters.
Figure 3. External load gain factor: a – depending on the jib length, b – depending on the bucket capacity.

The analysis performed on the graphs proves that the gain factor of the low-frequency random component of the external load is practically independent of operating equipment design parameters (with an increase in the jib length and bucket capacity, it increases insignificantly as per the relationship, close to the linear one). At the same time the gain factor of the high-frequency component decreases sharply, asymptotically tending to zero.

4. Conclusion
Proceeding from the above-mentioned, it becomes clear that it is reasonable to increase the design parameters of the operating equipment in order to decrease the dynamic loafs of the mining excavator hoisting mechanism.

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