

# On justification of efficient Energy-Force parameters of Hydraulic-excavator main mechanisms

Anatoliy Komissarov<sup>1,2</sup>, Yuliya Lagunova<sup>1,2</sup>, Viktor Shestakov<sup>1</sup> and Olga Lukashuk<sup>2</sup>

<sup>1</sup>Ural State Mining University, Yekaterinburg, Russia

<sup>2</sup>Ural Federal University, Yekaterinburg, Russia

\*E-mail: ptmir@inbox.ru

**Abstract.** The article formulates requirements for energy-efficient designs of the operational equipment of a hydraulic excavator (its boom, stick and bucket) and defines, for a mechanism of that equipment, a new term “performance characteristic”. The drives of main rotation mechanisms of the equipment are realized by hydraulic actuators (hydraulic cylinders) and transmission (leverage) mechanisms, with the actuators (the cylinders themselves, their pistons and piston rods) also acting as links of the leverage. Those drives are characterized by the complexity of translating mechanical-energy parameters of the actuators into energy parameters of the driven links (a boom, a stick and a bucket). Relations between those parameters depend as much on the types of mechanical characteristics of the hydraulic actuators as on the types of structural schematics of the transmission mechanisms. To assess how energy-force parameters of the driven links change when a typical operation is performed, it was proposed to calculate performance characteristics of the main mechanisms as represented by a set of values of transfer functions, i.e. by functional dependences between driven links and driving links (actuators). Another term “ideal performance characteristic” of a mechanism was introduced. Based on operation-emulating models for the main mechanisms of hydraulic excavators, analytical expressions were derived to calculate kinematic and force transfer functions of the main mechanisms.

## 1. Introduction

Given the current state of competitive development of mining industry, it is no wonder that the problem of increasing manufacturing machinery quality (in particular, operational-equipment energy efficiency) is becoming a top-priority issue.

The scientific basis and main principles behind efficient operation of hydraulic excavators (as the main type of excavation machinery) were developed and formulated in the papers authored by Melnikov N. N., Ranev A. V., Satovskiy B. I., Steinzeig V. M. and other scientists. Further development of the theory of hydraulic excavators could be found in [1-7].

In a general case, a preliminary estimate of the equipment energy efficiency could be provided by means of comparing capacity-specific energy consumption to theoretical energy intensity of the process in question.

But it does not allow to evaluate the rate of energy consumption without assessing formation dynamics for the energy-force parameters realized on an operating element – the performance characteristic of a mechanism determining its operating potential.



## 2. The goal of the research and its tasks

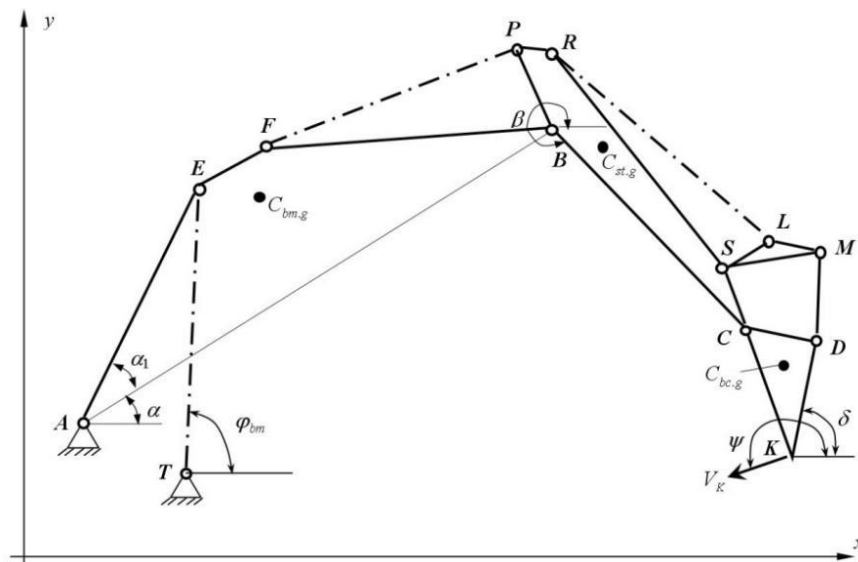
The goal of the research is to increase the operational energy efficiency of hydraulic excavators by defining the ideal performance characteristics of their main mechanisms.

The tasks which this paper is due to accomplish are:

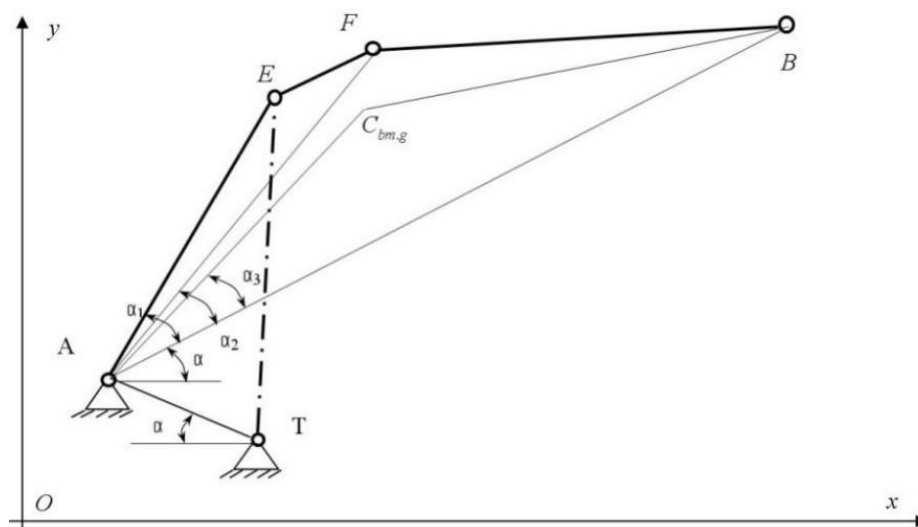
- to assess formation dynamics for the energy-force parameters realized on the links driven by the main mechanisms (a boom, a stick and a bucket);
- to determine performance characteristics of the main mechanisms;
- to define ideal performance characteristics of the mechanisms.

## 3. The solution of the research tasks

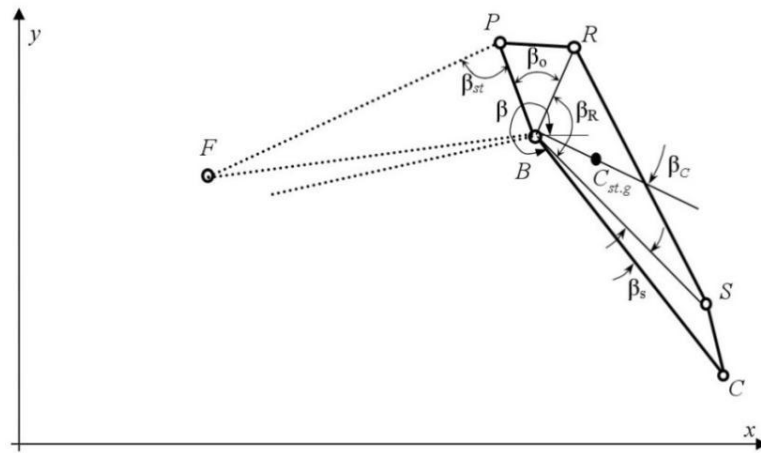
The object of the research are the main mechanisms of the operational equipment of a hydraulic excavator (the rotation mechanisms of its boom, stick and bucket). The diagrams for determining the coordinates of design-diagram points are shown in figures 1, 2, 3, and the ones for evaluating transfer functions – in figures 4, 5, 6.



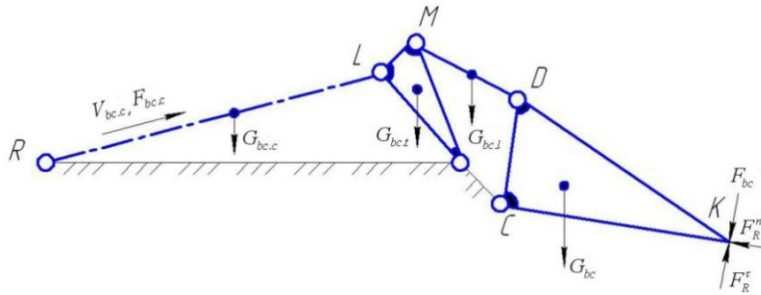
**Figure 1.** Schematic of the operational equipment.



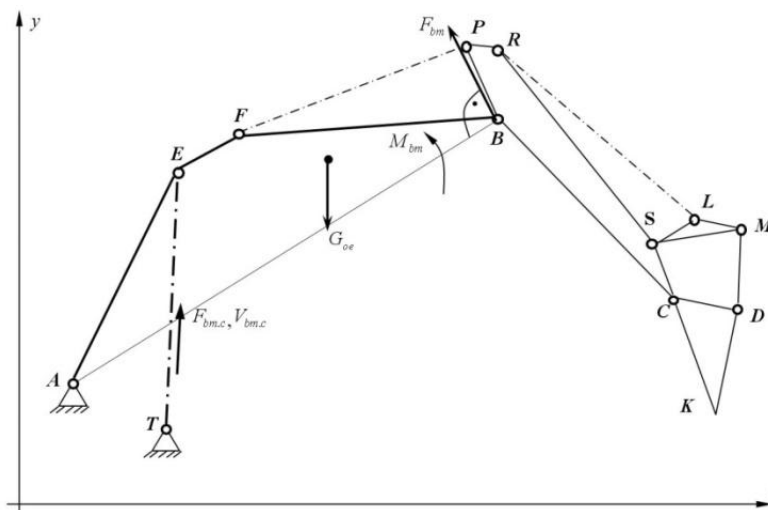
**Figure 2.** Diagram for determining coordinates of the boom elements.



**Figure 3.** Diagram for determining parameters of the stick.

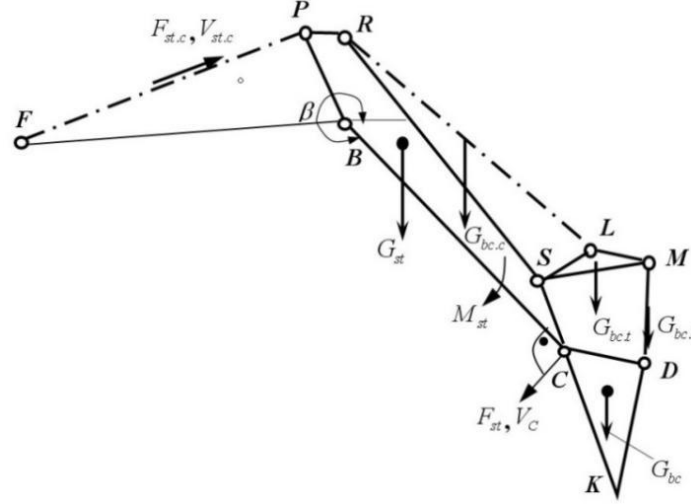


**Figure 4.** Diagram for determining transfer functions of the bucket-rotation mechanism:  $RL$  – length of the hydraulic cylinder (the generalized coordinate);  $V_{bc.c}$  – the speed of the cylinder rod;  $F_{bc.c}$  – the force on the cylinder rod;  $F_R^n$  and  $F_R^tau$  – the normal and tangential forces of excavation resistance;  $F_{bc}$  – the force on the cutting edge (teeth) of the bucket;  $G_{bc.c}$ ,  $G_{bc.t}$ ,  $G_{bc.l}$ ,  $G_{bc}$  – the gravity forces for the bucket cylinder, actuator (tipping) link, bucket link and bucket.



**Figure 5.** Diagram for determining transfer functions of the boom-rotation mechanism:  $V_{bm.c}$  – the speed of the cylinder rod;  $F_{bm.c}$  – the force on the cylinder rod;  $F_{bm}$  – the force on point  $B$ , which

generates  $M_{bm}$  – the torque relative to point  $A$  and equal to the moment of the force  $F_{bm,c}$ ;  $G_{oe}$  – the gravity force for the operational equipment



**Figure 6.** Diagram for determining transfer functions of the stick-rotation mechanism:  $V_{st,c}$  – the speed of the cylinder rod  $V_C$ ; – the speed of point  $C$ ;  $F_{st,c}$  – the force on the cylinder rod;  $F_{st}$  – the force on point  $C$ , which generates  $M_{st}$  – the torque relative to point  $B$  and equal to the moment of the force  $F_{st,c}$ ;  $G_{st}$ ,  $G_{bc,c}$ ,  $G_{bc,t}$ ,  $G_{bc,l}$ ,  $G_{bc}$  – the gravity forces for the stick, bucket cylinder, actuator (tipping) link, bucket link and bucket.

One feature of the main mechanisms of hydraulic operational equipment is kinematic linkage between hydraulic actuators (hydraulic cylinders) and driven links of the mechanisms. The linkage exists because the actuators themselves (or the cylinders, their pistons and piston rods) act as links of transmission mechanisms.

At the same time, relations between mechanical-energy parameters of the hydraulic actuators and energy-force parameters of the driven links (a boom, a stick and a bucket) depend both on the types of mechanical characteristics of the actuators and on the types of structural schematics of the transmission (leverage) mechanisms.

Based on operation-emulating models for the main mechanisms of hydraulic-excavator equipment [8, 9], analytical expressions were derived to calculate kinematic  $H_V$  and force  $H_F$  transfer functions of the main mechanisms, i.e. functional dependences between driven links and driving links (actuators).

The main excavating mechanism is the bucket-rotation mechanism. Its transfer functions are defined as:

$$H_{V.bc} = \frac{V_K}{V_{bc,c}} = \frac{CK \cdot MS \sin(\angle DMS)}{CD \cdot RS \sin(\angle LRS) \sin(\angle CDM)}; \quad (1)$$

$$H_{F.bc} = \frac{F_{bc}}{F_{bc,c}} = \frac{CD \cdot RS \sin(\angle LRS) \sin(\angle CDM)}{CK \cdot MS \sin(\angle DMS)} + \Delta; \quad (2)$$

$$\Delta = \frac{CD \cdot \sin(\angle CDM)}{F_{bc,c} CK} \left[ \frac{G_{bc}(X_{bc,g} - X_C) - 0,5G_{bc,l}(X_C - X_D)}{CD \sin(\angle CDM)} - \frac{G_{bc,t}(X_S - X_{bc,t,g}) + 0,5G_{bc,c}(X_S - X_L) + 0,5G_{bc,l}(X_S - X_M)}{MS \sin(\angle DMS)} \right],$$

where  $V_K$  – the speed of excavation (for point  $K$ );  $V_{bc,c}$  – the speed of the cylinder rod;  $F_{bc}$  – the force on the cutting edge (teeth) of the bucket;  $F_{bc,c}$  – the force on the cylinder rod;

$\angle DMS, \angle LRS, \angle CDM$  – angles;  $G_{bc}, G_{bc.l}, G_{bc.t}, G_{bc.c}$  – the gravity forces for the bucket, the bucket link, the actuator (tipping) link and the bucket cylinder;  $X_C, X_D, X_S, X_L, X_M$  – the coordinates of points;  $X_{bc.g}, X_{bc.t.g}$  – the center-of-gravity coordinates for the bucket and the actuator (tipping) link.

The positions of the bucket-rotation mechanism links and the angles  $\angle DMS, \angle LRS, \angle CDM$  are determined by the generalized coordinate  $RL$  and the fixed parameters of the mechanism – the linear and angular parameters of the links.

The point coordinates for the mechanism  $X_R, X_S, \dots, X_D$  and the center-of-gravity coordinates for the bucket and the actuator link  $X_{bc.g}, X_{bc.t.g}$  are determined in the Cartesian coordinate system with its center set at the point of intersection between the rotation axis of the rotating platform (axis  $Y$ ) and the datum level of the excavator (axis  $X$ ) (figure 1).

The value of  $\Delta$  is determined by the ratio between the gravity force of the equipment elements and the force on the cylinder rod  $F_{bc.c}$ , but actually it also depends on the bucket position.

$\Delta$  is at its maximum (for the bucket (line  $CK$ ) positioned horizontally) in two cases: in the beginning of excavation (when the forces  $G_{bc}$  and  $F_{bc}$  act in the same direction), with  $\Delta_{beg.max} = 0,02$ ; and in the end of excavation (when the forces  $G_{bc}$  and  $F_{bc}$  act in the opposite directions), with  $\Delta_{end.max} = -0,06$ .

On the whole, the position of the bucket also determines its load duty. In the beginning (when the force  $G_{bc}$  and the tangential force of excavation resistance  $F_R^r$  act in the opposite directions), the summation of external loads is at its minimum; in the end of excavation, it increases.

Table 1 cites the values of the transfer functions calculated for the bucket-rotation mechanism of a EGO-150 excavator by Uralmashplant (while the excavator in the process of excavation is extending the bucket cylinder rod [10]).

**Table1.** Calculation results of bucket-rotation mechanism transfer functions.

Transfer functions	Bucket-cylinder length, m					
	3,23	3,645	4,06	4,475	4,89	
$H_{V.bc}$	4,82	2,64	2,49	2,75	3,69	
$H_{F.bc}^*$	0,206	0,378	0,402	0,366	0,270	

The relative values of the force parameters (and accordingly, the forces on the bucket teeth) are increasing until the cylinder rod has been extended to an approximately one-half length of the rod stroke, and then they are decreasing. This means that the performance characteristic of that bucket-rotation mechanism does not provide correspondence between the values of the force parameters realized on the bucket and the bucket load duty (characterized by increase in external-load rate during excavation). It follows that, during soil excavation, the hydraulic actuator becomes overloaded by the end of the stroke, which could trigger its safety valve and stop the whole process.

Thus, the nominal mode of operation for the hydraulic actuator of the bucket-rotation mechanism (with the installed capacity of the actuator being used to its full) is realized (in the case of the ideal performance characteristic of the bucket-rotation mechanism) in the form of a monotonically increasing function.

For the boom-rotation mechanism, the transfer functions will be (figure 2):

$$H_{V.bm} = \frac{V_B}{V_{bm.c}} = \frac{AB \cdot ET}{AE \cdot AT \sin x}; \quad H_{F.bm} = \frac{F_{bm}}{F_{bm.c}} = \frac{AE \cdot AT \sin x}{AB \cdot ET}; \quad (3)$$

$$x = \arccos \left( \frac{AE^2 + AT^2 - ET^2}{2AE \cdot AT} \right),$$

where  $V_{bm.c}$  – the speed of the cylinder rod;  $V_B$  – the speed of point  $B$ ;  $F_{bm.c}$  – the force on the cylinder rod;  $F_{bm}$  – the force on point  $B$ , which generates  $M_{bm}$  – the torque relative to point  $A$  and equal to the moment of the force  $F_{bm.c}$ .

For the stick-rotation mechanism, the transfer functions will be (figure 3, 6):

$$H_{V.st} = \frac{V_C}{V_{st.c}} = \frac{BC}{BF \sin y}; \quad (4)$$

$$H_{F.st} = \frac{F_{st}}{F_{st.c}} = \frac{BF \sin y}{BC}; \quad (5)$$

$$y = \arccos \left( \frac{BF^2 + FP^2 - BP^2}{2BF \cdot FP} \right),$$

where  $V_{st.c}$  – the speed of the cylinder rod;  $V_C$  – the speed of point  $C$ ;  $F_{st.c}$  – the force on the cylinder rod;  $F_{st}$  – the force on point  $C$ , which generates  $M_{st}$  – the torque relative to point  $B$  and equal to the moment of the force  $F_{st.c}$ .

The values of the force transfer functions of the boom-rotation and bucket-rotation mechanisms have decreased by the end of their cylinder rod strokes. This means that, at that moment, the hydraulic actuators of the mechanisms become overloaded, which could trigger their safety valves.

#### 4. Conclusion

The analysis of the performance characteristics of operational-equipment mechanisms allows to assess the impact which the kinematic schemes of the mechanisms have on the operational efficiency of those mechanisms and a hydraulic excavator as a whole.

The synthesis of the ideal performance characteristics of operational-equipment mechanisms allows to increase usage of the installed capacity for the driving actuator and, ultimately, provide higher values of engineering-and-economical operational characteristics for hydraulic excavators.

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