ISSN: 1231-4005
e-ISSN: 2354-0133
DOI: 10.5604/12314005.1136589

ENERGY RECOVERY SYSTEM FOR EXCAVATORS WITH MOVABLE COUNTERWEIGHT

Artur Gawlik

Cracow University of Technology
Institute of Machine Design
Jana Pawla II Av. 37, 31-864 Cracow, Poland
tel.:+48 12 6283352, fax: +48 12 6283360
e-mail: agawlik@mech.pk.edu.pl

Abstract

Due to the high fuel costs are developed the different types of energy saving systems that are supposed to increase the productivity of machines per each burned liter of fuel. Paper describes the concept of an active counterweight system designed for excavators. Suggested solution includes both the kinematical structure and the hydraulic system. The hydraulic system of active counterweight is connected with the boom standard hydraulic system. The counterweight mechanism is homothetic to working mechanism of the excavator. The homothetic transformation applies to kinematical structure and positions of the centres of gravity. The homothetic transformation provides to static unloading of boom cylinder by the mated counterweight cylinder. The aim of investigations is saving energy by machines which can use potential energy of movable counterweight. The research tests are performed for selected work cycle to determine the key parameters of hydraulic system, such as: cylinders velocity, working pressure, oil flow etc. Primary test results compare power consumption for standard and modification system during the same work cycle. The level of energy recovery is promising especially at lower velocities of boom mechanism cylinder. The energy saving for higher boom cylinder velocity will be possible after reduce of pressure drop in hydraulic circuit.

Keywords: *active counterweight, excavator, homothetic transformation, energy saving*

1. Introduction

The development works of construction machinery are aimed at reducing operating costs while maintaining full functionality of the machines. One of the directions of research is energy recovery from swing mechanism of excavator [5]. Komatsu offers a model of excavator with hybrid system. This system converts energy generated when the upper structure reduces its speed while turning, stores the energy in the capacitor, and uses it to assist the power of the engine via the electric motor when the upper structure accelerates. Compared with the standard model of hydraulic excavator, the hybrid model achieves about 25% reduction of fuel consumption. Caterpillar chose the all-hydraulic approach because of the high power density of hydraulics. Their system reuses energy via the hydraulic hybrid swing system, which captures the excavator's upper structure swing brake energy in hydro accumulators, and then releases the energy during swing acceleration. This hydraulic hybrid system allows improve fuel efficiency up to 25 percent [7].

Next of the directions of research is energy recovery from boom mechanism [1]. It is to be implemented in machines such as excavators, where is repetition of the similar work cycles and the possibility of using the potential energy of the working equipment in the lowering phase. Recovered energy can be accumulated in hydro-pneumatic accumulators, but this solution does not allow for a smooth cooperation with the hydraulic system. In particular, the phases of charging and discharging of accumulators require the use of advanced control strategies. The author suggests a solution of system based on active counterweight. Typical counterweights are assembled quite often in the construction equipment structures but these elements are normally fixed.

A counterweight driven by a hydraulic cylinder is used in specialty machines such as the pipelaying cranes. It is however, only a passive solution which works apart from the boom mechanism. This system can only increase the static stability of the machine. However, port cranes have movable counterweight which is permanently driven by the boom mechanism. The counterweight is connected with boom mechanism through a mechanical links. This solution unloads the drive mechanism of crane boom in an approximate manner and increases the stability of the crane. Many experts announces the movable counterweight in construction of excavators next generation but it will be use only to improve of machines stability during digging.

2. System structure of moveable counterweight

Connection by hydraulic system of movable counterweight mechanism and excavator linkage is suggested. The excavator boom cylinder could collaborate with two units of cylinders which drive parallel parts of moveable counterweight. Presented solution of hydraulic system does not change functional properties of the excavator. To begin with, few assumptions were made: the mass of an active counterweight should not be bigger than the mass of the standard counterweight by reason of additional resistance to the motion of a machine and necessary modification of kinematics pairs. Moreover, the movable link of the counterweight should not exceed the superstructure contour. These assumptions limits homothetic transformation scale factor to level about k=0.5. After an analysis of linkages and fixed counterweights mass for different excavators, middle-sized excavators have value of factor the most close to definite factor. It is about 0.9. Due to the dimensions of the original cylinder of machine Cat 305 and dimensions of typical hydraulic cylinders, which have been used in laboratory stand, the scaling factor k = 0.77 for tested active counterweight system was established.

The centre of mass vector for excavator linkages and counterweight shows formula (Michalowski, S., Gawlik, A.):

$$\vec{r}_s = \left(x_s, y_s\right) = \frac{\left(x_o m_o g + x_p m_p g \quad , \quad y_o m_o g + y_p m_p g\right)}{g\left(m_o + m_p\right)}. \tag{1}$$

The first part of the numerator of a formula (1) presents sum total for gravitation force moments, which act on linkages and counterweight. The second part of the numerator shows the sum total of potential energy for excavator and active counterweight links. The correlation between the counterweight centre of a mass and the machine linkage centre of mass could be calculated by a method where components of the centre of the total mass vector are constants ($\vec{r}_s = \text{const}$). This situation gives two crucial effects:

- a static overturning moment will be constant for different linkage set up,
- a connection between excavator and counterweight systems allows for energy flow between described mechanisms and produces mutually static unloading of working linkages.

Vector of movable counterweight centre of mass at point P described by the equation (2):

$$\vec{r}_p = -k\,\vec{r}_o + \left(1+k\right)\vec{r}_s\,. \tag{2}$$

Taking into consideration that $\vec{r}_s = \text{const}$, result from equation (1) that point P of centre of mass for active counterweight trajectory should be homothetic to point O of centre of mass for excavator linkage trajectory. Homothetic transformation J_S^{-k} has centre in point S and scale factor –k (Fig. 1.). It is possible to achieve only if the counterweight mechanism is homothetic to the linkage mechanism. Fig. 2. presents simplified active counterweight mechanism in relation to typical linkages. First link, which is connected to excavator frame, is a result of homothetic to boom. The second element corresponds to motion of bucket and stick. Taking into consideration the kinematics of excavator links, system where the boom cylinder is connected to the main cylinder of the counterweight should have the greatest effect of energy savings.

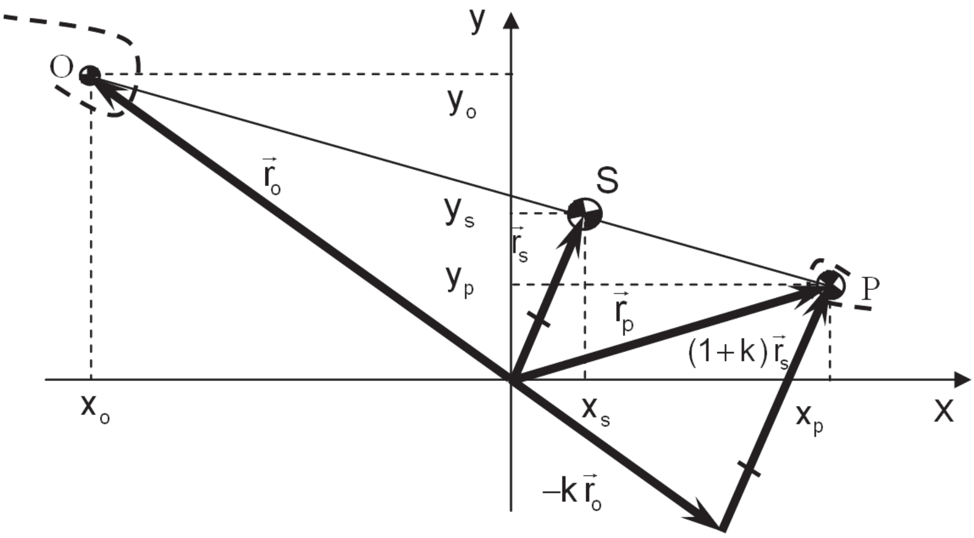

Fig. 1. Homothetic transformation of excavator and active counterweight system for scale factor k = 0.5

A static moment round the pin of the boom and the pin of the first link of counterweight allows calculated equation $F_o = k{\cdot}F_p$. The connected hydraulic system should ensure velocity for counterweight cylinder as the formula shows $v_p = k{\cdot}v_o$. The same goes for areas of cylinders $A_1 = k{\cdot}A_4$.

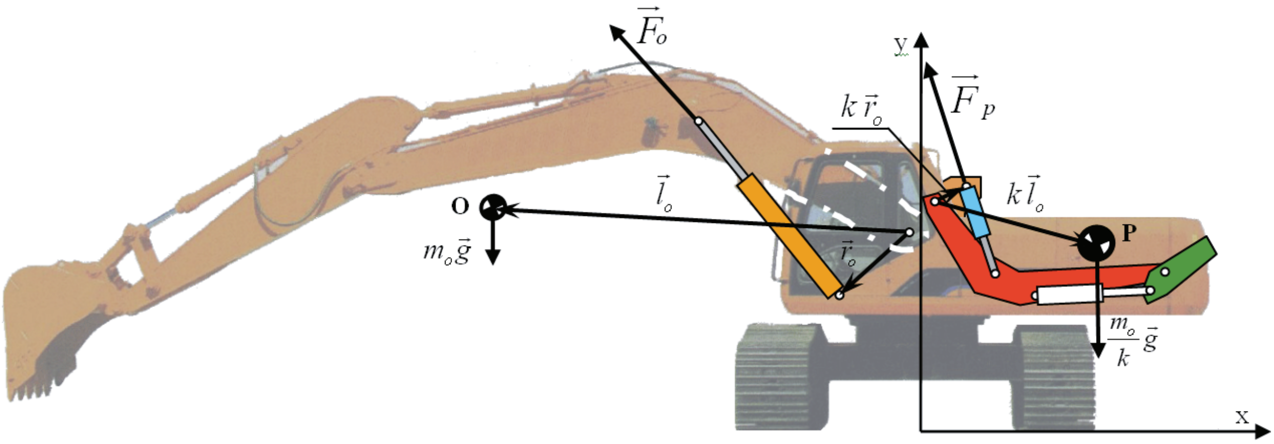

Fig. 2. Structure of excavator with active counterweight

3. Energy saving system with moveable counterweight

The mechanical part of test stand from original parts of Caterpillar 305 excavator was assembled. This solution has allowed to keep the kinematic structure which is typical for backhoe. Considering the limited field work in the laboratory, the test stand without bucket mechanism was used. The presented results for two configurations of excavator linkages with fixed minimal and maximal length of stick cylinder were obtained. Load on the end point of stick from 0 kg to 80 kg was changed.

In Fig. 3., main part of hydraulic system for connection between excavator mechanism and active counterweight mechanism is shown. The control strategy with quasi-constant level of working pressure (10 MPa) in the supply line was adopted for the hydraulic pump. Two units of Parker D3FP proportional directional control valve are used. This solution gives independent control of each cylinder power line. This allows reaching the parameters (pressure, flow) of the hydraulic system similar to those registered on the system which was built from factory hydraulic parts.

For comparison of the energy consumption in the standard hydraulic system of excavator and in the hydraulic system which incorporates the active counterweight, a specific work cycle was selected. Excavator boom was lowered by boom cylinder, later stopped for few seconds and in next phase was raised back to starting position. A ramp signal to determination actual stroke of the boom cylinder was used. Control signal for directional valve was generated by PID controller. Motion of excavator linkages for three different levels of cylinder velocity during boom raising and boom lowering was realized.

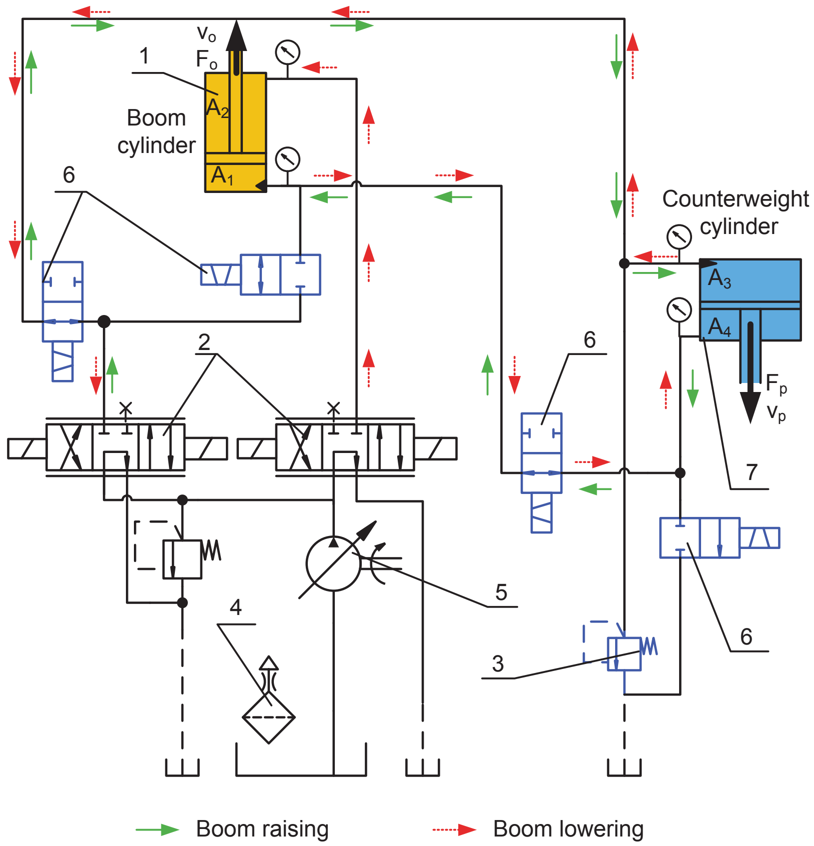

Fig. 3. Scheme of hydraulic system to test of mechanism with active counterweight: 1 – boom cylinder, 2 – distribution valve 4/3-way D3FP, 3 – relief valve, 4 – filter, 5 – variable pump PV046_UPG, 6 – valve 2/2-way, 7 – main counterweight cylinder

4. The results of comparative tests

The presented hydraulic circuit allows use the research stand as ordinary excavator system or as hydraulic system connected to main cylinder of counterweight. The same work cycles during comparative tests for both configurations were realized. Standard excavator system was tested to obtain reference data to comparison with parameters of active counterweight system. The all tests for the three speed levels of boom cylinder (0.023, 0.045, 0.09 m/s) were carried out. Selected parameters such as pressure, stroke and velocity of the boom cylinder in Fig. 4 are shown. Additionally, the calculated values of flow and hydraulic power of main cylinders are presented. Inlet power was calculated based on flow rate and pressure in active chamber of cylinder.

The computed area under power line (hatched area) allows determining energy consumption during each phase of the work cycle. Demand for energy is obviously higher in the lifting phase of the excavator linkages for the same velocity values of the cylinder. The oscillation of cylinder velocity has resulted from the operation of the PID controller in control system and was visible only at the lowest values of speed.

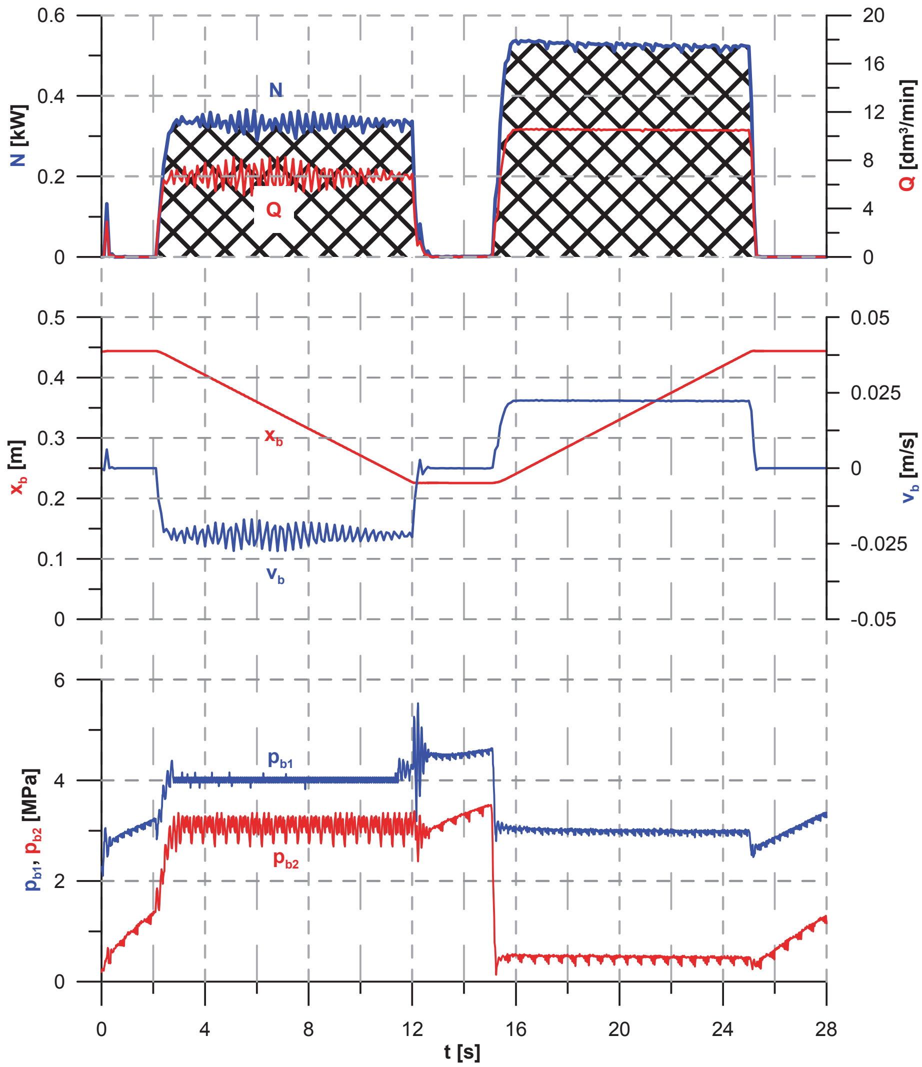

Fig. 4. Selected work parameters of the ordinary excavator system for low velocity of boom cylinder

Using the same control signals for a system with an active mechanism of counterweight was carried an experiment. Active chamber in the cylinder during raising excavator equipment was piston side chamber in cylinder of counterweight mechanism. On the other hand, during lowering the excavator working equipment, piston rod chamber of boom cylinder was an active chamber. Selected parameters for connected mechanisms of excavator and active counterweight in Fig. 5 are shown.

The test results a significant correlation for velocity of boom cylinder about 0.023 [m/s] and 0.045 [m/s], level of pressure p_{b1} and p_{b2} was similar to level of pressure which was observed during experiments with standard excavator system. Energy consumption was on the same level for both values of velocity but only during retracting of boom cylinder. Energy saving in raising phases of excavator linkages was received. It is result that system was actuated (in these phases of test cycle) by counterweight cylinder which had bigger piston area than piston area of boom cylinder and was supported by the forces of gravity of counterweight elements. In this way, the fluid pressure was reduced. Comparing the standard system of excavator and active counterweight system, 40% of energy saving for the boom cylinder velocity about 0.023 [m/s] and 30% for the higher cylinder velocity about 0.045 [m/s] were achieved. Decrease in the value of recovery energy justifies the increase in hydraulic losses in the lines connecting each element of system. Further increase in velocity of the boom cylinder and thus the oil flow rate progressively reduces the effectiveness of energy recovery system. For the fixed velocity value of boom cylinder about 0.09 [m/s], energy losses in the system was observed. Losses were result of the increasing flow resistance in the line between connected chambers of the both cylinders and the valves. The drop pressure was due to by to small size of cylinder ports, which also determined the diameter of the hydraulic lines between the connected cylinders. Mainly during lowering of excavator linkages this negative phenomenon was registered. At the time from the bigger chamber of main counterweight cylinder the oil flows to valve. Hydraulic losses in this line up to 2 [MPa] for flow rate about 60 [dm^3/min] were measured. Energy saving in lowering phase of boom cylinder is also less and global result for this value of cylinder velocity is negative. Calculated energy consumption for both systems in the Fig. 6 is shown.

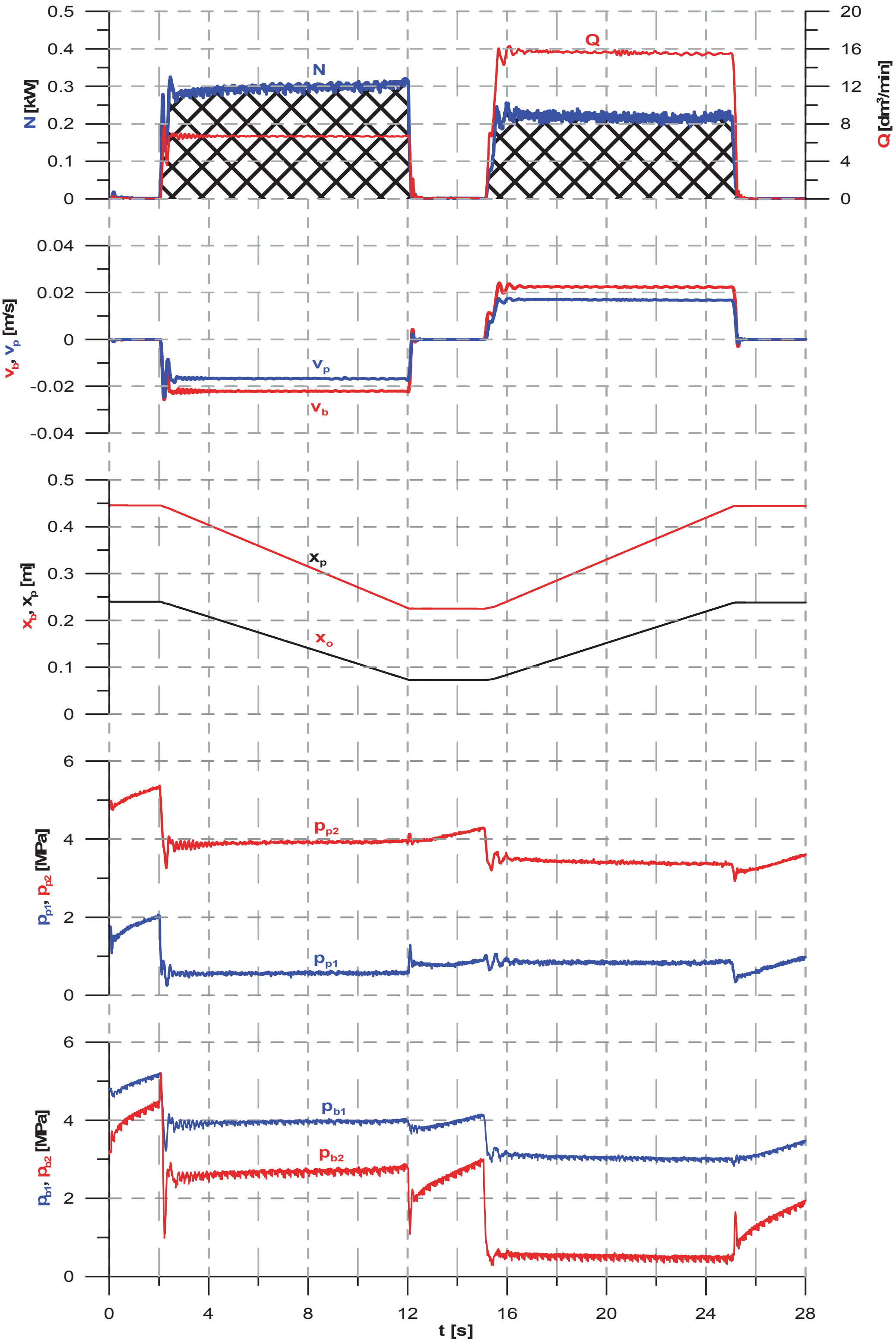

Fig. 5. Selected work parameters of the energy saving system for low velocity of boom cylinder

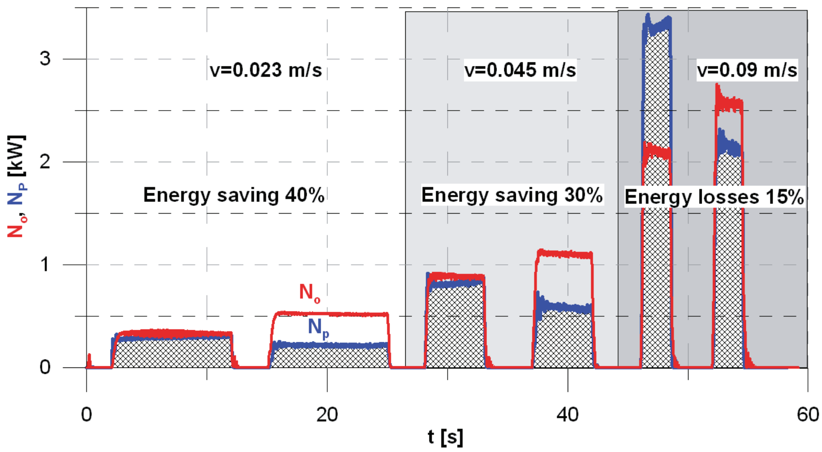

Fig. 6. Calculated energy consumption for the both tested systems without additional load

5. Conclusion

Test stand to compare of hydraulic systems for standard excavator mechanism and mechanism with movable counterweight was prepared. The same work parameters of hydraulic system as in ordinary excavator were obtained by using two units proportional directional control valve. Previous test results confirmed possibility of energy saving (up to 40%) by using active counterweight system for excavators. The main portion of saving energy during extension phase of boom cylinder was visible. This system for smaller values of boom cylinder velocity was ready to use without special modifications in the hydraulic system on the machine. In order to reduce energy losses for the larger flow rate in the supply lines hydraulic circuit requires some changes in the elements for example - the size of the connection ports of the cylinder should be increased. In the research stand was assembled one cylinder with huge area of piston and piston rod side. System with active counterweight requires two hydraulic cylinders operating simultaneously and therefore the hydraulic losses in the connection connectors (at the same flow rate) will be smaller. At this point, it seems important to underline that the scale factor for homothetic transformation of counterweight elements should not differ significantly from the value of 0.5. When the value of factor k goes up the oil flow increases and further increase in hydraulic losses is expected. Another way is to minimize the number of hydraulic elements in circuit and pipe lengths between the cooperating cylinders.

For both configuration of mechanical-hydraulic system mathematical model was prepared. This theoretical model of the hydraulic system must be expanded to include additional equations describing the pressure loss in more detail. Further research work is oriented for determination of energy saving level for different equipment configuration and load. The efficiency of an active counterweight system requires verification by simulation and further testing. Presented hydraulic system does not change the functional properties of the excavator and allows the operator to switch back to standard system when it is necessary (for example while working as a lift).

Tab. 1. List of notations

x_b	Boom cylinder displacement	m
x_p	Cylinder displacement of active counterweight	m
m_o, m_P	Total mass of excavator and counterweight linkages	kg
k	Scale of homothetic transformation ($k = m_o / m_P$)	
A_1,A_2,A_3, A_4	Area of piston and rod piston side of each cylinder	m^2
v_o, v_p	Linear velocity of boom and counterweight cylinder	m/s
p_{b1}, p_{b2}	Pressure in boom cylinder piston and rod piston side	MPa
p_{p1}, p_{p2}	Pressure in counterweight cylinder piston and rod piston side	MPa
Q	Flow rate	dm^3/min
N_o, N_p	Inlet power in ordinary and active counterweight system	kW

References

[1] Cetinkunt, S., Pinsopon, U., Chen C., Egelja, A., Anwar, S., *Positive flow control of closed-centre electrohydraulic implement-by-wire systems for mobile equipment applications*, Mechatronics 14, pp. 403-420, 2004.
[2] Liang, X., Virvalo, T., *Energy reutilization and balance analysis in a hydraulic crane*, IHA, Tampere University of Technology, 2003.
[3] Michalowski, S., Gawlik, A., *Concept of active counterweight system for medium size excavators,* Proc. Of 5-th FPNI-Phd Symposium, pp. 462-465, Cracow 2008.
[4] Qing, X., Qingfeng, W., Yanting, Z., *Control strategies of power system in hybrid hydraulic excavator*, Automation in Construction 17, pp. 361-367, 2008.
[5] Tianliang, L., Qingfeng, W., Baozan, H., Wen, G., *Development of hybrid powered hydraulic construction machinery*, Automation in Construction 19, pp. 11-19, 2010.
[6] Qing, X., Qingfeng, W., Yanting, Z., *Control strategies of power system in hybrid hydraulic excavator*, Automation in Construction 17, pp. 361-367, 2008.
[7] www.cat.com, 2013.