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APPLICATION OF A HYDRAULIC – LINKAGE MECHANISM TO BALANCE THE EXCAVATING EQUIPMENT IN HEAVY MACHINES

The author suggests that a mobile counterweight mechanism could be introduced to the excavator structure for coupling the hydraulic system with the excavating equipment. It is shown that the mobile counterweight mechanism reduces power demand, at the same time improving stability of the excavator.

1. Introduction

In certain machines, the centre of mass of particular machine components change its position whilst in service. Major variations of the horizontal coordinate of the centre of mass prompt us to search for new solutions to ensure stability of the machine. Significant change of the vertical coordinate is associated with higher power consumption required to lift those elements.

In some machines, specially designed counterbalances are provided which in order to stabilize the centre of mass position. Pipe-laying machines are equipped with mobile counterweights which ought to stabilize the machine and pipes. The solution applied in crane structures involves a mobile counterweight connected with the boom using by means of a linkage mechanism [5]. In this case, the mobile counterweight balances the boom drive, thus improving the crane stability. The motion of links in the excavator equipment causes that the centre of mass of the whole machine to be is displaced [2]. The designs of such machines incorporates an immobile counterweight in having the shape of profiled iron casts, placed on the rotary frame, on the opposite side to the excavating equipment. The position and mass of the counterweight are precisely controlled, such so that the counterweight should not protrude beyond the approved machine profile

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and the motion of the machine's centre of mass must would not cause any stability loss. Hydraulic drives in some machines enable the partial recovery of potential energy of the mobile machine elements. Instantaneous energy surplus can be stored in hydraulic, electric or inertial accumulators [4]. In the case, when accelerations of machine equipment elements are comparable with g , inertial forces and moments appears, increasing significantly the load on linkages and joints bolts load. It's It is possible to reduce this effect by the use of optimal mass arrangement of in the equipment linkages [1], [3].

2. Mobile counterweight mechanism

Let us consider an excavator design completed with a mobile counterweight coupled with the excavating equipment (Fig 1). The motion of the counterweight should be related to that of the machine centre of mass to immobilize the centre of the total mass. In this way, one can to ensure that:

- when the bucket is empty, the overturning moment when the bucket is empty should be constant for any arbitrary position of the excavating equipment,
- the excavating equipment and the counterweight mechanisms are coupled, enabling facilitating energy flow between those these mechanisms allowing the for static balancing of the drives with respect to specific weights of the excavating equipment and the counterweight.

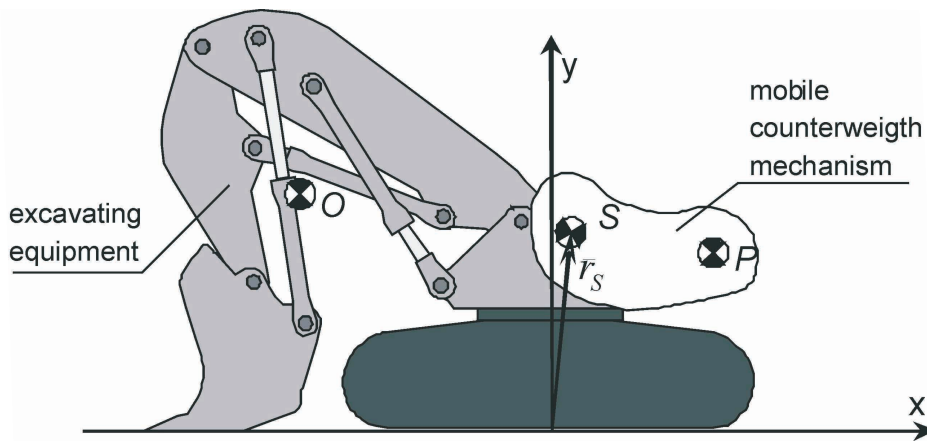


Fig. 1. Excavating equipment in a backhoe excavator with a mobile counterweight

One expects that Tthe coordinate's of vector of the total mass centre S is expressed asare:

$$(x_S, y_S) = \frac{(x_O m_O g + x_P m_P g, y_O m_O g + y_P m_P g)}{g(m_O + m_P)}, \tag{1}$$

where: (x_O, y_O) – coordinates of the point O – centre of mass of the working excavating equipment, (x_P, y_P) – coordinates of the point P – centre of mass of the counterweight, m_O – mass of excavating equipment links, m_P – mass of counterweight links.

The first coordinate in the numerator of (1) represents the sum of moments due to gravity forces acting upon the links of the excavating equipment and the counterweight, with respect to the origin of the coordinate system. The second coordinate expresses the sum of potential energies of the excavating equipment and the counterbalance links. Eq (1) yields the vector of instantaneous position of the point P (Fig 2):

$$\vec{r}_P = -k\vec{r}_O + (1+k)\vec{r}_S, \quad (2)$$

where: \vec{r}_O – vector of instantaneous position of point O , $k = m_O/m_P$ – proportion factor expressing the ratio of excavating equipment to the counterweight mass.

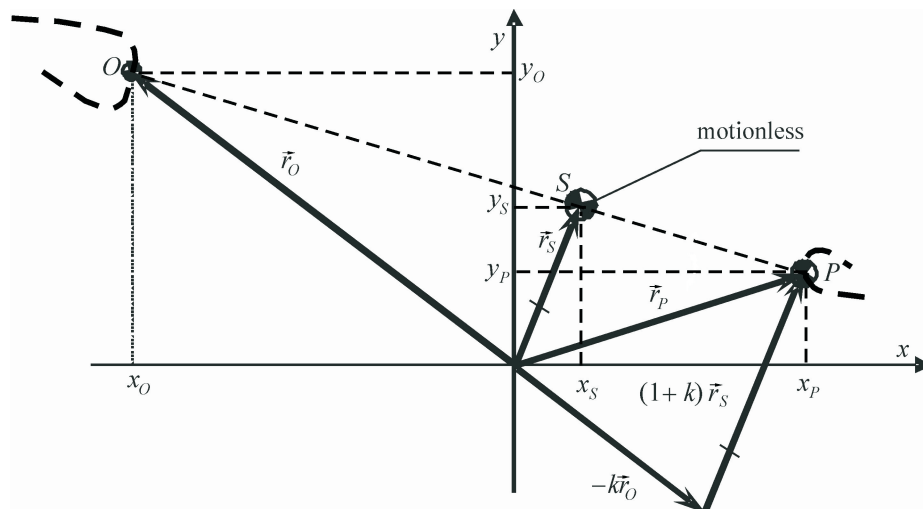


Fig. 2. Radius vector of the centre of mass points: excavating equipment, counterweight and the total mass

Assuming that $\vec{r}_S = \text{const}$, it follows from Eq (2) that the trajectory of the point P should be homothetic to the trajectory of the point O . The homothetic transformation is performed with respect to the centre S and in scale $-k$. It is achievable as long as the counterweight mechanism is homothetic with respect to the excavating equipment. Fig 3 shows the mobile counterweight mechanism formed by a homothetic transformation.

To find the relationship between the force acting from the cylinder on the boom F_O and the force acting from the cylinder onto the boom in the mobile

counterweight mechanism F_P , we derive equations are derived of moment balance around the pin bolts A and B (Fig 3):

$$\vec{r} \times m_O \vec{g} + \vec{l} \times \vec{F}_O = \vec{0}, \quad -k\vec{r} \times \frac{m_O}{k} \vec{g} - k\vec{l} \times \vec{F}_P = \vec{0} \quad (3)$$

Adding Eq (3) by sides yields:

$$F_O = k F_P. \quad (4)$$

The coupling system ought to should ensure the required linear velocities of particular points of the equipment mechanism and the counterweight links, the proportionality factor k being retained. This proportion applies also to velocities of the cylinders' motion in the excavator equipment and the counterweight:

$$v_P = k v_O, \quad (5)$$

where: v_P – velocity of piston protruding motion in the counterweight cylinder, v_O – velocity of piston protruding motion in the cylinder of the equipment.

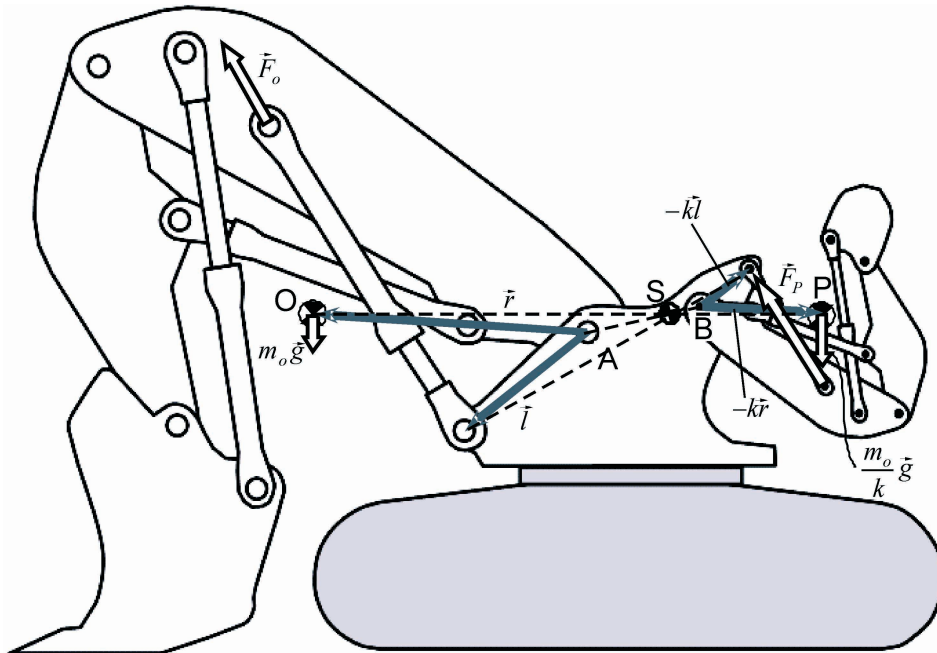


Fig. 3. Forces acting on excavating equipment and the mobile counterweight

The concept of hydraulic coupling of the pair of cylinders in the excavator equipment and the counterweight mechanism is shown schematically in Fig 4.

The hydraulic system incorporates two lines: the upper – high pressure line where the two cylinders support one another. The other- low – pressure line induces the piston motion. The position of a directional valve controls the direction of motion of the two cylinders (in the excavator equipment and in the counterweight mechanism). Eq (5) yields the relationship:

$$s_2 = k s_4, \quad (6)$$

where: s_2 – surface area of a the cylinder in a the excavator equipment, s_4 – rod area of the cylinder in the counterweight.

The required proportion of these two surfaces in Eq (6) leads to Eq (4), which is satisfied in each position for the considered mechanisms of the excavating equipment and the mobile counterweight, ensuring the full static equilibrium of the cylinder in the excavating equipment due to the pressure generated in the counterweight cylinder.

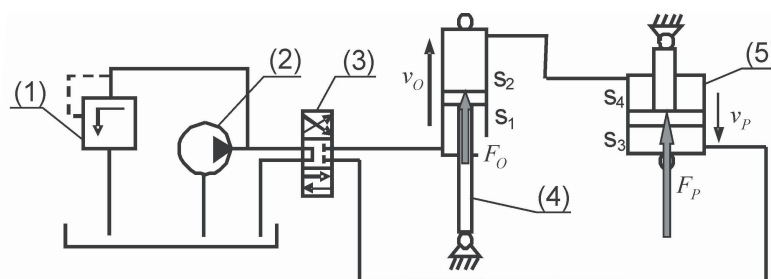


Fig. 4. Schematic diagram of hydraulic coupling of the cylinder in the excavating equipment with that in the mobile counterweight mechanism. (1) relief valve (2) pump, (3) distribution valve, (4) equipment cylinder, (5) counterweight cylinder

To find the potential energy savings, estimate calculations were performed for a backhoe excavator CAT 320D LRR. The summated mass of the excavating equipment is 2830 kg, the counterweight mass is 6500 kg. Assuming the same mass of the mobile counterweight, the value of factor k becomes $k=0.435$. For this value of k , the horizontal dimension of the mobile counterweight, when it is maximally protruded, approaches 4.1 m. when the bucket is lifted from the level of -5.89 m to 5.38 m, the position of the mass centre of mass of the excavating equipment changes in the horizontal direction by 7.08 m. Energy required to lift the equipment with an empty bucket roughly equals 196 kJ. For comparison, the energy required to lift the bucket with the maximal payload of 1800 kg amounts to 395 kJ. In this case, the energy savings associated with the transport function of the machine might approach 50%.

It is shown how particular cylinders in the excavator system contribute to energy consumption during the motion of a machine with an empty bucket,

and the contribution of the cylinder in the boom proves to be the greatest (cylinder in the boom: 83%, arm: 14%, bucket: 3%). These figures encourage us to focus on the boom cylinder. It is not necessary that to exactly emulate the mobile counterweight mechanism be exactly emulated on the basis of the structure of the excavating equipment. It can be simplified by eliminating the cylinder connected to the ultimate link, which corresponds to the bucket in the excavator. This link ought to be rigidly connected to the previous one, representing the arm. This simplification, however, might lead to a slightly weaker drive balancing performance.

3. Conclusions

The proposed modification of the excavator design has obvious benefits, though certain side -effects have to must be investigated. It might happen that a group of machines, where these structural changes are possible, shall would be slightly restricted. Modification of the hydraulic system will increase the losses due to flow resistance and friction forces occurring in additional cylinders in the counterweight mechanism. Incorporation of a mobile counterweight increases the number of mobile masses and hence the dynamic loading during the start-up, and braking will go up too. The mobile counterweight requires an extra space on the machine chassis. The smaller the ratio between the equipment mass to that of the counterweight, the less space shall be needed. In the case of backhoe excavators, the weight of the excavating equipment itself supports the earthmoving. Once the proposed modifications are introduced, this positive effect will be eliminated. The reverse is true for push shovels. When a mobile counterweight is incorporated, the bare weight of the equipment will no longer interfere with earthmoving tasks. Another solution is explored whereby the mobile counterweight mechanism should be radically simplified, incorporating a single mass supported on two cylinders controlled in such a way that the Eq (2) should be is satisfied. However, hydraulic coupling shall would require a much more complicated hydraulic system.

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Zastosowanie układu dźwigniowo-hydraulicznego do odciążenia osprzętu maszyn roboczych

Streszczenie

Artykuł zawiera propozycję umieszczenia w koparce mechanizmu ruchomej przeciwwagi, sprzęgniętego poprzez układ hydrauliczny z osprzętem. Wykazano że mechanizm ruchomej przeciwwagi zmniejszy zapotrzebowanie na energię, potrzebną do napędów osprzętu oraz polepszy stateczność koparki.