Povećanje radne efikasnosti mašina za premeštanje tla pomoću pneumatskog sistema za skladištenje energije

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Ovaj rad prezentuje ciklograme distribucije snage i šeme distribucije tokova snage kod radnog ciklusa buldozera, skrepera i grejdera, opremljenih sa pumpnim sistemima za skladištenje hidraulične energije.

Ključne reči: akumulator, energija, hidraulični pogon, snaga, tok snage

1. UVOD

U sadašnjem vremenu, teško je pronaći težak i naporan posao iz građevinskog, urbanog ili poljoprivrednog sektora koji nije mehanizovan. U svim pomenutim oblastima koriste se mašine za zemljane radove (MZR). Jedno od glavnih pitanja masovne proizvodnje je cena proizvoda. Cena goriva, koja neprestano raste, direktno utiče na cenu proizvoda. Prema tome, u epohi rastuće energetske krize, mogućnost smanjenja potrošnje goriva pri zadatoj produktivnosti, odnosno, povećanja produktivnosti pri zadatoj potrošnji goriva, ima sve veći značaj. U ovom pogledu, glavni i primarni zadatak sa kojim se suočavaju projektanti MZR u sledećoj deceniji biće povećanje njihove efikasnosti i konkurentnosti na globalnom tržištu. Osnovne radne karakteristike MZR su radni kapacitet, potrošnja goriva, materijala i energije, pouzdanost, društvena prihvatljivost (bezbednost, ljudski faktor i životna sredina) i, kao posledica, cena proizvedenih dobara u uložnog rada.

U vezi sa tim, smatra se da je poboljšanje efikasnosti MZR, kroz povećanje njihovog radnog kapaciteta i smanjenje potrošnje goriva, hitan i perspektivan pravac.

2. ANALIZA PUBLIKACIJA

Analiza dostignuća industrijske građevinske tehnologije omogućava da se izdvoji određen broj glavnih pravaca razvoja i istraživanja na polju opreme za zemljane radove, kao i metoda za povećanje učinka i efikasnosti. Ovi pravci su značajno predodređeni opštim uslovima razvoja mašinstva.

Prvi pravac pokriva probleme povećane pouzdanosti i konkurentnosti takvih mašina i masovno proizvedenih kompleksa, kao i mašina koje treba predstaviti [1]. Drugi pravac uključuje probleme unapređenja pogona i hidrauličnih sistema upravljanja radnog procesa mašina [5, 6, 7, 8, 9, 11].

Treći pravac karakterišu radovi na automatizaciji i načinima robotizacije MZR. Četvrti pravac je povezan sa

problematikom poboljšanja funkcionalnih karakteristika mašina i usavršavanja njihovih radnih priključaka na bazi dostignuća u tehnologiji i fundamentalnoj nauci, kao i u proizvodnji teških mašina i mašina za građevinske radove u prostorno ograničenom okruženju [1]. Peti pravac karakteriše jedna od najvažnijih tendencija u razvoju građevinarstva i inženjerstvu izgradnje i održavanja puteva, kao grana koja otvara proizvodnju nove opreme i uključuje probleme korišćenja načina i metoda automatizovanog projektovanja i istraživanja mašina u toku njihovog nastajanja i funkcionisanja.

Analiza, sprovedena nad radovima o povećanju efikasnosti opreme za zemljane radove i metoda skladištenja energije, pokazuje da:

– prema trenutnim preporukama za razvoj opreme za zemljane radove, postoji nedostatak sistema za skladištenje energije na bazi hidro-pneumatskih akumulatora. Postojeći pristupi obuhvataju mogućnost instaliranja pomenutih sistema kod mašina za transport i podizanje, gde se akumulira energija tereta koji se spušta [9, 10] ili razmatraju sisteme zamajca [5], koji su nepogodni za upotrebu kod MZR sa hidrauličnim pogonom, jer oni akumuliraju mehaničku umesto hidrauličnu energiju. Da bi se odabrao efikasan skladišni sistem energije, neophodna su dublja istraživanja.

– istraživanja povezanosti između vučnih karakteristika, geometrijskih i masenih parametara MZR i funkcionisanja hidrauličko-pneumatskog akumulacionog sistema nisu naglašena u publikacijama [7]. Ne postoji opšti kriterijum koji opisuje parametre hidrauličko-pneumatskog akumulacionog sistema u režimima vuče i akumuliranja energije, zajedno sa radnim uslovima MZR i njihovim tehničkim specifikacijama;

– ne postoje informacije o uticaju operativnog režima hidrauličko-pneumatskog akumulacionog sistema na određeni priključak mašine.

3. CILJ I ZADACI

Cilj ovog rada je povećanje efikasnosti MZR putem povećanja radnog kapaciteta u transportnom režimu zahva-ljujući uskladištenju energije koja je akumulirana u hidrauličko-pneumatskim akumulatorima na račun smanjenja vremena ciklusa i potrošnje goriva korišćenjem akumulirane energije u režimu kopanja tla.

Za dostizanje zadatog cilja, neophodno je:

- odrediti karakter distribucije snage u okviru radnog ciklusa MZR na osnovu postojećih teorijskih istraživanja i sprovedenih istraživanja na polju sistema za skladištenje hidraulične energije koji su pogonjeni pumpama [2, 3, 4];

– razviti teorijsko opravdanje za korišćenje sistema za skladištenje hidraulične energije koji su pogonjeni pumpama kod MZR.

4. TEORIJSKA OPRAVDANOST UPOTREBE PUMPNIH SISTEMA ZA SKLADIŠTENJE HIDRAULIČNE ENERGIJE KOD MZR

Kod upotrebe sistema za skladištenje hidraulične energije u radnom ciklusu buldozera, potreba za snagom motora će se menjati od etape do etape (Sl. 1) i dolaziće do preraspodele toka snage (Sl. 2).

Slika 1: Radni ciklus buldozera uz primenu pumpnog sistema za skladištenje hidraulične energije: E_{ak} – energija utrošena od strane akumulatora u fazi kopanja; $N_{p.ak}$ – snaga utrošena od strane akumulatora u fazi kopanja; E_{akx} – akumulirana energija pri neopterećenom hodu; $N_{3.ak}$ – akumulirana snaga pri neopterećenom hodu; $N_{DBC.kop}$ – snaga motora sa unutrašnjim sagorevanjem (SUS motor) kod iskopa bez primene pumpnog sistema za skladištenje hidraulične energije; $N_{AK.kop}$ – snaga motora sa unutrašnjim sagorevanjem (SUS motor) kod iskopa sa primenom pumpnog sistema za skladištenje hidraulične energije; N_{per} – snaga SUS motora kod premeštanja tla; N_{xx1} – utrošena snaga pri neopterećenom hodu sa primenom pumpnog sistema za skladištenje hidraulične energije; N_{xx} – utrošena snaga pri neopterećenom hodu bez primene pumpnog sistema za skladištenje hidraulične energije; t_{kop} – vreme iskopavanja; t_{per} – vreme hoda; t_{xx} – vreme neopterećenog (praznog) hoda; $t_{ц}$ – vreme trajanja ciklusa.

Slika 2: Shema toka energije u pogonskoj jedinici buldozera tokom radnog ciklusa: a – konvencionalni buldozer; b – buldozer sa pumpnim sistemom za skladištenje energije; M – motor; TP – transmisija buldozera; H – hidraulična pumpa; ГЦ – hidraulični cilindar; PO – radni priključak; AK – akumulator

Snaga motora SUS tokom radnog ciklusa je raspodeljena na snagu transmisije N_{tp} i snagu hidrauličnog sistema N_{rc}

$$N_{\text{ДБЗ}} = N_{rc} + N_{tp} . \quad (1)$$

Snaga predata hidrauličnom sistemu treba da se skladišti u hidrauličnom akumulatoru tokom praznog hoda

$$N_{rc} = N_{ak} . \quad (2)$$

Pri tome, snaga motora SUS, dopuštajući gubitke, će se u potpunosti predati transmisiji buldozera

$$N_{\text{ДБЗ}} = N_{tp} . \quad (3)$$

Stoga, snaga utrošena na kopanje imaće dve komponente

$$N_{\text{коп}} = N_{p.ak} + N_{\text{ДБЗ}_{\text{max}}} . \quad (4)$$

Da bi se postigao koristan učinak pomoću hidrauličko-pneumatskih akumulatora (HPA) pri prodiranju rezne ivice, neophodno je da akumulirana energija E_{ak} odgovara potrošenoj energiji od strane osnovne jedinice hidrauličkog sistema pri ovoj operaciji E_{ru} .

$$E_{ak} = E_{ru} . \quad (5)$$

Vreme pražnjenja HPA je jednako vremenu prodiranja rezne ivice

$$t_3 = t_{p.ak} . \quad (6)$$

Možemo postaviti relaciju

$$N_{ak} t_3 = N_{ii} t_3 . \quad (7)$$

Snaga potrebna hidrocilindru je povezana sa snagom pumpe na sledeći način

$$\frac{N_{ii}}{\eta_{ii}} = \frac{N_{rc}}{\eta_{rc}} , \quad (8)$$

gde je

N_{ii} – snaga hidro-pumpe; N_{ii} – zahtevana snaga za hidro-cilindar; η_{ii} – ukupan stepen korisnosti hidro-pumpe; η_{rc} – ukupan stepen korisnosti hidrauličkog sistema.

Upotrebljena snaga HPA pri operaciji kopanja mora biti jednaka snazi pumpe pri istoj operaciji. Na osnovu toga možemo napisati

$$\frac{N_{ii}}{\eta_{ii}} = \frac{N_{ak}}{\eta_{rc}} = \frac{N_{ii}}{\eta_{rc}} . \quad (9)$$

Jedan od glavnih parametara pumpnih sistema za skladištenje hidrauličke energije je potreban protok, tj. Zapremina radne komore HPA. Teoretski gledano, ona mora biti jednaka zapremini hidro-cilindra pri maksimalno izvučenoj klipnjači

$$V_{ak} = V_{ii} . \quad (10)$$

Međutim, tokom radnog ciklusa buldozera, klipnjača ne zauzima uvek krajnji položaj. Ovaj faktor zavisi od visine reznog sloja h_s . Zapremina HPA bi trebalo da obuhvati gubitke. Polazeći od specifične karakteristike rada HPA, on mora da obezbedi da se zapremina fluida ΔV_0 u potpunosti isprazni.

U fazi akumuliranja energije, neophodno je da se kontroliše ostatak radnog fluida u HPA u datom intervalu vremena.

Posmatrajmo snage N_{ii} i N_{ii}

$$N_{ii} = Q_{ii} p_{ii} \quad (11)$$

gde je Q_{ii} – protok hidraulične pumpe; p_{ii} – izlazni pritisak hidraulične pumpe

$$N_{ii} = R_{ii} \mathcal{G}_{ii} . \quad (12)$$

Uvedimo sada silu na klipnjači kao

$$R_{ii} = A_{ii} p_{ii} . \quad (13)$$

Brzina klipa iznosi

$$\mathcal{G}_{ii} = l_{ii} / t_3 , \quad (14)$$

gde je l_{ii} – pomeraj klipa; t_3 – vreme zarivanja rezne ivice.

Vreme zarivanja rezne ivice je vreme za koje HPA obezbeđuje potrebnu zapreminu fluida i iznosi

$$t_3 = \frac{T - G_6 (f \pm i)}{a_1 k_K B \mathcal{G}_3} , \quad (15)$$

gde je a_1 – koeficijent uticaja tipa zemljišta i konstrukcije mašine na putanju rezanja; $\mathcal{G}_3$ – brzina zarivanja rezne ivice.

Brzina zarivanja rezne ivice iznosi

$$\mathcal{G}_3 = \mathcal{G}_{ii} k_{k3} , \quad (16)$$

gde je k_{k3} – koeficijent koji uzima u obzir pogonsku vezu između klipnjače cilindra i rezne ivice.

Koristeći (3.23) – (3.25) može se napisati

$$\frac{Q_{ii}}{\eta_{rc}} t_3 = \frac{A_{ii} l_{ii}}{\eta_{rc}} . \quad (17)$$

na osnovu

$$\frac{Q_{ii}}{\eta_{rc}} t_3 = V_{ii} , \quad (18)$$

gde je V_{ii} – zapremina fluida koji treba da se dopremi cilindru u etapi zarivanja rezne ivice u tlo.

Zapremina radne komore HPA je izračunata uzimajući u obzir zapreminu fluida koju je potrošio hidro-cilindar. Imajući u vidu da HPA mora da dopremi hidro-cilindru količinu fluida V_{ii} može se napisati

$$V_{ak} = V_{ii} = \frac{A_{ii} l_{ii}}{\eta_{rc}} . \quad (19)$$

Dobijena vrednost se može smatrati minimalnom dopuštenom zapreminom, pa se, zbog pouzdanosti sistema, radna zapremina HPA mora uzeti kao

$$\Delta V = 2 V_{ak} . \quad (20)$$

Zapremina radne komore akumulatora je direktno proporcionalna broju cilindara kontrole pogonjene opreme i glasi

$$\Delta V_{ak} = n_{ii} \Delta V , \quad (21)$$

gde je

n_{ii} – broj cilindara jednakih zapremina.

Količina energije koja je sakupljena u akumulatoru mora biti dovoljna za obezbeđivanje učinka hidrauličnog pogona kontrole gonjene opreme i pražnjenja kašike

$$E_{ak} = E_{3H} + E_{3P} . \quad (22)$$

Snaga koja je utrošena na hidraulični pogon radnog priključka iznosi

$$N_{3.ak} = N_{p.ak1} + N_{p.ak2} . \quad (23)$$

U cilju osiguravanja visoke efikasnosti grejdera i motornih grejdera, neophodno je zahtevati redukovanje vremena radnog ciklusa, tj. zahtevati da se radi pri najvećem

ćim mogućim brzinama kretanja. Pored toga, neophodno je da se postigne povećanje zapremine zemljišta u prizmi koja se pomera pomoću rezne ivice. Međutim, preopterećivanje mašine nije efikasno, jer može izazvati proklizavanje gusenica vučnog traktora grejdera ili pogonskih točkova grejdera. Proklizavanje stajnog trapa ima za posledicu povećano habanje, prekomeran gubitak radnog vremena, stalnu potrebu za upotrebom kvačila i promenom brzine, kao i značajno smanjenje kapaciteta uređaja.

Operativni tok grejdera, motornih grejdera i skrepera tokom izgradnje i oblikovanja neasfaltiranih putevaje

identičan i ima cilični karakter kao kod buldozera. Predloženi pumpni skladišni sistem može biti upotrebljen kod motornih grejdera i skrepera u cilju smanjenja vremena radnog ciklusa po cenu smanjenja opterećenja na motor SUS u fazi zarivanja i podrivanja radnim priključkom, usmeravajući tako svu snagu ka pogonskom mehanizmu mašine, odnosno, ostvarujući rad pri velikim brzinama. Tako, ulazna snaga po etapama može biti promenjena (Sl. 3, 5) i tokovi snage preraspodeljeni (Sl. 4, 5).

Slika 3: Radni ciklus skrepera sa pumpnim hidro-skladišnim sistemom: N_H - snaga utrošena pri punjenju sanduka skrepera bez primene pumpnog hidro-skladišnog sistema; N_p i N_{p1} - snage utrošene od strane motora SUS pri pražnjenju skrepera sa i bez primene pumpnog hidro-skladišnog sistema respektivno; $N_{p.ak1}$ i $N_{p.ak2}$ - snage utrošene od strane akumulatora pri hidrauličnom pogonu radnog priključka pri punjenju i pražnjenju respektivno; E_{3H} i E_{3p} - energije utrošene pri hidrauličnom pogonu radnog uređaja pri punjenju i pražnjenju respektivno; t_H - vreme punjenja sanduka; t_p - vreme pražnjenja sanduka.

Slika 4: Shema toka energije u pogonskoj jedinici skreperatokom radnog ciklusa: a - skreper konvencionalne konstrukcije; b - skreper sa pumpnim hidro-skladišnim sistemom; M - motor; TP - transmisija skrepera; H - hidraulična pumpa; ΓЦ - hidraulični cilindar; PO - radni priključak; AK - akumulator

Slika 5: Radni ciklus motonog grejdera sa pumpnim hidro-skladišnim sistemom: N_{cn} , N_{cn1} – snaga utrošena od strane motora SUS pri rezanju i premeštanju tla pomoću radnog priključka sa i bez pumpnog hidro-skladišnog sistema respektivno; t_{cn} – vreme rezanja i premeštanja tla

Slika 6: Shema toka energije u pogonskoj jedinici buldozera tokom radnog toka: a – motorni grejder konvencionalne konstrukcije; b – motorni grejder sa pumpnim hidro-skladišnim sistemom; M – motor; TP – transmisija motornog grejdera; H – hidraulična pumpa; ГЦ – hidro-cilindar; PO – radni priključak; AK – akumulator

Moguće je, takođe, da će energija uskladištena u hidrauličkom akumulatoru dopuniti energiju motora SUS, što će omogućiti povećanje radnog kapaciteta mašine. U slučaju proklizavanja mašine, glavica energije može biti prenešena na transmisiju a snaga akumulirana u HPA prenešena hidrauličnom sistemu, gde se smanjuje snaga od strane motora SUS ka hidrauličnom sistemu a povećava snaga predata transmisiji buldozera. Prema podacima iz rezultata istraživanja, ulazna snaga hidrauličnog sistema, u zavisnosti od tipa pogona transmisije, čini do 50% ukupne snage motora SUS.

5. ZAKLJUČCI

1. Prvi put su prezentovani shematski dijagrami energetskih tokova za MZR sa cikličnim operacijama – buldozere, skrepere, motorne grejdere opremljen sa pumpnim sistemima za skladištenje hidraulične energije, čiji

princip rada leži u prekidanju toka energije od pogonske jedinice do radnog uređaja u režimu kopanja i korišćenju energije hidrauličnih akumulatora.

2. Teorijska istraživanja su pokazala da primena pumpnih sistema za skladištenje hidraulične energije baziranih na hidrauličko-pneumatskim akumulatorima omogućava da se smanji izlazna snaga pogonske jedinice za 4-6% kod buldozera, 5-6% kod skrepera i 4-7% kod motornih grejdера.

3. Izbor parametara hidrauličko-pneumatske akumulatorske jedinice je uslovljen takvim konstrukcijskim i tehničkim karakteristikama mašine kao što su kategorija vučne rude, broj i radna zapremina hidrauličnih cilindara, nominalni i maksimalni radni pritisak u hidrauličnoj instalaciji.

4. Određena je jednačina koja omogućava određivanje karakteristika pumpnog sistema za skladištenje hidraulične energije za date parametre MZR: zapremina

hidraulično-pneumatskog akumulatora je izračunata pomoću izraza $\Delta V_{ak} = n_u \Delta V$, pritisak pred-punjenja prema

izrazu $p_{naz} = \left(\frac{n}{zn-1} \right)^{\frac{n}{n-1}} p_{max}$ i vreme punjenja prema

izrazu $t_3 = t_{xx} = \frac{\Delta V_{ak}}{Q_h}$.

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Increasing the Efficiency of the Operation of the Batch-type Earthmoving Machines at the Expense of Using the Pneumatic Storage System

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This paper presents the cyclograms of power distribution and the schemes of distribution of power flows for the work cycle of bulldozers, scrapers and graders equipped by pumped hydro-storage systems.

Key words: Accumulator, Energy, Hydraulic drive, Power, Power flow

1. INTRODUCTION

At the present time not mechanized, heavy and labour - intensive work can hardly be found in the road building, municipal or agricultural sector. In all the mentioned industries earthmoving machines (EMM) are used. One of the main issues at large output volumes is the cost of the manufactured products. The cost of fuel, which is nowadays continuously increasing, has a direct impact on the cost of products. Therefore, in the epoch of progressing energy crisis the possibility to reduce fuel consumption at set productivity of a vehicle or increase its productivity at set fuel consumption is of growing interest. In this regard the major and primary task facing the creators of earthmoving machines in the next decade will be increasing of their efficiency and competitiveness on the global market. The basic performance characteristics of earthmoving machines are working capacity, fuel, materials and energy consumption, reliability, social adjustment (safety, human factor and environmental performance) and as a consequence, the cost of the manufactured products and the performed work.

In this connection, the urgent and perspective direction is considered to be advancing EMMs efficiency due to increase of their working capacity and decrease of fuel consumption that brings to reduction of the cost of manufactured products.

2. ANALYSIS OF PUBLICATIONS

The analysis of achievements of industrial construction technology allows to allocate a number of the main directions of development and researches in the field of earthmoving equipment, methods to intensify its operation and increase its efficiency. These directions are substantially predetermined by general provisions of mechanical engineering development.

The first direction covers problems of increasing reliability and competitiveness of such machines and mass-produced complexes as well as machines to be introduced [1]. The second direction includes problems of improvement of the drive and hydraulic systems of the ma-

chines work process control [5, 6, 7, 8, 9, 11]. The third direction is characterized by works on automation and ways of earthmoving machines robotization. The fourth direction is connected with problems of improvement of the machines functional properties and perfection of their working attachment on the basis of achievements in technology and fundamental sciences as well as creation of heavy machines and machines for construction works in space-limited environment [1]. The fifth direction characterizes one of the most important tendencies in development of construction and road mechanical engineering as a branch providing production of new equipment and includes problems of using the means and methods of automated design and research of the machines in the course of their creation and operation.

The carried-out analysis of works on increasing the efficiency of earthmoving equipment and methods of energy storage shows that:

- at present recommendations on development of earthmoving equipment with the energy storage system on the basis of hydro-pneumatic accumulators are lacking. The existing methods either consider possibility of installation of such systems on carrying and lifting machines to accumulate the energy of the load moving down [9, 10] or consider flywheel systems [5], which are rather inconvenient to be used on hydraulically operated EMMs for they accumulate the mechanical instead of hydraulic energy. To choose the effective storage system further researches are required;

- studies of interrelations between traction characteristics, geometrical and weight parameters of EMMs and operation of the hydro-pneumatic accumulating system are not highlighted in the publications [7]. There are no common criteria describing parameters of the hydro-pneumatic accumulating system in the traction and energy accumulation mode with EMMs working conditions and their technical specifications;

- there are no information on influence of the hydro-pneumatic accumulators operation mode on that one of the machine working attachment.

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3. PURPOSE AND TASKS

The purpose of the work is increasing EMMs efficiency due to storage of the energy accumulated in hydro-pneumatic accumulators at transportation modes to improve their working capacity at the expense of reducing the cycle time and fuel consumption through using the accumulated energy in the digging mode.

For achievement of the set purpose it is required to solve the following tasks:

– to determine the character of power distribution in the work cycle for EMMs of cyclic operation on the

basis of the existing theoretical research and the conducted field research of pumped hydro-storage systems [2, 3, 4];

– to develop theoretical justification for using of the pumped hydro-storage systems on EMMs.

4. THEORETICAL JUSTIFICATION OF USING PUMPED HYDROSTORAGE SYSTEMS ON EMM

At using the hydro-storage systems in the bulldozer work cycle, the stage-by-stage engine power demand will change (Fig. 1) and redistribution of the power flows will take place (Fig. 2).

Figure 1: The work cycle of a bulldozer with application of a pumped hydro-storage system: E_{ak} – energy spent by the accumulator at the stage of digging; $N_{p.ak}$ – power spent by the accumulator at the stage of digging; E_{akx} – energy accumulated at idle running; $N_{3.ak}$ – power accumulated at idle running; $N_{DBC.kop}$ – ICE power at digging operation without application of the pumped hydro-storage system; $N_{AK.kop}$ – ICE power at digging operation with application of the pumped hydro-storage system; N_{nep} – ICE power at earth handling; N_{xx1} – power spent at idle running with application of the pumped hydro-storage system; N_{xx} – power spent at idle running without application of the pumped hydro-storage system; t_{kop} – time of excavation; t_{nep} – travel time; t_{xx} – time of idle running; t_u – cycle time.

Figure 2: The scheme of energy flows in a bulldozer power unit during the work cycle: a – bulldozer of a conventional design; b – bulldozer with a pumped hydro-storage system; M – engine; TP – bulldozer power train; H – hydraulic pump; ГЦ – hydraulic cylinder; PO – working implement; AK – accumulator

The ICE power in the work cycle is distributed to powers spent on the power trains N_{tp} and the hydraulic system N_{rc}

$$N_{\text{ДБЗ}} = N_{rc} + N_{tp}. \quad (1)$$

The power spent on the hydraulic system should be stored in the hydraulic accumulator at the idle running

$$N_{rc} = N_{ak}. \quad (2)$$

At that the ICE power, making allowance for losses, will be completely transmitted to the bulldozer power train

$$N_{\text{ДБЗ}} = N_{tp}. \quad (3)$$

Thus, the power spent on digging will have 2 components

$$N_{\text{коп}} = N_{p.ak} + N_{\text{ДБЗ}_{\text{max}}}. \quad (4)$$

To perform the useful yield on the blade penetration with the help of hydro-pneumatic accumulators (HPA), it is necessary that the accumulated energy E_{ak} corresponded to the energy E_{ru} spent on this operation by the basic hydraulic system unit.

$$E_{ak} = E_{ru}. \quad (5)$$

The HPA discharge time is equal to the time of the blade penetration

$$t_3 = t_{p.ak}. \quad (6)$$

We set down

$$N_{ak} t_3 = N_u t_3. \quad (7)$$

The power required for the hydraulic cylinder is related to the pump power as follows

$$\frac{N_u}{\eta_u} = \frac{N_{rc}}{\eta_{rc}}, \quad (8)$$

where N_u – hydraulic pump power; N_u – power required for the hydraulic cylinder; η_u – total efficiency of the hydraulic pump; η_{rc} – total efficiency of the hydraulic system.

The HPA power spent on digging operations must equal to the pump power at the same operation. On this basis we can set down

$$\frac{N_u}{\eta_u} = \frac{N_{ak}}{\eta_{rc}} = \frac{N_u}{\eta_{rc}}. \quad (9)$$

One of the main parameters of the pumped hydro-storage system is a required liquid supply, i.e. the volume of the HPA working chamber. Theoretically it must be equal to the hydraulic cylinder volume upon condition of the maximally extended cylinder rod

$$V_{ak} = V_u. \quad (10)$$

But during the bulldozer work cycle the cylinder rod does not always move to the maximal possible position. This factor depends on the cut layer height h_s . The HPA volume should take account of losses. And proceeding from the HPA operation specific feature, it must keep the fluid volume ΔV_0 being completely discharged.

At the stage of the energy accumulation it is necessary to control the fluid residue in the HPA at a given time interval.

Let us consider the powers N_u and N_{rc}

$$N_u = Q_u p_u, \quad (11)$$

where Q_u – hydraulic pump consumption; p_u – hydraulic pump output pressure

$$N_u = R_u Q_u. \quad (12)$$

Let us present the force on the cylinder rod as

$$R_u = A_u p_u. \quad (13)$$

The speed of the piston movement makes

$$Q_u = l_u / t_3, \quad (14)$$

where l_u – piston movement; t_3 – time of the blade deepening.

The blade deepening time is the time of the HPA providing the required fuel volume equal to

$$t_3 = \frac{T - G_0 (f \pm i)}{a_1 k_K B Q_3}, \quad (15)$$

where a_1 – coefficient of influence of the soil type and the machine design on the deepening trajectory; Q_3 – blade deepening speed.

The blade deepening speed is equal to

$$Q_3 = Q_u k_{k3}, \quad (16)$$

where k_{k3} – coefficient taking into account drive connection between the cylinder rod and the blade.

Using (3.23) – (3.25) we can set down

$$\frac{Q_u}{\eta_{rc}} t_3 = \frac{A_u l_u}{\eta_{rc}}. \quad (17)$$

on the basis of

$$\frac{Q_u}{\eta_{rc}} t_3 = V_u, \quad (18)$$

where V_u – fluid volume to be supplied to the hydraulic cylinder at the deepening stage.

The HPA working chamber volume is calculated considering the volume of fluid consumed by the hydraulic cylinder. Taking into account that the HPA is to supply to the hydraulic cylinder the fluid supply V_u we can set down

$$V_{ak} = V_u = \frac{A_u l_u}{\eta_{rc}}. \quad (19)$$

The obtained value V_{ak} can be considered the minimum permissible volume, thus, for the system reliability the HPA working volume must be equal to

$$\Delta V = 2 V_{ak}. \quad (20)$$

The volume of the accumulator working chamber is directly proportional to the number of cylinders of the driven equipment control

$$\Delta V_{ak} = n_u \Delta V, \quad (21)$$

where n_u – number of cylinders with the same volume.

The amount of the energy accumulated in the battery must be sufficient for ensuring performance of the hydraulic drive of the driven equipment control and the bucket dump

$$E_{ak} = E_{3H} + E_{3P}. \quad (22)$$

The power spent on the work implement hydraulic drive is

$$N_{3.ak} = N_{p.ak1} + N_{p.ak2}. \quad (23)$$

For ensuring high efficiency of graders and motor graders it is necessary to seek for reduction of the work cycle time, i.e. to work at the highest possible speeds of motion.

Also it is necessary to achieve increase in volume of soil in the prism moved by the blade. However, it is not efficient to overload the machine, for it can cause skidding of caterpillars of the tractor towing the grader or of the grader driving wheels. The undercarriage skidding results in wear increase, excessive loss of working time, constant need to throw the clutch and change gears, as well as considerable reduction of the equipment working capacity.

The operation sequence of graders, motor graders and scrapers during construction and shaping of unpaved roads is identical and has a cyclic character as that of bulldozers. The proposed pumped storage system can be used for motor graders, scrapers to reduce the work cycle time at the expense of decreasing the load on the ICE in the modes of the working attachment deepening and shanking out thereby directing all the power to the machine power train, i.e. ensures working at high speeds. Thus, stage-by-stage power input of the machine can be changed (Fig. 3, 5) and energy flows redistributed (Fig. 4, 5)

Figure 3: The work cycle of a scraper with a pumped hydro-storage system: N_H – power consumed at filling of the scraper bowl without application of the pumped hydro-storage system; N_p and N_{p1} – powers spent by the ICD on the scraper dump with and without application of a pumped hydro-storage system correspondingly; $N_{p.ak1}$ and $N_{p.ak2}$ – powers spent by the accumulator on the hydraulic drive of the working attachment at its filling and dumping correspondingly; E_{3H} u E_{3p} – energies spent on the hydraulic drive of the working attachment at its filling and dumping correspondingly; t_H – time of the bowl filling; t_p – time of the bowl dumping

Figure 4: The scheme of energy flows in a scraper power unit during the work cycle: a – scraper of a conventional design; b – scraper with a pumped hydro-storage system; M – engine; TP – scraper power train; H – hydraulic pump; ГЦ – hydraulic cylinder; PO – working implement; AK – accumulator

Figure 5: The work cycle of a motor grader with a pumped hydro-storage system: N_{cn} , N_{cnl} – power spent by the ICE at cutting and handling of soil by the working attachment with and without application of a pumped hydro-storage system correspondingly; t_{cn} – time of soil cutting and handling

Figure 6: The scheme of energy flows in a bulldozer power unit during the work cycle: a – motor grader of a conventional design; b – motor grader with a pumped hydro-storage system; M – engine; TP – motor grader power train; H – hydraulic pump; ГЦ – hydraulic cylinder; PO – working implement; AK – accumulator

hydraulic accumulator will supplement the ICE energy, which will allow for increasing the machine working capacity. In the case of the machine skidding most of the energy can be distributed to the power train and the power accumulated in the HPA transmitted to the hydraulic system, thereby decreasing the power spent by the ICE on hydraulic system and increasing the power transmitted to the bulldozer power train. According to the research data the power input on the hydraulic system depending on the type of the power train drive makes up to 50% of the ICE power.

5. CONCLUSIONS

1. For the first time there have been presented a schematic diagram of energy flows for EMMs of cyclic

It is also possible that the energy stored in the operation – bulldozers, scrapers, motor graders equipped with a pumped hydro-storage system, which operation principle is in breaking the energy flow from the power unit to the working attachment in the digging mode and using the energy of hydraulic accumulators.

2. The theoretical studies demonstrated that application of pumped hydro-storage systems based on hydro-pneumatic accumulators allows to decrease a set output of the power unit on average for bulldozers by 4-6 %, scrapers by 5-6 % and motor graders by 4-7 %.

3. The choice of the hydro-pneumatic accumulator unit parameters is influenced by such construction and technical characteristics of the machine as the drawbar category, the number and volume of the hydraulic cylinder

ders, nominal and maximal pressure in the hydraulic system.

4. There have been obtained formula allowing to determine characteristics of the pumped hydro-storage system for the given parameters of EMMs: the hydro-pneumatic accumulator volume is calculated by the expression $\Delta V_{ak} = n_{ii} \Delta V$; the pre-charge pressure

$$p_{nai} = \left(\frac{n}{zn-1} \right)^{\frac{n}{n-1}} p_{max}; \text{ the charging time } t_z = t_{xx} = \frac{\Delta V_{ak}}{Q_{ii}}.$$

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