Analiza opterećenja guseničnih kretnih mehanizama hidrauličkih bagera

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Rad sadrži rezultate eksperimentalnih istraživanja opterećenja guseničnih kretnih mehanizama hidrauličkih bagera. Razvijen je dinamički matematički model za određivanje opterećenja podloge oslanjanja guseničnih kretnih mehanizama hidrauličkih bagera na osnovu merenih veličina stanja rada bagera u eksploatacionim uslovima. Kao merene veličine stanja rada bagera uzimaju se: odizanje oslonno-kretnog mehanizma, ugao okretanja i pritisci u vodovima hidromotora pogona okretanja platforme i hodovi i pritisci u vodovima hidrocilindara strele, ruke i kašike pogonskih mehanizama dubinskog manipulatora bagera. Definisani su matematički modeli određivanja sila u gusenici i sila u zahvatu lančanika i lanca gusenice. Kao primer, dati su rezultati analize opterećenja podloge oslanjanja hidrauličkog bagera guseničara mase 16000 kg opremljenog sa manipulatorom dubinske kašike zapremine 0,6 m³ i opterećenja lanca gusenica hidrauličkog bagera mase 24000 kg opremljenog sa manipulatorom dubinske kašike zapremine 1 m³.

Ključne reči: Hidraulički bageri, gusenični kretni mehanizmi, opterećenja

0. UVOD

Oslono-kretni mehanizmi hidrauličkih bagera se izvode sa jednim parom gusenica koje imaju sledeće standardne module i elemente: gusenični lanac 1 (sl 1), papuče 2, pogonski lančanik 3, zatezni uređaj 4, noseće 5 i vodeće 6 valjake objedinjene podužnim nosačem gusenice 7.

Zavisno od uslova rada, isti model bagera se može opremiti sa: standardnim (ST), produženim (LC) i ojačanim (HD) guseničnim kretnim mehanizmom. Navedene varijante kretnih mehanizama se međusobom razlikuju prema širini papuča, dužini naleganja L ili prema rasponu gusenica.

Gusenični oslonno-kretni mehanizmi bagera se odlikuju dobrim prijanjanjem na podlogu i mogućnošću postizanja velikih vučnih sila, malim brzinama kretanja, malim površinskim pritiskom na podlogu, velikom prohodnošću i stabilnim kretanjem.

U ovom radu su dati matematički modeli za određivanje opterećenja podloge oslanjanja i sila u lancu gusenice guseničnih kretnih mehanizama hidrauličkih bagera na osnovu merenih veličina stanja rada i kretanja bagera u eksploatacionim uslovima.

1. MATEMATIČKI MODEL BAGERA

Za određivanje opterećenja podloge oslanjanja bagera sa guseničnim kretnim mehanizmom razvijen je matematički model za opštu konfiguraciju kinematičkog lanca bagera sastavljenog od: oslonno-kretnog člana L_1 (sl.2a), obrtnog člana (platforme) L_2 i tročlanog ravanskog manipulatora sa: strelom L_3 , rukom L_4 i kašikom L_5 . Oslono-kretni član bagera je gusenični kretni mehanizam širine papuča b , dužine naleganja L i raspona gusenica B .

Matematičkim modelom član kinematičkog lanca bagera L_i je definisan, u svom lokalnom koordinatnom sistemu $O_i x_i y_i z_i$, gometrijskim, kinematičkim i dinamičkim parametara obuhvaćenim skupom veličina (sl.2) [1][2]:

$$L_i = \{e_i, s_i, t_i, m_i, J_i\} \quad (1)$$

gde je: e_i - jedinični vektor ose zglobova O_i kojim se član L_i vezuje za prethodni član L_{i-1} , s_i - vektor položaja središta zglobova O_{i+1} kojim se član lanca L_i vezuje za naredni član L_{i+1} (intenzitet vektora s_i predstavlja kinematičku dužinu člana), t_i - vektor položaja središta mase člana L_i , m_i - masa člana, J_i - moment inercije člana.

Dinamičke veličine koje deluju u središtu mase člana L_i su inercijalna sila F_i , koja se određuje pomoću drugog Newton-ovog zakona:

$$F_i = -m_i w_i \quad (2)$$

i moment inercijalnih sila M_i , koji se određuje na osnovu Euler-ovih dinamičkih jednačina:

$$\hat{M}_i = -\hat{J}_i \hat{e}_i + (\hat{\omega}_i \times \hat{J}_i \hat{\omega}_i) \quad (3)$$

gde je: w_i - linearno ubrzanje člana L_i , $\hat{J}_i$ - tenzor momenta inercije člana za koordinatni sistem vezan za središte mase

Sl. 1 Gusenični oslonno-kretni mehanizam hidrauličkih bagera

člana sa osama paralelnim lokalnim koordinatnim osama člana, $\hat{\omega}_i = A_{oi}\omega_i$, $\hat{\varepsilon}_i = A_{oi}\varepsilon_i$ - ugaona brzina i ugaono ubrzanje člana L_i u lokalnom koordinatnom sistemu $O_i x_i y_i z_i$, A_{oi} - transformacione matrice vektora iz apsolutnog koordinatni sistem bagera u lokalni koordinatni sistem člana.

Za određivanje kinematičkih veličina, ugaone ω_i brzine i ubrzanja ε_i i linernog ubrzanja w_i za središte mase člana lanca L_i , u odnosu na apsolutni koordinatni sistem $OXYZ$, koriste se rekursivne jednačine:

$$\omega_i = \omega_{i-1} + \dot{\theta}_i e_i \quad (4)$$

$$\varepsilon_i = \varepsilon_{i-1} + \ddot{\theta}_i e_i + (\omega_{i-1} \times \dot{\theta}_i e_i) \quad (5)$$

$$w_i = w_{i-1} + (\varepsilon_{i-1} \times (s_{i-1} - t_{i-1})) + \omega_{i-1} \times (\omega_{i-1} \times (s_{i-1} - t_{i-1})) + (\varepsilon_i t_i) + \omega_i \times (\omega_i \times t_i) \quad (6)$$

gde je: $\dot{\theta}_i, \ddot{\theta}_i$ - ugaona brzina i ugaono ubrzanje člana L_i u zglobu O_i .

Ukupna sila vezana za središte mase člana L_i kinematičkog lanca bagera, uzimajući i uticaj gravitacije, jednaka je:

$$F_{ui} = F_i + m_i g \quad \forall i = 1, \dots, 5 \quad (7)$$

a moment inercijalnih sila člana L_i u apsolutnom koordinatnom sistemu je određen izrazom:

$$M_{ui} = A_{io} \hat{M}_i \quad \forall i = 1, \dots, 5 \quad (8)$$

Redukcijom svih opterećenja bagera u središte O_1 oslone površine kretnog mehanizma, dobija se rezultujuća sila F_I i moment M_I koji opterećuju podlogu oslanjanja kretnog mehanizma bagera (sl.2a):

$$F_I = W + \sum_{i=1}^5 F_{ui} \quad (9)$$

$$M_I = ((r_w - r_1) \times W) + \sum_{i=1}^5 ((r_w - r_1) \times F_{ui}) + \sum_{i=1}^5 M_{ui} \quad (10)$$

gde je: W - vektor otpora kopanja (eksperimentalno određen na osnovu merenih veličina) [3], F_{ui} - ukupna sila u središtu mase člana L_i , uzimajući uticaj inercije i

Sl.2 Matematički model hidrauličkog bagera za eksperimentalno ispitivanje opterećenja podloge oslanjanja kretnog mehanizma bagera

gravitacije, M_{ui} - moment inercijalnih sila člana L_i u apsolutnom koordinatnom sistemu.

Inteziteti komponenta sile F_I i momenta M_I opterećenja podloge oslonu-kretnog mehanizma bagera su (sl.2b):

$$F_{Ix} = (F_I \cdot i), \quad F_{Iy} = (F_I \cdot j), \quad F_{Iz} = (F_I \cdot k) \quad (14)$$

$$M_{Ix} = (M_I \cdot i), \quad M_{Iy} = (M_I \cdot j), \quad M_{Iz} = (M_I \cdot k) \quad (15)$$

Za analizu opterećenja podloge kretnog mehanizma bagera, prema prethodno definisanom matematičkom modelu, potrebne su sledeće merene veličine stanja rada bagera u eksploatacionim uslovima: odizanje oslonu-kretnog mehanizma c_1 (sl.2), ugao okretanja platforme c_2 i pritisci p_{21} , p_{22} u vodovima hidromotora pogona okretanja platforme, hodovi c_i , i pritisci p_{i1} , p_{i2} ($i=3,4,5$) u radnim vodovima hidrocilindara pogonskih mehanizama: strele, ruke i kašike pogonskih mehanizama manipulatora [2].

Zavisno od izmerenih veličina c_i , korišćenjem prenosnih funkcija pogonskih mehanizama, određuju se generalisane koordinate θ_i položaja članova kinematičkog lanca, na osnovu kojih se, diferenciranjem, definišu ugaone brzine i ugaona ubrzanja u zglobovima članova kinematičkog lanca bagera:

$$\dot{\theta}_i = \frac{\theta_{i(t+\Delta t)} - \theta_{i(t-\Delta t)}}{2\Delta t} \quad (16)$$

$$\ddot{\theta}_i = \frac{\theta_{i(t+2\Delta t)} - 2\theta_{i(t)} + \theta_{i(t-2\Delta t)}}{4\Delta t^2} \quad \forall i = 2,3,4,5 \quad (17)$$

gde je: $\theta_{i(t)}$ - generalisane koordinate u trenutku t trajanja operacije kopa-nja, $\theta_{i(t+\Delta t)}$, $\theta_{i(t-\Delta t)}$, $\theta_{i(t+2\Delta t)}$, $\theta_{i(t-2\Delta t)}$ - generalisane koordinate u trenutku vremena koje je za jedan ili dva intervala vremena Δt veće ili manje od vremena t , Δt - interval vremena između dva uzastopna merenja veličina.

2.1. Određivanje opterećenja podloge oslanjanja

Korišćenjem razvijenog programa, na osnovu definisanog matematičkog modela, izvršena je analiza opterećenja podloge oslanjanja gusenice kretnog mehanizma hidrauličkih bagera mase 16000 kg sa dubinskim manipulatorom zapremine kašike $0,6 \text{ m}^3$ na osnovu merenih veličina stanja rada bagera u eksploatacionim uslovima [2].

Kao primer, dati su rezultati analize za manipulacioni zadatak sa operacijama: kopanje ($0-6 \text{ s}$), (sl.3a,b), prenos ($6-10 \text{ s}$ - uz okretanje platforme za 40°), istovar ($10-14 \text{ s}$ - na visini istovara oko $5,5 \text{ m}$) i vraćanje ($14-18 \text{ s}$ - u početnu ravan kopanja).

Sl.3 Opterećenja oslone površine: a) komponente statičkih F_{Is} i dinamičkih F_{Id} sila b) komponente statičkih M_{Is} i dinamičkih M_{Id} momenata

Rezultati analize pokazuju (sl.3a,b) da komponente kojima gusenični kretni mehanizam opterećuje oslone površine imaju izražene promene i pogledu inteziteta i smera delovanja. Najveće vrednosti sila i momenata opterećenja oslone površine javljaju se pri operaciji kopanja. Najveća po intezitetu je vertikalna komponenta sile F_{ly} koja tokom trajanja manipulacionog zadatka pritiska oslonu površinu.

Izrazito veliki intezitet i promenu smera, u odnosu na ostale, ima komponenta momenta M_{Iz} oko Oz ose - takođe tokom operacije kopanja. Promena smera momenta M_{Iz} nastaje pri operaciji kopanja usled promene pravca delovanja otpora kopanja.

Kod opterećenja oslone površine bagera razlika statičkih i dinamičkih opterećenja je izražena pri operacijama prenosa, istovara i ponovnog vraćanja u ravan kopanja kada kinematički lanac bagera ima otvorenu konfiguraciju a vremena trajanja operacija su relativno mala. Pri operaciji kopanja dinamička opterećenja su znatno manja zbog zatvorene konfiguracije kinematičkog lanca bagera (kašika je u dodiru sa zemljištem) a odvijanje procesa kopanja je relativno sporo.

2. ANALIZA SILA U GUSENICI

Pri mirovanju kretnog mehanizma gusenični lanac ima početnu silu zatezanja F_z (sl.4a). Pri kretanju na pogonski točak gusenice se dovodi obrtni moment M_o usled čega se opterećenje pogonskog dela gusenice uvećava za obimnu silu F_o tako da je, za male uglove prohodnosti α, β postoji jednakost [1][4]:

$$F_{kl} = F_{k2} + F_o \quad (18)$$

gde je: F_{kl} - sila u ogranku gusenice koji nailazi na pogonski lančanik, F_{k2} - sila u ogranku gusenice koji silazi sa pogonskog lančanika.

Iz poslednje jednačine sledi da je:

Sl.4 Dinamika guseničnog kretnog mehanizma: a) sile koje deluju na elemente guseničnog kretnog mehanizma [4], b) sile u zahvatu lančanika i gusenice [5]

$$F_o = \frac{M_o}{r_p} = F_{k1} - F_{k2} \quad (19)$$

gde je: M_o - obrtni moment koji deluje na pogonskom lančaniku guseničnog kretnog mehanizma, r_p - poluprečnik pogonskog točka.

Za zatezni točak nailazeći ogranak gusenice je ustvari silazeći ogranak sa pogonskog lančanika, tako da je:

$$F_{k2} = F_{z2} \quad (20)$$

Razlika sila u gusenici na zateznom točku daje momenat koji je jednak sumi momenata M_{zi} koji nastaje usled inercije okretanja zateznog točka i momenata M_{zt} usled sila trenja u ležištima zateznog točka:

$$(F_{z1} - F_{z2})r_z = M_{zi} + M_{zt} \quad (21)$$

odakle sledi:

$$F_{z2} = F_{z1} - \frac{M_{zi} + M_{zt}}{r_z} \quad (22)$$

gde je: F_{z1} - sila u ogranku gusenice koji silazi sa zateznog točka, r_z - poluprečnik zateznog točka.

Rezultujuća sila koja deluje na vučni deo gusenice, jednaka razlici sila $F_{kl} - F_{z1}$ i ima smer sile F_{kl} . Ova razlika sila predstavlja rezultujuću silu na vučnom delu gusenice kojoj se suprotstavlja tangencijalna reakcija (sila prijanjanja) podloge oslanjanja F_μ , sa smerom delovanja u pravcu kretanja guseničnog kretnog mehanizma tako da je:

$$F_\mu = F_{kl} - F_{z1} \quad (23)$$

Opterećenja guseničnog lanca se preko pogonskog lančanika i zateznog točka i nosećih valjaka prenosi na kostur mašine. Zbir svih horizontalnih komponenta sila u osama nosećih valjaka (F_{npx}, F_{nrx}) (sl. 4a), pogonskog lančanika (F_{klx}) i zateznog (F_{zlx}) točka po veličini je jednaka reakciji podloge F_μ .

Tangencijalna reakcija podloge u vidu potisnih sila prenosi se preko ležišta nosećih valjaka i pogonskog lančanika na kostur guseničnog kretnog mehanizma ostvarujući silu vuče mašine.

Reakcija podloge je ustvari sila prijanjanja gusenica na podlogu i može se izraziti jednačinom:

$$F_{\mu} = \mu_c mg + 2bh_r \tau \cdot z_r \quad (24)$$

gde je: μ_c - koeficijent trenja gusenica o podlogu oslanjanja, m - masa mašine, b - širina papuče gusenice, h_r - visina rebra papuče gusenice, τ - napon smicanja podloge, z_r - broj rebara gusenice u kontaktu sa podlogom.

Izvođenje guseničnog oslonu kretnog mehanizma može biti sa zadnjim i sa prednjim položajem pogonskih lančanika posmatrano u odnosu na pravac brzine kretanja mašine. U oba slučaja karakter promene sila zatezanja gusenice po konturi lanca nije jednak.

U prvom slučaju gornji slobodni ogranak gusenice ima malu veličinu opterećenja od horizontalnih sila uslovljenih samo sopstvenim ugibom guseničnog lanca. Kada se gusenični lanac približava pogonskom lančaniku, zatezanje guseničnog lanca počinje da raste, dostižući maksimum nailaskom na pogonski lančanik.

U drugom slučaju najveća zatezanja nastaju na gornjem ogranku gusenice, smanjujući se približavanjem nosećem valjku, gde se uravnotežavaju sa tangencijalnom reakcijom podloge. Varijanta guseničnog oslonu kretnog mehanizma sa prednjim pogonskim lančanicima zahteva veće zatezanje gusenica i ima veće opterećenje lanca.

Vertikalne kompenetne ($F_{k1y}, F_{z1y}, F_{npy}, F_{nzy}$) (sl.4a) sila koje naprežu ose pogonskih lančanika i zateznih točkova, kao i ose zadnjih i prednjih nosećih valjaka, obrazuju u vertikalnoj ravni sistem sila koji utiče na prerasodelu normalnih reakcija površine oslanjanja mašine.

2.1. Sile u zahvatu lančanika i lanca gusenice

Pri zahvatu pogonskog lančanika i članka lanca gusenice nastaju uzajamna opterećenja uz istovremeno relativno kretanje zuba lančanika i čaure segmenta lanca i relativno kretanje čaure i osovinice segmenta lanca. Kao posledica opterećenja i relativnih kretanja, nastaje habanje dodirnih elemenata, posebno čaure i osovinice pri čemu se, zbog povećanja zazora u spoju, povećava korak segmenta lanca na taj način i ukupna dužina lanca.

Da bi se održalo pravilno kretanje gusenice i pri neželjenom povećanju dužine lanca kretni mehanizmi imaju zatezni uređaj koji omogućuje potrebno pomeranje zateznog točka i na taj način održava neophodnu silu zatezanja lanca.

Za analizu sila koje nastaju u zahvatu pogonskog lančanika i guseničnog lanca, pri okretanju lančanika u oba smera, razvijen je model sa sledećim pretpostavkama: na lančanik i lanac deluju samo sile u podužnoj ravni gesenice xy (sl.4b), ugaona brzina pogonskog lančanika je konstantna, zanemaruju se inercijalne sile i sopstvena težina lanca, smer sile u lancu F_k (sl.4b) se ne menja pri ulasku u zahvat, središta zglobova lanca leže na podeonom prečniku lančanika [5].

Zbog malog vremena udara zanemarene su inercijalne sile i primenjen statički matematički model kojim su obuhvaćene sledeće sile sprežanja: sila u lancu F_k (sl.4b), sila normalna na bok zuba lančanika F_z , sila u zglobo segmenta lanca koji se spreže sa lančanikom F_{k+1} , sila trenja između zuba lančanika i čaure segmenta lanca

$\mu_z F_z$, sila trenja između čaure i osovinice segmenta lanca $\mu_b F_{k1}$. Koeficijent trenja između zuba lančanika i čaure segmenta lanca je μ_z , dok je koeficijent trenja između osovinice i čaure segmenta lanca gusenice μ_b .

Za članak I (sl.4b) lanca gusenice, koji se spreže sa lančanikom pri kretanju mašine napred, mogu se napisati statički uslovi ravnoteže u obliku [5]:

$$\begin{vmatrix} \cos \rho & \sin \rho & 0 \\ -\sin \rho & \cos \rho & 0 \\ 0 & -0,5d_2 & F_k \end{vmatrix} \cdot \begin{Bmatrix} F_n \\ F_r \\ h \end{Bmatrix} = \begin{Bmatrix} F_k \\ 0 \\ 0 \end{Bmatrix} \quad (25)$$

gde je: $\rho = \arctg \mu_b$ - ugao trenja između osovinice i čaure segmenta lanca gusenice, d_2 - prečnik osovinice segmenta lanca, h - visina podizanja zglobova lanca, F_n , F_r - normalna i tangentna komponenta sile F_{k+1} .

Za članak II (sl.4b) lanca gusenice, koji se spreže sa lančanikom pri kretanju mašine napred, mogu se napisati statički uslovi ravnoteže u obliku:

$$\begin{vmatrix} 0 & \sin(\tau + \gamma_w) & \cos(\tau + \gamma_w) \\ 1 & \cos(\tau + \gamma_w) & -\sin(\tau + \gamma_w) \\ 0 & 0 & t - 0,5d_2 \end{vmatrix} \cdot \begin{Bmatrix} F_z \\ F'_{k2} \\ F''_{k2} \end{Bmatrix} = \begin{Bmatrix} F_{k1} \sin(0,5\tau + \gamma_w - \varphi) \\ F_{k1} \cos(0,5\tau + \gamma_w - \varphi) \\ h F_{k1} \end{Bmatrix} \quad (26)$$

gde je: τ - ugao koji odgovara koraku lanca na podeonom prečniku lančanika (sl.4b), γ_w - ugao normale na do-dirnu ravan lančanika i čaure lanaca, φ - trenutni ugao okretanja lančanika, F'_{k2} , F''_{k2} - komponente sile F_{k2} koja deluje u zglobov kojim se vezuju članak II i naredni članak lanca.

Ugao koji odgovara koraku lanca na podeonom prečniku lančanika ima vrednost:

$$\tau = \frac{4\pi}{z} \quad (27)$$

gde je: z - broj zuba lančanika.

Ugao normale na dodirnu ravan lančanika i čaure lanca određen je jednačinom:

$$\gamma_w = \gamma - \left(\frac{\tau}{4} + \delta\right) = \beta - \frac{\tau}{2} - \delta \quad (28)$$

Pri analizi trenutni ugao okretanja lančanika φ se kreće u granicama:

$$-\frac{\tau}{2} < \varphi < \frac{\tau}{2} \quad (29)$$

2.2. Ispitivanje sila u gusenici

Kao primer daju se rezultati određivanja sila u lancima gusenica kretnog mehanizma hidrauličkog bagera mase 24000 kg sa hidrostatičkom transmisijom kretanja. Iste pogone gusenica čine integralni prenosnici sa hidromotorom i planetarnim reduktorom za koje su vezani lančanici. Hidromotore napajaju, u otvorenom hidrostatičkom kolu, hidropumpe sa regulacijom po kriterijumu konstantne hidrauličke snage [6].

Ispitivanje kretanja bagera je obavljeno na ravnoj i suvoj makadmskoj podlozi. Karakter kretanja je odgovarao načinu kretanja pri radu sa čestim prekidima na parvoliniskom i krivoliniskom putu, sa periodima ubrzanog i

Sl.5. Promene sila u lancu leve gusenice F_l i desne gusenice F_d i rezultujuće sile vuče F_r određenih ispitivanjem kretanja hidrauličkog bagera guseničara mase 24000 kg [7].

usporen kretanja, kretanja konstantnom brzinom i naglog zaustavljanja.

Pri ispitivanju kretanja bagera mereni su pritisci u vodovima hidromotora integralnih pogona obe gusenice kretnog mehanizma.

Na osnovu merenih veličina, za iste uslove ispitivanja, određene su sile u lancu F_{rd} , F_{rl} (sl.5) jedne i druge gusenice i ukupna sila vuče bagera F_r za sledeće periode i uslove kretanja: a) kretanje napred (0-4,5s); b) zaustavljanje bagera (4,5-6,5s); v) ponovno kretanje napred (6,5-9,5s); g) zakretanje bagera (gusenice imaju suprotne brzine) (9,5-13s) i d) zaustavljanje (13-16s).

Promene sila u lancima gusenica bagera, kako dijagram pokazuje, imaju vrlo dinamičnu promenu i izrazito udarni karakter naročito u prelaznim periodima promene pravca i smera kretanja i u slučajevima naglog zaustavljanja bagera [7].

3. ZAKLJUČAK

Definisani metod analize opterećenaja podloge oslanjanja guseničnih kretnih mehanizama hidrauličkih bagera je zasnovan na statičkom i dinamičkom modelu bagera i merenim veličinama stanja rada bagera u eksploatacionim uslovima.

Metod se može koristiti za određivanje karaktera promena i inteziteta delovanja bagera guseničara na podlogu oslanjanja, zatim za analizu reakcija podloge oslanjanja, odnosno analizu opterećenja guseničnih kretnih mehanizama i ocenu stabilnosti rada bagera.

Rezultati analize pokazuju da se pri kretanju hidrauličkih bagera guseničara, u zahvatu pogonskog lančanika i lancima gusenice, javlju dinamička opterećenja koja imaju udarni karakter kada se vrši promena pravca kretanja ili zaustavljanje bagera.

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Load analysis of the track type movement mechanisms of hydraulic excavators

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The paper contains the results of experimental research of loads of track type movement mechanisms of hydraulic excavators. Dynamic mathematical model was developed to determine the loads of surface reliance of track type movement mechanisms of hydraulic excavators based on the measured state sizes of work in the operating conditions. As the measured state sizes of the excavator there are taken: rising of supporting-moving mechanism, the angle of rotation and the pressure in the lines of hydromotor of drive of platform rotation and strokes and pressure in the lines of hydrocylinders of boom, arm and bucket of drive mechanisms of shovel excavator. Mathematical model is defined for determination of the forces in tracks, and forces in grips of sprocket wheel and chain of track. As an example, the results of analysis are presented: of the loads on the substrate reliance of hydraulic tracked excavator mass of 16000 kg equipped with a bucket, volume of 0.6 m³ and the loads of track chain of hydraulic excavator mass of 24000 kg equipped with manipulator that have bucket volume of 1 m³.

Key words: Hydraulic excavators, track type movement mechanism, loads

0. INTRODUCTION

Supporting-moving mechanisms of hydraulic excavator are performed with a single pair of tracks, with the following standard-module and elements: track chain 1 (Figure 1), shoes 2, the drive sprocket wheel 3, the anchoring device 4, carrier 5 and the guide rollers 6 unified with longitudinal tracks carrier 7.

Depending on the working conditions, the same model of excavator can be equipped with: standard (ST), extended (LC) and reinforced (HD) tracked moving mechanism. The listed variants of motion mechanisms are different among themselves according to the width of the shoes, length of reliance L or according to the range of tracks.

Tracked supporting-moving excavator mechanisms are characterized by good adhesion to the substrate and the ability to achieve a large tensile forces, low speeds of movement, a small surface pressure onto the substrate, great mobility and stabile movement.

Fig. 1 Track type movement mechanisms of hydraulic excavators

In this paper there are given mathematical models for determination of reliance surface loads and the forces in the chain of track type movement mechanism of hydraulic excavators according to the measured state sizes of work and the movement of excavator in the operating conditions.

1. MATHEMATICAL MODEL OF EXCAVATOR

For determine surface pressure at the substrate reliance crawler mechanisms of motion of hydraulic excavators mathematical model is developed for general configuration of the kinematic chain excavator composed of: the support and movement mechanism L_1 (Fig. 2a), the rotating platform L_2 and the three-plane load manipulators with: boom L_3 , stick L_4 , bucket L_5 . The support and movement member is crawler mechanisms of motion with width of the shoe b , length reliance caterpillars L , and ranging of caterpillars B .

Mathematical model of member of the kinematic chain excavator L_i was defined in its local coordinate system $O_i x_i y_i z_i$, geometric, kinematic and dynamic parameters covered by a set size (Fig. 2) [1][2]:

$$L_i = \{e_i, s_i, t_i, m_i, J_i\} \quad (1)$$

where: e_i - unit vector joint axes O_i which member L_i linked to previous member L_{i-1} , s_i - the position vector of the center joint O_{i+1} , which is a member chain L_i linked for the next member of the L_{i+1} (intensity vector is the length of the kinematic members), t_i - the position vector of the center of mass of member L_i , m_i - mass of member, J_i - moment of inertia of member.

Dynamic size operating in the center of the mass member L_i are: inertial force F_i which is determined by the second Newton's law:

$$F_i = -m_i w_i \quad (2)$$

and moment of inertial force M_i , determined on the basis

of Euler's dynamic equations:

$$\hat{M}_i = -\hat{J}_i \hat{\varepsilon}_i + (\hat{\omega}_i \times \hat{J}_i \hat{\omega}_i) \quad (3)$$

where: $\mathbf{w}_i$ – linear acceleration of member L_i , $\hat{J}_i$ – the moment of inertia tensor member for a coordinate system linked to the center of mass of members with axes parallel to the local coordinate axes of member, $\hat{\omega}_i = A_{oi} \omega_i$, $\hat{\varepsilon}_i = A_{oi} \varepsilon_i$ – angular velocity and angular acceleration of member L_i in local coordinate system $O_i x_i y_i z_i$, A_{oi} – transform matrices of vectors from absolute coordinate system of excavator to local coordinate system of member.

For the determination of kinematic quantities, angular velocity ω_i and acceleration ε_i and $\mathbf{w}_i$ linear acceleration of the center of mass of chain members L_i , in relation to the absolute coordinate system $OXYZ$, use the recursive equation [2] [3]:

$$\omega_i = \omega_{i-1} + \dot{\theta}_i \mathbf{e}_i \quad (4)$$

$$\varepsilon_i = \varepsilon_{i-1} + \ddot{\theta}_i \mathbf{e}_i + (\omega_{i-1} \times \dot{\theta}_i \mathbf{e}_i) \quad (5)$$

$$\mathbf{w}_i = \mathbf{w}_{i-1} + (\varepsilon_{i-1} \times (\mathbf{s}_{i-1} - \mathbf{t}_{i-1})) + \omega_{i-1} \times (\omega_{i-1} \times (\mathbf{s}_{i-1} - \mathbf{t}_{i-1})) + (\varepsilon_i, \mathbf{t}_i) + \omega_i \times (\omega_i \times \mathbf{t}_i) \quad (6)$$

where: $\dot{\theta}_i, \ddot{\theta}_i$ – angular velocity and angular acceleration of

the member L_i in a joint O_i .

The total force linked for the center of mass member L_i kinematic chain excavator, taking the influence of gravity, is equal to:

$$\mathbf{F}_{ui} = \mathbf{F}_i + m_i \mathbf{g} \quad \forall i = 1, \dots, 5 \quad (7)$$

and moment of inertial force member L_i in the absolute coordinate system is defined by expression:

$$\mathbf{M}_{ui} = A_{io} \hat{\mathbf{M}}_i \quad \forall i = 1, \dots, 5 \quad (8)$$

Reducing the loads of all excavators in the center O_1 reliance substrate crawler mechanisms of motion, obtained the resulting dynamic force $\mathbf{F}_I$ and dynamic moment load $\mathbf{M}_I$ substrate reliance excavator (Fig. 2a):

$$\mathbf{F}_I = \mathbf{W} + \sum_{i=1}^5 \mathbf{F}_{ui} \quad (9)$$

$$\mathbf{M}_I = ((\mathbf{r}_w - \mathbf{r}_I) \times \mathbf{W}) + \sum_{i=1}^5 ((\mathbf{r}_w - \mathbf{r}_I) \times \mathbf{F}_{ui}) + \sum_{i=1}^5 \mathbf{M}_{ui} \quad (10)$$

where: $\mathbf{W}$ – actual digging resistance (experimentally determined based on the measured values) [3], $\mathbf{F}_{ui}$ – the total force in the center of the mass member L_i , taking the

Fig. 2 The mathematical model of the hydraulic excavator for experimental testing of surface pressure crawler mechanisms of motion

influence of inertia and gravity, M_{ii} - moment of inertial forces members L_i in the absolute coordinate system.

The intensity of the component of the force F_I and moment M_I that cause surface pressure normal to the substrate reliance crawler mechanisms of motion of excavators are (Fig. 2b):

$$F_{Ix} = (F_I \cdot i), \quad F_{Iy} = (F_I \cdot j), \quad F_{Iz} = (F_I \cdot k) \quad (14)$$

$$M_{Ix} = (M_I \cdot i), \quad M_{Iy} = (M_I \cdot j), \quad M_{Iz} = (M_I \cdot k) \quad (15)$$

For load analysis of reliance of movement mechanism of excavator, according to defined mathematical model, required measured values of the state of kinematic chain excavator at work in the exploitation conditions are: uplift support and movement mechanism c_1 (Fig. 2), angle turning platforms c_2 and pressure p_{21}, p_{22} in lines of hydraulic motor slewing platform drive, strokes c_3, c_4, c_5 and pressures p_{i1}, p_{i2} ($i = 3, 4, 5$) in the working lines hydrocylinder driving mechanisms: boom, stick and bucket driving mechanisms of manipulators [2]. Depending on the measured values c_i , using the transfer functions driving mechanisms, determine the generalized coordinates θ_i , based on which, the double differencing, define the angular velocity $\dot{\theta}_i$, and angular acceleration $\ddot{\theta}_i$ of the joints members kinematic chain excavator:

$$\dot{\theta}_i = \frac{\theta_{i(t+\Delta t)} - \theta_{i(t-\Delta t)}}{2\Delta t} \quad (16)$$

$$\ddot{\theta}_i = \frac{\theta_{i(t+2\Delta t)} - 2\theta_{i(t)} + \theta_{i(t-2\Delta t)}}{4\Delta t^2} \quad \forall i = 2, 3, 4, 5 \quad (17)$$

where: $\theta_{i(t)}$ - generalized coordinates at time t duration of the operation digging, $\theta_{i(t+\Delta t)}, \theta_{i(t-\Delta t)}, \theta_{i(t+2\Delta t)}, \theta_{i(t-2\Delta t)}$ - generalized coordinates in a moment of time which is for one or two intervals Δt larger or less than the time t , Δt - the time interval between two successive measurements of the values.

2.1. Determination of loads of reliance surface

Using the developed program, based on a defined mathematical model, an analysis of the load surface pressure to the substrate reliance crawler mechanisms of motion of hydraulic excavators 16000 kg width backhoe mechanism volume of the bucket 0,6 m³ base on measurement value of state working excavator in exploitation conditions [2].

As an example, in this study, given the results for the analysis of manipulative task with operations: digging (0-6s) (Fig. 3a, b), transmission (6-10s - by turning the platform for 40°), unloading (10-14s - at the height of unloading around 5,5 m) and return (14-18s - in the starting digging plane).

Fig.3 The loads of reliance surface: a) the components of the static F_{Is} and dynamic F_{Id} force b) a components of static M_{Is} and dynamic torques M_{ID}

The analysis results show (Fig. 3a,b) component loads reliance substrate has expressed the changes and in respect intensity and direction of action. The highest values of forces and moments loads reliance substrate occur in the operation of digging. The highest intensity at the vertical force component F_{ly} that during the manipulative task pressure reliance substrate.

Exceptionally high intensity and the change of direction in relation to the other, has moment component M_{lz} around Oz axis - also during digging operations. Changing direction M_{lz} moment occurs when the digging operation due to changes in vector digging resistance.

When loads reliance substrate excavator difference of static and dynamic loads is expressed in operation of transmission, unloading and returning to digging plane when kinematic chain excavator has an open configuration and duration of operations are relatively small. When digging operation dynamic loads are significantly lower because of the closed configuration of the kinematic chain excavator (bucket is in contact with the ground) and the digging progress is relatively slow.

2. FORCE ANALYSIS IN TRACK

In steady state of movement mechanisms track chain have an initial tensile force F_z (Fig.4a). When track are moving on the drive sprocket wheel is supplied with M_o torque due to which the load of drive section of tracks is increased extensive force F_o , so that, for small angles of mobility α, β there are equality [1][4]:

$$F_{k1} = F_{k2} + F_o \quad (18)$$

where: F_{k1} - force in the branch of tracks which faces the drive sprocket, F_{k2} - force in the branch tracks that go down from the drive sprocket.

From the last equation it follows that:

$$F_o = \frac{M_o}{r_p} = F_{k1} - F_{k2} \quad (19)$$

where: M_o - torque acting on the drive sprocket of track type movement mechanism.

For tighten wheel encountering a branch tracks is actually descending branch of the drive sprocket, so that:

$$F_{k2} = F_{z2} \quad (20)$$

The difference of the forces of the anchoring wheel gives a moment that is equal to the sum of the moment M_{zi} which is caused by rotation inertia of the anchoring wheel and moment M_{zt} due to the frictional force of in the bearings of anchoring wheel:

$$(F_{z1} - F_{z2})r_z = M_{zi} + M_{zt} \quad (21)$$

which cause

$$F_{z2} = F_{z1} - \frac{M_{zi} + M_{zt}}{r_z} \quad (22)$$

where: F_{z1} - force in the branch tracks that come down from the anchoring wheel, r_z - radius of the anchoring wheel.

The resulting force which acting on the pulling part of the tracks, is equal to the difference of forces $F_{k1} - F_{z1}$ and has the direction of the force F_{k1} . This differential force is the resulting force on the traction part of the tracks where the tangential reaction are opposed (adhesive force) of the support reliance F_{μ} , with the direction of operation in the moving direction of the track type movement mechanism so that:

$$F_{\mu} = F_{k1} - F_{z1} \quad (23)$$

The loads of the track chain is transmitted to the skeleton of the machine via the drive sprocket, the tension wheel and the carrier rollers. The sum of all horizontal force components in the axes of the carrier rollers (F_{npx} , F_{nzx}) (Figure 4a), the drive sprocket (F_{k1x}) and the tension wheel (F_{z1x}) are the same as the reaction of the F_{μ} substrate.

The tangential reaction of the reliance surface in the form of push forces is transmitted through the bearing of the carrier rollers and the drive sprocket to the skeleton of the track type movement mechanism, achieving the pulling force of the machine.

Fig. 4 Dynamics of the track type movement mechanism: a) forces which acting on the elements of the track type movement mechanism [4], b) the forces in grip of the chain and the track [5]

The reaction force in reliance surface is in fact the force of adhering of tracks can be expressed by the equation:

$$F_{\mu} = \mu_c mg + 2bh_r \tau \cdot z_r \quad (24)$$

where: μ_c - friction coefficient of the track on the surface reliance, m - mass of the machine, b - the width of the shoes, h_r - the height of the shoe rib of track, τ - the shear stress of the surface reliance, z_r - the number of the rib of tracks in contact with the surface.

The performance of the track type movement mechanism can be with the rear and the front position of the drive sprockets in relation to the direction of the machine speed. In both cases, the character of the tightening force change of the tracks by contour is not the same.

In the first case, the upper free branch of the track has a small load size from the horizontal forces conditioned by its own deflection of the track chain. When the track chain approach to the drive sprocket, the track chain tension begins to grow, reaching the maximum on the drive wheel.

In the second case, the greatest tensile forces occurs on the upper branch of the tracks, reducing while approaching to the carrier roller, where they balance with the tangential reaction of the surface. The variation of the track type supporting mechanism with the front drive sprockets requires greater traction of the track and has a higher chain load.

Vertical components (F_{kly} , F_{zly} , F_{npy} , F_{nzy}) (Fig.4a) of forces which engage the axes of drive sprockets and tension wheels, as well as the axes of the rear and front carrier rollers, form in a vertical plane a force system that affects on distribution of normal reaction of the surface reliance of the machine.

2.1. The forces in the grip of sprocket and track chain

During the drive sprocket wheel and the track chain article, there are mutual loads with the same time relative movement of the chain sprockets and the roller of the chain and the relative movement of the roller and bush of the chain. As a consequence of the loads and relative motion, the wear of the contact elements is occurred, especially on roller and bush is caused, and with the increase in the gap in the joint, the step of chain is increased and so total length of chain. In order to maintain a correct movement of the track in the case of a non-desired increase in the length of the chain, the movement mechanisms have a tensioning device which enables the necessary displacement of the tension wheel and thus maintains the necessary force for tightening the chains.

For the analysis of the forces that are generated in the grip of drive sprocket and the track chain, when sprocket is rotated in both directions, a model has been developed with the following assumptions: on the sprocket and the chain acting only forces in the longitudinal plane xy of the chain (Fig.4b), the angular speed of the drive sprocket is Constant, the inertial forces and the own chain weight are neglected, the direction of force in the chain F_k (Fig. 4) does not change when entering the grip, the centers of the joints of the chain lie on the sub-line of the diameter of the sprocket [5].

Because of the small impact time, the inertial forces are ignored and a static mathematical model is applied which includes the following forces: the force in the chain F_k (Fig. 4b), the force normal on the flank of the drive sprocket F_z , the force of the chain segment that spun with the sprocket F_{k+1} , the friction force between the sprocket teeth and the chain roller $\mu_z F_z$, the friction force of between the roller and the pin of the chain segment $\mu_b F_{k1}$. The coefficient of friction between the sprocket teeth and the bush of the chain segment is μ_z , while the coefficient of friction between the pin and the bush segment is μ_b .

For segment I (Fig. 4b) of the track chain, which is in the grip with the sprocket when the machine is moving forward, static equilibrium conditions can be written in the form [5]:

$$\begin{bmatrix} \cos \rho & \sin \rho & 0 \\ -\sin \rho & \cos \rho & 0 \\ 0 & -0,5d_2 & F_k \end{bmatrix} \cdot \begin{bmatrix} F_n \\ F_r \\ h \end{bmatrix} = \begin{bmatrix} F_k \\ 0 \\ 0 \end{bmatrix} \quad (25)$$

Where: $\rho = \arctg \mu_b$ is the angle of friction between the pin and the roller of the chain segment, d_2 - the diameter of the pin, h - the lifting height of the chain joint, F_n , F_r - the normal and tangent components of force F_{k+1} .

For article II (Fig. 4b) of the track chain, which is in the grip with a sprocket when the machine is moving forward, the static equilibrium conditions can be written in the form:

$$\begin{bmatrix} 0 & \sin(\tau + \gamma_w) & \cos(\tau + \gamma_w) \\ 1 & \cos(\tau + \gamma_w) & -\sin(\tau + \gamma_w) \\ 0 & 0 & t - 0,5d_2 \end{bmatrix} \cdot \begin{bmatrix} F_z \\ F'_{k2} \\ F''_{k2} \end{bmatrix} = \begin{bmatrix} F_{k1} \sin(0,5\tau + \gamma_w - \varphi) \\ F_{k1} \cos(0,5\tau + \gamma_w - \varphi) \\ h F_{k1} \end{bmatrix} \quad (26)$$

Where: τ is the angle that correspond to the chain step on the sub-diameter of the sprocket (Fig.4b), γ_w is the angle of the normal to the plane of sprocket and roller of chain, φ - the current angle of rotation of the sprocket, F'_{k2} , F''_{k2} components of the force F_{k2} acting in the joint to which article II and the following article of the chain are bound.

The angle corresponding to the chain step on the sub-diameter and have value:

$$\tau = \frac{4\pi}{z} \quad (27)$$

where: z - number of sprocket teeth

The angle of the normal to the contact plane of the sprocket and roller of the chains is determined by the equation:

$$\gamma_w = \gamma - \left(\frac{\tau}{4} + \delta \right) = \beta - \frac{\tau}{2} - \delta \quad (28)$$

In the analysis, the current spin angle φ moves within the limits:

$$-\frac{\tau}{2} < \varphi < \frac{\tau}{2} \quad (29)$$

2.2. Testing the force in the track

As an example, the results of the determined force in the track chains of the movement mechanism are given, for hydraulic excavator in mass 24000 kg with hydrostatic transmission of motion. The same track drives are integral gearboxes with a hydromotor and a planetary gearbox for

Fig.5. The change of the force in the left track F_l and the right track F_d and the resulting pulling force F_r determined by testing of the movement track type hydraulic excavator in mass 24000 kg [7]

which the sprockets are attached. Hydraulic motors are powered, in an open hydrostatic circuit, hydropumps with regulation according to the criterion of constant hydraulic power [6].

The excavator's motion test was performed on a flat and dry macadam surface. The movement responded to the mode of moving with frequent interruptions on the linear and curvilinear path, with periods of accelerated and slow motion, constant velocity and sudden stop motion.

At the movement of the excavator the pressure in the hydraulic motors of the integral drives of the two tracks are tested.

Based on the measured values, for the same test conditions, the forces in the chain F_{rd} , F_{rl} (Fig. 5) of each and every track and the total pulling force of F_r for the following periods and driving conditions are determined: a) forward movement (0-4,5s); b) stopping the excavator (4.5-6.5s); v) re-forwarding (6.5-9.5s); g) rotating the excavator (the tracks have opposite speeds) (9.5-13s) and d) stopping (13-16s).

The changes in the power of the track chains of the excavator, as the diagram shows, have a very dynamic change and a striking impact character, especially in transition periods of change of direction of movement, and in cases of sudden stopping of excavators [7].

3. CONCLUSION

The defined method of load analysis of the reliance surface of the track type movement mechanism of hydraulic excavator is based on the static and dynamic model of the excavator and the measured sizes of excavator in exploitation conditions. The method can be used to determine the character of the changes and the intensity of the operation of the crawler excavator on the reliance surface, then to analyze the reaction of reliance surface, or to analyze the load of the track type movement mechanism and evaluate the stability of the excavator.

The results of the analysis show that at movement of the hydraulic track type excavator, in the the grip of drive sprocket and track chain, the dynamic loads that have a striking character when changing the direction of movement or stopping are occurred.

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