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Review Article

JUSTIFICATION OF THE HYDRAULIC SYSTEM PARAMETERS OF THE EXCAVATION BODY OF A MULTI-PURPOSE ROAD CONSTRUCTION VEHICLE BASED ON THE TTZ TRACTOR

Tulkin Askarhodjaev[1], Kamoliddin Rustamov[1], Farida Aymatova[1], Gulruh Husenova[1]

[1]Tashkent Institute of Projection, Building and Maintenance of Automobile Roads, Tashkent city, Uzbekistan, 100060, avenue Amir Temur 20.

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Abstract

About the strategy for further development of the Republic of Uzbekistan 3.2. The section considers the implementation of an active investment policy aimed at modernization, technical and technological updating of production, implementation of industrial, transport, communication and social infrastructure projects, further modernization and diversification of industry by transferring it to a qualitatively new level, aimed at accelerating the development of high-tech manufacturing industries, primarily for the production of finished products with high added value based on deep processing Local raw material resources.

The object of research is a road-building machine with the proposed mechanism for driving excavation equipment.

The article presents the results of a power, durable calculation of the components and parts of a hydraulic drive, the required kinematic power parameters are established taking into account external forces.

In solving the tasks, methods of numerical calculation are applied. The developed mechanical model of a single-bucket excavator with hydraulic drive, consisting of links, which are absolutely rigid structures, characterized by masses of links mi; impart by moments of inertia Jix, Jiy, Jiz relative to the axes of their own local coordinate systems; coordinates of the centers of mass of links in local coordinate systems. Modeling a proportional hydraulic drive of a single-bucket excavator is a two-stage process, which is associated with the development of a basic design hydraulic circuit of a single-bucket excavator (Fig. 2).

Keywords: hydraulic drive, modeling, automatic regulation, system, excavator, working body

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INTRODUCTION

The developed machine is universal and compact (Fig. 1). With high maneuverability and multifunctional that allow you to perform many types of work on road construction equipment.

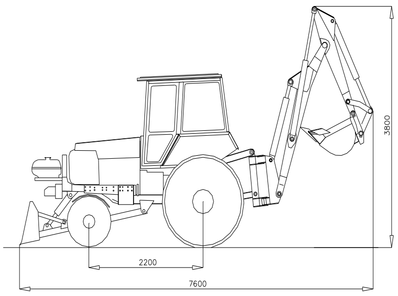


Fig. 1. The multi-purpose machine.

The working equipment of the machine, designed to perform certain operations, consists of a working body that directly interacts with the developed, overloaded or moved materials and cargoes, actuators that provide specified movements of the working body, and supporting structures that accept loads from the working body and transfer them to the base body cars. So in a single-bucket excavator, the working body is a bucket, the supporting structure is an arrow and a handle, and the executive mechanisms are traction, levers and other elements that control the movements of the bucket. Since the developed multi-purpose machine has a hydraulic drive, the driving force from the drive to the corresponding elements of the working equipment is transmitted using hydraulic cylinders.

The expansion trend in the use of hydraulic excavators is realized mainly through the use of various types of interchangeable working equipment.

The aim of the work is to develop a mathematical model of the hydraulic drive of the lifting mechanism of the working equipment of the developed multi-purpose machine. To achieve this goal, it was necessary to solve the following tasks: to develop a schematic and structural diagram of a system for automatically controlling the supply of hydraulic pumps for the lifting mechanism of excavators working equipment; make equations describing transients in the hydraulic system of an excavator, and converted for use in a simulation system.

METHOD

The work used the methods of engineering calculation of the mechanics of machines and hydraulic system drives, taking into account the kinematic, strength and dynamic characteristics, as well as the force factors acting on the nodes and parts of the drive of the excavation unit of the developed machine.

LITERATURE REVIEW

Scientific research on the problems of developing technological processes for multi-purpose machines based on the TTZ tractor multi-part machine was carried out by T.I.Askarhojaev [1], [2], [3], L.Ulmasova [3] and K.J.Rustamov [2], [8], research and theoretical calculations of the process of digging, drilling and moving learned by V.I. Balnevne [4], L.A.Hmara [4], Yu.A.Braykovskiy [5], S.I. Kornyushenko [5], I.P. Krutikov [7]. Hydraulic excavator operating equipment systems researched by Shapoval, Zaslavskiy, Balovnev, Pohvalov [9].

RESULTS AND DISCUSSION

The first stage of the kinematic calculation and analysis of the equipment being developed is the construction of its structural scheme (Fig. 2), which should provide the specified movements of the corresponding parts of the mechanism and, in particular, the necessary trajectory of the working body.

Cargo weight: G_W = 4000 N , Lifting speed: $\upsilon_{cp} = 0{,}2\, {}^{m}\!/_{s}$

Load tilt speed: v_{inc} = 0,12 m/s Lifting height: $H_m = 2m$

The friction force in the guides between the fixed and moving frames: $R = 1200\,N$

Carriage weight: $G_K = 700\,N$, Movable frame weight: $G_{n.p.} = 1000\,N$

Movable frame weight: $G_{н.р.} = 2200\,N$

Dimensions:

$$h = 0{,}9m$$

$$n = 0{,}6m$$

$$k = 0{,}7m$$

The maximum angle of inclination of the frame forward: $\alpha_2 = 7°$

The angle of the cylinder axis: $\beta = 20°$

Pressure rating: $P_{nom} = 16mPa$

Determination of loads in power cylinders

Lifting cylinder load P_1.

$$P_1 = \frac{2\cdot(G_{gr}+G_k)}{\eta_{bl}^2} + G_{n.r.} + R,$$

where $\eta_{бл}$ = 0.97 – KYW blocks.

$$P_1 = \frac{2\cdot(40000+700)}{0.97^2} + 1000 + 1200 = 88713H$$

Rod speed in the lift cylinder.

$$\upsilon_1 = 0.5\cdot\upsilon_{gr} = 0.5\cdot 0.2 = 0.1 \, {m}/{s}$$

The load on the tilt cylinder $P_{2\Sigma}$.

$$P_{2\Sigma} = \frac{(G_{gr}+G_k+0.25\cdot G_{n.r.}+0.75\cdot G_{p.r.})\cdot H_{gr}\cdot\sin\alpha_2 + G_{gr}\cdot n\cdot\cos\alpha_2}{k\cdot\cos(\beta+\alpha_2)}$$

$$P_{2\Sigma} = \frac{(40000+700+0.25\cdot 2200+0.75\cdot 1000)\cdot 2}{0.7\cdot\cos(20+7)}$$

$$\frac{\cdot\sin 7 + 40000\cdot 0.6\cdot\cos 7}{} = 54606 \quad N$$

The force on the rod of one hydraulic cylinder

$$P_{2\max} = \frac{P_{2\Sigma\max}}{2} = \frac{54606}{2} = 27303N$$

The speed of the rod is equal to the specified speed

$$\upsilon_{sht} = \upsilon_{nakl} = 0.12 \, {m}/{s}$$

Sizing of hydraulic cylinders

The diameter of the piston is determined from the condition

$$P_{c.p.} \ge P_1$$

$$D_1 \ge \sqrt{\frac{P_1\cdot 4}{P_{nom}\cdot\pi\cdot\eta_{g.m.}}}$$

where $\eta_{g.m.} = 0.95$ - hydromechanical cylinder efficiency.

$$D_1 \ge \sqrt{\frac{88713\cdot 4}{16\cdot 10^6\cdot\pi\cdot 0.95}} = 0.086m$$

according to GOST 12447-80 we accept D_1=90mm.

Piston stroke $h_{n1} = 0.5\cdot H_{gr}^{\max} = 0.5\cdot 2 = 1m$

Full stem extended cylinder length $L_{\max} = 2\cdot h_{n1} = 2\cdot 1 = 2m$

The diameter of the rod d1 is determined based on the longitudinal stability

$$d_1 = \sqrt[4]{\frac{k^2\cdot L_{\max}^2\cdot 64\cdot P_1}{\pi^2\cdot E}}$$

where k=1 – coefficient taking into account the design features of the hydraulic cylinder;

$$E = 2\cdot 10^5 MPa = 2\cdot 10^{11} \, {N}/{m^2}$$

- modulus of elasticity of the rod material.

$$d_1 = \sqrt[4]{\frac{1^2\cdot 2^2\cdot 64\cdot 88713}{\pi^2\cdot 2\cdot 10^{11}}} = 0.058m$$

according to GOST 12447-80 we accept d_1=63mm.

The dimensions of the tilt cylinders are determined from the condition $P_{c.n.} \ge P_{2\max}$

The diameter of the rod is determined based on the compressive or tensile strength

$$d_2 \ge \sqrt{\frac{4\cdot P_{2\max}}{\pi\cdot[\sigma]}}$$

where $[\sigma] = 120\cdot 10^6 \, {N}/{m^2}$ - permissible tensile stresses.

$$d_2 \ge \sqrt{\frac{4\cdot 27303}{\pi\cdot 120\cdot 10^6}} = 0.017m$$

according to GOST 12447-80 we accept d_2=18mm.

$$P_{sl} = 0.15\cdot P_{nom} = 0.15\cdot 16 = 2.4MPa$$

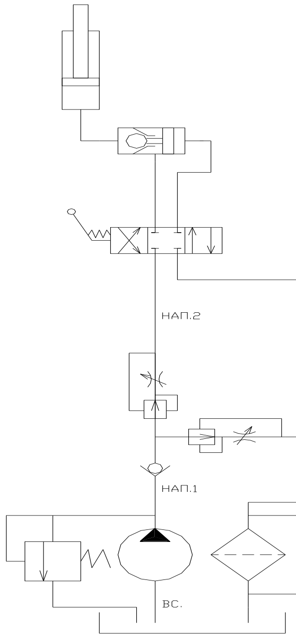
- discharge pressure at the exit of the hydraulic cylinder.

$$D_2 \ge \sqrt{\frac{4\cdot(P_{2\max}\cdot\eta_{g.m.} + \frac{\pi}{4}\cdot d_2^2\cdot P_{nom})}{\pi\cdot(P_{nom}-P_{sl})}} =$$

$$= \sqrt{\frac{4\cdot(27303\cdot 0.95 + \frac{\pi}{4}\cdot 0.018^2\cdot 16\cdot 10^6)}{\pi\cdot(16-2.4)\cdot 10^6}} = 0.053m$$

according to GOST 12447-80 we accept D_2=56mm.

$l_{NAP.1} = 1m$; $l_{NAP.2} = 5m$; $l_{NAP.3} = 6m$;



$l_{SL.} = 3m$; $l_{BC.} = 0{,}1м$

Fig. 2. The hydraulic circuit

Actual force on the ram for the lift cylinder

$$P`_1 = P_{nom} \cdot \frac{\pi \cdot D_1^2}{4} - \mu \cdot \pi \cdot D_1 \cdot \frac{1}{3} \cdot b_1{}^*$$

$$(P_{nom} - P_0) - \mu \cdot D_1 \cdot \pi \cdot \frac{1}{3} \cdot b_2 P_0$$

where μ = 0.1 – coefficient of friction between the sealing or wiper sleeve and the plunger;

b_1=10mm and b_2=8mm – respectively the width of the sealing and wiper sleeve;

P_0 =3 mPa – contact pressure arising from cuff deformation.

$$P`_1 = 16 \cdot 10^6 \cdot \frac{\pi \cdot 0.09^2}{4} - 0.1 \cdot \pi \cdot 0.09 \cdot \frac{1}{3} \cdot 0.01(16-3)$$

$$\cdot 10^6 - 0.1 \cdot 0.09 \cdot \pi \cdot \frac{1}{3} \cdot 0.008 \cdot 3 \cdot 10^6 = 100336..N$$

Actual pull force of the tilt cylinder rod

$$P`_2 = (P_{nom} - P_{sl}) \frac{\pi \cdot D_2^2}{4} - P_{nom} \frac{\pi \cdot d_2^2}{4} - \mu \cdot \pi \cdot d_2 \frac{1}{3} b_3 (P_0 + P_{nom})$$

$$-\mu \pi D_2 \frac{1}{3} b_5 (2P_0 + P_{nom} + P_{sl}) - \mu \pi d_2 \frac{1}{3} b_4 P_0$$

where b_3=4mm., b_4=4mm., b_5=10mm. – accordingly, the width of the sealing and wiper cuffs on the rod, the sealing cuffs on the piston.

$$P`_2 = (16 - 2.4) \cdot 10^6 \frac{\pi \cdot 0.056^2}{4} - 16 \cdot 10^6 \frac{\pi \cdot 0.018^2}{4}$$

$$-0.1 \cdot \pi \cdot 0.018 \frac{1}{3} 0.004(3+16) \cdot 10^6 -$$

$$-0.1\pi 0.056 \frac{1}{3} 0.01(2 \cdot 3 + 16 + 2.4) \cdot 10^6$$

$$-0.1\pi 0.018 \frac{1}{3} 0.004 \cdot 3 \cdot 10^6 = 27829N$$

Hydromechanical efficiency of a lifting hydraulic cylinder

$$\eta_{г.м.1} = \frac{P`_1}{P_{nom} \frac{\pi D_1^2}{4}} = \frac{100336}{16 \cdot 10^6 \frac{\pi 0.09^2}{4}} = 0.986$$

Hydromechanical efficiency of the tilt hydraulic cylinder when retracting

$$\eta_{г.м.2} = \frac{P`_2}{(P_{nom} - P_{sl}) \frac{\pi D_2^2}{4} - P_{nom} \frac{\pi d_2^2}{4}} =$$

$$= \frac{27829}{(16 - 2.4) \cdot 10^6 \frac{\pi 0.056^2}{4} - 16 \cdot 10^6 \frac{\pi 0.018^2}{4}} = 0.946$$

Choosing a pump for the hydraulic system

Feed for lift cylinder

$$Q_1 = \upsilon_1 \frac{\pi D_1^2}{4} = 0.1 \frac{\pi 0.09^2}{4} = 0.000636 \, m^3/s = 38.170 \, l/\min$$

Feed for tilt cylinder

$$Q_2 = \upsilon_2 \frac{\pi D_2^2}{4} = 0.12 \frac{\pi 0.056^2}{4} = 0.000296 \, m^3/s = 17.734 \, l/\min$$

Because the combination of operations is not provided, then we will calculate Q_1

$$q_{nas} = \frac{Q_1}{n_н \cdot \eta_{ом}}$$

where $n_н = n_{вм}$=1500 – pump speed;

$\eta_{ОМ} = 0.96$ - volumetric efficiency of the pump.

$$q_{nas} = \frac{38.17 \cdot 10^3}{1500 \cdot 0.96} = 26.507 \, sm^3/ob$$

Choose a pump:

Type: NSh -32

$$q = 32 \, sm^3/ob .... \frac{P_{nom}}{P_{\max}} = \frac{16}{20} MPa.$$

$$Q_{nom} = 43.2 \, l/\min .... \eta_{ob} = 90\%...$$

$$..... \frac{n_{nom}}{n_{\max}} = \frac{1500}{2000} ob/\min$$

$$\eta = 78\% .... m = 6.65kg ... \Delta = 40mkm$$

where q – hydraulic displacement;

P_{max} – maximum pressure;

N_{nom}, n_{max} – respectively rated and maximum permissible speed of the hydraulic machine;

Q_{nom} – nominal feed of the hydraulic machine or nominal flow of the hydraulic line and hydraulic unit under nominal conditions;

η_{ob} – volumetric efficiency under nominal conditions;

η - full efficiency of the hydraulic machine;

m – weight of hydraulic machine (without working fluid);

Δ - permissible fineness of the working fluid filtration.

$$N_{нас} = \frac{Q_{nas} P_{nom}}{\eta_{nas}} = \frac{0.00072 \cdot 16 \cdot 10^6}{0.78} = 14.769 kVt$$

Table I. Calculation and selection of hydraulic equipment

Safety valve	520.12	D_y=12 mm	$Q_н$=100 l/min	$P_н$=25 mPa	⍰		m=0,65 kg
Check valve	61 100	D_l=16 mm	$Q_н$=63 l/min	$P_н$=20 mPa	P=0,05 mPa	Qут=0,13 l/min	m=0,53 kg
Flow regulator	BG57-24	Dl=20 mm	$Q_н$=63 l/min	$P_П$=20 mPa	P_P=19 mPa	Qут=0,15 l/min	P=0,2 mPa
Flow regulator	BG57-24	Du=20 mm	$Q_н$=63 l/min	$P_П$=20 mPa	P_P=19 mPa	Qут=0,15 l/min	P=0,2 mPa
Raspre - divider	R-20	D_l=20 mm	$Q_н$=100 l/min	$P_н$=20 mPa	P=0,32 mPa	Qут=0,15 l/min	
Raspre - divider	R-20	D_l=20 mm	$Q_н$=100 l/min	$P_н$=20 mPa	P=0,32 mPa	Qут=0,15 l/min	
Filter	1.1.32-25	D_l=32 mm	$Q_н$=100 l/min	$P_н$=0,63 mPa	P=0,08 mPa	Δ =25 mkm	m=10 kg
Water lock	U4610.35	D_l=12mm	$Q_н$=50 l/min	$P_н$=16 mPa			m=1,7 kg

W=15l. – volume of the tank.

Verification calculation of the hydraulic system with the determination of pressure losses in the hydraulic lines; calculation of drive efficiency

Pipeline calculation

Oil: VMGZ (MG-15V), t=50 C.

$$\nu = 10^{-5} \frac{m^2}{s}, \rho = 865 \frac{kg}{m^3}$$

Pipeline inside diameter:

$$d_{tr} = \sqrt{\frac{4Q}{\pi \cdot \upsilon_{dop}}}$$

$\upsilon_{per.fl.} = 4 \, m/s$ – permissible fluid velocity.

$$\upsilon_{per.vs.} = 2 \, m/s \qquad \upsilon_{per.sl.} = 2 \, m/s$$

$$d_{tr.nap.1} = \sqrt{\frac{4 \cdot 0{,}00072}{\pi \cdot 4}} = 0{,}016m$$

$$d_{tr.nap.2} = \sqrt{\frac{4 \cdot 0{,}000636}{\pi \cdot 4}} = 0{,}016m$$

$$d_{tr.nap.3} = \sqrt{\frac{4 \cdot 0{,}000592}{\pi \cdot 4}} = 0{,}014m$$

$$d_{tr.sl.} = \sqrt{\frac{4 \cdot 0{,}00072}{\pi \cdot 2}} = 0{,}022m$$

$$d_{tr.vs.} = \sqrt{\frac{4 \cdot 0{,}00072}{\pi \cdot 2}} = 0{,}022m$$

We specify the actual speed

$$\upsilon_{tr} = \frac{4Q}{\pi \cdot d^2}$$

$$\upsilon_{tr.nap.1} = \frac{4 \cdot 0{,}00072}{\pi \cdot 0{,}016^2} = 3{,}581 \, m/s$$

$$\upsilon_{tr.nap.2} = \frac{4 \cdot 0{,}000636}{\pi \cdot 0{,}016^2} = 3{,}163 \, m/s$$

$$\upsilon_{tr.nap..3} = \frac{4 \cdot 0{,}000592}{\pi \cdot 0{,}014^2} = 3{,}846 \, m/s$$

$$\upsilon_{tr.sl.} = \frac{4 \cdot 0{,}00072}{\pi \cdot 0{,}022^2} = 1{,}894 \, m/s$$

$$\upsilon_{tr.вс.} = \frac{4 \cdot 0{,}00072}{\pi \cdot 0{,}022^2} = 1{,}894 \, m/s$$

Determine the Reynolds number

$$R_e = \frac{\upsilon \cdot d_{tr}}{\nu}$$

$$R_{e\,\text{nnap.1}} = \frac{3{,}581 \cdot 0{,}016}{10^{-5}} = 5729{,}6$$

$$R_{e\,\text{nnap.2}} = \frac{3{,}163 \cdot 0{,}016}{10^{-5}} = 5060{,}8$$

$$R_{e\,\text{nnap.3}} = \frac{3{,}846 \cdot 0{,}014}{10^{-5}} = 5384{,}4$$

$$R_{e\,\text{sl.}} = \frac{1{,}894 \cdot 0{,}022}{10^{-5}} = 4166{,}8$$

$$R_{e\,\text{VVC.}} = \frac{1{,}894 \cdot 0{,}022}{10^{-5}} = 4166{,}8$$

Determine the coefficient of pressure loss along the length

$$\lambda = \frac{0{,}316}{R_e^{0{,}25}} \qquad \lambda_{nap.1} = \frac{0{,}316}{5729{,}6^{0{,}25}} = 0{,}036$$

$$\lambda_{nap.2} = \frac{0{,}316}{5060{,}8^{0{,}25}} = 0{,}037$$

$$\lambda_{nap.3} = \frac{0{,}316}{5384{,}4^{0{,}25}} = 0{,}037$$

$$\lambda_{sl} = \frac{0{,}316}{4166{,}8^{0{,}25}} = 0{,}039$$

$$\lambda_{sl} = \frac{0{,}316}{4166{,}8^{0{,}25}} = 0{,}039$$

We determine the pressure loss along the length according to the Darcy formula

$$\Delta P_{trl} = \rho \frac{\lambda \cdot L \cdot \upsilon^2}{d_{tr} \cdot 2}$$

$$\Delta P_{trl.nap.1} = 865 \frac{0{,}036 \cdot 1 \cdot 3{,}581^2}{0{,}016 \cdot 2} = 12479 Pa$$

$$\Delta P_{trl.nap.2} = 865 \frac{0{,}037 \cdot 5 \cdot 3{,}163^2}{0{,}016 \cdot 2} = 50031 Pa$$

$$\Delta P_{trl.nap.3} = 865 \frac{0{,}037 \cdot 6 \cdot 3{,}846^2}{0{,}014 \cdot 2} = 101445 Pa$$

$$\Delta P_{trl.sl} = 865 \frac{0{,}039 \cdot 3 \cdot 1{,}894^2}{0{,}022 \cdot 2} = 8251 Pa$$

$$\Delta P_{trl.vc} = 865 \frac{0{,}039 \cdot 0{,}1 \cdot 1{,}894^2}{0{,}022 \cdot 2} = 275 Pa$$

$$\sum \Delta P_{trl} = 12479 + 50031 + 8251 + 275 = 172481\ Pa$$

We determine the pressure loss at local resistances

$$\sum \Delta P_{tr\xi} \approx 0{,}3 \cdot \sum \Delta P_{trl} = 0{,}3 \cdot 172481 = 51744\ Pa$$

We determine the pressure loss in hydraulic units at a given flow rate

$$\Delta P_{rasch} = \Delta P_{zad} \left(\frac{Q_{rasch}}{Q_{zad}} \right)^2$$

Check valve

$$\Delta P_{rasch} = 0{,}05 \left(\frac{43{,}2}{63} \right)^2 = 0{,}024 MPa$$

Flow regulator

$$\Delta P_{rasch} = 0{,}02 \left(\frac{43{,}2}{63} \right)^2 = 0{,}01 MPa$$

Flow regulator

$$\Delta P_{rasch} = 0{,}02 \left(\frac{43{,}2}{63} \right)^2 = 0{,}01 MPa$$

Distributor

$$\Delta P_{rasch} = 0{,}32 \left(\frac{38{,}170}{100} \right)^2 = 0{,}047 MPa$$

Distributor

$$\Delta P_{rasch} = 0{,}32 \left(\frac{38{,}170}{100} \right)^2 = 0{,}047 MPa$$

Filter

$$\Delta P_{rasch} = 0{,}08 \left(\frac{43{,}2}{100} \right)^2 = 0{,}015 MPa$$

$$\sum \Delta P_{rasch} = 0{,}024 + 0{,}01 + 0{,}01 + 0{,}047 + $$
$$+ 0{,}047 + 0{,}047 + 0{,}015 = 153000\ Pa$$

Total hydraulic losses

$$\Delta P_{\Sigma} = \sum \Delta P_{trl} + \sum \Delta P_{tr\xi} + \sum \Delta P_{rasch} =$$
$$= 172481 + 41744 + 153000 = 367225\ Pa$$

We determine the total leakage in hydraulic devices

$$\sum Q_{ut} = 0{,}13 + 0{,}15 + 0{,}15 + 0{,}15 + 0{,}15 = 0{,}73\ {}^{l}\!/\!_{\min}$$

We clarify the performance of the pump. Lift cylinder

$$P_{s1} = \frac{4 \cdot P_1}{\pi \cdot D_1^2 \cdot \eta_{g.m.}} = \frac{4 \cdot 88713}{\pi \cdot 0{,}09^2 \cdot 0{,}986} = 14{,}143 MPa$$

Tilt cylinder

$$P_{S2} = \frac{4 \cdot P_{2\max}}{\pi (D_2^2 - d_2^2) \eta_{g.m.}} =$$
$$= \frac{4 \cdot 27303}{\pi (0{,}053^2 - 0{,}018^2) 0{,}946} = 14{,}788 MPa$$

Determine the pressure in the pressure line of the pump

$$P_N = P_S + \Delta P_{\Sigma} = 14{,}788 + 0{,}367225 = 15{,}155 MPa$$

Determine the required pump flow

$$Q_{nomp} = Q_1 + \sum Q_{ut} = 38{,}170 + 0{,}73 = 38{,}9\ {}^{l}\!/\!_{\min}$$

Т.К. $P_N < P_{NOM}$ и $Q_{potr} < Q_{nas}$ => hydraulic equipment selected correctly.

Pressure setting relief valve

$$P_{pr.kl} = 1{,}06 P_N = 1{,}07 \cdot 15{,}155 = 16{,}2 MPa$$

Calculation of power and drive efficiency

Net power

For lifting gear

$$N_{P1} = P_1 \cdot \upsilon_1 = 88{,}713 \cdot 0{,}1 = 8{,}871 kVt$$

For tilt mechanism

$$N_{P2} = 2 P_{2\max} \cdot \upsilon_2 = 2 \cdot 27{,}303 \cdot 0{,}12 = 6{,}553 kVt$$

Power spent

$$N_{pr} = \frac{Q_{nas} \cdot P_N}{\eta_N} = \frac{0{,}00072 \cdot 15{,}155 \cdot 10^6}{0{,}78} = 13{,}989 kVt$$

Total drive efficiency

$$\eta_{pr} = \frac{N_{P1}}{N_{pr}} \cdot 100\% = \frac{8{,}871}{13{,}989} \cdot 100\% = 63{,}414\%$$

CONCLUSION

The developed hydraulic system allows you to save fuel due to the hydraulic pump under hydraulic pressure generated by the raised load.

As a result of the calculations, the following values of the system parameters were established: load on the lift hydraulic cylinder P_1=88713 N, rod diameter d_1=0.058 m, pump capacity Q_{H0}=72 l/min.

The development of a multi-purpose machine with a train of quick-detachable working bodies based on domestic TTZ tractors and on the basis of specific theoretical and experimental studies that ensure the construction of a reliable, resource-saving machine, with competitive indicators of metal consumption, energy consumption and labor costs, is an actual work and is important.

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