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Research paper

Signature Analysis of the Hydraulic Accumulators when Using in Energy Recovery Systems in Automobiles

Bogdan Korobko[1], Oleksiy Vasyliev[2], Ivan Rohozin[3*], Ievgen Vasyliev[4]

[1] *Poltava National Technical Yuri Kondratyuk University, Ukraine*
[2] *Poltava National Technical Yuri Kondratyuk University, Ukraine*
[3] *Poltava National Technical Yuri Kondratyuk University, Ukraine*
[4] *Poltava National Technical Yuri Kondratyuk University, Ukraine*
**Corresponding author E-mail: ria.workbox@gmail.com*

Abstract

The design of a car, which includes the hydrostatic transmission with a kinetic energy recovery system oriented at large-load freights under the conditions of frequent acceleration and repeated brakings has been proposed in this article. Rationale for choosing hydro accumulators for a dream-car (prototype vehicle) has been fulfilled, as well as energy storage battery capacity and the influence of these batteries on the car performance in different operating modes has been carefully analyzed.

Keywords: *energy recovery, hydraulic accumulator, hydrostatic transmission.*

1. Introduction

The kinetic energy recovery system is the technology that allows to accumulate the kinetic energy that occurs on application of brakes and provides an opportunity to use it for additional acceleration at the very moment of acceleration. In recent years, this system has been widely used in Grand Prix racing. Sports racing in this vehicle class has long been considered the breaking-in of innovative technologies for further implementation in everyday life. The kinetic energy recovery system was introduced into Grand Prix racing not only for visual appeal of these automobile races, but also for promoting hybrid environmentally friendly technologies [1, 2, 3]. It is expected that manufacturers will be further able to adapt the system to productionl cars.

There are three types of kinetic energy recovery systems: electric, mechanical and hydraulic.

The electric system is the most common. It is this technology that is being developed in in Grand Prix racing. This type uses the electric generator, connected to the racing car's driveline. It converts mechanical energy into electrical energy, which is accumulated in batteries.

The kinetic energy recovery system, apart from cars, is used in railway and electric transport. When electric locomotives, trams, trolleybuses and other automotive vehicles brake, electric generators accumulate energy for further use.

There are several significant disadvantages in the electric system: big losses in the transformation of mechanical energy into electric energy, high cost of batteries and their short battery service life. But, thanks to the relative simplicity of implementation of such technology, the electric version of the system came into Grand Prix racing in 2009 [4].

The mechanical kinetic energy recovery system has advantages based on the flywheel. When braking, the system receives kinetic energy on the flywheel. It spins up to several tens of thousands of turns of the wheel per minute. As soon as the system is turned on, the flywheel transfers energy to the driving axle of the car, and it receives the necessary acceleration. The most important benefit of the mechanical system is less energy loss. The point is that, when implementing such a kinetic energy recovery system, it will not be necessary to transfer one kind of energy into another. But such flywheels are high-tech and difficult-to-make [1, 2, 3].

The hydraulic system uses application of brakes in order to accumulate hydraulic pressure in hydraulic accumulators that can be transmitted to the drive wheels as necessary, and it can best be used for haulers and different heavy means of transport (commercial vehicles – lorries, trucks, motor-buses and etc.).

There are also systems which provide for derating (the power reduction of the engine), taking into account the fact that for a motor car and a city bus the peak output (maximum out-of-work) is required only to achieve intense acceleration/speeding up [5]. The car's working hour in a city takes place with a maximum out-of-work of less than 1%, therefore, for small motor-cars, it is possible to use power plants with an engine that has power of about 20 kW and energy accumulation during parking at the traffic light and during even movement at low speed. In the case of intense speeding-up (running-away) the energy, accumulated in the hydraulic accumulator, is added of the engine power.

Power plants with energy accumulation allow reducing potentially the fuel consumption by 38-50% [6]. Real fuel economy, of course, will be less, as it is necessary to take into account losses incurred during charging of hydraulic accumulators and during the implementation of the energy accumulated in them. This issue, as well as substantiation of rational parameters of the the hydrostatic transmission elements with a kinetic energy recovery system, require additional research.

The purpose of this work is to estimate the expediency of using hydraulic accumulators in the hydrostatic power drive of a car for the kinetic energy recovery during braking and idle running of the engine in comparison with the mechanical transmission.

The rationale for choosing hydraulic accumulators for a dream-car (prototype vehicle), research of the energy storage battery capacity, the influence of these batteries on the car performance in different accelerated modes and at braking, the comparison with the electrically driven system have been fulfilled in this article.

2. Main body

For the purpose of studying the kinetic energy recovery system with the hydrostatic transmission, we have chosen a wheeled semi-tractor for freight terminals (e.g., Figure 1), which belongs to a group of transport-traction cars with a power class of 130 horse power [7]. The maximum load on the fifth-wheel coupling is 64,000 N. The maximum weight of the towed semi-trailer, with a full loading, should not be more than 12000 kg. The tractive unit has a diesel engine MMZ D-245.9, produced at Minsk engine plant, with an engine yield of 100 kW (136 hp) with a crankshaft rotational speed of 2400 rpm. Its maximum rotational power is 460 N m with a crankshaft rotational speed of 1300 rpm. The tractive unit has hydrostatic transmission with an energy recovery system, which includes an adjustable pump, two high-frequency regulable hydraulic oil motors and two hydraulic accumulators (e.g. Figure 2). Hydraulic oil motors via a gear unit and a cardan drive transmit the torsion torque to the main drive gear. The authors of the article have also chosen the layout diagram of the cab's position over the engine and the wheel arrangement 4x2 with coupled wheels of the rear axle [8].

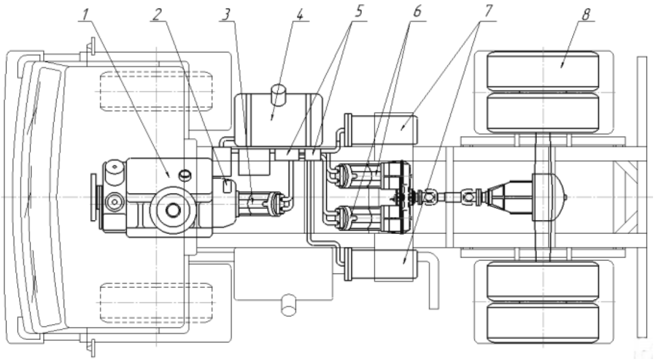

Fig.1: The layout diagram of a semi-tractor's transmission:

1 – engine; 2 – booster and control pump; 3 – hydraulic oil motor; 4 – the tank of the hydraulic drive system; 5 – hydraulic control valve blocks with back-pressure valves; 6 – hydraulic oil motors; 7 – hydraulic accumulators; 8 – the wheels of the rear axle.

The research was carried out on the basis of a hydraulic accumulator APX-16/32 with the following characteristics [9, 10]:

the gas chamber is filled with technical nitrogen of the 2nd grade according to all-Union State Standard GOST 9293 under the pressure up to 16 MPa;

power fluid - petroleum oils, which have the viscosity of 20 ... 500 cSt, at the temperature of 0 ... 60 ° C and at environmental temperature -30 ... + 60 ° C;

the power fluid purity grade is not more than 14 in accordance with the all-Union State Standard GOST 17216;

nominal pressure 32 MPa;

capacity 16 l;

dimensions 240x736 mm;

weight 106 kg;

technical conditions ТУ-053-1410-79.

Nitrogen pressure is adjustable and can vary from 1 to 16 MPa. Let us determine the approximate energy capacity of the battery at different initial gas pressures (without considering the thermal expansion of nitrogen). The energy given by the battery is equal to the work performed by it [5].

$$A = E_2 - E_1, \tag{1}$$

where A – work;

E_1, E_2 – initial and final energy, respectively.

$$A = F \cdot L, \tag{2}$$

where F – the force acting on the piston;

L – the distance passed by the piston under the influence of this force.

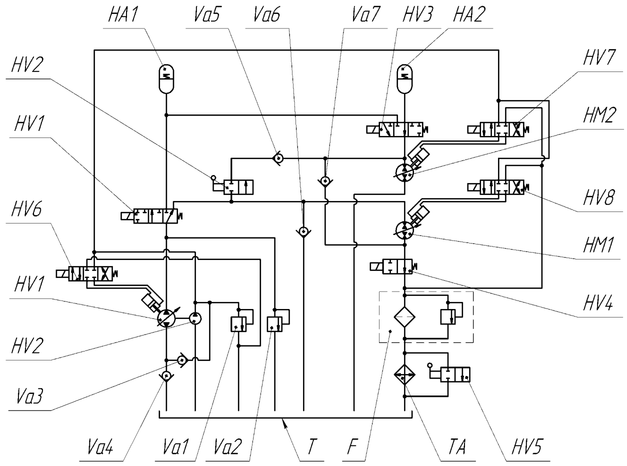

Fig.2: Hydraulic diagram of the hydraulic drive:

T – tank; HA1, HA2 – hydraulic accumulator APX-16/320; HM1,HM2 – A6V hydraulic oil motor; Va1, Va2 – safety valve 510.32; Va3 – back-pressure valve 61100; Va4-Va7 – back-pressure valve 61400; P1- pump A7V; P2 – pump NCH 6T-1; HV1-HV8 – hydraulic valve; TA – heat-transfer apparatus KM6-CK-1; F – filter 1132-25.

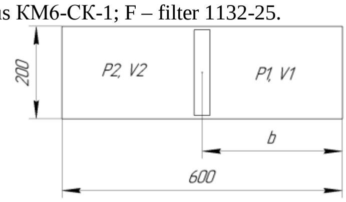

Fig.3: The scheme of the main power cylinder of the hydraulic accumulator:

P1, P2 – liquid and gas pressure respectively; V1, V2 – liquid volume and gas volume respectively; b – the distance between the piston and the bottom of the cylinder from the side of the gas; l = 600 mm – length of the working cylinder.

Force acting on the piston:

$$F = P \cdot S = P \cdot \frac{\pi \cdot d^2}{4}, \tag{3}$$

where S – the square of the piston's butt end;

d – the piston diameter.

The height of stroke of the piston from the charged state to the discharged state under the action of force F depends on the initial gas pressure in the cylinder. Let us find the energy capacity of the battery at different initial gas pressures.

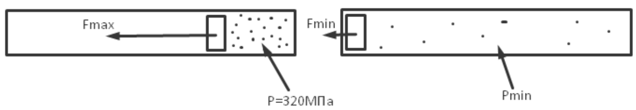

Fig.4: Gas pressure and force acting on the piston at its various positions

Having found the square A, we will be able to determine the work:

$$A = \int_b^l F_k db, \tag{4}$$

where F_k – the force acting on the piston from the gas side, depending on its position.

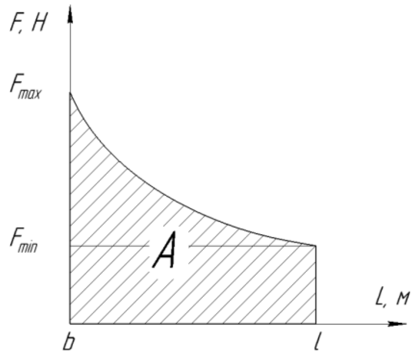

Fig. 5: The dependency diagram of the force acting on the piston from the path that it travelled, giving up energy:

$F_{\max}$ – maximum force acting on the piston from the gas side (when the battery is fully charged - at the pressure of 32 MPa); $F_{\min}$ – the minimum force acting on the piston (discharged battery, and it depends on the initial gas pressure); A – the square under the schedule, equal to the work performed by the piston; l – length of the working cylinder.

During the operation of the hydraulic accumulator, inits gas section, there takes course the polytropic process (adiabatic process). The equation of the polytropic process [11] is as follows:

$$PV^n = const\,, \tag{5}$$

where n – polytropic index, for the adiabatic process:

$$n = \frac{c_p}{c_v}\,, \tag{6}$$

where $c_p = 1050\dfrac{J}{kg \cdot K}$ – heat capacity of nitrogen at constant pressure;

$c_v = 750\dfrac{J}{kg \cdot K}$ – heat capacity of nitrogen at a constant volume.

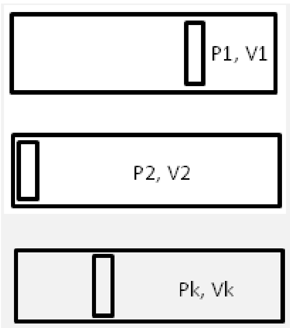

Fig.6: Nitrogen pressure and nitrogen volume at various piston positions

Polytropic equation for battery discharge process:

$$P_k \cdot V_k^n = P_1 \cdot V_1^n\,, \tag{7}$$

where P_k, V_k – the gas pressure and gas volume correspondingly at the position of the piston k (e.g. Figure 6),

$$P_k = \frac{P_1 \cdot V_1^n}{V_k^n}\,, \tag{8}$$

$$V_k = b_k \cdot S\,, \tag{9}$$

where b_k – the distance from the bottom of the cylinder to the piston from the side of the gas at the position of the piston k.

$$P_k = \frac{F_k}{S}\,, \tag{10}$$

$$F_k = \frac{P_1 \cdot V_1^n}{b_k^n \cdot S^{n-1}}\,, \tag{11}$$

Then work expansion of nitrogen is as follows:

$$\int_b^l \frac{P_1 \cdot V_1^n}{b_k^n \cdot S^{n-1}} db\,, \tag{12}$$

But the distance between the piston and the bottom of the cylinder from the gas side b in the charged battery will be different depending on the initial gas pressure. This is due to the fact that:

$$P_1 \cdot V_1^n = P_2 \cdot V_2^n\,, \tag{13}$$

In such a case we will obtain:

$$b = \frac{l}{\left(\dfrac{P_1}{P_2}\right)^{\frac{1}{n}}}\,, \tag{14}$$

Consequently, the work done by the battery during discharging is equal to:

$$\int_{\frac{l}{\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}}}^{l} \frac{P_1 \cdot V_1^n}{b_k^n \cdot S^{n-1}} db\,, \tag{15}$$

The calculations of the work performed by the hydraulic accumulator are given in table 1.

Table 1: Energy capacity of the battery depending on the initial gas pressure - $A(E)$ – the work performed by the piston during discharge is equal to the energy capacity of the battery)

P_1, MPa	P_2, MPa	b, m	V_1, m^3	V_2, m^3	$(V_1)^n$, $m^{4,2}$	k	A(E), J
32	1	0,050471	0,001585	0,01726	0,00012	15358,2	-973397
32	2	0,082807	0,0026	0,01624	0,00024	30716,39	-941997
32	3	0,110623	0,003474	0,01537	0,00036	46074,59	-910597
32	4	0,135859	0,004266	0,01457	0,00048	61432,78	-879198
32	5	0,159334	0,005003	0,01384	0,0006	76790,98	-847798
32	6	0,181496	0,005699	0,01314	0,00072	92149,18	-816398
32	7	0,202622	0,006362	0,01248	0,00084	107507,4	-784998
32	8	0,222899	0,006999	0,01184	0,00096	122865,6	-753598
32	9	0,242463	0,007613	0,01123	0,00108	138223,8	-722198
32	10	0,261415	0,008208	0,01063	0,0012	153582	-690798
32	11	0,279831	0,008787	0,01005	0,00132	168940,2	-659398
32	12	0,297775	0,00935	0,00949	0,00144	184298,4	-627998
32	13	0,315296	0,0099	0,00894	0,00156	199656,5	-596598
32	14	0,332435	0,010438	0,0084	0,00168	215014,7	-565198
32	15	0,349228	0,010966	0,00787	0,0018	230372,9	-533799
32	16	0,365704	0,011483	0,00736	0,00192	245731,1	-502399

The shaft speed of the hydraulic oil motor can be determined by the rotating frequency of the tractive unit's wheel:

$$n_{hm} = n_{k\max} \times u_{gen}\,, \tag{16}$$

the torsion torque, that the hydraulic oil motor can develop:

$$M_{hm} = \frac{P_{hm} \times V_{hm} \times h_{hm}}{2 \times p}\,, \tag{17}$$

where P_{hm} – the differential pressure on the hydraulic oil motor, MPa, will be determined after selecting the pump;

V_{hm} – swept-volume capacity of the hydraulic oil motor, cm^3;

η_{hm} – hydromechanical output-input ratio of the hydraulic oil motor, $\eta_{hm} = 0{,}97$.

Relative swept-volume capacity (regulation characteristic) of the adjustable hydraulic oil motor:

$$e_{hm} = \frac{V_{hm}}{V_{hm\max}}, \tag{18}$$

where $V_{hm\max}$ – the maximum swept-volume capacity of the hydraulic oil motor, $V_{hm\max}$ = 191.8 cm^3.

The hydraulic fluid consumption in the hydraulic oil motor is calculated on the basis of providing the required maximum speeds:

$$Q_{hm} = V_{hm} \times n_{hm\ \max}, \tag{19}$$

where V_{hm} – swept-volume capacity of the hydraulic oil motor at the maximum angular velocity, that is, the minimum swept-volume capacity at which smooth operation of the hydraulic machine is possible. We will be able to obtain the swept-volume capacity for the determination of hydraulic fluid consumption by dividing the maximum swept-volume capacity of the hydraulic oil motor by the adjustment range, which is 3.5:

$$V_{hm} = \frac{191{,}8 \times 10^{-6}}{3{,}5} = 54{,}8 \times 10^{-6}\ \ \text{m}^3, \tag{20}$$

The regulation characteristic then will be as follows:

$$e_{hm} = \frac{54{,}8}{191{,}8} = 0{,}286, \tag{21}$$

$$Q_{hm} = 54{,}8 \times 10^{-6} \times 5000 = 0{,}274\ \ \text{m}^3/\text{min.}, \tag{22}$$

This hydraulic fluid consumption will be maximal, since it has been determined under the conditions when the power output of the hydraulic oil motor is maximal (motion with maximum achievable speed). The the power output of the hydraulic oil motor begins to decrease when its swept-volume capacity becomes maximum and the pressure reaches its limit value. This is due to the fact that the consumption of the hydraulic oil motor is reduced (its angular velocity decreases), and the pressure and swept-volume capacity are no longer increasing.

According to the known pump rotational speed and necessary supply of hydraulic fluid (it is determined by the consumption of the hydraulic oil motor). The swept-volume capacity, which the pump should have under certain operating conditions, will be as follows:

$$M_p = \frac{Q_{hm}}{n_p \times h_{vp} \times h_{vhm}}, \tag{23}$$

where h_{vp} – the volume efficiency of the hydraulic pump, h_{vp} = 0,975;

h_{vhm} – volumetric efficiency of the hydraulic oil motor, h_{vgm} = 0,98.

When driving at maximum speed, swept-volume capacity of the hydraulic pump will be the following:

$$M_p = \frac{0{,}274}{2400 \times 0{,}975 \times 0{,}98} = 1{,}195 \times 10^{-4}\, m^3 = 119{,}5\, cm^3, \tag{24}$$

Under these conditions, the relative swept-volume capacity (regulation characteristic) of the adjustable hydraulic pump is equal to:

$$e_p = \frac{V_p}{V_{p\ \max}}, \tag{25}$$

where $V_{p\max}$ – the maximum swept-volume capacity of the pump, $V_{p\ \max}$ = 191.8 cm^3.

$$e_p = \frac{119{,}5}{191{,}8} = 0{,}623. \tag{26}$$

The actual pump flow is calculated by the formula:

$$Q_p = V_p \times n_p \times h_{vp}, \tag{27}$$

$$Q_и = 119{,}5 \times 10^{-6} \times 2400 \times 0{,}975 = 0{,}2796\ \ \text{m}^3/\text{min.} \tag{28}$$

The hydraulic fluid pressure, which will be created by the selected pump on the move at maximum speed, is calculated as follows:

$$P_p = \frac{2 \times p \times h_{hm} \times M_r}{V_p}, \tag{29}$$

where M_r – the reduced moment to the pump shaft, according to the external characteristic of the engine in the full power condition M_r = 398 N·m [8];

h_{hm} – hydromechanical output-input ratio of the hydraulic pump, = 0,964.

$$P_p = \frac{2 \times 3{,}14 \times 0{,}964 \times 398}{119{,}5 \times 10^{-6}} = 20{,}2\ \ , \tag{30}$$

The differential pressure on the hydraulic oil motor will be lower due to the capacity losses in the hydraulic oil system, which are characterized by its efficiency h_{hs}. In the previous calculations it was accepted h_{hs} = 0,95 [5]. Then the differential pressure on the hydraulic oil motor is calculated by the following formula:

$$P_{hm} = P_p \times h_{hs}, \tag{31}$$

Consequently, the differential pressure on the hydraulic oil motor when the tractive unit is moving with the maximum speed will be:

$$P_{hm} = 20{,}2 \times 0{,}95 = 19{,}2\ \ \text{MPa}, \tag{32}$$

Now we can find the torsion torque of the hydraulic oil motor under the same conditions:

$$M_{hm} = \frac{19{,}2 \times 54{,}8 \times 0{,}97}{2 \times 3{,}14} = 162{,}5\ \ \text{N} \times \text{m}, \tag{33}$$

Let us find the critical shaft speed of the hydraulic oil motor n_{hmc}, to which the pressure in the hydraulic system and, accordingly, the torsion torque on the hydraulic oil motor have the maximum values, and the frequency n_{hm} at gaining which the pump's swept-volume capacity, and, consequently, the regulation characteristic of the pump reach critical values. After that the pump operation is flawless and smooth, the pump flow and the discharge pressure level off. In the range of the angular velocity from n_{hmc} to $n\not{c}_{hm}$, the pump's regulation characteristic e_p changes with the constant regulation characteristic of the hydraulic oil motor e_{hm}, and the hydrostatic drive is capa-

ble of transmitting the maximum out-of-work, and from the n_{hm} to $n_{hm\,\max}$ there changes the regulation characteristic of the hydraulic oil motor e_{hm} for a constant regulation characteristic of the pump e_p.
The values of the considered rotational shaft speed of the hydraulic oil motor are determined by the following formulas:

$$n_{hmc} = \frac{60 \cdot N_p \cdot e_{hm} \cdot V_{hm\max} \cdot \eta_{vhm}}{P_{\max} \cdot V_{hm\ \max}}, \tag{34}$$

$$n_{hm} = \frac{60 \cdot N_p \cdot e_{hm} \cdot V_{hm\max} \cdot \eta_{vhm}}{P_{const} \cdot V_{hm\max}}, \tag{35}$$

where N_p – the pump power at engine performance with maximum out-of-work, N_p = 94 kW;
η_{vhm} – volumetric efficiency of the hydraulic oil motor, = 0,98;
$P_{\max}$, P_{const} – respectively, the maximum pressure and the pump delivery pressure at its established smooth operation with the constant volumetric displacement, are $P_{\max}$ = 40 MPa; P_{const} = 20.2 MPa;
$V_{hm\ \max}$ – maximum swept-volume capacity of the hydraulic oil motor, = 191.8 cm^3.

$$n_{hmc} = \frac{60 \cdot 94 \cdot 10^3 \cdot 0{,}98}{40 \cdot 10^6 \cdot 191{,}8 \cdot 10^{-6}} = 720\ \text{rpm}, \tag{36}$$

$$n_{hm} = \frac{60 \cdot 94 \cdot 10^3 \cdot 0{,}98}{20{,}2 \cdot 10^6 \cdot 191{,}8 \cdot 10^{-6}} = 1427\ \text{rpm}, \tag{37}$$

The characteristic curve of the hydrostatic transmission of the tractive unit are shown in figure 7.
Let us find the speed characteristics of the car:

$$V = a' = \left(\frac{F}{m}\right)' = \left(\frac{F_p - F_r}{m}\right)', \tag{38}$$

where a – acceleration of the car at a given time;
F – total force, acting on the car;
m – gross weight of the tractive unit;
F_p – driving force;
F_r – resisting force.

$$F_p = \frac{M_t}{r_d} = \frac{M_{hm} \cdot u_{gen} \cdot \eta_m}{r_d}, \tag{39}$$

where M_t – the torsion torque on the wheels;
r_d – the dynamic radius of the wheel mover;
η_m – mechanical output-input ratio of the drive arrangement from the HM (hydraulic oil motor) to the wheels.

$$M = \left(\frac{M_{hm} \cdot u_{зgen} \cdot \eta_m - F_r \cdot r_d}{m \cdot r_d}\right), \tag{40}$$

Using the graph (fig. 8), we can find the dependence of speed on time:

$$t = \frac{V - V_0}{a}, \tag{41}$$

where t is the time of changing speed from V to V_0.

The torsion torque on the shaft of the hydraulic oil motor, which perates from the hydraulic accumulator, will be as follows:

$$M_{hma} = \frac{P_{hm} \cdot V_{hm} \cdot \eta_{hm}}{2 \cdot \pi}, \tag{42}$$

Taking into account that $P_{гм} = \frac{P_1 \cdot V_1^m}{V_k^n}$, $V_1 = \frac{V_2}{\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}}$, $V_{hm} = e_{hm} \cdot V_{hm\max}$ we will get:

$$M_{hma} = \frac{P_2 \cdot V_2^n \cdot e_{hm} \cdot V_{hm\max} \cdot \eta_{hm}}{2 \cdot \pi \cdot V_k^n}, \tag{43}$$

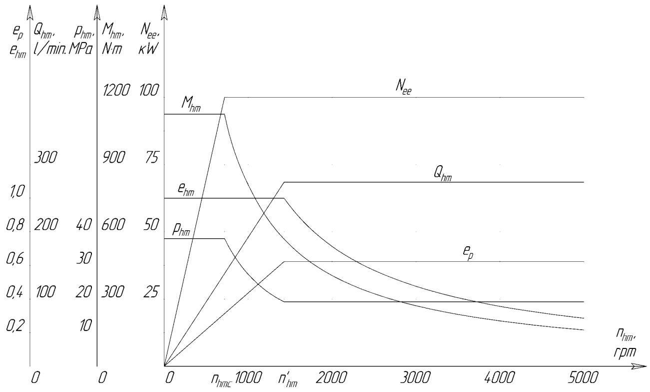

Fig.7: Initial characteristics of the hydrostatic transmission of the tractive unit with an adjustable pump and, regulable hydraulic oil motor:

N_{ee} – effective power of the explosion engine, M_{hm} – the torsion torque on the shaft of the hydraulic oil motor, P_{hm} – pressure difference on the hydraulic oil motor, Q_{hm} – consumption of the hydraulic oil motor, e_{hm} – control factor of the hydraulic oil motor e_p – the pump's control factor.

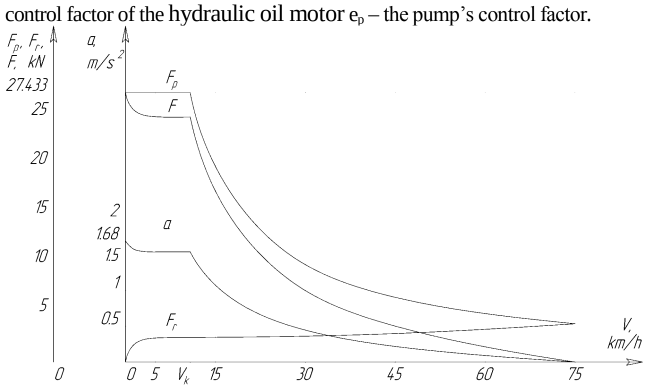

Fig. 8: Dependence of acceleration from speed

One hydraulic accumulator is fully charged and discharged at 7.33 turns of the tractive unit wheel (with the capacity of a hydraulic accumulator - 16 liters). That is, the gas volume in the battery is proportional to the path that the car has traveled since the start of the movement with the drive from the battery or since the beginning of regenerative braking.
The storage battery capacity depends on the initial gas pressure.

$$V_{ha} = (l - b) \cdot S, \tag{44}$$

where S – the square of the piston's butt end;
l – the length of the working cylinder;
b – the length of the gas compartment in the charged accumulator.
Number of drive shaft turns of the hydraulic oil motor until it is fully discharged [5]:

$$n_{hmg} = \frac{V_{ha}}{e_{hm} \times V_{hm\max}}, \tag{45}$$

The distance, covered by the car for one discharge of the accumulator:

$$L = 2 \times p \times r_d \times \frac{n_{hmg}}{u_{gen}}, \tag{46}$$

Then we will get the following results, which are given in tables 2 – 5 and the graphic dependence in figure 9.

Table 2: The distance, covered by a tractive unit on a single charge of a hydraulic accumulator, depending on the initial gas pressure and the hydraulic oil motor's control factor, without taking into account the strength of the resistance

P_2, MPa	b, m	l, m	S, m^2	V_{ha}, m^3	L, m		
					e_{hm1}=1	e_{hm2}=0,643	e_{hm3}=0,286
1	0,05047	0,6	0,0314	0,01726	22,4800851	34,9690213	78,680298
2	0,08280	0,6	0,0314	0,01624	21,1573085	32,9113687	74,05058
3	0,11062	0,6	0,0314	0,01537	20,019406	31,1412982	70,067921
4	0,13585	0,6	0,0314	0,01457	18,9870675	29,5354383	66,454736
5	0,15933	0,6	0,0314	0,01383	18,026736	28,0415894	63,093576
6	0,18149	0,6	0,0314	0,01314	17,1201442	26,6313354	59,920505
7	0,20262	0,6	0,0314	0,01248	16,2559321	25,2870055	56,895762
8	0,22289	0,6	0,0314	0,01184	15,4264167	23,9966482	53,992458
9	0,24246	0,6	0,0314	0,01124	14,6260897	22,751695	51,191314
10	0,26141	0,6	0,0314	0,01063	13,850829	21,545734	48,47790
11	0,27983	0,6	0,0314	0,01005	13,0974464	20,373806	45,84106
12	0,29777	0,6	0,0314	0,00949	12,3634103	19,2319716	43,271936
13	0,31529	0,6	0,0314	0,00894	11,6466675	18,1170384	40,763336
14	0,33243	0,6	0,0314	0,00840	10,9455229	17,026369	38,30933
15	0,34922	0,6	0,0314	0,00787	10,258556	15,9577537	35,904946
16	0,36570	0,6	0,0314	0,00736	9,5845612	14,9093174	33,545964

Table 3: The torsion torque on the shaft of the hydraulic oil motor, depending on the initial gas pressure, at the hydraulic oil motor's control factor $e_{hm} = 1$

V_k, m^3	M_{hma}, N×m			
	P_2=1 MPa	P_2=5 MPa	P_2=9 MPa	P_2=16 MPa
0,001585	947,8355	–	–	–
0,005003	189,6056	948,0282	–	–
0,007613	105,3406	526,703	948,0647	–
0,011483	59,2511	296,2555	533,2595	948,0176
0,013935	45,1866	225,933	406,679	722,9855
0,016388	36,01222	180,0611	324,1098	576,1956
0,01884	29,62515	148,1258	266,6262	474,0025
Total revolutions	89,96456	72,1424	58,53313	38,3571

Table 4: The torsion torque on the shaft of the hydraulic oil motor, depending on the initial gas pressure, at the hydraulic oil motor's control factor e_{hm} = 0.643

V_k, m^3	M_{hma}, N·m			
	P_2=1 MPa	P_2=5 MPa	P_2=9 MPa	P_2=16 MPa
0,001585	609,3226	–	–	–
0,005003	121,8893	609,4467	–	–
0,007613	67,71893	338,5948	609,4706	–
0,011483	38,08998	190,45	342,81	609,4399
0,013935	29,04851	145,2426	261,4367	464,7764
0,016388	23,15071	115,7536	208,3564	370,4114
0,01884	19,04473	95,22371	171,4027	304,7159
Total revolutions	139,9449	112,2215	91,05153	59,6666

Table 5: The torsion torque on the shaft of the hydraulic oil motor, depending on the initial gas pressure, at the hydraulic oil motor's control factor e_{hm} = 0,286

V_k, m^3	M_{hma}, N·m			
	P_2=1 MPa	P_2=5 MPa	P_2=9 MPa	P_2=16 MPa
0,001585	270,8101	–	–	–
0,005003	54,17304	270,8652	–	–
0,007613	30,09731	150,4866	270,8758	–
0,011483	16,92889	84,64443	152,36	270,8622
0,013935	12,91046	64,55228	116,1941	206,5673
0,016388	10,28921	51,44603	92,60286	164,6273
0,01884	8,46433	42,32165	76,17897	135,4293
Total revolutions	314,876	252,4984	204,866	134,2498

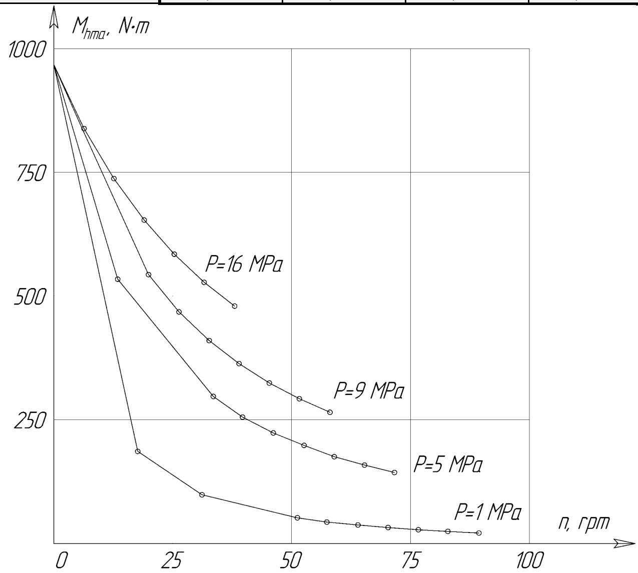

Fig. 9: The torsion torque on the shaft of the hydraulic oil motor, depending on number of revolutions from the beginning of the movement at different initial gas pressures and at various control factors of the hydraulic oil motor e_{hm} = 1

Although it has benn found out that at the initial gas pressure P_1 = 1 MPa the battery has the largest power capacity, it is evident from the graphs (fig. 9) that in this case the significant amount of time on the hydraulic motor shaft is small, as compared to cases where the initial gas pressure is higher. When operating from the battery, the moment on the motor shaft is larger than when it is running from the pump, but it occurs for a much shorter period of time. To prevent overloading of parts and for the purpose of the efficient use of battery power, the parameters e_{hm} (the hydraulic oil motor's control factor) and P_1 (initial gas pressure in the battery) must be selected from the middle of the range of values.

In the application of regenerative braking, battery charging takes place under the same laws as battery discharging. And the dependences of torsion torques on the shaft of the hydraulic oil motor will be reversed to these dependencies of the discharging process.

In the hydraulic oil system of the car there are two hydraulic accumulators and two hydraulic oil motors. During acceleration (running-away) without hydraulic accumulators, the liquid is fed from the pump to the motor. Initially, the pump control coefficient is e_p =0.286, which corresponds to its minimum working volume, and the hydraulic oil motor's control factor is e_{hm} = 1 (max working volume). With the increase in speed e_p increases too, and when e_p = 0.623 – e_{hm} , it begins to decrease. At e_p = 0.623 and e_{hm} = 0.286, the tractive unit reaches the maximum speed of 75 km / h. The hydraulic circuit also allows you to drive two hydromotors from the pump simultaneously, doubles the torsion torque on wheels and reduces the speed twice.

At acceleration, the parallel work of HM1 from the pump is possible, as well as the work of HM2 - from the battery. In this case, torsion torques will be summarized. After discharging of one HA, the automatic system connects to the second one. It is also possible to work sequentially - to move from a hydraulic accumulator and accelerate further from a pump drive. If you need a large torsion torque on wheels, two motors are connected to the battery simultaneously. When the battery is discharged, the pressure in it falls, so the control factor is initially the minimum (e_{hm} = 0.286) and it is increased to provide a regular torsion torque on the wheels.

In the application of regenerative braking, the law that changes the torsion torque on the shaft of the hydraulic motor operating from the battery is the same as in acceleration. Depending on the brake pedal pressure and the battery state of charge, the system automatically adjusts the working volumes of the hydromotors (pumps) and the degree of participation of the braking system. First, the working volume of one hydromotor (pump) increases, then of another, then the braking system is turned on. The decelerating torque increases with increasing the pump capacity per revolution, increasing the pressure in the hydraulic accumulators and on the great wheels of the car.

An important characteristic of the developed car is that its engine operates in optimal modes without frequent accelerations and temporary stops. This reduces the toxicity of the exhaust gases. The engine lifetime increases, and the costs of its maintenance and repair are reduced.

In accordance with the characteristics of the tractive unit's work in the use of the above mentioned scheme, more smooth motion of the machine from the place is provided as well as the continuous increase in speed, which occurs without interruption in torque delivery. It is quite convenient to work at low speeds. Handling the tractive unit becomes easier ad more convenient thanks to the automation system of gear ration of transmission and parameter checkout of the engine performance.

3. Conclusions

For the hydraulic accumulator APX-16/32 we have defined its energy storage capacity and its influence on the work of the tractive unit with hydrostatic transmission.

The initial gas pressure in the hydraulic accumulator APX-16/32 is adjusted to 16 MPa. The lower the initial pressure, the higher the energy storage capacity of the battery is, but the efficient use of energy is less. Consequently, the initial gas pressure should be selected from the middle of the range of values possible for APX-16/32 – 5-9 MPa.

To ensure the most efficient use of the energy recovery system, it is necessary to use hydroaccumulators actively. They should not be fully charged for a long time, because in this case, less energy will "return" into the system.

When using electrical energy recovery system the energy capacity of rechargeable lithium-ion batteries is two times higher, but their use is limited by the fact that their durability is much smaller and the cost is more than 30-50 times [12, 13] higher compared with hydraulic accumulators. The use of a hydrostatic power drive with hydraulic accumulators is appropriate. Such a kinetic energy recovery system has advantages over a mechanical system and over an electric system.

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