Development of energy efficient hydrostatic drives with energy recovery

Rozwój energooszczędnych napędów hydrostatycznych z odzyskiem energii

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Recovery of kinetic energy for its subsequent storage in hydraulic accumulators may be performed due to employment of **regenerative braking. It is due to two-directional energy flow** that the whole cycle of vehicle movement is made possible. Dynamic models, simulation results, and experimental tests of a electro-hydraulic hydrostatic systems with secondary control are presented, which can be used in hydraulic hybrid powertrains. Selection of control parameters of the secondary unit has the decidedly key meaning for improvement of **efficiency of the hydraulic hybrid drives. Today's hydrostatic** drives can handle much more power per unit mass than electric machines, which implies a considerable advantage of series hydraulic hybrids.
KEYWORDS: energy recuperation, energy accumulation, recuperative braking, hydrostatic powertrains, hydraulic hybrid drives

For many years, the search for energy-efficient solutions in hydrostatic drives of passenger vehicles, municipal and military vehicles, buses, work and construction machinery, lifting equipment and others has been ongoing. These solutions include: vehicle weight reduction, aerodynamic body design, use of electronic ignition or injection equipment, energy recovery and accumulator systems and hybrid drives. Such solutions lead to reduced noise and reduced fuel consumption and emissions to the environment: NOx, carbon monoxide and hydrocarbons (CO/HC) and particulate matter (PM).

In conventional vehicle drives, braking energy is irretrievably lost and the internal combustion engine must cover the power demand throughout the entire vehicle cycle. This clearly indicates the need to include in the vehicle movement recovery (recovery) and storage (accumulation) of braking energy, and then use it to accelerate the vehicle. In such drives, an internal combustion engine can be used to cover the average power requirement during traffic. This reduces the power of the engine cooling system. The main goal of hybrid vehicles is to improve fuel efficiency and reduce emissions for the benefit of the environment.

After the tightening of emission standards in 2000 and fuel price increases, the advantages of such drives have been recognized. Hybrid vehicles are classified based on the configuration of the power transmission system and the method of energy accumulation. The first hybrid electric vehicle, the Lohnerporsche (HEV), was built in 1900. Originally, the hybrid technology was designed for military vehicles, lorries, vans and buses. The Hydro-Bus concept was introduced in Hanover in the early 1980's and was subsequently researched at the Technical University of Lodz [16]. Loading and unloading equipment and construction machinery for short runs (with frequent stoppages), Bosch Rexroth has developed a hybrid hydrostatic regenerative braking system (HRB) [13].

The history of modern hybrid technology in cars began about 30 years ago – examples are hybrid drives used in the Toyota Prius (1997) and Honda Insight (1999). The research also included hydraulic hybrid drives in SUVs such as the Hummer H3, Nissan Titan, Pathfinder, Dodge Durango, Ford Explorer and GMC Yukon. One of the first hybrid parallel hybrid drives introduced into urban vehicles was the Parked Hannifin CBED (cumulo brake energy drive) system. On the other hand, the company's first hybrid hydrostatic drive was the CHD (cumulo hydrostatic drive) system. The latest hybrid hydraulic drive solution was developed by Bosch Rexroth with PSA Peugeot Citroën. This system, called hybrid air, was used in the Peugeot 2008 car, which was presented at the Geneva Motor Show in 2013.

The use of a hybrid drive (HHD) in vehicles is most appropriate when the vehicle is moving in a repetitive cycle (acceleration – fixed travel – braking – stopping), while high power instantaneous power with high energy efficiency is possible. little energy loss. In many cases the traffic of urban, municipal and commercial vehicles is characterized by such cyclic driving. In energy-efficient hydrostatic drive systems, the pump is used as the primary source of energy, and the battery as a secondary power source and reverse drive unit with secondary control, which, depending on the direction of transmission, operates like a motor or pump. In such systems there is a two-way flow of energy: the energy is transferred from the pump and the battery to the reversible motor as the engine, and the energy recovered from the braking (during so-called regenerative braking) is transferred from a reversible drive unit operating as a pump to a hydraulic accumulator, where

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potential energy is collected. In closed hydrostatic systems, it is possible to control the rotational speed *n* and the *M* torque of the reversible drive unit over the whole range of the vehicle movement, i.e. during acceleration and deceleration while driving forward and backward. The schematic of the range of rotation speed control *n* and torque *M* of the reversible drive unit depending on the phase of vehicle movement is shown in fig. 1.

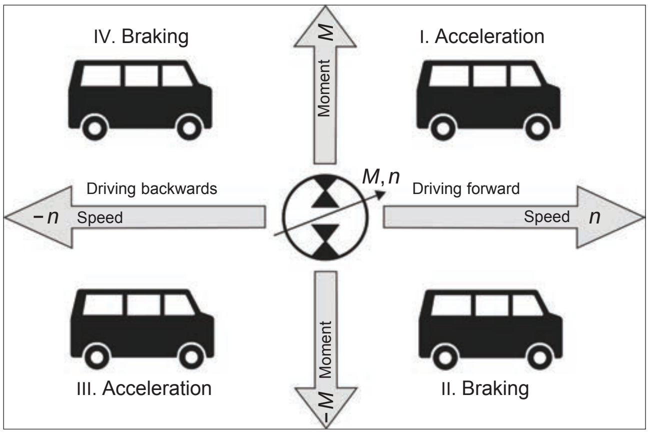


Fig. 1. Scheme of the hydrostatic control of the drive system depending on the phase of vehicle movement

Hydrostatic drives with recuperation and accumulative braking energy

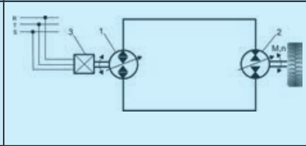
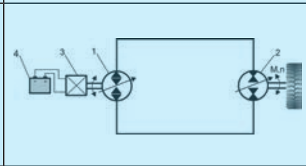
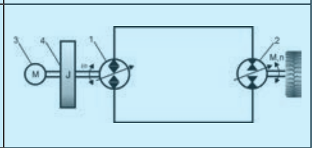
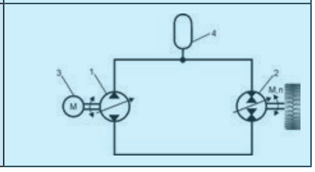
Recovering and accumulating (braking) energy (regenerative braking) are the most important features of hydrostatic systems used in hybrid drives. Table I shows basic closed hydrostatic systems with recuperation and energy transfer to traction and electrochemical, inertial and hydraulic batteries.

Hydrostatic drive with braking energy recuperation and return by the traction motor (working as a generator) to the overhead contact line can be used in traction vehicles (trams, trolley buses) used in public transport. Buses and trams with regenerative braking capability can reduce energy consumption in urban traction. In order to transmit braking energy to electric traction, two conditions must be fulfilled [2]: the voltage generated by the vehicle during braking must be higher than the traction power at the point of return and there must be a receiver of energy generated during braking, e.g. another traction vehicle that draws electricity.

Hydrostatic drive with regenerative braking energy and storage of energy in electrochemical accumulators is justified when these batteries have a high energy density (J/kg) [11]. To use the full capacity of the battery and to maintain high efficiency, however, a long charging time is required. Electrochemical batteries have a limited number of charges due to their degradation during the next charge.

The hydrostatic drive with regenerative braking energy and kinetic energy accumulation in the inertial battery consists of rotating the inertial element [9]. Additional inertia energy is used to increase the density of the inertia battery:

TABLE I. Composition of hydrostatic drives with recuperation and accumulation of braking energy

Drive type	Drive diagram
Recuperation and deceleration of braking energy to the overhead contact line: 1 - reversible pump-motor unit, 2 - reversible drive unit, 3 - traction motor (generator)	
Recovering and accumulating braking energy in the electrochemical battery: 1 - reversible pump-motor unit, 2 - reversible drive unit, 3 - motor (generator), 4 - electrochemical battery	
Recovery and accumulator braking energy in inertial battery: 1 - reversible pump-motor unit, 2 - reversible drive unit, 3 - electric motor, 4 - inertia	
Recovery and accumulator braking energy in the hydraulic accumulator: 1 - pump, 2 - reversible drive unit, 3 - electric motor, 4 - hydraulic battery	

special inlets and their housings, suitable bearings and auxiliary devices to reduce energy losses. Such accumulators pose risks associated with the high velocity of the rotating masses.

Hydrostatic drive with regenerative braking energy and potential energy accumulation in the hydraulic accumulator (bladder or piston) is suitable for storing the energy of the gaseous medium (nitrogen, air) [15]. Accumulation of energy takes place through gas compression, while energy transfer takes place by expanding the gas. Storage of energy in gas accumulators is only cost effective if the propulsion system is running in a suitable cycle and it is possible to use high momentary power when accelerating the vehicle. The efficiency of the gas battery depends on the thermodynamic changes and the heat losses.

Fig. 2 shows the energy density and power density of electrochemical, kinetic and hydraulic batteries. The hydraulic accumulators, compared to the lithium-ion electrochemical batteries used in electric cars, have a smaller

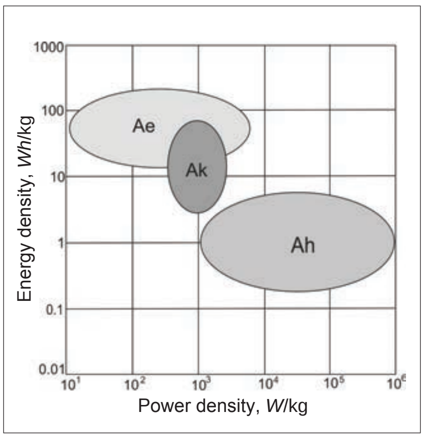


Fig. 2. Comparison of energy density and power density of electrochemical (*Ae*), kinetic (*Ak*) and hydraulic (*Ah*) accumulators according to [1]

capacity that translates to a smaller range of vehicles, but they charge much faster and can be more efficient. The data [18] shows that the efficiency of energy recovery in electric hybrid drives is η_{Rel} = 0.53 and in hybrid hydrostatic drives η_{Rhyd} = 0.69. The efficiency of regenerative braking energy for hydrostatic drive system with hydraulic accumulator is calculated from the formula [8]:

$$\eta_R = \eta_a \, \eta_s^2 \, \eta_n^2 \, \eta_m^2 \qquad (1)$$

where: η_a – efficiency of hydraulic accumulator, η_s – efficiency of control system, η_n – efficiency of secondary drive unit, η_m – efficiency of mechanical system.

Hydraulic hybrid drives

The basic solutions for hydraulic (hydrostatic) hybrid drives are given in the Table II.

In the **parallel hydraulic hybrid** (PHH), the internal combustion engine is mechanically connected to the wheels of the vehicle. When high power is required, the internal combustion engine and the hydraulic motor can operate in parallel. During braking, the hydraulic motor operates like a pump and transmits the recovered braking energy to the high pressure hydraulic pump.

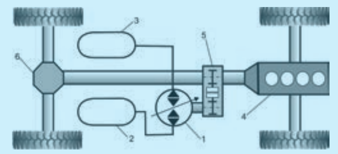
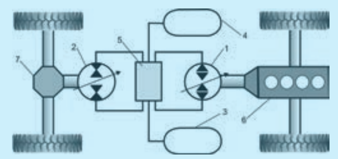
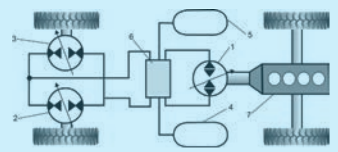
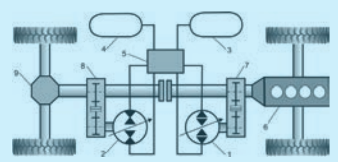
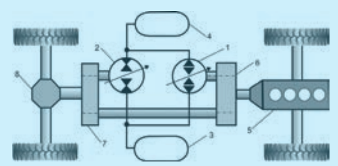
In the SHH **series hydraulic hybrid** engine, the internal combustion engine drives the hydraulic pump at its optimum speed, which transfers the hydraulic energy to the engine connected to the differential drive of the vehicle's wheel drive, and the excess energy is stored in the gas accumulators. If needed, the batteries can support the drive system. During braking, the engine operates like a pump and transfers the recovered energy to the high pressure hydraulic pump.

The **series full hydraulic hybrid** drive (SFHH) differs from the SHH system in that the wheels of the vehicle are directly connected to the hydraulic drive units.

Parker Hannifin's ASHH **advanced series hydraulic hybrid** engine [19] is based on the concept of power distribution (power-split concept). At low speed vehicles from 0 to 65 km/h use hydraulic drive. At speeds of 65 to 100 km/h direct mechanical drive is used and the hydraulic drive is disengaged. These two types of drives provide high performance throughout the vehicle speed range. The hydraulic energy regeneration system takes over the kinetic energy of the vehicle from 65 to 0 km/h.

The **hybrid air** drive consists of two engines (combustion and hydraulic) and a hydro-pneumatic accumulator. This system provides the highest possible performance regardless of driving conditions [10]. The battery-assisted hydraulic system covers the increased energy demand during start-up and acceleration, but it can also completely replace the internal combustion engine. The design of this hybrid drive utilizes the boost effect, which is an additional instantaneous power boost with higher demands. In turn, the powersplit concept allows for different types of propulsion: short distances can only be achieved with a hydraulically powered hydraulic drive.

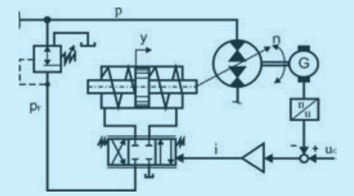
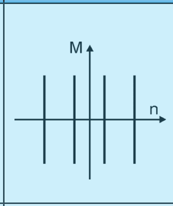
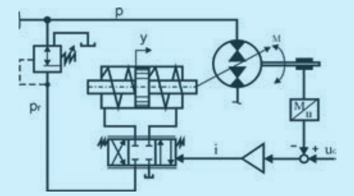
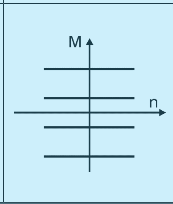
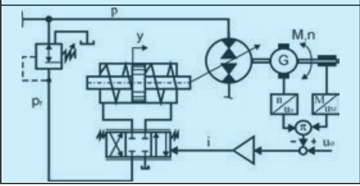
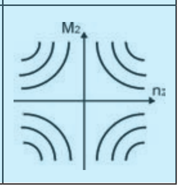
TABLE II. Hydrostatic hybrid drives

The type of hybrid drive	Hybrid drive scheme
Parallel Hydraulic Hybrid drive (PHH): 1 – reversible pump-motor unit, 2 – high-pressure rechargeable accumululator, 3 – low-pressure rechargeable accumululator, 4 – internal combustion engine, 5 – mechanical transmission, 6 – differential	
Series Hydraulic Hybrid drive (SHH): 1 – reversible pump-motor unit, 2 – reversible motor unit – pump, 3 – high-pressure rechargeable accumulator, 4 – low pressure accumululator, 5 – valve block, 6 – internal combustion engine, 7 – differential	
Full Hydraulic Hybrid drive (SFHH): 1 – reversible pump-motor unit, 2, 3 – secondary drive unit (motor-pump) 4 – high-pressure rechargeable accumulator, 5 – low-pressure accumululator, 6 – valve block, 7 – internal combustion engine	
Advanced Series Hydraulic Hybrid drive (ASHH): 1 – reversible pump-motor unit, 2 – secondary drive unit (motor-pump) 3 – low-pressure rechargeable accumululator, 4 – high-pressure rechargeable accumululator, 5 – valve block, 6 – internal combustion engine, 7, 8 – mechanical transmission, 9 – differential	
Hybrid Air drive: 1 – reversible pump-motor unit, 2 – secondary drive unit (motor-pump) 3 – low pressure pneumatic accumulator, 4 – high pressure accumulator, 5 – internal combustion engine, 6, 7 – Powersplit transmission, 8 – differential	

Constant-pressure hydrostatic systems

Constant- pressure hydrostatic dampers – used in energy-saving hydrostatic drives – are equipped with reversible drive units with secondary control. These vehicles are mounted on wheeled vehicles (city buses, work machines), tracked vehicles (tractors, work machines, tanks, military equipment), rail vehicles (shunting locomotives, mining vehicles) and stationary work equipment such as hoists and lifts. The advantage of hydraulic and electrohydraulic secondary control systems is the ability to control the reversible drive unit at a constant pressure *p* = const. in the hydrostatic system, which is not possible in conventional drives. Table III [12] shows schematics of basic electrohydraulic secondary control systems and their control characteristics for speed stabilization *n* = const, torque *M* = const. and power *P* = const. These mechanical parameters are converted to electrical parameters *u* (voltage from tachogenerator *G* and transducers) and *i* (current from amplifier to coil of control valve). In such control circuits the input signal is the voltage u_0 corresponding to the given mechanical parameter: n_0, M_0, P_0.

TABLE III. Electrohydraulic systems of secondary adjustment of reversing drive units

Type of adjustment	Diagram of adjustment system	Adjustment characteristicsji
n = const		
M = const		
P = const		

In constant pressure (p = const.) systems, the working volume V_r of the drive unit varies in the range of: $V_{rmin} < V_r < V_{rmax}$. The relationship between the torque M and the working volume V_r at the constant pressure difference Δp = $p - p_T \approx p$ = const. (where p_T – pressure in the tank corresponding to atmospheric pressure) is determined by the formula [7]:

$$M = \frac{p}{2\pi} V_r \tag{2}$$

$$M_{\max} = \frac{p}{2\pi} V_{r\max} \tag{3}$$

Comparing the formulas (2) and (3) taking into account the condition Δp = const., the torque relation is obtained from the change in working volume:

$$\frac{M}{M_{\max}} = \frac{V_r}{V_{r\max}} \tag{4}$$

The operating volume V_r of the reversible drive unit depends on the setting value y in the range: $y_{min} < y < y_{max}$, which is the adjustment parameter; then:

$$\frac{V_r}{V_{r\max}} = \frac{y}{y_{\max}} \tag{5}$$

After comparing the formulas (4) and (5), the torque dependence M is obtained from the setting y of the secondary drive unit:

$$\frac{M}{M_{\max}} = \frac{y}{y_{\max}} \tag{6}$$

The static characteristics $M/M_{max} = f(y/y_{max})$ of the secondary drive control set out in formula (6) for different differential pressure values Δp are shown in fig. 3. It is evident that the secondary control of the drive unit is characterized in that the change of torque M It is only due to the control of the working volume $V_r(y)$, that is, it is related to volumetric control.

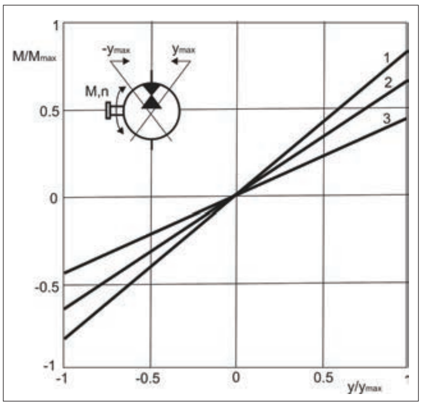


Fig. 3. Static characteristics $M/M_{max} = f(y/y_{max})$ of the secondary control of the drive unit at different values of pressure difference Δp: 1–21 MPa, 2–14 MPa, 3–7 MPa

Secondary adjustment also affects the change of flow rate q_v (engine capacity or pump performance), which is calculated from formula [6]:

$$q_v = V_r\, n \tag{7}$$

After substituting for formula (7) the working volume V_r of formula (2) gives:

$$q_v = V_r\, n = 2\pi \frac{n}{\Delta p} M = k_n M \tag{8}$$

where k_n is the rotational speed of the secondary drive unit:

$$k_n = 2\pi \frac{n}{p} \tag{9}$$

It should be noted that for $q_v > k_n M$ is so-called run, for $q_v < k_n M$ – self-braking of secondary drive unit. Such phenomena can occur with uncontrolled pressure changes.

Modeling of hydrostatic drive system

Model tests (simulation and physical) of a hydrostatic system with a secondary drive unit have been performed, which can be used in hybrid drives with recuperation and accumulative braking energy. This is a bidirectional flow of energy (power) that enables the conversion of hydraulic energy to the propulsion system during steady and accelerated motion, and the conversion of kinetic energy during braking from the propulsion system to the hydrostatic system and its storage in the hydraulic accumulator.

At this stage of simulation and physical model studies, a reversible drive control system in a closed hydrostatic system was analyzed, the functional diagram of which is shown in fig. 4. The selection of secondary control parameters of the reversible drive unit is crucial for improving the efficiency of the hydraulic transmission system. The hydrostatic propulsion system is composed of an adjustable 4-stroke piston pump type A4VG40, a reversible drive unit type A2V107, a high-pressure type bladder pressure hydraulic system and a hydraulic braking system HUO (brake and propulsion) of the pump A4VG40 and motor A6VM28 radiator and filters. The actuator 8 controlling the reversible drive unit 5 is supplied with a flow of liquid controlled by a proportionally proportional valve 9. The pressure p_r before the proportional valve 9 is set by a proportional control valve (reduction) 7. Fig. 4 also shows the transducers and control system [4].

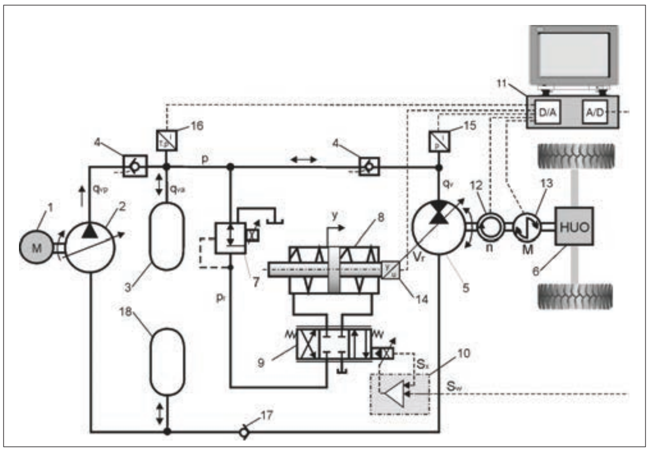


Fig. 4. Functional diagram of hydrostatic control station with high pressure hydraulic accumulator: *1* – electric motor (M), *2* – hydraulic pump with pressure control, *3* – high pressure hydraulic accumulator, *4* – check valve, *5* – reversible drive unit, hydraulic HUO, *7* – proportional pressure regulator, *8* – actuator, *9* – proportional valve, *10* – valve regulator, *11* – computer test and measurement cards, *12* – speed sensor, *13* – torque sensor, displacement sensor of actuator piston rod, *15* – pressure transducer, *16* – temperature and pressure transmitter in the battery, *17* – check valve, *18* – low pressure hydraulic accumululator

The purpose of the model study was to optimize the parameters of the hydrostatic system, the control parameters and the hydraulic accumulator. Secondary reversing drive unit should be capable of adjusting the rotational speed from the *M* torque to the load and duty cycle of the driveline, e.g. at acceleration and steady-state the reversible drive unit operates as a motor (M>0, q_v>0) pump (M<0, q_v<0). In the case of models and simulation tests included in works [3] and [5], a high-pressure hydraulic accumulator was included. Hydraulic accumulator modeling included changes in volume, temperature and pressure depending on the thermodynamic changes, but heat exchange with the environment was neglected (adiabatic shielding was used).

In the dynamic model of the electrohydraulic secondary control system of the reversible drive unit, the following state parameters are assumed: ω – angular speed of the drive unit, y – displacement of the actuator piston working as so- hydraulic balance, x – displacement of proportional valve control, Δp – differential pressure in the actuator, p – pressure in the hydrostatic system, T – gas temperature in the accumulator.

The dynamic model of the hydrostatic system with the secondary control of the drive unit and the hydraulic accumulator was recorded in the form of a system of differential equations:

$$\begin{cases} \dot{\omega} = a_1\, p\, y - a_2\, \omega - a_3\, M_h - a_4\, p \\ \ddot{y} = -a_{11}\, \dot{y} - a_{12}\, y - a_{13}\, \Delta p - a_{14}\, F_y \\ \ddot{x} = -a_{20}\, S_w - a_{16}\, \dot{x} - a_{17}\, x - a_{19}\, y + a_{18}\, p_r \\ \Delta\dot{p} = -a_{23}\, \Delta p - a_{22}\, \dot{y} + a_{21}\, f_1 \\ \dot{p} = \dfrac{1}{k_a}\left[q_v - q_{vp} + k_r \sqrt{p - p_r}\,\right] \\ \dot{T} = \dfrac{1}{\tau}\left(T_0 - T\right) \end{cases} \tag{10}$$

where: a_i – constant coefficients,

$$a_1 = \frac{1}{J}\frac{V_{r\max}}{2\pi y_{\max}},\quad a_2 = \frac{1}{J} f_\omega,\quad a_3 = \frac{M_h}{J},\quad a_4 = \frac{1}{J}\frac{V_r}{2\pi\,\eta_{hm}},$$

$$a_{11} = \frac{f_y}{m_y},\quad a_{12} = \frac{c_y}{m_y} + \frac{K_y}{y_{\max}},\quad a_{13} = \frac{A_y}{m_y},\quad a_{14} = \frac{1}{m_y},$$

$$a_{16} = \frac{f_x}{m_x},\quad a_{17} = \frac{c_x}{m_x},\quad a_{18} = \frac{A_x}{m_x},\quad a_{19} = \frac{c_{xy}}{m_x},\quad a_{20} = \frac{K_m}{m_x}$$

$$a_{21} = \frac{K_x}{C_{hc}},\quad a_{22} = \frac{2A_y}{C_{hc}},\quad a_{23} = \frac{2K_w}{C_{hc}},$$

f_1 – nonlinear function:

$$f_1 = (x_0 - x)\sqrt{\frac{1}{2}(p_r - \Delta p)} \tag{11}$$

Following parameters occur in fixed coefficients:

J – moment of inertia reduced on the reversible drive shaft,
f_ω – resistance on the reversible drive shaft due to viscous friction,
y_{max} – maximum setting position,
V_{rmax} – maximum working volume of the reversible drive unit,
M_h – braking unit moment
η_{hm} – hydro-mechanical efficiency of the drive unit,
m_e – mass reduced on the actuator piston,
f_y – viscosity coefficient of friction between piston and cylinder,
c_y – total spring stiffness of the actuator,
K_y – the gain factor determined from the characteristic $F_y = f(y/y_{\max})$,
F_y – force of the actuator,
A_y – the area of the actuator piston,
K_m – coupling gain factor from measuring system,
c_{xy} – the rigidity of the force feedback spring,
A_x – the area of the control slide,
m_x – weight of servo valve,
f_x – friction coefficient of viscous slide pair,
c_x – total servo spring servo stiffness,
K_x – flow factor, $K_x = C_d\,\pi\, d_x\sqrt{\frac{1}{\rho}}$,
x_0 – overlap of the control slide,
C_d – flow resistance coefficient,
d_x – the diameter of the control slide,
ρ – density of working fluid,
C_{hc} – hydraulic capacitance in the area between the valve and the actuator,
K – modulus of elasticity,
V_c – the volume of the area between the valve and the actuator,
K_w – leakage coefficient
τ – time constant of the accumulator,
T_0 – ambient temperature,
k_a – gas accumulator ratio,
p_r – pressure in the control system,
k_r – flow rate coefficient in the control valve,
q_{vp} – pump performance,
q_v – absorbency/efficiency of the reversible drive unit (6).

The values of the constants of the control system are: a_1 = 0,025 m/(Ns^2), a_2 = 97,006 1/s, a_3 = 0,001 m^2/(Ns^2), $a_4 = 1{,}073 \times 10^{-4}$ m^2/(Ns^2), a_{11} = 653,333 Ns/(kgm), $a_{12} = 1{,}28 \times 10^4$ N/(kgm), $a_{13} = 4{,}667 \times 10^4$ m^2/kg, a_{14} = 0,667 1/kg, a_{15} = 0,667 1/kg, a_{16} = 525 Ns/(kgm), $a_{17} = 7{,}5 \times 10^5$ N/(kgm), a_{18} = 0,0015 m^2/kg, $a_{19} = 2{,}85 \times 10^{10}$ N/(kgm), $a_{21} = 1{,}021 \times 10^{10}$ N$^{1/2}$/(m^{2}s), $a_{22} = 9{,}589 \times 10^9$ N/m^3, a_{23} = 45,205 1/s, a_{24} = 45,205 m^2.

During dynamic modeling, the initial pressure values p_0 = 15 MPa and torque of the braking unit M_{h0} = 25 Nm were introduced and the constant control parameters: pressure p_r = 4 MPa and actuator force F_y = 85 N.

During the simulation tests conducted in the Matlab/Simulink environment, the influence of hydrostatic drive parameters, expressed in coefficients a_i, on the stability of the control system after the introduction of the swing inducing signal S_w corresponding to the load torque of the reversible drive unit was analyzed. Two types of jumps have been reported: either strongly suppressed or unstable. During the simulation, only one factor was changed in turn – it was increased and decreased by 30%. Modification of the control system resulted mainly in the reduction or amplification of the amplitude and the period of pulsation in the transient state of the dynamic characteristics. After analyzing the influence of different coefficients on the stability of the hydrostatic propulsion system, it was found that some of these coefficients destabilize the course of the hydrostatic condition. These coefficients include: a_1, a_2, a_7, a_8, a_9, a_{18} and a_{19}. Other coefficients affect the course of particular system parameters, e.g. angular velocity ω(t), for which the coefficients a_5±30% and a_{20}±30% are most important. The dynamic characteristic $\omega(t)$ of a_5 and a_5±30% is shown in fig. 5, and the coefficients a_{20} and a_{20}±30% – in fig. 6. It follows that, after decreasing the coefficient a_5 by 30%, the overshoot δ_p by 0.46% and oscillation δ_o by 17.6%, while the increase of the coefficient a_5 by 30% decreased the overshoot δ_p by 0.3% and the oscillation δ_o by 6.33%. In turn, after the reduction of the coefficient a_{20} by 30% the oscillation increased by 6.8%, and after it increased by 30% the oscillation decreased by 4.18%.

Further model tests at the test bench consisted of the analysis of the angular velocity ω(t) of the secondary drive unit at various relative values of the S_w/S_{w0} control signals and various relative braking torque values M_h/M_{h0} as shown in fig. 7 and fig. 8.

Model tests of the hydraulic accumulator were also carried out during one virtual vehicle movement, for which: L = 1.5 km, T_c = 40 sec, braking time t_1 = 15%, stopping time t_2 = 25% , acceleration time t_3 = 20%, set driving time t_4 = 40%. For the accepted cycle, the pressure characteristic $p(t)$ and the temperature $T(t)$ in the vesicle gas cylinder (volume V_0 = 10 10^{-3} m^3) are shown in fig. 9 and fig. 10. From the presented characteristics, the time constant thermal τ = 10 s and pressure ratio p_2/p_1 = 4 (p_2 = 24 MPa, p_1 = 6 MPa). Actual characteristics $p(t)$ and $T(t)$ indicate the heat loss occurring in the hydraulic accumulator. The method of optimum gas accumulator selection is given in [6].

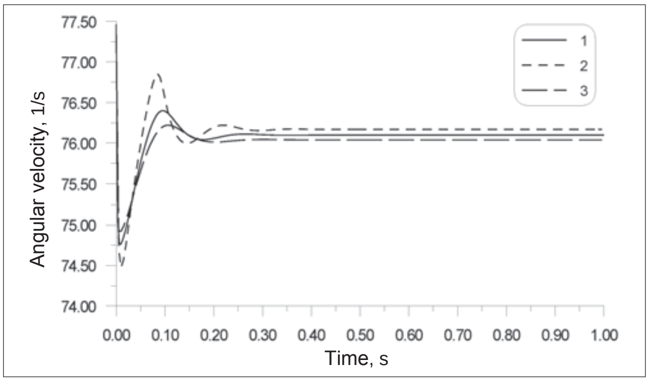


Fig. 5. Speed characteristics ω(t) for the coefficients: a_5 (1), a_5 – 30% (2), a_5 + 30% (3)

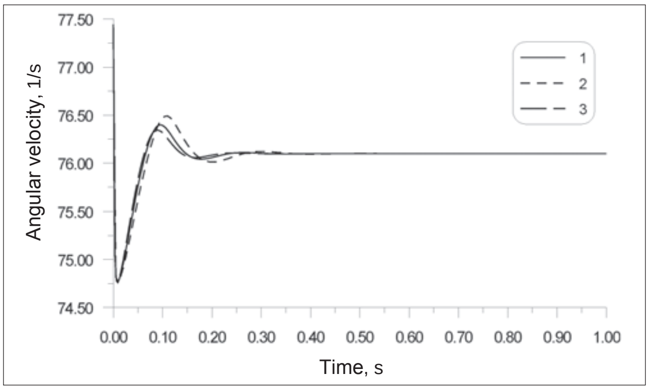


Fig. 6. Speed characteristics ω(t) for coefficients: a_{20} (1), a_{20} – 30% (2), a_{20} + 30% (3)

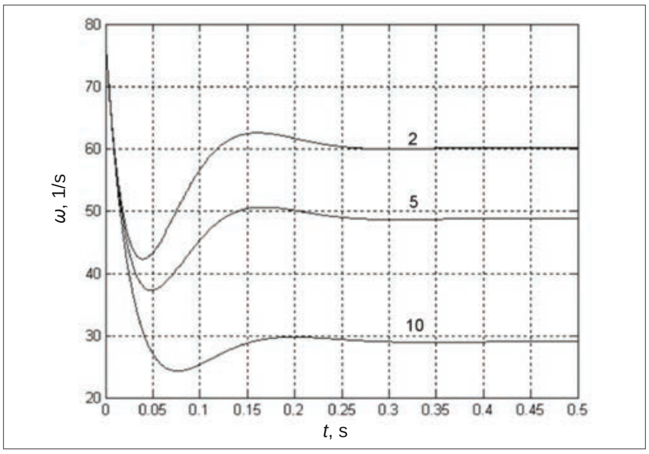


Fig. 7. Angular velocity ω(t) of the secondary drive unit for different values of S_w/S_{w0} = 2, 5, 10 and for M_h/M_{h0} = 1.

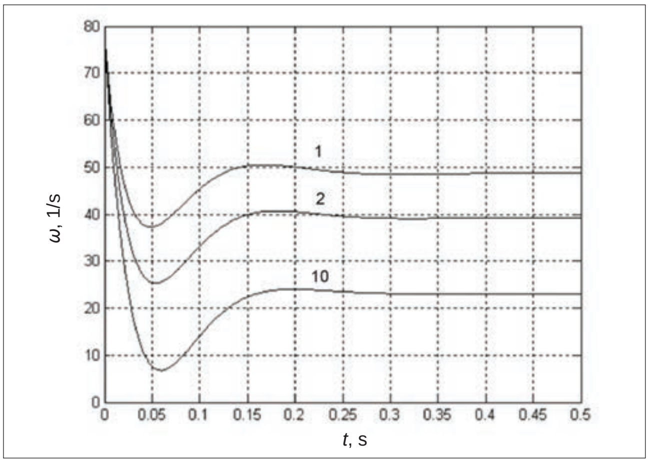


Fig. 8. Angular velocity ω(t) of the secondary drive unit for different values of M_h/M_{h0} = 1, 2, 10 and for S_w/S_{w0} = 5

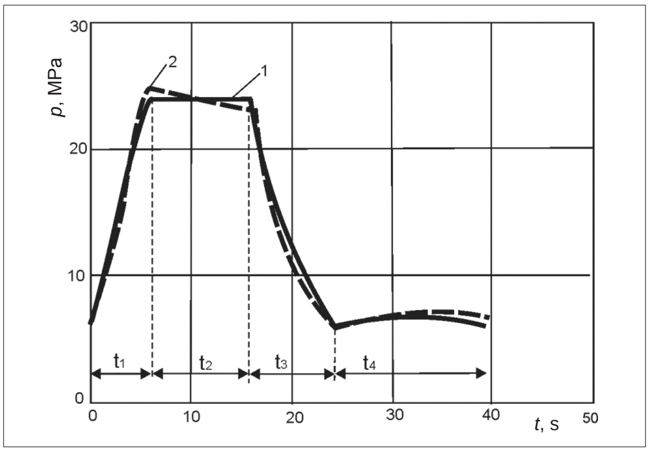


Fig. 9. Pressure change *p(t)* in a hydraulic accumulator during one driving cycle: 1 – simulation, 2 – experimental test

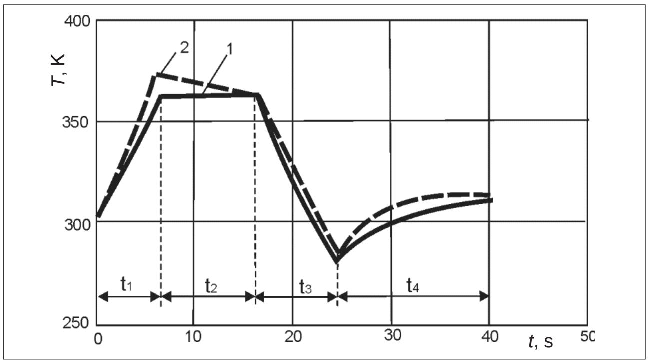


Fig. 10. Temperature change *T(t)* in hydraulic accumulator during one driving cycle: *1* – simulation, *2* – experimental test

Conclusions

From a review of power transmission systems and methods of recuperation and accumulation of braking energy in hybrid drives, hydraulic hybrid serial drives are becoming increasingly popular in light vehicles (passenger cars and near-by means of transport). There are not many hydrostatic test systems used in hybrid drives, especially electrohydraulic test drives for reversible drive units (motor/pump units). This demonstrates a number of technical barriers that require careful identification and overcoming. These barriers are mainly observed in the dynamic properties of hydrostatic systems with secondary control of the propulsion units and the accumulation of energy in the hydraulic accumulators, which will be facilitated by the simulated and physical model presented in the work.

At this stage of the study, focus was placed on the analysis of the dynamic electrohydraulic secondary control system of the reversible drive unit in a closed hydrostatic system with a high-pressure battery. The main purpose of the simulation and experimental studies was to determine the dynamic characteristics of the reversible drive unit depending on the parameters of the hydrostatic system. The obtained results provide the basis for further research on the development of electrohydraulic adaptive secondary control systems in hydrostatic series hybrid drives [20]. The research will also focus on the use of hydrostatic transmission in hybrid agricultural machines (for drive systems and agricultural tools). This is justified by the fact that agricultural machinery consumes a lot of fuel, and in the high season they often work up to 20 hours a day. Hybrid drives in these machines have not yet been used, although in Finland at Aalto University such studies have been undertaken.

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