The use of innovative solutions improving selected energy or environmental indices of hydrostatic drives

Z Domagała¹*, K Kędzia¹ and M Stosiak¹

¹ Wrocław University of Science and Technology, Faculty of Mechanical Engineering, Ignacego Łukasiewicza 5, 50-371 Wrocław, Poland

*E-mail: zygmunt.domagala@pwr.edu.pl

Abstract. This paper consists of three parts. The first part presents an exemplary application of a state-of-the-art proportional directional valve in hydrostatic gear starting. It is shown that through a proper directional valve control algorithm one can shape the pressure during the starting of the gear, reducing in this way the accompanying system noise. The second part describes the digital fluid power (DFP) technology. Its main advantage is the system’s simple design. One type of valve in 2 or 3 different sizes is sufficient to perform all the functions. In comparison with the often-used proportional valve, such a valve (called a binary valve), is characterized by simple circuitry, the absence of a position sensor, greater tolerance and lower sensitivity to impurities. An exemplary implementation of the DFP system as an additional drive in an off-road vehicle is presented. The third part describes the kinetostatic method for determining hydraulic system control parameters in order to increase the energy or environmental performance of the machine through the stabilization of the primary energy source and the selection of proper recuperation system components.

1. Introduction

Besides ensuring basic parameters, such as power output, driven element speed range, efficiency, etc., a drive system designer must also ensure proper dynamic properties specific to the machine being designed. Recently new-generation proportional directional control valves characterized by unprecedented properties have appeared on the market and are increasingly often used. The new directional valves achieve these properties owing to the novel way of driving the spool. In the patented design solutions the directional valve’s spool interacts with a movable coil, instead of (as previously) with an armature. Thanks to this solution the inertia of the moving components can be reduced, whereby the cut-off frequency of the directional control valve can be increased up to the values so far characterizing only two-stage electrohydraulic amplifiers. This way of driving the directional control valve spool is called Voice Coil Drive (VCD) [1]. When designing drive systems, one must also consider solutions which will limit the emission of noise by the operating machines since hydrostatic drive system have, besides their commonly known advantages, a major drawback – they are a source of noise, sometimes exceeding the valid noise standards. The designer must also meet the additional requirements stemming from the specific application of the system, e.g. in special-purpose vehicles. The design of a special-purpose vehicle must overcome two main difficulties. One of them is the ability to negotiate nearly any terrain, but not at the expense of an overcomplicated vehicle structure. This entails the other requirement – the vehicle should be effective enough for its practical use to be profitable. For this reason it is necessary to employ an internal-combustion engine as the primary
generator. Other solutions, e.g. electric motors, must be rejected since they necessitate the use of batteries with proper capacity, which are too heavy.

A hydrostatic transmission gear is one of the solutions very often used in agriculture and forestry. Its advantage is easy manoeuvrability making a continuous gear ratio within a certain range and gear change possible, which is useful in vehicles moving in steep terrain. Hence hydraulic systems based on the Digital Fluid Power (DFP) technology are increasingly often used in such applications.

In a DFP system the input and output parameters and the ones involved in the storage, transmission and processing of information can assume only specific values from a given range. There are two types of DFP systems: 1) based on the parallel connection and 2) based on the switching technology.

In each case, one should bear in mind the system operating parameters optimal for the adopted criterion. For this purpose one can use, e.g., the kinetostatic method of selecting operating parameters for a multisource drive system.

2. Use of proportional control technology for shaping hydrostatic gear starting

The throttling method and the volumetric method are the principle methods used to control the hydrostatic drive [2]. One of the reasons for the use of the proportional directional valve in the series throttling method to throttle the flow feeding the receiver is that thanks to this solution a change in the direction of the motion of the hydraulic receiver can be simultaneously effected. Figure 1 shows an exemplary schematic of a series throttling system for a hydrostatic gear with a proportionally controlled valve.

![Figure 1](image-url)

**Figure 1.** Schematic of series hydraulic system throttling hydrostatic gear:
- \( Q_{pt} \) – theoretical pump capacity 1, \( Q_{vp} \) – pump leakage losses, \( Q_z \) – flow through relief valve 3, \( Q_{RD} \) – flow through proportional valve, \( Q_{Cs1} \) – flow generated by compressibility within system volume between pump and valve controlling motion direction, \( Q_s \) – flow to hydraulic actuator, \( Q_{vs} \) – hydrostatic motor leakage losses, \( Q_{Cs2} \) – flow generated by compressibility within system volume between proportional directional valve and hydrostatic motor, \( Q_{RD1} \) – flow through proportional valve.
A mathematical model of the series throttling control of a hydrostatic gear consists of a system of two equations: (1) for the continuity of the flow in the particular points of the hydraulic system and (2) for the equilibrium of hydrostatic motor shaft torques [3]. In order to solve the system of equations initial conditions must be defined [4]. Using the symbols from Figure 1 one can write the following equation of the continuity of the flow in the discharge flange of the pump:

\[ Q_{p} = Q_{sp} + Q_{cp} + Q_{RD} + Q_{c} \]  

The balance of the rate of the flow through the throttling directional control valve has the form:

\[ Q_{RD} = Q_{x} + Q_{xv} + Q_{cst} \]  

If the hydraulic losses in the conduits are omitted, the rate of the flow through the proportional directional control valve is described by the equation:

\[ Q_{RD} = G_{RD}(t) \sqrt{p_{p} - p_{s1}} \]  

In the considered system the dependence between the surface area through which the fluid flows in the proportional directional valve and the displacement of the spool is assumed to be linear. The dynamics of the proportional control valve in this system are described by the 1st order differential equation identical with the inertial term:

\[ s(t) \cdot G_{RD_{max}} = G_{RD}(t) + T_{RD} \cdot \frac{dG_{RD}(t)}{dt} \]  

where:
- \( G_{RD}(t) \) – the conductivity of the directional control valve,
- \( G_{RD_{max}} \) – the maximum conductivity of the directional control valve,
- \( T_{RD} \) – the time constant of the directional control valve,
- \( s(t) \) – a control parameter.

The equilibrium of the proportional valve pressure drops is assumed:

\[ p_{p} - p_{s1} = p_{s2} \]  

The flow towards the hydrostatic motor has the form:

\[ Q_{s} = q_{s} \omega_{s}(t) \]  

The flow induced by the compressibility of the working fluid and by the deformations of the components between the pump and the proportional control valve is written as:

\[ Q_{cp} = c_{p} \cdot \frac{dp_{p}(t)}{dt} \]  

where:
- \( c_{p} \) – the capacitance of the fluid and of the conduits between the proportional control valve and the hydrostatic motor.

Between the proportional control valve and the motor the flow induced by the compressibility of the working fluid and by the deformations of the components has the form:

\[ Q_{cst} = c_{s1} \cdot \frac{dp_{s1}(t)}{dt} \]  

where:
- \( c_{s1} \) – the capacitance of the fluid and of the conduits between the proportional valve and the hydrostatic motor.

Between the motor and the proportional control valve the flow generated by the compressibility of the working fluid and by the deformations of the components is expressed as:
\[ Q_{c_{s2}} = c_{s2} \cdot \frac{dp_{s2}(t)}{dt} \]  

(9)

where:
\( c_{s2} \) – the capacitance of the fluid and of the conduits between the motor and the proportional control valve.

The losses due to motor leakages are defined by the relation:
\[ Q_{vs} = a_{vs} \cdot (p_{s1} - p_{s2}) \]  

(10)

where:
\( a_{vs} \) – a motor leakage coefficient, \( p_{s1}, p_{s2} \) – the pressure in the flanges of the hydrostatic motor.

The flow through the relief valve [4, 5], amounts to:
\[ Q_s(t) + T_z \cdot \frac{dQ_s(t)}{dt} = h_z \cdot [p_p(t) - p_0] \text{ for } p_p(t) > p_0 \text{ and } Q_s(t) = 0 \text{ for } p_p(t) \leq p_0 \]  

(11)

where:
\( T_z \) – the time constant of the relief valve, \( h_z \) – a valve amplification factor, \( p_0 \) – the relief valve opening pressure.

The hydrostatic motor shaft torques equilibrium condition is expressed by the relation:
\[ q_s \cdot (p_{s1} - p_{s2}) = M_h + f \cdot \omega_s(t) + I_{zr} \cdot \frac{d\omega_s(t)}{dt} \]  

(12)

where:
\( p_{s1} \) – the hydrostatic motor suction flange pressure, \( p_{s2} \) – the hydrostatic motor discharge flange pressure, \( q_s \) – the (geometric) specific absorbing capacity of the motor, \( f \) – the (experimentally determined) viscous friction coefficient [5], \( M_h \) – the motor shaft braking torque, \( \omega_s \) – the angular speed of the hydrostatic motor, \( I_{zr} \) – the reduced to the motor shaft solid moment of inertia of the rotating masses.

The initial conditions are:
\[ p_p(0) = p_0; \quad p_s(0) = 0 \]  

(13)

The boundary condition is defined as follows:
when \( q_s \cdot (p_{s1} - p_{s2}) \leq M_h \), then \( \omega_s = 0 \) and \((d\omega_s)/(dt) = 0 \)

(14)

After the mathematical model equations are parametrized and the initial conditions assumed, the model can be numerically solved and the time history diagrams of the pressure in the motor suction flange can be graphically presented (Figure 2).
A change in the control signal rise time causes a change in the maximum pressure value in the suction flange of the motor. The graphically presented solutions are for different values of control signal rise time $t_0$: 1 s, 2 s, 5 s. The change in the control signal fed to the proportional directional control valve’s coils over time ($s(t)$) is shown in Figure 3 and described by the relation:

$$s(t) = \left\{ \begin{array}{ll} \frac{i_0}{i_{\text{max}}} & t = 0 \\ \frac{1}{i_{\text{max}}} \left(1 - \frac{i_0}{i_{\text{max}}} \right) t & \text{for } t > 0 \end{array} \right.$$  \hspace{1cm} (15)

where:
- $i_0$ – the minimum value of the control current,
- $i_{\text{max}}$ – the maximum value of the control current,
- $t_0$ – the control signal rise time.

During the starting of the gear by means of the series throttling control method the positive displacement pump is loaded with the relief valve set point regardless of the rise time of the signal controlling the proportional control valve. But one can have an influence on the duration of this load – the faster is the valve reset, the shorter the duration of the pump load resulting from the operation of
the overflow valve. The proportional control valve resetting time affects the hydraulic motor feed pressure – mainly the maximum value of pressure \( p_s1 \) during starting. By reducing the maximum pressure values during the starting of the gear one can reduce the latter’s operational noisiness in this phase of motion [5].

3. Special-purpose vehicle design incorporating hydraulic components controlling system output power

A parallely connected DFP system includes many connections and the output effect is controlled through a change in the states of the combinations of the components. Such a system has a specific number of discrete output values and there is no need to employ switches. Figure 4a shows an assembly of valves connected in parallel, performing the function of a two-way valve. Such a structure is called a Digital Flow Control Unit (DFCU).

Since the valves in a DCFU can be only completely opened or completely closed, the rate of flow is adjusted in discrete steps, as opposed to the proportional valve in which proportioning is continuously adjusted.

![Figure 4. Digital flow control unit: a) schematic, b) symbol [6].](image)

Whereas the switching technology uses the fast and continuous switching of one or several elements and the output is adjusted through, e.g., pulse width. The symbol representing such a valve is shown in Figure 5b. This is one of the commonest ways of controlling a medium flow area. Theoretically the latter can assume any value, but the valve’s dynamics limit boundary work cycle values. The controllability also depends on the switching frequency. A low switching frequency improves the control of a medium flow area, but it contributes to pressure fluctuations. Usually it is necessary to use damping elements in order to reduce noise.

![Figure 5. On/off valve controlled by pulse width: a) signal graph (blue curve) and valve loading graph (black curve), b) valve symbol (based on [6, 7]).](image)

The DFP technology also involves the use of digital pumps (Figure 6). In a conventional pump the transport of a liquid is effected through cam disk deflection, which is set for the pump delivery best for the particular operating conditions. In the switching technology, the flow from a constant capacity pump is controlled by means to the on/off valve mentioned above. In the case of a parallel connection, the pump actually consists of several constant capacity pumps mounted on a single shaft.
Figure 6. Principle of digital pump operation with three pistons [8].

Figure 7 shows a design of a hydraulic drive incorporating digital technology elements and Figure 8 - schematic of drive using powersplit technology. The engagement and disengagement of the drives (a conventional drive and a hydrostatic drive) in a special-purpose vehicle will be effected by the powersplit gear used in electrical or hydraulic hybrids.

Figure 7. Hydraulic diagram of selected solution.

Figure 8. Schematic of drive using powersplit technology.
The solutions described above encounter difficulties in practice, mainly due to the small selection of digital hydraulics components on the Polish market. Since only one digital pump was commercially available the original design had to be somewhat modified, but the modification did not infringe the project’s guidelines. The next step is to develop a control program, which when properly designed will increase the operating efficiency of the vehicle.

4. Kinetostatic method of selecting operating parameters for multisource hydrostatic drive system

This method is used to determine – for a multisource drive system built of components with known characteristics, a load in the form of a machine duty cycle and an adopted criterion – the following parameters:

- the optimal primary energy source working point,
- the initial parameters of the secondary energy source,
- the modes of controlling the other drive system components.

This is done under the following assumptions:

- the primary energy source is to work in one point throughout the duty cycle of the device,
- the secondary energy source charge level is to be the same the beginning and end of the duty cycle.

The method yields a set of parameters for controlling the multisource drive system for the adopted criterion.

The method of selecting the operating parameters for a multisource drive system is based on the refined kinetostatic method [9, 10, 11, 12], shown in schematic way on Figure 9.

**Step 1.** On the motor performance characteristic define the possible allowable operation area. On this basis determine the considered set of engine operation points \( M_{sp}, \omega_{sp}, G_j, CO, NOx \) and exhaust gas opacity.

**Step 2.** Present, in the form of a table or in an analytical form, the machine load characteristic in the duty cycle of duration \( T \). The characteristic comprises dependences between load (force) torque \( M(t) \) and (linear) angular velocity \( \omega(t) \). Assume time increment \( \Delta t \) values. Select energy accumulator parameters: capacity and initial charge level \( E(t=0) \).

**Step 3.** Compare the values of internal-combustion engine power \( N_{sp} = f(M_{sp}, \omega_{sp}) \) for the adopted \( j \)-th allowable area point with power demand \( N_{oi}(M_{oi}, \omega_{oi}) \) over time \( t_i \), consistently with the load cycle.

Since:

- the primary source power and the load power are expressed by the respective values of their stress (torque) variables \( M_{sp} \) and \( M_{oi} \) and intensity variables \( \omega_{sp} \) and \( \omega_{oi} \),
- in the general case, the values of engine power \( N_{sp} \) and load \( N_{oi} \) are different,

the task of the hydrostatic power transmission system is to:

- appropriately transform the internal-combustion engine’s stress variable \( M_{sp} \) and intensity variable \( \omega_{sp} \) to the form required by the load’s stress variable \( M_{oi} \) and intensity variable \( \omega_{oi} \),
- balance the power supplied by the primary energy source and the load power demand by carrying off the excess energy to the accumulator or making up for an energy shortage by drawing energy from the accumulator.

The primary source power in the form of signals \( M_{sp} \) and \( \omega_{sp} \) is transmitted to the shaft of a displacement pump with a known capacity. For given pressure \( p \) input quantities \( M_{sp} \) and \( \omega_{sp} \) are transformed, by solving the equations describing actual pump energy characteristics \( M_p \) and \( Q_p \), into discharge flow rate controlling element tilt (\( \varepsilon_p \)) signals.

The instantaneous value of the load power in the form of signals \( M_{oi} \) and \( \omega_{oi} \) is transmitted to the shaft of the hydrostatic motor with a variable absorbing capacity. For given pressure \( p \) quantities \( M_{oi} \) and \( \omega_{oi} \) are transformed, by solving the equations describing the hydrostatic motor’s actual energy characteristics \( M_i \) and \( Q_{sh} \), into motor controlling signal \( \varepsilon_{sh} \).
The energy losses connected with flows $Q_p$ and $Q_{sh}$ in the hydraulic system, relative to the summing node in which pressure $p$ prevails, are mapped in the next blocks. At the given value of pressure $p$ power balancing takes place in the summing node through the satisfaction of the flow continuity equation: $Q_p + Q_{sh} + Q = 0$. In the case of primary source excess power relative to the load, the accumulator draws a liquid stream with power $p\cdot (Q_p - Q_{sh})$, while in the case a shortage, it gives up power $p\cdot (Q_{sh} - Q_p)$.

Pressure $p$ in the summing node is closely connected with the energy state of the accumulator. However, in the general case, this is not a simple and unequivocal relationship. It depends on the accumulator used and the way of controlling it. In the considered case, a hydraulic accumulator, incorporated directly into the hydraulic system, was used. The algorithm for this solution is shown in Figure 10.

1. IC engine characteristics
   $M_{sp} = f(\omega_{sp})$    $G_c = f(M_{sp}, \omega_{sp})$

2. Load characteristics
   in a form of $M_o(t_i)$ and $\omega_o(t_i)$ curves
   over machine duty cycle

3. Transformation, distribution and transmission of power in hydrostatic power transmission system

4. $N(t_i)_{out} < N(t_i)_{in}$

5. $E_{a2} = E_{a2} - \Delta E_{at}$

6. $E_{a1} = E_{a1} + \Delta E_{at}$

7. $t_i < T$

8. $E_{a2} - E_{a1} \leq \sigma_s$

9. $E_{a2} < E_{a1}$

10. System optimizing selection
    of engine characteristic points
    $M_{spj}$, $\omega_{spj}$

11. Optimal point $M_{opt}$, $\omega_{opt}$ for $G_c = \min$

Figure 9. Procedure for determining control parameters for prescribed hybrid system load – refined kinetostatic method [11, 12].
In **block A** pressure $p$ in the summing node is calculated for the given iteration ($k$-th). The pressure value depends on the accumulator charge status. Then the values of hydrostatic unit control parameters $\varepsilon_p$, $\varepsilon_{sh}$ are calculated for the given load $M_i$ and internal-combustion engine torque $M_{spj}$ (**block B**). The values, relative to the respective previous values, over time $t_i - \Delta t$ must satisfy the limit resetting speed condition.

**Figure 10.** Development and refinement of algorithm in block 3 dealing with power transformation and transmission in hydrostatic power transmission system [11, 12].

If the conditions in **block C** are not satisfied, go back to block A. If the maximum pump and hydrostatic motor resetting speeds are not viable, check whether the tilt of the hydrostatic motor disk is larger than the so-called hydrostatic motor self-braking interval. If this condition is satisfied, go to **block D**. Since the motor will find itself in the zone mentioned above, reduce disk tilt to 0 in the shortest possible time to minimize volumetric losses. In **block D** calculate the energy losses. First determine the real pump capacities $Q_p$ and motor absorbing capacities $Q_{sh}$.

Then determine the energy losses over time $t_{i-1} - t_{i-1} = \Delta t$ in:
- the displacement pump $-\Delta E_p(\alpha_p, p_p, A_p)$,
- the hydrostatic motor $-\Delta E_{sh}(\alpha_{sh}, p_{sh}, A_{sh})$,
- the hydraulic system $-\Delta E(Q_{sh}, Q_p)$,
- in the energy storage system $-\Delta E_{sh}(E_a, N_a)$ (using the Benedict Rubin Weber (BWR) model).

The power losses in the energy storage system depend on the accumulator charge status $E_a$, charging and discharging rates $\frac{dE_c}{dt}$, i.e. the supplied or drawn power, the duration of energy storage in the accumulator and the latter’s structural parameters: the wall thickness, the thermal conductivity coefficient of the walls, the ambient temperature, etc.
In block E decisions about going to the next (4th) step of the algorithm or the next \((k+1)\) iteration and a search for the next converter control are made. Thus the main problem consists in selecting such energy converter controlling signal \(E_a\) that for:

- internal-combustion engine working point \(M_{apj}, \omega_{apj}\) and
- load characteristic point \(M_{ap}(t_i), \omega_a(t_i)\) for instant \(t_i\)

energy loss sum \(\Sigma \Delta E_{at}\) is minimal.

**Step 4.** Compare the values of the instantaneous capacities reduced to the summing node, i.e. with energy transformation and transfer losses taken into account.

**Step 5.** Calculate values \(E_{a2}\) of the energy abstracted from the accumulator.

**Step 6.** Calculate the energy supplied to the accumulator over time \(\Delta t = t_i - t_{i-1}\), and reduced to the summing node. The operation of step 5 or 6 depends on the sign of inequality \(N(t_{i-1})_{\text{low}} < N(t_i)_{\text{up}}\).

**Step 7.** Stop the calculations for given cycle instant \(t_i\); when \(t_i < T\), repeat the algorithm loop for \(t_i = t_i + \Delta t\) until time \(t_i = T\). When duty cycle \(t_i = T\) ends, go to the next step.

**Step 8.** Test condition \(|E_{a2} - E_{a1}| < \sigma_a\). Depending on the result obtained for the given load cycle, do the following:

- if the condition specified in step 8 is satisfied, select new internal-combustion engine working point \(M_{apj}, \omega_{apj}\).
- if the condition is not satisfied, consider the following two options:

**Step 9.**

1. When for the given cycle the energy supplied to the accumulator is lower than the energy drawn from the latter, the energy balance of this cycle is negative, with the shortage exceeding \(\sigma_a\). This option is unacceptable and so go to step 10 and select a new internal-combustion engine working point \(M_{apj}, \omega_{apj}\).
2. When for the given cycle the energy supplied to the accumulator is higher and the excess energy exceeds \(\sigma_a\), go back to block 3 (Figure 10) concerning the energy converter control signal and repeat the load cycle calculations for the given internal-combustion working point \(M_{fp}, \omega_{fp}\).

**Step 10.** The energy analysis is performed by the optimizing system until point \(M_{ap}, \omega_{ap}\), \(M_{fp}, \omega_{fp}\), in which the quality criterion (fuel consumption \(G_e\) reaches the minimum) for the given cycle is satisfied, is found.

In the above method of selecting the parameters of a multisource hydrostatic drive system [9] one can adopt any control criterion, e.g. minimal fuel consumption \(G_e\) or minimal atmospheric harmful emission (CO, NOx, smoke pollution). Consequently, one can select a criterion proper for the place of machine operation, e.g. on throughways and in non-residential area one can adopt the minimal cost (min. \(G_e\)) criterion, increasing thereby exhaust gas emission, whereas in mines and built-up areas the fuel cost is of secondary importance.

**5. Conclusion**

The exemplary state-of-the-art hydraulic drive solutions presented in this paper indicate that the main advances are made in the areas of: the hydraulic drive and the control of hydraulic components, the continuous (proportional) control technology and the digital fluid power (DFP) technology. As a result, hydraulic systems can achieve unprecedented performance and their efficiency can be increased through the use of an appropriate control algorithm. The costs and availability of hydraulic components put some limitations on the use of the new technologies of controlling them. Moreover, the working fluid cleanness requirements for proportional valves are higher.

The kinetostatic method described in this paper can be used not only to determine the drive system control optimal for the energy or environmental criteria adopted for a given machine duty cycle, but
also to estimate the potential benefits from the use of multisource systems instead of the conventional ones. Thus, this method is a tool which can aid engineers in upgrading conventional drive systems and estimating the benefits gained by replacing a conventional drive system with a multisource one.

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