Considerations on Energy Losses in Hydraulic Drive Systems

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Abstract: *The article presents some considerations on energy losses in hydrostatic drive systems, identifying the elements and specific places where flow and pressure losses occur. Some considerations are made on flow regimes and calculation relations are presented for assessing the pressure and flow losses; also on an example of hydrostatic drive system there is assessed energy dissipation and system efficiency is calculated. In the end, there are presented some modern methods and concepts for design of hydraulic diagrams used for the purpose of increasing energy efficiency of hydrostatic drive systems.*

Keywords: *hydrostatic drive systems, energy efficiency, energy losses, pressure losses, flow losses*

1. Introduction

In recent years, hydrostatic drives have developed a lot and have penetrated into many areas. Technological facilities have become increasingly complicated and complex, and requirements on automation of their operation, on the one hand, and increased requirements regarding energy efficiency and productivity to increasingly higher levels, on the other hand, have imposed on the drive systems, and implicitly on the automation hydraulic equipment that are part of the former, new requirements and new performances. Among them the requirements with regard to increased operational reliability and accuracy, improved static and dynamic performance and, in particular, the **decrease in the consumption of energy** and materials are very important [1].
To achieve the required performance, there is a need not only for modern components with suitable static and dynamic parameters, but also a **modern concept** of the working diagram of a hydrostatic transmission. There also needs to be done a very careful analysis on mass and energy flow, identifying all critical points that may cause flow and pressure losses, **in order to increase energy efficiency** of the hydrostatic drive system.
Since a hydrostatic drive system is composed of various parts through which the hydraulic agent flows, it follows that the process of flowing is always accompanied by **load losses**. It is important, however, that these load (pressure) losses to be as small as possible, because the sum of all load losses in the system implicitly results in **unnecessary consumption** that the power supply of equipment has to cover in order to overcome those losses. Load loss manifests, eventually, in **dissipation of some of the energy of the conveying fluid**, because this energy is transformed into heat energy, which is taken up by the fluid, heating it. This process is more pronounced the higher the pressure drop on a system component is. Those presented before are phenomena that exist and they **always accompany the flowing phenomena of the working agent**, but the losses in the system should be as small as possible for the system to be a high efficiency one [2].
Worldwide there are noted intensive concerns with regard to developing hydrostatic drive systems **with low energy consumption.** Since in hydrostatic drive systems very high power is circulated, the energy consumption issue is particularly significant [1].

2. Assessment of local and linear pressure losses

It is known that in real (viscous) fluid flow via pipelines and equipment parts part of the total energy of the mass of fluid is used to overcome **viscous friction resistance forces**, turning irreversibly into heat. Another part of this energy is consumed when **turbulence phenomena occur**, when the flow velocity changes its direction and / or size (due to changes in the flowing section). These energy losses can be defined as **resistance mechanical work** due to the viscous and turbulent resistances of real fluids [1].

Bernoulli's principle, customized to the case of incompressible viscous fluids, leads to:

$$\Delta P_{tot} = \rho \cdot \Delta e_{tot} = (P_1 - P_2) + \frac{\rho}{2} \cdot (w_1^2 - w_2^2), \quad (1)$$

where the term ΔP_{tot} is total load loss on the section considered, and the term Δe_{tot} is precisely **specific energy lost** by the fluid when it moves between two sections, 1 and 2, characterized by pressures p1 and p2 and flow velocities w1 and w2, ρ being fluid density. Relation (1) highlights the two **components of load loss**:

- **Linear** or **distributed component** ΔP_d , due to the fluid viscosity;
- **Local component**, ΔP_l, due to turbulence, so:

$$\Delta P_{tot} = \Delta P_d + \Delta P_l \quad (2)$$

This division into linear and local losses is a conventional one, because, in reality, pressure losses are indivisible physically. Based on numerous studies and research, **it was agreed** that load loss to be related to the kinetic energy of moving fluid [1].

For this purpose, the strength or loss coefficient ξ has been defined.

$$\xi = \frac{\Delta e_{tot}}{e_c} = \frac{Specific\ energy\ dissipated}{Kinetic\ energy\ of\ mass\ flow\ unit}\ or\ \ \xi = \frac{m \cdot \Delta e_{tot}}{m \cdot \frac{w^2}{2}}, \quad (3)$$

And if density is constant:

$$\xi = \frac{\Delta P_{tot}}{p \cdot \frac{w^2}{2}} \quad (4)$$

According to the relation (4), **the coefficient ξ** can be defined as **the ratio of total load loss and dynamic pressure in the benchmark section.**

The above relation can be rewritten also as:

$$\Delta P_{tot} = \xi \cdot \rho \cdot \frac{w^2}{2} \quad (5)$$

In reality, the flow velocity through a narrowed section is not constant, therefore in the relation (5) it is considered the average speed of the fluid through the section considered, speed given by the relation: $w = q/S$.

In this case, **local load loss** can be written:

$$\Delta P_l = \xi \cdot \frac{1}{2} \cdot \rho \cdot (q/S)^2 \quad (6)$$

where ξ is a dimensionless numerical coefficient, dependent on the geometry of the hole, but sometimes also on the fluid, on its temperature and flow.

If one aims to express **linear / distributed load loss**, in a tube / pipe, it is obvious that its length must interfere, and the relation (6) will no longer be convenient.

In this case the following expression will be chosen:

$$\Delta P_d = \lambda \cdot (L/D) \cdot \frac{1}{2} \cdot \rho \cdot (q/S)^2 \quad (7)$$

where *L* is pipe length, *D* – its diameter (if the pipe has not a circular section, *D* is hydraulic diameter, defined as $D_h = 4 \cdot A/P_u$, where A is pipe section, and P_u - perimeter wetted by the fluid), λ a dimensionless numerical coefficient dependent on the pipe, fluid and flow velocity.

If one takes into account **load losses in the transitional regime** as well, they are expressed through the practical relation [5]:

$$\Delta p_{din} = L_H \frac{dQ}{dt} \quad (8)$$

where:

$L_H = \frac{m}{A^2}\left[\frac{daNs^2}{cm^5}\right]$ is **hydraulic inertance** coefficient, and $m\left[\frac{daNs^2}{cm}\right]$ is mass volume of fluid that changes the flow velocity through the flowing section with the area A, expressed by Q.

If there is also considered the **concentrated capacitance** of a volume of fluid flowing through the section, they are calculated by the relation

$$\frac{1}{C_H}\int Q\, dt \tag{9}$$

In the above relation, $C_H\ [cm^5/daN]$ is the **coefficient of capacitance,** according to the following relation:

$$C_H = \frac{V}{E} \tag{10}$$

Based on the similarity hydraulic-electric, it can be considered that the hydraulic drive circuit *CHA*, between the pump *P* and the linear hydraulic motor *MHL*, shown in Figure 1, is similar to an equivalent electric circuit *CEE*, shown in Figure 2.

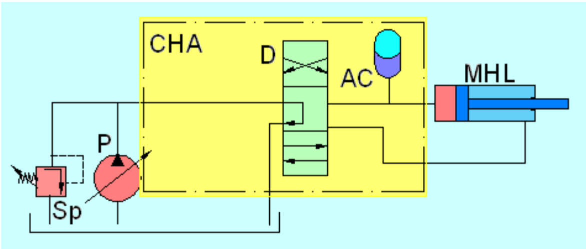

Fig. 1. Hydraulic drive circuit

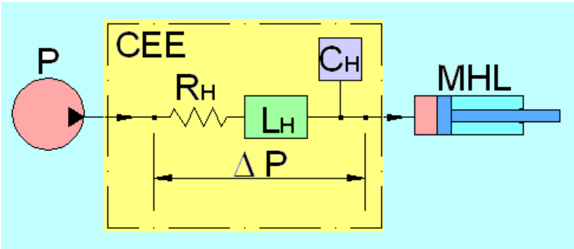

Fig. 2. Similar equivalent electric circuit

In this way, one can calculate a **generalized resistance equivalent to a hydraulic circuit [5].**
If fluid flow along the circuit in question takes place regularly, at a **frequency f,** the circuit impedes the flow by **total impedance** $Z_H\ \left[\frac{daN/cm^2}{cm^3/s}\right]$ which synthetically expresses flow-dependant pressure variation:

$$Z_H = \sqrt{R_H^2 + \left[2\pi f L_H - \frac{1}{2\pi f C_H}\right]^2} \tag{11}$$

Based on the above relations the characteristics of hydraulic circuit can be calculated using terms specific to electrical circuits, adjusted to the fluid power specific language.
In conclusion, assessment of load losses in a fluid drive system only requires **knowing the value of coefficients ξ or λ** for each system component. In a wider range, these coefficients are **characteristic constants** of the elements taken into account. In some cases, they depend on fluid flow, as a matter of convenience and in these cases one shall use the basic relations (6) and (7), but this time taking into account **the coefficients ξ and λ as variable**. Before giving numerical values to the coefficients ξ and λ, there will be defined two flow ranges corresponding to ξ and λ as constant and respectively as variable coefficients and the limits of these ranges shall be specified. [1].
There are known two types of flow in fluid dynamics: **laminar flow** and **turbulent flow**.
In laminar flow, adjacent fluid molecules advance with parallel speed vectors: one can imaginarily separate adjacent fluid layers between which there is no exchange of fluid. Due to the viscosity of the fluid, each layer tends to drag the adjacent one. The forces of friction being proportional to the viscosity coefficient and velocity gradient, **load losses**, in this case, will be proportional to the coefficient of friction and the fluid flow.

In turbulent flow, adjacent fluid molecules have no parallel trajectories. Speed vectors of molecules tend to depart from the direction of the average flow speed [1].
Laminar flow mode of the working agent in a finite circular pipe is shown in Figure 3, while **turbulent flow** is shown in Figure 4 [2].

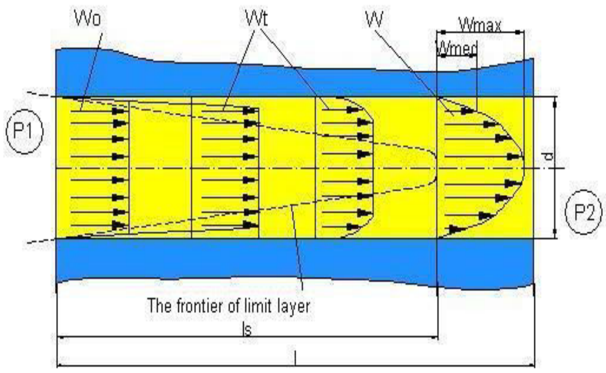

Fig. 3. Laminar flow in tubes

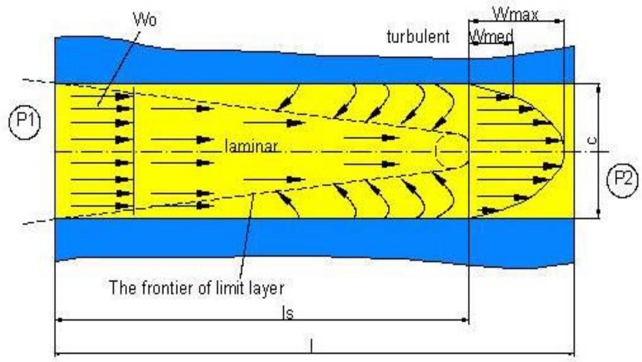

Fig. 4. Turbulent flow in tubes

Fluid flow mode depends on **the ratio** of **inertia forces** and **viscosity forces**, which is expressed by a dimensionless number, named **Reynolds number** [1]:

$$Re = \frac{\rho \cdot D_H \cdot w}{\eta} = \frac{D_H \cdot w}{\nu} \tag{12}$$

where: η – dynamic viscosity, ν – kinematic viscosity; D_H – hydraulic diameter, defined in a previous paragraph; w – fluid flow rate in a given point.
To calculate **Reynolds number**, suitable formulas are given in the literature for different situations. There are also charts and tables based on which there can be set the **coefficients:** λ, which characterizes linear load losses and ξ, which characterizes local load losses [1], [2], [5]. On these grounds, one can assess local and linear load losses in hydrostatic drive circuits.

3. Assessment of flow losses in hydrostatic systems

As regards **fluid losses in** complex **hydrostatic** drive **systems**, literature presents a series of specific elements through which there occurs leakage of working fluid and it also gives a series of specific mathematical relations and graphs for assessment of losses, [1], [2], [3], [4], [5]. From the literature investigated it can be noticed that **local flow loss,** through functional clearances of drive components, can be assessed by a relation of type [3]:

$$\Delta Q_i = K_{Qi} \cdot \Delta p; \;\; [m^3/sec]; \tag{13}$$

where: K_{Qi} is **flow loss coefficient**, which is dependent on the structural sizes (linear and angular dimensions) of the element in which leakage occurs and also on the **dynamic viscosity** of hydraulic fluid used; Δp – pressure difference between the high pressure enclosure and low pressure one, which causes the flow loss.
In a situation where in the construction of a component there are comprised several structural types of interstices, slots, clearances, subjected to the same differential pressure Δp, as in Figure 5, the relation for assessment of flow losses will be:

$$\Delta Q_i = \left(\sum_{i=1}^{s} K_{Qi}\right) \cdot \Delta p = K_Q \cdot \Delta p; \; [m^3/sec]; \tag{14}$$

If in the construction of a hydraulic component there are comprised several structural types of interstices, slots, clearances, which are connected in series, as in Figure 6, the global relation for assessment of flow losses will be:

$$\Delta Q_i = \sum_{i=1}^{s} K_{Qi} \cdot \Delta p_i;\ [m^3/sec]; \tag{15}$$

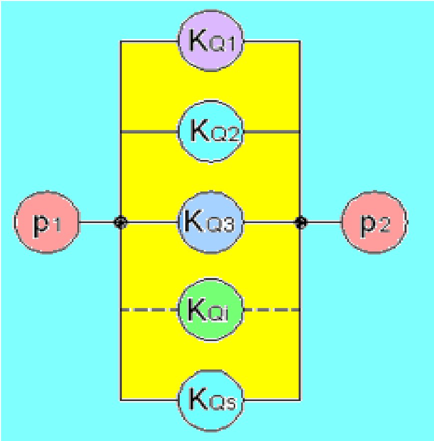

Fig. 5. Resistive elements in parallel

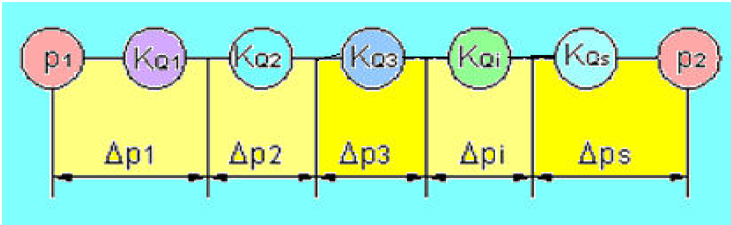

Fig. 6. Resistive elements in series

Both situations are commonly found in the structure of hydraulic equipment components [3].

4. Energy dissipation in hydrostatic drive systems

All the components of a hydrostatic drive system are impacted, more or less, by pressure and flow losses, resulting in significant energy dissipation in the system.
Energy dissipation in hydrostatic systems manifests itself by default in all drive components that, in order to convey energy in the system, use transporting fluids, in the form of liquids or gases [2].
Hydraulic power used in order to overcome load drop on a component of the system, at a circulated volumetric flow rate Q = constant, is given by the relation:

$$N_h = \Delta p \cdot Q = \xi\,(x)\ \frac{\rho}{2} \cdot \frac{Q^3}{A\,(x)};\ [Watt] \tag{16}$$

And for (i) components crossed by the flow Q, it results:

$$N_h = (\textstyle\sum_i \Delta p_i) \cdot Q;\ [Watt] \tag{17}$$

The above presented are phenomena that exist and always accompany the phenomena of working fluid flow through hydrostatic drive systems, but the amount of losses in the system should be as small as possible for the **system to be a high efficiency one** [2].

There are situations in which, **in order to adjust** some hydraulic parameters of the drive system, the phenomenon of **energy dissipation** described **is used** in spite of increased energy consumption in the system.
Such an example is a **hydrostatic drive system with resistive speed control.** This is done, for instance, in the case that the hydrostatic drive systems has **fixed displacement pumps,** by using pump flow control devices, which work by dissipating a part of the available energy. In other words, **part of the pump flow is discharged** to the system tank at a specified pressure drop. Such a flow control system is shown in Figure 7 and Figure 8. Since the pump P is a fixed displacement one, the only way to decrease the flow feeding the motor is discharging a part of it, at the pressure required by the motor, to the system tank (T). It results that the flow that must be discharged to the tank gets:

$$Q_T = Q_p - Q_p^{**} = \frac{\tau-1}{\tau}\, Q_p \tag{18}$$

To be possible to adjust the speed of the receiver it is considered the diagram in Figure 7, in which the flow limiter (fixed throttle valve) or the hydraulic resistance LD has been inserted, **on the active** circuit **line**, which serves as a agent **flow regulator**, while the valve S serves as a **pressure limiting device** [2].

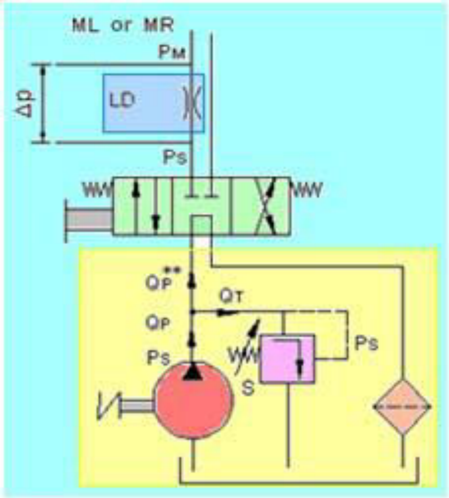

Fig. 7. Flow limiting device montage

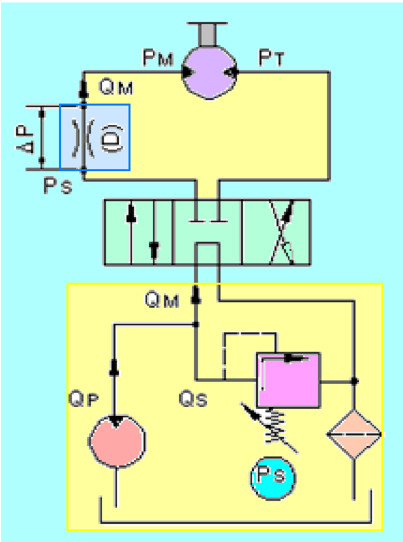

Fig. 8. Diagram of a hydrostatic system with fixed displacement pump and rotary hydraulic motor

If it is considered that the hydraulic circuit losses are negligible, then pressure drop on the throttle valve LD will be:

$$\Delta_p = p_s - p_M = \xi \cdot \frac{\rho}{2} \cdot \left(\frac{Q_p^{**}}{a^2}\right)^2; \tag{19}$$

where (a) is flow section of the throttle; p_s is control pressure of the pressure limiting valve; p_M is pressure required by the (linear or rotary) motor to develop the load at its shaft or rod. In these circumstances, if one assesses power consumption in the system, there results the following distribution of power on the system components:

- **Power supplied** by pump and taken over by the working agent:

$$N_p = Q_p \cdot p_s, [\mathrm{W}] \tag{20}$$

- **Power** effectively **taken over** by the hydraulic motor for overcoming load at the rod or shaft:

$$N_{MOT} = Q_p^{**} \cdot p_M; \ [\mathrm{W}] \tag{21}$$

where: $Q_p^{**} = Q_p/\tau$ and $p_M \cong \frac{p_s}{\gamma}$; τ and γ being supra-unitary numbers that define functions of pump flow, respectively, functions of adjusted pressure.

- **Power dissipated** by the pressure limiting device, namely the valve S, will be:

$$N_s = Q_T \cdot p_s = \frac{\tau - 1}{\tau} \cdot Q_p \cdot p_s; \tag{22}$$

- **Power dissipated** by the flow limiting device LD will be:

$$N_{LD} = Q_p^{**} \cdot (p_s - p_M) = \frac{Q_p}{\tau} \cdot p_s \left(1 - \frac{1}{\gamma}\right) = Q_p \cdot p_s \cdot \frac{\gamma - 1}{\gamma \tau}; \tag{23}$$

In the case shown in **Figure 8,** there is presented an application for a hydrostatic system with fixed displacement pump and rotary hydraulic motor, with throttle valve on the supply line [2].
In this case, we can calculate:
- Pressure drop on the hydraulic resistance (throttle valve):

$$\Delta p = p_s - p_M = \xi \ \frac{\rho}{2} \frac{Q_M^2}{Q^2} \tag{24}$$

- Flow circulated through the motor:

$$Q_M = Q_p - Q_s \tag{25}$$

- Theoretical angular speed at motor shaft:

$$\omega_M = \frac{Q_M}{V_{OM}} = \frac{Q_p - Q_s}{V_{OM}} \tag{26}$$

- Power taken over by motor, in (kW):

$$N_M = Q_M p_M \tag{27}$$

- (Theoretical) pump power:

$$N_P = Q_P p_s \tag{28}$$

- Power dissipated via throttle:

$$N_D = Q_M\,(p_s - p_M) \tag{29}$$

- Power dissipated via valve:

$$N_s = Q_s p_s = \left(Q_p - Q_M\right) p_s \tag{30}$$

- **Efficiency** of hydrostatic drive system with fixed displacement pump:

$$\eta = \frac{N_M}{N_p} = \frac{Q_M p_M}{Q_p p_s} = \frac{(Q_p - Q_s) p_M}{Q_p p_s} = \left(1 - \frac{Q_s}{Q_p}\right)\frac{p_M}{p_s} \tag{31}$$

From the above analysis it can be noticed that **efficiency of this hydrostatic system** is lower when the flow of hydraulic agent conveyed through the pressure limiting valve is greater and the ratio of motor pressure (p_M) and valve pressure (p_s) is lower, which is also shown in Figure 9.

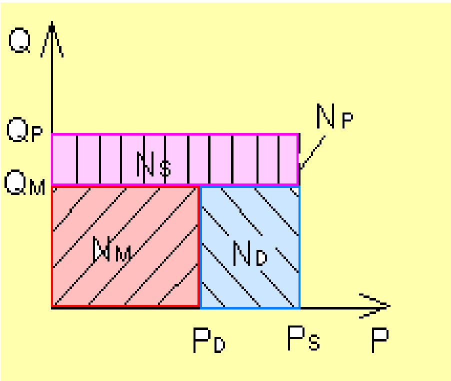

Fig. 9. The chart of power types in the system

Such a drive and control system is **totally ineffective** the more the power values required by the driven machine are higher, namely over 5 – 8 kW. Such system for drive and control of parameters **is used in control** or auxiliary **circuits** or as the main drive system in small power equipment or machines [2]. It should be mentioned that the flow limiter can be also installed on a bypass of the active circuit, as well as on the motor return line. All these variants are not efficient in terms of energy for high power hydrostatic systems.

In this chapter we presented **fixed** (constant) **displacement pump applications**, used in **open circuit,** in which **control** of energy and kinetic parameters of the circuit is based **on resistive principles,** with **large energy dissipation**, these systems being **ineffective in terms of energy**. These control principles are used in small power drives, fixed laboratory actuators, actuators for control circuits, and less in medium and high power drive equipment.
We will present below a series of modern methods for increasing energy efficiency of hydrostatic drive systems.

5. Methods for increasing energy efficiency of hydrostatic systems

Considering the above, to reduce load and flow losses, both local and linear/distributed, there have been proposed a range of techniques and methods to reduce these losses, which have been taken into account both during the design of components and when designing the functional diagrams of hydrostatic drive systems, in order to **improve energy efficiency** of these systems, as well as to **increase** their **functional performance**.

In the following we will present some of these techniques / methods and **modern concepts** for **increasing energy efficiency of hydrostatic systems**, by developing functional diagrams based on:

- **primary adjustment**, where variation of kinematic parameters is achieved by variation of the pump displacement / geometric volume, respectively by variation of flow;
- **secondary adjustment**, where variation of kinematic parameters is achieved by variation of the rotary hydraulic motor displacement / geometric volume;
- **mixed adjustment**, both pump displacement adjustment and rotary hydraulic motor geometric volume adjustment;
- **energy recovery**, where some of the energy is stored and reused in the next phase;
- **recirculation of** available **energy** between component equipment.

5.1. Hydraulic system with primary adjustment of displacement

Figure 10 presents a **primary adjustment diagram** of pump displacement, working in **open circuit,** for a system with open-circuit rotary motor, system that can be identical also for linear motor drives. The diagram presents a drive system for a pulling winch in which the pump *P* is set to zero displacement, on normal position (no control), displacement being adjusted by control of the operator between zero and maximum displacement [4]. Adjustment of winch speed is done from pump displacement value, and direction of motion - from directional valve *DP* control. The directional valve *DP* is an electrohydraulic pilot valve with the possibility of braking in the neutral position.

Figure 11 presents a diagram of a hydrostatic drive system with **primary adjustment**, but simple **closed circuit** and fixed displacement rotary motor, **with no motor brake.**

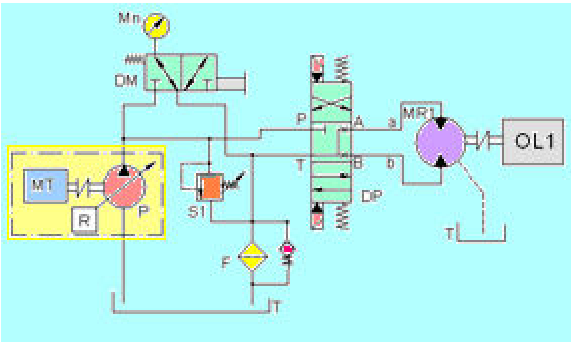

Fig. 10. Diagram of a hydrostatic system with primary adjustment and open circuit

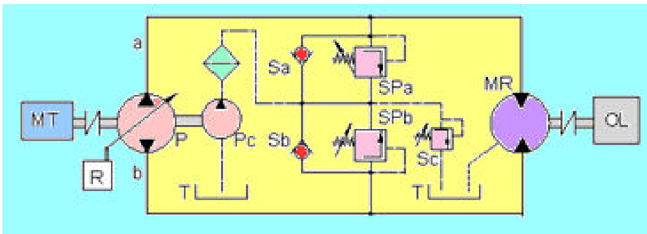

Fig. 11. Diagram of a hydrostatic system with primary adjustment and closed circuit

The diagram includes the **positive displacement pump** *P*, which is main pump, **with displacement primary adjustment by directional control valve *R***, which also performs the function of changing the distribution direction in the system. The pump is driven by the heat motor *MT*. The diagram includes a pump *Pc*, positive displacement, installed in tandem with the main pump, serving the system control and compensation circuit, which comprises the filter *F* and the drive pressure control valve *Sc*. To protect the power circuits *a* and *b* there is used the group of valves *SP* and *S*, *a* and *b*, which performs secondary and cavitation protection of system circuits. The final component of the system is the motor *MR* which does the actuation of the working body *OL,* or the equipment of the machine. **The directional control valve R** of the pump performs adjustment of pump *P* displacement, both according to the machine operator's order, and according to the adjustment requirements imposed by the technological process which the working body *OL* is involved in.

If the hydraulic units used in the diagram are **reversible units,** which can operate both as pumps and as hydraulic motors, the hydrostatic system **enables the use of braking energy** of a vehicle as motor brake, which it can capture and reuse at starting up. These energy recovery systems fall into the category of **hydrostatic regenerative systems**.

5.2. Hydraulic system with secondary adjustment of displacement

Secondary adjustment is an adjustment version in hydrostatic systems in which, for instance, an adjustable axial piston unit, with a predetermined operating pressure, seeks its adequate capacity to maintain a given rotational speed. In order words, there takes place capacity adjustment at constant pressure, so as to result in constant (or almost constant) motor speed [6], [7].
Figure 12 presents a hydrostatic drive with secondary adjustment at constant pressure. The capacity of the volumetric machine in the secondary unit of the winch drive system can be changed, in size and direction, by means of an adjustment device, type positioning cylinder, dependant on flow, controlled by a proportional solenoid valve, ensuring proportionality between geometric volume variation, consequently torque variation, and control signal [6].
In Figure 13 one can see the structure of a **hydrostatic servo system with secondary adjustment** under a **quasi-constant pressure** regime, maintained by the hydro-pneumatic accumulators (see Figure 12).
The structure is as follows: *1*-solenoid valve; *2*-positioning cylinder; *3*-secondary unit; *4*-electric speed voltage generator; *5*-speed regulator; *6*-amplifier; *7*-load; *8*-regulator of tilt angle; *9*-transducer of tilt angle; *10*-hydraulic coupling; *11*-pressure filter.

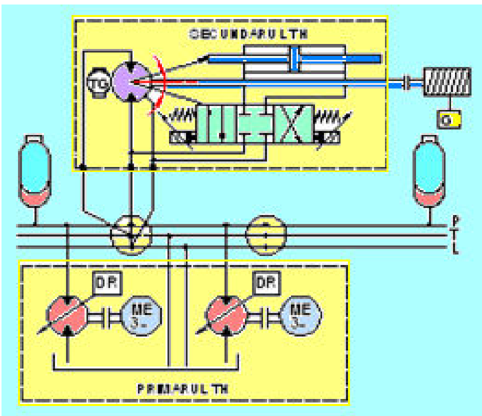

Fig. 12. Diagram of secondary adjustment

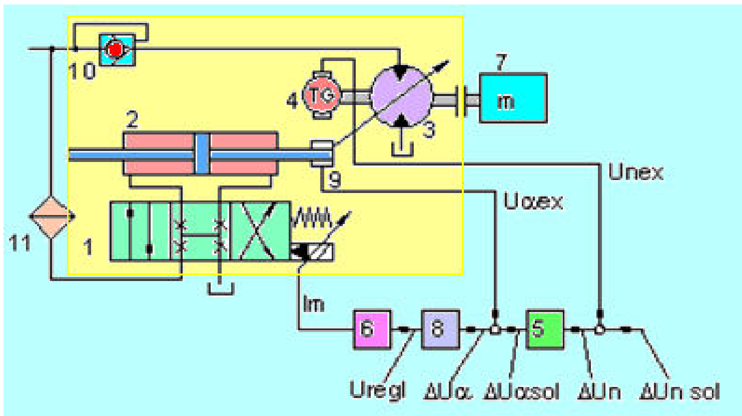

Fig. 13. Electrohydraulic control at secondary adjustment

Secondary adjustment is able to convert hydraulic energy into mechanical energy (operation as a motor) and mechanical energy into hydraulic energy (operation as a pump), **with almost zero losses**.
Secondary adjustment is possible both in **open circuit** (open-loop) and **closed** drive **circuit** (closed-loop).

5.3. Hydraulic system with mixed adjustment of displacement

For hydrostatic systems with special requirements concerning functional performance there is applied **the concept of mixed adjustment,** where one can find both **primary adjustment in pumps** and **secondary adjustment in hydraulic** drive **motors**. Figure 14 presents a hydraulic diagram with mixed adjustment, used in hydrostatic drive of a forging manipulator [4].
The forging manipulators require **precise movements** of the incandescent steel block, during processing by plastic deformation, on hydraulic forging presses. In addition to the **rotational movement** of the piece during forging, which is performed by the hydraulic motors M1 and M2, there must also be provided the manipulator **travel movement**, necessary to process and transport the ingot / piece between the oven and press, performed by the hydraulic motors M3 and M4. Figure 14, does not present the performing of translational movements that use hydraulic cylinders hydrostatically driven as well.

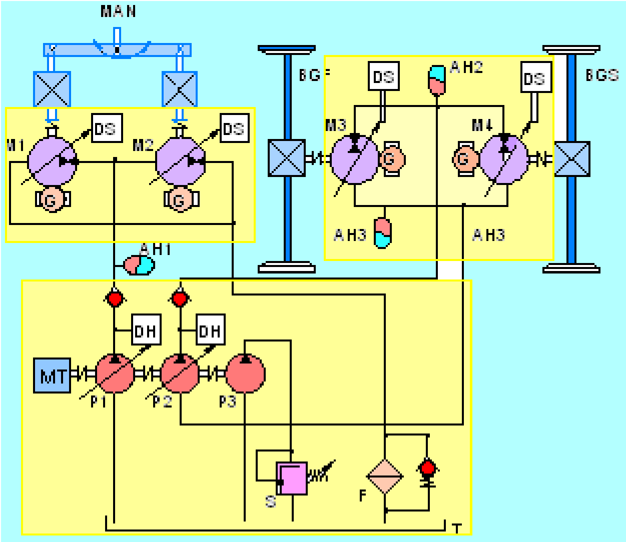

Fig. 14. Diagram of mixed adjustment used in forging manipulators

Diagram structure is as follows: *MT* – heat motor (or electric motor in some applications) *P1*, *P2*, *P3* – machine pumps; *P1* – pump serving the ingot rotation system, **in open circuit**, *P2* – pump serving the machine moving system, **in closed circuit**; both are **variable displacement pumps** with **primary adjustment** (*DH*), and *P3* is a **fixed displacement pump,** used for compensation of losses in the closed circuit of movement and for driving auxiliary equipment. *M1* and *M2* are **secondary adjustment** (*DS*) hydraulic motors, used for driving the ingot rotation system, while *M3* and *M4* are also variable displacement hydraulic motors with **secondary adjustment** (*DS*), used in the manipulator moving system.
The hydraulic motors, both for rotation (*M1* and *M2*) and for moving (*M3* and *M4*), work in **closed loops for tracking** the rotation angle, based on electrical signals given by speed voltage generators mounted on the rotation axes, processed by electronic control and adjustment systems, by which there is established correlation between the desired rotation angle and actual achieved angle.

5.4. Energy recirculation hydraulic system

Figure 15 presents a hydraulic diagram designed on the principle of energy recirculation. This refers both to **storing the energy** from pieces of equipment in an energy overplus (braking phases, load lowering, idle running etc.) and to **giving the stored energy** to other pieces of equipment that are in an energy shortage (acceleration phases, starting-up from rest, lifting etc.) [4].

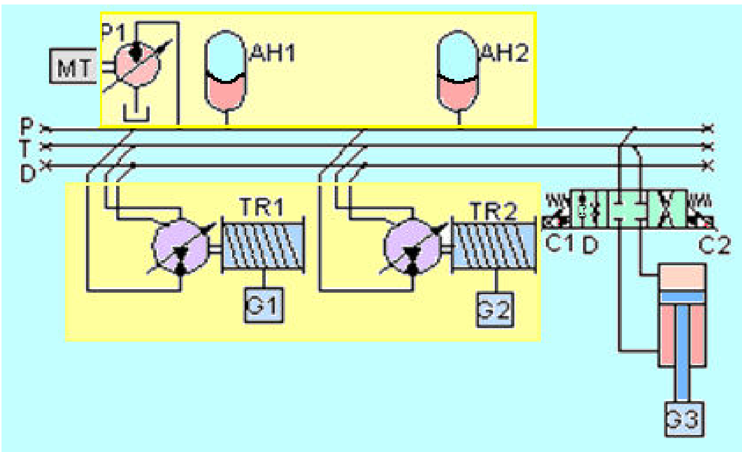

Fig. 15. Drive diagram with energy recirculation

The diagram consists of two winches *TR1* and *TR2*, which lift the loads *G1* and *G2*; these winches are driven by the hydraulic units *MR1* and *MR2*. The diagram also comprises the lever *G3*, driven by a linear motor *ML*, which is switched on or off the hydraulic buses *P* and *T* by use of directional control valve *D*. The machine is equipped with a heat motor *MT* and pump *P1*, which can work both as pump and motor. Both the pump *P1* driven by the heat motor *MT* and the rotary motors are **variable displacement,** and displacement adjustments are selected in such way that they facilitate energy delivery both from load to accumulators and from accumulator to load, as close as possible to technical and economical optimum, with minimum energy loss, that is **with higher** drive **efficiency**.

6. CONCLUSIONS

Based on the specialized literature, the paper presents some general considerations on pressure and flow losses in hydrostatic drive systems, relations for calculating them, as well as relations for assessing dissipated power and total efficiency of hydrostatic drive systems [8].
Following the presentation of energy losses and places where they occur, there is presented a set of **principles and modern concepts** for the design of hydrostatic drive diagrams, in order to minimize losses and **improve energy efficiency**.
By the above mentioned, the paper proves useful, especially to young professionals, in choosing how to design hydrostatic drive systems depending on actual requirements, with the aim of **developing high energy efficiency hydrostatic systems.**

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