

Hardware structures of hydronic systems for speed control

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Abstract. Most hydraulic actuating systems use constant flow pumps, for economic reasons. The resistive method is then used to control the speed of the actuated load. In the case of high performance systems the flow area is modified using analogical or numeric electric commands applied to proportional flow control devices. In the first part of the paper some hardware structures of hydronic actuating systems used for speed control are presented, and in the second part two experimental models of such systems are presented. Some aspects regarding the output improvement of such a system are also considered.

1. Introduction

Nowadays hydraulics technology has an increasing importance. Its qualities are used when high and very high forces must be developed with superior output, when loads rising and lowering must be rigorously controlled, when linear or rotational movements must be produced with important accelerations, accurate positioning of the actuated load, efforts transmission and working steps performing.

A hydraulic actuating system configuration may vary from simple actuating circuits to complex structures, controlled by programmable automatons or computers, which control, optimize and simulate the internal processes of the system [1]. The basic elements of a high performance hydraulic system are the proportional hydraulic devices. There are also used sensors and transducers, electronic circuits for signal processing, A/D and D/A converters, controllers and/or microprocessors.

These systems are developed on the basis of advanced control theories. The electronic control system gathers information from the system sensors, processes and structures the data, makes decisions on the basis of an internal program and then transmits control signals to the electromechanical actuators at the level of the proportional hydraulic devices. This way pressures and flow, displacements, speeds, accelerations, forces and torques developed by the motors of the system can be controlled.

2. The basic hardware structure of a hydronic system for speed control

The moving speed of the mobile assembly of a hydraulic motor is controlled by modifying the quantity of liquid passing through the motor in a unit of time. The variation of the quantity of liquid (the flow) may be obtained by two methods: *the volumetric method*, consisting in modifying the flow of the pump, at variable pressure according to the actuated load, and *the resistive method*, consisting in modifying the local resistance on the supply or the exhaust circuit of the motor, at constant pressure, using a variable hydraulic resistance. Most hydraulic actuating systems use constant flow pumps, for economic reasons. In this case the control of the flow is obtained using the second method, which is an



accurate one but it can lower the output of the system due to the loss of the local pressure in the variable resistance area and the discharge to the tank of the flow excess through the safety valve. The control of the actuated load speed is performed by a linear or rotational hydraulic motor *MHL* (figure 1), modifying the flow q_{aM} by means of a proportional hydraulic device *EHP* mounted between the points *a* and *b* of the circuit. This device can be chosen from table 1.

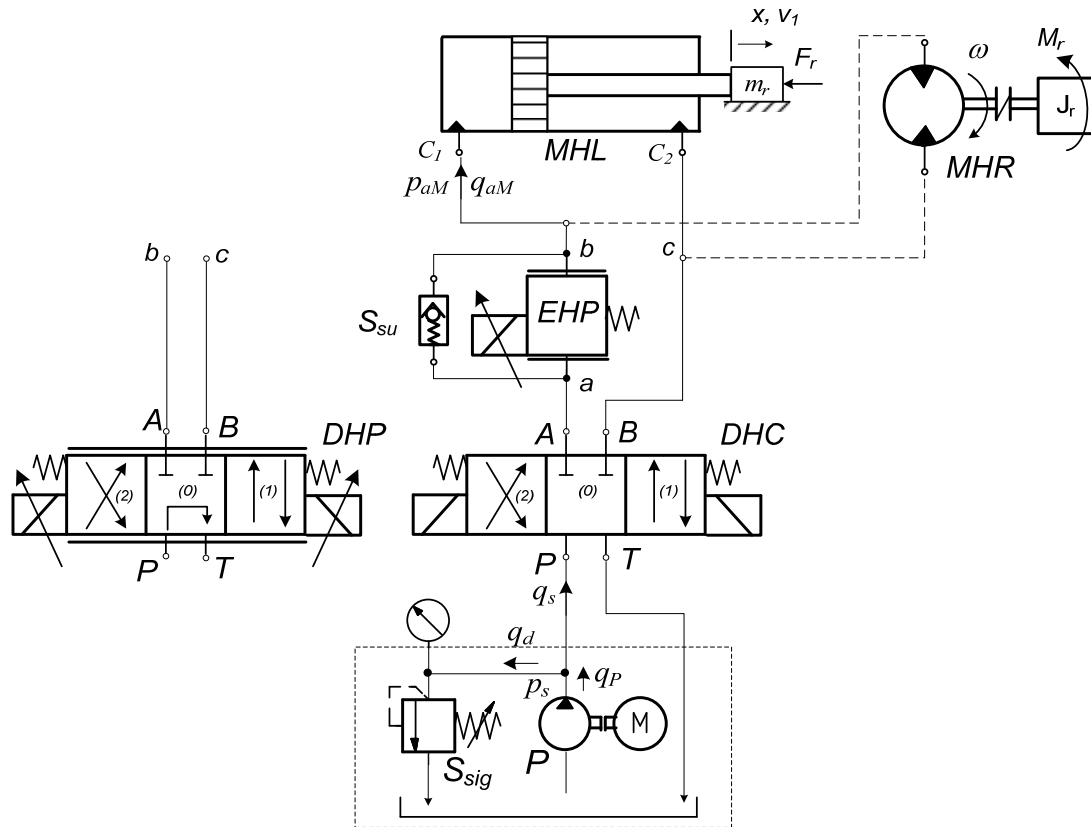


Figure 1. The speed control system configuration.

Considering a rotational hydraulic motor, the system can control the speed ω only for the shown direction, when the directional control valve *DHC* is in position (1). The motor is stopped when the directional control valve is in position (0), when the two consumers *A* and *B* (the active chambers of the hydraulic motor *MH*) are blocked. Because the directional control valve is a classic one, the blocking of the consumers is not perfect and the position of the shaft cannot be conserved. Some solutions for this inconvenient will be presented further.

The movement direction is inverted when the directional control valve *DHC* is in position (2), when the proportional device is short-circuited and the flow is exhausted from the motor to the tank through the one way valve S_{su} .

In these conditions, the rotation speed of the motor shaft for the shown direction is given by:

$$\omega = \frac{\eta_{VM}}{V_M} \cdot q_{aM} \quad (1)$$

The necessary pressure at the motor input is given by:

$$p_{aM} = \frac{1}{\eta_{VM} \cdot V_M} \cdot M_r \quad (2)$$

Table 1. Proportional hydraulic devices.

	Proportional throttle	Two ways flow regulator	
Principle scheme			
Symbol			
Characteristic $q = f(U_E)$			

The first variant of the system is therefore obtained using a proportional hydraulic throttle.

Using the proportional throttle-safety valve *Ssig* assembly, the supply flow of the motor q_{aM} can be controlled, and it is given by:

$$q_{aM} = \begin{cases} q_p & \text{if } p_s < p_r \\ \sqrt{\frac{\Delta p}{\Delta p_{nom}}} \cdot q_{nom}(u_c) & \text{if } p_s = p_r \end{cases} \quad (3)$$

where:

q_p is the flow supplied by the pump to the sistem;

p_s is the pressure before the safety valve *Ssig*;

p_r is the pressure regulated by the safety valve *Ssig*;

Δp is the pressure drop on the proportional hydraulic throttle, where:

$$\Delta p = p_r - p_{aM}(M_r) = f(M_r) \quad (4)$$

$q_{nom}(u_c)$ is the flow characteristic of the proportional throttle (table 1) and it is experimentally determined by the producer.

This characteristic is nonlinear at both ends; at the beginning the nonlinearity is given by the positive overlap of the slide (the variant with a slide is the most used), and at the end the nonlinearity is given by the hydraulic saturation.

The flow through the throttle is also dependent on the pressure drop on the throttle Δp ; usually the characteristic corresponds to the situation when $\Delta p = \Delta p_{nom} = 5 \text{ bar}$. This does not mean that the device cannot function for another pressure difference. In such a case the flow is given by:

$$q_{DHP} = \sqrt{\frac{\Delta p}{\Delta p_{nom}}} \cdot q_{nom}(u_c) \quad (5)$$

Equation (3) highlights the following aspects:

- the speed ω of the mobile assembly of the motor can be changed by modifying the flow (modifying the internal flow area) by means of the proportional hydraulic throttle *DHP*, coupled with a safety valve adjusted accordingly to the functioning conditions;
- the controlled speed also depends on the pressure p_{aM} , which depends on the load M_r ; the variations of the load will determine uncontrolled variations of the speed of the mobile assembly of the motor.

In order to control the speed, a pressure loss must be accepted, depending on:

- the flow q_d discharged through the safety valve to the tank;
- the pressure drop Δp on the throttle.

For this reason the discharged flow must not exceed 20% of the flow supplied by the pump, and the pressure loss on the throttle must be limited to 2...6 [bar].

A second variant of hydraulic system for controlling the speed of the actuated load is obtained using a two ways flow regulator as a proportional hydraulic device (figure 1), which allows the control of the flow necessary for a consumer to the desired value with a good accuracy and maintains it constant in some limits, even if the pressure of the fluid varies while functioning. The two ways flow regulator (table 1, column 3) consists of a proportional throttle *Dr* and a pressure variations compensating device *DC*, with an auto-adjustable area which is serially mounted with the throttle area. The device, which is in fact a pressure balance, reacts under the action of the pressures upstream and downstream the throttle and of a low rigidity spring.

The pressure difference on the throttle area is given by:

$$\Delta p = p_a - p' = p_a - \Delta p_{com} - p_b = \frac{F}{S_s} \quad (6)$$

where:

Δp_{com} is the pressure difference on the controlled area of the mobile element – the slide *s* of the compensating device *DC*; any variation of the pressures p_a and p_b determines the modification of the slide *s* position against the body and so the modification of the flow area controlled by the slide and of the pressure loss Δp_{com} ;

F is the force developed by the spring "a" in addition with the flow force on the area controlled by the slide *s*;

S_s is the slide area.

This way the pressure difference Δp is maintained approximately constant. The load loss on the throttle is not rigorously constant as desired, due to the hydrodynamic force; as a consequence the controlled flow may have variations of 5...10% depending on the flow value and on the values of the pressures upstream and downstream the throttle.

In the case of a hydraulic actuating system with a constant flow pump, this device accomplishes its task along with the safety valve of the system.

Figure 2 shows the characteristic $q = f(p_b)$ for such a device.

If the proportional hydraulic device *EHP* (figure 1) is a three ways flow regulator of the derivation type (table 1, column 4) a *third variant* of a hydraulic system for controlling the actuated load speed. This regulator has the same energetic advantages as the two ways regulator mounted in derivation [2] and assures an accurate control of the flow required by the motor. This flow is not influenced by the

pressure variations or by the variations of the flow required by the motor. The principle scheme of such a regulator shows that it differs from the two ways regulator by the way the throttle area and the area materialized by the slide of the compensator are disposed: they are connected in parallel, not serial as in the case of the two ways regulator.

In the case of the three ways flow regulator all the flow supplied by the pump enters into the regulator and is divided as follows: the flow required by the motor passes through the throttle area and the flow excess is directed to the tank by opening the compensator slide and through the third orifice of the device.

In this case the pressure difference on the throttle area is given by:

$$\Delta p = p_a - p_b = \Delta p_{com} - p_b \quad (7)$$

The increase of the pressure p_b , for example, will determine a decrease of the flow area controlled by the compensator and so an increase of the pressure difference Δp_{com} , having the effect of compensating the variation of p_b . In this case the pressures p_a and p_b have identical variations.

The characteristic $q = f(p_b)$ for such a device is presented in figure 3.

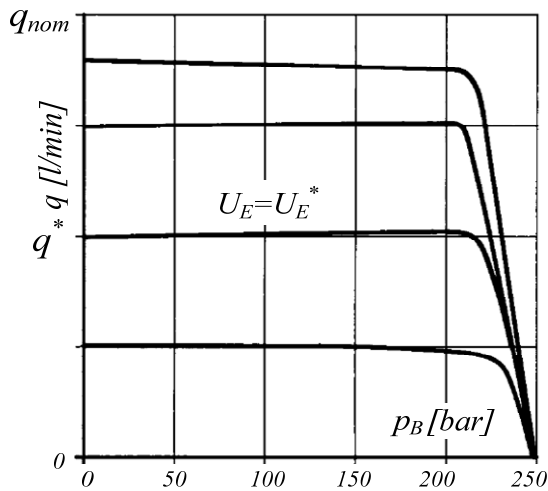


Figure 2. System with constant flow pump.

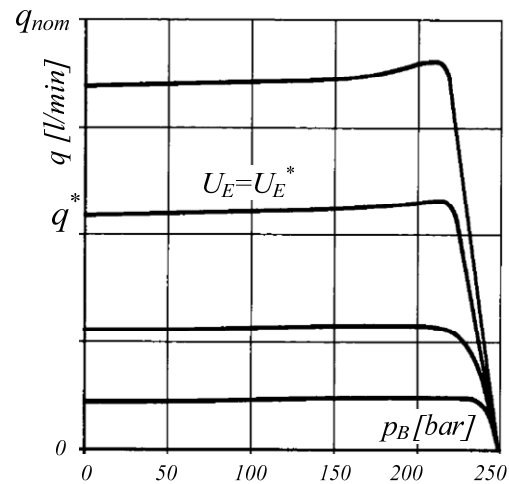


Figure 3. System with three ways flow regulator.

The flow variations at low and high working pressures are also due to the hydrodynamic force. The hydrodynamic force always tends to shut the flowing area and it is added to the spring force, producing an increase of the load loss and not it decrease as desired.

In order to assure the compensator functioning, the flow q_a at the input of the regulator must be superior to the controlled flow q_b (that is true for all types of regulators). Such a regulator can be mounted only upstream the motor; if the regulator is mounted downstream the motor or in parallel with it, the control of the mobile assembly speed is not possible, because this regulator does not limit the input flow. More than that, the orifice b cannot be totally closed (by a downstream directional valve, for example), because this would determine the simultaneously closing of the orifice t , and in this case the flow supplied by the pump would be directed to the tank through the safety valve of the system. On the other hand, the output orifice t must be connected to the tank; using this orifice for supplying another consumer would cause the loss of the pressure variations compensating capacity.

The resistant torque M_r may vary while functioning. Figure 4 shows, according to the system variant, the variation in time of the pressure difference Δp on the throttle area (figure 4b₁, 4b₂ and 4b₃), of the input flow q_{aM} of the motor and of the rotation speed of the load ω (figure 4c₁, 4c₂ and 4c₃) when the resistant torque M_r has a slope variation.

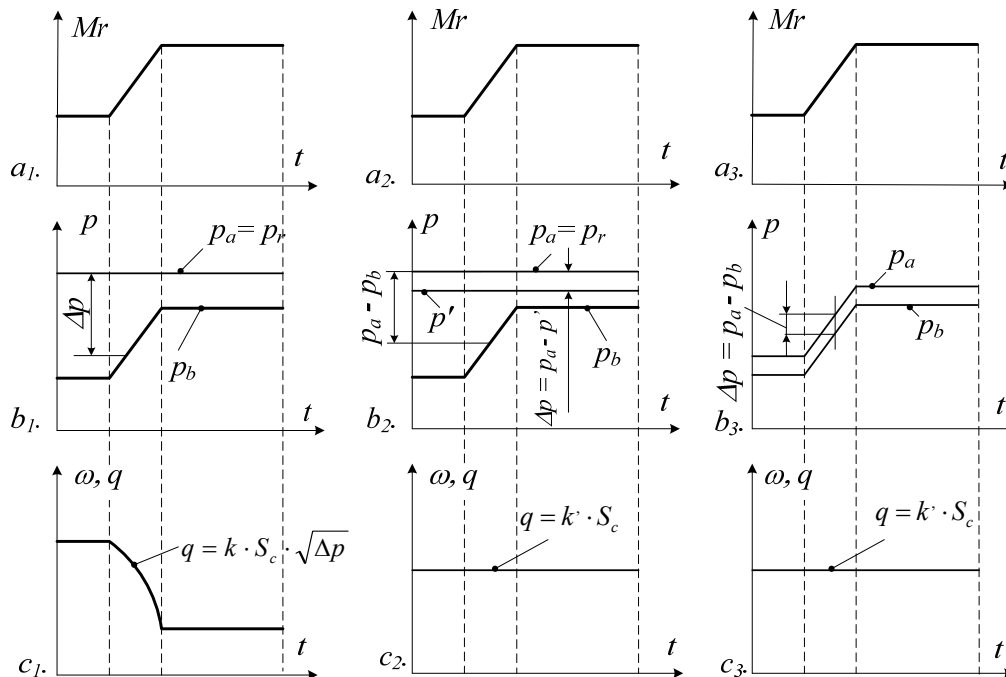


Figure 4. The variation in time of Δp , q_{aM} and ω for a slope variation of M_r .

Table 2. A comparative analysis of the energetic balance for variants I and III.

Variant I	Variant III
$N_e = N_i + \Delta N_{Sig} + \Delta N_{DrP}$ $\Delta N_{Sig} = q_d \cdot p_r = (q_P - q_{aM}) \cdot p_r$ $\Delta N_{DrP} = q_{aM} \cdot (p_r - p_b) = q_{aM} \cdot (p_r - p_{aM})$ $N_i = q_P \cdot p_r$ $N_e = q_{aM} \cdot p_{aM}$ $\eta_I = \frac{N_e}{N_i} = \frac{q_{aM}}{q_P} \cdot \frac{p_{aM}}{p_r}$	$N_e = N_i + \Delta N_{DC} + \Delta N_{DrP}$ $\Delta N_{DC} = q'_d \cdot p_a = (q_P - q_{aM}) \cdot p_a$ $\Delta N_{DrP} = q_{aM} \cdot (p_a - p_b) = q_{aM} \cdot (p_a - p_{aM})$ $N_i = q_P \cdot p_a$ $N_e = q_{aM} \cdot p_{aM}$ $\eta_I = \frac{N_e}{N_i} = \frac{q_{aM}}{q_P} \cdot \frac{p_{aM}}{p_a}$

where:

- N_i - hydraulic input power;
- N_e - hydraulic output power;
- ΔN_{Sig} - the power loss due to the discharge to the tank of the flow excess of the pump through the safety valve S_{sig} ;
- ΔN_{DrP} - the power loss due to the pressure difference on the area controlled by the throttle;
- ΔN_{DC} - the power loss due to the discharge to the tank of the flow q'_d through the third orifice of the device.

Table 2 presents a comparative analysis of the energetic balance for the first and the third variants of systems considered.

For the last proposed system (with three ways flow regulator) the pressure difference on the throttle $\Delta p = p_a - p_b$ (figure 4, b₃), remains approximately constant. In this case the device works independently on the safety valve of the system. The flow surplus $q'_d = q_p - q_{aM}$ is overflowed through the third orifice of the device t . The safety valve S_{sig} has now only the role of a safety element in the system (it limits the pressure to the maximum value p_r). While the system functioning it is closed and $q_d = 0$.

A comparison between the first variant and the third one shows that $\eta_{III} > \eta_I$.

This is also shown on the graphic from table 2, the second column, by ΔN^* , which represents the power economy obtained with variant III compared to variant I.

As mentioned, the system presented in figure 1 can control the speed only on one moving direction. In order to control the speed on both moving directions proportional directional valves with the center closed are most often used. The directional valve assures the moving direction reversal and in addition has the role of an adjustable throttle for flow control. Opposite to the proportional throttle, here two ways are simultaneously controlled, from P to A or B and from B or A to T .

Replacing the hydraulic classic directional valve DHC – proportional hydraulic device EHP assembly of the system in figure 1 with a proportional hydraulic directional valve DHP a new structure is obtained that can control the actuated load speed on both moving directions and assure the firm stop of the mobile assembly.

In order to compensate the influence of the actuated load variation on the controlled speed the system in figure 5 is used.

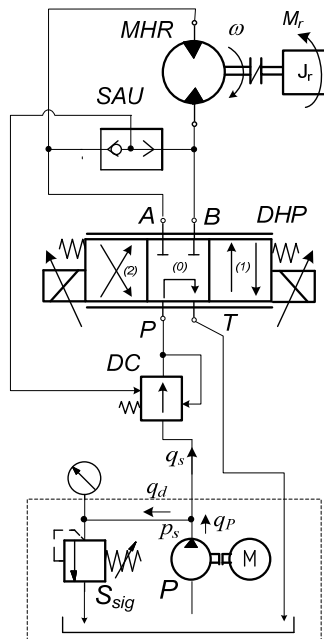


Figure 5. Controlled speed system.

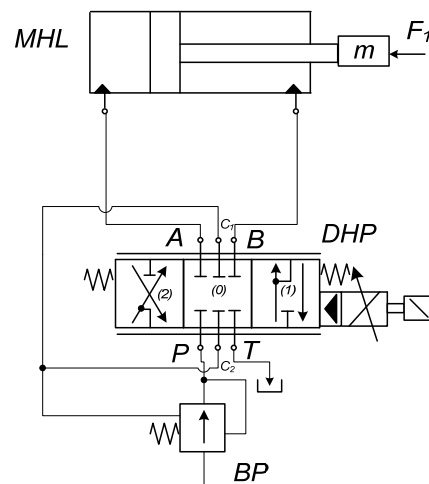


Figure 6. Proportional hydraulic directional valve with 6 orifices.

In this case, on the input circuit of the directional valve was inserted a compensating device (a “pressure balance”) DC which forms, together with the directional valve (in fact a one way throttle), a two ways flow regulator. In order to use only one pressure balance for both moving directions the valve SAU was used.

Rexroth produces a proportional hydraulic directional valve with 6 orifices (figure 6): the usual 4 orifices (pressure P , tank T , consumers A and B) and two more orifices C_1 and C_2 for connecting a pressure balance BP .

When the rigorous conservation of the actuated load position is required, a system as shown in figure 7 can be used. The system contains in addition the one way valves S_1 and S_2 , releasable, hydraulic controlled.

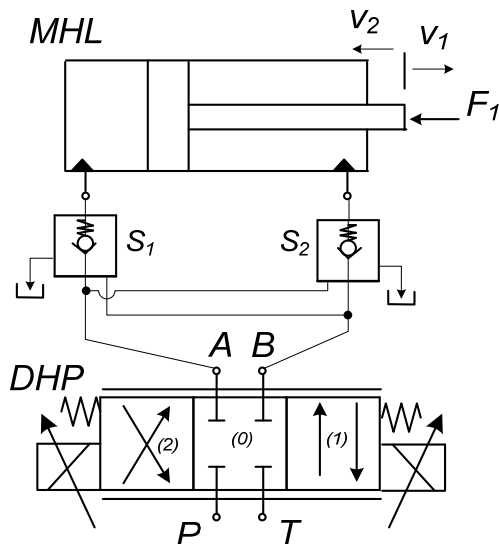


Figure 7. Hydraulic controlled system.

3. Experimental models of hydronic systems for speed control

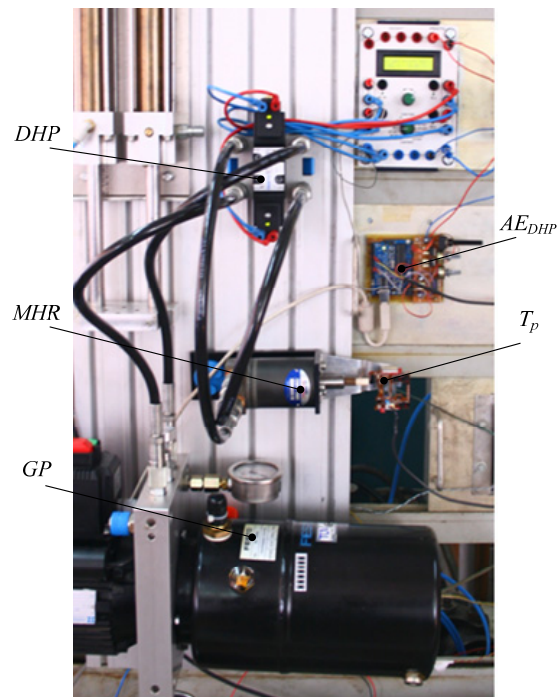
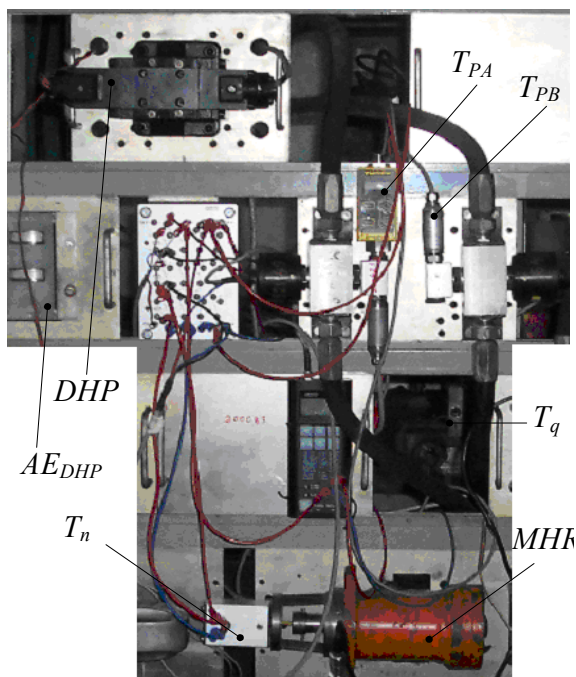
Two experimental models of hydronic systems developed by the authors are presented in Table 3. Both models, used in the laboratory of “Robotics, Actuating and Accurate Automation Systems” of the Mechanical Engineering and Mechatronics Faculty from POLITEHNICA University of Bucharest, have the functional scheme presented in figure 1, the variant with a proportional hydraulic directional valve.

Table 3. Experimental models of the hydronic systems.

Particularities:

- The system performs the control of the motor shaft rotation speed for both rotation directions, and there are also measured in real time the pressures in the active chambers of the motor; these tasks are achieved by means of three transducers integrated within the structure of the system, one for rotation speed and the other two for pressure;
- The hydraulic system is built using industrial hydraulic elements produced by Rexroth;
- The control of the system is performed with a PC and a modular I/O system type FieldPoint as an interface;
- The command program was developed using LabView graphic programming environment.
- The system performs the control of the motor shaft speed and position for both rotation directions; a transducer for angular position is used to achieve this task;
- The hydraulic system is built using didactic hydraulic elements produced by FESTO;
- The electronic control unit is built around a microprocessor; the rotation speed transducer is replaced by an angular position incremental transducer T_p ;
- More control algorithms were developed: for controlling the direction of rotation, for positioning, for controlling the speed and rotation direction of the motor using a given moving law, PID regulator for controlling the angular position of the motor shaft;
- These algorithms were implemented as a programs library; the programs are saved in the microcontroller memory and they can be run independently after the user sets the input value for every type of signal (amplitude, period, phase etc.); for the communication with the PC a RS232 bus is used;
- The microcontroller is programmed using C++ and the PC runs LabView graphic program.

Experimental models:



4. Conclusions

The informatization of the hydraulic actuating systems was an important step forward for them. The concept of „hydronics” was naturally developed by synergistic jointing of three domains: *hydraulics* – *electronics* – *informatics*. This way, *hydronic* actuating systems appeared. The configuration of such a system may vary from simple actuating circuits to complex structures, controlled by programmable automatons or computers, which control, optimize and simulate the internal processes of the system. The basic elements of a high performance hydraulic system are the proportional hydraulic devices. There are also used sensors and transducers, electronic circuits for signal processing, A/D and D/A converters, controllers and/or microprocessors. These systems are developed on the basis of advanced control theories and they can control pressures and flow, displacements, speeds, accelerations, forces and torques developed by the motors of the system can be controlled. In this paper some hardware structures of hydronic actuating systems used for the actuated load speed control are presented, and two experimental models of such systems built and tested by the authors are also presented.

5. References

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