Recirculation of Hydro-Mechanical Power at Stands for Testing Endurance of Hydraulic Cylinders

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Abstract

Endurance testing on volumetric machines and hydraulic equipment involves high energy consumption, as it takes a long time to carry them out and they are conducted at rated power. Energy consumption of the test stand can be reduced by a special construction of the pumping group. This includes two hydraulic volumetric machines, a pump and a motor, connected in a closed hydraulic circuit and mechanically coupled to an electric motor. Hydropower produced by the pump is reused in order to drive the pump through the engine. Thus, the power delivered in the system must cover the difference between the power consumed by the pump and the one supplied by the motor, and energy-saving process is called "hydro-mechanical power recirculation”.

The material shows the influence of hydro-mechanical power recirculation on energy consumption at stands for testing endurance of hydraulic cylinders, by: comparative analysis of energy consumption for two modes of operation of a mini-stand for testing endurance of hydraulic cylinders, coupled and uncoupled to hydro-mechanical power recirculation; demonstration of energy saving at these stands for testing endurance that operate on the principle of hydro-mechanical power recirculation.

KEYWORDS: testing endurance, hydro-mechanical power recirculation

1. The principle of hydro-mechanical power recirculation

Innovative aspect of this material is in extending the application of the principle of "hydro-mechanical power recirculation" to stands for testing endurance of hydraulic cylinders. In this respect, we start from considerations in the specialized literature on energy recovery during endurance tests on rotary volumetric machines /1/ and demonstrate the possibilities for energy recovery during endurance tests on hydraulic cylinders.

1.1. Energy recovery during endurance tests on rotary volumetric machines

Energy-saving process known as "hydro-mechanical power recirculation" can be materialized with several diagrams which differ in the way of compensation of power losses. In the case of mechanical compensation the power source is an electric motor. When testing endurance of two rotary volumetric machines, pump and motor, if one of them is adjustable the diagram in **Figure 1** is used, where the two machines are coupled through an electric motor, rotational speeds of the volumetric machines and the electric motor being equal /2/.

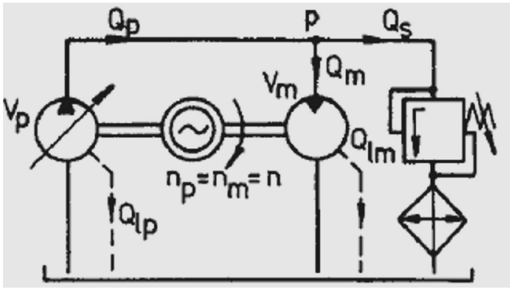


Figure 1: Testing endurance of two rotary volumetric machines

$$n_p = n_m = n \tag{1}$$

If the flow supplied by the pump, Q_p, is equal to the one consumed by the motor, Q_m, the pump discharge pressure, p, is practically null: as pump capacity, V_p, is increased compared to motor capacity, V_m, its discharge pressure increases to pump off the excess flow through clearances and gaps. Normally-closed pressure valve limits the pressure p at the rated value specific to the tested machines.

Power supplied by electric motor, N_e, represents the difference between the power absorbed by pump, N_p, and the one supplied by the hydraulic motor, N_m:

$$N_e = N_p - N_m \tag{2}$$

$$N_p = \frac{p \cdot Q_{tp}}{\eta_{tp}} = \frac{p \cdot n \cdot V_p}{\eta_{tp}} \tag{3}$$

$$N_m = p \cdot Q_{tm} \cdot \eta_{tm} = p \cdot n \cdot V_m \cdot \eta_{tm} \tag{4}$$

$$N_e = p \cdot n \cdot V_m \left(\frac{V_p}{V_m} \cdot \frac{1}{\eta_{tp}} - \eta_{tm} \right) = N_{tm} \left(\frac{V_p}{V_m} \cdot \frac{1}{\eta_{tp}} - \eta_{tm} \right) \tag{5}$$

$$Q_p = n \cdot V_p \cdot \eta_{vp} = Q_m = n \frac{V_m}{\eta_{vm}} \tag{6}$$

$$\left(\frac{V_p}{V_m}\right)_{\min} = \frac{1}{\eta_{vp} \cdot \eta_{vm}} \tag{7}$$

$$\left(\frac{N_e}{N_m}\right)_{\min} = \frac{1}{\eta_{tp} \cdot \eta_{vp} \cdot \eta_{vm}} - \eta_{tm} \tag{8}$$

For example, for volume yields

$$\eta_{vp} = \eta_{vm} = 0.95 \tag{9}$$

and total yields

$$\eta_{tp} = \eta_{tm} = 0.9 \tag{10}$$

is necessary for the minimum ratio of capacities of the two volumetric machines to be

$$\left(\frac{V_p}{V_m}\right)_{\min} \cong 1.1 \tag{11}$$

We obtain in this case a minimum ratio of powers of the two motors of

$$\left(\frac{N_e}{N_m}\right)_{\min} \cong 0.33 \tag{12}$$

1.2. Energy recovery during endurance tests on hydraulic cylinders

When testing endurance of hydraulic cylinders /3/, we propose to be used the diagram in **Figure 2**, where the electric motor can be connected directly to the two rotary volumetric machines, pomp and motor, or through gear transmission, one stage, with ratio 1:1. The diagram in Figure 2 includes: a pumping group with hydro-mechanical power recirculation, the pump capacity greater than capacity of hydraulic motor, and the electric motor **1**, two axes, is connected directly to pump **2** and hydraulic motor **3**; a hydraulic directional control valve 4/3, **5**; two hydraulic cylinders, one for testing, **6** and the other for load, **7**, their rods mechanically connected, **8**, fixed in a closed metal frame; two normally-closed pressure valves, one, **4.1**, for limiting pump pressure, and the other, **4.2**, for adjusting the pressure at which endurance tests are conducted; four way-valves, **9.1**, **9.2**, **9.3**, **9.4**, that allow alternatively inlet / outlet of oil into / from the chambers of variable volume of the load cylinder; a check- valve, **9.5**, allowing oil inlet into the hydraulic motor, directly from the tank, when the hydraulic directional control valve is not actuated, the electric motor works and pump discharges at the tank **10**.

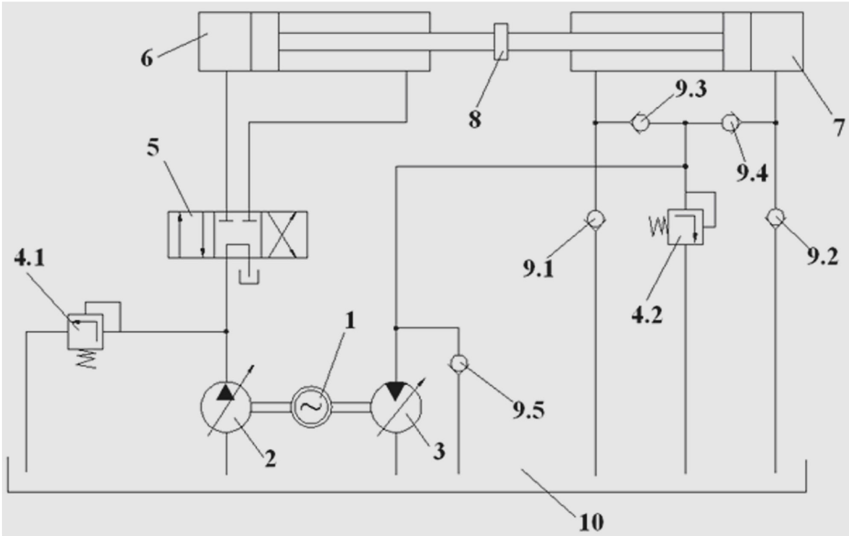


Figure 2: Hydraulic diagram for testing endurance of hydraulic cylinders

In this case the power supplied by the electric motor, N_e , is the difference between the sum of power absorbed by pump, N_p , and by the two hydraulic cylinders, N_c , and the power supplied by the hydraulic motor N_m :

$$N_e = N_p + 2N_c - N_m \tag{13}$$

Power consumption of hydraulic cylinders is mainly due to friction in sealing systems and is given by relation:

$$N_c = p \cdot n \cdot V_p \cdot \eta_{vp} (1 - \eta_{tc}) \tag{14}$$

Replacing in (13) N_p , N_c , N_m with relations (3), (14) and (4), we obtain:

$$\begin{aligned} N_e &= \frac{p \cdot n \cdot V_p}{\eta_{tp}} + 2p \cdot n \cdot V_p (1 - \eta_{tc}) - p \cdot n \cdot V_m \cdot \eta_{tm} = \\ &= p \cdot n \cdot V_m \left[\frac{V_p}{V_m} \cdot \frac{1}{\eta_{tp}} + 2\frac{V_p}{V_m}(1 - \eta_{tc}) - \eta_{tm} \right] \end{aligned} \tag{15}$$

Taking into consideration relation (7), we obtain:

$$\left(\frac{N_e}{p \cdot n \cdot V_m} \right)_{\min} = \left(\frac{N_e}{N_m} \right)_{\min} = \frac{1}{\eta_{vp} \cdot \eta_{vm} \cdot \eta_{tp}} + 2\frac{1 - \eta_{tc}}{\eta_{vm}} - \eta_{tm} \tag{16}$$

Considering that volume yields of pump and motor are given by (9), and total yields of pump, motor and cylinders are given by (10), we obtain:

$$\left(\frac{N_e}{N_m}\right)_{\min} \cong 0.54 \tag{17}$$

2. Numerical simulation of hydro-mechanical power recirculation during endurance tests on hydraulic cylinders

Diagram for testing endurance of hydraulic cylinders in Figure 2 was the basis for developing the simulation model in AMESim, presented in **Figure 3**.

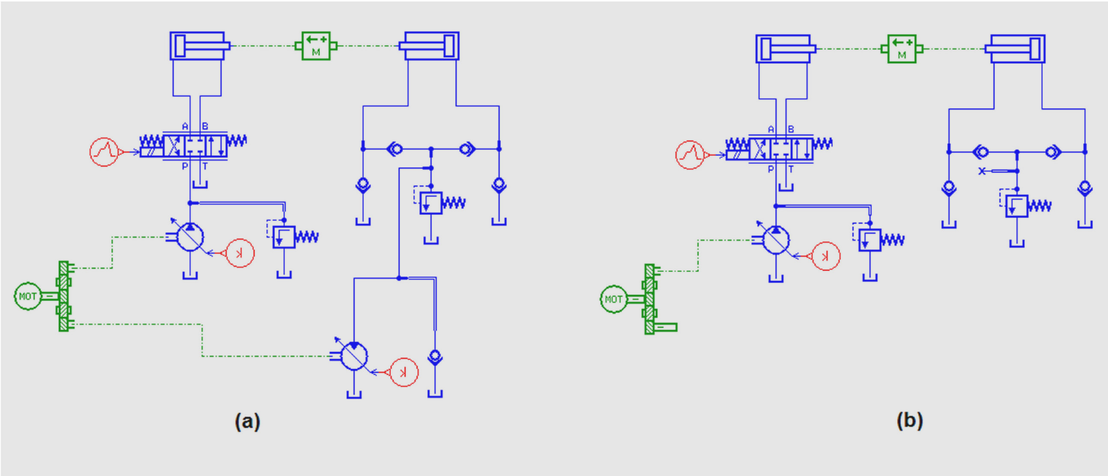


Figure 3: Simulation model in AMESim

The simulation model in Figure 3 has two versions: (a)- version with energy recovery, in which the two volumetric machines are coupled to the shaft of the electric motor through a gear transmission, and through the normally-closed pressure valve of the load cylinder passes the difference between the flow discharged by pump and the flow intake of the hydraulic motor and (b)- version without energy recovery, in which the hydraulic motor is decoupled from the gear transmission, and through the normally-closed pressure valve of the load cylinder passes all pump flow.

The real parameters of the main AMESim component sub-models are: for hydraulic cylinders – piston diameter = 50 mm, rod diameter = 28 mm, displacement of piston = 0.3 m; for the electric motor - shaft speed = 1450 rev/min; for the hydraulic motor-motor displacement = 4 cc/rev; for the hydraulic pump - pump displacement = 6 cc/rev; for the normally-closed pressure valve of the load cylinder - relief valve cracking pressure = 20 bar; for the piecewise linear signal source - output at start of stage 1 =40 null, output at end of stage 1 = 40 null, duration of stage 1 = 4 s, output at start of stage 2 = -40 null, output at end of stage 2 = -40 null, duration of stage 2 = 2.5 s.

Running the two versions of the simulation model was performed as follows: a control signal of +40 mA was applied to the hydraulic distributor, that is the slide valve was displaced at the maximum to one side, so that the hydraulic cylinder piston to perform a stroke of nearly 0.3 m, then a signal of -40 mA was applied to the distributor, that is the slide valve was displaced at the maximum to the opposite side, so that the piston to perform the same stroke in the reverse direction. During a dual piston stroke the following dynamic features were raised: piston rod displacement and variation of control signal applied to the distributor, **Figure 4**; pressure variation at the input of normally-closed pressure valve of the load cylinder, in versions with/without energy recovery, **Figure 5**; pressure variation in the junction "P" of the hydraulic distributor, in versions with/without energy recovery, **Figure 6**; variation of flow discharged through the normally-closed pressure valve of the load cylinder, in versions with/without energy recovery, **Figure 7**; variation of torque at the electric motor shaft, in versions with/without energy recovery, **Figure 8**.

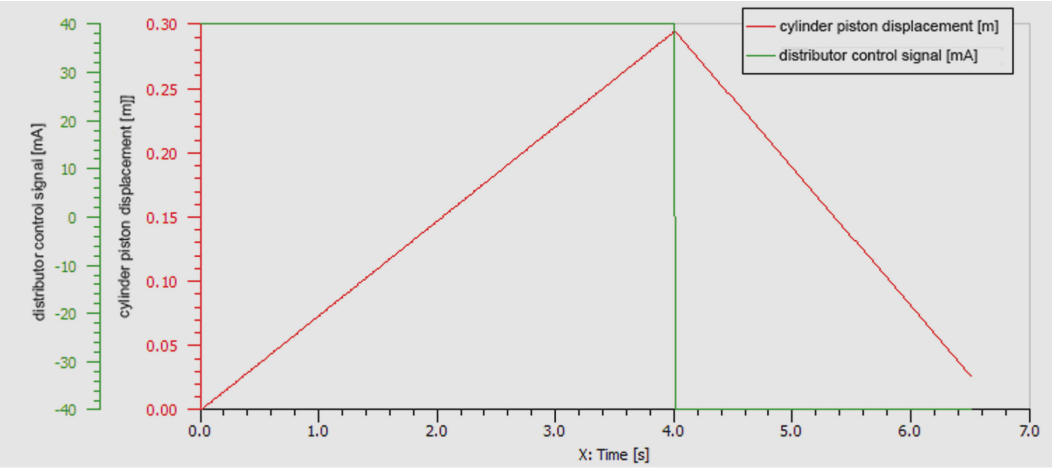


Figure 4: Variation over time of piston displacement depending on the control signal

Characteristics in the Figure 4 show the effect of applying at the hydraulic distributor two successive control signals, namely one of 40mA, over a period of 4 s, and one of -40mA, over a period of 2.5 s, upon the position of the piston of one of the two hydraulic cylinders. It can be noticed that under the effect of the first command piston moves from 0.00 m to 0.29 m, while under the effect of the second command piston moves from 0.29 m to 0.02 m. Periods for performing the forward stroke respectively the backward stroke of the piston are different because the hydraulic cylinder, being differential, has different active surfaces. This way of performing actuation of hydraulic cylinders stays identical for both types of AMESim simulation models, namely with and without energy recovery.

The second common characteristic of the two simulation models is represenred by maintaining an almost constant pressure at the input of the normally-closed pressure valve of the load cylinder, as shown in Figure 5.

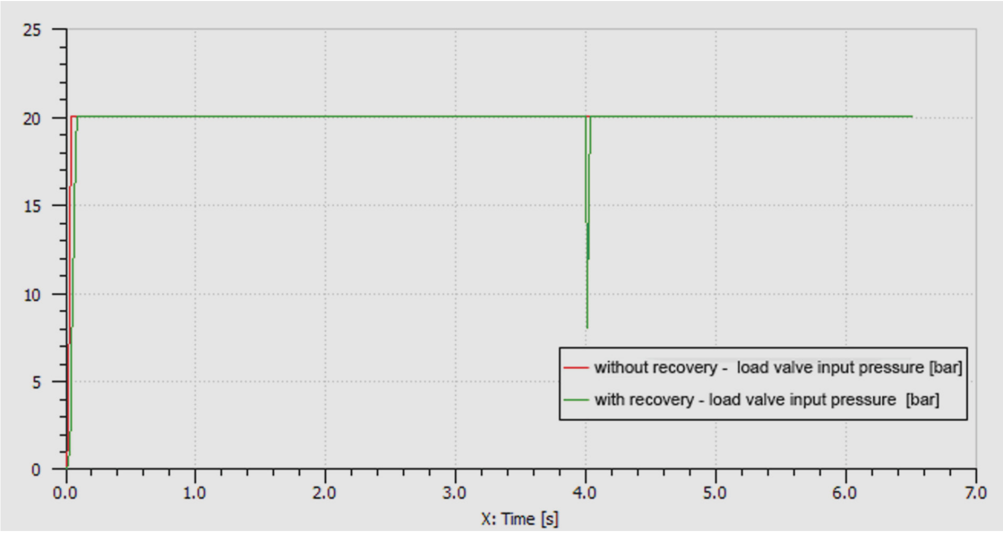


Figure 5: Variation of pressure at the input of the load simulation valve

It can be noticed, as shown in Fig. 5, that during the simulation of an endurance test the test pressure of the two hydraulic cylinders is of approximately 20bar, both in the simulation model version with energy recovery, and in the simulation model version without energy recovery.

The third common characteristic is given by the variation of pressure in the junction "P" of the hydraulic distributor, as shown in Figure 6.

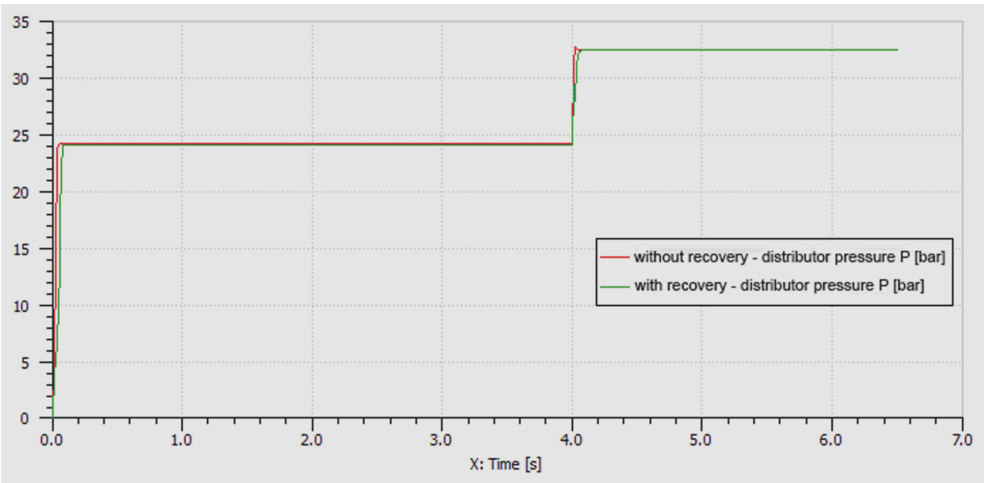


Figure 6: Variation of pressure in the junction "P" of the hydraulic distributor

As shown in Fig.6, depending on the type of active chamber of the differential hydraulic cylinder, piston chamber or rod chamber, which cause different friction resistances in the sealing systems, pressure in the junction "P" of the hydraulic distributor has the approximate value of 24 bar, and respectively of 33 bar.

The two simulation models can be distinguished in terms of variation of flow discharged through the normally-closed pressure valve of the load cylinder, as shown in Figure 7.

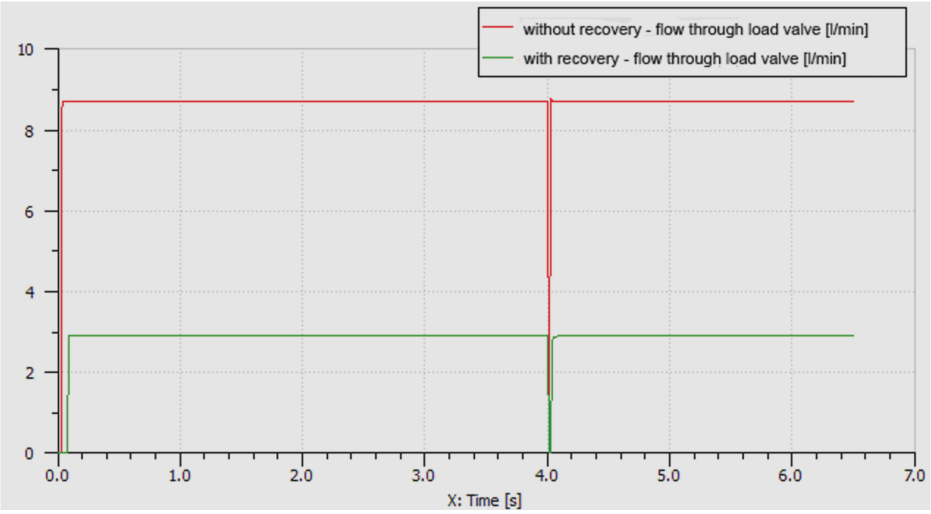


Figure 7: Variation of flow discharged through the load simulation valve

The rate of flow discharged through the load valve, as shown in Figure 7, for versions of simulation models with/without energy recovery, of approximately 3 l/min, respectively 9 l/min, can be explained by the fact that in the first version there is discharged through the valve the difference between pump flow and hydraulic motor flow, while in the second version there is discharged through the valve all the pump flow.

Characteristics of torque variation at the electric motor shaft, resulted from running the two simulation models in AMESim, as shown in Figure 8, show and approximate quantitatively the energy recovery. The simulation was performed under conditions of a constant load, given by the pressure value of about 20 bar on the input of the load valve, for an almost complete forward and backward stroke of piston of one of the two hydraulic cylinders. Energy recovery has an effect of reducing torque at the electric motor shaft by approximately 55%, namely from 2.3 Nm to 1.05 Nm on the forward stroke of hydraulic cylinder piston, and by by approximately 45%, namely from 3.1 Nm to 1.8 Nm on the backward stroke of piston. These results are valid for a value of the ratio of pump capacity and hydraulic motor capacity of 1.5, complying with the condition given by the relation (11).

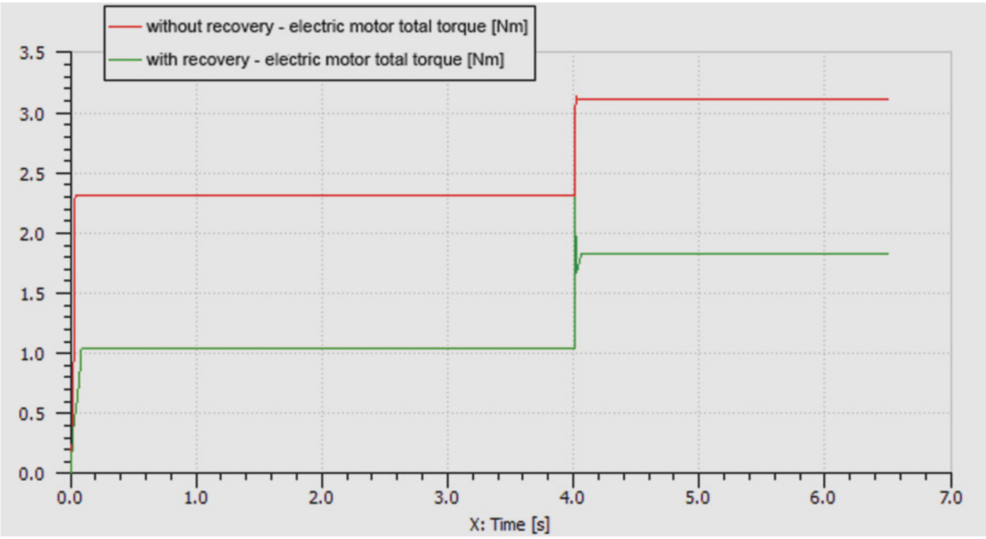
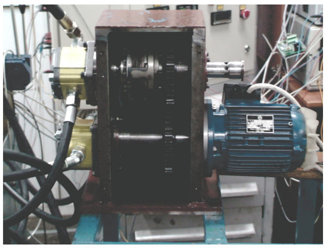


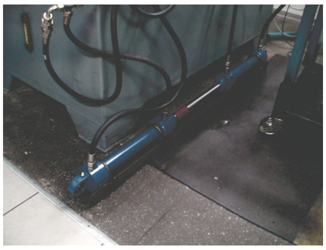
Figure 8: Variation of torque at the electric motor shaft

3. Experimental demonstration of hydro-mechanical power recirculation during endurance tests on hydraulic cylinders

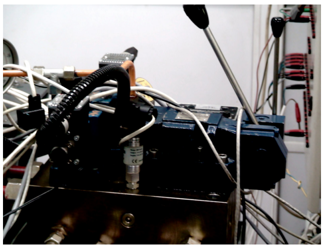
The hydraulic diagram for testing endurance of hydraulic cylinders, shown in Figure 2, and the simulation model in AMESim, shown in Figure 3, have been embodied in a pilot installation whose main modules are presented in **Figure 9**.



(a)



(b)



(c)

Figure 9: Modules of the pilot installation

Module in Figure 9 (a) includes: an electric motor with power = 0.37 kW and shaft speed = 1450 rev/min, a gear transmission, one stage, ratio = 1:1, a fixed pump of 6 cc/rev and a fixed hydraulic motor of 4 cc/rev.

Module in Figure 9 (b) includes two hydraulic cyliders, a test one and a load one, with piston diameter = 50 mm, rod diameter = 28 mm, displacement of piston = 0.3 m. The two cylinders are mounted in a closed metal frame.

Module in Figure 9 (c) includes a hydraulic distributor 4/3, manual control, used for starting, stopping and changing direction of displacement of hydraulic cylinders. Junction "P" of the distributor is connected to the fixed pump outlet, junction “T” to the oil tank, while junctions "A" and "B" to the test cylinder.

As shown in the diagram in Figure. 2, tests were conducted as follows:

a) For version with power recirculation

There was actuated the manual distributor that controls the displacement of hydraulic cylinders. Pressure within the normally-closed valve of the load cylinder was progressively raised, and there were measured pressure along the junction "P" of the hydraulic distributor, current absorbed by the electric induction motor and its rotary speed. Measurement results were recorded in **Table 1**.

I [A]	p [bar]	n [rev/min]
1.63	10	1280
1.68	12	1260
1.70	14.3	1240
1.74	16	1230
1.79	18.3	1210
1.83	**21**	**1190**

Table 1: Measurement results - version with power recirculation

b) For version without power recirculation

Hydraulic motor was decoupled from oil supply by installing a closed faucet on the supply route. Not being supplied with pressurized oil, the hydraulic motor can not generate torque for pump drive, although it stays coupled to the electric motor. To avoid hydraulic motor spinning without oil, a supply pipe has been mounted, as in the diagram in Figure 2, but without the way valve. Next we proceeded in the same way as in the first version, and measurement results were recorded in **Table 2**.

I [A]	p [bar]	n [rev/min]
1.63	9.2	1260
1.66	10.2	1250
1.73	12	1230
1.83	**14.1**	**1200**

Table 2: Measurement results - version without power recirculation

The test highlights that at the same input current of the electric induction motor and about the same rotary speed, so at the same torque, the pilot installation with power recirculation brings 50% additional energy compared to the one without power recirculation. This energy gain is due to the fact that 2/3 of pump flow is no longer discharged through the load valve but through the hydraulic motor, which generates torque that adds to the torque generated by the electric motor.

4. Conclusions

The results of numerical simulation and experimental identification have demonstrated an important method for rendering more efficient the energy consumption at the endurance stands of hydraulic cylinders, based on hydrodynamic power recirculation, which is of interest to the manufacturers for hydraulic cylinders.

For the described application, where the ratio of pump capacity and hydraulic motor capacity is 1.5, we obtain an energy gain of approximately 50%, which is due to the transformation of 2/3 of the dissipated energy, specific to the process of pressure adjustment based on discharging the excess flow through the load simulating valve, into useful energy for the endurance stand.

Numerical simulation shows that for the same pressure used in endurance tests on cylinders, generated on a stand with energy recovery and a stand without energy recovery, the electric motor of the first stand develops mechanical shaft torque of about 50% lower than the electric motor of the second stand.

Experimental tests demonstrate that for the same electric motor power, developed on a stand for testing endurance of hydraulic cylinders with energy recovery and on a similar stand, but without energy recovery, the first stand generates testing pressure of about 50% higher than the second stand.

5. Bibliographical References

/1/ T.C.P, I.L, & D.D.I.G. 2009. Recuperarea de energie la standurile de anduranta ale masinilor volumice rotative (Energy recovery at endurance stands of rotary volumetric machines), Proceedings of National Symposium with international participation HERVEX 2009, November 2009, Calimanesti-Caciulata, Romania, pp. 214-221.

/2/ N.V & D.V. 2005. Actionari hidraulice si pneumatice (Fluid Power), Tome I, 2005, Technical Publishing House, Bucharest, Romania, pp. 287–292.

/3/ T.C.P, P.D, D.D.I.G & I.B. 2010. Patent application no. A/0051-19.01.2010, Stand with power recirculation for hydraulic cylinders endurance, 2010, OSIM (STATE OFFICE FOR INVENTIONS AND TRADEMARKS), Bucharest, Romania.

6. Nomenclature

p	pressure	bar
n	electric motor rotational speed	rev/min
n_m	hydraulic motor rotational speed	rev/min
n_p	hydraulic pump rotational speed	rev/min
V_p	hydraulic pump capacity	cc/rev
V_m	hydraulic motor capacity	cc/rev
N_e	electric motor power	kW
N_m	hydraulic motor power	kW
N_p	hydraulic pump power	kW
N_c	hydraulic cylinder power	kW
Q_p	hydraulic pump flow	l/min
Q_m	hydraulic motor flow	l/min
η_{tp}	hydraulic pump total yield	-
η_{tm}	hydraulic motor total yield	-
η_{vp}	hydraulic pump volume yield	-
η_{vm}	hydraulic motor volume yield	-
η_{tc}	hydraulic cylinder total yield	-
t	simulation time	s
I_d	hydraulic distributor control signal	mA
d	cylinder piston displacement	m
I	current absorbed by the electric induction motor	A