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Preliminary and Experimental Research Focused on Linear Hydraulic Engine Used for Deep Pumps Driving

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Abstract

In this paper the author presents some experimental research conducted on linear motors used to drive hydraulic deep pumps. Research findings are highlighted and some conclusions have emerged for future research

Key words: *deep pumps, hydraulic engine*

Introduction

To study the interdependence which exists between the interested mesurements is necessary a simultanous registerings of all the mentioned parameters. For performing this experimental work in dynamic regime are necesary transducers capable to transform mechanical signals in proportional electrical ones. To perform the mesurements and the registerings of the laws of parameters varyings, which reflect the activity of hydraulic engine for diferent charging rates, are necesary three pressure transducers and a displacement one. Desining and to chosing of the proper transducer takes into account that ones whom sensitive element is a liniare function of the mesuring parameter and the responding time (time constant) has the smallest posible value. The proper transducers used for this work are the resistive strain sensors (transducers).

Scheme of Working Booth and of the Transducers

The experimental investigations were carried out „unloaded running” operation on the existing equipment (the equipment on the Department of Petroleum and Petrochemical Eqipment) .

The completed stand scheme is shown in Figure 1.

As shown in Figure 1, pressure transducers were used on the supply pipeline of linear hydraulic motor as well as registration of variation of pressure laws in the active rooms of the engine. Transducers are strain type, in the range of 0-8 MN/m^2. In Figure 2, some components of hydraulic linear motor, made by the author, are presented, [1, 2, 4].

The calibration was done by loading the transducers on the calibration bench (Figure 3). In the photo is shown the standard gauge mounted in parallel with transducer whose characteristic is rising. The output of the transducer is inserted into a bridge strain and then is conected to an oscilloscope with further loops. The strain marks stuck on the sensitive element of transducer are distorted in an elastic mode and the unbalanced of bridge strain determines a voltage that feeds the galvanometer loop, inside the oscilloscope. Current that flows through the loop is proportional to the pressure, the spot light following its variation (change).

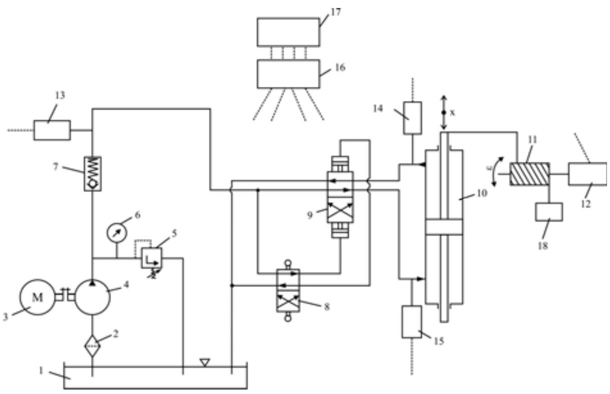

Fig. 1. Stand hydraulic scheme used for unload running used of linear hydraulic motor with embedded distribution: 1-tank; 2-filter; 3-DC electric motor; 4- hydrostatic generator; 5- safety-valve; 6-gauge; 7 - valve effect; 8.9 hydraulic distributors; 10-linear hydraulic motor; 11-reel; 12-displacement transducer; 13;14;15; pressure transducers; 16-bridge stain; 17-oscilloscope with loops; 18 – weight.

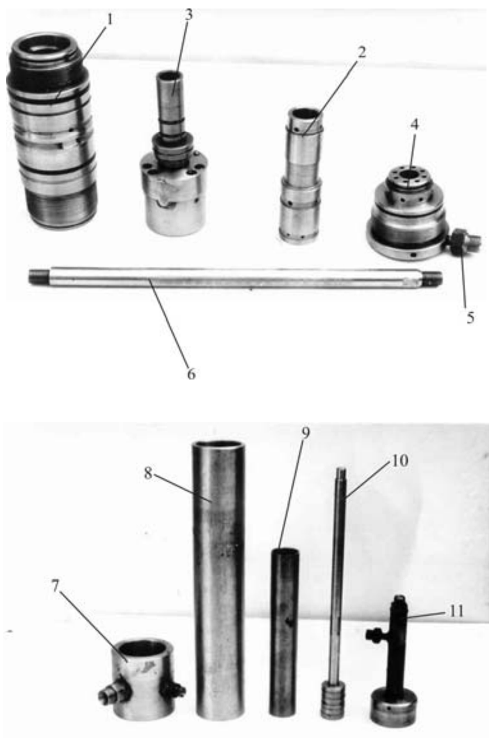

Fig. 2. Linear hydraulic engine components: 1- one-hand dealer, 2 - drawer distributor, 3- flat plain, 4- head pins, 5- socket mounting pressure transducer Hottinger type, 6- pin control, 7-collector exhaust, 8 - linear hydraulic motor casing, 9- cylinder engines, 10- pin connector, 11-feed nozzle of the engine.

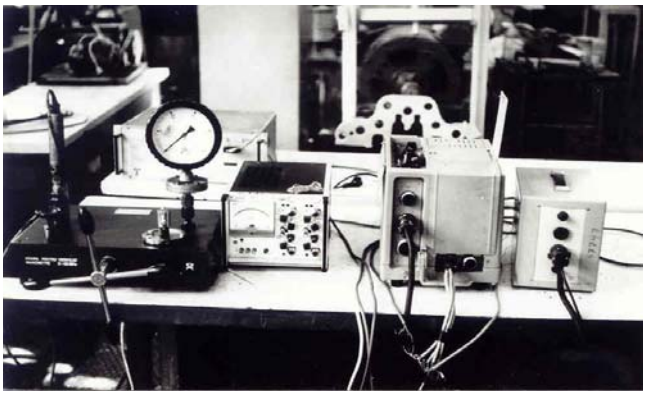

Fig. 3. Photo of stand for pressure transducer used for calibration

Calibration results are shown in Figure 4.

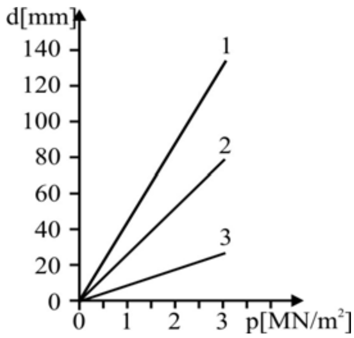

Fig. 4. Calibration of pressure transducers results:
1 - supply-pressure transducer; 2- pressure transducer the above chamber; 3- pressure transducer the lower chamber.

Piston rod movement was monitored using a displacement transducer. This transducer consists of a resistance arranged on the outer surface of a cylinder, over which it can be moveed a cursor. The cursor can be driven from outside and, as shown in Figure 1; piston rod connection is done through a thin steel cable from a reel. One end of the cable is connected to the piston rod and the other end is caught by a weight which stretch the yarnThe transducer assembly is done in such a way, that that comprising variable resistance, loop galvanometer and power supply are in series. Rod movement causes a rotation of the cursor and a change in current intensity of the circuit. By differently positioned of the piston rod, was obtained the dependence of the displacement rod and the size of the spot on the oscilloscope screen (Figure 5).Angular speed of DC motor used to drive the pump with the axial piston was measured with a tachogenerator connected to a tachometer.

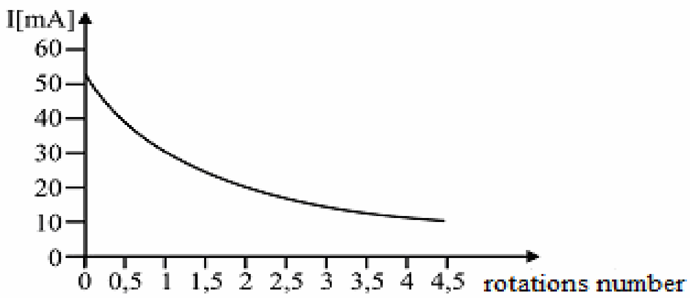

Fig. 5. The value of loop galvanometer current intensity for different positions of the transducer displacement

On Load Experimental Test Results

On load tests were performed according to the scheme in Figure 1. Given the complexity of hydraulic engine variant made, these attempts were aimed at highlighting the operation, engine sensitivity to changing conditions of supply.The available material conditions during that period [1] allowed to determine the law of variation of displacement x and the pressure of the supply rooms and linear hydraulic motor. A photo booth, with the fore deck strain gauges and oscilloscopes with photosensitive paper is given in Figure 6.

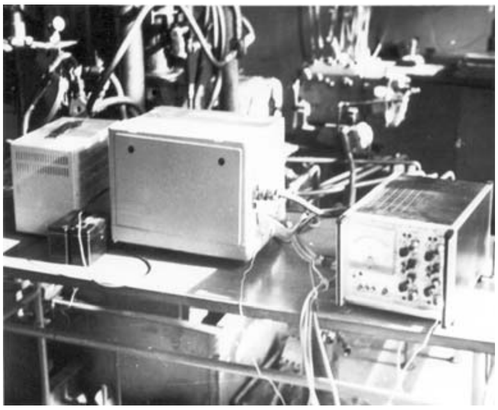

Fig. 6. Photo of working stand with –the measuring chain -bridge strain -oscilloscope in the fore.

A first set of tests at angular velocity ω = 149 rad / s pump (F112), which causes a power flow of Q = 0.334 l / s and frequency of the hydraulic motor racing f_m = 38 strokes / min with a period of motion T = 2.7 s. Also, the derivation was determined laws of variation of piston velocity and acceleration from on load, Figure 7.

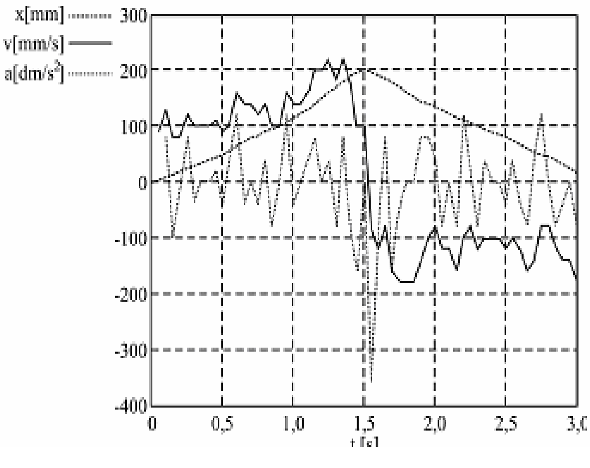

Fig. 7. Kinematic parameters of piston unload running

Calculation of kinematics parameters of the piston in the MATLAB program.

```
t=0:0.05:3;
x=[0  4.5 11 15 19 25 30 35 40 ...
    45.5 50 55 63 70 77 83 90 95 100 ...
    108 115 122 130 140 150 161 170 181 190 ...
```

```
 195 200 196 190 186 178 169  160  151 144 139 135 129 123 115 110 ...
106 100 95  90  85 79 74 68 60 53 ...
49 45 39 32 25 16];
plot(t,x,'r');
v=diff(x)./diff(t);
xlabel('t [s]');
ylabel('x [mm], v [mm/s], a [dm/s^2]');
title(kinematics parameters of the piston )
hold on;
grid on;
t1=0.05:0.05:3;
plot(t1,v,'g');
hold on;
a=10e–2*diff(v)./diff(t1);
t2=0.1:0.05:3;
plot(t2,a,'b');
```

As shown in Figure 7, piston displacement under these conditions, is to head off and travel, short breaks rather, about 0.1 s, or 1/27 movement period. Maximum piston displacement is 0.2 m / s and maximum acceleration is of 10 m/s^2.

Duration of lifting and lowering racing is about 1.35 s. The same discontinuities of kinematic parameters of the piston is attributable to: inadequate supply of linear hydraulic motor, inavecvate hydraulic balance of the linear hydraulic engine, unclear system for recording the movement of the piston in this phase of research. Interesting are the laws of variation of pressure in the hotel assets, Figure 8.

It is noted that pressures in the two houses are roughly constant, during the period when the room is active, reaching average values of 0.25 MN/m^2 for the lower house (upward trend) and 0.15 MN/m^2 for the upper chamber (downward stroke).

Apparently there is a slight difference between the average values, but overall it can be appreciated that friction forces are small, making it easy to move the piston. In times of change power mode of the rooms the pressures have developed an unregular evolution, mainly due to the mode of action of the distributor.

Supply pressure reaches higher values compared with the pressures of rooms, with a peak of about 4.0 MN/m^2. This development is understandable because of the liquid flow through the gaps and holes of engine power causes a pressure drop at the entrance of the engine.

Also, the lack of consistency between pump flow and flow consumed by the engine, resulting in removal of a quantity of oil from the circuit and therefore increase the pressure.

However, having regard to the two levels of evolution of supply pressure and the symmetry of active engine rooms it can be concluded that the main causes that create the difference between the average values of supply pressure on the two races (from about 2.8 MN/m^2 to 1 MN/m^2) are mainly the different supplyof the rooms and that the hydraulic engine is not balanced.

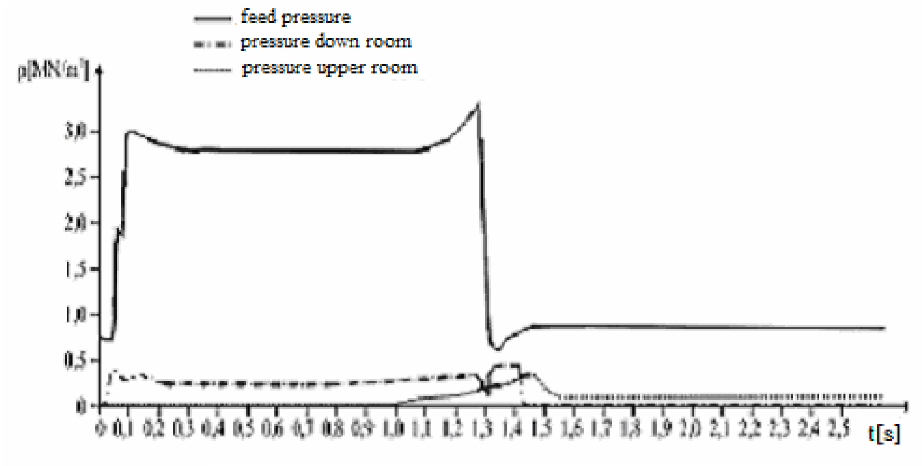

Fig. 8. Laws of pressure variation on suppling of the engine and in active rooms for unloaded running at ω_p = 149 rad / s Qp = 0.334 l / s

Changing of pump flow, by raising the angular speed of DC motor from 149 to 188 rad / s leads to a reduction of movement period from 2.7s to about 2.05 s (see Figure 9). The suppling with a higher flow, occurs an acceleration of pressure value on supply, while the pressure changes in the active rooms about the same as previously analyzed case.

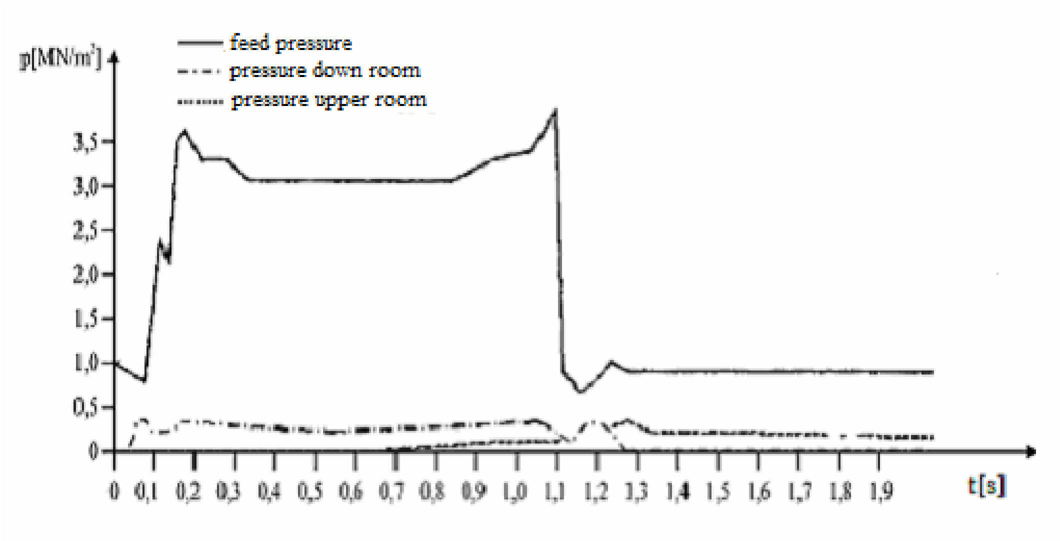

Fig. 9. Laws of pressure variation on supply of the engine and in active rooms, for unload running ω_p = 188 rad / s Q_p = 0.421 l / s

At this stage of research is tried „the loaded running" too. Its loading was accomplished by obstructing a valve mounted on the engine exhaust pipe.

Pressure on the exhaust pipe was 1.5 MN/m^2. Such a test is shown in Figure 10. On loaded running is found primarily an expansion of uniformity, at the race ends.

It is difficult to interpret the source of these variations in the absence of the information about movement of the distributor.

Increasing of pressure is found in the active cameras, which has a slightly different variation.

When the upper chamber is active, there is no a constant pressure as when on load, showing continuing oscillations.

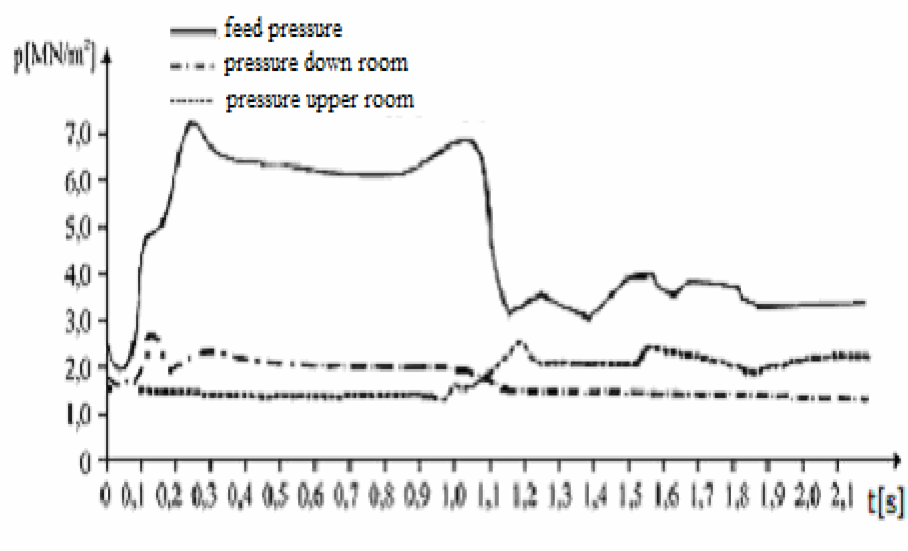

Fig. 10. Laws of pressure variation on supply of the engine and in the active cameras on loading running, for a power flow Q_p = 0.421 l / s , discharge pressure p_e = 1.5 MN/m^2

Same things have happened with supply pressure, which reaches quite high values (around 7 MN/m^2). Lack of information of flow in the engine, do not allow the formulation of comprehensive feedback about its operation.

There is a slight delay in carrying out the cycle period increased from T = 2.05 s at T = 2.2 s, keeping the same conditions of suppling for engine.

Conclusions

Given the weaknesses of this phase of research on specific findings and a series of measures such as:

- Determining of kinematic parameters proved a correct operation of the engine, the corresponding periods with the dead periods at the end of the race being short, of about 0.1 s and the velocity is constant during the race, so, the dynamic stresses in on load are small;
- The friction forces on engine are small, as is evident from the reduced value of pressure in engine rooms;
- Problems have appeared on engine suppling, being necessary to modify the suppling holes to reduce the loss of pressure into the engine input; the hydraulic balance will be achieved using balancing tube;
- The tracking system requires changing of the movement of piston rod which has not fully satisfied;
- Is necessary to track the moving of distributor drawer and how this it is correlated with the movement of the piston;
- Flow transducers mounted on the pipe of suppling of hydraulic linear engine and on the exhaust pipe.

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Cercetări experimentale preliminare privind utilizarea motoarelor hidraulice liniare la acționarea pompelor de adâncime

Rezumat

În această lucrare, autorul prezintă o parte din cercetările experimentale efectuate asupra motoarelor hidraulice liniare utilizate la acționarea pompelor de adâncime. Sunt evidențiate atât rezultatele cercetărilor, cât şi o parte din concluziile care s-au desprins, în vederea unor cercetări viitoare.