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EXPERIMENTAL STUDIES ON THE FRICTION OF MOBILE ELEMENTS IN HYDRAULIC EQUIPMENT

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Abstract: *In this study we will focus on the problems that arise due to the high travel speeds that the piston of a hydraulic motor develops during operation. These speeds lead to an increase in the value of the kinetic friction coefficient.*

Keywords: *hydraulic motor, frictional forces*

1. INTRODUCTION

A special contribution to the increase in the value of the total friction coefficient is also the very high values of the lubrication pressure of the piston rings (most hydraulic equipment is lubricated with oil at their operating pressure), as well as the hydrodynamic lubrication effect exerted on only one side of the piston ring [1], [2], the side opposite to the frictional forces (figure 1).

This fact leads to important frictions and to an increase of the temperature value inside the lubricating oil layer.

In the last period, hydrostatic or hydrodynamic lubrication was avoided, where possible, a mixed limit friction system is preferred (combination between dry and hydrodynamic or hydrostatic friction). This is possible by controlling the value of the lubricating oil pressure [3], [4].

Thus, in many cases, low values of the friction coefficient and implicitly of the wear of the equipment are obtained.

A test stand was designed to study the friction between the pistons and the body of the hydraulic motor.

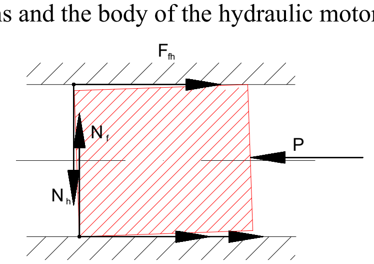

Figure 1. The forces acting on the piston, due to the hydrodynamic lubrication on one side

2. THE EFFECTS OF FRICTIONAL FORCES ON THE MOBILE CHAIN IN HYDRAULIC MOTORS

2.1. Strengths in the mobile chain due to technological errors

In the case of the balance equations for rotary hydraulic motors (especially the 20 MPa maximum inlet pressure motors that are most commonly used), neglecting frictional forces altogether is a wrong way of approaching the problems. Both the high forces in equilibrium and the eccentricities of the parts that cannot be excluded lead to the effect of frictional forces that cannot be neglected [5], [6].

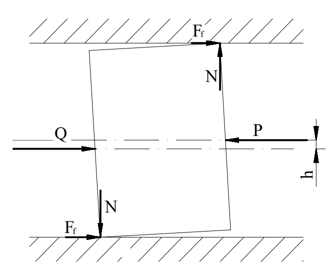

Figure 2. The forces acting on the piston, due to technological errors

P – force due to the working agent; Q – force in the connecting rod; N – normal forces on the guiding surface; F f – frictional forces opposite to the direction of movement of the piston.

2.2. Forces in the mobile chain due to the design of the hydraulic motor

Non-coaxialities are also introduced in the system, which are due to the very principle of operation of hydraulic motors with axial pistons.

The effect of these forces may not overlap the effect of the forces due to technological errors leading to the increase or decrease of the total effect, that is to increase or decrease the value of the frictional forces.

In the continuation of the work, only the effect of the forces acting on the piston and which are due to technological errors are considered.

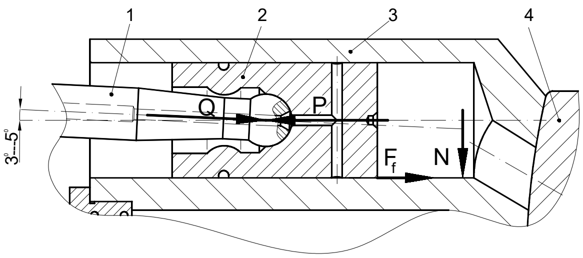

Figure 3. The forces acting on the piston, due to the principle of operation

1 - rod; 2 - piston; 3 – engine body; 4 – distributor; P – force given by the pressure of the working agent; Q – the force in the connecting rod that opposes the action of the pressure of the working agent; F f - frictional force between piston and guide; N – normal force on the surface of the guide

The effect of these forces may not overlap the effect of the forces due to technological errors, leading to the increase or decrease of the total effect, to increase or decrease the value of the frictional forces.

In the continuation of the work, only the effect of the forces acting on the piston and which are due to technological errors are considered.

From the force balance equation, it follows:

$$P = Q + 2 \cdot F_f = Q + 2\mu \cdot N$$

Considering some inclination of the piston due to the play, from the equation of moments with respect to the center of symmetry of the piston we obtain [7], [8]:

$$P \cdot h - N \cdot l = 0$$

whence $N = \frac{P \cdot h}{l}$ and considering: $N_1 = N_2 = N$

Substituting, the force P required for the movement of the piston results:

$$P = Q + 2\mu \cdot \frac{P \cdot h}{l} \quad \rightarrow \quad P\left(1 - 2\mu \cdot \frac{h}{l}\right) = Q$$

and $\quad P = \frac{Q}{1 - 2\mu\frac{h}{l}}$

in which: h – the force offset value P

l – the length of the piston

Movement is impossible when:

$$1 - 2\mu \cdot \frac{h}{l} = 0$$

the limit value of the h/l ratio results:

$$\left(\frac{h}{l}\right)_{\lim} = \frac{1}{2 \cdot \mu}$$

The necessary condition to avoid valve sticking is:

$$\frac{h}{l} \leq \left(\frac{h}{l}\right)_{\lim} = \frac{1}{2 \cdot \mu}$$

or in practice: $\frac{h}{l} \leq \frac{1}{2\mu \cdot c}$

in which: c – safety coefficient.

$c = 6.5$ for cylindrical guides.

$$\frac{h}{l} \leq \frac{1}{2 \cdot 0{,}2 \cdot 6{,}5}$$

$$\frac{h}{l} \leq \frac{1}{2 \cdot 0.2 \cdot 6.5} \quad \text{or} \quad \frac{h}{l} \leq \frac{1}{2.6}$$

This fact means that in the case of a piston with $h = 1$mm misalignment, its length must be at least 2.6 mm, otherwise the piston will block.

If the balance equation is written on the piston of a hydraulic motor, in the form:

$$P^{'} = P^{''} + F_f$$

$$P^{'} = \frac{\pi \cdot d^2}{4} \cdot p_{func}$$

where:

d - the diameter of the piston

p_{func} - the pressure of the hydraulic agent

2.3. Theoretical evaluation of the value of frictional forces

If the technical deviations from the first two quantitative laws of dry sliding friction are insignificant, regarding a third law that states that the friction force is independent of the sliding speed, these deviations are significant. Thus, it is necessary to consider two friction coefficients in practice:

- coefficient of static friction μ_s for the surfaces in contact at rest.

- the coefficient of kinetic friction μ for moving contact surfaces.

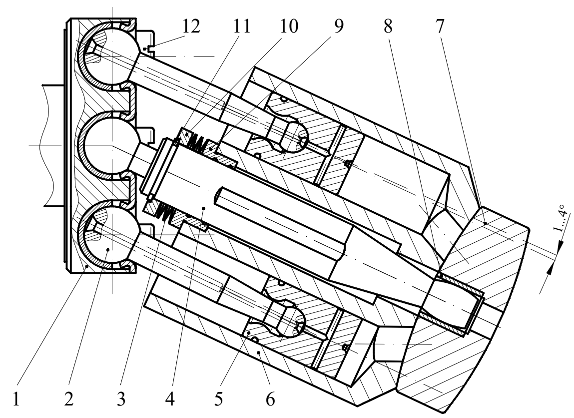

Figure 4. Misalignment between the connecting rod and the piston in the hydraulic motor

1 – rotating disk; 2 – connecting rod; 3 – disc springs; 4 – guide shaft; 5-pistons; 6-engine body; 7-distribution disk; 8-guide shaft bushing; 9-guide; 10-compression bushing; 11-elastic ring; 12-screw with notched cylindrical head.

Analyzing from an ideal point of view, the frictional forces for the case of ordinary hydraulic motors should, from a value point of view, be negligible because there are no major components of the forces acting in the kinematic chain in the motor that are perpendicular to surfaces in relative motion.
Thus, the basic component of the total friction force that acts during the operation of the hydraulic motor will be analyzed, the component introduced by the lack of coaxially of the landmarks that make up the kinematic chain.

To evaluate this component, we will take as an example the kinematic chain whose component is most often used in most hydraulic engines.

The following working assumptions will be made:

- The force *P* is given by the product between the working pressure and the frontal area of the piston and is calculated with the relation:

$P = p_{func} \cdot s$

in which: p_{func}- the value of the working pressure in the engine

s - the value of the frontal area of the piston, $s = \frac{\pi \cdot d^2}{4}$

d - the value of the piston diameter.

- For the theoretical evaluation, the dimensions of the moving chain parts of middle-class hydraulic motors, which are most often used, and which are also used for the execution of the experimental model, will be adopted.
- Because the angle of inclination of the piston has very small values, dimensions l_l between the two normal forces will be considered equal to the length *l* of the piston.
- We consider that the value of the friction coefficient, for all cases, is $\mu = 0.2$
- We consider that the value of the discount is $h = 0.01$, $h = 0.015$, $h = 0.02$
- Since the theoretical conclusions are obvious, they will be demonstrated on only four types of hydraulic motors that work at a pressure of 200 daN/cm^2, namely for those with the diameter of the pistons according to the tables.

Table 1.

Diam. of the pistons d [mm]	Length of the pistons l [mm]	Off-setting h [mm]	Frontal area s [mm^2]	The force P [N]	Strength value normal N [N]	The friction force F_f [N]
20	25	0.01	314	6280	2.512	0.502
30	35		706.5	14130	4.037	0.807
40	45		1256	25120	5.582	1.116
50	55		1962.5	39250	7.136	1.427

Table 2.

Diam. of the pistons d [mm]	Length of the pistons l [mm]	Off-setting h [mm]	Frontal area s [mm^2]	The force P [N]	Strength value normal N [N]	The friction force F_f [N]
20	25	0.015	314	6280	3.768	0.754
30	35		706.50	14130	6.056	1.211
40	45		1256	25120	8.373	1.675
50	55		1962.50	39250	10.705	2.141

Table 3.

Diam. of the pistons d [mm]	Length of the pistons l [mm]	Off-setting h [mm]	Frontal area s [mm^2]	The force P [N]	Strength value normal N [N]	The friction force F_f [N]
20	25	0.02	314	6280	5.024	1.004
30	35		706.50	14130	8.074	1.165
40	45		1256	25120	11.164	2.233
50	55		1962.50	39250	14.273	2.855

As can be observed from the tables, the theoretical values of the normal pressing forces and the friction forces are relatively small, compared to the values of the force *P*, for all three values of the offset, considered for the four dimensions of the diameter *d*.
Figure 5 shows the functions $F_f = f(P)$ for the three values of the offset *h*.

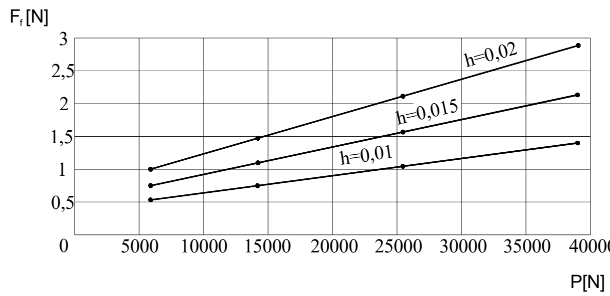

Figure 5. The graph of the function $F_f = f(P)$ for $h = 0.01mm$; $h = 0.015mm$; $h = 0.02mm$

3. EXPERIMENTAL RESULTS

3.1. Installation for determining frictional forces

The determinations regarding the values of the static friction forces between the elements of the mobile system of the engine (pump) and their guides are carried out using the installation presented below.

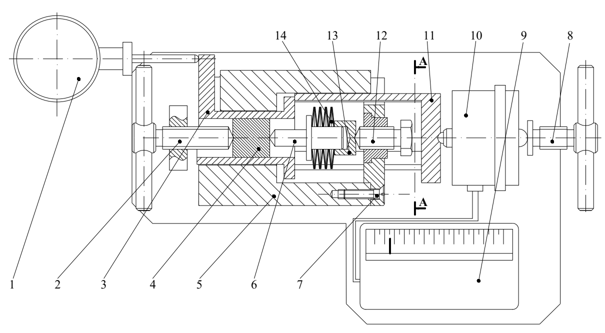

Figure 6. Device for determining the values of static friction forces

1-comparator clock; 2- future screw for force adjustment; 3- movable bore with spur; 4- piston; 5-case; 6-disc springs support; 7-screws; 8- future screw for moving the piston bore; 9-device indicating the value of the thrust force; 10-dynamometer; 11-piece with legs; 12-screw for piston position adjustment; 13-part for compression of disc springs; 14-package of disc springs.

The values of these forces are practically obtained by turning the screw (12) with the corresponding number of revolutions. The package of spring (14) has a known stiffness value. The number of rotations of the screw (12), resulting from the force relationship of the spring package:

$$F = c \cdot p \cdot n$$

in which:

c - the stiffness value of the springs package.

p - the value of the pitch of the screw.

n - the number of rotations of the screw necessary to obtain the necessary F, pre-calculated force of the spring package.

Results in the number of rotations: $n = \frac{F}{c \cdot p}$

After obtaining the value of the pressing force (in which case the piston is required as during the active stroke during the operation of the hydraulic motor) the screw (8) is actuated which actuates the electromagnetic dynamometer (10) and which, in turn, pushes the part (11), equipped with three with legs arranged at 120°, which pass through the cover (7) of the housing (5) and slide inside it pushing the movable bore with spur (3). This is how the relative movement between the piston and its bore is obtained.

The force acting on the movable bore with spur is measured by the dynamometer (10) and the displacement of the movable bore with spur facing the piston is measured by the comparator (1).

Thus, with the help of the dynamometer and the comparator, the values of the friction forces between the piston and the bore or the displacement of the piston against the bore can be determined. In this way, the friction force is measured for different values of the misalignment and for different values of the roughness of the piston and its bore.

3.2. Experimental evaluation of frictional forces

As was presented in the introduction of the work, it is necessary that in practice, in most situations, two coefficients of friction should be taken into account and, be careful, even when a perfectly dry friction is envisioned, which in fact does not even exist :

- **the coefficient of static friction** μ_s for the surfaces in contact at rest;
- **the coefficient of kinetic friction** μ for the surfaces in contact in relative motion.

Ideally, the frictional forces that occur in hydraulic motors of any type should be neglected, because there should be no major, normal components that act on the guide surfaces of the kinematic chain elements of these motors. In practice, however, the normal components in the direction of movement are introduced either by technological deviations, or conceptually, or intentionally so that friction forces, apparently hydrodynamic friction forces, have a permanent influence on the operation of hydraulic motors (for example, to prevent vibrations in during operation).

The experiments aim at analyzing one of the basic components of the total friction force in the kinematic chain, given component of the existence of eccentricities, games and the lack of coaxiality between the elements of any kinematic chain.

Table 4.

Diam. of the pistons d [mm]	Length of the pistons l [mm]	Off-setting h [mm]	Frontal area s [mm^2]	The force P [N]	Strength value normal N [N]	Starting friction force F_f [N]
20	25	0.01	314	6280	2.512	0.63
30	35		706.50	14130	4.037	1.02
40	45		1256	25120	5.582	1.40
50	55		1962.50	39250	7.136	1.78

Table 5.

Diam. of the pistons d [mm]	Length of the pistons l [mm]	Off-setting h [mm]	Frontal area s [mm^2]	The force P [N]	Strength value normal N [N]	Starting friction force F_f [N]
20	25	0.015	314	6280	3.768	0.94
30	35		706.50	14130	6.056	1.51
40	45		1256	25120	8.373	2.09
50	55		1962.50	39250	10.705	2.68

Table 6.

Diam. of the pistons d [mm]	Length of the pistons l [mm]	Off-setting h [mm]	Frontal area s [mm^2]	The force P [N]	Strength value normal N [N]	Starting friction force F_f [N]
20	25	0.02	314	6280	5,024	1.26
30	35		706.50	14130	8,074	2.02
40	45		1256	25120	11,164	2.79
50	55		1962.50	39250	14,273	3.57

4. CONCLUSIONS

As can be seen in the tables (last column), the practical values of the friction forces for the four types of hydraulic motors, when changing the direction of movement of the piston towards the active stroke, differ significantly from the theoretical ones.

These values of the friction forces, which are approximately 25% higher than the theoretical ones, cannot be neglected and measures must be taken to reduce them.

The figures below show the graphs of the function $F_f = f(P)$ for the three displacements studied ($h = 0.01\ mm$; $h = 0.015\ mm$; $h = 0.02\ mm$) and for the four dimensions of the piston diameter, which were also studied in the theoretical part. ($d = 20mm$; $d = 20mm$; $d = 20mm$; $d = 20mm$)

The value differences between the theoretically determined friction forces and the experimentally determined friction forces can be clearly observed, at least in the beginning part of the movement of the pistons.

Figures 7-9 show the dependence of the function $F_f = f(P)$ for the three displacements h studied.

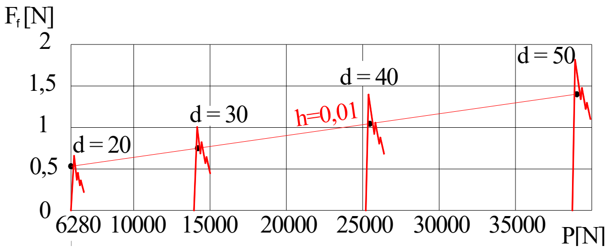

Figure 7. The experimental graph of the function $F_f = f(P)$ for $h = 0.01$ mm and $d = 20, 30, 40, 50$ mm.

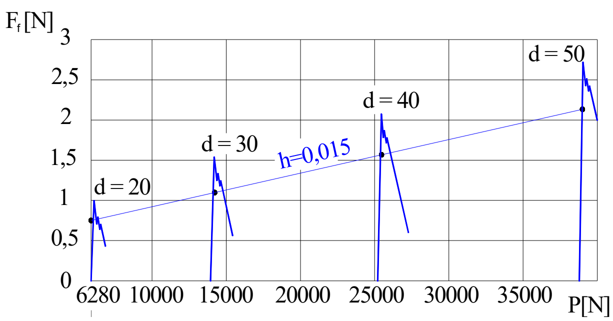

Figure 8. The experimental graph of the function $F_f = f(P)$ for $h = 0.015$ mm and $d = 20, 30, 40, 50$ mm.

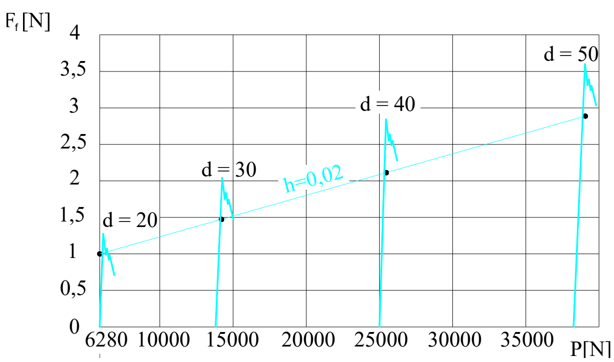

Figure 9. The experimental graph of the function $F_f = f(P)$ for $h = 0.02$ mm and $d = 20, 30, 40, 50$ mm.

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