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FRICTION LOSSES IN A LINEAR HYDRAULIC MOTOR AS A RESULT OF THE INFLUENCE OF THE CONTROL STRUCTURE AND OIL VISCOSITY

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Abstract

The problem of energy tests of hydraulic systems with hydraulic cylinders, seemingly simple displacement machines, is more complex than that of rotary hydraulic systems. The results of the researches provide an insight into the impact of external loads, required speed, structure of the power supply system, viscosity of hydraulic oil on friction loss, and the efficiency of cylinder drive. The hydraulic cylinder is the strongest structure in the system. Failure of the system is most likely due to failure of the pump supplying the cylinder. The high load of the pump is often caused by the very low energy efficiency of the cylinder, which, despite a relatively low external load, requires high inlet pressure due to large mechanical losses of friction between the piston and the cylinder and between the piston rod and gland. These losses depend on the type of seal used, its shape, the material it is made from, pre-clamp, and the operating parameters of the cylinder. Improperly sealed or assembled seals can cause energy losses of up to 25%. Due to the use of moving seals in the hydraulic cylinder, its energy behaviour is completely different compared to the energy behaviour of a rotary motor, which does not have any seals. The friction force connected with the work of the sealing joints and the mechanical efficiency of the cylinder are determined not only by the external load but also by the method of the applicable supply of the cylinder resulting from the throttling structure, and in particular the pressure level generated in the discharge chamber of the cylinder.

Keywords: *friction force, mechanical losses, energy efficiency, oil viscosity, hydrostatic drive, hydraulic cylinder*

1. Introduction

Hydrostatic drives due to numerous advantages are used in machines and equipment of many industrial branches. In technologically advanced areas of unit and bulk production, users are required to have an operational policy that allows them to maintain an adequate level of reliability in machinery and equipment, including their drive, at low total cost [3].

Cylinders are very common type of hydraulic motors in which the pressure energy is transferred to mechanical energy. Unlike rotary motors, cylinders perform straight-reverse movements with limited stroke, rarely with limited rotational angular rotation [an example may be a steering machine]. Cylinders due to their design are components in which it is generally difficult to provide both high efficiency and complete tightness. As a function of the application, one of these two aspects overrides the other, sometimes seeking a compromise between them.

The vast majority of energy losses occurring in the cylinder are caused by friction in the piston seals and piston rods.

Hydraulic elements have energy losses that are, among other things, a function of the viscosity of the hydraulic oil used, as well as energy losses. This oil transfers energy from the hydraulic drive, usually the pump, to the receivers – the cylinders and the hydraulic motors. It is important that the liquid in the hydraulic system have laminar flow rather than turbulent flow. Laminar flow causes much less resistance than turbulent flow. The nature of the flow significantly influences the viscosity of the hydraulic fluid. Viscosity is a measure of internal friction and results from the resistance of liquid molecules. It varies considerably with temperature, but depending on the type of liquid. A liquid with a low viscosity has a lower resistance when flowing than a liquid with a high-viscosity. At present, in hydraulic or stationary hydraulic systems, hydraulic fluids, refined mineral oils or non-flammable synthetic liquids are often used [1].

The manufacturer selects the viscosity of the hydraulic oil to provide optimal performance, efficiency and durability for the installation. For a wide range of temperatures, such as low starting temperatures and high operating temperatures, an oil with high viscosity coefficient should be selected. The choice of oil type will also depend on the starting temperature. For different liquids, with the same kinematic viscosity at 40ºC, the viscosity at negative temperatures may show considerable variations. The system should never be operated below the hydraulic oil flow temperature [1].

2. Construction of the hydraulic cylinder

The linear hydraulic cylinder is a simplified assembly formed of a piston fitted to a cylinder (Fig. 1). It is the simplest means of obtaining strength combined with linear motion. It can be in this sense an element with one direction of motion – one-sided cylinder or two directions of motion – two-sided cylinder.

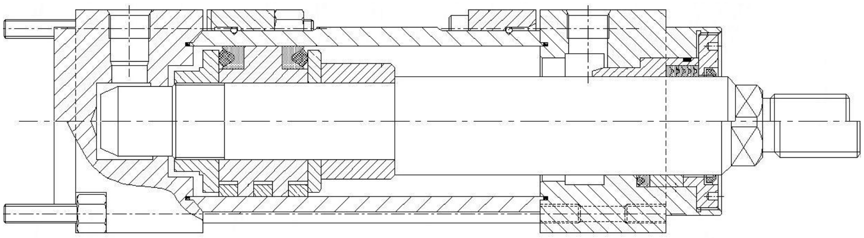

Fig. 1. One-sided hydraulic cylinder

Double-acting cylinders with one-sided piston rod are used most often. Due to the piston rod only on one side of the piston, the basic effective surface of the piston (from the rodless chamber) is larger than the surface of the effective piston in the piston rod chamber. For this reason, at the same pressure and flow rate, the piston rod of the cylinder is inserted into the cylinder at a speed higher than the speed of extension; the force that the piston rods then has in turn is less than the force at the time of extending the cylinder [2].

If the system is required to achieve exactly the same speed, with the same force during the piston rod extension and retraction, a double acting bilateral piston rod cylinder must be used.
In all cylinders may occur internal and external leakages with a different degree. They are called as volume losses.

Internal leaks are the result of fluid flow from the high-pressure chamber to the lower pressure chamber in the functional gap between the piston body and the inner surface of the cylinder.

External leaks – which are undesirable – are caused by a gap between the piston rod and the cylinder packing. In order to reduce these leaks, the piston body and gland in the cylinder are filled with sealing rings which, by being too tight, would result in excessive mechanical friction interfering with the piston movement (turning it into a step motion) and, above all, lowering the energy efficiency of the cylinder and the entire system too [4].

3. Mechanical losses of friction

Mechanical losses in the hydraulic cylinder are the result of the force of the friction F_{Mm} between the piston and the cylinder and between the piston rod and its lead in the stuffing box. The frictional force depends on the type of seal used, the material from which the sealing ring is made, the ring pre-setting, the pressure level in the working chamber, the contact surface condition, slip speed and oil properties.

The frictional force is practically independent of the velocity of motion, but below a certain critical velocity, of the order of v_{Mcr} = 0.03-0.10 ms^{-1}, and particularly at start-up, it can increase by 30 to 300% [1].

As a result of the friction force on the piston and the piston rod, which changes as a function of the pressure in the piston rod chamber or in the rodless chamber, different values of the mechanical efficiency η_{Mm} of the hydraulic cylinder are obtained [1]:

$$\eta_{Mm} = \frac{p_{Mi}S_M - F_{Mm}}{p_{Mi}S_M}, \tag{1}$$

where:
p_{Mi} – the pressure that prevails in the cylinder chamber,
S_M – active surface of the piston,
F_{Mm} – friction force on the piston and piston rod.

The energy efficiency η_M of the cylinder, which will equal the mechanical efficiency η_{Mm} can also be calculated from the formula [1]:

$$\eta_M = \eta_{Mm} = \frac{F_M}{F_{Mi}} = \frac{F_M}{F_M + F_{Mm}}, \tag{2}$$

whose measurable parameters can be written in the form of [1]:

$$\eta_M = \eta_{Mm} = \frac{F_M}{S \cdot \Delta p_M} \tag{3}$$

The designations used in the energy conservation considerations of the hydraulic cylinder are shown in Fig. 2.

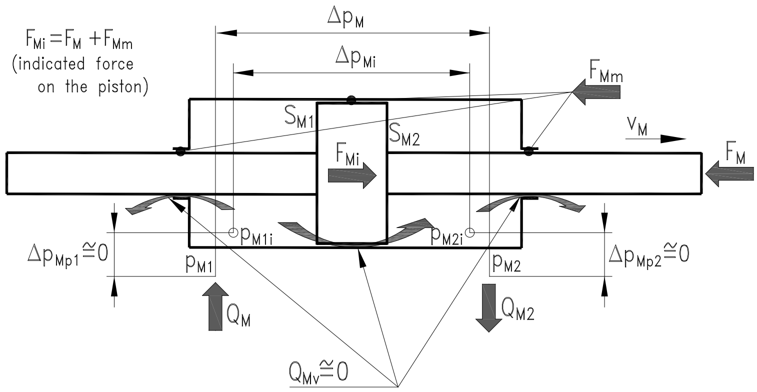

Fig. 2. Parameters adopted for describing the energy behaviour of the hydraulic cylinder

4. Friction loss proportions in the hydraulic cylinder controlled by a proportional directional control valve

Control of hydraulic cylinders by means of throttling valves (proportional directional control valves, servomotors) is based on the arrangement of these components, which slots simultaneously throttle the flows at the inlet and outlet of the cylinder. This implies conditions of an increase of pressure in the cylinder's drainage (outlet) line (Fig. 2 and 3), which depend on the control structure.

In the case of a piston cylinder and packing of piston rods equipped with sealing rings, the p_{M2i} pressure in the discharge chamber is clearly affected by the friction force F_{Mm}. Each increase in the pressure p_{M2i} results in an increase in the friction force F_{Mm} [1].

Figure 3 illustrates the operating parameters of the system on the example of a constant capacity pump system with an overflow valve that stabilizes the p_{P2} pressure level and the throttling valve on the inlet and outlet of the hydraulic cylinder.

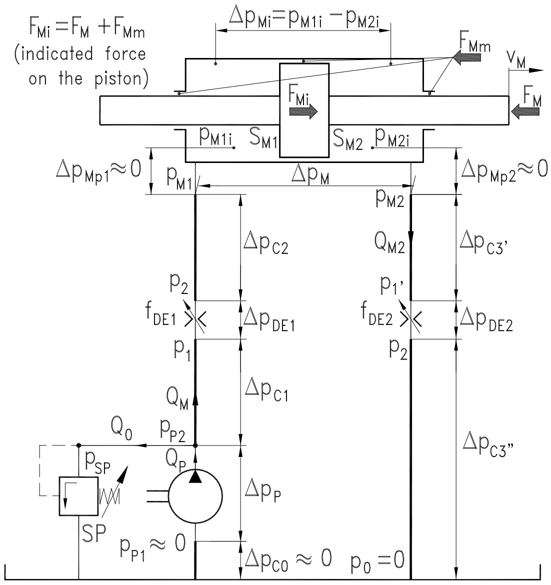

Fig. 3. Parameters of individual operation with constant capacity pump and with hydraulic cylinder control via throttling manifold (throttling in the inlet conduit and in the outlet conduit from the cylinder) [6]

For a given hydraulic cylinder and for a given piston speed higher than a certain critical velocity (below which the coefficient rises), the friction force F_{Mm} of the piston and piston rod friction do not depend on the speed v_m and the fluid viscosity υ but only on the piston rod force F_M (Fig. 4) depending on:

$$F_{Mm} = F_{Mm}|_{F_M=0} + \Delta F_{Mm}|_{F_M}. \tag{4}$$

The magnitude of friction $F_{Mm}|_{F_M=0}$ in the unloaded cylinder (both single and double cylinder) increases with increasing p_{M2i} pressure in the discharge chamber. This is due to the increase in friction between the packing rings in the stuffing box and the piston rod as well as between the piston sealing rings and the cylinder.

The increase in p_{M2i} pressure is primarily due to the use of a throttle control at the outlet of the cylinder. This can be implemented by a proportional directional control valve or a follow-up divider – a servo valve with two throttling slots – on the inlet and outlet from the cylinder.

In this situation, the F_M load exerted on the piston rod (Fig. 2 and 3) directly influences the pressure values p_{M2i} on the outlet and p_{M1i} on the inlet to the cylinder. This reasoning applies primarily to systems supplied by a pressure pump at a constant p_{P2} pressure equal to the nominal p_n pressure of the system.

In the case of supplying a pump running at p_{P2} pressure constant nominal $p_{P2} = p_n$ and throttling at the outlet from the cylinder, a decrease of the external load F_M of the cylinder causes an increase in pressure p_{M2i}.

The magnitude $\Delta F_{Mm}|_{F_M}$ resulting from the increase of external load F_M on the piston rod can be either positive or negative.

The positive value $\Delta F_{Mm}|_{F_M}$ of the increase of the friction force (Fig. 4), resulting from the increase in F_M force on the piston rod, occurs when the oil at the outlet of the cylinder is not throttled, i.e. under p_{M2i} low pressure. At this point, the increase in F_M load accompanied by the increase in p_{M1i} pressure in the inlet chamber results in an increase in friction between the piston sealing rings and the cylinder (and between the piston rod and the packing gland rings if the annular piston chamber is energized). The friction force of the sealing rings in the drainage chamber is constant as the p_{M2i} pressure in the waste chamber remains constant and low [4].

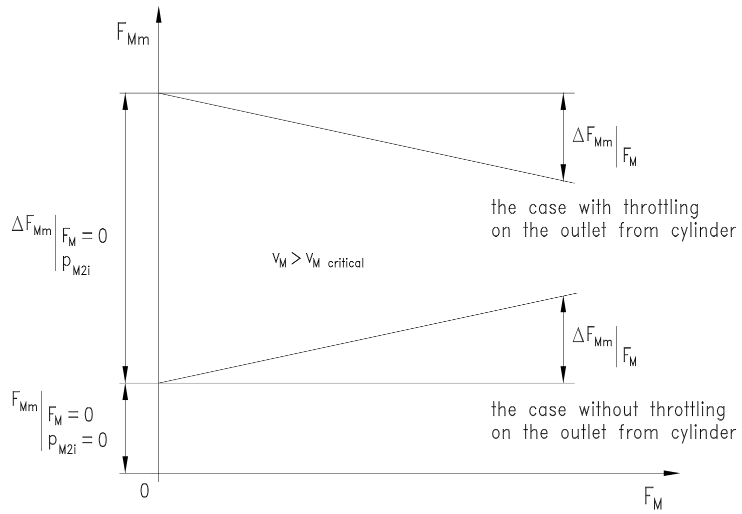

Fig. 4. Friction force F_{Mm} in the cylinder as a function of external load F_M on the piston rod. System powered by a constant capacity pump cooperating with an overflow valve stabilizing the pressure in its discharge line [1, 4]

The negative value $\Delta F_{Mm}|_{F_M}$ of the increase the friction force (Fig. 4) resulting from the increase in F_M force on the piston rod will take place in the case of speed control of the cylinder by throttling the flow at the outlet from the cylinder (for example a case of throttling valve with two throttling slots: on the inlet and outlet from cylinder – directional proportional control valve, servo valve).

Each increase of the external load F_M contributes, in this case, to the pressure reduction p_{M2i}, thus reducing the frictional forces of the sealing rings in the outlet chamber; this results in a negative value $\Delta F_{Mm}|_{F_M}$ [4].

Creation of pressure by the throttling distributor (for example by the directional proportional control valve, servo valve) influences the friction force F_{Mm} in the hydraulic cylinder [1].

5. Laboratory results

The tested linear hydraulic motor is a typical double-rod cylinder, (cylinder internal diameter d = 63 mm, piston rod outer diameter d=36 mm, stroke h=500 mm), which can be used in hydraulic or marine applications. In order to minimize the friction forces in the linear hydraulic motor, the piston rod packing sealing pressure was selected experimentally, providing a total absence of leakage at t = 43ºC (corresponding to a viscosity of ϑ=35 cSt) and the possibility of negligible leaks above the temperature of work t = 70ºC) [3].

There are possibilities to reduce energy losses in proportional control systems (in the pump, in the throttle control unit and in the hydraulic motor, especially in the linear motor), and thus to improve the energy efficiency of the throttle valve [1].

The hydraulic drive and proportional control of linear hydraulic motor can be supplied with a constant capacity pump cooperating with an overflow valve to stabilize the supply pressure of the proportional distributor at the nominal pressure level (Fig. 3) or a pump cooperating with the pressure control valve at the inlet to the receiver. A variable pressure system p = var reduces the losses in the pump, in the control unit and in the linear hydraulic motor [5].

Another arrangement that allows even a greater reduction in energy losses in components is a variable-output power supply structure fitted with a "Load Sensing" controller. The use of a variable capacity pump equipped with a "Load Sensing" controller in a proportional control system allows a simultaneous elimination of structural volume losses (occurring in an overflow valve), severe reduction of structural pressure losses (in the proportional directional control valve or in servovalve), reduction of mechanical losses and volumetric losses in the pump [6].

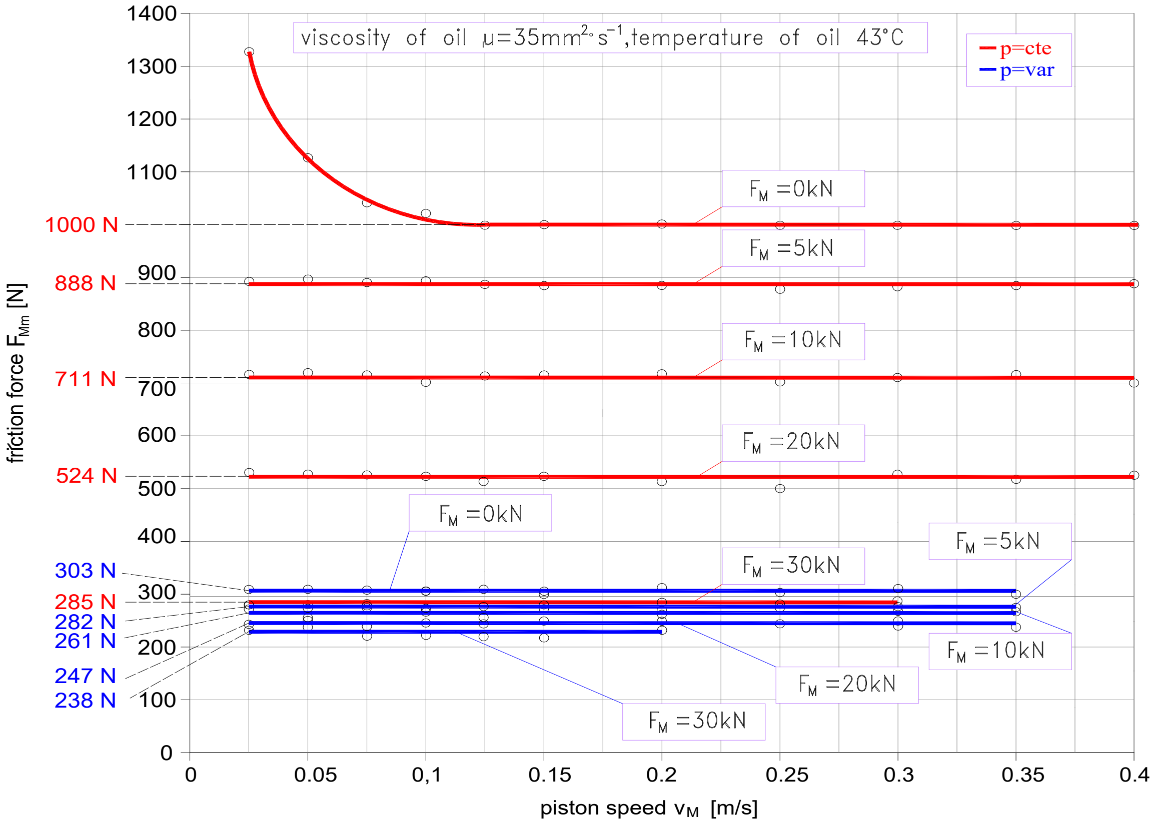

Fig. 5. The friction force F_{Mm} in the cylinder with throttling control as a function of the velocity v_M of the piston at oil viscosity $\vartheta = 35\ mm^2s^{-1}$ (t = 43ºC) – constant pressure system p = cte and variable pressure system p = var

The chart from Fig. 5 shows the dependence of the friction force F_{Mm} on the velocity v_M of the piston in p = cte and p = var systems. The obtained characteristics allow to state (in the tested object) no influence of the velocity v_M of the piston on the value of the friction force F_{Mm} resistance in the seals. The increase of the friction force F_{Mm} in the p = cte structure as a result of the cylinder's speed v_M decreases is only visible under no load, i.e. at $F_M = 0$. In this case, the friction force F_{Mm} reaches a value greater than 1300 N at a deceleration of v_M below 0.10 m/s. That means more than 30% growth. In contrast, the value of the friction force F_{Mm} in the p = var structure ranged from 238 N in the absence of load to 303 N at the cylinder piston rod load of 30 kN. Comparing the red and blue curves representing the studied structures (Fig. 5), a significant decrease in friction between the piston and the cylinder and between the piston rod and gland in the case of the pressure variant p = var can be noticed [4].

The charts from Fig. 6 and 7 show the dependence of the friction force F_{Mm} on the cylinder with throttling control as a function of its velocity v_M of the piston, with the external force $F_M = 0$ and $F_M = 30$ kN, respectively. As we can see on the graphs, the friction force F_{Mm} has practically no influence on the velocity v_M of the piston rod, while increasing the temperature of hydraulic oil, i.e. decreasing its viscosity. As the viscosity decreases, the oil has less dense and does not lubricate the co-operating surfaces i.e. the cylinder piston and the piston rods.

The graph from Fig. 8 shows the friction force F_{Mm} in the cylinder with throttling control as a function of the external load F_M at the velocity $v_M = 0.30$ m/s of the piston connected with Load Sensing structure. The graph shows that the characteristics tend to decrease as the load F_M on the piston rod of the cylinder increases. The friction losses are mainly influenced by the pressure in the discharge chamber of the cylinder. With external load $F_M = 0$ the outlet pressure is high and decreases as the load on the cylinder increases, resulting in the friction of the cylinder. The decreasing of the friction force F_{Mm} affects in turn on the mechanical efficiency η_M of the hydraulic cylinder.

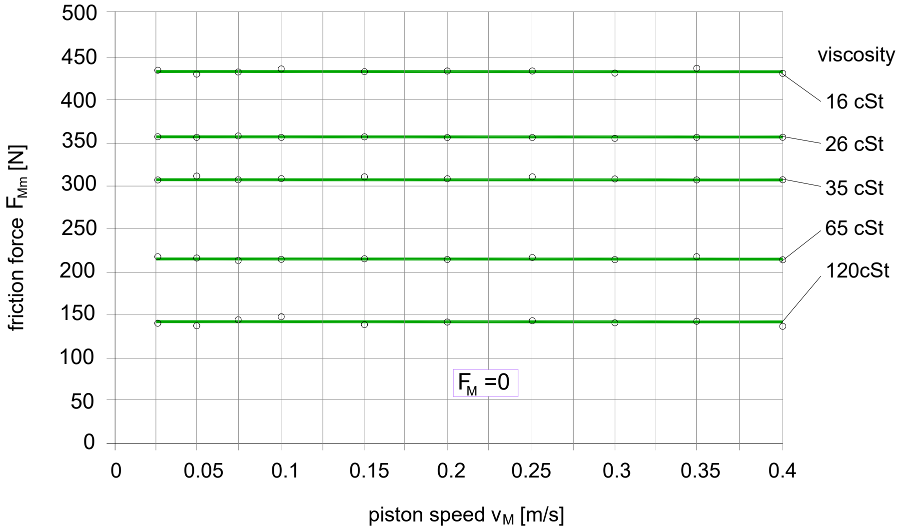

Fig. 6. The friction force F_{Mm} in the cylinder with throttling control as a function of the velocity v_M of the piston, with the force F_M = 0 obtained at different viscosities of the hydraulic oil – Load Sensing structure

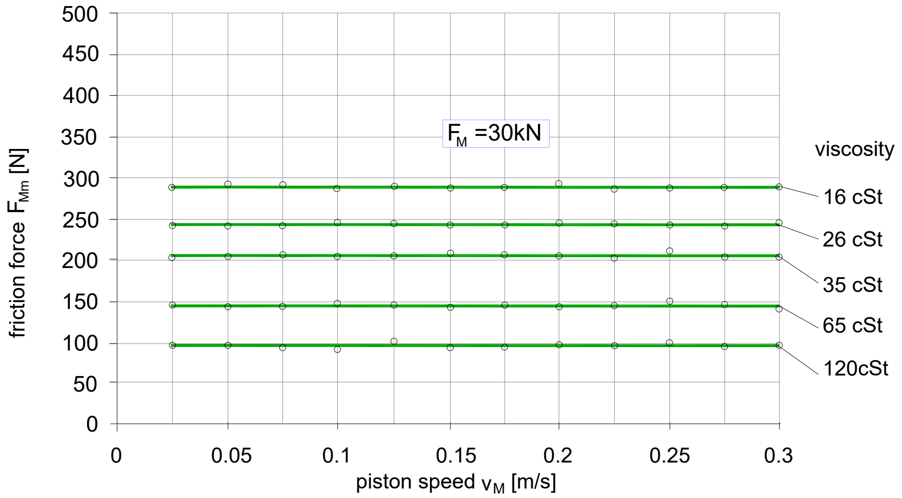

Fig. 7. The friction force F_{Mm} in the cylinder with throttling control as a function of the velocity v_M of the piston, with the force F_M = 30 kN obtained at different viscosities of the hydraulic oil – Load Sensing structure

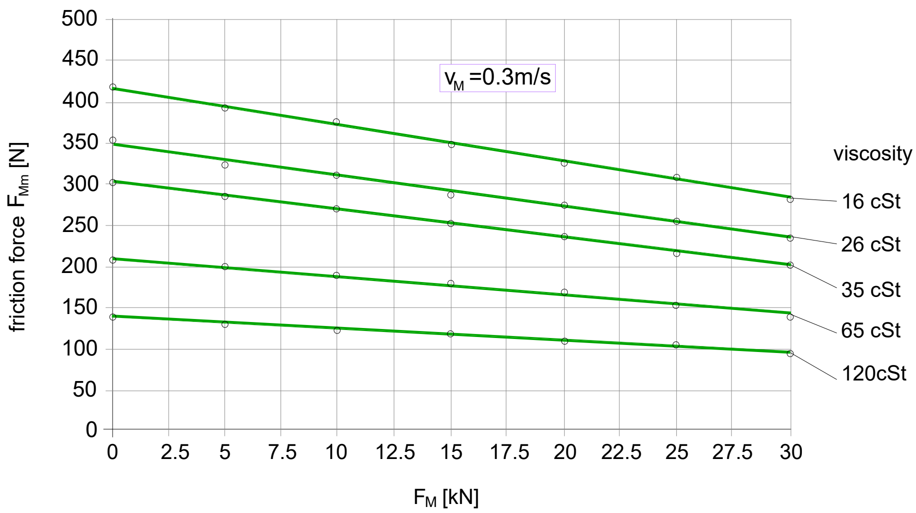

Fig. 8. The friction force F_{Mm} in the cylinder with throttling control as a function of the external load F_M at the velocity v_M = 0.30 m/s of the piston – Load Sensing structure

6. Summary

Losses connected with hydraulic cylinder depend on the type of seal used, its shape, the material it is made from, pre-clamp, and the operating parameters of the cylinder (piston speed, pressure, oil viscosity, temperature). Improperly sealed or assembled seals can cause energy losses of up to 25%. The friction of the elastic seals against the cylinder or piston rod is similar to the Stribeck graph, i.e. the coefficient of friction has the greatest value at zero relative velocity and is subject to intense variations at low speeds (decreasing with increasing speed). At cylinder rod speed $v_M = 0.3$ m/s has mixed friction. In this range, stick-slip vibrations may occur. Above $v_M = 0.3$ m/s the friction can flow smoothly, but pressure (or mechanical clamping) and temperature (liquid viscosity) are also important. As the temperature increases, the friction force rises.

As a result, modern cylinder designs focus on the sealing hinges they use to minimize mechanical losses on friction and thus improve overall efficiency. In manufactured cylinders, this efficiency is between 78% and 99%.

Due to the use of moving seals in the hydraulic cylinder, its energy behaviour is completely different compared to the energy behaviour of a rotary motor, which does not have any seals. The friction force F_{Mm} connected with the work of the sealing joints and the mechanical efficiency η_M of the cylinder are determined not only by the external load F_M but also by the method of the applicable supply of the cylinder resulting from the throttling structure, and in particular the pressure level p_{M2i} generated in the discharge chamber of the cylinder.

The factors determining the value of friction F_{Mm} resistance, which are the load F_M of the cylinder and the applied power structure, in turn have an influence on the pressure values in the inlet and outlet chambers of the hydraulic cylinder.

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