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THE POWER BALANCE DURING CUSHIONING FOR HYDRAULIC CYLINDERS

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ABSTRACT

Cylinder hydraulic brake systems to stop the stroke are built with moving parts to reduce the impact intensity. For correct sizing of hydraulic cylinder cushioning system it is necessary to know how components participate in energy dissipation. This comes from the hydraulic pump working fluid and the kinetic energy of the mechanical system operated.

KEYWORDS: hydraulic cylinders, mechanism, cushioning

1. INTRODUCTION

For the cases of mechanical systems driven by hydraulic cylinders, the initial impact velocity at the end of the stroke is larger than a critical value, mechanical stresses can occur whose effect is removal from service. Given the trend of increasing work rates, to avoid the impact of high speed, braking is performed at the end of stroke. Kinetic energy of moving masses is therefore reduced. Making an impact with lower speed than the critical speed, reduces risks of unwanted decommissioning.

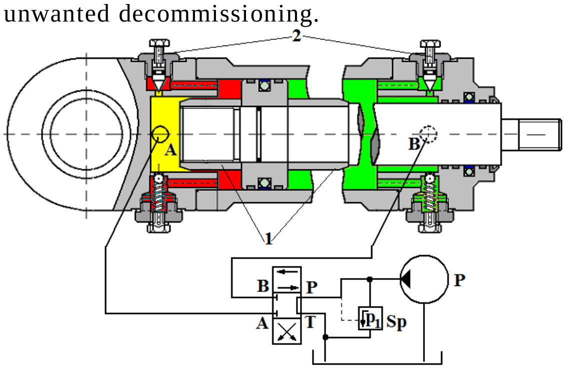

Fig.1 A hydraulic damping cylinder at the end of stroke: 1 - bushes, 2 - variable hydraulic resistance

The braking is achieved by placing to the end of stroke of a hydraulic resistance. Therefore, on the previously passive surface of the piston, it creates a high pressure, as can be seen in Figure 1. Hydraulic resistances convert the kinetic energy of mobile elements into heat. This paper presents an analysis of how energy is transformed during cushioning.

To study will use a simplified mechanical model represented in Figure 2. It is believed that the hydraulic cylinder equipped with speed reduction system to end the stroke works on a mobile mass (which includes the whole mobile mass of the cylinder) that can move on the cylinder axis.

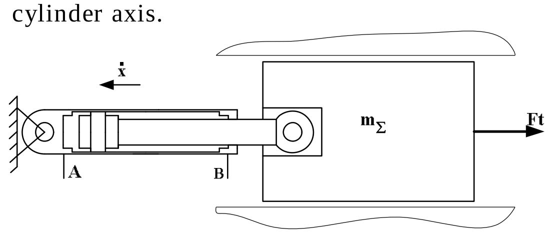

Fig 2. Mechanical system.

A constant force acts on the movable mass. It is a sum of forces formed by technological force and the friction forces to the frameworks.

2. ANALYSIS OF THE BRAKING PROCESS

It considers two periods in the evolution of the braking phenomenon:

It is considered two periods in the evolution of the braking phenomenon:

-The first period (phase "a") from a position of mobile BB plan (which includes front brake bush - Fig. 3.a). Is denoted by x distance from the front brake sleeve receiving hole (AA). The hydraulic oil passes through a conical surface and the hydraulic resistance circuit. Pressure decreases from p_2 to p_k. This period lasts until the AA and BB overlap;

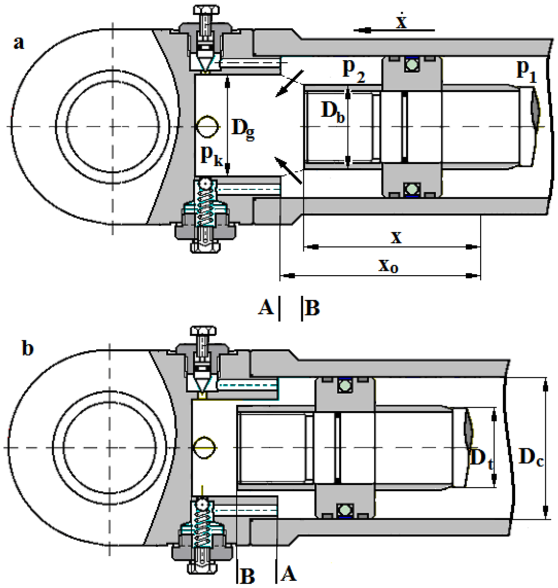

Fig. 3 Phases of hydraulic braking resistance

-The second phase, "b" starts from the moment of overlapping between planes AA and BB and lasts until the beginning of the mechanical impact. And during this period the hydraulic oil flow from the area under high pressure p_2 in the low pressure area p_k. Hydraulic circuits are being followed: I - the variable hydraulic resistance, II - through ring space formed between bush and wall opening.

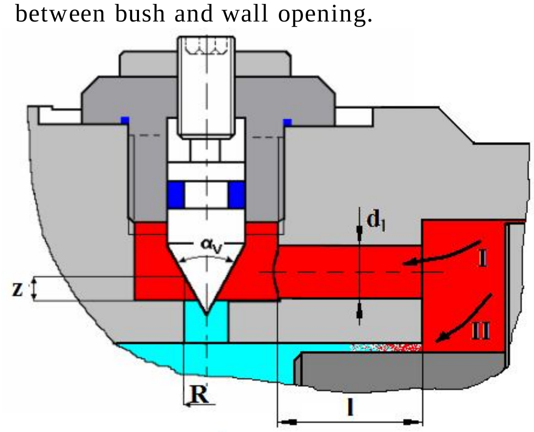

Fig 4. Resistant hydraulic operating in parallel

In [2] are presented mathematical models of hydraulic cylinders equipped with variable hydraulic resistance. In this paper we consider the existence of further useful technological forces. The analysis of forces acting on the whole mobile and the formation of the two-way flow (Figure 4) resulting parallel system

$$f = \begin{cases} f_a(x,\dot{x},\ddot{x}) & 0 \le x \le x_0 \\ f_b(x,\dot{x},\ddot{x}) & x_0 < x < x_0 + l_b \end{cases} \tag{1}$$

where

$$\begin{aligned} & f_a(x,\dot{x},\ddot{x}) = m_\Sigma \ddot{x} - p_k \cdot \frac{\pi}{4} D_c^2 \\ & -\frac{\pi}{4}\left(D_c^2 - D_b^2\right) \cdot \frac{\left[\dot{x}\frac{\pi}{4}\left(D_c^2 - D_b^2\right)\right]^2 \rho}{F^2(x,\dot{x})} - \\ & -\sum F_f(\dot{x}) + p_1 \frac{\pi}{4}\left(D_c^2 - D_t^2\right) + Ft(x,\dot{x}..) \end{aligned} \tag{2}$$

and

$$\begin{aligned} & f_b(x,\dot{x},\ddot{x}) = m_\Sigma \ddot{x} + p_1 \frac{\pi}{4}\left(D_c^2 - D_t^2\right) \\ & -\Delta p_{2k} \frac{\pi}{4}\left(D_c^2 - D_g^2\right) \\ & -p_k \frac{\pi}{4} D_c^2 - \sum F_f(\dot{x}) + Ft(x,\dot{x}..) \end{aligned} \tag{3}$$

Along the paths (I or II) of the agent that connects the hydraulic high pressure (p_2) to the low pressure area (p_k) it can be written the energy balance equation

$$c_v \cdot T_2 + \frac{p_2}{\rho} + g \cdot z_2 = c_v \cdot T_k + \frac{p_k}{\rho} + g \cdot z_k \tag{4}$$

Considering that $z_2 \cong z_k$ and $v_2 \cong v_k \cong 0$ 2 it resulted in an increase in temperature

$$\Delta T_{2k} = T_k - T_2 = \frac{p_2 - p_k}{\rho \cdot c_v} = \frac{\Delta p_{2k}}{\rho \cdot c_v} \tag{5}$$

regardless of the route that makes displacement hydraulic oil.

Power generated at a certain time by hydraulic pump, $P_{ph} = p_1 \cdot Q_{ph}$, is used for:

- Carrying payload

$$P_U = Ft \cdot v \ ; \tag{6}$$

- Flow losses through pressure valve

$$P_{SP} = \left(Q_{PH} - v \cdot \frac{\pi \cdot D_C^2}{4}\right) \cdot p_1 ; \tag{7}$$

- Cover all losses that appear due to friction

$$P_{Ff} = \sum F_f \cdot v \; ; \qquad (8)$$

- Heating the hydraulic oil by hydraulic resistance

$$P_Q = \Delta T_{2k} \cdot c_v \cdot \left(Q_I + Q_{II}\right) \cdot \rho \; ; \qquad (9)$$

- Acceleration of mobile masses

$$P_{EC} = m_{\Sigma} \cdot \frac{\left(v_i^2 - v_{i-1}^2\right)}{2} \, . \qquad (10)$$

The energies, introduced or consumed during this complex process is determined by integrating the components in the period considered

$$E_x = \int_0^{t_f} P_x \cdot dt \, . \qquad (11)$$

3 NUMERICAL SIMULATION

Given the complexity of the system (1) it was used the numerical integration Runge-Kutta method of order four [2]. Or made changes and additions that take into account the assumptions and parameters follow. Numerical integration step chosen took into account the need to ensure the convergence of process.

Table 1 shows the results of numerical simulations carried out in the conditions previously specified. The first columns describes the variation of kinematics parameters (x, v) and the next hydraulics parameters. Following powers and finally calculated mechanical efficiency and temperature variation of hydraulic agent. Rendering step is 1 of 32 results.

Table 1

t	x	v	p_2	v_{dr}	Q_{dr}	v_{cs}	Q_b	p_1	P_{ph}	P_U	P_{SP}	P_Q	P_{EC}	η_m	ΔT_{2k}
ms	mm	$\frac{mm}{s}$	MPa	$\frac{m}{s}$	$\frac{dm^3}{s}$	m/s	$\frac{dm^3}{s}$	MPa	kW	kW	kW	kW	kW		K
0	0	0.1	0.15	0	0	0	0	18	66	0.02	66	0	0.21	0.02	0
17	1.8	119.1	0.15	0	0	0.52	1.24	18	66	19.1	33	0	13.54	28.9	0
34	4.0	238.1	0.15	0.01	0	1.24	2.48	18	66	38.1	0.02	0	0.61	57.7	0
51	8.1	238.2	0.15	0.01	0	1.66	2.48	10.7	39.3	38.1	0	0	0	96.9	0
68	12.1	238.2	0.15	0.02	0	2.51	2.48	10.7	39.3	38.1	0	0	0	96.9	0
85	16.2	238.2	0.15	0.1	0	5.18	2.48	10.7	39.3	38.1	0	0	0	96.9	0
102	20.2	238.2	2.38	49.9	0.76	59.2	1.81	12.2	44.8	38.1	0	5.5	0	85	1.33
119	24.2	226.1	14.3	126.4	1.68	59.5	0.76	18	66	36.2	3.34	33.4	-8	54.8	8.47
136	27.7	189.9	13.5	122.5	1.63	59.5	0.35	18	66	30.4	13.4	26.4	-5.03	46	7.96
153	30.8	169.9	12.1	116	1.55	59.5	0.22	18	66	27.2	18.9	21.1	-2.08	41.2	7.14
170	33.6	160.7	11.5	112.9	1.5	59.5	0.17	18	66	25.7	21.5	18.9	-0.93	38.9	6.77
187	36.3	156.1	11.2	111.6	1.49	59.5	0.14	18	66	25	22.8	18.0	-0.49	37.8	6.62
204	38.9	153.4	11.1	111.1	1.48	59.5	0.12	18	66	24.5	23.5	17.5	-0.3	37.2	6.55
221	41.5	151.6	11.0	110.8	1.48	59.5	0.1	18	66	24.3	24	17.2	-0.21	36.8	6.52
238	44.0	150.4	11.0	110.6	1.47	59.5	0.09	18	66	24.1	24.3	17.0	-0.16	36.5	6.50
255	46.6	149.4	11.0	110.5	1.47	59.5	0.08	18	66	23.9	24.6	16.9	-0.12	36.2	6.48
272	49.1	148.6	11.0	110.4	1.47	59.5	0.07	18	66	23.8	24.8	16.8	-0.1	36	6.47
289	51.6	147.9	11.0	110.4	1.47	59.5	0.07	18	66	23.7	25	16.7	-0.08	35.9	6.47
306	54.2	147.4	11.0	110.3	1.47	59.5	0.06	18	66	23.6	25.2	16.6	-0.07	35.7	6.46
323	56.7	146.9	10.9	110.3	1.47	59.5	0.06	18	66	23.5	25.3	16.5	-0.06	35.6	4.66
340	59.19	146.5	10.9	110.3	1.47	59.5	0.06	18	66	23.4	25.4	16.5	-0.05	35.5	6.46

4 CONCLUSIONS

Numerical simulations show the possibilities of the mathematical model and o the computer program.
The evolution in time of powers introduced and consumed is shown in Figure 5.

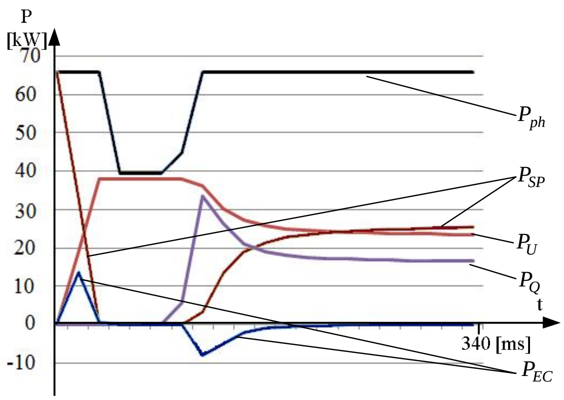

Fig. 5 Evolution of power over time

From the diagram it is found that:
- Hydraulic pump consumes the maximum power at startup (to speed up mobile masses) and after starting the process of effective brake;
- The useful power P_U is maximum in the period when the speed is high;
- The power dissipation P_Q due to hydraulic resistance increases rapidly after the start of braking. Then it decreases due to decrease speed of mobile masses and consequently the evacuated flow;
- Initial kinetic energy is stored (on acceleration) and then released (the brake).

Figure 6 shows the time evolution of mechanical efficiency calculated with relation

$$\eta_m = \frac{P_U}{P_{ph}} \tag{12}$$

It is found that braking by using hydraulic resistance is a significant efficiency reduction factor.

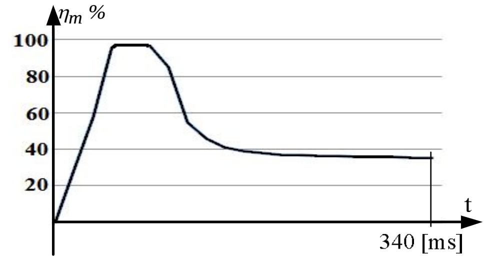

Fig. 6 The evolution of efficiency

Quantities of heat generated are significant and results in heating of the hydraulic oil. Theoretical the temperature increases by more than 8 C^o, as can be seen in Figure 7.

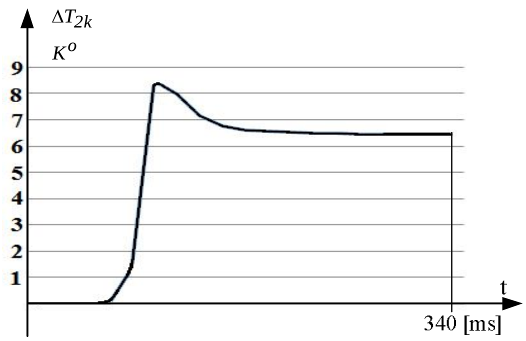

Fig. 7 Diagram of the theoretical temperature increase

In reality the temperature increase is smaller because some heat is transferred by convection to the hydraulic cylinder body.

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