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ON THE POSSIBILITIES OF REDUCING THE INTENSITY OF THE IMPACT AT THE END OF THE RACE IN THE CASE OF DRIVE WITH HYDRAULIC CYLINDERS

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ABSTRACT

Increasing speeds has led to an increase in the intensity of the impact between piston and cylinder body at the end of the race. Hence, there aroused a need for reducing mobile assembly speed before the impact starts. This paper tries to explain why only the cylinder equipped with hydraulic cushioning system at the end of the race is not satisfactory for carrying out cushioning. In fact, cushioning is based on the reduction of flow with replenished active room of the hydraulic cylinder. The main factors on which depends the cushioning process are: hydraulic power system, kinematics of the actuated mechanism and the geometry of the cushioning system placed inside the hydraulic cylinder. To develop a cushioning system without taking into account all these factors is not therefore possible.

KEYWORDS: hydraulic cylinder, cushioning, impact, actuated mechanism

INTRODUCTION

When assessing the behavior of a mechanical system driven by a hydraulic cylinder, the following aspects are to be considered:

- the hydraulic scheme of the hydraulic oil supply;
- the kinematic diagram of the mechanical system.

Hydraulic schemes used have different degrees of complexity. The pump type is a really important parameter when using hydraulic cylinders with cushioning at the end of stroke. The pump can be with a fixed (Figure 1) or variable (Figure 2) specific displacement volume.

In the first case, in order to reduce the relative speed of the piston inside the hydraulic cylinder, it is necessary that the deceleration valve begin to evacuate, to the tank, a part of the flow. The moment when the braking process begins depends on the regulated pressure p_{1m} of the deceleration valve.

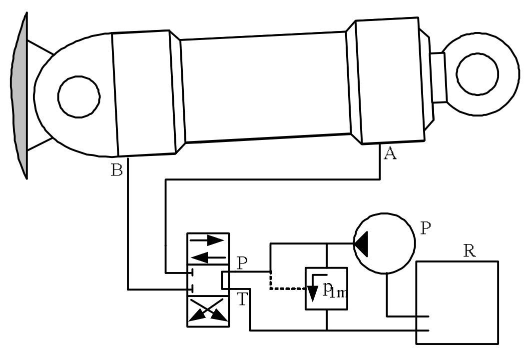


Fig. 1 Hydraulic scheme in which the pump is of a fixed displacement volume type

In the second case, the initial flow rate reduction is achieved by the own hydraulic pump systems. A further reduction in the flow rate occurs after the moment when the set pressure, p_{SG}, of the deceleration valve is reached.

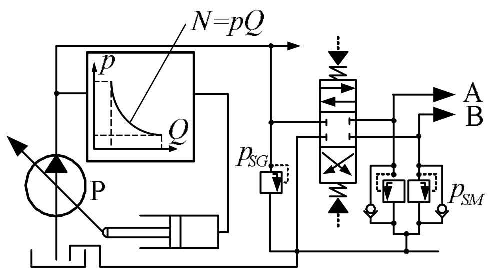


Fig. 2 Hydraulic scheme in which the pump is of a variable displacement volume type

Flow changes also occur due to:

- the altering of the rotation speed of the electric motor drive of the hydraulic pump;
- the hydraulic oil compressibility, flow changes being calculated according to the relationship:

$$Q_c = \frac{V_0 + \frac{\pi \cdot D_c^2}{4} \cdot x}{E_c} \cdot \frac{dp}{dt} \qquad (1)$$

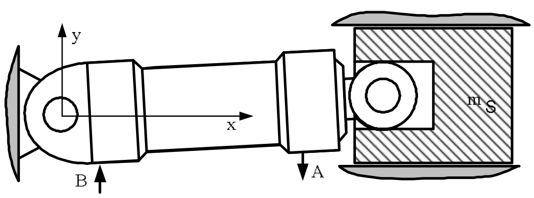


Fig. 3 Kinematic scheme of a rectilinear movement of the actuated element

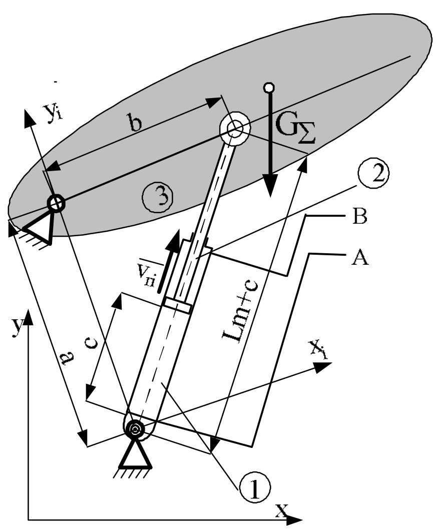


Fig. 4 Kinematic scheme of a fixed axle rigid body (3) revolved by a hydraulic cylinder having as main parts the cylinder body (1) and the piston rod (2)

The second important factor in the analysis of the movement of hydraulic cylinders with cushioning at end of stroke is the kinematic of the driven mechanisms.

Figures 3 and 4 show the most common applications. In figure 3, the hydraulic cylinder is used for a rectilinear movement of a mass, m_Σ. In this case, by neglecting the mass of the hydraulic cylinder, the force of inertia will manifest itself in the direction of motion.

Figure 4 shows a mechanism that includes item 3, a rigid body with fixed axle. In this case, inertia is of a rotational type and depends on the movement parameters and mass distribution characteristics of the actuated item 3.

2 CALCULUS OF THE CUSHIONING MOTION IN HYDRAULIC CYLINDERS

For the mechanisms operated by hydraulic cylinders fitted with cushioning systems at the end of stroke, the mathematical models which allow for numerical simulations of the mechanism behavior are based mainly on the following aspects:

- the mechanical equilibrium of the elements of the mechanism;
- the fluid flow and the Continuity Equation.

The results are complex functions that describe time variation of the parameters of interest for the cushioning process.

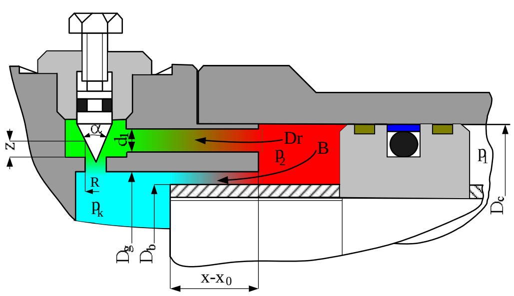


Fig. 5 Flow paths

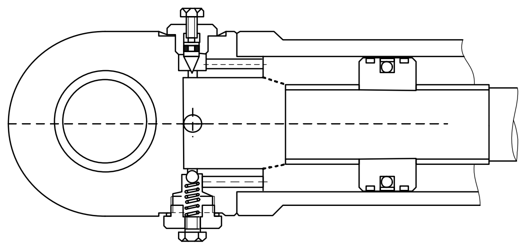
This paper analyzes a cushioning system at the end of stroke which consists of (Fig. 5):

- "B" flow path which is formed between the sleeve mounted on the piston and the cylinder gap of the cylinder body;
- "Dr" path which consists of a d_1 diameter hole and an adjustable hydraulic resistance.

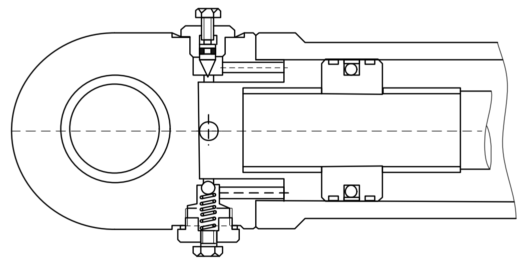
In the analysis of this cushioning system, we consider two phases of movement:

- the first phase the $x - x_0 < 0$ (Fig. 6a), when path "B" is in the form of tapering surfaces (marked by a dotted line);

- the second one, the $x - x_0 > 0$ (Fig. 6b), when path "B" is in the form of an annular clearance formed between the sleeve and the cylinder body.



a



b

Fig. 6 Phases of the assembly movement

The forces acting on the mobile assembly (piston - rod) are described by the axial force equilibrium equation, where:

$$\overline{F}_a + \overline{F}_{p_1} + \overline{F}_{p_2} + \overline{F}_{p_k} + \sum \overline{F}_f = 0 \qquad (2)$$

- $\overline{F}_a$ is the force that occurs in the joint of the cylinder rod. It is due to the technological forces and to the inertia ones acting on the acted item and results from their mechanical equilibrium;

- $\overline{F}_{p_1}$ is the pressure force acting on the active face of the piston;

- $\overline{F}_{p_2}$ is the pressure force acting on the passive face of the piston and contributes to the reduction of the mobile assembly speed;

- $\overline{F}_{p_k}$ is the pressure resistant force caused by the output hydraulic circuit to the tank;

- $\sum \overline{F}_f$ is the sum of pressure resistant forces from guides and seals.

Regarding the flows, they are:

- The input flow Q_i, entering the hydraulic cylinder. It is imposed by the characteristics of the hydraulic pump, pressure p_1, compressibility of the hydraulic oil and eventually, the characteristics of the deceleration valve (if it is open);

- The flow Q_a, evacuated along the two previously described flow paths:

$$Q_a = Q_B + Q_{Dr} \qquad (3)$$

Both Q_i and Q_a are bound by the velocity of the mobile assembly, $\dot{x}$, and by the two faces of the piston, from the corresponding chambers of the hydraulic cylinder.

Q_B flow depends on the phase in which the cushioning process is (phase a or b – Fig. 6)

Q_{Dr} flow passes over two hydraulic resistances connected in series, the last being an adjustable hydraulic valve.

It is important to note that, regardless of the path followed by the hydraulic oil, "B" or "Dr", the pressure drop is the same. This is used to estimate the parameters of step "n + 1" from the parameters at the known step "n".

The parameters determination of the hydraulic resistance is a difficult task. There are certain recommendations in this regard in the literature [6], although their level of generality is questionable. Therefore, it is more appropriate to conduct experimental determinations or, in the absence of the physical model, to make numerical simulations of flow, as it can be seen in Fig. 7 [5].

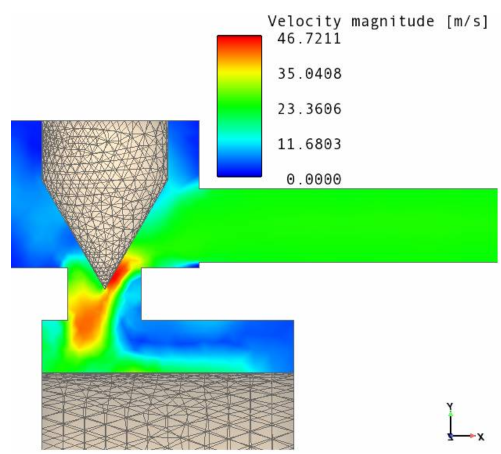


Fig. 7. The result of numerical simulation of flow passing through the adjustable hydraulic resistance

The data thus obtained are used to construct the functions describing the flow through the hydraulic cylinder, while cushioning.

By combining mechanical equilibrium equations and the fluid flow and continuity equation, in the two phases of the cushioning process, we obtain the following systems of differential equations:

$$f = \begin{cases} f_a\left(x, \dot{x}, \ddot{x}, p, \dot{p}\right) = 0 & \text{for} \quad x - x_0 < 0 \\ f_b\left(x, \dot{x}, \ddot{x}, p, \dot{p}\right) = 0 & \text{for} \quad x - x_0 > 0 \end{cases} \qquad (4)$$

Solving differential equations system was made using the Runge-Kutta algorithm of fourth order. The algorithm starts from the known initial values and estimates the appropriate values for the next step. The effective calculus of the cushioning motion was conducted with numerical calculation software.

3. CONCLUSIONS

Consequently, it is obvious that for each individual application, specific characteristics of the hydraulic cylinder cushioning system are required. It is not therefore possible to make a universal hydraulic cylinder, suitable to act adequately in any application.

It is possible that in some applications cushioning does not occur satisfactorily if the whole system was not considered in this respect. In some applications even a certain degree of temporary acceleration may arise, with undesirable effects on the integrity of the system.

It should be noted that without reducing the flow entering the hydraulic cylinder, one cannot talk about the cushioning process, or about the reduction of the piston speed that will occur with the impact at the end of the race.

The design of hydraulic cylinders uses numerical calculations and simulations of the functioning of the system. On this occasion shall be determined:

- the length of the sleeve mounted on the piston-rod assembly;
- the size of the space between the sleeve mounted on the piston and the cylinder gap of the cylinder body;
- the geometry of the adjustable hydraulic resistance (deceleration valve).

After the physical creation of hydraulic cylinders, the next step is to perform their adjustment and testing on specialized stands. On this occasion the hydraulic variable resistance adjustment is also made.

Addressing this kind of issue is necessary because even small changes in the geometry of flow areas can lead to significant changes in pressure drop.

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