RESEARCH ARTICLE | APRIL 24 2024

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Sasko Dimitrov ✉; Mihail Komitovski

AIP Conf. Proc. 3078, 060002 (2024)
https://doi.org/10.1063/5.0208519

Theoretic Research on Transient Process in Hydraulic Cylinder with Cushioning Device with Constant Gap

Sasko Dimitrov [1, a)] and Mihail Komitovski [2, b)]

[1]Faculty of Mechanical Engineering, Goce Delcev University, Stip, North Macedonia
[2]Faculty of Power Engineering and Power Machines, Technical University, Sofia, Bulgaria

[a)] Corresponding author: sasko.dimitrov@ugd.edu.mk
[b)] mkomitowski@web.de

Abstract. Hydraulic cylinders are devices that very often move large masses with high potential energy at high velocities. Abrupt stopping at its end positions, the hydraulic cylinders may damage or destroy itself or the whole system. To prevent the hydraulic system from damaging, the incorporation of the cushioning devices are introduced in the design of the hydraulic cylinder to help smoothly stop at the end position. In this work a mathematical model of a cushioning device with constant gap is developed and simulated the transient process at the end of its stroke.

INTRODUCTION - THEORETICAL BACKGROUND

Abrupt stopping of the mass driven by the hydraulic cylinder leads to unacceptable shocks and vibrations which can damage the cylinder or break the pipelines. The maximum permissible speed depends on the magnitude of the kinetic energy and usually does not exceed 0.02 m/s when stopping without damping in the cylinder head or 0.15 m/s when stopping suddenly in an intermediate position by closing the working chambers of the cylinder, for example by a directional control valve with direct electromagnetic control.

Many authors have researched the transient process in hydraulic cylinders at the end of its stroke. In [1] dynamic models are developed for describing the cylinder cushioning systems that when solved numerically predicts the pressure and velocity responses of the cylinder with time. The study includes modelling of cushion profiles and performing analytical analysis for achieving a constant deceleration during cushioning phase. It has been observed that performance of cushion is highly sensitive to variations in the spear shape profile. Cushioning device with more intricate and complicated structure with 9 steps using CFD analysis method is presented in [2]. The CFD model can be helpful for analysing the flow field in cushion chamber. In [3] a mathematical model for horizontally mounted cylinder with cushioning from the piston side has been developed and numerically solved. Few proposals have been given for improvement of the transient process. The work [4] describes the theoretical-experimental study of an auto-adjustable stroke end cushioning device utilized in hydraulic cylinders, focusing the characterization of the bush geometry effect on the cushioning achieved. A nonlinear model is presented which includes the physical phenomena that exert a significant influence on the performance of this hydraulic component, such as: friction, fluid compressibility and pressure energy loss in the cushioning section. The model is validated through the comparison between theoretical and experimental results, under different conditions of load, supply pressure and piston speed.

The acceleration at start-up $a_{st} = \dfrac{p_1 \cdot A_1 - F_c}{m}$ depends on the driving force $p_1 \cdot A_1$ and the resistance force F_c at the initial moment (when the back pressure is zero) and on the mass m of the moving parts. In case the maximum driving force $p_1 \cdot A_1$ is very large, a reduction in the starting acceleration can be achieved by gradually opening the

12th International Scientific Conference "TechSys 2023" – Engineering, Technologies and Systems
AIP Conf. Proc. 3078, 060002-1–060002-7; https://doi.org/10.1063/5.0208519

directional control valve when switching it and correspondingly slowing down the rate of increase in time of the pressure p_1 in the cylinder. Electro-hydraulically controlled proportional directional control valves with electronic control and servo valves have the possibility for delayed opening of the flow areas of the distributor.

Smooth stop at the end of the cylinder stroke is performed by cushioning devices built into its covers (Fig. 1) [5]. Upon reaching the distance L_d before the end of the stroke, a specially shaped protrusion to the piston closes a certain space from the return area of the cylinder, forcing the oil from this chamber to pass through a throttle with section area A_{ds}. The pressure in the enclosure chamber increases to p_d, creating a braking force. The deceleration is determined by the size of the throttle section area A_{ds} during the braking distance L_d.

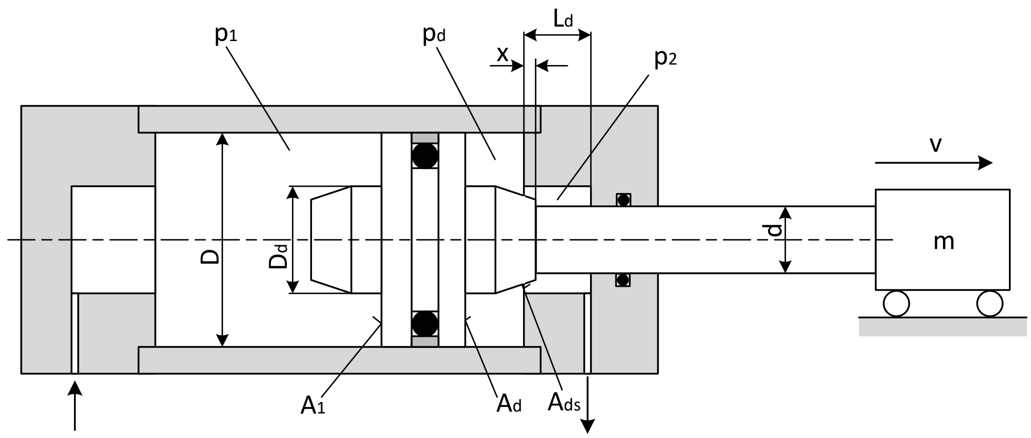

FIGURE 1. Cushioning device in hydraulic cylinders

MATHEMATICAL MODELING OF A CUSHIONING DEVICE WITH CONSTANT GAP FOR VERTICALLY MOUNTED HYDRAULIC CYLINDER

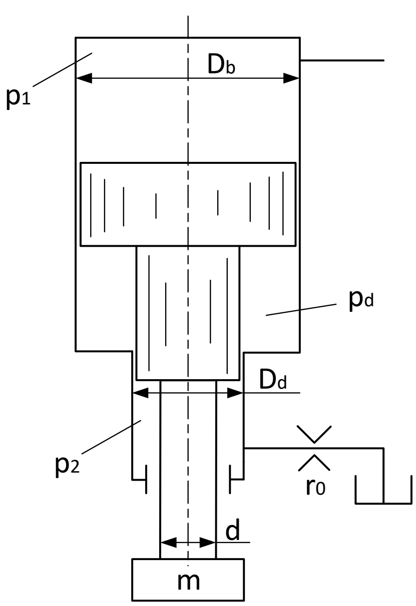

FIGURE 2. Vertical hydraulic cylinder with cushioning device

The subject of this research is mathematical modeling and simulation of the transient processes when a vertically placed hydraulic cylinder stops at the end of its stroke. The hydraulic cylinder with piston diameter D_b and rod diameter d is vertically mounted (Fig. 2). At the end of its rod a mass m is hanged which tends to move vertically under the influence of gravitational force. At the rod side chamber of the cylinder there is a restrictor r_0 to control the mass velocity v, i.e. to exert a counter – pressure p_2 ($p_2 = p_d$ in stationary motion) to prevent the mass m from falling uncontrollably. When the cylinder comes to a distance L_d (Fig. 1) to the end stroke, a special shape of the rod, so-called cushioning device begins to enter into the sleeve made in the cylinder head. At that moment the cushioning process begins.

On Fig. 3 a constant, concentric gap formed between a cylindrical cushioning device of length L_d and a cylindrical sleeve with diameter D_d is presented. The height of the gap is the same and constant along the length of the cushioning L_d .

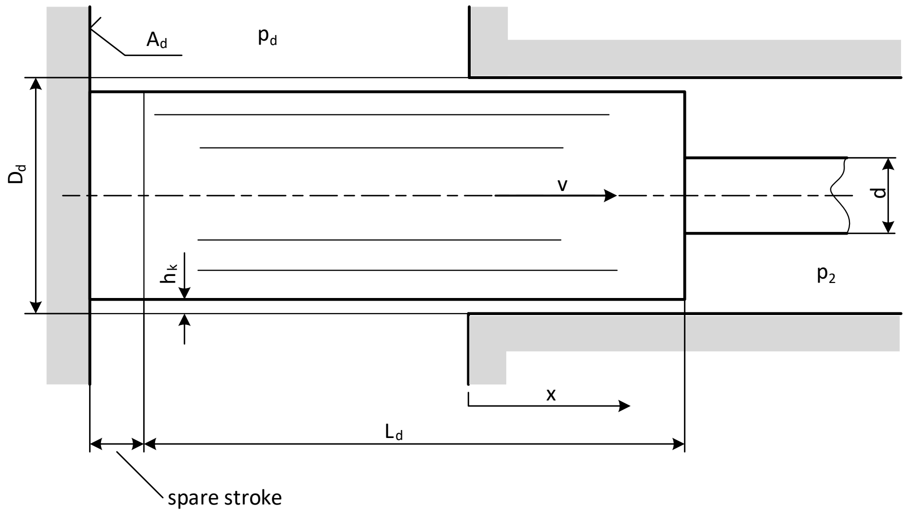

FIGURE 3. Cylindrical cushioning device with constant gap

In vertical position of the cylinder with mass m at its rod end and initial velocity v_0 , assuming laminar flow and neglecting the compressibility of the oil, the differential equation describing the motion of the mass has the form [5]:

$$m \cdot \frac{dv}{dt} = m \cdot g - p_d \cdot A_d - p_2 \cdot A_{ds} \tag{1}$$

Or

$$m \cdot \frac{dv}{dt} = m \cdot g - p_{d,2} \cdot A_d - p_2 \cdot A_2 \tag{2}$$

where: m – the moving mass; v – the velocity of the mass; g – the gravitational constant; p_d – the rod side pressure, the braking pressure (Fig. 3); p_2 – the pressure in the cylinder head cave; $p_{d,2} = p_d - p_2$ – the pressure drop in the restriction area A_{ds} of the cushioning device; $A_2 = \frac{\left(D_b^2 - d^2\right) \cdot \pi}{4}$ – the rod side area of the cylinder; $A_d = \frac{\left(D_b^2 - D_d^2\right) \cdot \pi}{4}$ - the rod side area without cushioning (Fig. 3); $A_{d.s} = \frac{\left(D_d^2 - d^2\right) \cdot \pi}{4} = A_2 - A_d$ – the restriction area of the cushioning device.

When moving in steady - state mode, before the cushioning, the equation of motion will be

$$m \cdot g = p_2 \cdot A_2 \tag{3}$$

and the pressure drop through the restrictor will be [6]

$$p_2 = r_0 \cdot q_2^2 = r_0 \cdot A_2^2 \cdot v_0^2 \tag{4}$$

where: r_0 – the restriction coefficient of the restrictor; q_2 – the flow through the restrictor; v_0 – the stationary velocity of the mass before cushion.

Combining the eq. (3) and (4), we deduce:

$$m \cdot g = r_0 \cdot A_2^3 \cdot v_0^2 \tag{5}$$

When cushioning begins to work, the pressure drop across the restrictor will be

$$p_2 = r_0 \cdot q_2^2 = r_0 \cdot A_2^2 \cdot v^2$$

And

$$p_2 \cdot A_2 = r_0 \cdot A_2^3 \cdot v^2 \tag{6}$$

Inserting eq.5 in eq.6 and then in eq. 2, we deduce:

$$m \frac{dv}{dt} = m \cdot g \cdot \left(1 - \frac{v^2}{v_0^2}\right) - p_{d,2} \cdot A_d \tag{7}$$

The pressure drop $p_{d,2}$ is determined by the solution of the differential equation of the flow in the cylindrical concentric gap of the cushioning device, when the cushioning device enters the sleeve (Fig. 3) [7], [8]:

$$p_{d,2} = \frac{12 \cdot \eta \cdot A_d \cdot v \cdot x}{\pi \cdot D_d \cdot h_k^3} \tag{8}$$

Inserting the eq. (8) in eq. (7) and moving to the dimensionless coordinates: $V = \dfrac{v}{v_0}$ – the dimensionless velocity; $X = \dfrac{x}{L_d}$ – the dimensionless displacement; $\tau = \dfrac{t}{T}$ – the dimensionless time, where $T = \dfrac{L_d}{v_0}$ – the time constant; the cushion differential equation (7) takes the form

$$\frac{dV}{d\tau} = A \cdot \left(1 - V^2\right) - B \cdot V \cdot X \tag{9}$$

where: $A = \dfrac{g \cdot t}{v_0} = \dfrac{g \cdot L_d}{v_0^2}$ and $B = \dfrac{12 \cdot \eta \cdot A_d^2 \cdot L_d^2}{\pi \cdot D_d \cdot h_k^3 \cdot v_0 \cdot m}$.

SIMULATION OF THE TRANSIENT PROCESS OF THE CYLINDER

The solution of the differential equation (9) represents the simulation of the transient process of the moving mass. The design parameters are the following: the moving mass $m = 3000$ *kg*, the initial velocity $v_0 = 0.0617$ *m/s* $= 3.7$ *m/min*, the oil kinematic viscosity $\nu = 46 cSt$, the piston diameter $D_b = 0.25$ *m*, the rod diameter $d = 0.11$ *m*, the diameter of the cushioning device $D_d = 0.145$ *m*, the length of the cushion $L_d = 0.25$ *m*, the height of the gap in the cylindrical section $h_k = 0.00012$ *m* $= 0.12$ *mm*.

The adaptive Runge-Kutta method was used to solve the nonlinear differential equations [9]. This method, based on the fifth order Runge-Kutta method, estimate the truncation error at each integration step and automatically adjust the time step size to keep the error within prescribed limits. Determination of a suitable step size in conventional

Runge-Kutta method can be a major headache in numerical integration. If the step size is too large, the truncation error may be unacceptable. If the step size is too small, higher computational resources are necessary. A constant step size may not be appropriate for the entire range of integration. In stiff differential equations, if the solution curve starts off with rapid changes before becoming smooth, it is recommended to use small step size at the beginning and increase it as it reaches the smooth region. This is where adaptive method come in.

The solution of the differential equation (9) and the transient process of the researched cylinder at the end of its stroke is presented at the diagrams on Fig. 4 and Fig. 5.

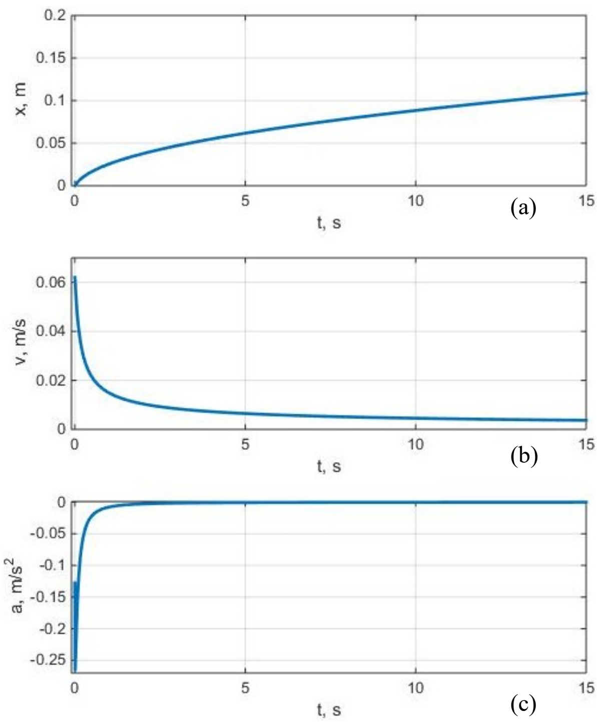

FIGURE 4. Transient process in terms of time: (a) displacement, (b) velocity and (c) acceleration

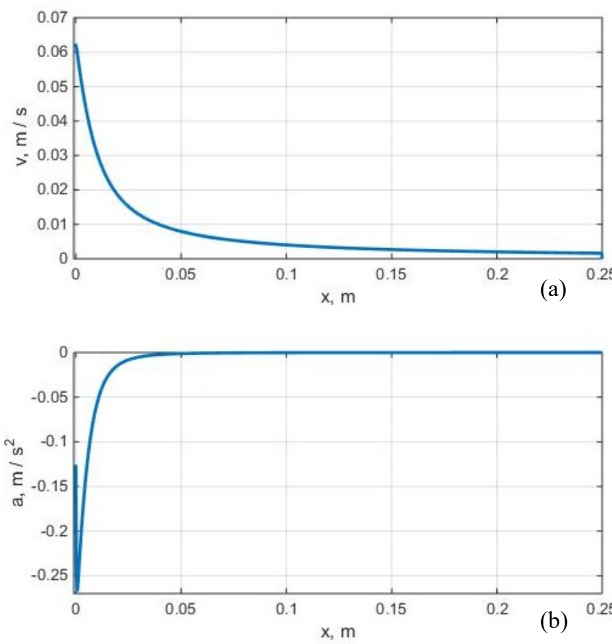

FIGURE 5. Transient response in terms of cylinder stroke: (a) velocity and (b) acceleration

Maximum deceleration is $a_{max} = 0.027\ m/s^2$ – Fig. 4. So, the hydraulic cylinder is loaded with additional inertial force of $F_a = m\ a_{max} = 8100\ N$. As a result of this force, undesirable shocks occur which can damage the hydraulic cylinder or the hydraulic system as a whole. Very often this additional force can not be accepted and must be decreased. Also, the diagram shows rapid decrease of the cylinder velocity. After 0.2 seconds of the beginning of the cushioning process, the velocity is halved to the value of $v = 0.032\ m/s$ and after 12 seconds the velocity is almost constant of around $v = 0.003\ m/s$ which is almost 20 times lower than steady-state velocity v_0 .

Maximum deceleration occurs immediately when cushioning process begins – Fig. 5. At that moment the velocity is still high of around $v = 0.055\ m/s$. At the end of the stroke the velocity is at acceptable value of around $v = 0.002\ m/s$.

A better transient process is obtained by increasing the gap to a value of $h_k = 0.02\ mm$ – Fig. 6 and Fig. 7. The maximum acceleration is significantly reduced, but the speed at the end of the stroke is still high of around $v = 0.01\ m/s$, which is unacceptable for the transitional process.

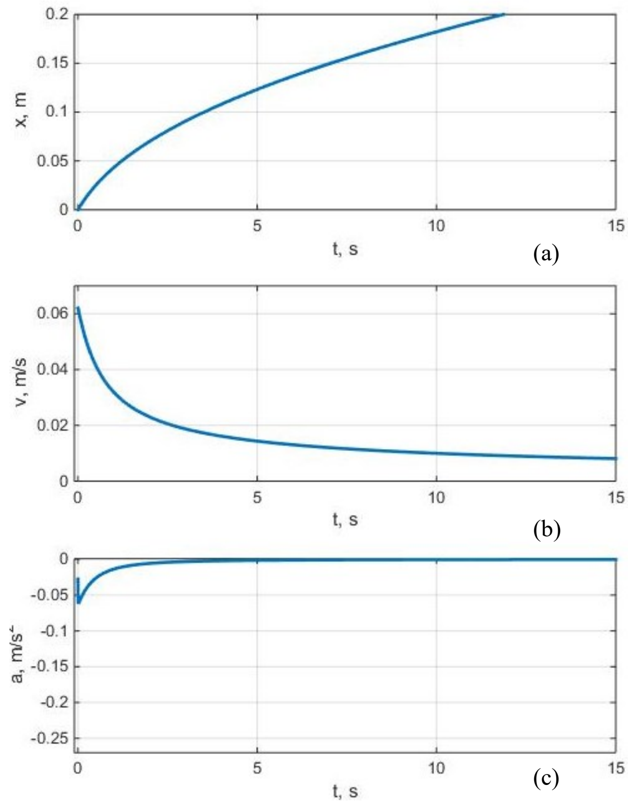

FIGURE 6. Transient process in terms of time: (a) displacement, (b) velocity and (c) acceleration

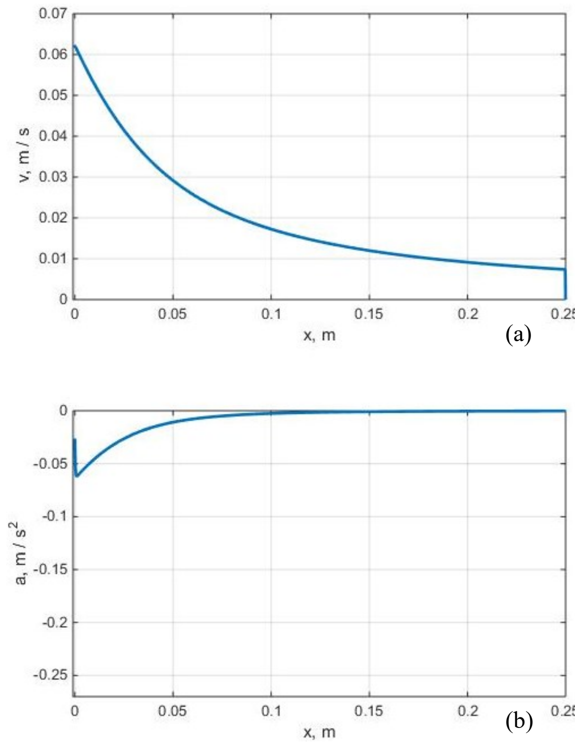

FIGURE 7. Transient response in terms of cylinder stroke: (a) velocity and (b) acceleration

CONCLUSION

Hydraulic cylinders are devices that very often move huge masses with high potential energy at high velocities. Abrupt stopping at its end positions, the hydraulic cylinders may damage or destroy itself or the whole system. To prevent the hydraulic system from damaging, the incorporation of the cushioning devices are introduced in the design of the hydraulic cylinder to help smoothly stop at the end position.

In this work a mathematical model of a cushioning device with constant gap in hydraulic cylinders is developed and simulated the transient process at the end of its stroke. The proposed cushioning device ensures rapid deceleration at the beginning and at the end of the transient process with relatively high additional overload on the hydraulic cylinder as a result of the deceleration – Fig. 4. This type of cushion provides undesirable velocity and acceleration profiles in time of the hydraulic cylinder – Fig. 5 and it is not acceptable as a technical solution of the cushion process in hydraulic cylinders. A better performances are obtained by increase of the gap h_k – Fig. 6 and Fig. 7, but the velocity at the end of the stroke is still high and unacceptable. More smoothly transient process gives modified cushioning devices with conical shape at the beginning of the cushioning [5].

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