Mechanisms and Machine Science

Milan Rackov
Aleksandar Miltenović
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Machine and Industrial Design in Mechanical Engineering

Proceedings of KOD 2024

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Mechanisms and Machine Science 174

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ISSN 2211-0984 ISSN 2211-0992 (electronic)
Mechanisms and Machine Science
ISBN 978-3-031-80511-0 ISBN 978-3-031-80512-7 (eBook)
https://doi.org/10.1007/978-3-031-80512-7

This Springer imprint is published by the registered company Springer Nature Switzerland AG
The registered company address is: Gewerbestrasse 11, 6330 Cham, Switzerland

Preface

Dear Ladies and Gentlemen, respectable Colleagues and Friends of KOD,

The 12th International Conference of Machine and Industrial Design in Mechanical Engineering—KOD 2024 is being organized by the Faculty of Technical Sciences, the University of Novi Sad from the 23rd till the 26th of May 2024 in Hotel Marina at Balatonfüred, Hungary.

The Conference Chair would like to extend special gratitude to the Editor of the Springer book series Mechanisms and Machine Science, Prof. Marco Ceccarelli, for supporting the Conference and giving the chance to all authors to publish a paper in an edition titled Machine and Industrial Design in Mechanical Engineering—Proceedings of KOD 2024.

The basic goal of this conference is to assemble experienced researchers and practitioners from universities, scientific institutes and different enterprises and organizations from within this field. Also, it should instigate more intensive cooperation and exchange of practical professional experiences in the field of shaping, forming and design in mechanical and graphical engineering, industrial design and shaping, product development and management. As there is a pressing need, under the cover of Industry 4.0, for more effective, simpler, smaller, cheaper, noiseless and more esthetically pleasing products that can easily be recycled and are not harmful to the environment, the cooperation between specialists in these fields should certainly be well developed and intricate.

Finally, we would like to thank all the people who have supported the Conference and have helped and encouraged us in all the activities.

We are very grateful to our reviewers whose enormous work of assessing the papers is gratefully appreciated.

We would also like to express our appreciation to our keynote speakers for their invaluable contribution.

We would like to thank the authors themselves for contributing research papers, without whose expert input there would have been no conference.

We are thankful to our sponsor Termometal d.o.o. Ada for helping and supporting the conference.

And, last but not least, we are pleased to acknowledge the assistance provided by the members of the International Scientific Committee and Technical Program Committee of the KOD 2024 Conference.

We wish You to have an interesting and stimulating conference that will bring about numerous fruitful discussions and lay the foundations for future collaboration.

Have an unforgettable stay in Balatonfüred, catch up with old friends and make some new ones.

We wish You good health and success in Your further research and great fortune and happiness in Your personal life.

Conference Chairs of KOD 2024

Balatonfüred
May 2024

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Theoretical Research on Transient Process in Hydraulic Cylinder with Conical Cushioning Devices

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Abstract. Hydraulic cylinders are devices that very often move large masses with high potential energy at high speeds. 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 is introduced in the design of the hydraulic cylinder to help smoothly stop at the end position. In this work a mathematical model of a conical cushioning device in vertically mounted hydraulic cylinder is developed and simulated the transient process at the end of its stroke.

Keywords: Hydraulic Cylinder · Cushioning · Transient Process

1 Introduction - Theoretical Background

High speed of hydraulic cylinders in combination with large moving masses can release large amounts of energy when the rod reaches the end position, which might destroy the hydraulic cylinder. In such cases, hydraulic cylinders with cushioning might be required which reduces the speed to the desire value. Ideally, a constant reduction of the velocity is desired. The cushioning design used should accomplish this objective with effectiveness in all of the operating ranges of the cylinder at the minimum pressure.

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 modeling of cushion profiles and performing analytical analysis for achieving a constant deceleration during the cushioning phase. It has been observed that the performance of the cushion is highly sensitive to variations in the spear shape profile. A 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 analyzing 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

M. Rackov et al. (Eds.): KOD 2024, MMS 174, pp. 717–726, 2025.
https://doi.org/10.1007/978-3-031-80512-7_70

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 work [5] performed a study of mechanical cushioning systems at the end of stroke of hydraulic linear actuators, analyzing various geometries (cylindrical, conical, double conical, etc.) with the objective of establishing a simulation mathematical model, being validated through experimental results. In [6] a relatively simple cushioning is presented. Its operation principle is the modulation of geometry of the discharge orifice (ports of the cylinder) using the piston body. Thus, once the piston reaches the ports, the outlet section is restricted, providing high impedance to flow around the interspace between the piston and the internal cylinder wall. Consequently, an increase of the pressure and a deceleration force are generated. Using Bond graph method, the solution of the theoretical mathematical model is presented and also CFD simulation of the flowing process has been done. Experimental records approved the obtained results by dynamic simulation.

The principal of operation of the cushioning theoretically researched in this paper has been shown on Fig. 1. Smooth stopping at the end of the cylinder stroke is performed by cushioning devices built into its covers [7]. 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.

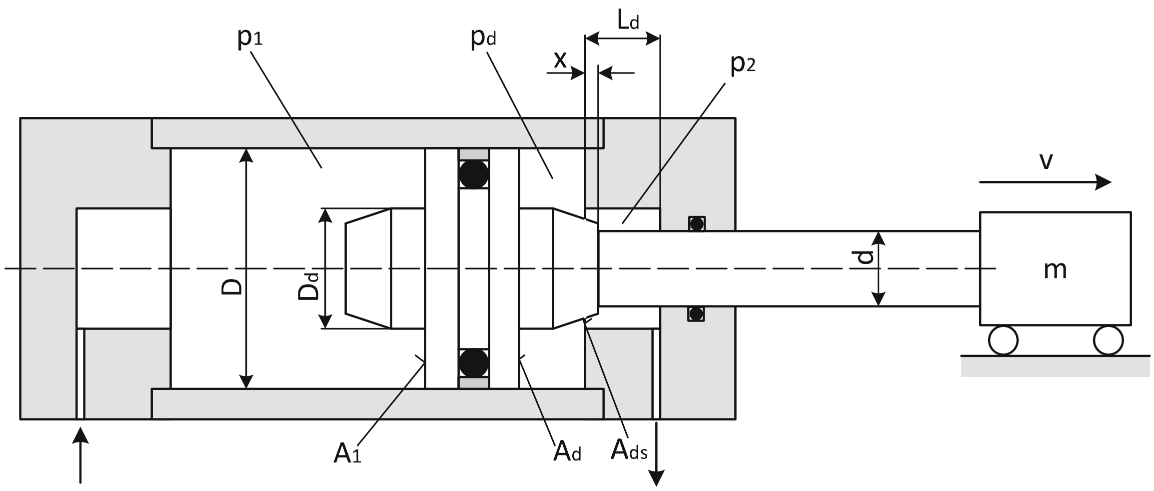

Fig. 1. Cushioning device in hydraulic cylinders.

2 Mathematical Modeling of a Cone Cushioning Device for Vertically Mounted Hydraulic Cylinder

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_d ($p_2 = p_d$ in stationary motion) to prevent the mass m from falling uncontrollably. When the piston rod 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.

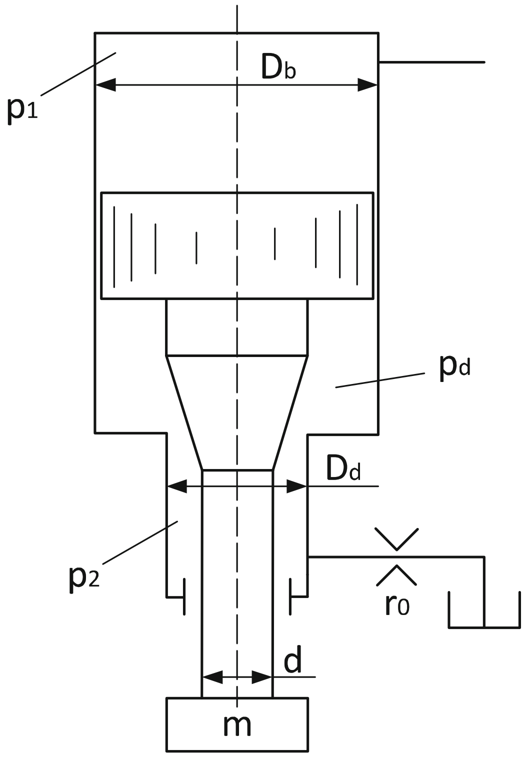

Fig. 2. Vertical hydraulic cylinder with cushioning device.

On Fig. 3 a conical, concentric gap formed between a conical cushioning device of length L_d and a cylindrical sleeve with diameter D_d is presented. The height of the gap at the beginning of the cone is h_1, and at the end the cushioning device ends with a cylindrical shape of length L_z and constant height of the gap h_k.

In vertical position of the cylinder with mass m at its rod end and initial speed v_0, assuming laminar flow and neglecting the compressibility of the oil, the differential equation describing the motion of the mass has the form [7]:

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

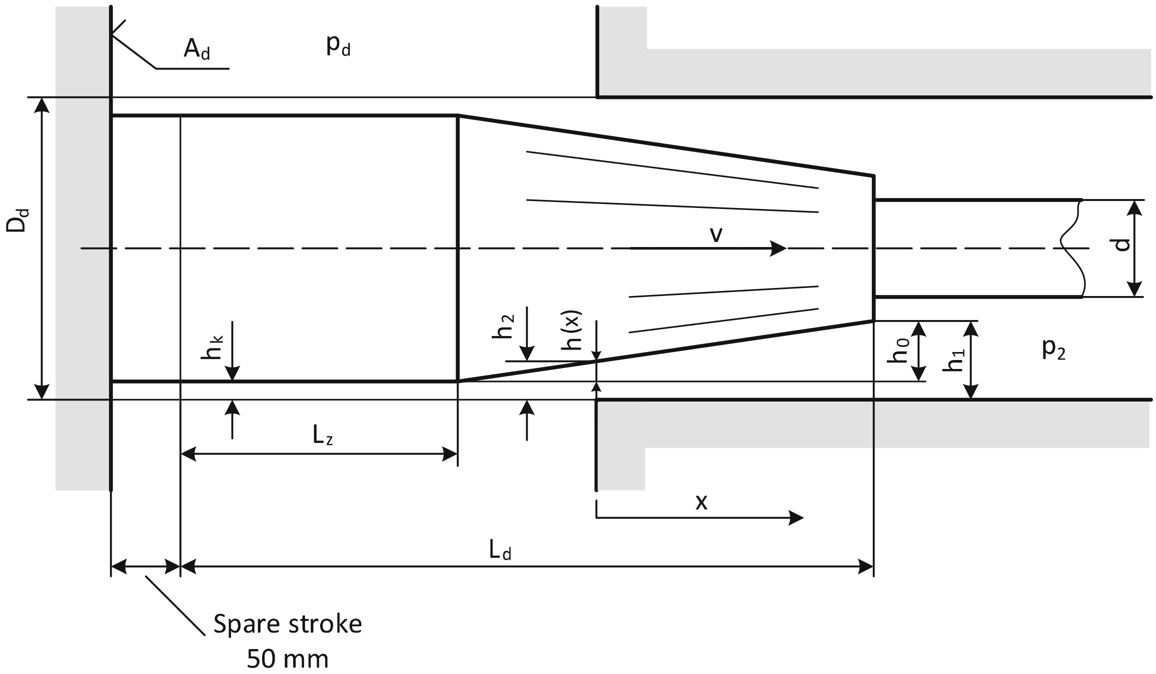

Fig. 3. Conical cushioning device with a cylindrical part and a cushioning sleeve.

or

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

where: m – the moving mass; v – the speed 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; $A_2 = \frac{(D_b^2 - d^2) \cdot \pi}{4}$ – the rod side area of the cylinder; $A_d = \frac{(D_b^2 - D_d^2) \cdot \pi}{4}$ – the rod side area without cushioning (Fig. 3); $A_{ds} = \frac{(D_d^2 - d^2) \cdot \pi}{4} = A_2 - A_d$ – the restriction area of the cushioning device.

When moving in steady - state mode, before the cushioning process begins, 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 [8]

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

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

Combining the Eqs. (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

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

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:

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

2.1 Mathematical Modelling of the Flow in the Conical Gap

Assuming incompressible laminar flow in the gap, equilibrium of forces on a fluid particle, Fig. 4-a, gives [9–11]:

$$\frac{dp_{d,2}}{dx} = \eta \cdot \frac{d^2v}{dz^2}$$

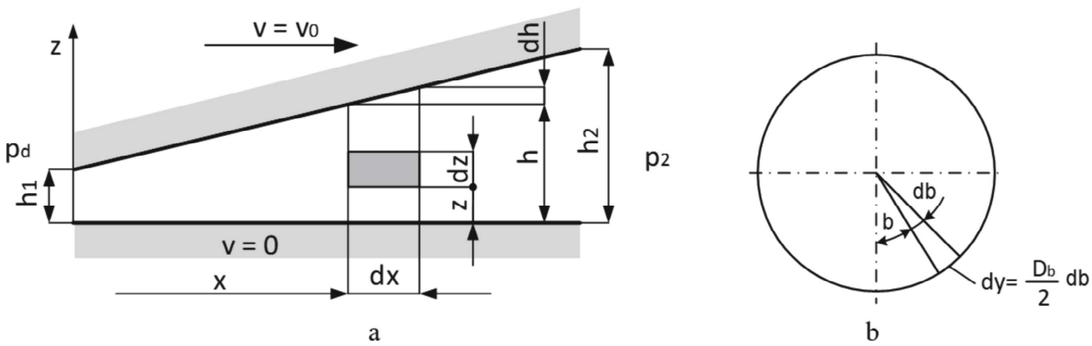

Fig. 4. Part of the conical gap.

Integrating twice, it gives

$$v = \frac{1}{2\eta} \cdot \frac{dp_{d,2}}{dx} \cdot z^2 + C_1 \cdot z + C_2 \tag{8}$$

Using the boundary conditions $v = v_0$ at $z = h$ and $v = 0$ at $z = 0$, the constants C_1 and C_2 can be solved:

$$C_1 = \frac{v_0}{h} - \frac{1}{2 \cdot \eta} \cdot \frac{dp_{d,2}}{dx} \cdot h$$

$$C_2 = 0$$

Substituting them in the Eq. (8), the velocity profile in the cone gap will be:

$$v = \frac{1}{2 \cdot \eta} \cdot \frac{dp_{d,2}}{dx} \cdot \left(z^2 - z \cdot h\right) + \frac{v_0 \cdot z}{h} \tag{9}$$

The flow through the conical gap will be:

$$\frac{dq}{dy} = \int_0^h v \cdot dz = -\frac{h^3}{12 \cdot \eta} \cdot \frac{dp_{d,2}}{dx} + \frac{v_0 \cdot h}{2} \tag{10}$$

Substituting $dy = \frac{D_d}{2} \cdot d\beta$ – Fig. 4-b, in (10) and integrating it around the circle form 0 to 2π, the pressure drop through the conical gap will be:

$$\frac{dp_{d,2}}{dx} = -\frac{12 \cdot \eta \cdot q}{\pi \cdot D_d \cdot h^3} + \frac{6 \cdot \eta \cdot v}{h^2} \tag{11}$$

The solution of the differential Eq. (11) for a laminar flow in a conical concentric gap with variable length x and $q = A_d \cdot v$ has the form:

$$p_{d,2} = \frac{6 \cdot \eta \cdot A_d \cdot v}{\pi \cdot D_d} \cdot \frac{x \cdot (h_1 + h_2)}{h_1^2 \cdot h_2^2} - \frac{6 \cdot \eta \cdot v \cdot x}{h_1 \cdot h_2} \tag{12}$$

where η – the dynamic viscosity.

The second component $\left(-\frac{6 \cdot \eta \cdot v \cdot x}{h_1 \cdot h_2}\right)$ in (14) is due to the entrained fluid from the surface of the cushioning device. It can be neglected compared to the first component, as it is about 100 times smaller. Thus the pressure drop $p_{d,2}$ depending on the speed v and the displacement x of the cushioning device, acquires the form

$$p_{d,2} = \frac{6 \cdot \eta \cdot A_d \cdot v}{\pi \cdot D_d} \cdot \frac{x \cdot (h_1 + h_2)}{h_1^2 \cdot h_2^2} \tag{13}$$

The pressure loss $p_{d,2}$ in the cylindrical concentric constant gap with height h_k in the second section of the cushioning will be

$$p_{d,2} = \frac{12 \cdot \eta \cdot A_d \cdot v \cdot [x - (L_d - L_z)]}{\pi \cdot D_d \cdot h_k^3} \tag{14}$$

The height of the gap h between the conical shape of the cushioning device and the cylindrical sleeve is not constant and depends on the displacement x of the cylinder and the slope of the cone $k = \frac{h_0}{L_d - L_z}$ and is determined by the expression

$$h = h_0 \cdot (1 - k \cdot x) \tag{15}$$

The heights in the conical gap of length x, in which the pressure drop $p_{d,2} = p_d - p_2$ is created, have the form

$$\begin{aligned} h_1 &= h_k + h_0 = h_0\left(1 + \frac{h_k}{h_0}\right) = h_0 \cdot a_1 \\ h_2 &= h_k + h(x) = h_0\left(1 + \frac{h_k}{h_0} - \frac{k}{h_0} \cdot x\right) = h_0 \cdot (a_1 - b \cdot x) \end{aligned} \tag{16}$$

where the constants are: $a_1 = 1 + \frac{h_k}{h_0}$; $b = \frac{k}{h_0} = \frac{1}{L_d - L_z}$.

Taking into account dependencies (16) and moving to the dimensionless coordinates: $V = \frac{v}{v_0}$ – the dimensionless velocity; $X = \frac{x}{L_d}$ – the dimensionless displacement; $\tau = \frac{t}{T}$ – the dimensionless time, where $T = \frac{L_d}{v_0}$ – the time constant; the cushion differential Eq. (7) takes the form

$$\frac{dV}{d\tau} = A \cdot \left(1 - V^2\right) - B \cdot V \cdot C(X) \tag{17}$$

where: $A = \frac{g \cdot T}{v_0} = \frac{g \cdot L_d}{v_0^2}$ and $B = \frac{6 \cdot \eta \cdot A_d^2 \cdot L_d^2}{\pi \cdot D_d \cdot h_0^3 \cdot v_0 \cdot m}$. $C(X)$ is a function of the displacement X and changes during the motion process, depending on the design of the cushioning device.

The scope of research in this work is a cushioning device with a conical shape with length $L_d - L_z$ and a cylindrical shape with length L_z, entering into sleeve with a length L_d equal to the sum of the lengths of the conical and cylindrical shapes.

a – First section when $x \leq L_d - L_z$.

When the conical part of the cushioning device $L_d - L_z$ enters the sleeve with length X, and the differential Eq. (17) will be

$$\frac{dV}{d\tau} = A \cdot \left(1 - V^2\right) - B \cdot V \cdot X \cdot \frac{(2 \cdot a_1 - b_1 \cdot X)}{a_1^2 \cdot (a_1 - b_1 \cdot X)^2} \tag{18}$$

where $a_1 = 1 + \frac{h_k}{h_0}$ and $b_1 = \frac{L_d}{L_d - L_z}$.

b – Second section when $L_d - L_z < x \leq L_d$.

When the cylindrical part of the cushioning device with the length L_z enters the sleeve and the differential Eq. (17) will be:

$$\frac{dV}{d\tau} = A \cdot \left(1 - V^2\right) - B \cdot V \cdot \frac{2}{(a_1 - 1)^2} \cdot \left[\frac{X - \frac{1}{b_1}}{a_1 - 1} + \frac{2 \cdot a_1 - 1}{2 \cdot a_1^2 \cdot b_1}\right] \tag{19}$$

3 Simulation of the Transient Process of the Cylinder

The solution of the differential Eqs. (18) and (19) represents the simulation of the transient process of the moving mass. The design parameters are the following: the moving mass $m = 30000kg$, the initial speed $v_0 = 0.0617m/s = 3.7m/min$, the oil kinematic viscosity $\nu = 46cSt$, the piston diameter $D_b = 0.25m$, the rod diameter $d = 0.11m$, the diameter of the cushioning device $D_d = 0.145m$, the length of the cushion $L_d = 0.35m$, the maximal height of the gap $h_0 = 0.001m = 1mm$, the length of the cylindrical section of the cushion $L_z = 0.12m = 120mm$, the height of the gap in the cylindrical section $h_k = 0.00012m = 0.12mm$.

For solving of the nonlinear differential equations, the adaptive Runge-Kutta method is used. This method based on the forth order Runge-Kutta method estimate the truncation error at each integration step and automatically adjust the time step size to keep the error within specified limits.

The solution of the differential Eqs. (18) and (19) and the transient process of the researched cylinder at the end of its stroke is presented at the diagrams on Figs. 5 and 6.

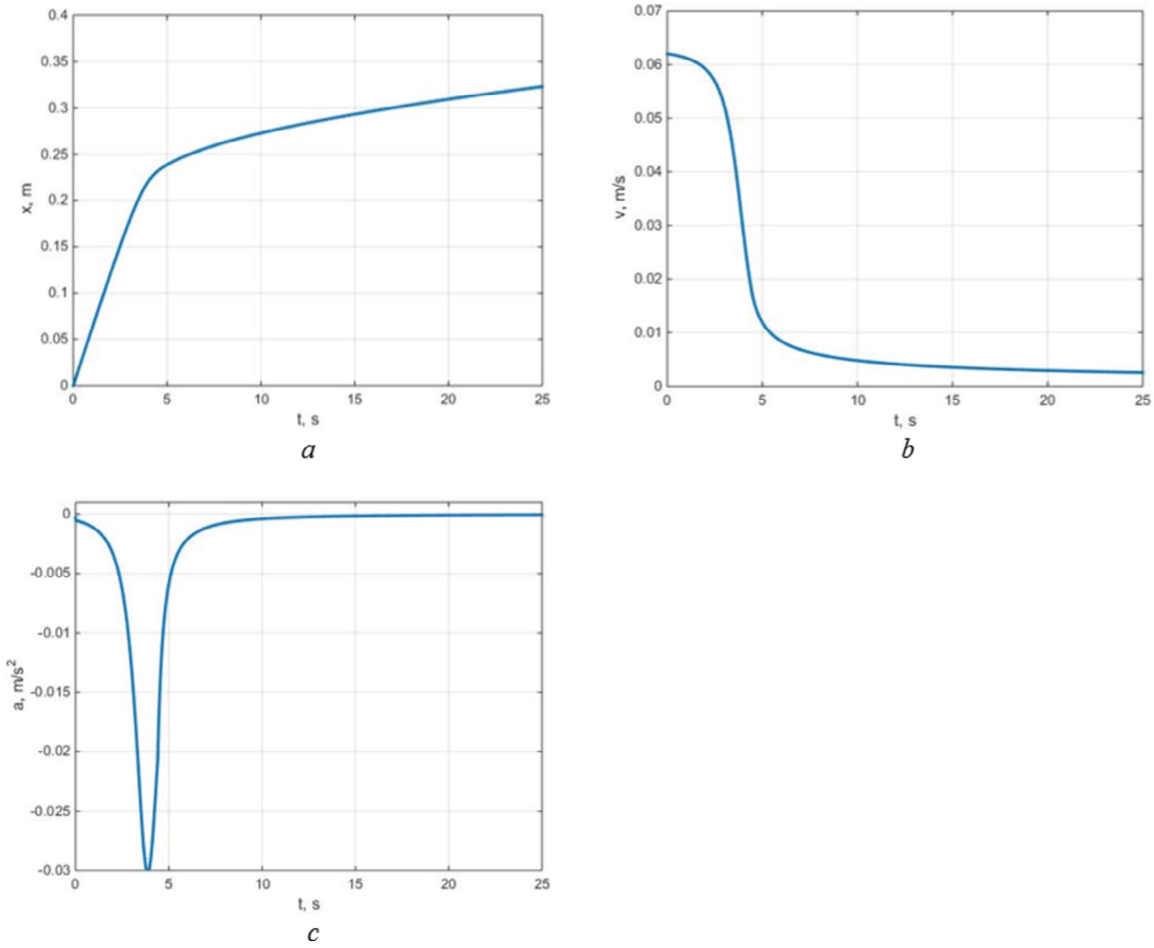

Fig. 5. Transient process in terms of time.

Maximum deceleration is $a_{max} = 0.03m/s^2$ – Fig. 5 – c. So, the hydraulic cylinder is loaded with additional inertial force of $F_a = m \cdot a_{max} = 900N$. This force can be neglected compared with the mass m. After 10 s of the beginning of the cushioning process, the speed is reduced to the value of $v = 0.005m/s$ which is 12.5 times lower than state-state speed v_0 and at the and of the cushioning process the speed is $v = 0.002m/s = 2mm/s$ which meets the requirements of the process quite satisfactorily.

Maximum deceleration occurs at a stroke of $0.22m$ – Fig. 6. At that moment the speed is $v = 0.032m/s$, it is half of the steady-state value v_0. At the distance of $0.23m$ when the whole cone shape is inside the sleeve and the cylindrical shape begins to enter the sleeve, the actual speed of the cylinder rod is around $v = 0.015m/s$ which is more than four times lower than the steady-state value of $v_0 = 0.0617m/s$.

In stationary mode, before beginning the cushioning, the pressure $p_2 = p_d$ is around $74bar$ – Fig. 7. It is a sufficient value to counterbalance the moving mass and to ensure it from falling uncontrollably. When the cushioning process begins, i.e. cone shape starts to enter into cylindrical sleeve, the pressure p_d is increasing to its final value of around $88bar$. The pressure p_2 is decreasing during the movement of the cushioning device into the sleeve and at the end of the stroke it is near zero.

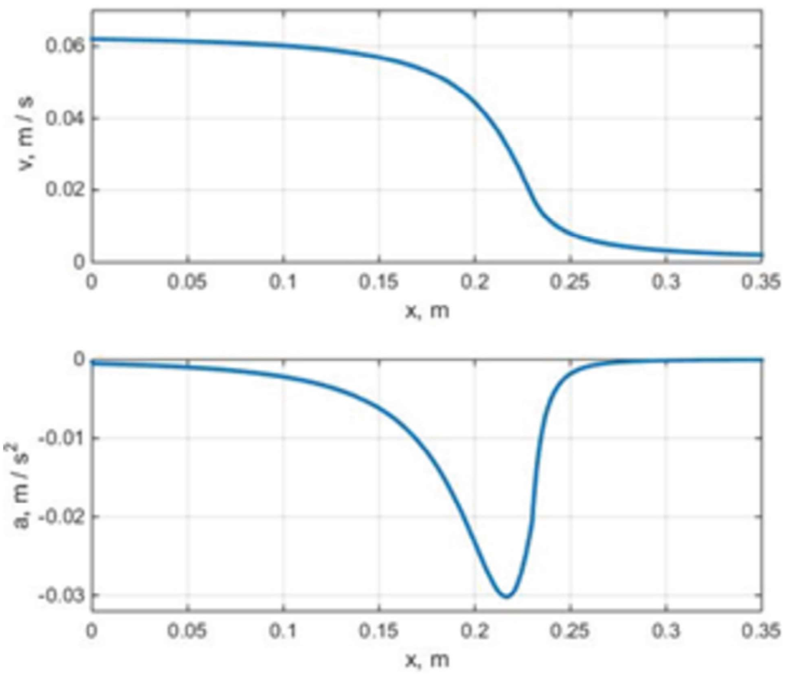

Fig. 6. Transient response in terms of cylinder stroke.

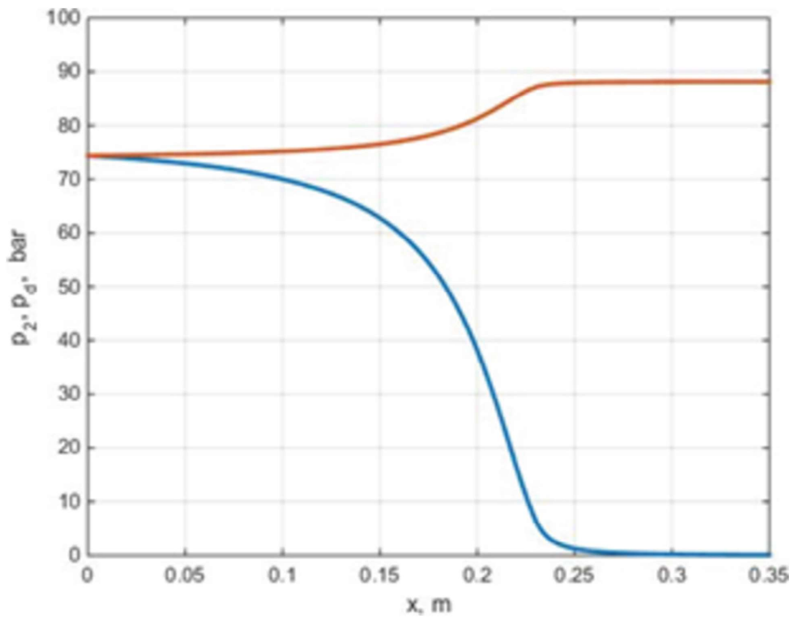

Fig. 7. Pressure distribution along the stroke.

4 Conclusion

Hydraulic cylinders are devices that very often move huge masses with high potential energy at high speeds. 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 is introduced in the design of the hydraulic cylinder to help smoothly stop at the end position.

In this work a mathematical model of a conical cushioning device in hydraulic cylinders is developed and simulated the transient process at the end of its stroke. The proposed cushioning device ensures smooth deceleration at the beginning and at the end of the transient process with relatively low additional overload on the hydraulic cylinder as a result of the deceleration – Fig. 5. This type of cushioning provides good cylinder speed profile in time without pressure peaks in the cylinder chambers – Figs. 6 and 7.

As it was presented in the diagrams above, including cushioning devices in hydraulic cylinders is essential when they move large masses with high speeds to absorb the high potential energy when they stop.

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