

# Transient Phenomena in Gradual Changes of Hydraulic Fluid Flow

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## ABSTRACT

Accelerations and decelerations, sometimes sudden stopping of moving masses are realized through hydrostatic drives during the working process of a machine. The consequence of the velocity changes are pressure changes inside the hydraulic system. In the case of underdamped motions of the actuators, the relevant part of the machine construction is exposed to oscillations. They should be avoided or, at least, minimized. More methods are available to do this. One of them is to use proportional valves, with which we can minimize the increases of pressure through the *ramp function* of these valves. But it is advisable to calculate the effect of it, which means calculating the reduced pressure amplitudes in advance with a relatively simple mathematical model and with the exactness which is acceptable for practical application. In addition, some final equations of this model and some results of measurements are presented in this paper.

## NOMENCLATURE

Q	hydraulic fluid flow – changing through time	$\text{m}^3 \text{s}^{-1}$
$Q_0$	constant flow – before or after transient phenomena	$\text{m}^3 \text{s}^{-1}$
t	time – time of transient phenomena	s

H	resistance to acceleration /Pro77/	$\text{N s}^2 \text{m}^{-5}$
D	resistance to deformation / Pro77/	$\text{N m}^{-5}$
K	coefficient of the resistance to motion	$\text{s}^{-1}$
p	pressure	$\text{N m}^{-2}$
$\Delta p$	pressure change	$\text{N m}^{-2}$
$\omega$	angular motion frequency of underdamped oscillation	$\text{s}^{-1}$
J	moment of inertia	$\text{kg m}^2$
$t_0$	time of restricting the fluid flow (deceleration time)	s
A	functional area of linear actuator	$\text{m}^2$
q	capacity of hydraulic motor per one rotation	$\text{m}^3/\text{rot.}$

## 1 INTRODUCTION

Through hydrostatic drives, linear or rotary motions are accelerated and decelerated during the working process of a machine. Motions are realized by actuators, thus by cylinders, hydraulic motors or limited angle rotary actuators. Hydraulic fluid flow is accelerated and decelerated through tubes and/or bores into these hydraulic actuators. The consequence of the flow changes in certain, mostly short time periods, are pressure changes inside the hydraulic system, or at least inside a part of the system. When hydraulic actuators move heavy masses the increases of pressure can be considerable. The increase of pressure, caused by decelerating masses, can be significantly higher than that caused by decelerating the fluid flow. Yet this is normally not problematic when the motion of the actuator is overdamped. But in the case of underdamped motions of the actuators, the relevant part of the machine construction is exposed to vibrations. These vibrations are the consequence of the flow and pressure vibrations inside the actuator and at least inside part of the hydraulic system. In numerous cases fluid flow, and thus the actuator, is stopped momentarily by valve closing. Not only the mass of fluid flow, but also the masses connected to actuators, are stopped. The masses moving by actuators are often more critical for pressure changes than mass of fluid flow. The pressure increases at momentarily stopping of the fluid flow and consequently the actuator are higher than in the cases of gradual stopping. Pressure vibrations can be overdamped, critically

damped or underdamped. The latter are the most problematic ones. Pressure increases – amplitudes of up to some hundred bars can occur.

Proportional valves are very often used to control accelerated and decelerated movements of hydraulic actuators and moving masses, and further to minimize the increases of pressure through the *ramp function* of these valves. The setting of the *ramp slope* or *duration of the ramp* is of critical importance in using these valves.

A mathematical model was derived for the instantaneous and for gradual stopping and accelerating of fluid flow. Some final equations of this model are presented here. They enable calculation of the fluid flow and pressure changes at the time of the accelerating and decelerating movement of the actuator for either instantaneous or gradual stopping or accelerating of the fluid flow.

A comparison between the increases of pressure occurring at instantaneous and at gradual valve opening or closing of duration up to some seconds is presented and evaluated in the paper. Furthermore, a comparison between the results of laboratory tests and the results obtained using mathematical models is described and discussed. The question how pressure changes in the time of the *ramp function* of a proportional valve is at least partially answered in this paper. No special knowledge is necessary to use the relatively simple equations, though they do not look so simple at first glance.

The effect of reducing pressure amplitudes by the use of flexible hoses compared to metallic tubes is considered in the paper. Comparison between results of test measurements and the results of mathematical models is done in this paper.

## **2 SOME RECAPITULATIONS FROM MATHEMATICAL MODEL**

Change in the rate of motion of the energy carrying hydraulic fluid and within it of the actuator, within a given time interval, involves acceleration or deceleration of that motion and of the moving masses depending to that motion through hydraulic fluid. Considering resistance to acceleration (H), resistance to deformation (D) and resistance to motion (R) we can write the following equation /Pro68/, /Pez84/:

$$H \cdot \frac{d^2 Q}{dt^2} + H \cdot \kappa \cdot \frac{dQ}{dt} + D \cdot Q = 0 \quad (1)$$

In the equation (1)  $Q$  is the fluid flow (changing through time) and  $\kappa$  represents the coefficient of the resistance to motion.

Final applicable equations for flow  $Q$  and pressure changes  $\Delta p$  through time at instant or gradual changes of fluid and actuators velocity can be derived from basic equation (1). These equations are identical or even the same for accelerations and decelerations. Yet in practice, for the machines using hydrostatic drives, decelerations are normally more problematic than accelerations. Therefore our experiments were done oriented mostly on decelerations. Especially instant stopping of movements can cause oscillations, through which heavy damages of the machine construction can occur because of the additional mass inertia forces. When the resistance to motion is low enough, the actuator with belonging masses has oscillatory damped movements. Deriving equation (1), considering conditions for underdamped oscillations, and furthermore considering, that the valve closing stopping the fluid flow is so quick, that it could be considered as instant, we obtain the following equation for pressure variations through time /Pez84/:

$$\Delta p = e^{-\frac{1}{2} \cdot \kappa \cdot t} \cdot Q_0 \cdot H \cdot \omega \cdot \sin(\omega \cdot t) \cdot \left[ 1 + \left( \frac{\kappa}{2 \cdot \omega} \right)^2 \right] \quad (2)$$

In the equation (2)  $Q_0$  represents constant flow at actuators movement with constant velocity before deceleration or after accelerations period. Yet it must be mentioned that some simplifications were used deriving equation (1). The coefficient of friction  $\kappa$  and some other values were considered as constant during deceleration, though they are really not constant. Otherwise mathematically the derivation would be far too complicated. We wanted to come up with not necessarily a very exact, but more importantly, a usable mathematical model.

Acceleration can be positive or negative, thus acceleration or deceleration. Resistance to acceleration  $H$  is for linear motion defined by following equation /Pro68/:

$$H_m = \frac{m}{A^2} \quad (3)$$

and for rotary motion by equation /Pro68/:

$$H_{HM} = \left( \frac{2 \cdot \pi}{q} \right)^2 \cdot J \quad (4)$$

Angular motion frequency  $\omega$  is defined by equation /Pro68/, /Pez84/:

$$\omega = \left( \frac{D}{H} - \frac{\kappa^2}{4} \right)^{0,5} \quad (5)$$

where resistance to deformation  $D$ , considering only the fluid, without elasticity of flow transmitters (tubes, pressure hoses, cylinders), is defined /Pro 68/:

$$D_f = \frac{K}{V_f} \quad (6)$$

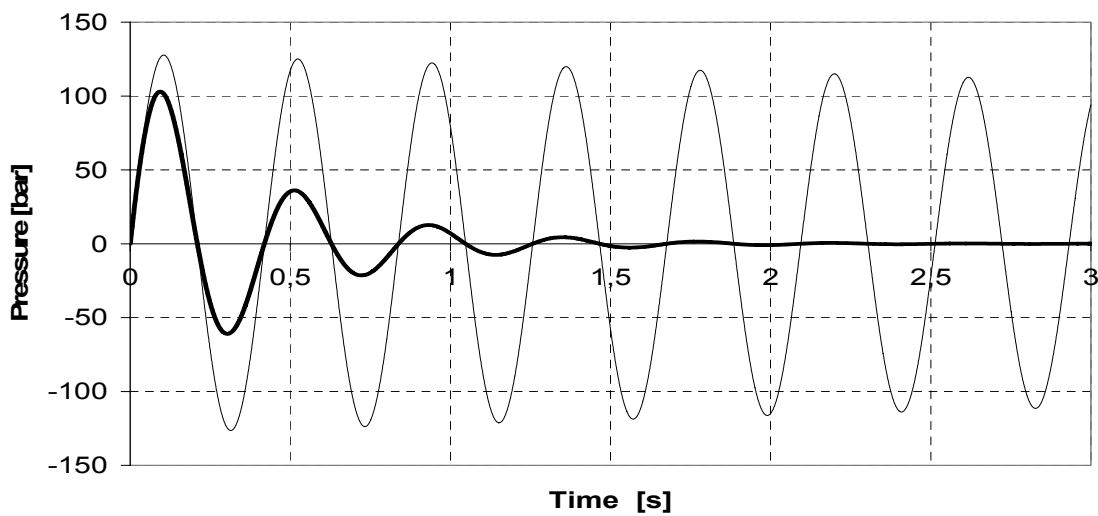
Sometimes fluid flow and to it depending actuators are decelerated in such time interval that it can not be considered as instant stopping. Thus we have gradual change of movement. Such cases occur at deceleration of masses through stroke end cushioning of cylinders and when using proportional control valves with ramp function. For the deceleration (or acceleration) time interval  $t_0$ , what could be time of closing the flow gap inside a proportional control valve, we obtain, through derivation of equation (1), the following final equation (7) /Pez84/ which expresses pressure

variations through time  $t_0$ . Pressure variations beyond  $t_0$  should be calculated using equation (2).

$$\Delta p = e^{-\frac{1}{2} \cdot \kappa \cdot t} \cdot \frac{Q_0 \cdot H}{1 - e^{-\kappa \cdot t_0}} \cdot \left\{ \omega \cdot \sin(\omega \cdot t) \cdot \left[ 1 - e^{-\kappa \cdot t} + \left( \frac{\kappa}{2 \cdot \omega} \right)^2 \cdot (1 - 3 \cdot e^{-\kappa \cdot t}) \right] + e^{-\frac{1}{2} \cdot \kappa \cdot t} \cdot \left( 1 - e^{-\frac{1}{2} \cdot \kappa \cdot t} \cdot \cos(\omega \cdot t) \right) \right\} \quad (7)$$

The derivation of this equation and the purpose of it was carried out at simplifications described already at equation (2). From the process of derivation of equation (7) comes, that it is applicable only for times  $t_0$  which are greater than  $\pi/2\omega$  what is one quarter of oscillation period.

**Figures 1 and 2** show pressure changes through time, considering only the changes caused by decelerating the actuator with a moving mass, without considering the deceleration of fluid flow. **Figure 1** shows pressure changes at instantaneous stopping of actuator for almost undamped and for underdamped oscillations. **Figure 2** shows pressure changes for almost undamped and for underdamped oscillations at gradual stopping of actuator with closing time interval  $t_0$ .



*Figure 1: Pressure change through time at instantaneous stopping of a moving mass with the coefficient of the resistance to motion  $\kappa = 0.1$  and  $\kappa = 5$ .*

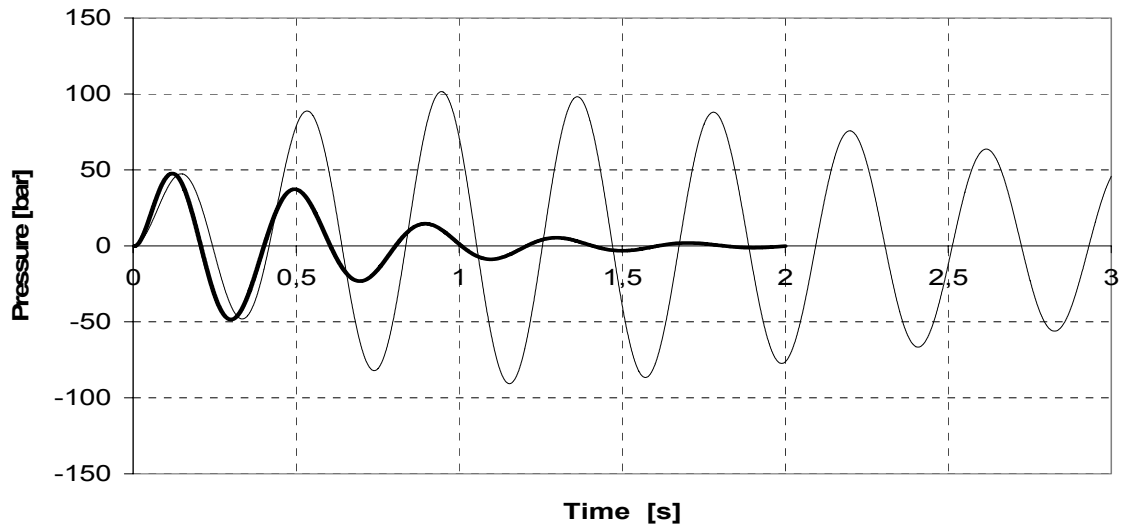


Figure 2: Pressure change through time at gradual stopping of a moving mass through decelerating time  $t_0 = 1.6$  s and with the coefficient of the resistance to motion  $\kappa = 0.1$  and  $\kappa = 5$ .

### 3 EXPERIMENTAL PART

A relatively simple test rig was constructed to carry out some adequate instantaneous and gradual stopping of the hydraulic motor's fluid flow. The test rig is shown in *Figure 3*. An additional mass was fixed on the axle of hydraulic motor. Four sensors for pressure (S1, S2, S3 and S4) were connected to the system. We have measured also the rotations of the hydraulic motor and the input fluid flow.

For the first example we hold energized the solenoid *a* (*Fig. 3*). Having constant rotation at pressures about 110 bar ( $p_3 = S_3$ ) and 120 bar ( $p_4 = S_4$ ) (see *Figure 4*) we deenergize solenoid *a* in a very short time (about 7 ms – measured) and that way we block the valve connections *A* and *B*. Also in a short time pressure in left line (flexible hose of 10 m) increases and in right line (also flexible hose of 10 m) decreases. The measured pressure change for both lines is shown in *Figure 4*. Sensors S1 and S2 were used only as control sensors.

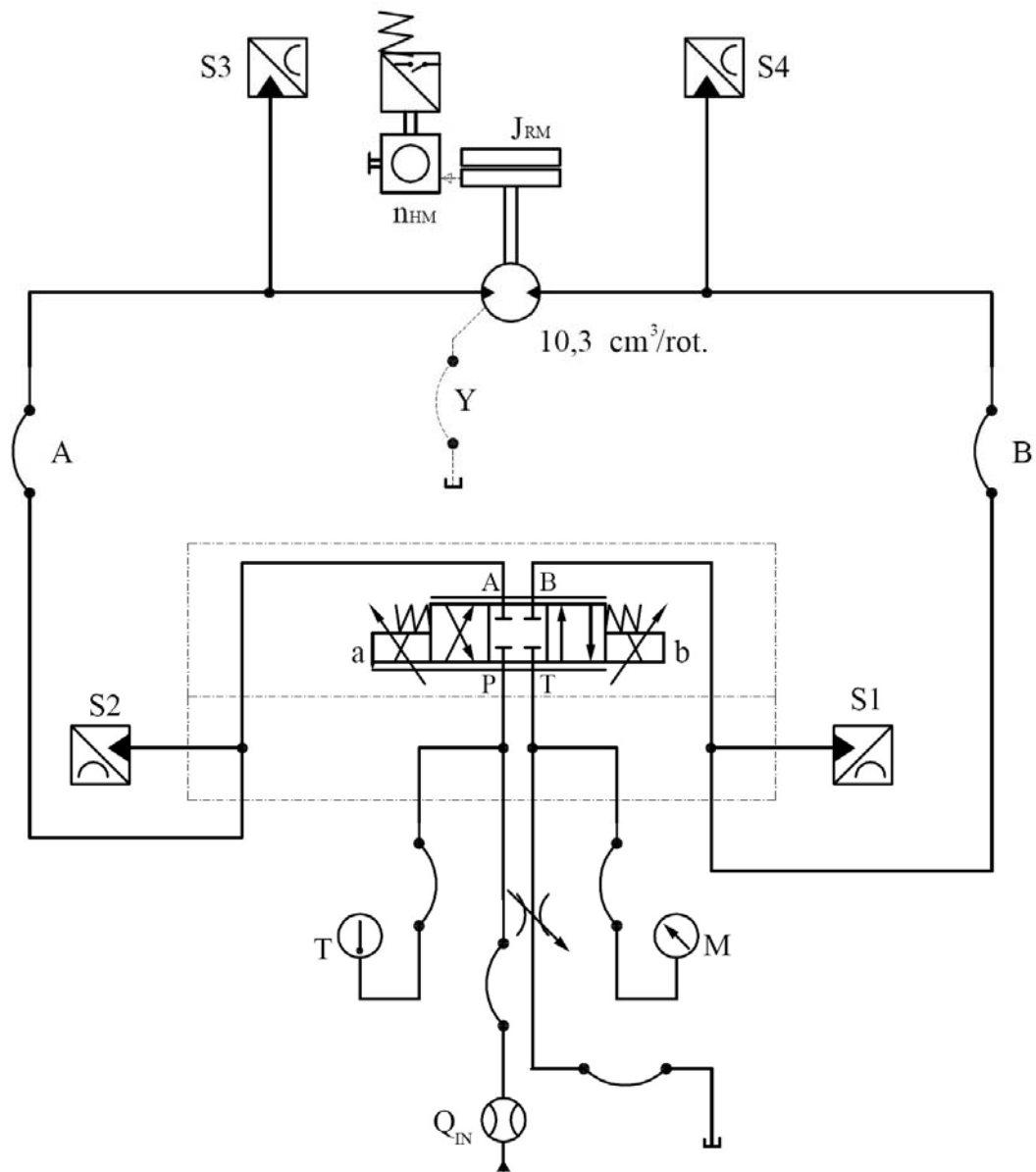


Figure 3: Experimental part of the hydraulic system for pressure changes simulation

Curve *a* from Figure 4 is shown in Figure 5 at greater time scale. Furthermore Figure 6 shows the curve defining pressure changes through time for the same parameters, but obtained through equation (7). We must mention, that elasticity of used pressure flexible hoses was measured and the results of these measurements were considered in calculation for resistance to deformation. Yet it can not be shown here as the extent of paper is limited.



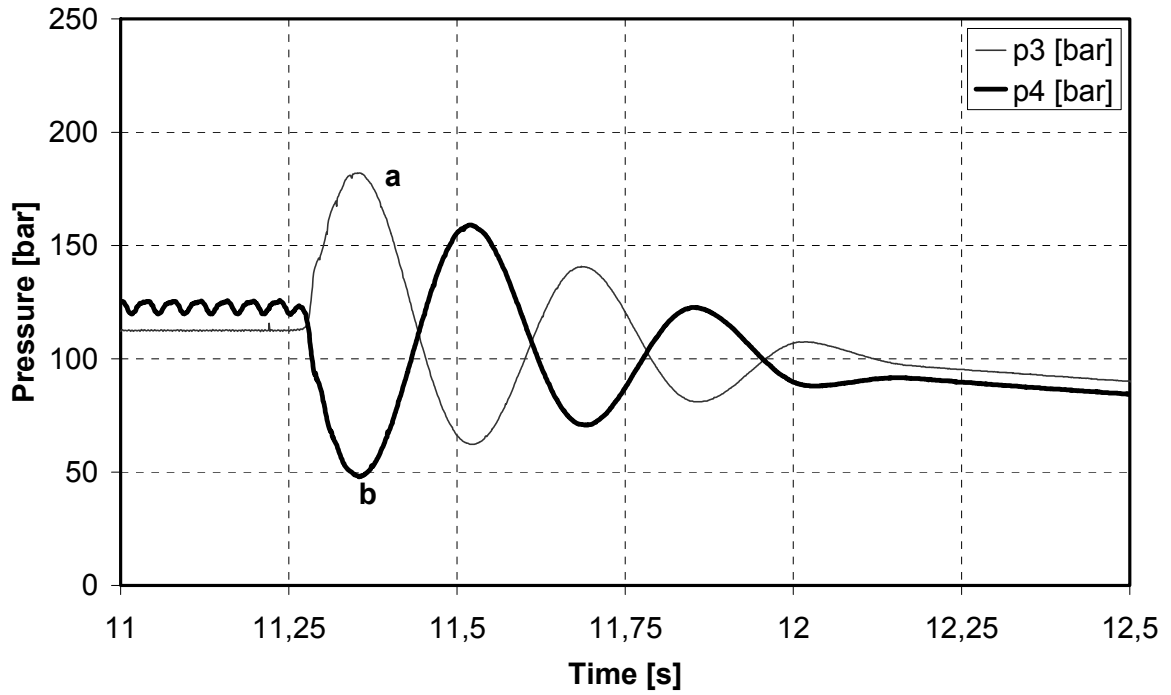


Figure 4: Pressure changes measured at instant stopping the fluid flow from and to the hydraulic motor; curve a is from sensor S3 and curve b from S4 (see Fig. 3)

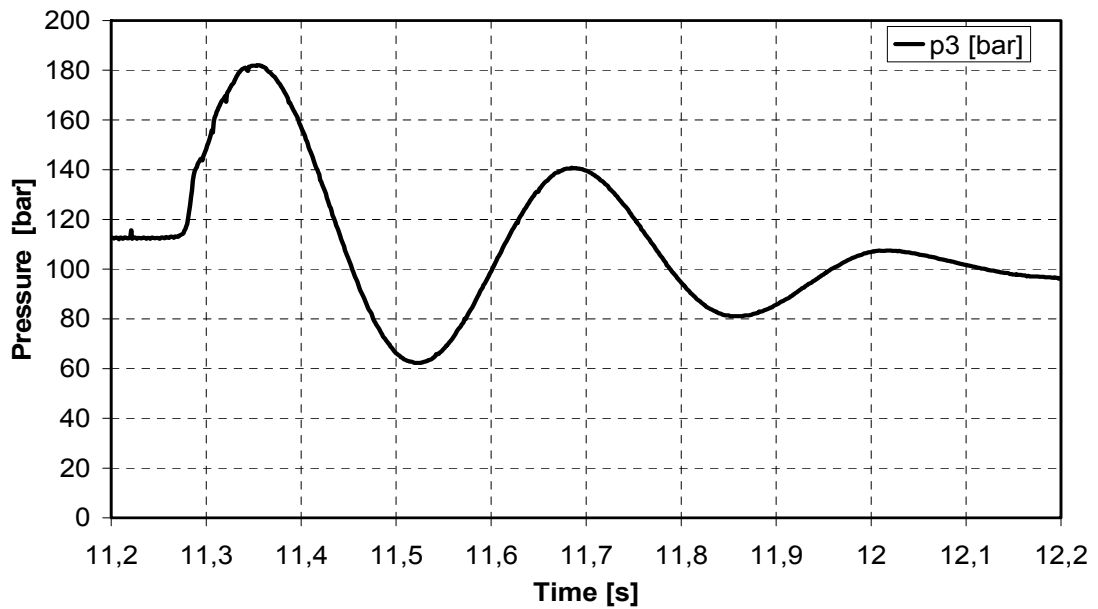


Figure 5: Curve a from Figure 4 in extended time scale

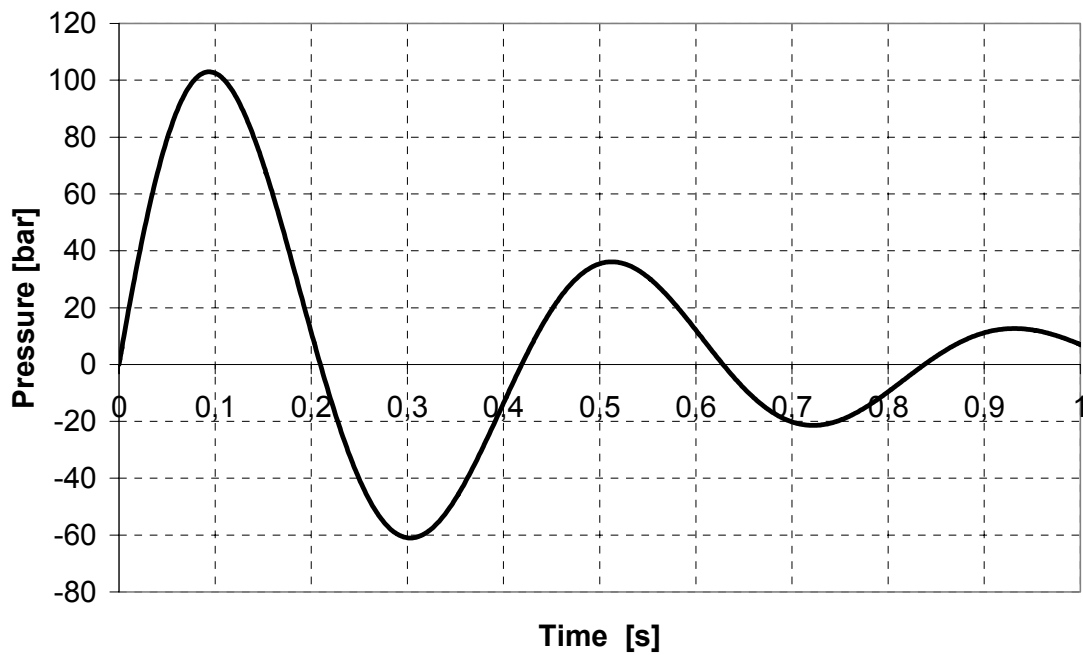


Figure 6: Curve obtained using mathematical model (equation (7)) for the same parameters which are valid for measurement shown in Figure 5

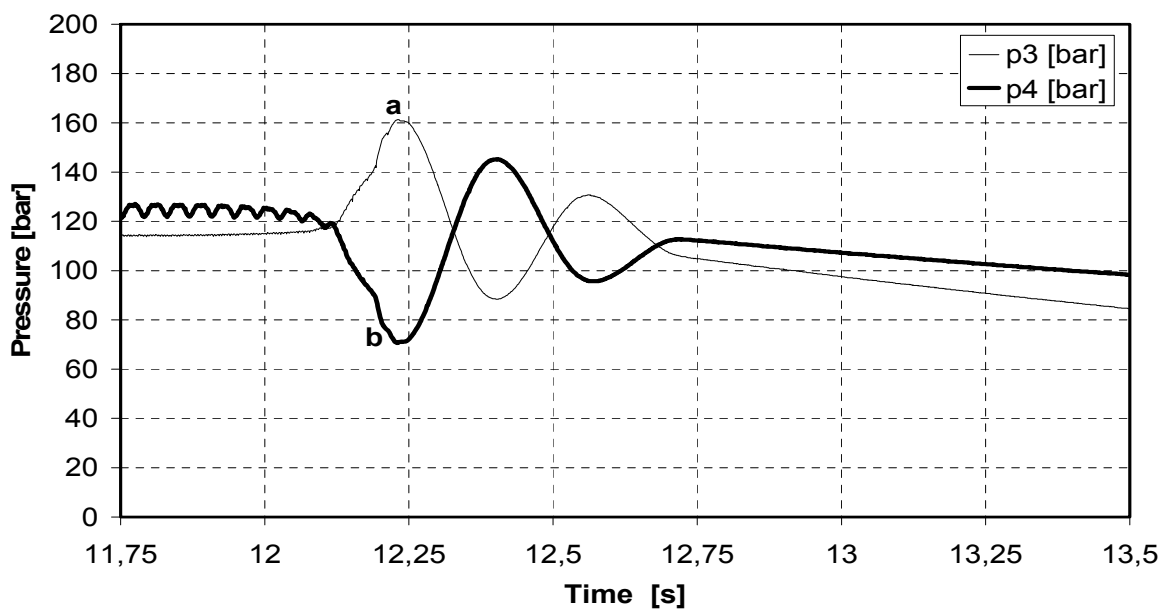


Figure 7: Gradual stopping of fluid flow (solenoid a deenergized – see Figure 3) through ramp function of the proportional directional valve

Figure 7 shows the result of measurement for the same parameters as are valid for the results shown in Figure 4 with the exception for the time of solenoid deenergizing. In this case the ramp time was set at  $t_0 = 1,6$  seconds. So there was gradual stopping of fluid flow and decelerating hydraulic motor and rotational mass.

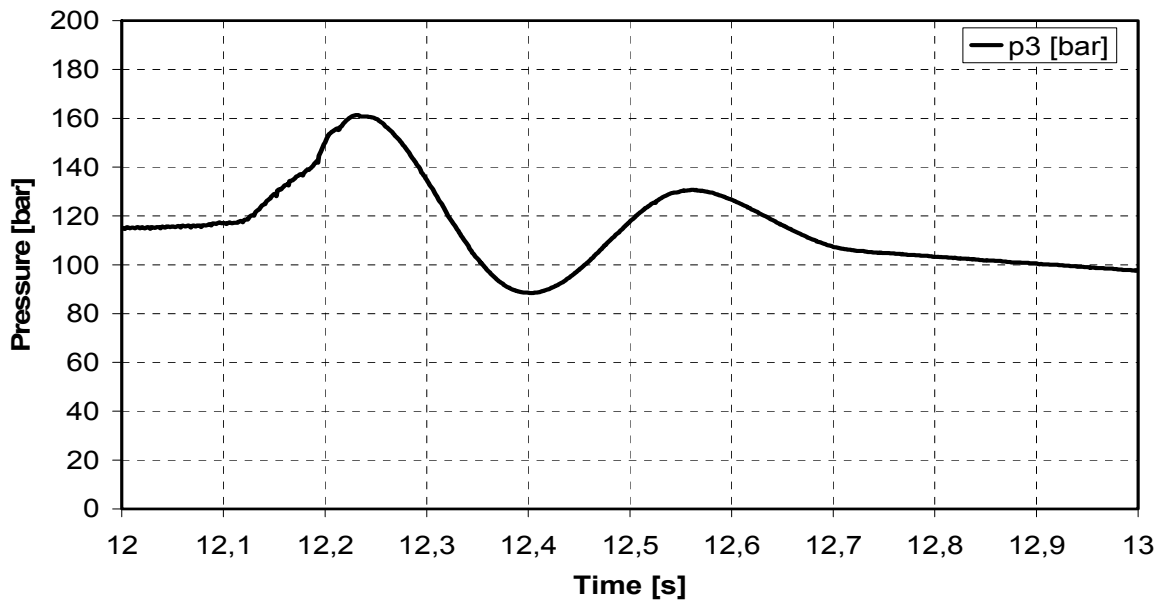


Figure 8: Curve a from Figure 7- pressure changes at gradual stopping of fluid flow (solenoid a deenergized – see Figure 3) through ramp function of the proportional directional valve

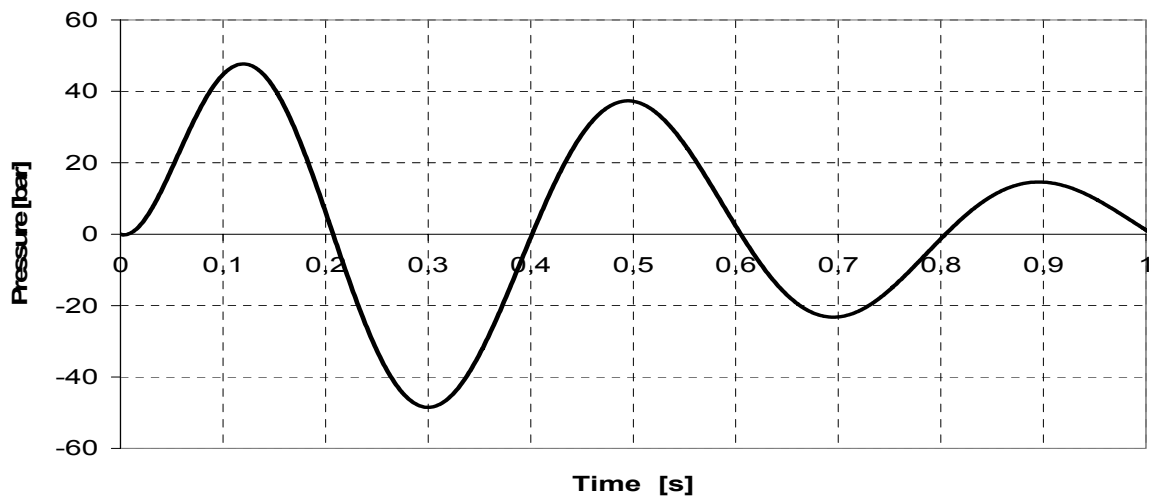


Figure 9: Curve obtained using mathematical model (equation (7)) for the same parameters which are valid for measurement shown in Figure 8

## CONCLUSION

Our measurements were carried out at poor knowledge of the resistance to motion, what makes rather great differences between the results obtained by measurements and others obtained by mathematical model. The derivation of it was simplified, what we pay through exactness. Yet the mathematical model is easy to apply. We consider that the results obtained through mathematical model are usable for practical application, especially for advance calculations when designing new hydrostatic drives moving heavy masses.

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