A model for active squeeze

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1 Introduction

This short report describes a simple model for the active squeeze problem. Although the author has little to do with the problem, so no details are known to him, it is possible to come up with a simple model.

A pipelay tower has a tensioner, to keep the pipe from falling to the seabed. The tensioner has separate tracks: One needs to position the pipe, the other one squeezes the pipe onto the position track. During active squeeze the active squeeze cylinders are controlled to keep a constant force on the pipe, while the position track keeps the pipe in its current position. In some situations the position cylinders are adapted and the squeeze cylinders must follow to keep the constant force on the pipe, which is the case for the simulation.

First the model and underlaying laws of physics are described, then how it is simulated and the results will be shown at the end of this article.

2 The Model

The model used during this simulation is based on both mechanics (second order differential equation) and the law of conservation of mass (first order differential equation). A schematic representation of this simple model is given in figure 1. For simplification, the three active squeeze cylinders are modeled as one, which is reasonable for this application, because only the hydraulics are taken into account. The external force is the force generated by the positioning track cylinders.

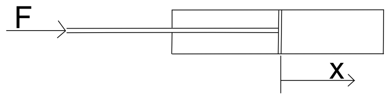

Figure 1: The cylinder with an external force

2.1 The mechanical model

The displacement of the active squeeze cylinder is simulated using Newton's law:

$$\sum F = m\ddot{x} \tag{1}$$

The piston and rod of the cylinder, together with the pipe and the piston and rod of the position cylinders, are taken as the mass which is accelerated. The forces acting on the squeeze cylinder are the force delivered by the position track cylinders and by the pressure at the rod-side of the squeeze cylinders, which is constant during measurements, so this pressure is not taken into account, but are part of the force on the cylinder. Furthermore damping is included. This leads to a differential equation as shown in equation

$$F_{in} - d\dot{x} - pA = m\ddot{x} \tag{2}$$

The force on the cylinder is also dependent on the displacement, due to the compression/decompression of the position cylinder. This is modeled as a spring, using $\Delta p = \int \frac{dp}{dt} dt$. By using the conservation of mass and bulk modulus this pressure is dependent on position. First the conservation of mass is rewritten.

$$\dot{m_{in}} = \frac{d}{dt}(\rho V) + \dot{m_{out}} \tag{3}$$

$$0 = \rho\frac{dV}{dt} + V\frac{d\rho}{dt} + 0 \tag{4}$$

$$\frac{dV}{dt} + \frac{V}{\rho}\frac{d\rho}{dt} = 0 \tag{5}$$

Now the bulk modulus is used:

$$\beta = -V\frac{dP}{dV} = -\rho\frac{dp}{d\rho} \tag{6}$$

$$\frac{d\rho}{\rho} = \frac{dp}{d\beta} \tag{7}$$

Using this in the conservation of mass gives:

$$\frac{dV}{dt} + \frac{V}{\beta}\frac{dp}{dt} = 0 \tag{8}$$

$$\frac{dp}{dt} = -\frac{\beta}{V}\frac{dV}{dt} \tag{9}$$

$$\frac{dp}{dt} = -\frac{\beta}{V}A\dot{x} \tag{10}$$

$$\tag{11}$$

Integrating this result gives a change in pressure caused by the movement of the pistons, acting as a spring on the mechanical system.

$$F_{in} = F_x - \int \frac{dp}{dt}dtA \tag{12}$$

$$F_{in} = F_x - \frac{\beta}{V}Ax \tag{13}$$

Now the mechanical system can be described as follows:

$$(F_x - \frac{\beta}{V}Ax) - d\dot{x} - pA = m\ddot{x} \tag{14}$$

2.2 The conservation of mass

As already shown in the previous subsection, the conservation of mass is used to calculate the pressure rise due to piston motion. In the previous chapter, especially from equation 3 up to equation 10, the conservation of mass is used when there no flow in or out of the cylinder, which is the case when the position cylinder is not moved. For the active squeeze cylinder, which are controlled and there is flow in and out of the cylinder, the flow needs to be included. This flow is used to control the cylinder. This flow is used in this model for the massflow in, but it can be negative. The massflow out of the cylinder is zero in this case.

$$\dot{m_{in}} = \frac{d}{dt}(\rho V) + \dot{m_{out}} \tag{15}$$

$$Q_{in} = \frac{dV}{dt} + \frac{V}{\rho}\frac{d\rho}{dt} \tag{16}$$

$$\frac{dp}{dt} = \frac{\beta}{V}(Q_{in} - A\dot{x}) \tag{17}$$

With these results and the results of section 2.1, the system can be simulated.

3 Simulation

For a simulation, the two differential equations need to be combined in three first order differential equations.

3.1 F_{in} becomes constant

In this simulation, the force on the cylinder becomes constant, because the position cylinder is moved from $t = 5[s]$ till $t = 12[s]$. This constant force replaces $F_x - \frac{\beta}{V}Ax_2)$ and is $1901837[N]$. This drop in force is equal to the pressure drops in the position cylinders during a measurement (bottom side pressure increases $15[bar]$, the rod side pressure increases $25[bar]$, meaning that the total force the position cylinder delivers drops approximately $21913[N]$).

$$\underline{x} = \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} = \begin{bmatrix} \dot{x} \\ x \\ p \end{bmatrix} \tag{18}$$

$$\frac{d\underline{x}}{dt} = \begin{bmatrix} \frac{(F_x - \frac{\beta}{V}A^2x_2) - dx_1 - x_3A}{m} \\ x_1 \\ \frac{\beta}{V}(Q_{in} - Ax_1) \end{bmatrix} \tag{19}$$

The flow in the cylinder is controlled by a proportional valve:

$$Q_{in} = k\left(\frac{F_{setpoint} - F_{in}}{A} - x_3\right) \tag{20}$$

For the other parameters the following is used:

A	0.0855	$[m^2]$
d	500000	$\left[\frac{kg}{s}\right]$
F_x	1923750	$[N]$
k	$\frac{2.19*10^{-8}}{60}$	$\left[\frac{m^5}{sN}\right]$
m	5000	$[kg]$
β	$1.43*10^9$	$\left[\frac{N}{m^2}\right]$
V	0.27	$[m^3]$
$F_{setpoint}$	3847500	$[N]$

The result is shown in figure 2.

The pressure drop is about 3 $[bar]$. From $t = 5[s]$ till $t = 12[s]$ the pressure is constant. All the flow to the cylinder is used to move the

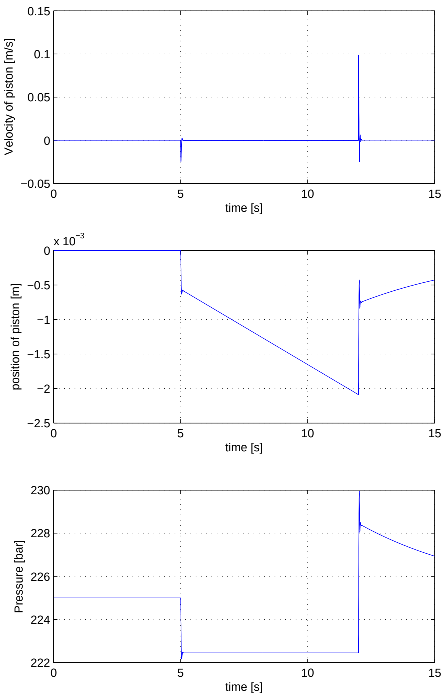

Figure 2: Simulation results for case 1

cylinder even further, because the pressure is pendent on the external force. When the fc from the bottom side pressure becomes largeı the external force, the cylinder starts to mov

3.2 F_{in} drops, but is not constar

In this section the simulation still uses $-\frac{\beta}{V}Ax_2$ term when F_x drops, so the sque cylinder can compress the bottom side of the sition cylinder. The simulation time is now ex tended to $60[s]$, the drop in force F_x is now from $t = 5[s]$. In figure 3 the result is shown.

In figure 3 the higher setpoint is seen, which is due to the fact that the controller wants to keep a constant force on the cylinder, so $F_{setpoint} = F_{in} + pA$. The force F_{in} drops, so the pressure p must rise to match the setpoint force. In reality the pressure at the position cylinder will become larger again, stopping the motion. The initial pressure drop is now $1.5[bar]$.

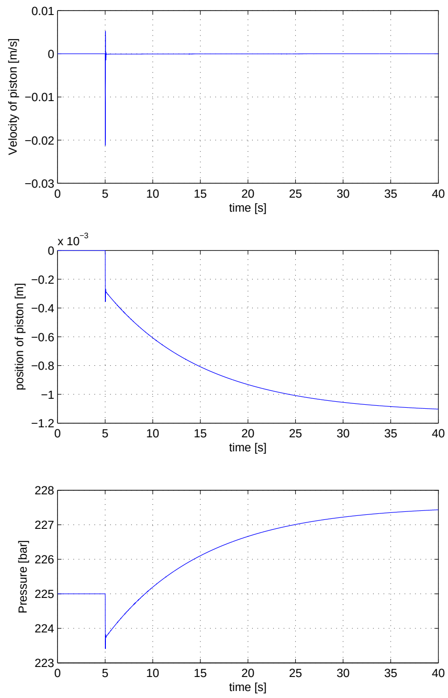

Figure 3: Simulation results for case 2