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Challenges in modelling and simulating hydraulic servovalve

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Abstract Modelling accurate response from hydraulic system in practice is difficult, especially establishing right mathematical model and getting all the parameters for certain hydraulic component. In this paper we mentioned two different different approaches for modelling hydraulic components, specifically Moog G761-series, made by company Moog. Approaches mentioned in this paper are using classical Matlab Simulink environment to show example of the first order model. Main focus is on using Matlab Simulink Simscape, which already includes some of the basic hydraulic components. We discussed problems and challenges when obtaining and simulating real components. We developed our own Simulink Simscape model for simulating Moog G761 servovalve. Simulation results were compared with the datasheet values from Catalog.

Keywords: • servovalve • modelling • Matlab • Simulink • Simscape •

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1 Introduction

Modelling of hydraulic system is an option, when it comes to simulating real physical components or system as a whole. It can be done either by setting up right mathematical equations for each system or subsystem. There are many hydraulic simulating software available, some known examples are: FluidSIM – Festo, Simulink (Matlab based), DSHplus simulation software, Visual Solutions, HydroCAD, Flowmaster 2, HyPneu, Hopsan, ACSL simulation software, Dynhax™ and many more. Specialised software for simulating hydraulic systems is easier to use, but there is still problem obtaining real process parameters, that must be set in simulation program. So simulating results should always be checked by conducting the experiment.

In this paper we focused on two approaches, first one being more "traditional" approach using Matlab Simulink, or more modern approach using libraries in Matlab Simulink Simscape environment, where some of the basic components are already developed and we can model more complex system/component behaviour by use of the basic Simscape library components. We focused on the Simscape model, because Li Lingjun [1] has already developed Classical Simulink model for hydraulic servovalve. Both approaches can provide similar results. While Simulink Simscape approach may be seen as more simple or user friendly, both approaches require degree of understanding the hydraulic component behaviour and some deepening in the field of establishing proper mathematical equations, especially in classical Matlab Simulink environment, where for each hydraulic component has its own or multiple equations are taken into consideration.

2 Servovalves

First appearance of electrohydraulic servovalve made appearance in the latter 1940s, developed specifically for aerospace, because fast response servo control was in need. At that time there wasn't much difference between electrohydraulic servo system and electrical servo, because electrohydraulic servo system lacked an element which could rapidly tranlate electrical signal into hydraulic flow. First series of servovales were actuated by small electric servomotor, which in comparison to magnet torque motors had large time constant, resulting in limited system performance [2].

W. C. Moog Jr. has developed in 1950 the two-stage servovalve using pilot stage without friction. Second-stage spool in a three way mode was driven by a flapper and nozzle variable orifice in conjuction with a fixed orifice. Such servovalve construction provided reduction in valve treshold, high dynamic response because of the lower mass of the first-stage parts. It was possible to use servovalves in high gain position. Hydraulic servovalves have developed and improvements were made both in reliability and life expectancy from the first introduction in 1951 [3]. One example is switching from steel ball to carbide ball on the feedback mechanism, which proved to not have any signs of wear after 1 billion cycles.

2.1 Behaviour and types of electrohydraulic servovalves

Similar to normal servovalves, also servovalves can come in different designs or stages. In single-stage servovalve consist of a torque motor which is directly attached to and positions a spool valve. The problem with single-stage servovalves is limited power capability, this limits the flow capacity of single-sate servovalve and can influence stability in certain applications [2].

Two-stage servovalves in comparison to single-stage servovalves have hydraulic preamplifier, which substantially multiplies the force output of the torque motor, leading to much more efficiency when overcoming flow forces, stiction forces and forces resulting from acceleration or vibration. Two-stage servovalves can be classified in three types, depending of feedback used which are spool position, load pressure and load flow feedback. Out of those, position feedback two-stage servovalves are the most common and even those are classified into three groups of construction, those are direct feedback, force feedback and spring centered spool. Direct position feedback is constructed that main spool follows the first stage valve in a one-to-one relation. In force feedback the position of main spool is converted into force by a feedback sping and this force is balanced at the torque motor armature against the torque due to input current. The third basic type uses stiff springs at the spool ends to center the spool against the pressure differential of the pilot stage, which is less frequently used [2].

2.2 Two-stage electrohydraulic servovalve from company Moog

As we mentioned single-stage two major faults which are limited flow capacity and stability problems depending on load dynamics. Focus of the paper went into structure and modeling of G761 servovalve made by company MOOG. Structure of Moog servovalve is seen on Figure 1. Servovalve is made of a polarized electrical torque motor and two stages of hydraulic power amplification. Two motor coils are surrounding the armature, one on each side of the flexure tube [3].

As seen on Figure 1, the flapper of the first stage is rigidly attached to the midpoint of the armature. Flapper then extends into flexure tube, which acts as seal between electromagnetic and hydraulic sections of the valve. The flapper passes two nozzles, which variable orifices are then used to control pressure in the end areas of the second stage spool. Input signal induces a magnetic charge in the armature and causes a deflection of the armature and flapper. With this the size of one nozzle orifice is increased and other one is decreased. This leads to differential pressure on each side of the spool, causing spool motion.

Servovalve uses cantilever feedback spring, which is fixed to the flapper and center of the spool. When spool is displaced, a linear force is induced on the feedback wire, which opposes the original input signal torque. Spool movement continues until the feedback wire force equals the input signal force [3] .

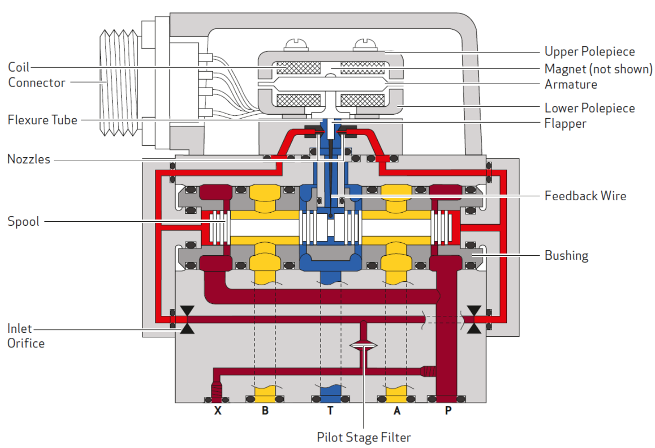


Figure 1: Structure of two stage Moog G761 servovalve [3]**.**

3 Modelling hydraulic servovalve

One of the problems when modelling hydraulic servovalve is nonlinearity, which has not been solved effectively yet [1]. Causes are throttle characteristics of electro-hydraulic elements, control element and problems with hydraulic power mechanism for example hyseresis, dead zone, limiting properties and more.

Servo valve is complex system to simulate, because several parameters can only be estimated within wider range or are completely unknown. According to previous research [1] it can be described as second order system.

$$\frac{Xv(s)}{I(s)} = \frac{K_{SC}}{\frac{s^2}{\omega_{sv}^2}+\frac{2\zeta_{sv}}{\omega_{sv}}+1} \tag{1}$$

Where I is the input of current, Xv is the output of spool displacement of servovalve, ω_{sv} is natural frequency, ζ_{sv} is damping ratio, K_{SC} is the gain of servovalve.

In some cases, servo valve was defined as first order system:

$$\frac{Xv(s)}{I(s)} = \frac{K_{SC}}{\frac{s}{\omega_s}+1} \tag{2}$$

If low frequency signal is used as reference signal, the dynamic characteristics of servo valve can be ignored and it can be treated as a proportional part:

$$x_v = K_{sc} \times \mathrm{I} \tag{3}$$

According to research work [1], the first order model can describe dynamic characteristics of the G761 series servovalves from MOOG company completely. Open loop transfer function is:

$$\frac{Q_0(s)}{I(s)} = \frac{K_{sv}}{\frac{s}{\omega_{sv}}+1} \tag{4}$$

K_{sv} presents flow-current gain of the servo valve: $K_{sv} = K_q K_{sc}$

3.1 Electro-hydraulic Servo Position Control

On Figure 2 example of simulating cylinder system with four-way valve, double-rod cylinder is shown.

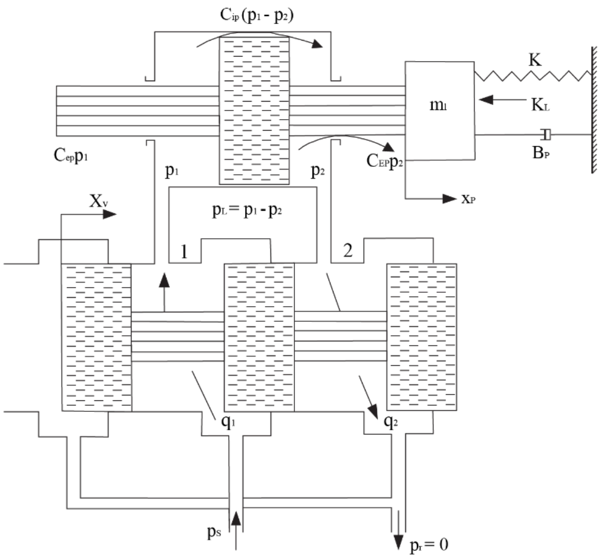


Figure 2: Schematic of valve-controlled cylinder system [1].

Simplified equation was taken into consideration. When modelling several assumptions were made, i. e. supply pressure p_s is constant, return pressure p_r is zero, pressure loss in pipes and valves were ignored, the flow coefficient in each valve port is the same and four-way spool is ideally centered. Also pressure loss in pipes is ignored, it is assumed that system runs at constant temperature, bulk modulus, internal and external leakage is considered laminar flow. Considering pressure flow equation of the control valve, the continuity equation of fluid and pressure-load equation and with neglecting spring stiffness, simplified equation was formed:

$$\frac{X_p(s)}{Q(s)} = \frac{\frac{1}{A}}{s(\frac{s^2}{\omega_h^2}+\frac{2\zeta_h}{\omega_h}s+1)} \tag{4}$$

Where ω_h represents natural frequency and ζ_h damping ratio.

When cylinder with bearing strip is used, influence of friction must be considered. There are a lot of friction model developed, but in this paper simple friction model was used, where F_f presents friction force and B_f is friction coefficient.

$$F_f = B_f \frac{dx_p}{dt} \tag{5}$$

Since high quality MTS R-Series position sensor is used, which has position measurement uncertainty under 0.01 %. Also non-ideal dynamic for this type of position sensor is greatly above the bandwidth of the cylinder and servovalve. Sensor-amplifier system thus is considered ideal and its open loop transfer function reduces to

$$F(s) = 1 \tag{6}$$

By combining derived transfer functions of the servovalve, the cylinder, the friction and the position sensor mathematical model of the electro-hydraulic servo position control system is formed [1].

$$\frac{X_p(s)}{I(s)} = \frac{\frac{K_{sv}}{A}}{s(\frac{s^2}{\omega_h^2}+\frac{2\zeta_h}{\omega_h}s+1)+(\frac{s}{\omega_{sv}}+1)} \tag{7}$$

3.2 Modelling in classical Simulink environment

Simulation example is seen on Figure 3. Electro-hydraulic servo position control system was simulated using transfer function (7). For that already measured hydraulic coefficientsw were used, where, for 40 kN cylinder with bearing strip from research [1] and available in laboratory, where ω_h= 2157.2 , ζ_h = 0.9766, ω_{sv}= 1005.3 and K_{sv}=0.0114.

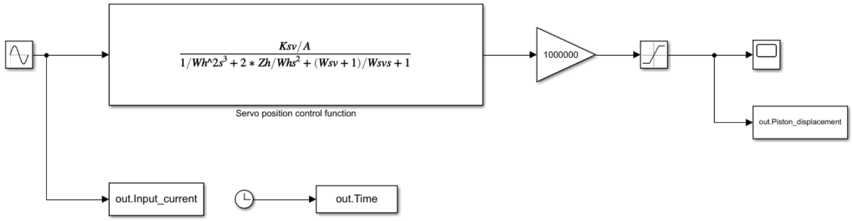


Figure 3: Matlab Simulink approach to simulate electro-hydraulic servovalve piston displacement dependent on input current.

Results are shown on Figure 4. Amplitude was set to 20 mA and frequency to 1 rad/sec. We can see that we do get slight delay of piston movement compared to input signal, which is also the case in real world applications because of response time of servovalves and time necessary for each piston chamber to fill up with hydraulic fluid.

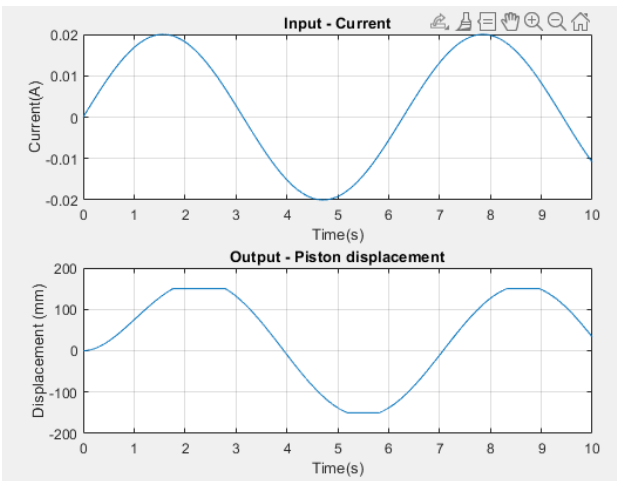


Figure 4: Example using classical Matlab Simulink approach to model servosystem.

3.3 Modelling in Matlab Simulink Simscape

Approach for modelling MOOG G761 hydraulic servovalve in Matlab Simulink Simscape is different from classical Matlab Simulink. Difference is that in Simulink Simscape there are already developed blocks from libraries to simulate mathematical equations. In both cases functioning of the hydraulic system must be broken down to basic components, for example physical behaviour of flow between different orifices, where certain parater must be considered, to get simulation as close we can to real world scenario.

3.4 Obtaining necessary parameters of the servovalve

Getting real world behaviour of hydraulic component or system can be difficult because companies do not provide all the necessary parameters. On Figure 5 we developed our own MOOG G761 servovalve model, where we tried to cover as parameters as we could.

3.5 The first option is to use the available data in the servo valve datasheet.

From the Moog G761 and D761 datasheet [3], certain parameters were obtained, as seen in Table 1:

Table 1: Parameters obtained from Moog G761 and D761 datasheet

Information in the Datasheet	
Nominal flow at 35 bar/spool land	19 l/min
Null leakage flow at *	2.3 l/min
Pilot leakage flow at *	0.7 l/min
Pilot flow * (max) for 100 % step input	0.3 l/min
Spool drive area	0.34 cm^2
*at 210 bar, 32 mm^2/s, 40°C	

Second option is to obtain the information from other literature. Certain values and parameters were obtained from book "Servohydraulik" from authors e. g. H. Murrenhoff [4], which can be seen in Table 2:

Table2: Parameters obtained from the literature

Information in the book	
Torque motor stroke	0.25 mm
Flapper movement	± 0.035 mm
Torque motor force	40 Nmm

3.6 Simscape model of Moog G761 servovalve

For Simscape model certain behaviours were neglected. System is configured to run under ideal pressure of 210 bars. Also the forces between nozzle, flapper and oil volume in pilot stage, were neglected. The coil of the torque motor also wasn't considered because the input signal is [N·mm], so electricity part of the torque motor, was also neglected.

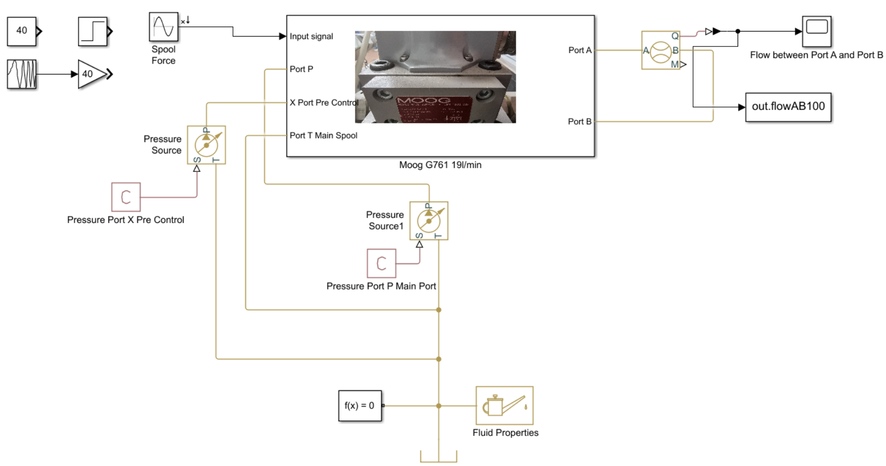


Figure 5: Simscape model of MOOG G761 servovalve.

Parameters can be defined either in the model in Simulink Simscape blocks itself or they can be set up in Matlab code and sent to Workspace. In this case parameters were insterted in Simscape blocks. On Figure 5 main system of the servovalve is shown, where pressure supply is connected to port P and X port of the servovalve. Input signal is in [N mm] and 40 means valve is fully opened in one direction.

Servovalve main system consists of the four subsystems as seen on Figure 6. These subsystems are for torque motor and flapper, hydraulic pre control, main spool mechanic and main spool hydraulic. As mentioned before, we replaced electric subsystem for torque motor with input of the physical signal in [N mm], which is from source [4]. Our system is configured to run simulation on 210 bars, with ISO VG 32 oil at temperature 40 °C.

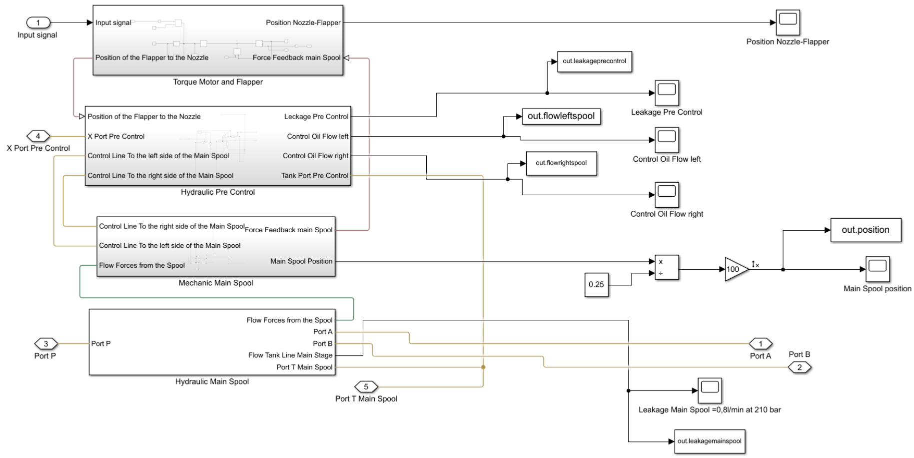


Figure 6: Subsystems of the MOOG G761 servovalve.

Figure 7 represents example of inserting necessary parameters into Simulink Simscape block. For example, leakage area was calculated with calculation for orifice (8).

$$Q = \alpha A \sqrt{\frac{2\Delta p}{\varrho}} \tag{8}$$

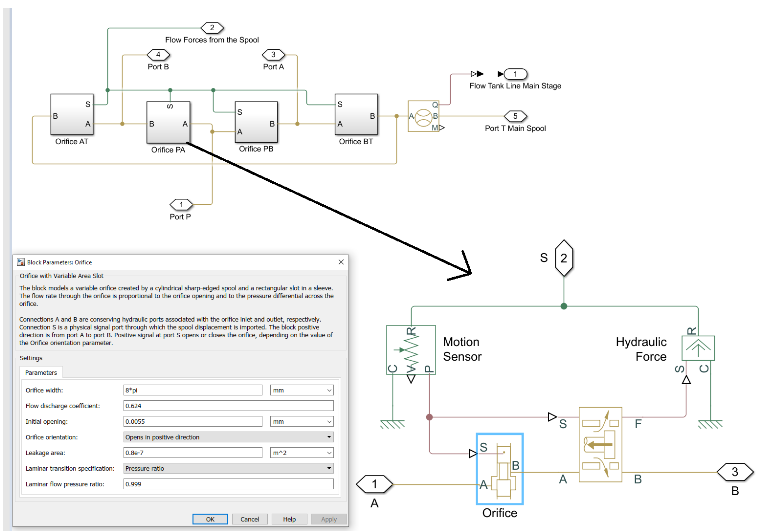


Figure 7: Orifice parameters inside of the orifice between port P and port A in the hydraulic subsystem fort he main spool.

4 Simulation results

Simulation of our Moog G761 were runned in Simscape model. In datasheet response time from 0 to 100 % stroke of the hydraulic spool was defined 5 ms for 19 l/min G761 Moog servovalve. Simulation results were compared to datasheet and the results are similar to those in the datasheet.

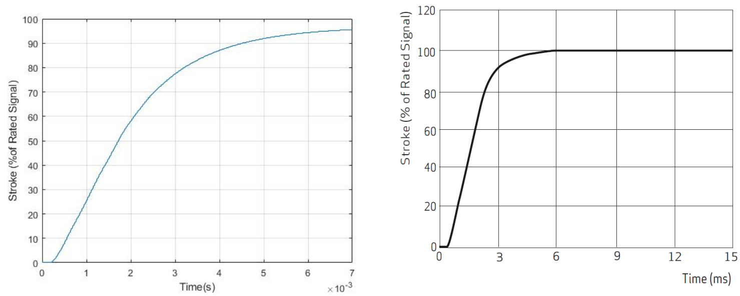


Figure 8: Simulated stroke (left), stroke value from the Moog G761 Datasheet (right) [3]**.**

Actual flow is dependent upon electrical command signal and valve pressure drop. Simulations were perfomed with 70 bar valve pressure drop and compared to datasheet value on Figure 9 and results look promising.

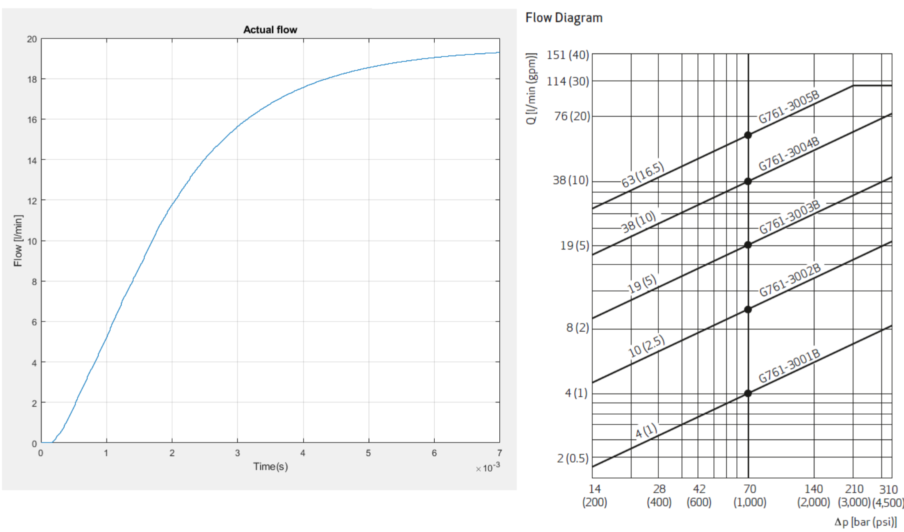
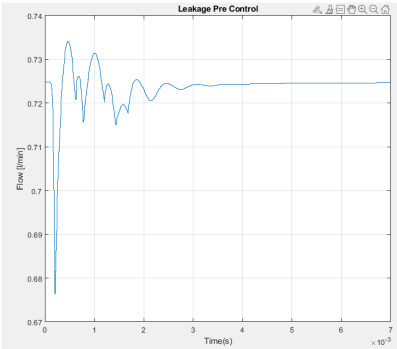
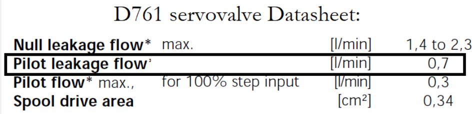


Figure 9: Simulated actual flow (right) and flow value from the Moog G761 datasheet

In Figure 10 pilot stage leakage flow was simulated and compared with data available in G761 datasheet [5], which is practical same servovalve, but older datasheet (D-series). Newer datasheet (G-series) does not provide these parameters. The structure of servoventil is same but G-series are newer models.





D761 servovalve Datasheet:

Null leakage flow*	max.		[l/min]	1,4 to 2,3
Pilot leakage flow'			[l/min]	0,7
Pilot flow*	max.,	for 100% step input	[l/min]	0,3
Spool drive area			[cm²]	0,34

Figure 10: Pilot stage leakage simulation (left) and pilot stage leakage flow value given in datasheet (right).

If we compare data from Figure 11, specifically maximum pilot flow which is defined at 0.3 l/min, simulated results show us maximum pilot flow around 0.2 l/min.

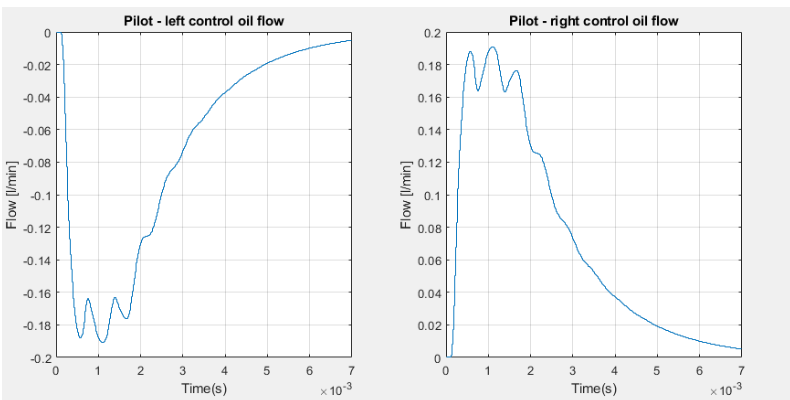


Figure 11: Simulating results of oil flow through pilot stage.

5 Conclusion

In paper Simulink Simscape model of Moog G761 servovalve with 19 l/min nominal flow was developed. The results were comparable with datasheet provided by company of the servovalve. Obtaining all necessary parameters for simulating hydraulic servovalve still remained the problem. Companies usually don't provide all the necessary data needed for hydraulic component simulation. Also it was noted, that in case of Moog G-761 datasheet newer datasheet contains less information than in older Moog d-series datasheet. So, the more information is provided in datasheet, better can the model be. This is the reason why servovalve is normally simulated only as one block, with first and second order model, compared to Simscape model developed in our paper. where some parameters are not given by the companies. In further research it would be necessary to compare simulated and datasheet values to the measured values, by conducting experiments on Moog G761 servovalve.

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