

# **Design of an Internally Pilot Operated Proportional Valve by Use of the Floating Spool Principle**

**Dipl.-Ing. Petr Mejsnar**

ARGO-HYTOS s.r.o., Dělnická 1306, 54315 Vrchlabí, Czech Republic,  
E-Mail: p.mejsnar@argo-hytos.com

**Dr. E. Englberth**

ARGO-HYTOS s.r.o., Dělnická 1306, 54315 Vrchlabí, Czech Republic,  
E-Mail: e.englberth@argo-hytos.com

**Dr. G. Schuster**

ARGO-HYTOS GmbH, Industriestraße 9, 76703 Kraichtal, Germany,  
E-Mail: g.schuster@argo-hytos.com

## **Abstract**

Nowadays, hydraulic systems are an integral part of various machines and equipment, both in stationary and mobile applications. Earth-moving machines like excavators of various sizes and designs use the benefit of hydraulic drives to a great extent.

One possibility to extend operation capabilities of excavators and loaders significantly is the use of a special adapter-head enabling the rotational and swinging movement of the working tool (roto-tilt). The adapter is installed between the excavator arm and the bucket and can be seen as a controllable joint.

Based on this context, the development of a CETOP03 proportional valve with an internal pilot stage is described in this paper. Therefore the boundary conditions have been given due to the application of an excavator rotational-tilting adapter.

During the development phase a Matlab/Simulink model has been built up to enable a better understanding of the valve behaviour. By an example, which occurred during the development phase, it will be shown how well measurement and simulation technique complement each other and help to find the solution of side effects in a shorter period of time.

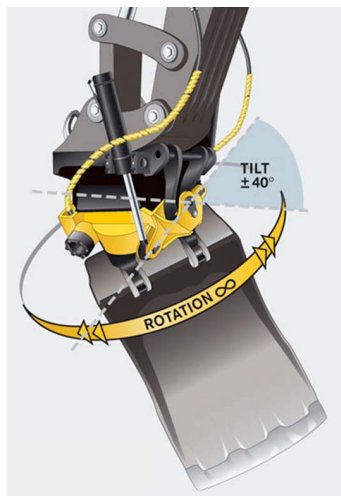
**KEYWORDS:** earth-moving machines, excavator, proportional valve,  
MATLAB/Simulink, CFD, simulation

## 1. Introduction

Hydraulic systems form an integral part of various machines and equipment, both in stationary and mobile application. Excavators as a part of earth-moving machines use also the advantages of hydraulic drives. One of the hydraulic equipment, which significantly increases operability of excavators, is a special adapter that enables rotary and swinging movement of the connected working tool e.g. a bucket.

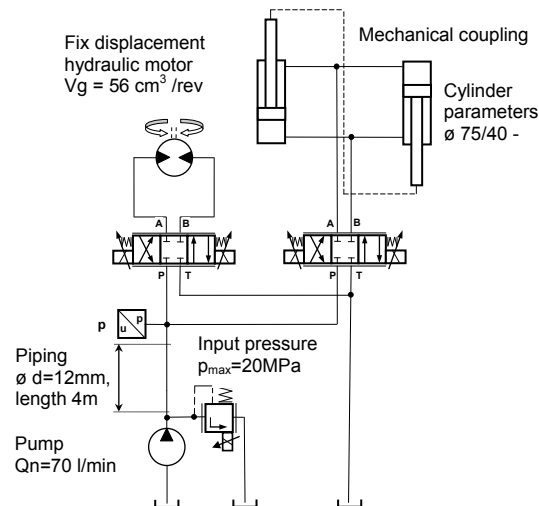
**Figure 1** shows the technical design of a rototilt adapter with two degrees of freedom, rotation and tilting. The importance of this special adapter is based on the fact of time saving which would be needed to change the position of the machine. By this the adapter increases work efficiency and safety at work.

To execute the rotational movement, a worm-gear unit with fixed-displacement hydraulic motor is used. The tilting movement is carried out by two double acting hydraulic cylinders. To control the direction and the speed of movement in particular axes, two modular electrical driven proportional directional control valves are used (CETOP03).



**Figure 1:** Rotating-tilting adapter of excavator with two degrees of freedom

**Figure 2** shows the hydraulic schematics of the adapter which has been built up for detailed laboratory studies of the system interactions with use of the real rototilt adapter. By this it was possible to perform tests without the necessity of the real excavator.



**Figure 2:** The scheme of laboratory hydraulic circuit of the rotating-tilting adapter with two degrees of freedom

## 2. Requirements for proportional valves controlling roto-tilt adapters

Before this, typical proportional valves in CETOP3 have been used to control the roto-tilt functions. Due to product portfolio extension of the roto-tilt manufacturer new requirements have been set. By increasing the power limits of the roto-tilt functions also some of the operating parameters for the proportional valves have been increased. Most challenging requirement has been to fulfill all these changes (see below) in the same build-in dimensions and compatible to the electrical parameters which have been used for the current proportional valve (CETOP03) solution.

New requirements have been:

- Maximum flow rate  $Q_{\max}=85 \text{ l/min}$  at a pressure drop of 120bar
- Range of supply pressure up to  $p_0=250\text{bar}$
- High sensitivity at small flow rate (easier regulation of desired position)
- Valve stability within the whole range of given pressure and flow rate
- Monotonous trend of power limit characteristics within the whole range of working pressure

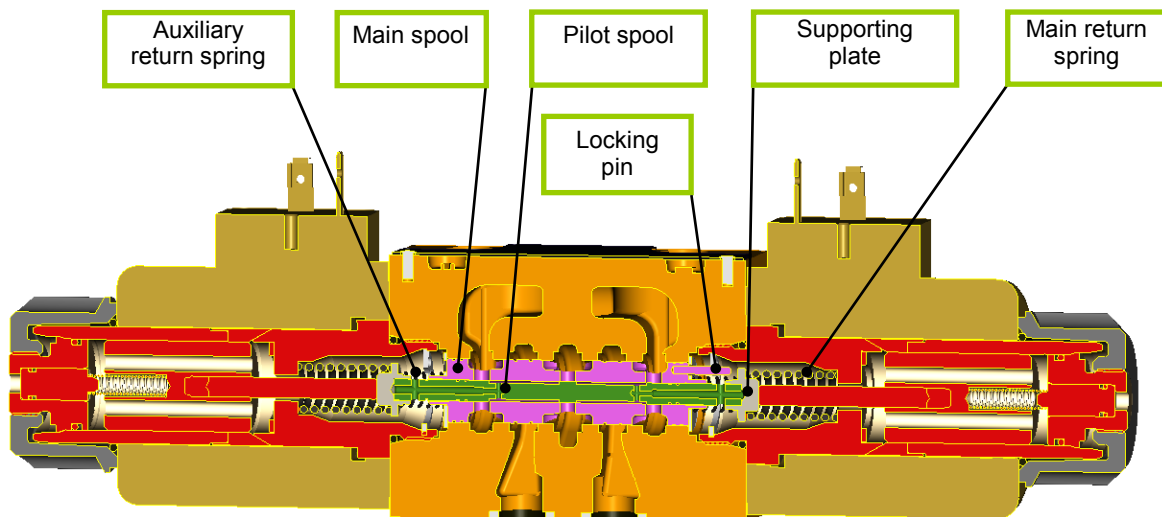
## 3. Approach of proportional pilot operated directional control valve

Based on the experience of earlier done optimization steps for proportional valves, it was necessary to find a completely new technical solution to satisfy the mentioned requirements. Therefore a layout has been chosen which is based on a two stage spool design (pilot and main) in a directional control valve:

The so called floating spool principle.

The main features of this approach are:

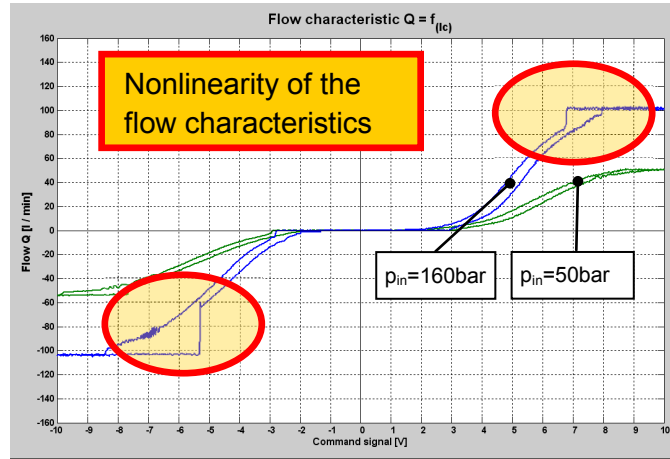
- Floating spool of the pilot stage is integrated into the spool of the main stage.
- To avoid rotation of the main spool it is equipped with a special pin.
- Radial drillings in the main spool at ports P and T create control edges of the pilot stage in relation to the main spool.
- The middle position of the main spool is provided by a couple of springs, which acts between washers of the main stage and supporting plates of the pilot stage.
- The middle position of the pilot spool is provided by a couple of auxiliary return springs in parallel configuration.
- Standard design of proportional solenoids for nominal size NG06 (CETOP03) has been used.



**Figure 3:** Cross section of proportional valve using the floating spool principle (PRM8)

The design of the valve led to a first prototype which is shown in **figure 3**. During testing of these samples it occurred that some of them showed an undesired performance. The behaviour of these valves is given by **figure 4**.

As one can see the unsteady performance is somehow connected with the level of the support pressure and emerge only in areas of a high volume flow. The effect was reproducible and didn't exist anymore at low support pressure levels.



**Figure 4:** Flow characteristics of functional sample

To identify the causes of the phenomenon and by this an explanation for the unsteady performance a MatLab/Simulink model has been used. The basic model had already been built up during the design phase for a better interpretation of the floating spool principle and the internal functions.

#### 4. Simulation model

The MatLab/Simulink model has been built up and a solution has been found based on the following steps:

1. Mathematic equation for the flow by the metering edges
2. CFD simulation to determinate flow forces as a function of flow rates / pressure difference and spool stroke
3. Experimental measurements of proportional solenoid force
4. Verification of the simulation model by comparing with the measurement results
5. Analysis of unsteady behavior that appears in flow characteristics.
6. Development of the solution and simulation of the impact to the power limit characteristics (feasibility study).

##### 4.1. Mathematical-physical model

The equations of motion describe the force equilibrium of pilot and main spool:

Pilot spool:

$$m_1 \ddot{x}_1 + b_1 \dot{x}_1 + (2k_1 + 2k_2)x_1 - k_2 x_2 = F_m(lc, x_1) + (p_3 - p_2)S_1 \quad (1)$$

Main spool:

$$m_2 \ddot{x}_2 + b_2 \dot{x}_2 + k_2(x_1 - x_2) = (p_3 - p_2)S_2 - F_h(dp, x_2) \quad (2)$$

Flow through the metering area of the pilot spool are expressed (/1/) as a non-linear function of pressure drop and orifice opening.

$$Q = C_{d\infty} \left( 1 + ae^{-\frac{\delta_1}{Cd\infty}\sqrt{Re}} + be^{-\frac{\delta_2}{Cd\infty}\sqrt{Re}} \right) \frac{S_{ekv} x_o}{1 - e^{-\frac{x}{d_o}}} \sqrt{\frac{2}{\rho} \Delta p} \quad (3)$$

Differential continuity equations are expressing control pressures of the main spool (/3/)

$$\dot{p}_2 = \frac{K}{V_A} (Q_{12} - S_1 \dot{x}_1 - S_2 \dot{x}_2) \quad (4)$$

$$\dot{p}_3 = \frac{K}{V_B} (Q_{13} - S_1 \dot{x}_1 - S_2 \dot{x}_2) \quad (5)$$

The force of the proportional solenoid is obtained by experimental measurements of the real system and is generally a non-linear function of two variables

$$Fm = f(lc, x_A) \quad (6)$$

General equation expressing function of flow forces acting on the main spool can be written as

$$Fh = f(dp, x_S) \quad (7)$$

and has been calculated by CFD.

Due to the mechanical layout it can be written

$$x_1 = x_A \quad (8)$$

$$x_2 = x_S \quad (9)$$

Flow through the control volume for one calculation step is given by integration of velocity  $u_i$  across the inlet area  $S_{int}$ , with normal vector  $v_i$ , through flow region, which can be expressed as

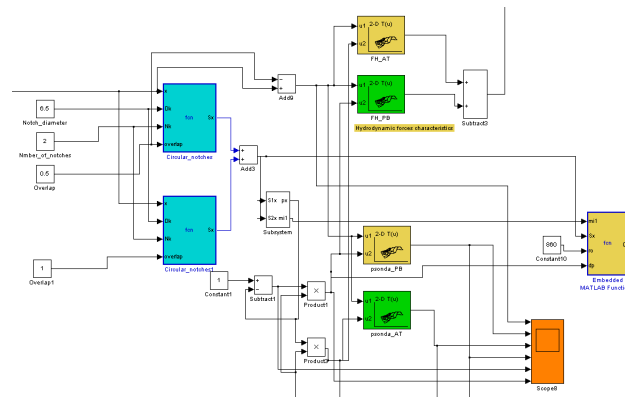
$$Q = \int_{S_{int}} u_i v_i dS \quad (10)$$

Flow force acting on the pilot spool was determinated from the momentum equation of stationary flow /2/

$$\int_S \rho u_i u_j v_j dS = \int_S \tau_{vi} dS \quad (11)$$

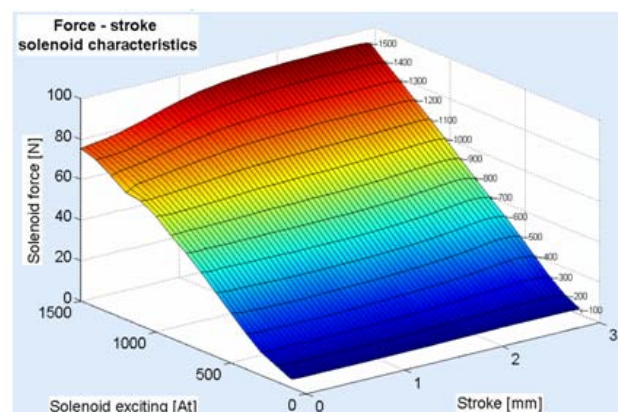
The inner space of the valve housing is divided by four main cavities, which are connected to the valve ports. Inlet port P, outlet port T, consumer ports A and B. By moving the spool from the middle position, the cavities of the housing are interconnecting each other.

The simulation model is based on the given equations and contexts. An example of the simulation model with subsystems for hydrodynamic force calculations using 2D look-up tables is shown in **figure 5**. To evaluate the quality of new designs two models have been carried out. The first model is used for calculating the power limit characteristics and the second model is used for calculating the flow characteristics.

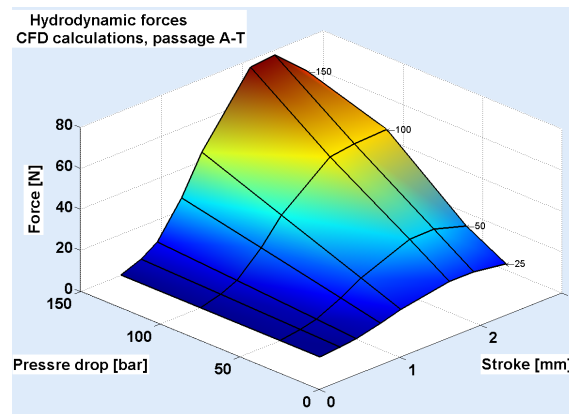


**Figure 5:** Example of the simulation model with subsystems for hydrodynamic force calculations using 2D look-up tables

The data gathered by the force measurements of the proportional solenoid (see **figure 6**) and the data of the flow forces calculated by using a CFD tool are imported into look-up tables (see **figure 7**) where the values are presented as two dimensional arrays.



**Figure 6:** Force characteristics of the proportional solenoid



**Figure 7:** Characteristics of the flow force, which acts on the main spool, e.g. interconnection A-T

The model verification is carried out by comparison of flow characteristics and power limit characteristics with measurement results. For a better understanding e.g. the value of the control pressure which moves the main spool and the spool position as a function of control current can be shown along others by the model.

The whole simulation is considered as a quasi-static process. The variation of the input parameters is based on measuring the static performance of the real valve. In this measurement the input parameters are changing also very slowly.

## 5. Results

The simulation results of the new solution regarding power limit (see **figure 8**) and flow characteristic (see **figure 9**) were made by a procedure that is similar to laboratorial measurements on a real valve. By comparing the simulation results with experimental measurements the model has been validated.

By simulation the problem of the unsteady flow characteristic was identified as a result of a limited main spool stroke.

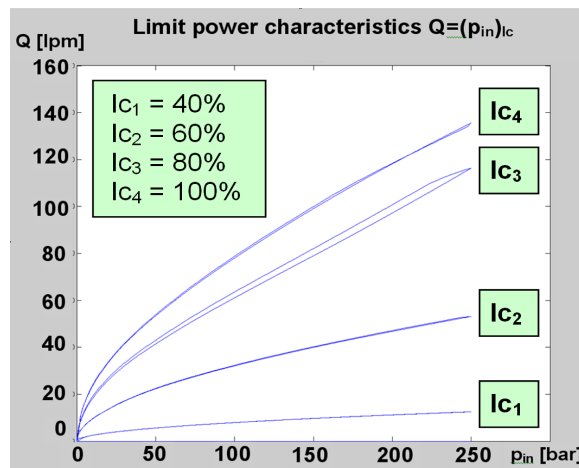
Because of the mechanical main spool stroke limitation a situation occurs, where in the area of the actuation system (the area where the control pressure acts on the main spool), the control pressure increased up to supply pressure level (which is equal to pressure in port P).

Usually when the main spool reaches the stroke of the pilot spool the pressure in the control volume decreases to the level of the T-Port. In this case of malfunction the main spool couldn't reach the predicted position due to the stroke limitation. Therefore the P-Port was still connected to the control volume and increased steadily the control

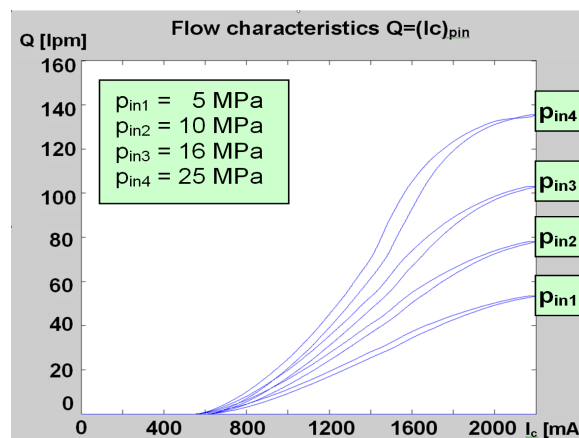
pressure to the P-Port level. This is the situation which occurred at maximum stroke and by that at maximum volume flow (figure 4).

The next step in the characteristic is to decrease the control current with the effect to bring the pilot spool slowly back into the center position. The only force which moves the pilot spool in this direction is the force given by the main return spring. This force has to deal with the force of the solenoid and the force given by the high pressure (close to P-port level) in the control volume acting on the pilot spool.

In case of a low P-Port pressure the force of the spring is strong enough to overcome the additional resistance. At a higher support pressure the equilibrium between spring force, pressure in the control volume in restriction to the pilot spool diameter and the solenoid force decides if or when the pilot spool begins to move. With this result the unsteady performance, which is shown in figure 4, and it's context to the dependency of the P-Port pressure can be shown.

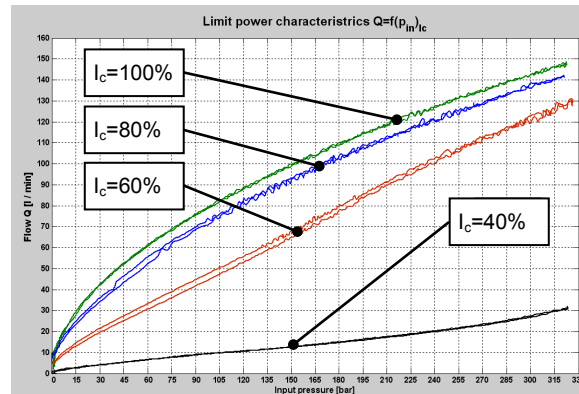


**Figure 8:** Limit power characteristics of the valve PRM8, results from simulation model

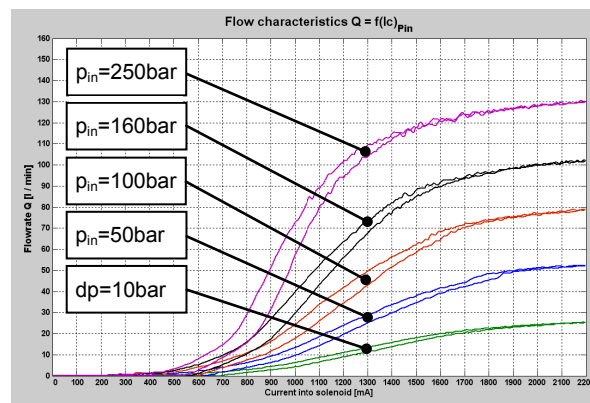


**Figure 9:** Flow characteristics of the valve PRM8, results from simulation model

A design change which removed the mechanical stroke limit of the main spool, led to desired function of the proportional valve which are shown in **figures 10 and 11**.



**Figure 10:** Limit power characteristics of the valve PRM8 after design change



**Figure 11:** Flow characteristics of the valve PRM8 after design change

By comparing the simulation (figure 8 & 9) with the measurement results (figure 10 & 11) one can see how well model and reality fits together.

## 6. Conclusion

A new design of a proportional pilot operated directional control valve based on the floating spool principle has been presented. The floating spool principle has been chosen due to the high restrictions of build in space and electrical parameter on one hand and distinctly increased power limits on the other. By the additional use of a simulation model an unsteady behaviour which occurred on some of the first prototypes could be identified and solved. Due to implementation of a specific design change a good performance of the proportional pilot operated directional control valve PRM8 was achieved. The PRM8 is already in use in the application of a roto-tilt adapter at the customer.

## 7. References

- /1/ Wu, D. – Burton, R. – Shoenau, G. – Bitner, D.: Modelling of orifice flow rate at very small openings. International Journal of Fluid Power, Vol. 4, No. 1, 2003, p. 31 – 39.
- /2/ LISOWSKI, E. - DOMAGALA, M.: Modelling of Hydrodynamic Interaction Forces in Direct Relief Valve by the Use of CFD Method. In: Proceedings of the 18-th International Conference on Hydraulics and Pneumatics, Prague 2003, p. 498-505.
- /3/ NOSKIJEVÍČ, P. Modelling and system identification. Ostrava: MONTANEX, a.s., 1999. 276p. ISBN 80-7225-030-2.
- /4/ MEJSNAR, P. – ENGELBERTH. E. 2008. Optimization of the proportional directional control valve for special hydraulic adapter with two degrees of freedom using physical and CFD modelling, Proceedings of the 20th International conference on Hydraulics and Pneumatics, Prague 2008, p.323-329. ISBN 978-80-02-02074-5.

## 8. Nomenclature

$a, b$	coefficients in the empirical model	-
$b_1$	pilot spool attenuation coefficient	kg/s
$b_2$	main spool attenuation coefficient	kg/s
$C_{d\infty}$	turbulent discharge coefficient	-
$d_0$	height of square type orifice at the zero position	m
$dp$	pressure drop across the control edge	N/m <sup>2</sup>
$F_h$	flow force of the main spool	N
$F_m$	force of proportional solenoid	N
$I_c$	actuating current of proportional solenoid	A
$K$	oil bulk modulus	N/m <sup>2</sup>
$k_1$	stiffness of main return spring	N/m
$k_2$	stiffness of auxiliary return spring of the pilot spool	N/m
$m_1$	mass of pilot spool	kg
$m_2$	mass of main spool	kg

$p_2$	control pressure of the main spool from the side of solenoid A	$\text{N/m}^2$
$p_3$	control pressure of the main spool from the side of solenoid B	$\text{N/m}^2$
$Q_{12}$	flow across the control edge from area 1 (P) to area 2 (A)	$\text{m}^3/\text{s}$
$Q_{13}$	flow across the control edge from area 1 (P) to area 3 (B)	$\text{m}^3/\text{s}$
$Re$	Reynolds number	-
$S_{ekv}$	rectangular orifice width	$\text{m}^2$
$S_{int}$	flow inlet area	$\text{m}^2$
$S_1$	pilot spool cross section area	$\text{m}^2$
$S_2$	main spool cross section area	$\text{m}^2$
$u_i$	flow velocity in control volume	$\text{m/s}$
$V_A$	volume connected to the main spool from A side	$\text{m}^3$
$V_B$	volume connected to the main spool from B side	$\text{m}^3$
$x_A$	position of the armature inside the proportional solenoid	$\text{m}$
$x_S$	spool position inside the housing of the valve	$\text{m}$
$x_o$	orifice (spool control notch) opening	$\text{m}$
$x_1$	pilot spool position	$\text{m}$
$x_2$	main spool position	$\text{m}$
$\dot{x}_1$	pilot spool movement velocity	$\text{m/s}$
$\dot{x}_2$	main spool movement velocity	$\text{m/s}$
$\ddot{x}_1$	pilot spool acceleration	$\text{m/s}^2$
$\ddot{x}_2$	main spool acceleration	$\text{m/s}^2$
$\delta_1, \delta_2$	attenuation coefficients of the empirical model modification - associated with discharge coefficient	-
$\Delta p$	pressure drop across orifice	$\text{N/m}^2$
$v_i$	normal vector of the flow via control volume	-
$\rho$	fluid density	$\text{kg/m}^3$