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# TUNING THE ELECTRO-HYDRAULIC SERVOMOTORS BY NUMERICAL SIMULATION

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*Lucrarea prezintă o metodologie de optimizare a structurii şi strategiei de comandă a servomotoarelor electrohidraulice cu reglare frontală, bazată pe modelarea matematică şi simularea numerică cu limbaje specializate. Optimizarea formei semnalului de reglare a capacităţii acestor motoare este impusă de utilizarea unui resort elicoidal pentru transformarea poziţiei pistonului cilindrului hidraulic într-o forţă de reacţie aplicată sertarului servovalvei. Aceasta este comandată printr-un electromagnet proporţional de forţă care nu poate amortiza autooscilaţiile sistemului inerto-elastic de reglare a capacităţii servomotorului.*

*The paper presents a methodology of optimizing the structure and control strategy of axial piston bent-axis motors with variable displacement with the aid of mathematical modeling and numerical simulation with modern software. The reason behind this optimization is determined by the spring feedback, used to turn the setting piston stroke into a force, applied on the proportional servo valve spool. The proportional force solenoid controlling the spool position is not able to increase the low damping factor of the displacement control system, which has a low natural frequency due to the small helical spring stiffness.*

**Keywords:** mathematical modeling, numerical simulation, optimization, electro-hydraulic servomotors

## 1. The design-performance relationship for hydraulic servomotors

The design of a hydrostatic transmission for a given application needs a deep knowledge of the system requirements and the real performances of the available hydrostatic machines. The main components of such a transmission have different performances. The hydraulic motors are defined by both static and dynamic parameters. The steady-state behavior is defined by hydraulic,

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mechanical, mixed characteristics, and efficiency diagrams. Complex diagrams are supplied by manufacturers for supporting the proper choice of a motor [1].

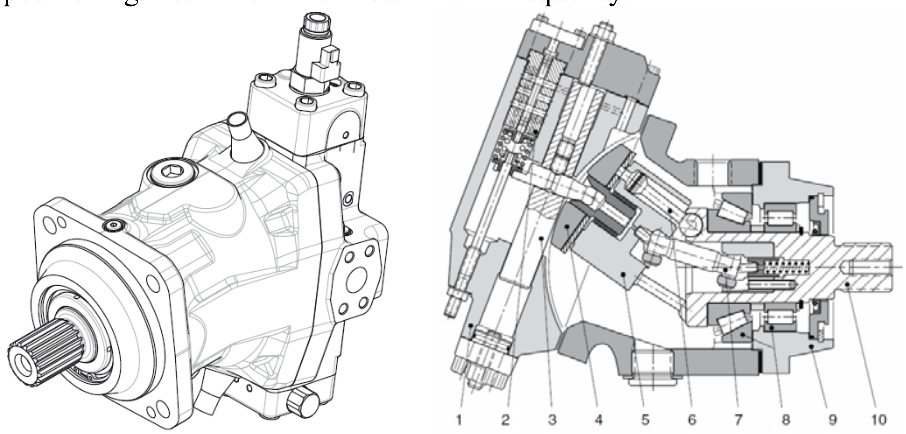
The dynamic performance of a hydraulic motor cannot be identified easily. A direct way involves the evaluation of the free transient induced by a step input flow, supplied by a high speed electro-hydraulic servo valve. Another common way is to find the frequency response. Parameters such as overshoot, response time, bandwidth and others can create an image of the dynamic capabilities of a motor.

A system designer needs to know the range of displacement, the normal and intermittent pressure, the normal speed operation range, the normal and intermittent flow, the specific power [kW/kg], the specific torque [Nm/kg] for a standard pressure drop across the ports (100 bar), the output power, the corner power, the mass moment of inertia, the mass, the maximum no-load start pressure drop, bearing life computation according to the shaft load, and the control system architecture.

From the user's perspective, the most important quality of a hydraulic motor is the stability at low speed operation. According to this parameter, the manufacturers offer low speed motors ($n_{min}$ = 1…10 rev/min), medium speed motors ($n_{min}$ = 10…50 rev/min) and high speed types ($n_{min}$ = 50…400 rev/min).

The "stick-slip" phenomenon can be avoided by lowering the leakages or by increasing the number of operational cycles per revolution for the same displacement. Another way is the use of the hydrostatic bearings instead of other types of bearings. The only way to keep the volumetric efficiency at a reasonable value for a wide transmission ratio range (1:8...1:20) is the use of high pressure variable displacement motors. The common option from this point of view is an axial piston motor.

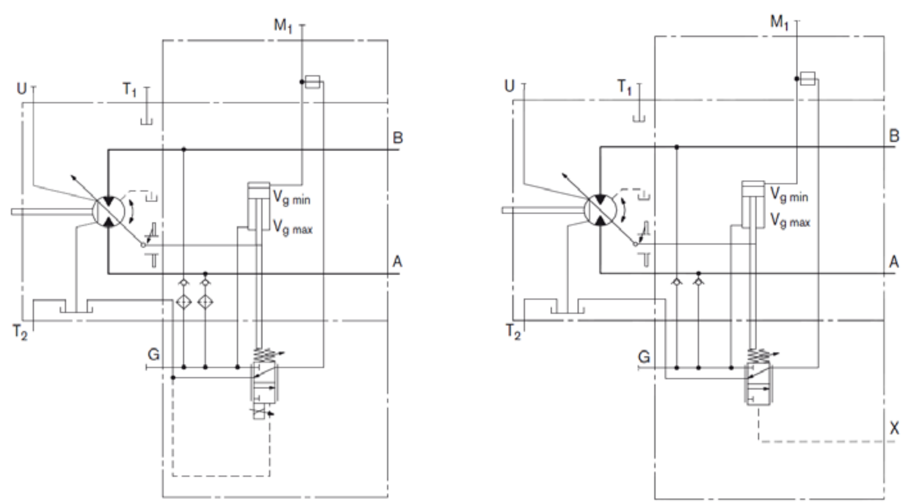
The swash plate motors have the lowest stable speed, but they need very clean oil (class 8 according NAS). They are used for high technology applications in close circuits mainly (Dennison-Gold Cup). The variable displacement increases the weight and the price of such a motor to the same level as a servo pump. The variable displacement bent axis motors (fig. 1) are the best option in heavy industrial applications for two reasons: the biggest specific power [W/kg] and the lack of a gear pump, for supplying the displacement control servomechanism. The combination of an electro-hydraulic variable displacement swash plate axial piston pump and an electro-hydraulic bent axis motor becomes nearly a standard in the field (Bosch-Rexroth). Two types of mechanical links between the cylinder barrel and the output shaft are currently used: linear contacts between conical pistons and cylinders (fig.1a); a universal two or three arms synchronizing shaft (fig. 1b). Many displacement control systems are available for this kind of motors. Most of them are including a servo valve actuated by a force proportional solenoid (fig. 2a) or by a pressure force (fig. 2b). The stroking piston position is turned into a force feedback by a low stiffness helical spring. The positioning mechanism has a low natural frequency.




a) b)

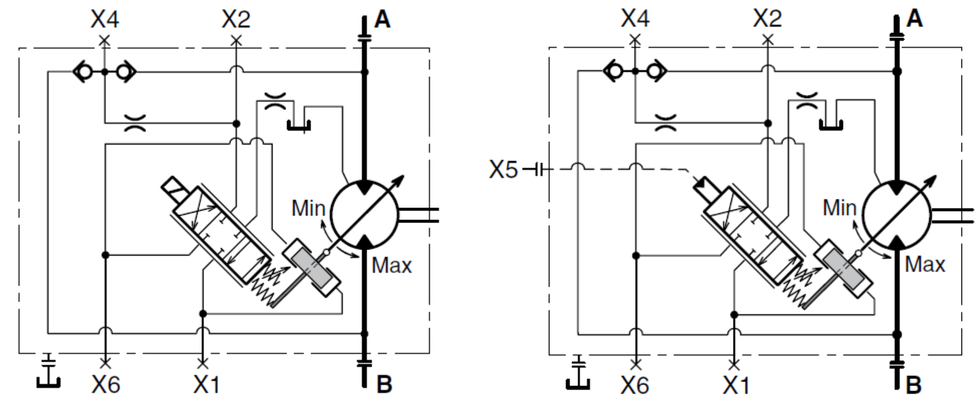
Fig. 1. Bent axis variable displacement axial piston motors (TRIMOT):
a) A6VM (BOSCH-REXROTH); b) V12 (PARKER).

The high displacement motors are controlled by a symmetric piston. They include a critical lap four way servo valve (fig. 3). The stability of the electro-hydraulic servo system and the hydro mechanical one is achieved by two restrictors sited on the input port and the output port of the servo valve.




a) b)

Fig. 2. The structure of the main displacement control systems for A6VM motors (BOSCH-REXROTH): a) electro-hydraulic servo system; b) hydro mechanical servo system.




a) b)

Fig. 3. Control systems for PARKER V14 hydraulic motor:
a) electro-hydraulic control system; b) hydraulic control system.

The highest accuracy of the displacement control system is obtained with the help of an additional electrical position feedback, like in mixed feedback two stage electro-hydraulic servo valve [2].

### 2. Mathematical modeling of force feedback bent axis motors

According to system theory, the displacement control device of a bent axis motor (fig. 4) is a force feedback servomechanism. The servo valve has a simple design (fig. 5): a spool with a single active shoulder is closing in a critical manner (zero lap) two round holes of the sleeve (fig. 5a). An increase of the

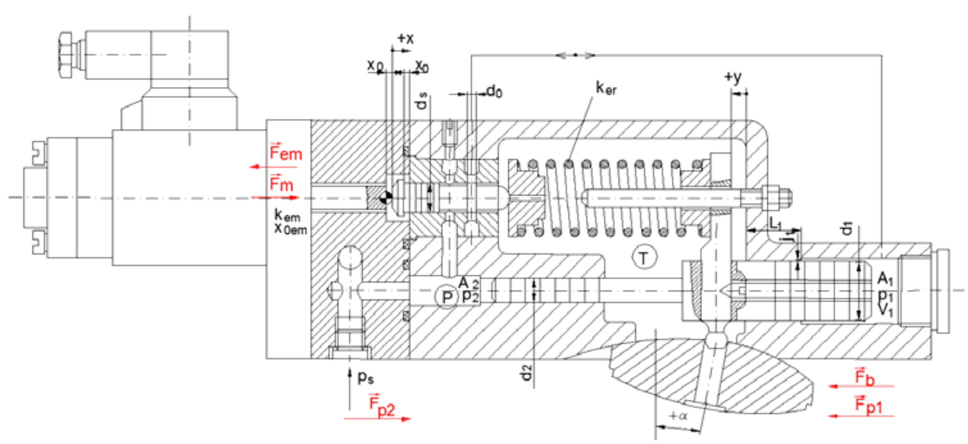



Fig. 4. Force feedback servomechanism for a bent axis motor.

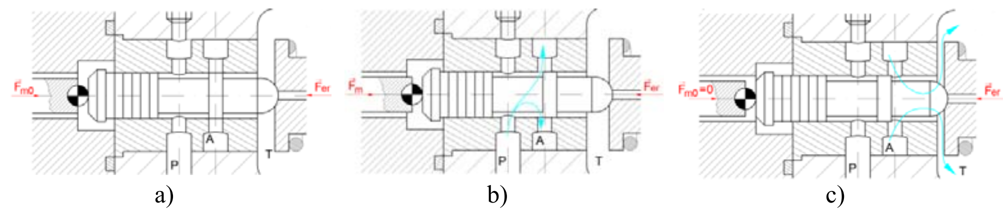



Fig. 6. The hydraulic connections of the servo valve:
a) all ports closed; b) P→A and T closed; c) A→T and P closed.

solenoid current (fig. 5b) creates the connection between the input motor port and the piston control chamber; in such a way, the piston reduces the tilting angle and compresses the feedback spring. A new steady-state is achieved, the motor displacement is reduced, and the shaft speed increases. The decrease of the solenoid current creates the connection between the piston control chamber and the output port, and the piston increases the tilting angle. The feedback spring is released and the motor shaft rotation speed decreases. The connection between the control current and the motor displacement is linear, with a negative slope.

The mathematical model of the servomechanism includes the motion equation of the proportional solenoid plunger and the servo valve spool, the stroking piston motion equation and the flow equation for the piston control chamber [3], [4].

*a) The motion equation of the proportional solenoid plunger and the servo valve spool has the following form:*

$$m_x \ddot{x} = F_m - F_{em} - F_{er} - F_h - F_{fv} - F_{fs} \tag{1}$$

where

$$F_m = K_m (i - i_0) \tag{2}$$

is the electromagnetic force corresponding to the difference between the solenoid current, i and the start current, $i_0$.

We denote by

$$I = i - i_0 \tag{3}$$

the useful current. The dynamic behavior of a force proportional solenoid can be approximated by a first order lag with a time constant of about 15...35 ms [4]. The steady-state constant of the solenoid, $K_m$ is given by the average slope of the static characteristics:

$$K_m = \frac{F_m(I_{\max})}{I_{\max}} \tag{4}$$

The solenoid plunger is retracted inside the coil with the force

$$F_{em} = K_{em}\left(x + x_{0em}\right) \tag{5}$$

where $K_{em}$ is the static stiffness of the reference helical spring; $x_{0em}$ - spring preload; x - spool stroke from the neutral position, defined by the connection between P, A and T by the clearances only ("hydraulic null point").

The feedback spring force,

$$F_{er} = K_{er} y \tag{6}$$

depends on the piston stroke, y, and on the feedback spring stiffness, $K_{er}$. The maximum value of the piston stroke can be computed by the relation

$$y_{\max} \cong R_d\left(\alpha_{\max} - \alpha_{\min}\right) \tag{7}$$

where $R_d$ is the radius of the cylinder surface in contact with the sliding valve plate. The flow force on the spool depends on the direction of the spool stroke from the neutral position. If x > 0,

$$F_h^+ = \rho Q^+ v^+ \cos\theta \tag{8}$$

where

$$Q^+ = 2c_d A^+(x)\sqrt{\frac{2\left(p_s - p_1\right)}{\rho}} \tag{9}$$

$$v^+ = c_v\sqrt{\frac{2\left(p_s - p_1\right)}{\rho}} \tag{10}$$

and θ is the angle between the liquid jet and the spool axis. The two round metering orifices from the servo valve sleeve have the diameter $d_0$. Relation (8) becomes:

$$F_h^+ = K_h \cdot A(x) \cdot \left(p_s - p_1\right) \tag{11}$$

Here $p_s$ is the pressure in the input port of the motor, and $p_1$ - the pressure from the control chamber of the piston. The constant $K_h$ of the hydrodynamic force has no dimensions:

$$K_h = 2c_d c_v \cos\theta \tag{12}$$

The hydrodynamic force tends to close the valve. For x < 0,

$$F_h^- = \rho Q^- v^- \cos\theta \tag{13}$$

Here

$$Q^{-} = 2c_d A^{-}(|x|)\sqrt{\frac{2p_1}{\rho}} \tag{14}$$

$$v^{-} = c_v\sqrt{\frac{2p_1}{\rho}} \tag{15}$$

The hydrodynamic force $F_h^-$ tends to close the valve also:

$$F_h^{-} = K_h A^{-}(|x|)\, p_1 \tag{16}$$

The viscous friction force between the moving assembly of the solenoid and the servo valve,

$$F_{fv} = K_{fv} \cdot \dot{x} \tag{17}$$

contains the coefficient $K_{fv}$ which can be found by experiments only. The same problem arises with regards to the static friction:

$$F_{fs} = K_{fs} \cdot sign\,\dot{x} \tag{18}$$

The overall equivalent mass of the plunger, $m_m$ , spool, $m_s$, and feedback spring, $m_{er}$ is:

$$m_x = m_m + m_s + 0{,}35\, m_{er} \tag{19}$$

*b) The stroking piston motion equation is*:

$$m_y\ddot{y} = F_{p1} - F_{p2} - F_{er} + F_b - F_{fvp} - F_{fsp} \tag{20}$$

where

$$m_y = m_p + m_t + m_{bc} \tag{21}$$

is the overall equivalent stroking piston mass, composed by the piston mass itself, $m_p$ , the feedback lever, $m_t$ and by the cylinder barrel mass, $m_{bc}$.
The other forces are computed in the following relations:

$$F_{p1} = p_1 A_1 \tag{22}$$

$$F_{p2} = p_2 A_2 \tag{23}$$

$$F_{er} = K_{er}\, y \tag{24}$$

$$F_{fvp} = K_{fvp}\dot{y} \tag{25}$$

$$F_{fsp} = K_{fsp} sign\,\dot{y} \tag{26}$$

$$F_b = p_s K_b - y K_y \tag{27}$$

The tilting force, $F_b$, depends both on the piston stroke and the input motor pressure. The increase of the tilting angle generates a positive feedback. The two coefficients can be found by experiments only. The influence of the motor load is greater than the control piston stroke. For example, in the case of the HYDROMATIK A6V motor $K_b$= 2.6 N/bar and $K_y$ = 1.35 N/mm [1], [8].

*c) The flow equation for the piston control chamber* depends on the spool stroke sign with respect to the neutral position. If $0 \le x \le x_{max}$ ,

$$Q_d^{\,+} - \dot{y}A_1 - Q_{l1} = \frac{V_1}{\varepsilon} \cdot \frac{dp_1}{dt} \tag{28}$$

The servo valve flow is given by the equation

$$Q_d^{\,+} = 2c_d A(|x|)\sqrt{\frac{2(p_s - p_1)}{\rho}} \ge 0 \tag{29}$$

The leakage flow can be approximated by the equation

$$Q_{l1} = K_{l1} p_1 > 0 \tag{30}$$

where

$$K_{l1} \cong 2{,}5 \frac{\pi d_1 j_1^{\,3}}{12\eta l_1} \tag{31}$$

and

$$V_1 = V_{10} + A_1 y \tag{32}$$

is the control chamber volume, including the volume of the supply channel. For y = 0, $V_1 = V_{10}$ (the minimum value). The flow equation becomes

$$\dot{p}_1 = \frac{\varepsilon}{V_1}\left(Q_d^{\,+} - \dot{y}A_1 - Q_{l1}\right) \tag{33}$$

If $x_{min} \le x < 0$, flow equation becomes

$$\dot{p}_1 = -\frac{\varepsilon}{v_1}\left(Q_d^{\,-} + \dot{y}A_1 + Q_{l1}\right) \tag{34}$$

where

$$Q_d^{\,-} = 2c_d A(|x|)\sqrt{\frac{2p_1}{\rho}} \geq 0 \tag{35}$$

The metering orifices area is a nonlinear one:

$$A(x_r) = \frac{A_0}{\pi}\left[\arccos(1-2x_r) - 2(1-2x_r)\sqrt{x_r - x_r^2}\right] \tag{36}$$

Here $x_r = |x|/d_0$ is the relative opening of the servo valve, and $A_0 = \pi d_0^2/4$ - the maximum metering orifice area.

### 3. Numerical simulation in SIMULINK

The above mathematical model was integrated in Simulink using the network from figure 7. The following parameters were used in the model: $i_0$=0.09934 A; $K_m$= 129 N/m; $1/m_x$ = 7.0922 $kg^{-1}$; $x_{0em} + x_0 = 1\cdot10^{-3}$ m; $K_{er}$= 1620 N/m; $K_h$= 0.428 m; $x_r$ =400; $A_0/\pi = 1.5625\cdot10^{-6}$ $m^2$; $K_{fv}$= 200 N/m; $K_{fs}$= 100 N/m; $A_1 = 3.14\cdot10^{-4}$ $m^2$; $A_2 = 1.57\cdot10^{-4}$ $m^2$; $K_b$= 2.6 N/bar; $K_{fvp}$= 4000 N/m; $K_{fsp}$= 2000 N/m; $1/m_y$ = 0.2717 $kg^{-1}$; $c_d$=0.61; $\rho$=900 $kg/m^3$; $K_{l1}$= $1.924\cdot10^{-13}$ N/bar; $A_d = 0.196\cdot10^{-6}$ $m^2$; $\varepsilon/V_1 = 4\cdot10^{13}$ $N/m^2$; $K_y$= 1350 N/m. The figure 8 presents the control piston stroke, and the servo valve stroke evolution for a input ramp of 2s with a maximum level of 75% from the nominal value, and $p_s$=100 bar.

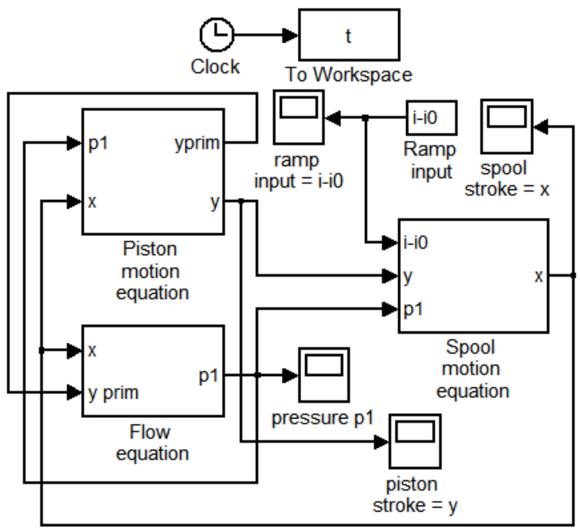



Fig. 7. Simulation network of the stroke control system for a ramp input.

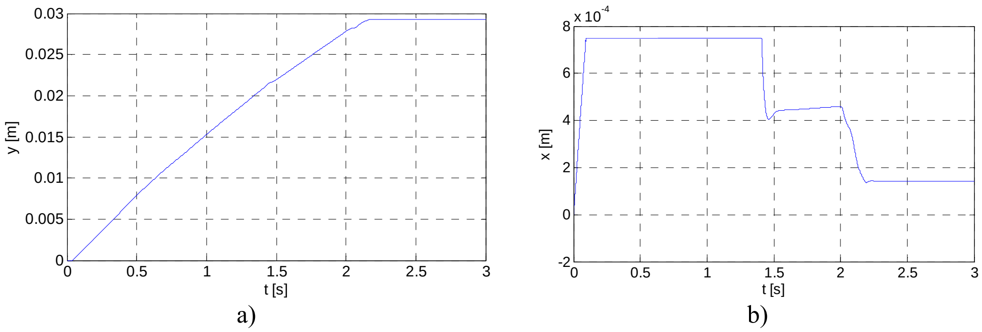



Fig. 8. Tipical Simulink integration result: a) servomechanism response for a ramp input of 75% from the nominal one; b) servo valve spool stroke evolution during the transient.

The usual hydrostatic powertrains are controlled during the start sequence by ramp inputs applied first to the servopump and than to the servomotor. From this point of view, the above motor servomechanism response is a good one. The same model indicates a very low overall damping for sudden inputs. This poor dynamic can be easily checked by a model built in AMESIM simulation language [5].

## 4. Modeling and simulation with AMESIM

A high degree of modeling efficiency can be obtained by the "compressed" components available in the libraries of AMESIM (fig. 9):

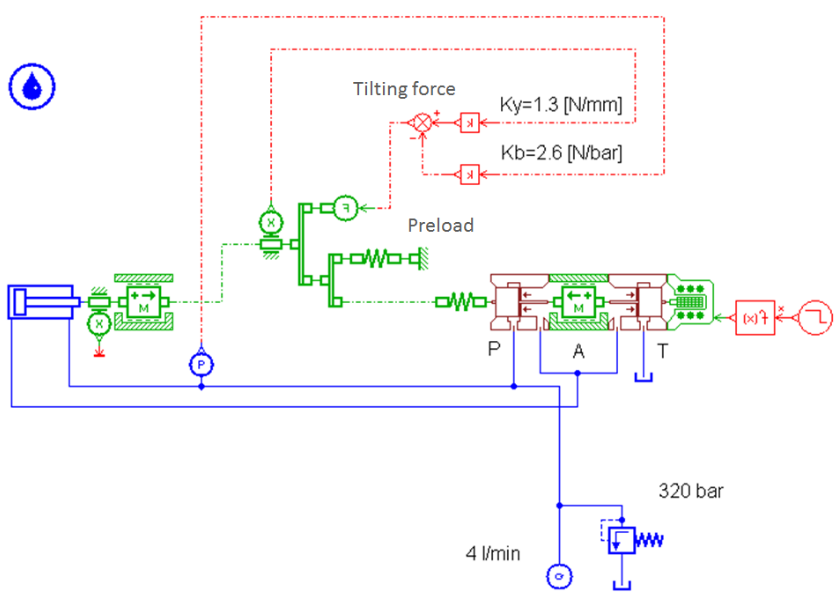
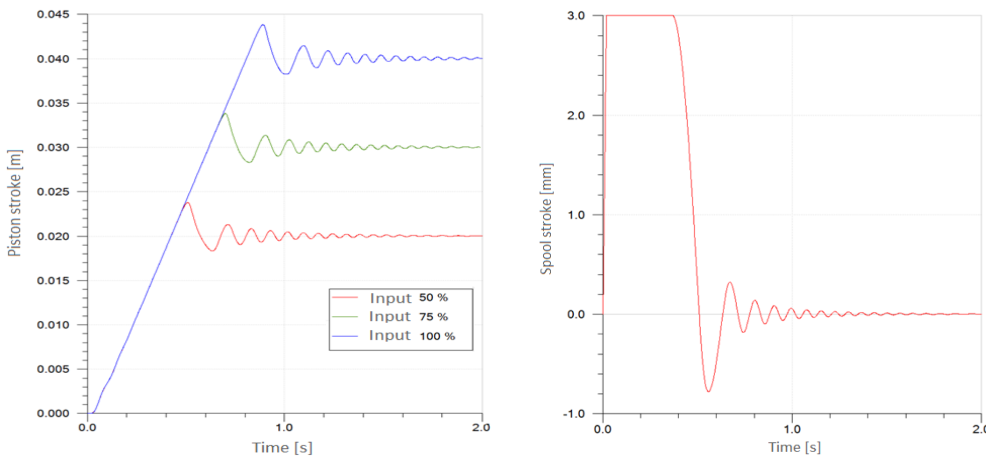


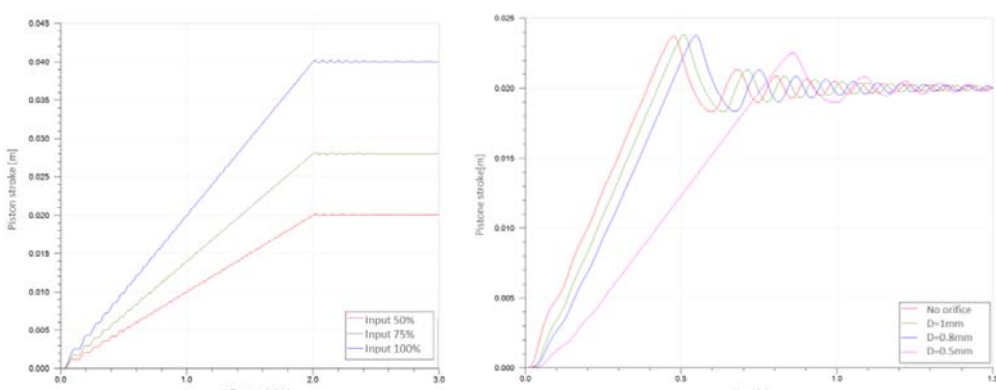

Fig. 9. AMESIM numerical simulation network of the electro-hydraulic servomechanism.

The step input response of the servomechanism (fig. 10) shows a very low overall damping factor. The ramp input response of the displacement control system (fig. 11a) shows a very low lag. The same result is supplied by the simulation in Simulink. The difference of about 0.2 s of the overall time response comes from the slightly different values of the steady-state constant of the solenoid, $K_m$, considered in the two models.




a) b)

Fig. 10. AMESIM simulation results: a) full step input current response ; b) typical spool stroke evolution during the transient generated by a step input current.




a) b)

Fig. 11. AMESIM results for a 2 s full ramp input current : a) control piston stroke variation; b) influence of a sharp edge metering orifice inserted in the supply line on the step input response.

The attempt to increase the control system overall damping leads to a very small metering orifice diameter, according to the small control chamber volume, and to a small force needed for positioning the cylinder barrel (fig.11b).

## 5. Conclusions

The main gain of the research presented in this paper is the promotion of modeling and simulation as preliminary tools for the design of any control system with uncertain physical parameters. The technical experience gain in high scientific fields like aerospace or automotive engineering is included step by step into modeling and simulation languages, in order to increase the speed of the design process in various new domains.

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