

# Effects of the Variation of Torque Motor Parameters on Servovalve Performance

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An electrohydraulic servovalve is an essential item of fluid power servomechanism where fast response, high power output and working fidelity are necessary. Based on detailed and experimentally verified mathematical model of two-stage spool position mechanical feedback electrohydraulic servovalves sensitivity analysis has been performed. The effects of variation of few torque motor electromagnetic parameters (air-gap length (thickness) at null, residual magnetic flux density (magnetic inductivity) of permanent magnet and number of turns of each coil) on dynamic performance of B.31.210.12.1000.U2V PPT servovalve have been studied. Obtained results are in accordance with servovalve engineering design practice.

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**Keywords:** electrohydraulics, servovalves, torque motors, sensitivity analysis

## 0 INTRODUCTION

Two-stage spool position feedback electrohydraulic servovalves with the torque motor as an electromechanical converter and a flapper-nozzle valve as the first stage of hydraulic amplification are the most utilised servovalves in electrohydraulic control systems for the last forty years. In spite of their relatively long period of application, all phenomena associated with their operation are not completely quantitatively and qualitatively explained. This is due to facts that a relatively complex mathematical apparatus, coming from fundamental laws of electromagnetism, fluid mechanics and general mechanics, is needed for describing their operation. Beside, these mathematical expressions include many parameters that are very difficult to precisely quantify.

In this paper authors study the influence of torque motor electromagnetic parameters on servovalve behaviour. Many authors gave contributions to the explanation of electromagnetic nature of servovalve torque motors. Still Merritt [1] explained in detail the effects of some non-linearities on the servovalve behaviour including torque motor non-linearities (magnetic hysteresis and saturation). In the paper [2], Arafa and Rizk made a special review on torque caused by electromagnetic forces. A non-linear mathematical model based on physical quantities was developed in [3]. This model includes non-linear relations for the torque motor

dynamics. From the experimental data and FEA analysis performed in [4], Fussell et al. state that magnetic flux leakage must be considered in the lumped model for torque predictions. In [5] and [6], Urata analysed the torque motor dynamics in detail and influence of unequal air-gap length thickness in null in servo valve torque motors.

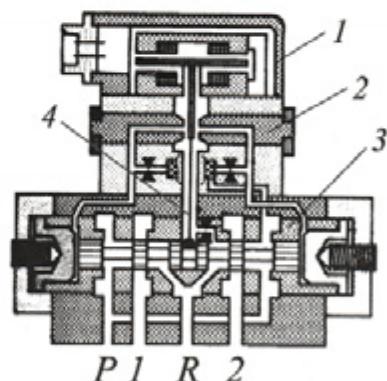


Fig. 1. Two-stage spool position feedback electrohydraulic servovalve (mechanical feedback) 1 – torque motor, 2 – first stage, 3 – second stage, 4 – feedback spring

## 1 THEORETICAL MODEL

Detailed theoretical model should be used in order to investigate the influence of torque motor electro-magnetic parameters on servovalve behaviour. It should include all physical phenomena and torque motor parameters that are

expected to be of influence on behaviour of analysed servovalves.

### 1.1 Torque Motor

When current is made to flow through torque motor coils, armature ends become polarised. A torque is thus produced on the armature which starts to move. It is common in practice, as it was proved in [7], to use the linear expression to calculate this torque:

$$T_e = K_i i + K_m \theta, \quad (1)$$

where values for  $K_i$  and  $K_m$  are obtained experimentally.

Taking into account expressions for electromagnetic forces in torque motor air gaps, where expressions for magnetic fluxes in air gaps (obtained using the first and the second Kirchhoff's laws for magnetic circuits) are implemented torque produced on armature can be calculated using (in the case of parallel coil connection):

$$T_e = \frac{r \cdot \mu_0 \cdot A_p}{x_{p0}^2} \left( M_m + N \frac{r \cdot \theta}{x_{p0}} i \right) \cdot \left[ \frac{M_m \frac{r \cdot \theta}{x_{p0}} + N(I+k_r)i}{I - \left( \frac{r \cdot \theta}{x_{p0}} \right)^2 + k_r} \right]^2, \quad (2)$$

where permanent magnet magnetomotive force is:

$$M_m = \frac{l_m B_{rm}}{\mu_m} \frac{l}{k_m}. \quad (3)$$

Linearising Eq. 2 about the null position ( $i=0$  and  $\theta=0$ ) one can write:

$$K_i = \frac{Nr \mu_0 A_p M_m}{x_{p0}^2 (I+k_r)}, \quad (4)$$

$$K_m = \frac{r^2 \mu_0 A_p M_m^2}{x_{p0}^3 (I+k_r)^2}. \quad (5)$$

Magnetic reluctance constant  $k_r$  considers influences of permanent magnet reluctance, the non-uniformity of a magnetic field through the permanent magnet volume, reduction of the magnetic field in the permanent magnet due to influences of end poles and the leakage flux in air gaps, as it was described in [5]. Following

expressions should be used for its calculation:

$$k_r = \frac{R_{mm}}{R_{mp0}} \frac{l}{k_l}, \quad (6)$$

where

$$R_{mm} = \frac{l_m}{A_m \mu_m} \frac{l}{k_m}, \quad (7)$$

$$R_{mp0} = \frac{x_{p0}}{\mu_0 A_p}. \quad (8)$$

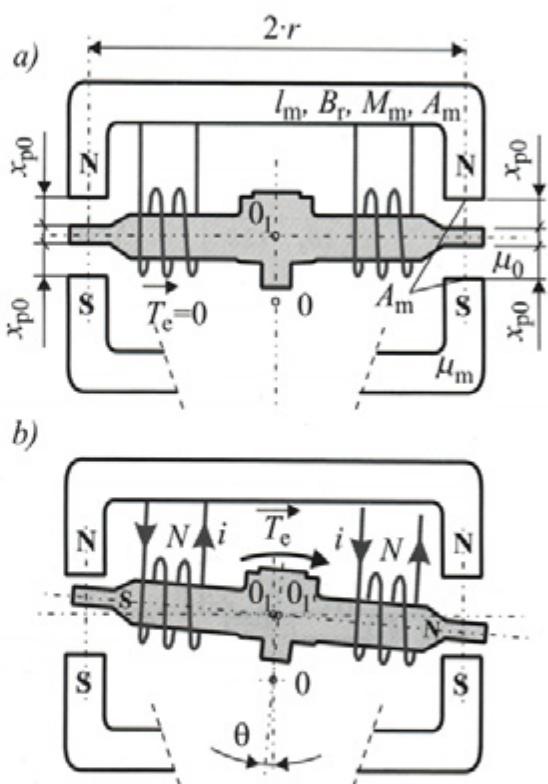


Fig. 2. Calculation of torque caused by electromagnetic forces in the torque motor: a) neutral, b) current runs through coils

### 1.2 Other Parts of the Servovalve

Only final equations of the model are presented in this paper for the reference. Presented equations are related to two-stage electrohydraulic servovalves with mechanical feedback. Detailed explanation of formulae can be found in [8] AND [9].

As long as the flapper does not hit a nozzle, armature dynamics can be described with:

$$T_e = J_a \ddot{\theta} + k_v \dot{\theta} + T_i + F_h l_a + T_f, \quad (9)$$

where:

$$l_a = l_{fl} - \frac{l_i}{2}. \quad (10)$$

The torque caused by flexure tube deformation can be written as:

$$T_i = \frac{B_i}{l_i} \theta. \quad (11)$$

Flow force on the flapper is given by:

$$F_h = (p_l - p_r) \frac{d_n^2 \pi}{4} + 8\pi \cdot \quad (12)$$

$$\left[ K_u^2 (x_0 - x)^2 (p_l - p_d) - K_w^2 (x_0 + x)^2 (p_r - p_d) \right]$$

where flapper tip displacement is:

$$x = \theta l_a - F_h \left( \frac{l_{fl}^3}{3B_{fl}} + \frac{l_i^3}{12B_i} \right). \quad (13)$$

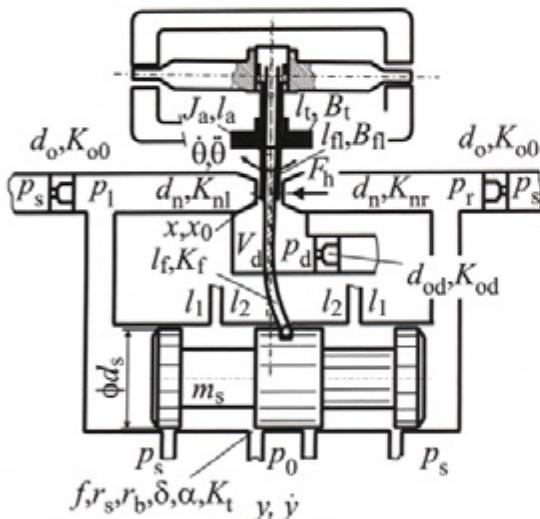


Fig. 3. Schematic of the servovalve with the parameters of its other parts

The torque caused by feedback spring deformation can be calculated with:

$$T_f = F_f l_f, \quad (14)$$

where the appropriate force is:

$$F_f = \begin{cases} 0 & |y + y_{ff}| < z \\ K_f [y + y_{ff} - z \operatorname{sgn}(y + y_{ff})] & |y + y_{ff}| \geq z \end{cases}. \quad (15)$$

where:

$$y_{ff} = x + l_f \varphi, \quad (16)$$

$$\varphi = \theta - \frac{F_h l_a^2}{2B_{fl}}. \quad (17)$$

Pressures in flapper-nozzle valve can be obtained using:

$$p_l = \frac{a^2 p_s + b^2 p_d}{a^2 + b^2} + \frac{d^2 (b^2 - a^2)}{(a^2 + b^2)^2} - \frac{2abd}{(a^2 + b^2)^2} \sqrt{(a^2 + b^2)(p_s - p_d) - d^2} \quad (18)$$

$$p_r = \frac{a^2 p_s + c^2 p_d}{a^2 + c^2} + \frac{d^2 (c^2 - a^2)}{(a^2 + c^2)^2} + \frac{2acd}{(a^2 + c^2)^2} \sqrt{(a^2 + c^2)(p_s - p_d) - d^2} \quad (19)$$

$$[(x_0 - x)\sqrt{p_l - p_d} + (x_0 + x)\sqrt{p_r - p_d}] K_{nd} d_n \cdot \pi \sqrt{\frac{2}{\rho}} = K_{od} \frac{d_n^2 \pi}{4} \sqrt{\frac{2}{\rho}} \sqrt{p_d - p_0} + \frac{V_d}{\beta} \dot{p}_d \quad (20)$$

where:

$$a = K_{o0} \frac{d_n^2 \pi}{4} \sqrt{\frac{2}{\rho}}, \quad (21)$$

$$b = K_{o0} d_n \pi (x_0 - x) \sqrt{\frac{2}{\rho}}, \quad (22)$$

$$c = K_{o0} d_n \pi (x_0 + x) \sqrt{\frac{2}{\rho}}, \quad (23)$$

$$d = \frac{d_n^2 \pi}{4} \dot{y}. \quad (24)$$

Algorithm for the calculation of flow coefficients can be found in [8].

The differential equation of spool motion can be written as:

$$\Delta p \frac{d_n^2 \pi}{4} = m_s \ddot{y} + k_n \dot{y} + F_c + F_a + F_f, \quad (25)$$

where:

$$\Delta p = p_l - p_r. \quad (26)$$

Dry friction force can be presented as:

$$F_c = \begin{cases} F_{cn} \operatorname{sgn}(\dot{y}), & \dot{y} \neq 0, \\ Z, & |Z| < F_{c0}, \\ F_{c0} \operatorname{sgn}(Z), & |Z| \geq F_{c0}, \end{cases} \quad (27)$$

where:

$$Z = \Delta p \frac{d_s^2 \pi}{4} - F_a - F_f. \quad (28)$$

In the case of the null lap condition for all four symmetrical and matched spool orifices of no-load servovalve, the following expression can be used for the axial component of the flow force:

$$F_a = 2 \frac{K_t^2}{K_c} f y_i \cos \alpha (p_s + p_0) + \\ + (l_2 - l_1) K_t \sqrt{2 \rho (p_s - p_0)} f \dot{y} \quad (29)$$

where the real value of spool valve opening is:

$$y_i = \sqrt{(\delta + r_s + r_b)^2 + (|y| + r_s + r_b)^2} - (r_s + r_b). \quad (30)$$

Numerical data from Dong and Ueno are used for the calculation of jet angle, contraction coefficient and flow coefficient [8].

The volumetric flow through the servovalve can be mathematically formulated as:

$$Q_{sv} = (K_t f y_i \sqrt{\frac{2}{\rho}} \sqrt{\frac{p_s - p_0}{2}} - Q_l) \operatorname{sgn}(y). \quad (31)$$

where the internal leakage flow can be calculated from:

$$\text{when } |y| < \left(\frac{q}{s}\right)^2 Q_l : \quad (32-a)$$

$$q \cdot Q_l^2 = \frac{p_s - p_0}{2} \quad (32-a)$$

$$\text{when } \left(\frac{q}{s}\right)^2 \cdot Q_l \leq |y| < y_{dl} : \quad (32-b)$$

$$s \cdot |y|^{1/2} \cdot Q_l^{3/2} = \frac{p_s - p_0}{2} \quad (32-b)$$

$$\text{and when } |y| \geq y_{dl} : \quad (32-c)$$

$$s y_{dl}^{1/2} Q_l^{3/2} + w(|y| - y_{dl}) Q_l = \frac{p_s - p_0}{2} \quad (32-c)$$

where

$$q = \frac{0.5 \rho}{K_t^2 f^2 (\sqrt{(\delta + r_s + r_b)^2 + (r_s + r_b)^2} - r_s - r_b)^2}, \quad (33)$$

$$s = \frac{23(\rho \eta)^{1/2}}{f^{3/2} \delta^{5/2}}, \quad (34)$$

$$w = \frac{12 \eta}{f \delta^3}, \quad (35)$$

$$y_{dl} = \frac{0.075 \delta Q_l}{\pi d_s v}. \quad (36)$$

## 2 EXPERIMENTAL VERIFICATION

The presented model was numerically solved and verified with experimental results for no-load servovalve B.31.210.12.1000.U2V - manufactured by PPT- Trstenik, Serbia (Fig. 4 and Fig. 5).

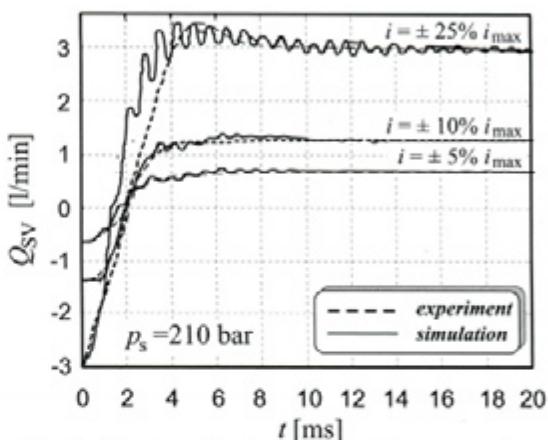


Fig. 4. The transient response of the servovalve (square periodic input current signal - 10 Hz frequency)

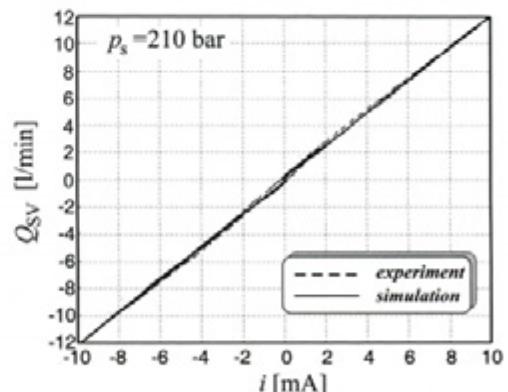


Fig. 5. The static flow characteristic of the servovalve

Parameter values of the servovalve are presented in nomenclature (in brackets). The manufacturer of permanent magnet provided values of parameters related to permanent magnet except its geometrical parameters, whereas values of constants related to permanent magnet and the

torque motor were calculated according to the procedure described in [5]. The friction coefficients were estimated to match the appropriate experimental transient responses. Values for other parameters were determined with a direct measurement on the disassembled servovalve. The equilibrium distance from the flapper tip to nozzles is an exception from this rule. It was calculated according to the assemblage requirement, which demands that pressures in both nozzles should be 70 bar when the supply pressure is 140 bar and when the flapper is in the zero position.

Static and dynamic experimental analyses of the servovalve were performed on the standard servovalve testing equipment, property of PPT (static - MOOG - Plotterstand D046-030, dynamic - MOOG - Frequenzgang - Prüfstand D046032).

The model satisfactorily agrees with the experiment and it can be used for the sensitivity analysis.

### 3 SENSITIVITY ANALYSIS

Torque motor parameters relate to its electromagnetic nature are length of each air gap at null  $x_{p0}$ , permanent magnet cross sectional area  $A_m$ , torque motor gap area  $A_p$ , magnetic flux density of permanent magnet  $B_r$ , permanent magnet length  $l_m$ , number of turns of each coil  $N$ , distance from armature pivot to the centre of permanent magnet pole face  $r$  and permanent magnet permeability  $\mu_m$ .

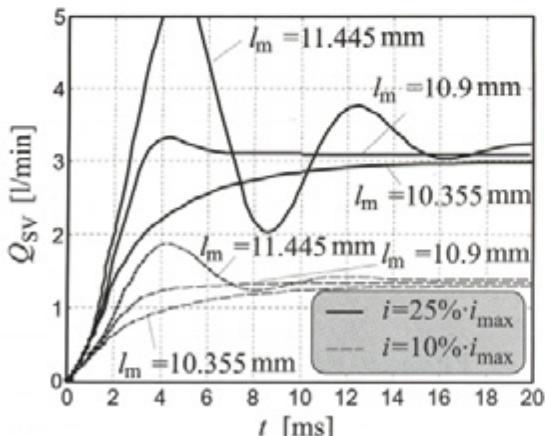


Fig. 6. The effect of the variation of permanent magnet length on servovalve performance

In order to analyse the influence of torque motor parameters on servovalve characteristics, the transient responses for step input current signal (from 0 to steady regimes determined with two input current signals  $i - 10\%$  and  $25\%$  of the rated  $i_{max}$ ) were numerically modelled. Three different values for the analysed parameters were taken in numerical modelling: value of the analysed servovalve, one higher and one smaller value. Results of the modelling are shown in Fig. 6 to 12.

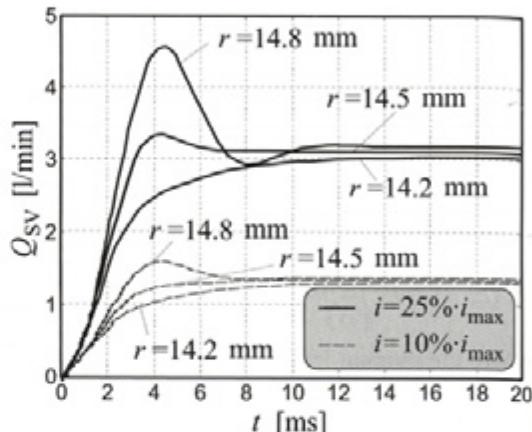


Fig. 7. The effect of the variation of distance from armature pivot to the centre of permanent magnet pole face on servovalve performance

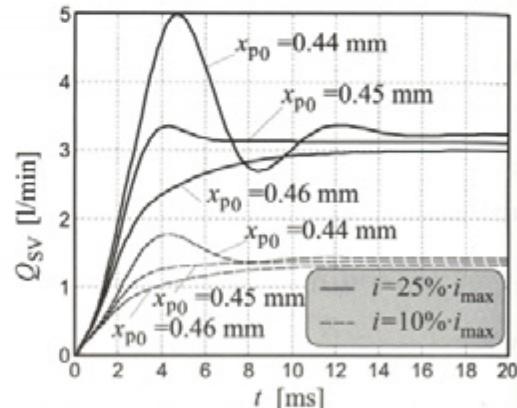


Fig. 8. The effect of the variation of air-gap length at null on servovalve performance

It can be noticed on diagrams that the increase (decrease) of  $x_{p0}$  generates the same effects as the decrease (increase) of other analysed parameters.

Analysing the influence of torque motor geometric parameters one can conclude that

length parameters ( $l_m$ ,  $r$ ,  $x_{p0}$ ) have greater influence on transient response than areas ( $A_m$ ,  $A_p$ ). Variation of  $A_m$  has bigger influence than variation of  $A_p$ .

It is especially important to analyse the effects of variation of  $x_{p0}$ ,  $B_{rm}$  and  $N$  on servovalve performance. These three parameters can be easily modified when a designer wants to change the characteristics of an existing servovalve.

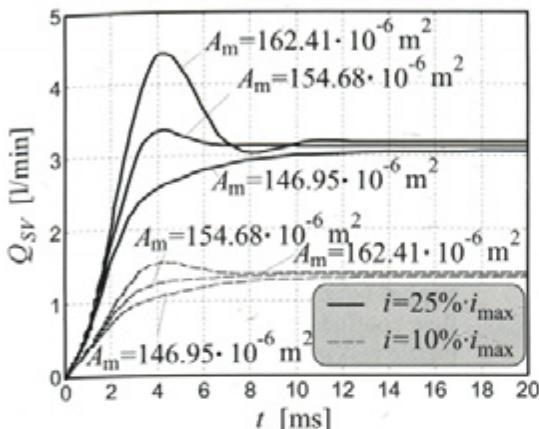


Fig. 9. The effect of the variation of permanent magnet cross sectional area at zero on servovalve performance

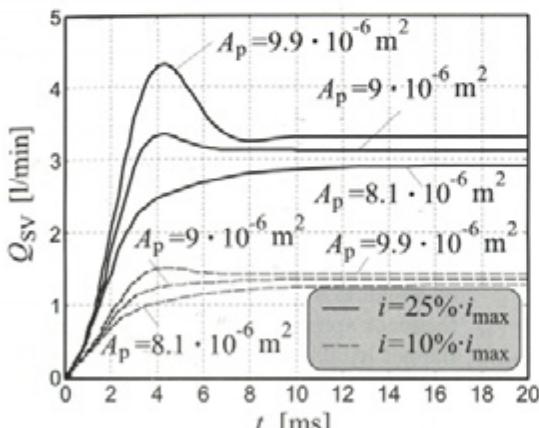


Fig. 10. The effect of the variation of torque motor gap area on servovalve performance

The change of air-gap length at null  $x_{p0}$  has the greatest influence on the quality of transient response and steady values of the servovalve flow (Fig. 8.). Small reduction of  $x_{p0}$  value (by 2 %) induce the decrease of rise time and existence (for  $i=10\% \cdot i_{max}$ ) i.e. the increase of overshoot (for

$i=25\% \cdot i_{max}$ ) and finally higher the stationary value of servovalve flow. If the  $x_{p0}$  has higher value then the transient response becomes more sluggish and stationary value of servovalve flow is being reduced.

With smaller changes for the same percentage variation of parameter value, similar trend can be noticed in Fig. 11 where the value of residual magnetic flux density is changed.

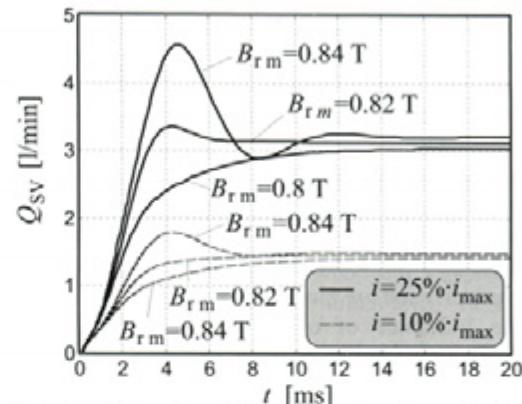


Fig. 11. The effect of the variation of residual magnetic flux density on servovalve performance

The change of number of turns of each coil  $N$  does not have influence on the quality of transient response. It only has small influence on the steady value of servovalve flow.

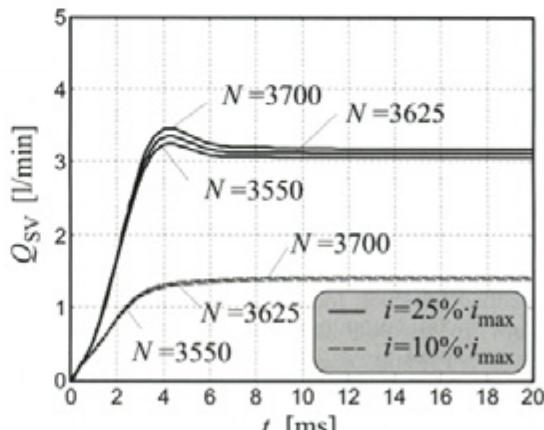


Fig. 12. The effect of the variation of number of turns of each coil on servovalve performance

### 3 CONCLUSION

Standard engineering practice considers that if servovalve designers want to redesign

existing or design a new servo valve they must modify some properties on physical object and then experimentally verify the results of this action. The model presented in the paper primarily gives a useful design tool, since all its parameters are physical quantities. When servo valve designers have such developed model that includes phenomena and parameters of influence and can predict their performance in a wide range of expected regimes, they can simply redesign existing servovalves without physical modification. They can numerically model all possible situations and choose results that are appropriate for their needs.

Using the detailed servo valve mathematical model authors analysed the influence of torque motor electromagnetic parameters. The analysis showed that the biggest influence on servo valve dynamic characteristics has variation of air-gap length at zero, although the influence of other parameters can not be neglected. Similar analysis can be performed for other servo valve parameters.

#### 4 NOMENCLATURE

$A_m$	permanent magnet cross sectional area ( $154.68 \cdot 10^{-6}$ )	[m <sup>2</sup> ]	$i_{\max}$	rate input current ( $10 \cdot 10^{-3}$ )	[A]
$A_p$	torque motor gap area ( $9 \cdot 10^{-6}$ )	[m <sup>2</sup> ]	$J_a$	armature assembly moment of inertia with respect of rotational centre ( $1.68 \cdot 10^{-7}$ )	[kgm <sup>2</sup> ]
$B_{fl}$	flapper bending rigidity (0.0256)	[Nm <sup>2</sup> ]	$K_c$	contraction coefficient of spool orifice	[-]
$B_t$	flexure tube rigidity (0.0302)	[Nm <sup>2</sup> ]	$K_f$	feedback spring stiffness (1962)	[N/m]
$B_{rm}$	magnetic flux density of permanent magnet	[T]	$K_i$	torque motor gain	[Nm/A]
$d_n$	nozzle diameter ( $0.28 \cdot 10^{-3}$ )	[m]	$k_l$	constant of leakage flux in air gap (0.27)	[-]
$d_o$	left (right) constant orifice diameter ( $0.18 \cdot 10^{-3}$ )	[m]	$k_m$	constant caused by the non-uniformity of magnetic field in permanent magnet (2.42)	[-]
$d_{od}$	drain constant orifice diameter ( $0.4 \cdot 10^{-3}$ )	[m]	$K_m$	torque motor electromagnetic spring constant	[Nm]
$d_s$	spool diameter ( $4.62 \cdot 10^{-3}$ )	[m]	$k_n$	spool viscous friction coefficient (3800)	[kg/s]
$f$	spool valve area gradient ( $4.8 \cdot 10^{-3}$ )	[m]	$K_{nl}$	flow coefficient of the left flapper-nozzle restriction	[-]
$F_a$	axial component of flow force on spool	[N]	$K_{nr}$	flow coefficient of the right flapper-nozzle restriction	[-]
$F_c$	dry friction force on spool	[N]	$K_{n0}$	flow coefficient of the flapper-nozzle restrictions at zero	[-]
$F_{c0}$	initial dry friction force on spool ( $3.2 \cdot 10^{-3}$ )	[N]	$K_{od}$	flow coefficient of the drain constant orifice	[-]
$F_{cn}$	nominal dry friction force on spool ( $2.2 \cdot 10^{-3}$ )	[N]	$K_{o0}$	flow coefficient of the left/right constant orifice at zero	[-]
$F_f$	force caused by mutual action of feedback spring and spool	[N]	$k_r$	magnetic reluctance constant (0.465)	[-]
$F_h$	flow force on flapper	[N]	$K_t$	flow coefficient of spool orifice	[-]
$i$	total current	[A]	$k_v$	armature assembly viscous friction coefficient ( $4 \cdot 10^{-4}$ )	[Nms]
			$l_1$	axial length between supply and actuator ports ( $5.4 \cdot 10^{-3}$ )	[m]
			$l_2$	axial length between reservoir and actuator ports ( $9.2 \cdot 10^{-3}$ )	[m]
			$l_a$	distance from the nozzle axis of symmetry to the axis of rotation of armature assembly ( $8.75 \cdot 10^{-3}$ )	[m]
			$l_f$	feedback spring length ( $13.3 \cdot 10^{-3}$ )	[m]
			$l_{fl}$	flapper length ( $13 \cdot 10^{-3}$ )	[m]
			$l_m$	permanent magnet length ( $10.9 \cdot 10^{-3}$ )	[m]
			$l_t$	flexure tube length ( $8.5 \cdot 10^{-3}$ )	[m]
			$M_m$	permanent magnet magnetomotive force	[A]
			$m_s$	spool mass ( $3.1 \cdot 10^{-3}$ )	[kg]
			$N$	number of turns of each coil (3625)	[-]
			$p_0$	atmospheric (reservoir) pressure ( $10^5$ )	[Pa]
			$p_d$	drain orifice inlet pressure	[Pa]

$p_l$	left nozzle pressure	[Pa]	$\nu$	fluid kinematic viscosity ( $14 \cdot 10^{-6}$ ) [m $^2$ /s]
$p_r$	right nozzle pressure	[Pa]	$\theta$	armature deflection [rad]
$p_s$	supply pressure ( $210 \cdot 10^5$ )	[Pa]	$\rho$	fluid density (871) [kg/m $^3$ ]
$r$	distance from armature pivot to the centre of permanent magnet pole face ( $14.5 \cdot 10^{-3}$ )	[m]		
$r_b$	radius of bushing control edge ( $10 \cdot 10^{-6}$ )	[m]	[1]	Merritt, H. (1967) Hydraulic Control Systems, John Wiley & Sons, New York.
$R_{mm}$	permanent magnet reluctance	[H $^{-1}$ ]	[2]	Arafa, H. A. and Rizk, M. (1987) Identification and Modelling of Some Electrohydraulic Servo-Valve Non-Linearities. Proceedings of the Institution of Mechanical Engineers, Part C: Mechanical Engineering Science, vol. 201, no. 2, pp. 137-144.
$R_{m0}$	reluctance of each air gap at the null	[H $^{-1}$ ]		
$r_s$	radius of spool control edge ( $10 \cdot 10^{-6}$ )	[m]	[3]	Wang, D., Dolid, R., Donath, M., Albright, J. (1995) Development and Verification of a Two-Stage Flow Control Servovalve Model, ASME The Fluid Power and Systems Technology Division (Publication), FPST, vol. 2, pp. 121-129.
$T_e$	torque caused by electromagnetic forces	[Nm]		
$T_f$	torque caused by feedback spring deformation	[Nm]	[4]	Fussell, B., Darwin, J., Prina S. (1999) Servovalve Torque Motor Analysis, Electrical Insulation Conference/Electrical Manufacturing & Coil Winding Expo, 1999.
$T_t$	torque caused by flexure tube deformation	[Nm]		
$V_d$	drain compartment volume ( $8.2 \cdot 10^{-6}$ )	[m $^3$ ]	[5]	Urata, E. (2000) Study of Magnetic Circuits for Servovalve Torque Motors, Bath Workshop on Power Transmission and Control (PTMC'00), Bath, UK, pp. 269-282.
$Q_l$	leakage flow through spool orifices	[m $^3$ /s]		
$Q_{sv}$	servovalve volumetric flow	[m $^3$ /s]	[6]	Urata, E. (2007) Influence of unequal air-gap thickness in servo valve torque motors, Proceedings of the I MECH E Part C Journal of Mechanical Engineering Science, Volume 221, Number 11, 2007, pp. 1287-1297(11).
$x$	flapper tip displacement	[m]		
$x_0$	distance from flapper tip to each nozzle at null ( $38.5 \cdot 10^{-6}$ )	[m]	[7]	Gordić, D., Šušteršić, V., Jovičić, N., Babić, M. (2004) Analysis of servovalve torque motor behaviour, Proceedings of the 29. NSS HIPNEF 2004, Vrnjačka Banja, Serbia, May, 19.-21., 2004, pp. 187-192. (in Serbian).
$x_n$	distance from flapper tip to nozzle	[m]		
$x_{p0}$	length of each air gap at null ( $0.45 \cdot 10^{-3}$ )	[m]	[8]	Gordić, D., Babić, M., Jovičić, N. (2004) The modelling of a spool position feedback servovalve, International Journal of Fluid Power, vol. 5, no. 1, March 2004.
$y$	spool displacement	[m]		
$y_f$	feedback spring end deflection	[m]		
$y_{ff}$	fictive feedback spring ball displacement if there is no spool	[m]	[9]	Gordić, D. (2002) The Analyses of the Two-Stage Electrohydraulic Servovalves With the Spool Position Feedback, PhD thesis, University of Kragujevac, Serbia (In Serbian).
$y_i$	spool orifice opening	[m]		
$y_d$	large transition length	[m]		
$z$	half the feedback spring ball clearance ( $2 \cdot 10^{-6}$ )	[m]		
$\alpha$	jet angle in spool orifice	[°]		
$\beta$	fluid compressibility ( $1.9 \cdot 10^{-9}$ )	[Pa]		
$\delta$	spool in bushing radial clearance ( $4 \cdot 10^{-6}$ )	[m]		
$\eta$	fluid dynamic viscosity (0.012)	[Pa·s]		
$\varphi$	flapper free end inclination	[rad]		
$\mu_0$	magnetic permeability of vacuum	[H/m]		
$\mu_m$	permanent magnet permeability (5)	[H/m]		

## 5 REFERENCES