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CONSIDERATIONS ON THE ELECTRO-HYDRAULIC SYSTEMS FOR THE STABILIZATION OF THE FEED SPEED OF LONGITUDINAL TURNING

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Abstract. The article focuses on a comparative analysis of two different structures of electro-hydraulic systems for the stabilization of the feed force of longitudinal turning based on numerical simulation experiments. The way the structure of these control systems can or cannot be used with compensated dynamics hydraulic motors is studied depending on their behaviour in the presence of cutting forces acting as motor resistance force. The article concludes with a series of recommendations with regard to the structure of electro-hydraulic systems for the stabilization of feed speed and to the field of application of compensated dynamics hydraulic motors, respectively.

Keywords: control, cutting process, electro-hydraulic systems

1. Introduction

Characteristic of the electro-hydraulic systems for the stabilization of the feed speed of turning is the fact that the feed is obtained by means of linear hydraulic motors. The basic structure of such a system has a linear hydraulic motor **MHL**, controlled by an electro-hydraulic servovalve **SVEH** ensures the longitudinal feed of the cutting tool **S**, figure 1. Therefore, the hydraulic motor speed represents the vey longitudinal feed speed of the cutting tool $v_M = w_L$, and the feed force represents the motor resistance force $F_a = F_{RM}$.

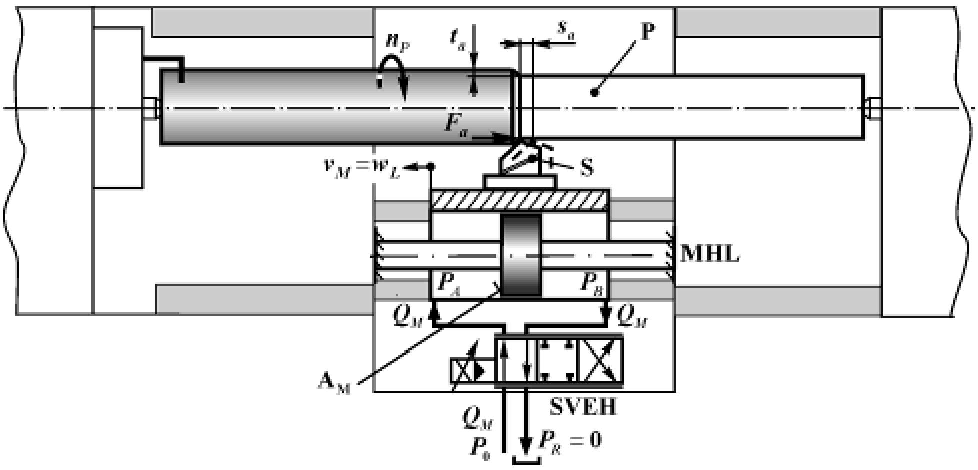

Figure 1. Electro-hydraulic system for the feed speed

2. The mathematical model and the numerical simulation of the stabilizing systems of the feed speed of longitudinal turning with uncompensated linear hydraulic motor

The system for the stabilization of the feed speed of longitudinal turning based on a symmetrical linear hydraulic motor consists of the following subsystems [1, 4, 7]:

- The electronic comparator with the following complex equation:

$$u_{\varepsilon v}(s) = u_v(s) - u_{Rv}(s) \qquad (1)$$

- The electronic regulator is of $(PD)_1$ type:

$$i(s) = K_R \cdot \frac{T_R \cdot s + 1}{T_1 \cdot s + 1} \cdot u_{\varepsilon v}(s) \qquad (2)$$

- The **A0 + A0** electro-hydraulic spool servovalve behaves like a $(P)_1$ element being supplied at a constant pressure:

$$Q_M(s) = \frac{K_{SV}}{T_{SV} \cdot s + 1} \cdot i(s) \qquad (3)$$

• The leak free, symmetric linear motor **MHL** is described by the system of equations below:

$$\begin{cases} P_L(s) = \frac{1}{T_c \cdot s} \cdot \left[Q_v(s) - A_M \cdot v_M(s) \right] \\ v_M(s) = \frac{k_M}{T_M \cdot s + 1} \cdot \left[A_M \cdot P_L(s) - F_a(s) \right] \end{cases} \qquad (4)$$

• The position transducer is represented by an element $(P)_1$:

$$u_{Rv}(s) = \frac{K_{Tr}}{T_{Tr} \cdot s + 1} \cdot v_M(s) \qquad (5)$$

• The cutting process is represented by the linearized expression of the feed force, the linearization introducing very small approximation errors and only in transient processes [2]:

$$F_a(s) = F_{a0}(s) + K_{Fw} \cdot v_M(s) + \Delta F_{ap}(s) \qquad (6)$$

Within the relations above the following notations:

u_v – reference tension of the feed speed stabilizing circuit,
$u_{\varepsilon v}$ – error tension,
u_{Rv} – reaction tension detected by the speed transducer,
K_R and T_R – transducer amplification and time constant,
T_1 – transducer delay constant,
K_{SV} and T_{SV} – amplification and time constant of the servovalve,
P_L – load differential pressure,
$Q_M = Q_v$ oil flow rate delivered by the servovalve to the motor,
F_a – feed force as motor resistance force,
A_M– motor useful area,
k_M, T_M and T_c – amplification and time constants of the motor partial transfer functions,
k_{Tr} and T_{Tr} – amplification and time constant of the speed transducer,
F_{aL0} – steady state feed force around which linearization is made,
ΔF_{ap} – disturbance component of the feed force introduced by variation Δt_a of the cutting depth and by variation ΔHB of the hardness of the processed component part,
K_{Fw} is the parameter of the viscous component of the feed force.

Based on the block diagram obtained through the representation of the equation given above and based on the numerical values of the dynamics parameters of the system we obtain the simulation diagram presented in figure 2 developed in Simulink – Matlab [5]. The numerical values of the dynamic parameters of the system are as follows:

$T_{sv} = 0.003$ [s], $K_{SV} = 2.24 \cdot 10^{-4}$ [m^3/s·A],
$A_M = 1.76 \cdot 10^{-3}$ [m^2], $k_M = 4.17 \cdot 10^{-3}$ [m^3/s·A],
$T_c = 3.3 \cdot 10^{-13}$ [s], $K_{FW} = 2.57 \cdot 10^{-4}$ [n·s/m],
$K_R = 0.2$ [A/V], $T_R = 10^{-3}$ [s], $T_1 = 0.1$ [s],
$K_{Tr} = 20$ [V·s/m], $T_{Tr} = 0.001$ [s].

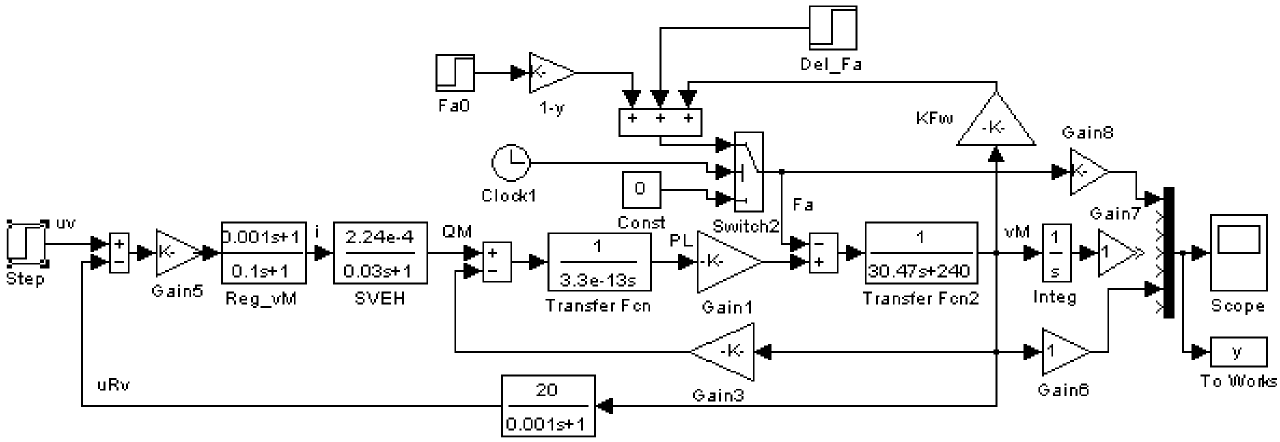

Figure 2. Simulation diagram (system with **MHL**)

The numerical simulation was performed for a input step $u_v = 1$ V and figure 3 graphically presents the results of the numerical simulation – feed force variation F_a and the evolution of the feed speed v_M. On the onset of the motion without any cutting, the cutting force is null and the response of the system is characteristic of an over damping system of second order. After approximately 0.25 seconds, the feed speed sets at the value of v_M = 8.5 mm/s. Cutting starts at time 0.3 s, whereas the disturbance occurs at 0.4 s, both occurring as step variations. Both transient processes of the indicial response have the same shape and extremely brief durations of 0.03 s. The engagement of the lathe tool determines a sudden increase of the cutting force immediately followed by a drop in the feed speed. Furthermore, the viscous component of the cutting force determines a drop in the cutting force. The reciprocating interaction between the two components will eventually lead to the stabilization of their values. The control circuit ensures a constant value of the cutting speed in steady state irrespective of the occurring disturbances.

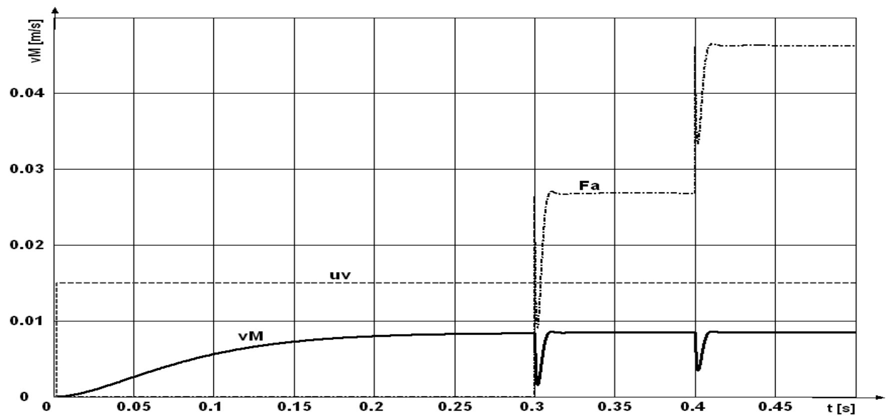

Figure 3. Indicial response (system with **MHL**)

3. Mathematical model and numerical simulation of the feed speed stabilization during longitudinal turning with compensated dynamics hydraulic motor

The only difference from the case studied above is the fact that here we have a compensated dynamics hydraulic motor fitted with a mechanical-hydraulic network for dynamics compensation. The most frequently used mechanical-hydraulic network for dynamics compensation is the one consisting of two pneumatic-hydraulic accumulators connected to hydraulic resistances for filtering the pressure oscillations. Therefore, the structure of the system simulation diagram (figure 4) gives the mathematical model of the dynamics compensating network **RcM** associated to the equation of the network connecting point [1, 4]:

$$\begin{cases} Q_M(s) = Q_v(s) - Q_{Rc}(s) \\ Q_{Rc}(s) = K_{Rc} \cdot \dfrac{T_{Rc} \cdot s + 1}{T_{Rc} \cdot s} \cdot P_L(s) \end{cases} \tag{7}$$

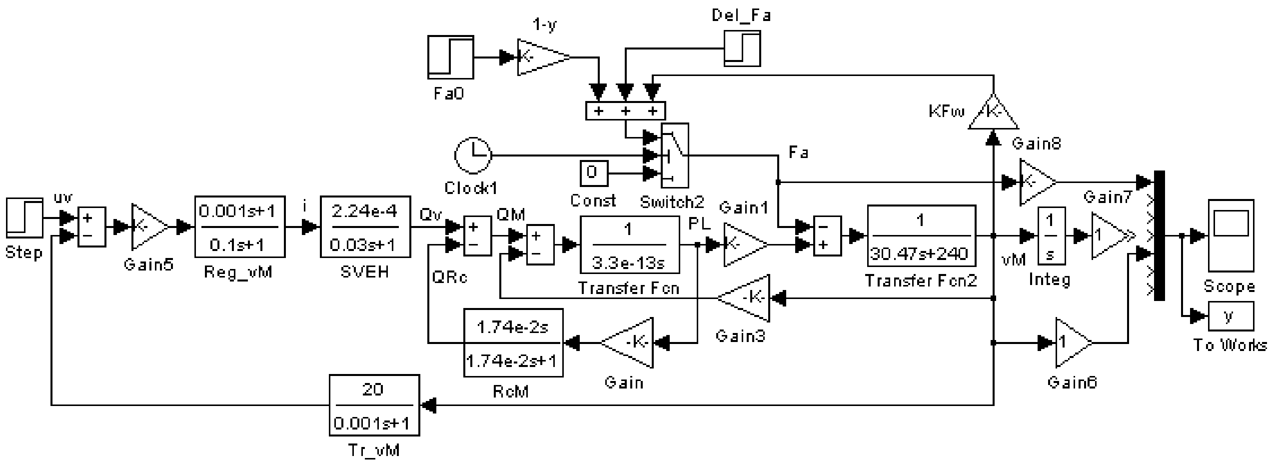

Figure 4. Simulation diagram (system with **MHL + RcM**)

The flow rate of the hydraulic motor, Q_M, results as the difference between the flow from the servovalve, Q_v, and the flow rate Q_{Rc} in the dynamics compensation network. The dynamics compensation network behaves like a high-pass filters which is characterized by amplification K_{Rc} and time constant T_{Rc}.

The numerical simulation was performed with the following values of the dynamics parameters of the dynamics compensation network:

$K_{Rc} = 5.4 \cdot 10^{-9}$ [m^5/N·s] and $T_{Rc} = 1.74 \cdot 10^{-2}$ [s].

Figure 5 shows the evolution of speed and feed force at a control step of u_v =0.505 V. It can be noticed that that the start is characteristic of a dampened ordinal system two and the stabilization of the idle running speed onsets only after 2 seconds. Once the cutting is engaged, i.e. after the occurrence of the disturbance, the stabilization of the cutting force and of the feed speed occurs after approximately 1 second with a null speed stabilization error.

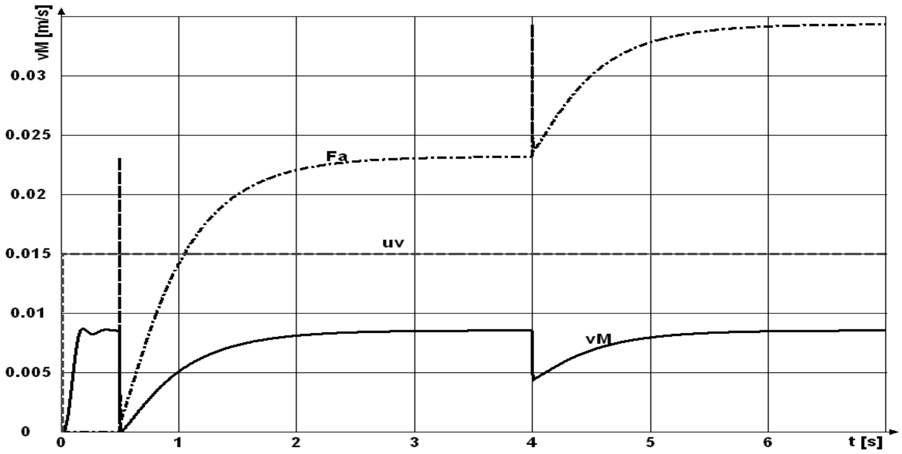

Figure 5. Indicial response (system with **MHL + RcM**)

4.Conclusions

The comparative analysis of the two variants of servo systems for the stabilization of the feed speed of longitudinal turning, with or without mechanic-hydraulic dynamics compensation network, allows us to make the following remarks:

- Both systems have null steady state errors of the feed speed which stabilizes after the transient processes;
- The uncompensated hydraulic motor system responds much faster but it also features higher drops in the feed speed in transient processes.
- On start-up, the presence of the compensated hydraulic motor induces its own oscillations in the transient process and, furthermore, the system becomes extremely slow.
- In conclusion, to ensure highly dynamic performances, the feed speed stabilizing system shall be designed with an uncompensated hydraulic motor.
- The compensated hydraulic motor system is recommended to be used to dampen the pressure oscillations in the motor when their frequency exceeds the filtering capacity of the mechanic-hydraulic dynamics compensation network.

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