The Braking Process at the Stroke end of Linear Hydraulic Motors

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Abstract: – The paper analyzes the transitory phenomena, which encounters at the braking at the stroke end of the hydraulic actuators from the structure of the hydraulic driving of the machines and industrial robotics. In order to point out, by numerical simulation, the transitory phenomena, which encounters at the braking at the stroke end of the hydraulic actuators, the mathematical model of the hydraulic actuator, in braking status, is first deduced. As a conclusion of the numerical simulation the motor response is analyzed and presented, for the particular case, when the braking is carried out by a hydraulic resistance.

Key-Words: braking of hydraulic motor, numerical simulation

1 Introduction
The structure of many hydraulic driving of machine tools and industrial robots contain linear hydraulic motors, with simple or double driving. In order to prevent mechanical shocks occurred upon the piston hitting the side covers of the motor, it is recommended to provide a braking system at the stroke end. The most frequent solutions [4, 5, 6] are the use of braking throttles placed on the side cover and braking through the play between the cover bore and braking dowel which continues the motor piston.

The braking of the hydraulic motor at the stroke end is accompanied by pulsation of the pressures of its two chambers. Consequently, by knowing transient phenomena related to the hydraulic motor braking is necessary to establish the size of the braking throttle able to provide the fast loss of pressure pulsations, a small overshoot of this pressure and adjustment of the interval required for braking.

2 Mathematical modeling of the hydraulic motor braking process
The linear mathematical model of the hydraulic motor under braking will highlight, as output size, the pressure \( P_B \) of the motors counter motor chamber. This will be done by accepting the following work hypothesis:

- the linear hydraulic motor is considered symmetrical,
- the adjustment of the hydraulic motor speed is done through the throttle \( \text{Dr} \) (Fig.1) located at the exit,
- the force resistant to motor, prior and during braking, is null \( F_{RM} = 0 \):
- the clearance between cover and braking dowel is null,

- the braking is exclusively provided by the braking resistance \( \text{DrF} \),
- to simplify the problem, both the motor speed adjustment throttle and the braking throttle are considered as being represented by flow laminar resistances,
- internal and external motor leakages are considered null \( \alpha_M = 0 \):
- The motor is considered fed at constant pressure \( P_0 \approx \text{const} \),
- The return pressure is considered constant \( P_{R0} \approx \text{const} \).
- The reference position of the motor is considered the position immediately near the point where the braking dowel which continues the piston enters the side cover. At this position the hydraulic motor speed is constant. In Fig.1.a the hydraulic motor is represented in the stage before braking, and in Fig.1.b the motor is represented in braking regime.

We will consider that prior the start of the braking process the motor is steady-state. Under these circumstances the flow through the braking throttle is null, and the flow crossing through the adjustment throttle is equal to the flow exiting from the hydraulic motor. Consequently the piston steady-state equilibrium and continuity of the flow to the motor and through the adjustment throttle:

\[
\begin{align*}
    c_M v_M^* &= A_M P_0 - A_M P_B^* \\
    Q_A^* &= Q_c^* = A_M v_M^* \\
    Q_c^* &= Y_{hd} (P_B^* - P_{R0})
\end{align*}
\]
Allow the calculation of the pressure steady-state values from the chamber of the counter motor and motor speed:

\[ P_B^* = \frac{A_M^2 P_0 + c_{RM}Y_{Hd} P_{R0}}{A_M^2 + c_{RM}Y_{Hd}} \]  

(2)

\[ v_M^* = \frac{A_M}{c_{RM}}(P_0 - P_B^*) \]  

(3)

In this relations \( A_M \) is the useful area of the hydraulic motor, \( Q_A \) - is the flow entering the motor chamber of the hydraulic motor, \( c_{RM} \) - low viscous friction coefficient of the hydraulic motor, and \( Y_{Hd} = \frac{1}{R_{Hd}} \),

where \( R_{Hd} \) is the hydraulic resistance of the adjustment throttle. The asterisk marks the steady-state values of the variables in question.

When the braking dowel enters the suitable cavity of the motor side cover, starts the braking process, which is characterized by the following equations:

- the dynamic equilibrium equation of the motor:

\[ A_M P_0 - a_M P_B - a P_F = m_{RM} v_M + c_M v_M \]  

(4)

- the equation of displacement of the motor piston,

\[ z_M = \int v_M \, dt \]  

(5)

- expression of calculation of the flow rate which enters the hydraulic motor,

\[ Q_A \approx A_M v_M \]  

(6)

- continuity equations of the flow associated to the counter motor chamber and cavity where braking dowel enters, respectively the knot upstream the adjustment dowel:

\[ \begin{align*}
-Q_B &= -a_M v_M + C_{HB} P_B^* \\
-Q_F &= -a v_M + C_{HF} P_F^* \\
Q_e &= Q_B + Q_F
\end{align*} \]  

(7)

- expression of the laminar flow crossing through the adjustment dowel:

\[ Q_F = Y_{HF} (P_F - P_B) \]  

(8)

where \( a_M \) is the area of the counter motor chamber at braking stage, \( a \) - area of braking cavity, \( C_{HB} \) - hydraulic capacity of the counter motor chamber, \( C_{HF} \) - hydraulic capacity of the braking chamber, \( Y_{HF} = \frac{1}{R_{HF}} \), where \( R_{HF} \) is the hydraulic resistance of the braking throttle, \( Q_F \) - laminar flow through the braking throttle.

With these considerations, the dynamic, in terms of time, of the hydraulic motor is described by the following equations:

\[ \begin{align*}
m_{RM} v_M + c_M v_M - A_M P_0 - a_M P_B - a P_F \\
\dot{z}_M &= \int v_M \, dt \\
\dot{Q}_A &= A_M v_M \\
C_{HF} P_F &= (a_M + a) v_M - C_{HB} P_B - Q_e \\
C_{HFR_B} + Y_{HF} P_B &= a_M v_M + Y_{HF} P_F - Q_e
\end{align*} \]  

(9)

When speaking of the linear hydraulic motor system – adjustment throttle, to the above equations the expression of adjustment throttle located on the exit is added:

\[ Q_e = Y_{Hd} (P_B - P_{R0}) \]  

(10)

One can note that the dynamic of the linear hydraulic motor - adjustment throttle, in the field of time, is described as a system of six equations. At the same time, in the structural diagram of Fig.2a, one can note the flow feedback transmitted to the throttle over the hydraulic motor.
By replacement of the relation (12) in the system of equations (11), the number of variables and implicitly the number of equations describing the system dynamics is decreased:

\[
\begin{align*}
MHL + DrF & \quad P_0 \\
\text{DrF} & \quad Q_a \\
\text{MHL} & \quad v_M
\end{align*}
\]

According to the structural diagram of Fig.2.b, drawn up based on these equations, the previous flow feedback was replaced by the return pressure feedback \(P_{R0}\).

By applying the Laplace transformation, the dynamics, in complex, of the system hydraulic motor – speed adjustment throttle is described by the following system of equations:

\[
\begin{align*}
\dot{v}_M(s) &= \frac{K_7}{T_3s+1} \left[ P_0(s) - K_5 P_B(s) - K_6 P_F(s) \right] \\
z_M(s) &= -\frac{1}{s} v_M(s) \\
Q_A(s) &= K_8 v_M(s) \\
\dot{P}_F(s) &= \frac{1}{T_4s+1} v_M(s) - \frac{T_5s+1}{T_6s} P_0(s) + \frac{1}{T_6s} P_{R0}(s) \\
\dot{P}_B(s) &= \frac{1}{T_5s+1} \left[ K_9 v_M(s) + K_{10} P_F(s) + K_{11} P_{R0}(s) \right] \\
\dot{Q}_c(s) &= K_{12} (P_B(s) - P_{R0}(s))
\end{align*}
\]

where:

\[
K_5 = A_M ; \quad K_6 = \frac{a_M}{A_M} ; \quad K_7 = \frac{\alpha}{c_M} ;
\]

\[
K_8 = \frac{A_M}{Y_{HF} + Y_{Hd}} ; \quad K_9 = \frac{Y_{HF} + Y_{Hd}}{Y_{HF} + Y_{Hd}} ; \quad K_{10} = \frac{Y_{HF} + Y_{Hd}}{Y_{HF} + Y_{Hd}} ; \quad K_{11} = \frac{Y_{Hd}}{Y_{HF} + Y_{Hd}} ; \quad K_{12} = \frac{Y_{Hd}}{Y_{HF} + Y_{Hd}} ;
\]

The block-diagram disclosed by Fig.3 reveals the internal connections within the system. One can note that in the case of the hydraulic motor controlled by the throttle placed at exit, the input size in the hydraulic motor is represented by the feeding pressure \(P_0\) of the hydraulic motor, as a difference from the case the motor is controlled by servovalve and the input size is represented by the motor input flow. Moreover the speed \(v_M\) influences both directly and indirectly the counter-motor pressure \(P_B\), which leads to the emergence of two direct ways of transmission of the information.

**3 Simulation of the braking transient process**

Based on the above and on the mathematical model of the linear hydraulic motor for the stage prior the braking [2] the simulation diagram was done, using Simulink language within the programming environment Matlab. Fig.4 discloses the evolution, prior and during braking, of pressure \(P_B\) from the motor’s counter motor chamber, of speed \(v_M\) of the hydraulic piston and its position \(z_M\).

The following constructive - functional parameters of the hydraulic motor were considered: \(D_M = 62 \ \text{[mm]}\), \(d_M = 40 \ \text{[mm]}\), \(L_M = 400 \ \text{[mm]}\), \(L_4 = 0.355 \ \text{[mm]}\), \(L_B = 0.050 \ \text{[mm]}\), \(L_F = 0.040 \ \text{[mm]}\), \(\alpha_{ill} = 0\), \(D_{dr} = 1 \ \text{[mm]}\), \(D_{df} = 0.5 \ \text{[mm]}\), \(P_0 = 50 \ \text{[bar]}\), \(P_{R0} = 3 \ \text{[bar]}\).

Upon the supply of hydraulic motor with pressure at \(P_0 = 50 \ \text{[bar]}\) and in presence of counter pressure \(P_{R0} = 3 \ \text{[bar]}\) and in the absence of the force resistant to motor, its speed will stabilize at the value \(v_M = 0.225 \ \text{[m/s]}\).
This is the speed when the braking process begins, the speed of input of the braking dowel entering the side cover. The transient braking process lasts approximately 0.1 seconds and is characterized by the following parameters:

- the speed represents negative overshoot of approximately 0.2 [m/s],
- the piston makes small absorbed oscillations, with initial amplitude of $\approx 0.3$ [mm].
- the transient braking process is continued by the steady-state displacement at the speed of $\approx 0.04$ [m/s]; at this speed the piston will hit the side cover and will stop.

### 4 Conclusions

Knowing the transient braking regime at the stroke end of hydraulic motors and its connection with the mathematical model supplies important details related to the establishment of the size of the braking throttle, able to provide the requested dynamic and steady parameters: duration and braking speed, pressure overshooting, speed and force of the piston hitting the side cover, etc.

### References: