Simulation Approach of an Automation System for Heavy Equipment Feed Drives

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Abstract: - This paper presents a new type of hydraulic servo-driven system. This device, driven by an electrical stepping motor can be used in the feed chain of heavy machine-tools, such as heavy milling machines. The authors developed in this work a theoretical model of the system, based upon transfer functions. In order to test the dynamic performances, a stability analysis was performed by mean of Bode diagram and root locus diagram.

Key- Words:- feed drive, functional stability, root locus, servosystem, synthesis, transfer function

1 Introduction

A large class of heavy equipment, such as heavy milling machines, main drives from flexible manufacturing systems or any other machine tool, which requires a good dynamic behaviour and an accurate positioning capability, are using automatic hydraulic driving systems. These types of drives have to satisfy specific conditions imposed by the machine dynamic, but also they have to satisfy conditions which are specific to the automation systems, meaning that they have to be stable, fast and accurate.

The problem of increasing performances of the machine tools is often solved by improving the control system. The force system is usually considered as an invariant "initial condition", or it has been proved that the most important possibilities of increasing the machine performances are contained in this system. Such automatic driving systems are constructed as linear hydraulic amplifiers with control elements as stepping micro-motors.

2 The system structure

A structural scheme of a driving servo system for a heavy milling machine feed chain is presented in figure 1 (only for one axis). The slide 1 is moved by the hydraulic linear motor 2. Both feed velocity and displacement are controlled by means of the tracking system 5, composed from a stepping micro-motor 6, a hydraulic distributor and a feedback mechanism 4. The CNC equipment 3 is also included in the system.

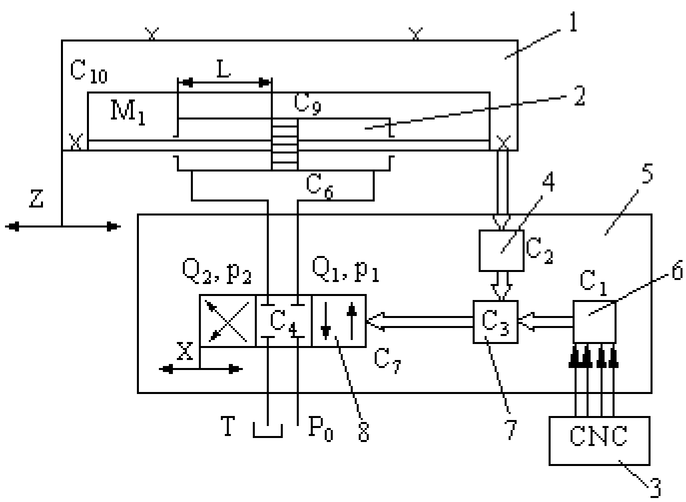

Fig.1 Scheme of a driving servo system of a heavy milling machine.

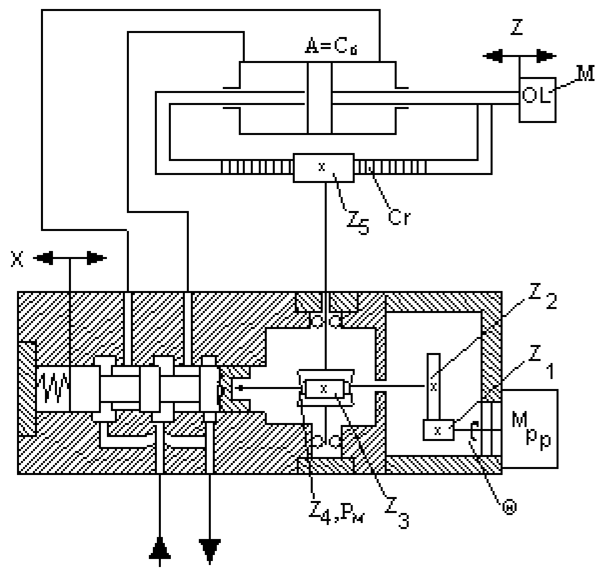

Fig.2 Structure of the tracking system

These kinds of servo systems present some advantages: the use of the hydraulic cylinder ensures low inertia of the moving parts and fast dynamic answer; the lack of intermediary transmission parts ensures a simple and robust construction; the structure of the tracking system (figure 2) as an autonomous system, with low dimensions allows a simple installation of it on the existing machines, and subsequently an upgrade of these machines. At the basis of designing the above-presented driving system are both the methods for calculating the productivity of driving systems and the specific designing methods for electro-hydraulic tracking systems.

The first method allows the designer to find the relationship between the characteristics of the driving system and its productivity. The second method allows establishing the dependency between the characteristics of the whole system and the parameters of its components, by means of transfer functions.

3 Researches regarding the system dynamics

The equations, which describe the behaviour of the system, are:

$$
\begin{aligned}
x &= \left(\theta - \frac{C_2 z}{C_1}\right) C_1 C_3 \\
Q &= C_4 x \\
Q &= C_6 \dot{z} + (C_7 + C_8)\,\Delta \dot{p} + C_9\,\Delta p \\
C_6\,\Delta p &= C_{10}\,\ddot{z} + C_{11}\,\dot{z} + C_{12}\,z
\end{aligned}
\tag{1}
$$

The open loop transfer function is:

$$
T_{D(s)} = \frac{K'_5 \cdot \dfrac{C_2}{C_1}}{K'_4 + K'_3 s + K'_2 s^2 + K'_1 s^3} \tag{2}
$$

where:

$$
\begin{aligned}
K'_1 &= C_{10}(C_7 + C_8) \\
K'_2 &= C_{11}(C_7 + C_8) + 2 C_9 C_{10} \\
K'_3 &= C_{12}(C_7 + C_8) + 2(C_6)^2 + 2 C_9 C_{11} \\
K'_4 &= 2 C_9 C_{12} \\
K'_5 &= 2 C_1 C_3 C_4 C_6
\end{aligned}
\tag{3}
$$

The meaning of the C_i coefficients is:

$C_1 = 1$;
$C_2 = (Z_4/Z_3)(1/\pi m_5 Z_5) = 1{,}8$ cm^{-1} (ratio of the feedback mechanism);
$C_3 = p_M$ [cm] (worm gear step – it will be used as parameter);
$C_4 = A_Q = 3\cdot10^4$ cm^2/s (flow gain);
$C_6 = 2A = 80$ cm^2 (piston surface);
$C_7 + C_8 = (4A\cdot L)/E_u = 0{,}01$ cm^5/daN;
$C_9 = 1$ cm^5/daN·s (loses due to the piston);
$C_{10} = M = 0{,}1$ daN·s^2/cm (inertial mass of the displaced object);
$C_{11} = 0{,}7$ daN·s/cm (viscous friction coefficient); $C_{12} = 0$;

The hydraulic environment is oil, with the following characteristics:
$E_u = 15000$ daN/cm^2, $\rho = 0{,}86\cdot0^{-6}$ daN·s^2/cm^4
With the above values, the values of K_i coefficients are:
$K_1 = 0{,}001$; $K_2 = 0.207$; $K_3 = 12801{,}4$; $K_4 = 0$; $K_5 = 4800000C_2$;
With these values the open loop transfer function $T_O(s)$ became:

$$T_D(s)=\frac{8640000\cdot C_2}{0.001s^3+0.207s^2+12801.4s} \quad (4)$$

We will use the worm step value as parameter in order to study the functional stability of the system by means of computer simulation using MATLAB & Simulink software.
A dual representation, including the root-locus and the Bode diagram of the open system will be presented. The root locus plot is used to analyse the open loop and shows the trajectories of the closed-loop poles when the gain varies from 0 to ∞. Three values for the gain (which is the worm step) will be taken into consideration: 0.1 cm, 0.2 cm and 0.4 cm. The root locus and Bode diagram for these values are presented in Figs.3, 4 and 5.

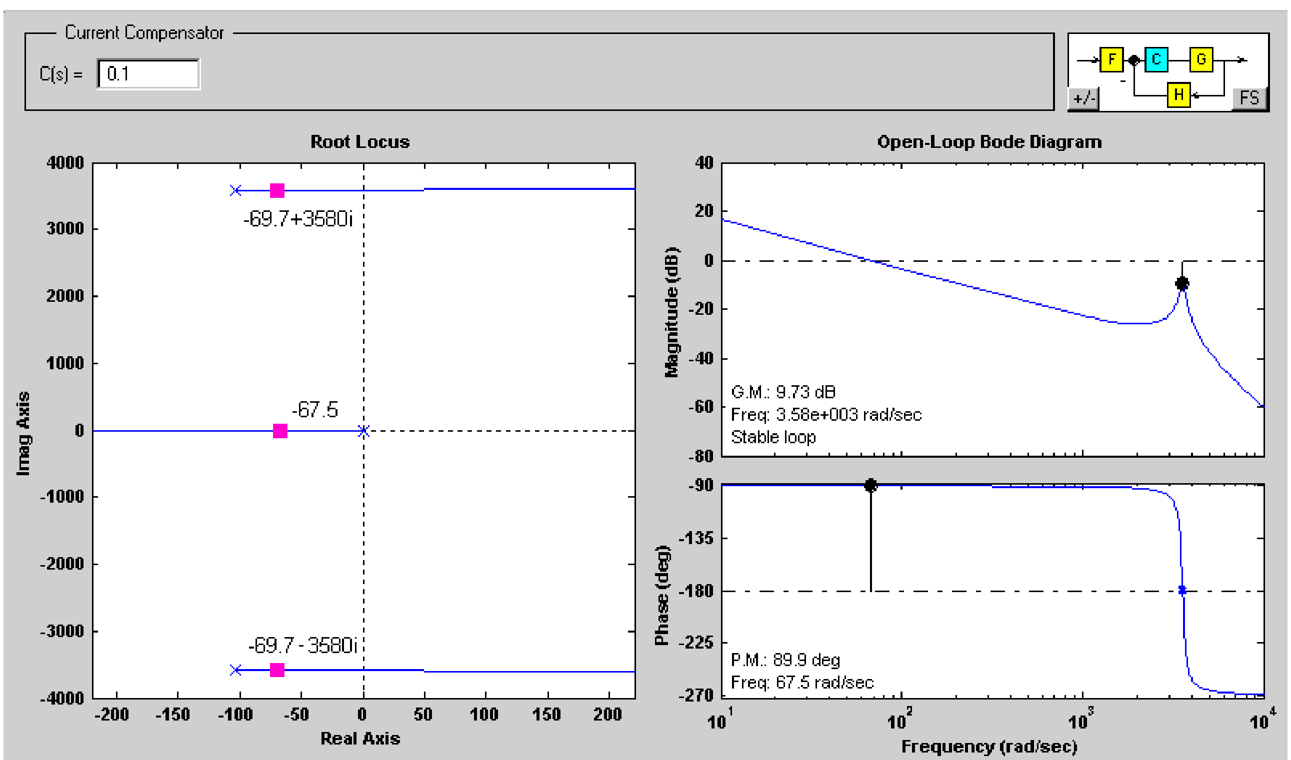

Fig.3

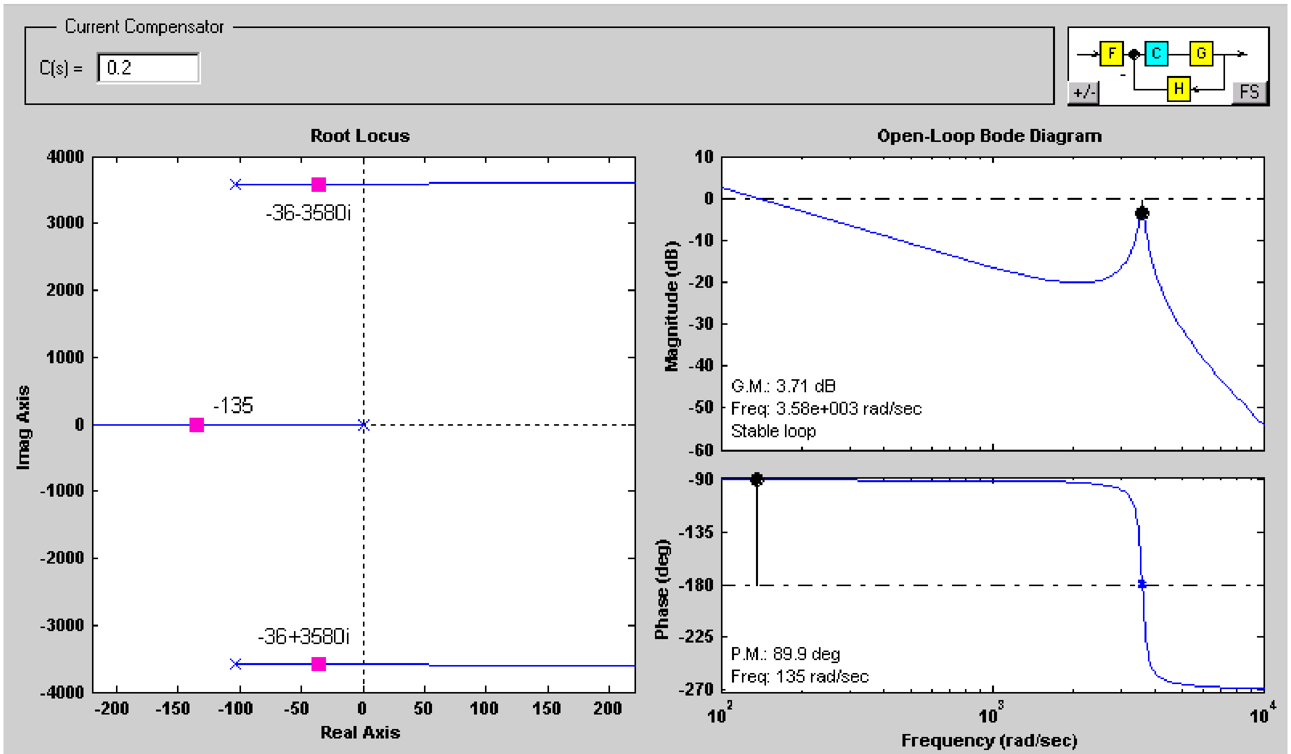
Current Compensator
C(s) = 0.2
F
C
G
H
+/-
FS
Root Locus
-36-3580i
-135
-36+3580i
Imag Axis
Real Axis
Open-Loop Bode Diagram
Magnitude (dB)
G.M.: 3.71 dB
Freq: 3.58e+003 rad/sec
Stable loop
Phase (deg)
P.M.: 89.9 deg
Freq: 135 rad/sec
Frequency (rad/sec)

Fig.4

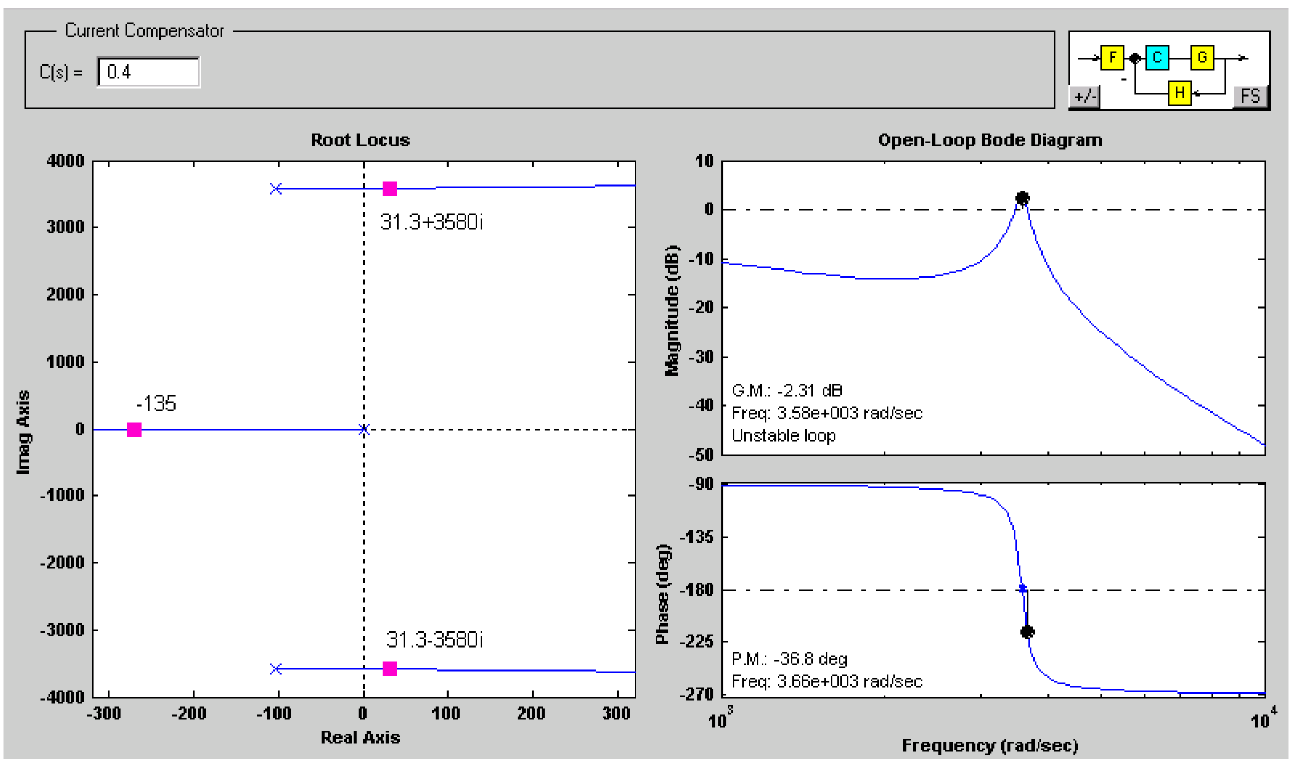
Current Compensator
C(s) = 0.4
F
C
G
H
+/-
FS
Root Locus
31.3+3580i
-135
31.3-3580i
Imag Axis
Real Axis
Open-Loop Bode Diagram
Magnitude (dB)
G.M.: -2.31 dB
Freq: 3.58e+003 rad/sec
Unstable loop
Phase (deg)
P.M.: -36.8 deg
Freq: 3.66e+003 rad/sec
Frequency (rad/sec)

Fig.5

From Figs.3-5 it can be observed that the system is stable for p_M = 0.1 cm and p_M = 0.2 cm and unstable for p_M = 0.4 cm. In the first and second case the gain margin (dB) and phase (degrees) margin from Bode diagram are positive (9.73 dB and 89,9 degrees, respectively 3.71 dB and 89,9 degrees). Furthermore, the closed-loop poles are situated in the left half-plane of the complex plane, both facts proving stability.
In the third case, p_M = 0.4 cm both the gain and phase margins are negative (-2.31 dB and –36.8 degrees) and only the real pole of the closed-loop is situated in the left half-plane, the complex poles are in the right one. These facts show that the closed-loop system is no more stable in this situation.
Further investigation have proven the fact that a value between p_M = 0.1 – 0.3 cm ensures the system's functional stability. Values under 0.1 cm are also ensuring stability but the manufacturing of a worm gear with this value of the step will became cumbersome and it's use is not suited for heavy equipment.

4 Conclusions

Feedback, one of the most fundamental processes existing in nature, is present in almost all dynamic systems. The theory of feedback control systems has been developed as an engineering discipline for analysing and designing practical control systems and other technological devices.
The objective of this paper was to present a comprehensive application of the theory of feedback control in solving the functional stability problem of a electro-hydraulic servo system.
The theoretical researches performed by means of open loop transfer function have shown the possibility to ensure the functional stability of the system by varying one of its constructive parameters.
The computer simulations performed during the researches helped the designers to find a range for the parameter p_M in which the closed loop system is stable, and the servo system may be used as driving system for the feed chains of heavy equipment.

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