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A Numerical Machine Tool Controller for Servomechanisms.

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Abstract- In industrial applications, when considering the performance of Computer Numerical Control (CNC) machines, it is worthy to note that the dimensional accuracies as well as surface finishes of parts produced by the machine tool rely strongly on the accuracy of the motion that each axis of the machine.The overall accuracy of the CNC machine tool is determined by the mechanical characteristics of the machine as well as the characteristics of the control system driving the individual tool. To maintain the quality of the finished product, it is necessary to optimise and sustain the level of accuracy of the machine tools so that defective parts can be prevented in manufacturing. A CNC Machine is programmed to travel along a predetermined path, and any deviation from the programmed path would constitute an error. The purpose of this project is to design a three-term (Proportional, Integral, and Derivative) digital controller to optimally improve the movement and accuracy of the control valve in anelectro-hydraulic servo-valve control system used in a numerical machine tool control and to accurately position the machine tool in the desired location on the workpiece. The three-term controller is designed to improve the phase margin lower than 0.498^o degree, less than 5 percent overshoot , a settling time below 0.0141 seconds and a rise time less than 5 seconds when subject to a unit step input, which reduces the damping ratio below 0.182.

Index Terms- Computer Numerical Control, PID Controller, hydraulic servo-valve control system, position a machine tool.

I. INTRODUCTION

Machining is basically removal of material, most often metal, from the workpiece, using one or more cutting tools to achieve the desired dimensions. There are different machining processes, such as, turning, milling, boring etc [6]. Normally, machining process is based on the relative motion between the workpiece and the tool. Generally, one of the two rotates at designated and generally high speed, causing the shearing of material (known as chips), from the workpiece [6]. In most modern industries, the performance of machines is determined by a Computer numerical control (CNC).

Computer numerical control could be defined as one in which the functions and motions of a machine tool are controlled by means of a prepared program containing coded alphanumeric data. CNC can control the motions of the workpiece or tool, the input parameters such as feed, depth of cut, speed, and the functions such as turning spindle on/off, turning coolant on/off [5].

A CNC machine is programmed to travel along a predetermined path, and any deviation of the actual path as compared to the programmed leads inaccuracies and errors. The basic function of a CNC machine is to provide automatic and precise motion control to its elements such work table, tool spindle etc [2]. Machine tools controlled by CNC are expected to perform within the specified limits of accuracy, especially for high precision operations[3]. If the two axes of a CNC machine are not perpendicular to one another, an oval path will result. In addition backlash in lead screws, axis straightness, pitch and yaw errors can all affect the accuracy of a machine tool[3]. Some CNC machines make use of servomechanism to achieve optimum performance.

A servo or a servomechanism is a control system which measures its own output and forces the output to quickly and accurately following a command signal. In this way, the effect of anomalies in the control device itself and in the load can be minimized as well as the influence of external disturbances[4]. The operation of an electro-hydraulic motion controller circuit is shown in figure 1.0a

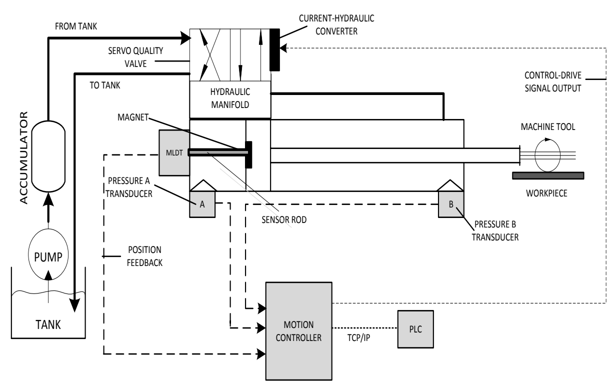

Figure 1.0a The electro-hydraulic CNC motion controller circuit

The essence of the electro-hydraulic motion controller circuit in figure 1.0a is to monitor the pressure in the Hydraulic Servo Valve [2]. A basic servo valve is one in which the control flow at constant load is proportional to the electrical input current. Flow from these servo valves will be influenced in varying degrees by changing load pressures [4]. Position control is used in this design to put the actuator in approximately the right position for applying a controlled amount of force. The electronic motion controller can be easily programmed into a PLC (using sequential or ladder logic) and can smoothly transition between controlling both pressure and position, depending on the control algorithm specified [2]. The motion controller in Figure 1a drives the hydraulic cylinder by sending analog signals to the proportional or servo-valve. The servo-valve is of high quality and is capable of making precise adjustments (sinusoidal or other waveforms) to pressure for controlling the cylinder's force or position. The accumulator stores hydraulic pressure to dampen pulses generated by the motor and also to ensure that consistent system supply pressure is available to operate the servo-valve during machine tool operation [4].

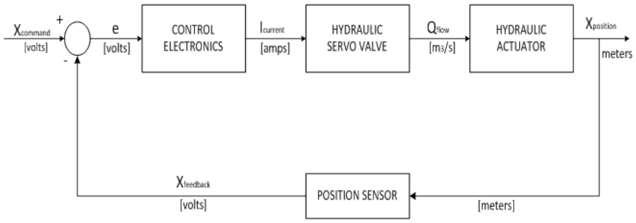

Figure 1.0b: Simplified diagram of the servo valve position control

Figure 1b is a simplified block diagram of the electro-hydraulic CNC motion controller circuit. The input command is a motion displacement which in turn moves the control valve spool resulting in the valve allowing hydraulic fluid flow into the actuator.

This research work focuses on the design of a digital controller that will significantly improve the operation or performance of the control valve and the accuracy in positioning a machine tool on a workpiece.

II. HYDRAULIC MACHINE TOOL POSITIONING SERVO TRANSFER FUNCTION.

In this design, a position servo which is made up of valve controlled cylinder in a constant pressure system will be used. The servo valve operating the machine tool is a 4-port valve with negligible dynamics and the cylinder is symmetric shown in figure 2.0.

In figure 2.0, the actuator or load position is measured by a position transducer, which converts the displacement of the piton X_p and produces an electric signal output (in volts) U_f. The piston position feedback gain factor, K_f is factored into the output signal. The servo amplifier compares the command signal U_c (in volts) with the feedback signal Uf and generate an error signal.

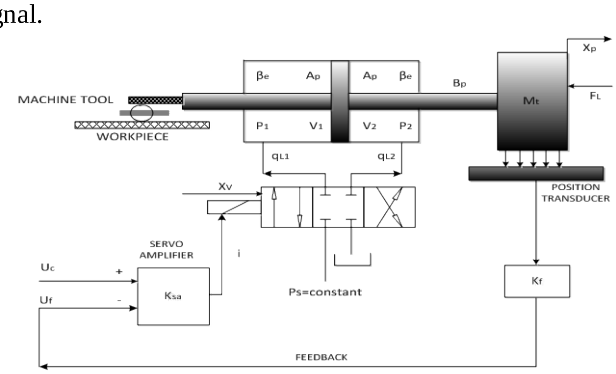

Figure 4.0: Schematic diagram of a complete position servo

The error generated by the servo difference amplifier signal is then combined with the gain factor *Ksa* to produce the output current signal *i*.

The amplifier signal is used to control the servo valve. The displacement of the servo valve *Xv* is a function of the output current signal, and determines the flow (*qL1*, *qL2*) of hydraulic fluid between the valve and the piton. The hydraulic pressure is supplied from a constant pressure source. The position servo carrying the machine tool is to be controlled by a computer control system and this design will be modeled with a series compensator to obtain the overall transfer function of the system, guided by the design specification.

The design of hydraulic machine tool positioning servo is carried out with parameters obtained from the simulation of position servo with mechanical springs to derive a steady state loop gain Kv = 10, and an amplitude margin of about 6 dB [7]. Table 2.0 below shows these parameter values that would be used in the design of the hydraulic servo position system:

Table 2.0: Simulation parameters

Parameter	*Description*	*Value*
A_p	*Piston Area*	*2,5·10-3 m²*
β_e	*effective bulk modulus*	*1,0X10⁹ Pa*
B_p	*viscous friction coeff. (cylinder)*	*0*
K_{ce}	*flow/pressure-coefficient of the valve and cylinder*	*1,0X10⁻¹¹ m⁵/Ns*
K_{qi}	*Internal flow gain (servo valve)*	*0,02 m³/As*
K_{sa}	*Servo amplifier gain*	*0,05 A/V*
V_t	*Total volume of cylinder*	*1,0·10-3 m³*
$K_1 = K_L$	*Spring constant*	*5,0·107 N/m*
K_f	*position feedback gain*	*25 V/m*
M_t	*Mass of piston and rod*	*1500 kg*

Using the parameters described above, we proceed to obtain the Open-loop transfer function of the hydraulic system, A_u using equation 1.5 where:

$$A_u = \frac{K_v}{s\left(\frac{s^2}{\omega_h^2} + 2\delta_h\frac{s}{\omega_h} + 1\right)} \quad \text{-------------------------} \quad 1.0$$

But the hydraulic eigen frequency, ω_h (rad/s) is given as:

$$\omega_h = \sqrt{\frac{4\beta_e A_p^2}{V_t M_t}} \quad \text{-----------------------------} \quad 1.2$$

Using parameters obtained from table 2.0: V_t=0.001m^3; M_p=M_t=1500kg; β_e=1.0 X 10^{9}N/m^2; A_p=2.5 X 10^{-3}m^2

Therefore,

$$\omega_h = \sqrt{\frac{4 \times 1.0 \times 10^9 \times (2.5 \times 10^{-3})^2}{0.001 \times 1500}} = 129 rad/s$$

ω_h =129rad/s and ω_h^2 =16.67X10^{3}rad/s

Also, the hydraulic damping factor is given as:

$$\delta_h = \frac{K_{ce}}{A_p}\sqrt{\frac{\beta_e M_t}{V_t}} \quad \text{-----------------------------} \quad 1.3$$

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Using the Parameters obtained from table 2.0 are:

$$\delta_h = \frac{1\times10^{-11}}{2.5\times10^{-3}}\sqrt{\frac{1\times10^{9}\times1500}{0.001}} = 0.155$$

$$\delta_h = 0.155$$

The hydraulic loop gain, k_v is calculated using equation 1.4:

$$K_v = \frac{K_f \times K_{qi} \times K_{sa}}{A_p} \quad \text{------------------------- 1.4}$$

Finally we parameters values input values obtained from the table, and we have:

$$K_v = \frac{25\times0.02\times0.05}{2.5\times10^{-3}} = 10$$

Then the open loop hydraulic system transfer function can now be calculated using equation 1.0:

$$A_u = \frac{10}{s\left(\frac{s^2}{\left(16.67\times10^3\right)^2} + 2\times0.155\,\frac{s}{129} + 1\right)}$$

$$A_u = \left(\frac{10}{3.6\times10^{-9}S^3 + 2.4\times10^{-3}S^2 + S}\right) \quad \text{---------------- 1.5}$$

The block diagram model of the controller and the hydraulic system is shown in Figure 3.0

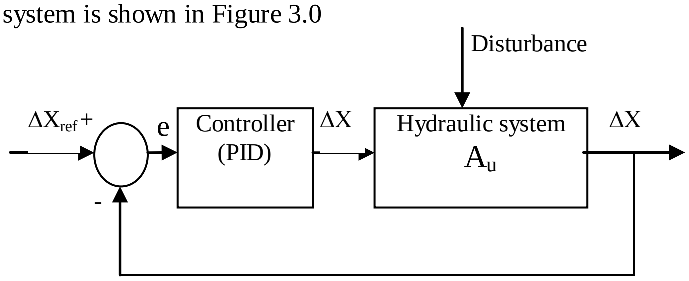

Figure 3.0: Block diagram model for the system design.

We have obtained the transfer function for the hydraulic part of the system in the s-domain. We will proceed next to design the complete motion controller.

III. DESIGN SPECIFICATION

a) Less than 5% overshoot to a unit step input.
b) Phase margin is 0.498° degree.
c) Settling time less than 0.2 seconds to a unit step input.
d) A damping ratio of 0.182.
e) Rise time of less than 5 seconds for a unit step input

IV. THE PID CONTROLLER TRANSFER FUNCTION.

The design of the control system would require the use of the three-term controller or Proportional, Integral, and Derivative (PID). Equation 1.6 is the transfer function of the PID controller.

$$K_P + \frac{K_I}{s} + K_D s = \frac{K_D s^2 + K_P s + K_I}{s}$$

where K_p is the proportional gain, K_i is the integral gain, and K_d is the derivative gain. From figure 3.0, the error e is the tracking error or the difference between the desired reference value (ΔX_{pref}) and the actual

V. THE RESPONSE OF THE SYSTEM TO PID CONTROLLER.

The various step responses to the PID controller resulting from the simulation of the system with equation 1.5 using MATLAB 2014 /Simulink is given as follows:

A. Initial Open-Loop Step Response

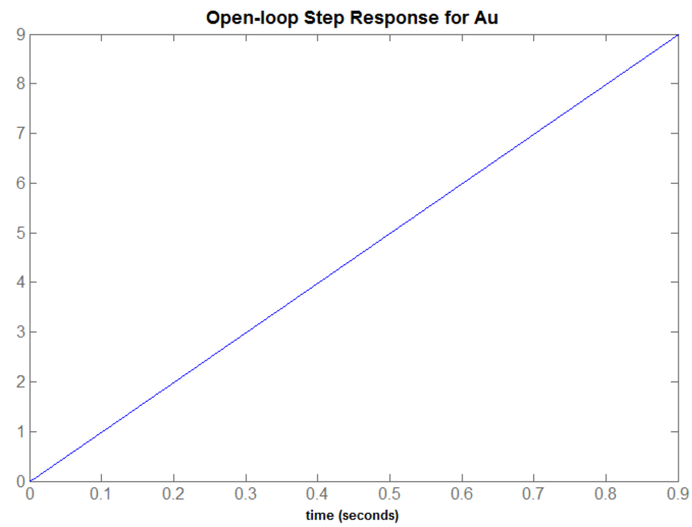

Figure 5.0: Initial Open-Loop Step Response of the System.

In the open-loop step response is in figure 4.0, the gain of the system transfer function is infinite, so 15000 is the final value of the output for a unit step input. This corresponds to an infinite steady-state error which is indeed too large. This response will not be adequate as it does not concur with our design objectives. Therefore, we need to add some sort of controller or compensator to improve on the system performance.

B. Step Response for System with PID Controller

The PID controller is introduced into the system and it is tuned to with MATLAB 2014 /Simulink to obtain the following values of controller gain: K_P = 9.31; K_D = 245.8878; K_I =23.52. From figure 5.0, the step response has a rise time of 0.0080 seconds, settling time of 0.0141 seconds and a peak time of 0.0772. These meet the design specification state in section III.

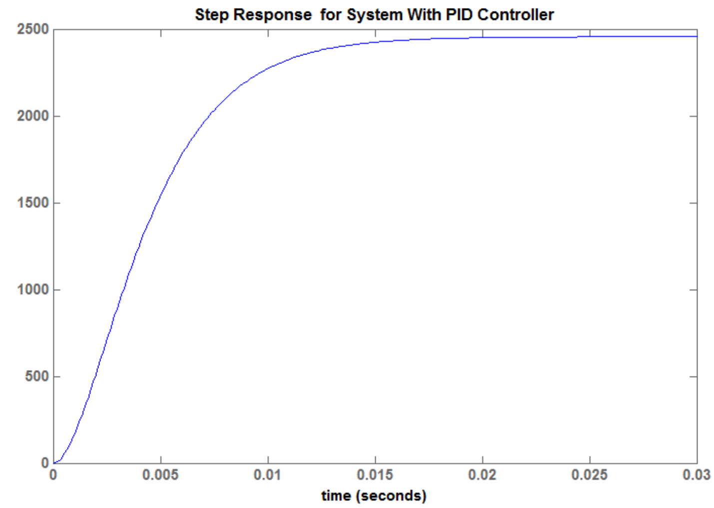

Figure 6.0: Step Response of the System with PID Controller.

C. *Step Response for System with PID Controller*

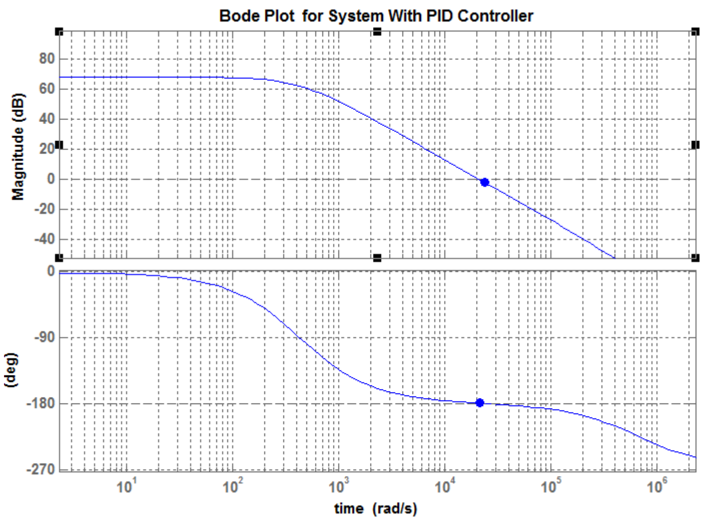

Figure 6.0: Bode Plot of the System with PID Controller.

As shown by the Bode Plot in figure 6.0, the gain margin, Gm is 1.2749dB, phase margin, Pm is 0.498^o and the minimum damping ratio obtained is 0.182. Considering the results achieved so far, the stability and performance of the machine tool has been improved by the addition of PID controller. Therefore, the suitable PID controller in s-domain is given as:

$$Gc = 9.31 + \frac{23.52}{s} + 245.8878s \quad \text{---------------} \; 1.7$$

VI. CONCLUSION

In this research, different linear controllers (proportional, integral, and derivative - PID) have been applied to the electro-hydraulic system controller transfer function to determine the performance of the system as the control parameters are fine-tuned to achieve the design specification. Introduction of the derivative controller dramatically improved performance by reducing both the rise time and settling time to within specification, but the outstanding steady state error was a concern. The PID-controller with closed loop feedback provided the best response and also proved to be very effective in the frequency domain when the Bode plot of the final controller gave a phase margin and damping ratio within specification.

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