A DESIGN TO DIGITALIZE HYDRAULIC CYLINDER CONTROL OF A MACHINE TOOL

by

P. B. U. Achi
Industrial and Manufacturing Engineering
Department of Mechanical Engineering
FEDERAL UNIVERSITY OF TECHNOLOGY, OWERRI,
P.M.B. 1526, OWERRI.

ABSTRACT
Conventionally hydraulic piston - cylinder servos are actuated using analogue controls for machine tool axis drives. In this paper a design of the axis control system of an NC milling machine which employs a small stepping motor to digitally actuated hydraulic piston - cylinder servo drives existing on the machines Y-axis is described. The digital control designed in this study was practically tested and found to be a linear servo, thus making it suitable for economic control from a computer.

LIST OF SYMBOLS

B = bulk modulus of hydraulic oil
V = 1/2 (volume of cylinder)
A = cross - sectional area of piston.
k_Q = rated servo valve flow gain
w_h = hydraulic natural frequency
ξ = hydraulic damping coefficient
^kLV = valve system leakage coefficient
f = viscous friction coefficient
Q_i = input angular displacement from stepping motor
Y_i = input linear displacement by precision ball-nut
Y_o = output linear displacement by hydraulic cylinder
M = mass carried by Y axis
G_1 = potentiometer gain
G_2 = electronic servo amplifier gain
G_3 = servo valve - hydraulic cylinder system gain
v = Voltage
w = rotational - speed of stepping motor
$\alpha = (1/W^2h)$
$\beta = (2\xi/w_h)$
N = number of stepping motor steps to be executed
Y = Linear positional value to be moved
r_m = machine resolution. (The value of the basic constants related to the components used in this investigation are listed in Table 1).

1.0 INTRODUCTION

Nigeria established her first car assembly plant over ten years ago but today in 1987 the country is still unable to produce a significant variety of vehicle spare parts. Also the manufacturing of the components for assembly of a new car is not in the predictable future. Therefore the vehicle assembly factories still depend on imported completely-knocked down (CKD) components to remain in production. With the drastic devaluation of the Naira following the advent of the foreign exchange market, vehicle prices in Nigeria have become unacceptably high! Also the prices of general machinery spare parts and components have soared beyond the reach of Nigerians.

The solution to the problem lies in the manufacturing of vehicle and other machine parts in Nigeria using Nigerian steel and Nigerian operators. The demand on skill for manufacturing using machine tools, however, is very high to ensure that the tolerances accessary for interchangeable assembly are met. This skill is in very short supply in Nigeria and requires several years of training and experience to acquire. The present solution would be to acquire the technology of numerically controlled (NC) machine tools which by their design have skill in-built in the control system.

The acquisition of this technology can only be by practical design and fabrication of NC machines and their control systems in Nigeria. One of the most feasible procedures to acquire NC technology is by redesigning (or retro - fitting) existing NC machine tools. It has been found (I) that the cost of retrofitted machines is about 20% of a brand new one. Many of the existing NC machines are equipped with hydraulic ram - cylinder drives which are controlled by electrical analogue comparators. When compared with digital controls, analogue controls are more expensive, more complicated, less accurate and are not easily amendable to control from digital computers. Therefore this study set out to design digitalization controls for the existing hydraulic ram - cylinder drive of the Y-axis of an existing NC machine.

1.1 The existing milling machine

This is an NC machine tool built by the H.P.E. (High precision Equipment) company of U.K. The milling machine possesses fixed bed layout and a vertical milling head. Unlike the Y- axis, the X axis is equipped with a hydraulic motor which is an entirely different design concept (2). The z-axis ram forms an integral part of the vertical quill (and spindle) assembly and the Y - axis slide is a cast block assembly having a cylinder housing for traverse movements, The Y-axis motion is on Dexter linear recirculating ball bearings on the top and bottom surfaces of the cast beam unit. The quill mounting casting incorporates a cylinder housing which holds a centrally mounted piston on a rod attached to the frame at both ends (fig.2). The Y (and Z) axis motion is caused by metering oil through a moog servo valve which varies the rate as well as the direction of flow of fluid to the hydraulic cylinder. The total traverse in the Y-axis is just over 30cm and the velocity in the axis is proportional to the rate of oil flow which is controlled by the driving signals from the principal control actuator, which would be stepping motors in the new design.

1.2 Stepping motors and electrohydraulic drives

1.2.1 Stepping Motor

A stepping motor is an electromagnetic incremental actuator which converts digital input signals (electronic pulses) to analogue outputs of shift motion.

In the rotary type of stepping motors, the output shaft (of the motor) responds to a train of input electronic pulses by equal angular displacements. Each pulse fed to the motor causes the output shaft to execute a constant angle called the "step angle" which is a characteristic of the particular stepping motor. This is in effect a digitalization of mechanical motion.

The advancement of electronic packages is simplifying digital incremental control, and is increasing the industrial interest in stepping motor technology. Thus stepping motors are being increasingly applied for many indexing purposes in machine tools, for example, as pallet drives in assembly lines, and as indirect or direct drives in numerically controlled machines. In this investigation small stepping motors (23 MSIO8) made by Moore Reed & Co. are used as actuators of hydraulic cylinder axis drives.

1.2.2 Electrohydraulic Drives

Before the industrial development of the d.c. servo motor the majority of NC machines built before the 1970's were equipped with electrohydraulic drives. The hydraulic power packs consist mainly of the hydraulic oil, pumps, pressure accumulators, electric motors driving the pumps, servo valves and suitable piping. The power pack supplies oil under suitable pressure to either hydraulic motor or cylinders or a combination of both to drive the various machine axes. Velocity (feed rate) control on the axes is achieved with servo valves which meter oil into the motors and cylinders at rates depending on the strength of electric signals (current) reaching the valves. To maintain a constant feed rate during a machining operation the difference (velocity error) between the momentarily commanded position and the executed position should be constant. In steady state this is possible only if the servo-mechanism is linear so that a constant velocity-following error by way of an error voltage is sent back to the servo valve. This is the case for the Y and Z axes described in this study.

2.0 THE NEW CONTROL SERVO SYSTEM OF THE Y (AND Z) AXIS

Figures 3 and 5 are electro-mechanical components and servo loops respectively for the Y axis and figure 6 shows the servo components of the Z axis. In figure 4, the manual control, electronic servo amplifier and stepping motor electronic drive logic boards are included in the "drive and control circuit" block. The manual control unit is necessary for tests to determine the accuracy of certain mechanical components (e.g ball screw pitch), and the machine resolution and its dynamic behaviour. .The miniature gear box is an anti-back- lash precision unit offering a 5: I of the input (stepping motor) speed. The output of this gear reducer drives the precision ballscrew and nut to offer a machine resolution of O.00254mm/step. A precision linear potentiometer provides the summing point of the servo valve controlled cylinder and the necessary feedback signals from this cylinder. The driving velocity-following error (voltage) is provided by the wiper arm of the potentiometer the terminals of which are connected to + 15V complementary voltages. The ballscrew and nut make up the rotary to linear translation block

shown in figure 5. The ballscrew itself has a pitch of 5. 08mm and is made of high tensile steel. To eliminate backlash and minimize the compliance of the servo system the ballscrew is of the preload recirculating ball nut variety, thus the accuracy and stability of the whole system is improved. During a machining operation, the stepping motor is commanded to execute N steps (digitally):
where

$$N = \frac{Y}{r_m}, \text{and } r_m = 0.00254\text{mm/step}$$

The miniature gear reducer output is 1/5th of the speed of the motor and this output drives the precision servo ballscrew which moves the ball nut linearly to offset a voltage at the linear potentiometer. The value of this voltage (i.e the velocity-following error voltage) depends on the speed of the stepping motor and on the response characteristics of the servo system. The voltage is an input into the electronic servo amplifier which outputs a current to the servo valve which opens to allow a proportional flow through the hydraulic cylinder. Thus the speed of the cylinder depends on the speed of the stepping motor which is determined by the rate of the digital pulses reaching the motor.

2.1 Piston/cylinder-servo valve transfer function;

In figure 1 considering an instant when the spool is displaced x. the differential pressure is P_m and the Laplace transform of the rate of increase of P_m is sp_m .Also the load has velocity, sY, and acceleration. s^2y. Generally, servo valve flow rate
= (K_Qx - $K_{Lv}P_m$). Pressure, P_1, increase at SP_I and since

$$B = \frac{\Delta P_1}{\Delta V/V}$$

$$\Delta V = \frac{V.\Delta P_1}{B}$$

$$P_m = P_1 - P_2$$

$$Therefore, \Delta p_m = \Delta P_1 - \Delta P_2$$

$$for\ equilibrium\ \Delta p_m = -\Delta P_2\ (stable\ servo\ system)$$

$$Hence, \Delta P_m = \Delta P_1 - (-\Delta P_1) = 2\Delta P_1$$

Therefore, rate of oil compression

$$SV = \frac{V}{B}.\frac{SP_m}{2}$$

Therefore the <u>flow equation</u> becomes:

$$K_QX - K_{LV} - \frac{V}{2B}.SP_2 = AsY \qquad (1)$$

Also the <u>force equation</u> is:

$$AP_m = Ms^2Y + fsY \qquad (2)$$

Substituting for P_m from (2) into (1):

$$K_Qx - K_{Lv}\frac{Ms^2Y}{A} - K_{Lv}fsY - \frac{vMs^2}{A} = \frac{AsY}{2BA} \qquad (3)$$

Therefore the transfer function becomes:

$$\frac{Y}{X} = \frac{\frac{K_QA}{A^2+K_{LV}f}}{S\left[\frac{VMS^2}{2B(A^2+K_{LV}f)} + \frac{(K_{LV}M+Vf/2B)}{A^2+K_{LV}f}S+1\right]} \qquad (4)$$

This is generally equivalent to :

$$\frac{Y}{X} = \frac{K}{S(\alpha S^2+\beta s+1)} \qquad (5)$$

where,

$$K = \frac{K_QA}{A^2+K_{LV}f} \qquad (6)$$

$$\alpha = \frac{VM}{2B(A^2+K_{LV}f)} = \frac{1}{W^2h} \qquad (7)$$

$$\beta = \frac{K_{LV}M+Vf/2B}{A^2+K_{LV}f} = \frac{2\xi}{Wh} \qquad (8)$$

The transfer function (eqn. 5) is shown in the block diagram, fig. 5.

2.2 Calculation of the steady state stability characteristics

From figure 5 the overall servo system closed loop transfer function can be derived:

$$\frac{Y_0}{Y_i} = \frac{G_1G_2K}{S(1/(S^2W^2)+2\xi/W_h+1)+G_1G_2K} \quad (9)$$

from equation 7, the hydraulic natural frequency,

$$W_h = \frac{2B(A^2+K_{LV}f)^{1/2}}{VM} \quad (10)$$

Using the appropriate values in table 1, the hydraulic natural frequency was found to be 494 rad/s or 78 H_Z

In the original system a maximum traverse speed of 38. 1 cm/min was specified for all the axes. In the new system the value of 38.1 cm/min was retained since no changes have been made in the original mechanical (hydraulic pressure) specifications of the machine tool system. Since the pitch of the servo ballscrew = 5.08 mm/rev: the rotational speed of the ballscrew at the machine traverse speed of 38.1 cm/min will be:

$$\frac{38.1 \times 10}{5.08}\frac{(rev)}{min}$$

Also since the miniature gear reduction of the stepping motor speed equals 5, the stepping motor speed at the maximum traverse speed of 38.1 mm/min is:

$$75 \times 5 = 375\, rpm$$

This represents the maximum speed at which the stepping motor can be commanded to run in the new control system. This limitation is mainly as a result of the maximum available hydraulic pressure in the system (which is 1200 N/cm^2).

By applying the Routh-Hurwitz (4) stability criterion on equation 9, it can be shown that for limiting stability:

$$2\xi w_h = (G_1G_2K)_{max} \quad (11)$$

Also by applying the "final Value Theorem" (4) to a ramp input function (which is representative of the stepping motor inputs, θ_i, in figure 5 during continuous motion) the maximum velocity following error (at maximum traverse speed of the machine tool axis) between the servo ball nut and the hydraulic cylinder following it is:

$$E_{max} = \frac{U}{G_1G_2K}(U = 38.1cm/min)$$

Using the values on table 1 on equations (11) and (12):

$$E_{max} = O.038mm\ approximately.$$

This represents the maximum path error during a machining operation (it occurs at maximum traverse speed) and it will be proportionately smaller for normal feed rates in the axis.

3.0 PROCEDURES TO STABILISE THE HYDRAULIC CYLINDER-PISTON SERVO SYSTEM

The stability of electrohydraulic drives used for axis positioning in machine tools can pose problems arising from the combined effects of coulomb friction and stiction in the system (3). Valve hysteresis and the compressibility of the oil especially for hydraulic cylinder drives cause even more serious problems.

This compressibility restricts the hydraulic cylinder-piston axis positioning systems to short stroke applications in machine tools.

On the existing milling machine in this

Table 1: LIST OF CONSTANTS RELATING TO COMPONENTS AND SYSTEM USED IN THIS INVESTIGATION

CONSTANT	UNITS	VALUE
Hydraulic natural frequency: valve cylinder (w_h)	rad/s	494
Damping coefficient ( )	-	0.170
Servo valve flow gain (k_Q)	$Cm^3/s/m^A$	5.118
Servo amplifier gain (G_2)	m^A/v	107
Miniature gearbox reduction (R)	-	+5
Potentiometer terminal voltage (v)	v	± 15
Potentiometer gain (G_1)	v/cm	11. 81
System leakage coefficient (k_{LV})	$(cm^3/s)/(n/cm^2)$	0.330
Viscous friction coefficient (f)	N/cm/s	52.560
Leadscrew pitch (p)	mm/rev	5.080
Mass carried by y-axis (M)	kg	284
Hydraulic system pressure (p)	N/cm^2	1200
Y -axis travel length	cm	30.500
Z-axis travel length	cm	15.250
Cylinder cross-sectional area (A)	Cm^2	45.610
Volume of cylinder (2v)	Cm^3	695.140
Balk modulus of oil (B)	N/cm^2	$1.38x10^5$
Machine resolution (r_m)	mm/step	0.00254

investigation, the traverse on the Y and Z axes were limited to 30cm and 15cm respectively and control by-pass leakages are incorporated in the valve-cylinder assemblies. These help to provide a value of damping coefficient,ξ to ensure stable operation and yet maintain adequate speed of response (equations 8 and 9)

The complication introduced by the inclusion of the stepping motor as the actuator of the servo system designed in the course of this as was that instability sets in whenever the stepping motor frequency coincides with value of the hydraulic natural frequencies (equation 10). This problem was taken care of by including viscous inertia dampers attached externally to the motor shaft to help reduce the stepping motor oscillations. To ensure that the overall gain margin of the servo system, (G,G_2K)max., is as high as is necessary to maintain stability and fast response a "current booster" in the electronic servo valve amplifier is included as the servo compensating circuit. This gives a high valve of G_2, i.e. the ratio,

$$\frac{(m^A),}{v}$$

of the amplifier gain. Thus the servo valve is continuously provided with adequate current to allow the appropriate oil flow necessary to maintain low velocity-following errors and fast responses by the hydraulic cylinder to the stepping motor inputs and hence adequate accuracy in machined shapes.

4.0 EXPERIMENTAL TESTS AND RESULTS

4.1 Tests and Measurement on the Axis of the Milling Machine

In order to test the system a manual control "black box" was created to generate discrete positional and velocity commands to the stepping motor on the axis. This "black box" contains a pulse-generator which provides the digital signals required to move the motors.

4.1.1 Oscillations and Resonance

Bench tests on the stepping motor alone had shown the presence of oscillations in the unit at low speeds. Later when the stepping motor was coupled to the new servo system on the axis, the oscillations of the velocity error were reduced by the inherent damping in the hydraulic servo system. A viscous inertia damper attached to the motor externally was later found to be effective in reducing the resonance amplitudes of the velocity-following error still further.

Thus the servo system is stabilised against the input oscillations of the stepping motor actuator. Operating the stepping motor in the half step mode (0.9°) not only improves control resolution and accuracy it also reduces the amplitude of oscillation.

4.1.2 Machine Resolution

The design machine resolution of 0.00254mm/step was confirmed during the manual (and later microcomputer) control tests. The preload ballscrew, the anti backlash miniature gearhead and the other precision units were completely effective in maintaining this resolution for motions in both the $\pm$ Y directions.

4.1.3 Feedrates

This was found to bear a linear relationship with the stepping rate, i.e. pulse rate supplied to the motor from an on-board manual pulse generators or as programmed from a microcomputer. Measurements of the traverse speed, the velocity following error as an analogue voltage at the potentiometer, the equivalent "step" following error and the servo amplified current were made at various feedrates.

Figure 7 shows the graphs of these test results on the servo system. The linear relationships confirm that this stepping motor-hydraulic piston/cylinder servo system is suitable for control from a digital processor.

5.0 CONCLUSIONS

As was mentioned in the introductory stages of this investigation it is possible for Nigeria (having got the steel industry) to join the "Retrofit" business which is still in its infancy. The present industrial objectives of the nation is the local assembly industries. This way Nigeria will dominate the general content of its industrial products (i.e. by adding value to her raw materials through the manufacture of the components locally). To be able to do this we need more of those machine tools which don't make excessive demands on operator skill. These are the numerically controlled machines. By joining the retrofit industry now we shall surely acquire the technology of modern machine tools needed for economic manufacture now and the competitive African and world markets in the future. The acquisition of the technology for this manufacture is hereby demonstrated by the success of this study. The digitalisation of the hydraulic servos facilitates the computer control of the machine tool axis motions and so further simplifies and optimises accuracy of manufacture.

REFERENCES

1. Production Engineer. "Are retrofits the answer to obsolete' NC?" *The Journal of the Institution of Production Engineers.* London, April 1979.

2. Achi, P.B.U. "Stepping Motor Hydraulic Motor Servo Drives for an NC Machine." *Nigerian Journal of Technology (NIJOTECH)* Vol. 6 No. 1, September 1982.

3. Parnaby, J. "Instability due to Static and Coulomb Friction in Autonomous Control System" *Proc. of "The British Hydromechanics Research Association,"* Cranfield, Bedford 1971.

4. Merrit, H.E., *"Hydraulic Control Systems"* John Wiley, New York, 1967.

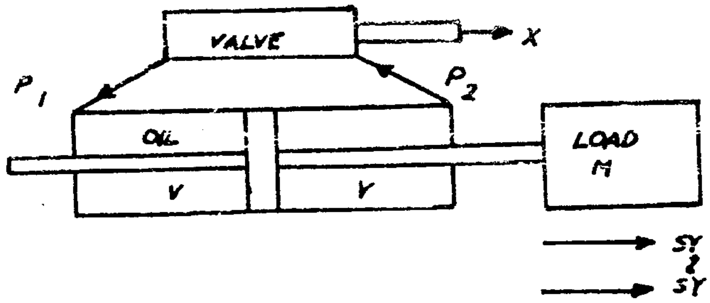
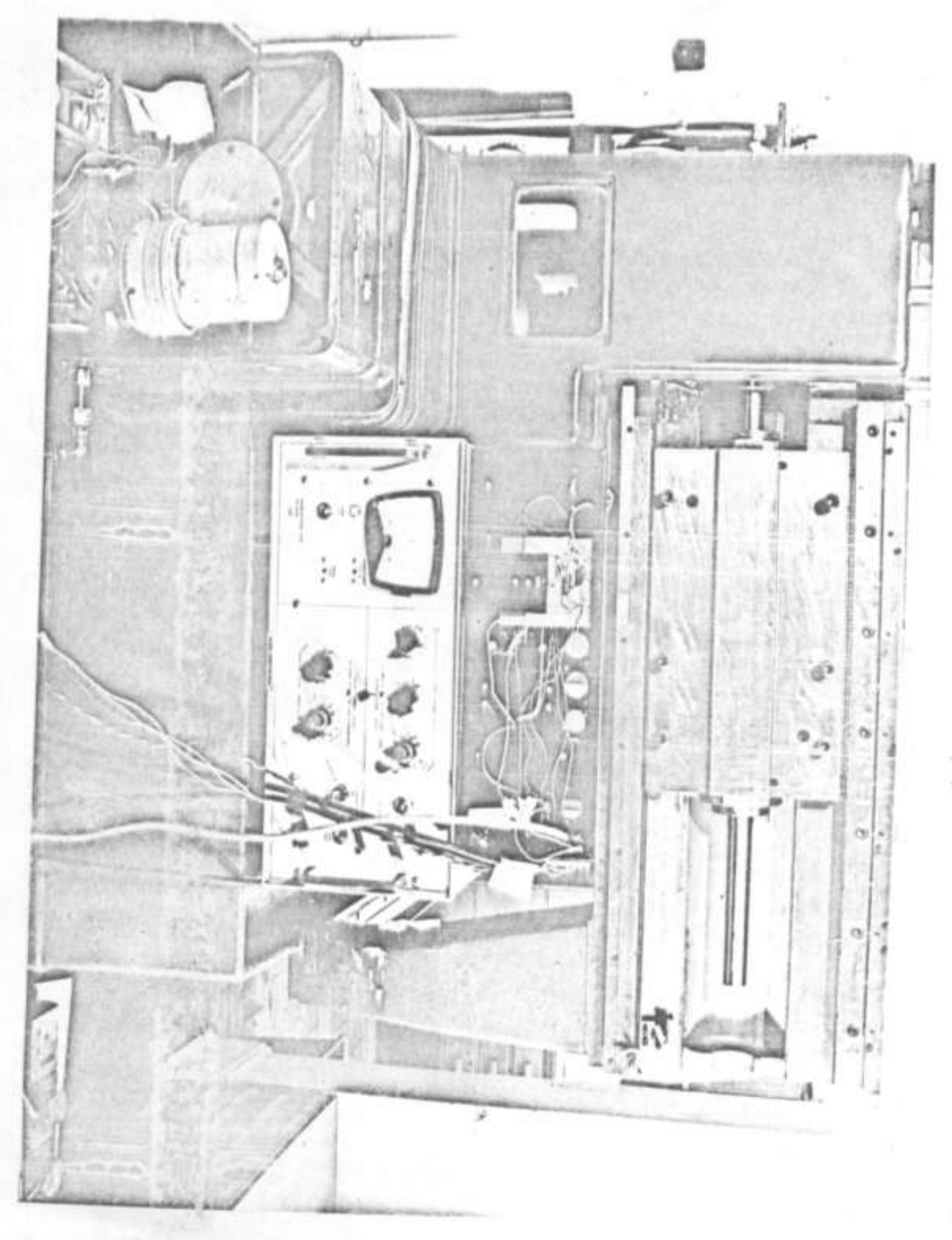


FIG. 1 SCHEMATIC OF VALVE - PISTON TRANSFER FUNCTION



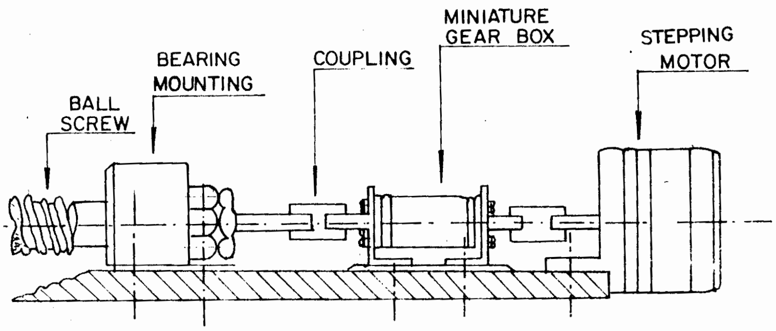


FIG. 3 STEPPING MOTOR SERVO DRIVE FOR HYDRAULIC PISTON-CYLINDER ON THE NC MACHINE

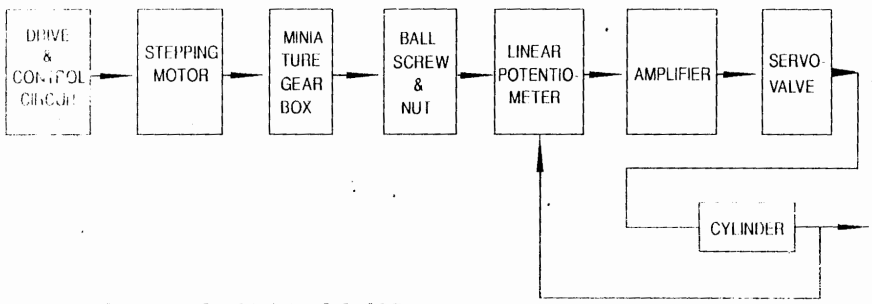


FIG. 4 Y-AXIS NEW SERVO SYSREM

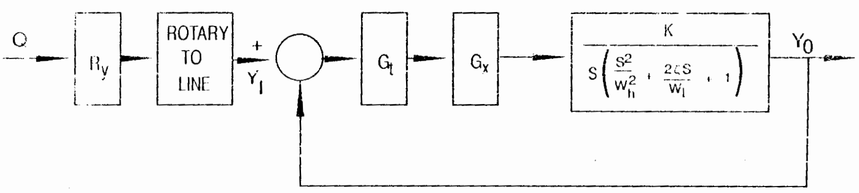


FIG. 5 THE Y-AXIS SERVO LOOP

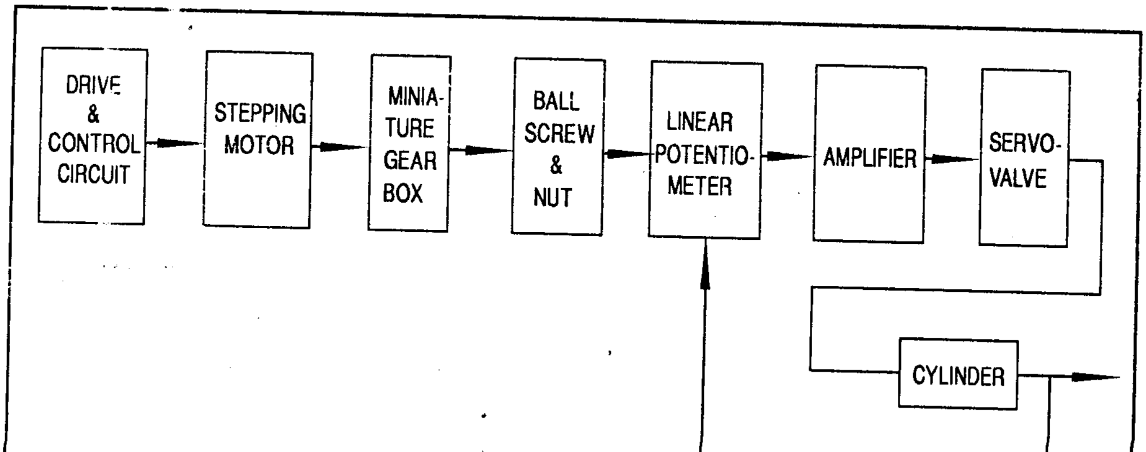


FIG. 6 Z-AXIS SEVO LOOP

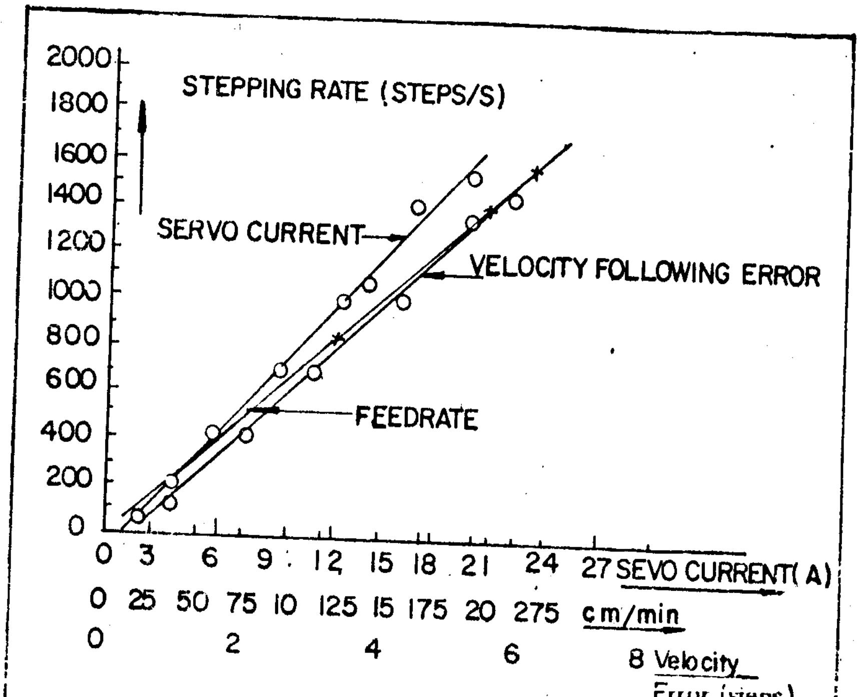


FIG. 7 STEPPING MOTOR HYDRAULIC CYLINDER SERVO SYSTEM GRAPHS