STEPPING MOTOR - HYDRAULIC MOTOR SERVO DRIVES FOR AN NC MILLING MACHINE

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ABSTRACT

In this paper the retrofit design of the control system of an NC milling machine with a stepping motor and stepping motor - actuated hydraulic motor servo mechanism on the machines X-axis is described. The servo designed in the course of this study was tested practically and shown to be linear - the velocity following errors being proportional to the axis feedrates. This property makes the servo system suitable for future on-line control from a microcomputer.

LIST OF SYMBOLS

ω_h = hydraulic natural frequency
ζ = hydraulic damping coefficient
R = miniature gearbox reduction
K_Q = servo value flow gain
d_m = hydraulic motor displacement
θ_i = input angle
θ_o = output angle
G_1 = potentiometer gain
G_2 = electronic servo amplifier gain
G_3 = servo valve - hydraulic motor system gain
V = voltage
$\alpha = 1/w^2_h$
$\beta = 2\zeta/\omega_h$
ω = rotational speed of stepping motor
k_L = System leakage coefficient
f = viscous friction coefficient.
(The values of the basic constants related to the components used in this investigation are listed in table 2).

1. INTRODUCTION

The shortage of skilled conventional machine tool operators is worldwide. In developing countries (like Nigeria) the scarcity of men in this area is *even* more acute. One of the ways to solve this problem would be to use numerically controlled (NC) machine tool systems wherever the existing production batch sizes and frequency of manufacture justifies it in a developing country. This is so mainly because numerically controlled (NC) machines have skill in-built into the control system and hence require only semi- skilled operators to produce components using prepared program sheet. Thousands of NC machines have become outdated in the developed countries because of the development of more versatile and economic control systems. The redesigning or updating of these machines with retrofits suitable for integrating into a flexible processor *[3]* has become a growing industry in its own right.

Because the NC machine is an expensive item of equipment it is often employed in double-shift operations initially. Therefore any redesigning procedures should include re-assessments of the precision and adequacy of the various members (gearboxes, leadscrews, valves, spindle assemblies) on the existing machine. It is possible that certain mechanical members may have to be replaced and this is likely to involve bought-out items initially but in some cases the machine may be found to be in good mechanical condition. This is the case with the machine used in this study.

1.1 THE OLD MACHINE TOOL (Fig. 1):

This is an "High Precision Equipment" NC milling machine which was fitted with Ferranti "Mark IV" 3-axes contouring control system. Magnetic tapes recorded by a computer were the sources of signal

generation and their cost was rarely justifiable for small batches and frequent "One-Offs". This, together with the added complication of the time taken for the preparation of tapes by outside bodies meant that the machine had not been much used since its purchase in 1964.
The design of the machine is based on a fixed bed layout with vertical milling head. It is equipped with hydraulic power to drive all the axes, the X-axis being by motor, gearbox and leadscrew. The X-axis motion is achieved by a hydraulic motor driving a recirculating ball screw through a 50:1 reduction gearbox. This traverse axis of nearly 122cm is the longest and its slide ways are integral with the fixed machine table.
The X-axis is actuated by metering oil through a MOOg servo valve. The velocity in this axis is proportional to the rate of oil flow through a hydraulic motor and is controlled by the driving signals derived from diffraction gratings fixed on the machine table. position feedback in this axis was by the optical diffraction gratings and scanning motor (Fig. 2a) as demanded by the earlier Ferranti system. In the X-axis there was an additional velocity signal from a tachogenerator at the motor shaft. The mechanical elements, spindle drive and quill, slide ways, gearing and leadscrew assembly had all suffered negligible wear and so it was justifiable to fit a new control system to extend the machine's period of use.

2.0 <u>THE NEW CONTROL SERVO SYSTEM OF THE X-AXIS</u>:

Figures 2b, 3a and 3b show the electro-mechanical components and the servo loop of the X-axis. In Fig. 3a the "drive and control circuits" are contained in a "black box" built by the author earlier and it includes the stepping motor electronic drive logic and manual control boards necessary for tests to be conducted later to determine the accuracy of the leadscrew gear-box assemblies and the machine resolution and its dynamic behaviour.

The rotary potentiometer which is driven directly by a small electric stepping motor is a very precise unit with 10k ohms resistance and a high linearity of ±0.2%. The miniature gearboxes in the feedback path is an anti-backlash precision unit and provides a reduction of 8:1. In the earlier control system, position sensing in all axes was by diffraction gratings and scanning motor (fig. 2a) as demanded by the original Ferranti system. In the X-axis there was an additional velocity signal from a tachogenerator at the hydraulic motor shaft. Spring - loaded anti-backlash spur gears (fig. 2b) were used to connect this hydraulic motor shaft to the tachogenerator shaft. In the new control system this tachogenerator was removed and the shaft retained to provide the feedback of the hydraulic motor motion through the miniature gearbox to the potentiometer. The main gearbox driven by the hydraulic motor provides a reduction of 50:1 and the output drives the X-axis leadscrew (fig. 3a).

2.1 <u>THE STEPPING MOTOR USED AS THE SERVO ACTUATOR</u>:

A stepping motor is an electromagnetic incremental actuator which converts digital input electronic signals (pulses) to analogu outputs of shaft motion. Each pulse fed to the motor causes the output shaft to execute a constant angle called "the step angle" which characterises a particular stepping motor. A train of pulses reaching the motor causes it to respond by rotating continuously until the pulses are cut off when the motor

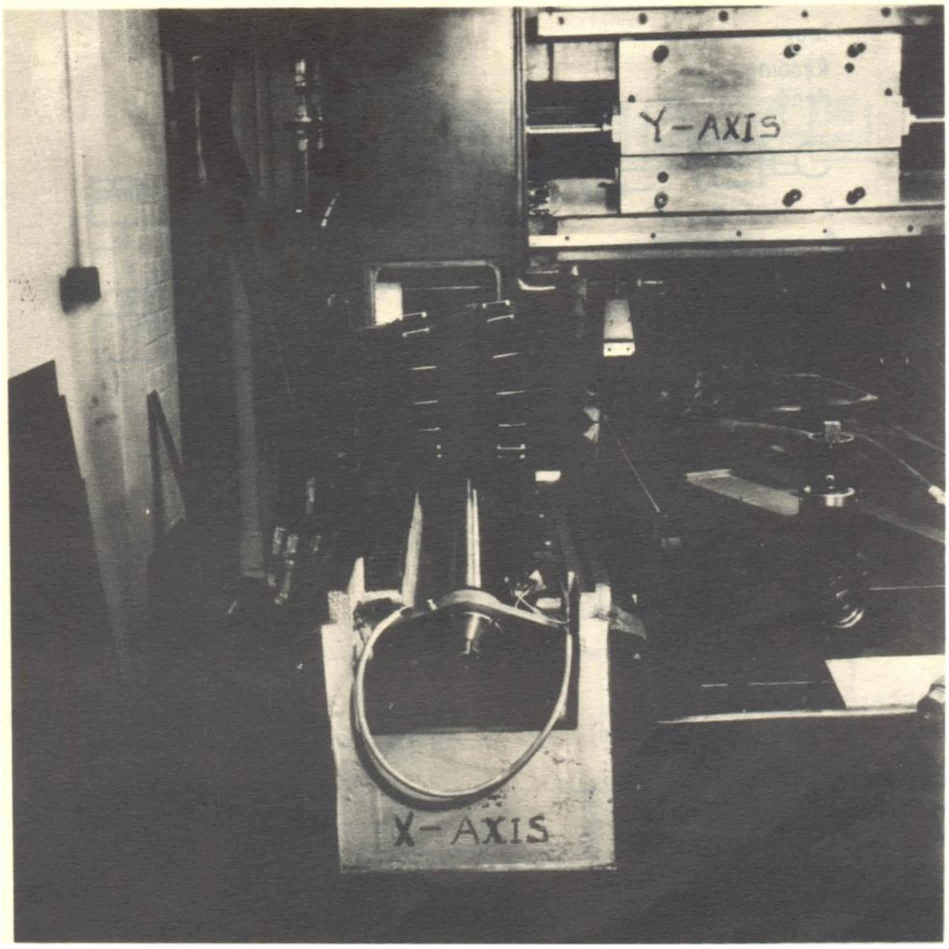


Fig. 1a - The Machine Tool Stripped Dow

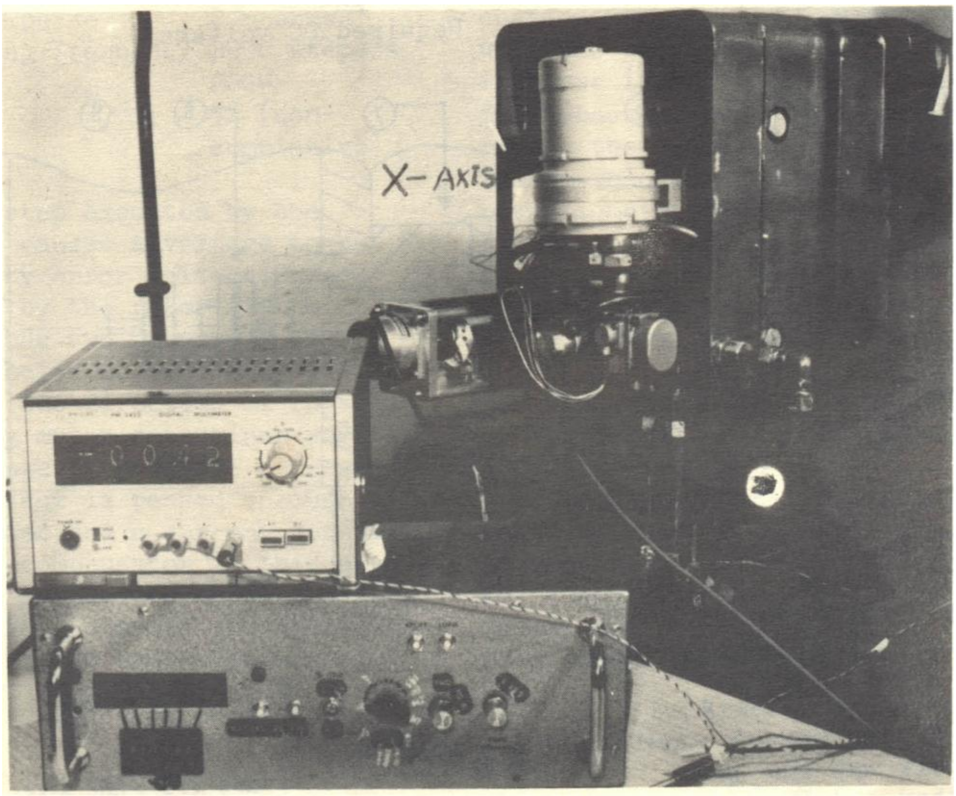


Fig. 1b: x-axis : main gearbox, new servo unity and control boxes.

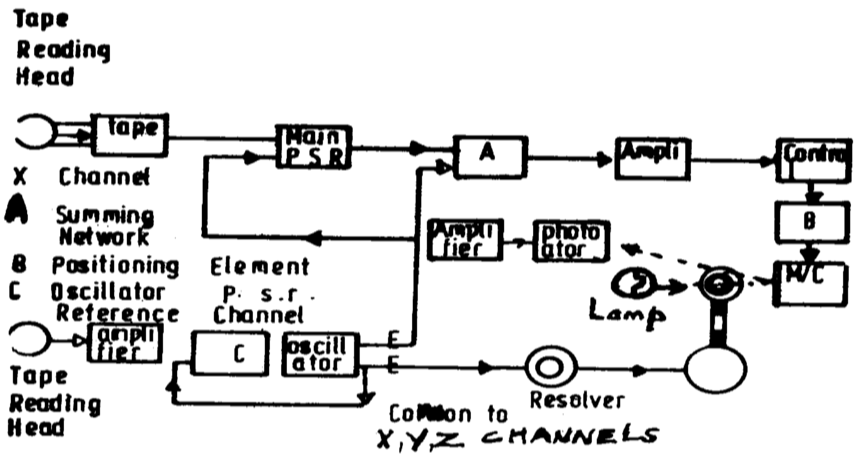


Fig. 2a Old Control System of the HPE Milling Machine

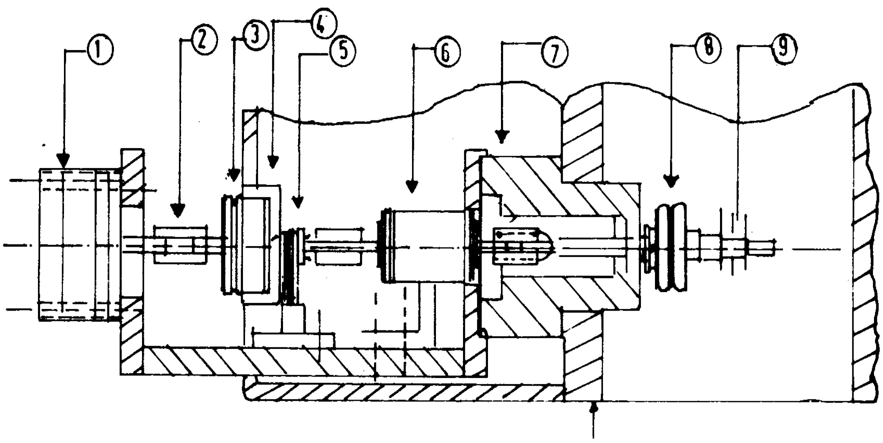
stops almost instantaneously (the deceleration being very high indeed).

For the purposes of the new X-axis control system for this machine tool a small electric steeping motor is required to perform the main control actuation for the electro-hydraulic servos on the axis.

The torque requirement of the motor is that to overcome friction in the servo rotary potentiometer which requires a torque of 0.0002N.M only. Therefore a small stepping motor with the following specifications was used:

TABLE 1: Characteristics of the Stepping motor.

Type	hybrid (23 MS108 by "Moore Reed")
Holding Torque	2600gm.cm
Step Angle (half steps)	0.9^0
Rotor inertia	70 gm .cm^2
NO of phases	4
Required DC voltage	24 v



1. stepping motor
2. Flexible coupling
3. Rotary potentiometer
4. Driving cup potentiometer casing
5. Slip rings and brushes
6. Miniature gear box (8.1)
7. Sleeve for feedback shaft
8. Anti bock lash spur gears
9. Feed bock shaft

FIG 2b

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Scale 2/3 X Full Size

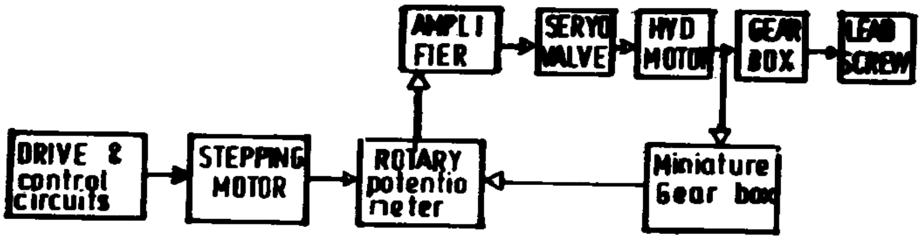

fig. 3.: axis new servo system

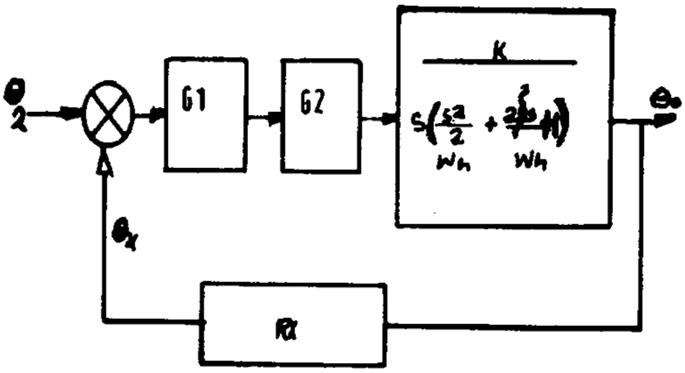

Fig. 3b: the X-Axis servo loop

Current per phase	275A
Resistance per phase	0.6 ohms
maximum starting frequency	220 steps/s
Power	200w
Step error	5 %(non- cumulative)

The first step executed by the stepping motor causes a voltage to be offset (velocity error voltage) near the zero point by the rotor of the potentiometer the two terminals of which are connected to +15V and -15V respectively. The direction of rotation of the stepping motor determines whether the error voltage is positive or negative. This error voltage is passed through a voltage amplifier and a current booster (used as servo compensator in this study) and the output is used to actuate a MOOg servo valve which controls the hydraulic oil flow through the hydraulic motor. When the stepping motor executes continuous rotation (at speeds consistent with the pulse rates) the hydraulic motor response is also continuous rotation which is fedback through the gear reducer to rotate the casing of the potentiometer. Thus a velocity error voltage exists at the potentiometer (its rotor and casing rotate relatively) and its magnitude is a function of the stepping motor RPM and the response characteristics of the hydraulic motor servo system in steady state.

3.0 THE CALCULATION OF THE STEADY STATE STABILITY CHARACTERISTICS (Fig. 3b)

In the original control system the maximum traverse speed **in** the axis is 38.1cm/min. and the coordinate accuracies are: 0.0127mm over the 122cm working range. In the new control system no changes are made in the specifications of the mechanical components and since the maximum traverse speed is limited by the available hydraulic power, the value of 38.1cm/min. is retained. However the machine resolution in the new design is to be 0.00254mm per step (0.0001 in/step) of the stepping motor.

Therefore the maximum required pulse rate is: –

$$\frac{38.1 \text{ X } 10}{0.00254} \text{ X } \frac{1}{6} \text{ (steps/s)} = 2.5\text{KHZ.}$$

For a servo valve - hydraulic motor unit the open loop system transfer function is given by:

$$\frac{k}{s\left(\frac{S^2}{\omega_h^2}+\frac{2\zeta\zeta}{\omega_h}+1\right)} \qquad (1)$$

where $K = \frac{(K_Q d_m)}{d^2{}_m + K_L f}$ = open gain of the valve-hydraulic motor system …(2)

In fig. 3b the overall transfer function of the machine axis servo system is given by

$$\frac{\theta_o}{\theta_i} = \frac{G_1G_2K/S\left(\frac{s^2}{\omega^2{}_h}+\frac{2\zeta\zeta}{\omega_h}+1\right)}{1+R_X G_1G_2K/S\left(\frac{s^2}{\omega^2{}_h}+\frac{2\zeta\zeta}{\omega_h}+1\right)}$$

$$\therefore \frac{\theta_0}{\theta_i} = \frac{G_1G_2K}{\left(\frac{1}{\omega^2{}_h}\right)s^3 + \frac{(2\zeta)}{\omega_h}s^2 + s + R_X G_1 G_2 K} \qquad (3)$$

Equations (2) and (3) serve as a basis for selecting maximum possible overall closed loop gain and this is given by:

$G = (R_X G_1 G_2 K)$ in (1/sce) (4)

(G was found to be equal to 2184 in this study). Knowing the feedback miniature gear reduction, R_x, the rated flow gain of the servo valve, (K_Q) and the values of d_m and $K_L f$ from motor and valve manufacturers specifications respectively (Table 2), G can be calculated. For the precision rotary (360^0) potentiometer $\pm$15V are connected to the two terminal (i.e. 30V total voltage). Therefore $G_1 = \frac{30}{2\pi}$(V/rad). Using Routh - Hurwitz stability criterion (equation 3) it can be shown that

$$G_1 G_2 K = \frac{\alpha}{\beta} = 2\zeta w_h \qquad (5)$$

(W_h is deduced from the physical specifications of the hydraulic motor). Applying the values in table 2 the maximum value of overall loop gain is calculated and hence the maximum setting of the amplifier gain, G_2, was found from equation (5): $G_2 = \frac{2\zeta\omega_h}{G_1 K}$ (6)

(in this study this was found to be 37mA/V). The maximum possible amplifier setting (gain) was used so as to give the fastest system response. This is desirable for the following reasons:

(i) To achieve certain motions in the shortest time period consistent with the pulse rate.

(ii) to minimise the lag between the electrical stepping motor motions and the hydraulic servo followers. This reduces the velocity following error and would avoid the significant end of a segment errors in circular interpolation (in two axes) mentioned by Plas and Blommaert [1].

TABLE 2:

CONSTANT	UNITS	VALUE
Hydraulic natural frequency (w_h)	Rad/S	422
Damping coefficient (ζ)	-	0.32
Miniature gearbox reduction R_X)	-	8
servo valve flow gain (kQ)	Cm^3/S/mA	9.2126
Hydraulic motor displacement (dm)	$Cm^{3/}$rev	0.7206
Potentiometer "gain" (G_1)	V/rad	4.9
Terminal voltages	V	± 15
Servo amplifier gain (G_2)	MA/V	37
System leakage coefficient (K_L)	(cm^3/S)/(N/cm^2)	1.54 X 10^{-2}
Viscous friction coefficient (f)	N/rad/s	0.86
Leadsrew pitch (p)	Mm/rev	6.35
Mass carried by x-axis (M_0)	Kg	908
System hydraulic pressure	N/cm^2	1200
Axis travel length	cm	122
Machine resolution	Mm/step	0.00254

3.1 THE STEADY STATE DRIVING ERROR BETWEEN THE STEPPING MOTOR AND THE HYDRAULIC FOLLOWERS:

The flow through the servo valve is directly proportional to the lag between the electric stepping motor and the hydraulic drive. This lag or driving error can be measured as a certain number of steps or driving pulses.

Treating the step inputs in this servo system as unit ramp functions it can be shown by the "Final Value Theorem" that the velocity following error:

$$E(S) = \frac{S\left(\frac{\omega}{S^2}\right)}{1+\frac{G_1G_2K}{S\left(\alpha S^2+\beta S+1\right)}} \qquad (7)$$

$$= \frac{\omega}{S+\frac{S\left(G_1G_2K\right)}{S\left(\alpha S^2+\beta S+1\right)}} \qquad (8)$$

Therefore in the limit

$$E_{max} = E(S) = \frac{\omega_0}{G_1G_2K} \quad S\to 0 \qquad (9)$$

Where w_o = velocity at input at maximum stepping rate of the stepping motor = 2500(steps/S). For the 0.9^0 step angle, 2500 steps per sec becomes $\frac{2500 \text{ x } 0.9}{360} \text{ X } \frac{60}{1} = 375$ RPM = 39.27 (rad/S) of the stepping motor. Therefore the maximum driving error (eqn. 9) is established as:

$$E = \frac{39.27(\text{rad}/S)}{273(1/S)}$$

(273 = $^R x^G I^G 2^K/^R x$ = 2184/8)

i.e. max = 0.14385 rad. at steady state. For the step angle of 0.9^0 the maximum (transverse) steady state error is equivalent to

$$\frac{0.14385(rad)}{\frac{0.9}{180}(rad)} = 9.15\, steps\ (puless)$$

This represents a path error of 0.023mm (machine resolution = 0.00254mm per step) and this is at traverse speed.

It follows from equation (9) that this error will be proportionally less for normal machining federates.

4.0 THE EFFECT OF THE STIFFNESS OF ATTACHMENTS TO THE CLOSED LOOP SERVO STRUCTURE:

Although the influence of the combined elasticity of the various mechanical attachments in the servo loop have not been quantitatively considered in the calculations of the maximum loop gain and the resulting maximum velocity following error, it is appreciated that there is an effect of the dynamic performance of the servo system. The material and size of connecting items like shafts, couplings and gears determine the overall stiffness of the mechanical unit.

In the servo loop (fig. 3b) let F_o = Hydraulic stiffness of the servo valve and hydraulic motor assembly,

ω'_0= the corresponding hydraulic natural frequency.

M = total mass of moving attachments in the servo drive.

It can be shown that generally,

$$\omega'_0 = \frac{(f_0)^{\frac{1}{2}}}{M}$$

Then $$\frac{1}{\omega'^{\,2}_0} = \frac{M}{F_0} \qquad (10)$$

If F is the stiffness of the other mechanical attachments, and w_T the overall system natural frequency, then

$$\frac{1}{\omega^2_T} = \frac{M}{F_0} + \frac{M}{F} \qquad (11)$$

From equ. 10

$$\frac{1}{\omega_T} = \frac{1}{\omega'_0}\left(1+F_0/F\right)^{\frac{1}{2}} \qquad (12)$$

$$\text{i.e. } \omega_T = \omega'_0/(1+F_0/F)^{\frac{1}{2}} \qquad (13)$$

Equation (13) shows that the natural frequency can be significantly reduced if the ratio F_o/F is high and equation (5) shows that the maximum system gain is proportional to the natural frequency. Therefore, a high value of (F_o/F) can lower the value of the system gain, increase the response time and the resulting velocity following error, ultimately affects the accuracy of NC contouring operations.

The effect of F_o/F was considered from the initial design stage when the material and size of components were selected. The original machine components were well conceived and with the added items minimum compliance was introduced, eg. silver steel for the feedback shaft and high tensile steel precision ballscrew for the axis, assembled with minimum length couplings. It has been correctly pointed out by Ertongur *[2]* that apart from the adverse effects of valve hysteresis on electro hydraulic system stability, backlash in the system also encourages the low frequency oscillations ("tweaks") which cause instability. Therefore, in the present system the gearboxes used are anti-backlash types and the leadscrew is of the preloaded re-circulating ball nut variety. All these help to increase the accuracy and stability of the whole servo system.

5.0 CONCLUSIONS:

5.1 TESTS AND MEASUREMENTS OF THE X-AXIS OF THE MILLING MACHINE:

(i) Stability: At the maximum servo amplifier setting (gain) the machine axis was found to be dynamically stable.

As long as the servo valve for which the system is designed is used no "hunting" will ever occur during axis motion.

The system responded stably to both unit step input and steady state input actuation.

(ii) Feedrate: This was found to bear a linear relationship with the pulse rate supplied to the motor. Measurements of the traverse speed, the servo current, the velocity following error (voltage) at the potentiometer and equivalent step following error were made at various stepping rates. These measurements were later repeated during microcomputer control (3).

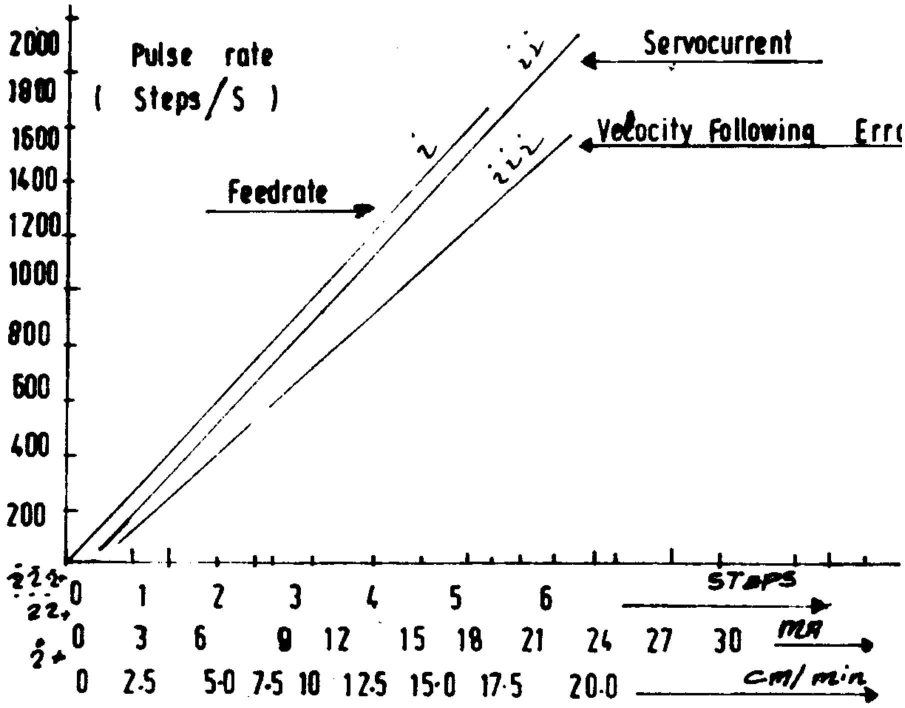

Fig. *3c:* The New Servo System Graphs Measured under both Manual and Micro-computer

controls. Fig. 3 C shows the graphs of these test results on the servo system. The linear relationships confirm that the system is suitable for microcomputer control. The brief deviation from the linearity of the servo current in the region of the origin (fig. 3c) is a result of internal friction in the servo drives and occurs at non-machining feedrates) Starting Speeds: Although the stepping motor itself is capable of a maximum

(iii) Starting frequency of 2KHZ, it was found that the maximum starting frequencies to which the X-axis could respond accurately was about 1000 pulses/s Suitable acceleration schemes [5] can then be employed to reach faster stepping rates (feedrates).

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