Design of a Programmable Jig for Automated Assembly and Machining

Neil C. Singer

Submitted to the Department of
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in Partial Fulfillment of the
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Degree of

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Abstract

The current thrust toward computerized automation has encouraged the development of more flexible and adaptive hardware. A computer controlled jig capable of holding a variety of parts while machining and assembly tasks are performed on them was developed. Its ability to generate large clamping forces coupled with the its measurement capability is invaluable for many robot applications. This fixture should be capable of speeding up the process of feeding parts to the robot, and reduce the time required to introduce new automated tasks.

Thesis Supervisor: Warren Seering

Title: Associate Professor of Mechanical Engineering

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1. Introduction

Currently, the use of robots and other types of computer controlled automation equipment is severely limited by hardware constraints. The programmable flexibility of the robot is negated by the requirement of specialized fixtures for holding parts. These specialized fixtures require many man hours of design and implementation, and therefore are not cost effective for short run applications.

Most small parts that are machined and assembled are made in the jaws of a common vise. Yet, the vise is not a practical jig for automated assembly, for it requires an operator to tend the manufacturing process. Therefore, a *programmable* jig that can repeatably hold a variety of parts for robot machining and assembly is invaluable.

The programmable fixture will be mounted on a table with two degrees of freedom (rotation and tilt). A robot with four degrees of freedom that is capable of light machining operations will perform various automation tasks on the table. This entire system must be considered in order to obtain the maximum performance from the jig.

This configuration suggests that the table can be effectively used to help load the jig. The part may be dropped onto the rotary table from a feeder or from the robot itself. The table can tilt and rotate to square up the piece, then the fixture can close on it. Next, a measurement of the piece is taken—the absolute distance between the jaws of the fixture. This measurement can be used to help determine the orientation of the piece and the table can readjust the part's position, if necessary.

This method would seem to be effective for a wide class of parts. For some, however, added sensors may be necessary.

2. Design Goals

The first step toward developing design goals was to study the size of the parts that can be handled by the rotary table. This determines the types of tasks that would be expected to be done by the robot. Next the forces that may be encountered by the jig were estimated. Thus the following specifications were set as goals:

- The jig must be able to take measurements of its jaw separation to within .002 inches.
- The jig should clamp with 180 pounds force.
- The jig should have at least a four inch stroke.
- The jig must fit on the nine inch diameter surface of the rotary table.
- The entire system *must* be extremely light. Its total weight should be only a small fraction of the fifteen pound maximum payload of the table.

- The system must be compact. This is necessary in order to maximize the available working space on the table and to avoid ruining the aesthetics of the high performance robot system.
- The added inertia of the jig must be minimized in order to retain the high performance of the rotary table.
- The closing speed of the gripper should be as fast as possible—at least 3 inches/second.
- The control system should be critically damped and thus have no overshoot.

3. System Description

The programmable fixture (Figure 1) consists of three major assemblies: linear bearings, a drive mechanism, and a set of fingers or walls. The structure of the device is supplied by a Design Components Incorporated precision crossed-roller slide with a four inch travel. This is a linear bearing with great side and vertical stiffness. This crossed-roller slide was then machined to allow the drive mechanism to fit through the center of it.

The drive system consists of a Warner M-510 $\frac{5}{16}$ inch diameter ball screw which is supported by two bearing blocks. one end of the shaft is supported by a high speed angular contact ball bearing. This bearing handles the full 180 pound thrust load of the device as it clamps. The other bearing is a miniature radial ball bearing which is preloaded against the thrust bearing. Attached to the thrust bearing mount is an Electroid EC-08B disk brake which is capable of holding up to 100 pounds of the clamping force. It is attached directly to the screw shaft in order to allow the motor to assist the brake in holding the screw shaft while minimizing the load on the timing belt.

The use of this timing belt enables the motor to be positioned parallel to the crossed-roller slide, saving space. The motor is a samarium cobalt rare earth DC servomotor which can produce a huge amount of power for its size—.1 H.P. in a 1.3 inch by 2.1 inch package. A Hewlett-Packard optical shaft encoder is attached to the other side of the motor for angular position feedback.

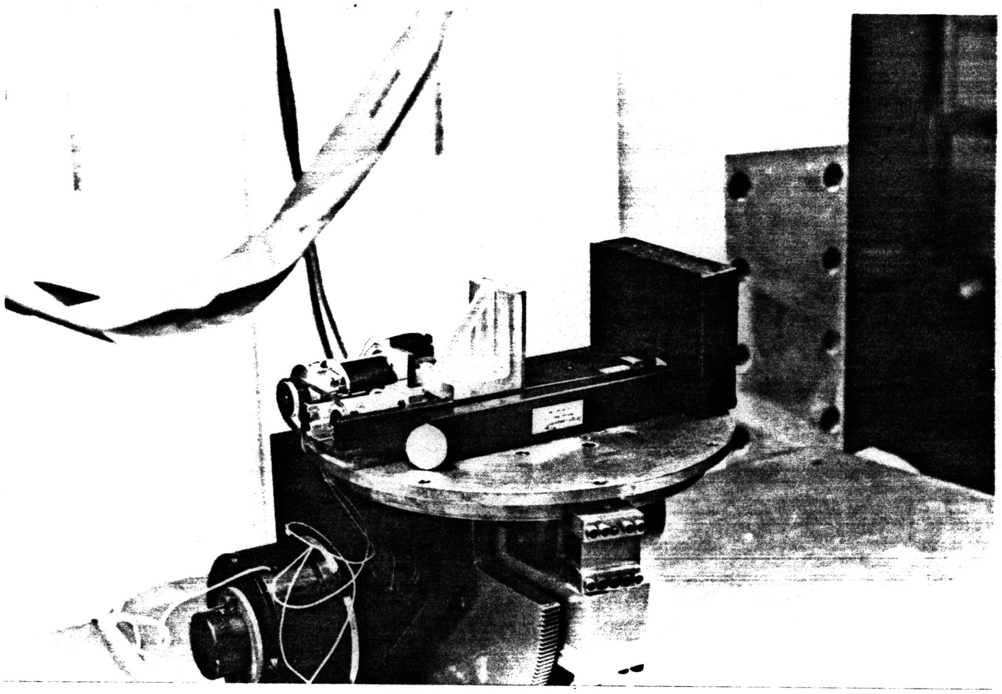
A finger which actually holds the workpiece is mounted on the crossed-roller slide. This is an aluminum wedge with a steel face to prevent surface marring. The finger is shaped to minimize deflection and can be easily replaced with a force sensing or other style finger.

4. Design

4.1. Mechanical Design

The mechanical design of this high performance system is complicated by the dependence of component selection on geometric design. The two processes must occur in parallel. Some of the many tradeoffs and design decisions are presented in the following sections.

Figure 1. Programmable Fixture Mounted on Rotary Table



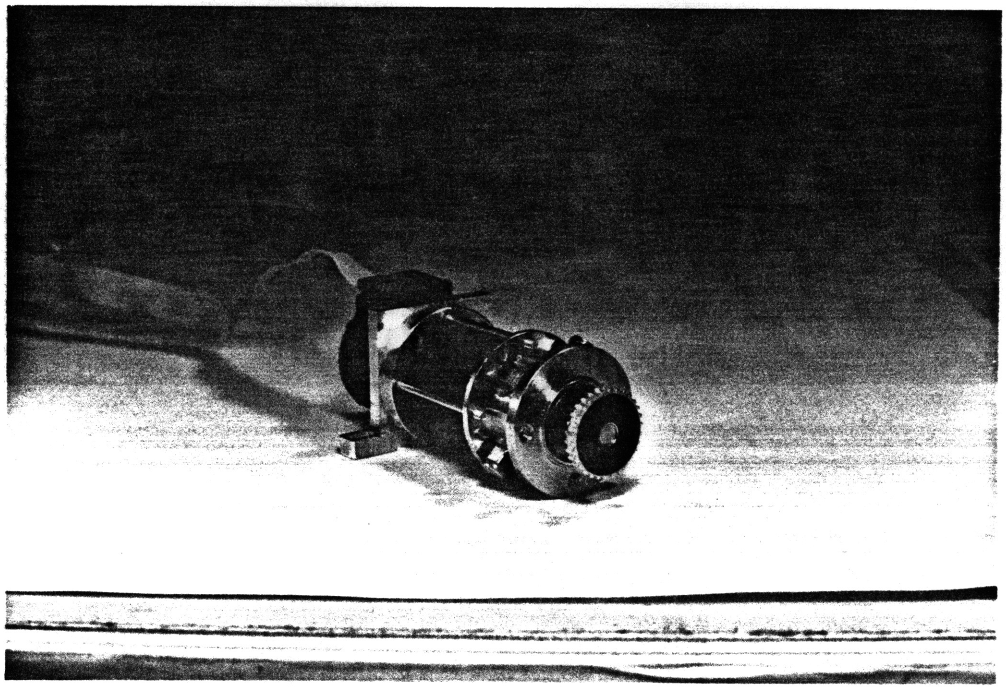
4.1.1. Component Selection

In order to design this jig, it was necessary to first examine different types of linear bearings that are capable of withstanding large, offcenter loadings. Rolling element bearings were desirable for a system that was to run under servo control. Ball or Crossed-roller slides seemed to be the best candidates for this application because of their load carrying capabilities and they provide great accuracy at a reasonable price. The choice between a ball slide and a crossed-roller slide was less obvious. Ball slides are preferred for most applications because they are less expensive and self cleaning—foreign particles are easily rejected from the balls. Crossed-roller slides, however, have much more load carrying capability for a given slide size. Therefore, the crossed-roller slide seemed to be the best option for a linear bearing as long as it is protected from dirt. In addition it is precision grade and thus has a straight line accuracy[1] within .0001 inches/inch of travel.

The Warner M-510 miniature ball screw was then selected. This screw was the largest of the miniature series of rolled screws. It was therefore relatively inexpensive and had a small diameter ball nut that would pose fewer difficulties in fitting in a tight place than any of the conventional screws in their industrial line. The rolled thread screws of the miniature series have a linear accuracy of .0005 inches/inch. Thus a precision grade screw—a screw which is significantly more expensive, is not required.

[1]Straight line accuracy is defined as the total amount of deviation the ground top surface of the slide moves from an imaginary plane as it rolls. This includes all effects of dipping and yawing.

Figure 2. Clifton Precision Motor



Motor selection was a difficult task. An examination of the various ball screws in the size range that could conceivably fit in this device suggested that leads over .125 inches were uncommon. This fact is understandable since most motors that would be used with these screws require small transmission ratios. Therefore, from the driving torque expression for a screw:

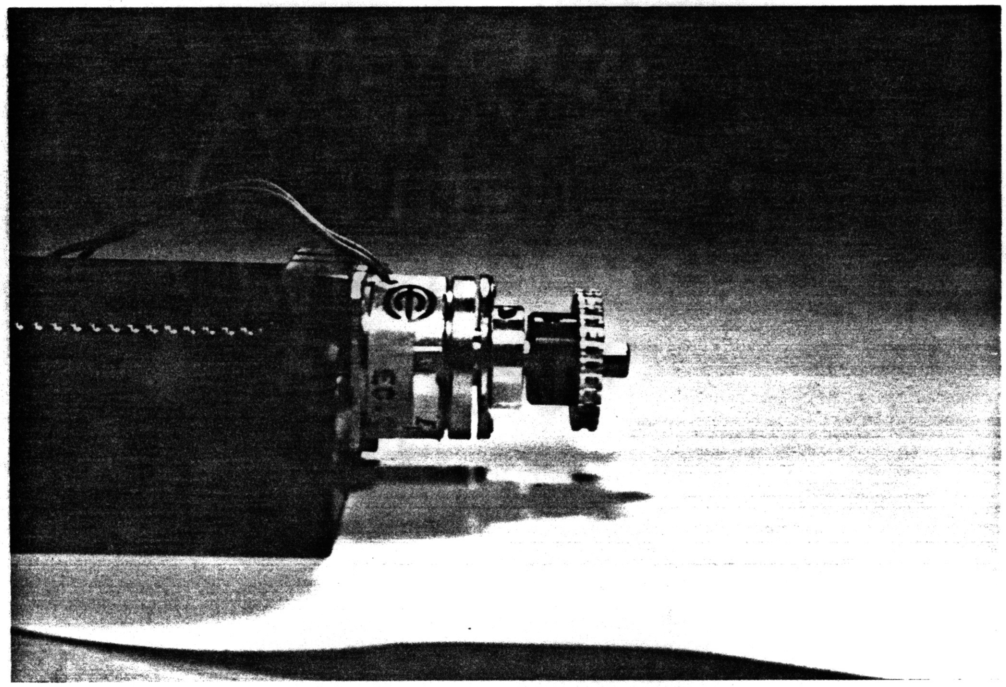
$$T_d = \frac{Fl}{2\pi e} \tag{1}$$

where T_d is the driving torque, F is the load, l is the screw lead (inches/revolution), and e is the ball screw efficiency (.90), the motor torque requirements at the screw shaft are approximately 50 ounce-inches at stall.

Over twenty different motors in this size range were studied. The manufacturers included Inland Motor, Clifton Precision, Aerotech, Pittman, B&B, and Ashland Electric. The motors were compared on several performance specifications including peak torque, continuous torque, weight, and rotor inertia. In this application the peak torque and weight become important parameters because the motor is used to clamp on an object and then have a brake provide the holding torque. Weight is important in order to maximize the rotary table's performance.

The motor with the highest performance with unquestionably the least weight was a Clifton Precision Rare Earth DC Servomotor. This motor was designed for use in high performance applications such as computer disk drives. It produces enough torque to run the jig at full clamping force with a direct drive. With Samarium Cobalt magnets it weighs only 5.2 ounces and provides more torque than motors with iron magnets weighing 42 ounces (Figure 2).

Figure 3. Electroid Brake



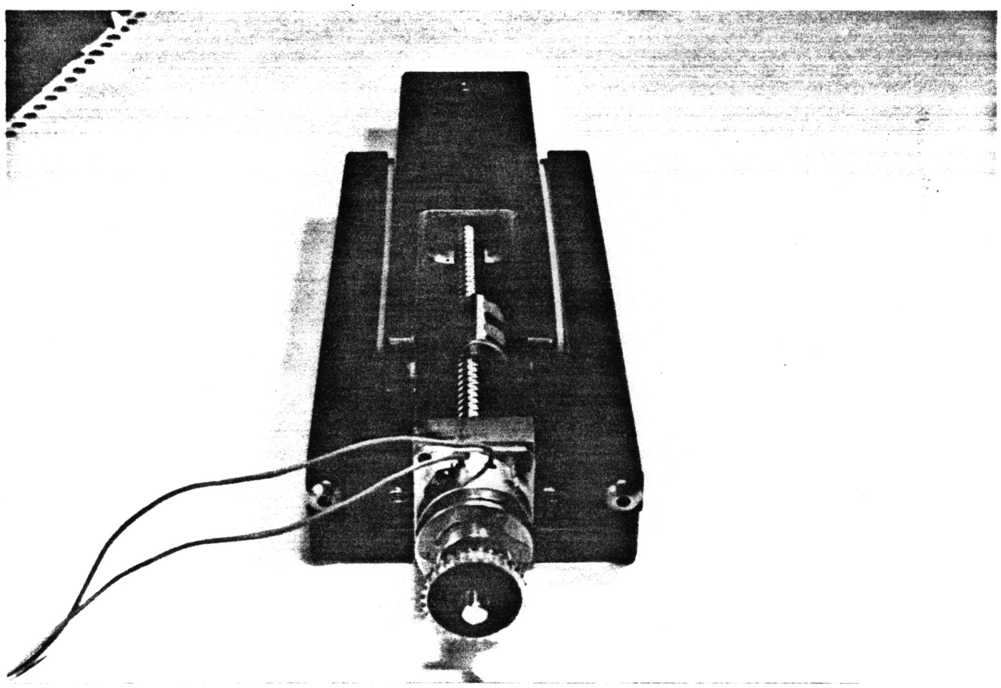
The screw shaft bearings are ABEC 7P precision ball bearings. On one end of the shaft is a .8661 inch (22 mm) O.D. angular contact bearing with a static thrust rating of 231 pounds. This bearing was selected because of its small size and capability of handling the 180 pound axial load. The other bearing is a .375 inch O.D. radial instrument bearing. The use of the crossed-roller slide minimizes radial load requirements of the screw bearings. Without this feature, the bearings would be so large that the present, compact configuration of the system would have to be abandoned (Figure 1).

The optical shaft encoder that was selected is a Hewlett-Packard model HEDS 5010 A05. This is a new product with 500 pulses per revolution (resolvable to 2000 counts per revolution) and an index pulse for calibration. This shaft encoder is ideal for this application because of its small size (28 mm diameter) and large quantity of pulses. With a .1 inch lead ball screw this translates to .00005 inches of linear travel per pulse.

An Electroid Corporation EC-08B electrically "on", flange mounted disk brake was purchased to assist the motor in holding the jig closed (Figure 3). It was selected so that it could hold slightly more than half of the rated clamping force ($\approx$100 pounds). The brake was included in the system because the motor cannot continuously hold the required torque without overheating. The brake was selected to minimize size while still protecting the motor. In the present configuration the system is optimal because both of these components are working at their maximum continuous output.

A miniature timing belt is used to tie the two halves of the drive system

Figure 4. Crossed-Roller Slide with Ball Screw Through the Center



together. In order to clear the ram the minimum minor diameter of the pulleys is .66 inches. Therefore two Winfred Berg 8TP3-30 (40 pitch, $\frac{3}{16}$ inch bore, .780 pitch diameter) timing belt pulleys were purchased. From belt length calculations:

$$L = 2C\cos\phi + \pi\frac{(D+d)}{2} + \pi\phi\frac{(D-d)}{180} \tag{2}$$

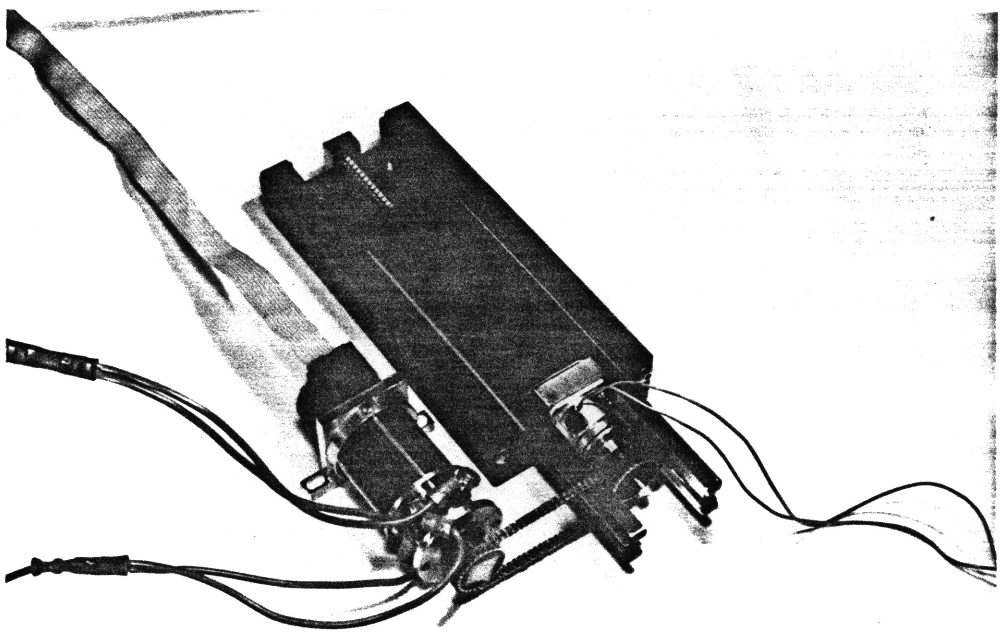
$$\phi = \sin^{-1}\frac{(D-d)}{2C} \tag{3}$$

where L is the pitch length of the belt, D is the pitch diameter of the large pulley, d is the pitch diameter of the small pulley, and C is the center distance, a 7.18 inch pitch length belt was selected to keep the motor and crossed-roller slide close together.

4.1.2. Geometric Design

The major problem with using a crossed-roller slide then became a problem of interfacing a drive system with the linear bearing. Since minimization of size and weight were important design specifications, placing a screw through the center of the crossed-roller slide was optimal. Though this significantly increased the difficulty of the system design, the advantages of using a screw and placing it there outweighed the problems introduced. In order to fit the nut through the slide, the slide was turned upside down. Thus, the original base became the moving portion or ram. In addition the nut on the ball screw had to be machined in order to make it fit within the crossed-roller slide. Since the working region of the nut is only .4 inches in diameter and the nut is .75 inches in diameter, the nut was flatted on two sides (Figure 4, Note: the nut has been removed from the hole in the moving ram).

Figure 5. Jig Fully Open



After the screw was fit through the center of the crossed-roller slide, the bearings for the screw shaft had to be located. One option was to extend the screw so that the bearings cleared the moving ram of the crossed-roller slide. In general this would be poor design practice since the screw adds considerable inertia to the system, the screw would have a much lower buckling load, and the machine would double in overall length. The only option was to design the bearings and their mounts so that they clear the ram as it rolls past the ends of the screw (Figures 5 and 6).

In this configuration the two bearings could then be attached directly to the crossed-roller slide. This enables the bearings to be preloaded before the system is mounted onto the rotary table. The preloading is accomplished by some shims and a wavey spring washer. The preload pushes the small radial ball bearing against the thrust bearing so that the jaw position is repeatable for movement in both directions. Since most of the measurements will be taken only when the jaw is closing, the preload is not an absolute necessity (Figure 7).

The journals on the screw required special machining since they were non-standard. The only bore available in the metric thrust bearing was .3150 inches, which is larger than the .298 inch diameter screw. In addition, all of the small components—the motor, encoder, brake, pulleys, and miniature bearing were selected to retain a $\frac{3}{16}$ inch standard bore. Therefore a custom adapter had to be manufactured. This adapter was pressed onto a $\frac{3}{16}$ inch diameter journal which was machined on the 17-4ph stainless steel screw. The machining of the steel adapter was critical since it was being used in an ABEC 7 precision ball bearing. The outer diameter must be within a tight tolerance, $.3105 \pm .0001$ inches, and since it has a

Figure 6. Jig Fully Closed

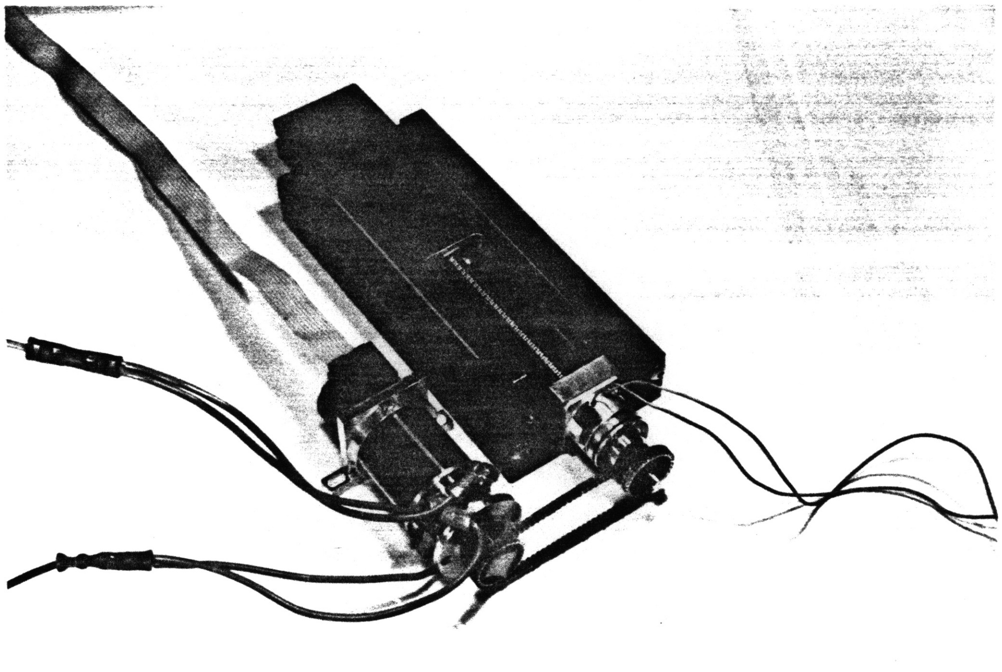
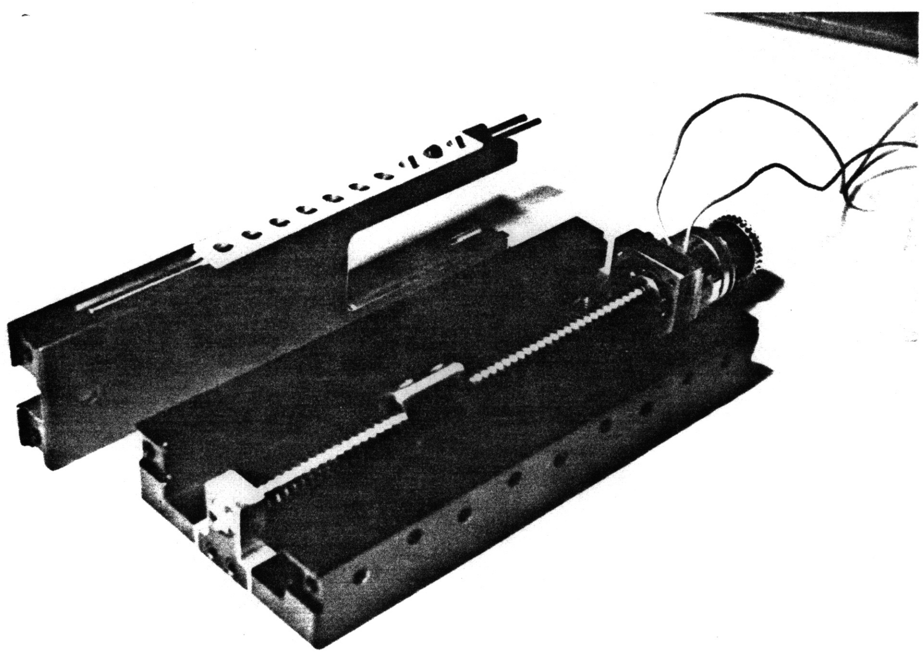


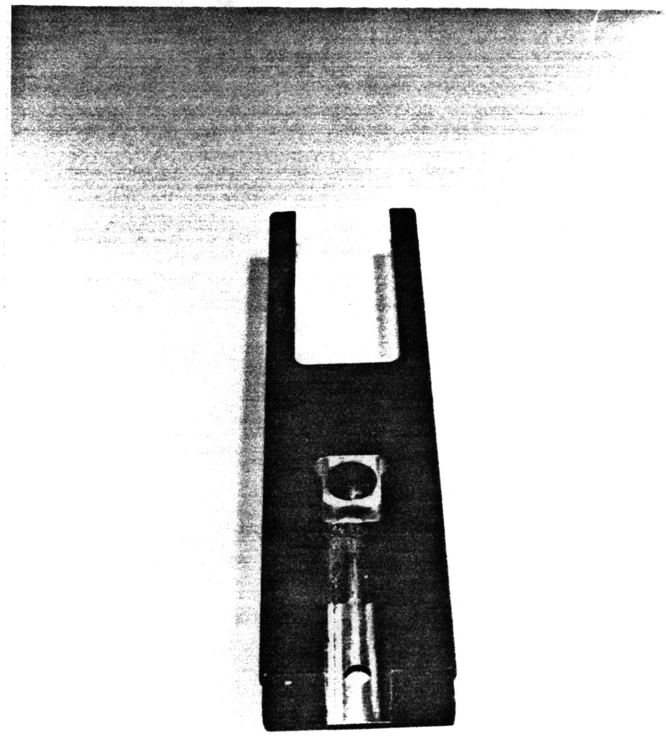
Figure 7. Screw Bearing Blocks



shoulder, it would be difficult to grind.

This design for the drive system raised some crucial design decisions concerning the moving ram of the crossed-roller slide. In cutting the ram so that it provided

Figure 8. Crossed-Roller Slide Ram



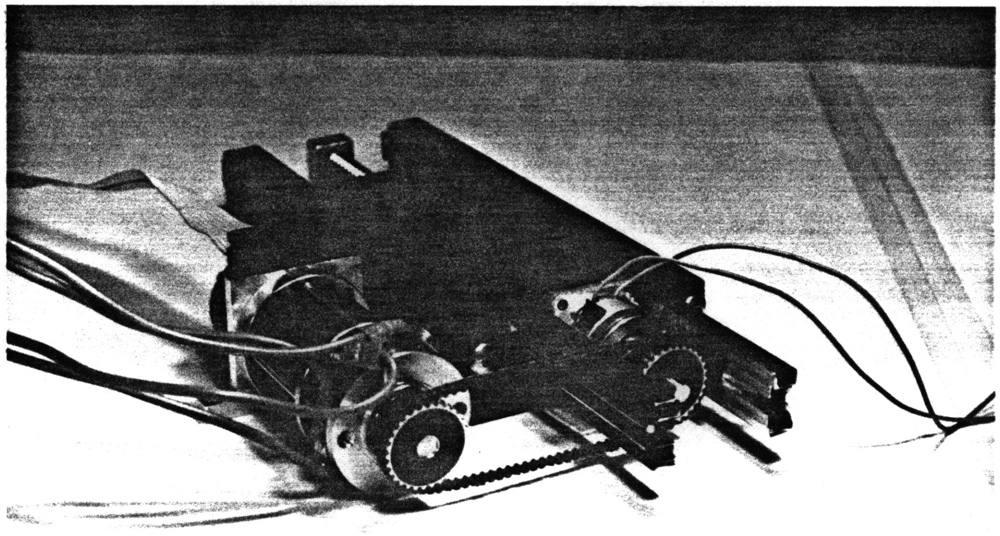
clearance for the drive system, care had to be exercised so as to minimize warpage of the side channels (Figure 8). If the 6061 aluminum bar had any internal stresses, cutting material away would create distortions. Care had to be exercised in the design so that distortions, of unpredictable magnitude, could not significantly affect the straight line accuracy of the crossed-roller slide.

On the end with the miniature bearing, a slot was cut away so that the bearing block could fit up inside the ram (Figure 8). The slot in this end was not cut all the way through the aluminum ram. A thin wall was left in order to restrain parts from falling into the crossed-roller slide and interfering with the screw. This wall had to be thick enough so that it would not crush under impact, yet thin so that it would not force a large distortion in the ways.

On the end with the thrust bearing and brake, a slot was cut through the ram (Figure 8). This was shaped this way in order to allow clearance and to keep distortions minimized to only some bowing of the "fork" and little actual misalignment of the ways which destroys straight line motion.

Since the ram passes by the bearing mounts, some care had to be taken

Figure 9. Timing Belt Arrangement

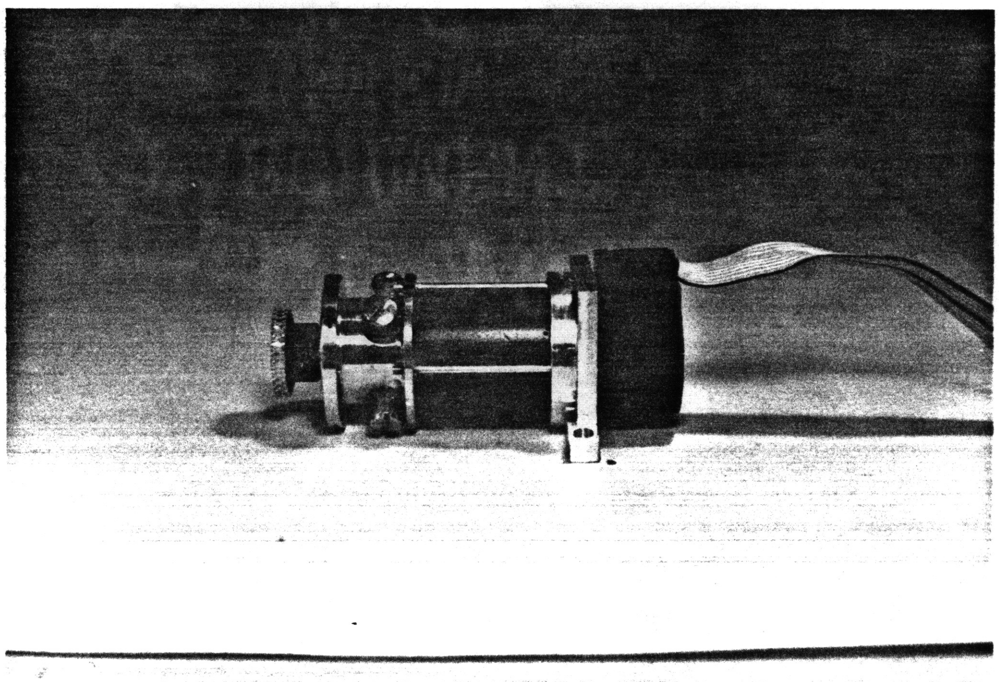


in attaching the three main drive elements—motor, optical shaft encoder, and brake. In order to reduce the length of the system and best utilize the table space, pulleys and a timing belt were used so that the motor could be located parallel to the crossed-roller slide. For clearance and mounting considerations, the brake was fastened to the outside of the thrust bearing housing. By squeezing the brake into this location, no extra space is wasted—the crossed-roller slide ram passes around the brake unit. The timing belt pulley then transmits the power from the motor shaft. In order for the pulley and belt to clear the top ram of the crossed-roller slide, the pulley on the screw must have at least a .66 inch minor diameter—large enough for the belt to clear the ram (Figure 9). In addition the pulley can not have more than a .97 inch pitch diameter so that the ram can clear the pulley as it rolls past.

The motor mount was designed so that it would also serve as an encoder adapter plate. It is a $\frac{1}{4}$ inch thick plate which was machined around the top to .163 inches (nominal) to leave .5 inch of motor shaft for the encoder. A locator hole for the motor and the hole pattern for the encoder were then machined into it. The base was slotted to allow for belt replacement and tensioning without removal of either piece from the table (Figure 10).

Both the moving and stationary walls were designed for minimum weight with acceptable amounts of deflection under a 180 pound load. Design of the moving wall posed several unique problems. The wall was required to be lightweight yet the face could not be aluminum for it would become marred. Therefore, a .1 inch steel plate was attached to an aluminum wedge. In addition, care had to be taken in mounting the wedge to the ram because these screws would have to transmit 180

Figure 10. Motor and Encoder Mount



pounds of force to the ram. This was accomplished by carefully locating 2 #10-32 and 2 #8-32 screws (the largest possible) in the ram in the most sturdy location on the ram. This was necessary because the intensive machining that was performed on the ram left relatively little material for tapping large screw holes.

The actual design of the aluminum wedge for the wall was also difficult—deflections had to be small compared to the .002 inch accuracy specification. First, the face of the wedge was modeled as a beam (Figure 1) and the deflection formula for a cantilevered beam under a distributed load:

$$\delta_{max} = \frac{w_a l^4}{8EI} \quad , \tag{3}$$

where δ_{max} is the maximum deflection, the quantity w_a is the total load on the wall, l is the length of the beam, E is the elastic modulus, and I is the area moment of inertia of the beam, was used. For a rectangular cross section the inertia is expressed by:

$$I = \frac{bh^3}{12} \quad , \tag{4}$$

where b is the width, and h is the thickness. Using the constants:

$$\begin{aligned} w_a l &= 180 \quad \text{pounds} \\ E &= 10^7 \quad \text{psi} \\ l &= 2 \quad \text{inches} \\ b &= 1.5 \quad \text{inches} \\ h &= .3 \quad \text{inches,} \end{aligned}$$

equation (4) gives:

$$I = 3.38 \times 10^{-3} \quad \text{inches}^4$$

and equation (3) therefore yields:

$$\delta_{max} = .0053 \quad \text{inches.}$$

By adding the gusset behind this wall (Figure 1) the model becomes that of a tapered beam. Interpolating from the tables of Roark and Young [1], where I_A is the inertia of the free end as calculated above and (from equation (4)),

$$I_B = .23 \quad \text{inches}^4 \quad ,$$

δ_{max} must be multiplied by a factor of $\approx .03$ or:

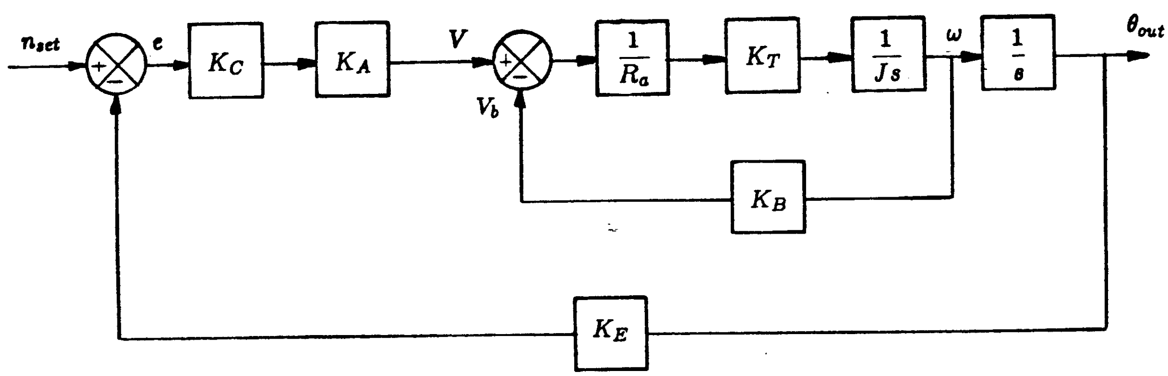
$$\delta_{actual} \approx .00016 \quad \text{inches}$$

for a 180 pound load.

The second, stationary wall (shown in Figure 1 as a 1 inch thick black rectangular wall) will actually be .6 inches thick and have a passage through it for the ram of the crossed-roller slide. The wall will be gusseted in the front and bolted to the crossed-roller slide base plate. It was also designed for deflections on the order of δ_{actual}.

4.2. Control System Design

Two types of controllers were implemented on the fixture, a proportional controller (P-type), and a proportional-plus-derivative controller (PD-type). Using the following block diagram of the system and controller:



where

$$
\begin{aligned}
K_C &= \text{Controller} \\
K_A &= \text{Amplifier Gain} \\
R_A &= \text{Motor Armature Resistance} \\
K_T &= \text{Motor Torque Constant} \\
J &= \text{Total Inertia}(J_m + J_s + J_l) \\
J_m &= \text{Motor Inertia} \\
J_s &= \text{Screw Inertia} \\
J_l &= \text{Reflected Inertia of the Load} \\
K_B &= \text{Back EMF of Motor} \\
K_E &= \text{Encoder Gain} \\
n_{set} &= \text{Set Point} \\
n_{out} &= \text{Output} \\
e &= \text{Error in Encoder Counts} \\
V &= \text{Voltage Output from Amplifier} \\
V_b &= \text{Back EMF Voltage} \\
\omega &= \text{Angular Velocity of Screw} \\
\theta_{out} &= \text{Angular Position of Screw,}
\end{aligned}
$$

the following transfer function is obtained (for a detailed discussion of the control of second order systems see Ogata [2]):

$$\frac{\theta_{out}(s)}{\theta_{set}(s)} = \frac{\frac{K_C K_A}{K_B}}{\frac{R_A J}{K_T K_B} s^2 + s + \frac{K_C K_A K_E}{K_B}} \quad . \tag{5}$$

This is of the form:

$$\frac{C}{s^2 + \frac{1}{\tau} s + \frac{C K_E}{\tau}} \quad . \tag{6}$$

Proportional Controller

For the P-type controller, the system natural frequency and damping ratio are given by:

$$\omega_n = \sqrt{\frac{K_C K_A K_E}{\tau K_B}} \tag{7}$$

$$\varsigma = \frac{1}{2\tau\sqrt{\frac{K_C K_A K_E}{\tau K_B}}} \quad . \tag{8}$$

From the values in Appendix I, and setting ς equal to one for critical damping,

$$\frac{1}{\tau} = 186 \quad \frac{1}{\text{rad} \cdot \text{sec}}$$

and

$$\omega_n = 8649 \ \frac{\text{rad}}{\text{sec}} \quad .$$

For a proportional controller this gives

$$K_C = .05 \quad .$$

This controller was implemented in FORTH on a PDP 11–23 computer (see Appendix III).

Proportional-Plus-Derivative Controller

For a PD-type controller the controller gain, K_C is replaced in equation (5) with $(K_p + K_d s)$. This type of control will increase the stiffness of the system. For this system, the natural frequency is given by:

$$\omega_n = \sqrt{\frac{K_p K_T K_E K_A}{R_a J}} = \sqrt{\frac{K_p}{A}} \tag{9}$$

and damping ratio by:

$$\varsigma = \frac{\frac{K_B}{K_E K_A} + K_d}{2\sqrt{\frac{K_p R_a J}{K_T K_E K_A}}} = \frac{B + K_d}{2\sqrt{K_p A}} \tag{10}$$

where A and B are system parameters. By choosing ς equal to one, a relationship between K_p and K_d is obtained. Finally, the controller constants can be determined by selecting either a desired natural frequency or open loop zero.

From some approximations of the motor's maximum stopping power, and the root locus plot of the open loop poles, an ω_n of 250 radians per second was chosen. This corresponds to a zero at 200 radians per second—slightly larger than the motor pole at 186 radians per second. Therefore:

$$K_p = .35$$
$$K_d = 1.75 \times 10^{-3}$$

These constants were converted into difference equations by a step-invariant approximation to the z-transform:

$$H(z) = \frac{(z-1)}{z} Z\left[\mathcal{L}^{-1}\{\frac{H(s)}{s}\}\right] \tag{11}$$

where $H(s) = K_p + K_d s$. This gives:

$$H(z) = \left[K_p + \frac{K_d}{T}\right] - \frac{K_d}{T} z^{-1} \tag{12}$$

were T is the sampling interval. Using a sampling time of 2 milliseconds the difference equation for the controller output becomes:

$$[1.22]e_{current} - [.88]e_{last} \quad .$$

This was the PD-type controller implemented on the system.

5. Preliminary Testing

5.1. Straight Line Accuracy

After the extensive machining of the moving ram, the straight line accuracy of the roller slide was in question. This was experimentally measured with a .0001 inch reading dial indicator. The crossed-roller slide was found to have less than .0002 inch deviation from the horizontal plane.

5.2. Positioning

The jig was operated with both types of controllers—P-type and PD-type. Both worked well, however, the PD controller seemed to have greater stiffness for seemingly the same amount of overshoot. The actual quantity of overshoot was not yet determined, thus the controller constants could not be adjusted for critical damping.

During preliminary testing, the ball screw was run as fast as 6000 RPM—three times faster than the rated speed. This most likely damaged the delrin deflectors in the ball nut. In addition, the runout of the screw journals was determined to exceed acceptable tolerances. The extent of the nut damage and the effect of these problems on the linear accuracy of the device have not yet been determined. Most likely, the screw will be used for continued testing and then a new one will be installed before placing the device into service on the robot.

6. Design Improvements

After construction and testing of the fixture, several design improvements became apparent. By changing some of the mechanical components in the system, performance can be greatly improved. In the present configuration, the clamping force is limited by the buckling load of the ball screw. In addition the ball screw has a delrin ball deflector which limits the speed to approximately 2000 rpm. These two limitations can be overcome by using a slightly larger screw ($\frac{3}{8}$ inch dia.) with a steel ball nut. Larger screws normally have critical speeds of approximately 3000 rpm.

Since 3000 rpm is as fast as the device should ever operate, the motor should be geared down with a 2.66:1 belt ratio. This would bring the 8000 rpm no load speed of the motor down to a 3000 rpm screw speed. Adding this transmission ratio the clamping force, in turn, increases by a factor of 2.66. However, this raises the clamping force beyond the maximum load rating of the thrust bearing. Therefore the screw lead may be increased. The increased screw lead improves the maximum closing speed of the jig while reducing its potential clamping force. A screw lead of .125 will increase the closing speed from 3.3 inches per second to 6.3 inches per second while simultaneously maintaining the clamping force over 200 pounds.This is not the maximum lead that can be used with the Clifton Precision motor while still having the ability to clamp with more than 200 pounds, however this combination of transmission ratios keeps the motor from overheating at maximum clamping.

Though the distortion of the ram seemed to be negligible, the ram should have been machined before the ways were cut. This would guarantee accuracy in straight line motion so that the risk of wasting many man hours of difficult machining would be minimized.

The construction and testing of this device suggests that a heavy duty version of this jig can be readily built without much redesign. The motor and transmission combination described above is capable of 400 pounds clamping force. For this type of loading, the angular contact bearing should be replaced by a combination of a .5 inch pure thrust bearing and a small radial bearing placed back to back. This would dramatically increase the thrust capacity of the clamp.

In addition, the aluminum ram of the crossed-roller slide would have to be reinforced. A thin steel plate added to the top of the ram should supply enough support so that the ram will not distort under load. This would add very little height to the clamp since most of the reinforcement would be required behind the moving jaw—a place where parts should never be placed.

The only remaining problem would then be to increase the braking torque. Several options are available to do this. One is to purchase a larger brake. This would solve the problem but would introduce many geometric problems. A second choice would be to move the existing brake to the motor shaft. This would increase the brake's holding torque by the belt ratio, thus the motor would have less power dissipation at full clamping.

This alteration, however, would require some hardware changes. Either the motor shaft would have to be extended to accommodate the brake or the brake would have to be mounted on a separate bracket so that only the rotating disk of the brake would be attached to the motor. Though pressing out and replacing the motor shaft seems simple, care must be exercised to avoid damaging the very delicate samarium cobalt magnets. Using the second mounting arrangement should theoretically be successful, however, alignment of the brake and the motor, especially after belt tensioning, becomes a burden if not a critical problem.

7. Conclusion

In designing this device, the specifications that were most difficult to meet simultaneously were the clamping force and measurement capabilities. Since large forces result in large deflections of the structure, designing a device that can both measure accurately and clamp firmly was a difficult task. Since the application dictates that the structure must be compact and lightweight, deflections, which can very easily approach the magnitude of the accuracy specification (.002") become a serious issue. Much of the effort was directed toward making the many components of the servo system fit in a small package while retaining the stiffness required to meet this accuracy specification.

The next step is to use this device for assembly and machining while carefully evaluating its effectiveness. This information could then be used to alter the design specifications. Thus the features that should be incorporated into an optimal fixture for programmable automation may be determined.

References

[1] Ogata, Katsuhiko, *Modern Control Engineering*, Prentice-Hall, Inc., 1970.

[2] Roark, Raymond and Warren Young, *Formulas for Stress and Strain*, McGraw-Hill Book Co., 1975.