A FOUR-AXIS MICROCOMPUTER-CONTROLLED PROBE POSITIONING SYSTEM FOR FLOW MEASUREMENT

by

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A thesis submitted to the Faculty
of the Graduate School of the State University
of New York at Buffalo in partial fulfillment
of the requirements for the degree of
Master of Science

February 1981

ACKNOWLEDGEMENTS

I would like to express my sincere appreciation and gratitude to Professor Andres Soom for his invaluable advice and encouragement throughout this project, without which it would have been interminable.

I would like to thank Professor William K. George for enthusiastically supporting my work, most tangibly with a Research Assistantship and funds to build the positioning system.

I would also like to thank Professor Hinrich Martens for his invaluable advice on microprocessors. I am very grateful to Norm Wagner who spent many long hours in the Machine Shop manufacturing the traverser mechanism, and to Eileen Graber for typing the thesis and helping with the purchase of innumerable hardware items for the project.

I gratefully acknowledge support for this project from the Air Force Office of Scientific Research through Contract No. AF49620-78-C-0047 entitled, "Cooperative Investigation of the Noise Producing Region of an Axisymmetric Jet", Contract No. F49620-80-C-0053 entitled, "Cooperative Investigation of Jet Flows" and the National Science Foundation Grant No. ENG76-17466 entitled, "Radiated Noise and the Structure of Jet Turbulence".

Finally, I would like to thank Edward Szczepanski for helping with the electrical design of the controller, Peggy Pfeiffer for helping with the documentation of the project, and Moog Inc., East Aurora for the use of their facilities and equipment at various stages of the project.

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ABSTRACT

This thesis describes the design and implementation of a four-axis microcomputer-controlled probe positioning system for automated flow measurements. The structural design aspects of a traverser mechanism consisting of three perpendicular linear axis and one rotary axis are examined. The process of selection of stepper motor drives for the traverser is presented in detail. The design of hardware and software for the microporcessor based controller is explained. Finally, the manufacture of the positioning system and its performance are described.

LIST OF FIGURES

CHAPTER I

1. INTRODUCTION

The purpose of this thesis is to describe the design and implementation of a computer-controlled multiple-axis positioning system.

The system is to be used to position hot-wire velocity probes within the flow field of a large air jet. Figure 1-1 shows a schematic diagram of the region of operation of the positioning system. The probes are required to be positioned at any point within a rectangular parallelepiped 6 ft. long and 4 ft. square at the ends. The actual positions of the probes must be accurate to within approximately 0.01 in. This positioning must be accomplished rapidly with a complete corner to corner traverse not exceeding 30 to 40 seconds. This requirement is imposed by the number of velocity readings to be taken within the jet, which are of the order of several hundred thousand. This large number of measurements also makes computer-controlled automatic operation a necessity in order to complete the task within a reasonable time period.

The positioning system designed to fulfill these requirements is shown in Figure 1-2. The entire structure of the traverser is supported by the base frame which rests on eight adjustable legs. The base frame consists of three large I-beams braced by five transverse I-beams to provide rigidity. The base frame also provides the mounting surface for the longitudinal axis slideway. The transverse slide rides on the longitudinal axis slideway and carries the transverse slideway on its upper surface. The transverse slide also supports the vertical column and the vertical slide onto which the probes are mounted. The slideways are fully supported ground circular shafts. The slides are mounted onto these shafts through linear ball bearin
All slides are driven by individual stepper motors acting through recirculat

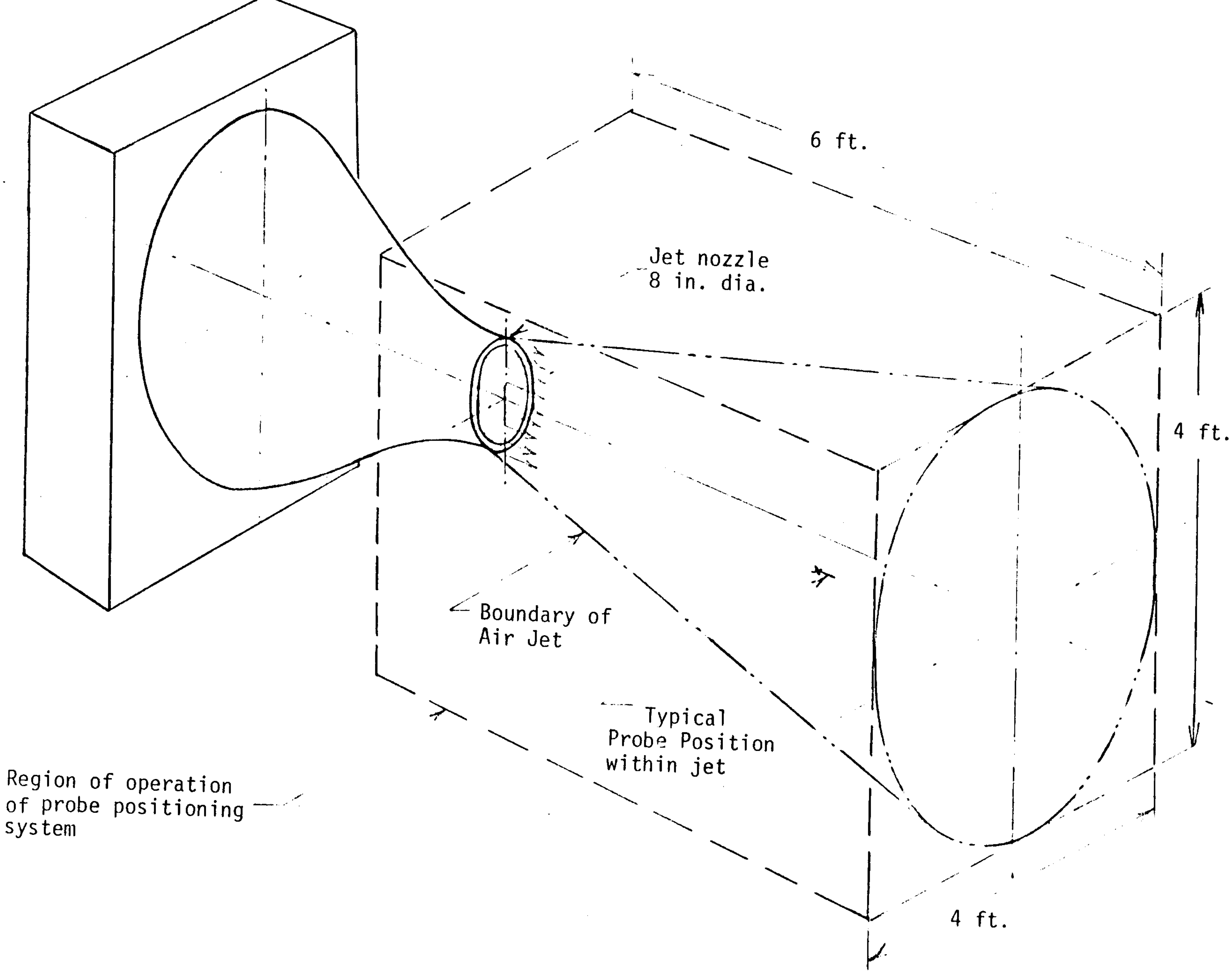


Figure 1.1. Schematic

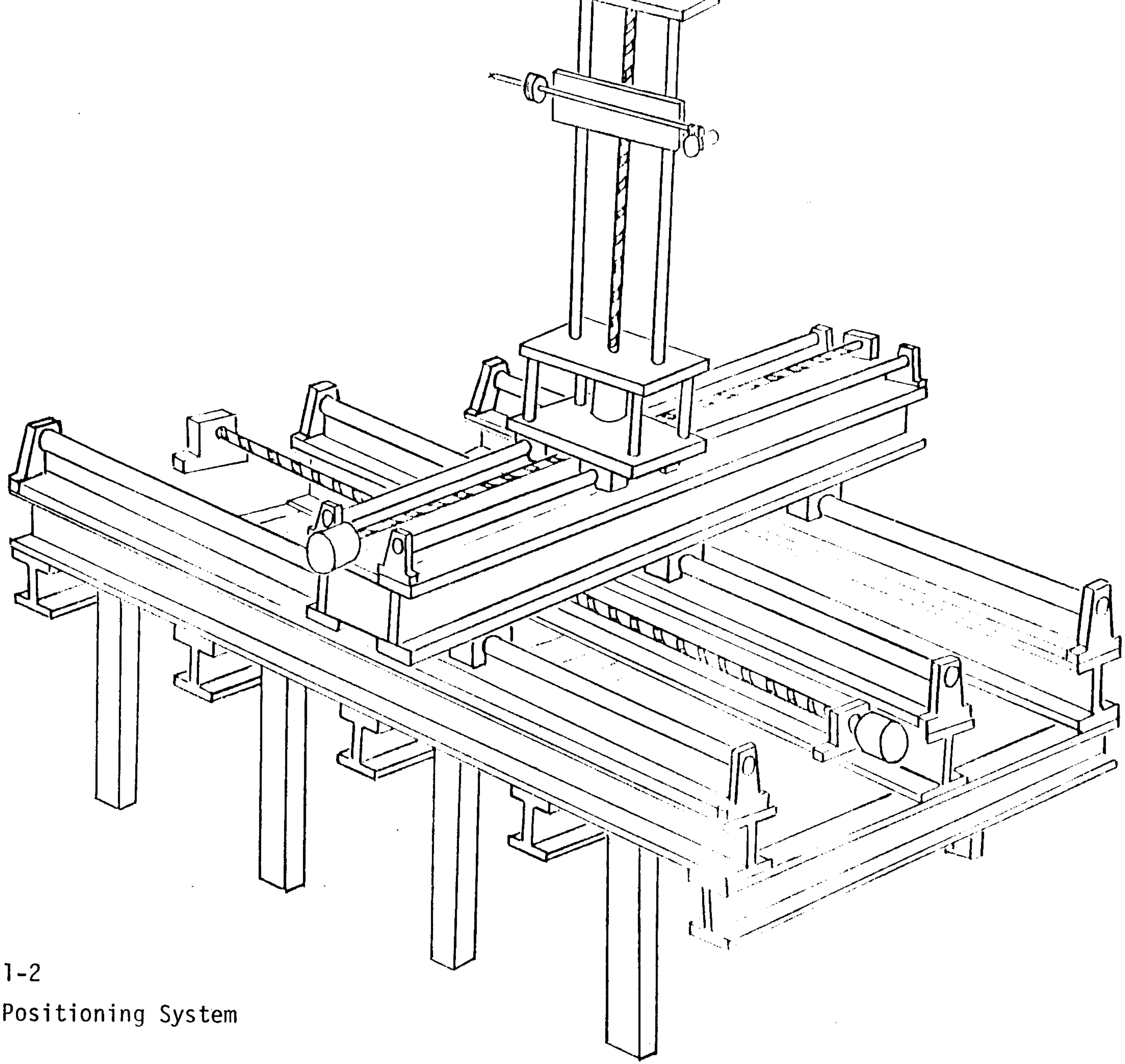


Figure 1-2
Schematic Diagram of the Positioning System

ball-type nuts and leadscrews. A block diagram of the drive system is shown in Figure 1-3. Drive for the stepper motor is supplied by electronic translator circuits which receive low level digital pulse inputs and convert them into high-power current pulses to drive the motors. The low-level digital pulses are supplied by a microcomputer dedicated to the task of controlling the positioning system. The microcomputer can receive positioning commands from either a human operator or another computer system. Manual mode operation is used to adjust and calibrate the positioning system whereas remote computer control is used to perform the main task of collecting flow data.

The following chapters discuss the design aspects of the various parts of the positioning system. Chapter 2 describes the design of the mechanical structure. Chapter 3 discusses the selection and design of the drive system. Chapters 4 and 5 describe the design of the microcomputer-based controller. Chapter 6 describes the performance of the entire device.

Although it has not seemed to be generally necessary to cite specific references in discussing the evolution of the design and in describing the system, bibliographic references on particular topics related to the design are given in Appendix A.

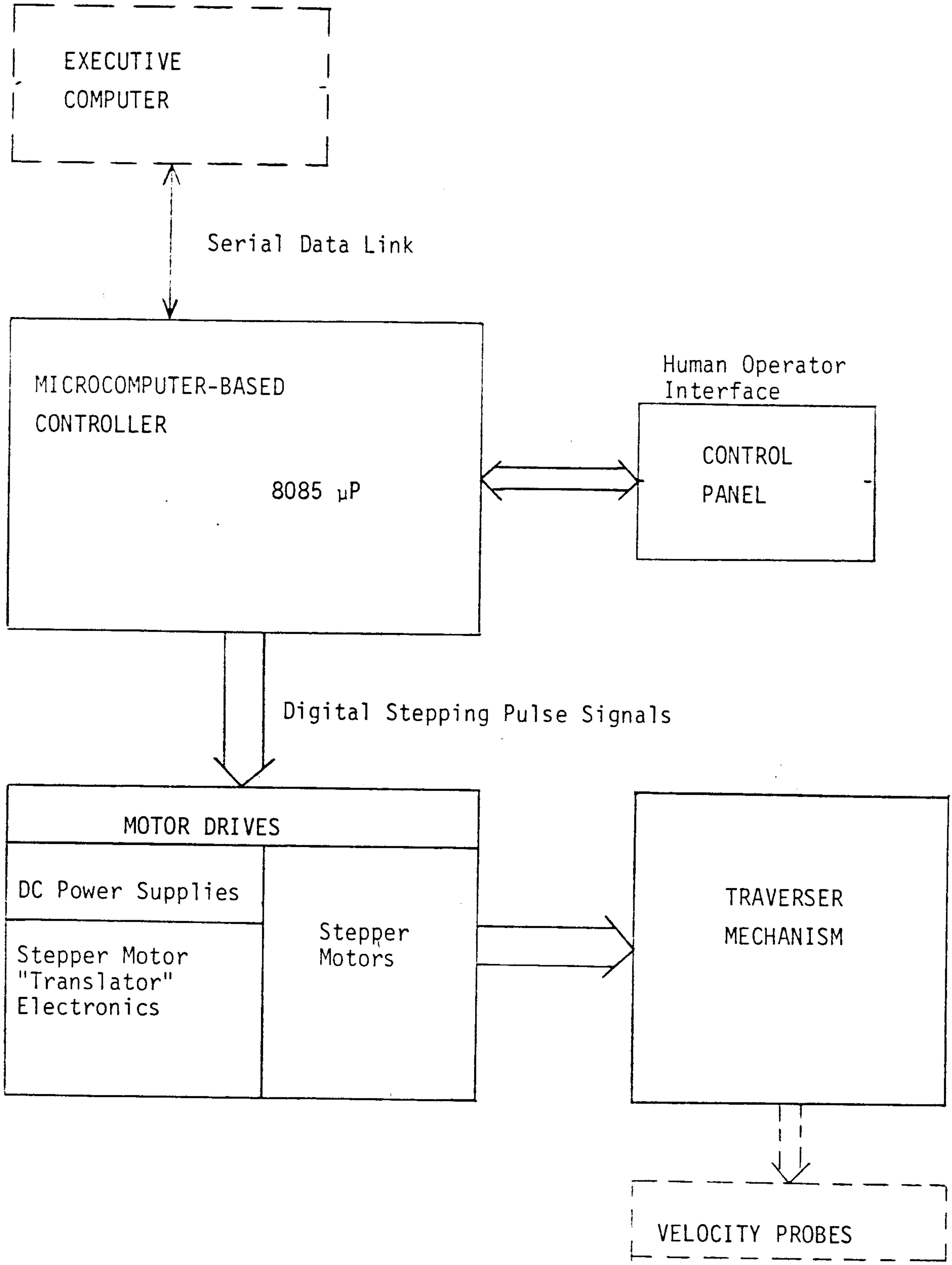


Figure 1-3 Block Diagram of Drive System

CHAPTER 2

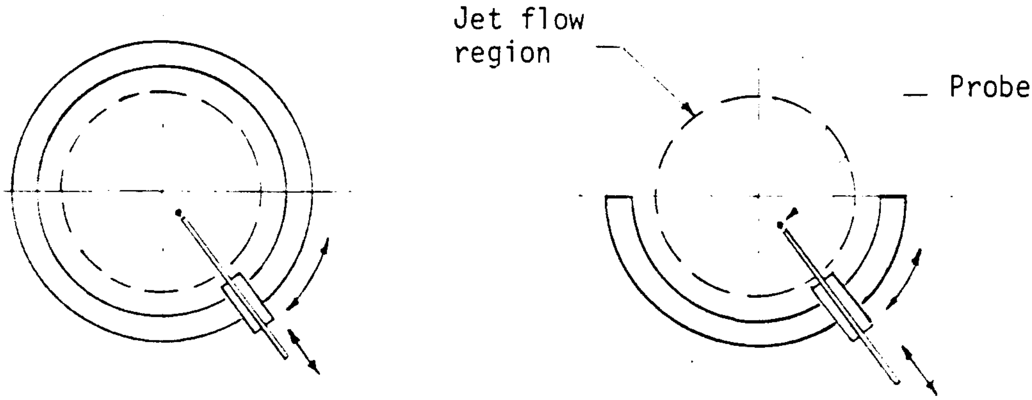
2. STRUCTURAL DESIGN OF TRAVERSER MECHANISM

2.1 Design Concepts

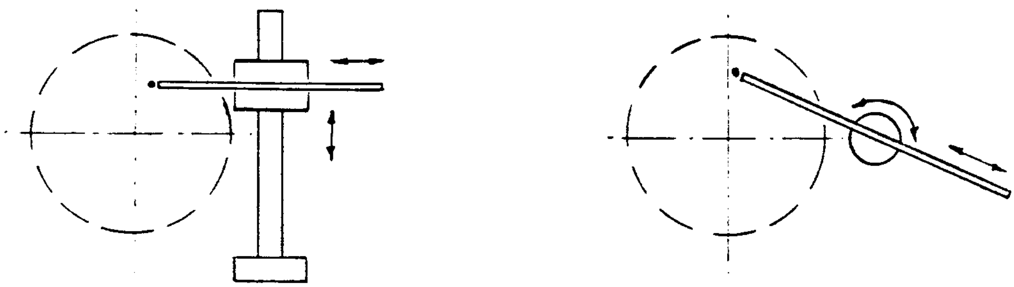
The mechanism used in the positioning system must be capable of positioning a probe at any point within a cylindrical zone 4 ft. in diameter and 6 ft. long in the direction of the jet. The positioning accuracy required is 0.01 in. (i.e. the actual position has to be within 0.01 in. of the desired position anywhere within the cylindrical zone).

Since flow disturbances would be caused by the presence of structural members within and around the jet flow region, the profile of the traverser mechanism must be designed to keep flow interference below a minimum level.

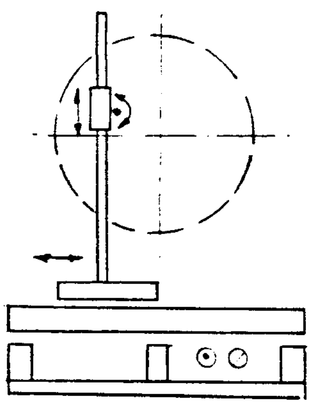
Figure 2-1 shows some of the mechanisms which were considered for this application. The arrangement finally chosen uses three mutually perpendicular linear slideways to translate the probe to any point within the operating zone. The probe can also be automatically rotated about its own axis. This arrangement is very flexible as it allows complex paths of travel to be synthesized out of successive positions in rectangular coordinates as in many numerically controlled machine tools. The linear slideways used here are easier to manufacture than circular or contoured slideways. This arrangement is also likely to be most reliable because of its simplicity. Other mechanisms using circular slideways would be bulky, expensive and hard to align with the axis of the jet at all points. Cantilevered mechanisms would keep the main structure out of the flow region but would have to be very rigid to maintain positioning accuracy. This rigidity requirement would make them very large and cause significant flow disturbances.



Circular Slideways



Cantilevered Slideways



Perpendicular
Linear Slideway

Figure 2-1 Types of Traverser Mechanisms

2.2 Description of the Structure and Slides

Photographs of the completed apparatus are included in Appendix E to which the reader may wish to refer.

The entire traverser mechanism is supported by the base frame which rests on eight adjustable vertical supports. The frame is built up of three longitudinal beams (7 ft. long and 6 in. deep wide-flange I-beams) bolted to five cross-members (4 in. deep standard I-beams) in a uniformly spaced grid. This construction provides very high vertical rigidity. The vertical deflection of the frame due to the weight of the traverser mechanism turns out to be negligibly small (less than 0.001 in.). The frame also provides large torsional rigidity by resisting the bending and twisting effects produced when the traverser slides are positioned halfway between supports.

Figure 2-2 illustrates the deflection considerations used in designing the base frame. The vertical deflections (Δx) are easily controlled due to the large vertical stiffness of the beam itself. The effect of angular deflections is large at the top of the vertical slideway (Δy) because of the long unsupported dimension, H.

Calculations show that the horizontal deflection effect due to θ is of the same order magnitude of the vertical deflection effect Δx.

The beam was designed to limit error due to angular deflection (while satisfying the vertical deflection criterion) to a maximum value of 0.005 in at the upper end of the vertical axis slideway. The details of the calculations are given in Appendix D.

The transverse slideway, which is perpendicular to the jet axis and lies in the horizontal plane carries the entire vertical axis and therefore has to be very rigid.

The deflection considerations used in designing this axis are similar to those used for the longitudinal axis slideway. However, unlike the

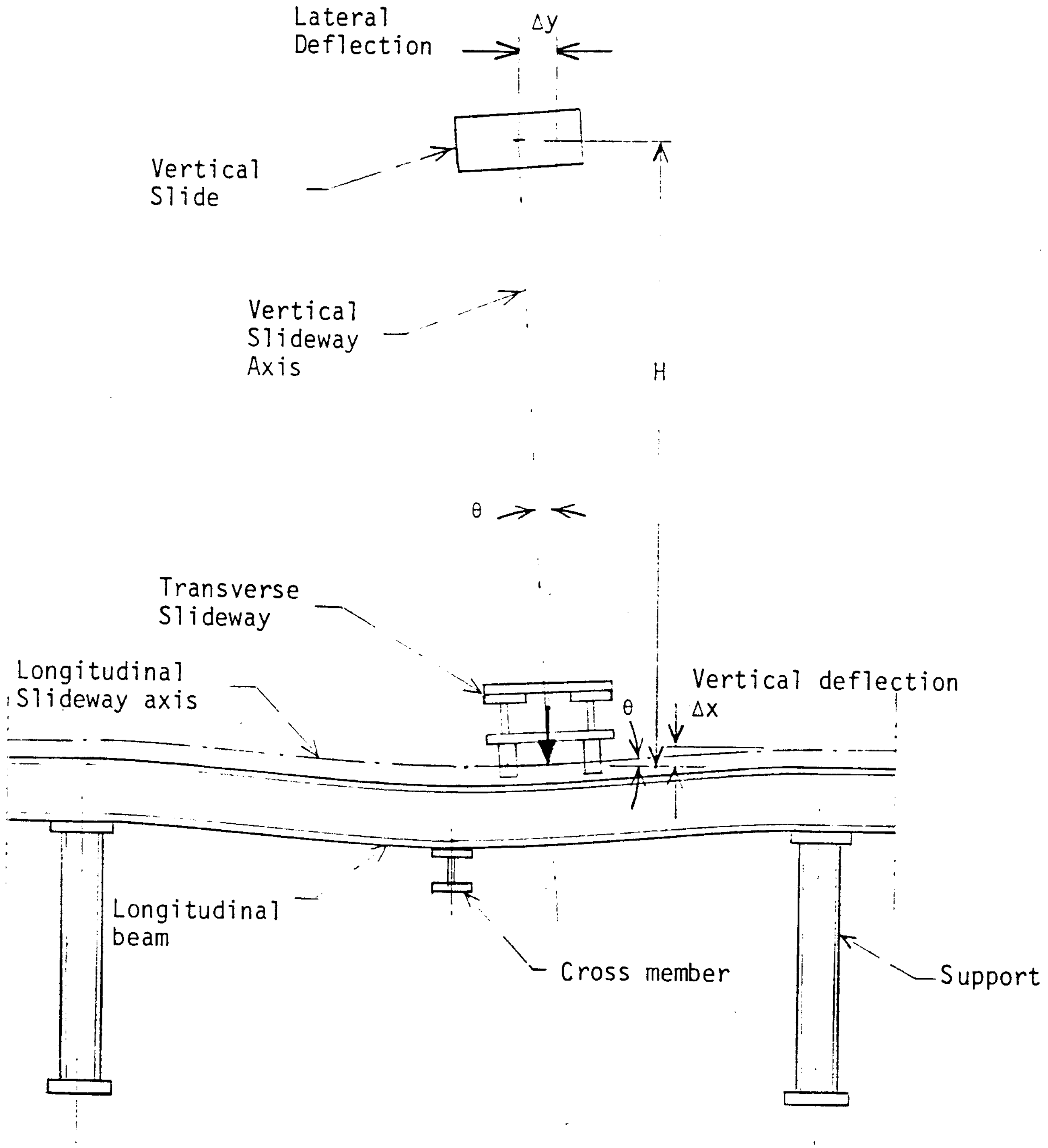


Figure 2-2 Deflection Considerations Used in Designing the Traverser Structur

longitudinal slideway, the transverse slide assembly has to traverse and be accelerated and therefore must be of low weight. To achieve both rigidity and low weight, a custom designed I-beam frame was built using aluminum bars 0.75 in. thick. The cross-section of the I-beam was designed to limit the deflection of the upper end of the vertical axis to 0.001 in. In the design of the longitudinal and transverse slideway frames, rigidity rathen than strength has been the main consideration.

The vertical slideway is mounted on top of the transverse slide and raised on stilts in order to keep the massive lower slideways out of the way of the air flow around the jet. Since the vertical axis has to stand directly inside the jet flow, it is very important to minimize the drag profile of the axis and to hold its upstream effects on the flow at the velocity probes to a negligible level. The design arrived at was a free-standing vertical slideway supported only at the lower end as shown in Figure 2-3. The slideway has two circular hardened and ground shafts which resist the twisting moment of the leadscrew and guide the slide vertically The leadscrew itself supports the weight of the vertical slide and probes. The vertical slide is built out of aluminum parts in order to minimize the inertial load during acceleration. The profile of the vertical slide facing the jet flow has been kept to a minimum. The vertical slide carries the probe on a shaft which can be rotated by a small stepper motor mounted on the slide itself.

The friction in the slideways associated with start-stop operation was minimized by using ball-bearing type linear bushings sliding over hardened and ground circular shafts. The leadscrews used to drive the slides are also ball-bearing type with recirculating ball nuts. These features allowed the use of relatively small motors. The leadscrew nuts have been preloaded to remove backlash. Preloading increases the torque required to turn the

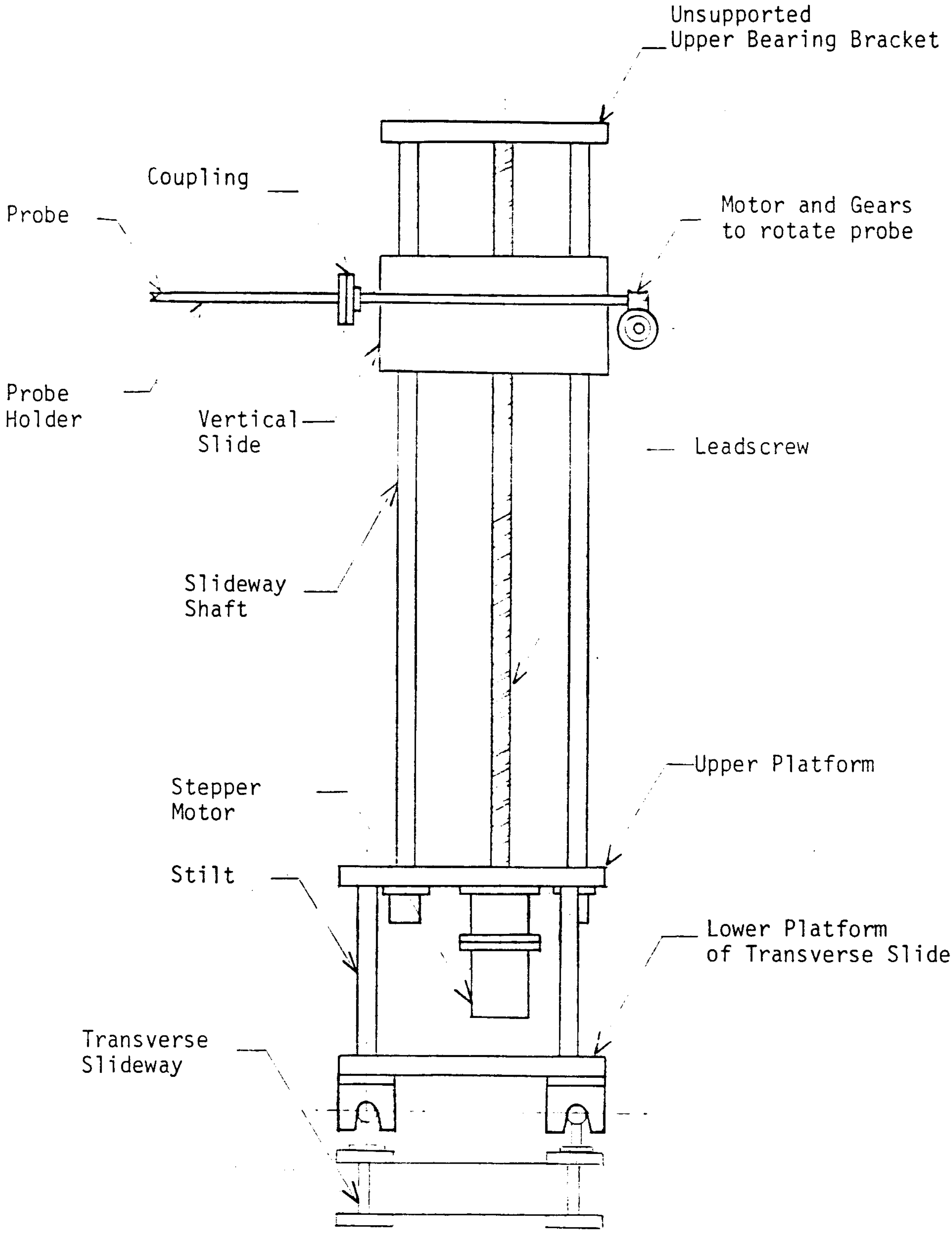


Figure 2-3 Vertical Slideway Structure

leadscrews, but provides a stiffer coupling arrangement.

The drive motors used for the slideways are stepper motors whose design is described in a later section. During operation, stepper motors produce a pulsating torque that rises to a peak and falls back to zero. At high stepping rates of 2000 steps per second, the torque pulsation causes fatigue loading of the mechanical coupling element between the motor and the leadscrews. Therefore the coupling has to be strong enough to withstand the fatigue loading. The coupling also has to tolerate small misalignments between the motor and leadscrew. If a flexible disc coupling is used it must be torsionally rigid enough to resist windup under motor action. The best commercially available coupling element for this purpose seemed to be a precision Oldham coupling whose backlash was well within the positioning accuracy requirements.

All the axes have been provided with means for aligning them with respect to the jet axis.

CHAPTER 3

3. POSITIONING DRIVES

3.1 Introduction

The function of the positioning drives is to accept position commands from the microprocessor-based controller in the form of electrical digital signals and physically move the slides to the commanded position. Two types of drive, stepper motors and DC servomotors were considered for use.

3.2 Stepper Motor Drives

Positioning drives using stepper motors were implemented as shown in Figure 3-1. The stepper motor drives the load through a leadscrew and ball nut arrangement. The translator electronics receive digital pulse inputs and energize the stepper motor windings in sequence to cause an angular step in position for every pulse. The position controller receives a digital position command signal from the main control computer or the manual control and decodes it to send the appropriate number of stepping pulses to the translator electronics. The operation of the controller is explained in detail in Chapters 4 and 5.

The stepper motor must, of course, be matched to its intended task. The first step in using the motor is to calculate the load inertia. The weight of the linear slide is calculated. This is converted into an equivalent rotary inertia seen by the motor due to the effect of the leadscrew. The rotary inertia of the leadscrew itself and the estimated inertia of the motor are added to the equivalent slide inertia to get the total inertia to be driven by the motor. Next the friction forces of the drive due to leadscrews, rotary bearings and the slideway are estimated. The required load-motion profiles of acceleration, velocity and distance over the fastest traverse are drawn. The maximum torque required to accelerate the load

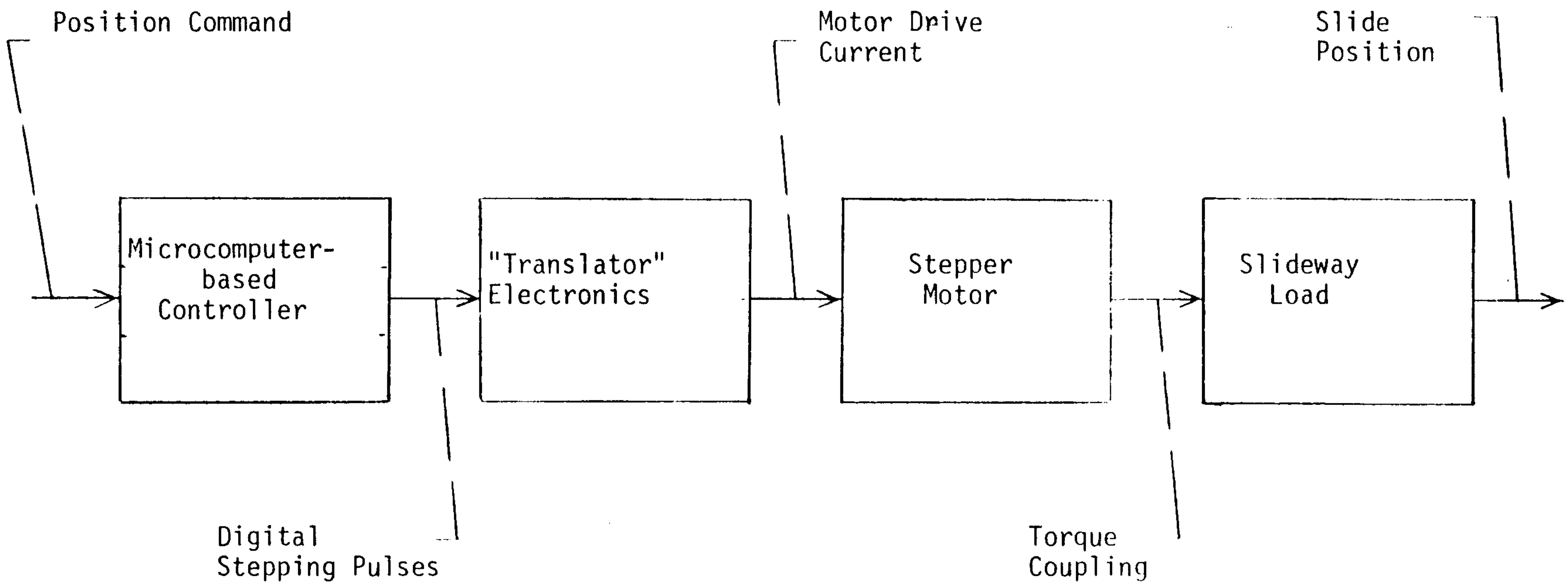


Figure 3-1 Block Diagram of Positioning Drives Using Stepper Motors.

inertia is calculated. If the motor lifts its slide against the force of slideway, the torque required to lift the slide is also calculated. The maximum torque required by the drive is the sum of the acceleration torque, friction torque and gravity torque. The least torque produced by the selected motor (usually at the maximum operating speed) should be greater than the calculated maximum torque.

The next consideration in selecting the motor is step size and operatin speed. The speed requirement is derived from the traverse time specificatio The number of revolutions that the motor has to make to perform an end-to-en traverse of the axis is dependent on the pitch of the leadscrew. If the pitch is small, the motor will have to make more revolutions to complete the traverse, but the inertia of the load seen by the motor as a rotary inertia will be reduced as an inverse of the square of the pitch. Also, if the pitch value is small, the motor has to have a higher speed capability in order to satisfy the traverse time specification. Keeping these factors in mind, several leadscrew sizes are chosen tentatively. The torque and speed requirements of the motor are calculated for each leadscrew. Stepper motors satisfying these requirements are chosen. The motor and leadscrew combination with the best price/performance chatacteristics is selected. An example of the step motor sizing calculations is shown in Appendix B.

Under normal operating conditions the stepper motor will take a step for every pulse sent out by the controller. Therefore the positioning syste can be operated in an "open-loop" mode. The controller need not measure the actual position to close a feedback loop. In order to operate in an open-loop, several factors have to be taken into account. The torque versus speed characteristics of the stepper motor shown in Figure 3-2 reveal that there is a significant falloff in torque as the speed increases and that the motor torque output is sensitive to the presence of load inertia.

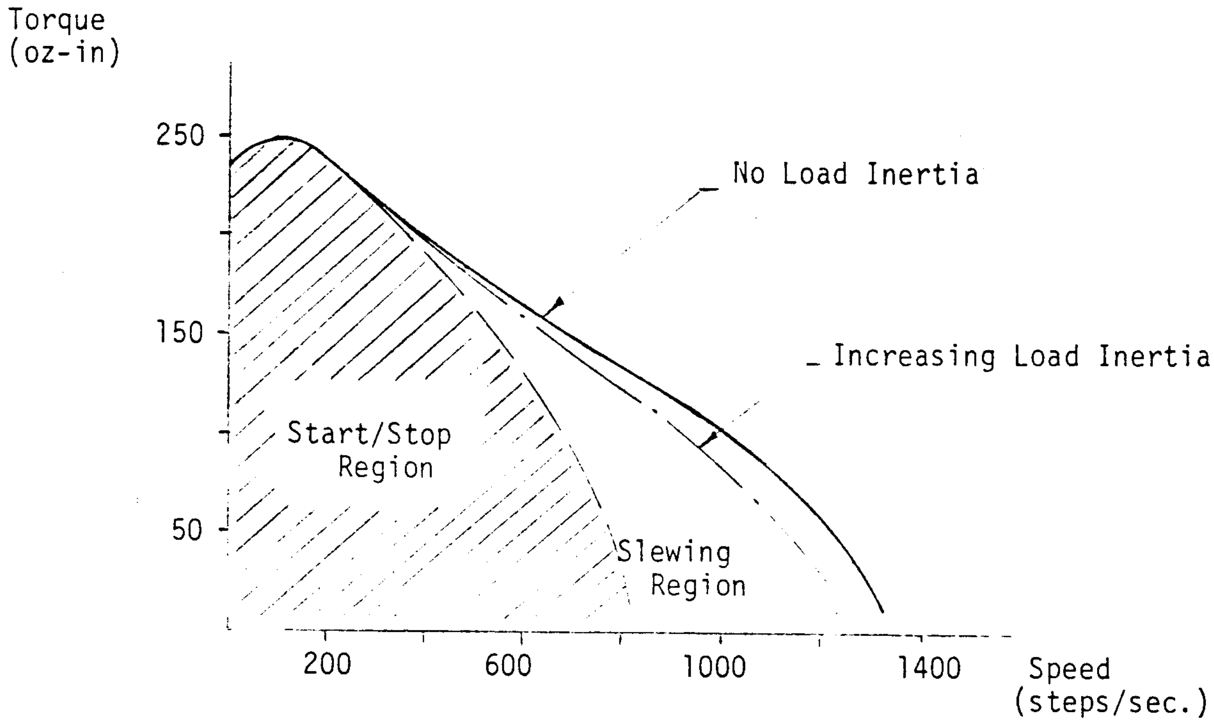


Figure 3-2 Torque vs. Speed Characteristics of a Stepper Motor

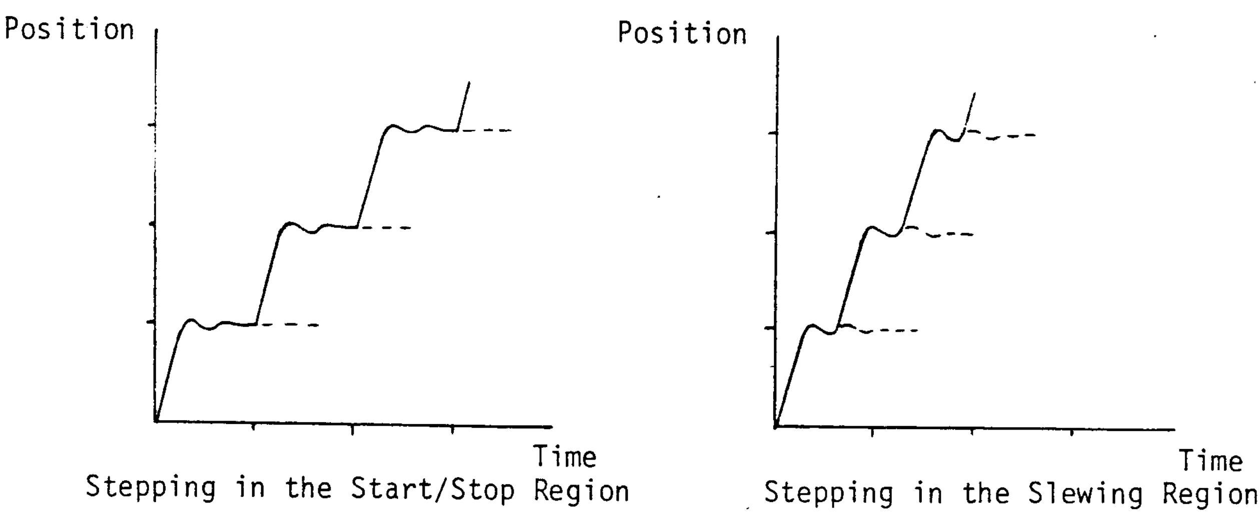


Figure 3-3 Dynamic Stepping Action

Within the start/stop region, the stepper motor will take a discrete step each time a pulse is received and if the pulses stop the motor will stop without taking any more steps. This is illustrated in Figure 3-3. In the slewing region the motor still takes a step for every pulse received, but the boundary between steps becomes increasingly blurred and if the input pulses stop suddenly the motor will take a few additional steps before coming to a stop, causing a loss of pulse-to-step integrity. If we wish to position the motor using a pulse train of constant frequency, then the speed must be within the start/stop region. If we start with a pulse rate beyond the start/stop region the motor cannot react fast enough and will lose pulse-to-step integrity. In order to use the stepper motor to its maximum capacity it has to be gradually accelerated through its start/stop region until it picks up enough momentum to respond to higher slewing rates. The rate of acceleration is limited by the response of the motor. The maximum slewing rate is usually limited by the torque required to maintain constant load motion. Before reaching its next position the motor must decelerate gradually into the start/stop region before stopping in order to avoid unwanted steps. Therefore, the controller must profile the pulse rates in addition to counting off the proper number of stepping pulses. If the motor is operated properly as explained above, we can assume proper pulse-to-step integrity and "open-loop" operation is adequate.

Stepper motor resonance is a factor that causes a loss of pulse-to-step integrity. At certain pulse rates during normal operation the motor torque drops off rapidly and the motor will not respond to stepping commands as shown in Figure 3-4. Instead the motor will stop and rotational vibrations start occurring. Though this resonance takes place over a very small range of speeds, it can cause significant positioning errors. This resonance is caused by the pulse rate coinciding with electromechanical natural frequency

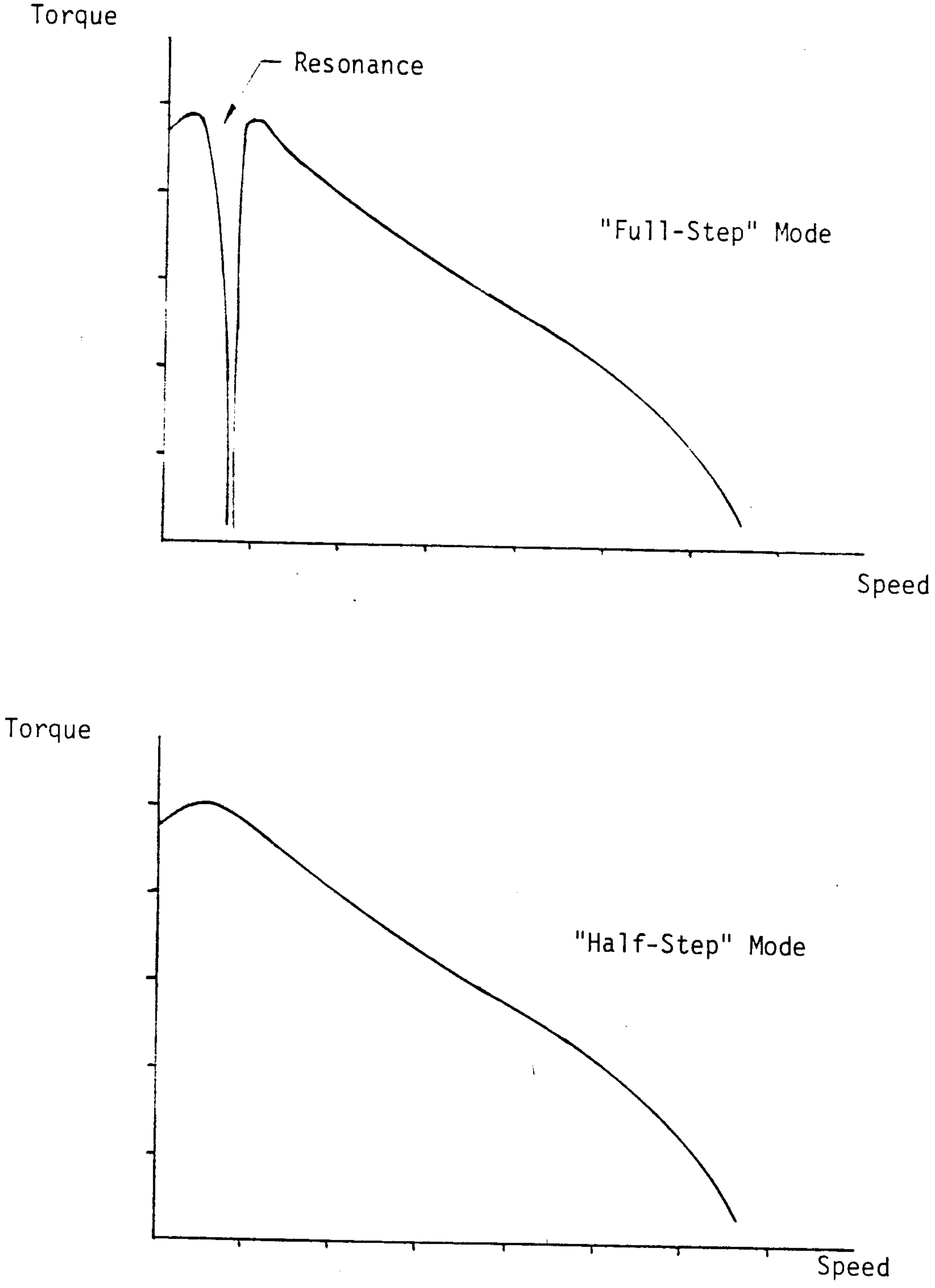


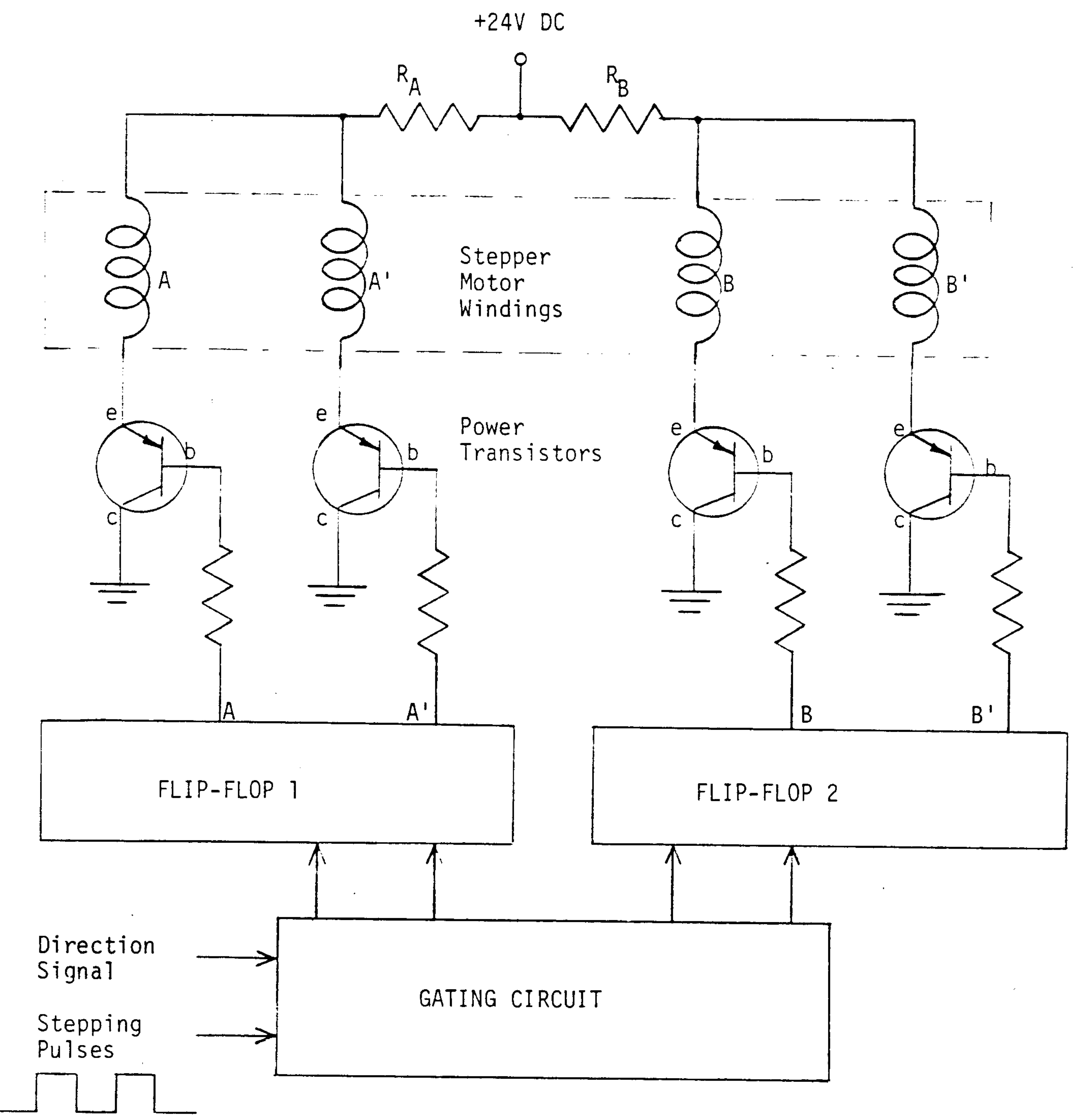
Figure 3-4 Stepper Motor Resonance

of the motor. The motor's natural frequency is a function of several factors including electrical parameters of the windings, rotor inertia and step size. In order to avoid resonance, the easiest factor to control is the step size of the motor. If two out of the four windings of the motor are always energized, the motor step size will be equal to the rated value called the "full step". If intermediate winding energization patterns are introduced having only one winding of the motor is energized, the motor will take "half-steps" for each command pulse. This reduction in step-size increases the motor's resonance frequency and moves it out of the operating speed range of the motor. The motor will now operate smoothly without resonance. The step-size modification is easily effected in the translator electronics described below.

The stepper motor is generally well-suited for computer control since the digital pulse outputs of the computer can be used with a fairly simple electronic interface to drive the motor directly as shown in Figure 3-5. Because of the discrete stepping action of the motor, there is a "detent" torque present even in the static condition which prevents position drift and other static errors. The position error of the stepper motor itself is only the error of one step and is not cumulative. Therefore the position error due to motor step size variation is unaffected by the length of traverse.

3.3 DC Servomotors

The other approach to positioning would be to use a DC servomotor in a "closed-loop" configuration with feedback. A block diagram of a closed-loop DC servo positioning system is shown in Figure 3-6. The DC servomotor would drive the slide through a leadscrew with reduction gears. This would reduce the effective load inertia seen at the motor and provide a mechanical



STEP	A	A'	B	B'
1	ON		ON	
2		ON	ON	
3		ON		ON
4	ON			ON
1	ON		ON	

L/R UNIPOLAR DRIVE

Figure 3-5 Translator Electronics

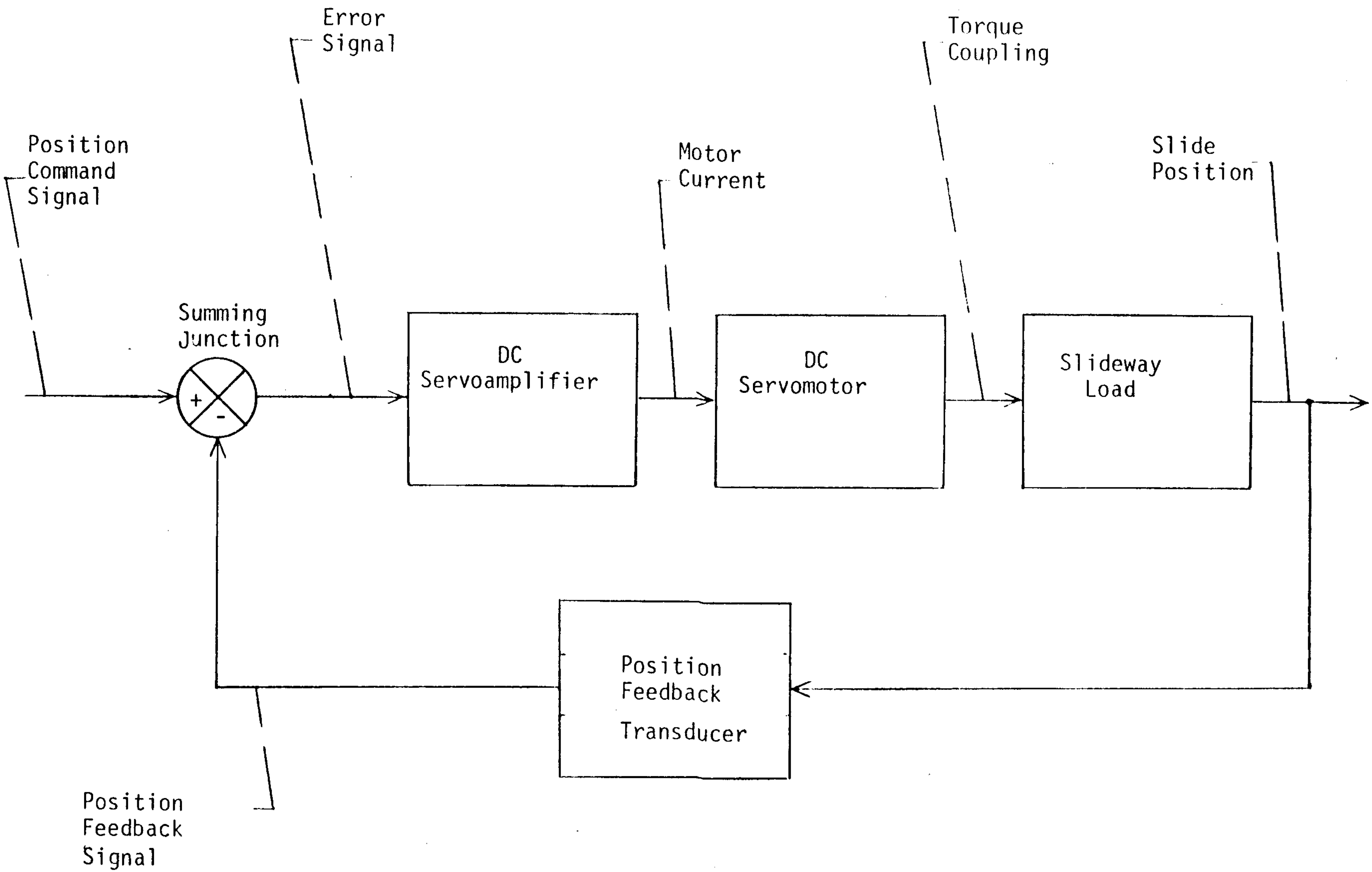


Figure 3-6 Block Diagram of Positioning Drives Using DC Servomotors

force advantage. The slide position has to be accurate to within 0.01 in. over 48 in. of travel. This calls for a minimum resolution of approximately 0.02 percent if an absolute position transducer is used. A linear position transducer of this length and accuracy would be prohibitively expensive, therefore we would choose to transduce the rotary position of the leadscrew and deduce the linear position from the leadscrew pitch. A typical leadscre pitch in the range of 0.25 in. per revolution would require a gear reduction of 192:1 to the position transducer in order to transduce absolute position over 48 in. of travel. Even a small backlash or nonlinearity in gearing would destroy the accuracy of the feedback signal. Therefore we would have to choose an "incremental" encoding scheme for position feedback. (The incremental feedback would be converted to absolute feedback at the summing junction). An analog servocontroller would also require electronic compensation to provide a position accuracy of 0.02 percent.

3.4 Comparison of Drives

Comparison of the two types of positioning systems resulted in the choi of the stepper motor system over the DC servomotor system. The stepper moto system only needs translator electronics to allow direct computer control of position. The DC servomotor needs a complex digital servoamplifier and the associated signal converters and compensation devices. The stepper motor system is therefore less expensive and less complicated than the analog DC servo system. The analog DC servo does have more torque capability and bett dynamic response characteristics than the stepper motor system. For the requirements of this application, the capabilities of the stepper motor are adequate. If torque and response were critical and could not be satisfied by stepper motors, we would need to choose DC servomotors.

We note that in either case, the final controlled value is the rotary position of the leadscrew shaft. The accuracy of linear position of the slide is dependent on the accuracy of the leadscrew pitch. Any pitch errors could be compensated for by calibrating the linear position of the slide versus the angular position of the leadscrew. Typically, leadscrew lead errors (about 0.001 in. per inch of screw length) are fairly constant over the length of the screw, therefore complete calibration at every point is unnecessary. In a computer-controlled positioning system it is a relatively easy task to store the calibration values and use them to modify the positio command appropriately should that be necessary.

CHAPTER 4

4. CONTROLLER DESIGN - HARDWARE

4.1 Microprocessor Control

A dedicated controller is used in the positioning system to enable convenient, manually operated alignment and calibration as well as remotely commanded positioning without tying up the time and resources of the executive computer system that manages the whole experiment. The controller independently accepts direct manual command inputs for free-running rapid traverse of the slides or for accurate point-to-point positioning. It also interfaces with the executive computer system for real-time automated operation of the flow experiment.

A microcomputer based system is an ideal candidate for this task. It easily generates the digital stepping pulses required to operate the stepper motor drives. It can be made to provide complex stepping pulse profiles, including acceleration and deceleration for rapid and accurate point-to-point positioning. It can be programmed to keep track of the absolute positions of the slides even though the actual positioning is done incrementally. By interfacing with a keyboard and display unit it accepts commands from a human operator. It communicates easily with the executive computer through a serial data link. By providing dedicated "intelligence" to the positioning system, it offloads the executive computer allowing better resource utilization. It can incorporate leadscrew calibration factors and backlash compensation which are "transparent" to the executive computer.

4.2 Preset Indexers

An alternative means of implementing the controller is to have a "preset indexer" control for each stepper motor. A preset indexer consists of two sections as shown in Figure 4-1. The Digital Pulse generation stage accepts

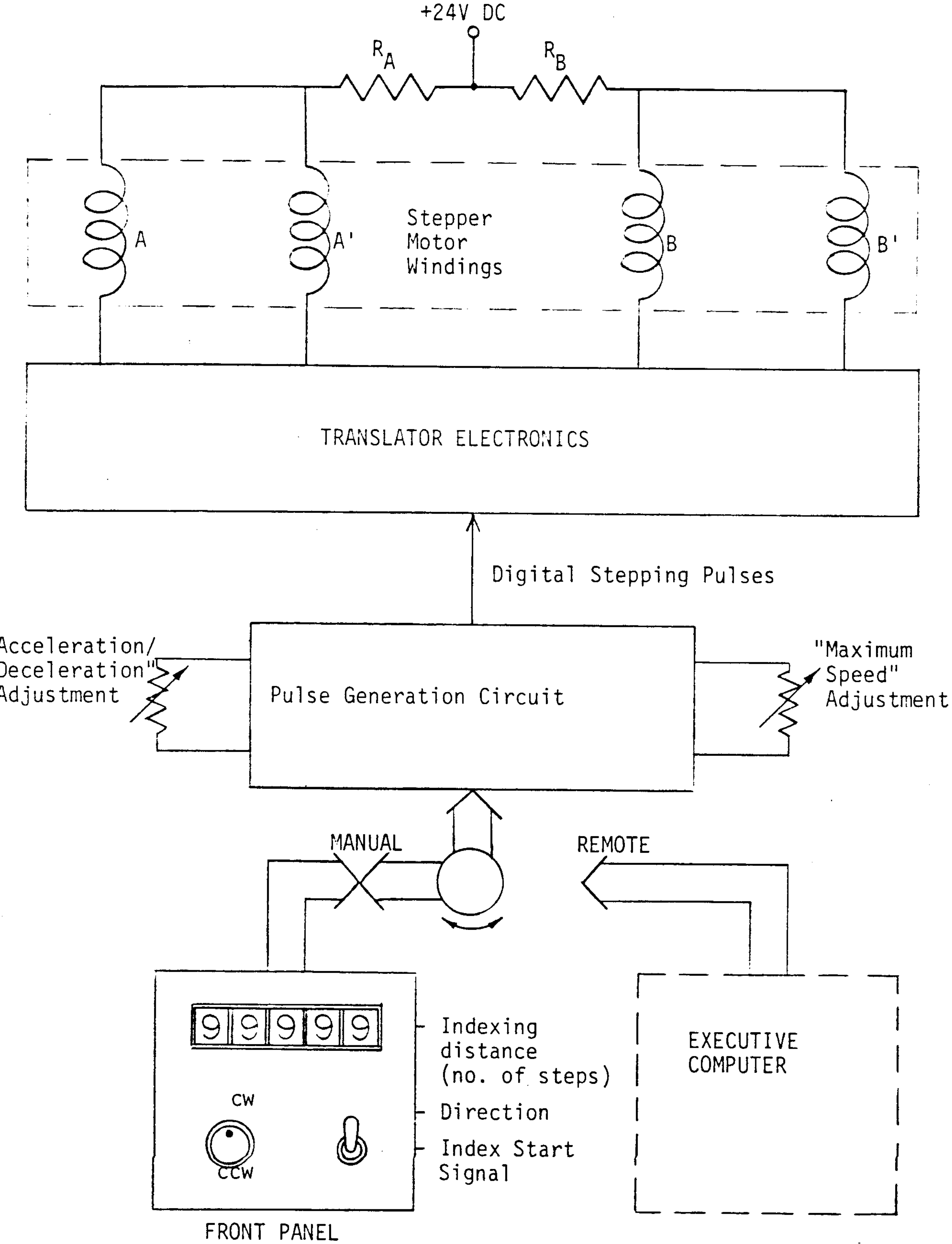


Figure 4-1 "Preset Indexer" for Stepper Motor Positioning indexing count inputs from BCD- input switches or a remote computer and upon command, sends the required number of pulses to the translator stage. The rate at which the pulses are sent is determined by the frequency of an internal oscillator which can be adjusted using an external potentiometer The pulses rate is also accelerated from zero to a maximum rate and decelerated back to zero at the end. These accelerations are adjusted using internal potentiometers. The acceleration and velocity controls are therefore in analog form and are not easily controlled by a remote computer. Furthermore the range of adjustment available does not provide enough flexibility to "tune" the traversing system to the required performance level as explained in Chapter 3. The indexing count is incremental which means that the executive computer system would need to keep records of the absolute position of the traverser. Preset indexers also provide motor "jogging" controls. Compared to the microcomputer system the preset indexer hardware is more expensive, less flexible and more difficult to interface with a remote computer system.

4.3 Description of the Microcomputer Configuration

The microcomputer-based controller chosen for the system consisted of the elements shown in Figure 4-2. The heart of the microcomputer system is a Central Processor Unit, with the associated EPROM erasable and programmable read-only memory for program storage and RAM read-write memory, for storage of variable data. The choice of the CPU is based on speed of instruction execution,cost and programming convenience. Speed of instruction execution is important in order to allow the microcomputer to simultaneously provide complex velocity profiles for all three motors, as explained later. This feature allows all the motors to run simultaneously at maximum capability and therefore keep the elapsed time for multiple-axis point-to-point

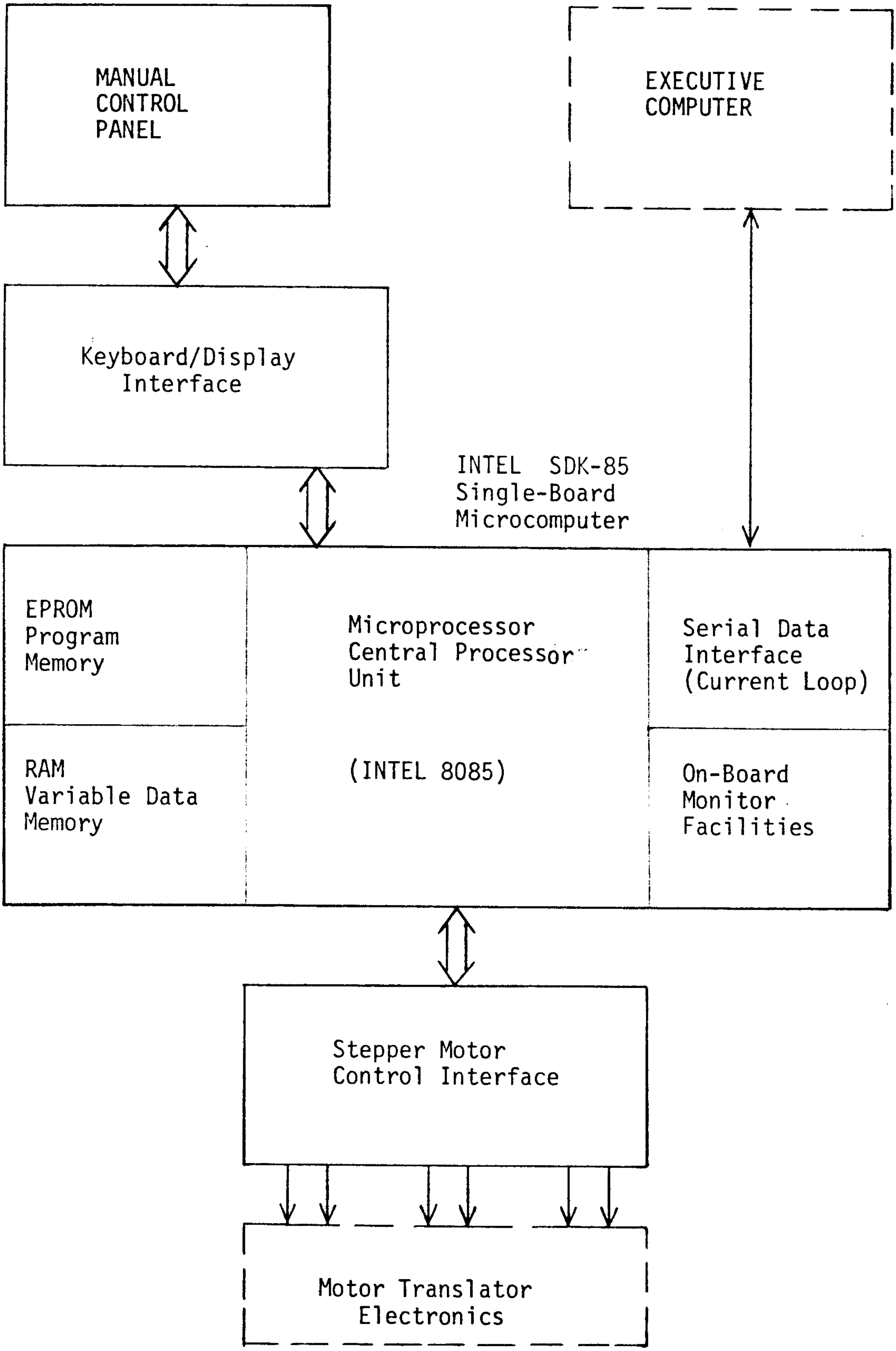


Figure 4-2 Block Diagram of Microcomputer-based Controller

positioning to a minimum. Timing calculations and preliminary programming showed that a INTEL 8085, 8-bit, CPU operating at a 3 MHz clock frequency was adequate. A single-board microcomputer with on board prototyping space and a "monitor" program is desirable for easy prototyping and testing. An Intel SDK-85 single computer was chosen. Familiarity with the 8085-type microprocessors and the ready availability of software development facilities were also important considerations in selecting this type of microprocessor.

The task of generating stepping pulses with the required waveform characteristics of pulse width and frequency is performed by a special interface circuit consisting of a programmable timer integrated circuit (INTEL 8253) driving discrete digital monostable multivibrator circuits as shown in Figure 4-3. The microcomputer programs the timer integrated circuit with the pulse rate count and starts it. The timer then issues a short trigger pulse (0.5 μs) at the end of every count cycle. This short pulse is used to trigger the monostable multivibrator which produces a long pulse of the duration required by the stepper motor translator circuit (10 μ sec. minimum). By changing the timer pulse rate during operation a velocity profile with controlled acceleration and deceleration can be created. The output pulses from the timer are also used to set a flip-flop memory. The CPU scans this memory continuously. When it sees the memory set it makes note that a step has been taken and resets the memory for the next pulse. Using this memory the computer keeps track of the distance traveled and the instantaneous position of the traverser slides. The timer is stopped by the CPU after the required distance has been traversed. As an additional safety measure, the output pulses are AND-gated with "enable" signals from the output ports of the single-board computer. The CPU has to set the "enable" signals to a logic level "one" before the timer pulses become effec

The above technique delegates the task of timing and shaping the steppi

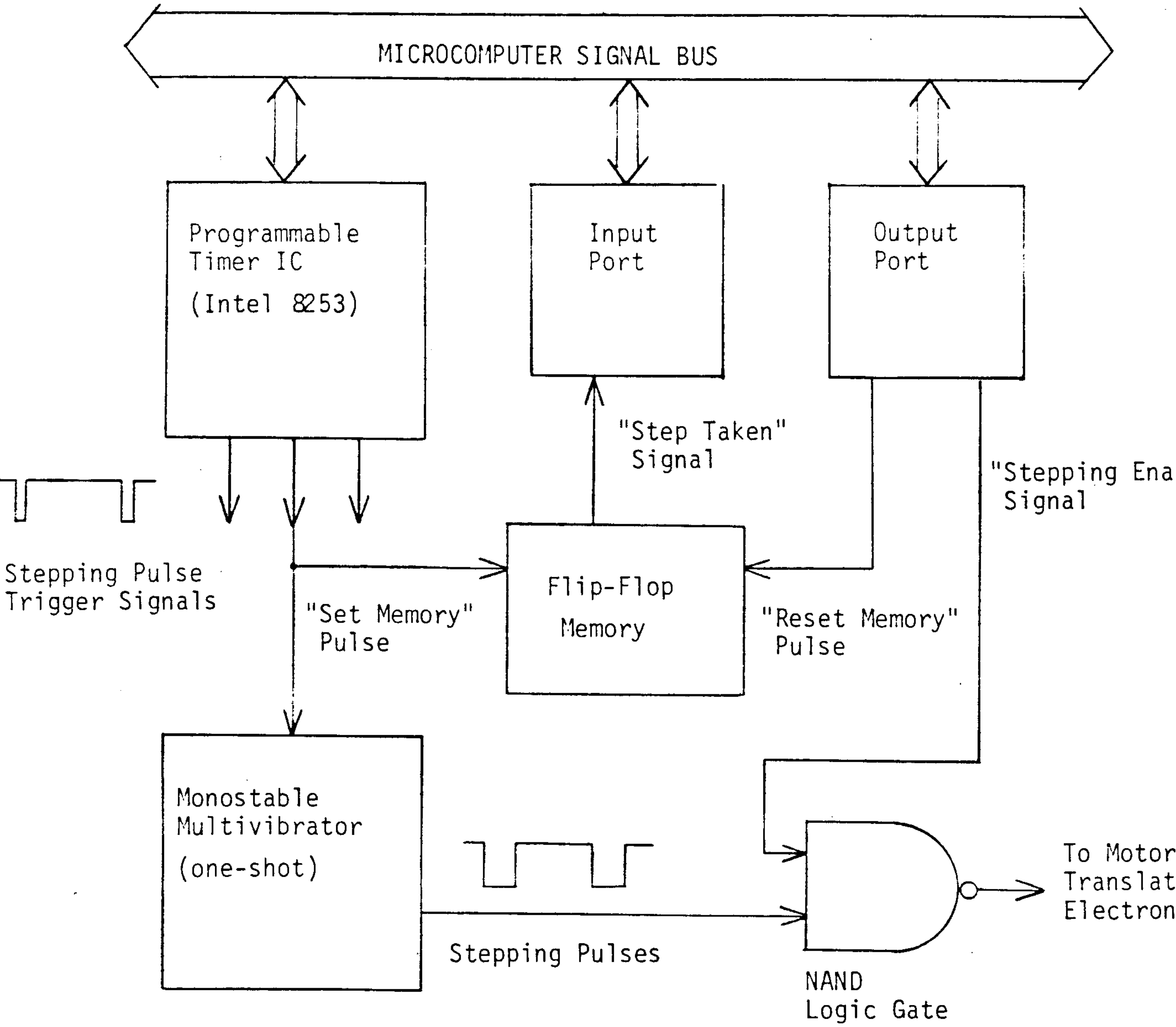


Figure 4-3 Block Diagram of Stepper Motor Control Interface

pulses to dedicated support circuits, allowing the microprocessor to spend its time more usefully in keeping count of the distance traversed and generating the required velocity profiles.

4.4 Manual Control

Communication with the human operator is performed through a keyboard and alphanumeric display shown in Figure 4-4. The keyboard allows the operator to enter commands and information into the computer. In the Manual control mode, the operator can jog or position any axis to any point by entering the proper key commands and numeric information. He can also command the computer to go into a remote control mode. In this Remote mode, positioning commands for all axis are received from a remote computer as explained later. The Manual mode of operation allows convenient calibration and set-up of the traverser, while the Remote mode provides automated long-term operation. A 32-character, alphanumeric display is used to echo the operator's key entries back to him allowing correction of erroneous entries. It is also used to display the absolute positions of all the axes. Errors detected during command entry and execution cause diagnostic messages in English language to be displayed, allowing easy troubleshooting.

The task of constantly scanning the keyboard for inputs and servicing the display devices is handled by a special purpose integrated circuit device called a Keyboard and Display Controller. The CPU communicates with this device through the data lines of the microcomputer and issues the necessary commands for the input and output of information. The Keyboard and Display Controller constantly scans the keys and when a key is struck by the operator it interrupts the CPU to inform it of this event. Thus, it relieves the CPU from having to wait for a key entry, allowing it to perform more important tasks.

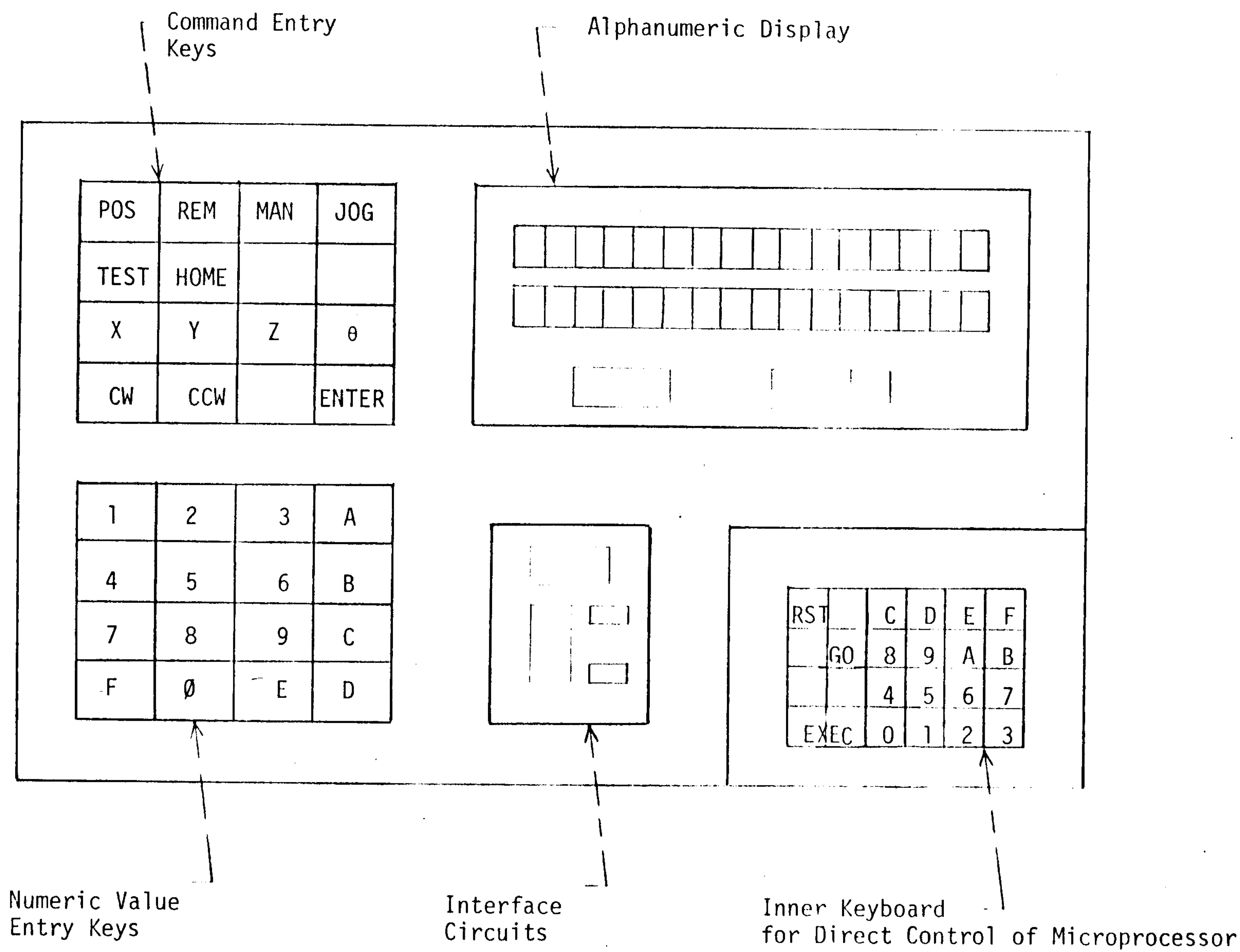


Figure 4-4 Manual Control Panel

4.5 Remote Control

Communication with the remote executive computer system is handled by the CPU itself through built-in single data lines for input and output. Since information is represented internally in the computer by binary numbers, it can be easily sent through a single-data line in a bit-serial fashion. Additional encoding information is used to mark the start and end of the serial data streams, and parity information is added to allow detection of data transmission errors. Since transmissions over large distances (up to 50 ft.) are involved, the digital data signals are boosted at both ends of the transmission lines by a current-loop interface. The data transmission rate is also limited to approximately 10 characters per second (110 bits/sec) to avoid transmission errors. Use of a serial data interface allows the amount of wiring between the computers to be reduced to a bare minimum of four wires.

4.6 Other Aspects

Liberal use of large-scale integrated circuits has considerably simplif the design of the microcomputer system and produced a compact and portable controller.

Finally, a DC power supply (providing enough reserve capacity for the addition of new circuits) had to be designed. A "linear" power supply was us with voltage regulators limiting voltage fluctuations to within five percent the specified levels of +5V, +12V, -12V.

5. CONTROLLER DESIGN - SOFTWARE

Once the microcomputer system hardware has been defined and the type of CPU selected, software has to be generated. Software is the set of machine instructions for the CPU of the microcomputer system which will cause the microcomputer to perform its various control and communications tasks.

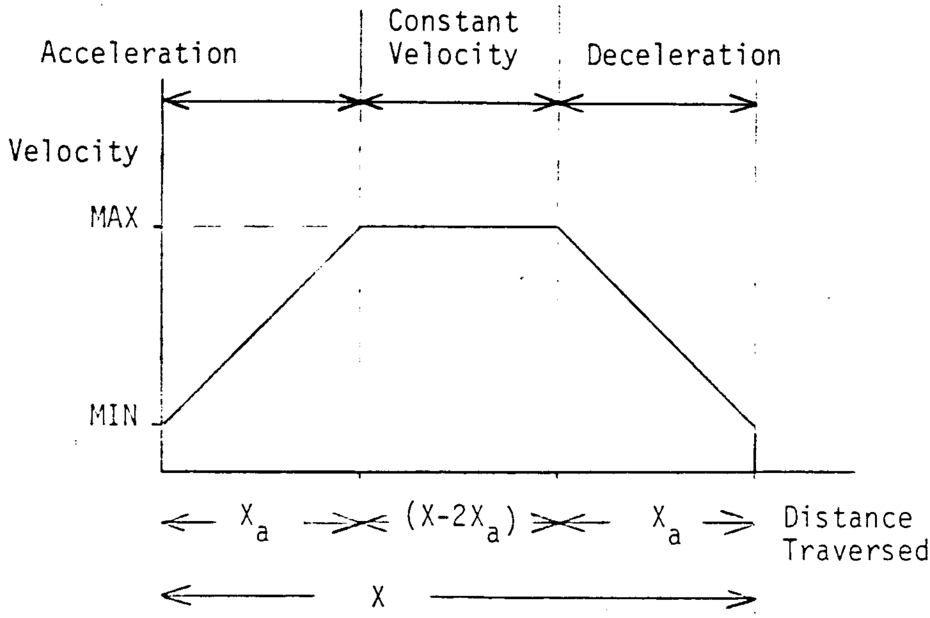
5.1 Definition of Tasks

The first stage in designing software is the definition of the tasks to be performed by the microcomputer. The primary task of the microcomputer is to control the motion of the stepper motors. Two types of motion control are required: constant speed rotation in either direction and point-to-point positioning over the desired distance in either direction with a profiled traverse including acceleration and deceleration. The second task of the microcomputer is to communicate with another computer system to provide remotely commanded automatic positioning of the traverser slides. This involves sending and receiving data and predefined commands over a serial data link. The third task of the microcomputer is to accept motor control commands from a human operator and displaying messages and information to him. This involves scanning and analyzing keyboard or switch inputs and displaying alphanumeric text strings on a multi-character LED display.

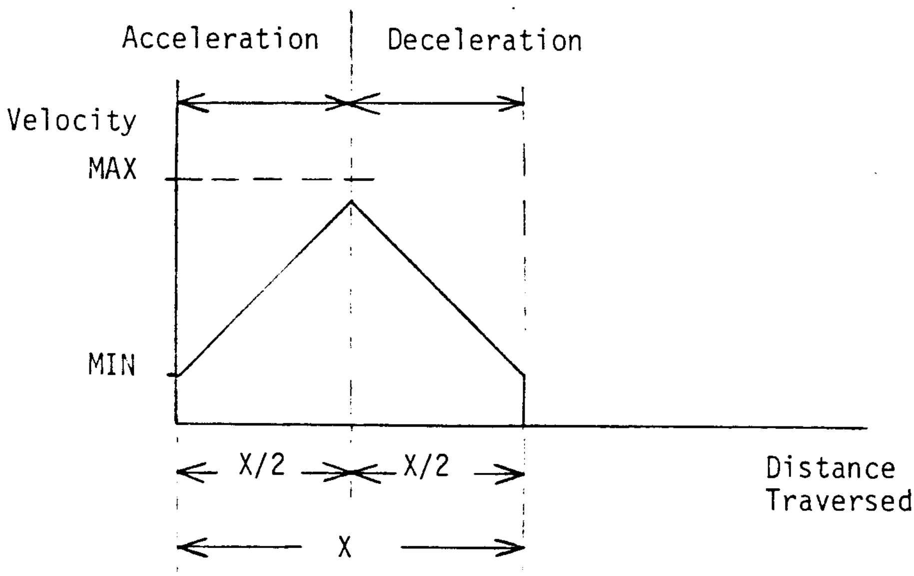
The second stage in software design is to design computer programs which will accomplish the above tasks. Flow charts are drawn showing the flow of control from each task to the next starting with the power-on reset initializations. The effects of all the control inputs (such as operator command inputs) on the flow of control are defined. All the flow charts have been grouped in Appendix C.

5.2 Program Descriptions

The operations of the microprocessor can be classified into three levels. At the uppermost level,a main program which controls the sequence of operation is used. The flow-chart of this program is shown in Figure C-1. The initialization of hardware and software is performed only once, when power is first applied. The input/output peripheral integrated circuits have to be initializ with specific data in order to operate properly. The main program waits until the operator enters a command at the keyboard. It analyzes this command and rejects it if errors are detected. If the command is valid the appropriate "sub program" is called upon to perform the required task. After the task is completed, control returns to the main program which again waits for a new com from the operator. There are four subprograms that can be called up by the ma program to perform specific traverser control tasks. 1) "Jogging" or slow rotation of the axes under manual control; 2) Positioning any axis at a specif position under manual command; 3) Positioning any one or all axes at positions specified by another computer; 4) Testing the microprocessor hardware to assur proper operation. The flow charts of these sub-programs are also shown in Appendix C. The axis positioning sub-program is called by the main programs i either the manual mode or remote control mode. It can simultaneously position one, two or three axis over different distances. This is achieved by handling the acceleration and deceleration profiles independently for each axis as foll The distance to be traversed by each axis is divided into three parts: acceler constant velocity and deceleration. The acceleration and deceleration rates a constant and identical for all three axis as shown in Figure 5-1 (a). The dis required to accelerate to maximum velocity, X_a, is constant. If the traverse distance is greater than $2x_a$, a constant velocity phase which covers the requi distance at maximum speed is inserted. If not, the traverse distance is split



a) Distance traversed > 2 X_a



b) Distance traversed $\leq$ 2 X_a

Figure 5-1 Velocity Profiles for "Open-Loop" Positioning of Stepper Moto into two equal acceleration and deceleration phases (Figure 5-1b). The stepping pulses for positioning are initiated by writing a count number intc the timer and starting it. The timer counts down at a fixed rate. When the count reaches zero a pulse is output and the timer is restarted with a new count value. The pulse is amplified and sent to the motor drives as explain in Chapter 4. During the acceleration phase the CPU monitors outgoing pulse and every time a pulse is sent out it decreases the count number in the time The pulse rate steadily increases, thus accelerating the motor. When the number of pulses required for the acceleration phase is completed, the count number is left unchanged and the motor is traversed at a fixed speed over th required distance. During the deceleration phase, the count number is stead increased with each outgoing pulse and the motor is slowed down to a stop. this time the required distance has been traversed.

In order to provide absolute positioning, a reference or "home" positio has to be established, usually at either end of the axis. A "Home" command that allows this operation to be performed on any axis is available. When this command is received, it sets an internal absolute position register equ to zero. When the axis is moved, this register is incremented or decrement according to the direction of travel to provide a measure of the absolute position of the axis with respect to "home" position.

At the lowest level of computer operations is a set of sub-programs which perform common chores required by the operation programs (e.g. sending a single data character to the executive computer). Flow charts showing the sequence of decisions and operations are prepared for these sub-programs too. The flow charts do not show any details of what machine instructions are used to implement these operations.

5.3 Programming

Once the flow charts are prepared, program coding is started. Each task on the flow chart is converted into a set of program instructions for the CPU. Since speed of execution is important and the entire program is relatively short (estimated at less than 2 kilo-bytes of program memory), program coding is done at the "assembly language" level. Assembly language uses symbolic representation of machine instructions by mnemonic "operation codes" which are more easily understood than the binary numbers actually used for machine instructions. Machine instructions in binary number codes are generated from the source programs written in asembly language using special computer programs called Assemblers which are run on a separate microcomputer development system. The machine instructions are transferred into integrated circuits memory devices called PROMs. (Programmable Read-Only Memories). These memory IC's are inserted into the microcomputer controller to complete the software installation.

The programs are tested in the actual system using the monitor programs available on the single-board computer. Machine instructions can be stepped through one at a time, and errors can be traced and debugged. Once the programs operate as designed the design of the microcomputer software can be considered to be complete.

6. CONSTRUCTION AND PERFORMANCE

Detailed drawings showing the shapes and dimensions of all parts of the traverser were prepared. The leadscrews, slideway structures of the vertica transverse and longitudinal axes, bearing housings and mounting structures were machined to close tolerances. The traverser structure was assembled starting with the base frame and supports and working up to the vertical axi The frames and slides were carefully aligned with respect to each other and to the axis of the jet. The stepper motors, leadscrews, bearings, couplings and mounting brackets were assembled separately into drive units which were then mounted onto the traverser structure after proper alignment.

The motor drive electronics were placed in an enclosure and wired to the stepper motors of all axes.

The microcomputer system, with the stepper motor interface circuits, ke board and display, remote computer interface and power supplies was wired to gether in a special enclosure. The microcomputer software was encoded into machine instructions in EPROM memory, using a microcomputer development syst This program memory was installed in the microcomputer and the controller was connected to motor drive electronics.

By this time the operation of the individual parts of the system, i.e. traverser mechanism, motor drives and microcomputer controller had been indi ually tested. The system was ready to be tested as a whole unit. A brief u manual describing how to operate the system was prepared and is included in pendix F. When the system, as designed, was initially assembled and tested, performance met all basic specifications. All axes could be easily "jogged" under manual control. Slide positioning under manual control as well as rem computer control was successfully achieved. All axes could be traversed end-to-end over a specified distance and returned to the starting point with excellent repeatability. Simultaneous positioning of all axes over differen distances was also successful. Some of the problems encountered and the solutions used to overcome them are described below.

(i) The slideway of the vertical axis was initially designed with a single cylindrical guide in parallel with the leadscrew. While the hardened and ground slide shaft was straight, the leadscrew which had been formed by rolling the threads was bowed. This caused the vertical slide to wobble as it traversed along the slideway. This problem was solved by adding a second slide shaft. The leadscrew bow was small enough to be overridden by the straightening action of the two slide shafts.

(ii) The torque couplings that were initially installed between the stepper motors and leadscrews were torsionally stiff metallic disc flexible couplings. These had a maximum torque rating of about 5 times the expected maximum motor torque. However the pulsating nature of the output torque of the stepper motor caused a coupling to fail by fatigue. The couplings were changed to Oldham-type couplings with 3 times the maximum torque rating of the metallic disc couplings. The Oldham couplings tolerate small misalingments of the motor and leadscrew while providing strength to withstand the fatiguing action of the stepper motor.

(iii) Stepping signal incompatibility between the microprocessor and th translator electronics was caused by ground loop noise. This is an electrica effect caused by the electrical signal "ground" potentials at the two ends o the cable being different. This problem was solved by connecting the "groun terminals of both the microprocessor and the drive electronics to the common ground line of the AC power supply.

(iv) The initial versions of the microcomputer software caused problem due to programming errors. These could be traced using the on-board monitor

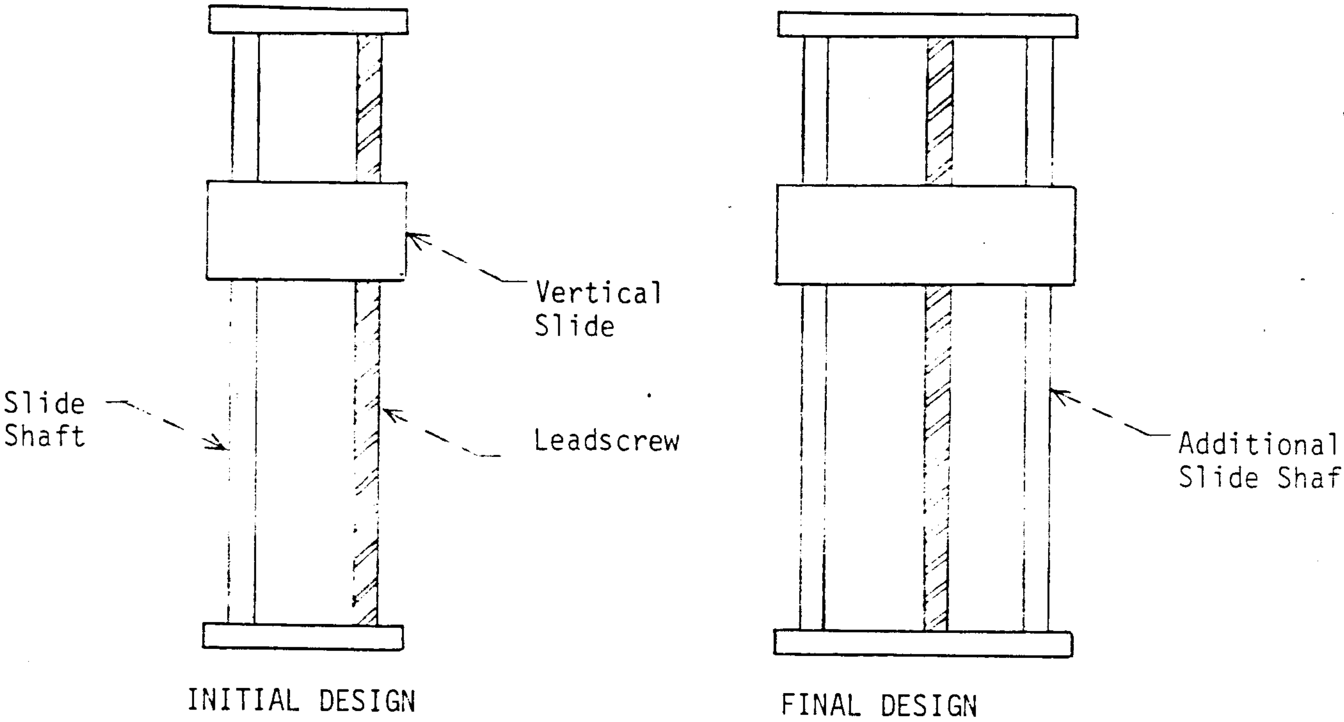


Figure 6-1 Modification of Vertical Slideway

program on the microprocessor to follow the program sequence and to isolate the problem area.

Except for application problems of the types described above the performance of the system was satisfactory. No major conceptual errors were discovered.

7. CONCLUSIONS

A multiple-axis automatic positioning system has been successfully designed, built and operated. The apparatus was developed to operate with stepper motors under direct microcomputer control. Stepper motors have low torque outputs and lower speeds than DC servomotors of comparable size, and therefore have a slower response to positioning commands. However when the response of the positioning system using stepper motors is adequate, they offer advantages over DC servomotors. Good accuracy can be achieved withou expensive transducers and closed-loop control. The control system is easie to design and less expensive. It also provides an easy interface with digital computers for automated operation.

A dedicated positioning system controller results in more effective use of the executive computer systems time and resources. Microprocessor-based systems provide a cost effective means of implementing dedicated controllers

While the degree of resolution in position is dependent upon the design the positioning drive and its controls, the accuracy of position actually achieved cannot be better than that allowed by the mechanical structure of the slideways and supports. Therefore, the traverser mechanism has to be carefully designed and built to minimize errors.

Careful electrical design and layout of the complex digital electronics involved is essential to ensure trouble-free operation of the system. In spite of the most careful design, "fine-tuning" adjustments invariably have to be made on the actual system itself, and ample allowances must be made for them in the design.

APPENDIX A

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APPENDIX B

STEPPER MOTOR SIZING AND SELECTION

Before the motor can be selected, the load inertia must be estimated and the expected motion profile must be specified. The example given is for the vertical axis slide.

Estimated maximum weight of the vertical axis slide (bearings, leadscrew nut, structural components, probe assemblies, fasteners)	=	50 lbs.

Specified motion profile (tentative):

Max. traverse distance	=	48 in.
Max. traverse time (end-to-end)	=	40 sec.
Max. acceleration and deceleration	=	physical limits of traverser e pected to lie beyond capabilit of stepper motor
Max. velocity	=	

Selection of motor step size:

Position accuracy required	=	0.010 in.
Motor step resolution should be better than the required position accuracy		
Using a factor of five, minimum motor step resolution	=	0.002 in. (mi
From a study of manufacturer's catalogs, we see that the widest range of selection of permanent magnet stepping motors is available with step series of 1.8°/step, i.e. 200 steps/rev.		

To achieve a linear resolution of 0.002 in. (min.) with a step size of 1.8° max. leadscrew pitch = $\frac{360}{1.8}$ x 0.002 = 0.4 in./rev. (max.). Tentatively choose a commercially available leadscrew with a pitch of 0.25 in/rev. Having selected the motor step size and leadscrew size, we chose a motor with the required torque-speed characteristics.

Motor speed requirements:

Average traverse speed = $\frac{48 \text{ in.}}{40 \text{ sec.}}$ = 1.2 in/sec.

Estimate acceleration and deceleration times at 5 sec. each.

Deduce max linear speed
to achieve above average speed = 1.5 in/sec (approx.)

Corresponding motor stepping rate = 1.5 in/sec x $\frac{1}{0.250}$ in/rev

x 200 $\frac{\text{steps}}{\text{rev.}}$ = 1200 $\frac{\text{steps}}{\text{sec.}}$ (min.)

Motor torque requirements:

Acceleration torque:

Slide Inertia seen by the motor $I_{SL} = W \times \frac{1}{p^2} \times 0.025$

w	=	weight of slide	=	50 lbs.
p	=	pitch of leadscrew	=	4 threads per inch
I_{SL}	=	$50 \times \frac{1}{16} \times 0.025$	=	0.078 lb-in^2
Inertia of leadscrew I_{SC}			=	D^4 x length x 0.028
D	=	Diameter of leadscrew	=	1.0 inch (from catalog product)
I_{SC}	=	$(1.0)^4 \times 48 \times 0.028$	=	1.344 lb-in^2
Inertia of motor rotor I_M			=	2.5 lb-in^2 (estimated from catalog

Total inertia to be driven by motor I_{TOTAL} = 4.0 lb-in^2 (approx.)

1. Torque required to accelerate slide $T_A = I_{TOTAL} \times \alpha \times \frac{1}{24}$

I_{TOTAL} = 4.0 lb-in^2

$$\alpha = V_{max}\left(\frac{\text{steps}}{\text{sec}}\right) \times \theta_s\left(\frac{\text{degrees}}{\text{step}}\right) \times \frac{\pi}{180}\left(\frac{\text{radians}}{\text{deg}}\right) \times \frac{1}{\text{Acc.Time(sec.)}}$$

$$= 1200\left(\frac{\text{steps}}{\text{sec}}\right) \times 1.8\left(\frac{\text{deg}}{\text{step}}\right) \times \frac{\pi}{180}\left(\frac{\text{radians}}{\text{deg}}\right) \times \frac{1}{5 \text{ sec}}$$

$$= 7.54 \text{ rad/sec}^2$$

$$T_A = 4.0 \times 7.54 \times \frac{1}{24} = 1.26 \text{ oz-in.}$$

ll. Torque required to overcome friction $T_F = \frac{\text{Linear Friction Force}}{0.393 \times p \times \eta_{sc}} +$ Rotar Frict

Linear Friction Force = estimated at 1 lb. with linear ball bearings

η_{sc} = efficiency of leadscrew = 90% for recirculating ball screw

p = pitch of leadscrew = 4 threads/inc

Leadscrew Rotary Friction = estimated at 5 oz-in.

$$T_F = \frac{1}{0.393 \times 4 \times 0.90} + 5 = 6 \text{ oz-in (approx.)}$$

lll. Torque required to raise slide against gravity $T_G = \frac{\text{weight}}{0.393 \times p + \eta_{sc}}$

Weight = 50 lbs.

$$T_G = \frac{50}{0.393 \times 4 \times 0.90} = 35.34 \text{ oz-in.}$$

Total torque T_{TOTAL} $= T_A + T_F + T_G = 1.676 + 6 + 35.34$

$= 43$ oz-in (approx.)

The above torque requirement must be met at all speeds up to the desired stepping rate of 1200 steps/sec.

A permanent magnet stepping motor satisfying this requirement (with an adequate margin) was chosen from inspection of manufacturer's catalogs. (Superior Electric SLO-SYN M092-FD09).

The actual values of motor inertia, step size, leadscrew compatability, acceleration capability and maximum velocity limit are compared with previously estimated values to ensure that all calculations were valid.

A few other combinations of motor sizes and leadscrews were chosen to compare with the initial design and to perform an ad hoc optimization. The initial design proved to be nearly optimal and was retained as the final choice.

Since recirculating ball type leadscrews are not self-locking, the "holding" torque of the motor (i.e. torque at standstill) had to be adequate to prevent movement of the slide under its own weight. Holding torque of selected motor = 300 oz-in. Torque required to hold weight against gravity T_G = 35.34 oz-in. Therefore, this requirement was met.

A few of the key parameters for the transverse and longitudinal axes are shown below.

	Transverse	Longitudinal
Weight of slide	140 lb.	300 lb.
Max. traverse distance	48 in.	72 in.
Max. traverse time (end-to-end)	40 sec.	40 sec.
Positioning accuracy required	0.010 in.	0.020 in.
Leadscrew pitch	0.25 in/rev.	0.25 in/rev.
Motor speed requirement	1200 steps/sec.	1200 steps/sec
Motor torque requirement (no gravity force)	< 15 oz-in.	< 20 oz-in.
Motor selected	SLO-SYN M092	SLO-SYN M092

Note that the same type of motor was chosen for all three axes in order to maintain equipment compatibility, since the performance requirements were not drastically different.

APPENDIX C

FLOWCHARTS OF MICROCOMPUTER CONTROLLER PROGRAMS

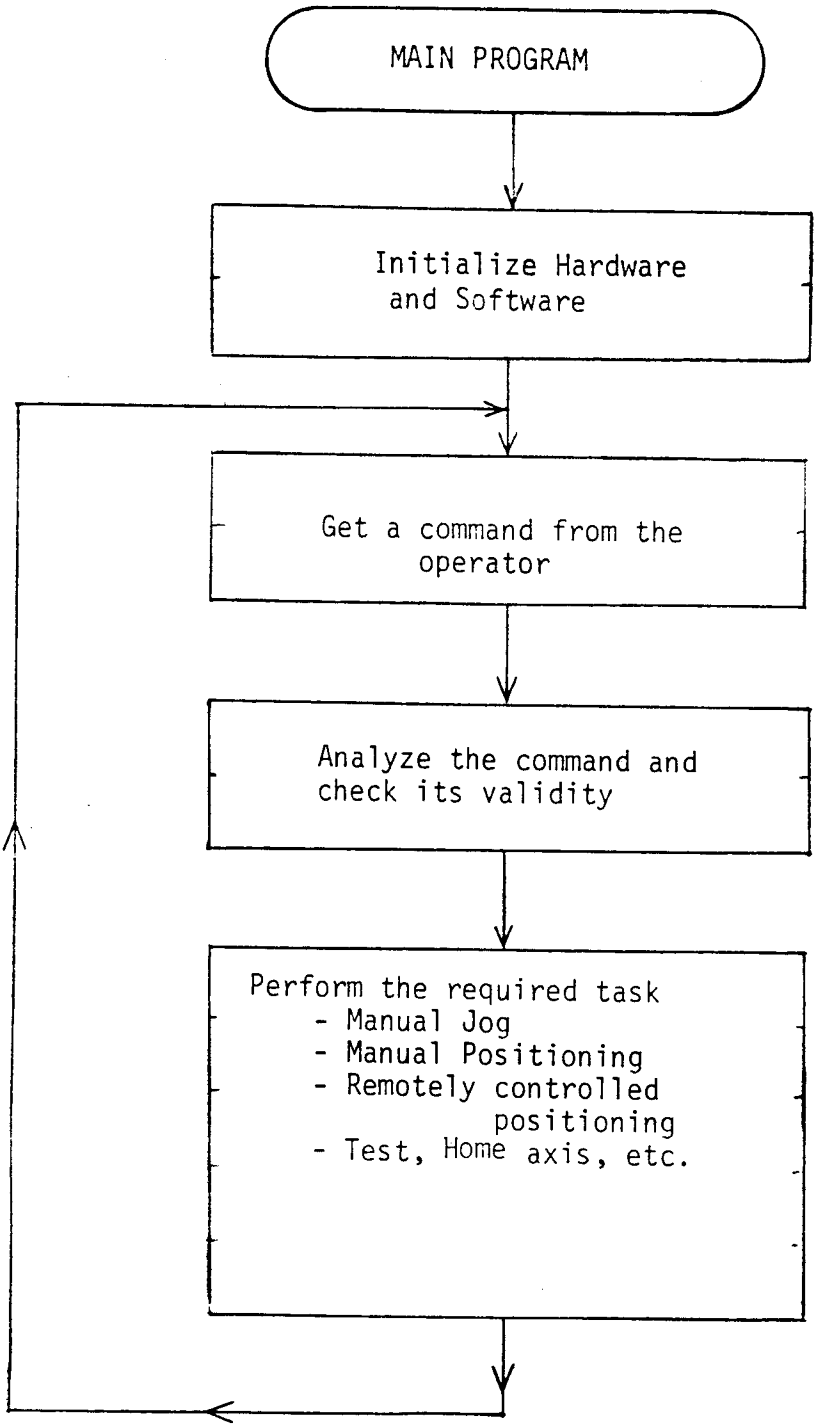

MAIN PROGRAM
Initialize Hardware and Software
Get a command from the operator
Analyze the command and check its validity
Perform the required task
- Manual Jog
- Manual Positioning
- Remotely controlled positioning
- Test, Home axis, etc.

Figure C-1

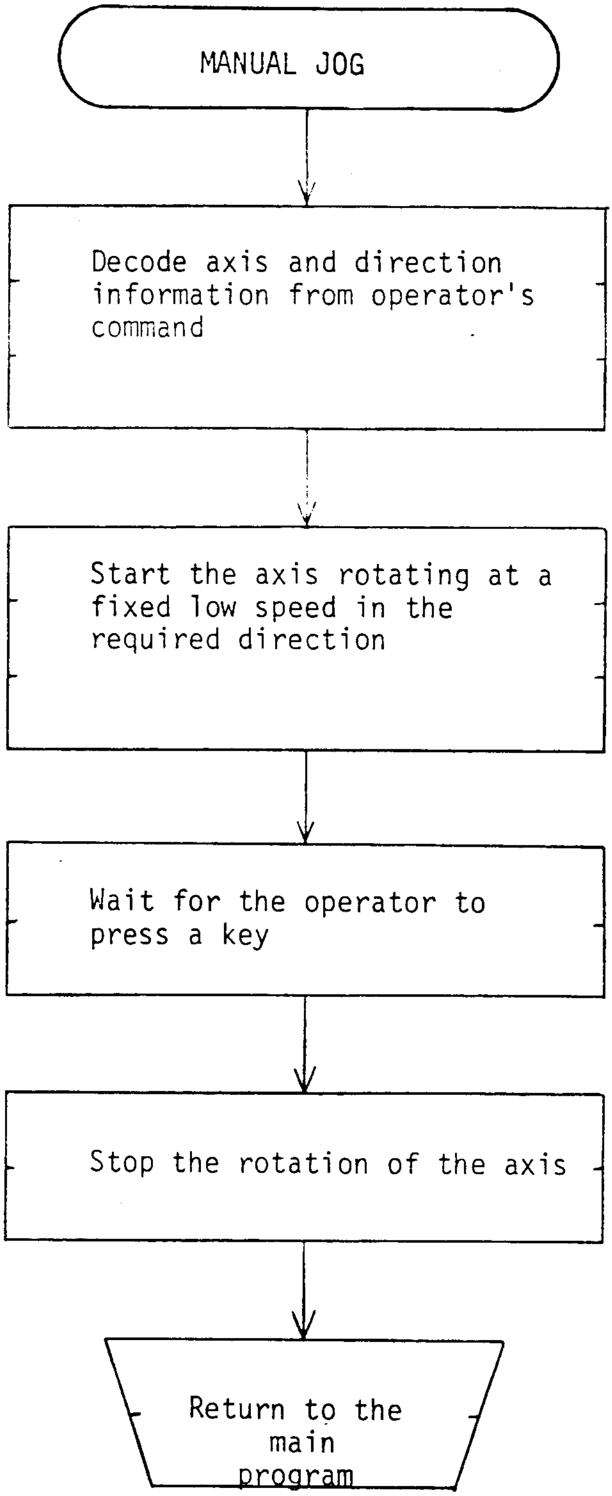

MANUAL JOG
Decode axis and direction information from operator's command
Start the axis rotating at a fixed low speed in the required direction
Wait for the operator to press a key
Stop the rotation of the axis
Return to the main program

Figure C-2

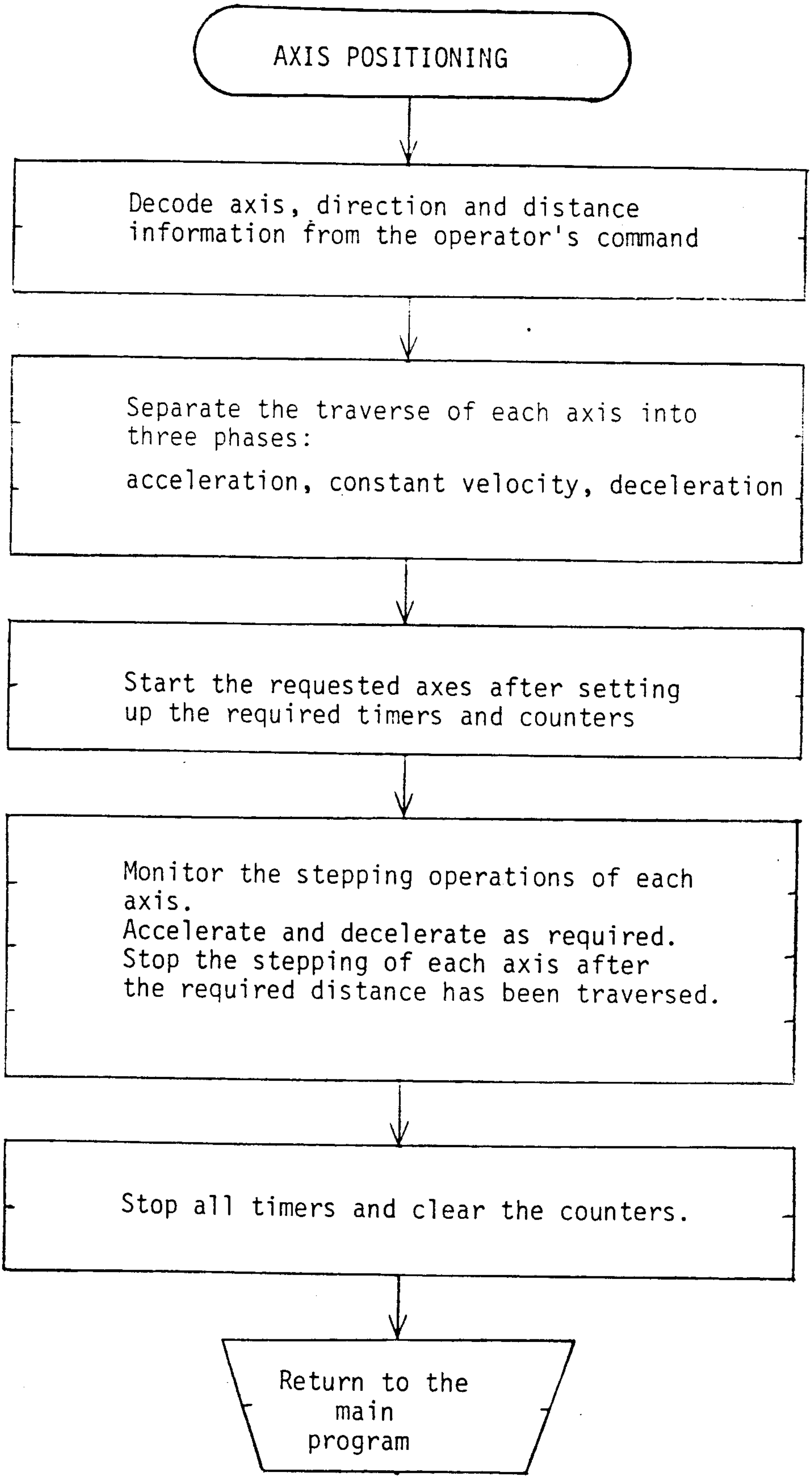

AXIS POSITIONING
Decode axis, direction and distance information from the operator's command
Separate the traverse of each axis into three phases:
acceleration, constant velocity, deceleration
Start the requested axes after setting up the required timers and counters
Monitor the stepping operations of each axis.
Accelerate and decelerate as required.
Stop the stepping of each axis after the required distance has been traversed.
Stop all timers and clear the counters.
Return to the main program

Figure C-3

APPENDIX D

STRUCTURAL DESIGN CALCULATIONS

Base frame design:

The initial structure of the base frame was sketched out with two long members and four cross members as shown in Figure D-1a. A third long member had to be added to support the transverse axis adequately. A fifth cross member was also added to provide more stiffness between support legs. Figure D-1b. Four support legs are used. The base frame design can be analyzed as several independent sections. The principal deflection considerations in the section A-B (Figure D-1b) are analyzed below.

Vertical deflection of the base frame:

Figure D-2 shows the loading of a typical section of the base frame. The load F comes from the combined weight of the transverse and vertical axis slideways and also the "lumped" effect of the weight of the base frame structure itself.

The worst case of this force is when the vertical axis is positioning directly over the center of this section. This worst case is estimated to be 255 lbs. If this section of the base frame is modeled as a simply-supported beam the vertical deflection at this point is calculated using the following equation:

$$\Delta x = \frac{F\ell^3}{48EI} \tag{D.1}$$

where E = modulus of elasticity of the long member = 20×10^6 psi for steel.

The long member was tentatively chosen as a 6 in. deep wide-flange structural I-beam. The moment of inertia of this beam's cross-section is

$$I = 21.8 \text{ in}^4.$$

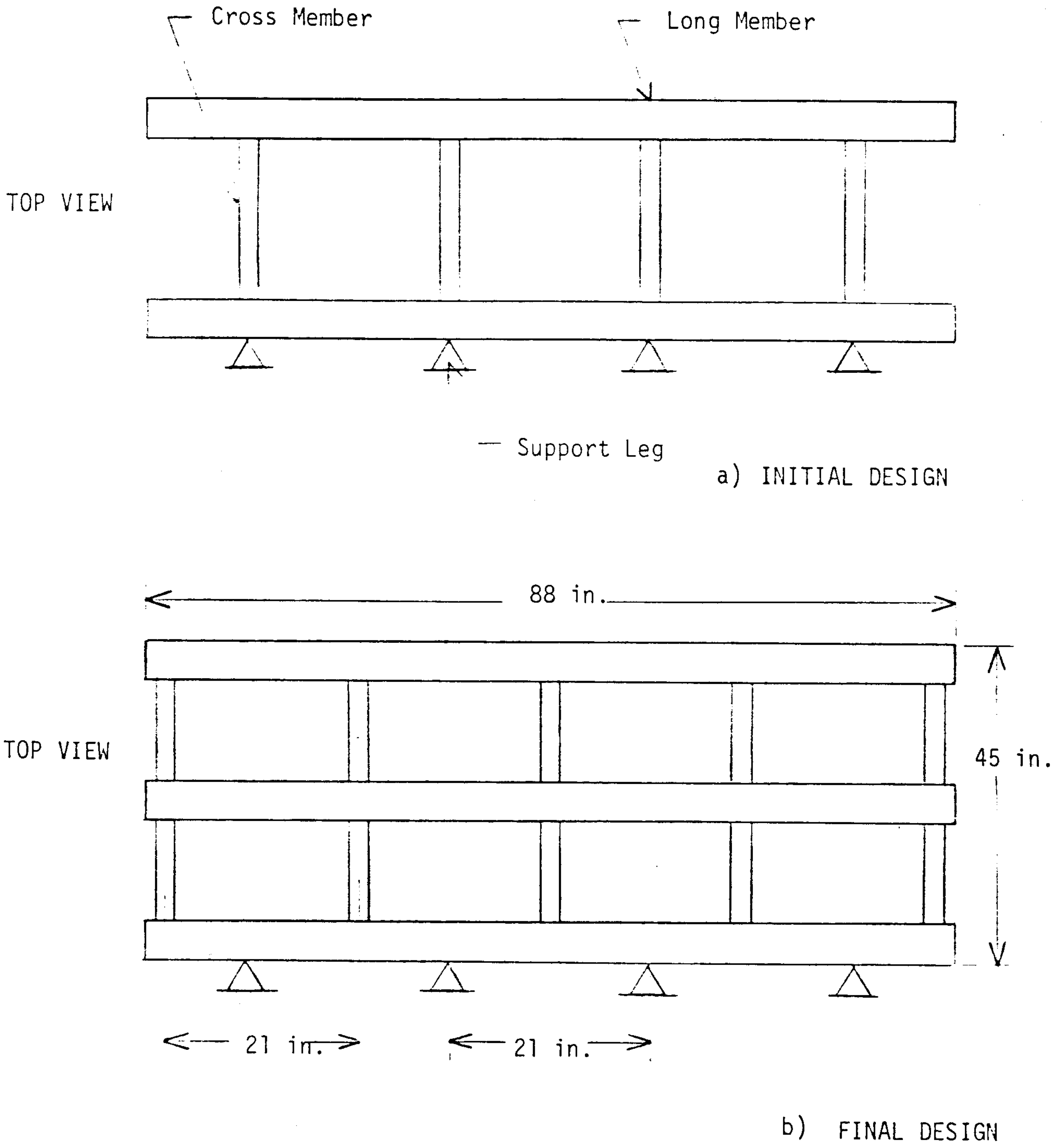


Figure D-1 Layout of Base Frame Structure

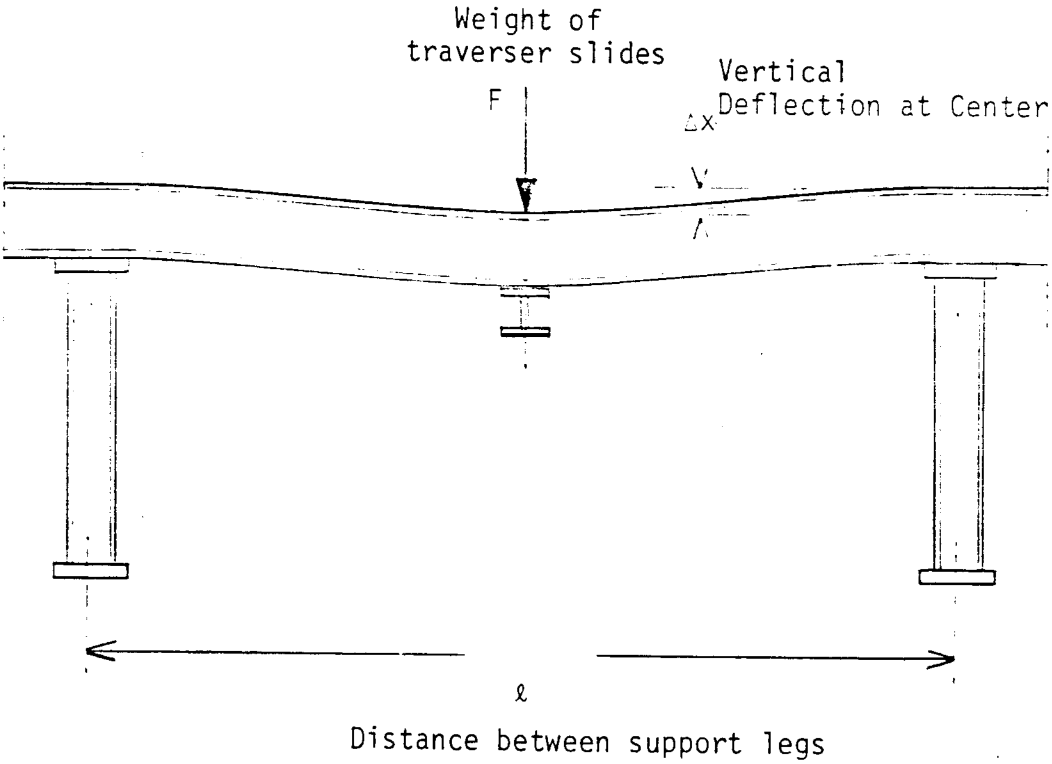
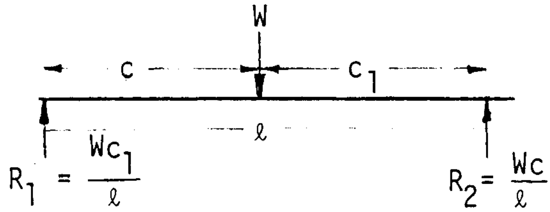


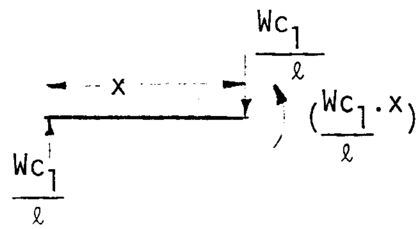
Figure D-2 Base Frame Deflection

With ℓ = 21 in. when four supports are used as shown in Figure D-1b.

$$\Delta x = \frac{225 \text{ lbs.} \times (21 \text{ in})^3}{48 \times (28 \times 10^6 \text{psi}) \times (21.8 \text{ in}^4)} = 8.060 \times 10^{-5} \text{ in.}$$

Angular Deflection under off-center loading





Bending Moment $M_x = \frac{Wc_1}{\ell} x$ (D.2)

Bending Equation $M_x = EI \frac{d^2y}{dx^2}$ (D.3)

$$\frac{dy}{dx} = \frac{Wc_1}{EI\ell} \int x \, dx = \frac{Wc_1}{EI\ell} \left(\frac{x^2}{2} + K \right)$$

$$\frac{dy}{dx} = 0 \text{ at } x = \sqrt{\frac{c(\ell + c_1)}{3}} \quad ; \quad K = - \frac{c(\ell + c_1)}{6}$$

$$\frac{dy}{dx} = \frac{Wc_1}{EI\ell} \left[\frac{x^2}{2} - \frac{c(\ell + c_1)}{6} \right]$$

At $x = c$ $\quad \left.\frac{dy}{dx}\right]_{x=c} = \frac{W}{6EI\ell} [3c^2 c_1 - cc_1 \ell - cc_1{}^2]$

Substituting $c_1 = (\ell - c)$

$$\left.\frac{dy}{dx}\right]_{x=c} = - \frac{W}{3EI\ell} \left[2c^3 - 3c^2 \ell + c\ell^2 \right]$$

Ignoring sign.

$$\theta_{\substack{\text{under}\\ \text{load}}} = \frac{dy}{dx} = \frac{W}{3EI\ell} \cdot \left[2c^3 - 3c^2\ell + c\ell^2 \right] \tag{D.4}$$

As the slide traverses, the value of c changes. Maximum under load with respect to c is at

$$\frac{d\theta}{dc} = \frac{W}{3EI\ell} \left[6c^2 - 6c\ell + \ell^2 \right] = 0$$

i.e. at $(Ac - \ell)\ (Bc - \ell) = 0$

Solving this equation,

$$\{(3+\sqrt{3})c-\ell\}\ \{(3-\sqrt{3})c-\ell\} = 0$$

$$\text{i.e.}\quad c_{\theta_{max}} = \frac{\ell}{(3-\sqrt{3})} \tag{D.5}$$

Therefore:

$$\max \theta_{\text{underload}} = 0.096\ \frac{W\ell^2}{3EI}$$

For the longitudinal axis

$W = 225$ lbs. $\ell = 21$ in. $E = 28\text{x}10^6$ psi (for steel)

$I = 21.8\ \text{in}^4$ (for a 6 in. wide-flange I-beam)

$\text{Max}\ \theta_{\text{underload}} = 5.895 \times 10^{-7}$ rad

max deflection at the top of vertical axis due to θ, $\Delta x_\theta = \theta h$

With $h = 10$ in., $\Delta x_\theta = 4.12 \times 10^{-5}$ in.

Base frame cross-member design

The purpose of the cross-member is to rigidly hold together the long members and to prevent torsional deflection of the base frame. They must also support the entire weight of the central long member in addition to their own weight. The deflection of the cross-members under loading from the transverse and vertical axes must be limited to 0.002 in. in order for proper traverser positioning tolerance to be maintained.

Weight of long member = 88 in. x 15 lb/ft. (for 6WE 15 I-beam)
= 110 lb.

1. Weight carried by a single cross member = $\frac{110 \text{ lb}}{5}$ = 22 lb.

2. Self-weight of cross member = 45 in. x 7.7 lb/ft. (for 4 in. std. I-beam
= 29 lb.

   Load from transverse axis = 160 lb.

3. Load shared by cross member through central long member = $\frac{160}{3}$ = 53 lb.

4. Load from vertical axis = 140 lb.

   Total load = 22 lb.+ 29 lb. + 53 lb. + 140 lb. = 245 lb.

Vertical deflection of cross-member = $\frac{W\ell^3}{48EI} = \frac{245 \text{ lb. x } (40 \text{ in.})^3}{48 * 28\text{x}10^3 \text{ psi x } 6.0 \text{ in.}^4}$

= 0.0019 in.

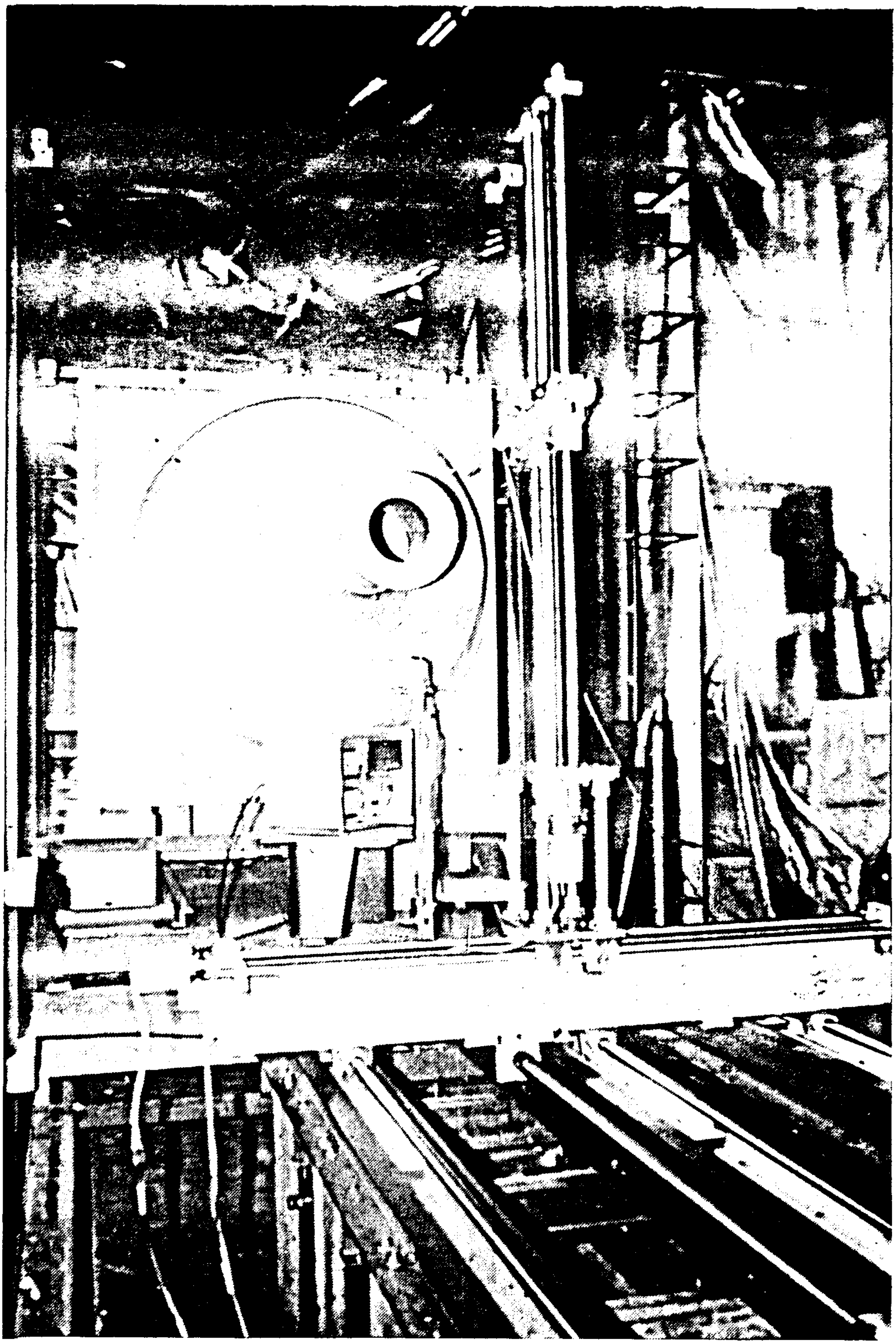


Figure E-1 Photograph of the Positioning System with the Jet and the Microcomputer-based Controller in the background

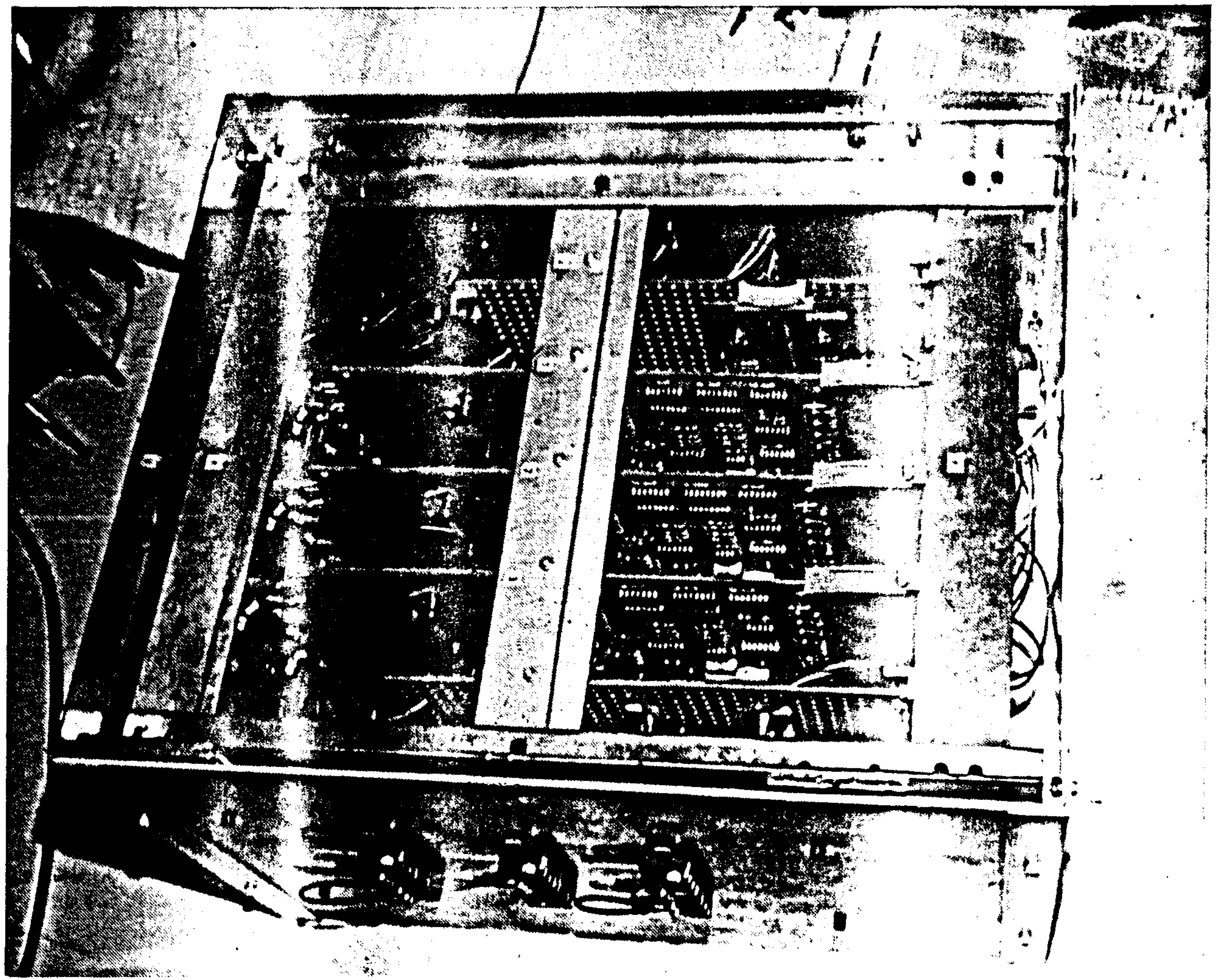


Figure F-2 Photograph of the "Translator" Electronics for the Stepper Motors

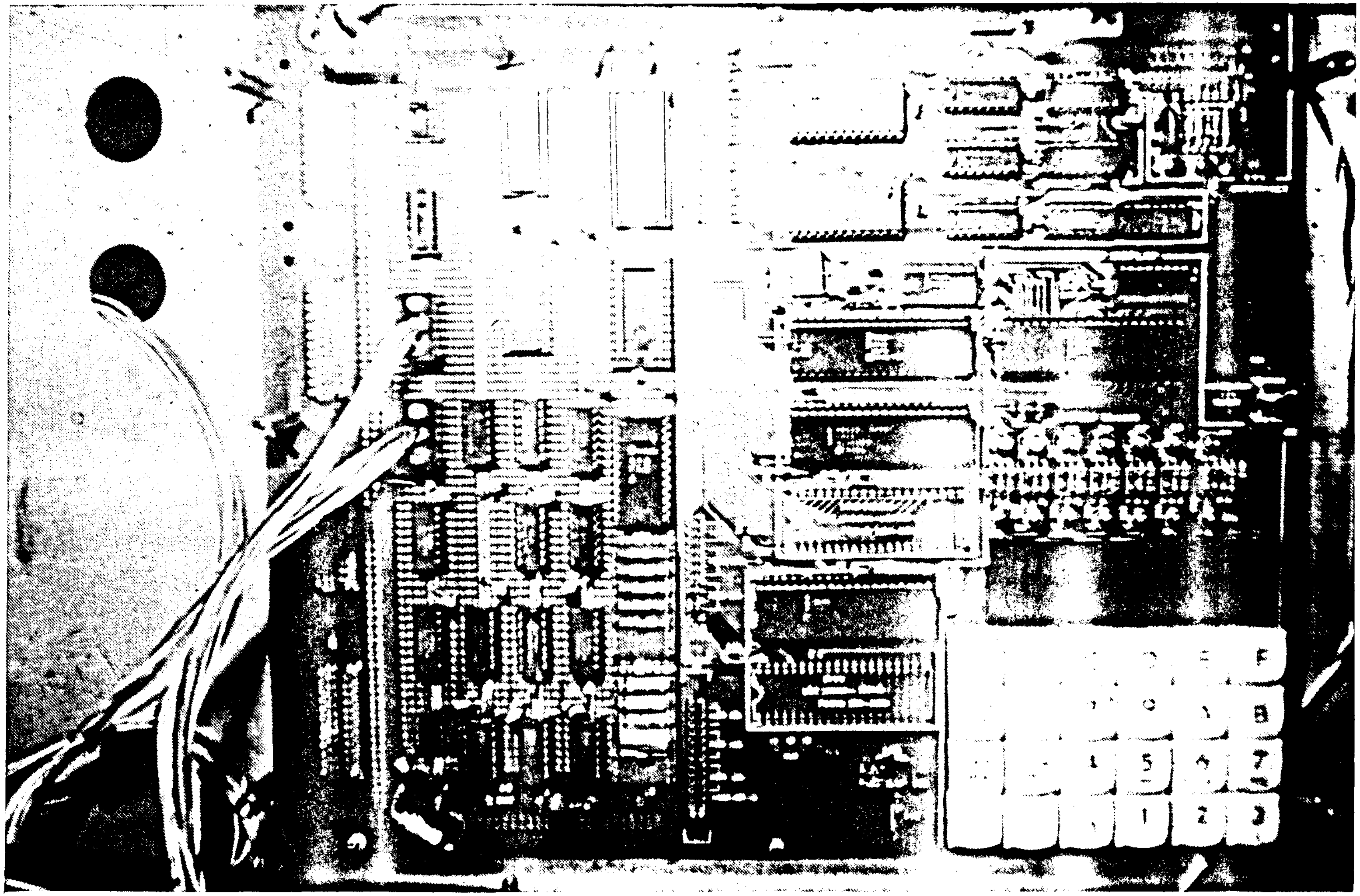


Figure 5.2 Photograph of the Microcomputer used in the Controller for the Positioning System

APPENDIX F

USER MANUAL

F.1 To Turn Power On:

Always turn power-on first at the microprocessor controller and start running the control programs by entering the following commands at the inner keyboard: RESET GO 8000 EXEC. Power to the motors must be turned on only after the control program is operating. This prevents stray electrical pulses at power-up from disturbing the position of the traverser. When the control program is initiated it defaults to the manual mode of operation.

F.2 To "Jog" the Slides:

Any axis can be "jogged" i.e. slowly traversed under continuous manual control. The axis to be jogged and the direction are selected while using the JOG command. Example: JOG X CW ENTER. To jog the vertical axis (X) with clockwise (CW) rotation of the motor. Jogging of the axis starts immediately after the ENTER key is pressed. Motor rotation will continue until any key is pressed again.

The axis are represented by the following keys:

X - for the vertical axis

Y - for the transverse axis

Z - for the longitudinal axis

θ - for the rotary axis

The directions are represented by:

cw - for clockwise rotation of the respective motor
when viewed from the nameplate end.

ccw - for counterclockwise rotation

F.3 To Position the Slides in the Manual Mode

The axis, direction of travel and distance must be specified using the "POSITION" command.

Example: POS X CW 100 ENTER

X	-	specifies the axis
CW	-	specifies the direction
100	-	is the number of steps to be traversed; each step = 0.25/400 in.

More than one axis can be positioned at a time by specifying the axis, direction and distance parameters before pressing the enter key.

Example: POS X CW 100 Z CCW 20 ENTER

The microprocessor will display a "DONE" message after positioning is completed.

F.4 To Position the Slides Under Remote Control from a Computer

The microcomputer controller is transferred into the remote control mode by pressing: REM ENTER.

Positioning commands now have to be entered from another computer through the serial data link. In this mode, the command format is identical to the manual mode positioning but the command keys are represented by single ASCII characters.

Example: P X + 100 Y - 40 Z + 80 RETURN

P stands for the position command.

X	=	vertical axis
Y	=	transverse axis
Z	=	longitudinal axis
R	=	Rotary axis
+	=	clockwise motor rotation
-	=	counter clockwise motor rotation.

The microcomputer will return a "DONE" character after the positioning is completed.

F.5 Home Position of Axes:

The absolute position value of any axis can be set to zero reference or HOME position at any point in its traverse range. The slide is positione at the desired point using either the JOG or POS commands. The HOME commanc is used to zero the absolute position.

Example: HOME Y ENTER

F.6 Test Commands:

The TEST command contains several subroutines which can be used to test the operation of the microprocessor controller, displays, keyboard, remote control communications and stepper motor interface.

F.7 Documentation:

The system hardware manual contains detailed drawings of the mechanical components and assemblies, microprocessor controller schematics and interconnects and stepper motor and translator drive wiring diagrams. Software listings of the microprocessor programs are also included. This documentati can be used for modification or troubleshooting of the positioning system.