A Lifting Mechanism for the J-750

by

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Submitted to the Department of Electrical Engineering and Computer Science

in Partial Fulfillment of the Requirements for the Degrees of

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ABSTRACT

The scope of this thesis addresses a particular need of the semi-conductor test industry. Specifically, it posits a means for automating vertical alignment between the machinery that tests and those that handle the silicon microchips to aid alignment between the two. This work will recommend a system that provides automated and compliant vertical motion with minimal expenditure. It will surpass the performance of Teradyne's current non-automated positioning system.

Thesis Supervisor: Alex Slocum
Title: Margaret MacVicar Faculty Fellow, Professor of Mechanical Engineering

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I Introduction

I.1 Preface

The unprecedented advancement of technology over the past century, and especially over the past few decades, owes much to the invention and commercialization of silicon-based hardware. The continued march of technology is fueled by the ubiquity of silicon in everything from discrete components to complex microchips. Designers and manufacturers continue to exploit the unique chemical characteristics of silicon, seeking to extract the maximum productivity from the minimum amount of silicon. The results: microchips the size of fingernails that power today's digital cameras, DVD players, and the wired world we live in.

The relentless zeal with which designers attempt to increase the usefulness of silicon inevitably leads to dilemmas of reliability in production. It is statistically impossible to guarantee the functionality of every chip. Errors in printing and fabrication render individual gates useless, even though the other millions of components function properly. Silicon manufacturers can ill afford to ship these faulty goods. For chips that often have tens and hundreds of input and output paths, manually verifying an individual chip's functionality is not feasible. Recognition of this fact gave rise to the Automatic Test Equipment (ATE) industry.

Launched by Teradyne's D133 diode tester in 1961, the ATE industry has since blossomed into a multi-billion dollar market niche and is integral to the continued development of silicon-based technology. Every fabrication floor in the world employs a fleet of automatic test equipment to verify the functionality of its products. Moreover, advances in the ATE industry have equipped testing machinery with the tools to both determine and rectify flaws on chips, thus saving microchips that would otherwise have been discarded. Additionally, such testing would provide manufacturers with a more efficient production process. The decreasing cost of test and the increasing performance of test equipment provide manufacturers with an irreplaceable tool. They have been able to produce more complex silicon structures while maintaining or decreasing their costs. As a result, a modern digital camera or DVD player costs less than a VCR used to cost. The demand for high-tech commercial goods will not subside, and the ATE industry ensures that the supply will flow cheaply and efficiently.

Since the industry's inception, Teradyne has been its leader. Starting with the D133 diode tester, Teradyne has since expanded into a variety of markets: Automotive Diagnostic Solutions, Circuit Board Testing, Network Test Products, and Semiconductor Test products, to name a few. Capturing the low end of the semiconductor market, the J-750 Microcontroller/Low-end VLSI Test machine has proved to be one of Teradyne's most successful products. The J-750 specializes in testing low-end semi-conductors quickly, accurately, and cheaply. For the bulk of products that do not operate at the limits of silicon's capability, the J-750 has proven to be the superior option for test.

For the J-750 to operate properly, a sound contact between the microchip or device under test (DUT) and the J-750's test electronics must be made to ensure signal integrity at frequencies exceeding 100Mhz. To that end, Teradyne has pioneered the design of Device Interface Boards (DIB) to mate a manufacturer's product with the J-750 Tester. A DIB is mounted to a machine called a handler. The handler stores trays of microchips and places the particular DUT in contact with one side of the DIB. The J-750 mates to the handler and contacts the other side of the DIB. As such, a signal path is completed from the chip to the Tester through the DIB.

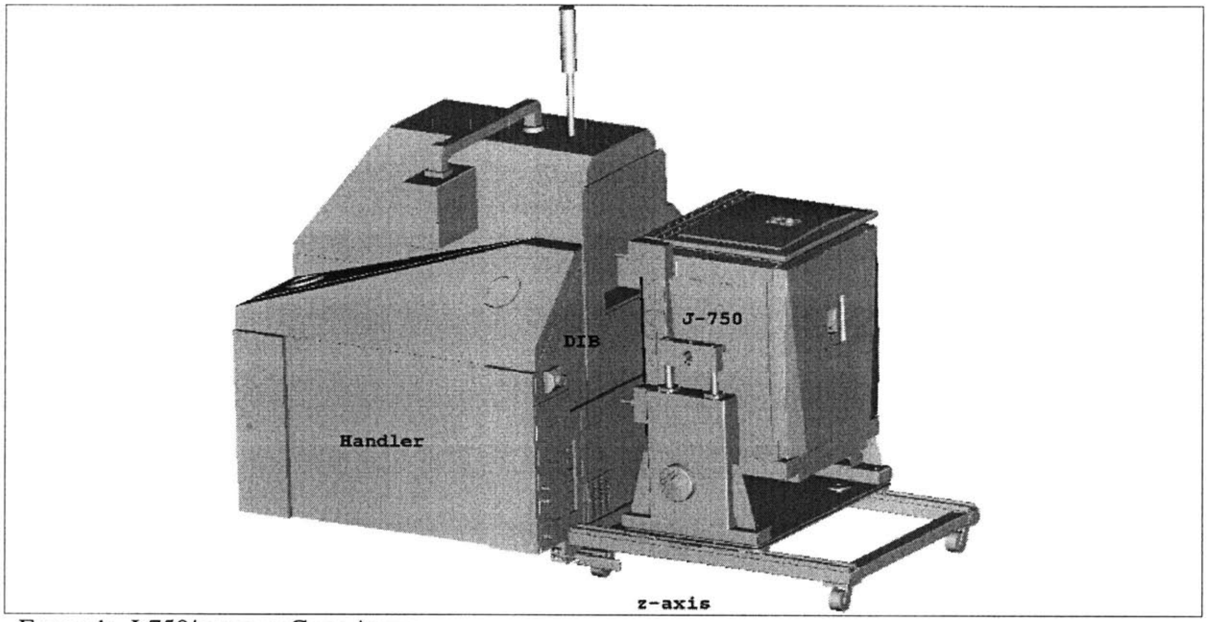


FIGURE 1: J-750/HANDLER CART ASSEMBLY

The Teradyne J-750 is mounted on a cart and mated to the Delta Design Castle handler. A DIB, not visible from this angle, completes the signal path between Tester and handler. The cart permits vertical motion and z-axis motion (toward and away from the handler).

Mating the Tester to the DIB is not a trivial task. When each DUT can have tens of channels and each DIB can provide contact with multiple DUTs, aligning the Tester's interface mechanics with the contact pads on the Tester side of the DIB is a complex mechanical problem. Proprietary Teradyne technology ensures fine alignment of the Tester and DIB, but the current solution for gross alignment is inadequate. Docking is limited to a few select handlers because of the available range of motion and the awkward design. The inconvenience of moving and docking the J-750 limits the Tester's performance in the field. A more easily manipulated Tester would allow customers to exact greater productivity from each J-750. As a result, demand, and Teradyne's revenues, will increase.

Teradyne's Blackstone project seeks to address this market need by automating the mating procedure and by increasing the manipulator's range of motion, thereby allowing even the most unskilled operator to successfully mate a Tester with any of various handlers. As part of Blackstone, this thesis documents the process of designing and selecting an actuator and control system to replace the current vertical positioning system. As the analysis will show, air cylinders prove to be the most robust solution. They perform well within the desired specifications, the compressibility of air can conveniently be used to achieve compliance, and they prove to be economical.

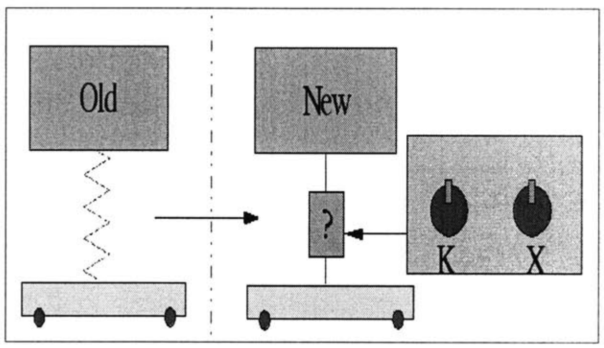


FIGURE 2: UPGRADING THE MANIPULATOR
Blackstone seeks to replace the existing hardware with a system that automates positioning and docking.

I.2 Thesis Outline

The thesis will review the existing technology, introduce possible replacements, and recommend a single system for further work. It will detail the process required to construct a prototype and assess its functionality once built.

Chapter II guides the reader through the existing technology. It gives an overview of the functionality a manipulator must provide, presents the relevant equations and design considerations, and highlights the deficiencies of the current art.

Chapter III is a detailed problem definition and specification for an acceptable system. It lists the performance requirements for an upgraded system.

Chapter IV presents four possible design solutions. The reader is introduced to four actuation techniques. Various design considerations are included. A detailed comparison followed by a recommendation for a particular system ends the chapter.

Chapter V serves as a manual for constructing and verifying the prototype's performance. A detailed Bill of Materials reflecting the device's cost structure, a guide to building the actuator, and a discussion of experimental results is included.

Chapter VI concludes the thesis. Thoughts on the entire design process and recommendations for future work are included here.

II Current Art

II.1 Introduction

Teradyne, Inc.'s current system provides limited functionality at a great cost to convenience. In this chapter, the current manipulator's workings will be explained. Specifically the positioning mechanism and compliance scheme will be addressed. The reader is encouraged to understand the necessity of having a manipulator to begin with, and to understand the failings of the current incarnation.

The existing machinery is depicted in figure 3. Two tower assemblies mounted on the cart's frame support the Tester. A manually turned crank connected to a worm gear assembly lifts the system. Springs mounted in each tower are the sources of the cart's compliance (Figure 3). The system works, but necessitates too much manual effort. Moreover, its total allowed height range approaches 4”, a value significantly less than the 10” needed to mate to the full compliment of existing handlers.

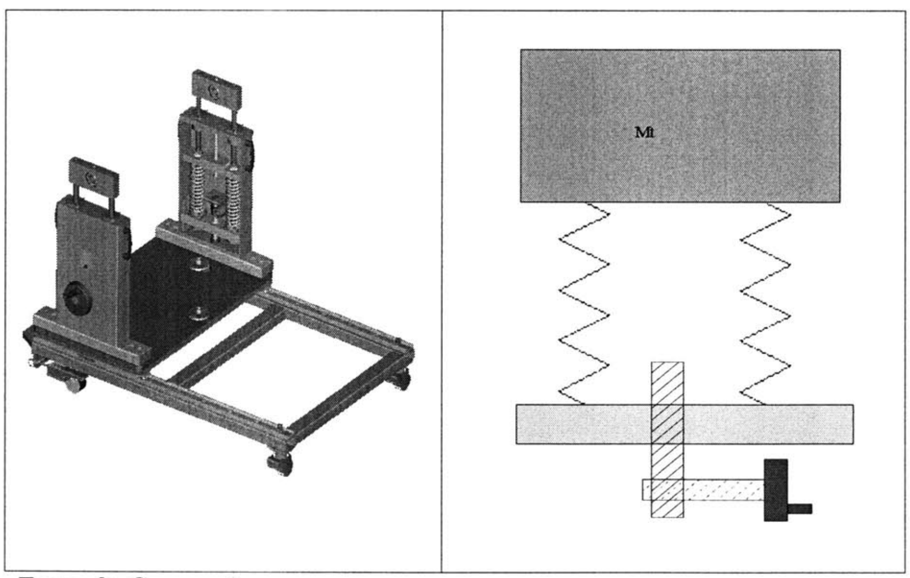


FIGURE 3: CURRENT SYSTEM

The tester is positioned by turning a crank on the Tower Assembly. The two springs mounted on each Tower provide compliance at the nominal height.

II.2 Compliance

Compliant vertical motion is needed for successful docking between a tester and handler. Due to slight variations in the floor or other uncontrollable phenomena, the difference between actual docking height and expected docking height can vary at least half an inch. To account for this discrepancy, the manipulator's vertical motion is made compliant by the addition of springs. Teradyne's proprietary fine adjustment mechanism exploits this compliance to mate the Tester and handler.

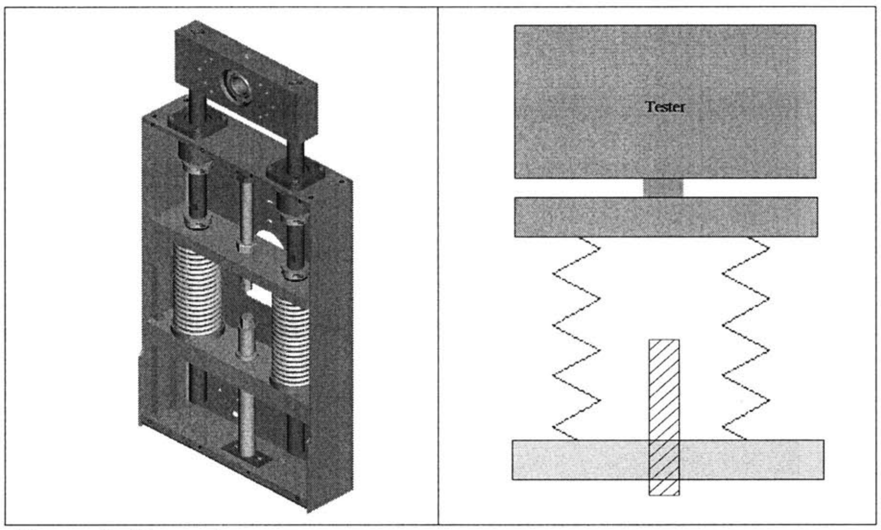


FIGURE 4: TOWER ASSEMBLY
Two springs mounted in parallel provide the tower's compliance.

Two springs in each arm assembly are used to provide the system's compliance. Each of the two linear springs are rated at a spring constant of K = 33 lb./in. Mounting the springs in parallel spreads the load and helps prevent the linear rails from jamming in their bearings. Moreover, two smaller springs can accomplish the same task as one larger spring. The parallel combination of two identical springs of constant K produces a system with K' = 2*K = 66 lb./in. A 600lb. Tester will load each arm evenly with F = 300 lb. This loading compresses the springs F/K = 4.5" to their bias point.

An arm constructed like this allows accurate and repeatable positioning while providing compliance about the nominal operating point. Though passive, the springs permit the manipulator to adapt to unpredictable changes in handler height and ensure a stable mate between the handler and the J-750 Tester.

II.3 Positioning

To vertically position the J-750 Tester, the operator turns the cranks located on each of the two arms. A worm gear assembly attached to the cranks allows a person to lift a fully loaded Tester with minimal strength. The worm gear turns a threaded rod, which in turn lifts and lowers the Tester. By positioning each side separately, the operator can move the Tester into proper position for docking.

Figure 4 illustrates the construction of the lifting mechanism. The two square threaded ACME rods, one on each side, bear the Tester's full load. Friction between the rod, the spring compression plate through which it is inserted, and the bearing retainer that contacts the bottom of the arm casing prevents the tester from dropping.

The static friction present in the mate between the ACME rod and the compression plate is well defined. The torque required to rotate an axially loaded, vertically-mounted, square-threaded joint is described by equation 1. In this example, a 1"-5 Acme rod is used to lift a system that can weight upwards of 1000lb. Since each arm is specified to support at least 75% of the system's full weight, the maximum axial load seen by the rod is 750 lb. Since the torque induced by gravity does not exceed the torque required to overcome static friction, the system will not fall on its own.

$$T = L\frac{d}{2}\frac{l \pm \pi d\mu}{\mu l \pm \pi d}$$

$L =$ load $=$ 1000lb $\quad d =$ pitch diameter $=$ 0.9 inch

$l =$ pitch $=$ 0.2 $\quad \mu =$ coeff . of sliding friction $=$ 0.15

$T_{lift} = 82\, in-$ lb $\quad T_{lower}\, 27\, in-lb$

$T_{lift} = T_{Friction} + T_{Gravity} \quad T_{lower} = T_{Friction} - T_{Gravity}$

$T_{Friction} = 54.5\, in-lb.$

$T_{Gravity} = 27.5\, in-lb.$

Equation 1: Lifting Torque for Square Threads

The Torque required to lift the Tester is calculated by using the plus signs. The Torque required to lower the Tester is calculated by using the minus signs.

Expectedly, the torque required to lift the system far exceeds that required to lower the system, because gravity must be overcome. Conversely, gravity aids in lowering. Any mechanism that seeks to lift and lower the tester must overcome the combination of gravity and friction. Therefore, designing a system that will allow an operator to lift the Tester ensures that the same operator will be able to lower it.

The torque required to lift the system far exceeds what could be considered convenient for a typical operator. To alleviate this problem, a gear ratio of 4:1 is built into the aforementioned worm gear assembly. The operator need only provide 20.5 in-lbs. to the crank. Since the handle used to rotate the crank provides a lever-arm of 1", the operator need only supply a force of 20.5 lbs. to overcome friction and gravity and lift the Tester.

Though highly inconvenient, the system provides a manageable interface for the user to position the system. The weight of a fully loaded J-750 and the torque it induces on the ACME rod are reduced four times by the gearing in the crank and worm-gear assembly. The reduction makes it possible for a typical operator to lift and lower the system.

II.4 Cost

Teradyne's paramount concern is developing a marketable product. An improvement in technology that costs too much does not suit their interests. Accordingly, the price tag of any design element looms over the heads of the engineering staff. In many cases, added functionality is compromised for a less expensive product.

Removing the handle and gear assembly is the most obvious example of reduction in functionality for the sake of saving money. As described earlier, the handle and worm gear assembly permit the operator to manageably maneuver the unwieldy Tester. Removing these does not prevent positioning, but it does make it more inconvenient. Teradyne dispensed with the worm gear assembly and molded a hex mount onto the end of the threaded ACME rod (Figure 4). Without a crank, the operator mates a socket wrench to the end of the rod and adjusts the height by turning the rod with the socket wrench. Without the gearing provided by the worm gear, the operator must now provide the full torque necessary to lift the Tester. This can be accomplished with a long socket wrench, but the system is no longer self-contained. Customers must stock a standard socket wrench and disassemble part of the cart to adjust height instead of simply turning a crank. Nonetheless, they prefer this option; customer interest has grown as a result of the design change. (A complete Bill of Materials for the existing system with and without the crank assembly is included in Appendix I.)

Total Cost of two Tower Assemblies	$3,283.19
Total Cost of two Tower Assemblies without Worm Gear Assembly and Crank	$2,641.06
Cost of Worm Gear Assembly and Crank	$642.13

TABLE 1: COST OF PRODUCING TWO ARMS FOR THE CURRENT SYSTEM WITH AND WITHOUT THE WORM GEAR ASSEMBLY AND CRANK

Customers are willing to sacrifice the convenience of a crank for near $640 in savings by removing the associated parts.

II.5 Conclusions

Though this design has found a small market, it is incompatible with Teradyne's strategy. As a preliminary product aimed at establishing a presence on the marketplace, it works, but its awkward user interface and inability to mate with many existing handlers preclude it from dominance.

To remedy this situation, any new design will need to address these concerns without adding a substantial premium to the price tag. A replacement should be able to mate with any of the standard existing handlers, do so conveniently, and do so without inflicting too much damage on the customer's bottom line.

III Problem Definition

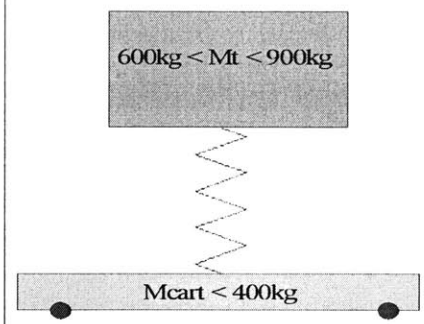
III.1 Overview

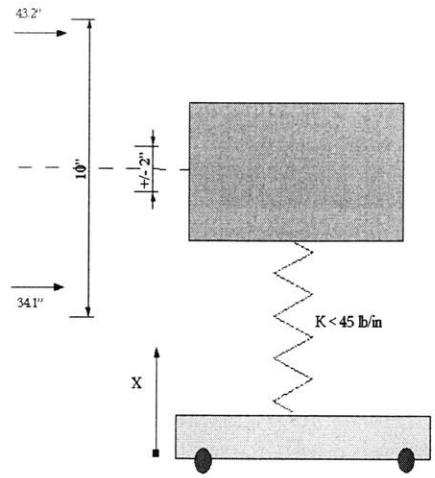
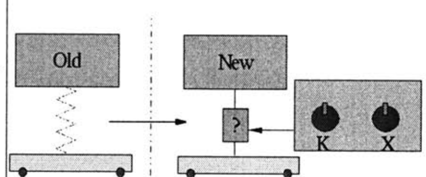
The existing Teradyne system significantly improves the mating procedure over that of previous manipulators, but its range of motion is severely limited by the manual crank used for positioning. Customers are impressed by the system's simplified manipulator but question its ability to mate with their full fleet of handlers. The Blackstone project, an effort on the part of Teradyne to both automate mating and increase the range of motion, is born of this market need.

The topic of this thesis, and the next step for Teradyne, is designing a manipulator that can dock with most existing handlers conveniently and automatically. As a matter of actual implementation, the automated vertical adjustment on Blackstone must be able to accurately move to any position in the range defined by the heights of existing handlers and to do so through a convenient and user-friendly operator interface. Moreover, once at the nominal docking height, the system must maintain compliance to account for any discrepancies between the nominal height and required height during docking. An upgraded cart that can perform these tasks for a modicum of added expense, would immediately make the J-750 assembly a far more attractive product.

To accomplish these goals, the system must meet myriad specifications. This section lists those specs. They include: weight requirements, covering the total weight supported by the manipulator; User Interface requirements, covering ease of use, safety, and convenience; Maintenance requirements, including definition of maintenance functions and the means for accomplishing them; Performance requirements, including speed, compliance, and susceptibility to unbalanced loads; Overall cost requirements, including a maximum price for any upgraded system; and Safety requirements.

III.2 Table of Specifications

Weight Requirements	• The assembly must support a J-750 configured for 512-pin test (600 lbs.) or 1024-pin test (900 lbs.). • The cart must weight less than 400 lbs.	
User Interface	Convenience • Regular operation will permit travel to eight programmed locations. • The programmed locations will match the heights of the customer's handler fleet. • The system will remember the programmed heights when unplugged.	
	Safety • The design will require the operator to use two hands. • The controls will be situated to prevent the operator's coming between the tester and handler as the two are mated.	
Maintenance Interface	• A maintenance panel will provide controls for moving the test head to an arbitrary height. • The maintenance panel will provide a means to program the current height as one of the eight allowable preset heights.	

<table>
<tr>
<td rowspan="4">Performance Requirements</td>
<td>Motion
• The system will have at least 10” of travel.
• The system will be able to dock with a handler whose Center of Test is as low as 34.1” above the ground.
• The system will be able to dock with a handler whose Center of Test is as high as 43.2” above the ground.
• At any point in the height range, the assembly will allow movement of +/- 2” about its position.</td>
<td rowspan="2">
</td>
</tr>
<tr>
<td>Compliance
• The assembly will be compliant at or below 45 lb./in. throughout its height range.</td>
</tr>
<tr>
<td colspan="2">System Response
• The system will be accurate to +/- 1/8th inch.
• The system will take no longer than 8 seconds to achieve its desired height from any starting point.</td>
</tr>
<tr>
<td colspan="2">Unbalanced Load
• Up to 75% of the load may be concentrated on one arm.
• The two arms will never differ in height by more than 1”.</td>
</tr>
<tr>
<td>Cost Requirements</td>
<td>Upgrading the system will cost less than $1500 in mass production. This includes:
• Replacing the existing crank with an automated positioning system.
• Improving the system's compliance.
• Outfitting any electronic hardware and user interface hardware needed to control the system.</td>
<td>
</td>
</tr>
<tr>
<td>Safety Requirements</td>
<td colspan="2">• System will maintain height/compliance after electrical power loss.
• System will maintain height/compliance after air loss.
• System will maintain height over 48 hours.</td>
</tr>
</table>

III.3 Weight Requirements

J-750 Testers come in multiple flavors. Different models test 512-pin and 1024-pin microchips. Expectedly, the systems that test more channels weigh more; it is a function of the added electronics. The cart must be able to support the heaviest of these systems while allowing an operator to move the entire assembly should the need arise.

III.4 User Interface

The Blackstone project's ultimate goal is an automated docking procedure that is as simple as possible for the operator to use. The goal is to allow an untrained operator to consistently and reliably dock the Tester and handler. To that end, the user interface must be simple and convenient and must ensure the operator's safety.

III.5 Maintenance Interface

On occasion, routine maintenance operations will need to be done to maintain the Tester Cart Assembly. Programming new heights and moving the tester to arbitrary heights are two functions that are envisioned for these maintenance operations.

III.6 Performance Requirements

The Tester Cart Assembly's task is to provide gross alignment over the full range of heights of the available handlers. This involves the Towers achieving their final height within a specified time limit, maintaining compliance about the nominal operating point, and doing so without tilting the tester to any exaggerated position.

III.7 Cost Requirements

A successful upgrade implies marketability. A design that work well, but is unaffordable, is inherently unsuccessful. The redesign must make minimizing cost a central concern.

IV Possible Design Solutions

IV.1 Introduction

As mentioned earlier, a replacement for the current system must provide accurate and compliant position control through a full 10" range at low cost. In addition to performance, cost is of paramount concern. A slightly less effective system that is significantly cheaper is more desirable than a system whose performance superiority is matched by its larger price tag. A number of possibilities for accomplishing the goals exist. This chapter will elaborate on four of the more promising designs and recommend a single one for further development.

Initial research has shown air driven actuators and motors to be the most promising systems. The compressibility of air and its abundance on manufacturing floors makes it a handy medium. Accurate positioning with air is achievable, and compliance can be obtained from its compressibility. A dilemma arises in choosing between available systems though. Both air cylinders and air springs seem promising. Investigating motor driven systems also appears worthwhile. Most of the required mechanical hardware already exists in the current Tower design. Mounting a motor and gear box to the existing ACME rods seems straightforward. Whether to continue to use springs or to develop an active force controller is not as clear.

This chapter will elucidate the design concerns of the potential air and motor-driven systems. The four systems analyzed are: active positioning/passive compliance with a motor and spring, active positioning/compliance with a motor, active positioning/passive compliance with an air spring, and active positioning/passive compliance with an air cylinder. The relevant theoretical background is addressed, selection of parts and an approximate bill of materials is presented, and conclusions are drawn for the various actuators.

IV.2 Active Positioning, Passive Compliance with a Motor and Springs

IV.2.A INTRODUCTION

Mounting a gear motor on the threaded shaft is a simple and direct method of automating this system. Assuming that a suitably sized motor can be found, the assembly will require minor mechanical changes. A mate between the motor and the rod will need to be devised and a mount for the motor designed, but otherwise, the existing hardware will remain virtually unchanged. The spring assembly will continue to provide the necessary compliance. This saves time for the designers involved and prevents any excess inventory from becoming obsolete. Moreover, once the system is characterized, a cheap commercial controller can be purchased and tuned to the appropriate performance levels.

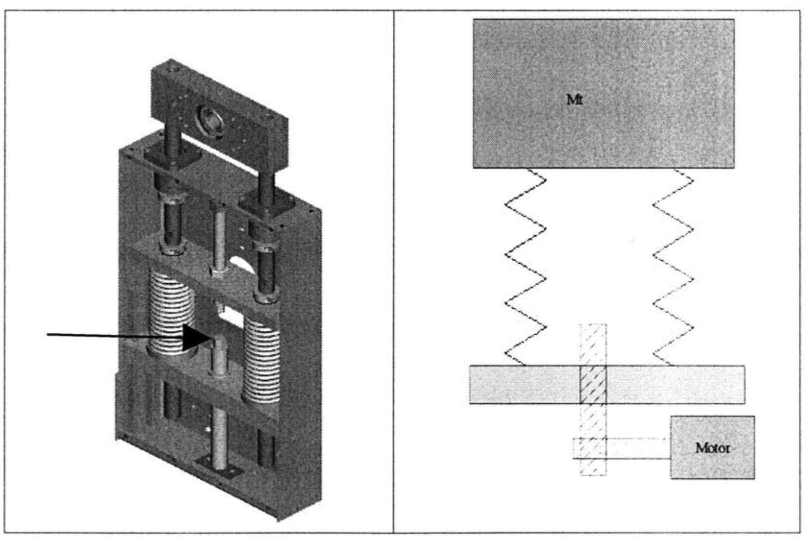


FIGURE 5: LOCATION OF MOTOR
A Parallel Shaft Gear Motor will be mounted onto the ACME rod for positioning. The springs from the previous system can be used to provide compliance.

IV.2.B Compliance

The same compliance scheme used in the current system is used here; however, in place of the 33 in./lb. springs, 22 in./lb. springs should be used. The stiffness in each of the arms will be reduced accordingly, from 66 in./lb. to 44 in./lb.

IV.2.C Motor Selection

Experience and intuition imply that a motor capable of performing the desired tasks exists; however, determining the best motor is a slightly more complicated procedure. Any acceptable actuator need meet a slew of requirements. Calculations in section II.3 established the torque necessary to lift and lower the Tester. An acceptable motor must have enough power to provide this torque and do so at a speed capable of meeting the performance requirements. Additionally, any such motor should fit within the available space within the tower.

Torque		***Pitch***	***Speed***				***Power***	
torque (inlb)	**torque (ftlb)**		**in/s**	**rps**	**rpm**	**rad/s**	**ft lb/sec**	**hp**
98.4	8.2	0.2	0.10	0.50	30.00	3.14	25.75	0.05
98.4	8.2	0.2	0.33	1.65	99.00	10.36	84.97	0.15
98.4	8.2	0.2	0.50	2.50	150.00	15.70	128.74	0.23
98.4	8.2	0.2	**1.00**	**5.00**	**300.00**	**31.40**	**257.48**	**0.47**
98.4	8.2	0.2	**2.00**	**10.00**	**600.00**	**62.80**	**514.96**	**0.94**

Table 2: Power Requirements for Motor Drive

As can be seen in this table, powerful motors are needed to lift the Tester at the desired speed. Motors that provide much more than ¼ hp can cost as much as $400 a piece, not including the added driver circuitry. Meeting the speed spec may be difficult with this actuator option.

Each arm ostensibly lifts half of the Tester's weight. However, the design specifications allow for up to 75% of the weight to be concentrated on one arm. Equation 1 demonstrates that a torque of 82 in-lb. is sufficient for this task. For reliable performance that does not tax the motor to the limit of its capabilities, the motor of choice should be able to provide at least 120% of this torque or 98 in-lb.

Additionally, the actuator is required to cover the full 10" range in less than 8 seconds. To drive a 1"-5 ACME screw, a screw with 5 threads per inch, at this speed, it must rotate at 375 rpm.

Gear motors are typically classified by the horsepower they can output. If the desired torque and speed are known, calculating the power is straightforward. Table 2 indicates that a motor capable of at least ½ hp is needed to move 10" in eight seconds. Motors that powerful are typically both large and expensive, totaling as much as $800 for the motors alone. Driver hardware can be nearly as much. Speed may have to be sacrificed for budgetary concerns.

IV.2.D CONTROL

Accurate position control with a servo motor is an old science. It is no less powerful for its age, and still requires careful design work. Controlling the motor that will actuate the system is a classical example of this procedure. The object is to overcome the system's inherently slow dynamics and replace it with a tailor made control system. The motor's capabilities and friction present obstacles, but a successful control scheme optimizes performance in spite of physical limitations.

The Tester's vertical position is the sum of the effects of two separate systems, the motor and the springs. Though the springs settle to a known bias point, their compliance adds to the system's transient response. A robust control scheme accounts for this existing compliance.

Compliant devices act in parallel with the motor. That is to say, the Tester's final position will be the sum of any effects due to the motor and any effects due to the spring. When placed in feedback, the compensator must account for this parallel construction. Figure 6 illustrates this point.

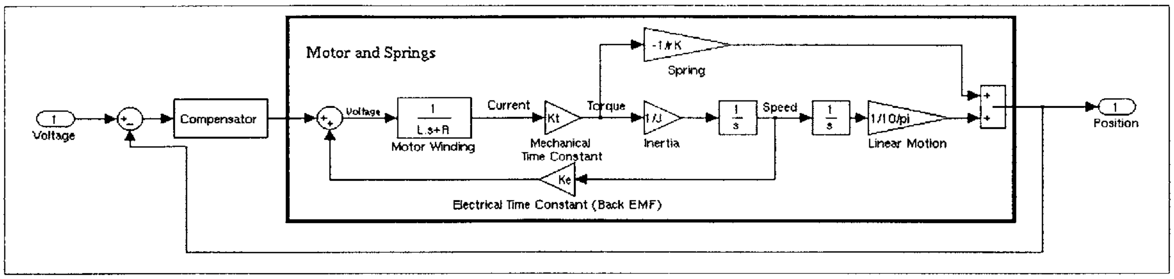


FIGURE 6: BLOCK DIAGRAM OF SYSTEM

$$\frac{\Theta}{V}=\frac{1}{K_T s\left(t_m s+1\right)}$$

K_t = motor constant

t_m = System Time Constant

EQUATION 2: PLANT EQUATIONS FOR HIGHLY GEARED DC MOTOR
This equation is an accurate simplification for a highly geared motors. Though it ignores the springs, it will nonetheless be accurate at the bandwidth considered.

Were the spring not present, a system like this can be compensated with a standard PID controller. However, the spring complicates matters. It adds two high frequency zeros to the transfer function. Qualitatively speaking, a spring is rigid at low frequencies. The two masses can be treated as one aggregate mass. Conversely, at frequencies beyond the system's natural frequency, the spring decouples the two masses. The 33 lb./in. springs used in this design are sufficiently large that the they can effectively be ignored when designing a motor controller. That is to say, at the motor bandwidth being explored here, the springs appear to be rigid.

That being the case, the task at hand is to compensate the standard motor transfer function described in equation 2, with the caveat that the system's bandwidth must be sufficiently low to obviate considering the spring's effect. In its most general form, the most suitable compensator is a PID controller. It can be reduced to a P, P+I, or P+D controller as necessary. In this application, the goal is to raise low frequency gain. There is no reason to raise high frequency gain, since the intention is to create a relatively low bandwidth control loop that bypasses the spring's effects. However, a phase increase is needed at crossover to counteract the phase of the motor's two poles. A standard lead compensator implemented through the PID interface suffices. The response pictured in figures 7 and 8 results. At approximately 12 rps, 2 Hz, the magnitude portion of the frequency response flattens out due to the spring. It would otherwise have continued to decrease. However, by this point, the closed loop magnitude has fallen to -25 DB. The step response of figure 8 verifies that the spring has negligible effect on the system.

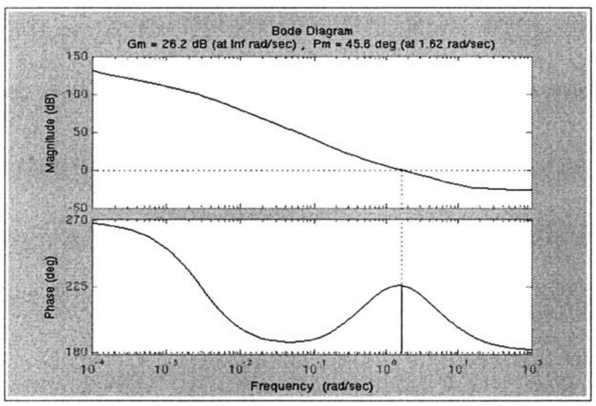


FIGURE 7: CLOSED LOOP BODE RESPONSES

As desired, the closed loop bode indicates good command following at low frequencies. The effects of the two zeros at 800 rps are visible at higher frequencies but they are radically attenuated.

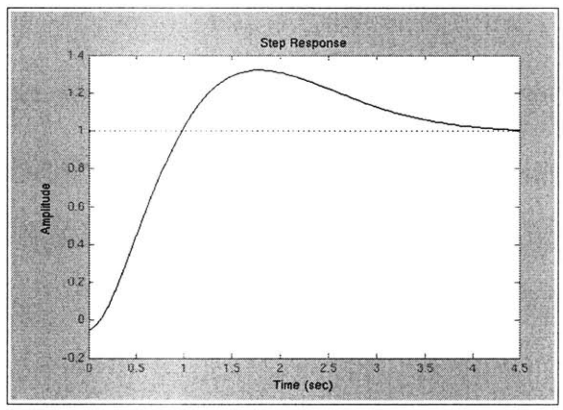


FIGURE 8: STEP RESPONSE

The compensated system achieves a height of 8 inches in under 10 seconds, as is required. In reality, the motor may not be able to keep up.

The two figures above illustrate the effectiveness of this compensation technique. Though the bode plot indicates a system bandwidth of around 1.5 rps, the actuator will never be responding to high speed inputs. In fact, it will only be required to respond to step inputs. As the step response suggests, in its current form, it will easily pass the overshoot and speed specifications laid out earlier.

In converting this theoretical model into practice, a number of concerns must be addressed. Firstly, the actual compensation will be done digitally within a PLC. The

Automation Direct DL05 PLC selected for the implementation, it combines the ladder logic capabilities needed for the user interface and the analog control circuitry needed for positioning for only $100, samples at well above 1MHz. Since the motor is operating a bandwidth well below 1kHz, the sampling period is sufficiently small for the digital controller to approximate a continuous-time system.

Additionally, though not pictured in the block diagram, a power amplifier will be required. The PLC is a low power device. It does not have the capability to power an industrial motor. Instead, a power amplifier capable of providing unity voltage gain with far more current is needed. Bodine markets such devices: they are included in the bill of materials at the end of this chapter.

Though the presence of a spring seems inconvenient, the analysis has shown that a standard P+D controller will accurately control the system if the loop bandwidth is made to be on the order of 0.25 Hz. At this low frequency, the spring appears rigid; positioning is straightforward.

IV.2.E Materials and Cost

The previous sections address the engineering design constraints present in this project. Included are the compliance mechanism, the control structure, the motor drive, and the existing hardware within the arm assembly. Whether or not the prototype works, the sum of its parts must cost less than the allotted amount.

Bodine-Electic offers the best option for motor control. Their package includes a ¼ hp DC-gear motor and associated electronic hardware. The motor is geared to produce sufficient torque, but insufficient speed to meet all performance requirements. More powerful motors are simply too expensive to contemplate. The cost of the added electronics - one board drives the 130V DC motor based on a 0-10V analog input, one board accepts analog inputs between 0-10V, and one board handles polarity to permit driving the motor in both directions – is almost enough to invalidate this option. Bodine's AC motor and speed-controller is $100 less expensive, but less controllable.

The hardware necessary to control the system is also a substantial cost. The least expensive controller on the market that combines analog signal control and digital logic for the user interface is AutomationDirect's DL05 PLC. Coupled with an Analog IO card, the PLC's internal electronics can provide the sort of closed loop control necessary to manage this system. A 24 Volt power supply is needed to power the system, and various switches are used as a rudimentary user interface.

As is evident in table 3, the cost of this system far exceeds that of the current design. It remains to be seen whether or not the added cost is offset by the added functionality. It is likely that this option is too expensive though.

Item	Vendor	Quantity	Cost per Item	Total
42AE - DC inline Gear Motor, Model 4583	Bodine-Electric	2	$247.32	$494.64
encoder 0941	Bodine-Electric	2	$78.28	$156.56
power amplifier cards	Bodine-Electric			
0888 - Analog Control		2	$68.80	$137.60
0890 - Remote Direction Control		2	$74.40	$148.80
0850 - Driver		2	$132.80	$265.60
2 in 2 out analog IO card	automation direct	1	$179.00	$179.00
PLC	automation direct	1	$99.00	$99.00
Linear Transducer	Unimeasure, Inc.	2	$150.00	$300.00
24 Volt DC Power Supply	mcmaster-carr	1	$62.00	$62.00
NO program button	digikey	1	$2.00	$2.00
On-Off-On rocker switch	digikey	2	$2.00	$4.00
rotary switch	Grayhill, Inc.	1	$20.00	$20.00
existing assembly	Teradyne	1	$2,641.06	$2,641.06
Total				$4,510.26
Difference with Current System				$1,869.20

TABLE 3: COMPLETE BILL OF MATERIALS FOR ACTIVE POSITIONING, PASSIVE COMPLIANCE WITH A MOTOR AND SPRINGS

Using a motor and the associated electronics adds a full $2,000 to the cost of the existing system. It is unlikely that Teradyne would be willing to spring for such an increase in cost.

IV.2.F Conclusions

Replacing a crank with a motor is a well-established means of automating rotary motion. The existing Teradyne system lends itself to this technique. The addition of a motor and some electronics transforms the current device into an easily controlled manipulator. In the absence of other options, this solution appears promising. However, its costly price tage begs further study of alternative solutions.

IV.3 Active Positioning and Compliance with a Motor

IV.3.A Introduction

For a motor driven system, the most elegant solution would include active position and force control. Active position control accurately moves the Tester to the desired position. Tailored force control provides constant stiffness independent of operating height. To make this work though, the system will require accurate sensors, two control schemes, and a host of electronics able to implement the controllers, collect the data, and account for variable weight testers.

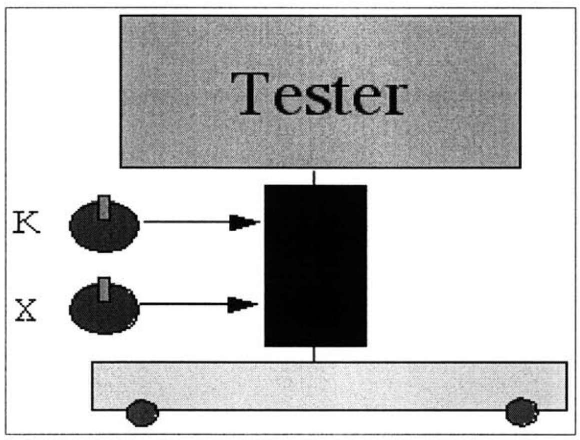


FIGURE 9: HIGH LEVEL DIAGRAM OF DESIRED SYSTEM. A motor will provide both position and compliance control.

An effectively designed system could overcome the inherently slow dynamics of the existing mechanics. The position controller would perform quickly and with very little ringing and the force controller would make stiffness controllable and independent of any system dynamics. Customer demand would no doubt surge.

Such a system comes with dilemmas though. High cost is a logical worry, and this analysis will certainly bear that out. The control scheme is not as simple as would be liked either. For a fixed load, tailoring a control system is straightforward. However, to position a system that can weigh anywhere between 600 lbs. and 1000 lbs. requires more intelligent methods. Adaptive control comes to mind. More complicated controllers mean more money, more time, and more parts. This chapter outlines what could be considered the dream machine, but as the analysis will show, it is too burdened by cost and complexity to be practical.

IV.3.B Motor Selection

There is no reason to expect the load requirements have changed from the previous chapter to this one. The same motor that suits active position control and passive compliance will suit here. The Bodine-Electric ¼ hp DC-gear motor will suffice.

IV.3.C Control

This system diverges from its predecessor only slightly from a theoretical standpoint. The position control scheme is practically identical to the one discussed earlier, but force control is now accomplished actively. Whereas the physical springs take over in the passive model, virtual springs provide force control in the active design. In the absence of springs, position control is somewhat simpler, but the additional force controller makes the whole system more complex.

The algorithm toggles between the two controllers depending on the Tester's state. If it is within a nominal distance of its desired position, force control is toggled on. In this mode, the motor is programmed to imitate the motion of a spring of some arbitrary stiffness. Should the Tester's desired height change, the position controller will switch on, move the Tester to its new desired position, and revert to the force controller once the new position is reached.

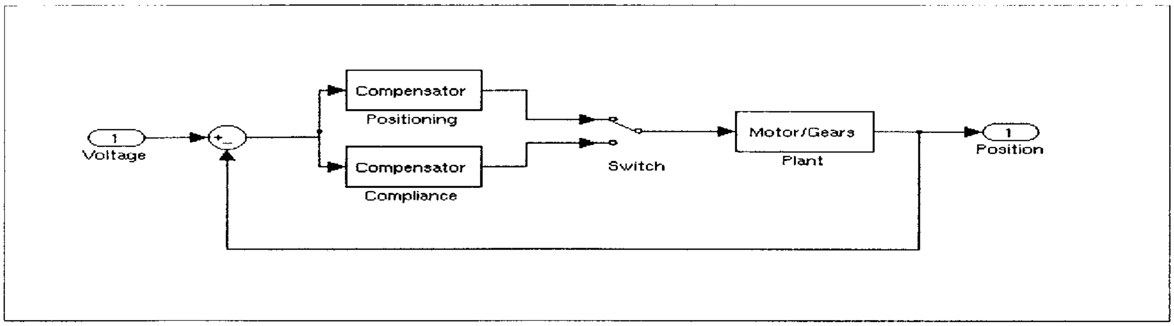


FIGURE 10: BLOCK DIAGRAM FOR ACTIVE POSITION AND ACTIVE FORCE CONTROL

Depending on the Tester's position, the control algorithm toggles between a position and force compensator. The position compensator functions much as the one described earlier.

As is described in section IV.2, the spring plays a very small part in designing the controller for the Active Positioning and Passive Compliance actuator. Consequently, the compensator used for that system can easily be used to compensate the positioning system here.

Designing the force controller is the next step. A suitable positioning system exhibits good command following over its bandwidth, that is to say, over some range of low frequencies, the closed loop system provides unity gain. The output follows the input. Force control is somewhat different. Instead of expecting the output to follow some signal input, the system should displace proportionally in the presence of a disturbance force. That is, F= -kx, or the system behaves like a spring.

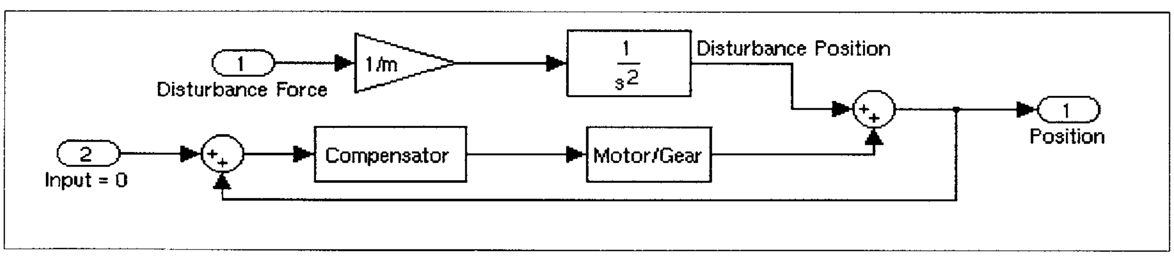


FIGURE 11: BLOCK DIAGRAM OF FORCE CONTROLLER AND DISTURBANCE

As the block diagram shows, disturbance force is related to disturbance position by 1/ms^2. Therefore, if the closed loop can be made to imitate ks^2 over some bandwidth, the relationship from Disturbance Force to Position will be k/m. That is to say, the force and position will be proportional and the system will have been made to look like a spring. Without delving into the full details, the general approach is to cancel the motor pole at s=1/tm and add another pole at s=0. The compensated loop transfer is of the form K/s^2. If K is sufficiently high, the closed loop will look like s^2 /K over its bandwidth. When multiplied by 1/ms^2, the full system is of the desired form 1/mK. The step for K=.03 is plotted below. After a quick transient, the system settles to a stiffness of 21 lbs./in., a number that is perfect for this Teradyne application.

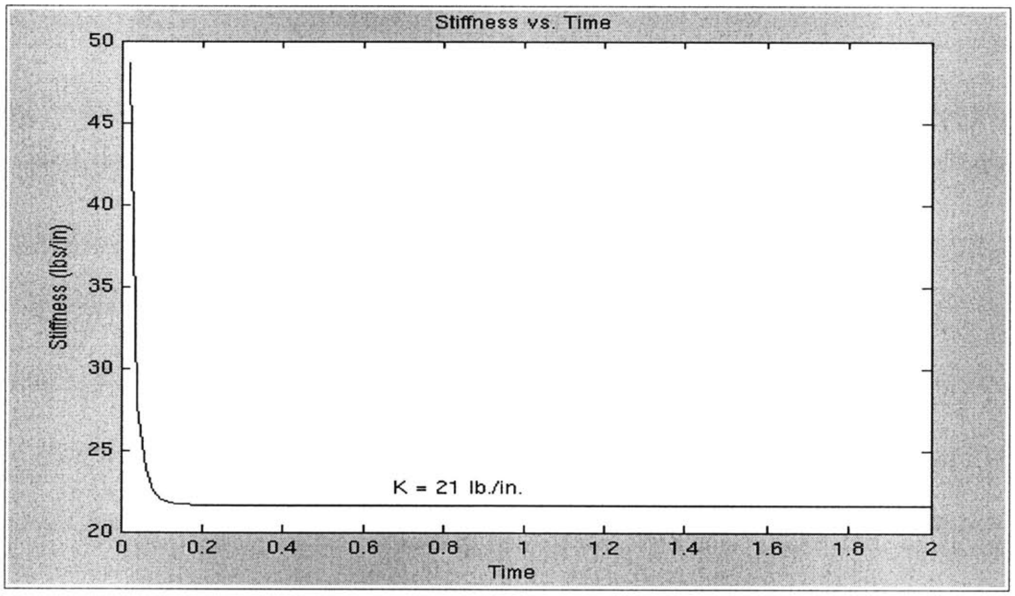


FIGURE 12: SYSTEM STIFFNESS VS. TIME

Unlike the previous system though, this variant requires power at all times. Active compliance depends on the motor receiving an uninterrupted supply of current. Should power fail for any reason, the gearing in the motor is sufficient to keep it from dropping, but the stiffness will inhibit motion of any sort.

IV.3.D MATERIALS AND COST

It should be no surprise that this system is far more costly than the one discussed in section IV.2. Though the actual springs present in the original have been removed, the addition of a more complicated controller, force sensors, and additional hardware, more than offset the savings.

Like the previous case, this design can be thought to have three major components. The original tower mechanics, minus the springs, is necessary here. The motor and the various drivers needed to make it work consume far less space than the towers, but nearly as much money. Lastly, the physical controller, a PID equipped PLC in this case; a position sensor, the same string-pot; a force sensor; and other hardware needed to complete the control scheme complete the bill of materials. The price tag is expectedly hefty.

Item	Vendor	Quantity	Price	Total
1" diameter, l=0.25, rolled ball screw	Nook industries	2.00	59.36	$118.72
24 Volt DC Power Supply	McMaster-Carr	1.00	$62.00	$62.00
Controller and Sensors				
DL06 PLC	automation direct	1.00	$199.00	$199.00
2 in 2 out IO card	automation direct	2.00	$149.00	$298.00
Linear Transducer	unimeasure, Inc.	2.00	$150.00	$300.00
Torque Sensor	Sentran	2.00	$100.00	$200.00
Buttons, Switches, Knobs				
NO program button	Digikey	1.00	$2.00	$2.00
On-Off-On rocker switch	Digikey	2.00	$2.00	$4.00
rotary switch	Grayhill	1.00	$20.00	$20.00
Motor and Parts				
42AE - DC inline - 4182	Bodine-Electric	2	247.32	$494.64
encoder 0941	Bodine-Electric	2	78.28	$156.56
power amplifier	Bodine-Electric			$0.00
0888 - Analog Control		2	$68.80	$137.60
0890 - Remote Direction Control		2	$74.40	$148.80
0850 - Driver		2	$132.80	$265.60
Teradyne Parts added and removed				
existing assembly	Teradyne	1	2641.06	$2,641.06
Thd Rod, ACME 1"-5	Teradyne	2.00	-$37.00	-$74.00
Spring, Compression, 2.812 od, .283 wd	Teradyne	4	-37.31	-$149.24
Plate, Spring Capture	Teradyne	4	-19.95	-$79.80
Total				$4,744.94
Difference with Current System				$2,103.88

TABLE 4: COMPLETE BILL OF MATERIALS FOR ACTIVE POSITIONING AND COMPLIANCE WITH A MOTOR

Though pretty, this system is far too expensive. It's a nice pipe-dream though.

IV.3.E CONCLUSIONS

Though indeed elegant, this solution is not practical. It's overkill. It's more likely to be found in a graduate student lab than in industrial application. It does too much for too much money. Accurate position and force control is exactly what Teradyne requires, but not for nearly $5,000.

Unlike the previous prototype, this design offers an active force controller. In robotics applications, that technology is absolutely required. For gross positioning, the springs present in both the existing and previous design are perfectly satisfactory. Using far more power and components, this system does exactly what four linear springs accomplish on the existing cart. Technology implemented intelligently improves functionality, it does not simply replace it with a more expensive version.

Of course, the drawbacks of the first prototype also apply here. It is bulky, it contains a great deal of parts, and it is unable to meet the desired performance requirements. Once these drawbacks are considered in combination with the amount of electronics necessary to control it, this system can rightly be ruled-out.

IV.4 Active Positioning, Passive Compliance with an Air Spring

IV.4.A Introduction

Until this point, only motor driven systems have been considered. There is no reason to constrain the design to one actuator though. Pneumatic systems are equally promising; air has been successfully employed to achieve compliant motion in the past.

Various air driven actuators exist, but air springs appear to be the best option. Air springs are glorified balloons. Though not quite as amorphous, an air spring under load will expand to the path of least resistance. They are constructed of materials far more rigid than those of balloons and can easily handle the loads present in this case, but any accurate axial motion must nonetheless be guided. The compressibility of air also ensures that the device will be compliant at its operating position. Air springs also require significantly less room than other pneumatic devices. When fully compressed, they can be as little as two inches tall.

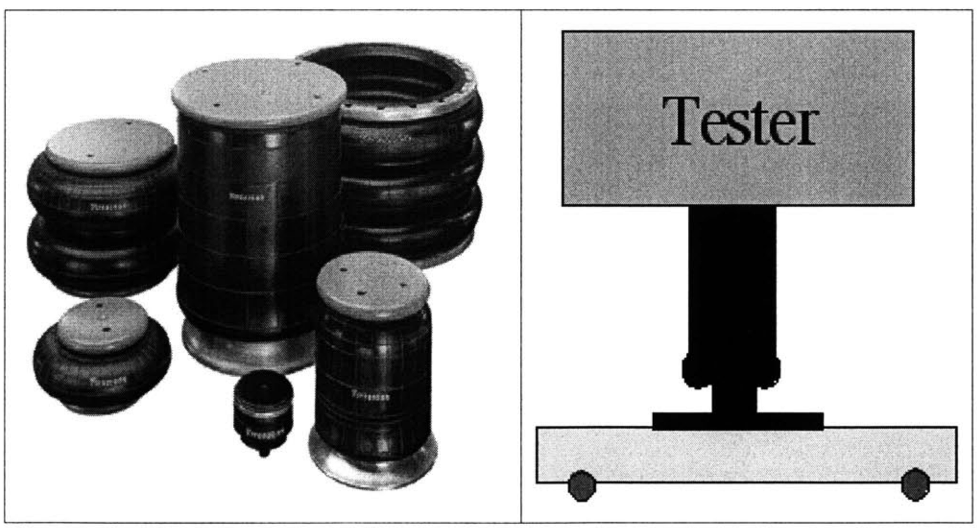


Figure 13: Firestone Airstroke Actuators

Both 'convoluted' and 'reversible sleeve' devices are pictured. Reversible sleeve air springs can be modeled as cylinders throughout much of their range.

IV.4.B Positioning

Though air springs can accept side loads, they are designed to operate axially. They present an advantage over standard air cylinders in that misalignment does not cause wear, but they must be guided to achieve accurate positioning. If a linear guide like the one already incorporated into the existing towers is present, the air springs can be used to accomplish accurate position control.

These actuators come in multiple styles. Figure 13 illustrates the difference between convoluted and reversible sleeve actuators. Convoluted actuators can shrink to negligible size, but reversible sleeve actuators are easier to control. As the image shows, reversible sleeve actuators are far more cylindrical in shape. At a given pressure, the load a reversible sleeve actuator is capable of handling is practically independent of height. Through most of its range, it can be modeled as an air cylinder with some constant surface area. Convoluted actuators are far less linear. To support the same load at its maximum height as at its minimum height, a convoluted device may need as much as twice the air pressure. As the device is inflated, its average radius decreases. Accordingly, the effective surface area with which it can 'push' back on a load decreases, necessitating more air pressure to support the same load. This effect does not plague reversible sleeve actuators.

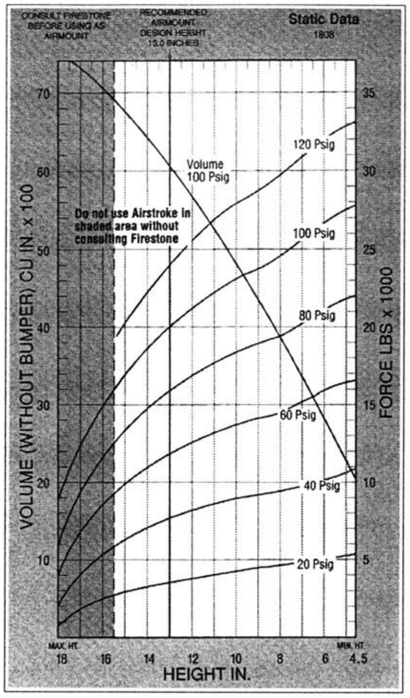


Figure 14: Load vs. Height for Convoluted Actuators

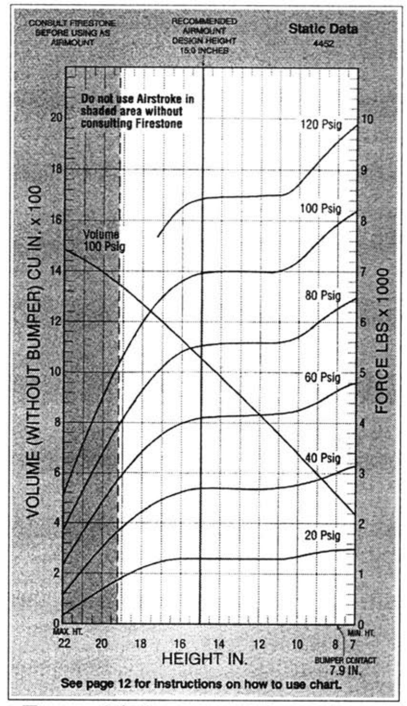


Figure 15: Load vs. Height for Reversible Sleeve Actuators

Whereas pressure vs. height under constant loading is a non-linear function for convoluted actuators, reversible sleeve actuators can support the same load at all heights throughout their range with the same internal pressure. The load lines for reversible sleeve actuators are flat through must of their range as compared with convoluted devices (Figures 14 and 15). The linear relationship between the system's total volume and its height is a further illustration of this point. As figure 15 shows, the reversible system's volume is essentially proportional to its height. The non-linearity of the same relationship for a convoluted device is clear in figure 14. The 'reversible sleeve' actuators are far more linear at lower air pressures, but as subsequent analysis will show, the air spring will not need a great deal of pressure for the current application.

Each tower need only support 750 lbs. at most. Since the device can support 1350 lbs. at 20 psi, only slightly above 10 psi will be needed to support the maximum expected weight. Assuming a linear relationship between pressure and load, as figure 15 would suggest, in fact only 11 psi is required. Under most normal conditions though, when a tower can be supporting as little as 300 lbs. of a 600 lb. Tester, the bellows will need far less than 10 psi.

Of the available options, reversible sleeve actuators are more controllable. It is also the case that the convoluted actuators capable of a full 10" of extension are more expensive and unnecessarily stiff. For long stroke distances under precise loading conditions, the reversible sleeve is a far more attractive option.

IV.4.c Compliance

Though air springs seem to present a convenient and compact means of positioning, they can not provide the compliance required. They are simply too stiff. These devices tend to be used to isolate vibration in trucks or in multi-ton industrial machines, not as springs designed for human use.

Determining the device's incremental performance about a particular operating point is impossible without an exact mathematical representation of its dynamics. In lieu of providing such a model, Firestone quotes the device's stiffness at differing heights and pressures. At a height of 15", 7.1" above the device's minimum height and 5" below its recommended maximum (i.e. A point in the typical range of use and one that would be used in this application), the actuator characterized in figure 15 has a spring rate of 395 lbs./in at 40 psi. The spring rate at 10 psi will be approximately 100 lbs./in. The specifications call for an assembly with a spring rate no greater than 45 lbs./in.

Though the air spring seemed promising, it is not designed to be particularly soft. Under normal design conditions, it is far more rigid than is required by the Blackstone project. The device will not work.

IV.4.D CONCLUSION

Air springs are not the solution. They are not able to provide the desired compliance. Moreover, Firestone specifically warns that these devices will not operate properly at internal pressures below 10 psi. As the above analysis has shown, the air springs will never be filled with more than 11 psi, and average pressure will be far below 10 psi. This actuator will not work.

IV.5 Active Positioning, Passive Compliance with an Air Cylinder

IV.5.A Introduction

Air powered pneumatic cylinders are another possible actuator for this system. Cylinders are capable of providing direct linear actuation at the required speeds, the needed distance, and the specified weight. Additionally, once at a nominal position, the compressibility of air can be used to achieve the desired compliance. Initial price surveys indicate that a full system will easily meet the price requirements as well. The primary considerations in taking this approach are discussed in the subsequent sections.

IV.5.B Compliance

By exploiting the properties of compressible fluids, pneumatics can be used to achieve compliant motion. Like a balloon, an air filled cylinder will be squishy about its operating point. The degree of this squishiness, the system's compliance, is a function of the cylinder's size and the air pressure. By correctly choosing these parameters, an air cylinder serves the dual purpose of position and force control.

The Ideal Gas Law accurately predicts the behavior of air in an enclosed volume. Imagine a cylinder of surface area SA and height h_{max}. When fully extended, the cylinder's total volume is $V_{max} = SA * h_{max}$. If a reservoir of fixed volume V_1 is attached to the cylinder, the system's maximum volume is $V_{max} + V_1$. Similarly, when the actuator is fully retracted, the minimum available volume is V_1. The important question is: if such a system exists and is loaded with some force parallel to the direction of motion, what is the resulting compliance at a height h? Essentially, with what force does the system resist attempts to displace the cylinder from height h to height h'?

Intuition indicates that the system's compliance can not be constant, as is the case in a linear spring. Assume for a second that there is no reservoir. When the cylinder is fully retracted, attempts to retract it further are impossible. It is infinitely stiff. Conversely, at some extension, the system's compliance is certainly finite. It stands to reason that the available volume in the system is in some way inversely related to the system's stiffness. Zero volume leads to infinite stiffness and increasing volume leads to decreasing stiffness. To mitigate the effects of changing volume, a constant volume chamber of size V_1 can be added. Though the stiffness will certainly change as V varies from its minimum at V_1 to its maximum V_1+V_{max}, the presence of the reservoir will minimize the effects. By appropriately selecting V_1, a system whose compliance is within the appropriate specifications throughout a particular 10" range can be engineered.

Intuition also expects the input pressure to play a part. Pressure, or lack thereof, is the cause of the difference between a flat tire and a filled one. The very essence of flatness is that the tire is too easily compressed. A filled tire is more resistant to compression. Consequently, we would expect greater pressure to coincide with greater stiffness. In the case of this system, the pressure must of course be a constant value that is capable of counteracting the weight of the Tester. Specifically, the force acting upward on the cylinder is equal to the pressure within the cylinder multiplied by the surface area upon which it acts. For a fixed Tester weight and cylinder surface area, the pressure must also be fixed.

Equation 3, presented below, gives a detailed mathematical analysis of the thought experiments presented above. Not surprisingly, the system's stiffness is proportional to pressure and inversely proportional to the available volume.

¿

$$F_O = P_O * SA = F_{Tester}$$
$$V_{O'} = V_O + \Delta V$$
$$\Delta V = SA * \Delta h$$
$$F_{O'} = F_{Tester} + F_{Displacement} = F_O + F_{Di.}$$
$$P_O(V_O + V_1) = P_{O'}(V_{O'} + V_1)$$
$$= P_{O'}(V_O + V_1 + \Delta V)$$
$$P_{O'} = \frac{P_O}{(1 + \frac{\Delta V}{V_O + V_1})}$$
$$for\ (V_O + V_1) >> \Delta V$$
$$P_{O'} = P_O(1 - \frac{\Delta V}{V_O + V_1})$$
$$F_{O'} = P_{O'} * SA = P_O * SA(1 - \frac{\Delta V}{V_O + V}$$
$$= F_O - P_O SA \frac{\Delta V}{V_O + V_1}$$

Equation 3: Compliance of Air Cylinder

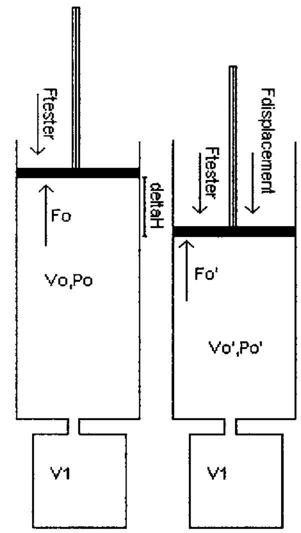


Figure 16: Model of Compressed Cylinder
All else constant, when Volume decreases pressure must increase. The system thus resists compression. About an nominal operating point, this resistance can be characterized as a spring constant.

As this analysis shows, the spring "constant" K varies but is limited. It will never exceed $K = P_0 \frac{SA^2}{V_1}$. Table 5 illustrates the change in stiffness as a cylinder with circular cross section and 80mm-bore size is extended from 1" to 13.5". A reservoir of size V_1=1L is attached to ameliorate the effect changing volume has on stiffness. At all points within the range of motion, the system's compliance surpasses the 45 lb./in. required in the specifications. Conveniently, the system becomes easier to manipulate as it gets higher. Experimentation has shown this to be a boon. Human physiology makes it easier to apply a vertical force at waist level than at shoulder level. Though the system's stiffness decreases as it rises, it is not perceptibly easier to manipulate. It is harder to apply the same force at chest-level as at waist-level.

Height of cylinder (in)	1.0	3.5	7.0	13.5
K predicted (lb/in)	34.0	26.5	20.2	14.1

TABLE 5: COMPLIANCE OF AIR CYLINDER AT VARIOUS HEIGHTS

These data assume an air cylinder of 80mm bore size with at least 13.5" of stroke. A reservoir of V_1 = 1L is used for calculations. The Tester used, a 512 pin J-750 weighs approximately 600 lb. If each tower bears half that weight, the pressure required to equalize 300 lbs. spread over the surface area of an 80mm bore is 38.5 psi.

It seems clear that the ability of air cylinders to provide accurate and compliant position control with a bare minimum of off-the-shelf parts works greatly in its favor. Bundling the two most critical performance requirements into one standard actuator makes for cheap and effective industrial practice.

IV.5.c Construction

The system specifications essentially determine the construction of the pneumatic system. Two cylinders will be required, one for each to the two towers. They must have at least a 14" stroke to satisfy the range of motion requirements. Moreover, they must be able to support the Tester's full weight. Finally, the system must accomplish motion with the specified performance requirements.

Festo, Inc. has provided sample geometry for the prototype system (Figure 17). The cylinders are sized to carry the Tester's full load at normal operating pressures without creating too large a footprint. To meet these goals, a cylinder with an 80-mm bore size has been selected for the prototype. An 80-mm diameter bore yields a surface area of 7.8 square-inches. If a 600lb.Tester will load each tower with 300 lbs., a pressure of 38.5 psi is required. Should the load fall entirely on one side, the cylinder can still withstand the aggregate 77psi.

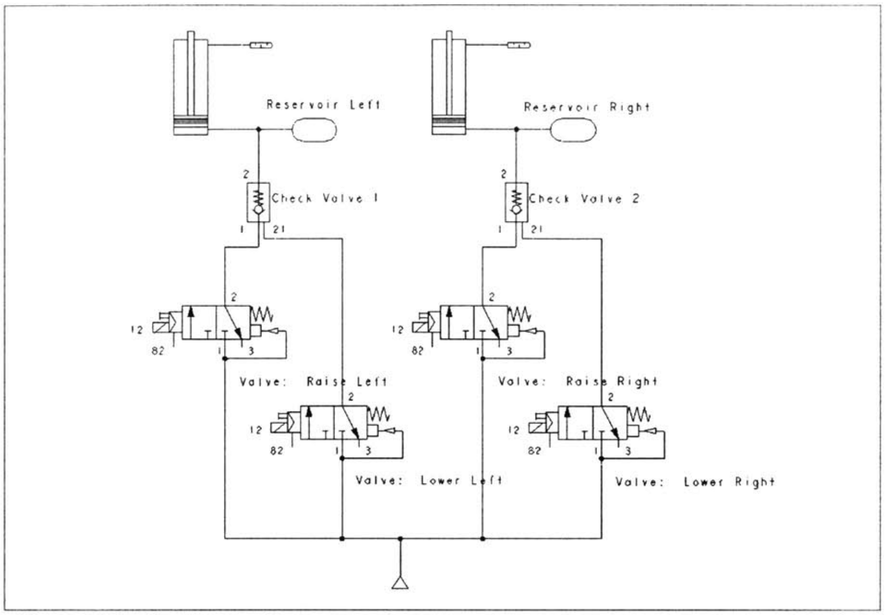


FIGURE 17: CIRCUIT FOR PNEUMATIC ACTUATOR
Two identical circuits are used to actuate the left and right arms. When the 'Raise' valves are toggled, air is forced into the cylinders. The check valves prevent any air escaping. When the 'lower' valves are toggled, the check valves are opened and air escapes the cylinders.

As Figure 17 illustrates, each of the two towers is controlled separately. When either *Valve: Raise Right* or *Valve: Raise Left* are triggered, air is forced from port 1 through port 2, through the associated check valve, and into the volume chambers and cylinders. In their standard state, the check valves only permit airflow in one direction. Therefore, raising an arm only requires actuating the *Raise* valve. The check valves permit air flow in both directions when an air signal is given on line 21. To lower the cylinders, the *Lower* valves need to be actuated. A pilot signal is fed through the *Lower* valves to the check valves, and air exhausts through the exhaust ports of the *Raise* valves.

Raise	Lower	Result	Explanation
Off	Off	Nothing	Air does not flow through any valve.
Off	On	Lower Tower	Air is provided to line 21 of the check valve allowing bi-directional air flow. Since the raise valve is not actuated, air exhausts through port 3 of the Raise valve.
On	Off	Raise Tower	The check valve allows only unidirectional airflow into the cylinders. The cylinders rise as air is forced in.
On	On	Raise Tower	The check valves permit bi-directional flow. However, the source pressure typically exceeds the pressure inside the cylinders and air is forced in. This configuration is not used in practice.

TABLE 6: LOGIC CHART FOR VALVES

Earlier, the effects a fixed volume has on mitigating the changes in stiffness due to changes in volume were discussed. Though this volume can be considered separate for theoretical purposes, there is no reason to use two separate chambers in practice. Instead, a single longer cylinder can be used. Table 4 lists the changes in stiffness in a system with an 80mm bore and 1L volume chamber. If the cylinder is specified to operate between a height of 3" and 13", a 15" stroke is required to allow two inches of motion about all operating points. A cylinder of height 22.8" will suffice to consolidate a 1L-volume chamber and a 15" cylinder with a bore size of 80-mm.

The system outlined in this section provides the desired performance with a minimum of parts. A single elongated air cylinder provides both positioning and compliance. Four electrically controlled solenoid valves and two air-piloted check valves provide the necessary logic. An external PLC need only provide the appropriate signals to ensure that the system behaves as designed.

IV.5.D CONTROL

Pneumatic systems are notoriously difficult to control. The relationship between input pressure and output position is highly non-linear. Moreover, degradation of the component pieces can vastly alter the system's dynamics. A system that is well behaved one day may cease to respond correctly the next. Engineers employ various hardware techniques to circumvent these problems.

The relationship between pressure and position is difficult to characterize, but the relationship between air flow and position is much simpler to develop. For a system at constant pressure and temperature, increasing the amount of air in the system must increase the volume. Since the cylinders have constant surface area, any increase in volume must be entirely due to increases in height. The airflow is thus proportional to the rate of change of height. If the system dynamics can be massaged to control air flow instead of pressure, the final position will be far more controllable.

The solution presents itself in the form of a flow controller. If the pressure across a flow controller is above a certain value, it will clamp the air flow at some value. In essence, it functions like a non-linear saturation. Pressure and air flow are related through the normal fluid dynamics laws for low pressure, but air flow will saturate for sufficiently high air pressure. If the source pressure is maintained at a high value, air flow can then be regulated by controlling the saturation point of the flow controller.

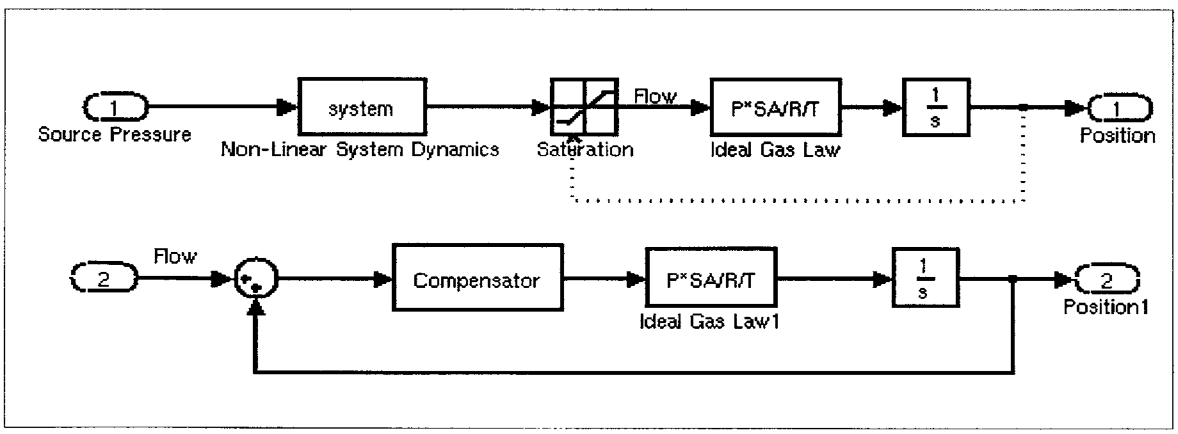


FIGURE 18: POSITION CONTROLLER FOR AIR CYLINDERS

As is directly evident from figure 18, this is a simple first order system with a single pole at the origin. Its dynamic behavior will be entirely described by the single closed loop pole location. Proportional gain will move the closed loop pole to an arbitrary location, thereby allowing the designer to customize the system's rise time and settle time.

IV.5.E Materials and Cost

This solution seems more promising if only because it can attain the appropriate performance at the desired cost. It consolidates compliant position control into one single actuator and replaces a multitude of parts in the existing assembly. A pneumatic system requires an arm casing, the cylinder and its valves, and some control hardware. Table 7 lists the Bill of Materials for a complete prototype of this system.

Item	Quantity	Cost per Item	Total
Air Cylinder	2	$306.90	$613.80
Valve Cables	4	$12.76	$51.04
3/2 Solenoid Valve	4	$46.20	$184.80
Check valve	2	$24.57	$49.14
Silencer	2	$2.44	$4.88
Swivel Flow Control	2	$11.32	$22.64
FRL	1	$84.30	$84.30
Tubing (clear)	10	$0.89	$8.90
Straight Connecter 1/8"	12	$1.41	$16.92
Straight Connecter ¼"	1	$1.46	$1.46
Linear Transducer	2	$150.00	$300.00
24 Volt/.3 A DC Power Supply	1	$62.00	$62.00
NO program button	1	$2.00	$2.00
On-Off-On rocker switch	2	$2.00	$4.00
rotary switch	1	$20.00	$20.00
PLC	1	$99.00	$99.00
Analog I /O option card	1	$149.00	$149.00
existing assembly	1	$2,641.06	$2,641.06
Components Removed from Existing Assembly	1	-$712.84	-$712.84
Total			$3,528.10
Difference with Current System			$887.04

Table 7: Bill of Materials for Pneumatic System

This table lists the individual parts, their quantities, and their costs for the prototype pneumatic system. The price will likely fall when large quantities of each item are purchased.

IV.5.F Conclusions

The analysis bears out the intuition air cylinders are an attractive option for actuation. The devices are compact, easily controllable, cheap, and fast. The cylinders position quickly and accurately. The compressibility of air provides ample compliance. Moreover, their compact and robust form makes them ideal for industrial applications.

IV.6 Comparison of Actuators

Four actuators have been considered, but air cylinders are clearly the best option. Air springs simply don't work. Active position and force control is very expensive and ineffective. Though the motion profile will be perfectly controllable, it cannot move quickly enough. Active position and passive force control with a motor and springs suffers similar dilemmas. It is only slightly cheaper, but it moves too slowly and is too stiff. Air cylinders are cheap, compact, and effective. They win in every category.

		Active Position/ Passive Compliance with a Motor		Active Position/ Compliance with a Motor		Air Spring		Air Cylinder	
		Comment	Grade	Comment	Grade	Comment	Grade	Comment	Grade
Weight	450lbs<Mt <900lbs		A		A		A		A
	Mc<400lb		A		A		A		A
User Interface	Same User Interface to be used for all actuators		A		A		A		A
Maint. Interface	Same Interface to be used for all actuators		A		A		A		A

		Active Position/ Passive Compliance with a Motor		Active Position/ Compliance with a Motor		Air Spring		Air Cylinder	
Performance Requirements	10" Travel	Requires Large Body	B	Requires Large Body	B	Not linear over 10"	D	Right on	A
	34.1"-43.2"		B		B	same	D		A
	+/- 2"		B		B		D		A
	K<45lbs/in	K = 66	F	arbitrary	A	K = 100	F	K ~ 25	A
	Accurate to +/- 1/8"		A		A		A		A
	Final Position in 8s	25 s	C	25 s	C	?	?	7.5 s	A
	Unbalanced Load		A		A		A		A
	1" Height DIfferential		A		A		A		A
Cost	$1,500.00	$1,869.20	D	$2,103.88	F	NA		$887.04	A
Safety	Maintain Height after electricity loss		A		A		A		A
	Maintain Height after air loss		NA		NA		A		A
	Maintain Height for 48 hours		A		A		A		A
	Maintain K after electricity loss		A		F		A		A
	Maintain K after air loss		NA		NA				
Final Score		**F**		**F**		**F**		**A**	

TABLE 8: COMPARISON OF ACTUATORS
Air Cylinders are clearly the best performers.

IV.7 Height Detectors

All of the actuation systems described above depend on position feedback. The complexity of the controllers that processes the signal vary, but all four are wholly dependent on 'knowing' the current height. In practice, a sensor of some sort is required to actually obtain the position information. And, as in the case of the actuators, a number of options exist. This section briefly describes some of the available technologies and compares their attributes.

String Potentiometers are the most economical option for determining an absolute measurement of the current height. Cross a tape measure and a potentiometer, and you have a string-pot. As the string is unwound, a wiper traverses the potentiometer's full range. A spring within the mechanism maintains tension on the string. As in the case of any linear potentiometer, the output voltage ranges between ground and the supply and is proportional to the fraction of the string that is extended. These devices have theoretically infinite resolution and are highly linear throughout their full range. A standard 20" device sold by Unimeasure Inc. runs for $150.

Linear potentiometers are the family sedans of the linear transducer market. A device with an 18" stroke and 1% accuracy is available for approximately $225. These devices are less compact and more robust than string potentiometers. A small truck rides atop a rod filled with resistive material. As the truck moves along the rod, it acts as a wiper moving along a standard potentiometer. The output is an analog voltage proportional to the distance the truck has traversed.

LVDTs capable of the same stroke length offer significantly improved accuracy. Their added performance comes with vastly increased size (the Hummer's of the market). The bodies of devices able to measure 20" of stroke are excessively long, nearly 60", and heavy, around 4.5 lb. They are more costly as well. In short, they are inefficient and expensive; not what is needed for this project.

Absolute encoders are the racecars of this industry. They are the most compact and the most accurate option available. Additional electronics are required to process the encoder's quadrature output though. The combination of sensor and hardware can run as much a $420, far more than a string potentiometer, and far more than is budgeted for the sensor.

Transducer Type	Resolution	Non-Linearity	Size	Cost
String Potentiometer	Infinite	0.5% full scale	small	$150
Linear Potentiometer	Infinite	0.15%	Large	$225
LVDT	Infinite	0.25%	Very large	$300
Absolute Encoder	Up to 65536 ticks per inch	Approaches 0%	Small	$420

TABLE 9: COMPARISON OF AVAILABLE LINEAR TRANSDUCERS

The budgetary constraints of this project make price a matter of paramount importance. The system's relatively low speeds and loose accuracy requirements permit using a relatively course measurement device. If the gross positioning mechanism must only be accurate to a quarter inch window, a string potentiometer will suffice perfectly for the purpose.

V Actual Prototype

V.1 Overview

Big picture of what the prototype looks like

Four options have been considered. As the analysis showed, air cylinders prove to be the best. They are compact, they require relatively few parts, and the entire system costs a fraction of the other options. It performs better, for less money. Selecting air cylinders for the physical prototype is the logical choice.

In this design, a PLC is used to separately control two air cylinders. The user interface, a set of buttons and knobs, commands the PLC to move the Tester. Check valves and solenoid valves route airflow in response to signals from the PLC, and move the Tester as desired.

This chapter will elucidate various implementation details for this design. The reader is encouraged to refer to section IV.5 for a complete discussion of the actuator. The subsequent sections will cover the actual PLC code used, the various wiring details, and the prototype's performance.

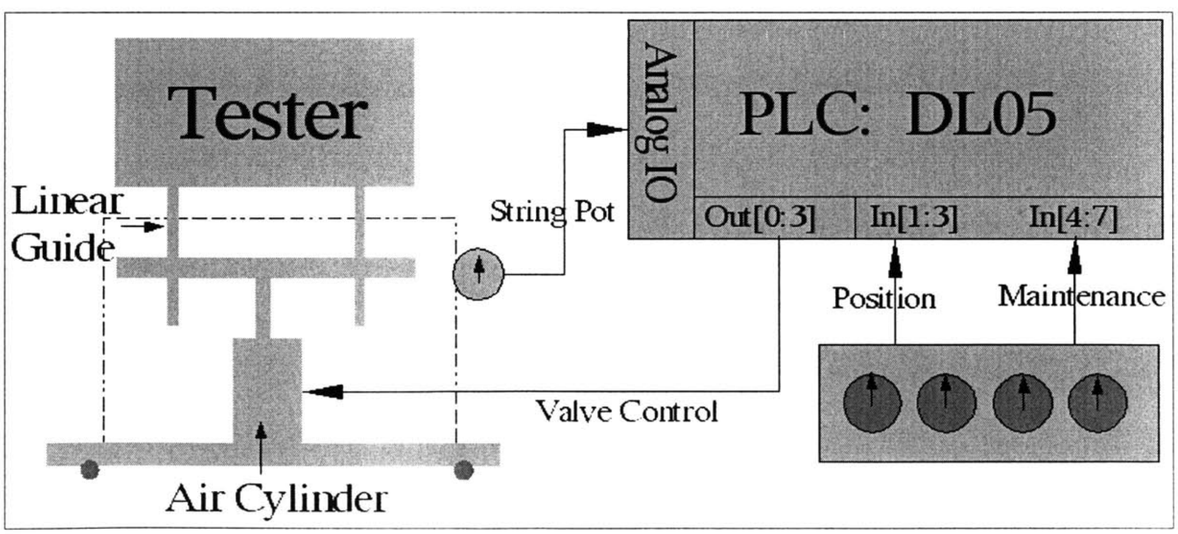


FIGURE 19: THE PROTOTYPE SYSTEM

The prototype includes a pneumatic cylinder as an actuator, a PLC as a controller, and a user interface.

V.2 Wiring

Controlling an automated system requires an electrical interface. Conveniently, the AutomationDirect DL05 PLC used in the prototype manages most of the difficult electronics. Nonetheless, the device's input and output ports need be connected to the various switches, valves, and power sources required to run the system.

The PLC's inputs and outputs relay the actions demanded by the user interface to the valves that permit air flow and actuation of the cylinders. These signals are broken by a dead man's switch, a normally open switch intended to prevent unintentional operation. Unless the operator has closed this switch, nothing will occur.

The PLC's inputs serve four separate functions. The lowest input recognizes whether or not the system is docked. The next three input bits code for a 3 bit binary number. The PLC decodes this value to recognize any of eight preprogrammed handlers and to load the appropriate information. Inputs X4, X5, and X6 permit the operator to manually adjust the height of either the right or left cylinder. Input X7 allows the operator to program the current height, as adjusted by X4:6, to the slot occupied by the handler coded by X1:3.

Input	Description
X0	Docked?: High indicates that the handler and Tester are Docked
X1	Rotary Switch Bit 0
X2	Rotary Switch Bit 1
X3	Rotary Switch Bit 2
X4	Right/Left: Toggles between right and left cylinder during manual operation
X5	Up?: If high, moves right/left cylinder up (depending on X4)
X6	Down?: If high, moves right/left cylinder down (depending on X4)
X7	Program: Used to program current height to of the handler coded by X1:X3

TABLE 10: PLC INPUTS

The PLC's output is far simpler. Each of the four solenoid valves pictured in figure 17 is attached to one of four outputs. If the PLC's logic triggers one of these outputs to change value, the attached valve will turn on or off accordingly.

Output	Description
Y0	Down Right: Toggles Valve Lower Right
Y1	Up Right: Toggles Valve Raise Right
Y2	Down Left: Toggles Valve Lower Left
Y3	Up Left: Toggles Valve Raise Left

TABLE 11: PLC OUTPUTS

In addition to PLC logic, the attached Analog-IO card receives a continuous analog signal from each of the two string potentiometers. The string potentiometers each have an internal resistance of 1k and are powered by the same 24 V source used to power the rest of the system. However, the analog IO card requires inputs to range between 0-10V. As such, a voltage divider circuit is used to reduce a 24V signal to 10V.

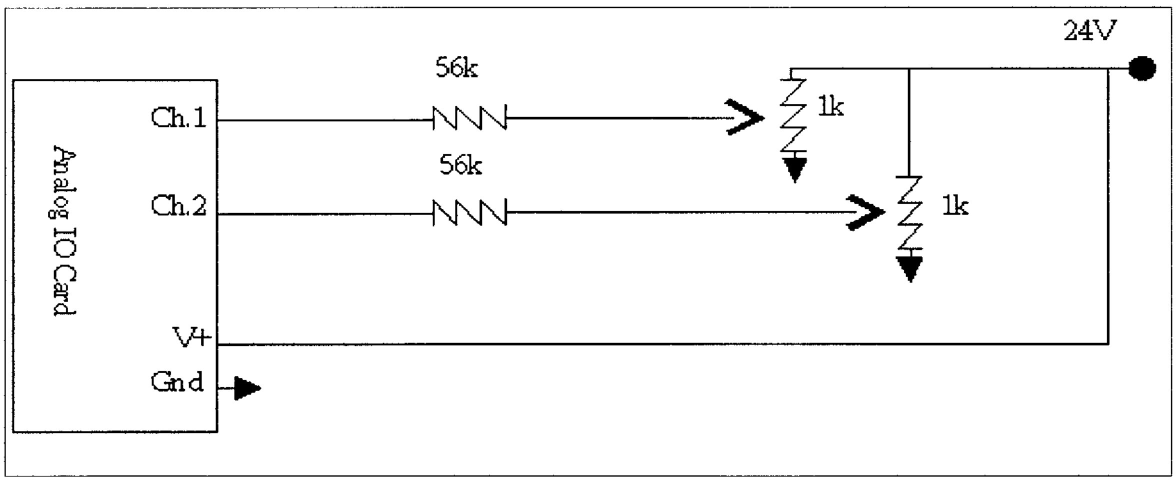


FIGURE 20: CIRCUIT SCHEMATIC OF ANALOG INPUT/OUTPUT CARD

Each of the two Input ports is preceded by a voltage divider to reduce the 24V analog signal to a range between 0-10V.

V.3 Control Scheme

Given the choice between an easy pragmatic solution or a complex but theoretically elegant solution, time and budget constraints create a preference for the first. Though section IV.5.D details a complete closed loop control scheme, it highlights various unmodeled difficulties that might hamper performance. Without the benefit of a physical device to verify and quantify these difficulties, determining the efficacy of the control scheme has been left in limbo. As the actual prototype has shown though, a much simpler model is far more robust.

The scheme used to position the Tester is far less complex than that described in chapter IV. As the design requirements state, the device's actual position must be within an eighth of an inch of the desired position. Once there, the system's compliance allows a full 2" of travel in either direction. These lax position requirements are easily met by the slow dynamics of the cylinders. Instead of closed loop PID control, a virtual comparator controls the entire cylinder. If the difference between the desired height and the cylinder's actual height is greater than an eighth of an inch, the cylinders are commanded to continue their motion (either down or up). Once the difference between the two values is less than an eighth of an inch, all airflow is ceased. A properly aligned cylinder will stop well within the allowable range. The performance details are discussed in subsequent sections. The control scheme is addressed here.

Modern PLC's offer the advantages of low cost, low power, and compact form, but they accomplish the same tasks as the decades old technology of relay logic. In place of actual relays, PLC's use a standard programming language called ladder logic to emulate the if-then statements that are the building blocks of relay logic. In lieu of replicating the full ladder logic program, Appendix III condenses the program into pseudo code with detailed explanations.

The control scheme detailed in the appendix can be reduced to a single comparator. There are subtleties permitting the PLC to be programmed and ensuring safe operation, but the actual motion control behaves much like a limit switch would behave. When the Tester approaches within a specified threshold of its final destination, all air pathways are closed and it glides to a stop. Experimental evidence has verified that it rarely glides beyond its allowed window, and if it does, it quickly returns. Even the most complex closed loop scheme permitted slight overshoot, so a simpler scheme can not be faulted for what appears to be identical performance.

V.4 Cost

Much to the designer's credit the Bill of Materials in chapter IV accurately predicted the cost of building a functional prototype. The analysis included the expected price of all pneumatic hardware, all electronics, and various odds and ends needed to complete the design. The final row of Table 7 is reprinted as Table 12 to refresh the reader's memory.

Total	$3,528.10

TABLE 12: TOTAL COST OF PNEUMATIC SYSTEM

Once in bulk order, the cost of this system will drop. Vendors frequently discount the price per item for bulk orders. Though no vendor has committed to a a final price structure, initial estimates reduce as much as $200 from the cost of building an entire system.

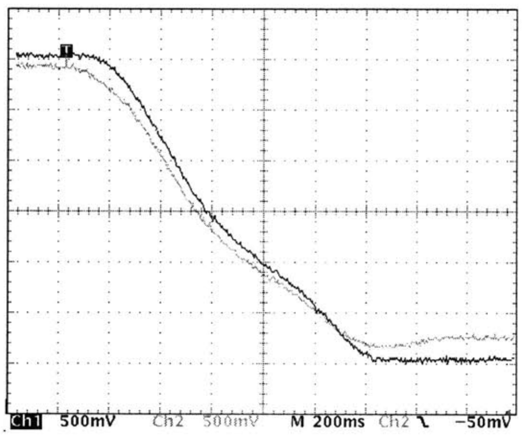
V.5 Experimental Results and Analysis

V.5.A INTRODUCTION

The theoretical development in chapter IV indicates that this system should meet the design requirements. The system's actual performance has been recorded to ascertain the validity of the hypothesis. Performance under a variety of source pressures is recorded. A statistical survey of each cylinder's ability to achieve its desired height is also included. Finally, measurement's of the system's stiffness at various heights has been recorded.

V.5.B STEP RESPONSES

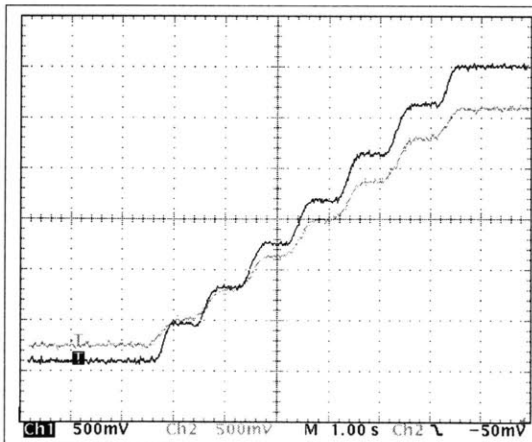
The cylinders' response to a step input of pressure verifies both the theoretical development presented earlier and the feared effects of unmodeled friction. The responses of the two arms illuminate the different response two otherwise identical systems can have in the presence of differing amounts of friction. As predicted, changing the source pressure can help override friction, and lowering the Tester is easier than raising it.



	Right Arm (Ch1)		Left Arm (Ch2)	
Quantity	Volts	Inches	Volts	Inches
Max	3.56	9.61	3.46	8.85
Min	0.50	3.49	0.64	3.21
High	3.54	2.49	3.42	8.77
Low	0.54	3.57	0.76	3.45
Amplitude	3.00	8.49	2.66	7.25
Pk-Pk	3.06	8.61	2.82	7.57
Fall Time	808 ms		766 ms	
Overshoot	1.3%		4.5%	

FIGURE 21: 'FALLING' RESPONSE OF THE TWO CYLINDERS WITH 73 PSI SOURCE PRESSURE

The cylinders begin at the top of their range of motion. At time t_o, air is allowed to exhaust to the atmosphere. Accordingly, the cylinders drop until airflow is stopped at t_f.



	Right Arm (Ch1)		Left Arm (Ch2)	
Quantity	Volts	Inches	Volts	Inches
Max				
Min				
High				
Low	0.58	3.65	0.74	3.41
Amplitude	2.92	8.33	2.34	6.61
Pk-Pk				
Fall Time				
Overshoot				

FIGURE 22: 'RISING' RESPONSE OF THE TWO CYLINDERS WITH 73 PSI SOURCE PRESSURE

The cylinders begin at the bottom of their range of motion. At time t_o, a source of 73 psi is connected to the cylinders' input. Because the pressure differential is insufficient to overcome the system's inherent friction, the two arms jump until the pressure source is disconnected at a later time.

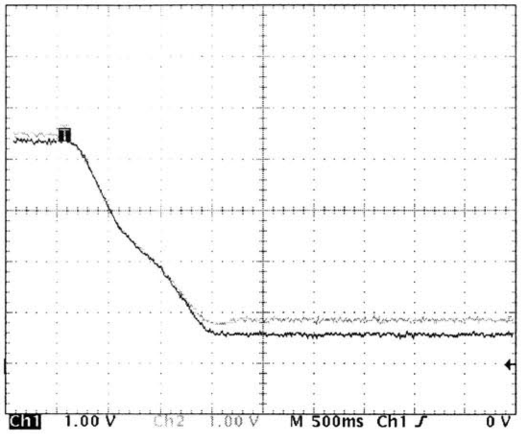
Figures 21 and 22 elucidate the differences between raising and lowering the Tester. In earlier work, the pressure required to balance a 600lb. Tester on these cylinders has been established to be 38.5 psi. Heavier Testers require more pressure. Therefore, the cylinders are responding to a pressure differential of 35.5 psi when the source pressure of 73 psi is administered. Likewise, they are subjected to a pressure differential of -38.5 psi when lowered, because the air is being exhausted to the atmosphere.

Given that these two values are nearly identical, it seems odd that friction should plague lifting motion but be barely discernible when lowering. The discrepancy is due to gravity. If motion were purely horizontal, only friction would resist actuation. The response would be identical for motion in either direction. It would be neither as smooth as 'falling' motion in figure 21 nor as punctuated as the rising motion in figure 22. However, gravity is present in vertical motion.

Consider three forces acting on the cylinder. The first, gravity, always acts downwards. The second, friction, always acts against the direction of motion. The third, a force due to pressure differential, will tend to raise the Tester should the external pressure exceed internal pressure and lower the Tester should the opposite be true. The effects of gravity are independent of direction of motion. The magnitude of friction is constant, but its direction will change to counter-act the direction of motion. Finally, since the pressure differentials are nearly equal but opposite, they will exert equal force but in opposing directions. When lifting, the force due to pressure must counteract both friction and gravity. When lowering, the force due to gravity works in concert with the pressure differential to overcome friction. Obviously, the net force on the cylinder is greater when falling. As such, the result is a smooth falling motion but a punctuated rising motion.

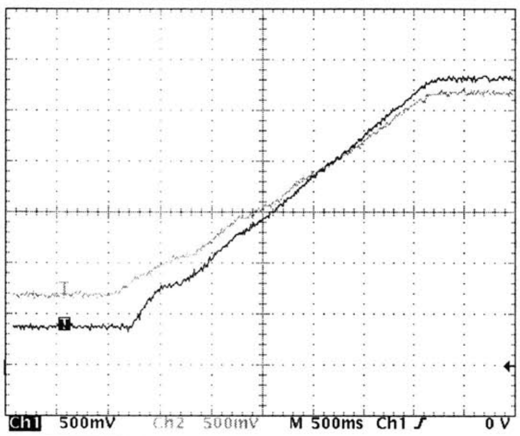
To further complicate matters, the friction in both arms is unequal. Figure 22 clearly shows both the right and left arms having difficulty rising; however, the right arm is clearly more jerky. It must wait for enough pressure to build up to overcome friction. Once the pressure hits this threshold, the arm jumps nearly 1/2 an inch, and stalls until the pressure can build up again. The left arm jumps as well, but to a lesser degree. This difference in friction is further manifested in figure 21. At the bottom of their motion, when the source of 73 psi is disconnected, the left arm bounces before settling. Like a highly damped spring, it overshoots before settling to a final position. The right arm is a clear demonstration of an over damped spring. Friction overpowers the air cylinder's compliance. It stops dead in its tracks.

If the cylinder has a more difficult time lifting the Tester than lowering it, a solution is to increase the source pressure. Though a cylinder will still need to overcome both friction and gravity to lift the Tester, the increased pressure differential will improve its overall performance. As shown by figure 24, using a source pressure of 90 psi. proves this assertion. Noticeably though, the falling response in figure 22 is practically identical to that in figure 21. When falling, the internal pressure exhausts to the atmosphere. Regardless of the source pressure used to lift it to a particular point, once at rest, the internal pressure will be exactly the 38.5 psi needed to balance the Tester's load.



	Right Arm (Ch1)		Left Arm (Ch2)	
Quantity	Volts	Inches	Volts	Inches
Max	4.44	11.37	4.5	10.93
Min	0.52	3.53	0.74	3.41
High	4.4	11.29	4.46	10.85
Low	0.6	3.69	0.82	3.57
Amplitude	3.8	10.09	3.64	9.21
Pk-Pk	3.92	10.33	3.76	9.45
Fall Time	1.04 s		1.02	
Overshoot	2.1%		2.2%	

FIGURE 23: 'FALLING' RESPONSE OF THE TWO CYLINDERS WITH 90 PSI SOURCE PRESSURE



	Right Arm (Ch1)		Left Arm (Ch2)	
Quantity	Volts	Inches	Volts	Inches
Max	2.84	8.17	2.65	7.23
Min	0.40	3.29	0.69	3.31
High	2.84	8.17	2.65	7.23
Low	0.40	3.29	0.69	3.31
Amplitude	2.44	7.37	1.96	5.85
Pk-Pk	2.44	7.37	1.96	5.85
Fall Time	2.5 s		2.4 s	
Overshoot	0.80%		3.10%	

FIGURE 24: 'RISING' RESPONSE OF THE TWO CYLINDERS WITH 90 PSI SOURCE PRESSURE

The system's response to 90 psi gives a much clearer picture of its dynamics when friction is overcome. Both the rising and falling responses exhibit a ramp response to a step input. That is to say, when the cylinder's are suddenly allowed to exhaust to the atmosphere, their height falls proportionally to time. Similarly, with the exception of the first few seconds, the cylinders ramp up to their final position when a source of 90 psi is connected to the cylinders. The first order dynamics predicted by the theoretical development in Chapter IV is vindicated by these results.

On another note, the system performs well within it specified speed requirements. At a source pressure of 90 psi, the cylinders are able to lift the Tester 5" in 3 seconds or 1.6" per second. It falls much quicker, dropping 4 inches in 1.5 seconds, or 2.66 inches per second. Even the jerky lifting action at 73 psi achieves its final position within the allowed time, if not faster.

V.5.c Accuracy

The step responses discussed above did not address any accuracy concerns. The analysis underscored the effects source pressure can have on lifting the Tester, and the adverse effects of friction in the presence of a low pressure differential. The results are an unacceptable jerkiness in the Tester's motion. As is made evident by figure 21, differing amounts of friction can exacerbate the problem of the jerkiness. Too much friction in the presence of too little pressure could result in the Tester missing its 1/4 inch target window.

The results bear this reality out. At low pressure, friction plays a more significant role in the system dynamics. Moreover, the frictional forces affecting the right arm are far more degenerative than those afflicting the left arm, as figures 21-24 illustrate. It is not surprising then, that while rising, the cylinders miss their target by a greater degree at low pressures, and that the right cylinder is the worst of the two.

The cylinder's are tested at two pressure settings: 73 PSI and 90 PSI. Testing involves commanding the cylinder to attain a certain height and then measuring the difference between the final and desired heights. Since the control algorithm permits a 1/4" window centered on the desired height, a valid final position will lie 1/8" above or below the desired position. Consequently, the mean of these differences is not a good reflection of the system's accuracy. If the cylinder routinely misses its target but in opposite directions, the mean difference will be near zero. Instead, the standard deviation or mean of the squared differentials is far more illustrative. Not surprisingly, this measure indicates that the cylinders are least accurate (when rising) at low pressures with the right worse than the left. At 90 PSI, the system performance improves markedly.

	Right Cylinder	Left Cylinder
σ at 73 PSI (in.)	0.071	0.063
σ at 90 PSI (in.)	0.062	0.052

TABLE 13: CYLINDER ACCURACY AT VARYING PRESSURES

High friction and low pressure combine to make for inaccurate performance. Conversely, low friction and high pressure allow for accurate positioning.

The analysis presented earlier also indicates that a system should fall faster than it rises due to the added effects of gravity. Figures 21-24 also graphically detail this point. fall times are significantly less than rise times. As a result, a command to lower a cylinder usually results in the cylinder undershooting its desired position and then rising again. Consequently, there is very little difference in accuracy between rising and falling motions.

V.5.D STIFFNESS

The point has been belabored, but vertical motion must be compliant to allow for fine adjustment of the Tester's height. The specifications require a stiffness at or less than 45 lbs./in throughout the range of motion. Table 13 indicates that the air cylinders should theoretically surpass this mark at every point within their range of motion. Experimental evidence verifies this point to great accuracy.

73 PSI			
Right Arm (Ch1)		Left Arm (Ch2)	
Height	Measured Stiffness (lb/in)	Height	Measured Stiffness (lb/in)
9.85	18.64	9.44	18.68
7.25	20.98	7.04	20.60
3.72	27.10	4.44	26.69
90 PSI			
Right Arm (Ch1)		Left Arm (Ch2)	
Height	Measured Stiffness (lb/in)	Height	Measured Stiffness (lb/in)
9.99	19.36	9.96	18.33
7.01	21.43	6.72	20.81
3.58	28.45	4.08	28.62

TABLE 14: STIFFNESS OF ACTUAL AIR CYLINDERS

As expected, the right arm proves to be stiffer than the left arm. The step responses discussed earlier indicated that the right arm has far more unmodelled friction than the left arm. As a result, the left arm is more compliant at equal heights.

Moreover, source pressure plays a part too. At a given height, the cylinders are more compliant at 73 PSI than they are at 90 PSI. The measurements for the right arm most clearly illustrate this fact.

V.5.E CONCLUSIONS

With some exceptions, the prototype system behaves as expected. Friction and misalignment adversely affected performance, but their effects are easily quantified. Future work will account for these two conditions and remove them.

The system's highlights include:

- It moves fast enough at 1.33 in/sec.
- It's cheap, adding only $887 to the cost of the existing system.
- It's compliant. The system's stiffness surpasses the 45 lbs./in spec.

- It's easily controllable. The complicated closed loop algorithms developed in earlier sections are supplanted by a much simpler technique in practice.
- It's built with a minimum of parts.

In all categories, this design meets its requirements. Though far from a production model, if Teradyne actually intends to implement an updated Vertical Motion system, this is the optimal product.

VI Concluding Remarks

VI.1 Design Problem

Teradyne retains its market share because it has established itself as a front-runner in technological innovation. Moreover, it has realized that equally as important as a technology's capability is its usability. This thesis has addressed on of Teradyne's most pressing usability issues.

The J-750 continues to find new niches. However, competition has made inroads into the J-750's market share. In spite of its power and technological superiority, customers have found it to be unwieldy. A manipulator that simplifies moving and docking the J-750 would vastly increase the J-750's marketability.

Chapter III outlines the specifications required of an upgraded system. A successful design would meet a slew of speed requirements, accuracy requirements, and compliance requirements. The goal is merge position and force control to provide accurate height adjustment and compliance at the nominal operating point.

VI.2 Solution and Results

Various possibilities for meeting the requirements are discussed in Chapter IV, constructing the best of these options is discussed in Chapter V. As the reader will recollect, pneumatic cylinders prove to be the single best actuator available. A PLC based control scheme that merges digital logic and rudimentary analog position regulation, controls the full system.

The experimental results, gleaned from testing a prototype system, verify the theoretical performance predicted by the model in Chapter IV. At high pressures, the tester achieves its final height within the allotted time. Its accuracy is acceptable and its motion profile is continuous and safe. Both closely match the theoretical predictions.

VI.3 Future Work

In spite of its success to date, the existing assembly is only a prototype. Customers are eager to see its completed form, but no one would purchase it in its current state. A number of details need to be addressed before Teradyne can begin to ship the cart as designed.

From a performance perspective, actuation could be improved on two fronts. For the sake of convenience, the reservoirs are implemented as separate volume chambers; that is what Festo had in stock. To consolidate space, remove extra parts, and avoid air leakage, the final product will need a single elongated cylinder. On another note, the prototype is built with existing linear bearings that rely on a low friction plastic to make contact. For minimal increase in cost but a great improvement in functionality, these bearings could be replaced with standard roller bearings. Any improvement in the coefficient of friction would be a boon to the actuator's dynamics.

These slight performance adjustments aside, the prototype needs to be taken through the process of creating a production model. This includes rigorous documentation, certification of various safety standards, and repackaging into a sturdy unbreakable form. The prototype is well on its way toward accomplishing these goals, but it requires an experienced hand in taking a product to market to apply the finishing touches.

VI.4 Additional Applications

Though this document has concerned itself only with addressing the needs of Teradyne's Blackstone project, compliant position control is a useful tool. The principles discussed here, apply equally well to designing active shock absorbers in cars or bio-mimetics. Few real world systems can handle, or produce, infinite stiffness. Combining position and forces control in manipulating a heavy and irregular load is a technique that will find broad applications beyond Teradyne's walls.

Appendix I: Complete Bill of Materials for Existing System

Description	Quantity	Cost	Total
Weldment Side	2.00	335.00	670.00
Shaft, Support Spring	4.00	57.00	228.00
Shft.Linearrace, Stl 1 00x30L	4.00		0.00
Block, Pillow	2.00	75.00	150.00
Retaining Rings, 2 5/8 00 shaft	4.00	1.38	5.52
SQ Flanged Bearing	8.00	52.77	422.16
Bearing Spherical, 1 3/16 bore	2.00	62.22	124.44
SCR, Shoulder, Socket, 10-32,5	16.00	0.00	0.02
Wahser, split lock 3/8	4.00	0.00	0.00
Screw, 3/8-16 x 1.50 lg.SHCS	4.00	0.00	0.00
Channel, Lift Plate, Spring	2.00	125.00	250.00
Bushing, Flg,1 ID, 1.25 OD, .62L	4.00	1.94	7.76
Collar, 2pc, Slit, .7/8 bore	4.00	4.13	16.52
Swivel Pad, Flat, SST, 3/4-10	2.00	12.10	24.20
Loctite #272 High Strength	0.00		0.00
Ret Ring, External 0.750" Shaft	2.00	0.40	0.80
Slotted Plug	2.00	0.00	0.00
Collar, 2pc, Split Hub 1"Bore	4.00	4.06	16.24
3/4-10 Hex Nut	2.00	0.24	0.48
1" ID Iglide Bearing	4.00	2.83	11.32
Screw, 10-32 x 3/8 lg Flanged Butt	4.00	0.19	0.76
Plate Compression, Spring	2.00	125.00	250.00
Spring, Compression, 2.812 od, .283 wd	4.00	37.31	149.24
Plate, Spring Capture	4.00	19.95	79.80
Thd Rod, Upper Stop, 3/4-10	2.00	22.50	45.00
!"-5 Acme Hex Nut	2.00	15.40	30.80
Bearing Retainer	2.00	50.00	100.00
Lower Adjustment Screw	2.00	29.00	58.00
Worm,10DP,Quad Thd w/ 3/16 Ke	2	54.6	109.2
Handwheel w/ foldaway,.50bore	2	49.77	99.54
Bearing,Sin Row,15mm,35mm,11m	6	8.75	52.5
Handle, Pull, 5/16-18 thd	4	6.66	26.64
Worm Gear,10DP,Quad w/3/16 Ke	2	6	12
Thrust Ball Brng,.75ID,.545 w	2	2.6	5.2
Key,1/8 sq x .75 lg	2	0.571	1.142
Key,3/16 sq x .75 lg	4	0.476	1.904
Bearing Support, Worm	2	84	168
Upper Bearing Support	2	54	108
Shaft, Worm Handle	2	29	58
Total Cost			$3,283.19
Total Cost per Arm			$1,641.59
Total Cost without Worm Drive			$2,641.06
Total Cost wo/WD per arm			$1,320.53

Appendix II: Complete Bill of Materials for Active Positioning, Passive Compliance with a Motor and Springs

Item	Vendor	Quantity	Cost per Item	Total
42AE - DC inline Gear Motor, Model 4182	Bodine-Electric	2	247.32	$494.64
encoder 0941	Bodine-Electric	2	78.28	$156.56
power amplifier cards	Bodine-Electric			$0.00
0888 - Analog Control		2	$68.80	$137.60
0890 - Remote Direction Control		2	$74.40	$148.80
0850 - Driver		2	$132.80	$265.60
2 in 2 out analog IO card	automation direct	1	$179.00	$179.00
PLC	automation direct	1	$99.00	$99.00
Linear Transducer	Unimeasure, Inc.	2	$150.00	$300.00
24 Volt DC Power Supply	mcmaster-carr	1	$62.00	$62.00
NO program button	digikey	1	$2.00	$2.00
On-Off-On rocker switch	digikey	2	$2.00	$4.00
rotary switch	Grayhill, Inc.	1	$20.00	$20.00
existing assemlby	Teradyne	1	2641.06	$2,641.06
Total				$4,510.26
per arm				$2,255.13

Appendix III: PLC Code

Code Stage

Comments and Explanations

Startup

On the First Clock Cycle:

- Inform the PLC that an Analog IO card with two input and two output channels is present.
- Store the heights of 8 handlers in memory

This logic initializes the PLC. It makes it aware of its Analog inputs and initializes the PLC's memory. The PLC will now respond to fluctuations on its analog inputs and will know the heights of eight handlers.

Current Height

On Every Clock Cycle:

- Determine the heights of the cylinders by reading the values on the analog input channels.
- Store these heights in memory

The PLC cycles every 10ms or so. On each cycle, it is to refresh its knowledge of the cylinders' current height to account for any motion.

Desired handler

On Every Clock Cycle:

- read input bits X1, X2, and X3
- decode 3 bit binary number to determine the desired handler

As the wiring section discussed, bit X[1:3] code for the desired handler. This code is deciphered for the PLC to know which handler to use.

Program the PLC

On Every Clock Cycle:

If the input X7, connected to the program button, is true

- Store the current height at the memory location associated with the desired handler.
- Toggle the Boolean variable 'program' to true.

The control scheme allows a trained user to program the PLC. When the program button is engaged, the PLC will store the current height in the memory location associated with the current handler and overwrite the value written during initialization.

Desired Height

On Every Clock Cycle:

If the PLC is not being programmed

- Load the height of the desired handler from the appropriate memory location.
- Add an $1/8^{th}$ of an inch and store this in a variable called 'Desired Height +'
- Subtract and $1/8^{th}$ of an inch and store this in a variable called 'Desired Height -'

The design specifications allow for a 1/4 inch window. This logic creates two variables a quarter of an inch apart centered about the desired height.

Difference in Height

On Every Cycle:

- calculate the difference in height between the right and left cylinders
- store this in a variable called 'Delta Height'

As a safety precaution, the difference in height between the two cylinders is prevented from exceeding an inch. A titled Tester is not comforting. This logic calculates the difference in height and stores it in a variable

Safe to Move?

On Every Cycle:

- if the right side is higher than the left side by more than one inch, make the Boolean variable 'Right too High' true. Otherwise, false.
- If the left side is higher than the rights side by more than one inch, make the Boolean variable 'Left too High' true. Otherwise, false.

The code here processes the 'Delta Height' variable to determine if it is safe to move the system. If 'Delta Height' indicates that one side is more than an inch higher than the other side, the appropriate Boolean variable is toggled.

Maintenance Panel

On Every Cycle:

- if right and up are engaged, toggle to Boolean variable 'right up' to true.
- If right and down -> 'right down'
- If left and up -> 'left up'
- If left and down -> 'left down'
- If any of these buttons are pressed, the toggle the Boolean variable 'maintenance' to true.

The program button allows a trained operator to program the current height. The maintenance panel allows the operator to adjust the height to some position other than the 8 heights associated with the 8 handlers. Once at a new position, the operator can use the program button.

This code does not actuate the cylinders so much as set variables that might call for actuation.

Motion

On Every Cycle:

1) If

- *[(The right cylinder is higher than the 'Desired Height +' and 'Maintenance' is false) OR (the maintenance panel has engaged 'right down')]*
- *AND 'Left too High' is false*
- *AND 'program' is false*

-> toggle output Y0 = 'down right' to on.

2) If

- *[(The right cylinder is lower than 'Desired Height -' and 'Maintenance' is false) OR (the maintenance panel has engaged 'right up')]*
- *AND 'Right too High' is false*
- *AND 'program' is false*

-> toggle output Y1 = 'up right' to on.

3) If

- *[(The left cylinder is higher than 'Desired Height +' and 'Maintenance' is false) OR (the maintenance panel has engaged 'left down')]*
- *AND 'Right too High' is false*
- *AND 'program' is false*

-> toggle output Y2 = 'down left' to on.

4) If

- *[(The left cylinder is lower than 'Desired Height -' and 'Maintenance' is false) OR (the maintenance panel has engaged 'left up')]*
- *AND 'Left too High' is false*
- *AND 'program' is false*

-> toggle output Y3 = 'up left' to on.

All the variables discussed in previous sections coalesce here to determine whether or not the cylinders will move.

As the code shows, three conditions must be met to allow motion. Starting with the final two conditions: if the difference in height is less than one inch and if the PLC is not being programmed, motion may proceed.

The first condition determines the source of the motion command. It differentiates between commands from the maintenance panel and those due to standard commands looking to move the Tester to a new height. If the maintenance panel is not engaged, then motion proceeds if the cylinders' actual height exceeds the limits set by 'Desired Height + or -'. If the maintenance panel is engaged, any motion commanded by altering the desired height is ignored and the instructions given be the maintenance panel are followed.

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