Low-Cost, Highly-Damped, Precision Linear Motion Using Porous Carbon Air Bearings and Epoxy Replication

by

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AUG 01 1994
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MICHAEL ALEC CHIU

Submitted to the Department of Mechanical Engineering on May 17, 1994 in partial fulfillment of the requirements for the Degree of Master of Science in Mechanical Engineering

ABSTRACT

This thesis presents a design and manufacturing method for producing high precision linear air bearing systems. This 'vacuum-replication' method reduces the need for expensive precision manufacturing and assembly methods. This method was used to produce bearings with zero initial clearance which yields motion of higher performance than that of conventional mechanical and aerostatic bearings, including improvements in stiffness, straightness of motion, damping and friction.

The thesis is presented in the form of a case study of the redesign of a high speed linear actuator used in the turning of non-axisymmetric parts. The original system used rolling element bearings and had several regions that require optimization. A low cost, high performance air bearing system was designed to rectify these problems and manufactured using the methods described above. A prototype system was developed with porous carbon air bearings and assembled using a novel vacuum-replication method. Laboratory tests of stiffness, straightness and dynamic response demonstrate that the air bearing system is superior to the existing design. The bearing system and data are analyzed and a design model is created to allow this technology to be reproduced in other systems. Using these results, a retrofit system with lower cost and complexity was designed to replace the existing actuator and was implemented in a production machine tool.

Thesis Supervisor: Dr. Alexander Slocum
Title: Alex and Britt d'Arbeloff Associate Professor of Mechanical Engineering
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# Table of Contents

Abstract ......................................................................................................................... 2  
Acknowledgments ........................................................................................................ 3  
Table of Contents ......................................................................................................... 4  
1 Description of Design Problem ............................................................................... 6  
  1.1 Machine Requirements ..................................................................................... 8  
  1.2 Existing Machine ............................................................................................. 10  
  1.3 Goals of Redesign ............................................................................................ 13  
  1.4 Existing Design Problems ................................................................................ 14  
2 Precision Components and Techniques ................................................................... 21  
  2.1 Air Bearings .................................................................................................... 21  
    2.1.1 General Theory .......................................................................................... 23  
    2.1.2 Porous Bearings ....................................................................................... 27  
  2.2 Potting and Replication in Manufacturing ....................................................... 28  
3 Prototype Design ..................................................................................................... 30  
  3.1 Initial Design options ....................................................................................... 30  
    3.1.1 Linear Bearings ......................................................................................... 30  
    3.1.2 Tool carriage ............................................................................................. 32  
    3.1.3 Linear Motors ............................................................................................ 36  
    3.1.4 Sensor Systems ......................................................................................... 37  
  3.2 Methods and Techniques .................................................................................... 39  
    3.2.1 Vacuum Assembly Method ....................................................................... 41  
    3.2.2 Zero Clearance ......................................................................................... 42  
    3.2.3 Bearing Assembly ..................................................................................... 43  
  3.3 Test Systems ...................................................................................................... 48  
    3.3.1 Devitt System ............................................................................................ 48  
    3.3.2 Ceramic System ......................................................................................... 50  
    3.3.3 Prototype System ...................................................................................... 52  
    3.3.4 36” Bearing Guideway ............................................................................. 54  
4 Analysis ................................................................................................................... 55  
  4.1 Stiffness ............................................................................................................ 55  
    4.1.1 Structural Stiffness .................................................................................... 56  
    4.1.2 Bearing Stiffness ....................................................................................... 57  
    4.1.3 System Stiffness ....................................................................................... 57
<table>
<thead>
<tr>
<th>Section</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.2</td>
<td>Straightness</td>
<td>59</td>
</tr>
<tr>
<td>4.3</td>
<td>Dynamic Response (Damping)</td>
<td>60</td>
</tr>
<tr>
<td>4.4</td>
<td>Friction</td>
<td>60</td>
</tr>
<tr>
<td>5</td>
<td>Testing</td>
<td>62</td>
</tr>
<tr>
<td>5.1</td>
<td>Static Stiffness</td>
<td>62</td>
</tr>
<tr>
<td>5.2</td>
<td>Straightness</td>
<td>69</td>
</tr>
<tr>
<td>5.3</td>
<td>Dynamic Response</td>
<td>72</td>
</tr>
<tr>
<td>5.4</td>
<td>Friction</td>
<td>76</td>
</tr>
<tr>
<td>6</td>
<td>Final Design</td>
<td>77</td>
</tr>
<tr>
<td>6.1</td>
<td>Part Modifications</td>
<td>79</td>
</tr>
<tr>
<td>6.2</td>
<td>New Parts</td>
<td>79</td>
</tr>
<tr>
<td>6.3</td>
<td>Jigs and Fixtures</td>
<td>80</td>
</tr>
<tr>
<td>6.4</td>
<td>Assembly Drawings and Instructions</td>
<td>81</td>
</tr>
<tr>
<td>7</td>
<td>Future Work</td>
<td>83</td>
</tr>
<tr>
<td>8</td>
<td>Conclusions</td>
<td>85</td>
</tr>
<tr>
<td>A1</td>
<td>Linear Motors</td>
<td>86</td>
</tr>
<tr>
<td>A1.1</td>
<td>Linear motor design issues</td>
<td>87</td>
</tr>
<tr>
<td></td>
<td>Range of motion</td>
<td>87</td>
</tr>
<tr>
<td></td>
<td>Accuracy, repeatability and resolution of motion</td>
<td>87</td>
</tr>
<tr>
<td></td>
<td>Straightness of motion</td>
<td>88</td>
</tr>
<tr>
<td></td>
<td>Performance</td>
<td>88</td>
</tr>
<tr>
<td>A1.2</td>
<td>Description of linear motor types</td>
<td>89</td>
</tr>
<tr>
<td></td>
<td>Continuous range motors</td>
<td>90</td>
</tr>
<tr>
<td></td>
<td>Linear induction motor</td>
<td>90</td>
</tr>
<tr>
<td></td>
<td>Linear synchronous motors</td>
<td>91</td>
</tr>
<tr>
<td></td>
<td>Intermediate range motors</td>
<td>92</td>
</tr>
<tr>
<td></td>
<td>DC homopolar motors</td>
<td>92</td>
</tr>
<tr>
<td></td>
<td>DC motors</td>
<td>96</td>
</tr>
<tr>
<td></td>
<td>Short Range Motors: PZT actuators</td>
<td>99</td>
</tr>
<tr>
<td></td>
<td>Piezoelectric effect</td>
<td>100</td>
</tr>
<tr>
<td></td>
<td>Magnetostrictive effect</td>
<td>103</td>
</tr>
<tr>
<td>A1.3</td>
<td>Selection of actuator for fast tool servo</td>
<td>104</td>
</tr>
<tr>
<td>A2</td>
<td>Final Design Drawings</td>
<td>106</td>
</tr>
<tr>
<td>Bibliography</td>
<td></td>
<td>129</td>
</tr>
</tbody>
</table>
1 Description of Design Problem

This thesis focuses on the development of a linear tool servo used to actuate a cutting tool used in the turning of non-axisymmetric parts.

![Diagram](image)

**Figure 1.1 Non-axisymmetric Turning Operation**

Non cylindrical parts can be turned on a lathe by coordinating the angular position of the spindle with the radial motion of the cutting tool, as shown in Figure 1.1. This is accomplished using a linear servo mechanism to actuate the cutting tool. The achievable part complexity and accuracy of the radial features depends on the spindle speed and the bandwidth of the linear tool motion. The application under investigation required parts with 0.05 to 2 mm (0.01 - 0.04 ") ovality and accuracies of approximately 1-5 \( \mu m \) (40-200 \( \mu \text{in} \)). This precision is beyond the realm of most production lathes, which can typically achieve 5 \( \mu m \) (200 \( \mu \text{in} \)) accuracy and generally cannot coordinate spindle and tool position owing to the large mass and low bandwidth of the typical cross slide.
Specialty machines do exist that are capable of producing parts of this precision. However, they are not capable of the sustained throughput required for high volume production applications. Compared to precision machines for diamond turning or lithography, the relatively large cutting forces and moderate accuracy requirements place the required operations outside the realm of most of these machines. The use of high speed linear servo technology as applied to non-axisymmetric turning lies between the realm of precision instruments and production machine tools.

This application of this technology is not limited to the simple case of non-axisymmetric turning. High speed and high precision linear actuators have a wide range of applications including high resolution printers, precision x-y tables and lithography equipment. They are also an integral part of test equipment such as precision shakers use for modal analysis and accelerometer calibration.
1.1 Machine Requirements

The requirements of the linear actuator are determined by the part profile, accuracy requirements and spindle speed. Figure 1.2 outlines the specific requirements for a typical aluminum part, cutting forces for a steel part are up to 20 times larger.

<table>
<thead>
<tr>
<th>Requirement</th>
<th>Value</th>
<th>Unit</th>
</tr>
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<tbody>
<tr>
<td>Spindle Speed</td>
<td>2000 (204)</td>
<td>RPM (radians/sec)</td>
</tr>
<tr>
<td>Actuator Bandwidth</td>
<td>70</td>
<td>Hz</td>
</tr>
<tr>
<td>Feed Force</td>
<td>50</td>
<td>N</td>
</tr>
<tr>
<td>Tangential Force</td>
<td>150</td>
<td>N</td>
</tr>
<tr>
<td>Radial Force</td>
<td>65</td>
<td>N</td>
</tr>
<tr>
<td>Amplitude of Motion</td>
<td>0.5</td>
<td>mm</td>
</tr>
<tr>
<td>Total Stroke</td>
<td>25</td>
<td>mm</td>
</tr>
<tr>
<td>Accuracy</td>
<td>5</td>
<td>µm</td>
</tr>
<tr>
<td>Resolution</td>
<td>0.5</td>
<td>µm</td>
</tr>
</tbody>
</table>

**Figure 1.2 Typical Actuator Requirements for Cutting Aluminum**

The parts are finish turned from a circular profile in two passes of the tool. The first pass is a rough cut which generates the non-circular profile and the return pass produces a surface finish with a continuous helical groove. The spindle speed of 2000-3000 rpm is typical of an aluminum-turning operation and corresponds to a throughput time of approximately 20 seconds. Higher speeds and throughput are limited by the bandwidth of the tool servo, whose motion must be coordinated with the position and speed of the spindle. More complex geometries can be cut with this method if the spindle speed is reduced or the linear tool servo bandwidth is increased. The stiffness, load capacity and power of the linear tool servo system are determined by three force components; the
tangential and feed forces must be supported by the actuator bearing system and the radial force is supplied by the actuator. The bearings for the system must be able to support the 150 N tangential force plus any additional dynamic loads.

The amplitude of motion is the distance that the tool must travel to generate the non-circular profile during cutting. The total stroke of the tool is much greater. It includes the amplitude, as well as additional travel to adjust for different part sizes, clearance for loading and compensation for tool wear. The accuracy of the system is the tolerance of the final part, and the resolution is the smallest increment that the servo can make. Resolution is important in generating accurate profiles and adjusting for tool wear. In this case, the resolution was determined by the logical resolution of the optical encoder used for position feedback.

From the specification outlined above, the motion requirements of the tool holder can be calculated as follows:

\[
\begin{align*}
\text{position:} & \quad x = A \sin(\omega t) & 0.5 \times 10^{-3} \text{ m} \\
\text{velocity:} & \quad v = \frac{dx}{dt} = A \omega \sin(\omega t) & 0.2 \text{ m/sec} \\
\text{acceleration:} & \quad a = \frac{dv}{dt} = A \omega^2 \sin(\omega t) & 88 \text{ m/sec}^2
\end{align*}
\]

where \(\omega\) is twice the spindle speed (if the part has two lobes)

\(A\) is the amplitude of motion.

The linear actuator has moderate linear velocity and acceleration requirements (10 g). The high velocity, coupled with bearing friction, can consume power and generate a significant amount of heat in the bearings. The dynamic forces due to high acceleration can be reduced by minimizing the mass of the moving components.
1.2 Existing Machine

![Diagram of a common lathe](image)

**Figure 1.3 Schematic of a Common Lathe**

The existing machine is similar to a common engine lathe, as shown in Figure 1.3. It has a spindle, tailstock, bed and cross-slide and can be configured with a horizontal or vertical spindle. The only significant difference between this machine and a standard lathe is the addition of the linear tool servo, which is mounted orthogonal to the axis of rotation of the spindle. The tool fast servo functions as the cross slide on a standard lathe, it automates the radial motion of the tool relative to the part.

The existing linear tool servo consists of a toolholder supported by an arrangement of bearings and actuated by a linear motor. The entire system is enclosed in a cast iron housing which also contains feedback sensors and control electronics.

---

1 Picture from Serope Kalyakjian, *Manufacturing Engineering and Technology*
Accurate linear motion of the cutting tool is achieved with the use of a slender tool holder supported by a quasi-kinematic precision roller bearing arrangement, as shown in Figure 1.4. The tool holder is made of 6061-T6 aluminum which has been ground flat and parallel on four sides. Aluminum was used to reduce the inertial mass of the system, the finished part was heat treated and hard anodized to increase its strength. Pairs of precision roller bearings running on a hardened surfaces were used to constrain the motion of the tool holder. These bearings are intended to deterministically support the tool holder: the two pair of front bearings are allowed to pivot and the upper pairs of bearings float on a leaf spring to provide a preload force. This arrangement is repeated on the sides of the tool holder and the result is a quasi-kinematic constraint of five degrees of freedom.

The linear motor used in the existing tool servo was a moving voice coil design. The stator is constructed of a ring of magnets and an iron core. There is a 0.75 mm nominal air gap between the stator magnets and the moving coil. Velocity feedback information used in the control of the motor is measured with a linear velocity transducer, mounted in the center of the voice coil. Position information is obtained with a 50 line/mm glass scale mounted near the tip of the tool holder. The optical encoder system has a resolution of 0.5 micron using a 4X quadrature decoding and 10X electronic interpolation.
Figure 1.5 Schematic of the Existing System in its Housing

The entire package, including bearings, motor and electronics is enclosed in a cast iron housing as shown in Figure 1.5. A flexible diaphragm seal permits the tool carriage to move while protecting the bearings from the stream of high pressure coolant that floods the tool and workplace. Low-pressure air is forced through the housing to remove excess heat generated in the linear motor. The entire package is somewhat larger than a shoe box and can be easily mounted in any position, allowing flexibility in machine configuration.
1.3 Goals of Redesign

The overall goal is to produce a cost-effective actuator that has increased precision of motion and increased reliability. In addition to accuracy and cost, other considerations such as durability, efficiency and ease of implementation were also considered.

The performance of any precision machine can be evaluated by examining three qualities; accuracy, repeatability and precision. Accuracy is the maximum error between any two points in a machine tool’s work volume, which equates to the uncertainty in the dimension of any part made. Poor accuracy is usually caused by repeatable phenomenon within the machine and are sometimes referred to as systematic errors. If the source of these repeatable errors can be found and modeled, the accuracy can be improved with feedback compensation. Repeatability is the error between a number of successive attempts to move a machine to the same position. Repeatability errors are considered random and are difficult to predict. Resolution is the smallest step or increment that a machine can move. It is limited by either the electrical sensing equipment or by a mechanical phenomenon such as stiction or cogging. All of these factors were addressed in the redesign of the linear tool servo.

When discussing machine kinematics it is important to distinguish between sensitive and non sensitive directions of error motion. For a turning operation like that shown in Figure 1, small motions tangential to the part will result in a second order (sine) error in the finished part geometry. Motion in this direction, as well as any type of rotation about the tool point is considered non sensitive to the turning operation. Error motion in the radial direction will linearly affect the part geometry and is considered sensitive to the operation. The feed direction may or may not be considered sensitive depending on the part geometry (i.e. if it has a shoulder or other axial feature) When designing precision machines the greatest results will be achieved by concentrating on improving motion in the sensitive direction. In this case, motion in the sensitive direction is determined mainly by the actuator and feed-back mechanisms as well as the friction and resolution of the linear
bearing system. Error motions due to poor stiffness and straightness are second order effects but are still very important because they effect factors such as surface finish and tool wear. Poor dynamic stiffness can have a pronounced effect on dynamic stability, causing tool chatter and surface damage.

The term precision machine generally invokes the image of a delicate, sensitive instrument that is carefully maintained in a clean laboratory environment. This is rarely true for any machine tool, especially those that operate in a production environment. Most machine tools are subjected to abuse, coolant, dirt and crashes, so it is important that they are designed robustly. Robust design means more than that they shouldn't break or wear out under use. They also must maintain their degree of precision or they will be useless as a precision tool. During the design process a conscious effort was made to make the machine as robust and problem free as possible.

Cost and manufacturability were also addressed in the redesign process. Design for manufacturing and assembly strategies were considered. However, due to the low production volume of this machine (less than 100 per year) most of them did not directly apply. A more appropriate approach for a next generation modifications would be 'design for retrofit.' This includes using existing parts and bolt patterns whenever possible and designing modifications that are backwards compatible so that existing machines can be easily updated.

1.4 Existing Design Problems

In addition to the broad goals of the redesign described above, the linear tool servo had several fundamental design weaknesses that reduced the machine's accuracy and performance. The majority of the machine problems rest on the over constrained arrangement of the roller bearings used to support the tool carriage. If this over constrained design is corrected, most of the other problems will be resolved.
There are generally two design approaches used to achieve accuracy and repeatability in machine tools. Kinematic design attempts to constrain a body using the minimum number of contact points required to locate two bodies. This eliminates forced congruence and deformation and therefore makes the interface more predictable. Elastically averaged systems take the opposite approach. They use large contact areas to increase the strength of the interface. This method relies on deformation between the contact surfaces, or forced congruence. This is not deterministic, but the large contact area gives it much strength and therefore it may be more predictable under heavy loads.

![Diagram of linear bearing design](image)

**Figure 1.6 Existing Linear Bearing Design**

The original bearing design attempts to kinematically support the tool carriage by using pivoting bearing trucks and a floating leaf spring as shown in Figure 1.6 above. Ideally, a kinematic linear bearing system would have only five contact points. This would constrain five of the six degrees of freedom and allow motion in one direction only. The existing system has 20 roller bearings that contact the tool carriage. Kinematically, they are reduced to eight points with the use of flexures and pivots. This over-constrained design is the source of many of the bearing’s problems, including unbalanced stiffness, bearing surface damage and hysteresis.
The biggest drawback of the spring preloaded roller bearing design is its lack of uniform stiffness.

**What the designers were thinking:**
Kinematic arrangement of preloaded springs yields bidirectional stiffness.

**Reality:**
Kinematic preload acts as a pivot point and acts through the center of stiffness ...

**Results:**
Spring rotates and system has poor stiffness in one direction.

**Solution:** Increase preload => preload is limited by contact stresses

*Figure 1.7 Unbalanced Bearing Design*

The poor stiffness is due to improper design of the bearing preload mechanism. Ideally, the leaf spring preload force should be evenly distributed between the two pairs of upper bearings. As long as the preload force is not exceeded, the bearings should have equal stiffness regardless of whether they are loaded against the fixed or the floating bearings. This would have worked if each of the upper bearings were preloaded with individual springs. However, the leaf spring preload acts as a fulcrum and can pivot about its mounting point. When the system is loaded against the fixed bearings it will perform as expected, but when it is loaded against the bearings floating on the spring pivots, the lower bearing loses contact and the preload is lost. This results in very poor stiffness in one direction, as shown in Figure 1.7. The original designers were aware of this fact and compensated by rough-cutting against the compliant bearings and finish-cutting against the
stiff bearings. Ideally the bearing should have high stiffness in both directions. This will increase the cutting accuracy and the machines flexibility.

The data presented in Section 5.1 shows that bearings also exhibited a hysteresis curve during loading/unloading cycles. This is likely due to the nonlinear bending/pivot motion of the leaf spring preload coupled with motion of the balls in the roller bearings. The small hysteresis error is not a significant problem in this application because it is in a non-sensitive direction, however it is an indication that the bearings do not behave well.

Close inspection of the aluminum tool carriage revealed that the surface was damaged throughout the area of contact with the roller bearings. Carriage surface has a series of cylindrical indentations on the track where the bearings run. This damage indicates that bearing tool bar was being overloaded throughout its range of motion. Hertzian contact stress theory can be used to calculate the area and magnitude of contact stress between the bearing and tool carriage.

![Hertzian Theory](image)

Contact stress due to preload and cutting forces:
\[ P_{\text{max}} \approx 60 \text{ kpsi} \]

Yield Strength of 6061 aluminum:
\[ \sigma_y = 37 \text{ kpsi} \]

**Results:**
- Deformation of bearing surface
- Loss of precision & resolution
- Loss of controllability

**Figure 1.8 Calculating Hertzian Contact Stress**
These calculations, shown in Figure 1.8 show that the contact stress is actually higher than the yield stress of the base aluminum metal. The leaf spring preload force of approximately 120 Newtons is divided between the four upper bearings and the six lower bearings. The kinematic design relies on a few small contact points to distribute the load and therefore the contact stresses at those points can be quite large. The tool carriage surface has been hard-anodized to strengthen the interface, but anodization process is only a few microns deep and does not add significant strength to the surface of the base metal. The anodization process essentially creates a very thin, brittle surface on top of a much thicker, softer material, much like a thin layer of ice on top of a puddle of mud. The interface between the bearing and carriage is sufficiently fragile that the combination of static preload and dynamic cutting forces are enough to damage the surface. A sudden crash of the tool into the part would severely indent the surface of the carriage.

![Diagram of a roller bearing with labels indicating preload force, damage from overload, bearing surface, and friction force with a note indicating surface features cause variations in frictional force.]

**Figure 1.9 Affects of Surface Damage on Friction**

Lateral error motion caused by the damaged surface is on the order of 1-2 microns and since this motion is in a non-sensitive direction, it will not significantly affect the part accuracy. However, the indentations will effect the controllability of the tool servo. Figure
1.9 shows how a deep indentation located near the operating point can significantly increase the bearing impedance and resolution. It is difficult for a controller to compensate for this type of change in servo controller and the result will be a loss of positioning accuracy.

\[ \epsilon = L \alpha \Delta T \quad \alpha_{\text{aluminum}} = 23.6 \, \mu m/m/°C \]

Thermal Error for Aluminum Tool Carriage = 4.6 \, \mu m/°C

**Figure 1.10 Affects of Temperature Gradients: Thermal Expansion**

Another common problem that plagues many precision machines and instruments is that of unmeasured differential thermal expansion. In this case, heat generated by electrical resistance in the motors coil is conducted directly into the aluminum tool carriage. Because of the motor air gap and small bearing contact area, the majority of this heat must escape through the tip of the tool carriage which is exposed to a constant flow of coolant. This heat flow creates a temperature gradient across the carriage and causes thermal expansion of the tool carriage. Thermal gradients are usually very undesirable because they cause parts to warp and generate angular errors. In this case, the tool carriage is symmetric along the direction of the gradient so no warping will occur. However, there still will be significant thermal expansion in length. If all the components were made of similar materials, the thermal expansion would not be an issue because all of the components would expand equally. This is not the case because the aluminum is made of lighter-weight aluminum and
the remaining parts are steel or cast iron. Figure 1.10 shows the tool point error generated by a small change in temperature in the tool carriage. The use of closed-loop control eliminates all of the thermal expansion error to the rear of the encoder but there remains 50 mm of aluminum between the encoder and tool point with uncompensated expansion. This small length can generate significant errors, aluminum has a thermal expansion coefficient of $24 \times 10^{-6}/^\circ C$, a 2 degree temperature gradient will cause a 2 micron error at the tool point.

The basic goal of the redesign was to improve performance and robustness and to reduce cost. Many of the above design issues can be easily addressed by applying standard methods and techniques used in precision machines. In addition several new design and manufacturing methods were developed to further reduce cost and increase performance.
2 Precision Components and Techniques

Many different linear bearing and linear motion systems were considered during the design stage of this project. The final combination of porous carbon air bearings and a linear voice coil motor was found to have the best performance and fewest drawbacks. Section 3 discusses the various linear motion systems considered for this design. The result of this investigation was that an air bearing system would offer the best overall performance for this application. The following sections discuss the attributes of air bearing systems and an efficient method of manufacturing precision components.

2.1 Air Bearings

![Diagram of Air Bearing Pad and System](image)

**Figure 2.1 Schematic of Air Bearing Pad and System**

There are many different designs and configurations of air bearings, but the most common in machine tools and inspection equipment is the externally pressurized, opposed pad configuration. Externally pressurized refers to the use of high-pressure air that is forced into the gap between the two moving surfaces, usually through an orifice or slot.

---

2 Picture from Alexander Slocum, *Precision Machine Design*
Opposed pads are used to preload the bearing, generate bi-directional stiffness and increase stability. A schematic of a simple air bearing pad and an opposed pad system are shown in Figure 2.1.

Externally-pressurized air bearings offer many advantages over conventional sliding and rolling element bearings, especially for machine tools and inspection machines. They exhibit zero stiction and very low, predictable operating friction. Because there is no contact between the moving parts there is no wear and negligible heat generation. The effect of film-averaging and squeeze-film damping results in a reduction in high frequency motion errors and the transmittance of machine vibrations. The constant flow of air out of the bearing gap also prevents contamination and reduces the need for seals or operating precautions.

Air bearings do have limitations. The principle disadvantages are: poor dynamic performance; expensive precision manufacturing; and the need for an external source of clean, dry, pressurized air. The load capacity and stiffness of an air bearing system is also somewhat limited compared to other bearing systems, and their use is usually restricted to light machining and measurement applications.

Section 2.1.1 below discusses the general operating principles of air bearing systems and some details on how they are commonly applied. Section 2.1.2 discusses a special type of air bearing that uses a porous media to distribute the air.
2.1.1 General Theory

Figure 2.2 Orifice Bearing and Pressure Profile (operating gap, zero gap)

Figure 2.2 shows a cross section of a simple-orifice compensated air bearing pad and the pressure profile generated across the pad. The effect of this pressure distribution, integrated across the surface, is a lifting force on the bottom of the pad. During operation, air at pressure $P_o$ flows through the orifice and into the pocket where the pressure is constant at $P_p$. From the pocket it flows across the bearing land where it escapes to atmospheric pressure $P_a$. There is a significant pressure drop across the orifice and a more gradual, pressure drop across the bearing land. The profile of these pressure drops depends on the flow rate and flow resistance of the orifice and bearing land. As the bearing gap gets smaller, the flow decreases. The decrease in flow reduces the pressure drop across the orifice, which increases the average pressure across the land. Therefore, as gap decreases, lift increases. The limit of this relationship occurs when the gap is completely closed. In this case, there is no flow across the land, and the only lift is that due to pressure in the pocket. This initial lift, or load capacity, is solely a function of the supply pressure $P_s$ and the pocket area $A_p$, and determines the maximum load that the bearing can support at startup.

3Ibid.
The relationship of pressure drops and flows between the orifice, pocket and land is analogous to that of a voltage divider electrical circuit. Figure 2.3 shows the fluid and electrical models. Voltage is analogous to pressure, current to flow, and resistance increases with decreasing bearing air gap. As a load is applied to the bearing, the bearing air gap is reduced and the resistance across the land increases. This results in lower flow through and a smaller pressure drop across the orifice. The lower pressure drop increases the pocket and land pressure and creates a restoring force opposing the applied load. In this simple model lift is maximized by reducing the orifice resistance and increasing the pocket area. Actually, this is not the best design strategy, because both the orifice
resistance and the pocket volume have a profound affect on the dynamic performance of the air bearing. In practice, the pocket area is kept to a minimum, and the majority of the lift is generated in the pressure profile across the bearing land. A large bearing pocket has a large capacitance, and due to the compressibility of air the capacitance can produce a dynamic instability called pneumatic hammer. Air bearing are usually constructed with several orifices across the face of the pad, like that shown in Figure 2.4. This increases initial lift of the pad without requiring a large pocket and distributes the pressure gradient more evenly across the surface.

![Figure 2.4 Typical Thrust Bearing Pad Design](image-url)
An orifice restrictor is required for stable operation of opposed-pad configurations. Figure 2.5 shows a model of two bearing pads in an opposed-pad configuration, with the electrical analogy of a Wheatstone bridge circuit. If a load is applied to bearing rail, the lower air gap will decrease and the upper gap will increase. Without the orifice compensation, the lower gap will quickly close, the bearing will touch down and the entire flow will pass through the upper bearing. If the restrictors are properly designed the reduced flow in the lower bearing will increase the bearing lift and compensate for the applied load. This inherent compensation is common to all types of opposed pad air bearing systems though the exact method of restriction can vary. Common designs use hypodermic needles, tuned slots or porous plugs to produce a compensating pressure drop.

The stiffness of any air bearing system is strongly dependent on the air gap, or clearance between the bearing pad and the bearing surface. The bearings act by squeezing a film of air and because the viscosity of air is quite low, the bearing gap must be quite small to achieve adequate performance, usually 1-10 μm. These clearances can be very difficult to manufacture and maintain, especially for production machine tools. The bearing pads and bearing surface are typically made of a hardened steel and/or anodized aluminum.
Contact of bearing surfaces during operation, such as a crash, overload or loss of supply pressure, may result in serious damage to the bearing pads and surface. The bearings also require very clean, dry air; typically 1 µm or better. If the supply air is not properly maintained, the restrictor may clog, causing changes in the performance of the bearing, or possibly causing the bearing to fail. Because of these problems, air bearings have historically been restricted to the instrument and measurement field.

2.1.2 Porous Bearings

![Porous Bearing Pad](image)

**Figure 2.6 Porous Bearing Pad**

One approach to addressing some of the problems described above is the use of a full face porous air bearing pad, like that shown in Figure 2.6. Rather than use a single orifice, it uses a porous material, usually carbon or sintered bronze, that acts like a continuous distribution of orifices across the entire surface. With this design, there is no need to have a bearing pocket for initial lift as the entire face of the porous pad will be used during liftoff. The elimination of the bearing pocket reduces the capacitance of the bearing which results in more stable performance. The distribution of air over the entire surface also gives the porous pad superior tilt stiffness, which allows the pad to self-align to a surface if it is free to rotate. Because of their large distribution area and large number of flow paths,
porous bearings are less susceptible to contamination and require less stringent air supply requirements.

Bearing pads made from porous carbon have an additional advantage in that carbon is a natural lubricant. In the event of an overload or loss of supply pressure, contact between the carbon and the bearing surface will not cause serious damage. At worst, the contact will damage the softer porous material, rather than the precision-ground, hardened bearing surface. Usually this bearing contact will just cause the bearing pad to conform to the bearing surface and thereby be lapped even smoother.

The porous pads themselves are relatively simple to construct. They consist of a simple backing pad with distribution grooves glued to a thin slab of porous material, which can be easily lapped flat with a surface plate. Before assembly, however, the permeability of the porous material must be tuned, usually with a lacquer or other solvent based resin. When assembled, the bearing pads still require a very small air gap for proper operation. Several methods have been developed to using flexures and pivots to reduce the need for high precision assembly. These are discussed in section 3.2.

### 2.2 Potting and Replication in Manufacturing

![Diagram of potting and replication](image)

**Figure 2.7 Example of potting and replication⁴**

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⁴Ibid.
Potting and replication are two similar methods used to lower manufacturing costs by reducing the need for precision machining. Potting, also known as grouting, uses an epoxy polymer to rigidly and accurately locate two parts without using precision surfaces. For example, if two cast iron parts are bolted together, the rough, as-cast surfaces would not mate very well and the actual contact area is probably less than 30% of the total surface area. Contact area between mating parts greatly affects the strength and stiffness of the joint. By filling the interface with an epoxy grouting compound prior to assembly, the contact area can be greatly increased, and likewise the joint strength, stiffness and damping. This is illustrated in Figure 2.7. The potted compound completely fills the voids between the two mating surfaces and effectively keys the two surfaces together, preventing micro slip of the joint. The epoxy potting is generally only a light mechanical bond between the two surfaces, not an adhesive and mechanical fasteners are still required to support the static load on the joint. Since the potted joint has two perfectly mating surfaces, the two parts are less likely to deform when the bolts are tightened. Potting is commonly used to attach a machine tool to a floor, or to affix a bearing rail to a machine bed with minimal deformation.

Replication uses epoxy to generate a complex precision surface with the use of a mold or a master surface. This is different than potting in that the master surface is usually coated with a mold release so that the two surfaces may be separated once the epoxy has cured. This technique can be used to reduce precision machining by casting complex shapes such as hydrostatic bearing pads and machine threads. Low friction replicating material is often used to rebuild machine tool bearing ways; rather than grind and scrape a surface flat, it is cast to conform to the opposing reference surface.

Both methods use special epoxies that are formulated for either high stiffness, low shrinkage or low friction, depending on the application. Epoxies can be purchased in putty or liquid form. The putty can be applied with a trowel before the parts are assembled. The liquid can be poured into the void, and thin gaps can be filled by injection it with a syringe.
3 Prototype Design

This chapter discusses the design process and design decisions from concept to prototype. The design options considered for each component are summarized and the final design choices are justified. This is followed by descriptions of the vacuum-replication assembly and zero-clearance design methods used to construct the air bearing systems. The four prototype air bearing systems built for testing are then described.

3.1 Initial Design options

Several different designs were evaluated before an initial prototype was built. Most of them were quickly eliminated because they would be difficult to implement or prohibitively expensive. Most of the viable designs considered were evolutionary and focused on modifying the existing system in an effort to solve the major design issues described above. These design options focused on four basic areas: linear bearings, tool carriage design, linear actuators and sensor systems.

3.1.1 Linear Bearings

The linear bearings are the critical element in the linear actuator design because they effect or determine almost all of the performance characteristics of the actuator. The standard measures of a precision linear bearing are stiffness, straightness and resolution. Due to the high operating speed, friction is also important because it can generate heat and thermal gradients that will deform the structure. Durability and serviceability must always be considered when designing components that will be used in a production environment, Cost and ease of manufacture were also evaluated.
<table>
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<td>Existing design</td>
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<td>Fluid Leaks Too bulky</td>
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</table>

* considers orifice type air bearings. porous bearings will have improved performance

| Good         | ●          | ●          | ●          | ○           | Bad           |

**Figure 3.1 Comparison of Linear Bearing Systems**

Figure 3.1 shows the design criteria for the various linear bearing technologies considered for this system. Sliding, roller bearings and linear guides were eliminated from consideration because of their high friction and stiction problems. There is a trade-off between stiffness (determined by preload) and friction when using these bearings. This is evident in the design of the existing bearing system: it suffers from poor stiffness due to a low preload, but the preload cannot be increased due to contact stresses on the bearing surface. Hydrostatic bearings showed the most promise of all the systems considered, however it would be very difficult to implement in this application. The compact size of the bearing package and the presence of sensitive electronic sensors would make the plumbing and sealing of a hydrostatic bearing assembly difficult and expensive.

Initially, air bearings also had several design obstacles, mainly the tight manufacturing requirements and the susceptibility to damage in the event of a crash. The use of porous bearing pads instead of an orifice-compensated pad addresses the robustness concerns.
The manufacturing requirements can be significantly reduced using the vacuum-replication method described below. A porous air bearing system also addresses all of the design goals outlined in the above: improved stiffness, low friction and stiction, lower contact forces and higher precision motion.

### 3.1.2 Tool carriage

The design and construction of the tool carriage was also considered in the evaluation of the linear actuator system. The design issues addressed were stiffness, surface strength, thermal expansion, mass and vibration. Several different materials, geometries and configurations were considered.

<table>
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</tr>
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**Figure 3.2 Properties of Potential Tool Carriage Materials**

Figure 3.2 shows the relevant properties of several alternate materials considered for the tool carriage. High stiffness (E) is required to reduce bending deflections and surface damage due to bearing contact stress. Since the existing tool carriage is made of aluminum, which has the lowest stiffness, any of the other materials would present an improvement. Mass must be minimized in oscillatory machinery and therefore density is important. Here aluminum is the best performer; steel and iron have nearly three times the density. The best measure of dynamic performance is the stiffness-to-weight ratio. It addresses mass and bending strength as well as natural frequency of vibration. Alumina is by far the best performer in this category.
In addition to mechanical properties, the thermal performance of the candidate materials was also considered. Ideally, it would be best to build the entire machine out of a material with a very low thermal expansion, such as invar, to eliminate thermal errors. This is clearly not possible, so the next best alternative is to use materials with similar coefficient of thermal expansion (\(\alpha\)) throughout the design so that all thermal growth around the structural loop will cancel. The large disparity in \(\alpha\) between the aluminum tool carriage and cast iron housing is a significant source of error. Thermal conductivity and thermal heat capacity can be used as a guideline to approximate the time required for a temperature gradient to dissipate in a part. High conductivity and low heat capacity will result in small thermal time constants and thus short warm-up cycles.

The data above suggest that alumina ceramic would be the best candidate for improving the performance of the tool carriage. It has a very high strength, and although it is slightly more dense than aluminum, its' high stiffness-to-weight ratio will permit much lighter structure. The dynamic thermal response of the ceramic is not as good as that of aluminum, however the thermal expansion is similar to the cast iron support structure, so differential thermal growth will be less significant.
Figure 3.3 Tool Carriage Cross-Section Designs

The higher strength of alumina allows the use of a lighter, thinner structure. The original part was simply a square bar with the center bored out for weight savings. The stiffness and weight of the carriage can be optimized by using an internally ribbed structure as shown in Figure 3.3. This shape can easily be extruded while the ceramic is still in its green form. The external surfaces can be accurately ground after the part is fired.
Figure 3.4 Composite Tool Carriage Design

Complex ceramic components are difficult to manufacture because accurate surfaces must be ground after the part is fired. Cast surfaces cannot be considered accurate due to shrinkage during firing. This problem can be solved by using a composite design, like that shown in Figure 3.4. High-strength alumina with internal ribbing can be easily extruded for the body of the part and the complex features on the ends can be machined from lightweight aluminum or beryllium. The components can be assembled using a high-strength epoxy and finish-ground to dimension. If necessary, additional structural damping could be achieved by integrating viscous shear dampers into the internal structure of the alumina body.

Ultimately, the ceramic composite tool carriage design was abandoned because of cost and lead time considerations. It was decided that the existing tool carriage would be used in the prototype design. The aluminum carriage was the lightest option and the lack of
surface strength would be addressed by replacing the roller bearings with air bearings. This would simplify the design and construction and would make the transition between designs less expensive and time consuming. The more advanced composite design can be easily implemented in the future if further improvements in performance are required.

3.1.3 Linear Motors

The linear actuator used in the design has a large effect on the performance of the tool servo. A number of linear actuator technologies were investigated for this design but only a few were capable of the accuracy, speed and length of travel required for this application. The existing system uses a fixed magnet, moving voice coil linear motor. This design is lightweight, compact and can fulfill the force, stroke and accuracy requirements of the application. However, energy dissipated due to resistance in the motor coils is the major source of heat generated in the actuator. Because the coil is moving, there is no way to thermally ground the coil and remove the heat, which is subsequently conducted from the coil and into the tool carriage, causing thermal growth errors. The motor coils can be thermally grounded by inverting the design, i.e. a fixed coil, moving magnet motor or solenoid. While this design addresses the thermal issues, the heavy magnets also greatly increases the moving mass of the system, thus reducing its dynamic performance.

Other types of linear motors and standard linear actuators were also considered, but did not favorably compare with the existing voice coil motor. DC linear motor designs required either brushes, which are prone to wear, or expensive commutation circuitry for control. AC synchronous and inductance motors are heavy, bulky and suffer from poor resolution due to cogging. All of the other standard linear actuators surveyed, such as hydraulic cylinders, ball screws and mechanical linkages, do not have adequate accuracy and performance for this precision application.

Consideration was also given to solid-state piezoelectric and magnetostrictive actuators. These actuators are very attractive because they are compact, simple to design and would be
easy to implement. The most significant advantage to these actuators is that they would eliminate the need for precision linear bearings. However, they have a very limited strain rates, the 0.5 mm amplitude of motion would require a PZT stack over 1/2 meter long. In addition to this, they require expensive, high voltage amplifiers and control is difficult due to hysteresis and non-linearity.

Ultimately, it was decided that the original moving voice coil linear motor design would be retained. None of the options above presented a significant improvement or advantage over the existing system and retaining the existing motor would eliminate the cost and effort required to design, build and control a new linear actuator. A more complete description of all of the above linear actuator technologies and their properties can be found in appendix A1.

### 3.1.4 Sensor Systems

![Figure 3.5 Feedback Sensor System](image)

**Figure 3.5 Feedback Sensor System**

The last area of consideration was the feedback system used to control the position of the tool carriage during operation, which is shown schematically in Figure 3.5. The existing design used an optical encoder mounted to the housing with a glass scale mounted on the moving carriage. The grating on the glass scale had a pitch of 50 lines/mm and
using quadrature decoding and 10X interpolation the system had an electronic resolution of 0.5 μm. A linear velocity voltage transducer (LVT) was also mounted in the center of the voice coil. The velocity feedback was used to add derivative damping term to the control loop.

Ideally, the position feedback device would read the position of the tool point directly, to eliminate any errors between the measurement and tool point. This is very difficult to achieve in practice, however, because the tool point is extended beyond the structure, and is operating in a harsh environment. The inability to directly measure the tool point position can increase the position error. The scale is mounted approximately 150 mm behind the tool point. Temperature changes causing thermal expansion will change the length of the tool carriage, thus changing the position of the tool. Expansion behind the scale will be measured and compensated for in the control loop, but expansion beyond the scale will cause an error at the tool point. As discussed above, this error can be quite significant; approximately 1 μm/°C.

Two methods of eliminating this thermal growth error were investigated, repositioning of the optical scale and temperature feedback compensation. The original position of the scale on the tool carriage was determined by the clearance between the rolling bearings. If they were to be replaced with air bearings, it would be possible to move the scale forward. Temperature feedback compensation could be achieved thermally mapping the tool point error as a function of carriage temperature. During operation, the controller could then continuously adjust the tool position as the machine warmed up, eliminating the need for warm-up cycles at the beginning of a shift. Thermal tests on the system using a thermocouple mounted in the tool carriage confirmed that the tool point error was linearly tracked by the carriage temperature.

The results of the thermal compensation tests suggested that temperature feedback would be adequate and it was not necessary to change the position of the position encoder scale. This would eliminate the need to extensively modify the existing tool carriage for
use in the retrofit design. The only change would be the addition of a thermocouple mounted in the center of the tool carriage. If additional accuracy was required, the encoder scale could very easily be moved forward, reducing the unmeasured error due to thermal expansion.

After considering all of the options described above, it was determined that the design and development efforts would be focused on improving the linear motion by implementing air bearings. Most of the problems with the original design stemmed from the linear bearing system and a proper design of the bearings will address most of the problems. Additional improvements might also be gained by modifying the tool carriage, motor and sensors, but they would involve greater modifications and therefore would be more difficult to implement as a retrofit. If additional performance was required, many of the designs concepts, such as thermal compensation and the use of a composite tool carriage, could be implemented independently after the air bearing modifications were complete.

3.2 Methods and Techniques

Having chosen porous carbon air bearing pads as the support bearings for the tool carriage, a method had to be developed to fixture the pads in the proper position relative to the bearing surface. The fixtureing method must be capable of accurately positioning the pad parallel to the bearing surface with extremely small air gaps (< 5 μm). Simply bolting the pad to the structure is not practical. It would require that all mating surfaces be ground flat and parallel to better than 5 μm, which can be very expensive, if not impossible.
Figure 3.6 Previous Bearing Support Schemes

Previous designs used either a ball and socket arrangement or a thin flexure to support the bearing pad. Both of these designs, shown in Figure 3.6, fulfill the parallelism requirements by allowing the bearing to tilt or pivot about the support. This flexibility is not without a cost. Both of these designs suffer from poor stiffness and stability due to their limited support and both may require precision machining and tedious adjustment to produce the desired air film.
3.2.1 Vacuum Assembly Method

![Diagram of vacuum assembly method]

**Figure 3.7 Vacuum-Assembly Method**

A method was developed that incorporates the self-alignment properties of the methods without sacrificing strength or stability. An accurate air gap is created during assembly by using a shim between the bearing pad and bearing surface. If the shim is perforated, then a vacuum drawn through the pad will temporarily fixture the pad and shim to the bearing rail. The pads can then be permanently fixtured in the proper orientation by replicating them in place with a grouting epoxy, as shown in Figure 3.7. The cured epoxy accurately fixtures and provides full support for the bearing pad, thus eliminating the poor stiffness and instability inherent in the flexural and ball/socket designs. The epoxy film also eliminates the need for accurate surface finish on the back of the bearing pads and the inside of the structure. The bearing faces, however, must be lapped flat to provide proper vacuum-adhesion. After the epoxy has cured, the vacuum and shims can be removed and the bearing is completely operational.

Stiffness losses due to compliance in the epoxy can be effectively eliminated by minimizing the epoxy support thickness. Epoxy films of less than 0.030" can be achieved by properly fixturing the bearing assembly relative to the support structure and then injecting the epoxy into the gap with a syringe. The gap size is limited by the pressure available to inject the epoxy and the viscous forces required to squeeze the epoxy between
two surfaces. Also, if the assembly is not rigidly fixtured during injection, the pressurized epoxy can force the components out of alignment.

An additional benefit of the replication assembly method is the serviceability of the system. The air bearing pads can be easily removed by using a jack screw in the injection hole to force them out and this can be further facilitated by coating the pad with mold release before assembly. The pads can be accurately replaced because the epoxy has formed a perfect mating surface with the pad. Screws can also be used to hold the pad in place with little fear of pad deformation because the replicated surface yields 100 percent contact area between the pad and epoxy. If the pad requires replacement, it can be re-cast using new epoxy and the vacuum method outlined above.

3.2.2 Zero Clearance

Bearing performance, i.e. stiffness and damping, is strongly dependent on the size of the air gap between the pad and the bearing surface. Generally, the gap should be as small as the surface roughness will allow. If no shims are used during vacuum assembly, it is possible to produce an air bearing system with zero initial clearance. Ideally, when the vacuum is removed and pressure is applied, the bearing will have infinite stiffness. (recall that stiffness is proportional to 1/gap) In practice, however, the lift force generated by the bearing will produce a small deformation in the support structure. This deformation and the resulting air gap are shown schematically (exaggerated) in Figure 3.8.
**Figure 3.8 Expansion of Bearing Shell Due to Air Pressure (exaggerated)**

This deformation produces a bearing air gap and therefore the bearing has a finite stiffness. The bearing gap, and therefore the bearing performance, depends greatly on the stiffness of the support structure. Small gaps can be produced using very stiff structures, and larger gaps with more compliant structures.

### 3.2.3 Bearing Assembly

A simple, scalable linear bearing guideway can be built using the vacuum-replication assembly method and zero-clearance design. This guideway has three basic components: air bearing pads, a bearing rail and a structural shell. Plumbing fittings and tubing are also required during the assembly and operation of this bearing. Figures 3.9-3.12 describe the assembly of a simple bearing system using the methods described above.
Figure 3.9 Bearing Components

Figure 3.9 shows the bearing rail and a single bearing pad. The bearing rail has been ground flat and parallel, and the face of the bearing pad has also been lapped flat. Not shown is the bearing shell/structure. With the exception of the finishing described above, none of the shell or pad surfaces require and finish machining for this assembly.
Figure 3.10 Using a Vacuum to Assemble Bearings

Figure 3.10 shows how the bearing sub-assembly is constructed. The bearing pads are fixed directly to the bearing surface with a vacuum. Because the zero-clearance design is being used, no shim is required to produce the bearing gap. Figure 3.11 below shows how this assembly is inserted into the bearing shell/support structure. The bearing assembly is accurately positioned within the shell using parallels or a custom jig.
Figure 3.11  Inserting Bearing Assembly into Shell

Figure 3.12  Potting Bearing Pads with Epoxy
Figure 3.12 shows the final assembly step, potting the bearings with structural epoxy. The epoxy is injected into the gap through a hole in the shell using a syringe or caulking tube. The vacuum can be removed after the epoxy has cured (approx. 24 hours) and the bearing assembly jig can also be removed. At this point, pressurized air can be supplied to the pads and the bearing should be fully functional. The assembled bearing will look similar to that shown in Figure 3.12.

In addition to being an inexpensive manufacturing method, this can also be a very powerful design tool. Performance of previous air bearing systems was dictated by the cost or the ability to manufacture an accurate air gap. Bearing performance is now controlled by varying the structural stiffness. Initial design equations for implementing the zero clearance design method are discussed in the analysis section below.
3.3 Test Systems

Four porous air bearing systems were developed using the methods outlined above. All of the bearing systems used a porous air bearing sleeve that runs on a ground bearing guideway, like that shown schematically in Figure 3.13. The first two were built for proof-of-concept purposes and to evaluate different bearing configurations. The third bearing was designed as a prototype retrofit for the linear tool servo discussed above. A fourth system, currently under development, will be used for a high-precision printing application. Each of these systems is briefly described below, with further emphasis on the design of the prototype retrofit for the linear actuator.

![Schematic of Test Bearing System](image)

**Figure 3.13 Schematic of Test Bearing System**
3.3.1 System 1: Devitt System

The first air bearing system was built prior to the inception of the linear tool servo project by Devitt Machinery of Aston, PA. It consisted of a 150 mm x 40 mm x 750 mm anodized aluminum rail, and a ‘C-shaped’ aluminum bearing bushing. The bushing contained 12, 50 mm x 100 mm porous air bearing pads, 4 on the narrow side and 8 on the wide side, in an opposed pad configuration. This system was developed as a proof-of-concept to demonstrate the vacuum-replication assembly method outlined above. It was designed to act as a replacement for similarly sized precision bearing bushing systems. This system is shown in Figure 3.14 below.

![Devitt Bearing System with Instrumentation](image)

Figure 3.14 Devitt Bearing System with Instrumentation
3.3.2 Ceramic System

A second system was assembled that was of similar to, but slightly larger than the final prototype. The bearing, shown in Figure 3.15 and 3.16, consisted of a cast iron housing and a ceramic bearing rail. It was designed to be used as a test system; its larger size simplified access to the components. The system used two distinct bearing configurations. The horizontal orientation had one pair of full length bearings and the vertical used two pair of smaller pads.

![Diagram of Ceramic Bearing System](image)

**Figure 3.15 Schematic of Ceramic Bearing System**
Figure 3.16  Ceramic Bearing System

This permitted the testing of the effect of surface area and geometry on stiffness and damping. Figure 3.17 below shows the two bearing configurations and the location of the center of stiffness for each pair of pads (solid line). The two pad system has the same system center of stiffness (dashed line) as the one pad design, but the distance between the two pads may provide improved moment stiffness over the single pad design. Higher damping is expected from the single-pad design due to the larger surface area. However, this larger surface area will produce greater lift and a larger air gap, thus reducing the damping. These stiffness and damping issues are resolved in the tests discussed in Section 5 below.
3.3.3 Prototype System

This system was designed as a retrofit to the existing system. It was required to be compact enough to fit in the existing casting, stiff enough to support the cutting loads and had to meet all of the other criteria outlined above. The system, show schematically in Figure 3.13 above and 3.18 below, consists of the original tool holder, eight bearing pads and a steel support structure, or shell. The original toolholder was used to simplify the design and retrofit process as discussed above. The eight pads were 1.7 in. square and were made of varying thickness to position the tool holder in the proper position relative to the other tool servo components. The steel shell was a 3.5” square steel tube with 0.25” wall thickness.
The dimensions and scale of the prototype system are such that it can fit directly into the existing actuator system as a retrofit. The similar size and the fact that the retrofit system also uses the original toolholder, allows accurate comparison testing between the two systems.
3.3.4 36" Bearing Guideway

A fourth linear bearing system was built to investigate the applicability of this technology to precision low-speed velocity control. Conventional roller-bearing and ball-bushing systems suffer from stiction and bearing noise that can degrade the velocity control. The low dynamic friction and lack of static friction make air-bearings an attractive option for this type of system. Tests are currently being run on this system to test its’ performance relative to a similar roller bearing system. This system is shown in Figure 3.19 below.

Figure 3.19 36" Bearing Guideway system
4 Analysis

There are several important performance criteria that must be considered when designing a linear air bearing system. The analysis of this type of system can be very complex, however, using some simple assumptions the basic design rules can be derived. The section below discusses the basic calculations required to adequately design or analyze a ‘zero clearance’ porous air bearing system.

4.1 Stiffness

The stiffness of the bearing system is a combination of the air bearing film and structural stiffness. In conventional air bearing systems, the structural stiffness is usually several orders of magnitude greater than that of the air bearing, and thus can be ignored. However, for a system using the zero-clearance method described in Section 3.2.2, the structural stiffness may be of the same order as that of the bearing stiffness and therefore both must be considered for accurate analysis. The modeling and analysis of the system stiffness and interaction between the air bearing and structure is quite complex. For first-order design purposes however, it can be simplified with several approximations. Design or analysis of a bearing system proceeds as follows:

<table>
<thead>
<tr>
<th>Design</th>
<th>Analysis</th>
</tr>
</thead>
<tbody>
<tr>
<td>Choose desired bearing stiffness</td>
<td>Calculate structural stiffness</td>
</tr>
<tr>
<td>Choose operating pressure and pad area</td>
<td>Calculate lift force based on pressure and area</td>
</tr>
<tr>
<td>Calculate required air gap for desired stiffness (assume infinitely stiff structure)</td>
<td>Calculate air gap based on lift and structural stiffness</td>
</tr>
<tr>
<td>Calculate required structural stiffness based on lift force</td>
<td></td>
</tr>
</tbody>
</table>

55
The following section discusses methods for calculating structural, bearing and total system stiffness.

4.1.1 Structural Stiffness

\[
K_s = \frac{EWT}{L} \\
K_b = \frac{24EWT^3}{L^3} \\
K_{total} = \frac{K_b K_s}{K_b + K_s}
\]

For \( \frac{T}{L} \approx \frac{1}{12} \) \( K_s \gg K_b \) \( \therefore K_t \approx K_b \)

Stretching Mode \hspace{1cm} Bending Mode

Figure 4.1 Calculation of Structural Stiffness

The bearing support structure for the systems described above generally consisted of a closed-shell structure, the one exception to this is the Devitt system which was an open-C structure. The analysis of the closed-shell system is shown schematically in Figure 4.1. The bearing air gap is generated by two dominant modes of structural deformation, plate bending of the top, and tensional elongation of the sides. In general, the plate bending compliance dominates and the problem is reduced to a simple beam deflection.
4.1.2 Bearing Stiffness

A simple first order approximation of the static stiffness of a single bearing can be calculated using equation 1.

\[ k = \frac{\Delta W}{\Delta H} = \frac{1}{2} \frac{P A}{h} \]  \hspace{1cm} (1)

P: supply pressure
A: pad area
h, \Delta H: bearing gap

Equation 1 is derived by dividing the maximum load capacity \( W \), by the maximum deflection (gap height) \( h \). The constant factor in front of the equation is empirical, and compensates for the pressure loss through the porous material and for the pressure profile across the bearing face. Equation 1 is sufficient for most design approximations and initial analysis. A more complex model of the air bearing stiffness is required for more accurate analysis. This analysis shows that there is a \( 1/h^3 \) and \( 1/h^4 \) dependence between stiffness and pressure.

4.1.3 System Stiffness

The zero clearance assembly method relies on the deformation of the machine structure to produce the air bearing gap and therefore the bearing stiffness is directly coupled to the structure stiffness. The system stiffness includes the deformation in the structure as well as the compression of the air film. If this is modeled as two springs in series, the system stiffness can be found using equation 2.

\[ K_{\text{system}} = \frac{K_{\text{air}} K_{\text{struct}}}{K_{\text{air}} + K_{\text{struct}}} = \frac{K_{\text{air}}}{1 + \frac{K_{\text{air}}}{K_{\text{struct}}}} \]  \hspace{1cm} (2)
The curve in Figure 4.2 shows how the bearing gap height and bearing system stiffness vary with the housing structural stiffness. When the housing is very compliant, the bearing will have large air gaps and poor stiffness. Increased housing stiffness yields smaller bearing gaps and a stiffer system. This relationship changes abruptly at $K_{\text{max}}$, which is the point where the bearing has grounded out. The structural stiffness, $K_{\text{design}}$, should be somewhat lower than this value so that a minimal air gap $h_{\text{min}}$ can be achieved. If the structural stiffness is greater than $K_{\text{design}}$, the air gap will be less than $h_{\text{min}}$ and grounding problems may arise.

![Figure 4.2 Effect of Structure on System Stiffness](image-url)
4.2 Straightness

The straightness performance of a linear bearing system depends smoothness and straightness of the surface and bearings, as well as the geometry of the bearing itself. The straightness performance of a typical linear roller bearing system, like that shown in Figure 4.3 is affected by the condition of the bearing surface, the roller bearing roundness and the distance L, between the two rollers. Roller bearing non-circularity and bearing surface finish generate high frequency straightness errors. Increasing the distance between the bearings has an averaging effect, and tends to reduce these errors, as well as lower frequency straightness errors in the bearing surface.

![Diagram of bearing straightness components]

**Figure 4.3 Bearing Straightness Components**

The straightness performance of any air bearing system is dependent mainly on the low-frequency straightness error of the bearing surface. High frequency straightness error, or surface roughness will tend to be averaged out by the air bearing film as long as the peak surface feature is less than the minimum bearing gap height. The low frequency errors will also be averaged by the bearing system, this averaging effect depends on the distance between the front and back pads, as well as the compliance in the bearing structure. The
bearing will be able to accommodate small errors in flatness and parallelism of the bearing guideway if the structure has sufficient compliance.

4.3 Dynamic Response (Damping)

Static stiffness test are a good tool for evaluating the capacity of bearing systems, but they do not evaluate how the bearing will perform dynamically. During operation, the bearings will be subject to large dynamic loads over a large range of frequencies, especially at the harmonics of the spindle speed. It is very important that these frequencies be well damped to avoid dynamic instability.

The dominant form of damping in aerostatic bearings is squeeze-film damping, and is essentially the compression and expulsion of the thin film of air between the pad and bearing surface. The characteristics of this form of damping can be derived from the model presented in Appendix A1. The damping is proportional to the viscosity of air and inversely proportional to cube of the gap. Because of the low viscosity of air, careful design is required to build a bearing with adequate damping. The poor damping, coupled with the compressibility of air, can lead to bearing instability known as pneumatic hammer. The strong dependence on gap height suggests that bearing damping can be improved by reducing the gap height. Very small airgaps can be achieved using the vacuum replication and zero-clearance design methods described above. Because of this, these bearings should have very good damping and stable dynamic response.

4.4 Friction

Bearing friction has a significant effect on the controllability and power consumption of the tool servo system. High static friction, is the main cause of stick-slip motion and effects the maximum resolution of the actuator system. Kinetic friction consumes power and generates heat which reduces the efficiency and accuracy of the system. Static and dynamic friction in a roller-bearing system depends on surface finish, bearing geometry
and preload. Friction in air bearing systems is due only to the viscous shear of the air film. Viscous shear forces depend on velocity, at zero velocity there is no friction. Therefore there is no static friction and no problems with stiction. Dynamic friction is also small because of the low viscosity of air.
5 Testing

A variety of tests were run on each of the bearing systems. These include static stiffness, horizontal and vertical straightness, dynamic response and friction. The measurements for these tests were taken at the end of the tool carriage, as close to the location of the tool point as possible. The existing system was tested with the bearings inside the support housing. The air bearing retrofit was tested without the motor or sensor systems attached.

5.1 Static Stiffness

![Diagram of Bearing Stiffness Test System]

Figure 5.1 Bearing Stiffness Test System

Static stiffness of the bearing systems was measured using the test setup shown in figure 5.1. The test setup used a pneumatic cylinder and load to apply a known, variable load and displacement was measured using a capacitance probe. Tests were taken on the bed of a knee mill, with the pneumatic cylinder mounted in the spindle chuck. The original actuator was bolted directly to the machine bed with T-nuts. Toe clamps were used to fix the air bearing system. Because of the small airgaps and high compliance of the support
shell, the toe clamps could not be tightly fastened or the air gap would close. This
additional compliance resulted in low system stiffness measurements. This effect can be
removed by modeling the system as two springs in series; the air film and the structure.
The structural stiffness can be approximated by testing the system at 0 psi. The air film
stiffness can then be calculated using the structural and system stiffness, as shown in figure
5.2.

Figure 5.2 Measuring Stiffness of Air Film

Stiffness is calculated by taking the derivative of the slope of the force-displacement
curve. These stiffness numbers cannot be directly compared to those calculated in Section
4. These tests involve the angular stiffness of the entire system and the calculations in
Section 4 deal solely with a single pad and an orthogonal load. The air bearing system was
also tested using several different supply pressures. The results of these tests are shown in
the graphs below.
Figure 5.3 Effect of Supply Pressure on Air Bearing Stiffness

The curves in Figure 5.1 show the effect of supply pressure on the bearing system stiffness. The 0 psi curve was taken as a reference system stiffness, it measures the compliance of the tool carriage, structure and mounting system. The 60 and 90 psi tests have very similar curves, indicating that there is not a strong relationship between pressure and stiffness. This is contrary to the linear relationship predicted in Section 4.1.2, but not unexpected considering the non-linearities introduced by the compliant support structure. The 60 and 90 psi stiffnesses do behave linearly with displacement up to approximately 300 μinch. This is the point where the air gap is closed and further deflection is purely structural. This is confirmed by the similar slopes between the pressurized and unpressurized curves. The 90 psi curve extends slightly beyond that of the 60 psi curve, thus the effect of the higher stiffness is solely a larger gap, but no increase in stiffness.

The air bearing stiffness can be calculated using the values from Figure 5.3 and the equations in Figure 5.2. This calculation yields a stiffness of 165,000 lbs/inch, which is
the same as the structural stiffness alone. It is coincidence that these two numbers are equal, but because they are, the bearing system stiffness is maximized.

![Graph comparing force-displacement curves for air and ball bearings](image)

**Figure 5.4 Comparison of Bearing Stiffnesses**

Figure 5.4 above compares the force-displacement curves for the air bearing and roller bearing systems. The air bearing stiffness exhibits the same non-linear response as that shown in Figure 5.3 and therefore the actual bearing stiffness can be assumed to be approximately 160,000 lbs/inch. This test shows that the air bearing system, if properly mounted, can provide a 5 times improvement in stiffness over the existing design.
Figure 5.5  Roller Bearing Stiffness

Two other important results arose from the stiffness testing of the roller-bearing system. Figure 5.5 shows the stiffness response of the bearing when it is loaded in two different directions. The upper curve is stiffness data from when the bearing was loaded against the fixed bearings. The lower curve is loaded against the leaf spring-supported preload bearings. Ideally, these two curves should be equal, yielding equal stiffness in either direction. However, due to the rotation of the leaf spring, as explained in Section 1.4, the bearing system does not have bi-directional stiffness.
A second problem with the roller-bearing system is illustrated in Figure 5.6. This is a hysteresis curve generated by loading and unloading the bearing against the leaf spring. This curve indicates that significant error is generated when cutting against this bearing.

Both the bi-directional stiffness issue and the hysteresis are dealt with by using the leaf spring bearing for rough cutting only. This is acceptable, because the accuracy requirements are not crucial for rough cutting and the fixed bearing is accurate enough for finish cutting. However, if the hysteresis were eliminated and the stiffnesses were balanced, the rough/finish cutting restriction could be eliminated and the flexibility of the system would be increased.
Figure 5.7  Effect of Pad Geometry on Stiffness

Figure 5.7 shows the effect of pad area and geometry on the bearing stiffness. The stiffness of the single-pad design is approximately 50% greater than that of the two pad design. This demonstrates that the moment stiffness is determined by the pad area and not the center of stiffness as discussed in Section 3.3.2.
5.2 Straightness

Bearing straightness was measured using the configuration shown in Figure 5.8. A potentiometer with a constant voltage source was used to measure the position of the axial position of the bearing. Lateral displacement was gauged with a capacitance probe. A Zerodur™ straightedge with a straightness of λ/10 was used as a reference surface.

![Diagram of Straightness Test Setup]

Figure 5.8 Straightness Test Setup
Figure 5.9 compares the straightness of the air bearing and roller bearing systems. The two systems used different tool carriages, but both were manufactured to the same specifications. This graph shows that the air bearing system has an order of magnitude improvement in straightness performance. In addition to the large peak-to-valley straightness errors, the roller bearing also exhibited large, high-frequency errors. This is due to the surface damage and bearing noise discussed in Section 1.

Figure 5.9  Straightness Test Data
Figure 5.10 shows the straightness performance of the Devitt bearing system. Unlike the other systems tested, bearing surface straightness data was available for this system. The original bearing surface had an average straightness of 20 μinch/inch and a peak-to-valley error of 150 μinch. The air film averaging and geometric averaging effects of this bearing yielded a 3 times improvement in operating straightness.

![Graph showing straightness performance](image)

**Specifications of bearing surface**
- Straightness: 20 μinch/inch
- Total: 150 μinch

**Performance of air bearing**
- Straightness: 7 μinch/inch
- Total: 38 μinch

Figure 5.10  Straightness Test Data on Devitt System
5.3 Dynamic Response

![Diagram of dynamic test setup]

**Figure 5.11 Dynamic Test Setup**

Dynamic response of the bearing systems was measured using drive point impulse response measurement techniques. An accelerometer was mounted at point on the tool carriage was chosen that was close to the actual tool point, as shown in Figure 5.11. Using a calibrated impulse hammer, a known impulse was applied to the tool carriage near the accelerometer and the response was recorded with a signal analyzer. Several different frequency ranges were tested using different hammer points and accelerometers. By varying the instrumentation and frequency ranges, data with high (95%) coherence could be attained.
Figure 5.12 Dynamic Response of Prototype Air Bearing System

Figure 5.12 shows the gain and phase data for a 250Hz - 4KHz drive point response test. The phase curve comes close to crossing the 90° (π/2) axis three times between 1250 and 1750 Hz. This indicates that there may be resonances in this frequency range. The plot of frequency gain also has a small peak in this frequency range, supporting the possibility of a pole. Overall, the response curve is very smooth, the peaks are very flat and wide. This indicates that the system is very well damped and resonance should not be a problem.
Figure 5.13 Comparison of Dynamic Response: Original and Prototype Systems

Figure 5.13 compares the response of the roller and air bearing systems. The number of large, narrow peaks on the roller bearing response curve indicates that this system has several poorly damped resonances. Because these resonances can cause dynamic instabilities if excited, the existing controller contains several notch filters to reduce their effect. Overall, the air bearing offers a significant improvement in dynamic response, which will result in improvements in accuracy and reduction of controller complexity.
Figure 5.14 Frequency Response Comparison of Bearing Configurations

Another dynamic issue investigated was the effect of pad configuration on damping. Figure 5.14 compares the frequency response plots of the two bearing pad configurations of the ceramic system. The plots show that the full length pads are slightly better damped than the two discrete pads. This effect was predicted above based on the increase bearing surface area. This result, coupled with improved stiffness suggests that the full length pads have better performance over the smaller, discrete pads.
5.4 Friction

![Diagram of friction test measurement](image)

**Figure 5.15 Friction Test Measurement**

Frictional resistance of the bearing systems was measured with a hand held spring scale, as shown in Figure 5.15. This was not a high precision test setup, however, it did allow rough approximations of force and qualitative comparisons between the two systems. The frictional force in the air bearing was below the resolution of the scale, which was 0.1 lbf. Friction in the roller bearing system varied between 1 and 3 lbf, depending on the position and velocity. The 3 lbf figure was the force required at startup, and can be considered an approximate value for the stiction force. These measurements are clearly quite rough and should not be used for quantitative calculations. However, they are valid for qualitative comparisons between the two systems. They demonstrate that the air bearing system offers a significant reduction in bearing friction. This The elimination of friction will reduce the power consumption and heat generation while increasing the accuracy and controllability.
6 Final Design

This prototype addresses all of the design goals outlined above. The use of air bearings eliminates the preload, stiction and stiffness problems. Replication reduces number of parts from over 100 to less than 10, thereby significantly reducing manufacturing and assembly effort. Precision manufacturing requirements are all but eliminated. The only part requiring an accurate finish is the tool holder, which is carried over from the original design. An important feature of this design is its compact size. It is designed to fit directly into the existing actuator casting as a retrofit. Other than the shell and bearing pads, no additional parts are required to retrofit the existing design to accept the air bearing system.

The assembly of the bearing cartridge, sensor systems and the complete actuator also comprised significant design effort. Special consideration was required to assure that the tool carriage was properly aligned with the sensors and that the housing structure did not exert additional forces on the bearing shell. Replication reduces the number of accurate mating reference surfaces, making this task more difficult. This problem was solved by using potting epoxy once again. The bearing cartridge was aligned inside the housing using a fixture and potting epoxy was used to hold it in place. This was done with the bearing pressurized so that the structural stiffness would not change due to the mounting. This method is described in the assembly drawings below.
Figure 6.1 Final Design

In what follows, the various components comprising the final design will be described. This also includes modifications to original parts, special fixtures or jigs and assembly instructions. Italic names refer to specific part drawings, copies of the final design drawings and assemblies can be found in Appendix A2.

With the exception of some small changes, the prototype and final designs are quite similar. As mentioned above, the design used several of the components from the original system. Some of these parts required modification to be used in the final design.
6.1 Part Modifications

Tool Carriage Modifications:

- Groove for détente bearing position removed, will not work with air bearings.
- Bearing surface near flange needed to be undercut to remove material inaccessible to grinding wheel.
- Holes for dowel pins are removed. Vacuum assembly method does not allow pins projecting above surface.

Lower Housing Modifications:

- All holes and pockets for leaf spring preload assembly can be removed.
- All features inside the front section of the casting need to be removed to allow clearance for the air bearing cartridge.
- Additional holes were added to accommodate the injection of epoxy and the fastening of the cartridge to the housing.
- A reference surface is added to accommodate gauging during assembly.

6.2 New Parts

Air Bearing Pads:

- Six different pads are used to support the tool carriage, each is slightly different.
- The top and bottom pads run the full length of the bearing cartridge. Different pad thicknesses are used to properly position the tool carriage with respect to the motor and sensors. These pads are also slightly smaller than the width of the tool carriage. This is to accommodate the proximity sensor tab and to avoid interference with the other pads at the corners.
- The left pads have slots cut in them to accommodate the motion of the encoder scale.
- The right pads are modified to produce lift properties similar to the opposing slotted pad.
Shell:

- Recesses are cut in the sides of the shell to accommodate sensors and for tool clearance.
- A 3/8" notch is cut along the length of the carriage to provide additional clearance for the optical scale.
- Epoxy injection holes are re-arranged to accommodate the slots cut in the bearing pads.
- Holes are added to the bottom of the shell for bolting it to the housing.
- Tapped holes are added to the bottom and a slight taper is ground in the side to facilitate removal of the shell from the housing.
- Two tapped holes are added which mate with the assembly jig.

6.3 Jigs and Fixtures

In addition to the part designs, several additional jigs and fixtures were designed to aid in the assembly of the final system. These jigs are described below.

Scale Jig:

- The scale jig is used to properly position the encoder scale on the tool carriage during assembly. This is required because the dowel pins were removed from the tool carriage.

Base Jig:

- This is used to accurately position the tool carriage with respect to the shell during replication.
- Reference surfaces and are provided so that the parallelism between parts can be measured with an indicator.
- Holes in base of jig mate with holes in flange of tool carriage.

Spacer Block:
• This is used during replication to set the depth of the bolts inserted into the replication holes. The bolts act as plugs, preventing the epoxy from flowing back out, but if they are set too deep, they may dislodge the bearing.

Bearing Plugs:

• These plugs are inserted into the slots of the left-side bearings to prevent epoxy from filling the slot.

Top Jig Plate:

• The jig plate is used to accurately position the tool carriage with respect to the housing during replication.
• A reference surface is provided to check the position and parallelism of the two components.
• Holes in plate mate with injection holes on shell and existing holes in housing.

6.4 Assembly Drawings and Instructions

The system is assembled in several discrete steps. First the bearing cartridge is assembled using the air bearing pads, tool carriage, shell and potting epoxy. Next, the lower housing assembly is manufactured by potting the bearing cartridge into the lower housing. Finally, the instrumentation is installed into the finished system.

Bearing Assembly:

• This drawing outlines the assembly of the bearing pads using the vacuum assembly method.

Base Jig Assembly:

• This drawing outlines the assembly of the bearing cartridge components on the base jig as well as the injection of the epoxy potting compound.

Bearing Assembly Section:

• This drawing is provided to clarity the design of the bearing cartridge.

Lower Housing Assembly:
• This drawing show the position of the bearing cartridge inside the modified lower housing. The jig plate used during assembly is also shown.

• Instructions for the assembly of this system are found on the drawing Lower Housing Assembly Instructions.

**Lower Housing Assembly Instructions:**

• This drawing contains step-by-step instructions for potting the bearing cartridge inside the lower housing.

• Figure 1 shows how the bearing cartridge is accurately fixtured and indicated to the Top Jig Plate.

• Figures 2 and 3 show how the bearing cartridge is potted into the lower housing and how the position can be checked with an indicator.

**Instrumentation Assembly and Instructions**

• This drawing describes the installation of the encoder scale and read head.

• The Scale Jig is used to position the encoder scale parallel tool carriage.

• A ground spacer block is added to allow adjustment of the encoder position.

**Bolt and Washer List**

• This contains a list of all the fasteners required for the assembly of the actuator.

As of this writing, the final design drawings and the results of the prototype tests have been submitted to the original machine tool manufacturer. They are in the process of manufacturing the system described above. When completed, this new system will be tested in a production environment.
7 Future Work

The test results presented above show that the air bearing systems designed and built using the vacuum-replication and zero-clearance methods have superior performance over the existing roller bearing system. This is sufficient justification to recommend applying this technology to the specific machine under investigation. However, the study is not yet strong enough to make broad recommendations concerning machine tools in general. Before this is possible, the final design must be built and tested in a production environment as planned. The system must be evaluated in terms of performance, manufacturability and durability.

The porous bearing system showed significant improvements in performance over the existing roller-bearing design. This is not unexpected, air bearings have long been used over roller-bearings in precision applications. The real advantage of this design is the ease of which it can be manufactured. What needs be shown is that this system can perform as well as or better than a conventional orifice air bearing system. If this is true then the vacuum-replication and zero-clearance design methods have the potential significantly reduce the cost of standard precision linear bearings without sacrificing performance. Therefore, tests between two similar air bearing systems must be run to compare the porous and orifice bearings.

Accurate dynamic models also need to be developed that allow the prediction of dynamic response, damping and pneumatic instability. A full analysis, involving fluid film analysis and structural deformation will be required to obtain accurate results. It is likely that a closed-form solution will not be achievable and therefore computational based models (i.e. finite element) will be required to achieve usable solutions.

In order for this technology to be accepted and used by machine designers and engineers, an elementary model must be developed that incorporates all of the significant design parameter. This would include operating pressure, bearing pad geometry and
structural geometry and material. Using these parameters, bearing gaps and stiffnesses could then be accurately approximated. Tools should also be developed to integrate this knowledge into a CAD library, permitting the integration of the air bearing into the analysis of the entire machine structure.

Finally, the technology and design methods need to be implemented in additional machine tool and instrument applications. This will allow additional test cases to be investigated and used as a means to develop and optimize the process and technology. It will also help to prove the concept and introduce it to the design and manufacturing community. Currently, a linear bearing is being developed for a printing application, as mentioned above and another is being considered for a high performance shaker application. Both of these projects can benefit greatly from this new technology.
8 Conclusions

A case study was presented that discussed the redesign of a linear tool servo system used for non-axisymmetric turning. The existing system was evaluated and various design options were considered. Based on this analysis, a prototype test system was designed using porous air bearings and assembled with a novel vacuum-replication method. The replication assembly method permitted the manufacture of bearings with no initial clearance, which resulted in improved performance. Tests comparing the existing bearings and the prototype air bearing design showed that this technology is superior and can be successfully implemented in a machine tool application. The porous air bearing systems built using zero-clearance design have been shown to have improved stiffness, straightness, damping and frictional properties over that of the original roller bearing system. The stiffness and straightness performance will result in improved machine accuracy. The improved damping and the elimination of stiction will simplify control requirements. In addition to the improved performance, the vacuum replication method greatly reduces the cost, manufacturing effort and part count of the bearing system. Based on these results, a retrofit system was designed to replace the existing bearings and is currently under construction.
Appendices

A1: Linear Motors
A2: Final Design Drawings
A1: Linear Motors

A1.1 Linear motor design issues

There are many criteria used to judge the quality and performance of a linear motion system. The evaluation process is complicated by the fact that it is often difficult to separate the motor from its support system. For example, all linear motors require additional bearings to maintain a constant air gap between the fixed 'stator' and moving 'rotor', and the performance of piezo-actuators depends mainly on the quality of the power source. However, the accuracy of the bearings and the quality of the power supply are often the critical components in determining the performance of the linear motion system. Despite these parameters, it is important to develop a list of criteria with which to evaluate high-speed, high-precision linear motion systems. The most relevant criteria are as follows:

Range of motion

The range of motion required for a linear motion systems can be the factor the determines which type of system is used. Generally, the different types of linear motors are confined to work in one of three ranges. Solid state actuators, such as PZT and Terfenol, have a very short stroke, generally less than 1 mm. Voice coils and stepper motors generally have an intermediate range of travel, up to 10 mm for voice coils to several meters with a stepper motor. The range of linear induction and synchronous motors is limited only by the length of the track that can be built. However, problems with cogging and construction make it very difficult to produce high precision over such a large range of travel. For these reasons high speed, high precision machines are generally confined to operate in the low to intermediate ranges of travel.

Accuracy, repeatability and resolution of motion

Because motors convert electrical energy to mechanical force, not mechanical displacement, accuracy and repeatability of a linear motion system usually depends on the controller and method of position feedback as well as the bearing system. However, some systems can simultaneously act as both a motor and position sensor and thus work 'open loop'. This is true for linear step motors as well as PZT and Terfenol crystals. Linear step motors achieve this by taking the product of the step count and the step size. Piezo systems utilize the linear relationship between strain and input voltage to track position. This has limited accuracy due to hysteresis and these systems must be frequently reset to account for drift.
Linear motor resolution is an important consideration, especially when considering step motors and some voice-coil motors. Resolution is the smallest step that a motor can accurately move. For step motors, this is determined by arrangement of the magnets and motor windings. All motion systems with mechanical contact bearings will have some limit of resolution due to stiction control problems and some motor designs may also experience problems with cogging, such as tubular synchronous motors. Piezoelectric and magnetostrictive actuators are generally limited by the resolution of the power supply electronics.

**Straightness of motion**

The straightness of motion of a linear motor is generally a function of the support bearings. Because most linear motors require some sort of support structure to maintain a uniform air gap, straightness is usually dependent on the bearing system used to separate the primary and secondary.

Straightness of solid state actuators is generally dependent on the orientation of the crystalline structure. However, motion from these actuators is generally transmitted through a kinematic mount, such as a wobble pin, and therefore, any straightness errors are generated in the support or bearing system. End effect forces in linear induction and flux-focusing voice coil motors also can contribute non-axial forces that will cause additional straightness errors.

Non-axial magnetic forces and variations in magnetic fields also contribute to straightness error motion. Periodic variations in magnetic attraction between the rotor and stator will occur as the motor moves between poles. This will change the bearing preload and thus generate a periodic straightness error. Voice coil motors

**Performance**

Because of the wide array of designs and configurations, several performance parameters need to be considered when choosing a linear motor. These include peak force, average, or RMS force, peak velocity, resolution and range of motion. It is also important to consider other design factors such as size, weight, cost and environmental requirements.

Peak force is the sum of all forces that the motor must overcome. These include the cutting force at the tool point, inertial load of the tool, motor and other moving components(bearings, linkages) as well as frictional and gravitational forces.

\[ F_{peak} = F_{load} + F_{friction} + F_{inertia} \]
In the application of a fast tool servo, all of these factors may be relevant. Therefore, it is important to try to minimize mass and frictional forces, and, if possible, minimize gravitational forces by using a horizontal configuration.

Most motor applications do not operate continuously at peak output. A typical motion cycle might consist of an acceleration at peak output, followed by a constant velocity step, deceleration and finally a period at standstill. The RMS (Root Mean Square) load value is a measure of the expected average continuous load on the motor. It is essentially a weighted average of the motor loads, using the load duration as the weights. Operating above the RMS limit may cause damage to the motor due to excess heat generation.

Linear velocity and acceleration requirements of the motor depend on the motion and force requirements of the system. Three common tasks are constant force, step positioning and controlled oscillation. Generally, these tasks are mutually exclusive and linear motors are optimized to perform one of these operations well. For example, positioning systems generally require high accelerations and high velocity but lower load force requirements, constant force applications are usually low velocity. Because it is very expensive, if not impossible to simultaneously have high speed, force and precision, high speed, high precision applications generally require very little force. Also, most applications have fairly short strokes, which implies large accelerations and moderate velocities.

Thermal performance of linear motors is usually a function of the applied current through the coils. In most motors, temperature is only a concern if it is high enough to damage the coil or lower performance. Heat from motors can cause unwanted thermal growth in the machine, which can be a significant problem in high precision machines.

### A1.2 Description of linear motor types

There are many different configurations of linear motors, all of which use one of the basic operating principles described above. Although there are many possible and useful designs, only a few of them are practical for high speed, high precision systems. These motors can be classified into one of three categories, based on the type and range of motion produced. Continuous range motors are capable of continuous motion over large range of travel. The length of travel is determined only by the size of the motor and support bearings, or track. Theoretically, all types of linear motors can be scaled to have an infinite range of travel, however, the motors in this category are those types that can be practically and economically produced. Limited range motors are those that are capable of short to moderate strokes (less than 1 mm to greater than 100 mm). These motors include voice coils for the shorter range (1 to 20 mm). Brushed and brushless, as well as synchronous motors for the longer ranges. The third type of 'motor' can be classified as short range,
these include the piezoelectric and magnetostrictive actuators. These have a very limited range of motion (less than 1 mm) and nanometer or better resolution.

Although most linear motors have conceptually similar rotary counterparts, the concept of linear rotor and stator is a bit ambiguous. In many linear systems it is unclear which side of the motor is fixed and which is moving, and to further complicate matter, the conventional arrangements of magnets and windings are often reversed in linear motors. For the sake of clarity, the motor components that are fixed to mechanical ground will be considered the stator, and those moving relative to the stator will be considered the rotor.

Continuous range motors

Both linear induction and linear synchronous motors can be constructed by conceptually 'unrolling' their rotary counterparts and extending either the rotor or the stator to the desired length of travel. Both of these types of motors exhibit a net normal magnetic force between the rotor and the stator. This force can be used as a bearing preload, or as the bearing itself in the form of magnetic levitation.

Linear induction motor

Linear induction motors (LIM), like their rotary consist of multi-phase windings, or primary and electrically conductive bars, often referred to as the secondary. LIM's can be configured with the stator as either the primary or the secondary. The least expensive of the two arrangements has a continuous track of bars as a stator with the primary windings moving perpendicular to the bars. The more common configuration, however has the windings fixed as the stator. This reduces the moving mass and eliminates the need for commutating the power to the rotor. In this case, the secondary can be built in to the design of the machine and generally consists of a strip of aluminum, brass or copper that is backed by a strip of iron, or the machine frame (if it is cast iron). To eliminate the losses due to end effects, the rotor must completely cover one phase the stator windings.

The normal attractive force between the rotor and stator varies with the speed of the rotor and can range from 0 to 3 times the thrust force. This variation in force will affect the accuracy and stiffness of the support bearing system (mechanical or magnetic) and therefore, a two sided design that counteracts these forces is best for precision applications.

It is also possible to build a tubular LIM that has a moving primary that is wound around an iron shaft which moves within a tubular secondary. This arrangement eliminates losses due to end effects and has a zero net normal force. The magnetic attraction between the rotor and stator is balanced by the cylindrical design, and the result is a simple
integrated magnetic bearing. This design has a limited length of travel due to the mechanical alignment required to maintain a reasonable air gap.

LIMs are capable of generating large forces, the output thrust is proportional to the square of the input voltage. However, the use of AC excitation in the primary makes them difficult to use with position or velocity control. Also, the heat generated by this motor is evenly distributed between the primary and secondary, requiring both to be thermally grounded or cooled. For these reasons, LIMs are generally used as force motors and in materials handling applications.

Linear synchronous motors

Linear synchronous motors (LSMs) consist of a multi-phase primary that similar to that of a LIM primary. The secondary is an arrangement of notched magnetic strips, which can either be permanent magnets, or have separate field windings. Either the primary or secondary can be used as the fixed stator, in either case the moving rotor will be fairly massive. For precision and high speed applications, the best design uses a permanent magnet rotor and a fixed primary. This will generate the least heat in the rotor and eliminates wiring between the rotor and ground. Figure A2.1 shows a single sided, moving primary LSM.

![Diagram of a Linear Synchronous Motor](image)

**Figure A2.1 Single Sided AC Synchronous Motor**

The speed and control of an LSM depends on the pole pitch of the rotor and stator and the frequency of the primary AC excitation. Because of this, precise position and velocity control require frequency control of the primary power supply, which is difficult and
expensive. One solution to this problem is a variation of the LSM called a Sawyer, or linear step motor, which is discussed later.

Linear synchronous motors suffer from many of the design problems as LIM, namely large moving mass and poor control qualities. Like the LIM they also have a varying attractive force between the primary and secondary. This force varies with the power angle of operation and is best dealt with by using a two sided design.

Because induction and synchronous motors use continuous frequency current, they are best suited for continuous, unidirectional operation. Accurate control of these types of motors requires, among other things, precise control of the input frequency, which is usually difficult and expensive. In addition, the moving member must either have permanent magnets, back iron or three-phase windings that are supported by an iron core. All of these are relatively massive and reduce the dynamic response.

Therefore, these motors are generally restricted to low to moderate speed, continuous operation applications, such as rail transportation and conveyor systems that require low precision.

It is also possible to produce a continuous track linear hysteresis motor by building a stator track of magnetically hard material which supports a three phase rotor above. The inverse of this (multiphase track with magnetic rotor) is also possible, but is considerably more expensive to build. The hysteresis design is similar to the linear induction design, except that the rotor is made of magnetically hard material, rather than softer iron. Hysteresis motors generally are not used in linear design due to their poor efficiency and because they do not present any strong advantages over linear synchronous or induction designs.

**Intermediate Range Motors**

Intermediate range motors are the most versatile class of linear motors. They are capable of high precision throughout a long stroke and over a large frequency range. There are several practical motor designs that fall into this category, all of which are DC motors, with the exception of the stepper motor.

**DC homopolar motors**

As the name implies, the windings of a homopolar motors share a common magnetic pole, which can be either the fixed stator, or the moving rotor. Although field windings can be used for this pole, it is generally generated using a permanent magnet. If the coil is fixed as the stator, the motor is called a moving magnet, and if the magnet is the stator, it is a voice coil. These motors are sometimes referred to as a non commutated DC motor.
A simplified version of the moving magnet actuator is the solenoid. Solenoids generally use a soft magnetic material, such as iron, as a moving member, rather than a permanent magnet. The solenoid coil generates a magnetic field that creates an attractive force on the iron. The cylindrical design equalizes the normal components and the net result is an axial force on the iron shaft. Figure A2.2 shows a typical solenoid actuator, although there are many variations in design.

![Diagram of Moving Magnet and Solenoid Actuator](image)

**Figure A2.2 Moving Magnet and Solenoid Actuator**

Due to end effects, the force on the shaft can vary greatly with position, depending on the design. However, it can be approximated by the following equation:

\[
F = \frac{N^2 i^2 A \mu}{2 h^2} \quad [N]
\]

where

- \(N\) = number of turns in coil
- \(i\) = current in coil (A)
- \(A\) = area of pole (m\(^2\))
- \(h\) = air gap (m)
- \(\mu\) = permeability of air (\(4\pi \times 10^{-7} \text{ N/A}^2\))

Solenoid actuators are only capable of generating force, and therefore are very difficult to control in position and velocity applications. Their application is generally limited to two point positioning and applying forces to mechanical stops, such as a preload for a kinematic coupling.
If the iron core of the solenoid is replaced with a magnetic material, the result is a position controllable homopolar moving magnet motor. The position is controlled by the current in the coil, a particular value of current corresponds to a specific magnetic equilibrium position between the coil and magnet. All homopolar motors (moving magnet and voice coil) have 'infinite' electrical resolution. The actual resolution may be limited by the quality of bearings and power supply.

The force generated by the solenoid motor can be calculated using the equation:

\[ F = B L i N \]

where \( B \) = the magnetic field of the moving member.

The moving magnet design is acceptable for low speed, low bandwidth applications. However, as the speed and force requirements increase, the strength and the mass of the moving magnet must also increase, thus limiting its response time. One advantage of this design is that the coils can be fixed to a thermal ground, thus reducing the thermal effects on the motors accuracy.

![Diagram of a voice coil actuator](image)

**Figure A2.3 Voice Coil Actuator**

The magnetic inverse of the moving magnet motor is the voice coil. The name voice coil is in reference to its common usage in audio speaker drivers. Moderately priced speaker systems are capable of dissipating over 100 watts of power at frequencies ranging from 10 Hz to 30 KHz. Therefore, a properly designed voice coil actuator should be expected to perform at least this well. The moving coil design minimizes the moving mass of the system and therefore maximizes the bandwidth. However, this design has poorer thermal performance due to the fact that the heat generating coil cannot be thermally grounded.
Because it uses the same operating principle as the solenoid motor, the force generated by the voice coil actuator can be found using the same equation, \( F = BLiN \). In both design, this equation holds in the absence of end effects, however, this assumption only holds true if either the magnet is considerably longer than the coil or the coil is longer than then magnet. When the coil's length exceeds the length of the magnet by at least the stroke length, the response is more linear, heat is dissipated better, and efficiency is increased, however, the increased mass and coil inductance decreases the system bandwidth. When the coil is shorter than the magnet assembly, the reverse is true.

Motor efficiency depends greatly on the losses due to end effects. Two effective methods of reducing these losses involve maximizing the flux lines near the operating point and minimizing the air gap.

![Iron Shell](image)

**Figure A2.4 Flux Focusing Voice Coil**

The flux focusing design shown in Figure A2.4 maximizes the force to weight ratio, and therefore the system bandwidth. This design increases the magnet surface area and volume without the corresponding increase in coil sizes. This focuses more flux through the air gap, which generates greater forces. This design, however, can only be used in very short stroke applications.

Motor efficiency can also be increased by minimizing the air gap between the coil and magnets. The smaller air gap reduces the losses due to end effects and allows the coil to run at a lower temperature. Maintaining a small air gap (.005 to .050 inches) becomes more difficult as the length of the stroke and coil increase. For this reason it is important to use a linear bearing support system that can maintain adequate straightness over the length of travel and have sufficient stiffness to support any transverse loads.

For limited range of motion applications, voice coil actuators are generally the best choice. In addition their high bandwidth, they have no mechanical hysteresis, torque ripple
or backlash. With a proper bearing support system, such as an air or hydrostatic bearing, voice coils can operate at high speeds with nanometer resolution over a range of many millimeters.

**DC motors**

DC linear motors (DCLM) are the most common and versatile configuration for high precision high speed systems. Like AC linear motors, most DCLMs have analogous rotary systems. There are three possible configurations for DC linear motors, which depend on the number of poles and the form of commutation. The single poled, or homopolar linear motors described above are the simplest of the three. Because they only have one pole, they do not require commutation. Multi-poled, or heteropolar motors require some sort of commutation to switch the current as the motor moves across the windings. This is achieved either electronically or by using mechanical brushes.

Brushed DCLM, like their rotary counterparts, consist of two or four phase windings on the primary and steel backed permanent magnets as the moving secondary. This is illustrated in Figure 16. Current is switched between the windings using mechanical brushes and a commutator that runs the length of the windings. The length of travel is unlimited by design, but may be constrained by the straightness of the linear bearings that are used. Because of the use of permanent magnets and a small air gap, there will be a significant (50 to 500+ lbs) attractive force between the primary and secondary. This force can be reduced significantly by using a double-sided configuration, or the force can also be used to preload the bearing system. The single sided design may suffer from large straightness errors due to variations in the magnetic field between poles and at the ends of travel. These motors are capable of high speeds and high forces, but the large inertial mass limits its acceleration and bandwidth. Table A1 shows the specifications of some typical brushed DC linear motors.
<table>
<thead>
<tr>
<th>Property</th>
<th>Unit</th>
<th>Value 1</th>
<th>Value 2</th>
<th>Value 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Force Constant</td>
<td>lbs/amp</td>
<td>0.9</td>
<td>3</td>
<td>6.1</td>
</tr>
<tr>
<td>Continuous Force</td>
<td>lb</td>
<td>4.2</td>
<td>15</td>
<td>31</td>
</tr>
<tr>
<td>Peak Force</td>
<td>lb</td>
<td>13</td>
<td>47</td>
<td>96</td>
</tr>
<tr>
<td>Inductance per Phase</td>
<td>mh</td>
<td>1.2</td>
<td>4.1</td>
<td>8.2</td>
</tr>
<tr>
<td>Back EMF</td>
<td>V/inch/sec</td>
<td>0.1</td>
<td>0.34</td>
<td>0.68</td>
</tr>
<tr>
<td>Electrical Time Constant</td>
<td>msec</td>
<td>1.2</td>
<td>2.2</td>
<td>2.7</td>
</tr>
<tr>
<td>Magnetic Attraction</td>
<td>lb</td>
<td>45</td>
<td>160</td>
<td>315</td>
</tr>
</tbody>
</table>

Table A1 Properties of Typical Brushed DC Linear Motors

Resolution and accuracy of brushed DC linear motors is limited only by the position encoder used for closed loop control, however, the mechanical contact between the brush and commutator may add unmodeled dynamics to the system, reducing the usable resolution.

The brushed DC motor does present an improvement over conventional linear motion systems (i.e. ballscrew or belt) due to its direct drive configuration that eliminates backlash and simplifies control. However, its applications in high speed, high precision machine tools is limited. The large inertial mass of the moving magnet secondary limit the speed and acceleration of the motor. Mechanical brushes present several problems, including errors due to intermittent contact, particle generation and additional maintenance. These motors are generally used in moderately accurate high speed applications such as pick and place equipment, water and laser cutters or part transfer systems. They are also used in high precision applications such as CMMs and positioning tables that do not require high speeds.

The commutation problem can be resolved by a position sensor and controller to electronically commutate the windings. The armature windings usually have two or four phases and are turned on successively by the controller as the carriage travels across the windings. Position information is read using a magnetic or optical scale, which is usually built in to the linear bearing system.

In order to achieve high resolution and accuracy, brushed DC linear motors require a linear position encoder for closed loop control. Since this position information is already available, it is possible to use it to eliminate the need for mechanical brushes and use electronic commutation instead. This is essentially what a brushless DCLM does, as shown in Figure 17. Brushless motors have the same power, speed and resolution as brushed motors, but can be built in several different configurations. A typical single sided
brushless DCLM has a multi-phase primary and an iron backed permanent magnet secondary. These motors are capable of generating large forces (20-500+ lbs) and have unlimited travel. Table A2 shows the specifications for some typical brushless DCLMs.

<table>
<thead>
<tr>
<th>Number of Poles</th>
<th>6</th>
<th>8</th>
<th>14</th>
</tr>
</thead>
<tbody>
<tr>
<td>Force Constant</td>
<td>lbs/amp</td>
<td>11.7</td>
<td>11.6</td>
</tr>
<tr>
<td>Continuous Force</td>
<td>lb</td>
<td>20</td>
<td>30</td>
</tr>
<tr>
<td>Peak Force</td>
<td>lb</td>
<td>60</td>
<td>90</td>
</tr>
<tr>
<td>Inductance per Phase</td>
<td>mh</td>
<td>59</td>
<td>47</td>
</tr>
<tr>
<td>Electrical Time Constant</td>
<td>msec</td>
<td>4.2</td>
<td>4.2</td>
</tr>
<tr>
<td>Moving mass</td>
<td>lb</td>
<td>4.5</td>
<td>5.5</td>
</tr>
<tr>
<td>Magnetic Attraction</td>
<td>lb</td>
<td>140</td>
<td>190</td>
</tr>
</tbody>
</table>

**Table A2  Properties of Single Sided Brushless DC Linear Motors**

Like the brushed motor, this design has high inertial mass and large magnetic attraction forces. An alternate design, which is more applicable for high speed, high precision applications, is the two-sided BDCLM. This configuration has the heavy magnets attached to a U-shaped primary. The secondary consists of a short moving coil, with a three phase winding, that is positioned in the air gap of the primary. This design eliminates the magnetic attraction between the primary and secondary and significantly reduces the moving mass of the system. This increases the bandwidth of the system and decreases the load on the support bearings. Like all DCLM, this design has unlimited length and the resolution is limited only by the resolution of the encoder and control system. A properly designed system can have micro inch resolution and accelerations of up to 10 G's. Table A3 shows that these motors are smaller and lighter, but are not capable of producing very large (less than 20 lbs) forces.
Number of Poles | 2 | 4 | 8
---|---|---|---
Force Constant | lbs/amp | 1.8 | 3.6 | 7
Continuous Force | lb | 2.5 | 10 | 20
Peak Force | lb | 7.5 | 30 | 60
Inductance per Phase | mh | 1.5 | 3 | 3
Back EMF | V/inch/sec | 0.18 | 0.36 | 0.7
Electrical Time Constant | msec | 0.2 | 0.2 | 0.3
Moving mass | lb | 0.4 | 0.7 | 1.6

Table A3  Properties of Double Sided Brushless DC Linear Motors

A third type of BDCLM is the linear step motor. It operates on the same reluctance principles as the rotary step motor. The moving primary has either two or four phase windings and travels along a transversely grooved secondary, or platen. Step motion is achieved by sending controlled pulses to the primary windings, each pulse will move it exactly one step, which is generally five to ten thousandths of an inch. More sophisticated controllers are able to control the motion between steps, yielding resolutions of about 5-10 microns. Air gap between the primary and platen is on the order of 0.001 inch and is maintained using either ball or air bearings. Because the step motor is capable of discrete motion, it can be operated open-loop, without concern for wear, hysteresis or backlash. The drawbacks of this type of motor include poor power efficiency, low force/weight ratio and long settling time between step. These factors, coupled with the relatively low resolution, limit the application of this type of motor to high speed, high precision machines. Another common implementation of this type of motor is the dual axis motor, which is capable of accurate X-Y motion over a large area. Commonly referred to as a Sawyer motor, the primary is modified to have windings that correspond to both axes, and the platen consists of an array of square teeth, rather than grooves. Air gap is generally maintained with an air bearing which is preloaded by the magnetic attraction.

This type of motor is common in light duty assembly applications, such as printed circuit boards and inspection systems, but does not yet have the speed or resolution to be compete with other high speed, high precision systems.

**Short range motors: PZT actuators**

PZT and magnetostrictive actuators are best suited for small excursions, usually less than 0.1 mm. A PZT crystal operating at a maximum of 500 μstrain would have to be 20
cm long to achieve this displacement. Larger motions can be achieved through mechanical amplification, but then some of the benefits of simplicity and fatigue resistance are lost.

Piezoelectrics are generally used for limited range of motion applications, such as alignment of optics, high frequency error compensation, and fast tool servos. This is due to the inherently low strain of the PZT materials, generally on the order of

Like ferro-magnetics, there are both 'hard' and 'soft' PZT materials. Soft PZT materials have typical displacements of 0.6 μm/V and require excitation voltages between 0 to 150 V. This results in a maximum displacement of about 90 microns. Actuators made from this 'soft' material generally are used in high load, low precision applications, such as electro-mechanical switches. Higher accuracy can be attained using 'hard' PZT materials, which have typical extensions of 0.2 μm/Volt. These materials exhibit better controllability and are capable of more accurate positioning.

Piezoelectric effect.

A piezoelectric material has a crystalline structure that produces a voltage at its surface when strained. When pressure is applied to the material along specific crystalline axis, a charge is generated in the material. This effect, shown in Figure A2.5, is also reversible, and the result is a mechanical strain, or expansion, of the crystal when it is subjected to a voltage across its surface.

![Simple Piezoelectric Crystal](image_url)

Figure A2.5 Simple Piezoelectric Crystal

To achieve this expansion, an electric field must be generated that is parallel to the direction of polarization of the crystal. The electric field creates a torque on dipoles in the crystal, causing them to rotate, which results in expansion of the crystal. Some
piezoelectric materials, called ferroelectrics, exhibit large hysteresis, in that the crystal will retain a portion of its strain after the electric field is removed. Generally, these materials are used in acoustic applications, where the hysteresis effects are not problematic. However, this nonlinear effect is undesirable in position applications, and may lead to control problems. Although all piezoelectric materials have some hysteresis, new materials have been developed that are designed specifically for motion applications that reduce this effect, however it is still significant. The most common of these is lead-zirconate-titanate, or PZT. This is a sintered ceramic who's composition can be specifically tuned to fit a particular application.

Piezoelectric materials have several properties which determine their performance. The expansion of a PZT crystal is described by its d-value, which is measured in meters/volt. It is a function of the exact composition of the crystals, and can vary between 2 - 600 x 10⁻¹². The change in length, ∆L is found using the equation:

$$\Delta L = E \cdot d \cdot L_0 + \frac{F}{c_t}$$

where:
- L is the length of the crystal
- E is the electric field strength
- F is the tensile load on the crystal
- c_t is the effective stiffness of the crystal

The PZT crystal will saturate at a certain field strength, and damage due to sparkovers will occur if too high of a voltage is applied (typically 2kV). This saturation limits the maximum expansion of the crystals, and generally, it is possible to achieve maximum strains (∆L/L_0) of 0.10 to 0.15%

Other physical constants that affect the performance of a piezo crystal are the piezostress constant, g, coupling constant k and dielectric constant K.

The piezoelectric stress constant relates the axial electric field to the stress applied in the radial direction. The piezoelectric coupling k is a measure of a crystal's ability to convert electrical energy to mechanical energy. This value is not a measure of efficiency, very little energy is lost in the crystal due to its high resistance. The coupling is better described as a transmission ratio, and generally has values ranging from 0.5 to 0.8. Much like a capacitor, piezoelectric crystals are able to store electrical energy, the difference is that piezomaterials store the electric potential in the form of mechanical strain. This ability to hold a charge is described by the dielectric constant K, which is used to find the electrical
capacitance of the crystal. Using this value, the piezo can then be accurately modeled as a capacitor in an electrical circuit.

The expansion of the PZT crystal is also affected by the applied load. Because of this, the response of the actuator depends on how the load varies with position. The simplest case is a constant load, such as one due to gravity or spring preload. This does not affect the piezoelectric expansion capacity, but it does shift the position of the positioning zero point. This length of this shift, $L_n$, is equal to $Fc_i$, which is just the mechanical strain due to compression, where $c$ is the compliance of the material. The upper limit to the static preload shift is determined by the depolarization effects, as well as the mechanical strength of the crystal.

Another common loading application is when the load is proportional to the displacement, such as the compression of a spring or against a rigid body. The effective expansion of the crystal is then a function of the opposing spring $c_s$ and the internal stiffness, $c_l$

$$\Delta L = L_0 \frac{c_l}{c_l + c_s}$$

where $L_0$ is the maximum change of length of an unrestricted crystal.

The maximum force which a crystal can produce is a function of the internal compliance $c_i$, and its expansion capacity. The maximum, or blocking force is achieved when the crystal is rigidly constrained and is simply the product of the maximum elongation $\Delta L$, and the internal stiffness $k_t$. ($1/c_t$)This value of force is only achievable with zero displacement.

A piezoelectric crystal can be modeled as a simple mass-spring system, and therefore will have resonant operating frequencies. As expected, the resonant frequency of an unconstrained crystal is simply $f_o = \frac{1}{2\pi} \sqrt{\frac{c_l}{m}}$, however, any mass that is coupled to the crystal will reduce this value.
Magnetostrictive effect.

Magnetostrictive materials are very similar to piezoelectrics, except for the fact that magnetostrictive materials are actuated with a magnetic, rather than an electric field. Low strain magnetostrictive materials have been available for the last 100 years, but only recently have new alloys been developed that are competitive with the PZT crystals. Terfenol-D is the newest of such magnetostrictive materials and is capable of strains that are an order of magnitude larger than PZT.

Mechanical strain in a Terfenol crystal is achieved by creating a properly aligned magnetic field around the crystal. This is best accomplished by using a simple coil wrapped around the cylindrical crystal. A return path for the flux can be provided with a soft iron core, as shown in Figure A2.6.

![Diagram of magnetostrictive actuator](image)

**Figure A2.6 Simplified Magnetostrictive Actuator**

In addition to having higher strains, Terfenol also has lower voltage requirements. A typical piezo system may require a 1KV voltage source to drive the actuator, a typical Terfenol actuator requires only a 250 - 500 V	ext{e} magnetic field, which is easily generated with a small coil and a few milliamps of current. Ideal Terfenol actuators are single crystals, however internal defects are always present and will lower the maximum strains achievable. Current processing techniques yield crystals capable of strains of 2000 micrometer/meter, which is about 75% of the theoretical maximum. However, the linear range of these crystals extends only through about 1000 ppm, above which, the crystal
saturates. In addition to saturation effects, Terfenol also experiences hysteresis effects and after the magnetic field is removed, some displacement will remain.

Like piezo materials, Terfenol will generate large forces when the ends are clamped. Because of the greater internal strains, the forces produced by a Terfenol crystal will be proportionally larger than those generated with a PZT material. Force calculations for Terfenol are similar to piezo crystals and can be found simply by using the stiffness and deflection of the material.

\[ F = E A \frac{\Delta L}{L_0} \]

where

- \( E \) = modulus of Terfenol \( (5 \times 10^6 \text{ psi}) \)
- \( A \) = area
- \( \Delta L / L \) = strain

and strain \((\Delta L / L)\) is a function of the input magnetic field.

The Magnetostrictive effect is essentially instantaneous, and therefore, the response time of a Terfenol crystal is limited only by the speed at which magnetic field can be changed. This speed is generally limited due to the inductance of the magnetizing coil, and therefore magnetostrictive materials will have a lower bandwidth than its piezo counterpart. For low force, small displacement systems, it is possible to achieve response times as short as 10\(\mu\)s. Like piezo systems, crystals can be tuned with spring preloads and mass loads to operate at frequencies in excess of 2KHz. Terfenol crystals can be damaged by overload, however, because they are single crystals, they do not experience fatigue. Except at high frequencies, the crystals do not generate heat during operation. This effect can be reduced through the use of laminations. Terfenol is somewhat temperature sensitive, however, and performance degrades linearly with temperature. Crystal composition can be tuned for a specific temperature range, and generally, performance is stable between 0 and 300 °C.

### A1.3 Selection of actuator for fast tool servo

There are many applications of linear motion systems in the machine tool and manufacturing industries. Applications such as CNC milling, pick and place assembly, and coordinate measuring machines require high precision as well as moderate lengths of travel. Design improvements on these machines generally focus on higher precision or higher throughput, meaning higher operating speeds.

A typical application for case study is a fast tool servo used in the turning of non-axisymmetric parts or to compensate for radial spindle errors. The tool servo actuates the cutting tool and its motion is coordinated with the motion of the spindle to produce the
desired profile on the workpiece. Increase in throughput requires higher spindle speeds, and therefore faster response of the tool servo.

This type of applications requires micron accuracy over a working stroke of about 0.5 mm. However, the tool must be retractable between parts to avoid interfering with the part changing robots and chuck. One possible design might involve using two stages, a PZT actuator for the fine motion and a solenoid or induction motor for the larger steps. However, the compounded error associated with dual stages will most likely be too great for the high accuracy requirements. A better design would involve the use of a homopolar voice coil motor. The voice coil is lightweight and has a good frequency response. It is also capable of producing the required accuracy and stroke length. In order to maintain an acceptable air gap between the coil and magnets, a stiff accurate bearing system must be developed, as discussed above.
A2: Final Design Drawings
NOTE: MANUFACTURE PART AS SPECIFIED IN DRAWING FS041873
WITH THE FOLLOWING MODIFICATIONS:
1. MACHINE PART AS SHOWN ON DRAWING EXCEPT:
   DO NOT DRILL HOLES 602 & 603
   DO NOT MACHINE BEARING GROOVE AS SHOWN IN VIEW A,
   DRAWING FS041873
2. MILL SIDE FACES NEAR THE CARRIAGE BASE FLANGE TO REMOVE
   FILLETS FROM GRINDING OPERATION SO THAT SURFACES ARE
   FLUSH OR UNDERCUT AS SHOWN IN "VIEW IN CIRCLE A"
   THIS OPERATION CAN BE PERFORMED BEFORE OR AFTER GRINDING
3. ANODIZE AND FINISH AS SPEC.

TOLERANCES
0.01 +/- 0.005
0.02 +/- 0.01
0.005 +/- 0.0005
0.0005 +/- 0.0005

MATERIAL: ALUMINUM
QUANTITY: ONE (1)

TOOL CARRIAGE
MODIFICATIONS

DRAWN BY: Michael Chu, Precision Eng. Research Group
Massachusetts Institute of Technology, Cambridge, MA

SCALE 1" = 1" DATE: 11-1-93
ASSEMBLY INSTRUCTIONS:
1. ATTACH BARS FITTINGS TO MANIFOLD AND BEARING PADS.
2. CONNECT PADS GRL.110 & GRL.111 WITH 3 3/4" TUBING.
3. CONNECT PADS GRL.113 & GRL.114 WITH 3 3/4" TUBING.
4. ARRANGE PADS ON TABLE SIDE BY SIDE AS SHOWN IN DRAWING.
5. CONNECT BEARING PADS IN PARALLEL TO MANIFOLD WITH 1/8" TUBING. EACH SIDE SHOULD BE INDIVIDUALLY SUPPLIED BY THE MANIFOLD.
6. CONNECT THE MANIFOLD TO 60 PSIG AIR. LIGHTLY SPRAY EACH PAD WITH ALCOHOL TO INSURE THAT THE PADS ARE OPERATING PROPERLY.
7. CONNECT MANIFOLD TO VACUUM SOURCE.
8. POSITION PADS ON TOOL CARRIAGE AS SHOWN. VACUUM SHOULD CAUSE PADS TO STICK TO SURFACE.

NOTE: HIDDEN LINES HAVE BEEN REMOVED FOR CLARITY TOBING AND MANIFOLD ARE NOT SHOWN.
FITTINGS ARE LOCATED IN CENTER OF BACKING PAD
ALL FITTINGS ARE #10-32NC

NOTE:
PURCHASE FROM
DEVITT MACHINERY CO.

NOTE: CUT SLOT WITH 5/16" END MILL THROUGH PAD
NOTE:
PURCHASE FROM
DEVITT MACHINERY CO.

BEARING MANIFOLD SHOULD BE DESIGNED TO
YIELD SIMILAR LIFT PROPERTIES AS OPPOSING
BEARING, GNL111B

FITTINGS ARE LOCATED IN CENTER OF BACKING PAD
ALL FITTINGS ARE #10-32NC

TOLERANCES
0.X +/- 0.05
0.0X +/- 0.01
0.00X +/- 0.005
0.000X +/- 0.0005

MATERIAL: STEEL, CARBON

QUANTITY: ONE (1)
AIR BEARING PAD (RIGHT FRNT)

DRAWN BY: Michael Chiu, Precision Eng. Research Group
Massachusetts Institute of Technology, Cambridge, MA

SCALE 1" : 1" DATE: 11-1-93
NOTE:
PURCHASE FROM DEVITT MACHINERY CO.

BEARING MANIFOLD SHOULD BE DESIGNED TO YIELD SIMILAR LIFT PROPERTIES AS OPPOSING BEARING, GNL110B

FITTINGS ARE LOCATED IN CENTER OF BACKING PAD
ALL FITTINGS ARE #10-32NC

TOLERANCES
0.X +/- 0.05
0.0X +/- 0.01
0.00X +/- 0.005
0.000X +/- 0.0005

MATERIAL: STEEL, CARBON
QUANTITY: ONE (1)
AIR BEARING PAD(RIGHT REAR)

DRAWN BY: Michael Chiu, Precision Eng. Research Group
Massachusetts Institute of Technology, Cambridge, MA

SCALE 1" : 1"  DATE: 11-1-93
Fittings are located in center of backing pad.
All fittings 10-32 NC

Steel side

Top view

End view

Side view

Graphite side

Note:
Purchase from Devitt Machinery Co.

Hole for air fitting

Material: Steel, Carbon

Tolerances
0.X +/- 0.05
0.0X +/- 0.01
0.00X +/- 0.005
0.000X +/- 0.0005

Quantity: One (1)

Air bearing pad (bottom)

Drawn by: Michael Chiu, Precision Eng. Research Group
Massachusetts Institute of Technology, Cambridge, MA

Scale 1" : 1" Date: 11-1-93
NOTE: CHAMFER ALL EDGES 1/16"
APPROX.

DRILL AND TAP 10-24 THRU

R. 5/32" APPROX.

TOLERANCE
0.X +/- 0.05
0.0X +/- 0.01
0.00X +/- 0.005
0.000X +/- 0.0005

MATERIAL: NYLON

QUANTITY: ONE (1)

BEARING PLUG, FRONT

DRAWN BY: Michael Chiu, Precision Eng. Research Group
Massachusetts Institute of Technology, Cambridge, MA

SCALE 2" : 1"  DATE: 11-1-93
NOTE: CHAMFER ALL EDGES 1/16" APPROX.

R. 5/32" APPROX.

DRILL THRU, TAP 10-24 1/2" DEEP

TOLERANCES
0.X +/- 0.05
0.0X +/- 0.01
0.00X +/- 0.005
0.000X +/- 0.0005

MATERIAL: NYLON

QUANTITY: ONE BEARING PLUG

DRAWN BY: Michael Chiu, Precision Eng. Research Group
Massachusetts Institute of Technology, Cambridge, MA

SCALE 2" : 1" DATE: 11-1-93
LOWER HOUSING ASSEMBLY INSTRUCTIONS. ALSO REFER TO DRAWING #GML300

1. Remove the bolts from the injection holes. Remove the bearing assembly from the base JG and the vacuum line from the manifold.
2. Apply 60 psi line pressure to the manifold.
3. Gently tap on the tool carriage to free it from the bearing pads. Remove carriage from assembly.
4. Remove bearing flange and modeling clay. Be sure that no epoxy has filled the slots in the bearing pads. If this has happened, remove excess epoxy with the drill, grinder.
5. Clean the tool carriage with a soft cloth and acetone. Remove any excess epoxy from the inside of the bearing assembly with a small chisel or the drill, tool. Use compressed air to clear the excess particles.
6. Reinsert the tool carriage into the bearing assembly.

Refer to Figure 1 below.

7. Mount the bearing assembly to the bottom of the JV plate with 1/4-20 screws. Use 1/4" washers as spacers or grind ends of bolts if necessary. Fasten screws finger tight, do not use wrench.
8. Check the position of the bearing assembly by indicating along the ground JV plate surface and the exterior of the bearing shell. Adjust until the TIR is less than 0.001" across the entire length of the bearing shell and tighten screws.
9. Spray three exposed surfaces of bearing shell and the threads of six 1/4-20 screws with hole release. Insert two of the screws into the threaded holes in the base of the bearing shell so that the threaded end of the screw is flush with the outer surface of the shell.
10. Use a putty knife and a flat surface to thoroughly mix the epoxy resin and hardener. Pour the epoxy into the stringer.

Refer to Figure 2 below.

11. Lay a generous bead of epoxy around each of the four holes in the lower housing casting and along the center, front and rear of the casting. See diagram.
12. Install the bearing assembly into the lower housing. Insert four bolts with washers through the holes in the bearing shell and into the threaded holes in the lower housing. Tighten them until they are finger tight, then back off 1/2 turn.
13. Insert the four #10-24 screws thru the JV plate, into the lower housing and finger tighten. Run indicator along ground edge of the casting and ground edge of the JV plate. Adjust position of plate until indicator reads less than 0.001" TIR, then tighten bolts.

Refer to Figure 3 below.

14. Thread the stringer into one of the injection holes on the lower housing. Slowly inject the epoxy into the hole by applying steady moderate force on the stringer plunger. Continue until epoxy is squeezed out between the shell and the casting.
15. Remove stringer and insert a 3/8-16 screw into the injection hole.
16. Repeat Step 11 for the other 3 holes. Allow system to dry overnight.

FIGURE 1

FIGURE 2

FIGURE 3

TOLERANCES
Material: USS
Scale 1" = 1'-0"
100-SERIES HOLE CHART

<table>
<thead>
<tr>
<th>INCH</th>
<th>HOLE</th>
<th>X (mm)</th>
<th>Y (mm)</th>
<th>DIA</th>
<th>DEPTH</th>
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<td>9.409</td>
<td>104</td>
<td>264.00</td>
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<td>0.201</td>
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<td>212.00</td>
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<tr>
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<td>108</td>
<td>385.00</td>
<td>108.00</td>
<td>0.201</td>
</tr>
<tr>
<td>10.394</td>
<td>3.189</td>
<td>111</td>
<td>264.00</td>
<td>81.00</td>
<td>0.201</td>
</tr>
</tbody>
</table>

REFERENCE FOR HOLES 104, 107, 108, 111
CORRESPONDS TO REFERENCE ON DRAWING
FS136004 (LOWER HOUSING), VIEW 100

NOTE: THIS PART MATES WITH DRAWING No N10002 (FS136004, LOWER HOUSING)

NOTE: USE 3/4" STOCK CAST IRON JIG PLATE

MATERIAL: IRON, STEEL OR ALUM.

0.X +/- 0.05
0.0X +/- 0.01
0.000X +/- 0.005
0.0000X +/- 0.0005

QUANTITY: ONE (1)

TOP JIG PLATE

DRAWN BY: Michael Chiu, Precision Eng. Research Group
Massachusetts Institute of Technology, Cambridge, MA

SCALE 1" : 1"  DATE: 11-1-93
NOTE: THE AIR SUPPLY TO THE BEARINGS PADS SHOULD REMAIN AT 60 PSI THROUGHOUT ALL OF THE FOLLOWING STEPS

1. MAKE MEASUREMENTS AS DESCRIBED IN STEP 1, C/W # 53065358, PART B. GRIND SPACERS AS SPEC. IN STEP 4B.
2. REMOVE BOLTS FIXING THE JG PLATE TO THE CASTING AND BEARING ASSEMBLY.
3. REMOVE JG PLATE.
4. REMOVE ANY EXCESS EPOXY WITH A SMALL CLEAVER OR THE DREMEL TOOL. CAREFULLY CLEAN THE INSIDE OF THE LOWER HOUSING WITH COMpressed AIR. CLEAN THE SURFACE OF THE TOOL CARRIAGE WITH ALCOHOL AND A SOFT RAG.
5. REMOVE TWO JACK SCREWS IN THE CENTER OF THE STEEL SHIELD. TIGHTEN THE 4 BOLTS AT THE BASE OF THE TOOL CARRIAGE.
6. COMPLETE STEP 4 FROM FN65358B, PART II.
7. ATTACH TAIL (FS041492) TO CARRIAGE (FS041875, GNL001)
8. MOUNT MOTOR COIL (FS011595-1) TO BACK OF CARRIAGE.
9. COMPLETE STEPS 7, 8 & 10 FROM FN65358B, PART II.
10. ATTACH GLASS SCALE (FS041867) TO TOOL CARRIAGE. POSITION CARRIAGE WITH SCALE JG (GNL401)
    AND TIGHTEN SCREWS. THE CARRIAGE CAN BE MOVED FORWARD OR BACK IF NECESSARY.
11. ATTACH BLOCK FS041880 TO LOWER HOUSING
12. MEASURE DISTANCE, R, BETWEEN THE BLOCK FS041879 AND THE CARRIAGE. THIS DISTANCE SHOULD BE 1.000 INCHES.
    IF REQUIRED, GRIND SURFACE "B" ON BLOCK FS041880 TO OBTAIN THIS DISTANCE.
13. TAKE MEASUREMENTS OF THE DISTANCE S AND T. THE DIFFERENCE BETWEEN THE READINGS SHOULD BE 0.045 INCHES.
    GRIND SURFACE "A" OF BLOCK FS041880 TO OBTAIN THIS DISTANCE.
14. REMOVE THE GLASS SCALE FROM THE CARRIAGE.
16. INSTALL PROXIMITY SWITCH (FS011536) ON BRACKET (FS041878). IF NECESSARY, HOESE SECTION BE REMOVED AND REPLACED WITH A LONGER SECTION TO AVOID INTERFERENCE.
17. COMPLETE ASSEMBLY ACCORDING TO FN65358B, STEPS 22-23 & 28-34.

FIGURE 1 SCALE JIG IN POSITION WITH GLASS SCALE

FIGURE 2
<table>
<thead>
<tr>
<th>Part</th>
<th>Qty</th>
<th>Size</th>
<th>Length</th>
<th>Type</th>
<th>Descr. of Purpose</th>
</tr>
</thead>
<tbody>
<tr>
<td>Top Jig Plate</td>
<td>4</td>
<td>10-24</td>
<td>1 1/4</td>
<td>Socket Cap Screw</td>
<td>Bolt plate to casting</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>1/4-20</td>
<td>5/8</td>
<td>Socket Cap Screw</td>
<td>Bolt plate to shell</td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>1/4-20</td>
<td>Washers</td>
<td>Lt-Duty</td>
<td>Spacers for above</td>
</tr>
<tr>
<td>Shell</td>
<td>4</td>
<td>1/4-20</td>
<td>5/8</td>
<td>Socket Cap Screw</td>
<td>Bolt Shell to casting</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>1/4-20</td>
<td>Washers</td>
<td>Lt-Duty</td>
<td>For above</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>1/4-20</td>
<td>5/8</td>
<td>Socket Cap Screw</td>
<td>Jack screws to remove shell</td>
</tr>
<tr>
<td></td>
<td>11</td>
<td>1/4-20</td>
<td>3/4</td>
<td>Socket Cap Screw</td>
<td>Plug screws for shell injection holes</td>
</tr>
<tr>
<td>Lower Housing</td>
<td>4</td>
<td>3/8-16</td>
<td>3/4</td>
<td>Socket Cap Screw</td>
<td>Plug screws for housing injection holes</td>
</tr>
<tr>
<td>Bearing Plugs</td>
<td>2</td>
<td>10-24</td>
<td>1/2</td>
<td>Socket Cap Screw</td>
<td>Screws for removing bearing plugs</td>
</tr>
<tr>
<td>Base Jig</td>
<td>1</td>
<td>10-24</td>
<td>1/2</td>
<td>Socket Cap Screw</td>
<td>Screw for clamping shell to jig</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>10-24</td>
<td>3/8</td>
<td>Socket Cap Screw</td>
<td>Screw for clamping shell to jig</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>4-40</td>
<td>3/4</td>
<td>Socket Cap Screw</td>
<td>Screws for attaching carriage to jig</td>
</tr>
</tbody>
</table>
Bibliography

Boothroyd, G., Dewhurst P.  Product Design for Assembly.  Boothroyd Dewhurst Inc., Wakefield RI.


Lawrence Livermore National Laboratory Report UCRL-53643.
