

# Chapter 1

## DESIGNING THE AXTRUSION

The Axtrusion is a new linear motion concept developed by Professor Alexander Slocum and Roger Cortesi of the Massachusetts Institute of Technology's Mechanical Engineering Department.

It is intended for applications where the emphasis is on high speed, no wear, and very low error motions. It is designed to enable air bearing systems to be competitive in price with high performance ball bearings systems. At the same time, it exploits all the advantages of a non-contact motion system.

Envisioned applications include:

- Precision high speed material handling
- Machining and turning centers
- Optical equipment.

The functional requirements of the Axtrusion system are:

- **No Contact:** No contact between the way and the carriage allows for very high speed operation and NO wear. The elimination of grease will reduce machine downtime and mess. Non-contact is also the primary means of reducing error motion.
- **Moderate Stiffness:** The exact stiffness requirement will be determined by the specific application.
- **Thermal Robustness:** Many linear guide systems are very sensitive to large changes in temperature due to the very tight tolerances between the parts. The Axtrusion should be insensitive to temperature changes.
- **Minimal Precision Surfaces and Parts:** The geometry should remain simple, and minimal precision parts should be used to keep the manufacturing inexpensive and easy.

## 1.1 Axtrusion Components

1. The Way: This is the base of the Axtrusion. It is the surface on which the carriage slides back and forth.
2. The Top and Side Precision Surfaces: These are the two critical surfaces of the way. The air bearings slide over these two surfaces, hence the necessity for higher tolerances on straightness and surface finish.
3. Linear Motor Permanent Magnet Track: This component is bolted to the angled groove in the top surface.
4. Position Encoder Scale: This component is a piece of tape that allows the position encoder read head to measure the carriage's position.

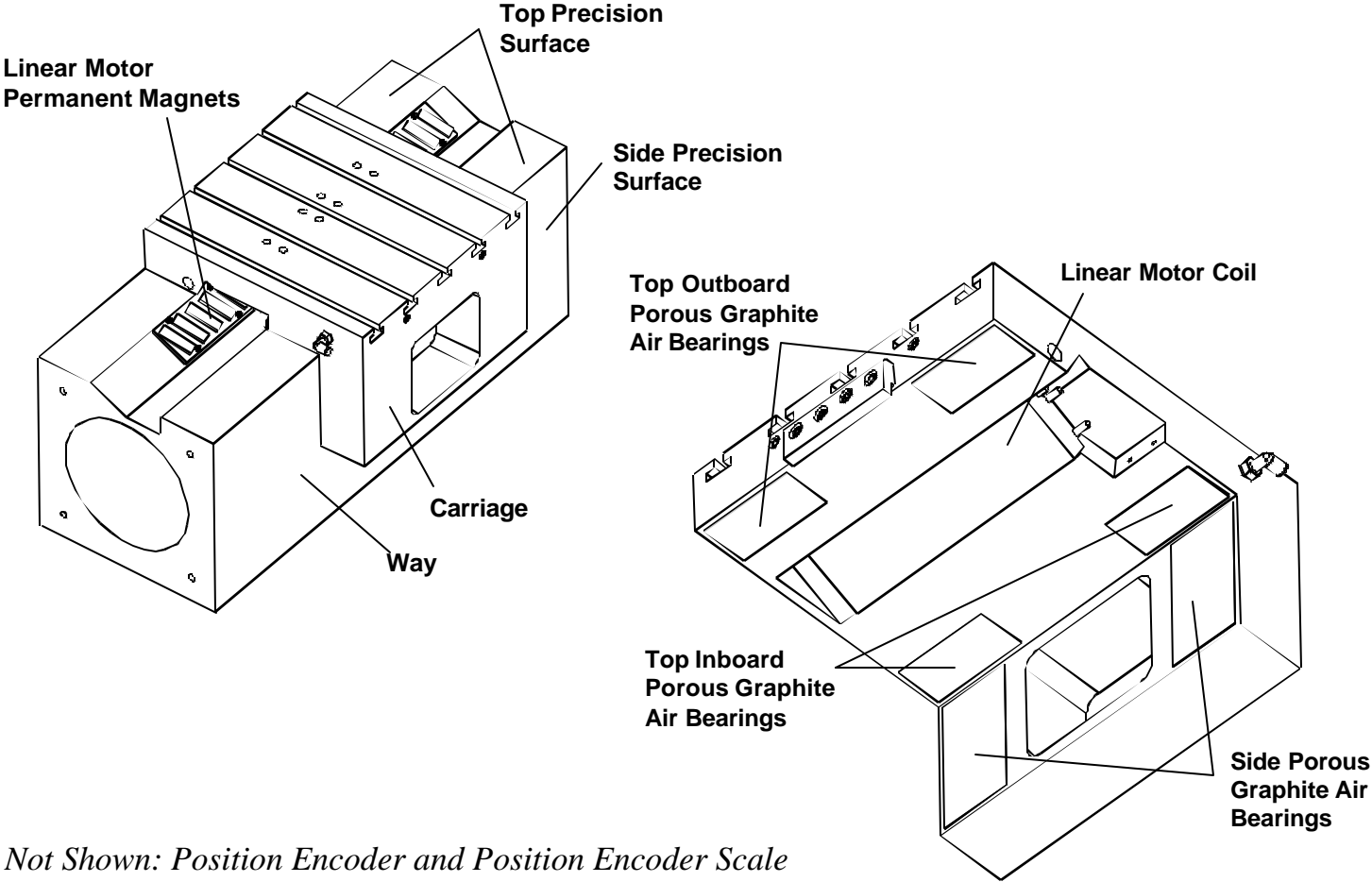
The carriage is made up of a number of smaller components.

1. The Carriage Base: This is the structure to which the other carriage components are attached.
2. Porous Graphite Air Bearings: These six (6) bearings allow the carriage to slide with no friction over the way surfaces. They also determine the maximum load capacity of the carriage.
3. The Linear Motor Coil: Since this component is essentially a large block of iron with coils of

wire inside, it is very strongly attracted to the magnet track on the way. This attractive force preloads the air bearings. When the coils are energized by the controller, the motor coil provides the motive force to move the carriage.

4. Position Encoder Read Head: This is what senses the position of the carriage and transmits it to the motor controller.

# Axtrusion Components



*Not Shown: Position Encoder and Position Encoder Scale*

## 1.2 How the Axtrusion Works

The attractive force between the motor coil and the magnet track preloads the bearings, which support the vertical and horizontal loads. By choosing the groove angle ( $\mathbf{q}$ ) and motor location ( $y_m, z_m$ ) the designer can specify the amount of preload on each bearing pair. The pairs are the top outboard pair, the top inboard pair and the side pair.

Summing the forces in the horizontal direction yields,

$$F_{side} = \frac{-F_m \sin \theta}{2}, \quad (1.1)$$

where  $F_{side}$  is the preload forces on each of the side bearings,  $F_m$  is the motor attractive force, and  $\mathbf{q}$  is the motor angle. Summing the forces in the vertical direction yields,

$$F_{top1} + F_{top2} = \frac{-F_m \cos \theta}{2}, \quad (1.2)$$

where  $F_{top1}$  and  $F_{top2}$  are the forces on each of the inboard and outboard pairs of top bearings. Summing the moments yields,

$$F_{side}z - F_{top1}y_1 - F_{top2}y_2 = \frac{F_m}{2}(\cos(\theta)y_m - \sin(\theta)z_m), \quad (1.3)$$

where  $y_m$  and  $z_m$  are the motor location in the horizontal and vertical directions. In the prototype configuration the vertical motor location is determined by

$$z_m = w_m \sin \theta, \quad (1.4)$$

where  $w_m$  is the width of the motor track.

Equations 1.1 through 1.3 have not taken into account the 20 kg (44 lbs) mass of the carriage. It is not significant compared to the magnetic preload force. If the mass of the carriage is significant with respect to the linear motor attractive force, then it must be included in the calculations.

Equations 1.1 through 1.3 can be solved as a linear system for  $F_{side}, F_{top1}, F_{top2}$ , yielding

$$\begin{bmatrix} -2 & 0 & 0 \\ 0 & -2 & -2 \\ 2y & -2x_1 & -2x_2 \end{bmatrix}^{-1} \begin{bmatrix} F_m \sin \theta \\ F_m \cos \theta \\ F_m (\cos(\theta)y_m + \sin(\theta)z_m) \end{bmatrix} = \begin{bmatrix} F_{side} \\ F_{top1} \\ F_{top2} \end{bmatrix}. \quad (1.5)$$

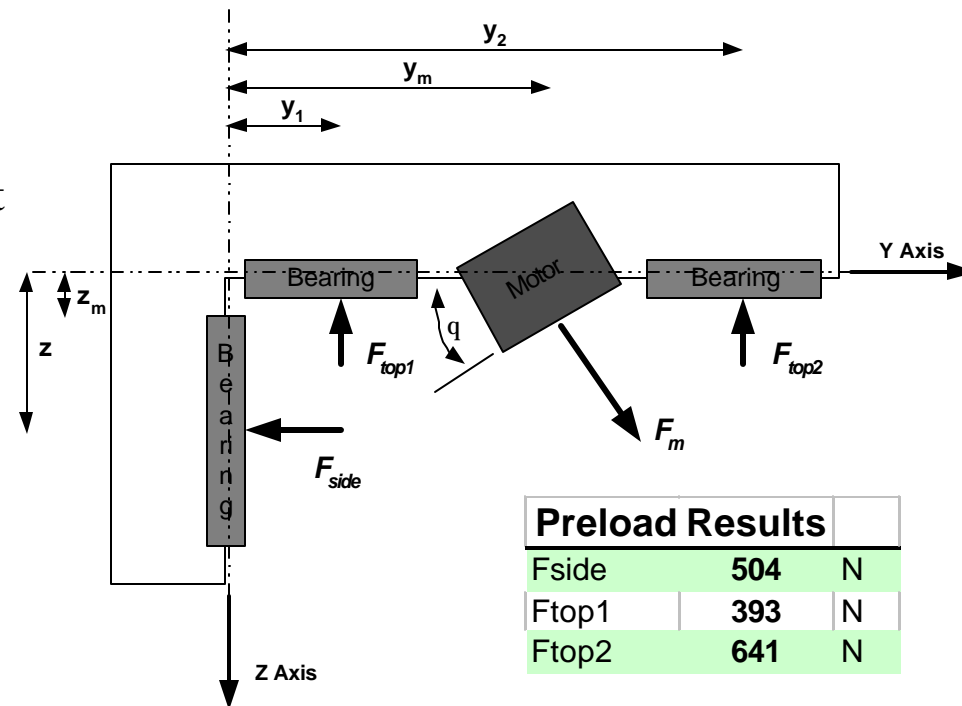
The values for  $F_{side}, F_{top1}, F_{top2}$  will determine the preload, and hence air gap and stiffness, of the three pairs of bearings.

# How the Axtrusion™ Works

The attractive force between the motor coil and magnets preload the air bearings.

Changing the values of  $q$ ,  $y_m$ , and  $z_m$  the values for  $F_{side}$ ,  $F_{top1}$ , and  $F_{top2}$  can all be set independently

Prototype Parameters		
Theta	26	degrees
Fm	2300	N
Y1	30	mm
Y2	260	mm
Ym	145	mm
Z	81	mm
Zm	25	mm



Preload Results		
Fside	504	N
Ftop1	393	N
Ftop2	641	N

### 1.3 The Bench Level Prototype

The bench level prototype (BLP) of the Axtrusion uses five cam rollers as the bearings: three (3) on the top surface and two (2) on the side surface.

The gap between the magnet and the way can be adjusted using the magnet adjustment screw. This adjusts the amount of attractive force preloading the bearings.

This BLP demonstrates the viability of the Axtrusion concept, and justifies further development.

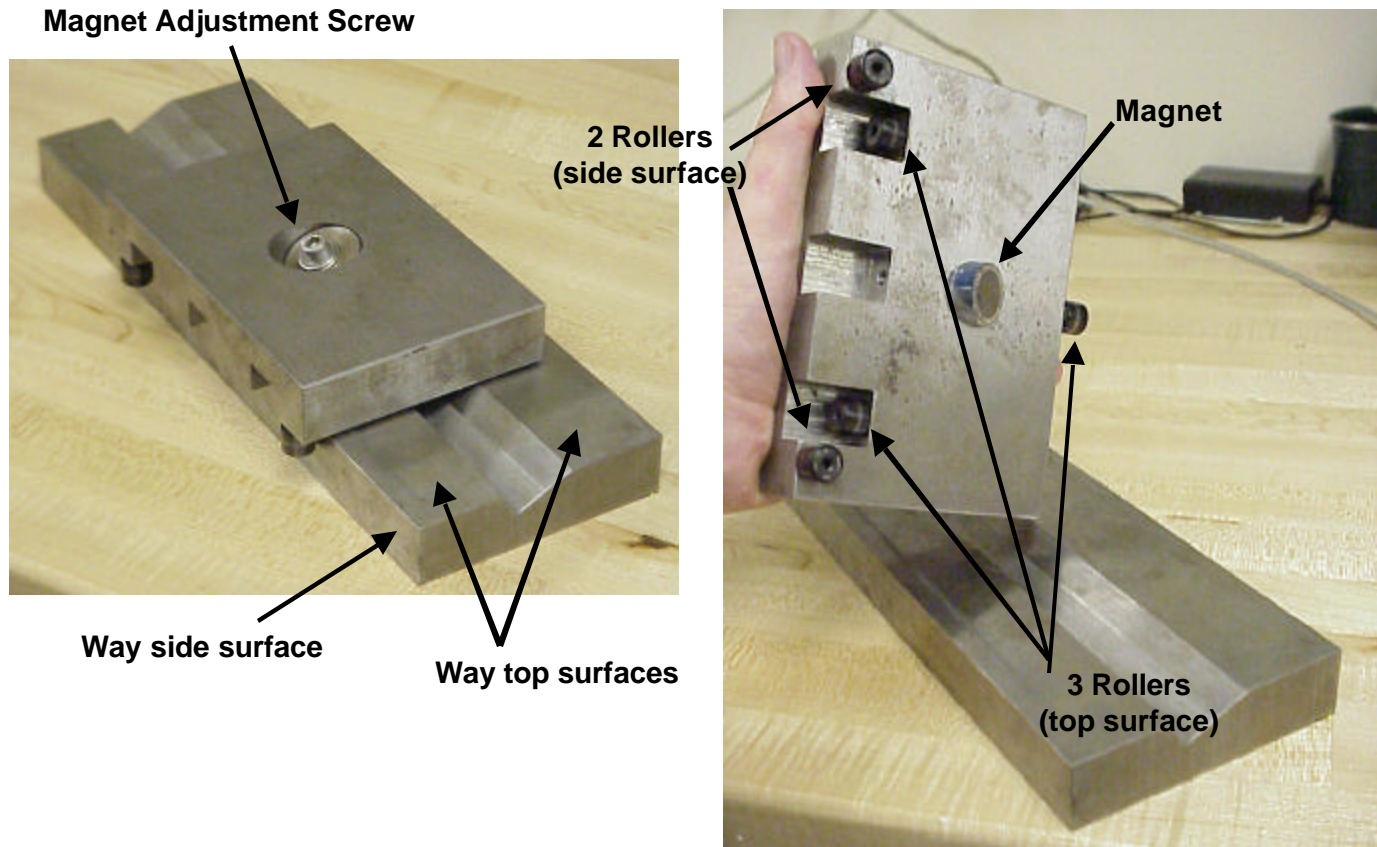
The five (5) rolling element bearings are a good setup to demonstrate the Axtrusion concept. However, they leave a lot to be desired for precision applications, including:

1. The five point contacts will make the carriage extremely sensitive to surface roughness and way straightness. Any dirt on the way will affect the motion of the carriage as the rollers roll over it.
2. The load capacity is limited by the hertz contact stress between the roller and the way. If the stress is too high, the way will be damaged, especially under impact loads.
3. The roundness of each roller directly affects the motion of the way.

These problems can be reduced by using more rollers, but this increases the complexity of the system and its sensitivity to dirt remains.

Other bearings considered for use in the Axtrusion are discussed in Section 1.4.

# The Bench Level Prototype



*The first Axtusion ever built! It uses 5 CAM rollers as bearings*

## 1.4 Bearing Selection

The functional requirements for the bearing system are:

- Easy to mount in the carriage assembly
- Robust with respect to dirt and surface scratches
- Robust when the carriage is “unpowered”
- Independent from extensive support equipment

### 1.4.1 Rolling Elements

The problems with rolling element bearings are covered in Section 1.3.

### 1.4.2 Hydrostatic Bearings

Hydrostatic bearings would provide a very stiff non-contact bearing system for the carriage. However, the fluid pumping systems are expensive and the prototype would be messy. Hydrostatic bearings would be worth considering if the application was submerged in a fluid environment.

### 1.4.3 Orifice Air Bearings

Orifice Air Bearings require a very smooth underside of the carriage to maintain the small air gap needed to support the carriage. This requires precision machining of the carriage. If the way surface is scratched, the scratch could “short” the bearing as the orifice passes over it.

### 1.4.4 Porous Graphite Air Bearings

Porous graphite air bearings were selected for the following reasons:

1. Clean: No fluid captive systems are required.
2. Non-Contact: None of the friction or wear associated with rolling element bearings is experienced.
3. Self-contained: Unlike the orifice bearings, the porous graphite bearings will function regardless of the structure that holds them. This allows them to be replicated in place, which greatly simplifies the manufacturing process.
4. Large Discharge Area: This allows bearings to travel over scratches and craters without shorting.

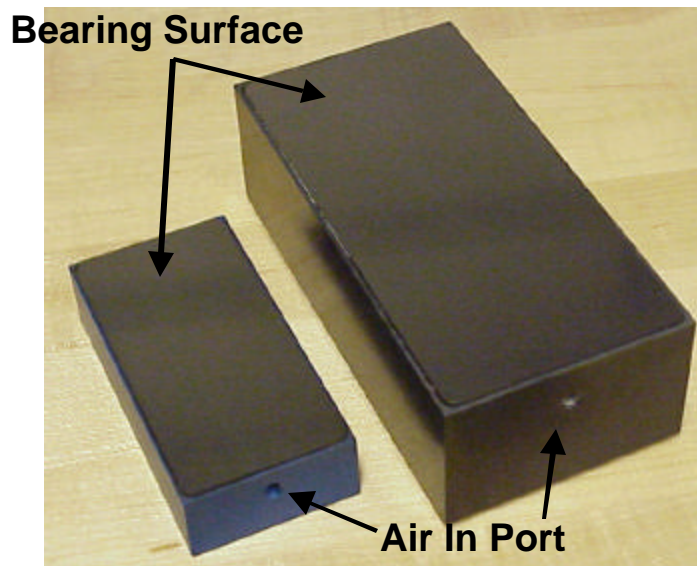


# Bearing Selection

Bearing Types	Design Considerations			Additional Equipment	Mess
	Non-Contact	Stiffness	Sensitivity to way surface		
<b>Porous Graphite Air</b>	no contact	ok	ok	some	none
<b>Orifice Air</b>	no contact	ok	poor	some	none
<b>Rolling Element</b>	contact	good	good	none to minimal	none
<b>Hydrostatic</b>	no contact	excellent	poor	a lot	a lot

Modular Air Bearings (**porous graphite bearings**) help make the Axtusion a cost effective **high performance** linear motion system.

*Left: Newway™ Porous graphite air bearings in the two sizes used in the Axtusion Prototype.  
[Http://www.newwaybearings.com/](http://www.newwaybearings.com/)*



## 1.5 Way Surface Selection

The functional requirements for the way material are:

- No upward divots when the way is cratered
- Nonferrous
- Easy to finish to the needed tolerances

### 1.5.1 Granite

Granite has some of the best divot properties. If cracked, the granite will fracture completely downward. This allows the air bearings to continue to slide over the fracture unimpeded. In single piece quantities, the granite base for the prototype cost \$2800.

### 1.5.2 Polymer Concrete

Polymer Concrete cratering resistance is inferior to granite's, but it is better than metal's. A wood prototype mold for the base costs about \$3800, and makes 3 to 4 parts. A steel production mold costs about \$12,500, and makes 600 to 800 parts. It might be possible to cast the parts to the required surface finish, thereby eliminating the need for additional grinding. The price per casting is \$1000. Polymer

concrete is an area that should be researched further for mass production of both the way and the carriage.

### 1.5.3 Metal

There is a wide range of metals available for the way, including steel, cast iron, and aluminum. All of these will have an upward divot when they are cratered, and many of these are ferrous. A ferrous way will become magnetized over time by the permanent magnets. This will prevent the air bearings from blowing metal particles aside.

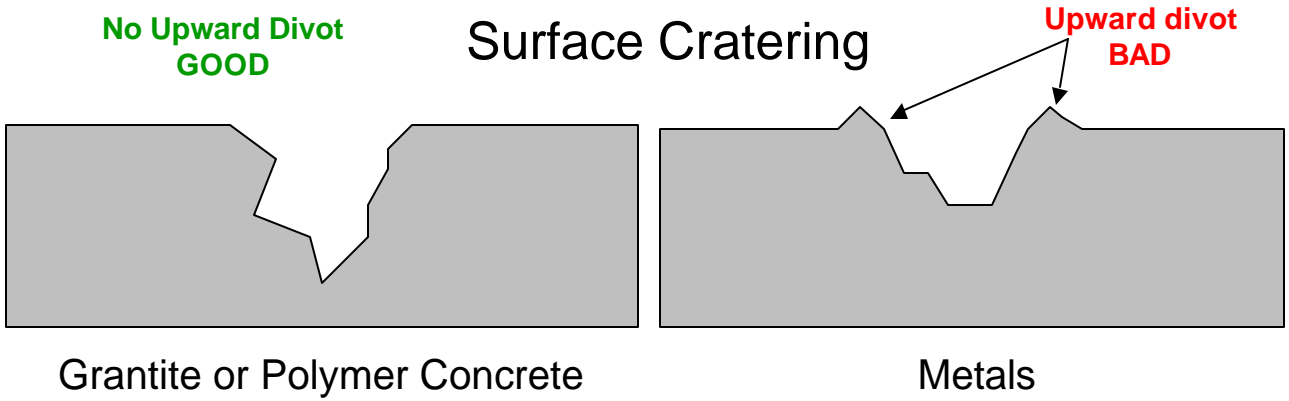
One of the most promising ways to get a light weight low cost way is to extrude the Axtusion profile from aluminum, grid the top and side precision surfaces to the correct shape, and finally hard anodize them.

### 1.5.4 Aluminum Oxide

Because the geometry of the way is very simple, it could be cold pressed in aluminum oxide. This would provide a way that is 4 times stiffer than aluminum, and whose surface is virtually indestructible.

# Way Surface Selection

Material	Design Considerations			
	Divot Properties	Mass Producible	Stiffness	Process
Granite	Excellent	Medium	OK	Cutting/Grinding
Polymer Concrete	Good	Excellent	Poor	Casting/Grinding
Aluminum	Poor	Excellent	OK	Extrusion/Grinding
Aluminum Oxide	Are there any?	Medium	Excellent	Cold Pressing



## 1.6 Motive Power Selection

The motive power functional requirements are:

- Minimal contact between moving elements to minimize wear and noise in the motion.
- Robust installation and operation
- Ability to provide the preload force for air bearings.

### 1.6.1 Linear Motors

Open face permanent magnet linear motors meet all the motive power functional requirements. There is no contact between the motor coil and magnet tracks so there is no wear. The strong attractive force between the coil and magnet track provides the force to preload the bearings. The tolerances for the alignment of the motor coil and magnet track are much looser than that of a gear drive or ballscrew system.

Currently linear motors are designed to minimize the attractive force between the coil and the magnets. In conventional rolling element linear bearing systems, this high attractive force causes increased wear. In the Axtrusion, the higher

attractive force is a benefit, because the higher preload force improves air bearing stiffness. Whether linear motors would become more compact and/or higher performance if their design was no-longer constrained by the need to minimize their attractive force? This is a topic for further study.

### 1.6.2 Ball Screw

It is possible to design an Axtrusion that uses a ball screw for motive power. This would increase the number of moving parts. A magnet track would still be required to preload the air bearings. And mounting the ballscrew in the vicinity of the magnets and motors presents some tricky design issues. This choice of motive power increases the error and repeatability of the carriage motion.

### 1.6.3 Belt Drive

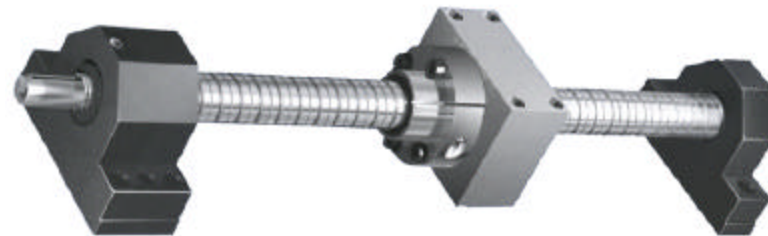
If stiffness in the direction of motion is not a major concern, a belt drive can be used to move the Axtrusion. This is considerably easier to mount than the ball screw option, and is the least expensive per unit length of the three options. However, it is also the least precise of the three.

# Motive Power Selection

Motors	Design Considerations			
	Contact	Scalable	Stiffness	Wear
<b>Linear Motor</b>	None	Few Discrete Sizes	Excellent	None
<b>Ball Screw</b>	Many	Excellent	Excellent	Most
<b>Belt Drive</b>	Some	Excellent	poor	Yes
<b>Rocket Motor</b>	None	Poor	N/A	Yes



*An open face linear motor*



*Typical ballscrew assembly*

## 1.7 Sizing the Carriage (Load Capacity)

The design parameters that determine the carriage load capacity are:

- The maximum load capacity of the bearings is determined by the minimum allowable gap height between the bearings and way.
- The working load and preload applied to each bearing must not exceed its maximum load.

The maximum load capacity of the air bearings is determined by their surface area and their air pressure. The bearings reach their maximum capacity when the gaps decrease to about 3-4 microns.

The carriage working load capacity in the horizontal direction is determined by

$$2L_{bmaxside} - F_m \sin \theta = L_{cmah}, \quad (1.6)$$

where  $L_{bmaxside}$  is the maximum load each of the side bearings can support,  $F_m \sin \theta$  is the preload component in the direction in question, and  $L_{cmah}$  is the maximum load capacity of the carriage in the horizontal direction. The carriage load capacity in the vertical direction is estimated by

$$4L_{bmaxtop} - F_m \cos \theta = L_{cmav}, \quad (1.7)$$

where  $L_{bmaxtop}$  is the maximum load each of the top bearings can support, and  $L_{cmav}$  is the maximum load capacity of the carriage in the vertical direction.

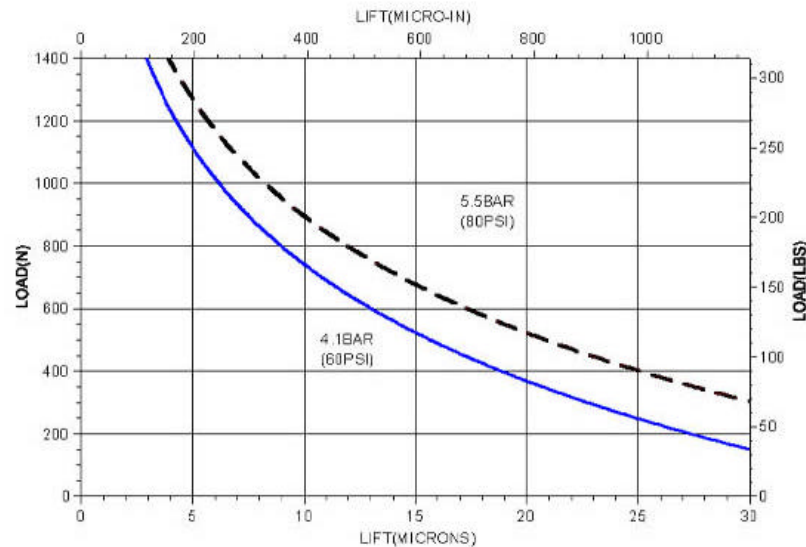
If the mass of the carriage is significant with respect to the preloading or anticipated forces then it should be included in equation 1.6 or 1.7.

NOTE: The values used for  $L_{bmaxside}$  and  $L_{bmaxtop}$  are determined by the minimum allowed gap between the air bearing and the way. The estimates in the prototype assume that the minimum gap is 3-4 microns. If this gap is too small then one should use the load vs. gap charts supplied by the bearing vendor to calculate the correct values for  $L_{bmaxside}$  and  $L_{bmaxtop}$ .

# Carriage Load Capacity

Carriage Load Capacity is determined by:

- Air Pressure
- Bearing Surface Area
- Minimum Tolerated Gap
- The Amount of Preload



*This plot shows the load vs. gap for a single 50x100 mm Air Bearing*

Prototype Axtusion Carriage Load Capacity							
	n	Max Bearing Load (N)	Total (N)	Preload (N)	Angle	Preload Componet (N)	Carriage Capacity (N)
Vertical	4	1400	5600	2300	26	2067	<b>3533</b>
Horizontal	2	3000	6000	2300	26	1008	<b>4992</b>

*These values assume that the carriage can run with an air gap of 3 to 4 microns*

## 1.8 Sizing the Carriage (Roll and Normal Stiffness)

The design parameters that determine the rotational and the normal stiffness of the carriage are:

- The stiffness of the individual air bearings
- The number of air bearings
- The bearing's distance from the center of stiffness of the carriage.

Carriage stiffness is a function of the stiffness of each individual air bearings and the distance between them. We are interested in stiffness in two directions normal to the direction of travel (horizontal and vertical), and the roll, yaw, and pitch stiffness.

### 1.8.1 Stiffness of the Individual Bearing Pads

The stiffness of an individual bearing pad is the derivative of the load capacity vs. the gap thickness function for each bearings. The stiffness of the bearings is approximately

$$K_{50,x100} = -0.0258h^3 + 1.489h^2 - 29.196h + 223.35 \quad (1.8)$$

or

$$K_{75,x150} = -0.158h^3 + 7.543h^2 - 122.644h + 786.51, \quad (1.9)$$

where  $h$  is the gap height [microns] and  $K$  is the stiffness [Newtons per micron]. SectionA.1 gives the full derivation.

### 1.8.2 Stiffness Normal to the Direction of Travel

The stiffness normal to the direction of travel is simply the sum of the individual stiffnesses of each bearing in the direction in question. Given a load in a particular direction. Equations 1.8 and 1.9 can be used to calculate the stiffness of the individual bearings.

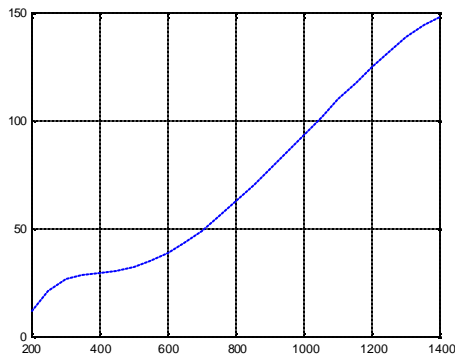
### 1.8.3 Rotational Stiffness

There are three axes of rotation that the stiffness needs to be computed about. The rotational stiffness of the carriage is proportional to the distance between the bearings squared. Therefore increasing the distance between the bearings pads can increase the roll stiffness dramatically. See SectionA.1 on page 113 for more detailed information on the theoretical stiffness of the carriage. See Section1.22 on page57 for the measured stiffness data and its implications.



# Carriage Stiffness

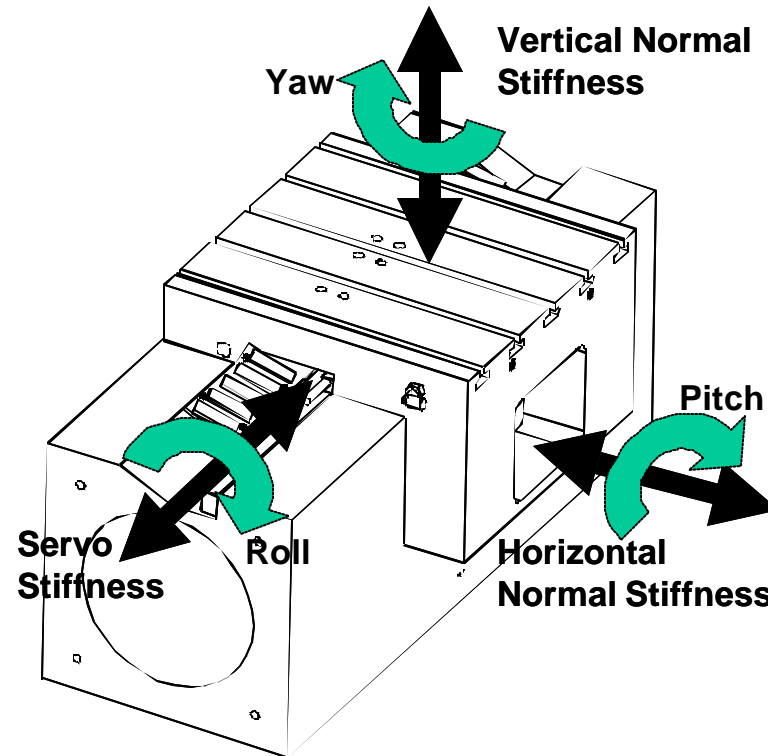
Calculate the bearing load vs. stiffness curves for the individual bearings



From the preload values calculate the stiffness of each bearing pair.

Enter the individual stiffnesses in the carriage compliance model (see appendix)

Calculate the displacement at other points



## 1.9 Casting the Carriage Base

The functional requirements of the carriage base are:

- To provide a ridged mount for the linear motor and air bearings.
- Light weight to allow rapid accelerations.
- Nonferrous, so the carriage does not become magnetized by the magnet track over time.

The rough shape of the carriage for the prototype was cast in Magnesium AZ91-T6 alloy. Magnesium's density is 60% of Aluminum's and is about half the Young's Modulus. The drawings for the prototype casing are in Chapter3. The pattern for the casting costs \$2300 and each casting costs \$600.

The cross section of the carriage had to be large enough to provide plenty of stiffness despite the low Young's Modulus of the Magnesium.

The carriage casting was then cleaned up and detailed features were added on a milling machine. The drawings for the carriage machining are in Chapter3.

Other options for manufacturing the casting base considered include:

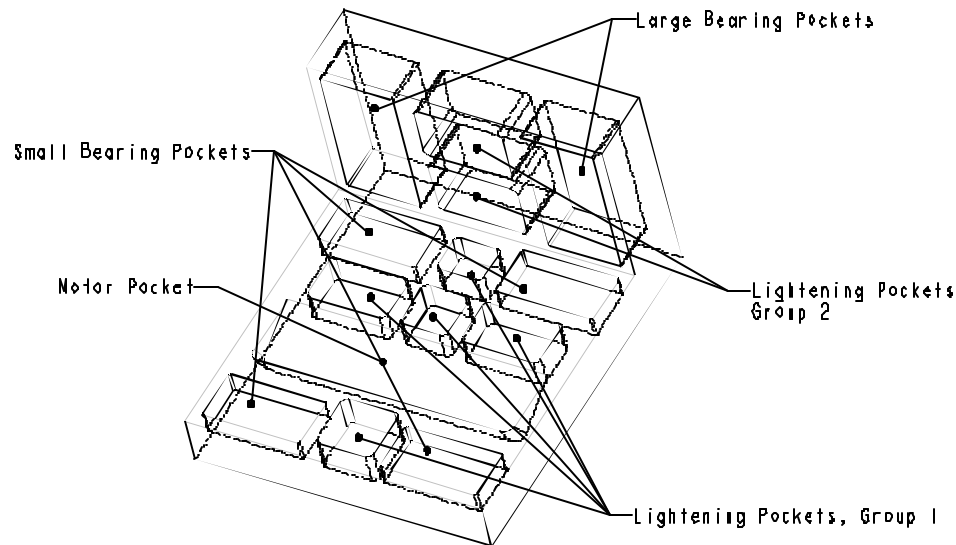
1. Hogging the base out of a solid metal block. This method was discarded because it offers no time saving when making multiple copies of the carriage.
2. Machining the top and side pieces out of separate pieces of metal and then welding or bolting them together. Again this method offered no time savings when making multiple copies.

It is important for the carriage to be made of a nonferrous material. A ferrous carriage is more difficult to assemble. Additionally, the strong permanent magnets on the way will magnetize a ferrous carriage over time.

# Manufacturing the Carriage Base

(the casting)

It is important that the carriage base be made of a stiff **nonferrous** material



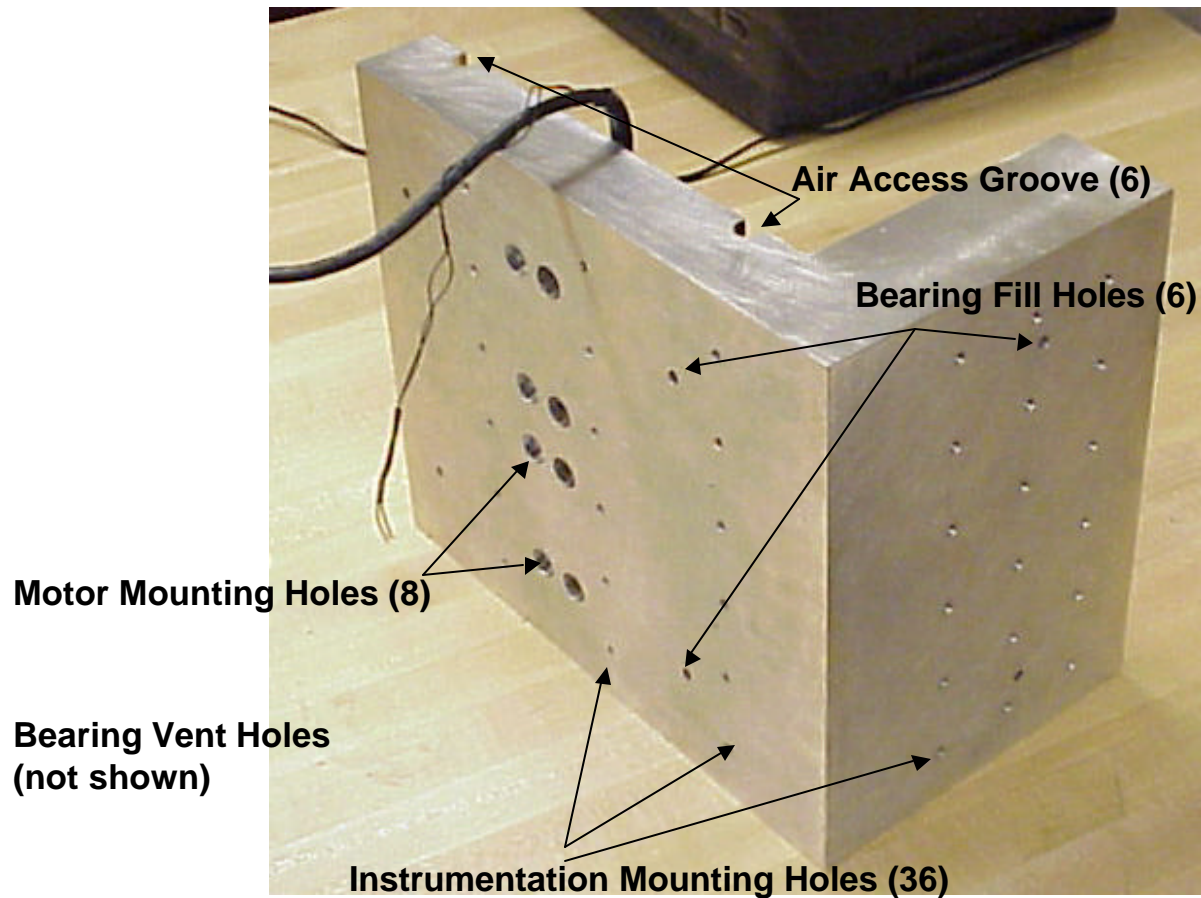
## 1.10 Machining the Carriage Base

The carriage base casting had more features machined into it. These features include:

1. Mounting holes on the linear motor.
2. Access grooves for the air fittings for each of the air bearings.
3. Fill and Vent holes for replicating the air bearings in place.
4. Threaded holes on the two outside faces for mounting test instrumentation.

# Manufacturing the Carriage Base

(the machining)



## 1.11 The Carriage Fixturing

The functional requirements for the carriage fixturing are

- Support the carriage near the bearing points during bearing replication.
- Align the carriage with respect to the way
- Be able to remove the carriage when the full preload force is applied.
- Allow access to the fill holes and air lines

Because the preload forces on the carriage are high the carriage will deflect. If the bearings are replicated in place without the preload force, the carriage will deflect and move the bearings slightly out of alignment when the preload force is applied. This will reduce their effectiveness. The solution is to support the carriage near the bearing points, while the bearings are replicated with the full preload force in place.

The fixturing will also determine how parallel the carriage top and sides are with respect to the way. This parallelism was not a major concern in the prototype Axtrusion, so the prototype fixturing was not very precise.

The prototype fixturing consists of the following: A top plate that is bolted to the top surface of the carriage; Top blocks which bolt to the top plate to support the carriage in the vertical direction; And a pair of side “L” blocks to support the carriage in the horizontal direction. The height of the top blocks and L block determine the gap between the back of the bearing and the carriage.

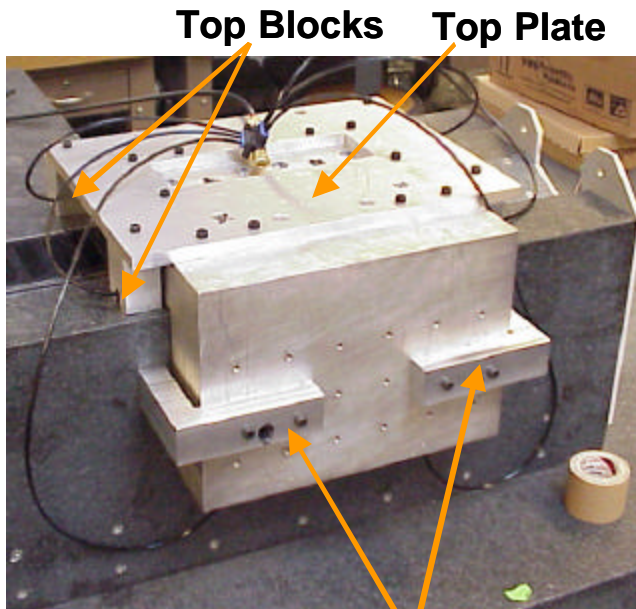
After the bearings have been replicated in place the fixturing is needed to remove the carriage from the way under the full preload force. Five (5) M6 screws are used to raise the carriage off the way in both the vertical and horizontal directions. It is important to lift the carriage off both way surfaces to prevent the air bearings from getting scratched. As the carriage is raised the magnetic preload force drops by the distance squared, so it is not difficult to get the preload to a manageable level.

The next generation carriage should be designed to take advantage of standard fixturing (i.e. a 1-2-3 block or parallels) during replication. This would allow the carriage to be aligned with greater accuracy in less time.

# Carriage Fixturing

Replicating Fixturing Features:

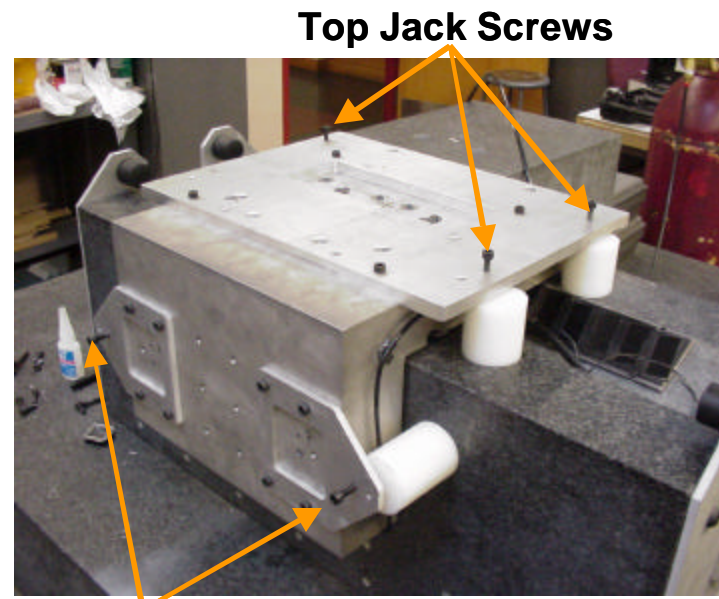
- “Correct” Stress Distribution
- Square’s Carriage with respect to the Way



**Top Blocks**   **Top Plate**  
**Side “L” Blocks**  
*Replicating Fixturing*

Removal Fixturing Features:

- Removes Carriage vertically and horizontally to prevent scratch of the bearings



**Top Jack Screws**  
**Side Jack Screws**  
*Removal Fixturing*

## 1.12 Replicating the Bearing Pads to the Carriage Base

Appendix Section A.2 contains all the details available on replicating the bearings into the carriage. The important points are summarized here. This procedure took 2 hours 15 minutes during the prototype assembly. It is expected that it will get faster as the carriage is improved. An outline of the procedure is listed below.

Thoroughly clean and degrease all surfaces that will come in contact with epoxy. Make sure the surface of the way and bearings are completely free from particles. These particles could damage the bearings if they are caught between the bearings and way.

Draw a vacuum through the bearings when they are positioned correctly on the way to ensure that they remain in contact and aligned with the way as the epoxy cures.

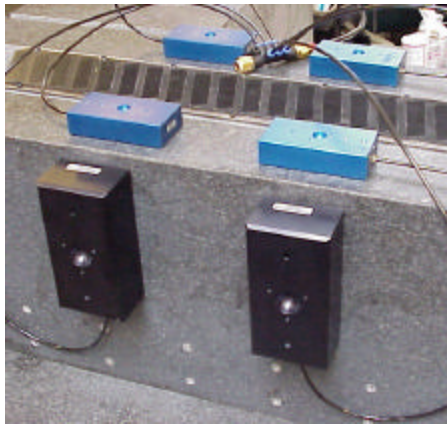
A means of lowering the motor coil to the correct air gap must be provided. At full preload force (500+ lbs) the carriage assembly will be difficult to handle by hand. In the prototype this is accomplished by drawing the motor coil

completely up into its pocket. Once the carriage is in place and aligned, the coil is lowered to the correct air gap (measured using a piece of non-ferrous shim stock), thus applying the full preload to the carriage. Then the bearings are replicated in place, and then space behind the motor is filled with epoxy giving the motor a secure mounting face.

It is also very important that one calculates an estimate for the amount of epoxy that should be injected into each pocket. Overfilling the pockets can lead to epoxy leaking around the bearings and gluing the carriage to the way. The estimate for the amount for each pocket is obtained by multiplying the surface area of the bearing face by the gap between the pocket and back of the bearing. It highly suggested the hat distance between the bearing and the pocket be measured as a double check to any estimate from CAD models.



## Replicating the Bearing Pads into the Carriage



**1**

*Place Bearings on the way. Draw a vacuum through them to temporarily secure them in place.*



**2**

*Lower motor coil to the correct air gap. This applies the full preload force to the carriage.*



**3**

*Inject epoxy into bearing and motor pockets. Calculate the amount of epoxy needed before injection to prevent gluing the carriage to the way!!*

Please see the text for a more detailed description of the bearing replication process.

### **1.13 Assembly Lessons Learned**

The assembly of the prototype carriage went well. However, there are quite a few improvements needed for the next generation.

Rather than having one fill hole per bearing pocket there should be two. This will allow the epoxy to cover more of the back surface of the bearing. The hemispherical feature on the back of the bearing would not need to be covered because they will no longer be under the injection holes. Furthermore, the bearing vent holes can be eliminated.

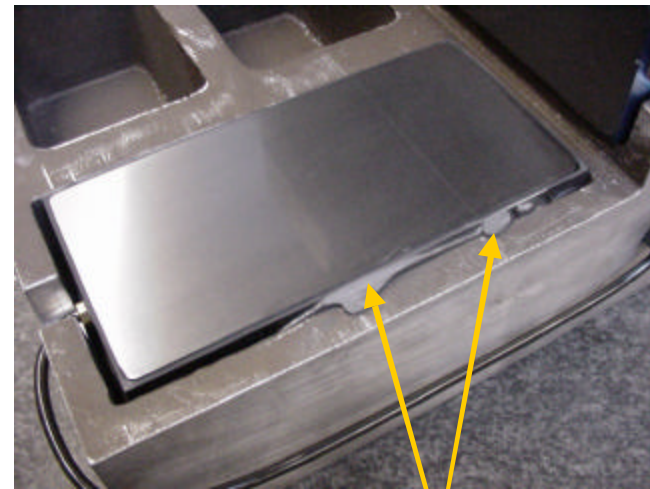
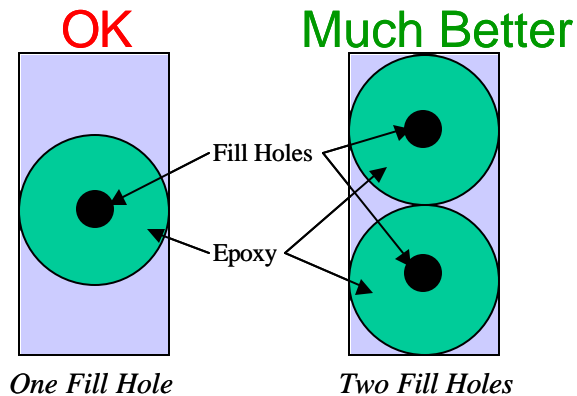
The distance from top and side faces of the carriage from the way surface in its final position should be the height of some standard form of fixturing (1-2-3 block for example). This will make it less expensive and easier to square the carriage with respect to the way without custom made fixturing.

Once the motor and bearings have been replicated into the carriage, removing carriage without mechanical assistance is impossible. In the prototype the alignment fixturing was modified to allow the carriage to be removed. It may be

desirable to have the removal features built into the carriage in the future.

DO NOT overfill one of the bearing pockets and glue the carriage to the way. However, should this occur it can be corrected by removing the carriage and filing down the place where the epoxy made contact with way.

# Assembly Lessons Learned



***DON'T GLUE THE CARRIAGE TO THE WAY! It had to be filed down to allow the carriage to slide.***

- Center Fixturing Holes on Bearings
- Two Fill Holes Per Bearing
- No Vent Holes Needed
- Design Carriage to use 1-2-3 Blocks for Fixturing
- Build removal features into carriage base
- DO NOT overfill the one of the bearing pockets and glue the carriage to the way.

## 1.14 Modal Analysis Setup

The first performance analysis of the prototype was a modal analysis. The data was collected using a three axis accelerometer and a Hewlett Packard 35670 Dynamic Signal Analyzer. The data was analyzed in the Star Modal software package. The impact hammer was a PCB model 086C03, and the accelerometer was a PCB model 356A08 three axis accelerometer.

First the carriage was removed from the base and the accelerometer was glued to the way. The vibration modes were found for the way on the table. This allows one to differentiate between modes in the carriage and those in the way/table combination.

The carriage was replaced on the way, and eight (8) points were marked on the surface. Point 1 was the excitation point. This is where the carriage was struck with a hammer for all tests. Points 2 through 8 were measurement points. The accelerometer was glued to each point in turn while the carriage was excited.

Measurements were made when the carriage was both floating and not floating.

The data was then imported into the Star Modal software. A stick figure model of the carriage was made where each vertex is one of the measurement points.

The data was analyzed by finding the frequency, damping percent, magnitude, and the dynamic compliance of prominent vibration modes. The stick figure carriage model was animated to show the vibration modes.

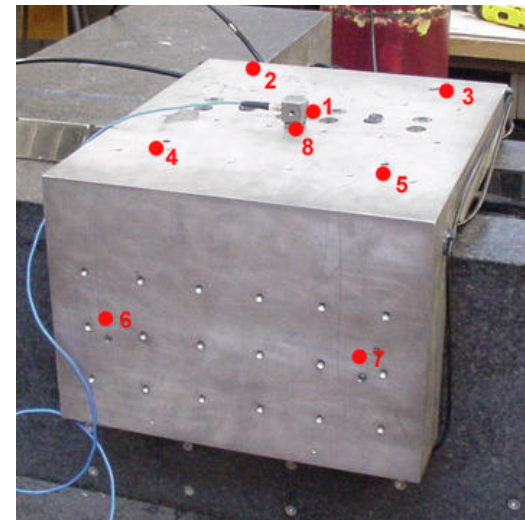
# The Modal Analysis Setup

Measurement were taken for:

- Way Only
- Carriage Floating
- Carriage Not Floating

Point 1: Impact point  
 Points 2-8: Measurement Points

Data Measured with 3  
 Axis Accelerometer



## 1.15 Modal Analysis Results

Several interesting modes were found in the Axtrusion. These are summarized in the tables below.

**TABLE 1.1** Carriage Floating Modes

Frequency [Hz]	Damping [%]	Magnitude [Output/ Input]	Comments
362	3.8	48	Top left front air bearing oscillates much more than others
608	3.3	33	Carriage deformation is like a “hinge” and top center vibrates a lot
487	1.9	5	Carriage deformation is like a “hinge” and top center vibrates less

**TABLE 1.2** Carriage Not Floating Modes

Frequency [Hz]	Damping [%]	Magnitude [Output/ Input]	Comments
1430	0.6	300	Top center of carriage oscillates up and down
501	1.3	2.3	Whole carriage moves up and down

Some important results of the modal analysis are:

In almost all the dominant modes the top center of the carriage vibrates in the Z axis much more than the rest of the carriage. In the prototype this mode occurred at 607 Hz while the carriage was floating, and at 1430 Hz while it was not.

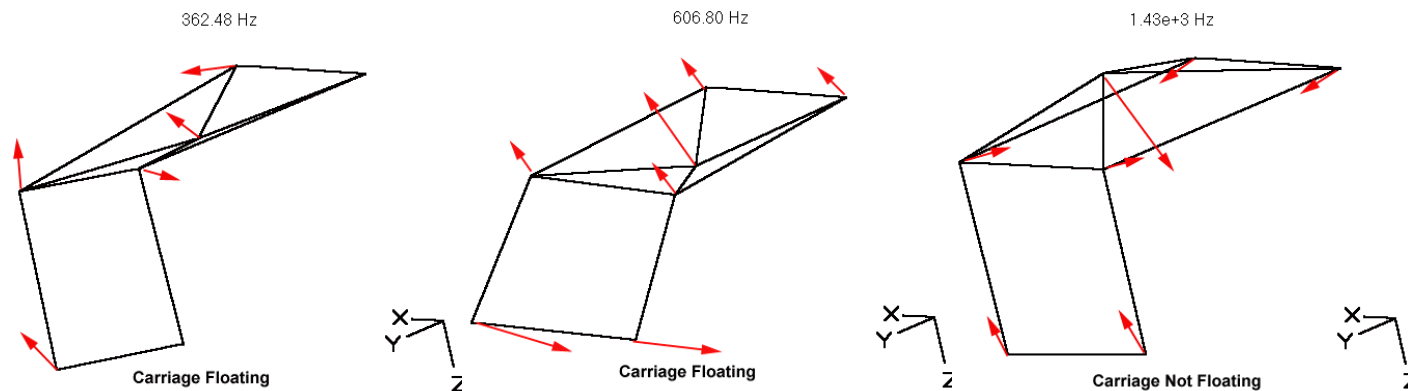
The mode at 362 Hz in the floating prototype is very interesting too. In this mode the top front left corner of the carriage is oscillating in the Z axis much more than the rest of the carriage. This suggests that air bearing in this corner is much less stiff than the other three top bearings. This reduced stiffness could be caused by uneven preloading. It is expected that the inboard top bearings will be preloaded less (therefore less stiff) than the outboard pair. See the Vertical Stiffness Section 1.22 on page 57 for explanation of what might be causing the left or right sides of the carriage to be preloaded unevenly.

The modal analysis also shows the effect on the angled preload by the linear motor. Even though the carriage was excited only in the Z direction, the carriage oscillates normal to the angled motor track.

# The Modal Analysis Results

The modal analysis provided the following information:

- Resonant frequencies of the carriage.
- Which bearings are not preloaded equally
- Good and poor locations for mounting sensitive equipment.



*See text or web page for table of modes.*

Go to the Axtusion web site <http://pergatory.mit.edu/cortesi/index.htm> for the modal animations.

## 1.16 The Dynamic Stiffness

The modal data was also used to calculate a dynamic stiffness of the carriage in the floating and not floating configurations. The dynamic stiffness was calculated for the top center of the carriage (point #8 when the model data was taken).

The raw data from the accelerometer is in volts. This must be converted to an acceleration by multiplying it by an appropriate conversion factor for the accelerometer used. Then the data must then be divided by the conversion factor for the impact hammer. Now the data has units of  $\frac{Ns^2}{m}$ . Dividing the data by the frequency (in Hz) squared yields stiffness as a function of frequency.

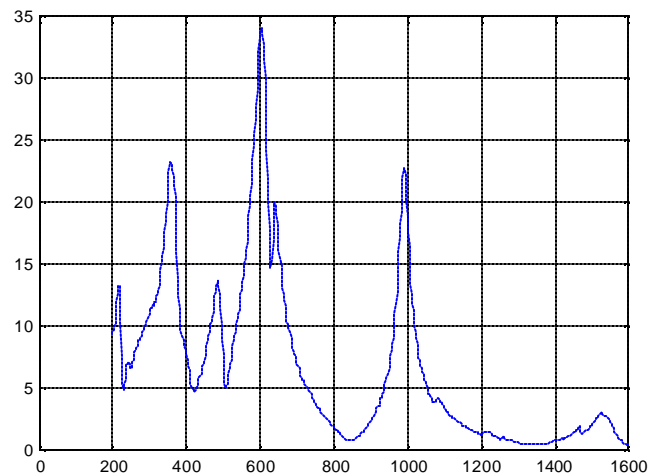
**TABLE 1.3** Modal Equipment Conversion Factors

Equipment	Factor
PCB 3 Axis Accelerometer	10 mV/(m/s <sup>2</sup> )
PCB model 086C03 Impact Hammer	2.3 mV/N

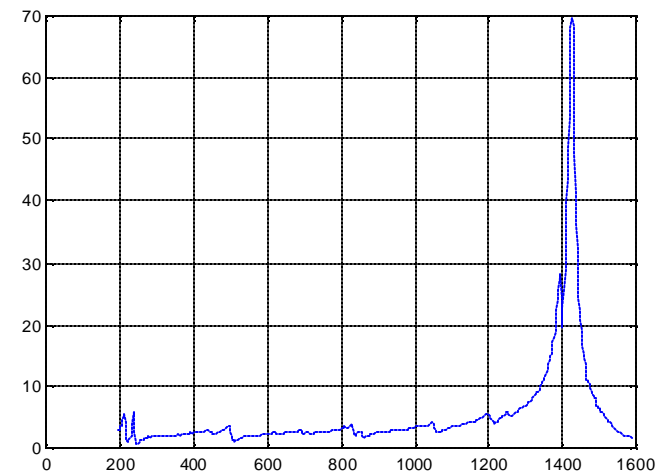


# The Dynamic Stiffness

The modal analysis also revealed the dynamic stiffness of the carriage over a range of frequencies.



*Top center of the carriage while floating*



*Top center of the carriage while not floating*

## 1.17 Measurement Setup

The pitch, yaw, linear accuracy, straightness, and stiffness measurements were done at Dover Instruments in Westboro Massachusetts. All data was taken with the air bearings running at 60 psi (4.14 Bar).

The pitch, yaw, and linear accuracy measurements were made with a Hewlett Packard 5519A Laser System. Four data sets were taken for both pitch and yaw. The first three data sets consisted of six (6) passes, three (3) in each direction, using 320 mm of travel (the carriage has a total travel of 330 mm). The measurements were taken every 10 mm. Two data sets were run with the carriage at continuous speeds of 10 mm/s, 40 mm/s. A third data was run with the carriage stopping every 10 mm to take a measurement at rest. Finally for both pitch and yaw, a fourth pass was made to take measurements every 0.1 seconds, while the carriage traveled at a continuous speed of 10 mm/s. This provided a higher resolution image of what the carriage was doing in pitch and yaw.

The linear accuracy test was done to determine the difference between where the controller thought the carriage was and its actual position. The same HP laser system was used with slightly different optics. The carriage was moved in 10 mm increments and the difference between the controller position and the actual position was recorded.

The vertical straightness measurements were made by suspending a capacitance probe (ADE-2102 probe with ADE Microsense 3401 Amp) above a plane mirror on the carriage. As the carriage moved back and forth (10 mm/s) the vertical displacement of the carriage was recorded at 0.1 second intervals.

The stiffness measurements were fairly crude. A pair of 25 lb. (111.2 N) weights were placed on the center of the carriage (single and together). The displacement was measured above each of the four (4) top bearings pads by a dial indicator. From the known force and displacement the vertical stiffness of the carriage was estimated.

Due to a lack of fixturing, horizontal straightness and horizontal stiffness were not measured.

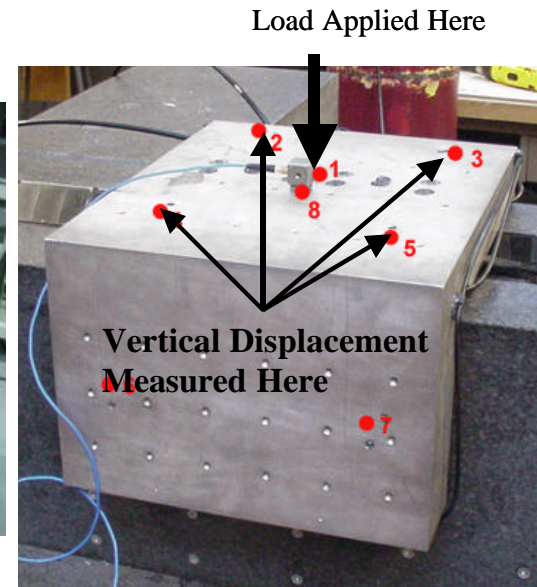
## The Pitch, Yaw, Position Accuracy, Vertical Straightness, and Vertical Stiffness Setup.



*The basic setup for the pitch, yaw, and accuracy measurements. A laser and a variety of optics was used to make each measurement.*



*The basic setup for the vertical straightness measurement. The probe is suspended above the straight edge on the carriage.*



*The basic setup for the vertical stiffness measurements. The load is applied in the center of the carriage and measurements made above each of the top bearing pads.*

## 1.18 The Pitch Data

The pitch of the carriage, as it traveled down the length of the way, varied between 2.38 and 2.44 arc seconds, with a repeatability of between 0.19 and 0.50 arc seconds.

The pitch error has a couple of obvious components. The most noticeable is the periodic oscillations. These oscillations have a period of approximately 29.9 mm and a magnitude of about 1.4 arc seconds. These oscillations are due to the motor coil traveling over the alternating magnetic poles of the magnet track. These poles are spaced 31 mm center to center.

The other significant component of the pitch data is that at the beginning of the carriage (position 0 mm) it starts out with an average pitch of -0.5 arc seconds. Over the length of travel this value changes to +0.5 arc seconds. This indicated that the top surface of the way is slightly concave.

Please see SectionA.3.1 on page123 for the complete plots of all the pitch data taken.

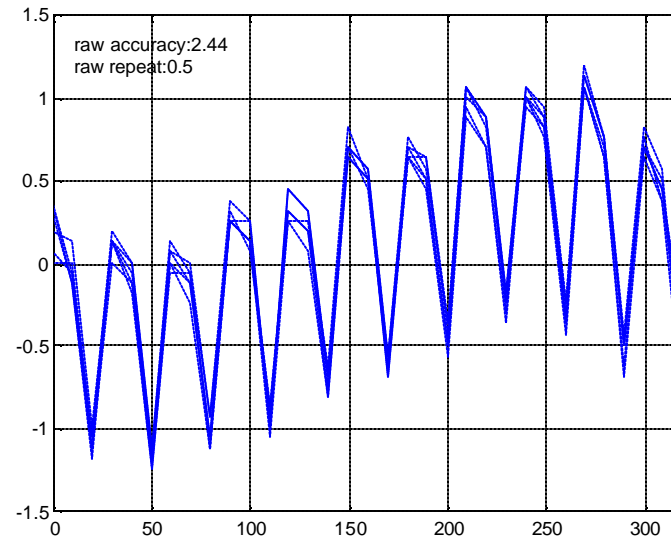
## The Pitch Data

Raw Accuracy: **2.44 arc sec**  
Raw Repeatability: **0.50 arc sec**

Period of Variation 29.9 mm

This period is half the pitch of the magnet track!

The way also appears to be slightly curved (from  $-0.5$  arc seconds to  $+0.5$  arc seconds).



Measurements every 10 mm  
Carriage Speed of 10 mm/sec  
6 passes (3 in each direction)

*Pitch data was also taken at a carriage speed of 40 mm/s, after incremental movements, and as a function of time. There were no major differences. Please see the appendix for the complete data.*

## 1.19 The Yaw Data

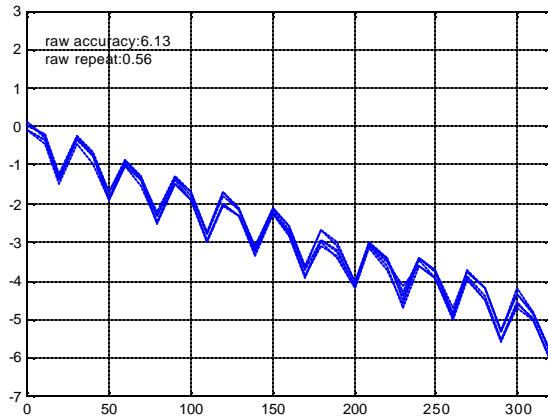
The yaw data was collected in the exact same manner as the pitch data, except the optics of the inferometer were reoriented to measure the angular displacement about the vertical axis of the carriage.

The most striking feature of the raw yaw data is the very linear trend from 0 arc seconds to about 6 arc seconds. If this linear trend is removed the yaw error motion ranges from 1.59 to 1.70 arc seconds, with a repeatability of between 0.26 to 0.56 arc seconds. It is not clear at this time whether this linear change in yaw is due to instrumentation error or an actual change in yaw. Having the controller map out such a linear error and compensate for it is fairly straightforward, but is not even necessary if the trend is an artifact of the instrumentation.

Again the oscillation with a period of about 30 mm due to the magnet poles is visible.

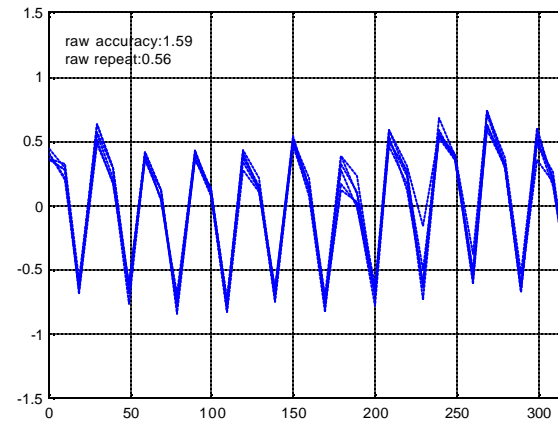
Please see Section A.3.2 on page 127 for all the yaw data taken.

# The Yaw Data



Raw Yaw Data

Measurements every 10 mm  
 Carriage Speed of 10 mm/sec  
 6 passes (3 in each direction)



Yaw Data with Linear Slope Removed

Raw Accuracy:	<b>1.59 arc sec</b>
Raw Repeatability:	<b>0.56 arc sec</b>

Period of Variation 30.1 mm

*Pitch data was also taken at a carriage speed of 40 mm/s, after incremental movements, and as a function of time. There were no major differences. Please see the appendix for the complete data.*

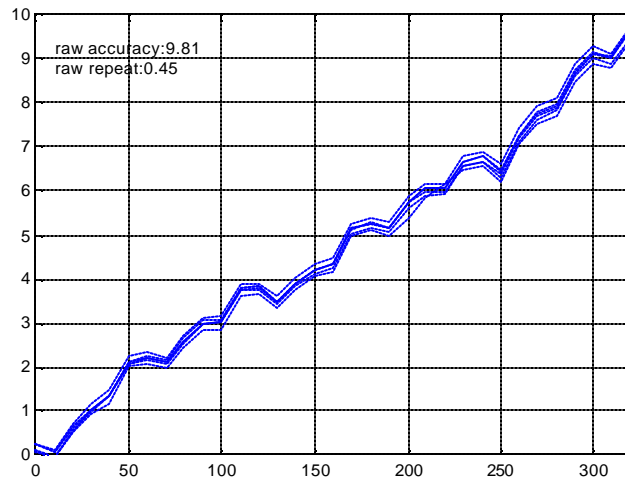
## 1.20 The Linear Position Accuracy Data

The linear position accuracy measures how accurately the controller can position the carriage. This data had a strong linear trend. Since this linear trend is very easy to correct within the controller, the real significance is in the position errors that are left once this trend is removed.

With the linear trend in the data removed, the carriage was consistently positioned to within 1.34 microns, with a repeatability of about 0.33 microns. The 0.33 micron repeatability seems to be an artifact of one or two data points. The repeatability over most of the travel is closer to 0.1 microns. The position encoder has a number resolution of 0.1 microns.

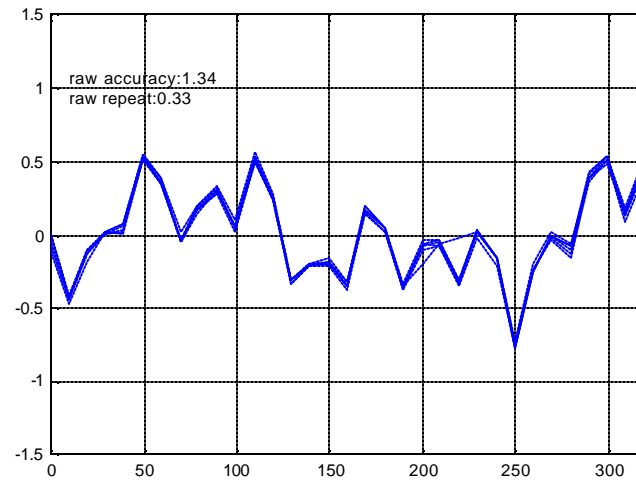


# The Linear Accuracy Data



Raw Linear Accuracy Data

This is the difference between the actual carriage position and the position that the controller thinks it is in.



Linear Accuracy with Linear Slope Removed

Raw Accuracy:	<b>1.34 micron</b>
Raw Repeatability:	<b>0.33 micron</b>

## 1.21 The Vertical Straightness Data

The raw vertical straightness data also had a strong linear trend. This was due to the plane mirror not being level. The slanted mirror gave a vertical displacement reading as it traveled with the carriage under the capacitance probe. This linear trend was removed from the data.

The carriage was moved at 10 mm/s and a measurement was taken every 0.1 seconds. Four (4) passes were made two (2) in each direction. The data from one forward and one reverse pass was filtered by the data collection software to remove the noise; the other two passes were not filtered.

To plot both filtered plots on the same graph, it was necessary to reverse the data taken in the reverse direction and to shift it in time to align the common features on the single plot.

Notice the hour glass shape of the data. Due to a lack of fixturing the plane mirror had to be placed on the carriage with the probe suspended above it. Due to the Abbe error caused by the pitching of the carriage, larger vertical displacements were measured as the measurement point was moved fur-

ther away from the center of rotation of the carriage. This is confirmed by multiplying the pitch by the distance from the center of the carriage to the edge, yielding

$$\frac{1.5[\text{arcsec}]}{3600} \frac{\pi}{180} 152\text{mm} = 1.1\mu\text{m}. \quad (1.10)$$

The vertical translation of the carriage can be estimated by looking at the displacements when the probe was near the middle of the carriage. From the data gathered this appears to be on the order of 0.3 microns.

This test should be redone with the probe attached to the middle of the carriage and the plane mirror suspended above the carriage to get a much better picture of the vertical translation of the carriage over its length of travel.

# The Vertical Straightness Data

An Abbe error of 1.2 microns vertical displacement at both edges of the carriage.

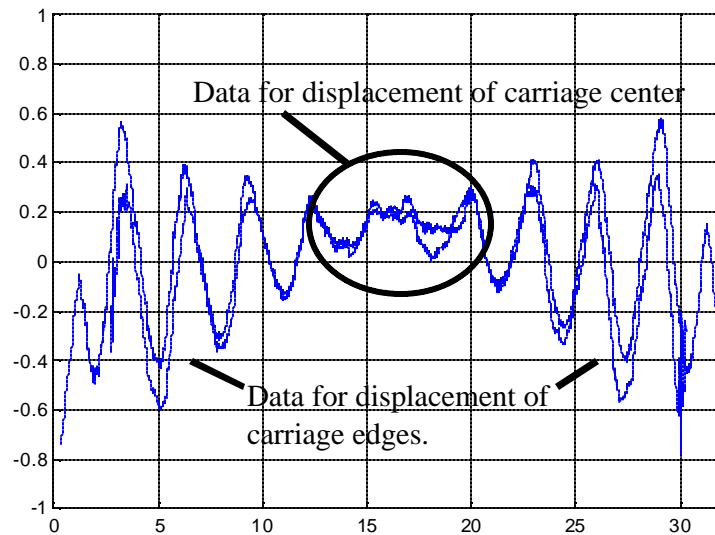
Period of Variation is 28.6 mm

Notice that the hourglass shape of the data is due to pitch errors measured away from the center of pitch rotation.

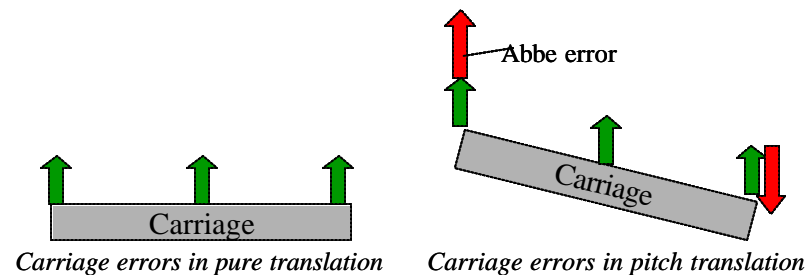
To get better data, the test must be rerun with the probe mounted in the middle of the carriage and the mirror suspended above it.

The data from the center of the carriage shows the pure translation of the carriage to be **about 0.3 microns**.

Because the plane mirror could not be leveled perfectly, any linear slope in the data was removed.



Two passes are shown for a carriage speed of 10 mm/s



## 1.22 The Vertical Stiffness Data

The vertical stiffness of the carriage was measured by placing weights on the top center of the carriage, and measuring the displacement above each of the four top bearing pads.

Because of more preloading on the outboard bearing pair it is expected that the outboard side of the carriage (points 2 and 3) should have a higher stiffness than the inboard side (points 4 and 5). The measured data supports this. Point 2 is stiffer than point 4, and point 3 is stiffer than point 5.

Another interesting feature of the data, is that the left side of the carriage (2 and 4) is much stiffer than the right side (3 and 5). This effect was not predicted. One can see a possible explanation by looking at the pitch data. There are periodic attractive forces that pitch the carriage forward and backwards. This changes the loading on the left and right pairs of top bearings and increases the stiffness of the bearing on the loaded side. The position that the carriage was placed in to perform the stiffness test was chosen without considering this effect. So it is no surprise that the left and right sides are loaded unevenly. This hypothesis is easy to test. Moving the

carriage half a period (15 mm) in either direction should pitch the carriage to the other extreme and load the right side of the carriage. Moving the carriage to a position where the pitch is zero should allow both bearing pairs to be preloaded evenly.

Averaging the stiffness measured at the four point yields a vertical stiffness of **422 Newtons per micron**. Therefore each of the top bearings has a stiffness of 106 N/ $\mu\text{m}$ . Using the estimates for bearing load vs. stiffness derived in SectionA.1.1 (illustrated in FigureA.1 on page115 for the bearings in the prototype) each top bearing is loaded with, approximately 1000 N (225 lbs). The resulting total vertical load on the carriage is about 4000 N (900 lbs) or twice the predicted vertical load of 2000 N (550 lbs). Since the magnetic attractive force varies inversely to the distance *squared*, setting the motor coil 0.2 mm closer to magnet track could be enough to almost double the attractive force. This hypothesis could be checked by collecting data for the horizontal stiffness of the carriage, and comparing the measured load to the preload.

# The Vertical Stiffness Data

Point	Deflection [microns]		Average Stiffness [N/micron]
	25 lbs (111 N)	50 lbs (222 N)	
2	0.15	0.4	649
3	0.4	1.0	250
4	0.2	0.4	556
5	0.5	0.9	235
<b>Average Vertical Stiffness of Carriage 422 N/micron</b>			

Masses were placed on the top center of the carriage and the displacement above the four (4) top air bearings was measured.

