

Design and Control of a Linear Bearing Durability Tester

by

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Submitted to the Department of Mechanical Engineering  
in Partial Fulfillment of the Requirements for the Degree of

Bachelor of Science in Mechanical Engineering

at the

Massachusetts Institute of Technology

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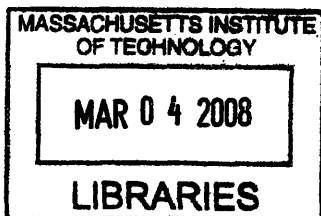
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## ABSTRACT

In order to better understand the characteristics of linear bearings under high load and high speed conditions, a machine capable of testing bearings under these conditions was created. The machine is flexible in its design, allowing testing of a wide variety of bearing styles and under many different load conditions. The maximum load that the machine can provide in any single direction is over 11 kN, with a maximum moment-application capability of over 850 N-m. Additionally, an open-loop load control system and closed-loop motion control system were designed for this application. Finally, a LabView based program integrates the control systems and allows the user to easily calibrate the applied loads, run tests, and analyze data. The overall result is a function test machine capable of testing linear bearings under high load and high speed conditions, either for short term testing of durability verification.

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## **Introduction**

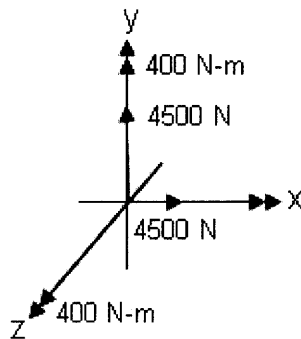
This goal of this thesis is to design a machine capable of carrying out lifetime durability tests of linear bearings. Largely, bearing manufacturers have well-characterized the performance of bearings. As a result, many failure modes are well understood and protected against. However, certain applications require that relatively small bearings operate while high forces and torques are applied. One potential use is in within automotive suspensions, and it is with this application in mind that this testing machine is designed.

This document begins by describing the functional requirements and specifications of the system (both mechanical design and software). Subsequently, the mechanical design of the system is described, beginning with the overall architecture of the mechanical system and progressing through each module. Additionally, the integration of the modules is discussed. Also described are the methods considered and used to control the system, including both the motor and pneumatic loading elements. Furthermore, the software used to interact with the machine is described.

## **Chapter 1: Testing Goals and Machine Specifications**

Within an automotive suspension, testing has shown that a lifetime can be approximately 1 million cycles of a travel up to  $\pm 0.110$  m and speeds approaching 5 m/s. The purpose of this machine, however, is not to recreate the full range of conditions within the wheel well, just most of them. To that end, this machine must be able to complete full lifetime tests within a reasonable period of time (nominally 1 million complete cycles within 500 days which relates to approximately 1.8 seconds per cycle) and be adaptable such that the normal “prime mover” may be replaced and the entire machine may be placed onto a larger testing apparatus capable of producing very high speed motion.

In addition to the motion of the wheel, the machine must mimic the loading conditions which the bearing is likely to undergo. While in use, load is applied to suspension elements by different types of driving activities including braking, cornering, and driving over difficult terrain. Forces transferred to the bearing approach 4500 N (in both directions) and torques approach 400 N-m. Please see Figure 1, below, for loads that the machine must be able to apply.



*Figure 1. In this case, the prime mover will oscillate along the z axis. Forces, signified by single arrows, are required in the other two directions, while torque, signified by double arrows, are required about all three axes.*

As mentioned previously, the principle type of testing that this machine will be used for is durability testing of bearings. So, the entire system must be exceedingly durable, far more so than any potential device under test. The load applied to the bearings need not be highly dynamic. Rather, only quasi-static loading is required. However, it is desired that the user be able to safely vary the loading conditions upon the bearings while a test is being run. Furthermore, the loading profiles and position of the bearing do not need to be coupled. For the purpose of this testing, loads will be applied for a specified number of oscillations and then changed, the load will not vary within each oscillation.

Since this is a testing machine, obviously data will need to be collected. In this case, the parameter that will be of most interest is the coefficient of friction ( $\mu$ ) of the bearings under test. This data should be correlated with the loading condition at the time of the measurement and, ideally, the position of the bearing as well, though this is not absolutely necessary.

In addition, flexibility of the machine is highly desirable. Principally, this means that the machine must be configurable to test several different types of bearings with minimal hardware adjustment. Furthermore, in addition to standard testing, the machine should be able to withstand more extreme testing, such as high and low temperature testing.. Therefore, all machine components must be capable of operating regularly up to 200 degrees Fahrenheit. Additionally, the entire machine (except any extension of the prime mover) must be able to fit into standard on-site environmental chamber no larger than 1.22 meters in each dimension.

Overall, there are many constraints imposed on the hardware and software components of the system. In addition to these constraints, which are summarized in Table 1, below, the machine must be safe for users to operate, especially if an item under test fails violently. This will require several safety measures including protecting against any pinch points and shielding against possible projectiles.

*Table 1. System requirements.*

Size	No greater than a cube of 1.22 m (though a leadscrew might protrude)
Device Under Test	Must be highly flexible to accommodate a wide range of bearings.
Travel	> 200 mm, 220 mm is desirable
Cycle Time	1.8 seconds per cycle
Load	4500 N in directions orthogonal to travel
Moments	400 N-m in three axes
Load Profile	Quasi-static loading capability, variable while machine is in operation
Temperature	Up to 200 Fahrenheit
Data Collection	Collect data on friction in devices under test for long periods of time

## Chapter 2: Mechanical Design

### Section 2.1 – Overall Architecture and Module Planning

In order to create the appropriate relative motion between bearing truck and rail<sup>1</sup>, as well as accurately apply loads to the device under test, the overall architecture of the system must be well developed. To begin, the basic requirement is considered: there must be bearing rails and bearing trucks, between which there is relative motion. Furthermore, in order to faithfully recreate in-situ conditions, forces and torques must be applied to the bearing trucks. These are the only requirements.

Initially, it would be simple to conceive of an architecture that largely mimics a typical bearing setup. That is, hold the bearing rails fixed while moving the bearing trucks. Upon further consideration, this system has a major disadvantage: if the bearing trucks are moving, the elements that apply loads and torques to the bearing trucks must also move. This is undesirable for several reasons. Practically, moving any load application elements would be complicating. Not only would the prime mover<sup>2</sup> be necessarily very large to move the significant mass at the desired velocities. Additionally, moving any pneumatic or hydraulic elements, and their supply and exhaust tubing, would be a dangerous proposition. In terms of testing requirements, the coupling of the bearings to additional moving mass would introduce inertial loading to the bearings, a condition that does not faithfully represent the conditions within the vehicle.

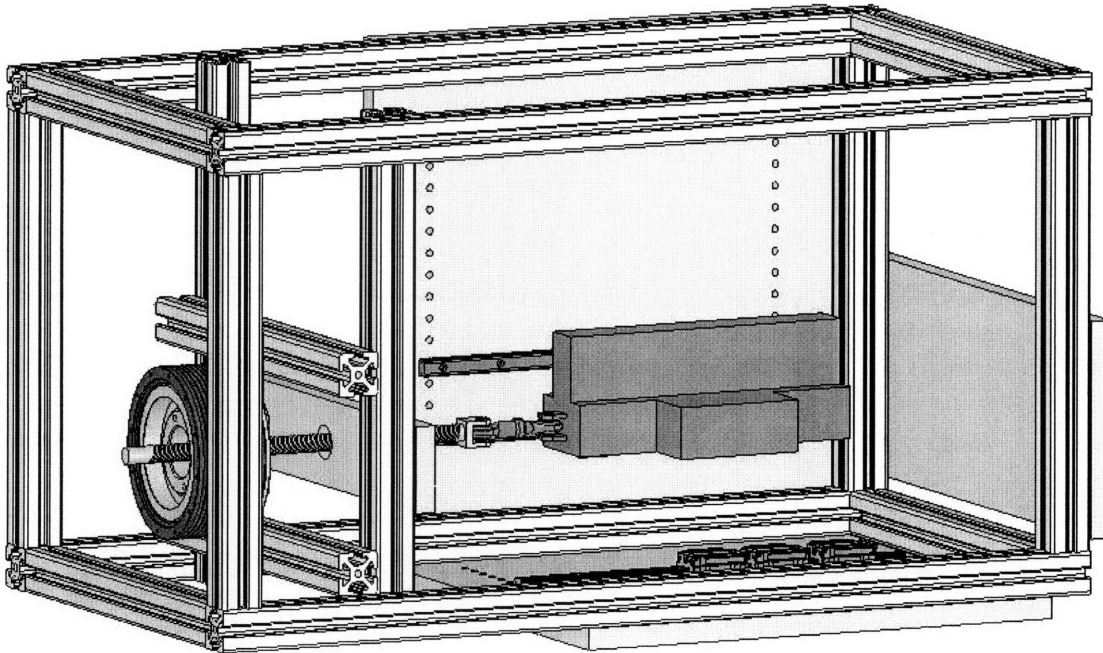
Since a cursory analysis renders the fixed bearing rail method to be undesirable, it therefore makes sense to examine the opposite: constrain the bearing trucks while moving the bearing rails. This solution does not have either of the major problems discussed in the previous paragraph. Other complications are, however, introduced. Specifically, the bearing rails, now

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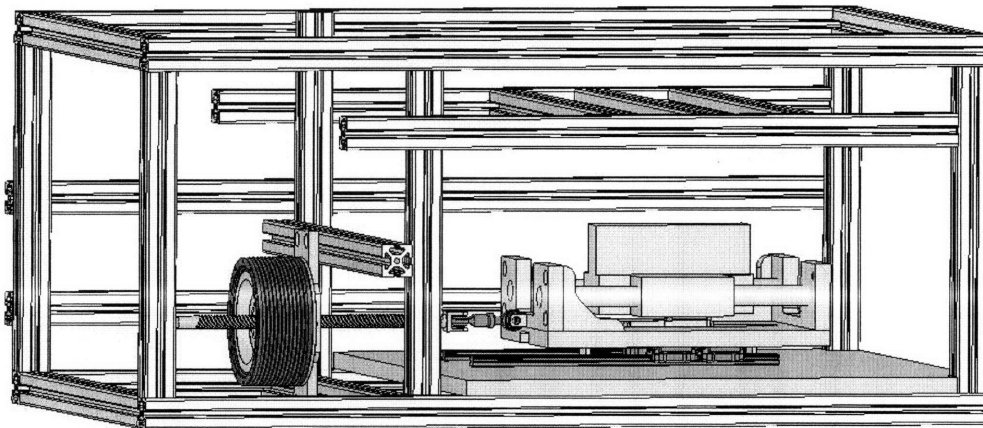
<sup>1</sup> Here the bearing truck is defined to be the component that contains the ball bearings while the rail is the component that guides the truck.

<sup>2</sup> The “prime mover” is defined as the motor that creates the oscillatory testing motion.

moving, must themselves have a bearing system on which to travel. In order to ensure that this bearing system does not become the limiting factor in the test, these bearings will be sized significantly larger than any other bearing that may be tested on this machine. Please see Figure 2 and Figure 3, below, for a comparison of the two architectures.



*Figure 2. Moving carriage<sup>3</sup> concept; notice the ball screw is coupled directly to the carriage. In this design, load application elements would ride on bearings below and to the side of the carriage. The carriage itself would be supported by the device under test (not shown).*



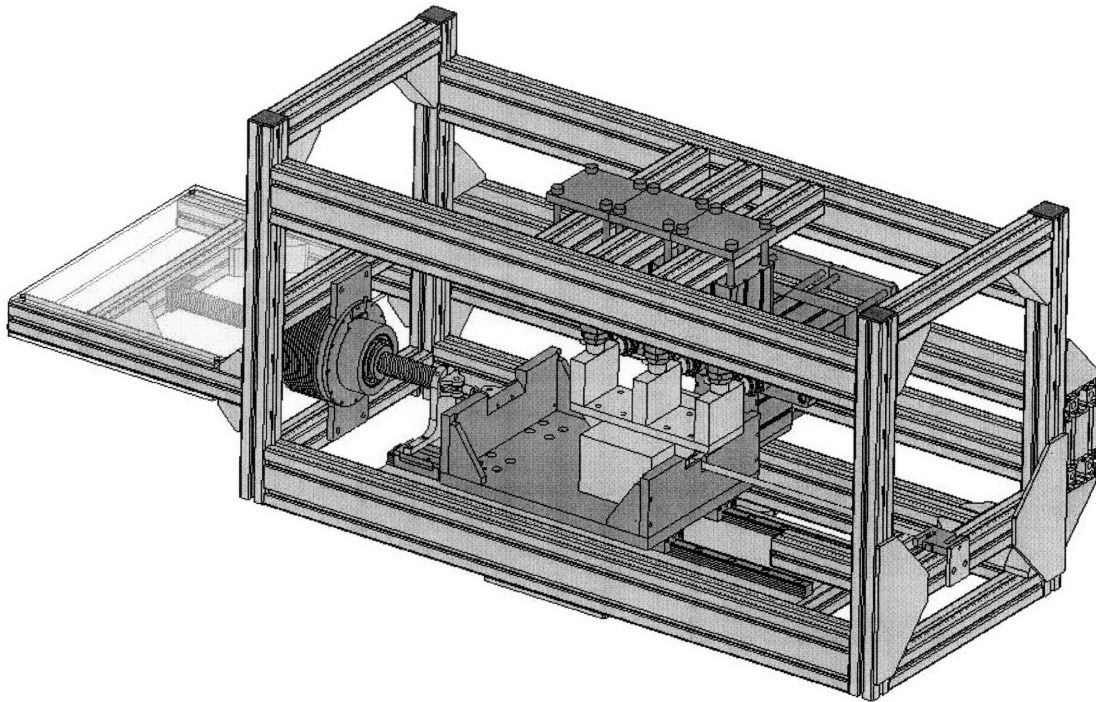
*Figure 3. Static carriage design concept; notice the ball screw is not coupled to the carriage. Again, the device under test supports the carriage (not shown).*

<sup>3</sup> The carriage is defined as the module that carries the bearing trucks under test.



Having decided upon an overall system design, the rest of the architecture may be described and designed as modules. First, there must be a prime mover to move the bearing rails back and forth. Additionally, there must be a load application system of some kind. Also, there will be a module under test: the bearing rails, the bearing trucks, and an apparatus to apply load to them. In order to contain these modules, there must be a skeleton of some kind that will allow the flexibility to adjust between several testing configurations. Finally, there will be several elements that allow the coupling and constraint of these modules (e.g. oversized bearings between the bearing rails and the skeleton and the flexure that constrains the bearing trucks).

Figure 4, below, shows the completed mechanical design.



*Figure 4. The completed mechanical design. On the left, the motor provides oscillation back and forth while several pneumatic cylinders apply loads to the device under test.*

## Section 2.2 – Individual Module Design

### *Prime Mover*

Having decided that the best overall architecture involves moving the bearing rails while constraining the bearing trucks, it becomes possible to specify a prime mover for the system. In

this case, a rotary motor with a ball screw was selected for its high speed, flexible travel length, and limited cost. Several choices are possible including fluid based actuators (i.e. pneumatic or hydraulic), linear electromagnetic motors, or conventional motors coupled to a lead screw. Initially, I explored linear electromagnetic motors. Currently, commercially available gantries can achieve very high speeds and withstand considerable loads and are extensively used in automating manufacturing facilities. Unfortunately, achieving both the desired speed and length of travel incurred unacceptably large costs. Next, I explored available fluid based actuators. Several companies offer pneumatic pistons with integrated position control. However, stock models offer either insufficient speed or unattractive package size. Though not completely discounted, I moved on from fluid based actuation to explore the application of a conventional motor.

After exploring several commercially available packages incorporating a motor and lead screw, I determined that none were available that fit the system requirements. I believe this was because the worm gears or rack and pinion assemblies used by these packages limited the speed and durability of the system. I did, however, find brushless ring motors with high torque output manufactured by Aerotech, Inc. These motors, if coupled with a rotating ball screw could potentially provide the necessary speed and travel requirements with significantly less friction. I created the following model in order to verify that this was indeed the case, and to select the ball screw appropriate for my design.

THK Corporation offers several different large lead rotating ball screws. In each case the diameter of the screw is equal to its lead. In theory, then, the larger the screw, the faster maximum speed it would have. However, the larger screws also have larger nuts, and more moving mass. This becomes a factor when calculating the maximum possible acceleration rate.

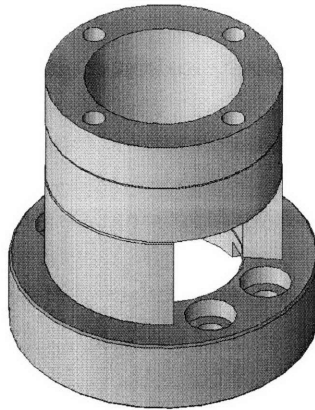
From this data, a theoretical time per cycle can be calculated. The ball screw which yields the shortest cycle time would be the optimal choice. Table 2, below, shows the results of this calculation. It can be seen that the ball screw with a lead of 32 mm is the optimum choice and was, therefore, selected.

Table 2. Results from calculating the time per half cycle and time per million cycles of different ball screw sizes.

Pitch [mm]	16	20	25	32	36	40
Approx Time Per HalfCycle [sec]	0.6	0.5	0.43	0.37	0.38	0.4
Approx Time Per Million Cycles [hours]	333.3	277.8	238.9	205.6	211.1	222.2

Having selected the appropriate motor and ball screw combination, further mechanical design was still necessary. Aerotech provides an OEM package only, which consists of a stator and a rotor. Therefore, a housing for the stator, a method to couple the stator and the rotor, and a method of mounting the motor to the machine needed to be designed. Previously, another engineer that I worked with had designed a body for the stator to press into, providing both heat dissipation for the motor and a bolt circle to interface with. To reiterate, the stator must be coupled to the ball nut, and the ball nut must be coupled to the rotor. The rotating ball nut allows the transfer of rotary motion from the motor to the linear motion of the ball screw, which passes through the center of the rotor.

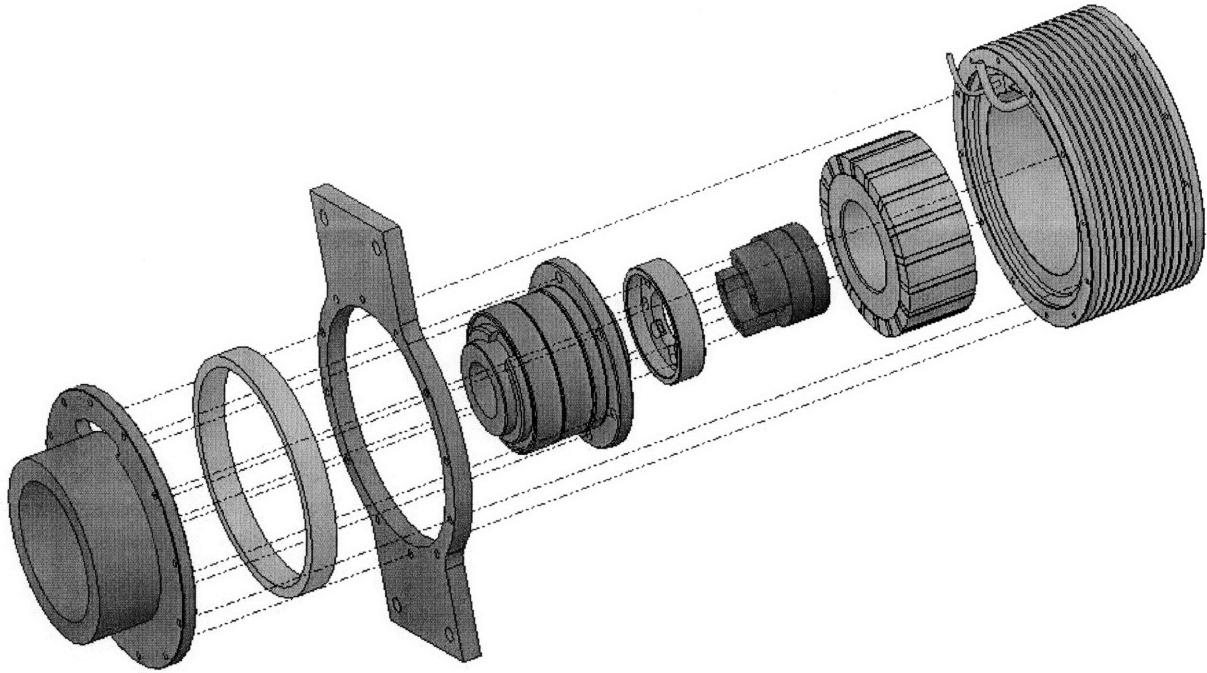
The ball nut is mounted to the rotor by means of two pieces. One piece is attached to the ball nut by a bolt circle. Due to the tight tolerances inherent in the motor’s assembly, the bolt circle does not serve to align the two pieces. Rather, the piece fits around a protrusion in the ball nut. The second piece attaches to the first piece, which is then pressed into the rotor. In theory, the two pieces could be one. The slots allowing access to the bolts that attach the ball nut to the coupling piece, however, could be potentially difficult to machine. Therefore, dividing the part into two greatly simplifies the machining process. Please refer to Figure 5, below, for a detail of the two-piece coupling.



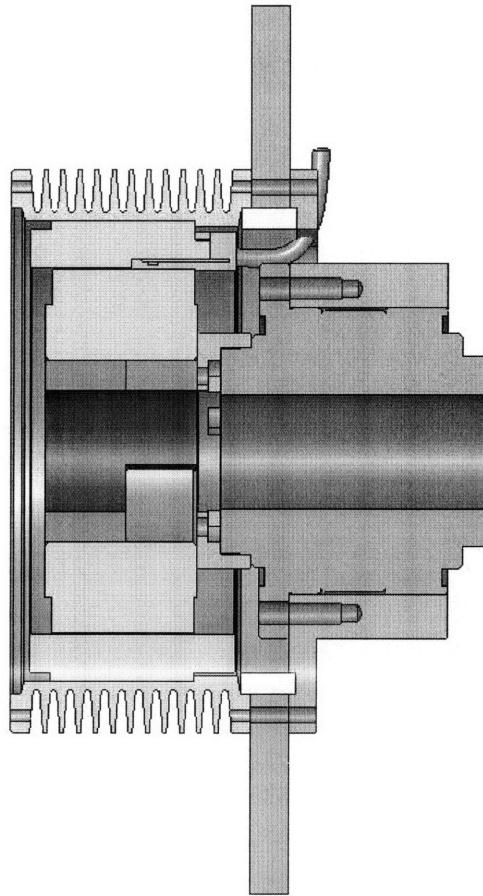
*Figure 5. Two piece coupling from motor rotor to ball nut. The ball nut attaches through the countersunk holes in the coupling's base. Afterwards, the rotor is pressed over the coupling. The tapped holes on the top are for an optional end cap.*

The ball nut is then set within its own housing. Again, it is attached by a bolt circle and aligned by concentric cylinders. This housing has a bolt circle matching the bolt circle of the stator. The two are then attached by machine screws around the circumference. Sandwiched in between the ball nut housing and the stator housing is the piece that allows the motor to be mounted to the body of the machine.

When trying to assemble these parts, it is very quickly evident that there is a problem. There is a strong magnetic attraction between the stator and the rotor. As a result, using a bolt circle to align the rotor within the stator is unacceptable. It is simply impossible to achieve the necessary tolerances around the circumference of the rotor. Therefore, a key alignment ring is added, as shown in an exploded view and a section view in Figure 6 and Figure 7, respectively.



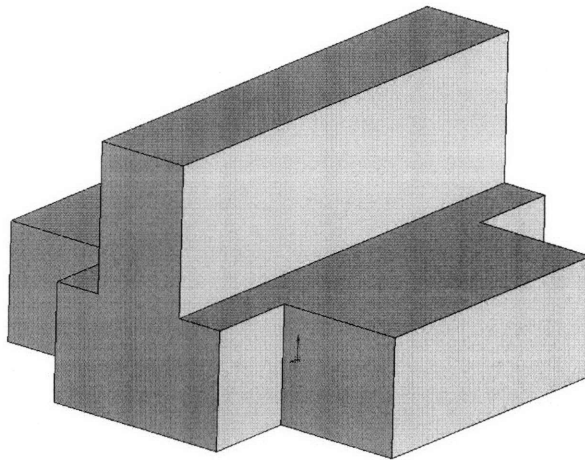
*Figure 6. Exploded view of motor assembly.*



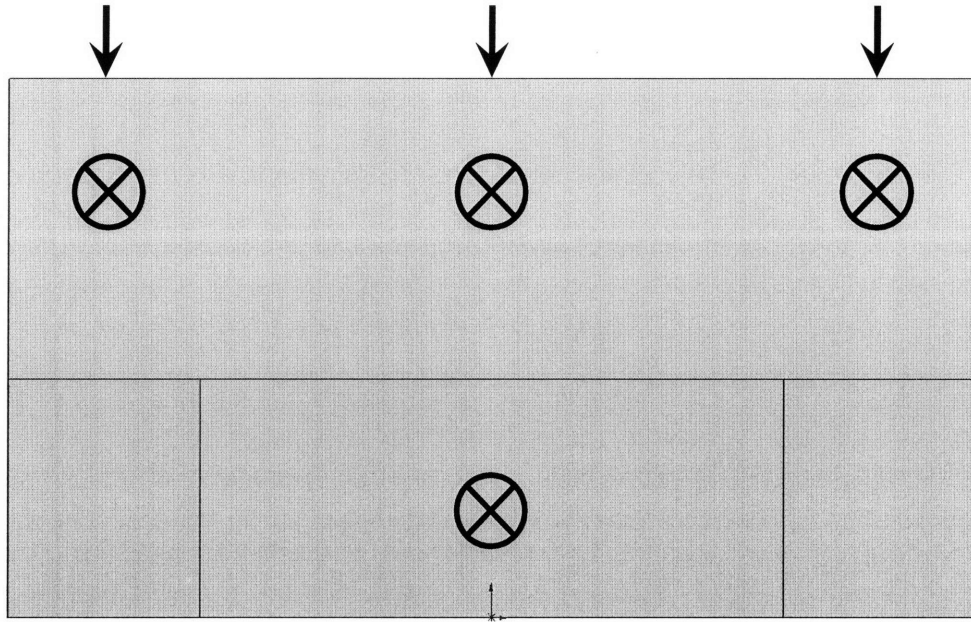
*Figure 7. Section view of the motor assembly. Note, the alignment ring constrains the stator, mounting ring, and ball nut housing, therefore maintaining the motor's required tolerances.*

### *Box and Carriage*

The box and carriage is the module that incorporates the device under test. The carriage contains the bearing trucks; it is to this component that load is applied to mimic the force and torque loading that the bearing undergoes in the automobile. While under test, the carriage must accommodate force along two axes (both of those perpendicular to the direction of travel) and torques in all three directions. In order to incorporate these loading conditions seven actuators have been employed. Correspondingly, there must be seven points on the carriage to receive the forces imparted by these actuators. Please refer to Figure 8 and Figure 9, below, for a preliminary carriage design and a description of the general loading structure.



*Figure 8. Early 'carriage' design. It shows the general shape requirements for the loading points.*



*Figure 9. A side view of the previous figure; the arrows and circles represent where loads will be applied. Direct loads will be placed above (central arrow) and on the side (lower circle). Torques are created by equal and opposite forces on the 'wings.'*

Understanding that these loading points must be conserved, the manufacturability of the carriage still needed to be improved. In order to reduce both machining time and material cost, the box was broken up into component parts. A central section to house the bearings and accommodate one of the side loads served as the main component. Upon it are added an upper section to accept the other loads and triangular sections on either side to reinforce the loaded arms. Please see Figure 10 and Figure 11, below, for the completed assembly as well as an exploded view, respectively.

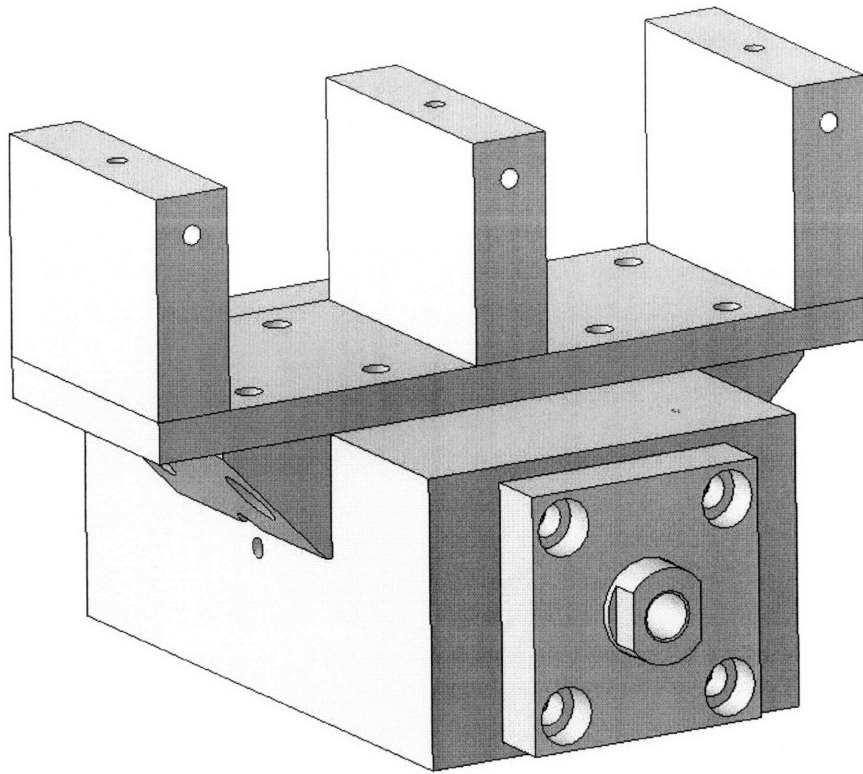


Figure 10. Revised carriage design, including a compliant mount for loading on the side.

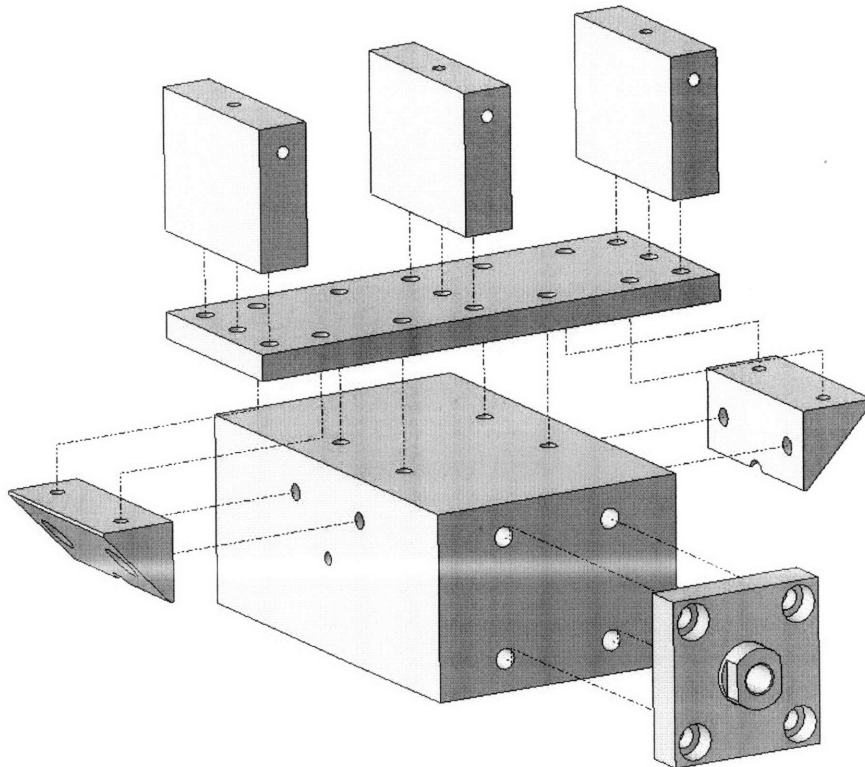
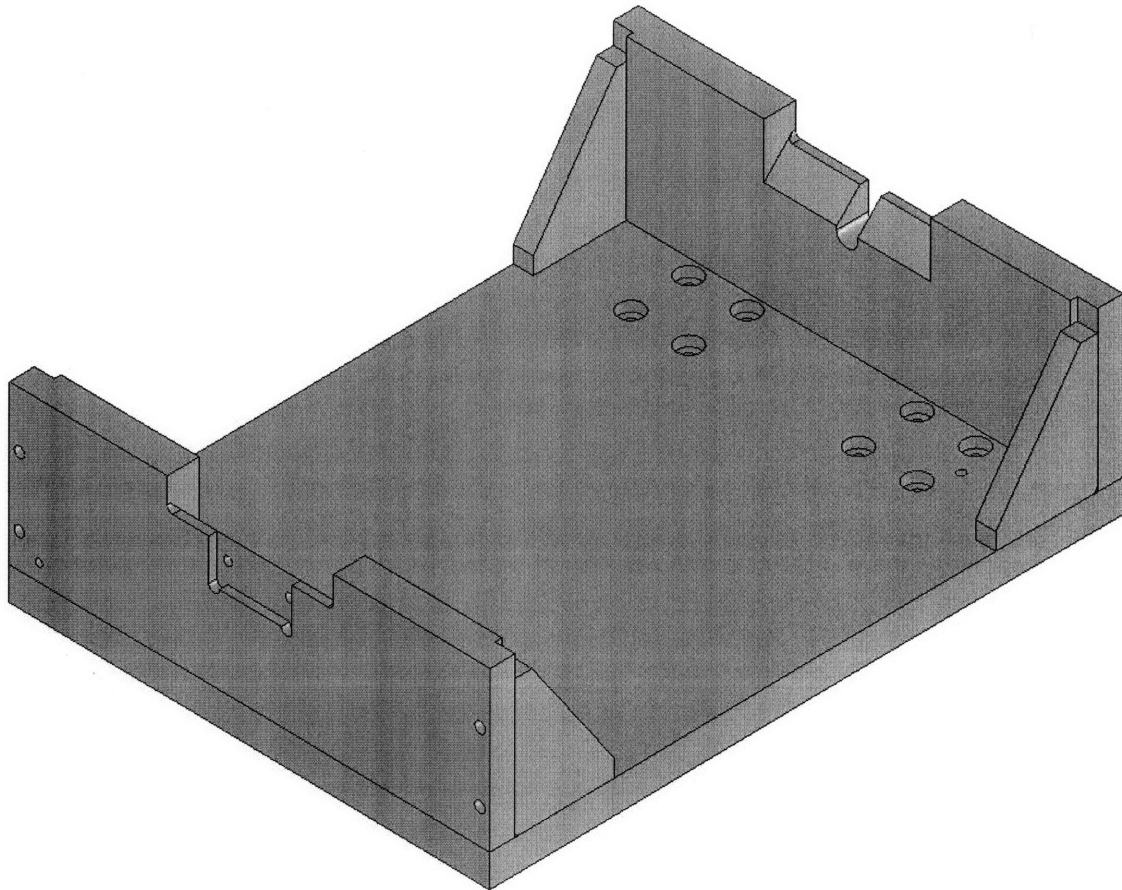


Figure 11. Exploded view of the carriage. Please note, each of the component pieces is very easy to machine, and the overall design conserves a great deal of material compared with the original.



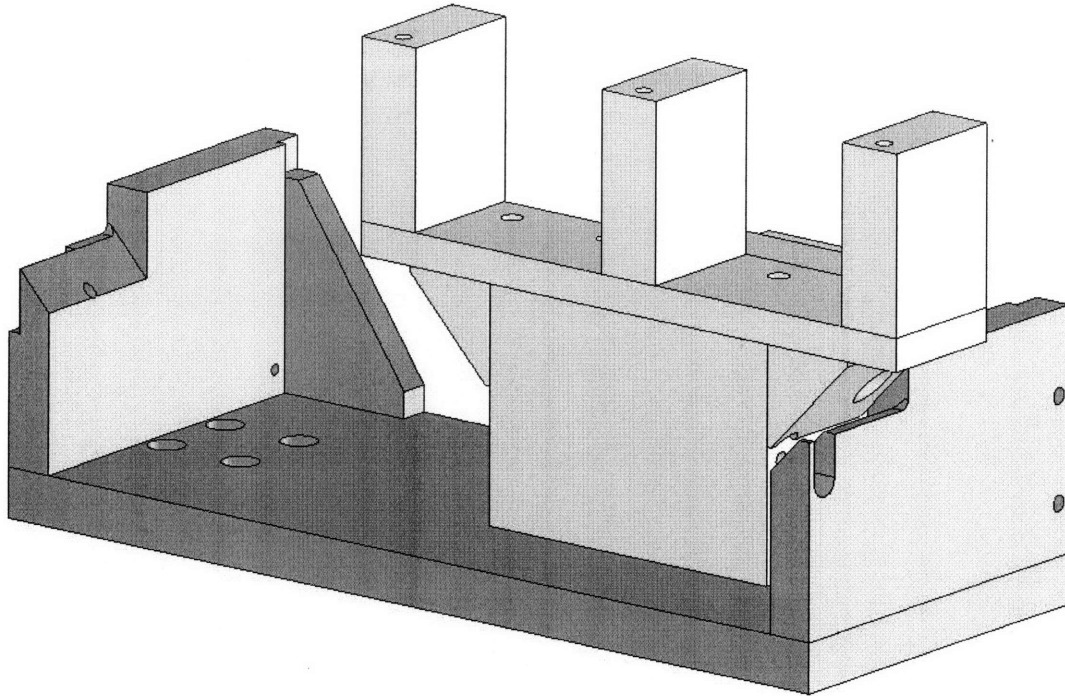
The box contains the bearing rails and is coupled to the lead screw. The box may take different forms in order to most easily conform to the type of testing that is required. A preliminary version, however, is explained here. Please refer to Figure 12, below.



*Figure 12. Potential design for the 'box' section. Gussets provide stiffness while the open sides and top allow loads to be applied to the bearing under test.*

In the design of the carriage, close attention was paid to ensure that the bearing rails under test are the same length as those in the automobile. This is particularly important in case the rails flex since curvature of the rails can introduce severe loading within the bearing trucks. This can be difficult, however, since the box must be allowed to travel the full length of the carriage without interference. Therefore, the box is matched to the carriage with areas cut out of its ends to allow the passage of the box, as seen in Figure 13, below. Similarly, while gussets are

included in the box to provide strength and stiffness, they must be sized carefully such that they do not interfere with the load application elements.



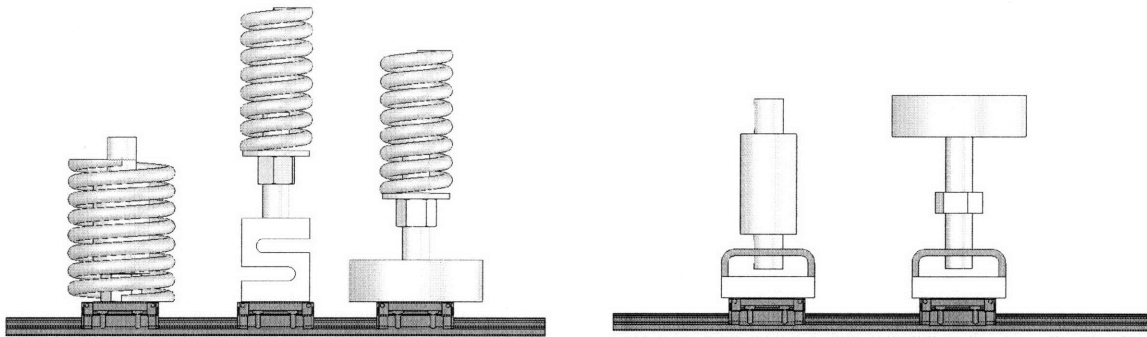
*Figure 13. Section view of 'box' and 'carriage.' Please note the clearance between the brace and the end of the box, while still allowing the carriage fits full travel length.*

### *Load Application*

As discussed previously, the load application elements are designed to be stationary. Therefore, the decision factors include the effectiveness and accuracy by which the actuator can provide the desired force output, the size of the actuator and its ability to integrate within the machine, and the ability to vary the load while the prime mover is operating. Also as discussed previously, there are to be seven force actuators, thereby making makes both size and manufacturability of critical importance.

One potential solution involves the use of mechanical loading elements. More specifically, springs could be placed in tension or compression. If placed in series with a load cell, it would be a fairly simple task to statically set the desired loads. Using normal coil springs does not satisfy the task, however, due to the necessity of applying both tensile and compressive

forces. Using an appropriately tuned leaf spring and a load cell modified to act as a turnbuckle could conceivably achieve the desired task. Several configurations using springs are displayed below in Figure 14.



*Figure 14. Several force application concepts utilizing mechanical springs. On the left, compression springs are used with several types of load cells (l-r turnbuckle, S- and pancake styles). On the right, leaf springs, considerably more compact and capable of force in both directions, are shown with a turnbuckle and a pancake style load cell.*

Mechanical springs, however, include several drawbacks. Principle among these is the difficulty in adjusting the desired load. In the displayed configuration, it would be impossible to remotely adjust the load applied to the carriage. Rather, an operator would need to manually move a nut or turnbuckle. Should this need to be done while the prime mover is in operation, it could potentially be very hazardous. While it could be possible to include a small motor of some kind to achieve remote load variation capability, such a design begins to become prohibitively complex in light of simpler options.

The simpler option selected in this case was a compact pneumatic cylinder capable of push-pull operation. Cylinder sizing was based upon the requisite system loads. As described earlier, maximum loads reach approximately 4500 N and 400 N-m. Assuming a maximum air pressure of approximately 90 psi, a piston diameter of 100 mm is necessary in order to achieve the desired load. In order to reduce the overall size of the machine, compact pneumatic cylinders were selected since more than an inch of travel is not necessary.

Next, there are two principle methods by which the load from each cylinder can be controlled: proportional flow control and proportional pressure control. In the proportional flow control scheme, if the load drops below the desired level, flow to one side of the cylinder, and away from the opposite side, is increased until the necessary load is achieved. Similarly, if the desired load is too high, the opposite would occur. By switching the compartment into which the higher pressure is directed, both push and pull operation can be achieved. As a result, a total of 14 solenoid valves and 7 proportional flow valves are necessary and switching from tension to compression, or vice-versa, could potentially be a tricky crossover. In addition, since the control elements are simple valves, a closed loop control scheme would definitely need to be created in order to effectively apply accurate loads, and smooth any crossover discontinuities. Please see Figure 15, below, for a diagram of this pneumatic circuit.

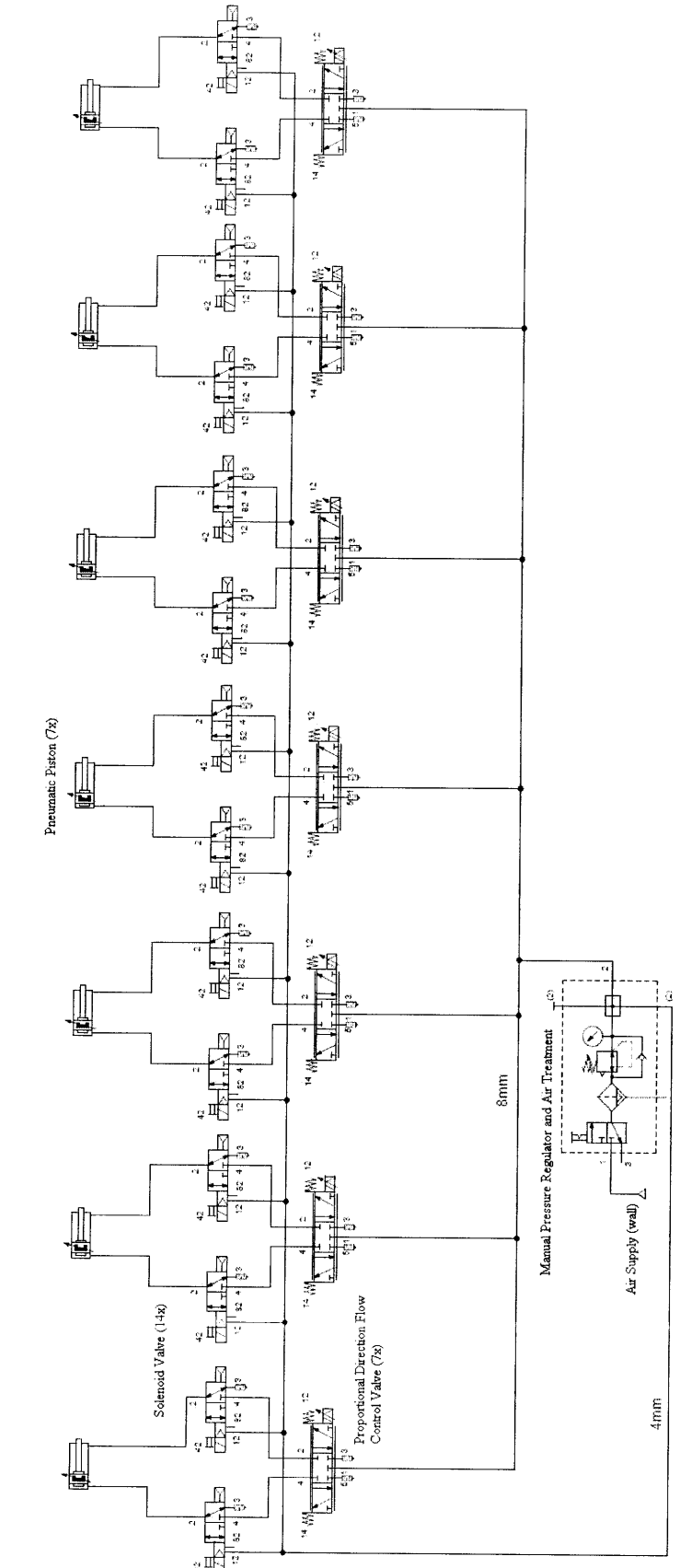


Figure 15. Proportional flow system pneumatic circuit diagram.

The second strategy, that which has been selected, is the proportional pressure system. In this case, the pressure on the other side of the cylinder is regulated by a control unit into which an analog voltage signal is sent. This unit incorporates an internal control system that regulates flow into and out of the cylinder in order to maintain the desired pressure. This integrated closed loop system means that while a closed loop control system could be written, an open loop system is could potentially suffice.

If one controller were used for each side of each pneumatic cylinder, a total of 14 controllers would need to be used. However, this would incorporate the same problem of a potentially problematic crossover as mentioned before. An alternate solution is described in the pneumatic circuit in Figure 16, below. This solution involves pressurizing one side of every cylinder at approximately 90 psi using a simple regulator. The other side of the circuit has a pressure varied from 0 psi to approximately 180 psi as regulated by the control units. In order to achieve the 180 psi level from line pressure of approximately 100 psi, a pressure doubler is used.

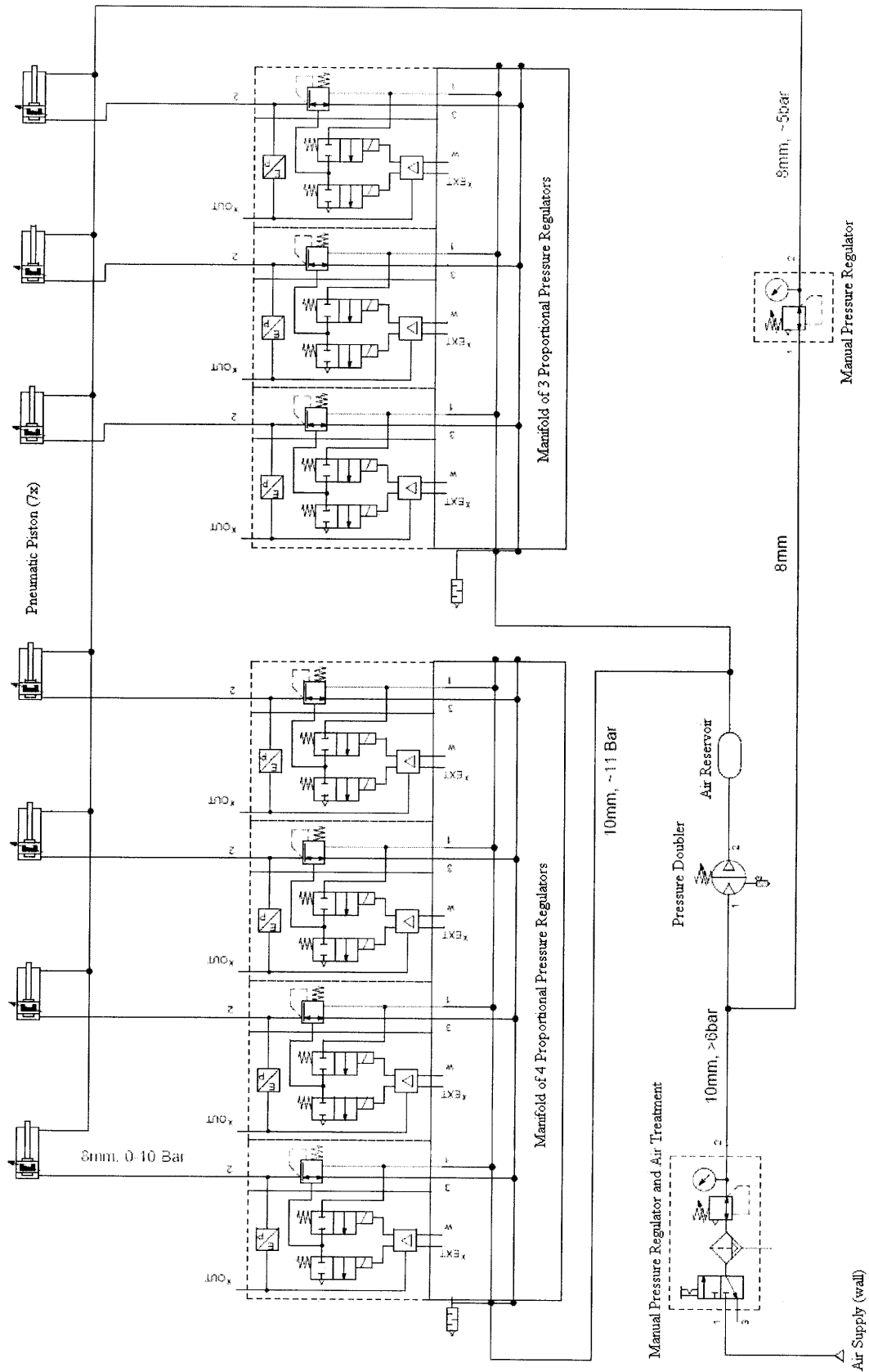
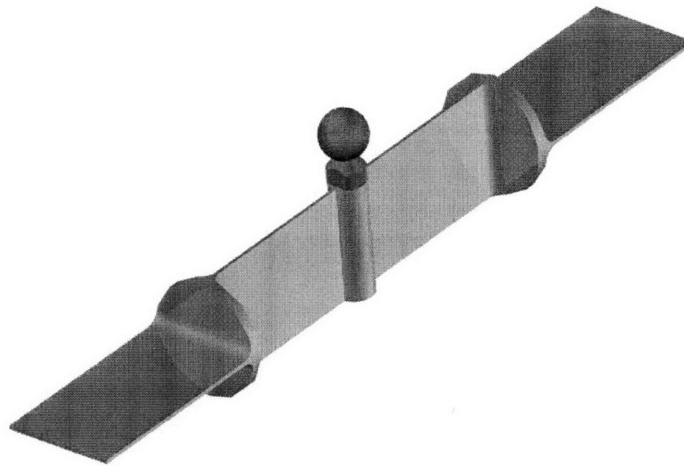


Figure 16. Proportional pressure system pneumatic circuit diagram.

This solution allows a smooth transition from tension to compression while still allowing the full range of required forces. Furthermore, the proportional pressure system is simpler to control and less costly, thereby making it the clear choice.

#### *Carriage Constraint*

In order to allow for small deflections in the bearing rails, the carriage must be allowed to move slightly in each of the two axes in which loading occurs, and rotationally in all three directions. Therefore, the carriage must be constrained in one degree of freedom: the direction of travel of the carriage. One initial concept for the design (seen in Figure 17) involved attaching a flexure to the bottom of the carriage, through the base of the box, as seen below in Figure 18. As can be seen, the shape of the flexure allows motion along each axis and the ball joint atop the flexure allows free rotation in all directions.



*Figure 17. Initial flexure concept providing motion in 5 degrees of freedom, but constraint along the long axis. It would have been supported on pedestals at either end.*



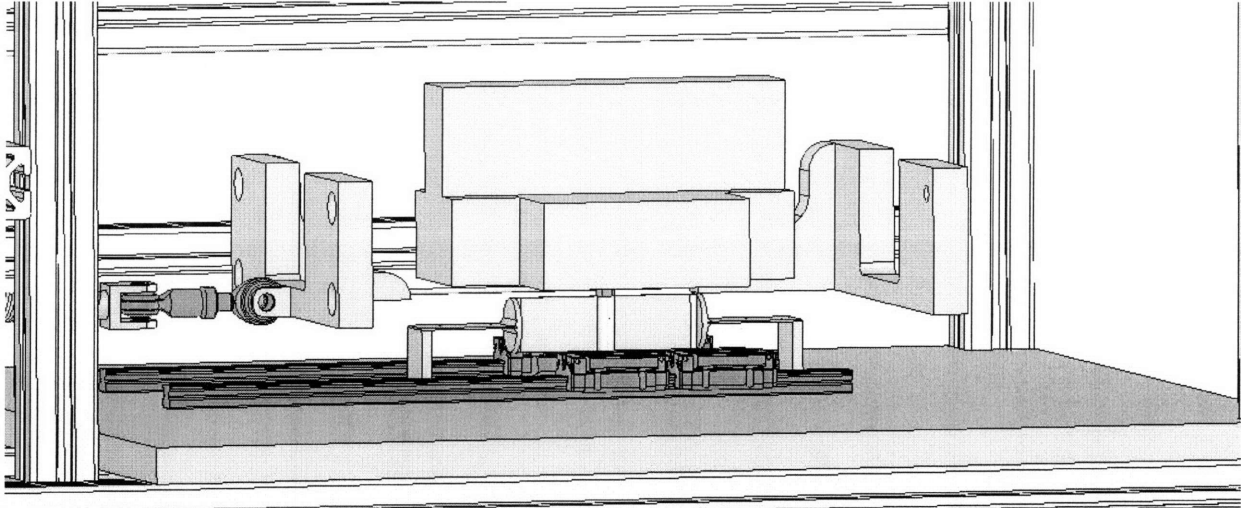


Figure 18. An early design concept. The bottom and front side of the box have been removed revealing the flexure installed below the base of the box.

There is, however, a far simpler method, that also allows the desired measurement. The coefficient of friction ( $\mu$ ) can be derived from the load between the box and the carriage. The combined solution is a “stinger” that attaches the carriage to the outside skeleton (please refer to Figure 19, below). If engineered appropriately, the long and narrow rod does not constrain the carriage in any direction except the axial. Furthermore, any axial load on the rod is that imparted by the friction in the bearings between the carriage and the box. Any side-loading of the rod is excluded by the load cell that is placed in series with the rod.

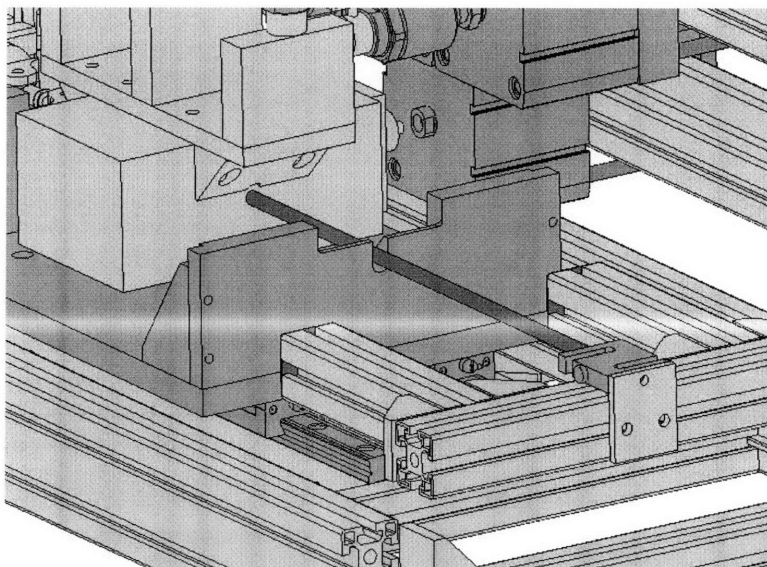


Figure 19. The constraining rod, spans from the carriage on the left to the S-shaped load cell on the right. In turn, the load cell is mounted to the extrusion by means of a simple plate.

In order to be useful, the constraining rod must be flexible in the appropriate directions, but also strong enough to withstand the loads imparted by the box upon the carriage. This means that the rod must be of large enough dimensions to not buckle. Furthermore, the constraining rod cannot transfer too much of a sideload to the load cell in the case that the bearing rails flexed. Finally, the rod had to be able to survive in the fatigue conditions introduced by testing cycles. Figure 20, below, illustrates the design envelope created by these constraints.

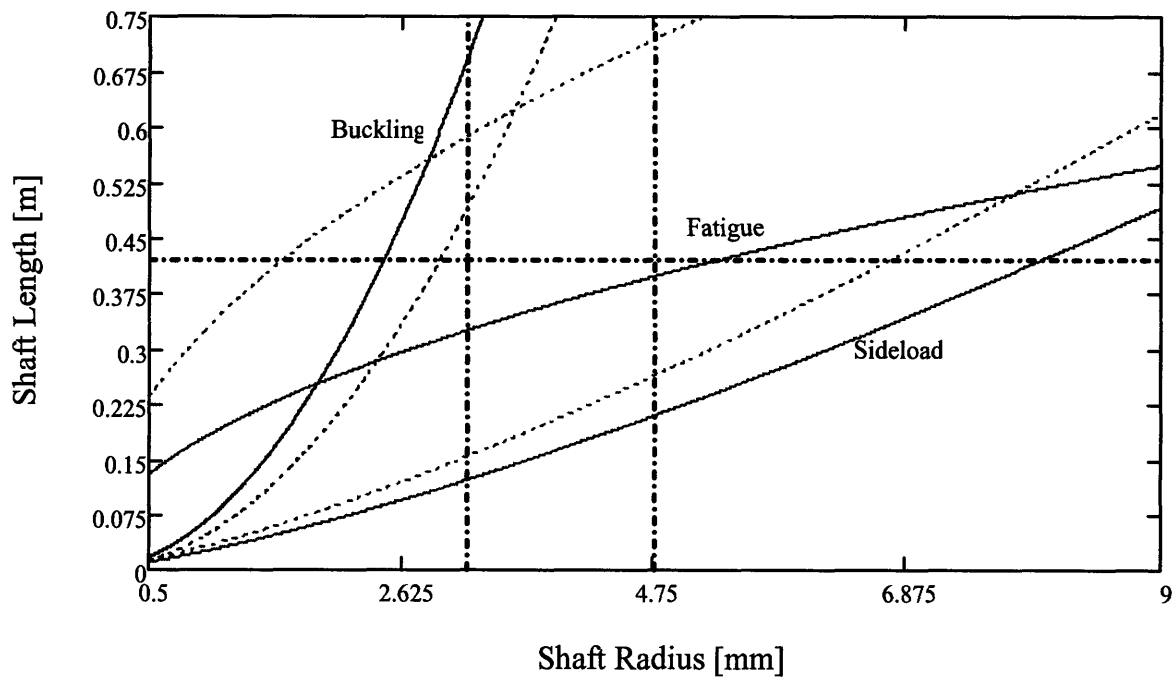


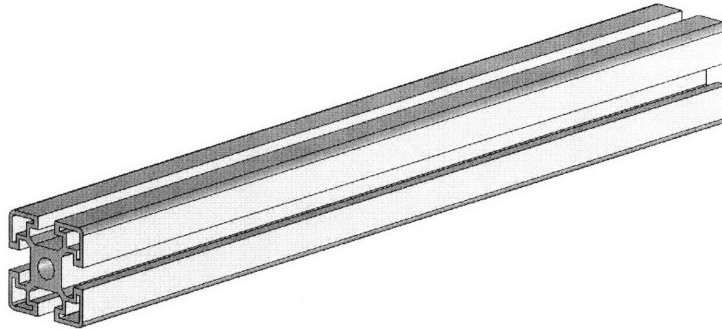
Figure 20. The solid lines form the design envelope, bounded by fatigue (green), buckling (blue), and sideload (red) limits. The dotted lines represent the acceptable design direction.

### Skeleton

The skeleton that contains all of the other modules had several key requirements, beyond merely connecting to the other modules. Principle among these is flexibility. In other words, the system needs to be able to adjust to a wide array of different bearing test configurations.

Different setups will require different lengths, different heights and different widths. One material that was both met that need and was readily available was slotted aluminum extrusion,

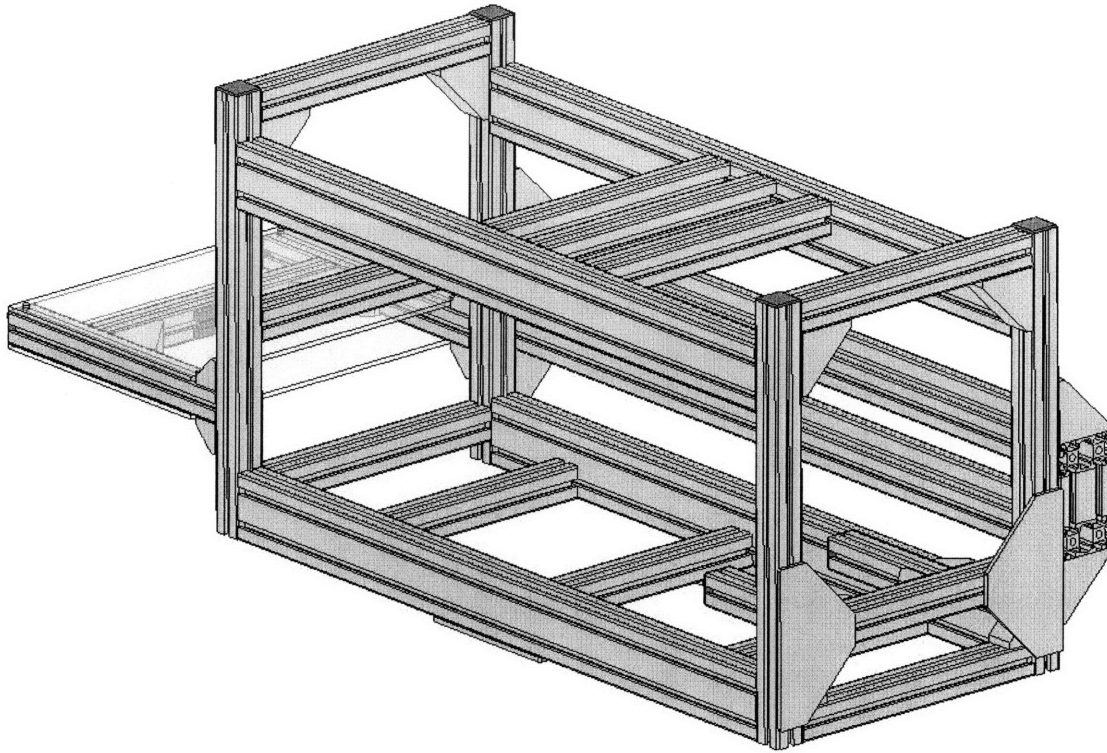
an example of which is shown in Figure 21, below. (The extrusion is often referred to by the brand name 80/20, though it is sold by many manufacturers.)



*Figure 19. Example of square extrusion, 45 mm square on the end. The T-slotted profile allows the user to easily connect several pieces or mount other hardware.*

Having decided upon a frame composed of aluminum extrusion, the frame itself needs to be designed. On one end, the prime mover must be attached. On the opposite end the constraining rod must be attached via a load cell. On the upper “ceiling” of the frame are mounted three of the pneumatic cylinders. On an adjacent side, between the two ends holding the prime mover and constraining rod, are the other four pneumatic cylinders. Finally, the carriage assembly rests upon the bottom of the frame.

Much of the frame is built of 45 mm square extrusion of varying lengths. However, several of the longer spans are made from extrusions that measure 45 mm x 90 mm, in order to minimize their flexing when the machine is fully loaded. The core of the final frame design is shown in Figure 21, below.

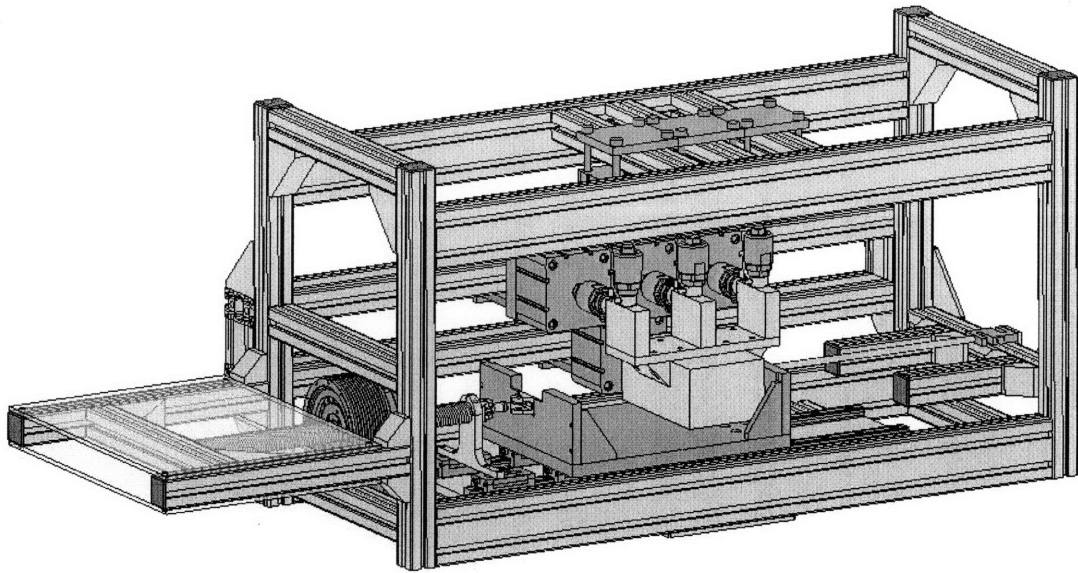


*Figure 20. The 'skeleton' is composed primarily of aluminum extrusion. Load bearing areas are made with thicker extrusion to reduce flexing of the frame and potentially vulnerable joints are stiffened with gussets or plates. Additionally, hard stops are incorporated at both ends to ensure the ball screw remains within the ball nut.*

In order to allow testing of a wide variety of bearings in many configurations, the machine was made as long as possible, while still fitting within an environmental chamber (1.22 meters). Similarly, the height and width of the machine are designed to maximize the number of different types of bearings and configurations that might be tested while still allowing the machine to fit upon a workbench.

### Section 2.3 – Module Integration

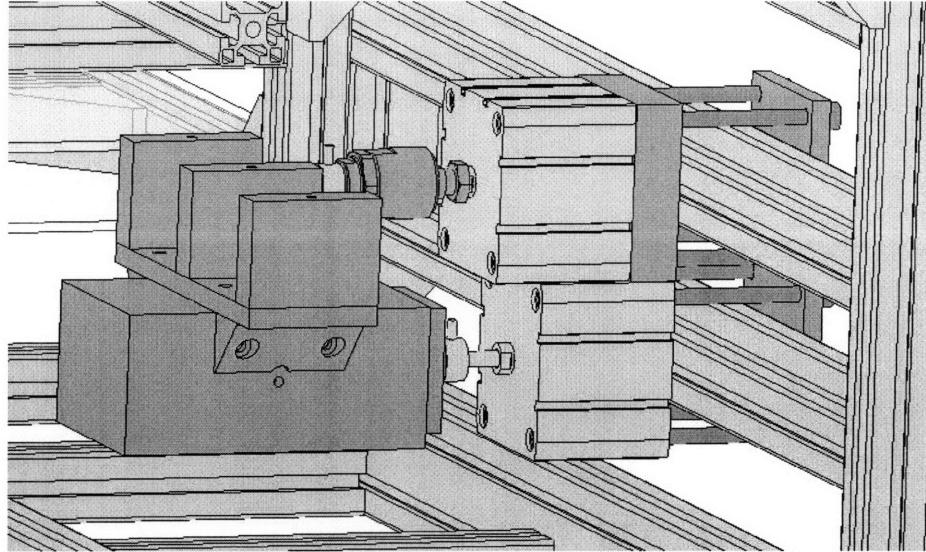
In order for the machine to work successfully, instead of being a mere jumble of parts, the modules must be integrated to synthesize the whole. Figure 4 previously displayed the assembled machine. Figure 23, below, offers another viewing angle.



*Figure 21. Completed bearing durability test machine.*

The carriage is constrained in the axial direction by means of the flexure discussed previously. Its tapped end simply screws into the body of the carriage. In turn, the flexure screws into a load cell on its opposite end. It is this load cell that measures the frictional load within the bearings. The load cell is rested upon a piece of extrusion while its end is bolted to a plate that in turn is bolted to the underlying piece of extrusion, as seen previously in Figure 19

The carriage is further constrained by the cylinders to which it is attached. In most cases, one terminal of the compact load cell screws into carriage while the other terminal screws into a compliant coupling. The compliant coupling allows small misalignments between the carriage's mounting points and the pneumatic cylinder. In some cases, a smaller coupling is desirable or, for some reason, the load cell cannot be screwed directly into the carriage at the loading point. An alternate design incorporates a smaller of coupling, though it only compensates for translational offsets, and not radial ones. In this case, the coupling is attached to the carriage and the load cell bridges the coupling and the pneumatic cylinder. Figure 24, below, details both mounting schemes. The pneumatic cylinders themselves are simply clamped to sections of extrusion. If necessary, spacing blocks can be inserted between the cylinder and extrusion.

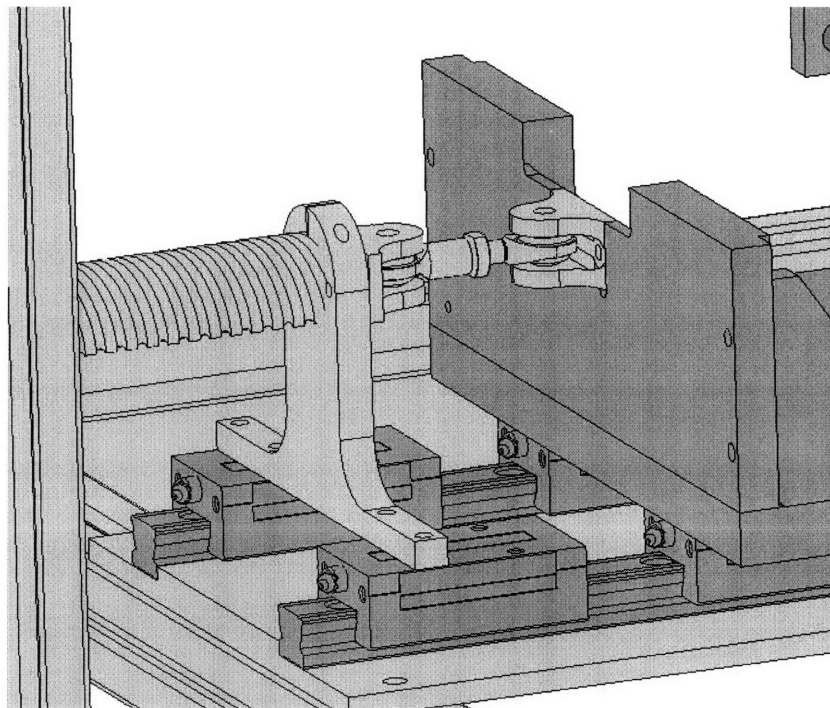


*Figure 22. Two cylinder-carriage coupling schemes. The upper cylinder uses a coupling that allows for both translational and radial misalignment. In addition, the cylinder is clamped to the extrusion with a spacer block as an offset. The lower cylinder is clamped directly to the extrusion. The coupling is attached directly to the carriage and allows for only translational, not radial, misalignments.*

Finally, the carriage is coupled to the box through the device under test. The box however, rests upon its own, very sturdy, bearing system. These very large bearings are sufficiently sized to withstand potential loads and moments over many lifetime tests, ensuring that they will not appear in the data for the devices under test. The box mounts to the bearings below by means of several bolts. Pins pressed through the base of the box provide fine positioning for the bearing trucks. The rails below mount to a plate that runs the length of the bearing rails. Corners machined into this plate provide mounting points for alignment for the rails.

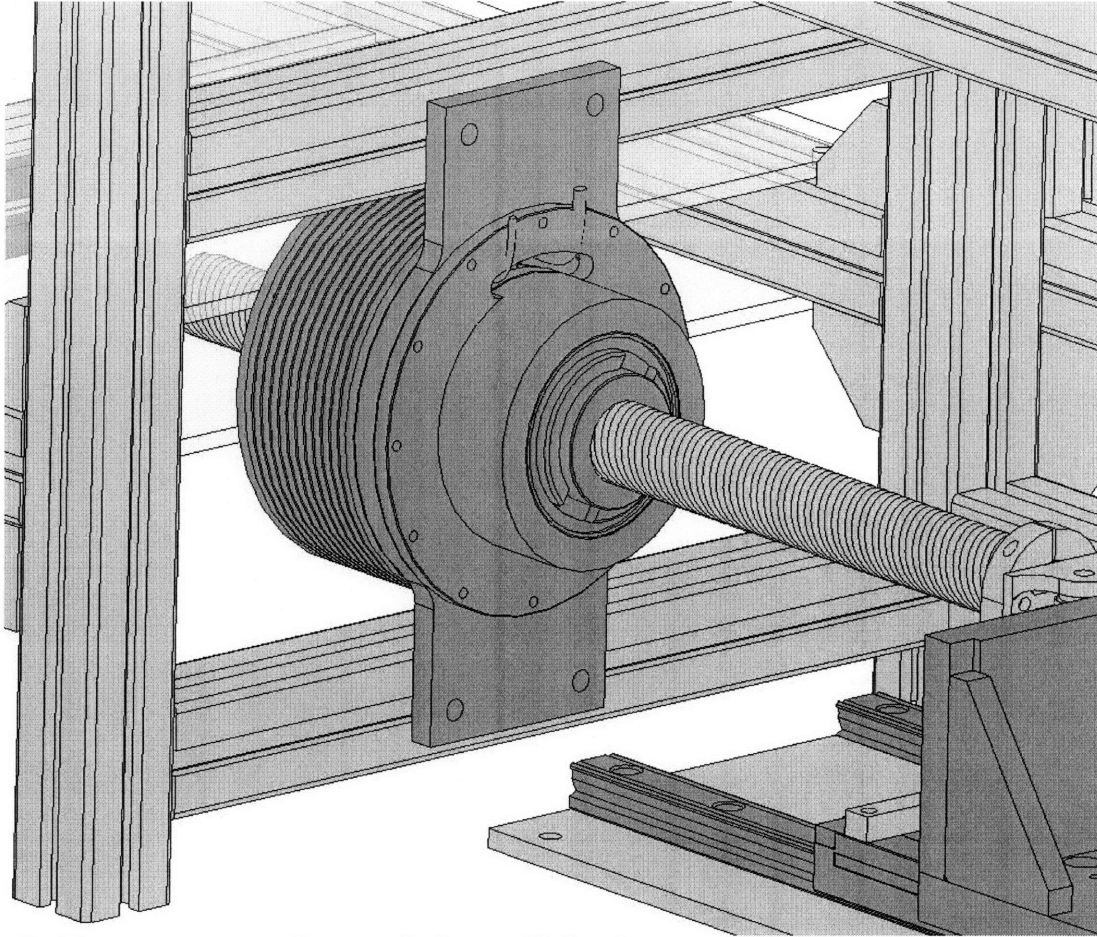
The box itself is supported by two pairs of bearing trucks. A third pair of bearing trucks is also included on the same rail system. This pair guides the ball screw. A mounting plate is bolted to the bearing trucks and clamped around the outside of the ball screw. Since the screw is composed of case-hardened steel, instead of attempting to tap into the end of the screw, a portion at the end was turned down to a smaller radius, around which the mounting plate could be clamped. The mounting plate couples the ball screw to the box by means of two rod end

bearings. This interface is shown, below, in Figure 25. The fact that the two are on the same bearing rails accounts for some potential alignment errors. However, the double rod end bearing system allows for possible vertical misalignment and, more importantly, mitigates loading to the ball screw in any direction except the axial. While the ball nut used is sized to withstand high loads, the less off-axis force on the ball screw, the higher the acceleration and maximum speed it will be capable of, therefore increasing the capacity of the machine.



*Figure 23. The ball screw connects to the box via two rod-end bearings. These bearings ensure that the motor receives minimal side loading and account for any vertical misalignment between the ball screw and the box.*

Finally the motor assembly is attached to the machine's skeleton by means of several bolts, as shown in Figure 26, below. Installing this part requires patience and precision in order to properly align it with the large bearings that guide the ball screw.



*Figure 24. The motor mounting plate extends above and below the motor housing. The plate bolts pass through the plate, holding it to the extrusion framing.*

### **Chapter 3: Controlling the Test**

#### *Section 3.1 Load Control*

One critical aspect of obtaining meaningful test data is successfully recreating the conditions that the bearings undergo while in use. Therefore, a control system that can accurately recreate those loads is essential. Essentially there are two potential architectures: open-loop or closed-loop control. While each strategy has its benefits, open loop load control system was selected because of its simplicity and reliability.

The open loop system is easily controlled. A voltage is applied to a variable pressure regulator. A signal from 0 to 10 VDC corresponds proportionally to a pressure from 0 to 10 bar. The dimensions of the cylinder, specifically the surface area, are known. Therefore, controlling the pressure is directly controlling the load.



A key component in this system is the constant load on one side of the cylinder, as discussed previously. To reiterate, in each of the seven cylinders, there are two compartments. One compartment is varied in pressure from 0 to 10 bar. The other side of the cylinder is held constant at a user defined value. This load is set by hand using a simple, manual pressure regulator. Inherent in this process is a significant amount of variability. In order to prevent this from impacting the test, a calibration feature was included in the software program which measured the manually set “bottom side” pressure.

Prior to beginning any testing, both sides of the cylinder are evacuated to atmospheric pressure. Each load cell is averaged over 30 seconds at five hertz in order to determine the DC offset. Subsequently, a constant pressure is supplied to the same compartment of each pneumatic cylinder. Again, 150 load readings are collected over a period of 30 seconds and averaged. Knowing the total load, cylinder dimensions, and pressure on one side of the cylinder (i.e. atmospheric pressure), the pressure on the other side of the cylinder can be calculated. The DC offset for each cylinder and the constant pressure are each stored as global variables for reference throughout the remainder of the program’s execution.

For the sake of completeness, a closed loop solution was also considered. However, it was deemed unnecessary. Primarily, the open loop system was able to satisfactorily control the load on each of the cylinders. As a result, the more complex closed loop system was not necessary. One such complication is “cross-talk” between the load application elements. In other words, if load is applied through only one cylinder, load is registered not only in the cell that is in series with that cylinder, but with the surrounding cylinders as well. In theory, it would be possible to construct a matrix plotting the cross talk between each of the components, and incorporate it into a control system. That level of precision, however, was not necessary.

Finally, the inclusion of a closed-loop component within the open-loops system allowed compensation for potential changes in the physical plant over time, solving one of the problems that closed-loop control solves. The variable pressure regulator has a closed loop controller pre-programmed within it, thereby ensuring a constant load over the length of the test.

### *Section 3.2 Motion Control*

In closing the position loop, I worked closely with another engineer who has significant experience in automated robotic systems. I proposed a relatively simple closed-loop system to him, incorporating a Renishaw non-contacting linear encoder. This encoder is easy to apply to the system and can be used in several different geometries. Furthermore, it is highly accurate, meaning that a wide range of tests can be run, from sinusoidal sweeps to high frequency dither motions.

In implementing the system, we determined that the LabView interface was not the ideal platform to perform closed loop control. Instead, a micro controller was used. Several different test profiles may be written in any text editor and loaded to the controller. The micro-controller communicates with the controlling PC via a serial interface (RS-232), and is capable of loading different profiles as necessary. In addition, LabView stores position data in order to correlate it with frictional loading data to better understand how load varies with the relative position of the bearing rails and bearing trucks.

### *Section 3.3 Software Architecture*

The user interface is organized into a tabbed environment. Three tabs are organized based on the major phases of testing: calibration, test execution, and data analysis. The tabbed interface offers easy navigation within the software and also serves to guide the user through a test procedure.

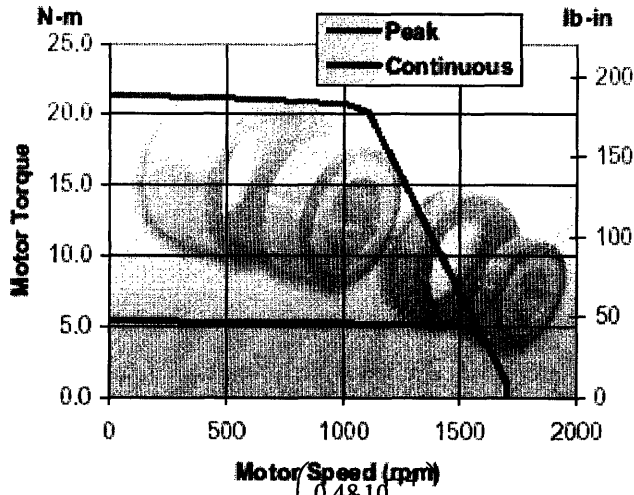
Overall, the software is designed in a way similar to the hardware: flexible. For example, the calibration system discussed previously allows the user to set the constant pressure, thereby increasing test flexibility. Similarly, two types of interface are available while running the test. For the casual user, there is a simple interface which allows control of only the major attributes (i.e. force in each of two directions and moments in three directions). The software takes the load readings and calculates the relevant parameters. For the more advanced user, each cylinder can be controlled independently.

### **Conclusion**

Overall, the goal of this project was to design and build a machine capable of testing bearings as well as write the software to control the machine and run tests. Since the bearings of interest are for use in automotive suspensions, the machine needs to be capable of applying significant loads and moments in all directions as well. Furthermore, the bearings need to be moved at speeds approaching those that they would experience in an automotive setting. These characteristics resulted in the design that has been presented here. The skeleton allows significant flexibility in designing different tests for different bearings. The load application structure reduces the moving mass in the design and increases the safety of the machine. Finally, the prime mover and load application elements have been controlled in order to successfully run tests. Further work could focus on several areas including detailed design on the bearing box and carriage for specific applications and optimization of the software for long-life testing. Nevertheless, the result of this process is a fully functional machine capable of testing bearings under conditions that they might see in automotive suspension applications and recovering pertinent data from those tests.

# Appendix I: Ball Screw Selection

## Aerotech S-130-60-A



$$P := \begin{pmatrix} 16 \\ 20 \\ 25 \\ 32 \\ 36 \\ 40 \end{pmatrix} \text{ mm} \quad I_{nut} := \begin{pmatrix} 0.48 \cdot 10^{-4} \\ 1.44 \cdot 10^{-4} \\ 3.23 \cdot 10^{-4} \\ 6.74 \cdot 10^{-4} \\ 16.8 \cdot 10^{-4} \\ 27.9 \cdot 10^{-4} \end{pmatrix} \text{ kg} \cdot \text{m}^2 \quad M_{shaft} := \begin{pmatrix} 1.41 \\ 2.25 \\ 3.52 \\ 5.83 \\ 7.34 \\ 9.01 \end{pmatrix} \frac{\text{kg}}{\text{m}} \quad I_{rotor} := 0.003 \text{ kg} \cdot \text{m}^2$$

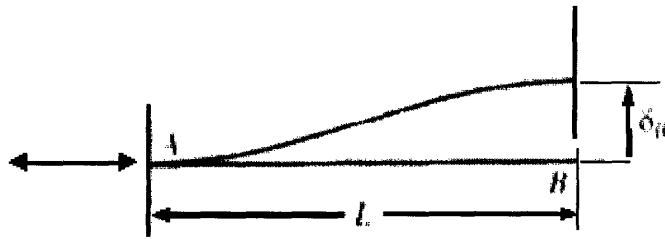
$$I_{connect} := 0.000 \text{ kg} \cdot \text{m}^2$$

$$\omega(t, i) := \begin{cases} \frac{\text{torque} \cdot t}{I_{rotor} + I_{connect} + I_{nut}_i + \frac{M}{\eta} \left( \frac{P_i}{2\pi} \right)^2} & \text{if } \frac{\text{torque} \cdot t}{I_{rotor} + I_{connect} + I_{nut}_i + \frac{M}{\eta} \left( \frac{P_i}{2\pi} \right)^2} < \text{max\_rot} \\ \text{max\_rot} & \text{otherwise} \end{cases}$$

$$v(t, i) := \omega(t, i) \cdot \frac{P_i}{2\pi} \quad x(t, i) := \int_0^t v(t, i) dt \quad a(t, i) := \frac{d}{dt} v(t, i)$$

Pitch [mm]	16	20	25	32	36	40
Approx Time Per HalfCycle [sec]	0.6	0.5	0.43	0.37	0.38	0.4
Approx Time Per Million Cycles [hours]	333.3	277.8	238.9	205.6	211.1	222.2

Appendix II: Flexure Rod Selection



Material/Geometric Properties:  $E := 69\text{GP}\epsilon$      $K_t := 1$     Rod Length: flexure\_length := 0.42r  
 Buckling Loads:  $F_{b_1} := 112.5\text{N}$      $F_{b_2} := 225\text{N}$     Fatigue Loads:  $\sigma_{\text{bend}_1} := 95\text{MPa}$      $\sigma_{\text{bend}_2} := 310\text{MPa}$   
 Sideload:  $F_{s_1} := 900\text{N}$      $F_{s_2} := 450\text{N}$     Vertical Displacement:  $\delta_B := 25\text{mm}$      $\delta_{b_1} := 25\text{mm}$

$$L_s(r, i) := \left[ \frac{3 \cdot E \cdot \pi \cdot (r \cdot 10^{-3})^4 \cdot \delta_B}{F_{s_i}} \right]^{\frac{1}{3}} \quad \text{RED - sideload}$$

$$L_b(r, i) := \sqrt{\frac{E \cdot \pi^3 \cdot (r \cdot 10^{-3})^4}{4 \cdot F_{b_i}}} \quad \text{BLUE - buckling}$$

$$L_{\text{bend}}(r, i) := \sqrt{\frac{6 \cdot K_t \cdot E \cdot (r \cdot 10^{-3}) \cdot \delta_{b_i}}{\sigma_{\text{bend}_i}}} \quad \text{GREEN - fatigue}$$

BLACK DOTTED - shaft length, stock shaft radii

The "design envelope" is bounded by the red, blue and green curves.  
 The "acceptable design direction" is towards thi respective dotted lines.

