

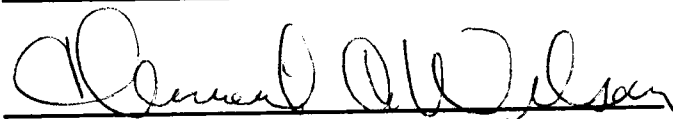
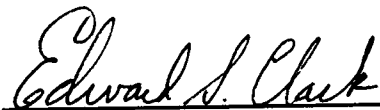
To the Graduate Council:

I am submitting herewith a thesis written by Jeremy Allen Davis entitled, "Development of a Force Guided Translation System For Automated Fender Finishing". I have examined the final copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Mechanical Engineering.



Frank H. Speckhart, Major Professor

We have read this thesis
and recommend its acceptance:



Accepted for the Council:



Associate Vice Chancellor
and Dean of the Graduate School

**Development of a
Force Guided
Translation System For
Automated Fender Finishing**

A Thesis

Presented for the

Master of Science Degree

The University of Tennessee, Knoxville

Jeremy Allen Davis

May 1995

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I would like to acknowledge the many people who have contributed to the success of this project. Through each step of my education I have been blessed with just the right mentors, friends, and family who have unselfishly contributed the necessary ingredients required for me to attain my goals.

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There is no substitute for a supportive family, and that is certainly the case with this

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ABSTRACT

The translation system has been developed as a test-bed to perform parting-line removal experiments on Saturn's plastic fenders. This project will be combined with sanding and cutting tool end-effectors in an effort to produce a fast and reliable parting-line removal method.

Since the fender parting-line represents a complex trajectory, a guidance system was developed that follows the parting-line based on the fender itself. The top of the fender acts as the template by which two specialized guide wheels follow its path. The forces exerted against these wheels are translated through a linear rail system which selectively restricts some motions while allowing other motion the freedom to follow the fender.

As the guide car follows the complex trajectory it is important that the end-effector maintains a constant force with the parting-line so that it finishes the surface consistently. Force control is achieved by attaching the end-effector to a compliant air cylinder. Through preset pressure levels, this air cylinder maintains forces between the fender and the end-effector. In addition, it can be set to add or subtract the downward force of the end-effector; therefore, allowing for precise and variable control between the end-effector and fender.

The force compliant guidance system and end-effector attachment system is a very

economical approach while also maintaining the robustness required in an assembly environment. Given these features, a wide variety of end-effectors can be attached and used to develop the best method for removing the parting-line.

The proto-type translation system has the ability to traverse 91% of the fender in 30 seconds (a speed of 4 in/sec). The effectiveness of the sander or cutting tool varies when the translator is operated at full speed, therefore, the system is slowed down according to the needs of a particular end-effector. Test results have shown that the cutting tool can perform its desired cut while moving at 1 in/sec. Similarly the sander can be operated at 0.25 in/sec and the desired fender profile is obtained. The proposed final system will use these two methods in sequence to successfully remove the fender parting-line.

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CHAPTER 1

BACKGROUND

Problem Statement

The Parting-Line Removal Investigation Team was formed in 1993 and charged with the development and demonstration of an automated parting-line removal system. Saturn's injection molded fenders are currently produced with an unavoidable parting-line located on a "Class A" surface (defined by Saturn to be a surface that must be cosmetically perfect and appealing to the customer). The manual removal of this parting-line has proven to be a tedious, expensive, and inconsistent process. With these as the motivating factors, a complete investigation was launched to explore various automated solutions.

As the investigation of a parting-line removal method progressed, it soon became apparent that an inexpensive and reliable test-bed mechanism would be required to facilitate the proper evaluation of proposed end-effectors. This test-bed would assist in the development of a removal system by integrating end-effectors into a complete parting-line removal process. The development of such a test-bed is the focus of this master's thesis.

History of UT's Investigation

Phase 1 of the Parting-Line Removal Investigation was initiated in May of 1993. The goal of this phase was to investigate and develop a process for removing the parting-line

and shaping the fender. A complete discussion of Phase 1 results can be obtained from Tim Wyrick's thesis [Wyrick 95].

Phase 2 began in January of 1994 and was charged with implementing a translator to move the end-effectors along the parting-line. The following thesis describes the investigation, design, and conclusions that were developed as a result of Phase 2.

It is interesting to note that between the inception of Phase 1 and the completion of Phase 2, Saturn's fender design changed three times. With each new design change Saturn provided only driver side fenders with the expectation that passenger side fenders would simply require a mirror solution. Figure 1.1 shows a side view of the SL-2 and the 1996 fender (an SL-3 view was not available). Although we aspired to create a general fender solution, each design change presented new and unpredicted challenges. Phase 1 started investigating the fender design SL-2, which placed the parting-line approximately 0.25 inch from the top of the fender. The SL-3 concept fender soon followed with the parting-line placed directly on the top of the fender at the fender-hood surface. We finished the project using the 1996 fender which also places the parting-line in the same position as the SL-3. The 1996 fender presented additional challenges, most of which were a result of the complex curvature at the front of the fender. Details on how this complication was dealt with are discussed in later chapters.

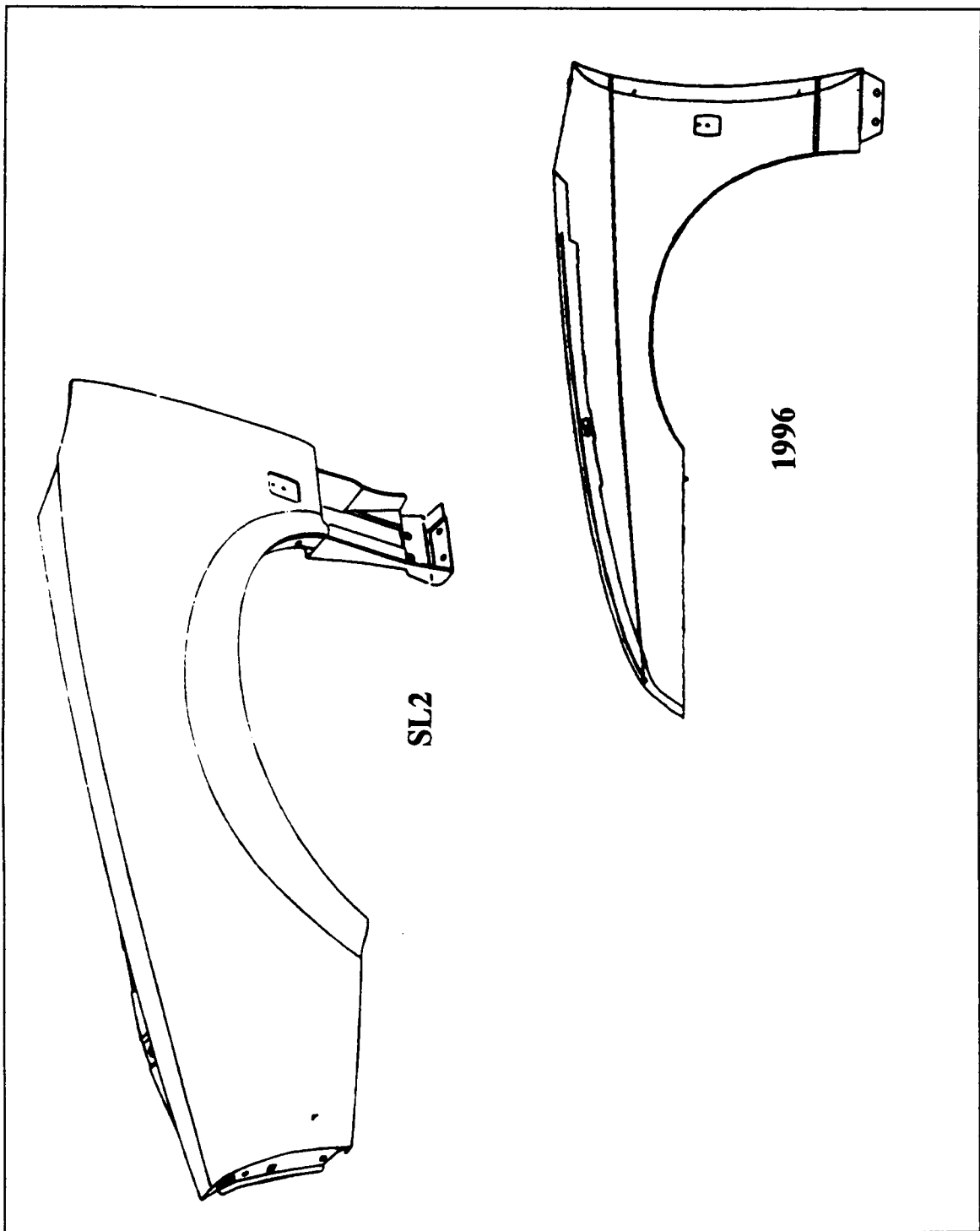


Figure 1.1: Changes in Fender Style: SL-2, 1996.

Source: Saturn Corporation (1995)

Throughout these style changes, our team worked closely with Saturn engineers to find ways of dealing with the unexpected complications. The fender design changes turned out to be useful design tools because each new style required the development of a more robust parting-line removal process. By March of 1995, the hardware and software for the Phase 2 translator was completed and is currently being used to test various end-effector combinations.

Current Manual Method of Parting-line Removal

Saturn's front fenders are made of a nylon polymer (presented in Appendix E) that is injection molded using a 5000 ton injection molding machine. As is common with this process, a small rib of plastic is left on the product where the two halves of the mold come together. Many different configurations have been attempted for the design of these fender molds; however, there is no known method of eliminating this parting-line from the class A surface during the molding process. Figure 1.2 shows a cross section view of the parting-line that has been the focus of this project.

Until an automated alternative is qualified and implemented, the parting-lines are being removed using manual methods. This manual solution involves a three step process of scraping, sanding, and polishing after the fender has been removed from the injection molder for a period of at least 20 minutes (the usual curing time is over one day due to

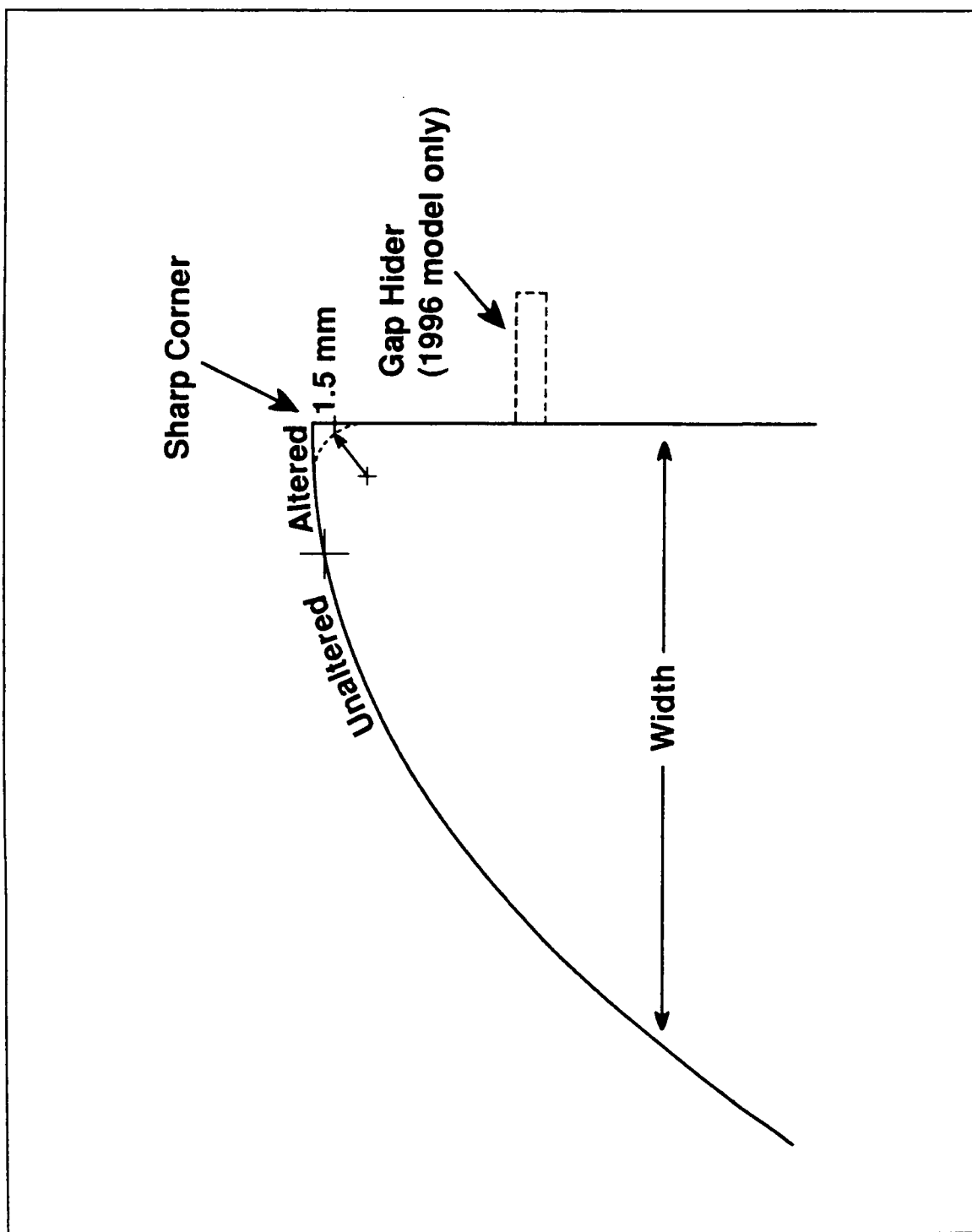


Figure 1.2: Cross-Section Showing Parting-line

Source: Timothy William Wyrick, *The Development of a Parting-line Removal Technique for the Saturn 1996 Model Fender*, M.S. Thesis in Mechanical Engineering, The University of Tennessee, Knoxville, TN, May 1995.

the manufacturing process).

- 1) Scraping - a flat-blade metal cutting tool is used to remove the bulk of the parting-line. This process can result in an uneven and inconsistent surface, yet, is necessary since the next step would take too long to remove all of the material.
- 2) Sanding - an orbital air sander is used to remove and blend the rest of the parting-line. In addition, a small radius must be added to this sharp edge to assist in proper paint adhesion. This step requires a complex set of hand and wrist operations which can lead to such problems as carpal tunnel syndrome and inconsistency in the final fender geometry.
- 3) Polishing - the final step is a quick rub down with a 30 micron hand polishing pad. This step is necessary to remove any small scratches that may have been left by the sanding process.

The cycle time of this three step process is approximately two minutes and includes removing and returning the fender to a storage rack. An additional amount of time that is difficult to account for is spent transporting these fenders to an off-site location where the manual operation is actually performed. The primary motivation for this project is caused by the inconsistencies and quality variation associated with the manual operation.

Therefore, the goal of this project is to focus on improving the over all paint quality and reducing variations in the shape of the fender.

Phase 1 Results

Phase 1 of the project investigated various methods and techniques for removing the fender parting-line. Fender design specifications call for a 1 to 2 mm radius to exist at the top edge of the fender, also described as the fender-hood mating line. The position of this radius also corresponds to the fender parting-line that must be removed and shaped in preparation for painting (shown previously in Figure 1.2).

Results of Phase 1 recommend the use of a contoured cutting tool used to remove the majority of the undesired material. This is to be followed by a belt sander or a manual sanding operation which will remove the final material while also shaping the fender to the desired form [Wyrick 95].

Scope of Phase 2

The goal of Phase 2 was the construction of a translation mechanism capable of moving an unknown end-effector along the path of the fender parting-line. This required the development of a process that can follow the complex surface of the fender with robust repeatability. In addition, the force of the end-effector on the fender must be controlled by a constant force control system.

The goals and scope of Phase 2 can be summarized as follows:

- 1) Design and demonstrate a method for following the parting-line such that the end-effector will always be located in the same position relative to the parting-line. The solution should be as general as possible as to not limit the mechanism to one specific fender design or to one specialized end-effector.
- 2) Design and test a mechanism for controlling the force between the end-effector and the fender surface. This subsystem must be capable of increasing or reducing the effective weight of the end-effector and then holding it constant throughout the process.
- 3) Design and construct a prototype translation mechanism that will work as a test-bed for investigating various parting-line removal techniques. This mechanism will include sub-assemblies that satisfy goals 1 and 2.

It was anticipated that the development of this test-bed may require minor modifications or further investigation of parting-line removal methods. As a result, the system was designed with the greatest amount of flexibility and generality to its environment as possible. The following chapters describe the requirements and summarize the effectiveness of the prototype translator in fulfilling these goals.

CHAPTER 2

DESIGN REQUIREMENTS

General Overview

Phase 2 specified the development of a full-scale prototype translating mechanism to be used as a test-bed for demonstrating various parting-line removal methods. In this context, the word prototype is being used to depict the expectation that a production level translator will require additional safety devices and upgrades to insure durability.

To work as an effective test-bed, it is important that the translator be flexible to changes in fender design, uncertainties in the manufacturing processes, and variations in end-effector requirements. Changing from one fender design to another may require both software and hardware changes, but should not require excessive amounts of reprogramming or re-fixturing. The translator must provide five d.o.f. to the end-effector and provide a constant contact force between the cutting tool and the fender parting-line. The following sections look more in-depth at how these specifications were developed and why they will insure a successful translation mechanism.

Kinematic Requirements of an End-effector

The most important requirement of the entire system is the orientation of the end-effector relative to the parting-line. If this orientation is maintained within a certain tolerance

limit, then successful parting-line removal will depend only on the chosen end-effector.

During the development of these design requirements, it was expected that a sander would be used as the primary end-effector, however, this requirement was changed by the end of the project to focus on the cutting tool as the primary end-effector. Figure 2.1 shows a coordinate system attached to the Phase 1 sander. This coordinate system will be used in the following discussion.

The most obvious movement will be along the length of the fender (noted as X''). The Y'' and Z'' degrees of freedom must also be available so that the end-effector stays in constant contact with the parting-line. Pitch and Yaw are required to maintain the proper orientation when following the curves of the fender.

The only degree of freedom that will not be necessary is the roll (rotation about X''). Based on the SL-3 design, it was determined that the amount of roll that would be necessary was not large enough to effect the needs of the end-effector. However, if the design requirement had been developed with the 1996 fender design in mind, this restriction may not have been made. The 1996 fender does possess a rather distinctive twist (roll and pitch combined) at the front of the fender. The final translator has been successfully adapted to accommodate the 1996 fender; however, the lack of a roll d.o.f. may limit the use of some end-effectors, thus reducing the effectiveness of this test-bed. Based on the kinematic requirements of the sander, a conceptual design was proposed that

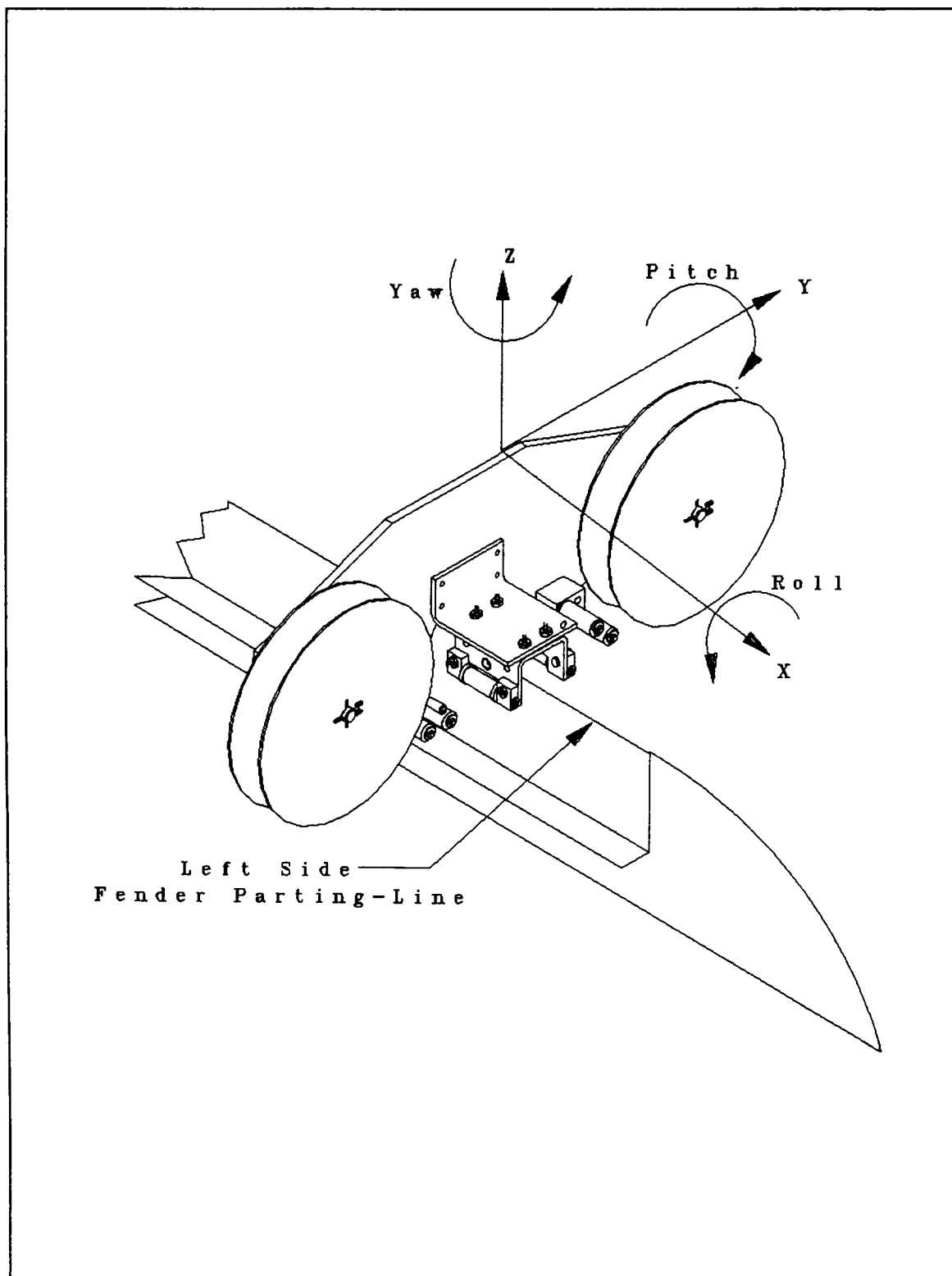


Figure 2.1: Coordinate System Attached to the Sander

allowed for the necessary degrees of freedom. Figure 2.2 shows the four primary subsystems and the kinematic relationships between them. Also, Table 2.1 summarizes the direction of movement that each subsystem must provide. The Z" d.o.f. might be viewed as redundant; however, the following section describes the specialized need of this d.o.f.

Table 2.1: Kinematic Relationship of the Subsystems

Subsystem	Motion Provided or Controlled
Frame	None (stationary)
Platform	X, Y, yaw (Z')
Vertical Rod	Z'
Guide Car	pitch (Y')
End-effector Attachment	Z"

The minimum range of motion required to successfully traverse a fender, is summarized in Table 2.2. These values include a factor of safety that will allow for the maximum expected change in fender design features.

Table 2.2: Minimum Required Range of Motion

Direction	Minimum Travel	Actual Designed Travel
X	85.0 in	100.0 in
Y	10.0 in	40.0 in
Z	7.5 in	22.5 in
pitch	± 20 degrees	± 75 degrees
yaw	± 90 degrees	± 180 degrees
Z"	± 1.0 in	± 1.0 in

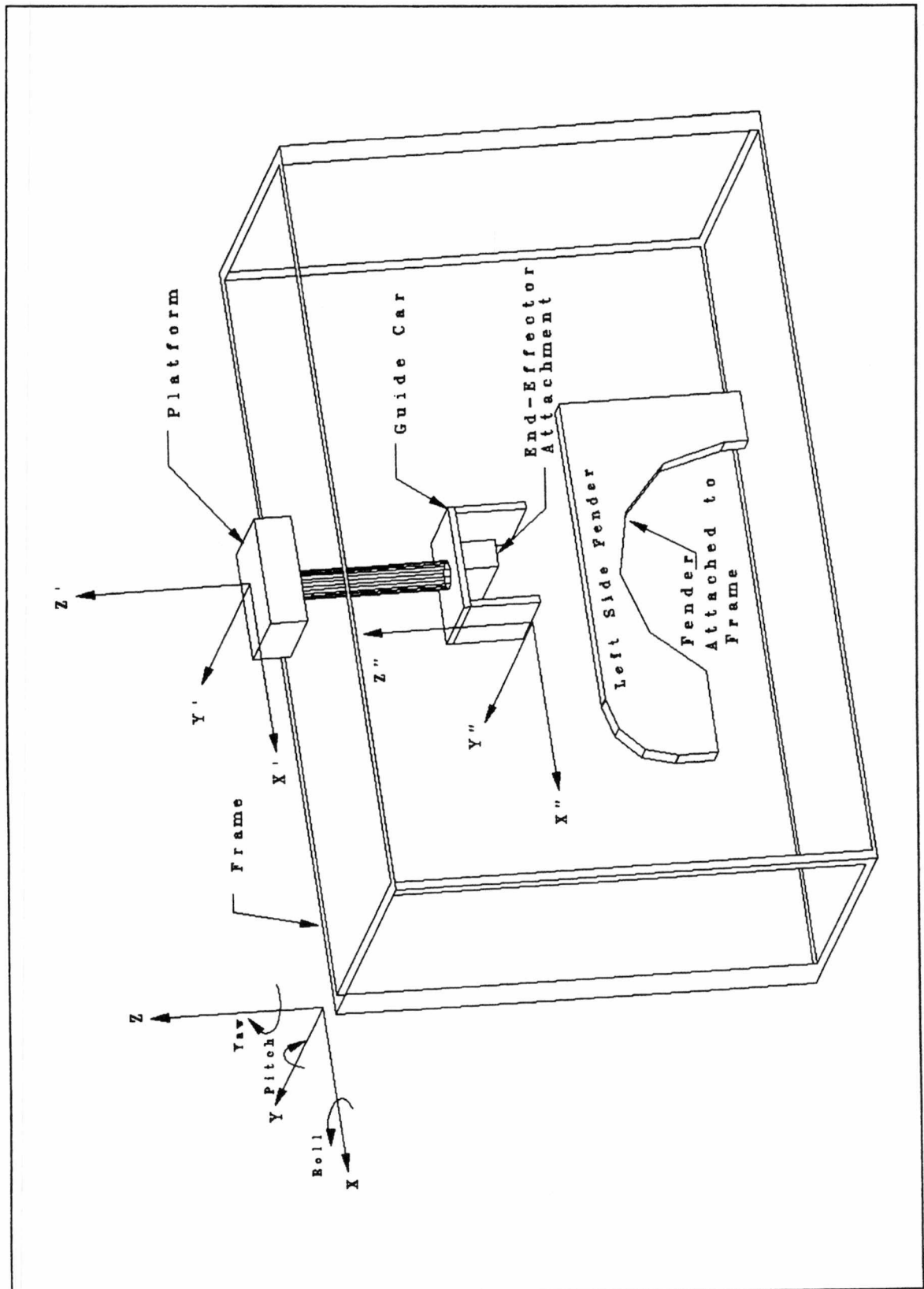


Figure 2.2: Coordinate Systems and Kinematic Relationships for the Translator

The preceding kinematic requirements were carefully considered against a wide range of potential end-effectors. It was concluded that a test-bed providing these degrees of freedom would allow for the investigation of end-effectors including belt sanders, orbital sanders, cutting tools, routers, or any combination of these tools.

Sixth D.O.F: Perpendicular Force

The next requirement of the system is to maintain the contact force between the end-effector and the fender. This can be thought of as a sixth d.o.f. in that the end-effector must move relative to the guidance car so that it does not lose contact with the fender or dig into the fender. Figure 2.3 shows a hypothetical situation that may occur if the end-effector were held rigid inside the guidance car and the distance between the end-effector and fender was allowed to change. The force-regulating system must compensate for these relative distance changes and maintain constant force.

The goal of constant force can be obtained through two different approaches. The least practical would be to specify a programmed translation system that would keep the end-effector in perfect orientation relative to the fender. Due to the uncertainties of a real-world system, this approach is too costly and impractical. The more practical solution is to design a system that will maintain constant perpendicular force between the end-effector and the fender by mechanical feedback.

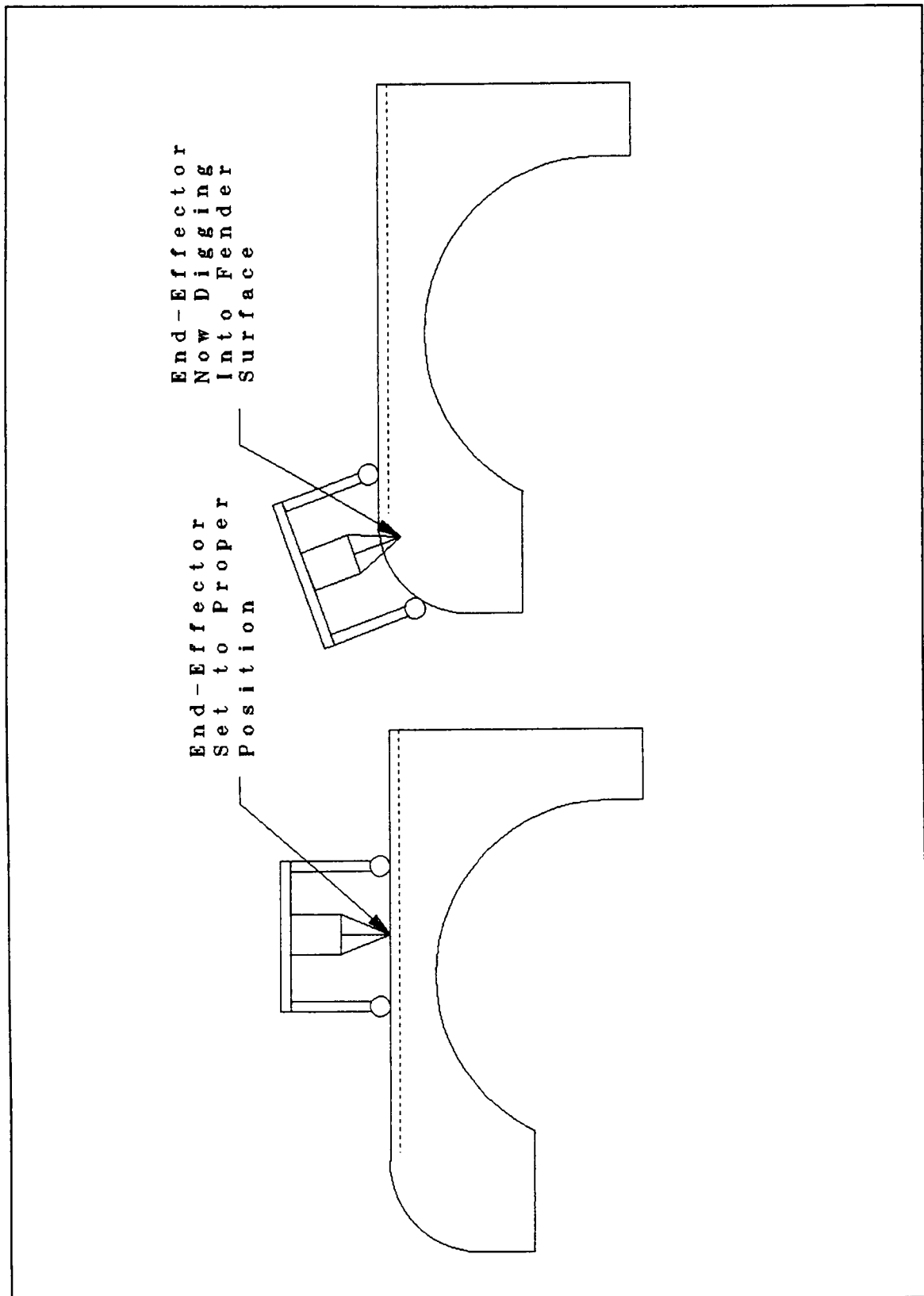


Figure 2.3: Example Showing the Necessity of End-effector Force Control

An additional requirement is that the perpendicular force must be adjustable by the user and remain constant throughout the parting-line removal operation. This feature will allow the user to increase or decrease the effective weight of an end-effector based on the characteristics of that end-effector.

Additional Requirements

To insure a successful project and to help focus the design, the following design requirements were also specified.

1. The fender must be manually loaded and remain stationary during the parting-line removal process. The main reason for this requirement is to obtain as small a foot print as possible for the entire machine, reduce safety risks, and eliminate the added complexity of automated loading.
2. Desired cycle time is approximately one minute per fender. This can be broken down into 30 seconds for the manual loading and unloading of a fender and 30 seconds for the parting-line removal. The automated portion of the cycle may change as a result of the end-effector; however, the design specifications require that the translation mechanism accurately traverse the fender within 30 seconds.

3. The reliability and robustness should be such that the prototype can demonstrate the effectiveness of the tracking method, force-regulating mechanism, and various end-effectors. It is also understood that a production level machine will require additional safety devices and upgraded hardware to increase durability.
4. Size of the translator should be kept to a minimum. This will conserve floor space while making the mechanism easier to move and relocate.
5. One of the key elements to creating a robust translator will be its rigidity. While the goal is to make a mechanism that is not dependent on a coordinate system, positional stability of the various components (i.e. fender stand and fender mounting hardware) will be a very important concern.
6. Weight will most likely not be a concern since the translator will be located in a manufacturing environment capable of dealing with heavy equipment. The use of weight must be balanced between a desire to create rigidity and eliminate vibration while also conserving material. Weight distribution should be of concern to insure that the device is not "top heavy" or prone to cause injuries.

Armed with these requirements, a thorough investigation was launched to examine every possible approach for a translator. One idea that was discarded was the use of a six d.o.f. general positioning robot. This approach required a great deal of precise coordinate-based programming that would result in a solution biased to one fender design. In addition, general coordinate-based robots are not robust when dealing with large payloads (such as a 27 lb sander) or positional uncertainties due to manual loading and unloading.

Other ideas included mounting the fender upside down; however, this approach would favor only a limited number of end-effectors. Although many ideas were considered and pursued, the most robust and promising concept was implemented. The initial results and detailed design for the translator are described in the following chapters.

CHAPTER 3

PROJECT RESULTS

Upon completing the development of the first prototype translator (shown in Figure 3.1), each of the three goals has been completed with varying degrees of success. To reiterate, the goals of Phase 2 were to prove a method for following the parting-line, control the force between the end-effector and fender, and demonstrate the mechanism as a test-bed. The most successful and primary result of the translator is its usefulness as a test-bed and its ability to motivate a large variety of end-effectors. The following sections provide a quick summary of the quantified results that have been obtained from the prototype translator. Further discussion of the results is presented in Chapters 6 and 7 and results specific to the end-effectors can be obtained from Tim Wyrick's thesis [Wyrick 95].

Translator Results

Without considering end-effectors, the translator has been demonstrated to move from its home position, traverse 91 percent of the fender, and return to its home position within 30 seconds. Reliable tracking of the front nine percent of the fender (four inches) has not been achieved with this prototype (see Figure 3.2), and Chapter 7 presents recommendations for improving and reaching this goal. To achieve the desired 30 second cycle time, the translator is set to a constant speed of 4 in/sec. This speed, however, is not optimal for every end-effector, and therefore, must be adjusted to obtain the best

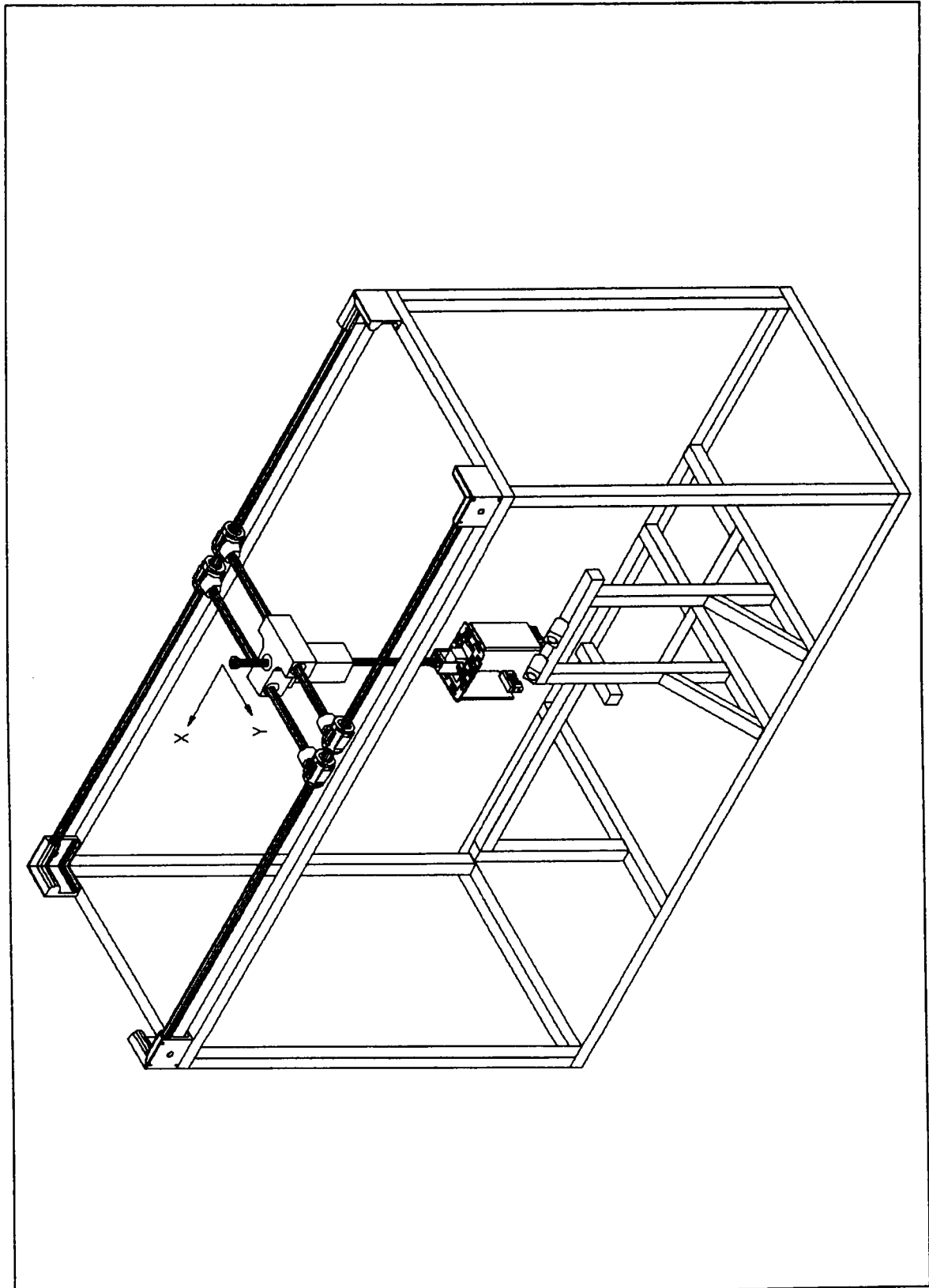


Figure 3.1: Composite Drawing of Final Translator

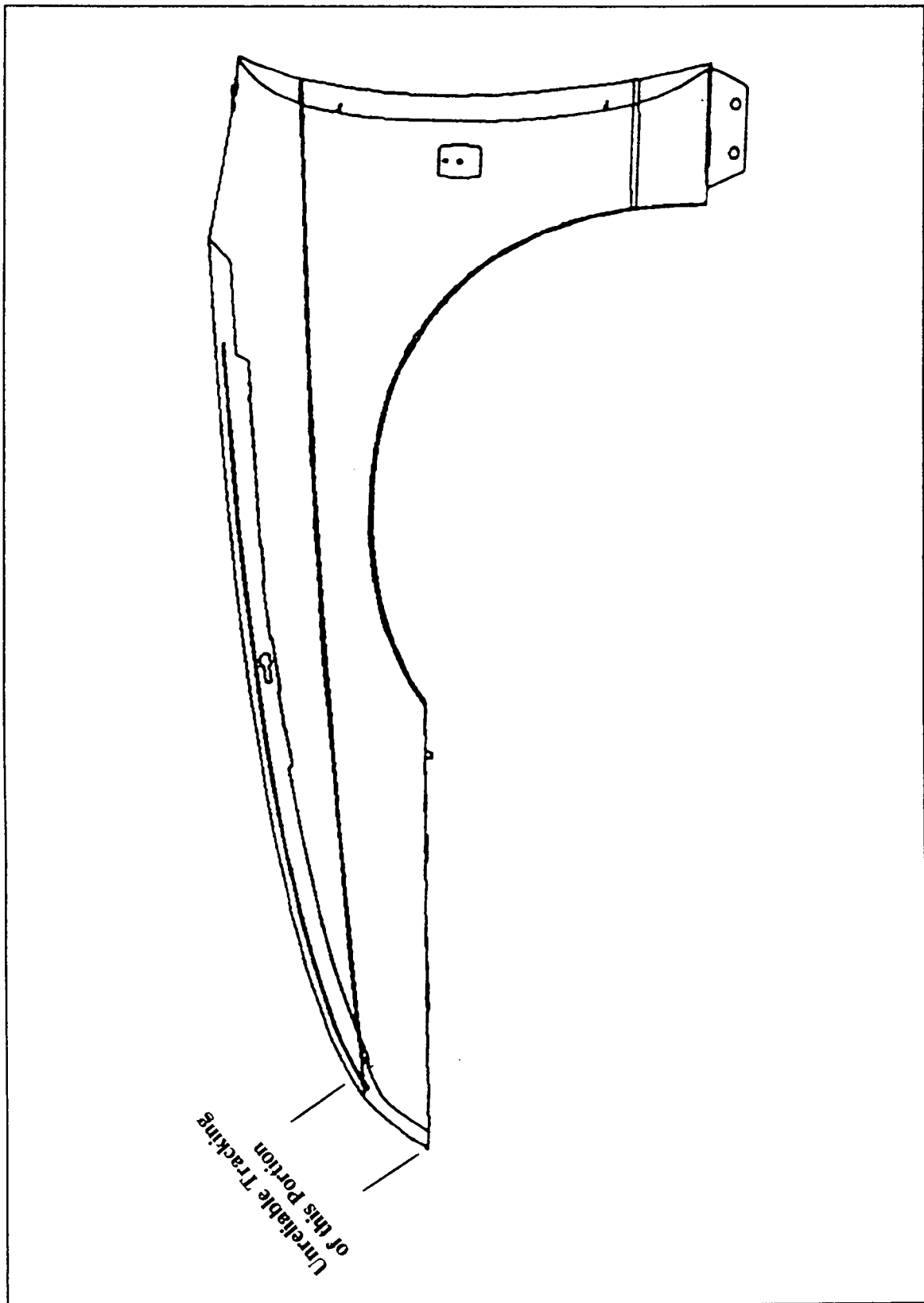


Figure 3.2: Section of Fender that is Not Reliably Traversed

performance and desired finish. The translator's speed capabilities have been experimentally tested, and the available range is 0.02 in/sec to 5.6 in/sec. To ensure reliable operation and maintain proper phase within the motors, the software currently restricts the speed to a recommended range of 0.1 in/sec to 4 in/sec.

The translator follows the parting-line with a relative positional accuracy of $\pm .010$ inches. Again, this claim is only valid for the portion of the fender that is accurately traversed. As the second major goal of the project, the guidance mechanism has demonstrated the high reliability and tracking abilities that were expected. Using two contoured wheels, the guide car follows the top ridge of the fender regardless of fender style, position on the translator (within reason), or uncertainties that may arise from manufacture. This mechanical feedback system has also proven to be the most effective means of translating high weight end-effectors such as the sander (weighing 27 lbs) across a complex profile. The next step in developing the guidance system will be to increase the robustness of the tracking system to complete the front four inches.

The final innovation of the project, that of maintaining constant force between the fender and the end-effector, has met with numerous difficulties. On the positive side, tests have shown that this feature may not be as critical as initially suspected during the development of the product specification. The purpose of the force control subsystem is to allow the user to choose a downward force value and for the system to maintain this force throughout the translation of the fender. The mechanism used in this prototype

provided a force range of -0.5 lbs to +6 lbs. A very small reduction in force between the end-effector and the fender will result in almost immediate correction (this is mainly due to the assistance of gravity). However, an increase in force between the end-effector and the fender can reach as much as 6 lbs before the system will move to reduce the force. The arguments for developing a force control system were discussed in Chapter 2, Product Specification. Further testing will be necessary to determine if an improved system will be required, and recommendations for such improvements are presented in Chapter 7.

Cutting Tool Results

One of the two end-effectors tested on the translator was a cutting tool designed to remove the majority of the parting-line. Iterative experiments showed that the maximum speed at which the cutting tool could be moved and still provide a consistent, controlled cut, is 1.0 in/sec. Based on this speed, a fairly consistent removal of fender material was performed over specific portions of the fender. The final conclusion for the cutting tool is that it does produce the desired result; however, a more sophisticated and robust design will be needed for mounting it to the translation system. Binding caused by the attachment mechanism has limited the cutting tool to completing only 57 percent of the fender.

Sander Results

The sander that was developed in Phase 1 has been thoroughly tested on the translator and the following conclusions can be made. Most importantly, the recommended translation speed while operating the sander is a constant 0.25 in/sec. This speed is in conjunction with the sanding belt moving at it's maximum velocity of 14.44 in/sec. Using these two settings, and a downward air pressure of 10 psi, the successful sanding of a fender will require 6 feet of sanding belt per inch of fender. The sanding operation will, therefore, require 4 minutes per fender and will consume 270 feet of sanding belt. It is important to note that these results have been determined based on the best observations of the graduate students involved. Final approval of the surface finish will be required by Saturn.

Figure 3.3 shows the recommended orientation of the sander with respect to the translator. In addition, a better surface finish was achieved by the sander when it was translated from the front of the fender to the back (left to right with respect to Figure 3.1 or Figure 3.3).

This chapter was presented as a quick overview of the most important and significant project results. The next chapter describes the design and implementation of the actual translator. Following this, a more in-depth discussion is presented concerning the effectiveness and reliability of the system as well as recommendations for further development.

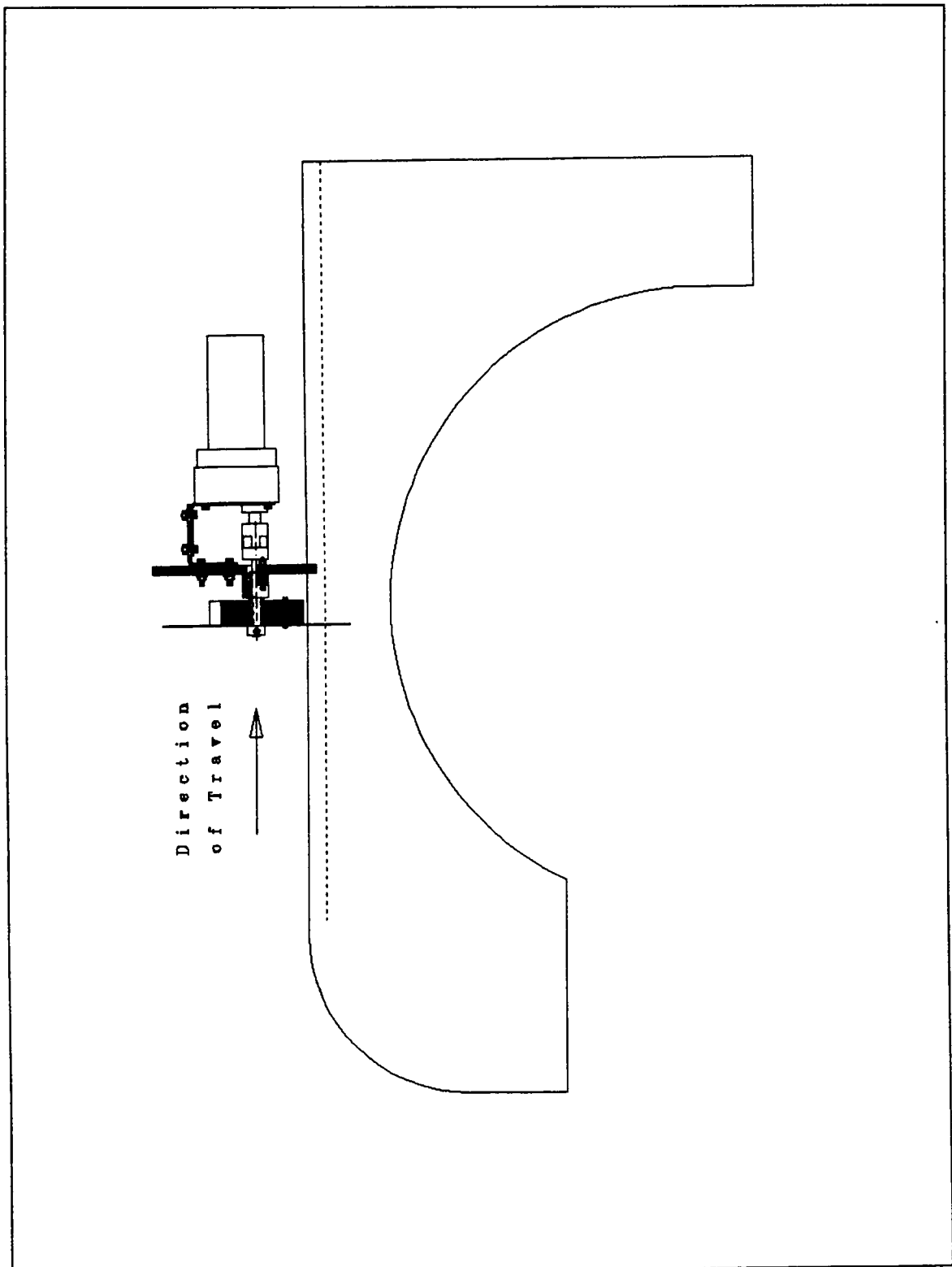


Figure 3.3: Orientation of Sander on Fender

CHAPTER 4

DETAILED DESIGN

The following section describes the detailed design and implementation of the translation system that successfully completed this project. Discussion of the translator will be broken down into its major sub-assemblies which consist of the frame, rail system, guidance car, and supporting hardware. Included for each subassembly will be a complete description of how the design was generated and any minor adjustments that were required for implementation.

Frame

The main supporting structure for the translator is taken from the "UT Robot" developed by a number of graduate students originating in 1990 [Barrett 92, Kennedy 91, Paulus 92, Williams 92]. Using Kennedy's design drawings and Barrett's construction methods, the major change required was to adjust the size of the frame to encompass an entire fender and additional system hardware. The result is a frame that is more than twice the length in the X and Z directions and similar in size in the Y when compared to the UT Robot (see Figure 2.2 for coordinate notation).

Before final construction was initiated, the results of Tony Barrett's thesis project were incorporated. This addition allowed the top and bottom of the frame to be welded.

Although this reduced the break-down capability of the robot, it greatly enhanced the strength and rigidity of the structure.

The top and bottom of the frame consist of 2 inch equal-leg angle "iron" with a 0.125 inch wall thickness. In retrospect, 0.25 inch wall thickness should have been used to facilitate a better welding process and increased stiffness. The vertical support beams (consisting of 2 inch square tubing, 0.25 inch wall thickness), although difficult to manufacture due to their length, serve their purpose quite well without flexing.

The final frame measures 110 inches in X, 44 inches in Y, and 68.5 inches in Z. The addition of 2 inch leveling feet does not alter the workings of the translator and could be eliminated if necessary. Based on preliminary observations, the chosen frame system has proven to be a good selection for this application. While the obvious advantage is its enormous cost savings, it also facilitates easy installation and mounting of robot components.

Primary Guide Rail System

This section discusses the primary X, Y, and Z degrees of freedom and why they are necessary to perform the required translation task. The guide system can be further broken down into the drive system, (consisting of the motors, motor mounts, corner blocks, belt system, and X shaft), the Y system (including the sliders, Y shaft, and platform), and the Z system (simply a shaft running vertically through the platform).

Corner Blocks

The drive system is very similar to that used in the "UT robot". Again the research and design has already been proven on a previous robot, and since the application was very similar, only minor modifications were needed. The primary alteration was actually a simplification. Since the translator requires only one stationary axis, the corner blocks were only milled and bored on one side. The largest possible diameter, 1.25 inch, was used for the X guide system. While the design did not require more than a 1.00 inch shaft, the increased rigidity of the 1.25 inch was determined to be worth the increase in cost. Figure 4.1 shows the revised corner blocks.

Drive System

With the corner blocks mounted on the frame, the motor plates and motors are attached in the same manner as the "UT robot". Motivational power is provided by two 200 step/revolution stepper motors operating in half step mode.

When operated in the recommended speed range, the two stepper motors provide adequate, synchronized torque and force. The torque of the motors is further enhanced by the use of a 4:1 reduction gear and belt system. The larger gear is then directly connected to the belt drive system. Originally, there was a great deal of concern over the possible stretching of the nylon reinforced belts that are directly connected to the sliders. Although the belt lengths are considerably longer than the ones used on the "UT Robot", the drive system has performed flawlessly. Endurance testing may prove that these belts

Modified Corner Blocks

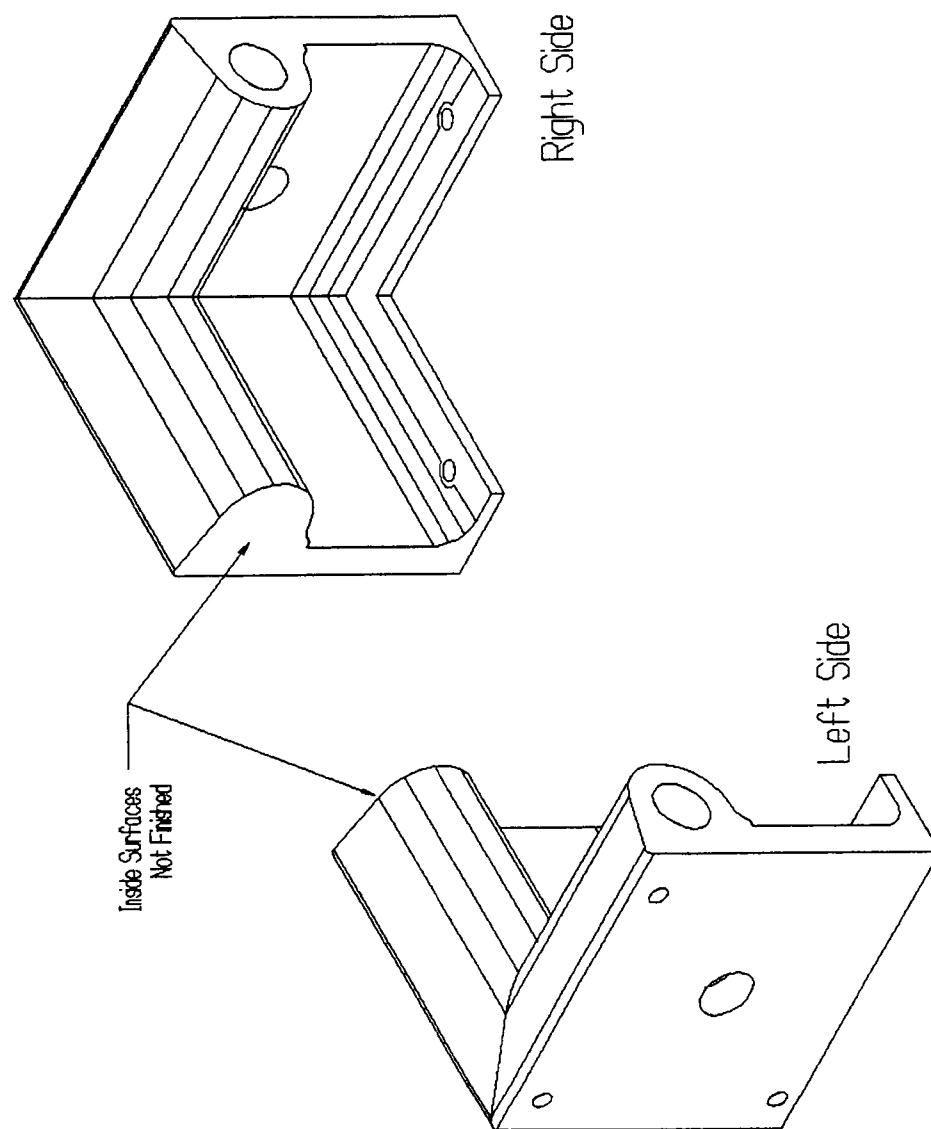


Figure 4.1: Revised Corner Block

actually do stretch, and a more durable belt may be necessary for a production-level device.

Synchronized control of the two stepper motors is accomplished using a Turbo C program to generate pulses for two IMS IB 463 stepper motor drivers. This driver system was not used by the "UT Robot", and the wiring schematic can be found in Appendix D.

The software code that controls the translator provides the user with complete control over the distance and speed at which the translator will travel. Since this mechanism is meant to be used as a test-bed, the software control is very flexible. Variation of the code presented in Appendix C can be adapted for specific applications and end-effectors.

Rail System

As shown in Figure 4.2, four sliders are mounted on the X guide rails. These sliders support two parallel 1.00 inch shafts which allow the platform (discussed later) to move back and forth freely. Force from the drive system is directed to these sliders, and they in turn move the platform (which indirectly moves the end-effector) in the X direction. There was a great deal of concern over whether or not the sliders should be rigidly coupled together to maintain proper parallelism and allow free movement of the platform. Experimental results have shown this to be unnecessary, and there are a number of reasons for this robustness. First, the belt drive system is only attached to one slider on each side, thus, only one of the parallel shafts is being driven. The other shaft is free

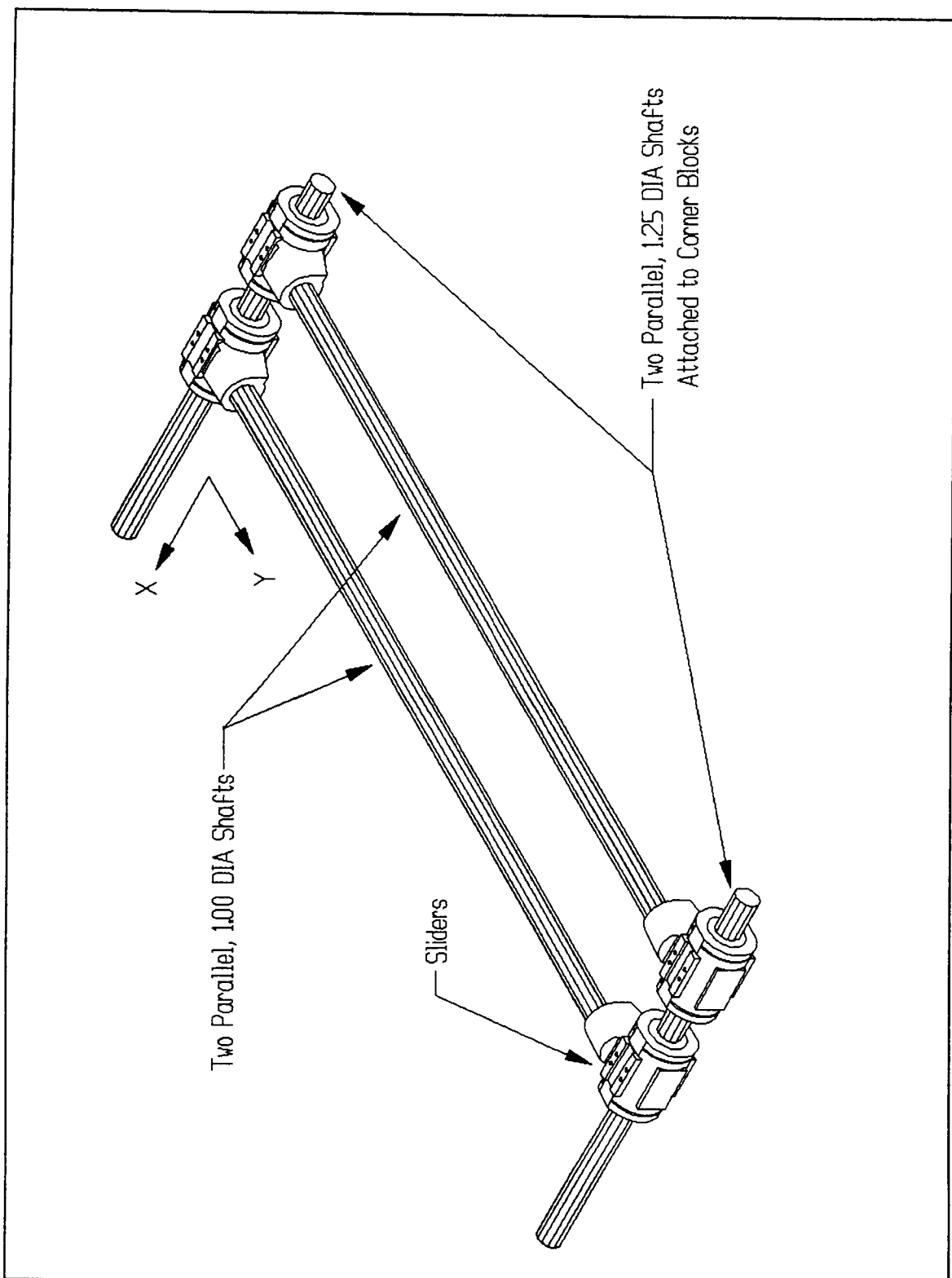


Figure 4.2: X and Y Rail System, with Sliders

floating and, therefore, does not encourage binding. Second, two stepper motors are being driven in close synchronization which can be attributed to the motor driver system discussed previously. Finally, the platform (which actually transfers force to the second parallel shaft) is constructed in such a way as to reduce potential binding.

Platform

The platform is designed to allow free, side to side movement along the two parallel Y shafts. The necessity of this movement will be realized as the guidance system is discussed later. It can be seen that the platform (shown in Figure 4.3) is arranged so that the side with two bearings rides along the Y shaft that is being driven by the motors. The other side of the platform has only one bearing which promotes less friction, less mass, and less binding.

Finally, the Z degree of freedom is used to accommodate vertical changes in the fender profile and is accomplished by allowing a 1.00 inch shaft to rotate and travel vertically through linear bearings in the platform. The first implementation of the platform, shown in Figure 4.4, used only one bearing placed in the center of the platform. The torque created when the end-effector was at the front of the fender proved to be too great and resulted in binding and destruction of the bearing.

A mathematical model was developed to help understand this problem, and an explanation is presented in Figure 4.5. Part number P002-PEX (shown in Figure 4.4 as an addition

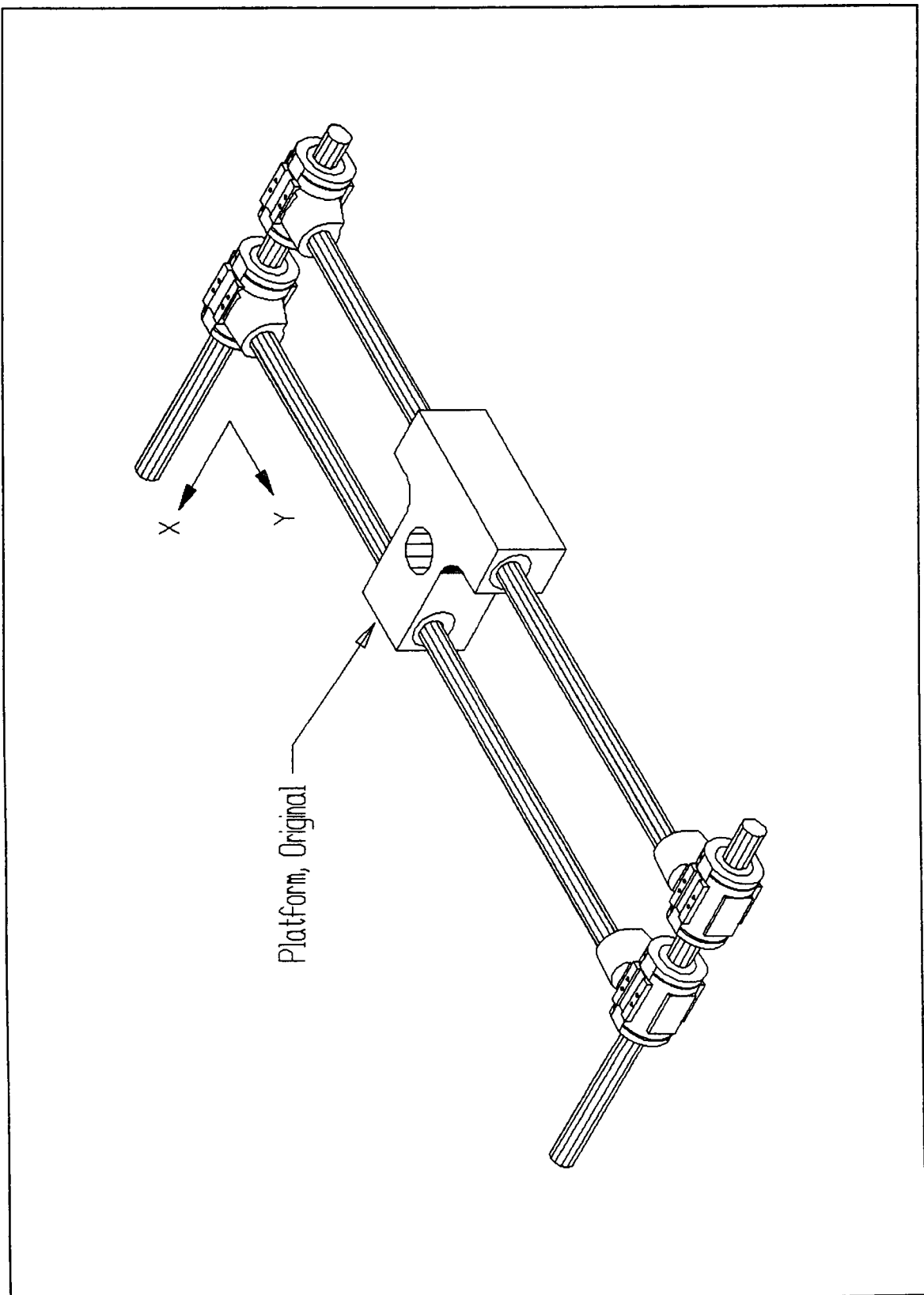
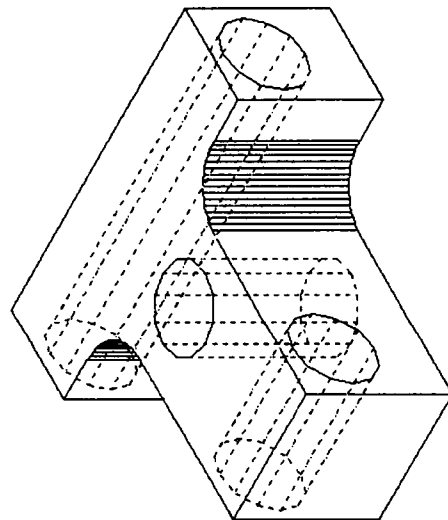


Figure 4.3: X and Y Rail System with Platform

Original
Platform



Platform
With Bearing
Extension

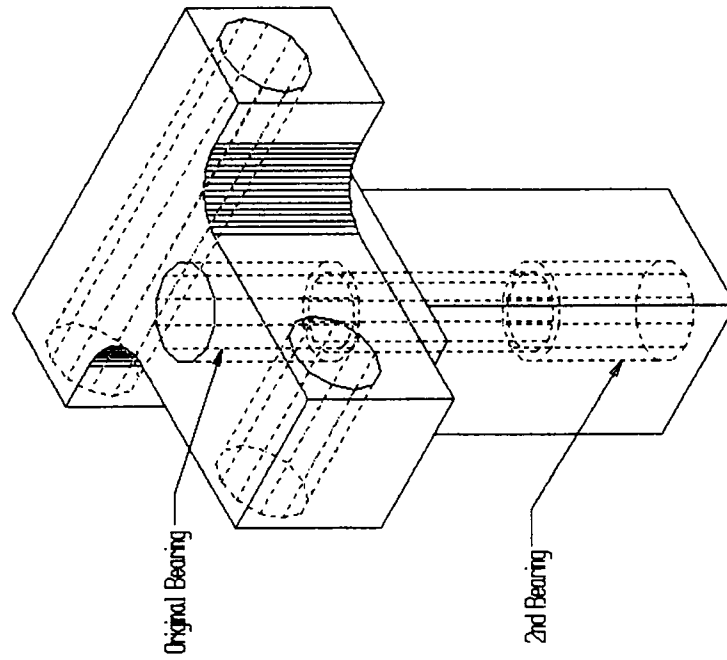


Figure 4.4: Original and Retro-fitted Platform

Variable Definitions

d = distance between bearings

** critical variable

L = total vertical distance

$l = L-d$, exposed shaft

r = inside bearing radius

F = force applied to guide car

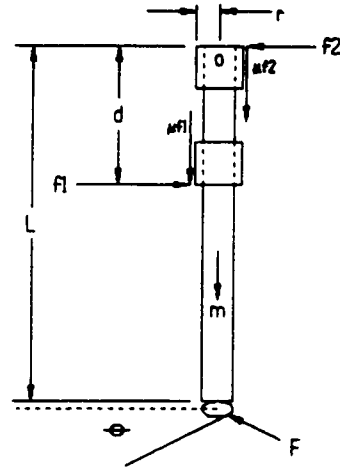
f = resultant torque force

m = mass of shaft and guide car

θ = angle of applied force

μ = static coefficient of friction

a = acceleration provided by drive motors



Three initial equations can be formulated from the diagram above:

$$F \cos \theta - \mu(f_1 + f_2) = ma \quad (1)$$

$$F \sin \theta + f_2 = f_1 \quad (2)$$

$$\sum M_o = -(l+d)F \sin \theta + df_1 + r\mu f_1 - r\mu f_2 = 0 \quad (3)$$

Equations (1), (2), and (3) can be combined and arranged into one equation:

$$\frac{F}{m} [\cos \theta - \mu \sin \theta (1 - \frac{2(l-r\mu)}{d})] = a \quad (4)$$

The final design suggested that $d = 9$ in, therefore $l = 13.75$ in. Using these values with $r = .5$ in and $\mu = .104$ (determined experimentally), equation (4) can be reduced to:

$$\frac{F}{m} (\cos \theta - .213 \sin \theta) = a \quad (5)$$

Figure 4.5: Mathematical Analysis of Vertical Platform Bearings

The result of equation (5) is best analyzed by looking at the efficiency of the bearings when a force (F) is applied at various angles (θ). The efficiency of the bearing can be expressed as:

$$eff = \frac{acceleration(\theta=x)}{acceleration(\theta=0)} = \frac{\frac{F}{m}(\cos\theta - .213\sin\theta)}{\frac{F}{m}}$$

$$eff = \cos\theta - .213\sin\theta$$

The following table shows the bearing efficiency at 15 degree increments.

θ	Efficiency (percent)
0	100
15	91.08
30	75.95
45	55.65
60	31.55
75	5.31
77	1.74
78	-0.04

This table is interpreted to show that a maximum angle of 77 degrees will still allow a positive acceleration, however, the efficiency of the bearing will be very low. Including a small measure of safety, it can be argued that the maximum angle at which the proposed system will function properly is 75 degrees. This angle is more than sufficient for the requirements of the system, and therefore, the design was implemented with $d = 9$ inches.

Figure 4.5: (Continued)

to the original platform) was designed and has been added to the platform to alleviate the binding and self-destruction problem. This part adds another linear bearing approximately 6.5 inches below the original and provides a very rigid, yet low friction guide for the system. As with many retro-fits, the new part also added a desired side effect of further restricting side to side movement of the Z shaft.

Guidance Car

The key to the force-guided translation system is the guidance car. This car is designed to transfer changes in force between the car and the fender into positional changes in the Y, Z, roll and yaw degrees of freedom. By properly attaching an end-effector within the car, the relative orientation between the end-effector and fender is maintained throughout the operation. This constant orientation is essential for producing a robust interaction between the end-effector and the fender. The following discussion of the guidance car describes each subsystem in the order of assembly.

1 Degree of Freedom Hinge

Connected to the Z shaft is the 1 degree of freedom hinge (referred to as the 1 D.O.F. hinge) shown in Figure 4.6. The primary purpose of this hinge is to allow the car to pivot (referred to in Figure 2.2 as "pitch") and follow the vertical slope of the fender. All other flexibility is restricted at this hinge; however, all other forces translate directly through the hinge to the platform.

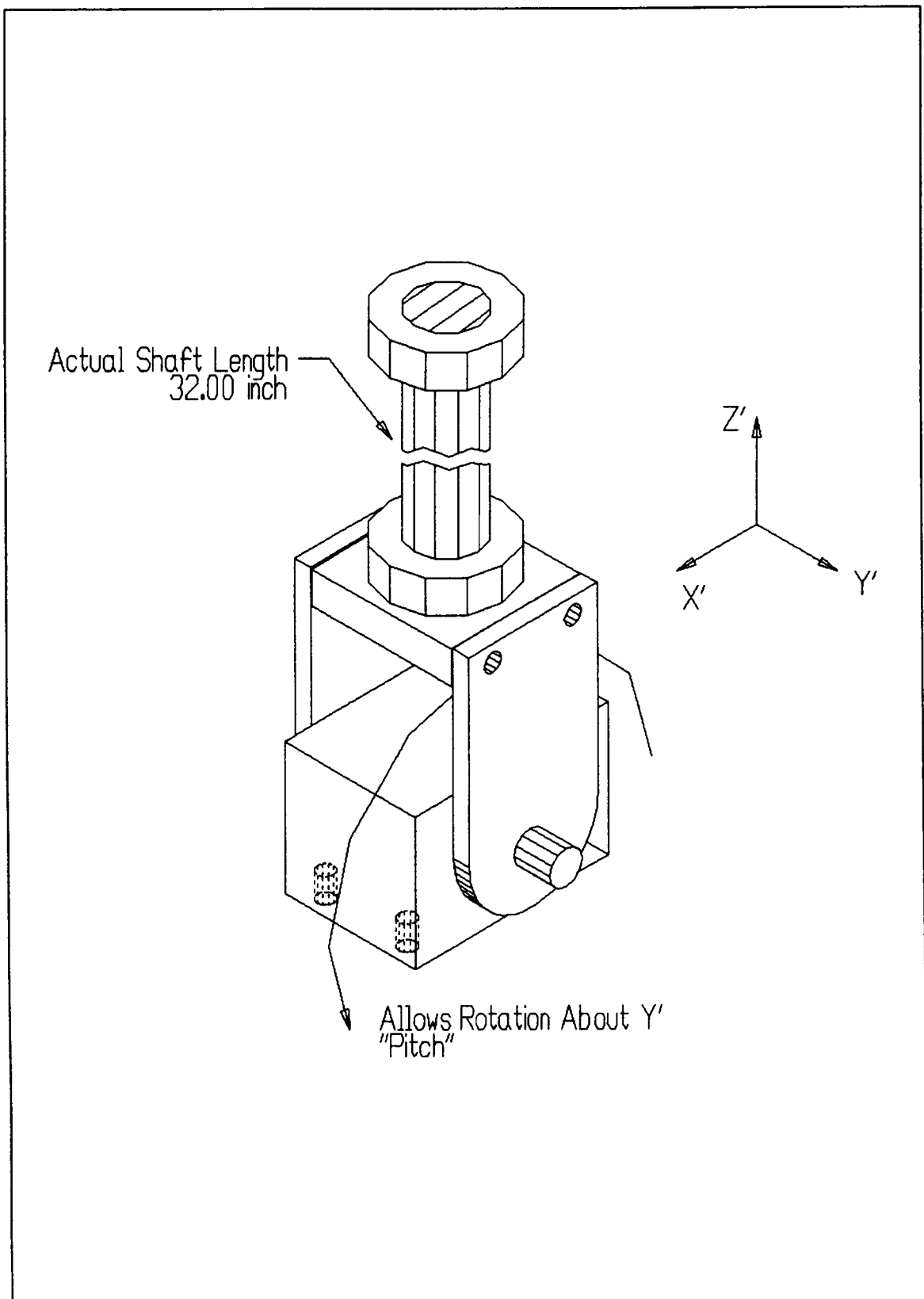


Figure 4.6: 1 D.O.F. Hinge, Shown Attached to Z' Shaft

Guide Car Adjustments

The 1 D.O.F. hinge is rigidly connected to the top side of the base plate. Figure 4.7 shows the complete guidance car structure with the 1 D.O.F. hinge attached. The slots available on the X and Y adjustment plates are necessary to accommodate various end-effector's size and function requirements. By moving the length (or X) adjustments, a maximum wheel span of 12 inches or a minimum of 9 inches can be obtained. The width (or Y) adjustments are used to control how the wheels track the fender relative to the end-effector and can move off center ± 0.75 inches.

Preliminary designs called for an adjustment in the height of the guide car; however, this feature was eliminated by using spacers between the X and Y adjustment plates to vary height. Current end-effectors do not show a need for any additional vertical adjustments, and the current solid plate is much more rigid than an adjustable version.

Guide Wheel Mounting and Geometry

The guide wheels are constructed from a 1.500 inch rod of polyurethane into the shape shown in Figure 4.9 (hardness grade 85-90, P/N C020-WHL in Appendix E). Originally, the wheels were mounted in rigid bearing blocks attached to each vertical Z support. To improve its ability to track some of the abrupt contour changes, the final guidance car design calls for the front set of the wheels to have pivoting ability (shown in Figure 4.8). Although some binding can occur in the design shown in Figure 4.8, the pivoting wheel has improved the tracking system and its ability to properly follow the fender surface.

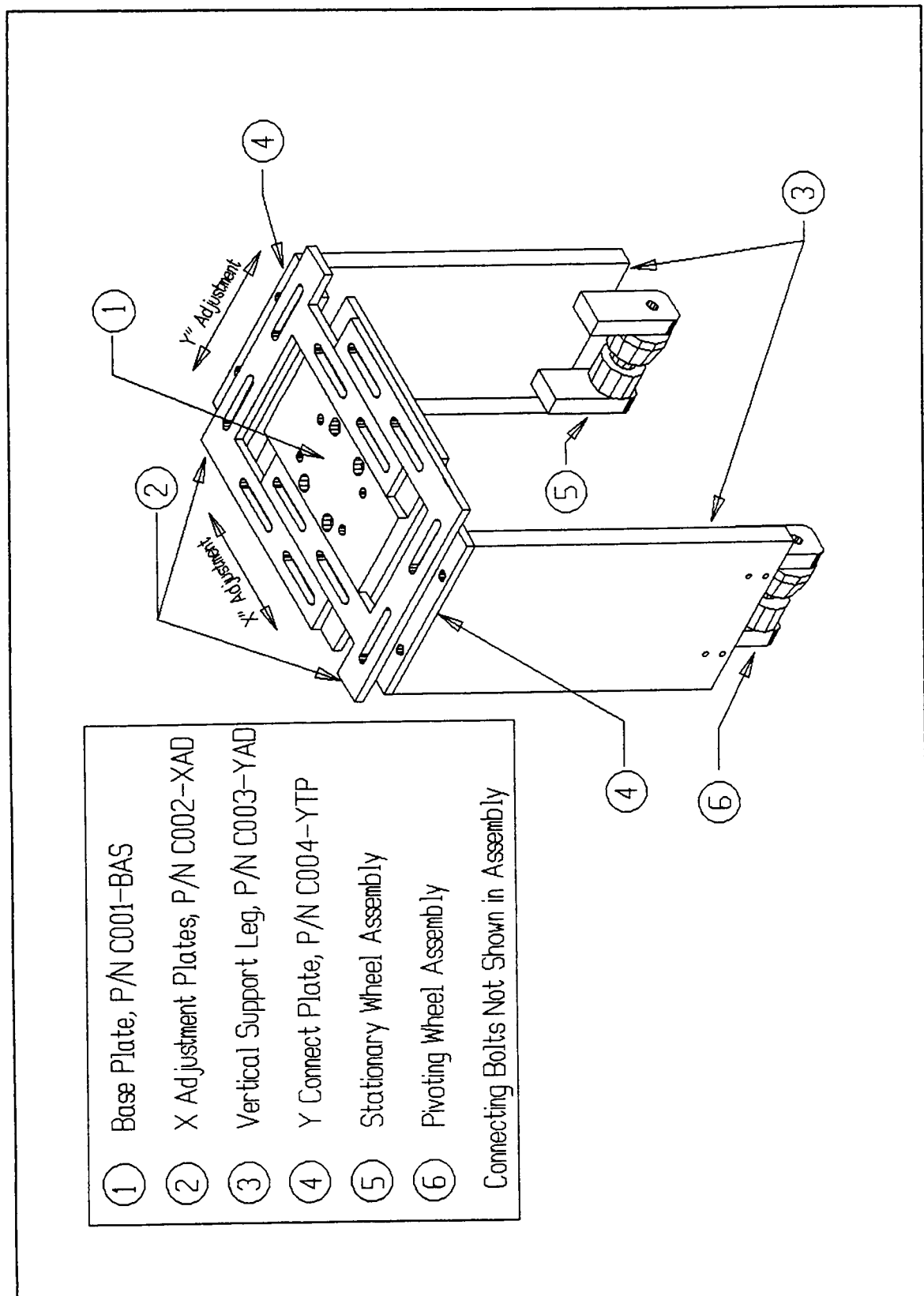


Figure 4.7: Guidance Car, Illustrated

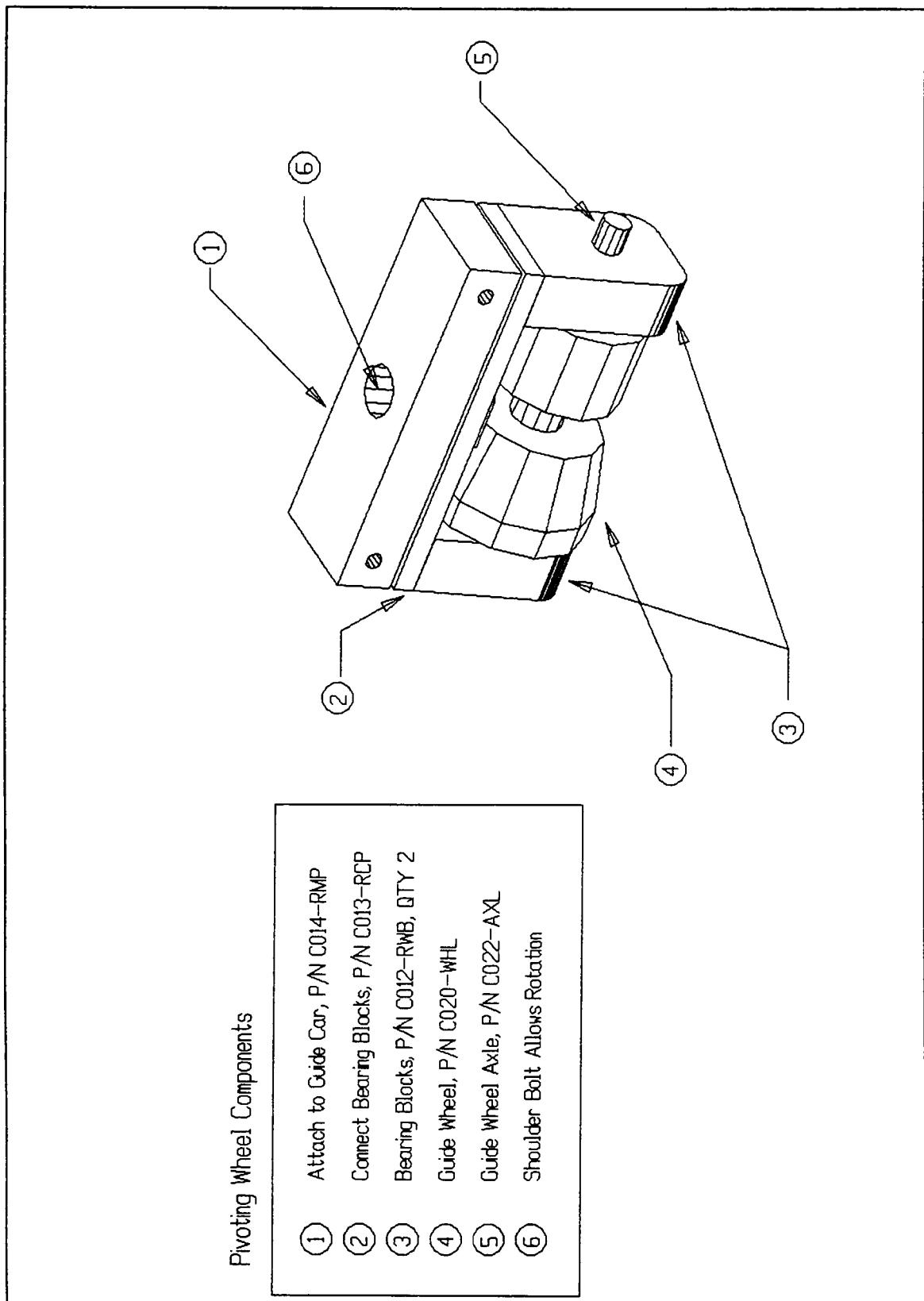


Figure 4.8: Pivoting Wheel Subassembly

Illustration of P/N C020-WHL

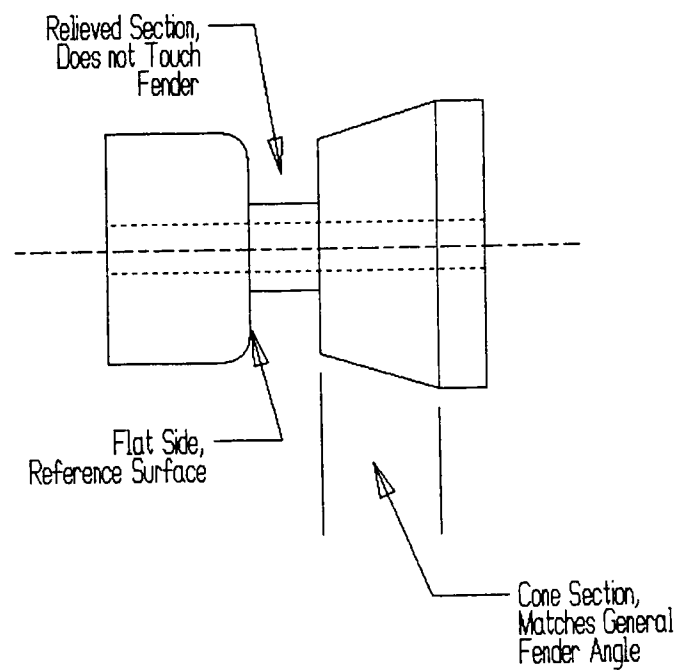


Figure 4.9: Guide Wheel Illustration

Additional improvements could be incorporated by using bearings in the pivoting system rather than the shoulder bolt currently being used.

Figure 4.9 shows the final shape of the guide wheels which are also shown mounted on the guide car in Figure 4.7. This shape was achieved through an iterative trial and improvement method. The cone side of the wheel closely matches the general angle of the fender. The flat side of the wheel is used as the reference guide such that the wheel always tracks the vertical surface on the inside of the fender. Figure 4.10 shows a sectional view of how the guide wheel interacts with the fender. Notice that the parting-line is always in the same relative position to the vertical edge being used as the guiding surface.

Another result of the iterative wheel design process is the relieved portion between the cone and the flat side. This part of the wheel never touches the fender and was shown to improve tracking performance over the sharp turn at the end of the fender. These prototype wheels consist of polyurethane that was chosen for its flexibility and inability to damage the fender surface. The milling process, however, leaves the material ragged and un-even, resulting in high friction. A lower friction guide wheel may improve tracking ability, provided its increased hardness does not scar the fender.

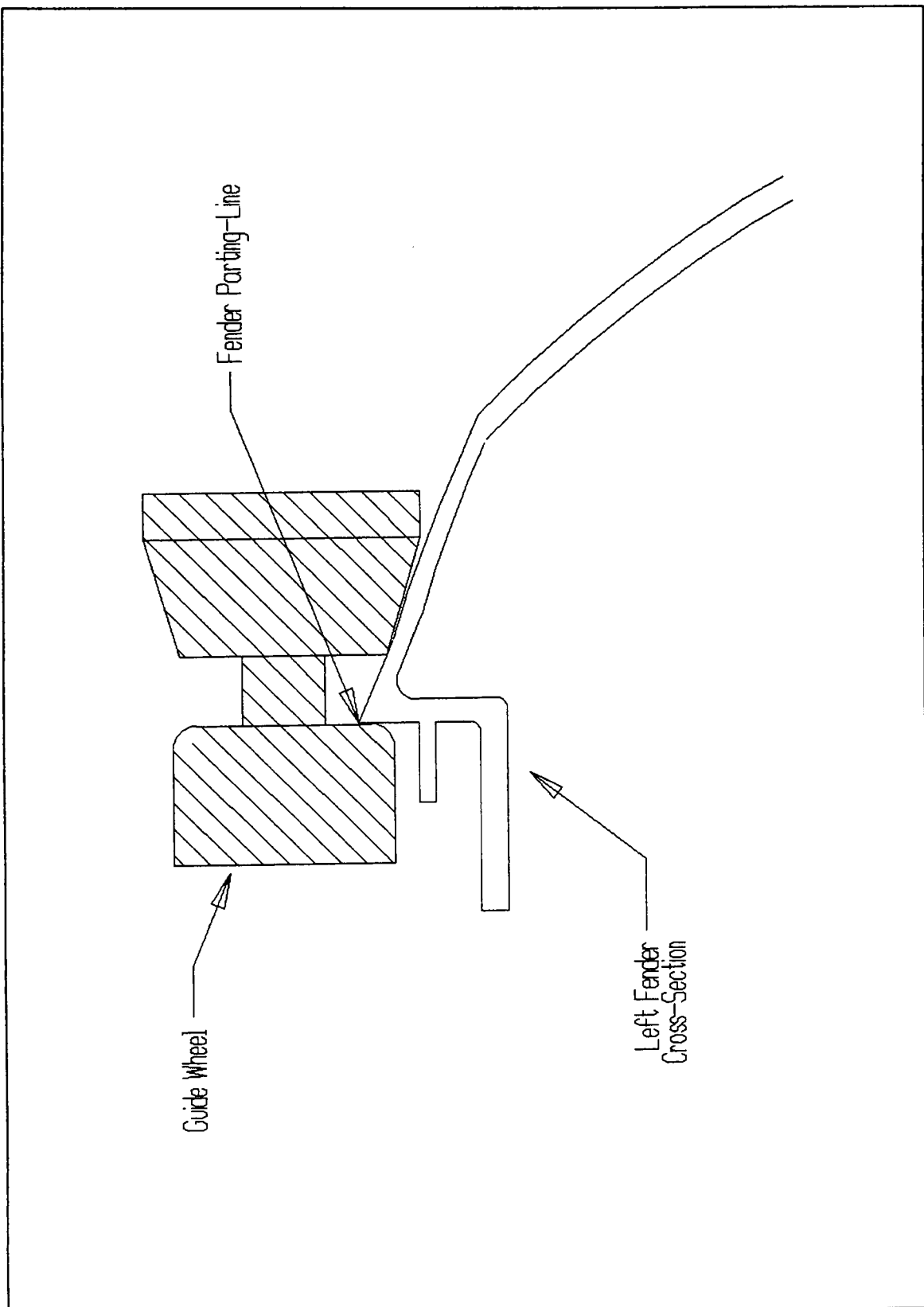


Figure 4.10: Guide Wheel Contact on Fender

Supporting Hardware

Supporting hardware addresses the issues of mounting a fender within the translator and providing the guidance car with a home base (right-hand run-off). Figure 4.11 shows the supporting infrastructure of the fender stand. Designed and constructed from pre-fabricated aluminum extrusions, this system allows for extreme flexibility in its configuration while providing structural rigidity. Next, an aluminum fender casting (supplied by Saturn) is attached to the fender stand using two protruding studs. The fender casting is currently used by Saturn in the painting process and provides a dimensionally accurate pattern on which the fender can be mounted. Adjusting the angle of the fender casting is accomplished by utilizing the two universal hinges into which the fender casting is mounted. This hinge can be used to adjust the angle at which the fender is presented to the end-effector; once set, it remains constant throughout the process.

Located at each end of the fender is additional supporting hardware referred to as "run-offs". These structures act as continuations of the fender profile and are necessary to allow fender loading and unloading as well as allow the end-effector to reach the entire parting-line. Starting with the guide car completely on the right-hand run-off (as shown in Figure 4.12), a finished fender can be unloaded and a new one loaded. At this stage, the run-offs help the operator position the fender correctly.

When the parting-line removal process is initiated, the guide car directs the end-effector along the run-off and onto the fender. When the guide car reaches the left side of the

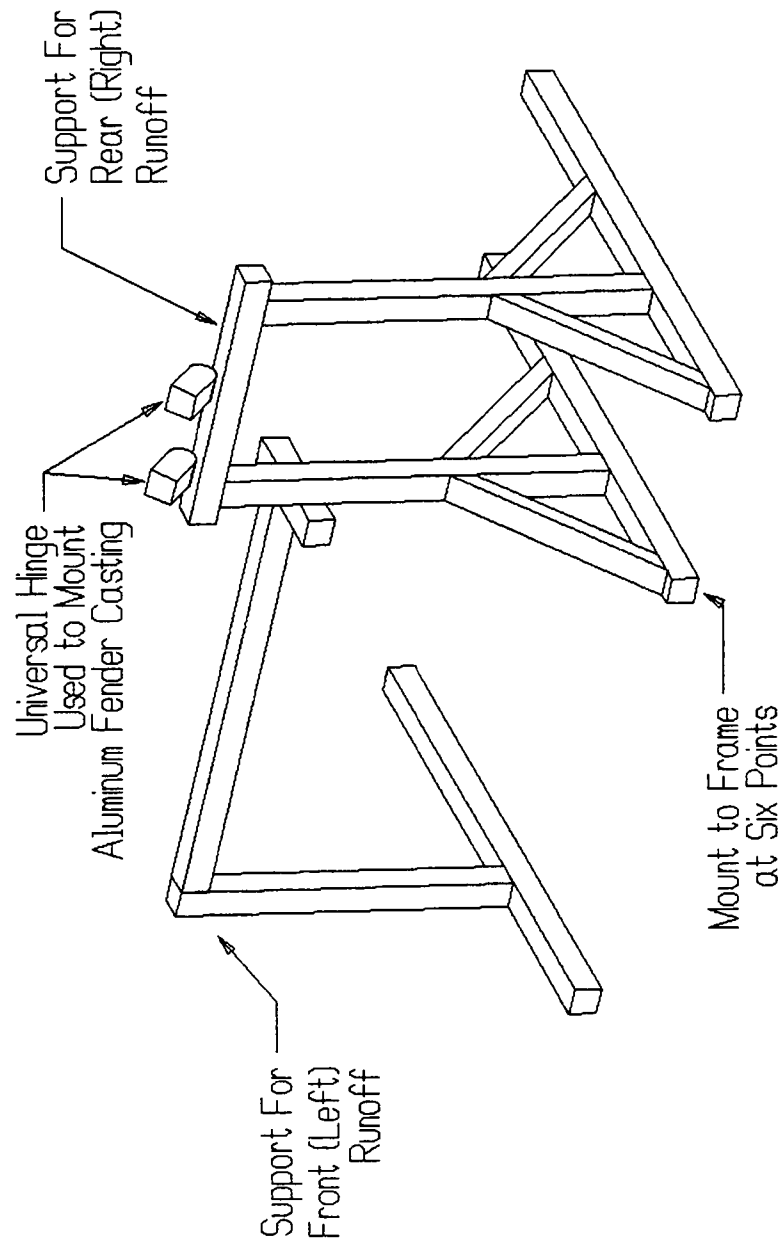


Figure 4.11: Fender Stand Supporting Structure

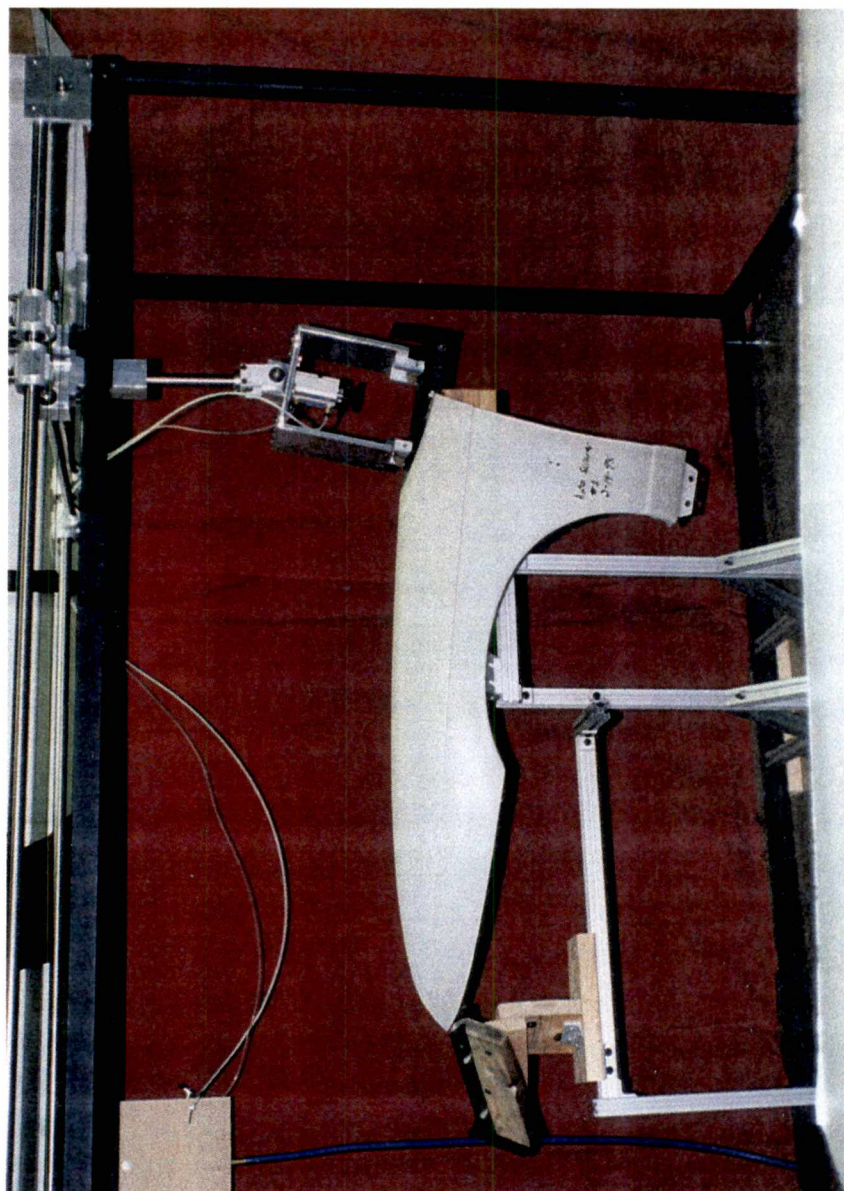


Figure 4.12: Car Positioned on Right-hand Run-off (Home Position)

fender it will continue onto the run-off so that the end-effector can reach the entire length of the fender. Finally, the end-effector is lifted from the fender, and the guide car returns to its original home position on the right. It is possible that some end-effectors may work equally as effectively in both directions; in this case, operating the end-effector during the "return trip" may be a more efficient use of time.

Due to the orientation and slope of the front end of the fender, maintaining alignment between the front (left) run-off and the fender has been unreliable. Chapter 7 will further discuss this problem and propose possible solutions.

CHAPTER 5

END-EFFECTORS

After completing the translator, Phase 1 and Phase 2 were brought together by demonstrating two end-effectors on the system. First was the sander, which has received the greatest amount of research and development, and a cutting tool, which will require additional modifications (refer to chapter 1 for an explanation of how the sander and cutting tool are to be used to remove the parting-line). The following is a summary of the observations and test results relating to how well the translator worked with these end-effectors. Both end-effectors were constructed as a part of Phase 1, and a more complete discussion of their surface finish capabilities can be obtained from Tim Wyrick's master's thesis [Wyrick 95].

Constant Contact Force

In order to achieve the design requirement of constant force between the end-effector and the fender (see Figure 2.3), an air cylinder is used to connect the end-effector to the guidance car. The theory behind this approach is the basic $\text{force} = \text{pressure} * \text{area}$ equation. By applying a particular air pressure to the cylinder, the effective weight of the end-effector could be reduced or increased (depending on which side of the cylinder the air is applied). As was shown in the design requirements, a fully constrained system would result in changes in the force realized between the fender and the end-effector.

The effectiveness of using an air cylinder to control the contact forces is presented in Chapter 6.

As a secondary function, the air cylinder is used to lift and lower the end-effectors to the fender. This is an essential operation for end-effectors that are designed to operate in only one direction.

Sander Attachment

The sander is rigidly attached to the air cylinder through a 90 degree steel bracket. It is important that the air cylinder be located directly above the fender-sander contact point. Testing has shown that binding will occur when forces that are not directed along its support rods are applied to the air cylinder.

Since the air cylinder must be located directly above the sanding belt, it can not be attached at the sander's natural center of gravity. This also causes a problem with binding and resulted in the addition of a spring balancing system. As shown in Figure 5.1, an adjustable spring system is mounted between the platform and the motors. The connecting spring is set to reduce the downward weight of the motors and to shift the sander's center of gravity to be directly underneath the air cylinder. Successful adjustment of this spring can be judged by raising and lowering the sander. The sander should raise and lower very smoothly with the high air pressure set to 50 psi (to raise the sander) and the low air pressure set to 10 psi (applied so that it increases the downward

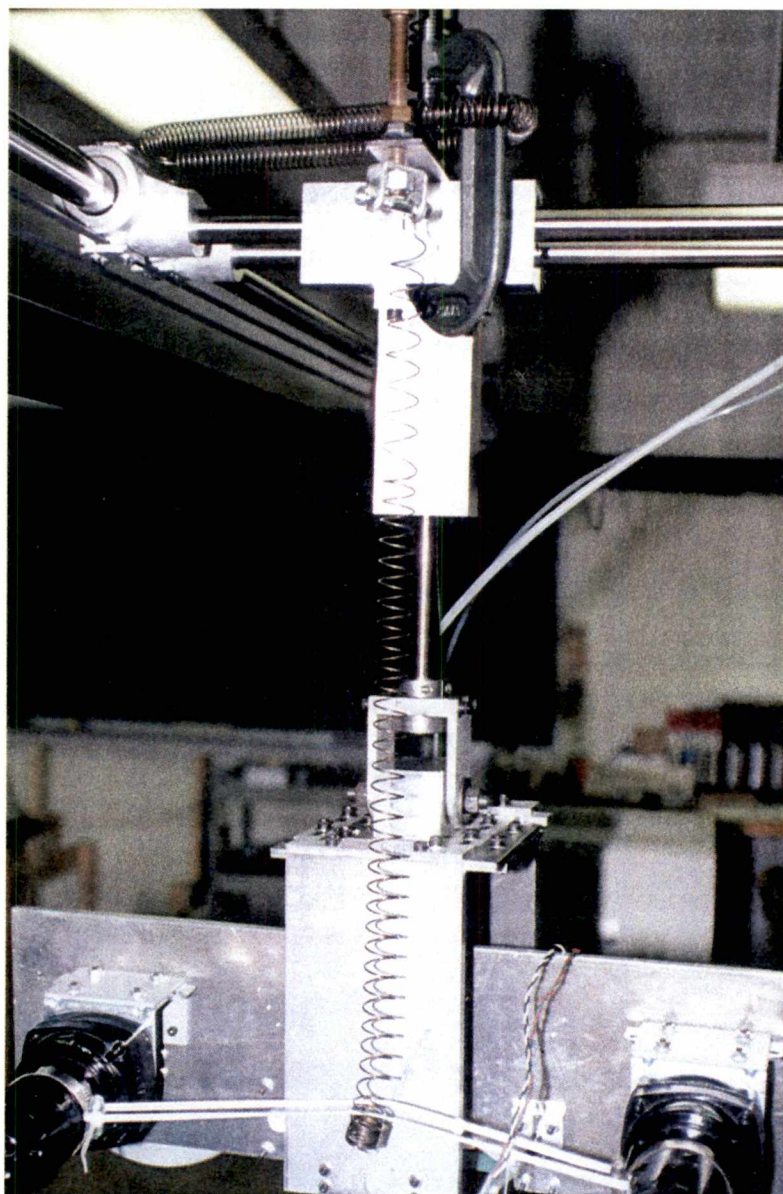


Figure 5.1: Sander Center of Gravity Spring System

force of the end-effector).

Cutting tool Attachment

The cutting tool required a more specialized attachment method due to the complex forces that must be applied. Similar to the sander, the cutting tool uses the flat inner surface of the fender as a reference surface; however, it requires a constant force to be applied perpendicular to this surface, which is contrary to the natural abilities of the air cylinder.

A prototype bracket, pictured in Figure 5.2, was developed to apply the necessary force vector to the cutting tool. A downward force is added to the cutting tool by applying 10 psi to the air cylinder. The result of this prototype mounting bracket has been acceptable, yet limited. Chapter 6 describes in more detail the interaction between the cutting tool and the translator and some improvements that must be made to the mounting bracket.

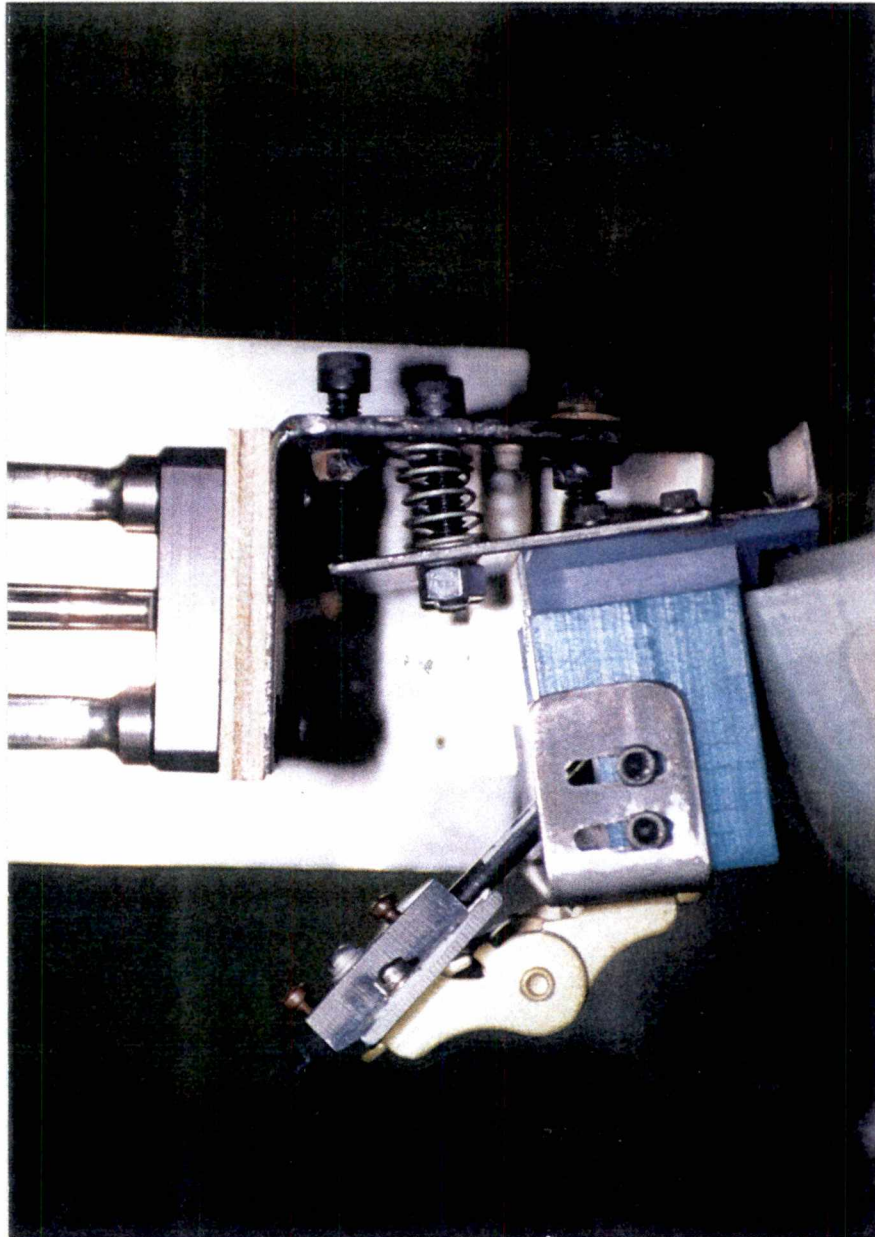


Figure 5.2: Cutting Tool Attachment Bracket

CHAPTER 6

DISCUSSION OF RESULTS

At the completion of this translation system, all of the primary goals were achieved and investigated. A large array of tests were run to determine the effectiveness and reliability of the innovative subsystems that are essential in demonstrating the system methodology. The two key subsystems are the tracking system and the constant force end-effector system. Robustness and appropriateness to task are discussed in this section as well as the effectiveness of the entire system in reaching its goals.

Fender Tracking Results

Evaluating the guidance car without any end-effectors, the tracking capabilities have proven very effective. The position of the parting-line on the fender has proven to be convenient for this approach to tracking. By using the flat inner surface of the fender, the guide wheels are designed to always remain in contact with this reference surface. This approach has eliminated the need of a complex, computer controlled positioning system while providing a relative position accuracy of ± 0.010 inches (this result is accurate over the 91 percent of the fender that is reliably traversed). In addition, this system has displayed a great deal of robustness to the uncertainty of position and fender style. For example, the system is equally capable of tracking right side fenders as it is left side fenders with only minor changes to the mounting hardware. Similar robustness

should be experienced during change over to new model year styles.

Air Cylinder

The air cylinder used to control the contact force between fender and end-effector has met with the greatest amount of difficulties. After a great deal of investigation, an air cylinder was chosen with a two inch stroke and two guide rods that maintain orientation of the attachment plate. It has been determined that the air cylinder's problems stem from friction caused by the brass bushings used to support these guide rods. After a large number of tests and experiments, it has been concluded that the air cylinder can only fulfill its design requirements in a very limited range of conditions. This narrow operating space has proven to be impractical and probably unattainable in the final operating environment.

When a mass is supported directly below the air cylinder such that its center of gravity is in-line with the piston rod, 4.5 to 5 lbs of force is required to move the air cylinder upward. These results were determined experimentally using air pressures between 10 and 37 psi and suspended weights of 5 to 25 lbs. Figure 6.1 depicts these results showing that the force required to move the air cylinder can reach as much as 6.25 lbs. The primary conclusion that can be drawn from the experiments is that the air cylinder can deliver a force resolution of approximately 5 lbs regardless of the mass being supported. This resolution, however, is only available within the perfect condition where the center of gravity of the mass is perfectly in-line with the piston rod.

Suspended Weight	Pressure Applied to Air Cylinder (psi)	Minimum Force (lbs.): 5 Trials					Mean
5	7.50	3.50	4.00	3.25	3.75	3.50	3.60
10	15.00	4.75	4.50	4.75	5.25	4.50	4.75
15	20.00	4.50	4.00	4.25	4.50	4.25	4.30
20	26.50	4.75	4.50	4.50	4.25	4.00	4.40
25	35.00	5.25	6.25	4.75	5.50	5.75	5.50

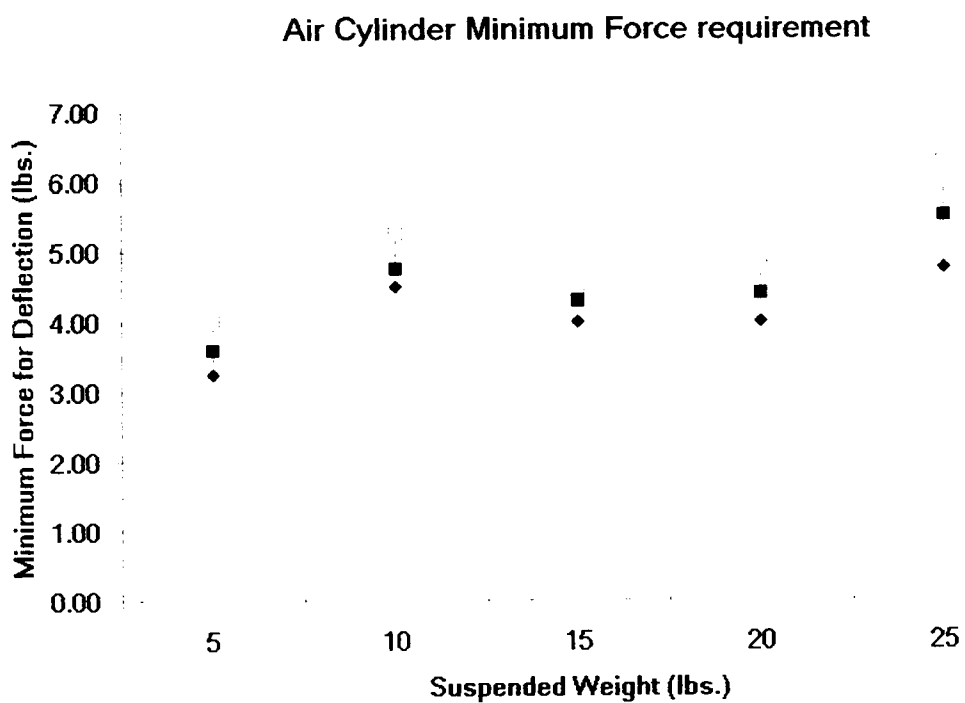


Figure 6.1: Air Cylinder Minimum Force Requirement

Further tests show that a force as small as 5 pounds not applied directly in-line with the piston rod causes the minimum required force to climb as high as 20 lbs. This means that each end-effector has to be attached such that its center of gravity and contact point with the fender are in-line with the piston rod. If this is not possible, weights and springs must be added to move the center of gravity to a more desired location.

Although the air cylinder has not been able to perform its force control task with the desired robustness, it has also been shown that critical force control may not be necessary for some end-effectors. For example, the downward force applied to the sander has been varied as much as 11 lbs. without any noticeable difference in surface finish. The more crucial variable is actually the amount of sanding belt used per length of the fender, rather than the contact force.

Testing has shown that by placing the center of gravity in-line with the piston rod, the desired force control can be delivered and the end-effector will maintain contact with the fender at all times. Further investigation will be required to determine how much variation in contact force is acceptable and whether or not a new approach will be necessary.

Result of Trial Runs with Sander and Cutting Tool

The software and drive system designed to move the translator can be used with several end-effectors. Velocity is the main variable that needs to be considered and should be set at the maximum velocity that will still allow the end-effector to function properly. In addition, the distance needs to be considered since each end-effector will be mounted differently in the guidance car. Both of these parameters can easily be set through interactive menus in the control program. When desired velocity and distance values have been determined, it is recommend that the program be changed to reflect these new settings on start-up.

Currently, the cutting tool does not perform well in the transition area between the right run-off and the beginning of the parting-line. Therefore, start the cutting tool just on the fender and operate the translator at 1 in/sec over a distance of 35 inches. Three passes are currently necessary to achieve the desired fender shape. Experiments were performed that used an acceleration ramp to bring the cutting tool from 0 to 1 in/sec. These tests did not show any noticeable difference in the way the end-effector performed; however, the use of an acceleration ramp may prove valuable when trying to approach speeds as high as 4 in/sec.

As referred to earlier, the cutting tool begins to bind and will not follow the front end of the fender robustly. This binding is caused by two factors. First, the cutting tool guide does not make the turn in the X-Y plane. This problem should be alleviated by reducing

the wheel base of the guide car and shortening the cutting tool fixture. Second, as the guide car follows the front curve of the fender, the relative position between the cutting tool and the parting-line changes very slightly. One solution to this problem could be the addition of a d.o.f. in the Y" direction (refer to Figure 2.2). The cutting tool attachment hardware would provide this d.o.f. by using a preloaded mechanical system that maintains a constant orientation of the cutting tool relative to the parting-line.

This additional d.o.f. may not be necessary if the wheel base of the guide car can be reduced significantly. Based on empirical results, the cutting tool can vary as much as 0.1 inches from its desired position and still perform the cutting process. The smallest arc (greatest curvature) on the fender has been measured to have a radius of 35 inches.

Knowing that the current wheel base is nine inches, the cutting tool can move as much as 0.29 inches from its desired position. In contrast, the straightest part of the fender has a radius of 160 inches and results in position variation of 0.05 inches. Figure 6.2 shows the relationship between fender curvature and guide car wheel base. The conclusion that can be drawn from Figure 6.2 is that a wheel base of just under 6 inches will keep the cutting tool position variation inside the 0.1 inches limit. Such a wheel base should be achievable through simple modifications to the guide car and cutting tool designs.

As opposed to the cutting tool, the sander can be translated at 4 in/sec, however, the desired finish is not achieved. As a result, the best speed for the Phase 1 sander is 0.25 in/sec. In addition, it can cover a distance of 40 inches if it starts in the home position.

These settings result in a sanding time of 3 minutes per fender which will obviously have to be improved to reach design specifications.

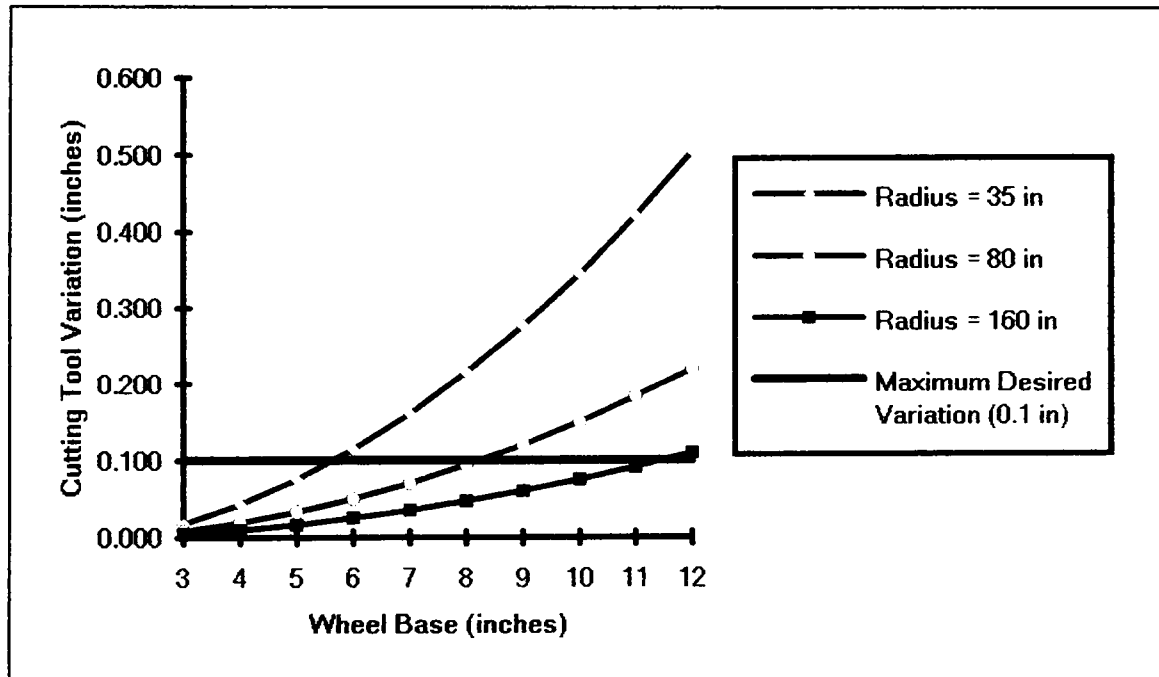


Figure 6.2: Cutting Tool Variation vs. Wheel Base

Cost of the Prototype Translator

Considering the difficult task of following a complex trajectory and motivating a 27 lb end-effector, the force-compliant rail guide system was an economic choice. Figure 6.3 shows a summary of the cost associated with each subsystem. These totals include the cost of vendor parts and machining of specialized designed parts. A complete listing of all the parts and vender part numbers is presented in Appendix B. The final system cost of \$5,716 is approximately 14 percent over the original \$5,000 budget.

Mechanical Subassemblies	
Frame Assembly	\$ 265.00
Fender Stand and Run-offs	380.00
Corner Supports	755.00
Sliders	1,332.00
X, Y, and Z Shafts	331.00
Motor Mounts, Belts, and Pulleys	277.00
Platform	684.00
Guidance Car	954.00
Sander and Cutting tool	457.00
Attachments	
Electronics and Motors	281.00
Total System Cost	\$5,716.00

Figure 6.3: Cost Summary for Prototype Translator

CHAPTER 7

RECOMMENDATIONS

As the essence of the word prototype implies, the translator has been a major stepping stone toward solving the parting-line removal dilemma. The following is a summary of observations and ideas that were generated through the construction and testing of this prototype. It is very possible that some recommendations will not require implementation depending on the results and changes in end-effector design.

Increase Stability of Frame and Fender Mount

While the chosen frame is quite adequate for a prototype, the final production model will have to eliminate the vibrations and flexibility that can exist in the horizontal frame members. The best means for absorbing these vibrations will be to add supports and stiffen the system. Increasing the thickness of the horizontal angle iron sections from .125 inch gauge steel to .250 inch and strategically placing braces may provide significant improvement to the stability of the system.

Improve Fender Mounting and Run-Off Method

Improving the fender mounting and run-off system will also be required for a production-level translator. The current fender stand contributes to the flexibility of the system as well as allowing the fender position to change after a short number of cycles. Once

again, adding supports to the system as well as attaching it to something other than the translation frame should substantially reduce this problem.

The flexibility of the run-offs and the mounting hardware has lead to additional positioning difficulties. These run-offs need to be attached so that the weight of the guidance car does not cause deflection or misalignment with the fenders. Usually, the run-offs have to be adjusted between each fender cycle to maintain the proper alignment and function. This misalignment could be avoided by rigidly attaching the runoff to the fender mounting system so that if the system flexed, it would move together and maintain alignment. In addition to stability, the shape and contour of the run-off will need to be altered as discussed in the next section.

Continuous Improvement of Tracking System

The tracking system has proven quite reliable over a large portion of the fender. There are still two primary areas that need to be improved with respect to the tracking system. First, the end-effector attachments must be improved so they do not restrict the movement and functionality of the guide car. Currently, the cutting tool limits the tracking ability of the guide car to only 57 percent of the fender. Second, the problems associated with the front of the fender may be solved with improved geometry and stability of the run-offs. However, additional changes may have to be made to the guidance car itself. Possibilities to investigate include both front and back pivoting wheels (this may have a negative effect by increasing positional variation), changes to wheel geometry, or

additional hardware that will provide improved guidance over this difficult portion of the fender.

Wheel Design

Changes to wheel geometry and design have proven to be the most promising thus far in improving tracking capability. The relieved area between the cone and flat side, shown again in Figure 7.1, allows the wheel to make the difficult turn while not touching the cone section. When the cone surface begins to touch the top of the fender, the guidance car has a tendency to "climb" up the fender and fall off. The current wheel design has a relieved section (as shown in Figure 7.1) that alleviates this tendency. The next step in improving the wheel design would be to eliminate the relieved section altogether and replace it with bearings. This would allow each half of the wheel to turn independently, therefore insuring that the wheel would not need to "slip" on the fender in order to make the difficult turn.

The material used to make the guide wheels has also come under minor questioning. The current polyurethane wheels are much softer than the fender material. After a number of cycles it has been seen that the polyurethane wheels pick up particles that can then be deposited on the next fender. It has been suggested that a harder wheel, one more similar to the material that the fender is made out of (see Appendix F), may help reduce this concern. The final system may need to use pressurized air or a vacuum system to help evacuate material as soon as it is removed from the fender.

Illustration of Guide Wheel

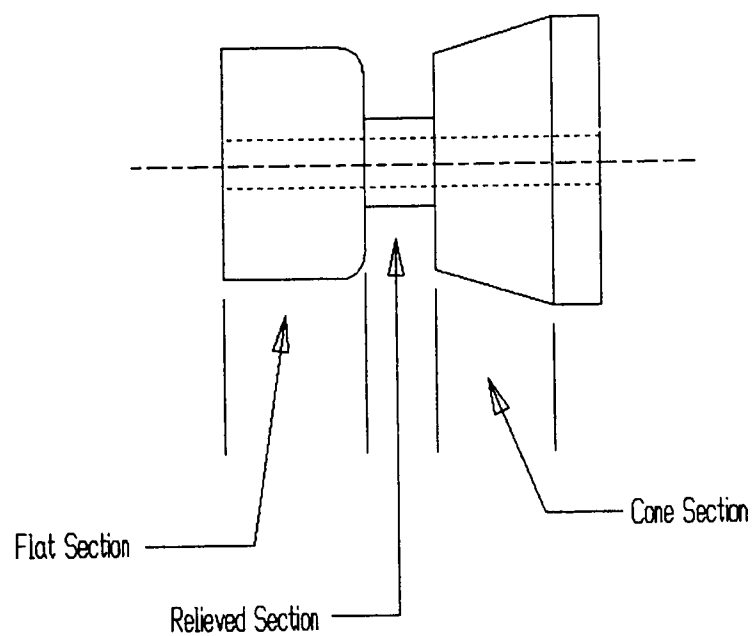


Figure 7.1: Guide Wheel Illustration

Fender Mounting Position

Additional improvements in the tracking system may be accomplished by increasing the angle at which the fender is mounted (rotating clockwise), shortening the wheel base of the guide car, and reducing the height of the guide car. Each of these changes should provide a minor improvement, that when used together, may allow robust tracking along the front of the fender.

A more radical, yet very possible solution, was presented at a recent design review and involves mounting the fender on its side. This orientation should provide a much gentler slope in which gravity would not be such an adversary. The obvious drawbacks to this concept is the need to redesign most of the guide car as well as solve the inherent mounting problems.

Final design reviews also brought up some questions concerning the position of the 1 D.O.F. hinge relative to the guidance car (shown in Figure 4.6). Experiments with the cutting tool suggest that moving the 1 D.O.F. hinge closer to the trailing guide wheel should improve the tracking and the function of the cutting tool. Currently, weights have been added to the guide car to insure consistent contact between the guide wheels and the fender. An analytical study of the 1 D.O.F. position is recommended and should lead to improved robustness when using the cutting tool.

Constant Force System

As stated before, the current force control method may prove to be quite inadequate for the application, and the air cylinder may have to be replaced with a more robust system. One inexpensive solution is to use an end-effector attachment system that utilizes springs and dampers. This approach has some inherent difficulties, mainly a lack of adjustments and the fact that the spring force will change as the spring is compressed or extended. Other more elaborate solutions may exist involving computer controlled monitoring of a constant force system. The cost and software integration were considered to be prohibitive factors that ruled out the implementation of such complex solutions.

CHAPTER 8

CONCLUSIONS

The achievements of this project can be summarized into two major components. First, the translator has demonstrated its ability as a test-bed for investigating various parting-line removal methods. Second, the tracking system has proven to be a robust and reliable method of traversing the fender parting-line while motivating a wide spectrum of end-effectors.

The primary goal of developing a test-bed and providing an accurate and easy to use method of testing numerous end-effectors has been accomplished. This test-bed allows control over translation speed and contact force in order to determine the best settings for various end-effectors. Once determined, this information will be directly transferable to the production-level parting-line removal machine. Current results have been summarized in Table 8.1. These results are extrapolated in Table 8.2 to describe the cycle time of a 3-head cutting tool that could traverse 100 percent of the fender. Notice that the limiting factor is the speed at which the cutting tool can be reliably operated. The sander can be added to this operation if the belt speed is increased so that the sander can be translated at 1 in/sec.

Table 8.1: Translator Capability Results

End-effector	Recommended Speed	Percent of Fender Finished	Cycle Time
None	4 in/sec	91%	30 sec
Cutting Tool	1 in/sec	57%	26 sec one pass
Sander	0.25 in/sec	79%	240 sec one pass

Table 8.2: Predicted Cycle Time for a 3-Head Cutting Tool

Recommended Speed	Complete Cycle Time
1.0 in/sec	61 sec (return to home position)

Following the complex trajectory of the fender parting-line while carrying a 27 lb end-effector would be quite difficult and software intensive for a traditional cartesian coordinate robot. The tracking system used in this system has shown that reliable, relative positioning can be obtained by using the fender profile as the trajectory guide. In addition, the robustness of this mechanical feedback loop will result in a system that can handle a wide range of manufacturing variations as well as fender style changes.

The parting-line removal process developed in phase 1 and the fender tracking capabilities of the translator have shown that a fully automated process is practical and realizable.

A final step will be necessary in the prototype development to further integrate the process and the automation as well as define a complete operating space that will produce reliable, high quality finishes.

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LIST OF REFERENCES

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APPENDICES

APPENDIX A

Instructions for Operating the Translator

Instructions for Operating the Translator

General Operating Procedure :

At the completion of this thesis, the following operating procedures produced the best results for the respective end-effectors.

Cutting tool End-Effector

- 1) Attach the cutting tool to the air cylinder using the cutting tool attachment bracket (P/N SC01-SMB).
- 2) Install a new fender onto the fender casting
- 3) Position the Guidance Car so that the cutting tool is located approx 0.2 inches on the fender parting-line.
- 4) Verify that the run-offs are properly aligned with the fender. Make any necessary adjustments.
- 5) Set the air pressures to 50 psi to the lift side (lower input on the air cylinder) and 10 psi to the push side (top input on the air cylinder).
- 6) Locate the file "trans.exe" on the IBM computer. Currently located in directory C:\SATURN\TRANS\
- 7) Initiate the program and change the setting. (Further instructions for operating the "trans.exe" program can be found by executing the program and also reading the code documentation (this program was written in Turbo C).
- 8) Change the settings of the program to:
Speed = 1 in/sec
Distance = 30 in
- 9) Turn on the power to the motor drive system.
- 10) With the cutting tool in the lowered position, initiate the process by selecting M on the keyboard.
- 11) When the process is complete, lift the cutting tool off of the fender. Return the guidance car to its original position by choosing H on the keyboard.

Sander End-Effector

- 1) Attach the sander to the air cylinder using the sander attachment bracket (P/N SN01-SMB). Also attach the center of gravity compensation spring as shown in Figure 5.1.
- 2) Verify that the Guidance Car is located in the "home" position (see Figure 4.12).
- 3) Verify that the run-offs are properly aligned with the fender. Make any necessary adjustments.
- 4) Set the air pressures to 50 psi to the lift side (lower input on the air cylinder) and 10 psi to the push side (top input on the air cylinder).
- 6) Locate the file "trans.exe" on the IBM computer. Currently located in C:\SATURN\TRANS\
- 7) Initiate the program and change the setting. (Further instructions for operating the "trans.exe" program can be found by executing the program and also reading the code documentation (this program was written in Turbo C).
- 8) Change the settings of the program to:
Speed = .25 in/sec
Distance = 40 in
- 9) Follow the instructions provided in (Wyrick 95) to initiate the sanding end-effector.
- 10) Turn on the power to the motor driver system.
- 11) Start the sander running and then set the translator into motion by selecting M on the keyboard.
- 12) When the translator has moved the length of fender, the sander should be lifted off of the fender. The system can be returned to the "home" position by selecting H on the keyboard.

APPENDIX B

Prototype Part Numbers and Cost Summary

Translation Mechanism - Cost Summary/ Parts List

REQD QTY	P/N	PART NAME, DESCRIPTION	IBMCAD FILE	VENDOR	VENDOR P/N	UNITS	UNIT COSTS	TOTAL COST	TOTAL COST PER SECTION
FRAME ASSEMBLY - ASY300-XXXX-YYY-ZZZ									
4	---	Frame Support Feet	---	Ohio Nut and Bolt	FB3348	1	\$0.00	\$0.00	
2	F101-ALP	Member, Horiz., X Top	HORIZBAR R4	UT FMC	---	110	\$0.05	\$20.00	\$0.00
2	F102-YLP	Member, Horiz., Y Top	HORIZLPLY R4	UT FMC	---	40	\$0.05	\$20.00	\$24.00
2	F103-XDN	Member, Horiz., X Bottom	HORIZDNNR R4	UT FMC	---	110	\$0.05	\$20.00	\$31.00
2	F104-XDN	Member, Horiz., Y Bottom	HORIZDNNY R4	UT FMC	---	40	\$0.05	\$20.00	\$24.00
4	F105-VLT	Frame Member, Vertical ("Z")	ZVERTENJ3	UT FMC	---	68	\$0.40	\$40.00	\$148.80
32	VEN451-075	SHCS, 0.312-24 UNF, 0.75 long	SHCS312 R2	McMASTER-CARR	91252A581	1	\$0.12	\$0.00	\$3.99
32	VEN487	Washer, Flat, for 0.312 ID	WSHR312 R2	McMASTER-CARR	98019A185	1	\$0.08	\$0.00	\$2.50
FENDER STAND AND RUN-OFF PROFILES									
1	F110-FST	Fender Mounting Stand	F110-FSTLFI	BERTELKAMP	80 20	1	\$330.00	\$330.00	
1	---	Aluminum casting 1996	---	SATURN	---	1	\$0.00	\$0.00	
2	---	Run-Off Profiles	---	UT	---	1	\$25.00	\$25.00	\$380.00
X-Y MOTION ASSEMBLY - ASY516-XXXX-YYY-DDD									
4	N001-CRL-R	Support, Corner	N001-CRL-TR1	UT FMC	---	1	\$42.00	\$582.14	\$750.14
2	VEN406-XXX	X Shaft, 1.25 DIA	GSHAFT R1	THOMSON	---	102	\$1.03	\$20.00	\$230.12
8	VEN451-075	SHCS, 0.312-24 UNF, 0.75 long	SHCS312 R2	McMASTER-CARR	91252A581	1	\$0.12	\$0.00	\$1.00
8	VEN491	Nut, Hexagonal, 0.312-24 UNF	HNT312 R2	KNOX BOLT	---	1	\$0.00	\$0.00	\$4.00
SLIDER ASSEMBLY - ASY521-DDD									
4	MEK021-DDD	Housing, Slider	SLIDER R16	UT FMC	---	1	\$0.00	\$1,030.88	\$1,030.88
4	VEN401-DDD	Bearing, Linear, 1.25 Tube Dia.	LINBRG R3	THOMSON	A-162536	1	\$58.00	\$58.00	\$232.00
8	VEN402-DDD	Seal, Linear Brg, 1.25 Tube Dia.	LINSEAL R3	THOMSON	S-1000	1	\$8.70	\$0.00	\$69.60
X MOTOR MOUNT AND ASSEMBLY									
2	MEK011	Plate, Motor Mounting	MTRPLATL R4	UT FMC	---	1	\$0.00	\$10.00	\$20.00
4	VEN448-050	SHCS, 0.190-32 UNF, 0.50 lng	SHCS190 R2	McMASTER-CARR	91252A242	1	\$0.08	\$0.00	\$0.32
4	VEN484	Washer, 0.190 norm D (0.438 OD)	WSHR190 R2	KNOX BOLT	---	1	\$0.06	\$0.00	\$0.24
X BELT AND PULLEY COMPONENTS									
2	ASY040	Assy, Belt Attachment	BLTCLASRY R2	UT FMC	---	1	\$0.00	\$10.00	\$20.00
1	---	Belt 1.5 pitch	---	STOCK DRIVE	A6R3-C037	38	\$2.24	\$0.00	\$85.12
2	---	Reduction Pulley (3.820 in)	---	STOCK DRIVE	A6A3-60NT037L	1	\$18.21	\$0.00	\$36.42
2	---	Motor Belts (103 grooves)	---	STOCK DRIVE	A6R3-103037	1	\$4.70	\$0.00	\$9.40
6	---	Direct Drive Pulley (955)	---	STOCK DRIVE	6A3-15DF03712	1	\$6.18	\$0.00	\$37.08
8	---	Bearing, Flanged for corner block	---	NATIONAL BEAR	FR6-2RS	1	\$7.85	\$0.00	\$62.80
1	---	Spacers, 125	---	McMASTER 97	3088A511	10	\$3.13	\$0.00	\$31.30
3	---	Shaft Clamp	---	McMASTER 98	6166K23	1	\$0.71	\$0.00	\$2.13
									\$256.08

Translation Mechanism - Cost Summary/ Parts List

REQD QTY	P.N.	PART NAME, DESCRIPTION	IBACAD FILE	VENDOR	VENDOR P.N.	UNITS	UNIT COSTS ----- "RAW" MATERIAL (S UNIT)	UNIT	MACHINE	TOTAL COST PER PART NUMBER	TOTAL COST PER SECTION
PLATFORM ASSEMBLY											
1	P001-PLT	Platform	P001-PLT.TR4	UT FMC	---	1	\$20.00	ca	\$250.00	\$270.00	
1	P002-PEN	Platform Bearing Extension	P002-PEN.TR1	UT FMC	---	1	\$0.00	ca	\$200.00	\$200.00	
2	VEX-403-100	Rectic Ballbearings, Linear, 1.00 Dia	SUPERBRG-R2	THOMSON	SUPER-16	1	\$23.00	ca	\$0.00	\$46.00	
3	VEX-403-100	Rectic Ballbearings, Linear, 1.00 Dia	SUPERBRG-R2	THOMSON	A-162536	1	\$29.30	ca	\$0.00	\$87.90	
7	VEX-402-100	Seal, Linear Brg, 1.00 Tube Dia	LINSEAL-R3	THOMSON	S-1000	1	\$8.00	ca	\$0.00	\$56.00	
2	---	Shaft, Y Guide, 1.00 DIA	SHAFT-R1	THOMSON	---	35	\$0.79	in	\$20.00	\$75.30	
3	---	Stainless Steel Collars	---	McMASTER 97	6435K38	1	\$8.02	ca	\$0.00	\$24.06	\$759.26
GUIDANCE CAR											
1	D00N-X-TRN	1 inch Shaft	---	THOMSON	---	32	\$0.79	in	\$0.00	\$25.28	
1	---	I.D.O.F. Hinge	---	UT FMC	---	1	\$30.00	ca	\$200.00	\$250.00	
2	---	Tapered Roller Bearing	---	DIXIE-TINKEN	A4050-A4138	1	\$11.63	ca	\$0.00	\$23.26	
1	---	Car Hardware and Adjustments	---	UT FMC	---	1	\$100.00	ca	\$450.00	\$550.00	
2	C020-WHL	Poly Urethane Guide Wheels	C020-WHL-TR3	UT FMC	---	1	\$10.00	ca	\$30.00	\$50.00	
4	C010-BHR-L	Bearing Blocks for Guide Wheels	C010-BHR-TR2	UT FMC	---	1	\$5.00	ca	\$30.00	\$50.00	
4	---	Ballbearings for Guide Wheels	---	DIXIE BEARING	---	1	\$7.72	ca	\$0.00	\$30.88	\$979.42
END-EFFECTOR ATTACHMENTS											
1	---	Air Cylinder	---	BERTELKAMP	GC3000-25 50	1	\$250.00	ca	\$0.00	\$250.00	
2	---	Air Regulators	---	BERTELKAMP	---	1	\$40.00	ca	\$0.00	\$80.00	
1	---	Air Valve Switch	---	BERTELKAMP	41411-7002	1	\$45.00	ca	\$0.00	\$45.00	
1	---	1 4-28 Bolts 75 inch	---	McMASTER 97	91252A540	1	\$11.81	ca	\$0.00	\$11.81	
1	SC01-SMB	Scraper Air Cyl Bracket	SC01-SMB-TR1	UT FMC	---	1	\$10.00	ca	\$30.00	\$40.00	
1	SN01-SMB	Sander Air Cyl Bracket	SN01-SMB-TR1	UT FMC	---	1	\$10.00	ca	\$20.00	\$30.00	\$456.81
X-STEP MOTOR, MOTOR DRIVER, POWER SUPPLY, ELECTRONICS, etc.											
2	---	Step Motors (180 step rev)	MOTNYZ1-R1	UT FMC	---	1	\$0.00	ca	\$0.00	\$0.00	
2	---	Drive Cubes	---	INS	BH463	1	\$135.50	ca	\$0.00	\$271.00	
1	---	30 Volt Power Supply	---	UT FMC	---	1	\$0.00	ca	\$0.00	\$0.00	
1	---	Drive Cube Mounting Plate	---	UT FMC	---	1	\$10.00	ca	\$0.00	\$10.00	
1	---	PC Computer for programming	---	UT FMC	---	1	\$0.00	ca	\$0.00	\$0.00	
Total Motor Driver Components										\$281.00	

Total System Cost

\$5,716.16

APPENDIX C

Software Code

```

/*****
*           File Name : TRANS.C
*           Date      : Jan 02, 1995
*           Programmer: Jeremy A Davis
*
* Purpose: This program is to be used with the Phase 2 Translator.
*          It will allow the user to quickly set the travel distance
*          and a constant rate of speed. In addition, the delay
*          value and number of motor steps are displayed for the
*          users benefit.
*
* Operation: Use the following hot keys to manipulate the program:
* c) Change the distance and velocity settings. This key
*     will allow you to change these settings by displaying
*     the current value as well as an acceptable range.
*     To accept the current value, simply press ENTER.
*
* m) Move. Pressing this key sets the motors into motion
*     according to the current distance and velocity
*     settings. Remember that this move is only
*     relative to the current position of the translator.
*
* h) Home. This key moves the translator back to the position
*     it was in when the program was initiated.
*
* s) Stop. This key stops the program
*****/
#include <stdio.h> /* for printf() */
#include <dos.h> /* for inport() ,outport() & delay() */
#include <conio.h> /* for getch() */
#include <math.h> /* for pow10() */
#include <time.h> /* for time functions */

void delayx(int i);
int StoF( char string[20], float *p_result);

void main()
{
    int PARA_PORT=888; /* constant value for parallel port */
    int i, /* basic counting variable */
        count, /* keeps track of current step count (position) */
        steps, /* number of steps to be moved next time */
        asteps, /* absolute value for number of steps */
        newstep, /* temp storage of new step value */

```

```

    delay,                /* holds the delay to be sent to delayx() */
    newdelay,             /* temp storage of new delay */
    dirbit,               /* controls direction of motors */
    loop;                 /* acts as a boolean, set to 0 finished */

float res,                /* distance the car travels for one motor pulse */
    dist,                 /* distance entered in by the user */
    distnow,              /* calulation of current position */
    vel,                  /* velocity entered in by the user */
    ldelayv,              /* delay adjustment to make delayx()in seconds */
    temp,                 /* temporary variable used by Stof() */
    etime;                /* calculation of estimated elapsed time for move*/

/* time variables */
long t1,                  /* starting time of a move */
    t2,                   /* ending time of a move */
    atime;                /* actual time of a forward move, based on t2-t1*/

char dir, string[20];

/*set initial values*/
loop = 1;                 /* set loop to true */
ldelayv = 7*pow10(-6);    /* delay adjustment in seconds for delayx() */
res = .001875;            /* resolution for one motor pulse (inches) */
dist = 10;                /* distance of move specified in inches */
vel = .25;                /* velocity of move specified in inches/sec */
count = 0;                /* set inital position to 0 */
atime = 0;                /* set first move time to 0 */

/***** Main control loop *****/
while( loop == 1)
{
    /* calculate numbers that the program can use */
    steps = (dist/res);    /* number of motor pulses for next move */
    delay = (res/(vel*ldelayv)); /* delay between each pulse */
    distnow = (count * .001875); /* distance traveled since start program*/
    etime = abs(dist/vel);    /* estimated time for next move */

/**** status printout, performed after each move or change ****/
    clrscr();
    printf("\nCurrent position is: %6.4f inches\n",distnow);
    printf("Last Move required %ld sec\n",atime);
    printf("\n\nCurrent settings are:\n");
    printf("                Distance = %6.4f inches\n",dist);

```

```

printf("                Velocity = %6.4f inches/sec\n",vel);
printf("\nThe following values will be used by the program:\n");
printf("                Delay = %5i \n",delay);
printf("                Steps = %5i \n",steps);
printf("                Estimated Elapsed Time = %6.4f sec\n",etime);
printf("\nChoose a Command [C=Change settings; M=Move; H=Home; S=Stop
                Program]\n");

dir = getch();
printf("\n");

/*****begin if 'c'*****/
if (dir == 'c' || dir == 'C')
{
do
{
clrscr();
printf("Enter a new value or choose ENTER to retain current value\n\n");
printf("Distance can range (-50 to 50), current value [%6.4f inches]:
                ",dist);

gets(string);
if (string[0] == '\0')
temp = dist;
else
StoF( string, &temp);
}
while( temp < -50 || temp > 50);
printf("\n\n");
dist = temp;
do
{
printf("\n\nVelocity can range (.1 to 4), current value [%6.4f in/sec]:
                ",vel);

gets(string);
if (string[0] == '\0')
temp = vel;
else
StoF( string, &temp);
}
while( temp < .1 || temp > 4);
printf("\n\n");
vel = temp;
} /*****end if 'c'*****/

/*****begin if 'm'*****/

```

```

if (dir == 'm' || dir == 'M')
{
    if (steps < 0)
    {
        asteps = steps * -1;
        dirbit = 2;
    }
    else
    {
        asteps = steps;
        dirbit = 0;
    }
    time(&t1);
    for(i=0; i < asteps; i++)
    {
        output(PARA_PORT, (0x00 + dirbit));
        delayx(delay/2);
        output(PARA_PORT, (0x01 + dirbit));
        delayx(delay/2);
        ++count;
    }
    time(&t2);
    atime = t2-t1;
} /*****end if 'm'*****/

/*****begin if 'h'*****/
if (dir == 'h' || dir == 'H')
{
    if (count < 0)
    {
        asteps = count;
        dirbit = 2;
    }
    else
    {
        asteps = count * -1;
        dirbit = 0;
    }
    for (i=0; i< asteps; i++)
    {
        output(PARA_PORT, 0x00 + dirbit);
        delayx(delay/2);
        output(PARA_PORT, 0x01 + dirbit);
        delayx(delay/2);
    }
}

```

```

        }
        count = 0;
    } /*****end if 'h'*****/

    /*****begin if 's'*****/
    if (dir == 's')
    {
        loop = 0; /* stop loop by setting loop to false */
    } /*****end if 's'*****/
    } /* end while*/
} /* end of main */

/*****
function delayx()
    This function produces a delay and is used to produce
    a square pulse wave to prompt the motor drivers. This
    delay routine is linear, however, we can not know exactly
    how long it delays simply by looking at the code. Therefore,
    a delay adjustment value is used (variable ldelayv) to adjust to seconds.
*****/
void delayx(int dly)
{
    int j=0;
    for(j=0; j<dly; j++)
    {
        j=j;
    }
}

/*****
function StoF()
    This function is used to read decimal value numbers from the user
    and put these the value in the p_result variable. The usefullness
    of this procedure is in its ability to accept a carriage return
    and then retain the current value. Actual opperation of the program
    will help to illustrate this feature.
*****/
int StoF( char string[20], float *p_result)
{
    int    i, count, subnum, stop;
    float  number;

    number = 0.0;
    stop = 0;

```

```

if (string[0] == '-')
    i = 1;
else i = 0;

subnum = string[i] - '0';
while(subnum >= 0 && subnum <= 9 && stop == 0)
{
    number = (10 * number) + subnum;
    ++i;
    subnum = string[i] - '0';

    if (string[i] == '\0')
        stop = 1;
}

if (stop == 0 && (subnum < 0 || subnum > 9) && string[i] != '.')
{
    printf("\n    You have entered a non numeric value.\n");
    return 1;
}
else if (string[i] == '.')
{
    ++i;

    if (string[i] == '\0')
        stop = 1;
    else stop = 0;

    subnum = string[i] - '0';
    count = 1;

    while(subnum >= 0 && subnum <= 9 && stop == 0)
    {
        number = number + (subnum / pow(10, count));
        ++count;
        ++i;
        subnum = string[i] - '0';

        if (string[i] == '\0')
            stop = 1;
    }
}

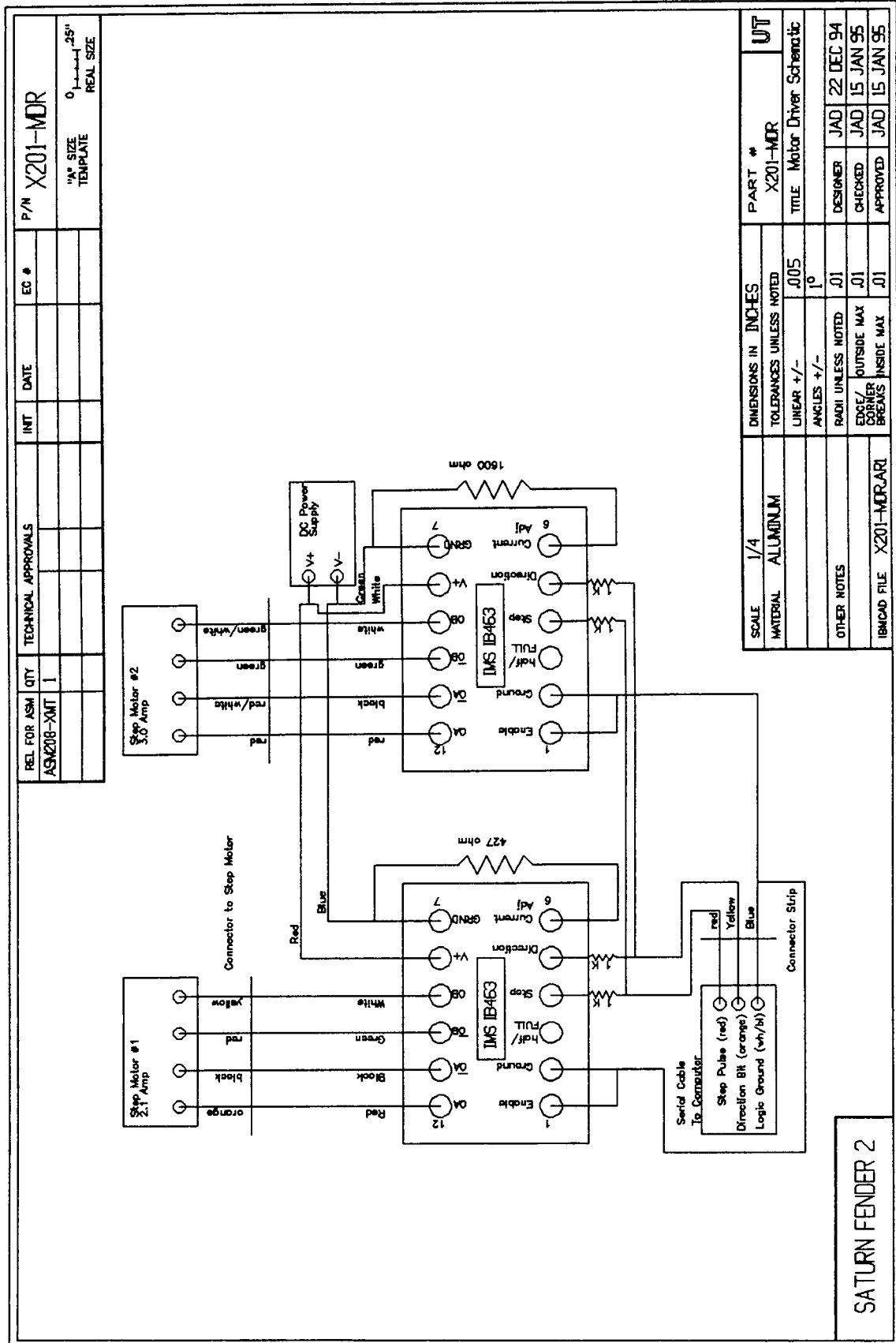
if (stop == 0 && (subnum < 0 || subnum > 9))

```

```
    {
    printf("\n    You have entered a non numeric value.\n");
    return 1;
    }
else
    {
    if (string[0] == '-')
    number = number * -1;
    *p_result = number;
    return 0;
    }
} /*end StoF*/
```

APPENDIX D

Motor Driver Wiring Diagram



REL FOR ASM	QTY	TECHNICAL APPROVALS	INT	DATE	EC #	P/N
AS2018-XMT	1					X201-MDR

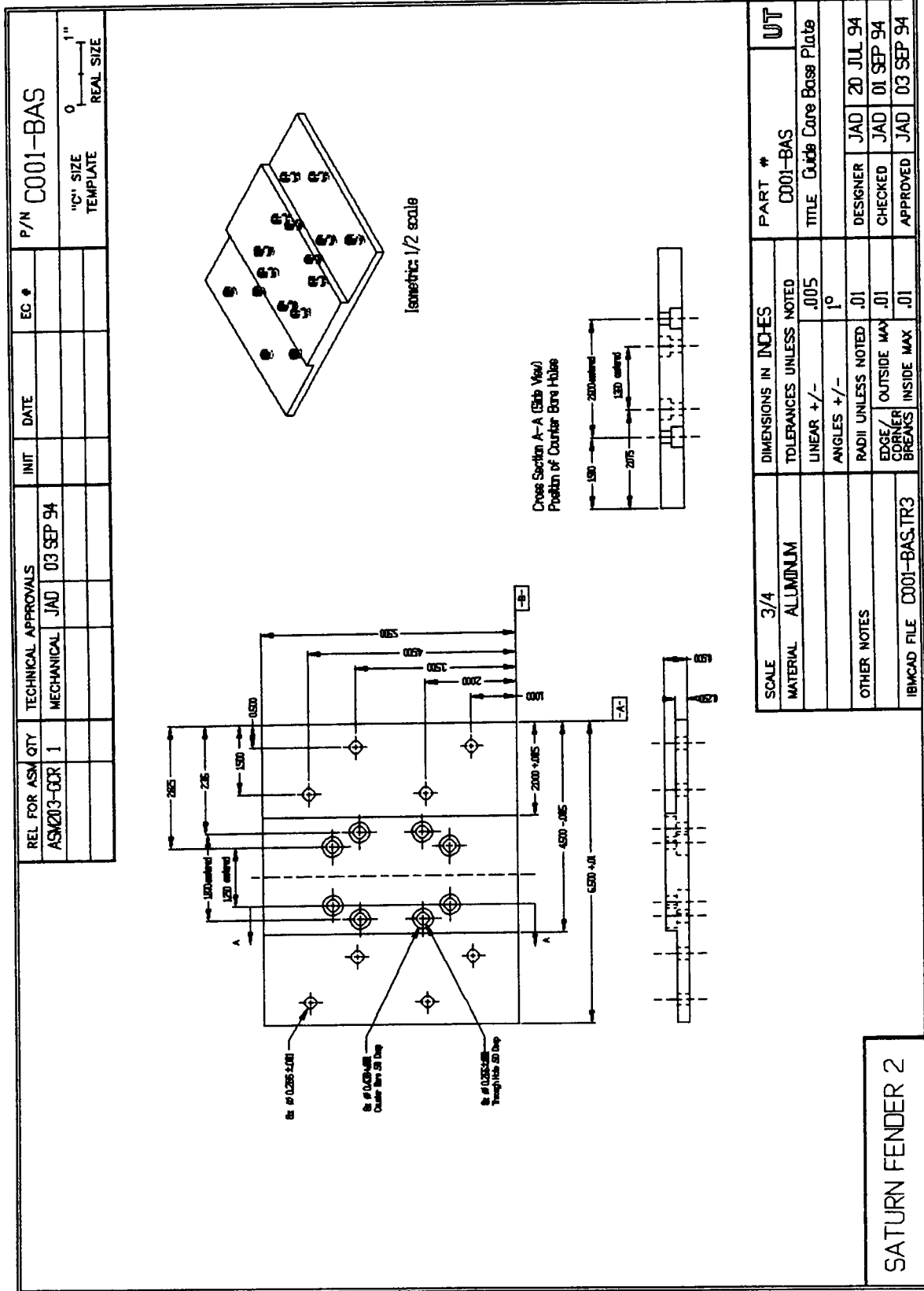
"A" SIZE
TEMPLATE
REAL SIZE
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40 41 42 43 44 45 46 47 48 49 50 51 52 53 54 55 56 57 58 59 60 61 62 63 64 65 66 67 68 69 70 71 72 73 74 75 76 77 78 79 80 81 82 83 84 85 86 87 88 89 90 91 92 93 94 95 96 97 98 99 100

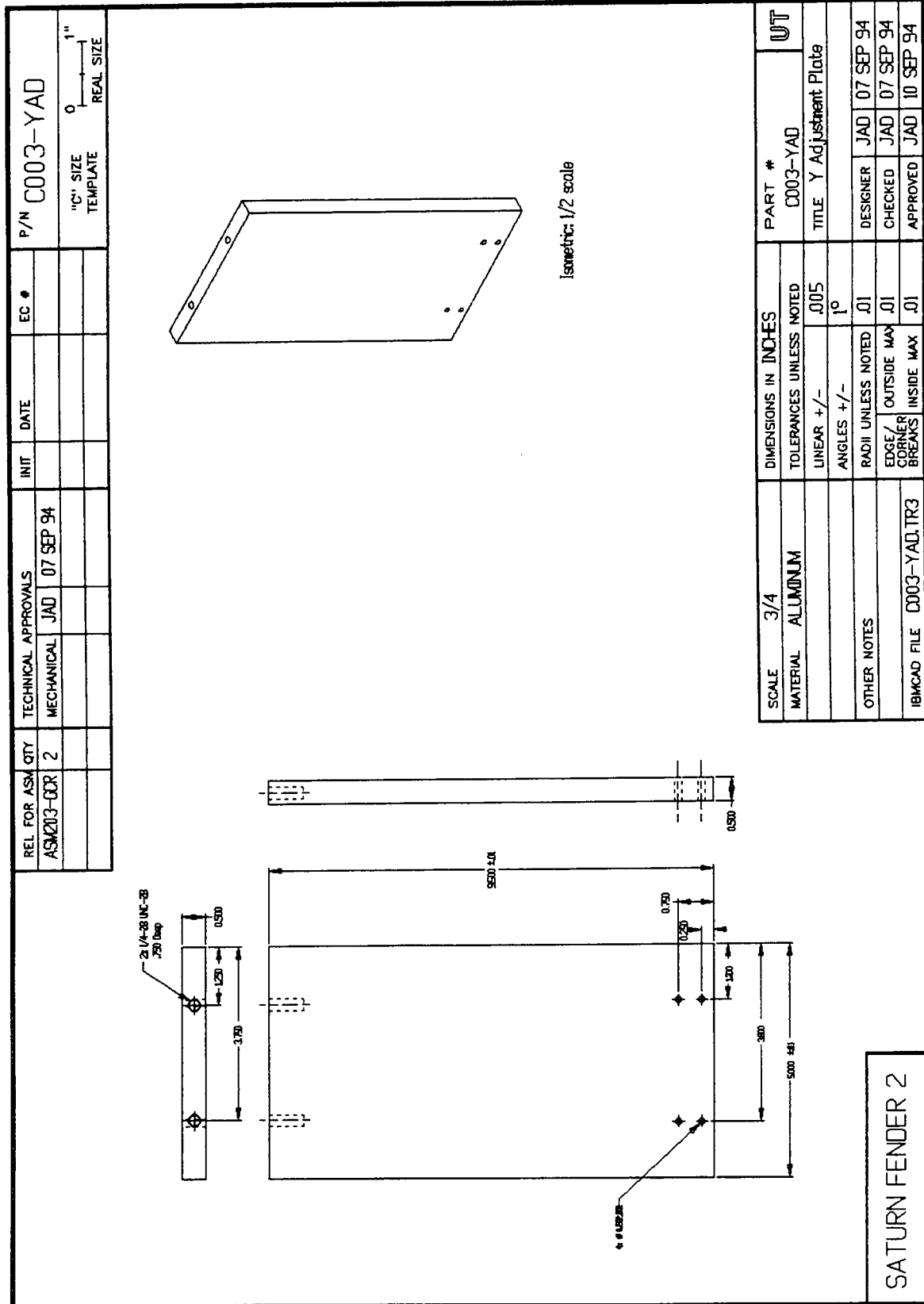
SCALE	1/4	DIMENSIONS IN INCHES	PART #	UT
MATERIAL	ALUMINUM	TOLERANCES UNLESS NOTED	X201-MDR	
		LINEAR +/-	.005	
		ANGLES +/-	1°	
OTHER NOTES		RADI UNLESS NOTED	.01	
		EDGE/ CORNER	OUTSIDE MAX	
		BREAS	INSIDE MAX	
IBCAD FILE	X201-MDR.ARI	DESIGNER	JAD	22 DEC 94
		CHECKED	JAD	15 JAN 95
		APPROVED	JAD	15 JAN 95

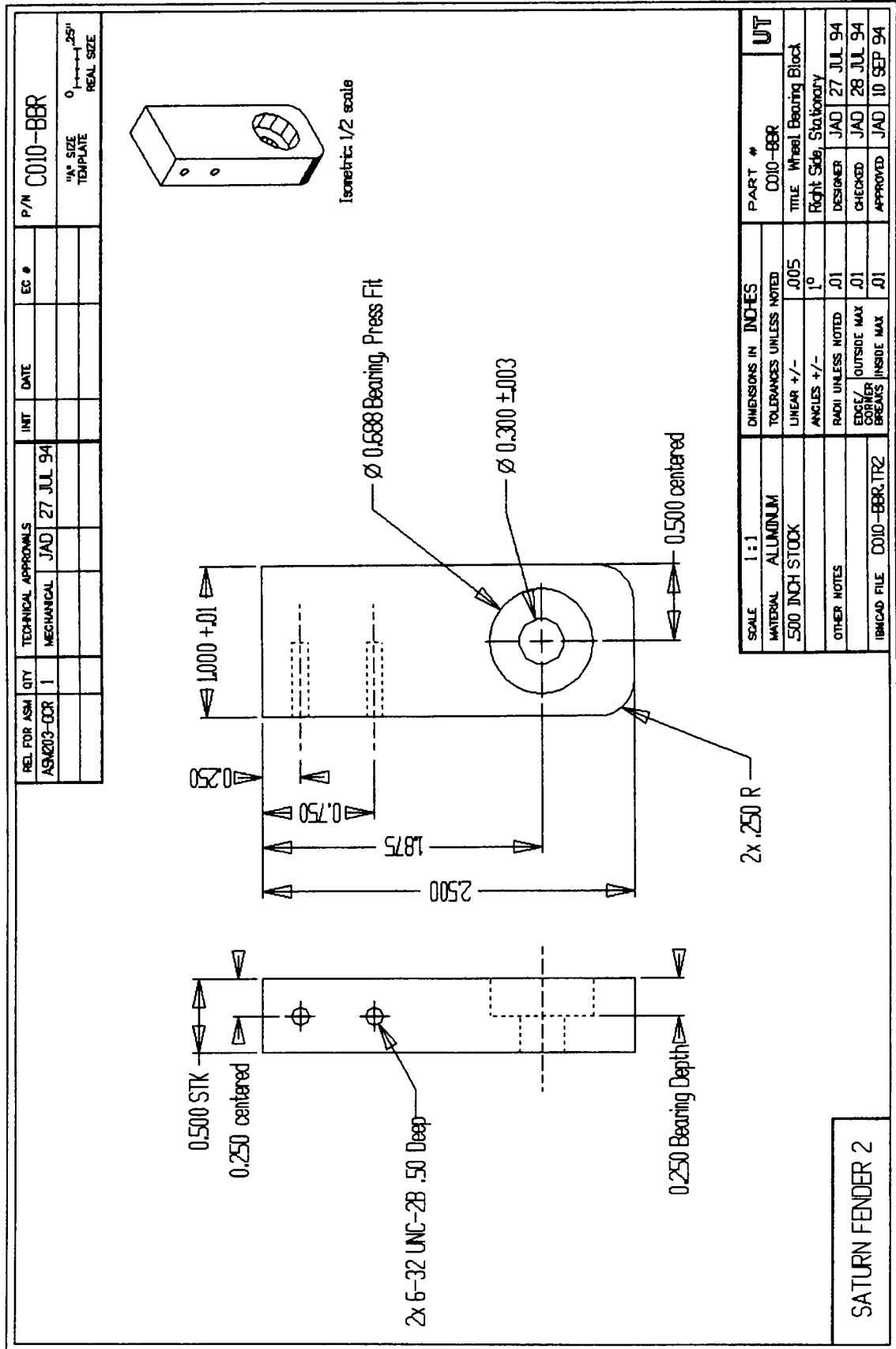
SATURN FENDER 2

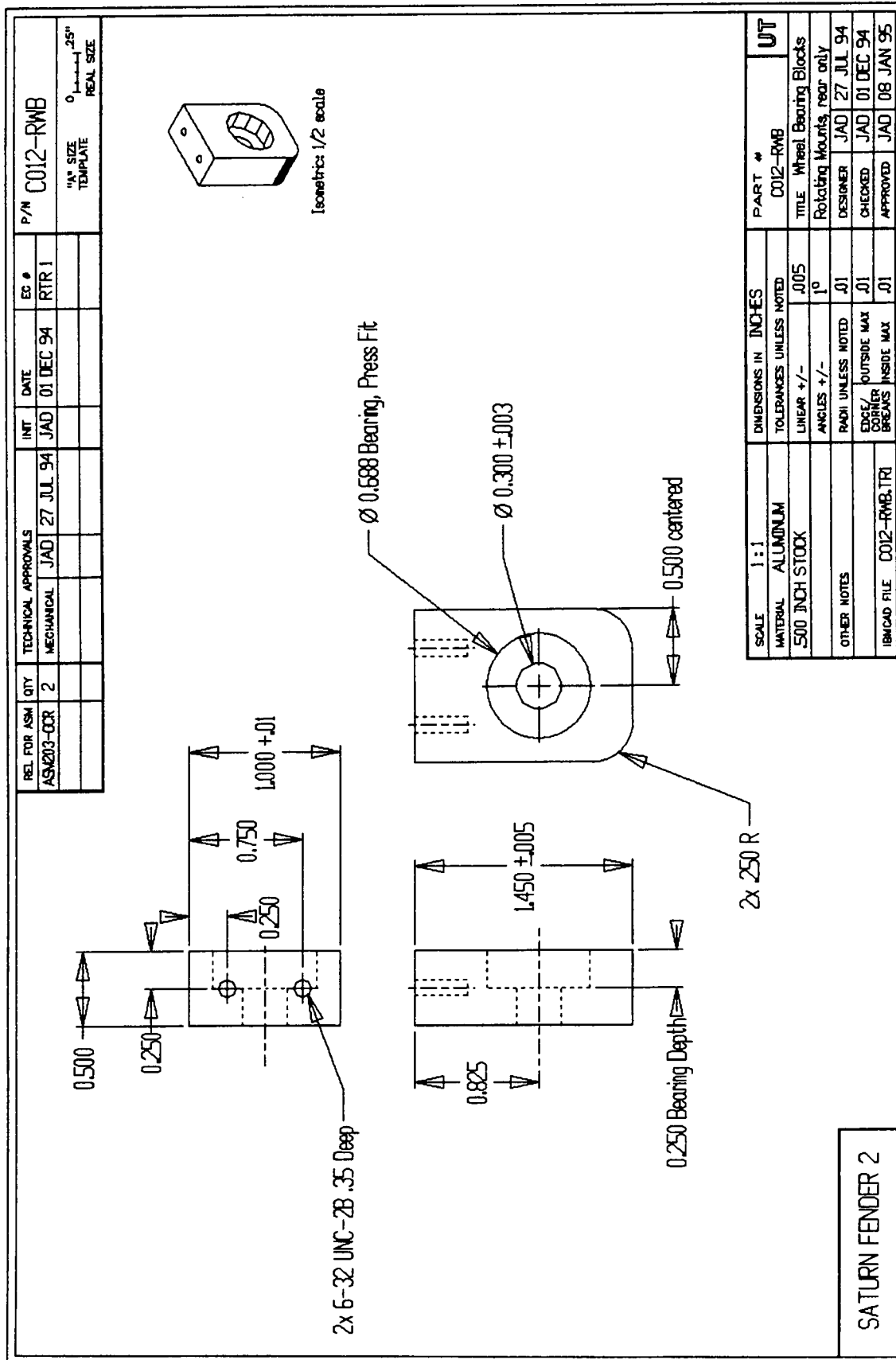
APPENDIX E

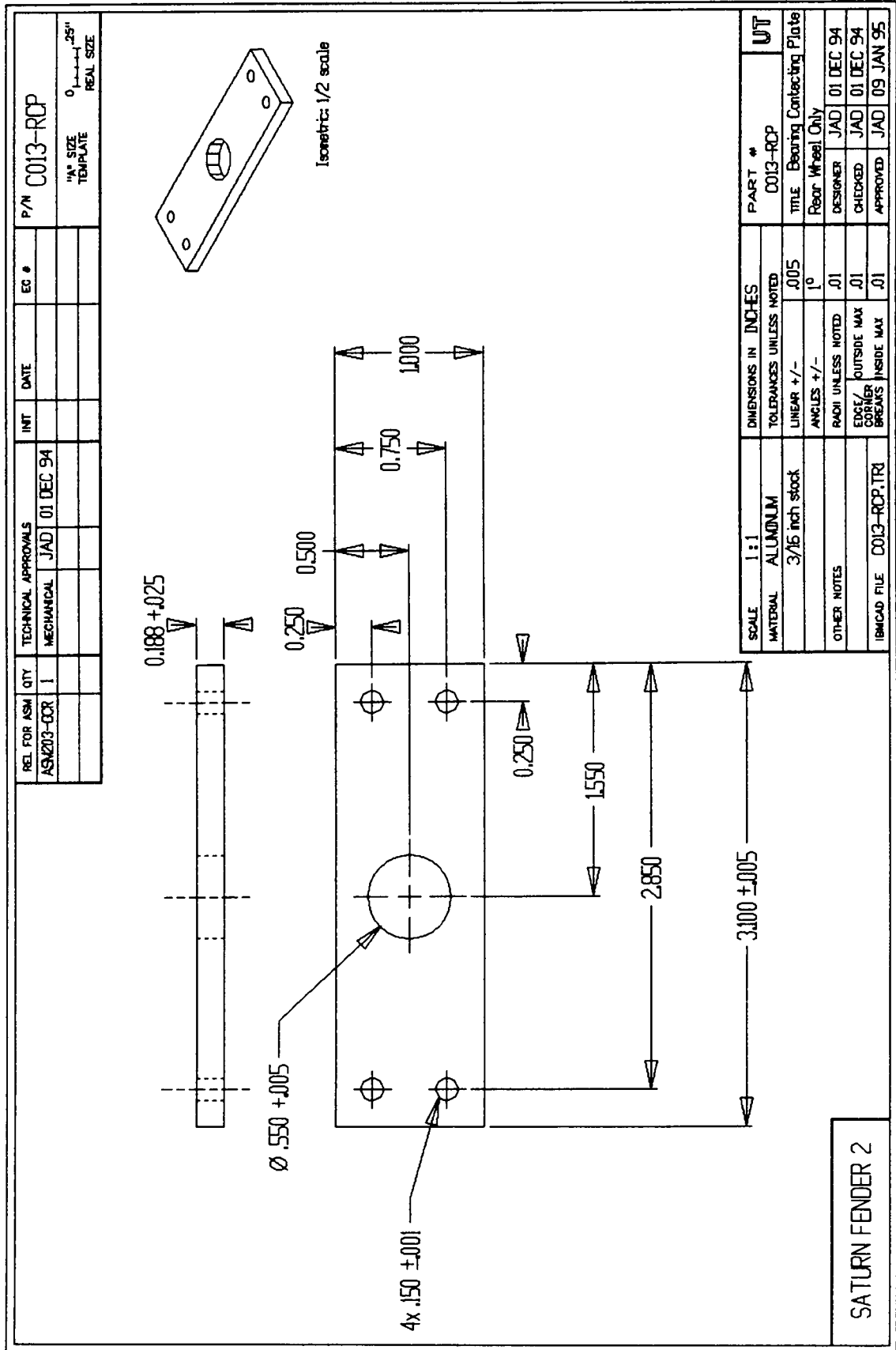
Part and Assembly Drawings

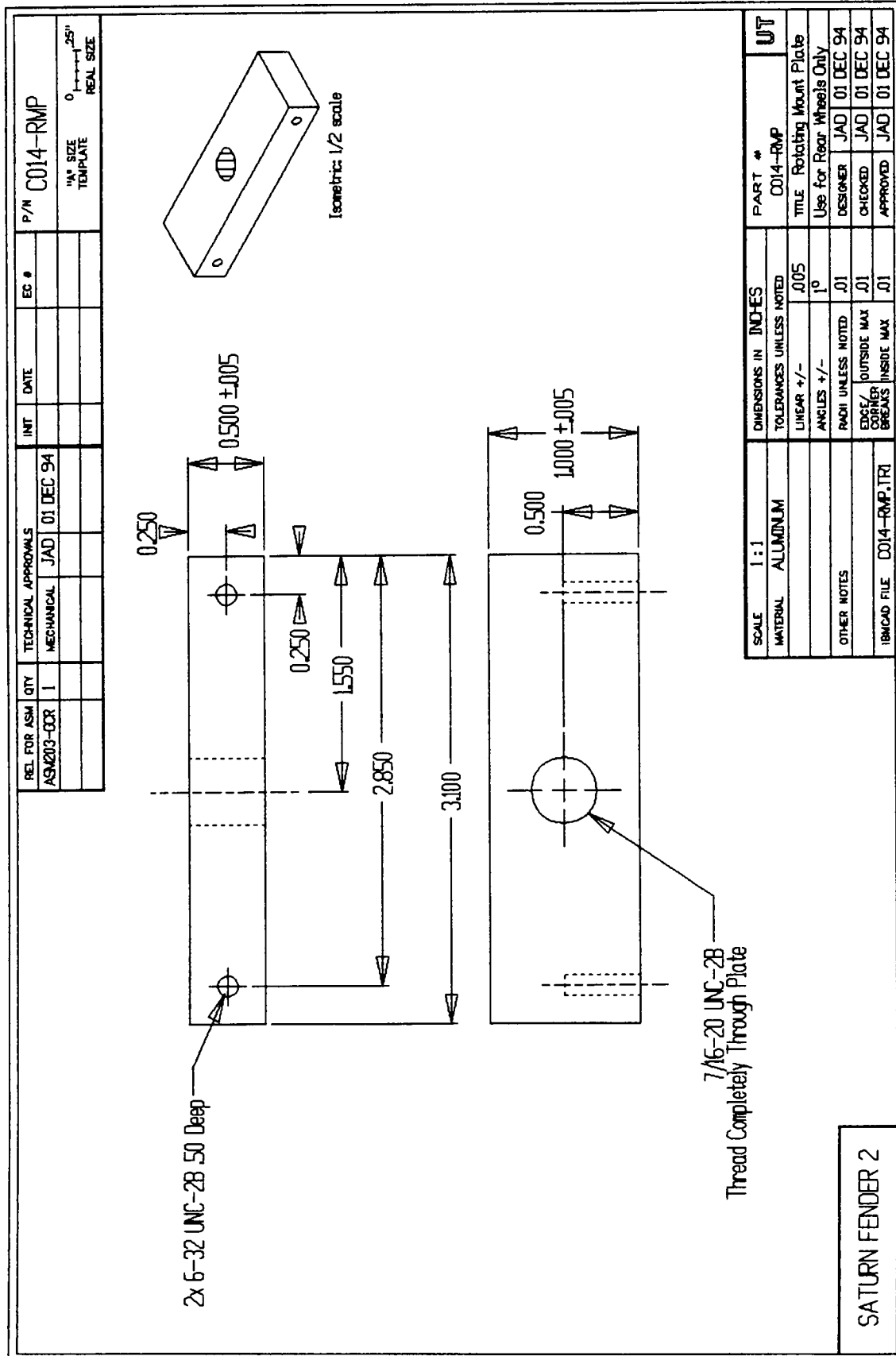


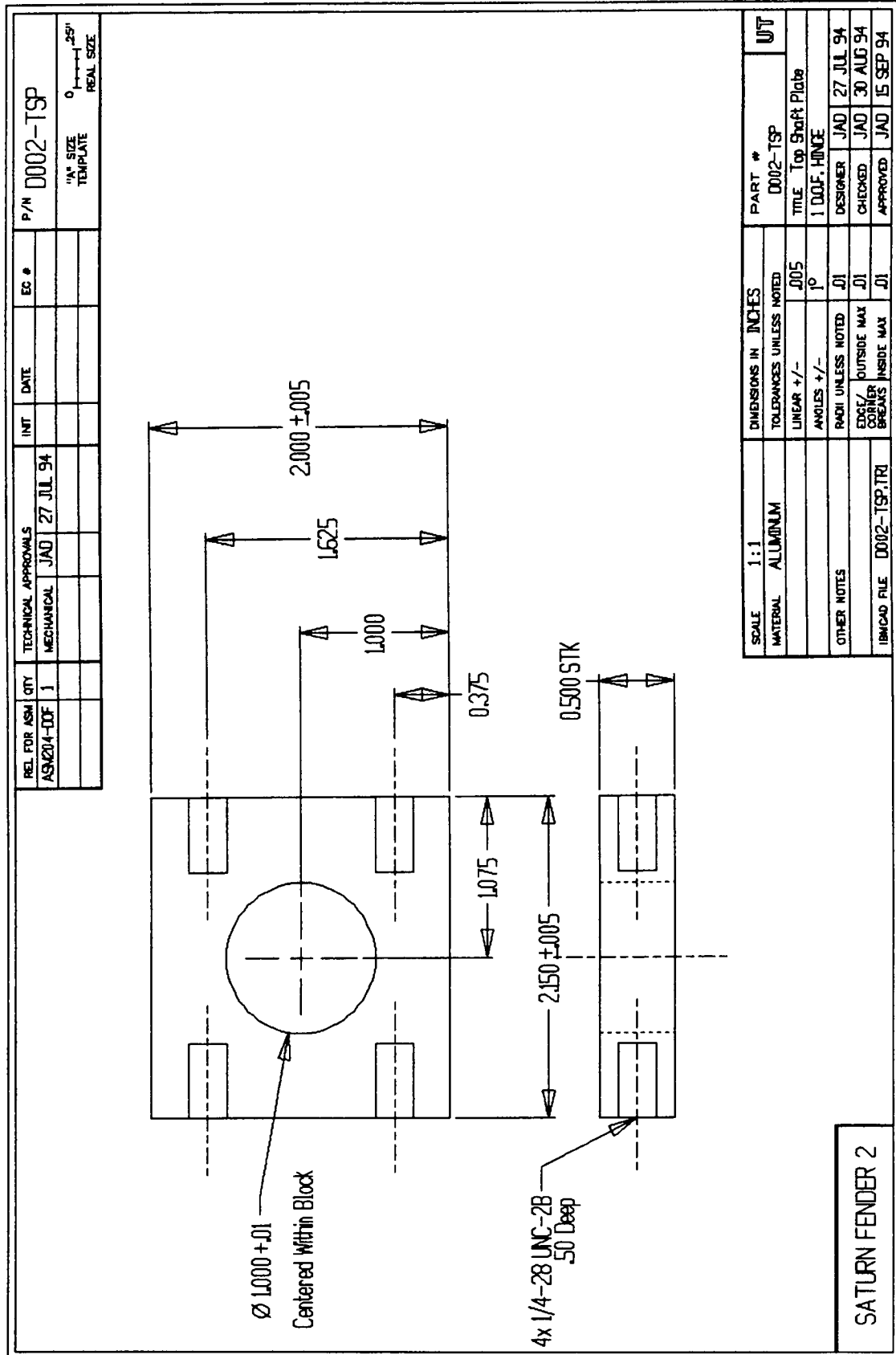




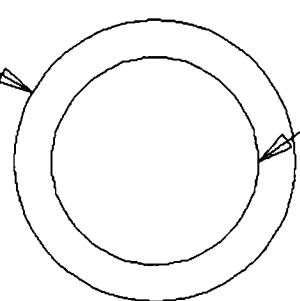




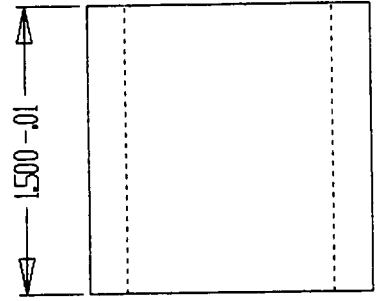




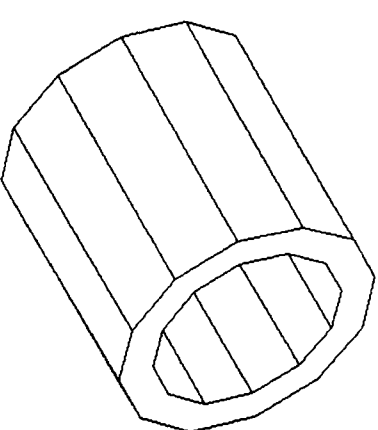
REL FOR ASM	QTY	TECHNICAL APPROVALS	INT	DATE	EC #	P/N
ASM205-PLT 1	1	MECHANICAL JAD 21 AUG 94				P003-SPC
						"A" SIZE TEMPLATE
						0 1/4" 25" REAL SIZE



Ø 1.500 +.050



1.500 -.01

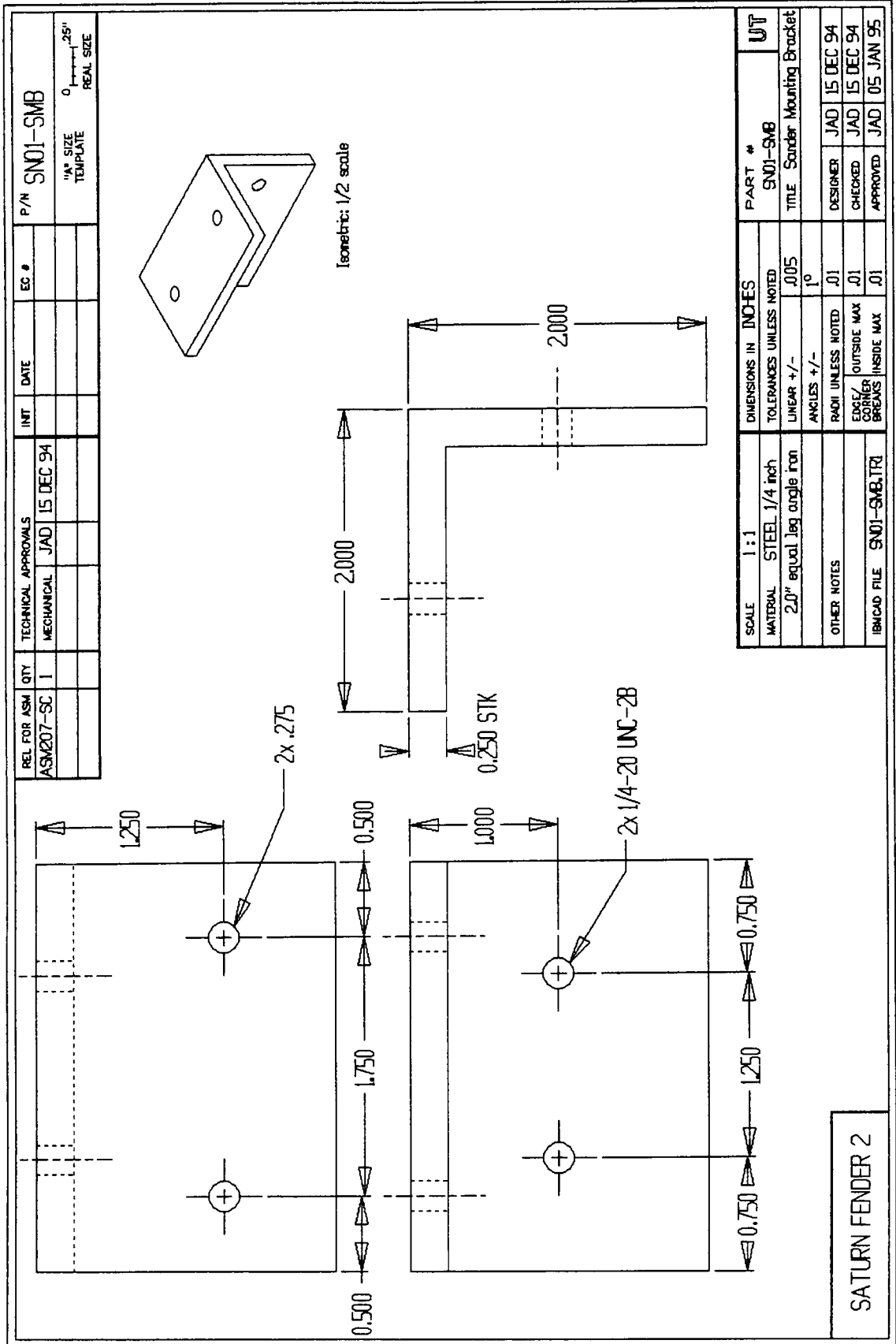


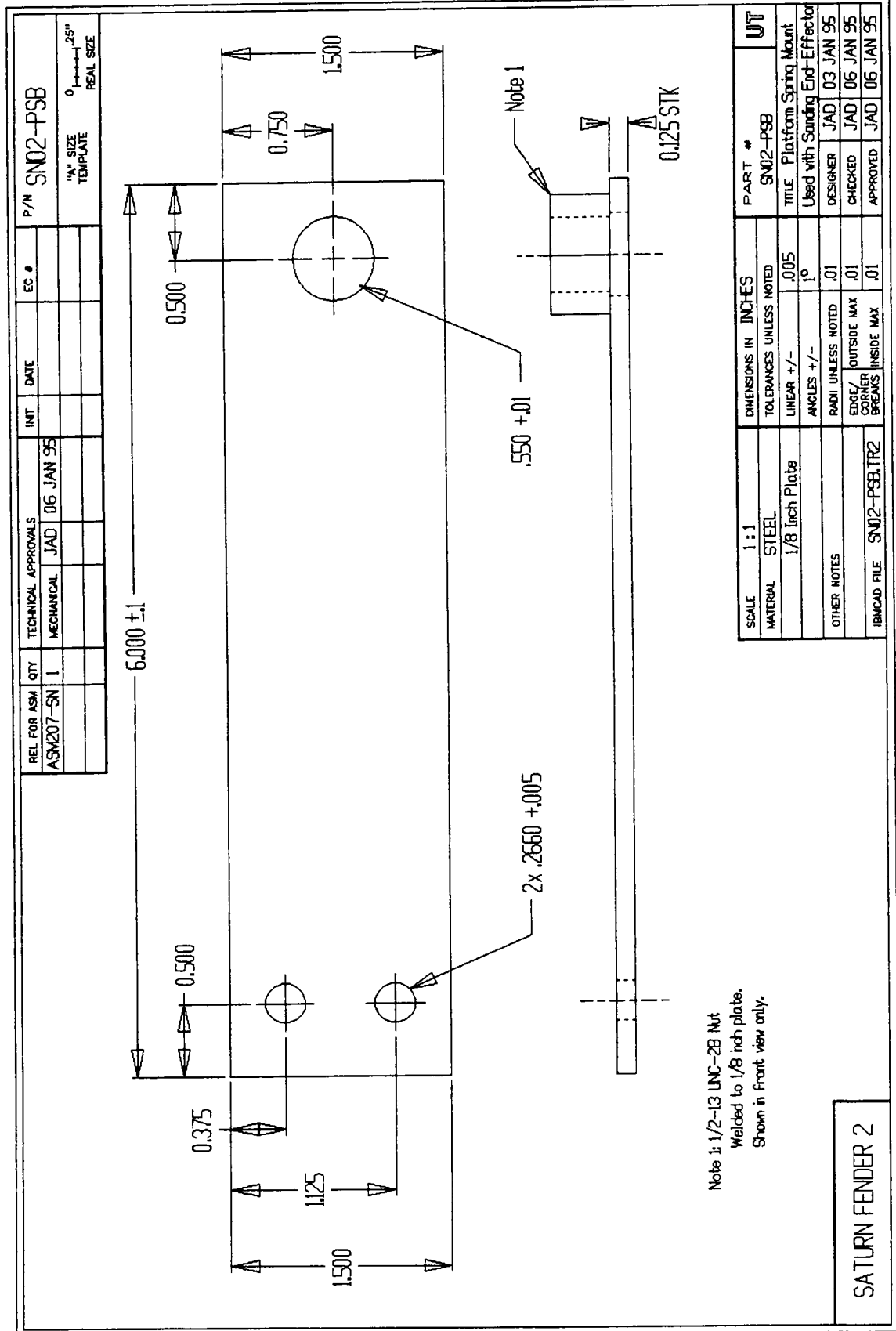
Isometric: 1:1 scale

Inner Diameter Range is
1.100 to 1.300
Smaller bore is desired

SATURN FENDER 2

SCALE	1:1	DIMENSIONS IN INCHES	PART #	UT
MATERIAL	ALUMINUM	TOLERANCES UNLESS NOTED	P003-SPC	
		LINEAR +/-	.005	TITLE Bearing Spacer
		ANGLES +/-	0°	Used in part P001-PLT
OTHER NOTES		RADI UNLESS NOTED	.01	DESIGNER JAD 21 AUG 94
		EDGE/ CORNER BREAKS	OUTSIDE MAX .01	CHECKED JAD 21 AUG 94
IB/CAD FILE	P003-SPC.TRI	INSIDE MAX .01	APPROVED JAD 10 DEC 94	





APPENDIX F

Fender Material Specifications

The fenders used in this project consist of a GE plastic, Noryl GTX910. This is a blend of polyphenylene ether and polyamide 66. The following is the basic property specifications for the GTX family.

NORYL GTX Resin Grades
English Units (SI Units)
Range of Typical Property Values

PROPERTY	ENG(SI) UNITS	TEST METHOD	GTX810 GTX820 GTX830	GTX901 GTX910 GTX917
PHYSICAL				
Specific Gravity, solid Density	— g/cc (oz/cu in)	ASTM D 792 ASTM D 792	1.10-1.33 1.11(0.040)- 1.33(0.48)±	
Water Absorption, 24 hours @ 73F (23C)	%	ASTM D 570	0.3-0.5±	
Water Absorption equilibrium, 73F (23C)	%	ASTM D 570	1.0±	
Mold Shrinkage, unannealed, 0.125" (3.2 mm)	in/in E-3	ASTM D 955	5.0 to 12-16±	
Mold Shrink, annealed, 325F (163C), 1hr, 0.125" (3.2 mm)	in/in E-3	ASTM D 955	6.5 to 15-17±	
MECHANICAL				
Tensile Strength, yield, Type I, 0.125" (3.2 mm)	psi(MPa)	ASTM D 638	8,000(55)- 22,000(150)	
Tensile Strength, break, Type I, 0.125" (3.2 mm)	psi(MPa)	ASTM D 638	8,000(55)- 23,000(160)±	
Tensile Elongation, break, 0.125" (3.2 mm)	%	ASTM D 638	70-60.0	
Flexural Strength, 0.125" (3.2 mm)	psi(MPa)	ASTM D 790	13,000(89)- 32,500(225)	
Flexural Modulus, 0.125" (3.2 mm)	psi(MPa)	ASTM D 790	315,000(2,200)- 1,200,000(8,300)	
Compressive Strength	psi(MPa)	ASTM D 695	11,500(80)- 26,000(180)±	
Shear Strength	psi(MPa)	ASTM D 732	9,000(60)- 11,000(75)±	
Hardness, Rockwell R	—	ASTM D 785	116-120±	

VITA

Jeremy Allen Davis was born in Brighton, Colorado on May 12, 1970, and grew up in the mountains of Grand County, Colorado. He graduated from Middle Park High School in May, 1988 and began his engineering education at Colorado State University that same year. While working on a Colorado Advanced Software Institute Undergraduate Research Grant, he began investigating novel ideas for high-level robot control. As a conclusion to the research, he co-authored a technical paper entitled "Task-achieving Modules for Robot Programming". He also worked two summers as an engineering intern with Storage Technology Corporation in Louisville, Colorado. In his senior year, Mr. Davis was elected as president of the Tau Beta Pi chapter, Colorado Delta. He received his B.S. degree in Engineering Science from CSU in May, 1993.

Mr. Davis continued his educational goals in the following semester by enrolling in the University of Tennessee, Knoxville and in May of 1995, completed his M.S. degree in Mechanical Engineering. During this time he was awarded the Saturn Sponsored SEDP Graduate Fellowship and used this as the foundation of his thesis project. Mr. Davis will now pursue a career in the industry of product manufacturing and design for manufacturing.