FLEXIBLE AUTOMATION FOR NEW PRODUCT INTRODUCTION

by

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ABSTRACT

Shorter product lifecycles are forcing manufacturers to re-consider the way their products are assembled. In current practice, new product families are often assembled manually with the aid of simple jigs and fixtures. Once demand for the new product greatly increases, manufacturers move into high-speed, automated assembly through the use of dedicated machines specifically designed for the new product. However, the transition from low demand to high demand presents a problem manufacturers often have difficulty addressing. Also, because of shorter product lifecycles, manufacturers find it more and more difficult to justify the time and funds required to build such dedicated assembly systems, particularly if they are to be used only for intermediate volume production. A clear need exists for assembly systems which are flexible enough to accomodate entire product families with little or no setup.

This thesis describes a design concept for a flexible assembly system consisting of seven modules that perform the primary operations required to assemble a class of products. These operations are: feeding, handling, serial insertion, parallel insertion, bending, and trimming. It also describes in detail the process through which a prototype of the bending module was designed and fabricated. This prototype bending module can bend multiple types of parts for products of different length. The prototype is capable of measuring and compensating for springback on-line.

Thesis Supervisors:
Doctor David Hardt, Professor of Mechanical Engineering
Doctor Andre Sharon, Executive Officer of The Manufacturing Institute

To My Parents:
Gonzalo Barrientos and
Corina Gutierrez de Barrientos

Acknowledgments

As I finish what has been an incredibly rewarding period in my life I look back and thank many people at the Massachusetts Institute of Technology without whose support I would not have been able to make it this far. My advisors, Andre Sharon and David Hardt, provided the funding, the advice, and more importantly, the trust that helped me learn what it takes to succeed. Moris Fresko shared his experience and his ideas very generously every time I turned to him. Wayne Dempsey, Charles Scheck, Brandon Gordon, and Tiina Hämeenanttila enriched my life with their friendship. Fred Cote took the time to teach me almost everything I know about machining, and at the same time became a good friend. Leslie Regan gave me the encouragement I needed just at the right time.

Outside of MIT I thank Amit Desai, Kate Baty, Ed Hanley, Dan and Barbara Harris, who gave me the type of friendships one could never forget. Claudia Soria, far away geographically but always very close to my heart, who has been my source of inspiration for more than six years. Last, but not least, I thank my family -Gonzalo, Corina, Velma, and Daniel- for all their love and unconditional faith in me.

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Chapter 1 - Introduction

Flexibility, defined as the ability to change over to produce multiple products within a family quickly and economically, is increasingly becoming more important as a design criterion for the development of modern manufacturing systems [1]. Shorter product lifecycles, driven by technological innovation and intense competition, and the well documented fragmentation of mass markets into niche markets, have made the flexibility of a manufacturing system a valuable source of competitive advantage[1].

In many manufacturing operations, new types of products are initially produced in small quantities. They are often assembled by hand, with the aid of various jigs and fixtures. As demand for the product increases, hand assembly becomes too costly. Thus, manufacturers will seek economies of scale by automating the process. Typically, manufacturers will jump directly into hard automation by modifying old machines, or by designing a new set of machines. In either case, the assembly machines are optimized around the new product design.

Both the development of new machines and the adaptation of existing machines are lengthy and costly processes. Using dedicated machines for the assembly of products is cost effective only if demand for the product is high during the life of the machines. However, demand for many types of products seldom jumps from quantities for which hand assembly is appropriate to quantities for which hard automation is justifiable. There is a period during which both hand assembly and hard automation are not cost efficient. Shorter lifecycles are also making the development of product-specific assembly systems a very risky option. Demand for a particular type of product may peak and decline long before a high-speed assembly system can be designed and built. Consequently, a clear need exists for multiple-product assembly systems that will bridge the gap between hand assembly and hard automation.

This thesis will describe the development of an automated assembly system for intermediate volume production that allows for rapid product changeover. The system was designed to assemble a class of products which require common manufacturing operations: feeding,

[1] See Appendix A for a detailed classification of flexibility.

handling, insertion, bending, and trimming. The development plan consisted of the following stages:

- Characterizing the assembly process
- Establishing concepts for fully flexible assembly
- Selecting and developing a full system concept
- Designing in detail the most promising components of the system

1.1 Assembly Process Characterization

The product assembly process was characterized by proposing simple product classification schemes, by identifying the process' constituent operations, by characterizing the process flow, and by identifying the degree-of-freedom requirements.

1.1.1 Product Classification

The first step toward developing a flexible assembly system consisted of classifying the products to be assembled so that common characteristics could be identified. Since the products to be assembled come in thousands of different types, this task could have easily become a project by itself. Thus, a deliberate effort was made to develop a very high-level classification scheme.

The product selected for assembly is typically comprised of a plastic casing and metallic inserts or parts[2]. Casings come in many different shapes; however, for the purposes of assembly, they all share a few common characteristics (see Figure 1).

[2] Casing and part terminology is included in Appendix C.

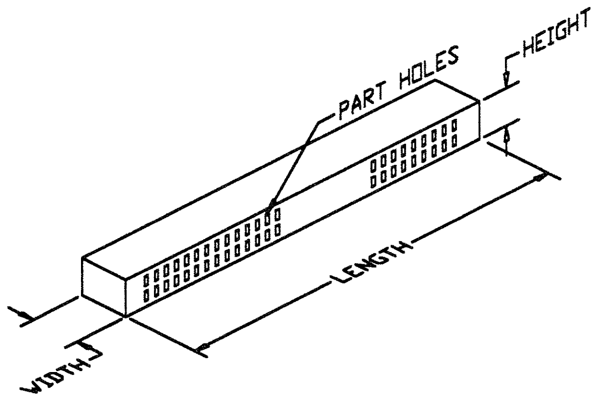

Figure 1: Typical product casing

First, their length to height, and length to width ratios are seldom smaller than two to one, and can be as high as twenty to one. Second, the mating face, i.e. the face on which parts are inserted into, usually consists of a flat surface regardless of the shape and size of the part holes. Third, if very small features are not taken into account, most casings have two planes of symmetry: the x-z plane, and the y-z plane[3].

Parts are even more diverse than casings, and thus harder to classify. One possible classification scheme would group parts by their carrier strips. Most parts come in one of three different types of carrier strips: side, center, and ladder (see Figure 2 thru Figure 4).

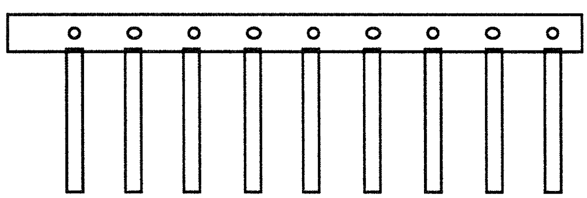

Figure 2: Side carrier strip

[3] The following conventions have been used throughout this document:

x-axis: axis parallel to plane of casing mating face, horizontal with respect to ground.

y-axis: axis parallel to plane of casing mating face, vertical with respect to ground

z-axis: axis perpendicular to plane of casing mating face.

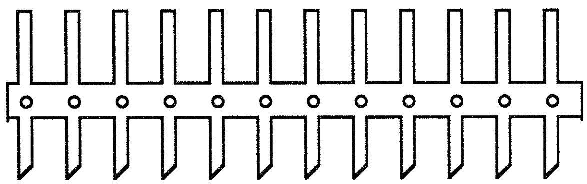

Figure 3: Center carrier strip.

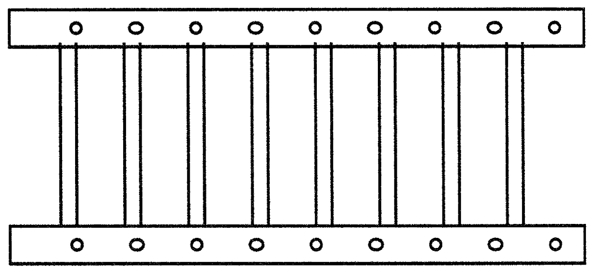

Figure 4: Ladder carrier strip.

A closely related classification method would group parts by how they are separated from the carrier strips. Parts are sheared off, or broken off (in cases where the parts have been pre-notched).

A third classification method would group parts by how they are inserted into the casings. The two methods of insertion are serial insertion (parts inserted one at a time), and parallel insertion (multiple parts inserted at once).

Finally, parts could also be grouped by how they fit into a casing. The most common types of fits are snap-on fit, press fit (where the hole's cross sectional area is smaller than the part's cross sectional area), and crimping fit (where the casing is crimped after the parts have been inserted).

It must be noted that the classification methods outlined above are not meant to be comprehensive when used in isolation. Each is particularly suited for one or two assembly operations; thus, they should be combined together to provide an accurate description of the different types of parts. For example, for the insertion operation, a part may be fully described by specifying the type of carrier it comes mounted on, how it is separated from the carrier, the insertion method (serial or parallel), and the way the part fits into a casing.

1.1.2 Assembly Operations

Four primary and two secondary assembly operations have been identified. The primary operations are:

- Casing feeding and positioning
- Part insertion (parallel and serial)
- Bending
- Trimming (to final length)

It should be noted that the part insertion operation includes part feeding, and part separation from carriers. Moreover, the bending and trimming operations are not always necessary; some types of parts are trimmed to their final length as they are inserted, and need not be bent for the product to be fully assembled.

The secondary operations are:

- Inspection (during and after assembly)
- Labeling

Inspection during assembly (in-line inspection) is performed after insertion to ensure that parts and casings have not been damaged, to verify that no parts are missing, and to check that all specified tolerances have being met.

1.1.3 Process Flow

The flow of the product assembly process may be described through the flowchart shown in Figure 5.

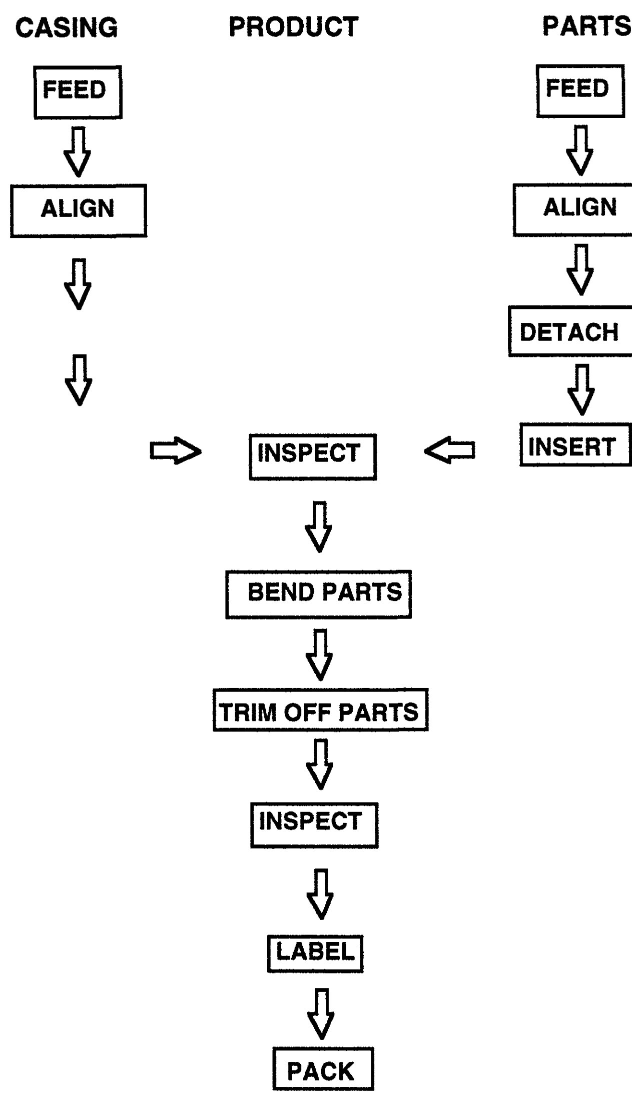

Figure 5: Flow chart of the product assembly process.

The figure shows that operations are performed at three levels. Some operations are performed at the casing level, others at the part level, and the rest at the product level. The figure also shows that operations such as alignment and feeding of parts and casings may be performed simultaneously.

1.1.4 Degree of Freedom Requirements

Most product assembly operations require a maximum of five degrees of freedom (dof). Such degrees of freedom are:

- Translation along the x-axis
- Translation along the y-axis
- Translation along the z-axis
- Rotation in the x-y plane
- Rotation in the z-y plane

The first two are required primarily for positioning of the casings prior to and during the insertion operation. Translation along the y-axis is also needed for the trimming operation. Translation along the z-axis is required to perform part insertion, and also to position the casing with respect to the bending tool. Rotation in the x-y plane is needed to bend the parts. Finally, rotation in the z-y plane is needed to flip the casings so that multiple rows of parts may be inserted, bent, or trimmed.

The degree of freedom requirements are displayed in Figure 6.

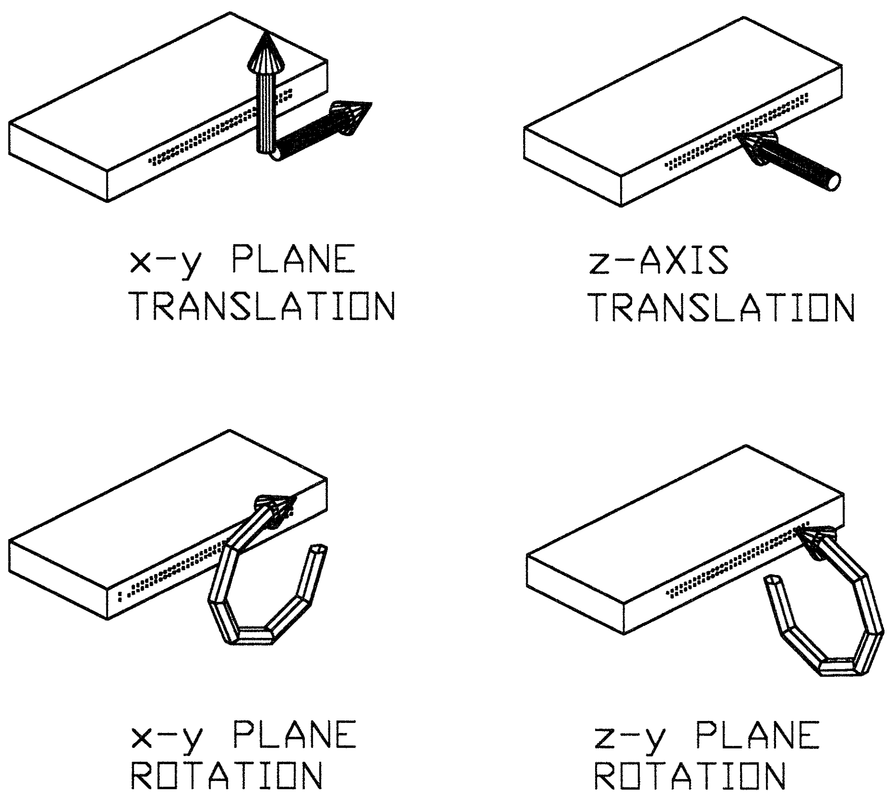

Figure 6: Degrees of freedom required for product assembly.

Chapter 2 - System Design

System design started by considering two options. The first option consisted of developing a single multi-purpose machine that would be capable of performing all the assembly operations. To some extent, such a machine would mimic what a human operator is capable of accomplishing equipped with a pair of pliers. The second option consisted of developing a set of modules, each performing a specific operation, that acting in unison would assemble a product from start to finish with the required flexibility.

Comparison of the two options showed that the modular approach was better suited to the task. First, a modular system could be customized, through addition or removal of modules, for the assembly of both simple and complex products. In contrast, a single multi-purpose machine would have to be designed with the most complex product in mind. Second, a modular system could be easily upgraded as better methods for each of the assembly operations are developed, since each of the modules could be replaced independently. Finally, a modular system would be more robust to failure. If one of the modules failed, it would be easier to isolate and fix the problem. It may even be possible to quickly replace the broken module with a new one, a very expensive solution if a monolithic system were to be used.

2.1 Concept Generation

Selection of a modular approach to assembly led to the development of two high-level system concepts. The first system concept consisted of a relatively simple casing handling mechanism (e.g. a conveyor belt supplied by a feeder) that would transport the casings to relatively complex insertion, bending, and trimming modules. The second system concept is simply the opposite of the first, i.e. a complex casing handling system coupled with simple insertion, bending and trimming modules.

2.2 Concept Comparison & Selection

The primary selection criterion at this stage was complexity, which has direct impact on cost and robustness of the system. One way of measuring the complexity of a system is to sum the number of degrees of freedom (dof) of all the components. The larger the number of dof, the higher the complexity. A system with a simple casing handling mechanism would necessarily require that the other modules have all the degrees of freedom needed to assemble the parts. Since some operations share dof requirements with others, repetition of dof would be unavoidable. In contrast, a system with a more complex casing mechanism would avoid repetition by consolidating dof common to multiple operations into one module. Given these considerations, the latter system concept was chosen for further development.

The next step in the development of the system consisted of deciding how the modules were to be arranged. Two options were considered, a circular arrangement, and a linear arrangement[4].

The circular rotary system, shown in Figure 7, clusters the modules around a rotational actuator, which transports the casings to each module. Advantages of a rotary system include the opportunity to assemble multiple casings at a time (through use of multiple gripper arms), and the lower cost associated with rotary actuators. The primary disadvantages are the lack of expansion flexibility, and the lack of precision that may be encountered when displacing small angular increments during the insertion process. An additional actuator would be required in order to allow small movements along the x-axis.

[4] This section and the remainder of the chapter was prepared in collaboration with Wayne R. Dempsey.

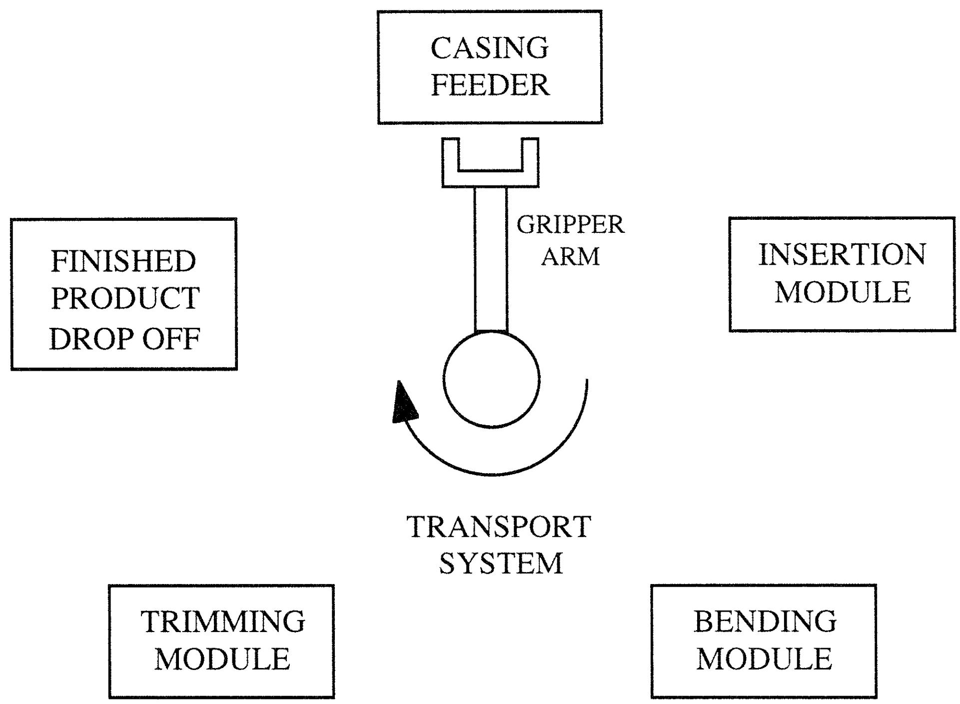

Figure 7: Circular arrangement.

A linear arrangement of the modules, shown in Figure 8, requires a relatively long transport mechanism, which picks up the casings from the casing feeder and transports them to the assembly modules. As products become increasingly more complex, their assembly may require additional modules to be added to the assembly system. The use of the linear transport system along the x-axis allows for convenient expansion of the system with additional modules. One possible disadvantage of the linear arrangement is that the transport module will have to return empty to the feeding tray after each product is assembled, whereas with a circular arrangement the last module (finished product drop-off) would be next to the first module (casing feeder).

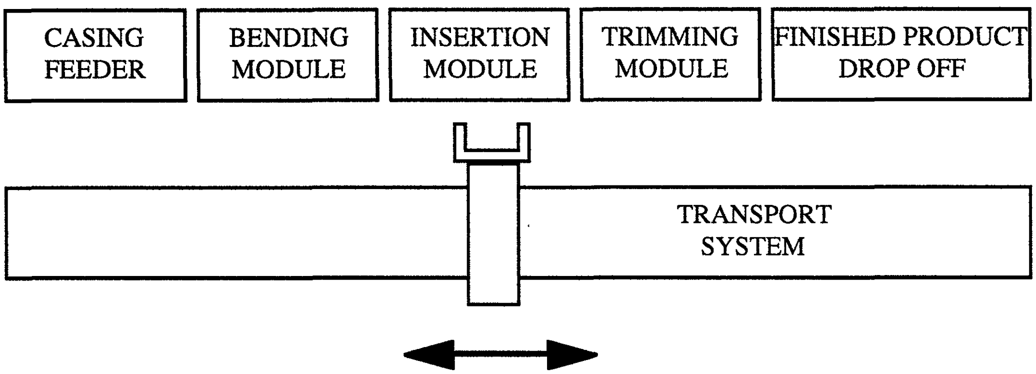

Figure 8: Linear arrangement.

The prevailing criterion at this stage was expansion flexibility. Different types of the products to be assembled require different modules, and thus the ability to expand or contract the assembly system is crucial. Consequently, the linear arrangement was selected for further development.

2.3 System Overview

The modules involved with the assembly of the products are described in the following sections. They include:

- **Casing Transport Module.** This module is responsible for gripping the casing housing, and provides a majority of the motions required for the various assembly operations.
- **Universal Casing Feeding Tray.** This module stores casings in a queue, and provides the Transport Module with the current casing for assembly.
- **Mass Insertion Module.** This module is designed to insert an entire row of parts at once into a casing.
- **Bending Module.** This module is designed to form the complex bends required for many products.
- **Universal Trimming Module.** This module is a multi-purpose trimming tool that trims parts to their final lengths.
- **Serial Insertion Module.** This module is designed to insert a single part at a time into a casing.

The modules are shown in Figure 9.

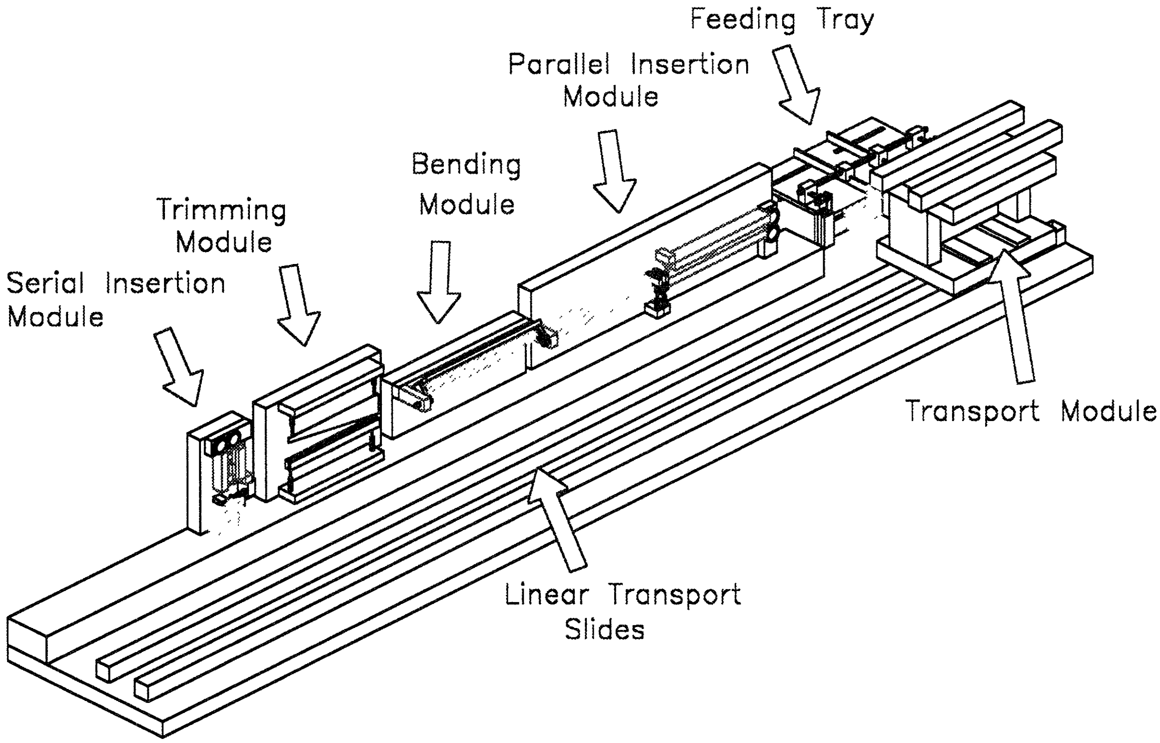

Figure 9: Flexible Assembly System Integrated View.

2.4 Universal Casing Feeding Tray

The universal casing feeding tray, shown in Figure 10, supplies one casing at a time to the transport module, while storing a finite supply in a queue. The feeding tray was designed to hold any size or type of casings, and feed them into a transport mechanism for use within the assembly process. Multiple casings can easily be stacked in the queue by an operator, and the design of the tray requires no casing-specific tooling for operation.

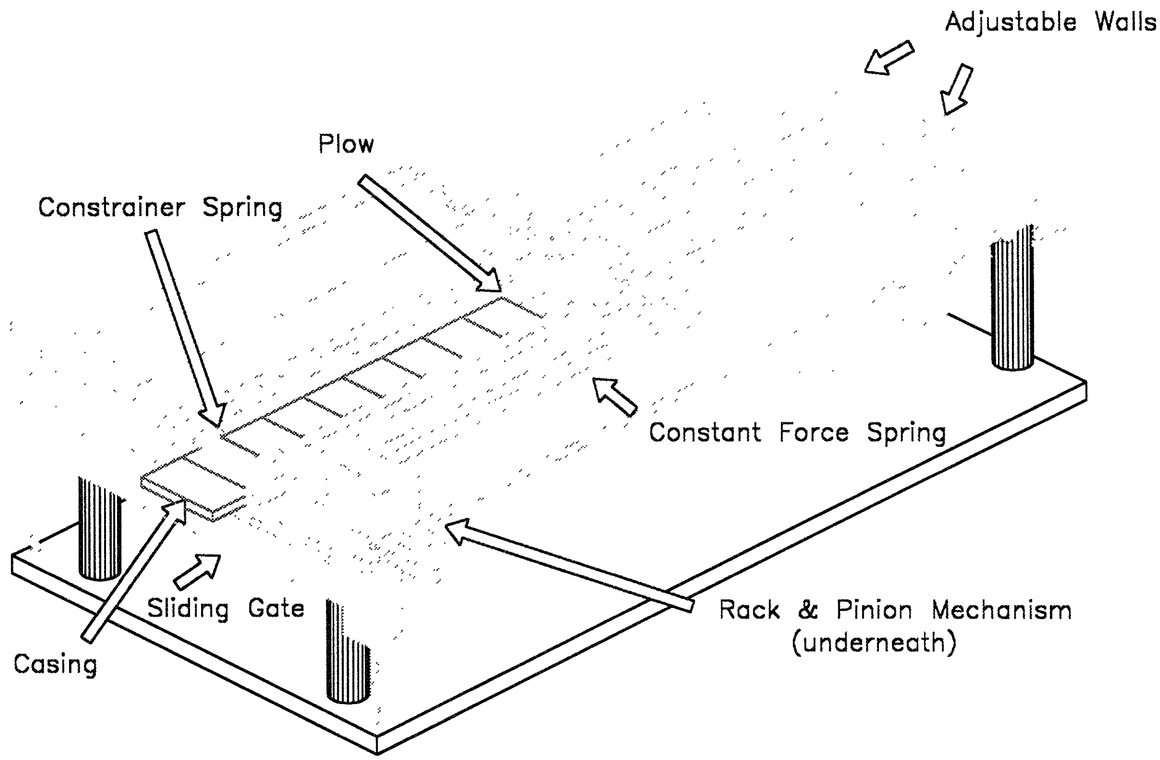

Figure 10: Feeding Tray

The feeding tray was designed to accommodate various length and width casings. Two adjustable sliding walls are used to vary the tray width to the appropriate casing length. The two walls are constrained with two rack and pinion mechanisms located underneath the tray. One on either end of the tray, the dual rack insures uniform motion along the length of the tray. The mechanism constrains the motion of the walls so that each wall is always an equal distance from the center of the tray. The symmetrical motion of the adjustable walls insures that the casing always lies in the center of the tray.

The casings are supported in the tray using a system that applies a constant force on the last casing in the queue. Two plows, connected through an upper rod, uniformly apply the constant force to the casings. The constant force is applied to the connecting rod using a pair of constant force springs. The first casing is restrained by a spring loaded gate. When grasping the casing the gate is shifted downward by the motion of a gripper mechanism. The first casing is pushed out of the tray by the force of the constant force springs. After the gripper is moved upward with the casing in its grasp, the gate is returned to its original position.

The overall length of the tray determines the number of casings that can be stored in the queue. In the final design concept, the tray length will be extended to the minimal length required for continued assembly, without compromising space requirements. Additionally, varying amounts of force may be required within the constant-force spring system to insure smooth feeding of casings within the tray.

Although complex, the feeding mechanism of the tray is better suited to the types of casings the system will be assembling than a gravity feeder, also considered during the brainstorming phase. Feasibility tests showed that a gravity feeder could easily jam because some of the casings have interlocking features built in.

2.5 Transport Module

The primary purpose of the transport module is to manipulate and move the casings to and from each assembly module. The module consists of a three degree of freedom manipulation device attached to a linear slide. A flexible gripping mechanism is attached to the transport module and is used to secure the casing during the assembly process. This flexible gripper, combined with the motion of the transport module and the various assembly modules, can assemble a wide variety of casings.

The casing transport module, displayed in Figure 11, consists of the following elements:

- Y-axis linear slide mounted on a x-axis linear slide,
- Rotational servoed actuator mounted on the y-axis slide
- Z-axis slide mounted onto the rotational servoed actuator
- Interchangeable grippers mounted on the z-axis slide.

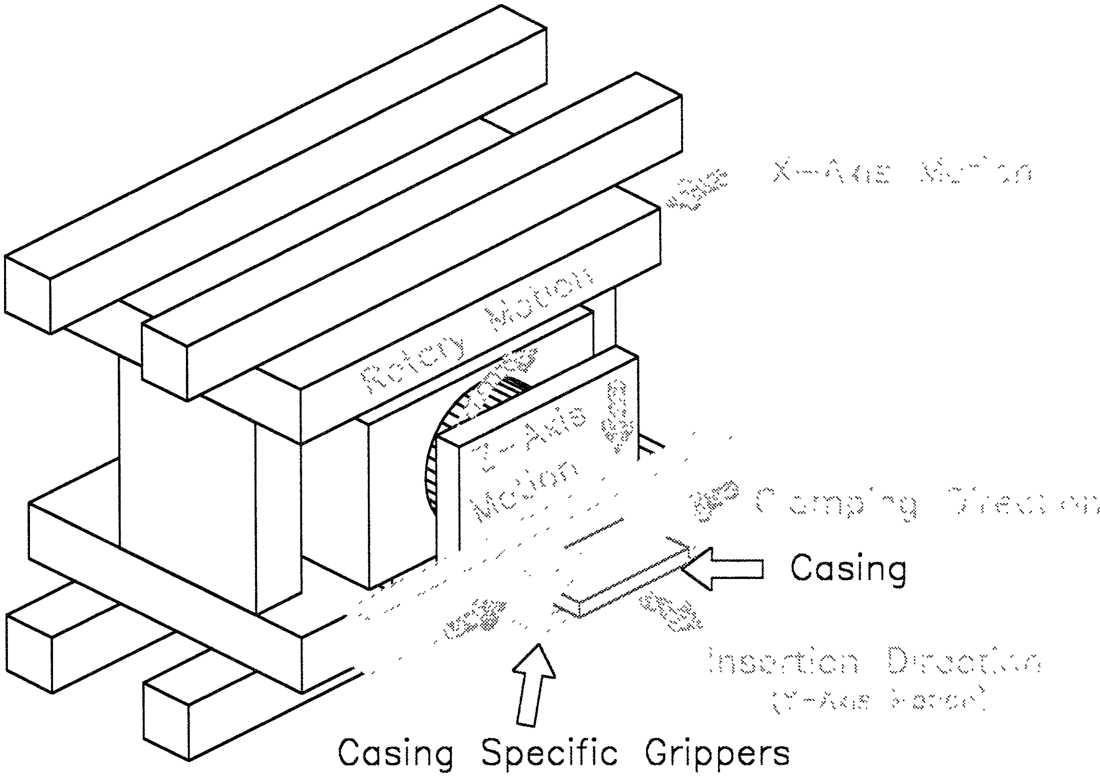

Figure 11: Casing Transport Module

The Casing Transport Module moves the casing along the x, y, and z axes. Additionally, it provides rotation about the y-axis. The x-axis linear slide is used to transport the casing from one module to another. It can also position the casing in small increments during the serial insertion operation. The y-axis linear slide is primarily used for insertion operations. The heavy-duty y-axis actuator pushes the casing into the parts that are held stationary by one of the assembly modules. The y-axis is also used to precisely position the casing prior to the bending and carrier removal operations. The z-axis linear slide is used to align the casing vertically, and also to manipulate the spring-loaded gate on the feeding tray. Customized grippers, designed to grasp a particular type of casing, are attached to the z-axis linear slide. The rotary actuator is used to flip the casing 180 degrees to facilitate different assembly operations.

The transport module picks up one casing at a time from the feeding tray. The gripper is designed to grasp housings after sliding the spring-loaded gate on the tray out of the way. Once the tray mechanism inserts a casing into the gripper, the gripper clamps down on it, securing it along every axis. The casing is then ready to be transported to the assembly modules.

Consolidation of the four primary degrees of freedom required to assemble a product within the transport module simplifies the design of the other assembly modules. Only a few additional actuators have to be incorporated into the design of each assembly module, reducing both cost and complexity. However, the design of a four degree-of-freedom positioning mechanism will require a complex control system to enable it to perform all the motions required by each type of product. In order to simplify the design of the transport module, the y-axis actuator (used to insert the parts into the casing) may be mounted behind the assembly modules that require insertion operations. Leaving this actuator off of the linear slide will reduce complexity and weight, and most likely will increase overall accuracy and repeatability.

2.6 Mass Insertion Module

The Mass Insertion Module is designed to insert an entire row of parts into a casing. The module consists of a part indexer, a set of part cutters, and a set of discrete flexible grippers. The module was designed so that the indexer, cutters, and gripper fingers can be easily interchanged to accommodate large variety of part types. The Mass Insertion Module is displayed in Figure 12.

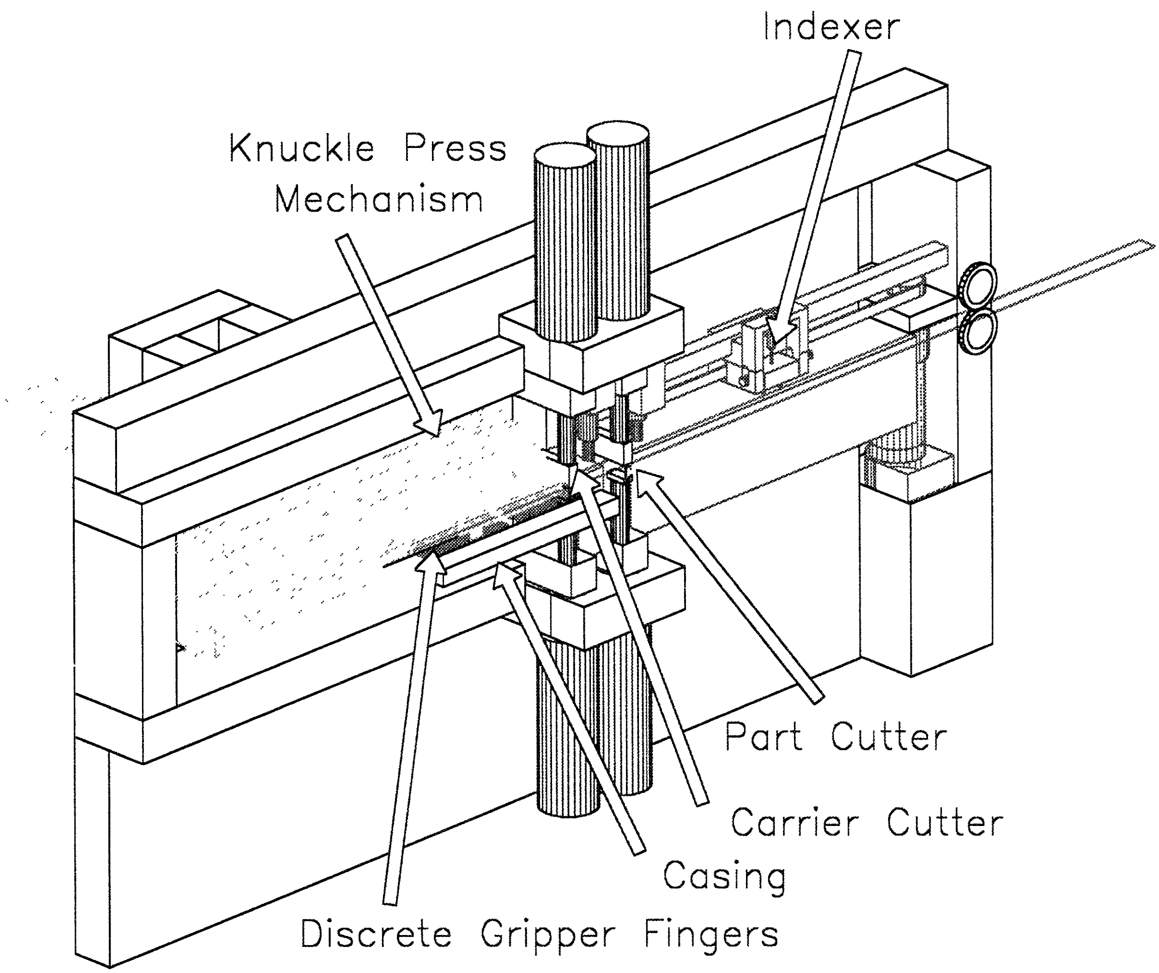

Figure 12: Mass Insertion Module

The mass insertion module is designed to insert a set of parts attached to a flat carrier strip. The motion of the gripper, indexer, and cutters is controlled through software. The part indexer begins the insertion process by removing the part carrier strip from the part storage wheel and positioning it correctly for the trimming operations. The carrier strip is indexed until the excess center parts are placed below the first part cutter. The cutter then shears off the unneeded parts and the reel is fed forward until the entire set of parts to be inserted are located between the gripper fingers. The grippers then clamp down and the carrier cutter shears the carrier off from the strip of parts wound about the wheel. The reel is then rewound to move the unused parts out of the way. Finally, the casing is brought forward until the parts are inserted into the casing. The gripper releases the set of parts and the cycle begins again.

2.6.1 Indexing Mechanism

The parts required for insertion are transported to the cutters and grippers by the part indexing mechanism (see Figure 13). The indexer is responsible for feeding and positioning the parts as they are removed from the part storage wheel. The parts are attached to a carrier strip, which contains small feeder holes used to grasp and manipulate the strip. The indexer has a carriage containing a set of conical cleats which engage the feeder holes and constrain the strip tightly. The distance between the conical cleats can be adjusted to accommodate the feeder hole pitch of any carrier strip. A linear actuator moves the carriage to wind the parts off of the part storage wheel. When the carriage has reached its maximum travel, a rubber stopper engages the carrier strip, and the cleats are removed. The carriage can then be transported back to the starting position, where the cleats are once again engaged, and the rubber stop is removed. The part indexer has the ability to flexibly wind and rewind the carrier strip through any specific displacement up to the length of travel of the carriage.

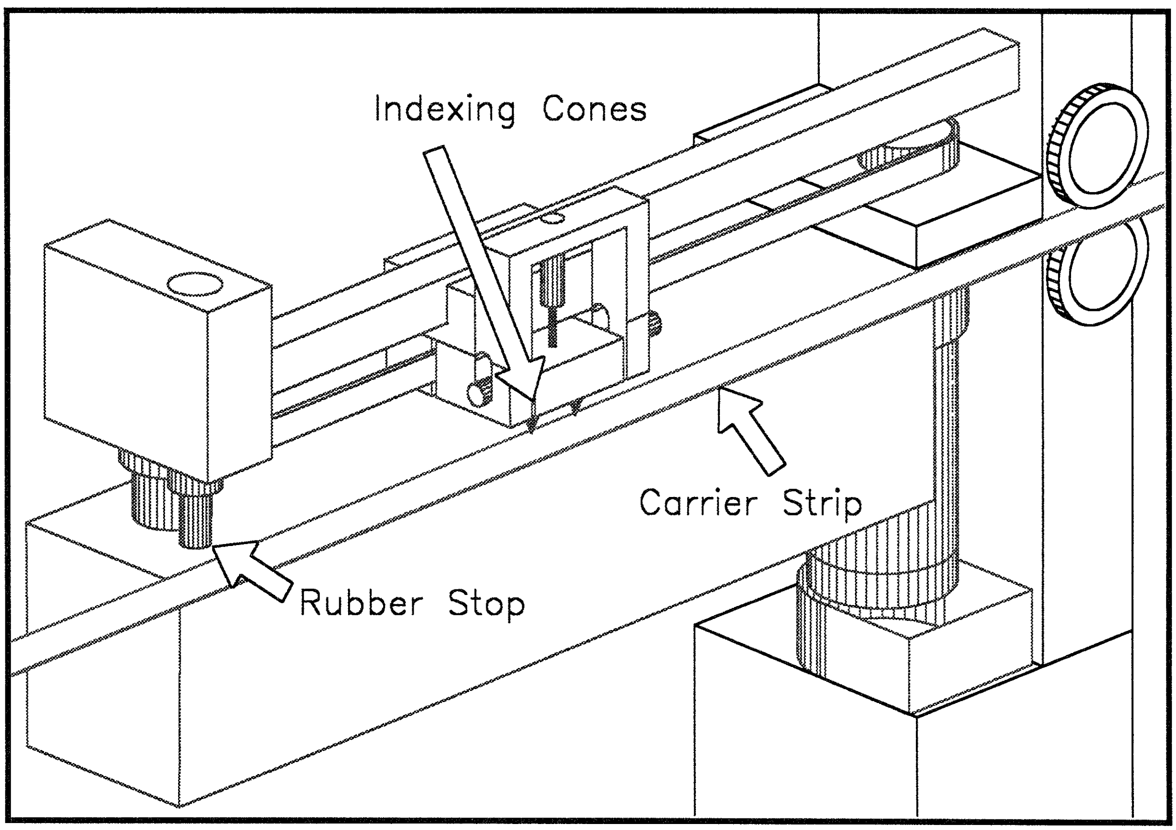

Figure 13: Part indexer.

Use of the adjustable indexing cones is a big improvement in terms of flexibility with respect to current part indexing mechanisms. Current machines use wheels with conical teeth to index parts. Spacing between teeth is fixed, and thus each set of wheels can only be used with carrier strips whose feeder holes have the same pitch.

2.6.2 Discrete Flexible Grippers

The parts are supported in discrete flexible grippers. A detailed illustration of the discrete flexible grippers is shown in Figure 14. The length of the grippers is adjusted by using varying thickness gripper blocks that can be combined together, much like in a press brake. The flexibility of the system is incorporated in the use of multiple thickness gripper blocks which can be combined to achieve the desired length. The limiting factor in the discrete block concept is the minimum size block that can be used. The overall dimensional length of the grippers can only be incremented by the thickness of the smallest block. Additionally, a minimum set of each size block must be available to insure that any length gripper can be created.

An advantage of the discrete gripper concept, other than its inherent flexibility, is the ability to define a set of grippers for a particular product and then store them when not in use. By having a reserve set of grippers, changeover time can be dramatically reduced by avoiding the assembly and measurement of the discrete gripper set. Once a product is assembled on the Flexible Assembly System, the grippers can be stored and reused in the future if another production run is required.

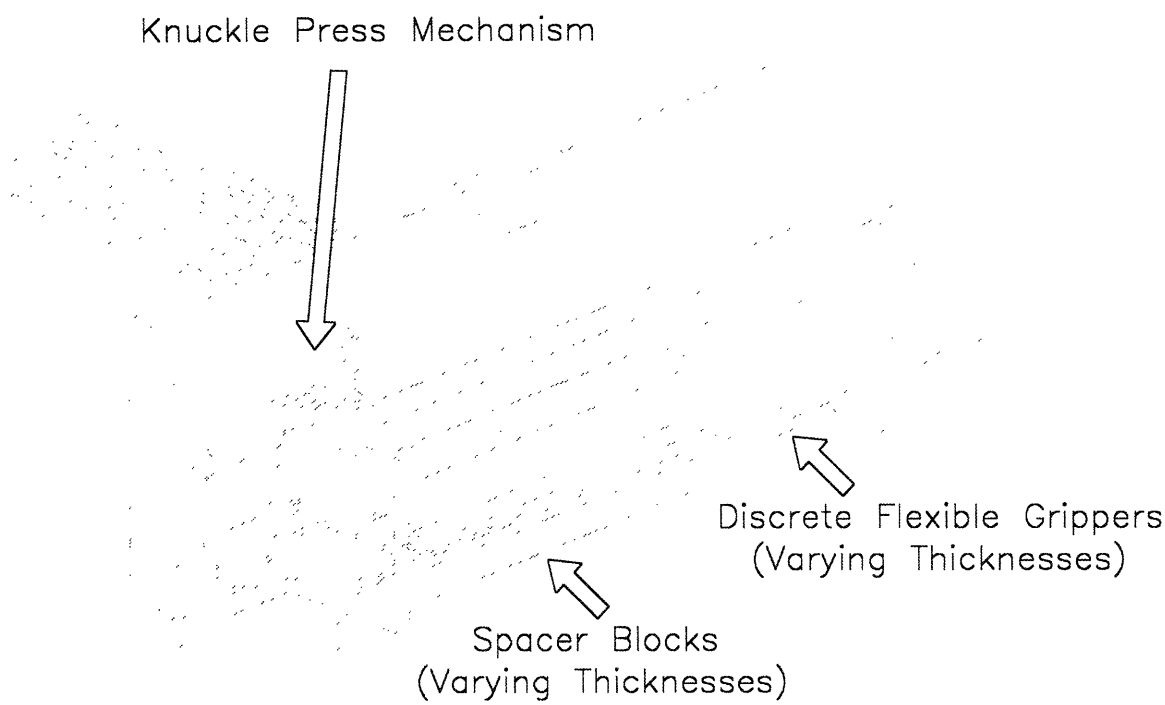

Figure 14: Knuckle press and discrete flexible grippers.

2.7 Bending Module

The bending module provides a means for the formation of complex bends often found in products. Using the coordinated motion of the module's actuator coupled with the motion of the casing transport and guidance system, complex part bends can be achieved.

The bending module consists of a set of gripper fingers that are mounted on a rotary actuator. The gripper rotates the parts about a fixed axis in space, through angles varying from 0° to 130°. The clamping force required to hold the parts is provided by a knuckle-press mechanism that uses a pneumatic cylinder coupled with mechanical advantage to provide a high clamping force. The overall simplicity of the bending module is derived from the fact that most of the required degrees of freedom have been consolidated within the casing transport system. By coupling the motion of the casing transport, with the rotary motion of the bending module, complex bends can be achieved. The bending module is displayed in Figure 15.

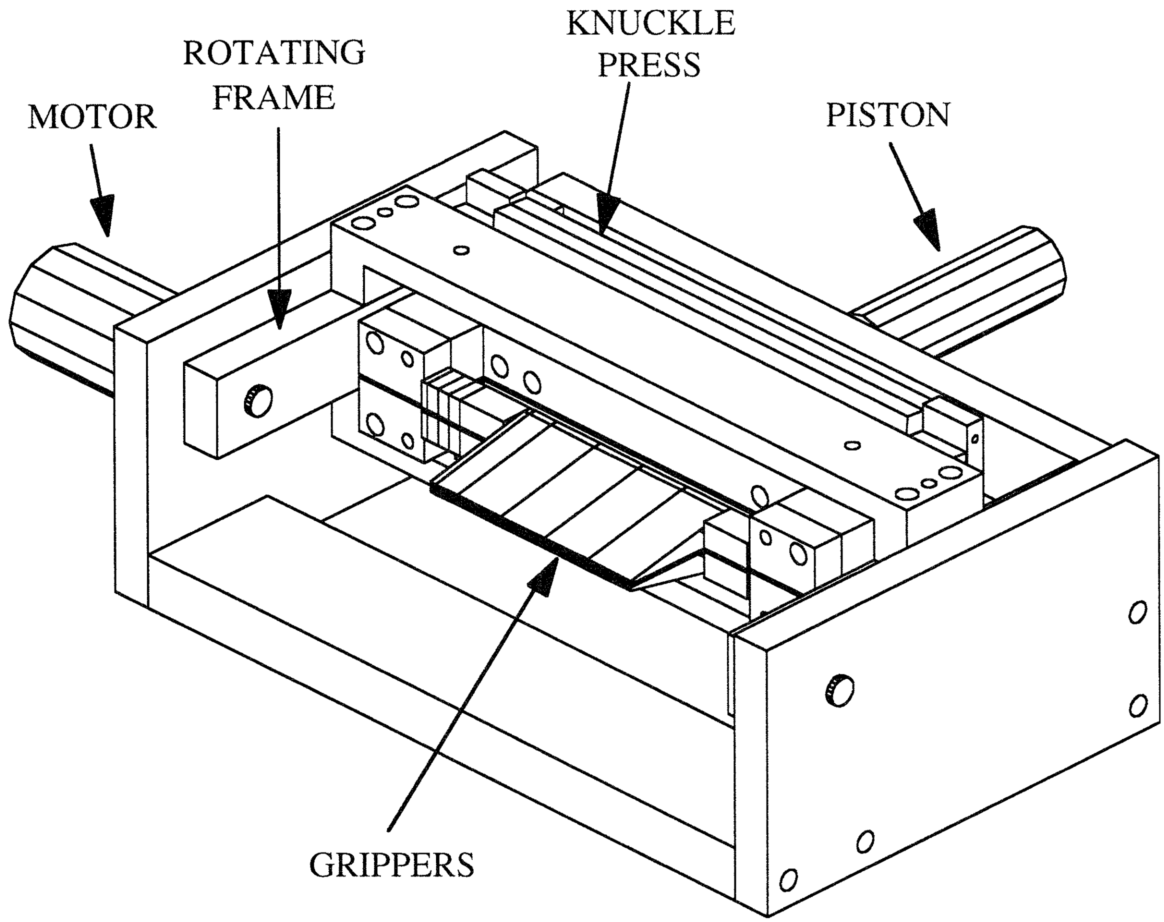

Figure 15: Bending module.

The grippers found on the Bending Module are similar to the ones found on the Mass Insertion Module (Section 2.6.2). The grippers can be adjusted in length by choosing various length gripper sub-units. In some cases, the grippers may require some type of part-specific surface in order to hold and affix the parts properly.

The bending process entails the coordinated motion of the Casing Transport System and the rotary actuator of the Bending Module. With combined motion from the two, four-degree of freedom bends can be achieved. The starting position of the bend can be regulated by the amount that the y-axis moves into the grippers. The rotary axis of the bending module can also be used in the removal of excess carrier strips. Additionally, the rotary actuator may be used as a multi-purpose tool for performing other secondary assembly operations and part manipulation.

2.8 Universal Trimming Module

The universal trimming module was added to the final system design to satisfy the need for secondary trimming operations that could not be performed with the other modules. Its primary use centers around the trimming of parts following the insertion operation. Various depth insertion procedures leave a wide variety of part lengths emerging from the casing. The trimming module can trim the parts down to an even length following the insertion operation. The trimming module was also designed to be flexible enough to trim an entire row of parts to a specified length, or to cut only a single part at a time.

The trimming module consists of a triangular shaped blade, and two adjustable support plates. The blade is mounted on a linear slide and moves in the vertical plane. A small gap between the two support plates allows the blade to penetrate without severely deforming the parts. The blade slides in-between the two support plates and shears the parts as they are supported evenly on both sides. The trimming module is depicted in Figure 16.

The triangular shaped blade incorporates flexibility into the system in that it can trim any given number of parts. The total number of parts sheared is controlled by limiting the amount of vertical travel of the guillotine-type blade. The overall length of the sheared parts can be controlled by manipulating the transport mechanism that grips the casing.

The trimming module may encounter problems when trimming parts that have unique geometries that do not fit well on the slat support plates. In this case, part-specific support plates may be required for proper trimming. Additionally, complex part geometries may require additional support during the trimming operation. The parts may need to be held from both top and bottom, while suspended upon the support plates. Casings may also require profile clips to be placed on the tips of the parts. This may require a different type of blade, or perhaps a rotary grinding motion to achieve the desired profile.

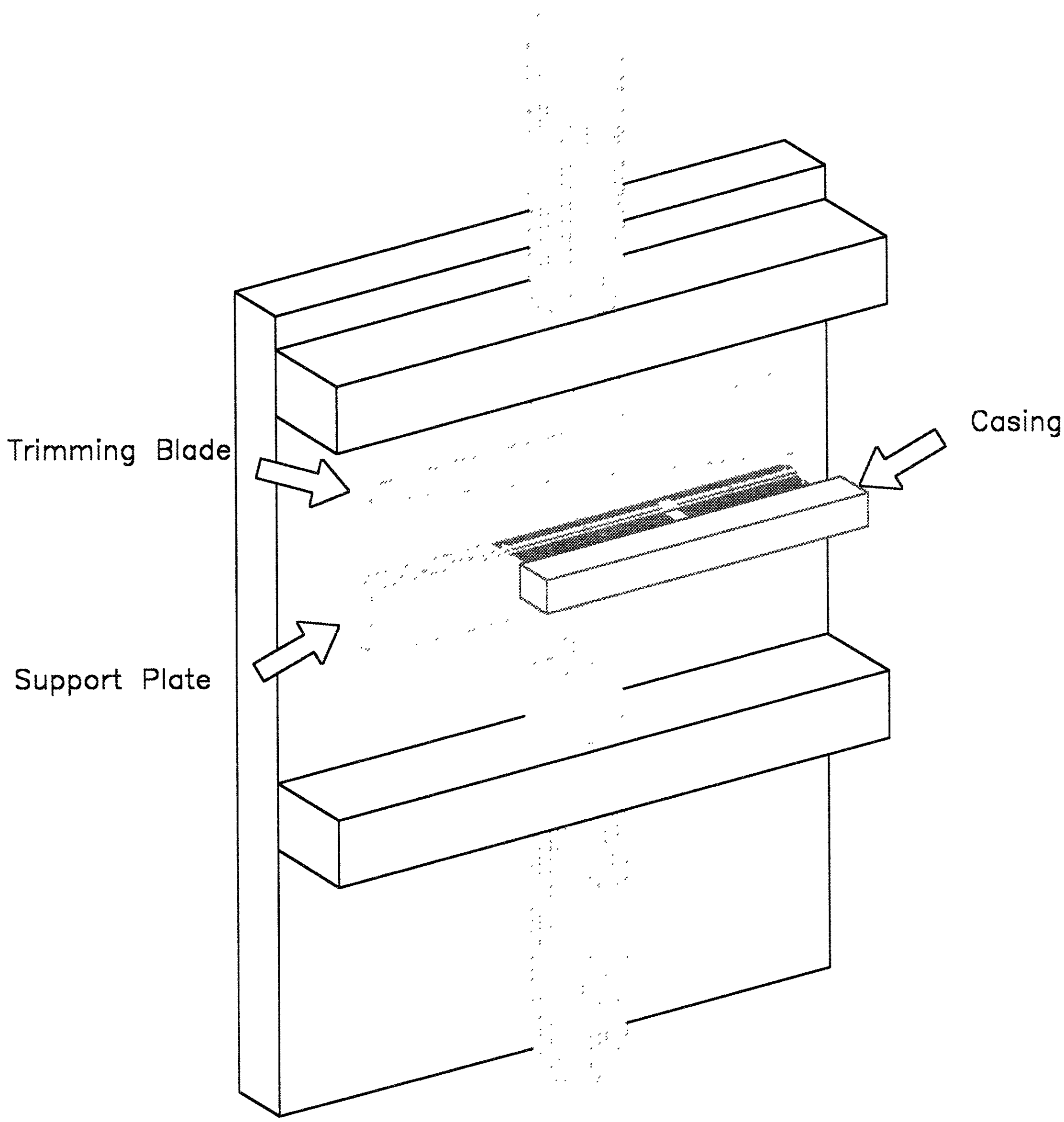

Figure 16: Universal trimming module.

2.9 Serial Insertion Module

The Serial Insertion Module is designed to insert and trim parts attached to a flat carrier strip one at a time. The parts are fed into the serial insertion module using a linear transport mechanism similar to the one used in the Mass Insertion Module (Section 2.6.1). The indexer transports the carrier strip until the first part is in the correct position. Then the gripper grasps the first part by its middle section and shears it from the carrier strip. Afterwards, the part is transported downward, where an insertion tube is used for the final insertion process. A vacuum pressure within the tube secures and constrains the part from moving. The holding tube is then transported forwards until the part is inserted. The cycle begins again as another part is indexed into place and the casing is re-positioned one notch by the transport module.

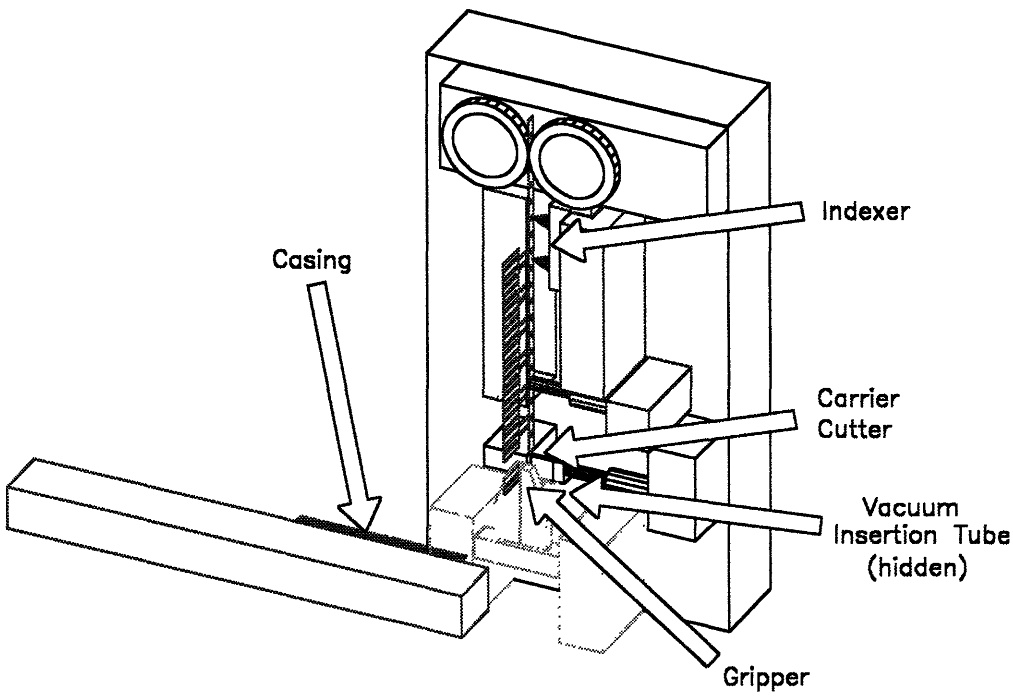

Figure 17: Universal Serial insertion Module

Chapter 3 - Bending Module Design

This chapter includes a description of the initial concepts for a bending module, selection of a concept for further development, and detailed design.

3.1 Initial concepts

Flexible pliers

The first step of the concept generation process consisted of brainstorming of the most flexible assembly mechanism the design group could think of. The purpose of this mental exercise was to develop a benchmark against which subsequent concepts could be measured.

After attempting to assemble a few products, it became clear that the most flexible solution would consist of a human operator equipped with a pair of pliers. Armed with a pair of pliers, the operator could separate a part from its carrier, insert the part into a casing, bend the part to its final shape, and trim it to its final length.

Although this approach would be extremely flexible, it would suffer from two major limitations: low rate and poor repeatability. These shortcomings could be overcome, at least in theory, by substituting the human operator with a computer controlled robot arm. Both the arm and the control system, however, would very likely have to be extremely complex (and thus costly) to be able to successfully mimic the capabilities of a human operator.

Wiping

A second concept that was considered is shown schematically in Figure 18.

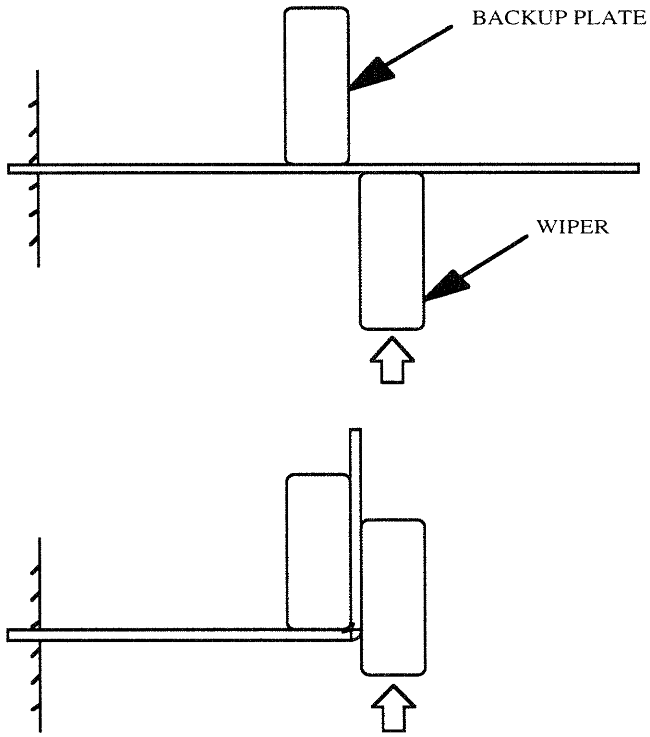

Figure 18: Part bent by wiping action.

This concept consists of a moving wiper plate and a stationary backup plate. The part is bent by being forced against the backup plate by the upward motion of the wiper. Parts of different thickness could be bent using this method by adjusting the size of the horizontal gap between the wiper and the backup plate. Multiple bends could be made by adding a second wiper and backup plates.

Feasibility tests performed on sample parts showed that this concept suffered from two major drawbacks. First, the wiper could easily damage the surface of the parts. Second, the gap between the wiper and the backup plate would have to be minimized to reduce springback. Since friction between the wiper and the part increases as the size of the gap decreases, reducing the gap would aggravate damage to part plating.

Bending + Support Plate

Consideration of the wiping concept showed that rubbing the parts against a hard surface should be avoided. Thus a third concept, shown in Figure 19, was developed.

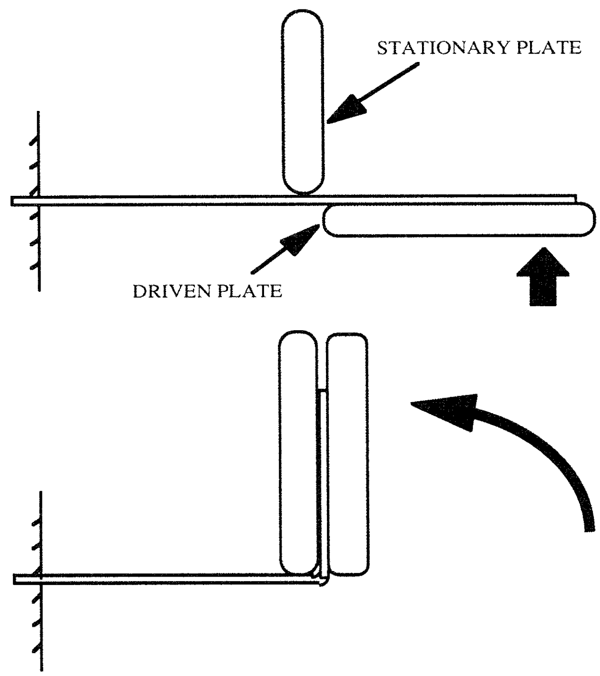

Figure 19: Part bent using a bending and a support plate.

The part is bent by rotating the driven plate until the part touches the stationary backup plate. Once again flexibility could be attained by adjusting the gap between the driven and the stationary plates. Multiple bends could be made by alternating which plate is driven and which plate is stationary.

Use of this concept would minimize damage to the surface of the parts; however, the concept may require a complex controlling system to be able to position and displace the bending and support plates independently.

Rotating Gripper

The rotating gripper concept, shown in Figure 20, is a simplified version of the flexible pliers concept. To bend a part, the gripper grabs the part at the right distance from the housing. It then rotates about an axis located halfway between an imaginary line connecting the tips of the upper and lower grippers. Flexibility may be attained by varying the gap between the upper and lower grippers, and by grabbing the part at different distances from the housing. A possible disadvantage may be that the grippers will not fit between tightly spaced rows.

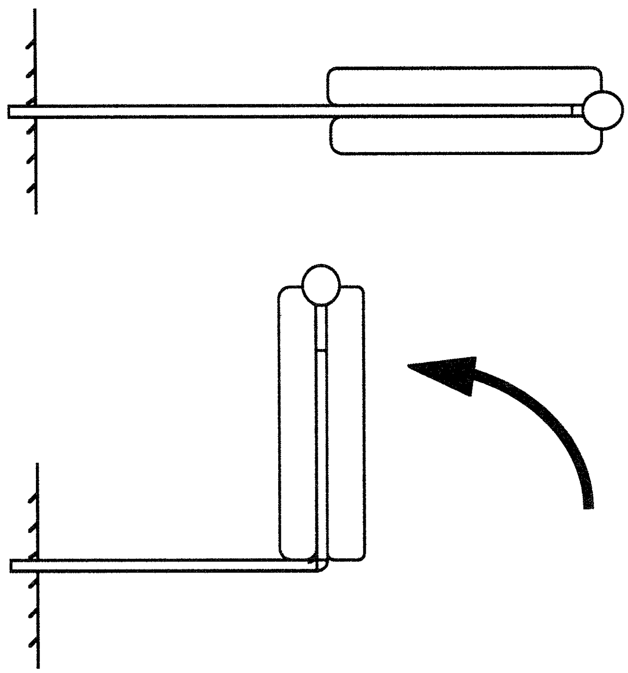

Figure 20: Rotating gripper concept.

Rotation about a Remote Center

This concept is another variation of the flexible pliers concept. By grabbing the part only by its tip, and placing a backup plate at the appropriate position, multiple rows could be bent more easily than by grabbing a longer segment of the part. In some cases, the casing itself could replace the backup plate. The concept is illustrated in Figure 21.

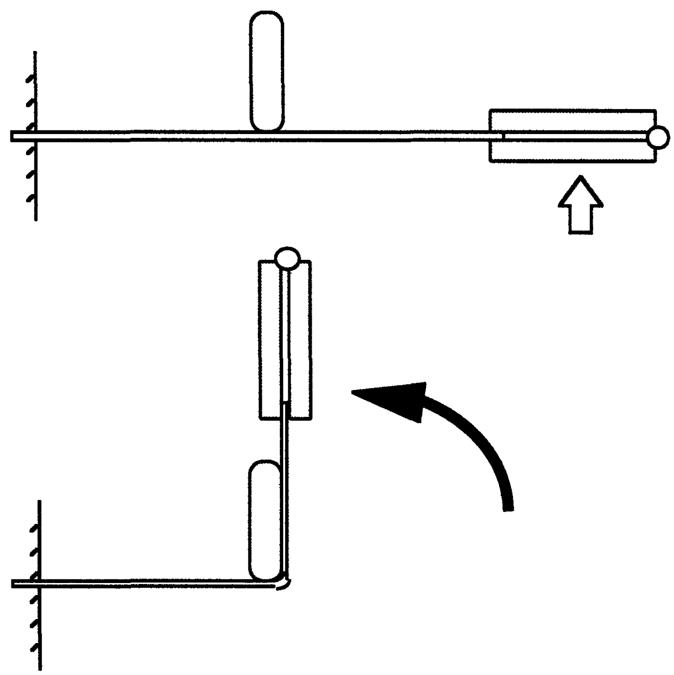

Figure 21: Part bent through remote-center rotation.

Two-step forming

The two-step bending concept, displayed in Figure 22, uses the combined motion of two driven plates to bend the part against a backup plate. One of the driven plates moves vertically, bending the part just enough so that the second driven plate, moving horizontally, can push the part against the backup plate.

This method would exert smaller frictional forces on the part than the wiping concept, thus reducing the risk of damaging the surface of the part. On the other hand, it would require a more complex actuation mechanism, since two plates need to be actuated independently.

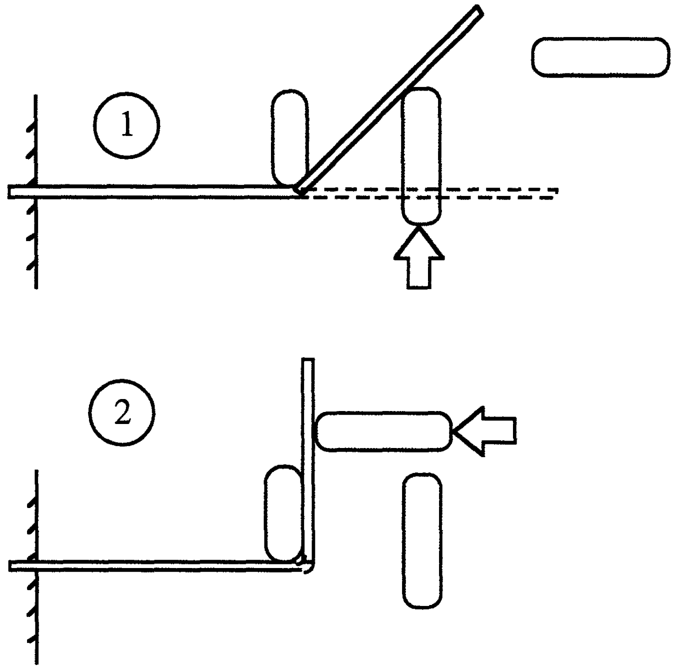

Figure 22: Part bent in two steps.

Die-forming

Perhaps the least flexible of all the concepts, the die-forming concept would require making matching die sets shaped so that the part is bent to its final shape when the driven die presses the part against the backup die (see Figure 23). As in the wiping concept, it is very likely that the surface of the part would be damaged.

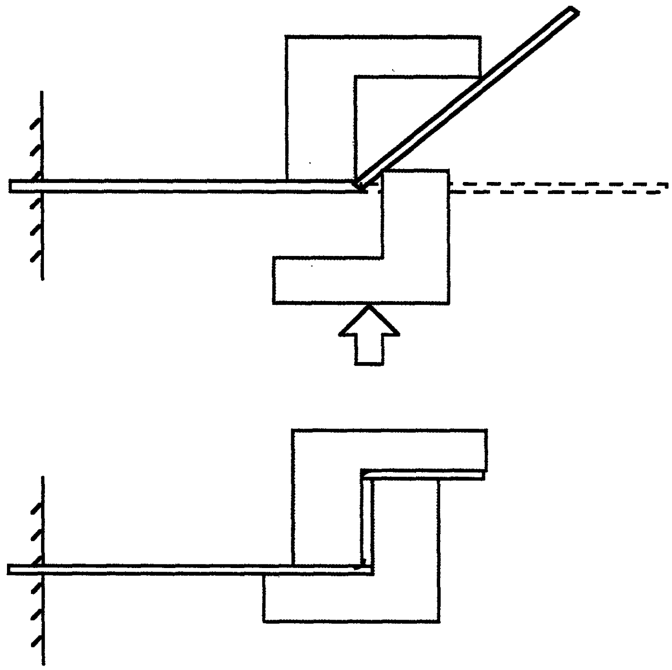

Figure 23: Part bent through the use of dies.

3.2 Concept Selection

Once the initial concepts were generated, it was necessary to select one concept for further development. The selection criteria used at this stage included flexibility, friendliness to parts, and perceived complexity. The Pugh concept selection method [4] was used to identify the better concept.

The wiping concept was used as a benchmark because it seemed to be one of the easiest concepts to implement. The remaining concepts were judged using a qualitative ranking system. In this system, a plus sign (+) means that the concept being judged is superior to the benchmark, an equal sign (=) means that it is just as good, and a minus sign (-) means that it is inferior. The purpose of using qualitative rather than quantitative rankings was to maintain the boundaries of the design space[5] relatively fuzzy so that strong features of the seemingly 'inferior' concepts would not be discarded and lost.

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[5] See Appendix B for an explanation of what is meant by design space.

The Pugh concept selection matrix, shown in Figure 24, reveals that both the rotating gripper and the remote-center rotation concepts are on the whole superior to the other concepts. Furthermore, comparison of the two concepts revealed that the rotating gripper concept could be easily modified to work like the remote-center rotation concept. Thus, both concepts were combined into a single concept for further development.

Criteria	**Wiping**	Bending + support plate	Rotating gripper	Remote-center rotation	Two-step forming	Die-forming
Flexibility		+	+	+	+	-
Damage to parts		+	+	+	+	-
Complexity		-	=	=	-	=

Figure 24: Pugh concept selection matrix.

3.3 Design Guidelines and Specifications

After the initial round of concept generation, it became necessary to prepare a set of design guidelines to guide the next step of the development process. The design guidelines for part bending are described below.

Maximize flexibility. Flexibility was defined as the ability to bend (or detach) multiple types of parts without having to go through a complicated setup process. Consequently, the goal of maximizing flexibility was broken into two more specific goals:

- Maximize the number of different part types the module is capable of handling.
- Minimize the number of setup operations required for product changeover.

Minimize damage to parts. The surface of the parts to be assembled can be easily damaged; thus, particular attention had to be paid to developing a 'part-friendly' bending mechanism.

Minimize manufacturing costs. This guideline includes designing for manufacture and designing for assembly.

Design for manufacture. For the purposes of this project, this requirement was meant to encompass the following list of guidelines:

- Use standard parts
- Use commonly available materials
- Use materials that can be machined easily
- Design symmetrical parts
- Avoid using odd dimensions
- Avoid specifying tolerances that are difficult to achieve

Design for assembly. There were two reasons for stressing the importance of designing the module so that it could be put together easily. First, it would reduce the chances of malfunction due to improperly assembled parts. Second, it would make maintenance and repair a less time consuming process (operators in many manufacturing plants spend a considerable amount of time taking apart and then putting together machines that have broken down).

One way of designing for assembly is to constantly optimize the complexity of parts to number of parts ratio. It is usually the case that fewer parts result in easier assembly. On the other hand, it is difficult to achieve part count reduction without increasing part complexity and cost.

Minimize Operating Costs

This guideline includes two components: minimization of maintenance requirements, and maximization of the assembly rate of the module.

Minimize maintenance requirements. This requirement may be met by using parts that require low maintenance, designing the machine so that every part is easily accessible, avoiding the use of parts susceptible to wear, and making the machine robust to failure (jamming of parts, dirty environment)

Maximize assembly rate. The objective associated with this requirement is to maximize the rate of the bending cycle so that the overall assembly rate is less than or equal to one product per minute. While it was agreed that a faster is better, it was also agreed that a total cycle time of two seconds per 90 degree bend would be a realistic goal to aim for.

Design Specifications for Flexible Bending

The design specifications for flexible bending may be summarized as follows:

- The module must be capable of bending an arbitrary number of parts, given a maximum casing length of seven inches.
- Single bends: the module must be capable of achieving bends up to 90 degrees in each direction (clockwise and counterclockwise).
- Multiple bends: the module must be capable of performing up to two 90 degree bends per part.
- The resolution of the module must result in angular displacements as small as 0.1 degrees. The tolerance on part tip position should not exceed +- 0.005 inches.
- The module must bend parts at a rate of one 90 degree bend every two seconds.

The types of bends that the module must be capable of making are displayed in Figure 25.

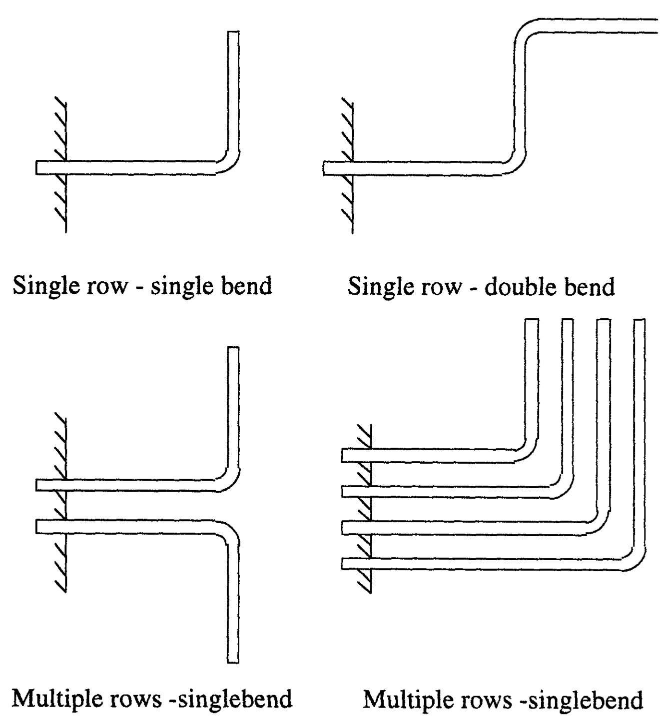

Figure 25: Common types of bends.

3.4 Detailed Design

Detailed design of the bending module was accomplished in eight rounds, each having a higher level of detail than its predecessor. Most of the parts were originally sketched on paper, and then drawn in three dimensions using AutoCad release 12. Description of what was accomplished in each round follows below.

Round 1 - Physical embodiment

Once the rotating gripper concept had been selected for further development, a few sketches were made illustrating how the concept could be implemented in practice. One of such sketches is shown in Figure 26.

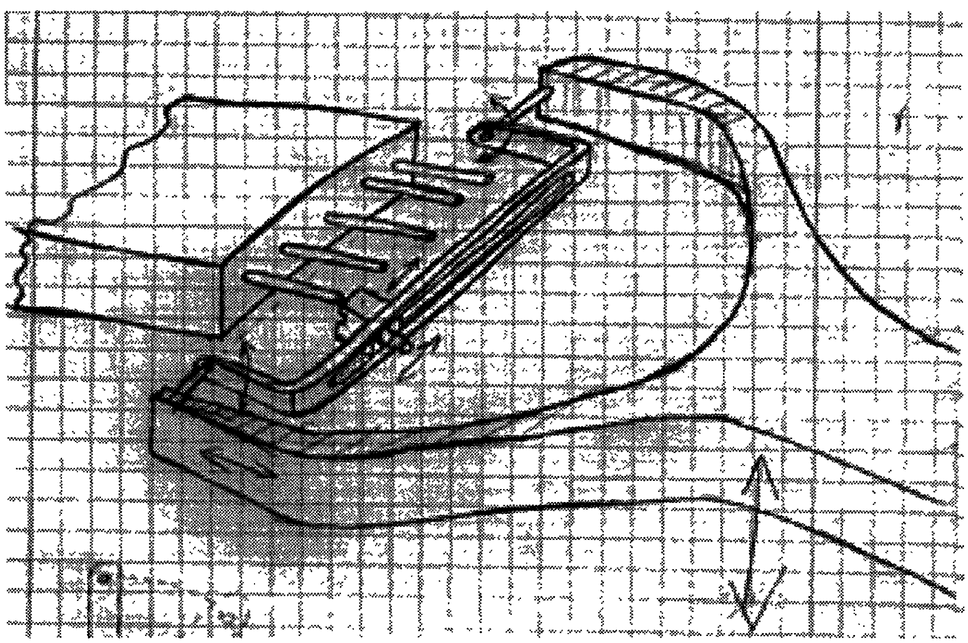

Figure 26: Pencil sketch of part bending concept.

The sketch shows a gripper mounted on a rotating c-shaped frame. The gripper shown in the sketch would grab one part at a time. To bend a row of parts, the gripper would move parallel to the mating face of the casing.

The low level of detail reveals some of the issues that were being considered at that stage. For instance, the absence of any type of actuator shows that only the degree of freedom requirements had been defined. Also, the size of the gripper shows that the issue of whether bending one part at a time was better than bending multiple parts was being debated.

At the same time that bending concepts were being developed, the total system concept described in Chapter 2 was being defined. Once it was decided that most of the degrees of freedom would be consolidated on the transport module, the idea of using a moving single part gripper was discarded. Instead, it was proposed that grippers of different widths would be mounted on a common rotating frame to accommodate different product lengths (see Figure 27).

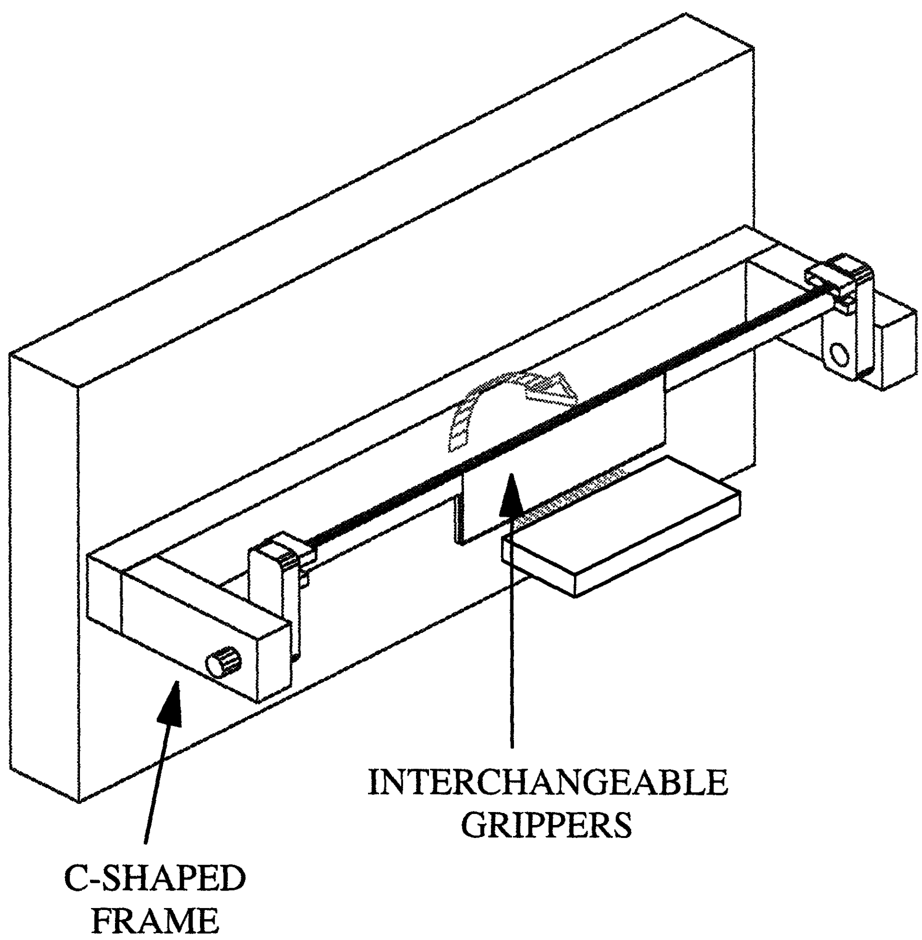

Figure 27: First round concept.

As the figure reveals, it was still not clear how the grippers would be attached to the frame, nor was it clear how the gap between the upper and lower gripper would be adjusted.

Round 2 - Discrete grippers introduced

The issue addressed in this round was gripper design. The engineering specification asked for a gripper that could bend parts on products up to seven inches long. A minimum length was not specified; however, it was understood that flexibility could be increased by designing a gripper that could bend one part at a time if necessary.

Two solutions were considered. The first consisted of designing very simple grippers, custom-sized for each product length. Such grippers would be very easy to make, and would share a common mounting mechanism so that they could be easily attached to the rotating frame. The second solution consisted of using discrete gripper blocks, i.e. grippers with the same cross section but different lengths, added together to obtain the required total length.

At this stage it became necessary to define the flexibility requirement better so that an informed decision could be made. The question that had to be answered was how often the product width will be changed. In some manufacturing environments, machines are expected to be able to switch between as many as three different product lengths per day. Such product changeover frequency made it clear that the most flexible solution, i.e. discrete gripper blocks, had to be developed further.

Once the decision to develop the discrete grippers had been made, a preliminary concept was created (see Figure 28). The figure shows gripper and spacer blocks mounted on two rods. It was hoped that accurate mounting of the gripper blocks would be achieved by using a locational fit.

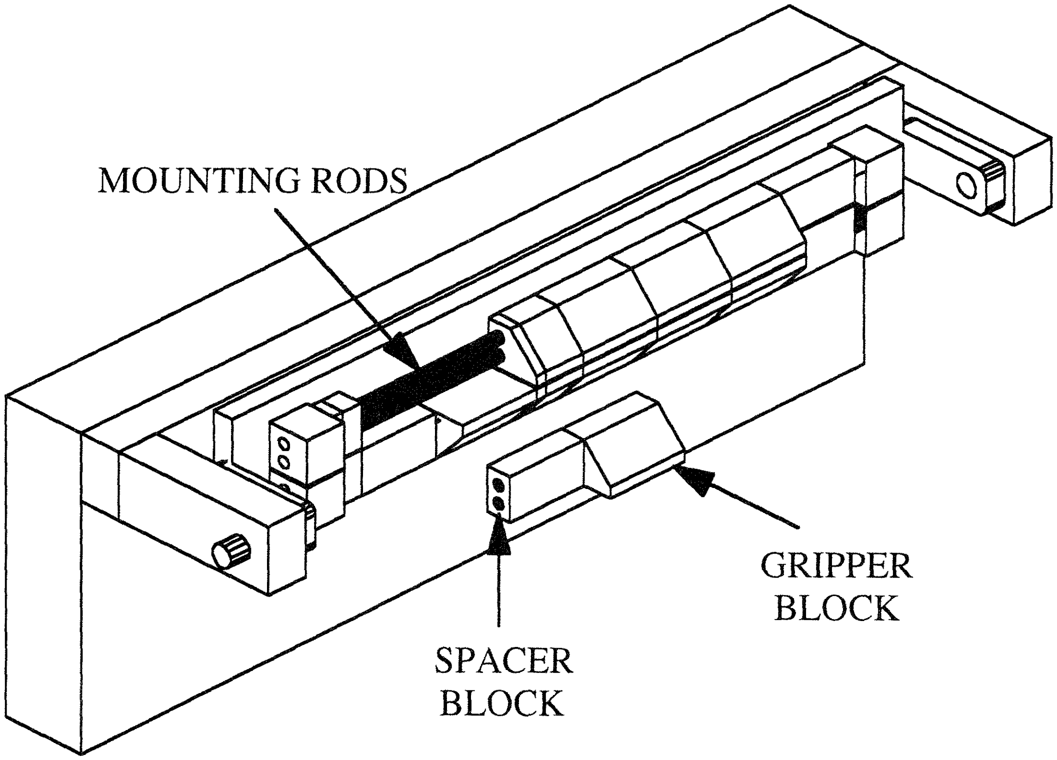

Figure 28: Discrete grippers concept.

Round 3 - Basic design completed

The main elements of the bending module were defined during this round. They included grippers mounted on linear carriages, and a knuckle-press and pneumatic piston combination to actuate the grippers (see Figure 29).

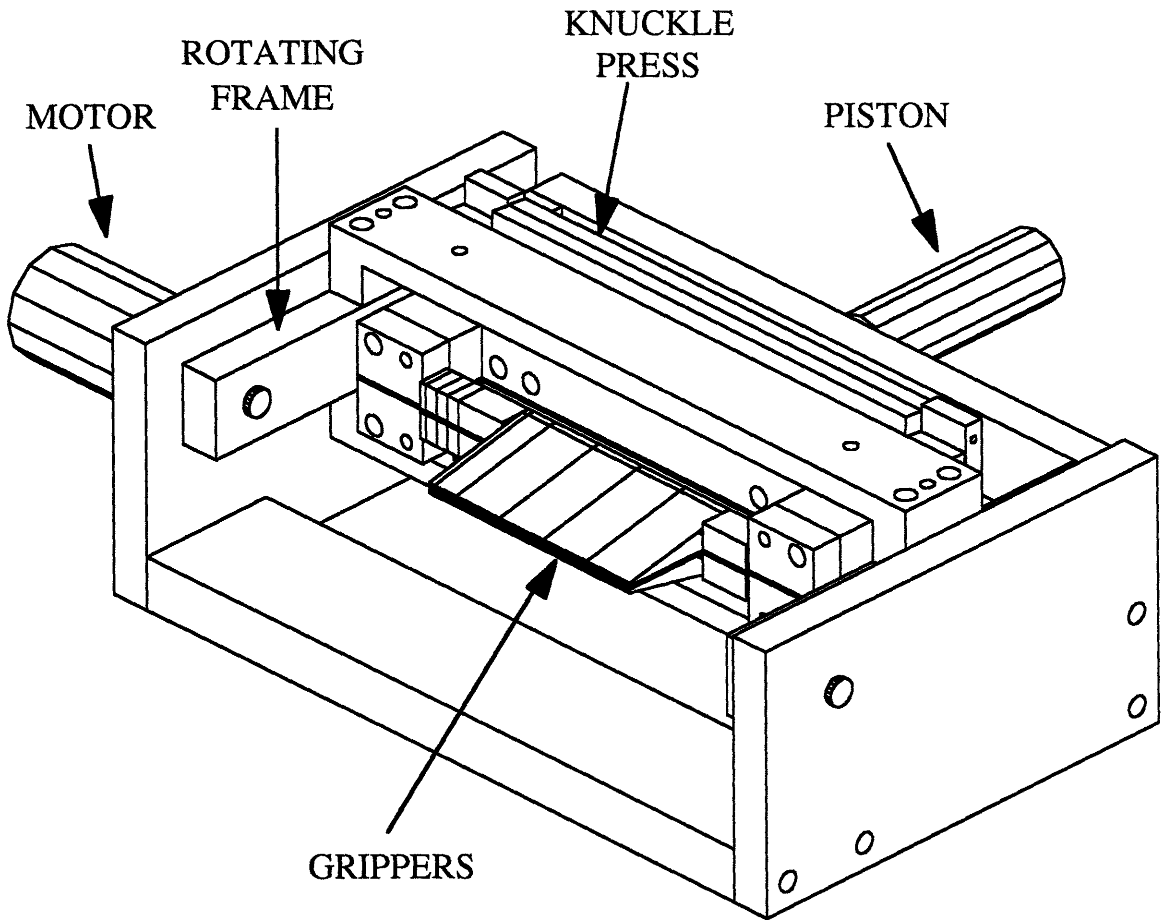

Figure 29: Third round bending module.

Grippers

During this round the problem of how to quickly add or remove gripper blocks was addressed. As shown in Figure 30, the rods would be supported at both ends by a support block. The support block would be mounted on a c-shaped frame, via a dowel pin and a cap screw.

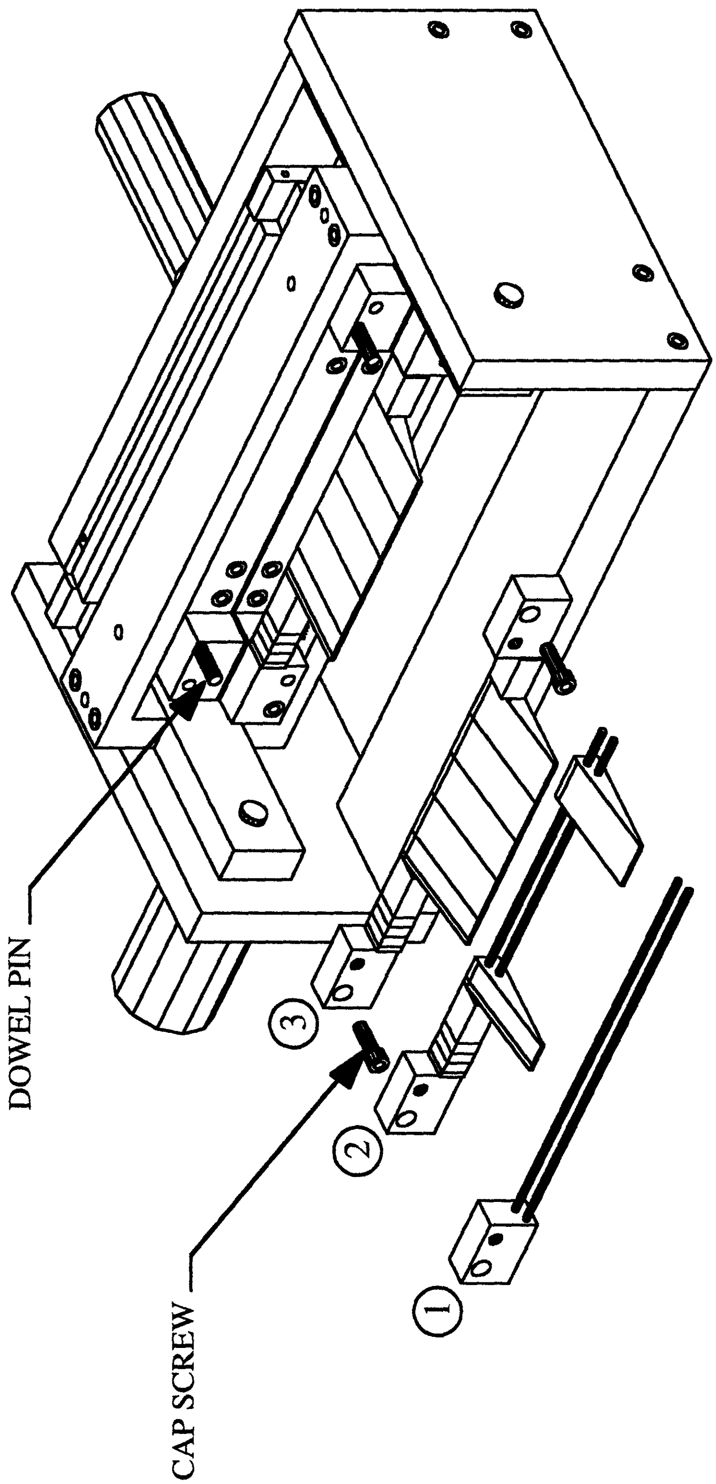

Figure 30: Concept for mounting of grippers.

Clamping Mechanism

A second issue addressed during this round was the mechanism by which the grippers would open and close. The requirements for gripper motion were: a) that the gap between the upper and lower grippers be easily adjustable, b) that the surfaces of the upper and lower gripper be parallel to each other and to the parts for any size gaps, c) that the grippers clamped down symmetrically about the plane containing the center axis of the parts, and d) that the grippers could support the necessary force to grip and bend a seven inch long row of parts.

The issue of parallelism was addressed by attaching the grippers to carriages riding on linear bearings, as shown in Figure 31.

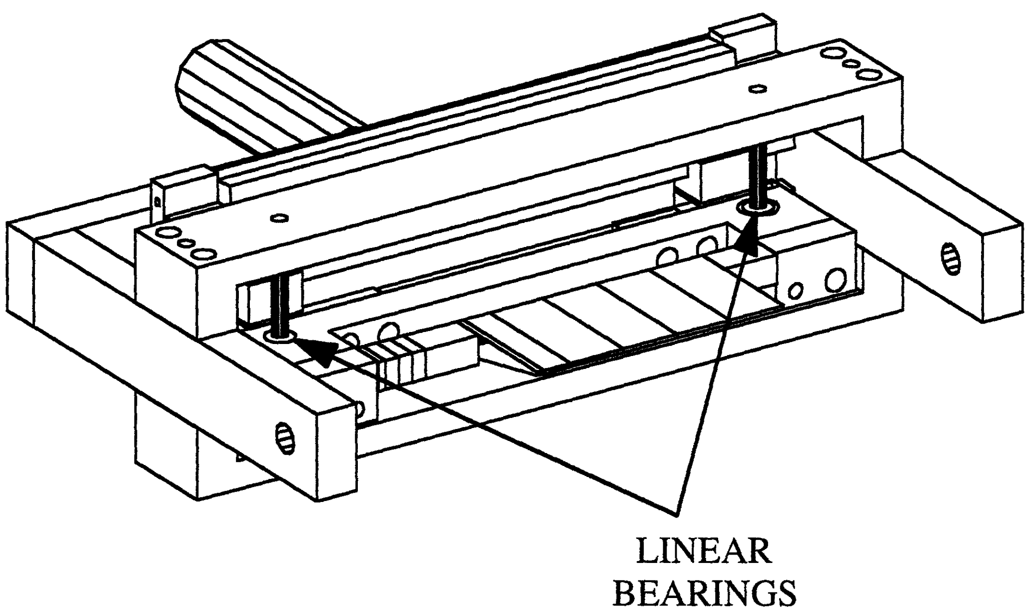

Figure 31: Gripper carriage mounted on linear bearings.

The linear bearing rails are press fit into two u-shaped beams, which are part of the main frame.

In addition to the u-shaped beams, the main frame is comprised of two side beams bolted onto a back-beam. A shaft is press fit into each side beam. Each shaft slides into bearings that are part of a support frame. One of the shafts is attached to an electric motor that supplies the torque needed to rotate the main frame.

Once it had been decided that the grippers were going to be mounted on linear carriages, the next step consisted of designing the actuation mechanism. Since the requirements for gripping the parts for bending are similar to the requirements for the insertion operation, it was decided that the bending and insertion modules would share the knuckle-press mechanism shown in Figure 32 and Figure 33. The knuckle-press met the requirements for easy gap adjustment, axi-symmetrical motion, and gripping force provision. The pneumatic piston supplying the gripping force was mounted on the main frame.

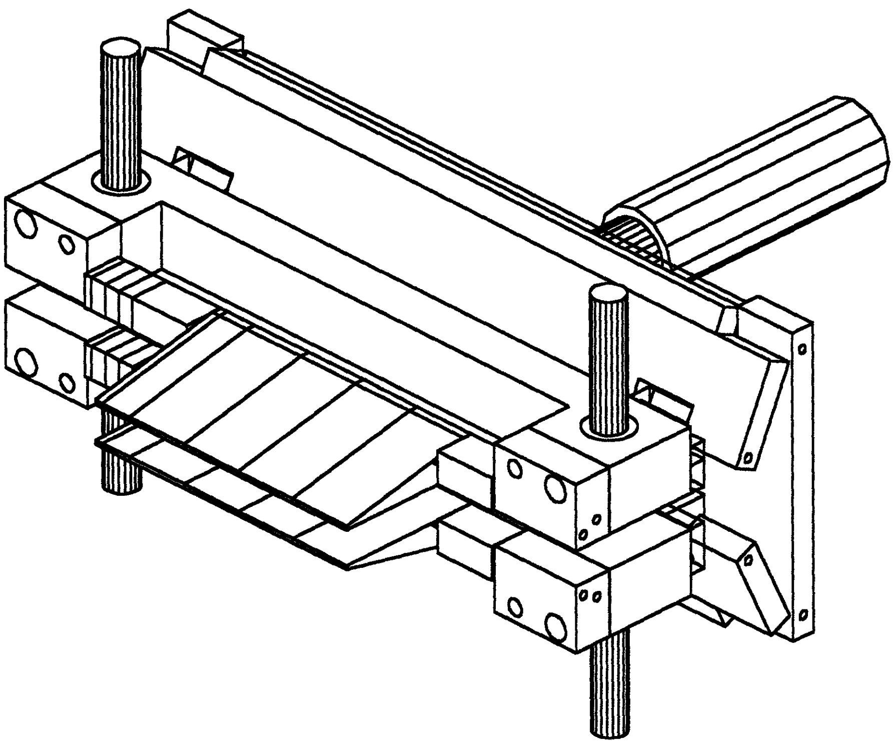

Figure 32: Isometric view of knuckle press and gripper assembly.

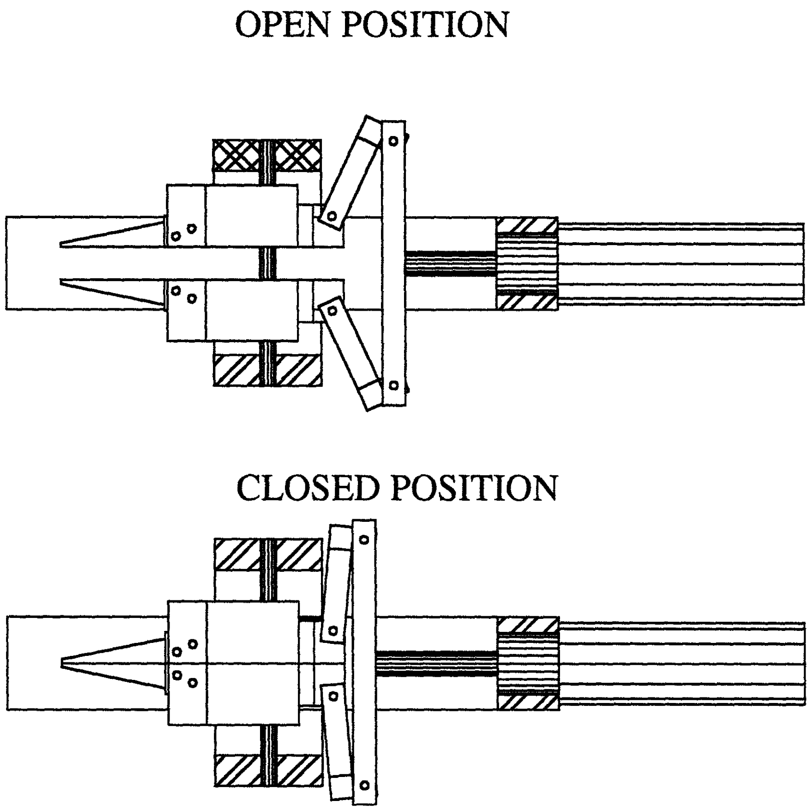

Figure 33: Side view of knuckle-press and gripper assembly.

Round 4 - Concept evaluation and improvement

The completion of the third round marked the end of the conceptual design phase and the beginning of the detailed design phase. Up until the end of the third round the shape and dimensions of all the parts had been chosen based on intuition. It was now time to design the parts that were going to be machined, and to select the parts that were going to be purchased.

Before jumping into the detailed design phase, however, it was necessary to review the conceptual design carefully to make sure that it met all the requirements. Evaluation of the third round design resulted in four major changes. First, the actuation mechanism for the grippers was simplified. Second, the mounting of the grippers was modified to make it easier to manufacture. Third, the shape of the grippers was defined. Fourth, counterweights were added to the main frame to reduce torque requirements on the motor, and to make the system easier to control. The improved concept is shown in Figure 34.

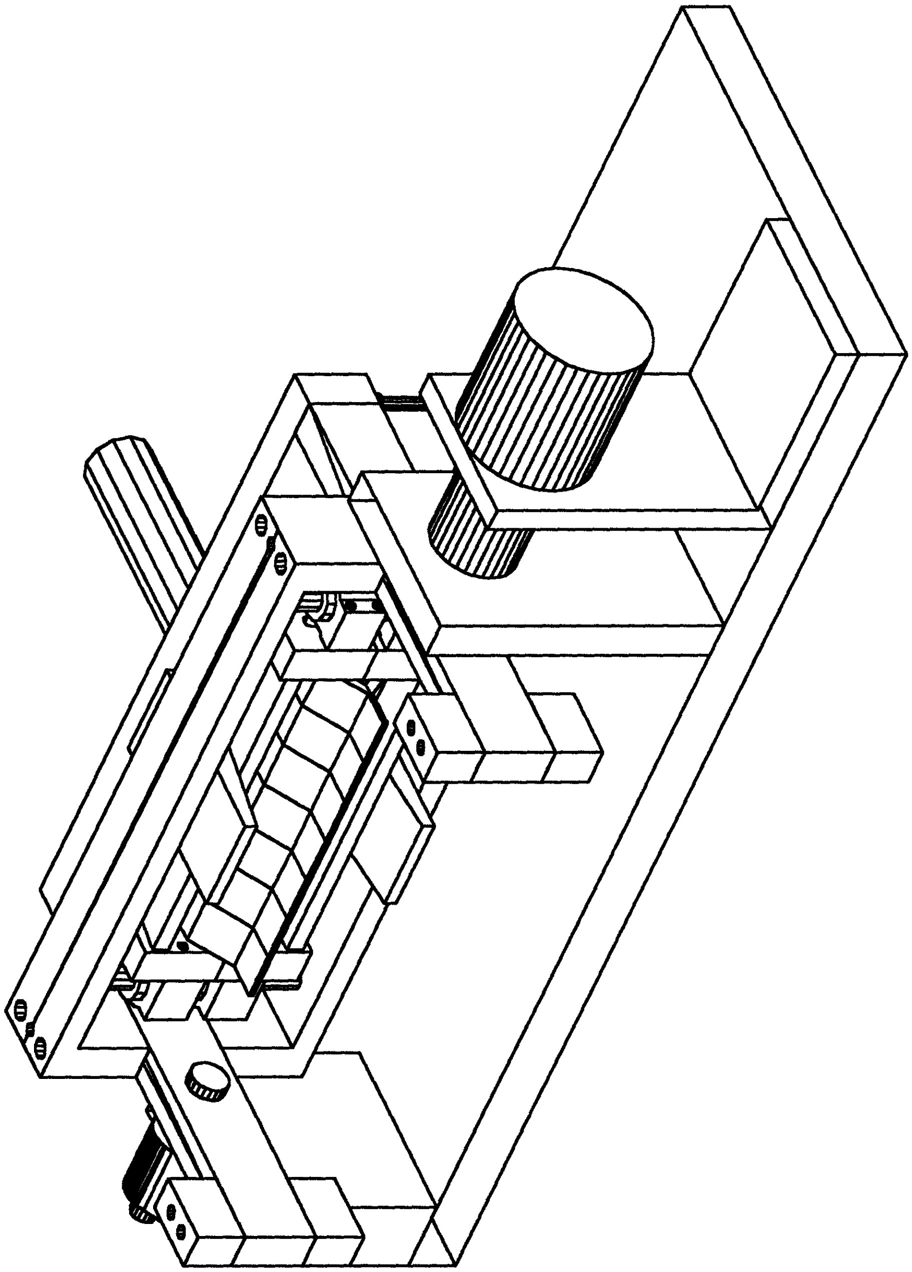

Figure 34: Bending module at the end of round 4.

Clamping Mechanism

The first element to be reviewed was the knuckle-press. A closer look at what was required to make it showed that the knuckle-press could easily become the most complex part of the entire module, because of the number of parts involved. Since the grippers had to be actuated approximately sixty times per minute, it was clear that careful attention had to be paid to the design of the hinges connecting the different links of the press. Poorly designed hinges would wear out quickly. A proposed design for the hinges consisted of flanged brass bushings press fit into the linkages, as shown in Figure 35.

At this point in the review it became clear that the knuckle-press was not a very good solution because of three reasons. First, preliminary tests showed that high gripping forces are not needed for bending. Second, the large number of parts of the knuckle press translates into too many potential failure points. Third, manufacture of the links would also present problems. Drilling the holes for the brass bushings so that their axes were collinear seemed particularly difficult.

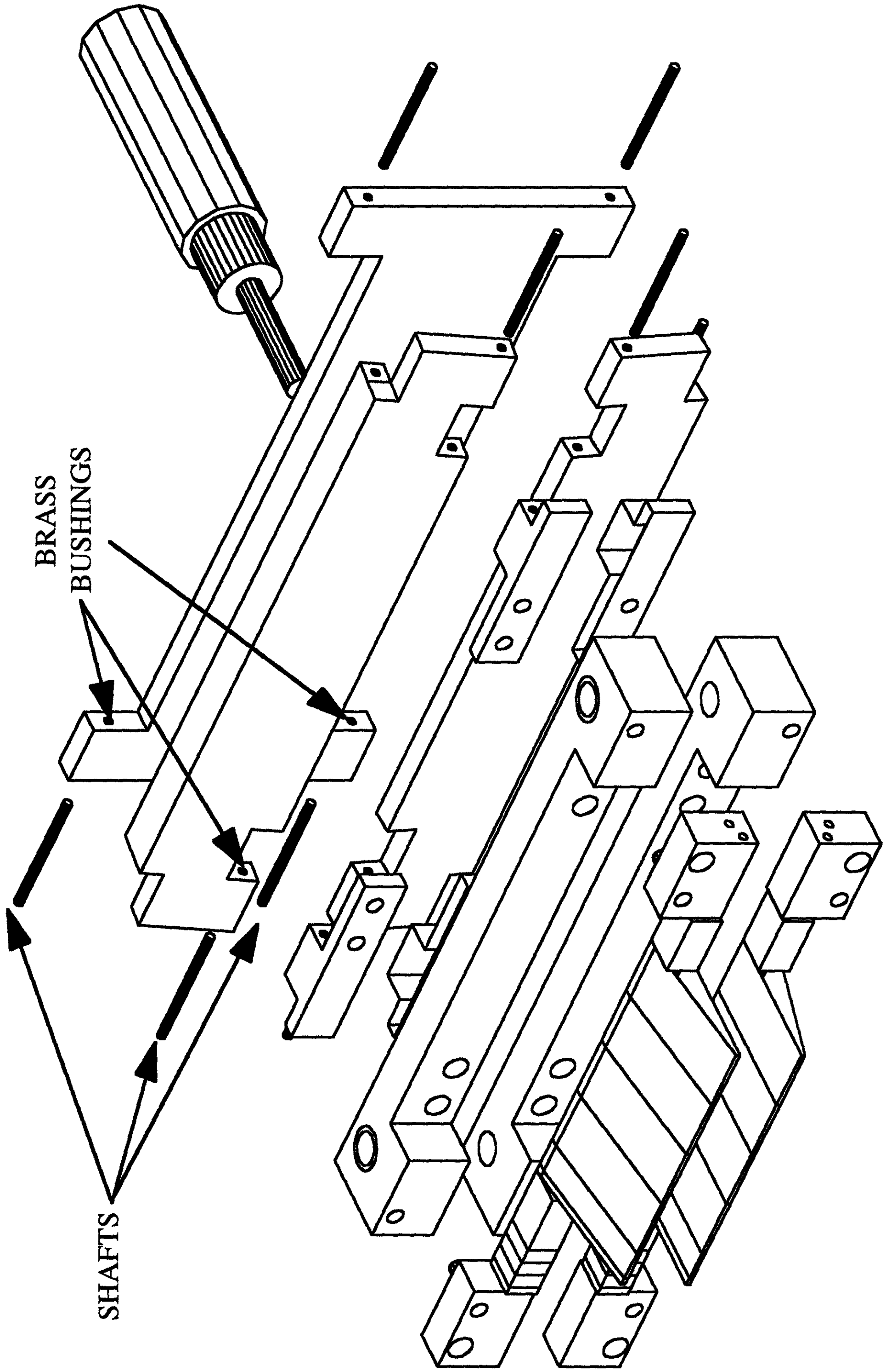

Figure 35: Exploded view of knuckle-press.

An alternative to the knuckle-press mechanism was proposed at this point. As shown in Figure 37 and Figure 36, the knuckle-press is replaced by a wedge-shaped clamp. The clamp is attached to the pneumatic piston. As the piston moves forward, the carriages on which the grippers are mounted are wedged in by the clamp. Springs placed between the upper and lower gripper mounting blocks provide the force needed to open the grippers when the piston is retracted.

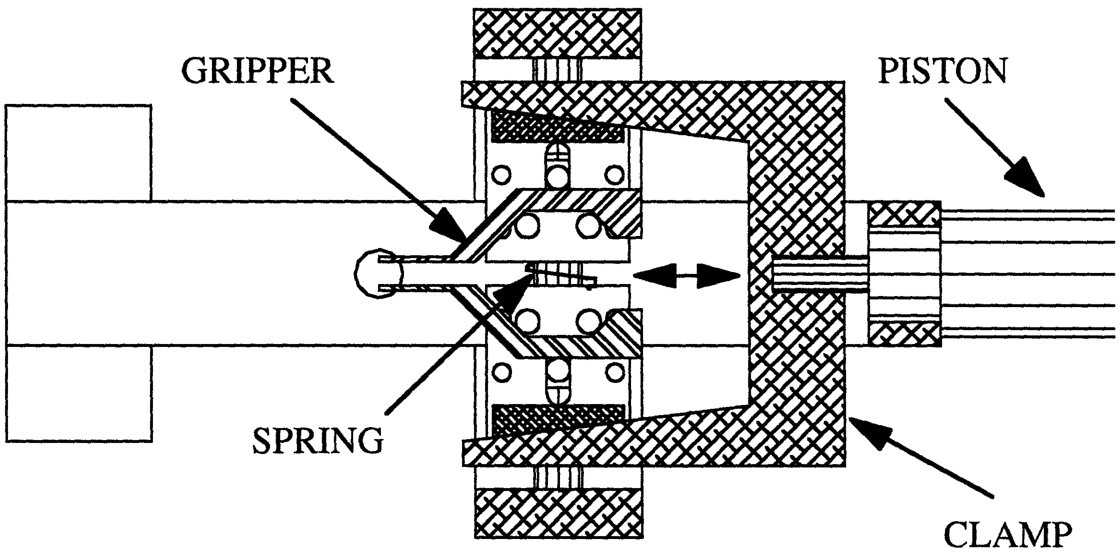

Figure 36: Clamping mechanism - open position.

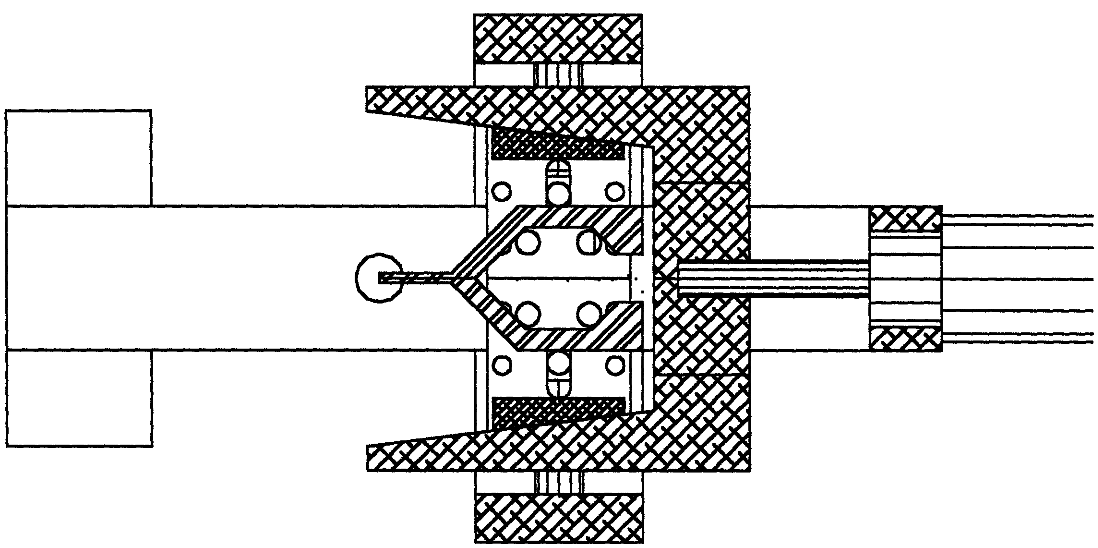

Figure 37: Clamping mechanism - closed position.

A second way of using the wedge-shaped clamp is shown in Figure 38.

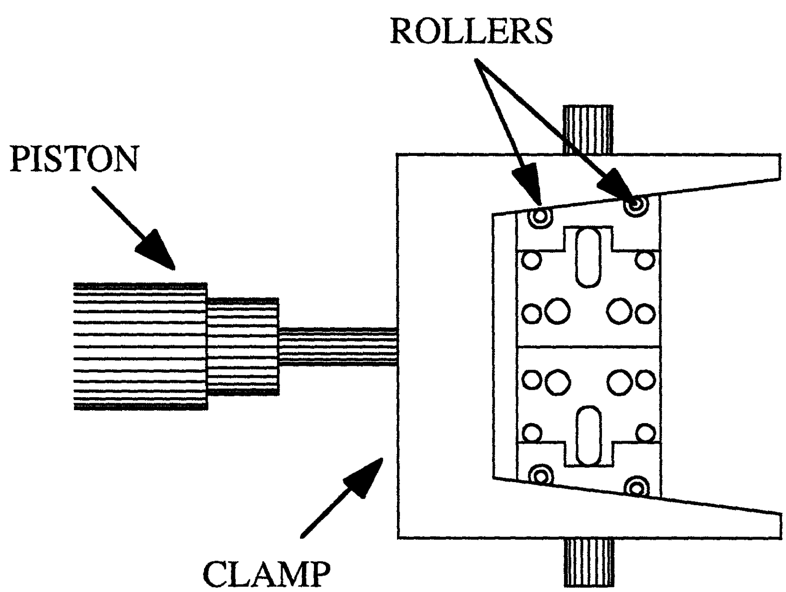

Figure 38: Alternative wedge clamping mechanism.

Here the clamp rides on four rollers, made of rods supported by brass bushings. Since the rods turn as the clamp moves forward, friction between the rods and the clamp is minimized. On the other hand, adding bushings and rotating rods adds complexity to the concept.

Grippers

The design of the gripper sub-assembly was also reviewed. Analysis of the design from a manufacturing standpoint led to the conclusion that the mounting mechanism was flawed. Precise alignment of all the gripper blocks would be possible only if all the holes were exactly aligned and had the same diameter, goals which are very difficult to meet in practice. The seven inch length requirement eliminated the option of machining the gripper blocks from a single piece, drilling the holes, and then cutting the machined block into several pieces. Discussing how the grippers could be made, however, led to developing the concept shown in Figure 39.

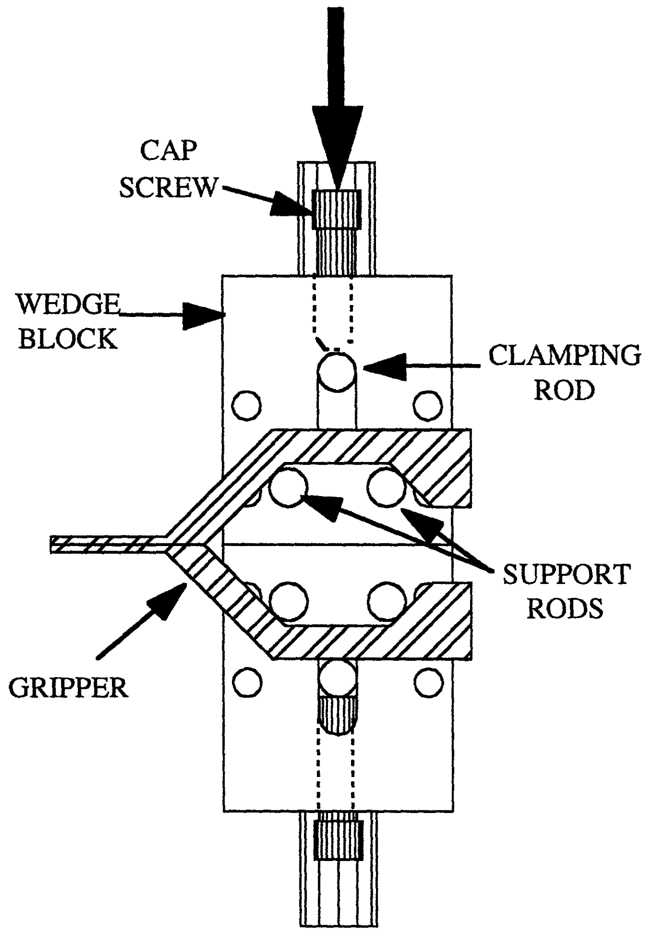

Figure 39: Three-point contact gripper mounting.

The two holes for the rod have been replaced by a 'channel' or slot with tapered sides. The gripper block is pressed against the two support rods by a clamping rod. The tapered sides of the channel makes this mounting mechanism self-aligning. The open channel also makes it possible to machine a set of grippers from a single block.

At the same time that the mounting mechanism was being modified, the design of the prototype grippers was being developed. The requirements affecting the shape of the grippers were:

- Long enough to bend parts where distance from bend to tip is equal to 1.250 inches
- Thin enough to fit between rows (minimum row spacing = .040 inches)

- Stiff enough to withstand forces applied during bending
- Easy to manufacture

The shapes chosen for the prototype gripper and spacer blocks are shown in Figure 40.

Figure 40: Prototype gripper and spacer blocks.

Frame

Sizing the frame and gripper sub-assemblies made it clear that it would take a fairly large torque to rotate the main frame. While the magnitude of the torque was not a problem by itself, the fact that the torque would be a function of the angular displacement placed some difficult demands on the controller. This problem was resolved by adding counterweights to the side beams. Adding the counterweights moves the center of gravity of the system closer to the axis of rotation, thus decreasing the static torque requirements on the motor. On the other hand, the counterweights increase the inertia of the system, thus increasing the dynamic torque requirements. Given that the rotational speed of the system is relatively low (< 30 rpm), such compromise was deemed acceptable.

Round 5 - Clamping mechanism completed

During this round the design was improved further by substituting the single, central clamp with two clamps mounted on a beam as shown in Figure 41. The central clamp was discarded to eliminate the possibility of interference between the clamp and the casing transport module. To eliminate bending loads on the piston rod, the clamps were mounted on linear carriages, also shown in Figure 41.

Figure 41 also shows the Delrin wedges added to the clamps. Delrin was selected because of the low coefficient of friction of Delrin on aluminum. It was envisioned at this point that the wedges would be glued to the clamps.

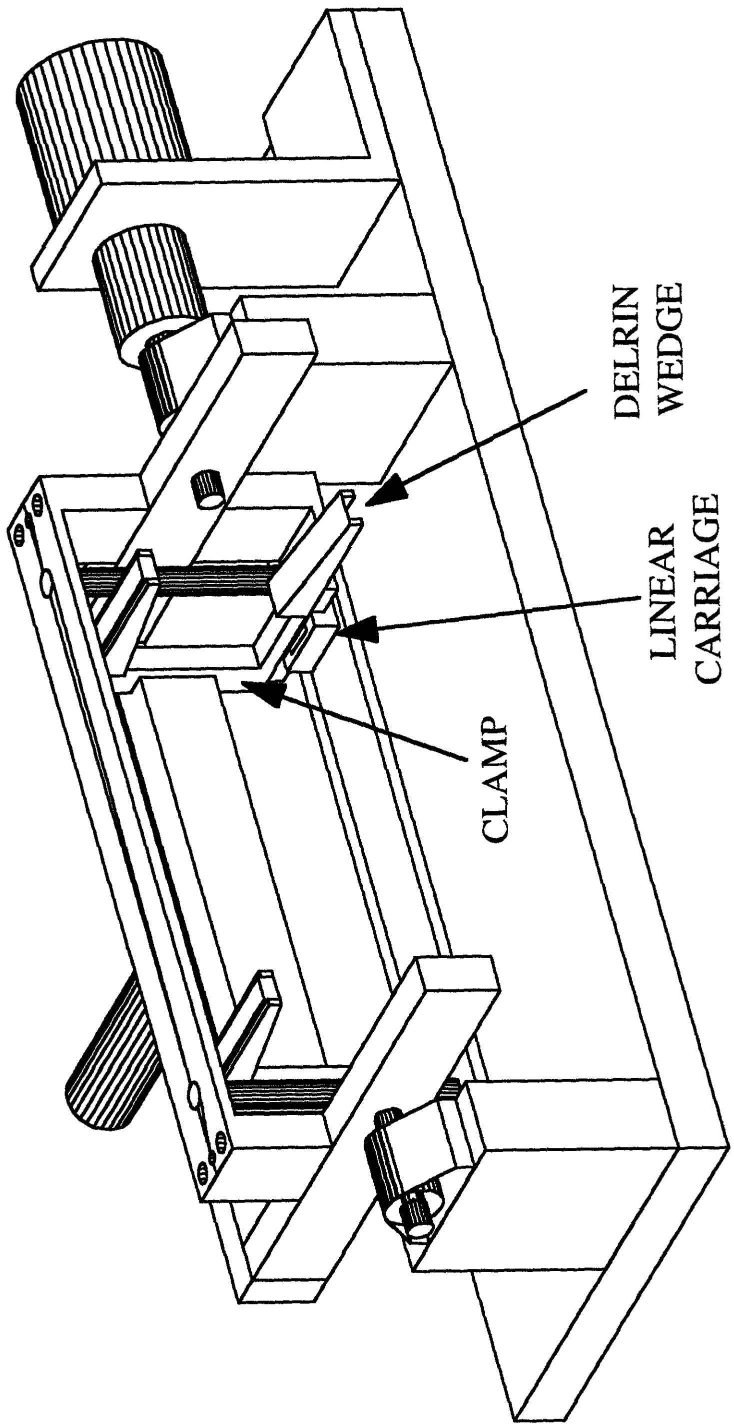

Figure 41: Improved clamping mechanism.

Splitting and moving the central clamp also forced the re-design of the gripper mounting blocks (see Figure 42).

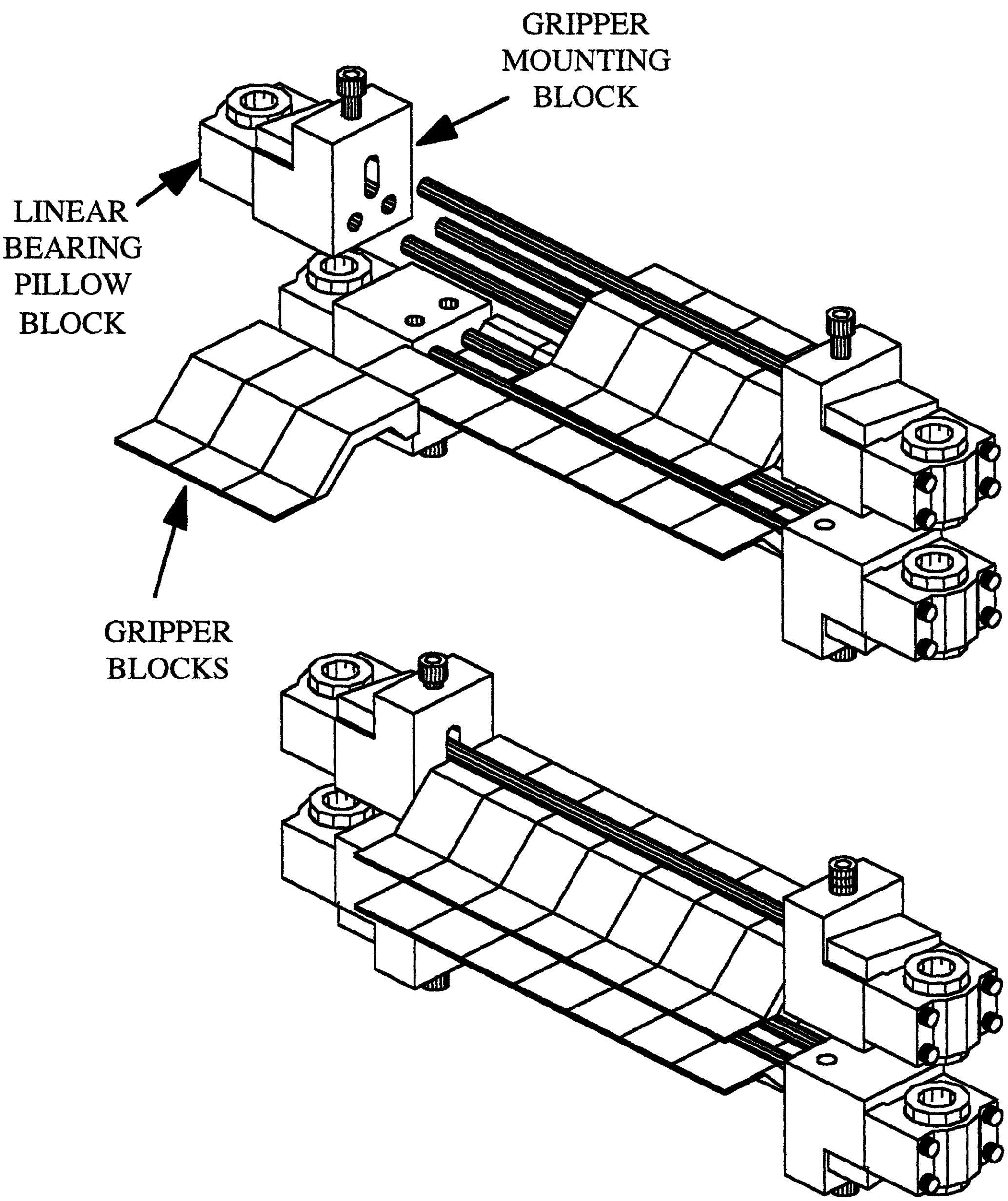

Figure 42: Gripper mounting block - linear bearing pillow block assembly.

The gripper mounting blocks, mounting rods, linear bearing pillow blocks, and clamping mechanism are shown mounted on the main frame in Figure 43. The design concept at the end of round 5 is shown in Figure 44

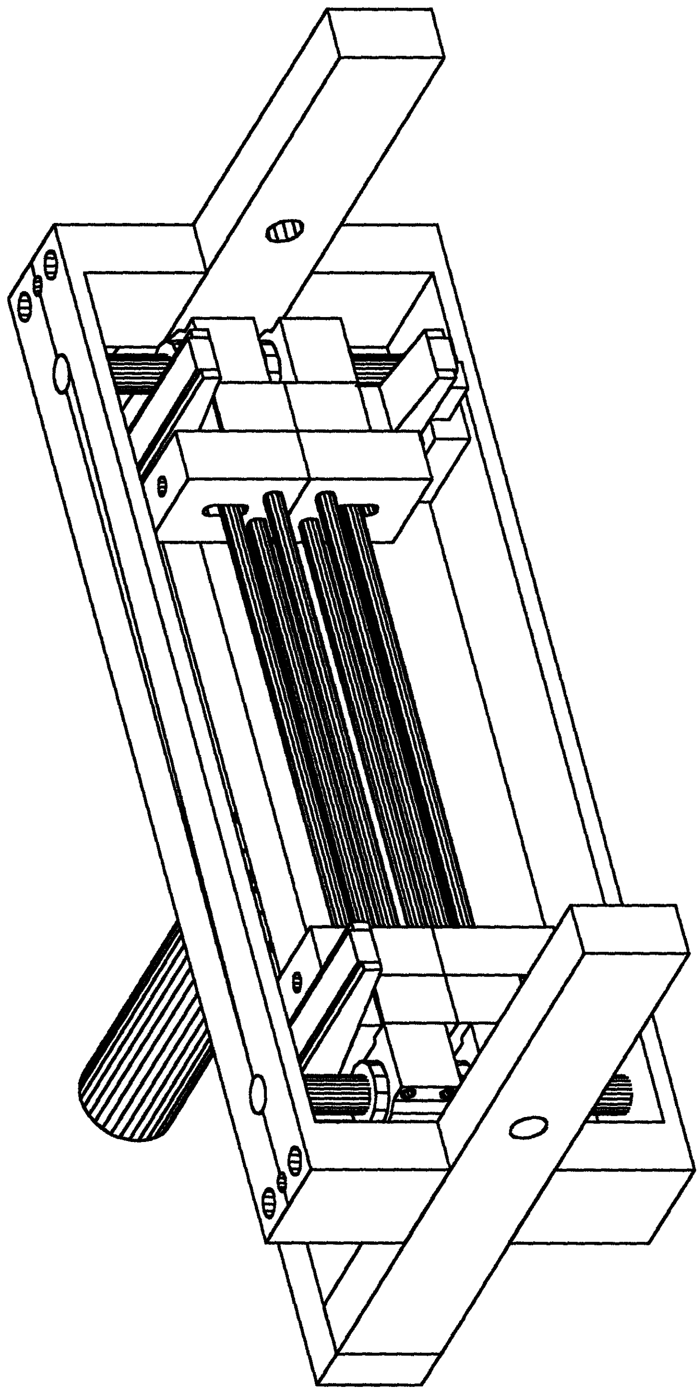

Figure 43: Gripper mounting assembly, clamping mechanism, and main frame.

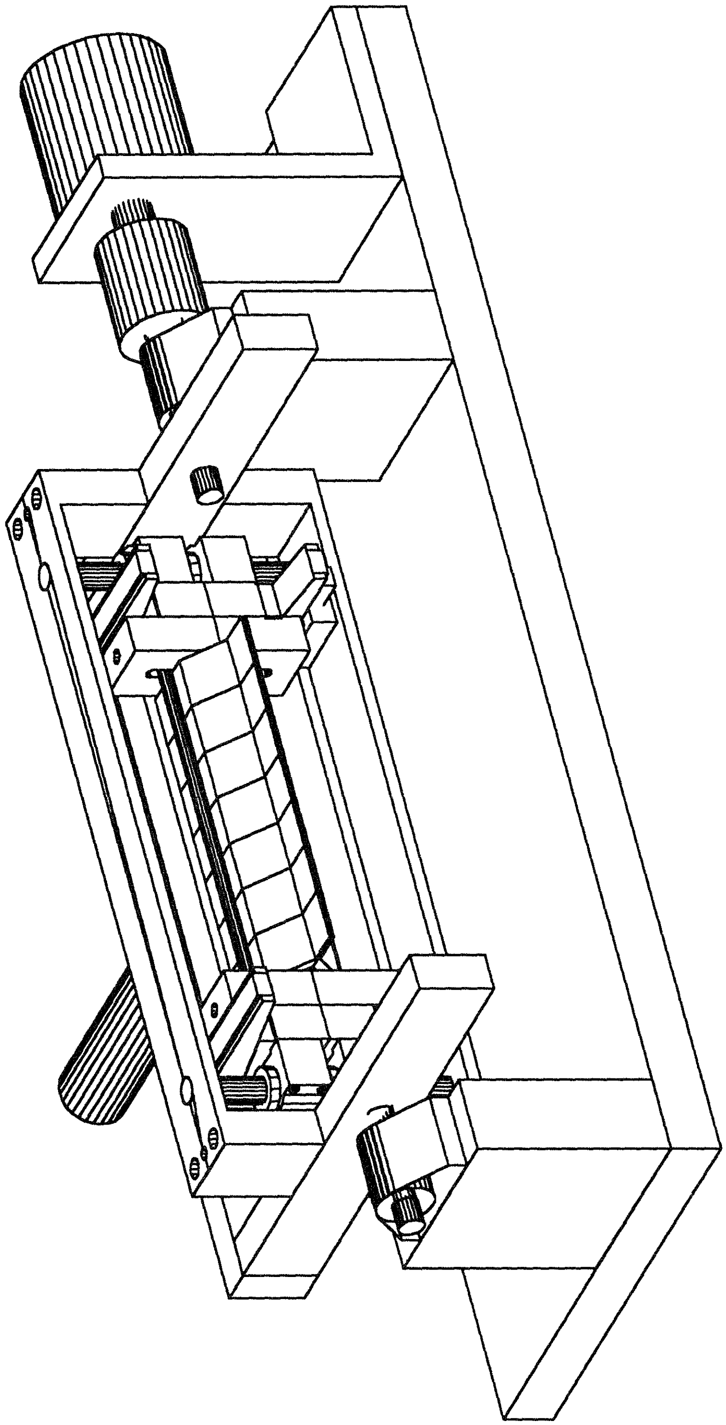

Figure 44: Round 5 design concept.

Round 6 - Designing for assembly

The sixth round consisted of making relatively minor changes to the module to facilitate assembly and to insure proper alignment between the different parts. The main change consisted of adding steps to the lower u-bar. The purpose of the steps is to serve as registration surfaces for the clamp's linear carriages. The linear carriages will be registered against the steps, assuring parallelism between the two carriages and the frame (see Figure 45).

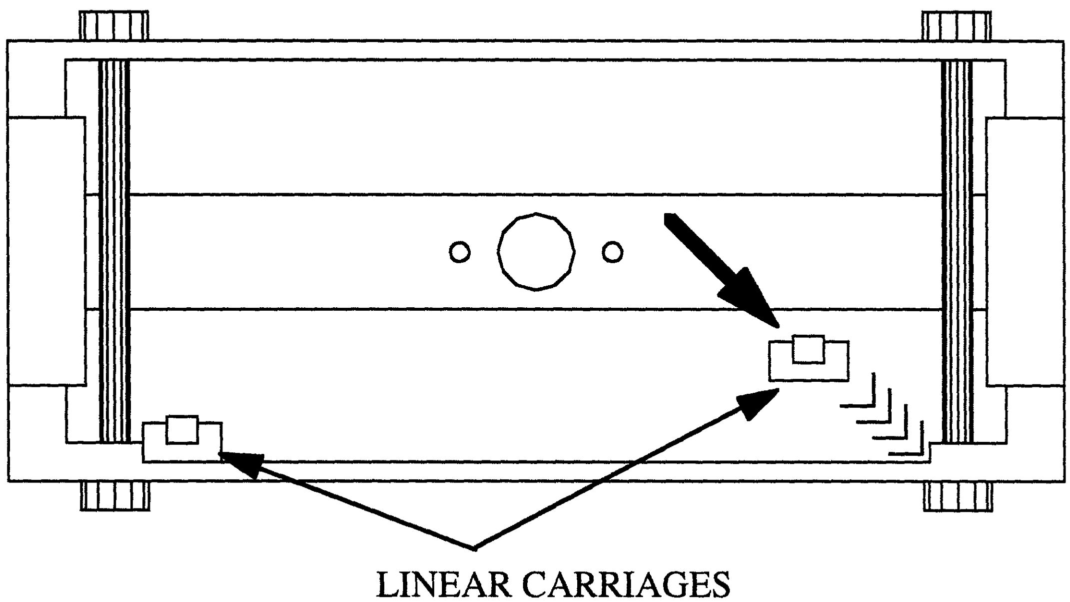

Figure 45: Linear carriages registered against steps.

Sizing of the motor, and designing of the motor bracket, pillow blocks, mounts, counterweights, and base were also completed during this round. The base, motor bracket, and pillow block mounts are shown in Figure 46. The pillow block and motor mounts are aligned with respect to each other by backing them up against pockets milled into the base.

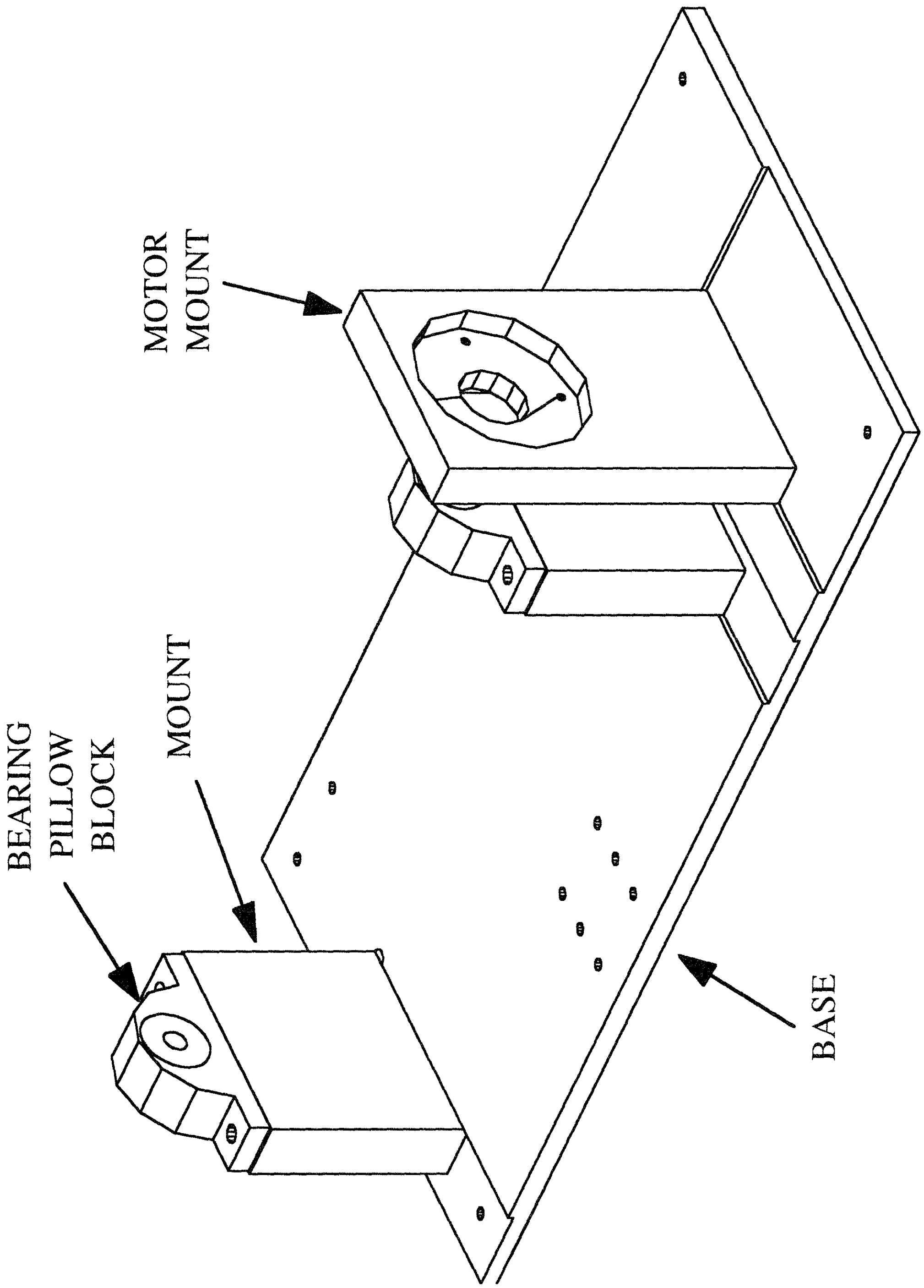

Figure 46: Base, pillow block and motor mounts.

The completed design at the end of round 6 is shown in Figure 47.

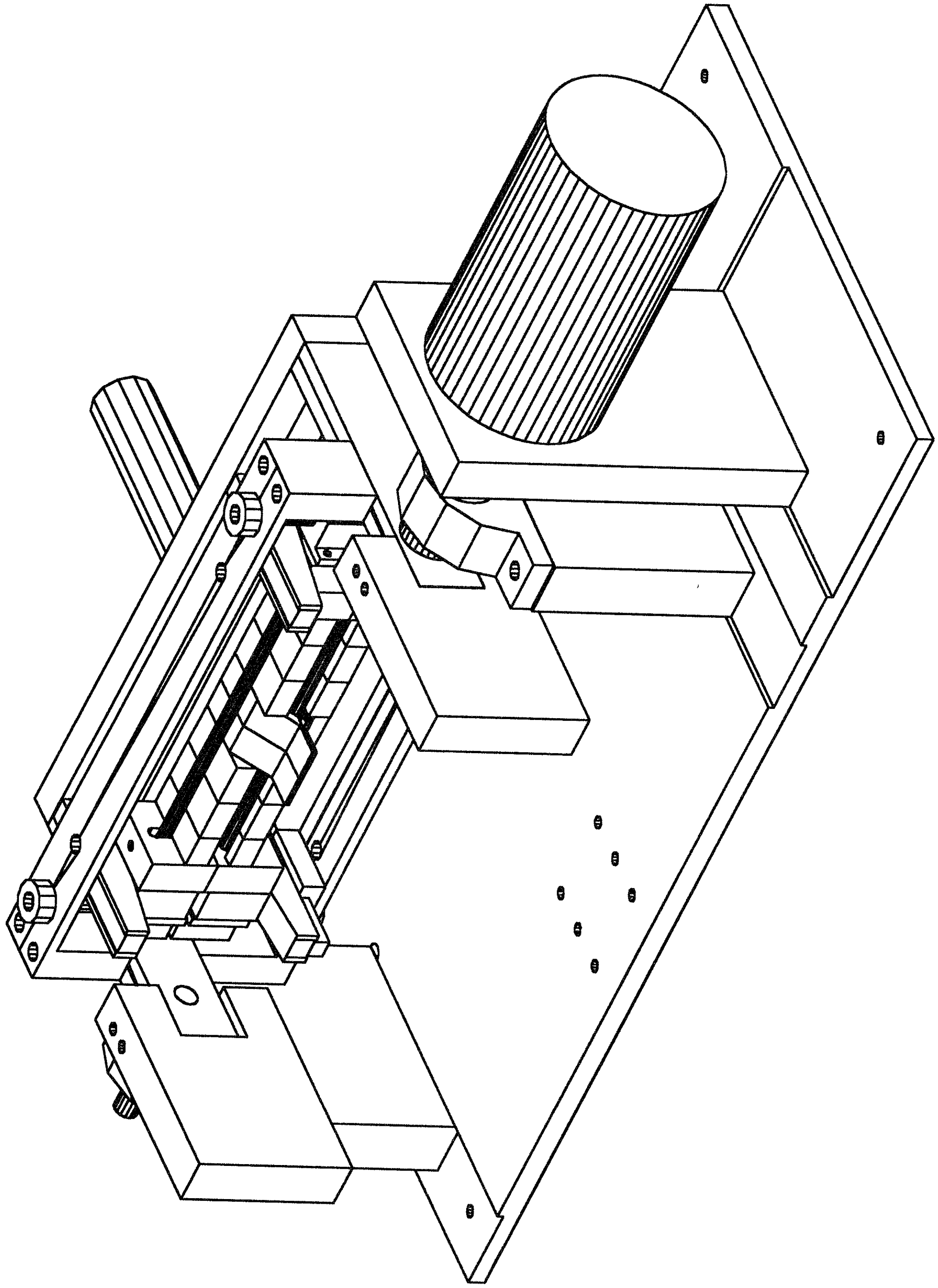

Figure 47: Bending module design at the end of round 6.

Round 7 - Final prototype design

Round seven of the prototype design process consisted primarily of simplifying gripper mounting. Some minor changes were also incorporated into several parts to facilitate fabrication.

Gripper mounting

Although the gripper mounting concept developed during Round 4 was very appealing for its apparent simplicity, building of a test prototype revealed two problems. First, because of the long moment-arm, it only took a small force applied at the tip of the gripper block to counteract the force applied by the upper rod. Second, the tapered sides increased the machining cost considerably. The shortcomings were eliminated by substituting the support rods with a rectangular cross section mounting beam, as shown in Figure 48.

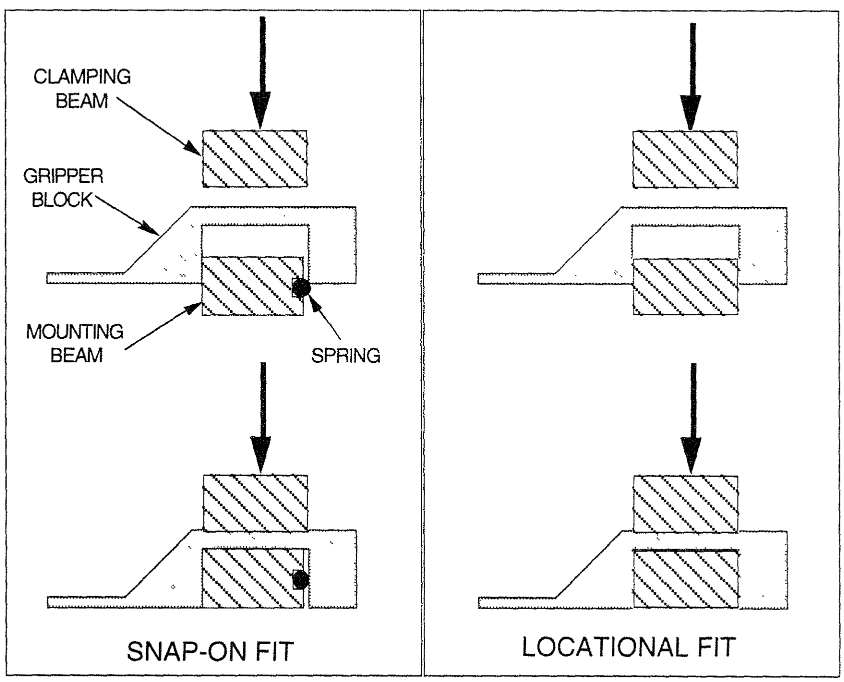

Figure 48: Grippers mounted on rectangular beam.

The grippers are pushed against the mounting beam by a clamping beam, which is bolted onto the wedge blocks. The figure shows the two options that were considered to achieve accurate positioning of the gripper blocks. On the left, the gripper block is forced against one side of the beam by a spring placed inside a slot on the opposite side. Such spring would have to exert a relatively high horizontal force, to keep the gripper from moving horizontally, and a small vertical force, to facilitate mounting of the gripper block. The concept shown on the right relies on a locational fit and on the frictional force between the clamping beam and the gripper block to constrain the gripper block horizontally. The latter option was selected for the prototype because of its simplicity. The parts designed based on this concept are shown in Figure 49.

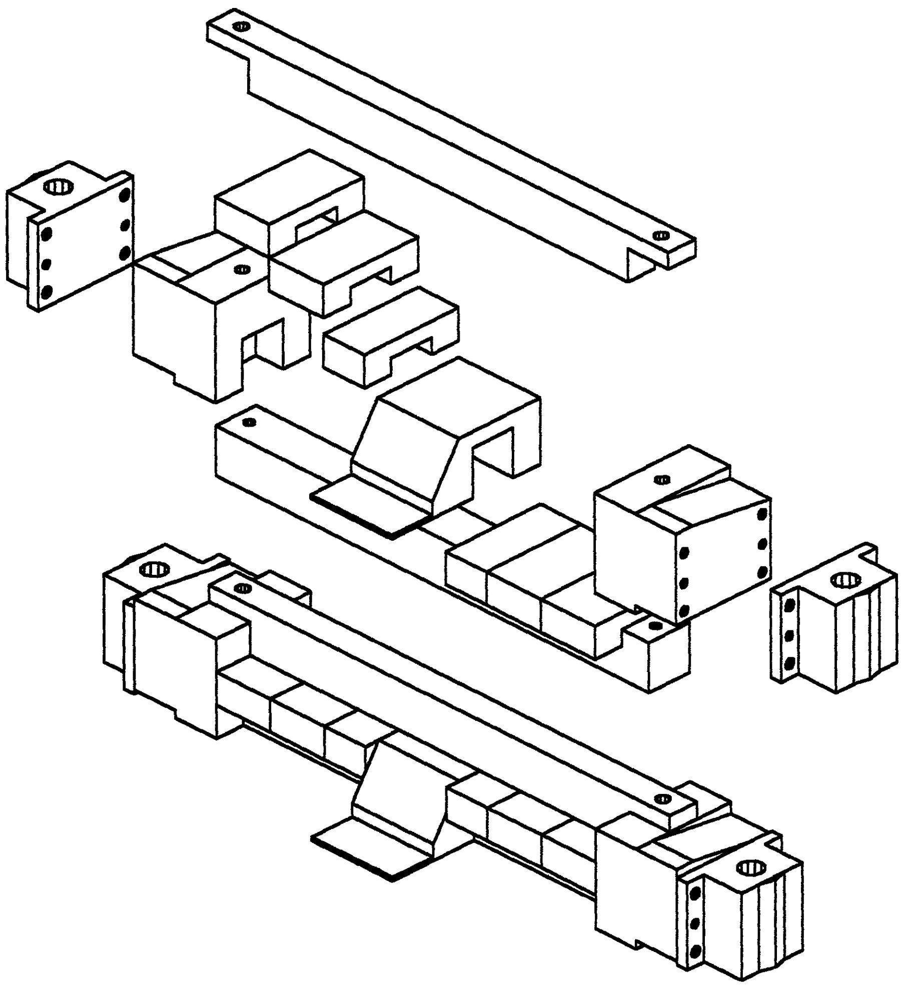

Figure 49: Improved gripper mounting assembly.

3.5 Robust Design

3.5.1 Sensitivity to Parameter Variations

The assembly system described in Chapter 2 needs to be robust to variations both in the casings and in the parts. Casings vary mainly in their gross dimensions and in their straightness along their length. As far as the parts are concerned, the sources of variation can be traced back to material parameter variations (e.g. yield strength, strain hardening coefficient), dimensional variations (e.g. asymmetrical or non-uniform cross sections, undersized or oversized diameters, widths or heights, etc.), and damage due to handling.

Although addressing every source of variation fell beyond the scope of this thesis, it was considered important to select a few that could lead to better understanding of the assembly operations. In the bending operation, material and geometrical property variations have a direct effect on the final shape of the part because of springback.

Springback

Springback in bending may be defined as the difference between the loaded and the unloaded bend angles of the bent part, as shown in Figure 50. Such difference is due to the fact that plastic deformation is followed by elastic recovery when the load is removed. Figure 50 also shows that the bend radius[6] of the unloaded part is larger than the bend radius of the loaded part [3].

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[6] Bending terminology is explained in Appendix C.

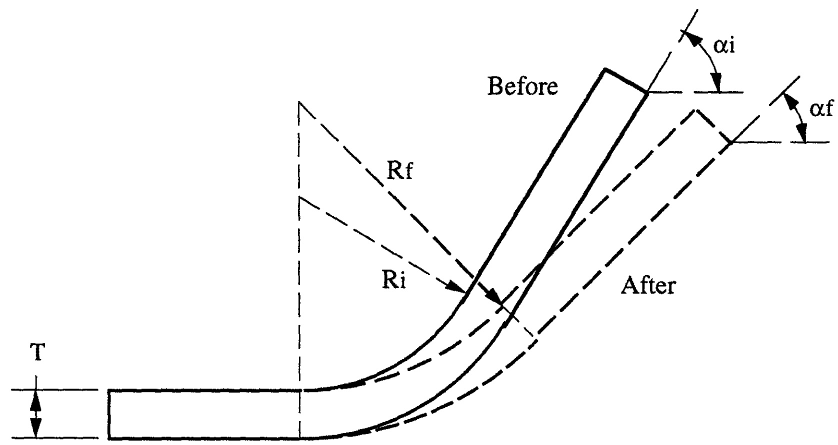

Figure 50: Springback in bending.

A finite elements model was developed to study in more detail the sensitivity of springback to variations in part material properties, part dimensions, and gripper dimensions. The question to be answered through the analysis was whether variations between part batches are large enough to justify on-line springback compensation. The analysis, described in detail in Appendix D, revealed the following:

1. Sensitivity to part radius: A 10% change in the cross-sectional radius of a part results in a springback variation of approximately 0.6 degrees (see Figure 51).

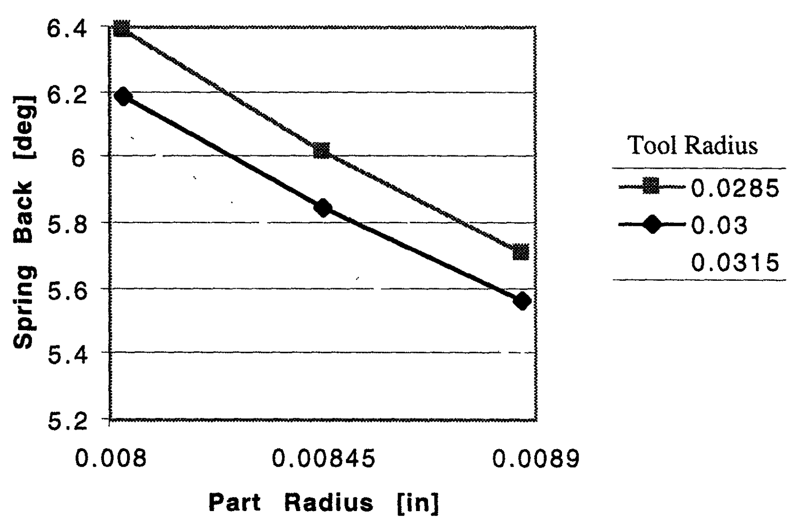

Figure 51: Spring-back vs. part radius for different gripper radii.

2. Sensitivity to yield strength. A 10% change in the yield strength of the material of the part produces a springback variation of approximately 0.3 degrees (see Figure 52).

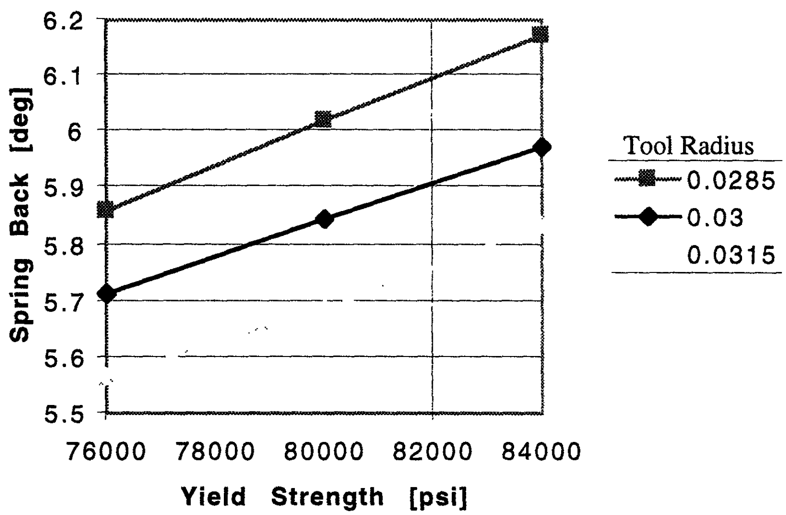

Figure 52: Spring-back vs. yield strength for different gripper radii.

3. Sensitivity to tool radius. A change of 10% in the tip radius of the gripper fingers causes a variation of approximately 0.3 degrees on the springback angle (see Figure 53).

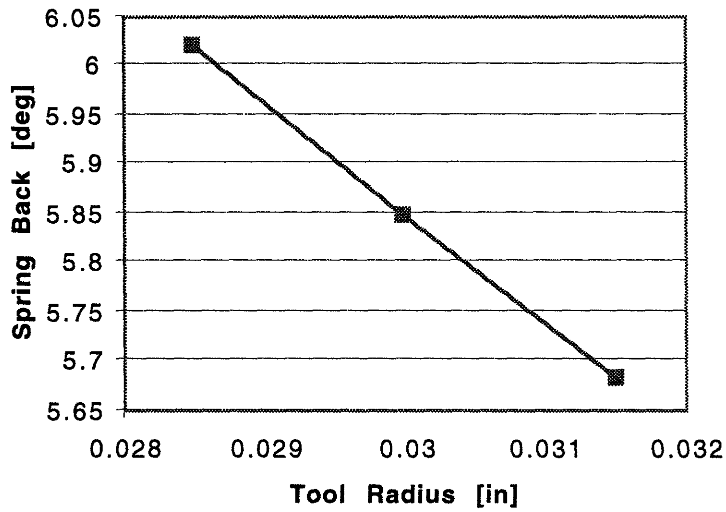

Figure 53: Springback vs. tool radius.

The significance of these results can be better understood by considering that to achieve the specified tolerance on the part tip position, the following condition must be true:

$$\Delta\Theta \leq \frac{TOL}{2L}$$

where:

$\Delta\Theta$ = Angle error in radians (springback)

TOL = specified tolerance in inches

L = distance from bend to tip

These parameters are shown in Figure 54.

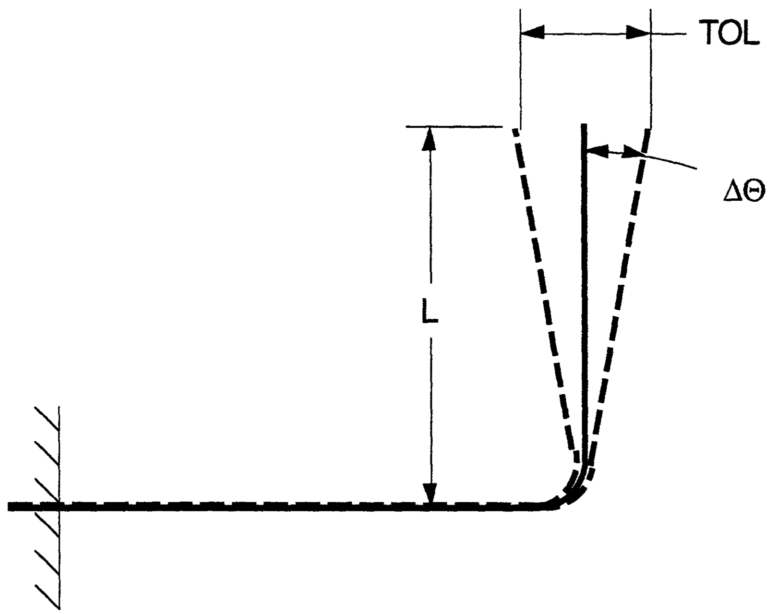

Figure 54: Maximum springback.

This condition was tested for a sample part for which TOL = 0.010 inches and L = 0.75 inches. For these values, $\Delta\Theta$ must be less than or equal to 0.38 degrees. Going back to the results of the FEA analysis, it is clear that a variation in the part radius of approximately 5% would be sufficient to exceed the specified tolerance.

Based on these results and the fact that increased robustness would greatly enhance the appeal of the bending module, a concept for an on-line springback compensation method was developed. Such concept is described in the following section.

3.5.2 Springback Measurement and Compensation

Problem Overview

In current practice, springback is often compensated by overbending the part, coining the bend area by subjecting it to high localized compressive stresses, and subjecting the part to tension while being bent [3]. Current approaches to springback measurement typically rely on statistical process control (SPC). Process data is acquired through post-assembly inspection of tip end-position, typically using machine vision systems. Products with bends that exceed the tolerance limits set by the inspection system are scrapped. If the data suggests that the bending process is getting out of control, the machine must be stopped until the source of the variation is isolated. In most cases, assembly can be resumed once the machine has been re-calibrated.

Calibrating a machine to compensate for springback is a time consuming task even when the machine bends only a single type of part. Because of material variations, the overbend angle needs to be changed frequently. Given that the overbend angle must be determined experimentally, adjusting the machine requires stopping the assembly process, which reduces the production rate. Since material variations are not tracked, the adjustments are made only after a number of defective parts have been assembled.

While time consuming, compensating for springback in a dedicated machine is a fairly straightforward task because the machine bends only one type of part. In a flexible bending machine, in contrast, the type of part becomes a new variable that has to be dealt with during calibration. Since the flexible bending module may have to assemble as many as three different types of products per day, a quick and robust system of springback measurement and compensation becomes necessary.

Concepts

Springback may be measured directly, by comparing the desired end-position versus the actual end-position of the part tip, or indirectly, by comparing the desired versus the actual angle the part has been bent to. In this section, concepts that use both approaches will be described.

Torque Measurement

Let us assume that the torque needed to bend a given number of parts can be measured accurately and in real-time. Furthermore, let us also assume that the angle of the motor shaft can be measured accurately and in real time. Finally, let us assume that the relationship between torque and angular displacement is linear in the elastic region. Then, a plot of torque vs. angular displacement would be similar to the one shown in Figure 55.

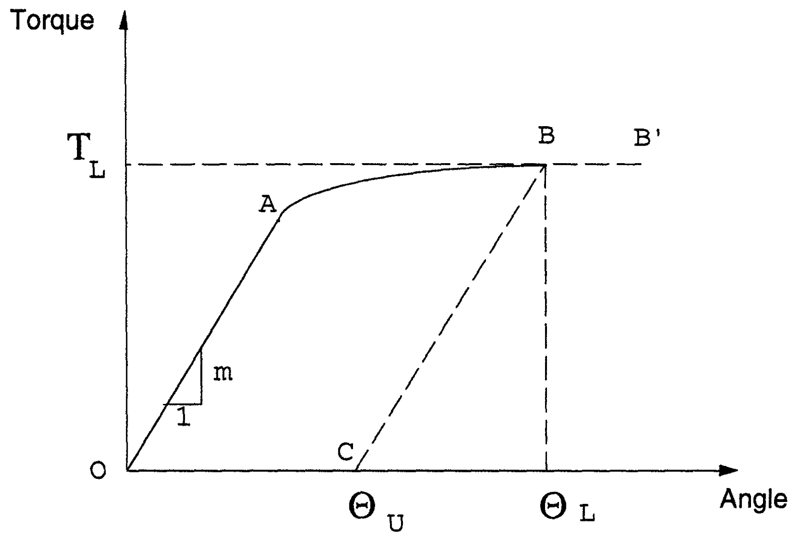

Figure 55: Input torque vs. angular displacement.

Given such a relationship, springback (Θ_L - Θ_U) could be predicted and compensated for by sampling the torque input and the angular displacement while traversing segment OA, i.e. while the part is being bent. Such measurements would be used to calculate the slope m ($dT/d\Theta$). Once the elastic region has been left behind, Θ_U could be calculated continuously using the following equation:

$$\Theta_U = \Theta_L - T_L/m$$

Segment AB would be traversed until the calculated Θ_U were equal to the desired angular displacement. Once such condition became true, the part would be released. A similar approach was taken in [5] for the process of roll bending, where material property changes also have a direct effect on final shape.

Pressure / Force Sensors

Springback compensation could be achieved by embedding pressure or force sensors on the gripper fingers. This approach would rely on sampling the force that the part exerts on the gripper during the bending operation. The controller would monitor both such force and the gripper angle, and it would rotate the gripper until the force was equal to zero when $\Theta_{gripper}$ was equal to $\Theta_{desired}$.

Proximity Sensors

An alternative to using pressure or force sensors would be to embed proximity sensors on the gripper fingers. Ideally, the proximity sensors would send a signal to the controller as soon as the gap between the parts and the gripper fingers became larger than zero. The angle at which such signal was sent would be checked against the desired angle. If the angles were equal, the bending operation would be complete. If not, the controller would determine whether it had underbent or overbent the part, and would adjust the overbend angle accordingly.

Vision System

A vision system would directly determine the position of the part tip and feed back the information into the controller. The controller would then determine whether it had overbent or underbent the part, and would adjust the overbend angle accordingly.

Electrical Continuity

Springback could be measured and compensated for by running a current through a circuit consisting of the grippers and the parts. The circuit would be open (or closed) until the part stopped touching the gripper fingers. The opening (or closing) of the circuit, would signal the controller when the gripper angle and the part angle were approximately equal (provided that a small gap existed between the gripper fingers and the parts). The concept could be implemented as illustrated in Figure 56.

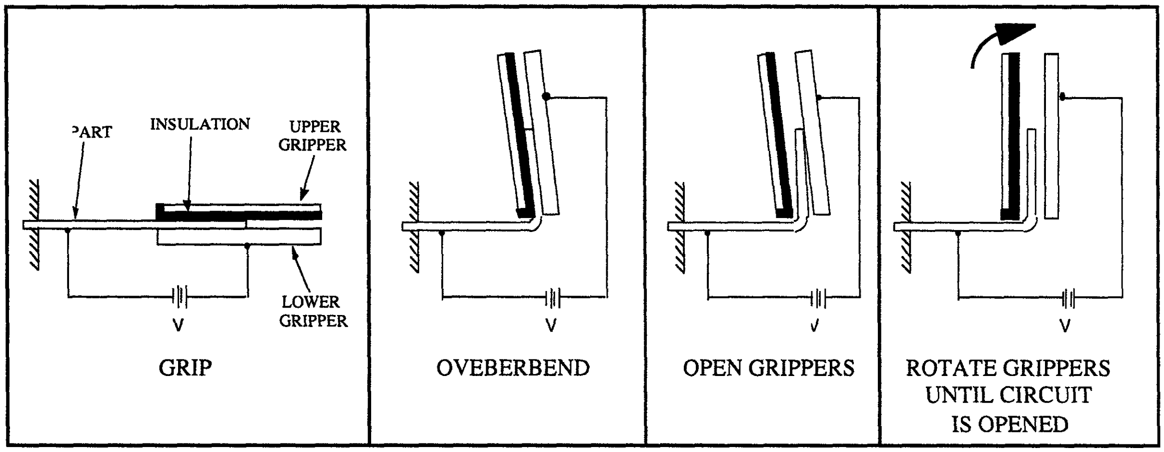

Figure 56: Conceptual implementation of a springback compensation electrical circuit.

After the grippers bent the part to the desired overbend angle, the grippers would be opened until the gap between them was equal to the part diameter plus the specified tolerance. Then, the grippers would be rotated back until the tip stopped contacting the lower gripper, or until the tip made contact with the upper gripper.

Alternatively, the parts could be bent without completely closing the grippers. In this case, the gap between the upper and lower gripper would have to be equal to the part diameter plus one half of the specified tolerance.

Compensation algorithm

Regardless of the type of sensor used, an algorithm similar to the one shown in Figure 57 could be used by the controlling software to converge to the overbend angle that would produce the desired bend angle.

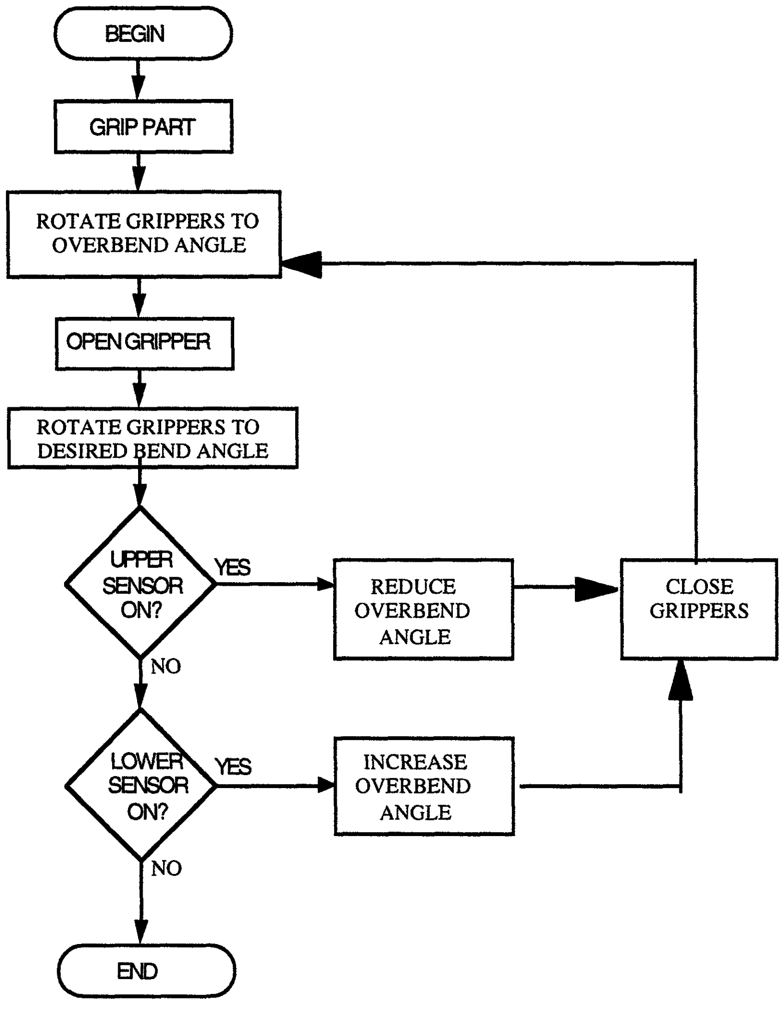

Figure 57: Springback compensation algorithm.

The algorithm shows the procedure that the controller would follow to converge to the desired bend angle. After gripping the part(s), the gripper would be rotated to an arbitrary overbend angle. Then, the gripper would be opened to a pre-determined gap, and rotated to the desired bend angle. At this point the controller would check if the sensor embedded on the upper gripper is on. If it were, the controller would know that the parts had been overbent. To correct the overbend, the grippers would be rotated to a smaller overbend angle. The controller would then query the upper sensor again. This loop would be executed until the upper sensor responded with an off signal to the query.

Once the upper sensor responded with an off signal, the controller would query the lower sensor. It the lower sensor were on, it would mean that the part had been underbent. The controller would then increase the overbend angle. This loop would be repeated until the lower sensor sent an off signal. The bending operation would stop once both the lower and upper sensors sent off signals.

This approach is feasible because experimental observation and the FEA results show that the segments of the part at either side of the bend remain straight after the part has been unloaded. Thus, the position of the tip can be calculated by measuring the angle at which the gripper last makes contact with the part.

Concept Development: Electrical Continuity

The electrical continuity concept was selected for development for three reasons. First, because of its simplicity, it seemed the least expensive approach. Second, it would be relatively easy to implement. Third, it had the potential to measure the variable of interest (tip position) directly, instead of relying on indirect measurements such as the torque measurement system would.

The electrical continuity concept was developed by using a part bending jig consisting of a pair of manually adjustable grippers mounted on a rotating c-shaped frame. Two sets of experiments were performed with the jig. In the first set, a circuit was setup so that current would flow until the part being bent stopped touching the upper gripper. In the second set, the circuit was setup so that current would start flowing once the part touched the lower gripper. Furthermore, the gap between the grippers was decreased to a value that enabled to bend the parts to a tighter tolerance. The two sets of experiments are explained in more detail in the following sections.

Results

First Set

In the first set of experiments, a circuit was set up to measure and compensate springback as shown in Figure 58.

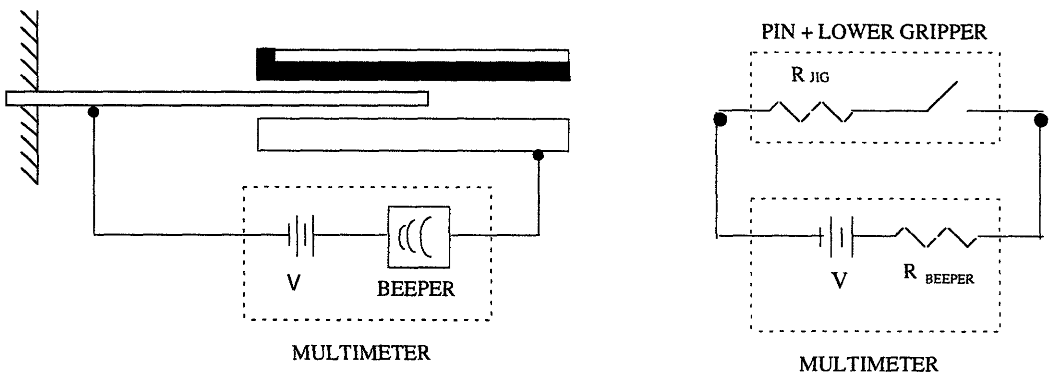

Figure 58: Experimental setup and electrical circuit equivalent.

The upper gripper was fully insulated. A multimeter was set to check for continuity. One of the multimeter's prongs was connected to the part, while the second prong was connected to the lower gripper.

To characterize bending, the angles shown in Figure 59 were defined. The overbend and circuit-off angles were measured using a protractor attached to the test jig. The final bend angle β was measured using a second protractor. The measurement error was estimated to be plus or minus one degree.

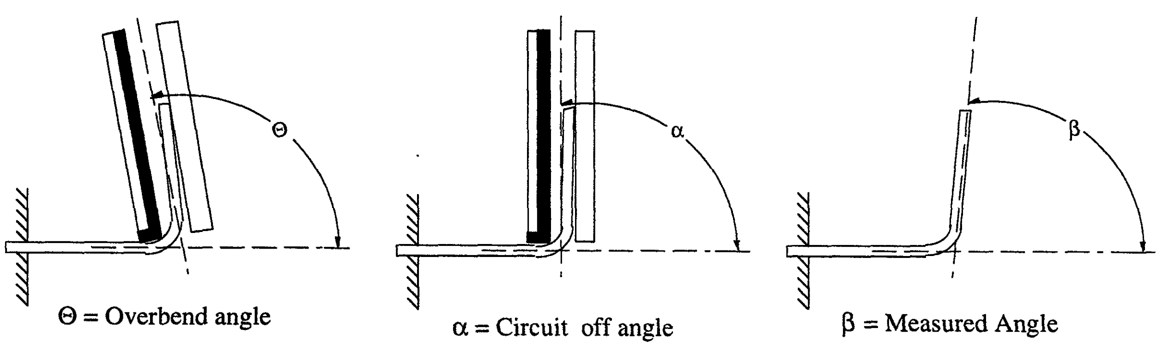

Figure 59: Overbend, circuit-off, and measured angles.

Since the grippers of the test jig could not be easily adjusted, bending was performed without gripping the part. The gap between the gripper fingers was set to 0.028 inches, a value that would enable the part to be bent to the specified tolerance if it is positioned exactly halfway between the two gripper fingers prior to bending.

The grippers were rotated manually to the desired overbend angle. Then, the grippers were rotated back until the multimeter continuity beep stopped. Once the beep stopped, the part was released, and its actual angle was measured. This procedure was repeated for overbend angles of 100, 102, 104, 106, 108, and 110 degrees.

Initially, a large discrepancy (in the order of 10 degrees) between the circuit off and the measured angle was detected (see Figure 60). The expected difference (calculated from geometrical constraints) however, was less than one degree. Such inconsistency between expected and actual angles suggested that good electrical contact was not being established, results were being distorted due to the crudeness of the experimental jig, or the multimeter could not detect current flow below a certain value.

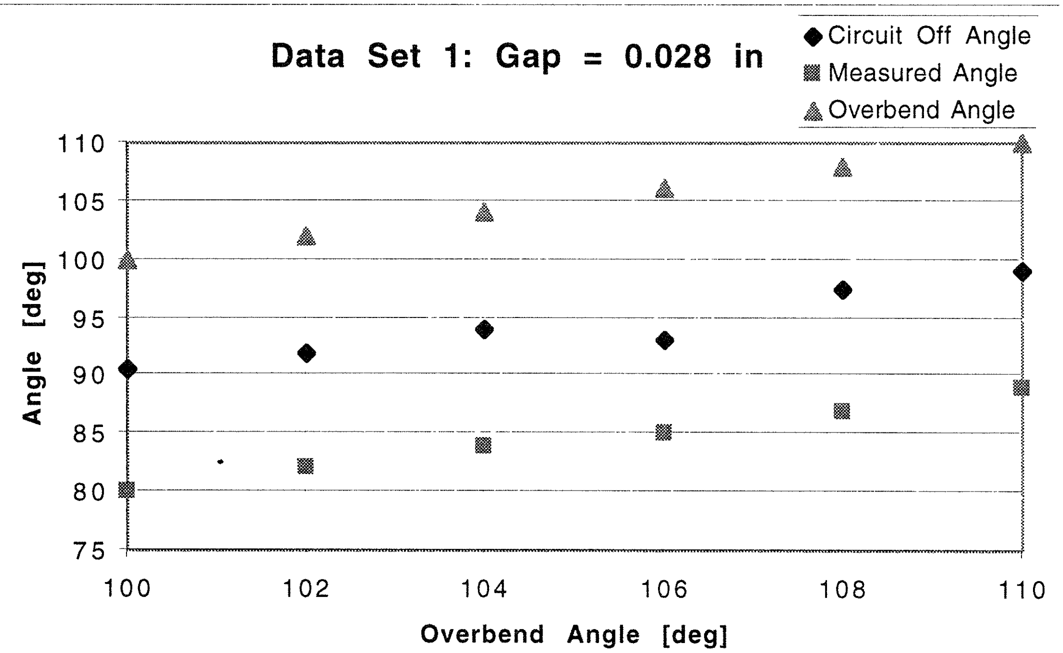

Figure 60: Experimental results for a gap of 0.028 inches

Closer observation showed that a region of intermittent contact existed, i.e. the beep would stop and start as the gripper was being rotated back to the position where the part was unloaded. Thus, the part was being released before it had been fully unloaded. The problem was corrected by backing up the grippers until the beep stopped altogether. Once such correction was made, the difference between the circuit off angle and the measured angle became much smaller[7], as shown in Figure 61. The average error, defined as the difference between the circuit-off and the measured angles, was approximately equal to 0.6 degrees.

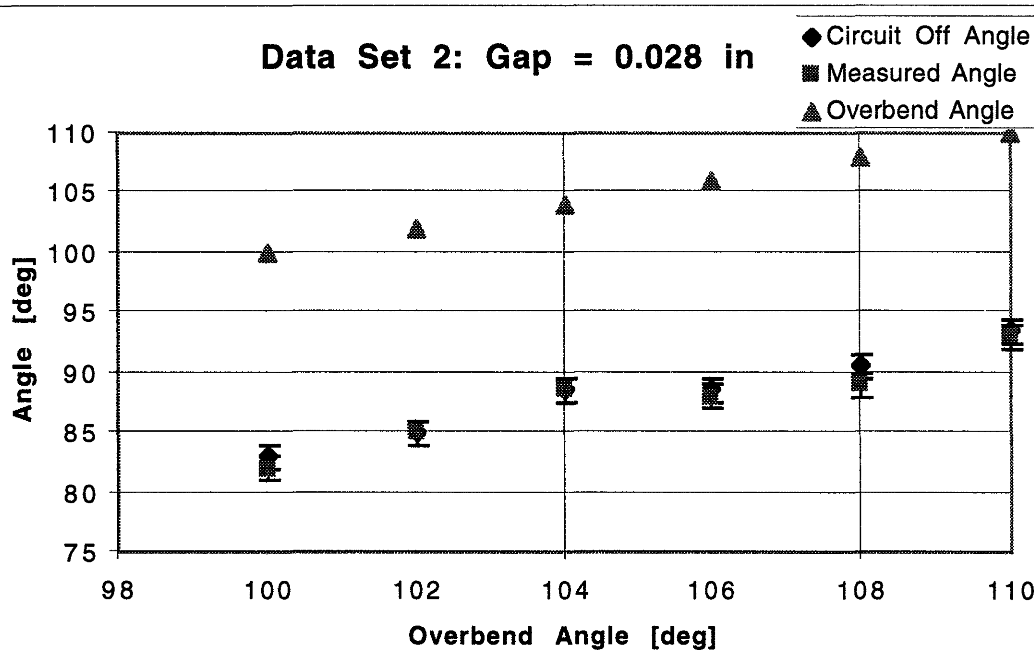

Figure 61: Improved experimental results for a gap of 0.028 inches.

A secondary result from the experiments was that a good bend could be made without gripping the parts. Since eliminating the gripping action would result in a faster cycle time, a second set of experiments were conducted to verify whether the electrical continuity concept would work with a smaller gap between the gripper fingers.

Second Set

The size of the gap to be used was determined assuming that, in the worst case, the position of the part with respect to the grippers is known to within 0.005 inches. Since the diameter of the sample part is 0.017 inches, the size of the gap must be less than or equal to 0.022 inches.

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[7] This result was verified by conducting an extra set of experiments. The results are included in Appendix E.

At the same time that the gap was reduced, it was proposed that a more robust approach to sensing tip position would be to fully insulate the lower gripper, and to partially insulate the upper gripper. With such approach, the beep would start when the part first touches the gripper, thus eliminating the uncertainty caused by the intermittent contact region.

The approach outlined above was tested during the second set of experiments. The circuit was setup such that the beep started when the part first made contact with the upper gripper. The setup is shown schematically in Figure 62.

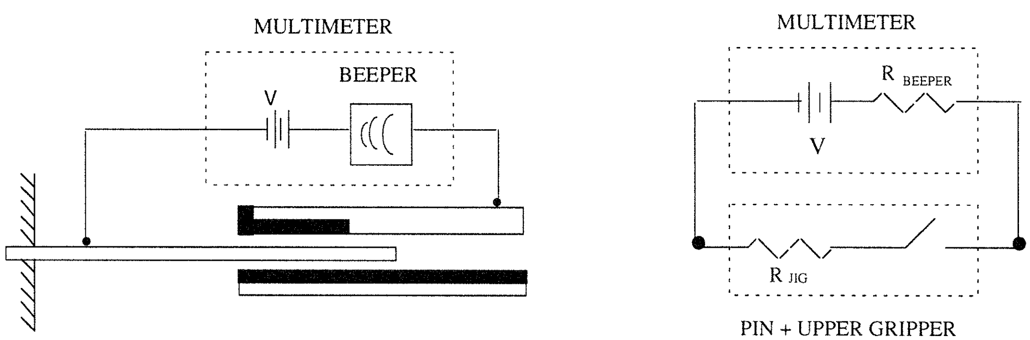

Figure 62: Experimental setup and electrical circuit equivalent for second set of experiments.

The results of the experiments are shown in Figure 63.

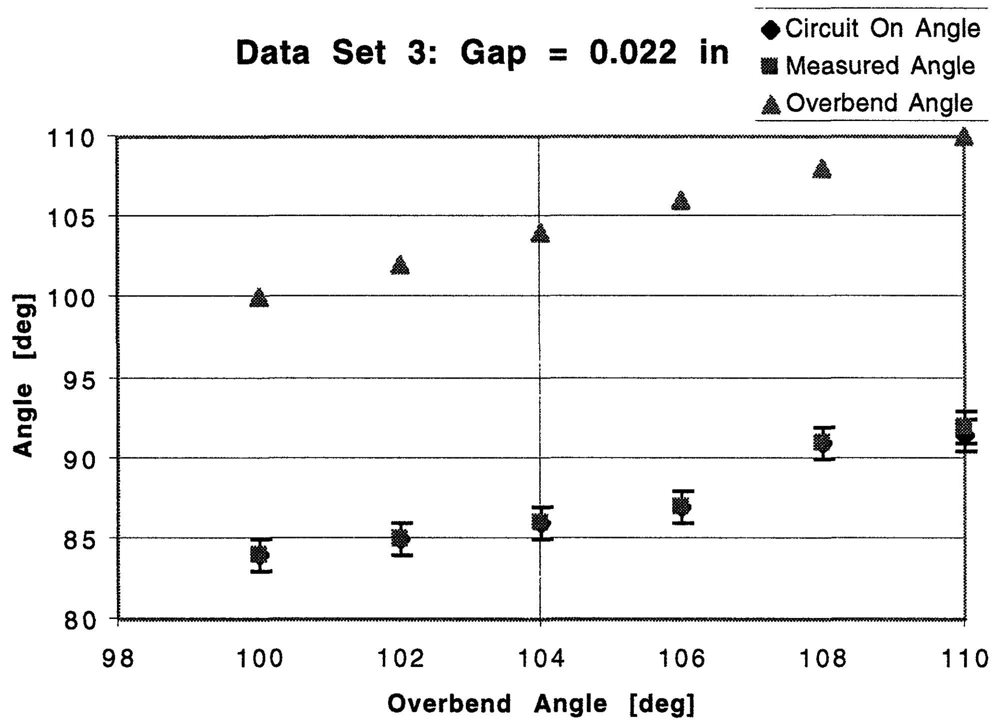

Figure 63: Experimental results for a gap of 0.022 inches.

The figure shows that the difference between the angle at which the circuit was closed and the measured angle is within the bounds of the estimated measurement error (+- 1 degree) and thus negligible. The figure also shows a kink in what otherwise seems to be a linear relationship. Both the circuit-off and the measured angles are somewhat lower than what the previous points lead to expect for an overbend angle of 106 degrees. A similar kink can be observed in Figure 62. This unexpected result may be explained by the fact that the grippers did not close symmetrically about the axis of rotation of the test jig frame, which introduces an error to the circuit-off and measured angles at angles as they approach 90 degrees. This hypothesis was confirmed by performing experiments with the prototype bending module, which is a much more precise machine than the test jig. The results, showing the expected linear relationship, are shown in Figure 83 and Figure 84.

Conclusions & Recommendations

The experiments proved the feasibility of the electrical continuity concept as a springback measurement and compensation technique; however, the concept must be refined, and its robustness improved, before it can be implemented in factory environments.

The concept's capabilities may be enhanced by adding a second circuit that would be activated when the part is overbent (see Figure 64). The robustness of the springback measurement system may be increased by improving electrical contact between parts and grippers. One way of improving contact would consist of reducing the contact area (thereby increasing the contact pressure) by machining tightly spaced grooves on the grippers, also shown in Figure 64.

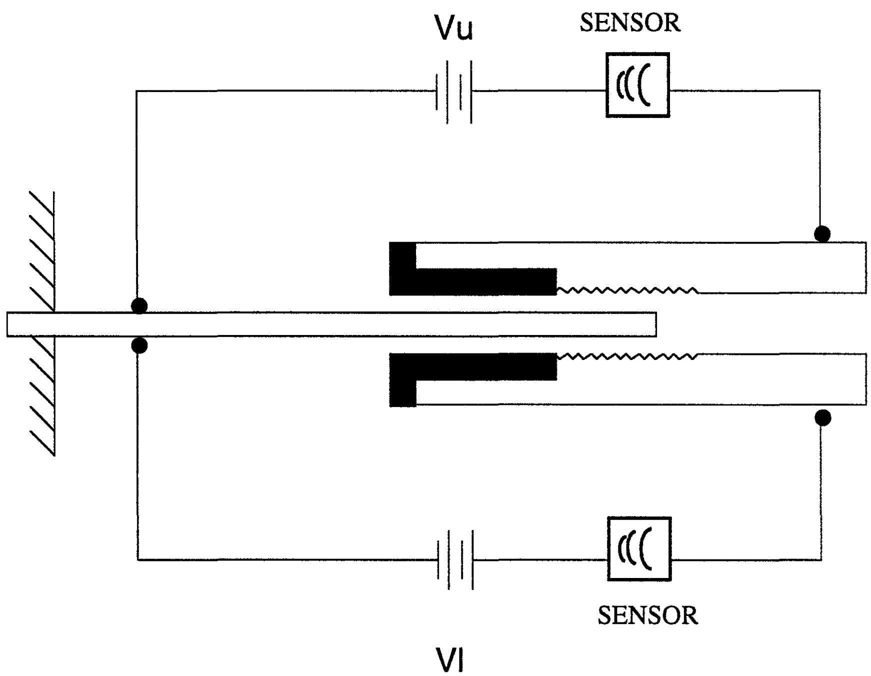

Figure 64: Gripper design for improved electrical contact.

The system could be used to detect damaged parts before bending is done. As the gripper approached the product, parts that are not straight enough to fit in the gap would contact the gripper fingers, thus closing the circuit. The control software could then reject the product or increase the gap until the damaged parts fit in. Once all the parts are between the grippers, the gap could be closed and the bending operation would proceed as usual. The latter option would be viable only if bending the parts fixes the misalignment of the damaged parts.

3.6 Control System Model

3.6.1 Lumped Parameter Model

A position control system for the bending module was designed based on a lumped parameter model of the machine. The rotating assembly was modeled as a flywheel with inertia J_L. The pillow blocks were modeled as a single rotational damper with coefficient b_L. The motor and the gearbox were modeled as a single inertia J_M, with damping coefficient b_M. The coupling was modeled as a torsional spring with stiffness k_c. The parts bent by the module were modeled as a torsional spring to ground with constant[8] stiffness k_t. A schematic of the lumped parameter model is shown in Figure 65. A free body diagram of the system is shown in Figure 66.

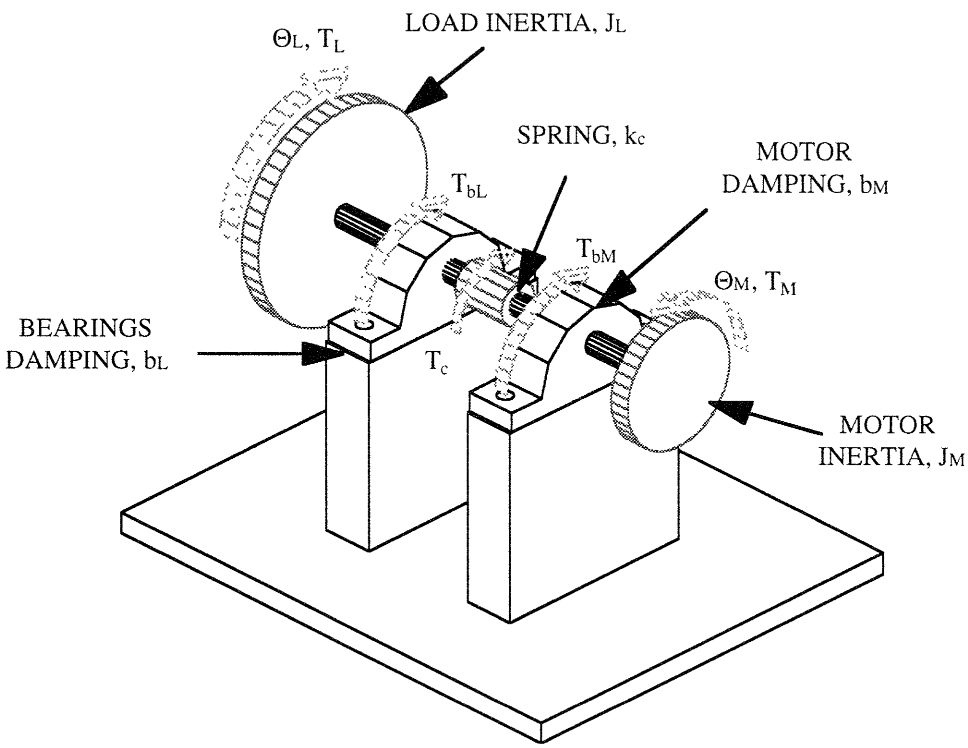

Figure 65: Schematic of the lumped parameter model.

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[8] Note that this approximation is valid only in the elastic region. However, for the sake of simplicity, the stiffness was kept constant even for bend angles that produce plastic deformation.

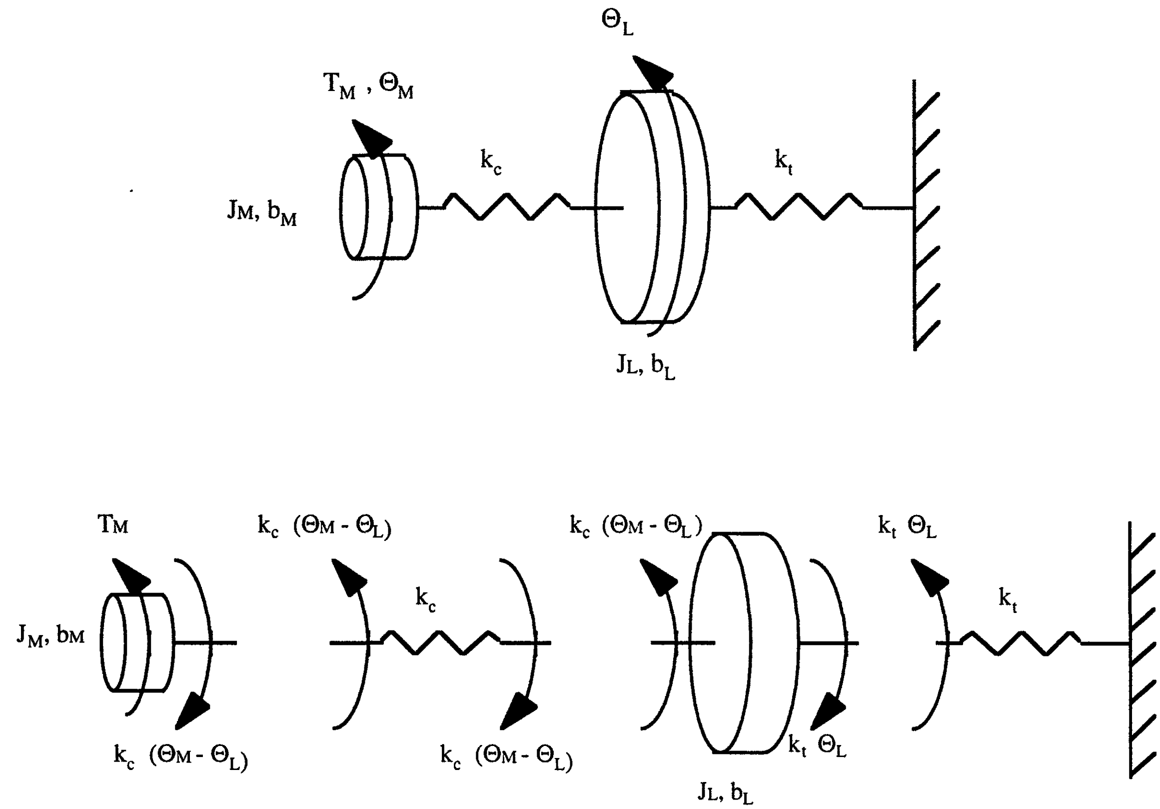


Figure 66: Free body diagram of the system.

Equations of Motion

The equations of motion of the system are derived to be:

$$T_M - k_c(\theta_M - \theta_L) - b_M \frac{d\theta_M}{dt} = J_M \frac{d^2\theta_M}{dt^2}$$

$$-k_t\theta_L - k_c(\theta_L - \theta_M) - b_L \frac{d\theta_L}{dt} = J_L \frac{d^2\theta_L}{dt^2}$$

Transfer Functions

The angular displacement transfer functions of the system may be obtained by taking the Laplace transform of the equations of motion:

$$T_M - b_M s\Theta_M - k_c(\Theta_M - \Theta_L) = J_M s^2 \Theta_M$$

$$-k_t\Theta_L - b_L s\Theta_L - k_c(\Theta_L - \Theta_M) = J_L s^2 \Theta_L$$

Combination of both equations leads to the following transfer functions for the load and motor angular displacements:

$$\frac{\Theta_L}{T_M} = \frac{\left(\frac{k_c}{J_M J_L}\right)}{\left[s^4 + \left(\frac{b_L}{J_L} + \frac{b_M}{J_M}\right)s^3 + \left(\frac{k_c + k_t}{J_L} + \frac{k_c}{J_M} + \frac{b_M b_L}{J_M J_L}\right)s^2 + \left(\frac{b_L k_c}{J_M J_L} + \frac{b_M(k_c + k_t)}{J_M J_L}\right)s + \frac{k_c k_t}{J_L J_M}\right]}$$

$$\frac{\Theta_M}{T_M} = \frac{\left(\frac{1}{J_M}\right)\left[s^2 + \left(\frac{b_L}{J_L}\right)s + \frac{k_c + k_t}{J_L}\right]}{\left[s^4 + \left(\frac{b_L}{J_L} + \frac{b_M}{J_M}\right)s^3 + \left(\frac{k_c + k_t}{J_L} + \frac{k_c}{J_M} + \frac{b_M b_L}{J_M J_L}\right)s^2 + \left(\frac{b_L k_c}{J_M J_L} + \frac{b_M(k_c + k_t)}{J_M J_L}\right)s + \frac{k_c k_t}{J_L J_M}\right]}$$

Simplified expressions for the transfer functions may be obtained by making the following substitutions:

$$a = \left(\frac{b_L}{J_L} + \frac{b_M}{J_M}\right)$$

$$b = \left(\frac{k_c + k_t}{J_L} + \frac{k_c}{J_M} + \frac{b_M b_L}{J_M J_L}\right)$$

$$c = \left(\frac{b_L k_c}{J_M J_L} + \frac{b_M(k_c + k_t)}{J_M J_L}\right)$$

$$d = \frac{k_c k_t}{J_L J_M}$$

$$K_{SYS} = \left(\frac{1}{J_M}\right)$$

$$A = \left(\frac{b_L}{J_L}\right)$$

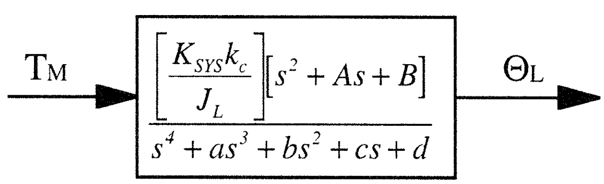
$$B = \frac{k_c + k_t}{J_L}$$

Simplified transfer functions:

$$\frac{\Theta_L}{T_M} = \frac{\left(\frac{K_{SYS} k_c}{J_L}\right)\left[s^2 + As + B\right]}{s^4 + as^3 + bs^2 + cs + d}$$

$$\frac{\Theta_M}{T_M} = \frac{K_{SYS}\left[s^2 + As + B\right]}{s^4 + as^3 + bs^2 + cs + d}$$

The corresponding block diagrams are shown in Figure 67.



$$T_M \longrightarrow \boxed{\dfrac{K_{SYS}\left[s^2 + As + B\right]}{s^4 + as^3 + bs^2 + cs + d}} \longrightarrow \Theta_M$$

Figure 67: Block diagrams of the lumped parameter model.

The values for the different parameters were estimated to be:

J_L = 15 oz-in sec^2
b_L = 10 oz-in-sec / rad
J_M = 1 oz-in sec^2
b_M = 300 oz-in-sec / rad
k_c = 12.0 E6 oz-in / rad
k_t = 13 oz-in / rad

The value for the load inertia J_L was calculated by summing the inertia of the prototype parts. The load damping coefficient b_L was obtained from ball bearing manufacturer specifications. The motor inertia J_M and damping coefficient b_M were estimated from the motor manufacturer specifications. The coupling stiffness k_c was calculated from the dimensions and material properties of the coupling. The load stiffness k_t was estimated by measuring the load necessary to bend the parts of a sample product to a final bend angle of 90 degrees.

3.6.2 Position Controller Model

The hardware used to control the machine consists of an 386 IBM compatible personal computer, a Technology 80 controller card (model TE5650), an Advanced Motion Controls servo amplifier, and a 4000 count quadrature optical encoder mounted on the rear shaft of the motor. The system is shown schematically in Figure 68.

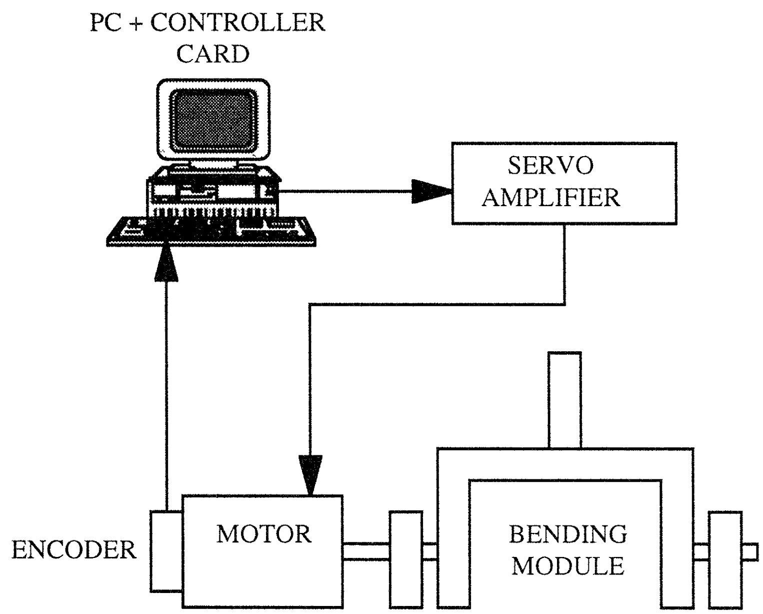


Figure 68: Schematic of the bending module's control system.

A block diagram of the position control feedback system is shown in Figure 69. Note that this block diagram neglects the fact that a discrete control system was used (a valid approximation for fast sample times such as the 0.1 msec sample time of the TE5650 controller card).

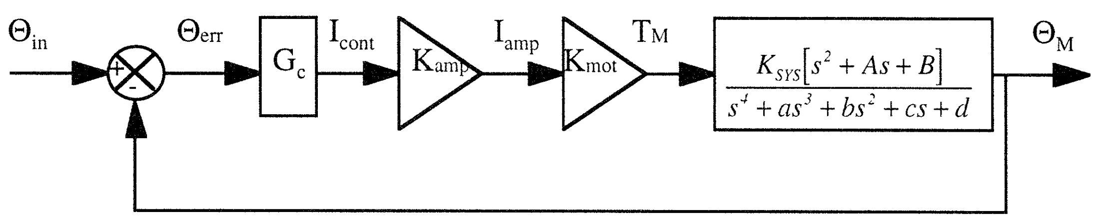


Figure 69: Block diagram of the closed loop system.

The transfer function of the complete system is given by:

$$\frac{\Theta_M}{\Theta_{in}} = \frac{G_c K[s^2 + As + B]}{s^4 + as^3 + (b + G_c K)s^2 + (c + G_c KA)s + G_c KB + d}$$

where $K = K_{amp} K_{mot} K_{SYS}$

It should be noted that the output of interest for the bending operation is the load angular displacement Θ_L; however, since an encoder mounted on the motor was used in the prototype, the controller was designed based on measurement of the motor angular displacement Θ_M. The static error introduced by such approximation may be estimated by calculating the difference between Θ_M and Θ_L. Assuming that the maximum static torque on the bending module is T_L, and given the coupling stiffness k_c, the static error is given by:

$$\Theta_M - \Theta_L = T_L / k_c$$

A conservative estimate of the error may be obtained by using a static torque of 200 oz-in, a value which is ten times larger than the torque needed to bend the parts of the sample product used to test the prototype. For a coupling stiffness of 12E6 oz-in / rad, the resulting error is approximately 0.001 degrees. Such error is two orders of magnitude smaller than the specified final bend angle tolerance and thus can be neglected. To verify whether using Θ_M instead of Θ_L would have any dynamic effects for the range of frequencies and gains at which the bending module will operate, the Bode plots and step responses of the system transfer functions were compared after the controller had been tuned. The plots are shown at the end of this chapter.

To understand the behavior of the system, a pole-zero plot of the transfer function was obtained (see Figure 70).

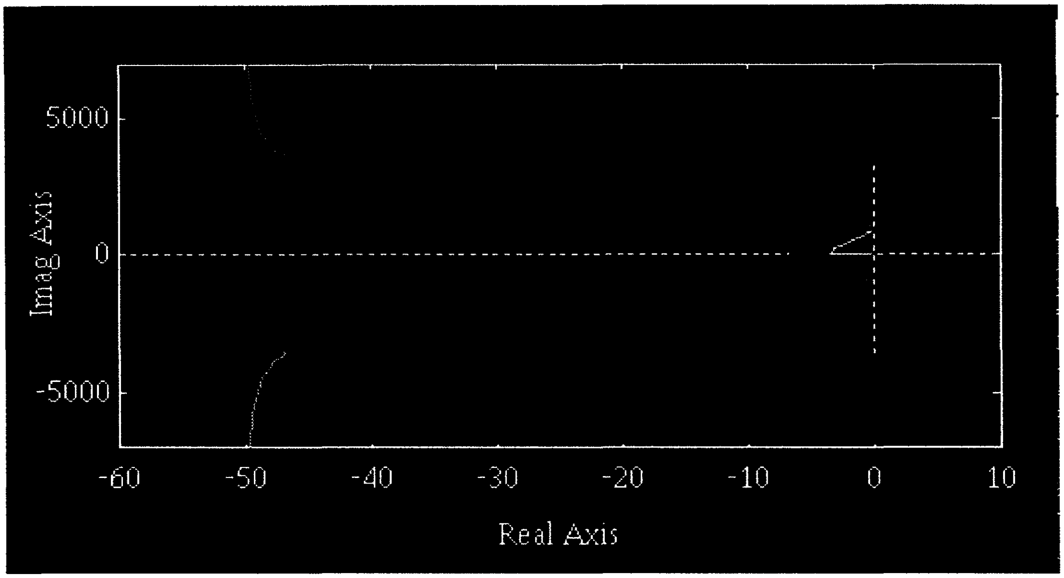


Figure 70: Pole-zero plot.

The plot shows that the system has two zeros, and four open-loop poles. The poles closer to the imaginary axis are pulled and canceled by the zeros as the proportional gain increases. Thus, the poles located farther away from the imaginary axis dominate the response of the system as the gain is increased.

The controller was designed to meet three performance requirements. First, the system should be capable of performing moves with no overshoot. Second, the steady state error should be close to zero. Third, the system should be able to operate at a frequency of up to 1 Hz, which will guarantee that 90 degree bends will be completed in one second[9].

As a first step in designing the controller, a Bode diagram of the open-loop transfer function was constructed (see Figure 71). The Bode diagram suggests that the system may operate satisfactorily at frequencies as high as 6 Hz (~ 40 rad /s). However, the phase margin woul be very low, resulting in an oscillatory response.

[9] In normal operation, the module will complete a full bending cycle in two seconds. Use of a safety margin of two results in a frequency of one hertz.

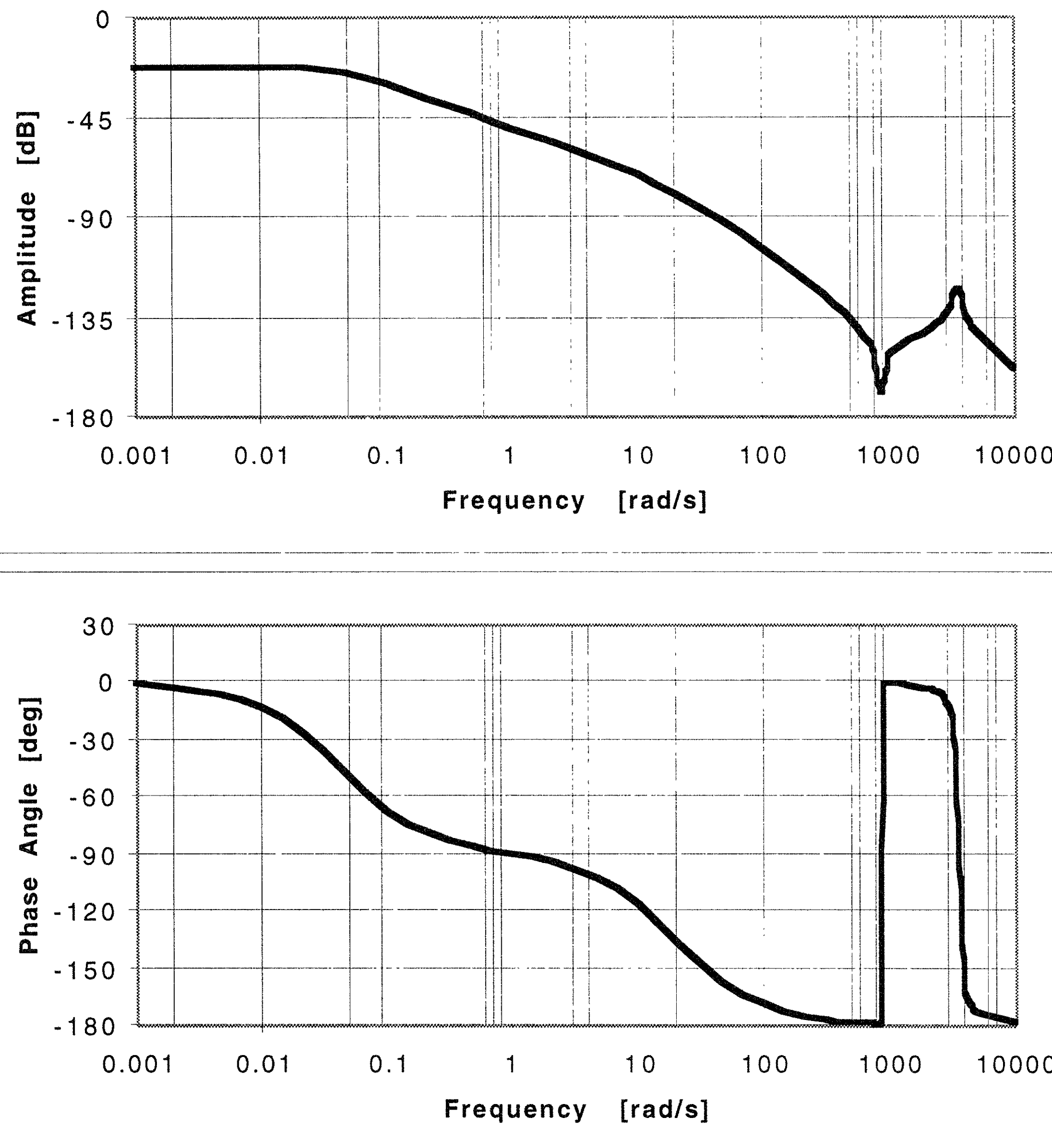


Figure 71: Bode diagram of the open-loop motor displacement transfer function.

This problem is alleviated by adding derivative feedback, and adjusting the corresponding gain until an acceptable phase margin at a frequency of 40 rad/s is obtained. The open-loop Bode plot for the system with derivative control is shown in Figure 72.

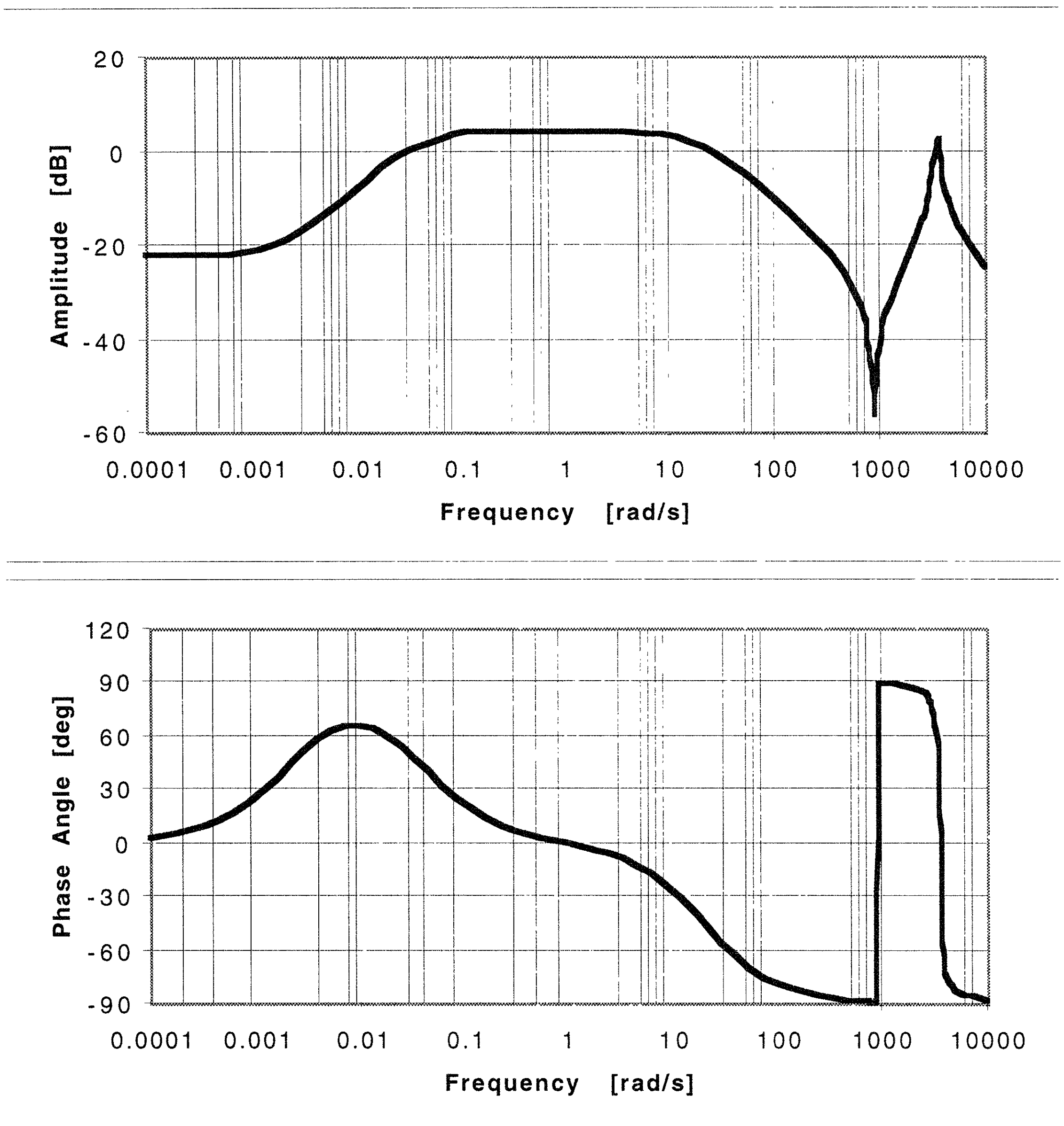


Figure 72: Open-loop Bode diagram of the motor displacement transfer function with derivative control (Kv = 500).

Proportional control, an adequate choice for many position control systems, was then added to close the loop. The proportional gain was set to obtain a crossover frequency of 40 rad/s. The open-loop Bode diagram of the system with proportional and derivative control is shown in Figure 73.

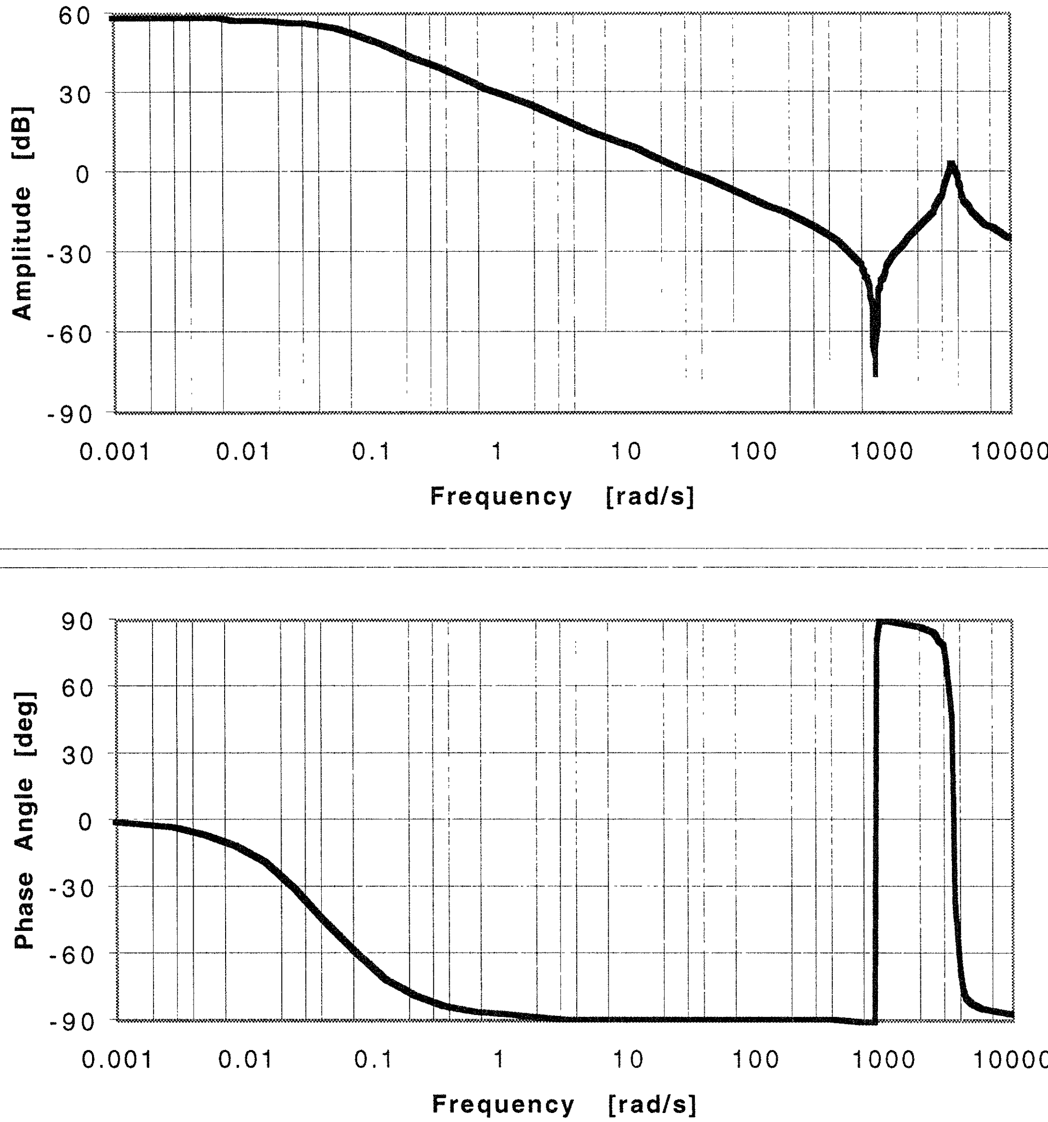


Figure 73: Open-loop Bode diagram for PD control (Kp = 9900, Kv = 500).

The Bode diagram of the closed-loop transfer function was then constructed to check what the theoretical cutoff frequency of the system is (see Figure 74). The plot shows that the cutoff frequency is approximately 5.6 Hz (35 rad /s).

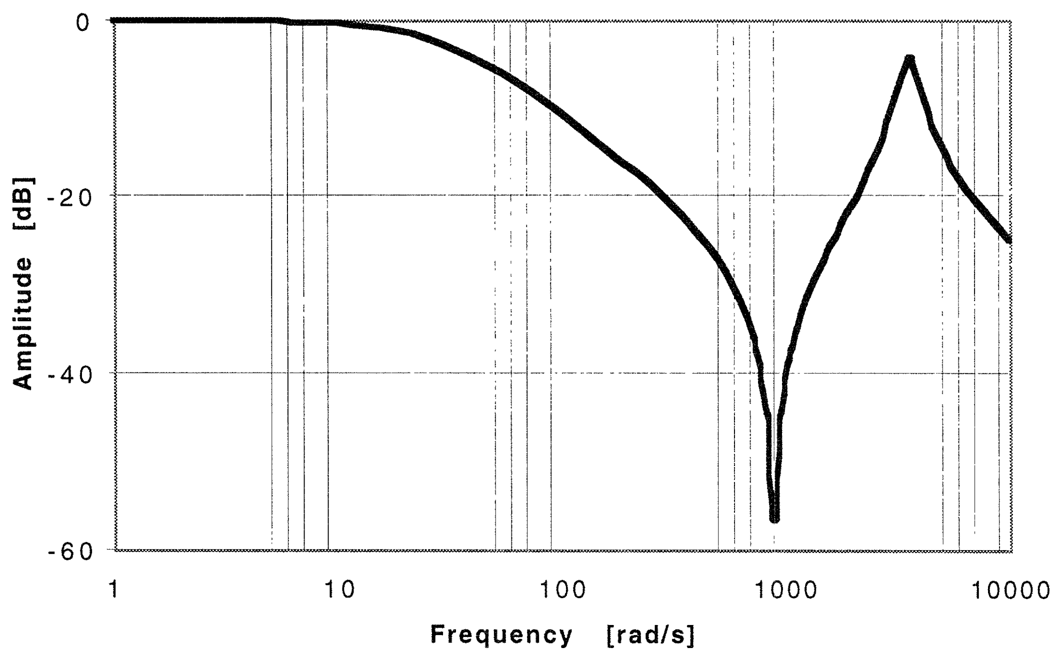


Figure 74: Closed-loop Bode diagram for system with PD control.

Since derivative control worsens the noise rejection characteristics of a system, it was proposed that only proportional control be used in the prototype bending module. As mentioned previously, one of the performance requirements calls for satisfactory performance at a frequency of one hertz. To verify whether such performance requirement could be met, a Bode diagram of the open-loop transfer function with proportional control was constructed. The proportional gain was adjusted to obtain a crossover frequency of one hertz. The Bode diagram for the selected gain (Kp = 2100) is shown in Figure 75. The plot shows that for the chosen gain, the 3 dB bandwidth of the system is approximately 1.6 Hz (10 rad/s), and the phase margin is approximately 60 degrees. Thus, it can be expected that proportional control will produce the necessary response time.

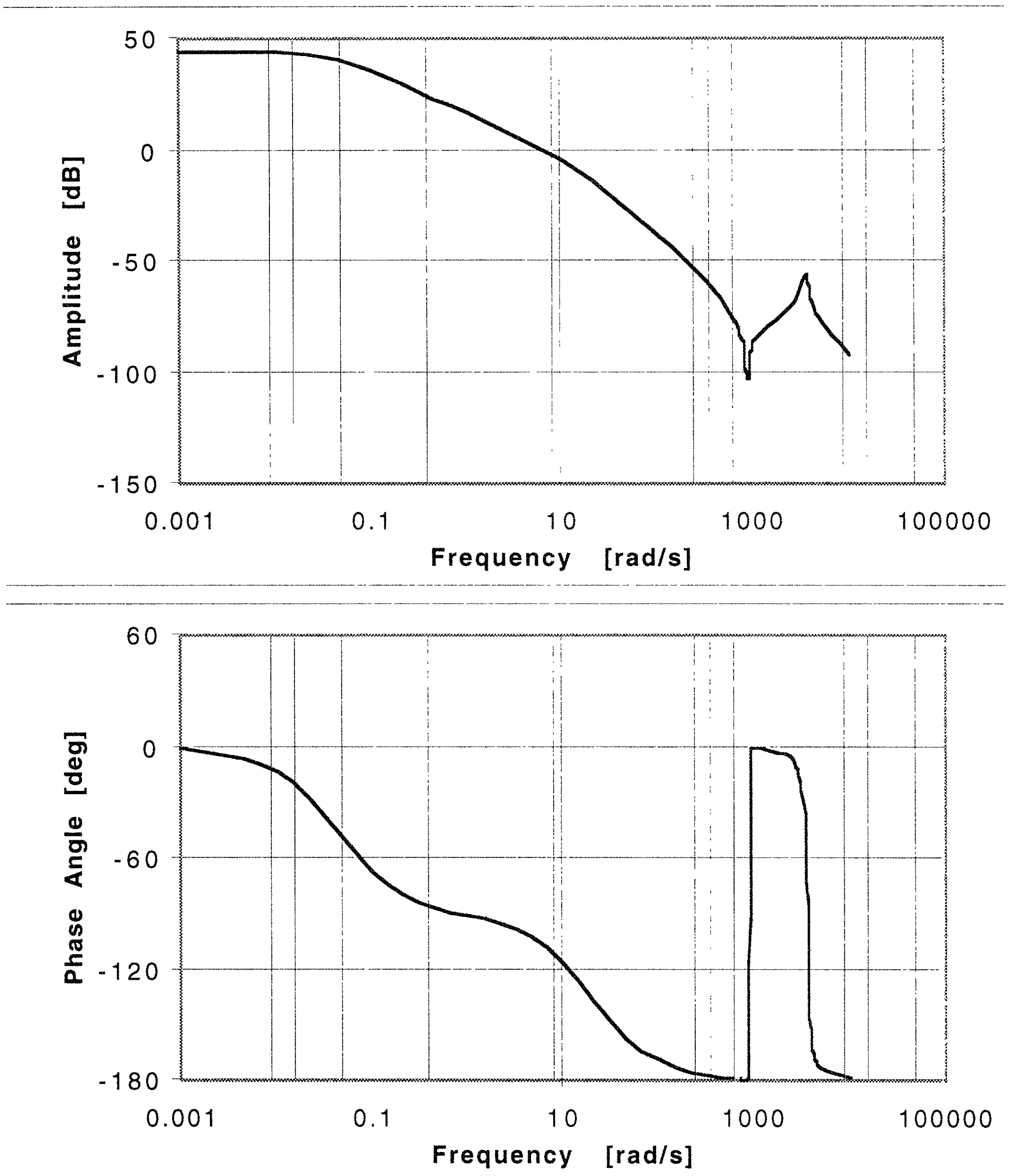


Figure 75: Open-loop Bode diagram of the motor displacement transfer function with Kp = 2100.

To verify that the gain determined from the Bode diagrams would result in satisfactory performance, the step response[10] of the system model was obtained using Matlab/Simulink, a dynamic systems simulation software package (see Figure 76).

[10] The magnitude of the step input was set to 5 degrees to eliminate the possibility that friction effects distorted the prototype response.

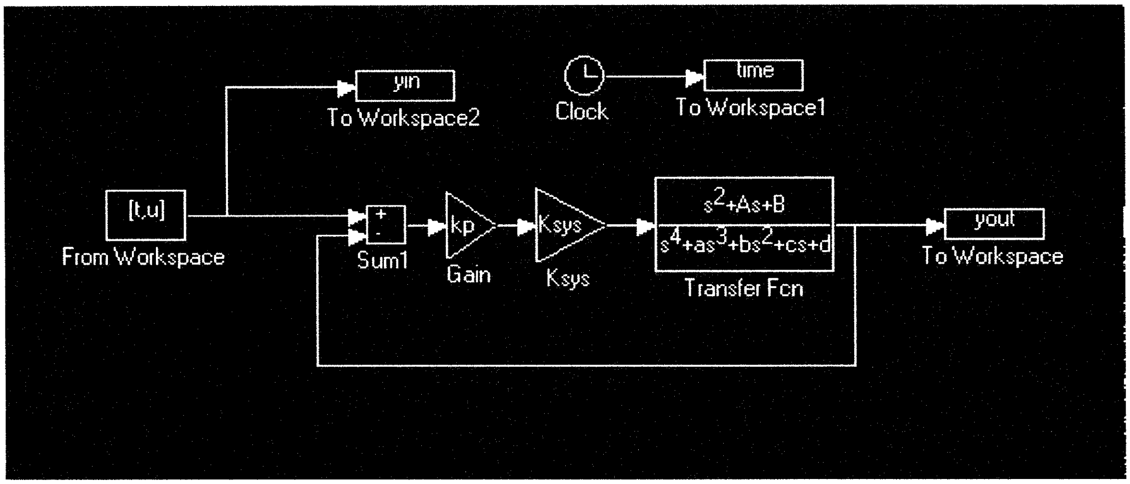


Figure 76: Simulink model of the system.

The proportional gain was scaled to account for the amplifier, controller card, and the motor gains. The model and the bending module step responses are shown in Figure 77. The bending module response settles faster than the simulation model response with proportional control. Such difference may be due to excessive damping in the model. For comparison purposes, the response of the system with PD control is also included in the figure. As expected, the addition of derivative feedback shortens the settling time of the response.

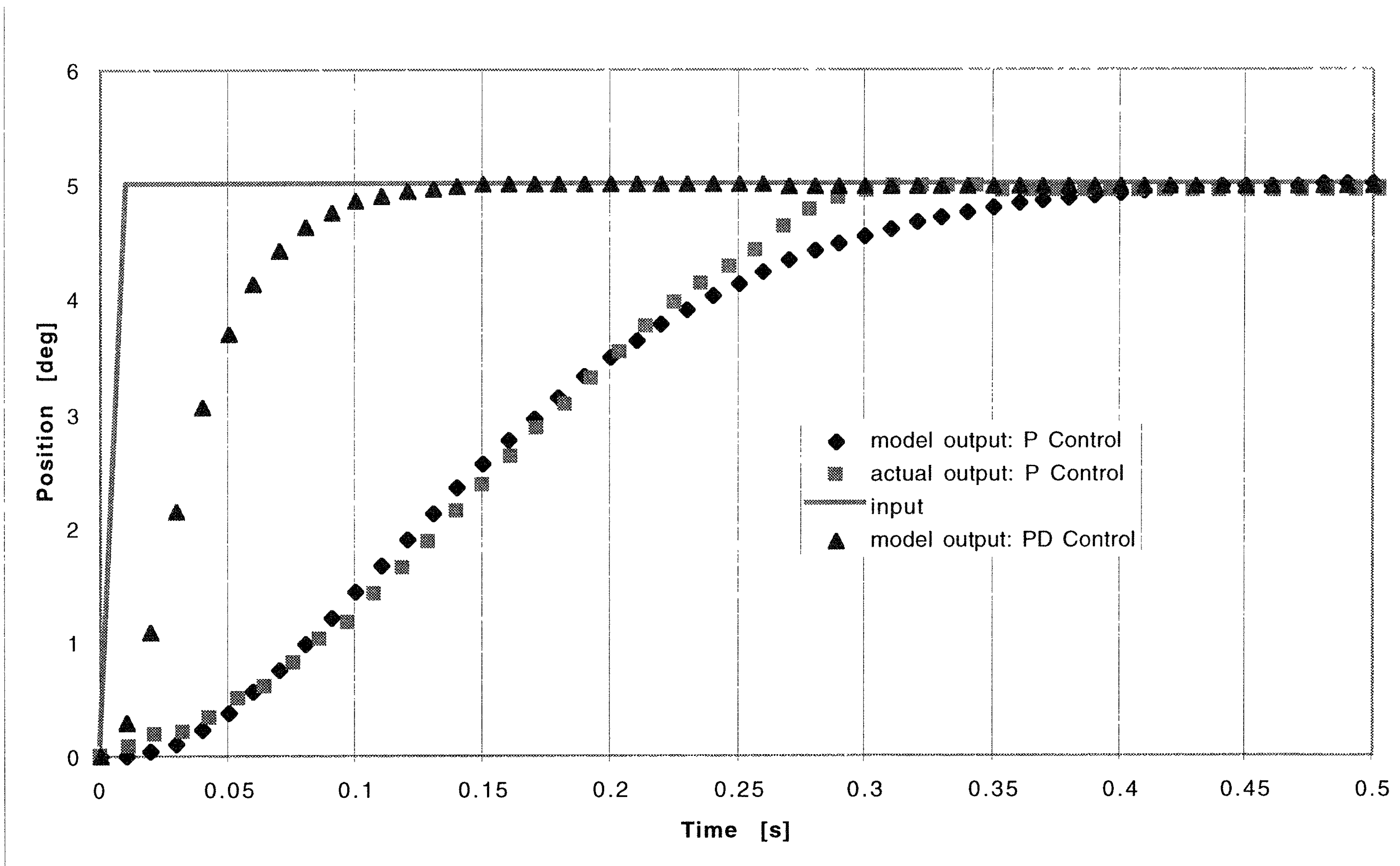


Figure 77: Comparison of theoretical and experimental step responses.

After the proportional gain had been tuned, a 90 degree bending operation was simulated by entering the position input values generated by the TE5650 controller card into the model. The desired time to complete the move was set to one second[11]. The simulations showed that the system should be able to meet the performance specifications, i.e. a 90 degree displacement in approximately one second, solely under proportional control. Armed with this knowledge, the response of the bending module prototype was tested under proportional control. The responses of the model and the bending module are shown in Figure 78.

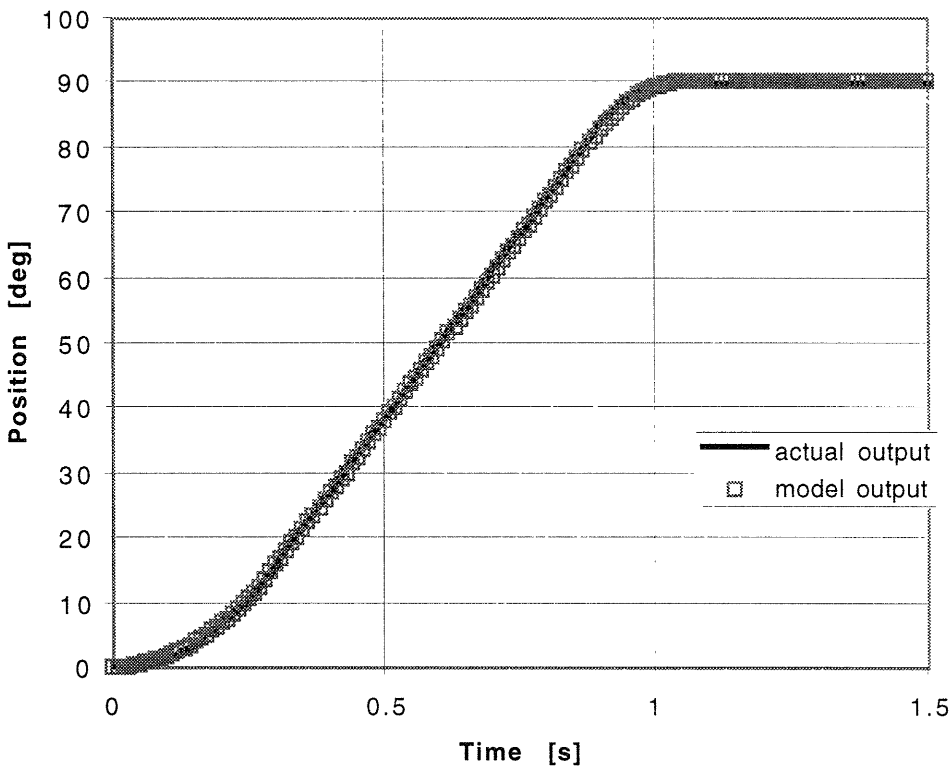


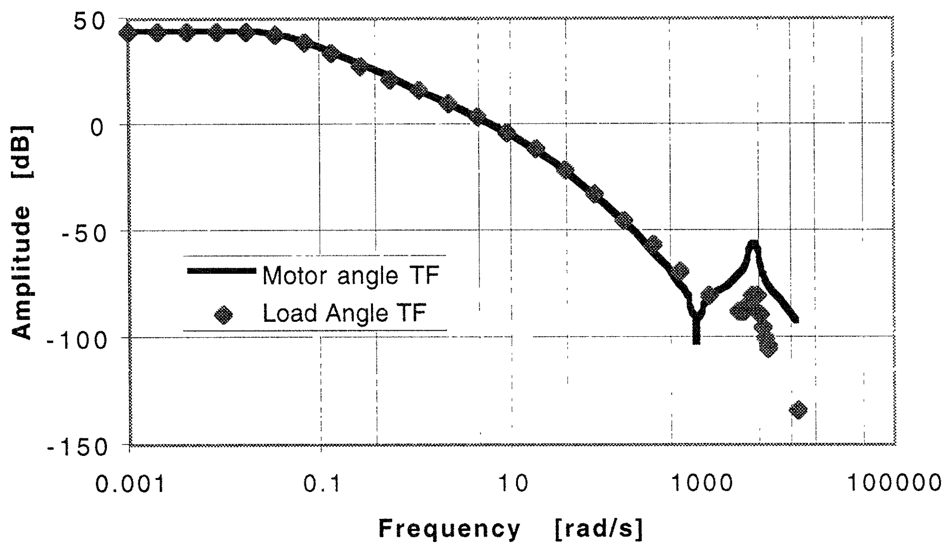
Figure 78: Comparison of theoretical and experimental responses to a 90 degree move command generated by specifying a trapezoidal velocity profile.

[11] Since the card was set to trapezoidal velocity mode, the duration of the move was specified by entering maximum velocity and acceleration values.

The figure above shows that the system meets the performance specifications, namely no overshoot, close to zero steady-state error, and completion of a 90 degree bend in approximately one second, under proportional control.

3.6.3 Comparison of Load and Motor Displacements

As mentioned before, Θ_L may be approximated by Θ_M provided that the static error is negligible, and that there is no introduction of any significant dynamic effects for the range of frequencies and gains at which the bending module will operate. The latter condition was verified by constructing Bode diagrams for the load and motor angle transfer functions after the proportional controller was tuned. The Bode diagrams, displayed in Figure 79, are indeed very similar for frequencies lower than 1.6 Hz (10 rad/s). Thus, at the operating frequency of 1 Hz (3.14 rad/s), the motor displacement is a very good approximation of the load displacement.



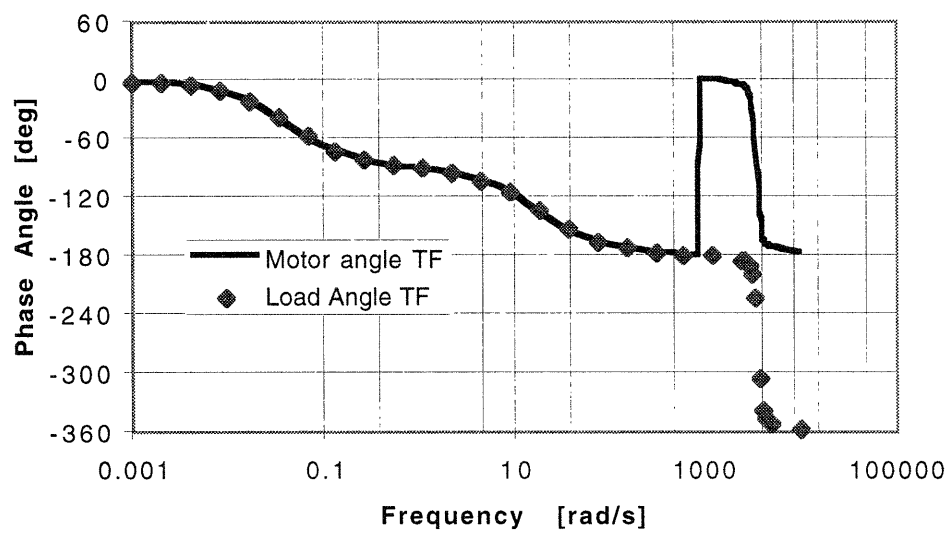


Figure 79: Open-loop Bode diagrams of the load and motor displacement transfer functions with proportional control.

The validity of the approximation was verified further by obtaining the response of the load and motor displacement transfer functions to a step input. As shown in Figure 80, the step responses are virtually identical for the proportional gain at which the system will typically operate. Thus, controlling the motor angular displacement Θ_M in the frequency and gain ranges at which the system will operate is equivalent to controlling Θ_L, with the added benefit that control of Θ_M is easier to implement in practice.

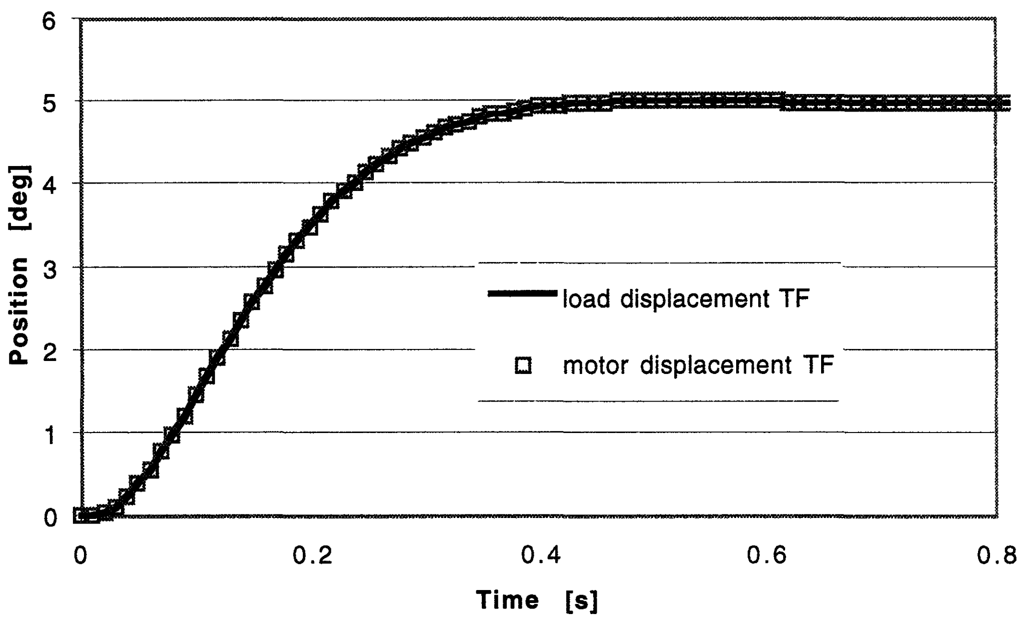


Figure 80: Step responses of the load and motor displacements transfer functions.

Chapter 4 - Prototype Building and Testing

A prototype of the multi-purpose bending module described in Chapter 3 was built to verify the soundness of the design (see Figure 81). Fabrication, assembly, and testing of the prototype helped identify which components of the design should be improved. To test the prototype, the springback measurement and compensation concept described in section 3.6 was implemented. A summary of test results, and a list of recommendations for improvement of the prototype design are included in this chapter.

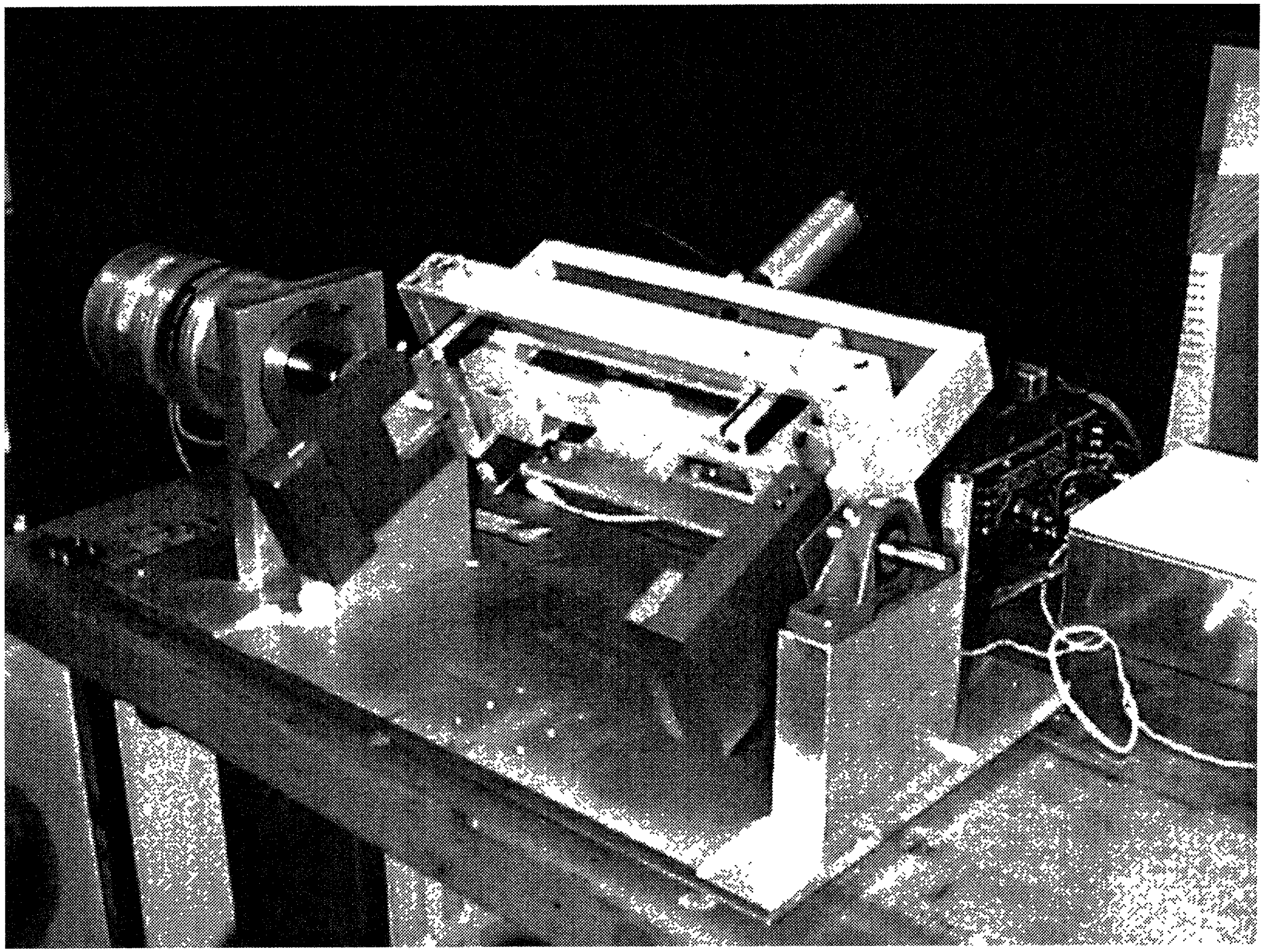


Figure 81: Prototype bending module.

4.1 Results

Experiments similar to those described in section 3.6 were performed to test the prototype. The overbend and circuit on angles were measured with the motor encoder. The measured angle was determined once again by visually projecting the final shape of the parts into a protractor. The measurement error was estimated to be +/- one degree. In the first set, a digital multimeter was used to detect the angle at which the electrical circuit was closed. The results are shown in Figure 82. The average springback, defined as the difference between the overbend and the measured angles, is 25 degrees. The average error, defined as the difference between the circuit-on angle and the measured angle, is approximately -0.87 degrees.

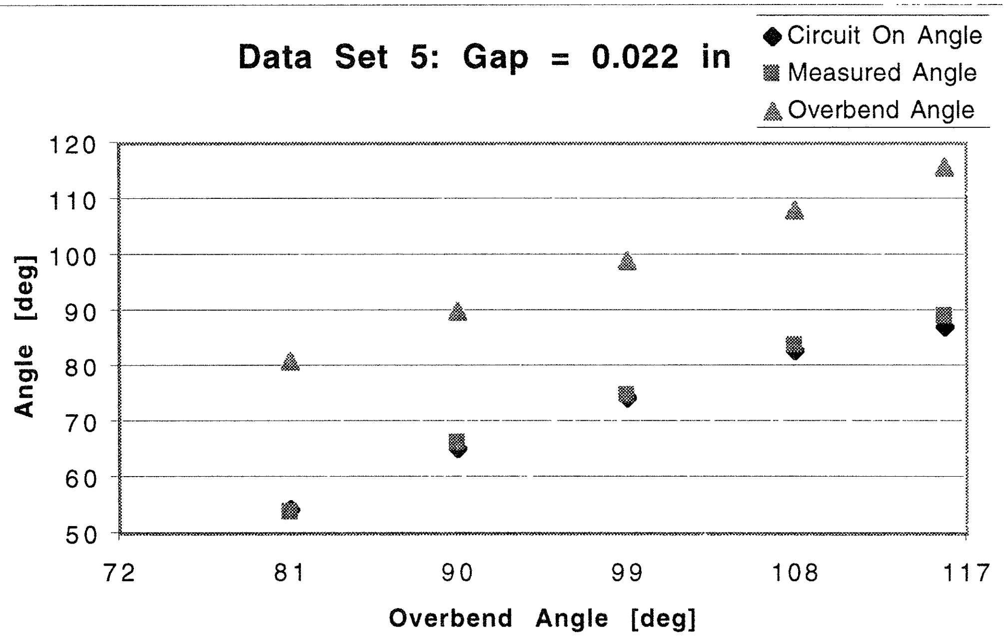


Figure 82: Prototype test results.

Another set of experiments was performed once the controlling software had been fully implemented. Closing of the circuit sent a signal to the computer controller board. The software processed such signal and calculated the Circuit On angle based on the encoder position. The results are shown in Figure 83. The average springback is 26.4 degrees. The average error is 0.84 degrees.

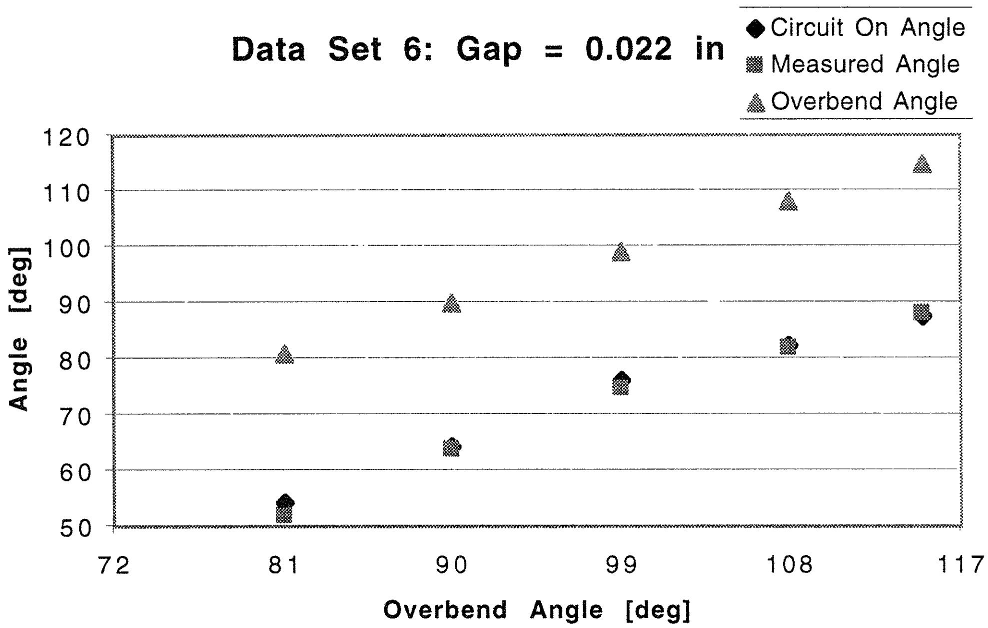


Figure 83: Prototype test results with software control.

A final set of experiments was performed in which the parts were gripped during bending. After the overbend angles was reached, the grippers were opened to a gap of approximately 0.022 inches. The grippers were then rotated back until the circuit was closed. As expected, the resulting springback was lower than for bending done with the grippers open. The results for this set of experiments are shown in Figure 84. The average springback is 23 degrees. The average error is 0.91 degrees.

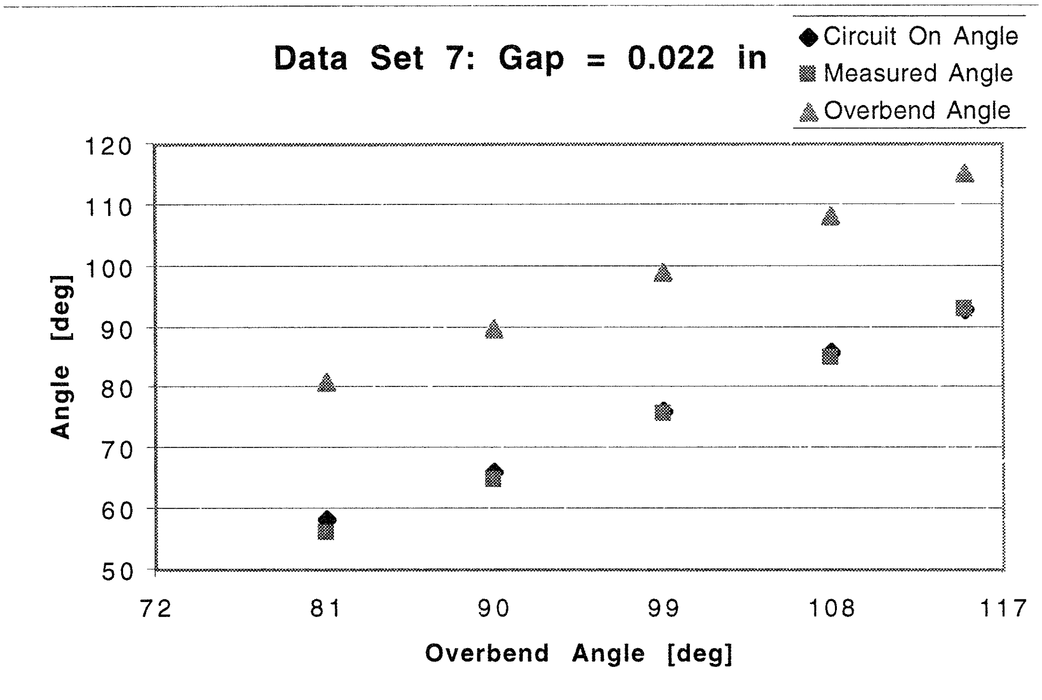


Figure 84: Prototype test results for closed-gripper bending.

4.2 Prototype Design Recommendations

The basic feasibility of the design as a flexible bending module was confirmed through implementation of the springback measurement and compensation concept. The tests showed that the module is capable of bending parts to different angles with no hardware setup. As a next step, the flexibility of the module should be tested by bending parts of different thickness, length, and shape, by changing the number of parts bent each time, and also by performing multiple bends per part once a casing transport module becomes available.

Testing of the prototype also helped identify areas for future work. They include hardware and control software changes. The key areas for future work are described below:

The pillow blocks on which the frame rests should be replaced with custom made bearing housings to facilitate alignment between the motor and frame shafts. Increasing the height of the pillow block mounts (or the custom made housings) will increase the range of motion of the machine. It will also eliminate the possibility of damaging the machine if it gets out of control and hits the base.

If the electrical contact concept is used for springback compensation, a robust way to insulate the grippers electrically must be found. Although the thin adhesive tape used in the tests insulated the gripper effectively, the tape got scratched and peeled off easily. Springback compensation may also be improved by implementing an algorithm into the software that could automatically converge to the desired final bend angle once an initial overbend angle is specified. The speed of the convergence process could be optimized by including a learning loop that would give an estimate of the initial overbend angle based on prior measurements.

A better mechanism to regulate the stroke of the pneumatic piston (and thus the gap between the grippers) should be added to the machine. The cap screw stops used in the prototype work well, but it would be more convenient to have a mechanism that could be adjusted through software. A precise mechanism to adjust the gap between the upper and lower grippers must be developed if it is decided that parts will be bent without being gripped. A possible solution might be to use shims as hard stops for the grippers.

The delrin blocks may not need to be glued to the clamps. In the prototype, they were held in place by washers attached to the front of the clamps. Avoiding the use of glue will facilitate easy replacement of the delrin blocks.

Currently, the home position for the frame is determined by backing it up against a hard stop. The hard stop should be designed so that it can be easily removed out of the way once the module starts operating. Alternatively, the hard stop may be replaced with a limit switch.

References

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Appendix A - Classification of Flexibility

There are several different ways of defining the flexibility of a manufacturing system. A detailed classification [2] includes the following types of flexibility:

- *Machine flexibility:* the ease of making changes required to produce a given set of part types.
- *Process flexibility:* the ability to produce a given set of part types, possibly using diferent materials, in different ways.
- *Product flexibility:* the ability to change over to produce new (set of) products very economically and quickly.
- *Routing flexibility:* the ability to handle breakdowns and to continue producing a given set of part types.
- *Volume flexibility:* the ability to operate profitably at different production volumes.
- *Expansion flexibility:* the ability to expand the system easily and in a modular fashion.
- *Operation flexibility:* the ability to interchange ordering of several operations for each part type.
- *Production flexibility:* the universe of part types that the manufacturing system can produce.

Quantitative measures for each of the different types of flexibility listed above have been proposed [1]. They include a flexibility index, which measures the available number of options and the freedom to select them; a weighted average efficiency, which measures the efficiency with which the system can perform tasks in a refence set; a versality index, defined as the number of times per year that the system can be reconfigured to asemble a new model; discounted cash flow techniques, which estimate the savings achieved by reducing change over and lead times; and a penalty for change measure, which combines measures of the costs required to make a change, and the probability that such change may occur. By selecting the types of flexibility that are relevant to any given case, these quantitative measures may be combined to measure the overall flexibility of an assembly system.

Appendix B - The Design Space

The process through which a flexible bending module was developed can be best described through a 'Design Space' analogy that the author of this thesis has created.

New designs are created to satisfy a set of requirements. Such requirements could be very vague, e.g. 'the device must be flexible', or very specific as in 'the device must be able to exert a downward force of 50 Newtons at a distance of 30 millimeters from the tip'.

After studying the design requirements, the designer can generate some concepts and then define a boundary around the most promising ones. The boundary is created by judging the concepts against the requirements. Concepts which do not meet the requirements are left outside the boundary. This boundary encompasses a 'design space' within which more work is done until a more detailed concept is created.

The analogy is explained in Figure 85.

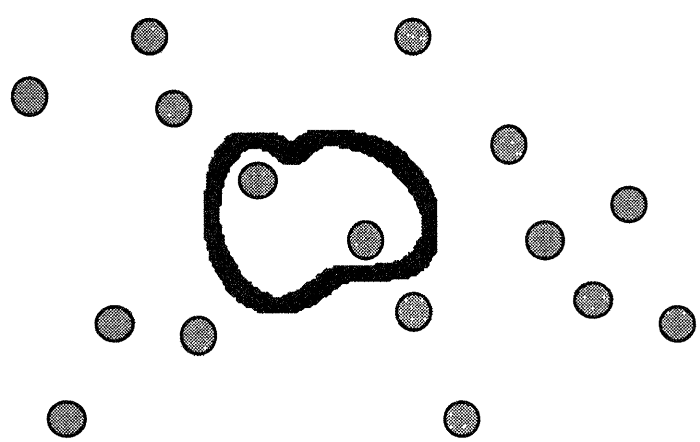

Figure 85: Graphical representation of the Design Space.

The dots represent the concepts generated by the designer. Initially, when there are only a few, relatively vague requirements, a large boundary is drawn around the most promising concepts. As the number of requirements increases, the perimeter of the boundary decreases, leaving out more concepts until only a few remain inside. At the same time, the fuzziness of the boundary is changing. A light boundary indicates that the requirements are vague, while a dark boundary indicates that the requirements are very specific. A light boundary is more permeable to the flow of features between concepts, i.e. features from concepts outside the boundary are more easily incorporated into the concepts inside. Thus, the flexibility of the selected concepts may be maintained by drawing a relatively light boundary around them.

Appendix C - Bending and Product Terminology

Bending Terminology

The terminology used in bending [3] is shown in Figure 86. Note that the bend radius is measured to the inner surface of the bent part.

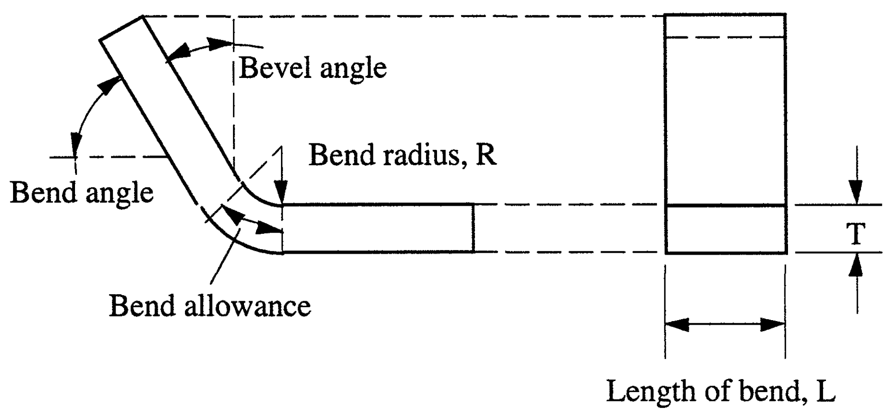

Figure 86: Bending terminology.

Casing Terminology

Casing terminology is shown in Figure 87.

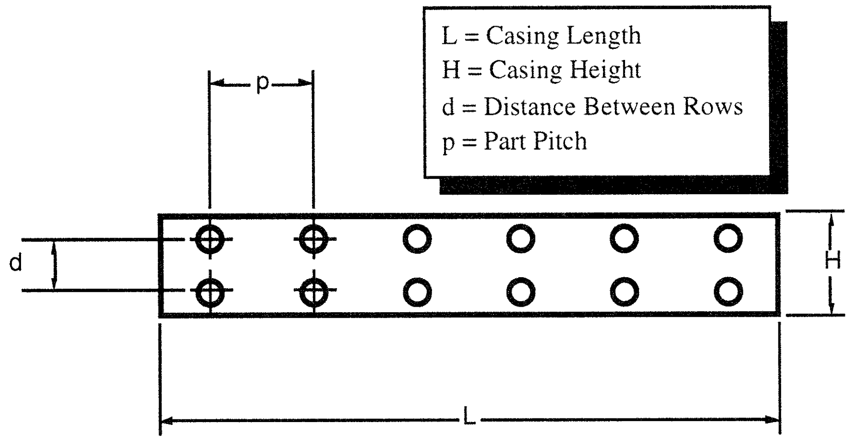

Figure 87: Casing terminology.

Part Terminology

Part terminology is shown in Figure 88.

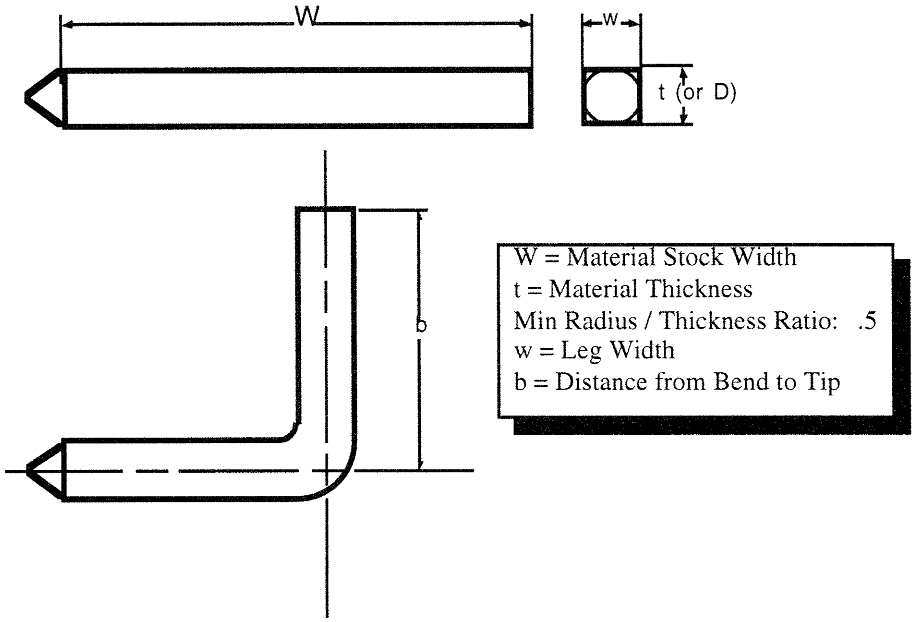

Figure 88: Part terminology.

Appendix D - Finite Element Analysis of Part Bending

Objective: Determine sensitivity of final angle (or springback) to variations of gripper dimensions, part dimensions, and part material properties.
Variables of interest: part diameter, yield strength, tool (gripper) radius

Two models of part bending were analyzed. In the first model, the bend was made by simply constraining the degrees of freedom of specific nodes along the part's length. In the second model, the part was bent using a bending tool with characteristics similar to the grippers that could be used in practice.

Simplified FEA Model

A part with the following characteristics was bent:

Material:	Phosphor bronze wire UNS No. C51000 (1/2 hard)
Diameter:	0.0169 in
Young's modulus:	16E6 psi
Poisson ratio:	0.35

The analysis was performed using ABAQUS FEA software[12]. A part model of 101 nodes and 50 beam elements of circular cross section was generated. The part was modeled as a cantilever beam of length L = 0.894 inches. Plastic behavior of the part material was characterized by specifying two points on the plastic region of the stress and strain curve: zero plastic strain at a stress equal to the yield strength (80,000 psi), and plastic strain equal to one at a stress equal to 120,000 psi.

The part was bent by specifying displacement of the node located at the tip. The tip was rotated around an axis intersecting the plane of the beam at a distance of 0.354 inches from the fixed end of the beam. A 90 degree rotation was accomplished in six steps of 15 degrees. Springback was investigated by programming a seventh step in which all boundary conditions on the tip node were relaxed. The boundary conditions for the seven steps are summarized below:

---

[12] The model was prepared with the help of Dr. Jian Cao, to whom the author is greatly indebted.

Boundary Conditions:

Steps 1 through 6:

Node 1 constrained in directions 1, 2, and 6 (x, y, and Ø)

Node 101's 1, 2 coordinates given by R cos Ø, R sin Ø

where R = 0.894 in - 0.354 in = 0.54 in

Ø = 15, 30, 45, 60, 75, 90

Step 7:

Node 1 constrained in directions 1, 2, and 6 (x, y, and Ø)

Results for simplified model

The resulting shapes of the part for steps 6 (Ø = 90 degrees) and 7 (free shape) are shown in Figure 89.

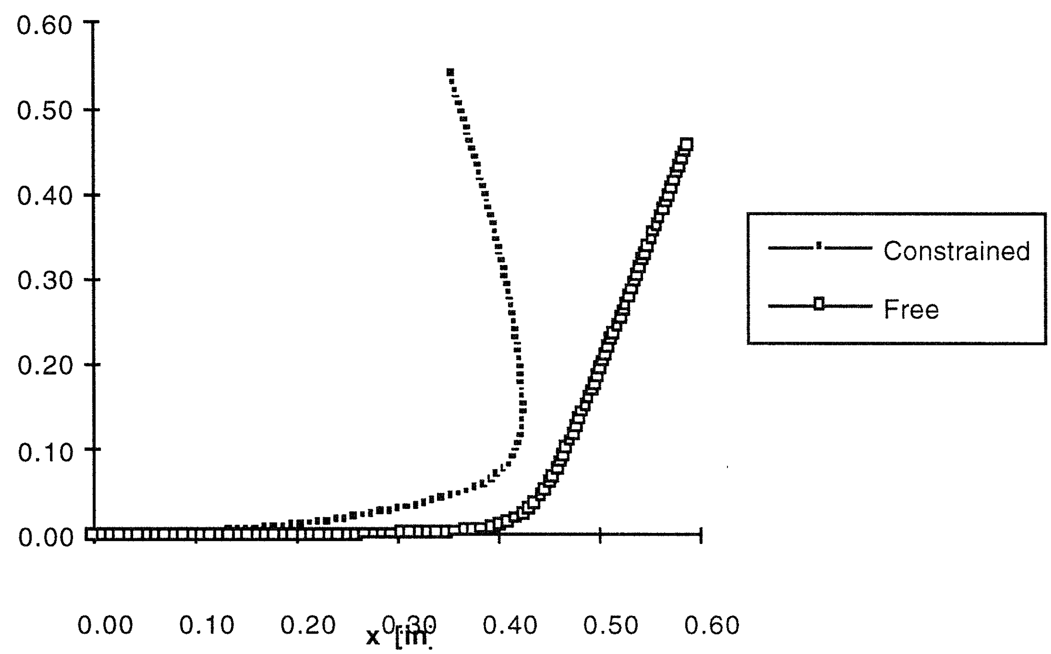

Figure 89: Constrained and free shapes of simple finite element model of bent part.

The figure shows that plastic deformation is restricted to a section starting at Node 35 (0.30 inches from the fixed end) and ending approximately at Node 54. The figure also shows that springback is significant; while the coordinates of the end node specified in step 6 were (0.354, 0.54), the actual final coordinates were (0.59, 0.46).

Complete FEA Model

Since the simplified model did not account for the shape of the bending tool, a second model was developed. The second model consisted of the same part/cantilever beam plus a bending tool, modeled as a rigid surface with the shape shown in Figure 90. The radius of the tip of the bending tool was chosen so that the constrained part had a bend radius equal to the radius given by the part manufacturer specifications (R = 0.030 inches).

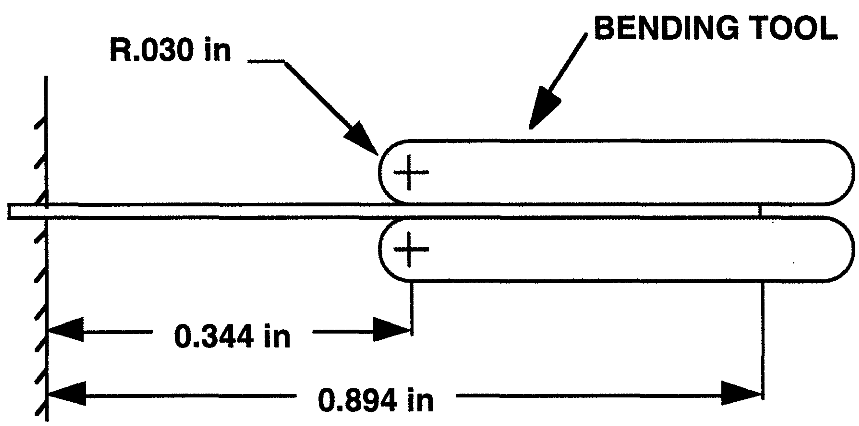

Figure 90: Finite element model of part and bending tool.

Contact between the bending tool and the part was allowed by "attaching" interface elements to the part model. Surface interaction between the bending tool and the part was allowed by specifying a coefficient of friction of 0.3. The part was bent by rotating the gripper about a point on the axis of the part, 0.314 inches from the constrained end, in six steps to an angle of 95 degrees. Then, the gripper was rotated back to 90 degrees in two steps. In the final step, the part was released by moving the gripper vertically upwards.

Results for complete model

1. Sensitivity to part radius: A 10% change in the cross-sectional radius of a part results in a springback variation of approximately 0.6 degrees (see Figure 91).

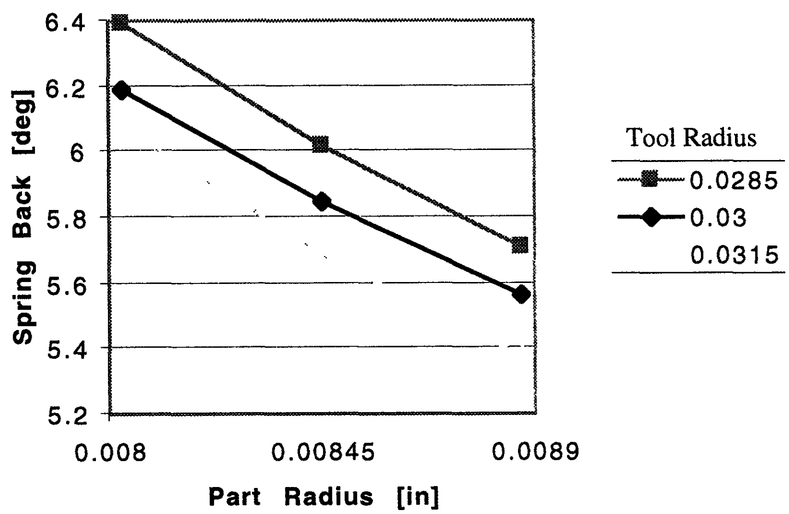

Figure 91: Spring-back vs. part radius for different gripper radii.

2. Sensitivity to yield strength. A 10% change in the yield strength of the material of the part produces a springback variation of approximately 0.3 degrees (see Figure 92).

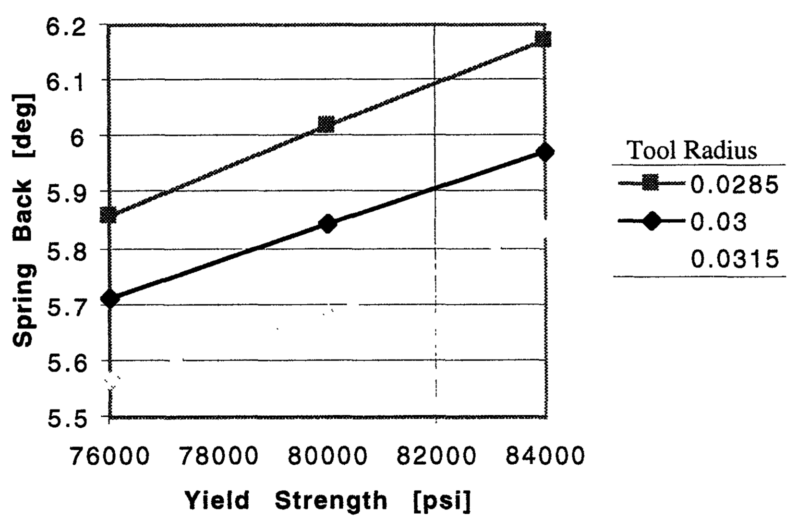

Figure 92: Spring-back vs. yield strength for different gripper radii.

3. Sensitivity to tool radius. A change of 10% in the tip radius of the gripper fingers causes a variation of approximately 0.3 degrees on the springback angle (see Figure 93).

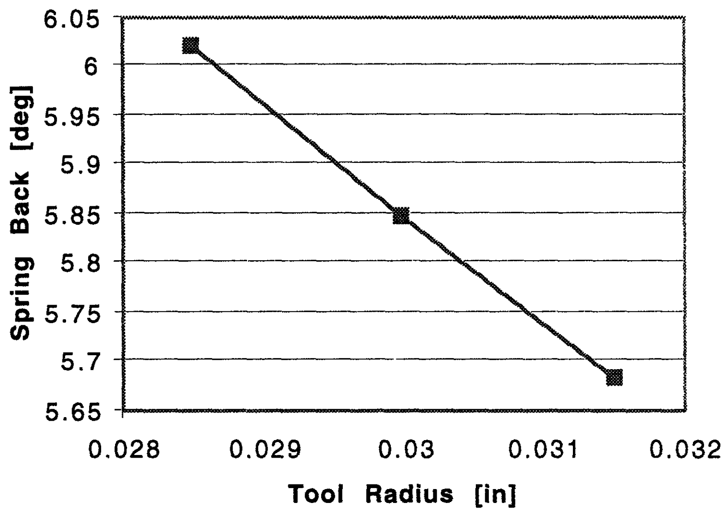

Figure 93: Springback vs. tool radius.

Appendix E - Repeatability Checks

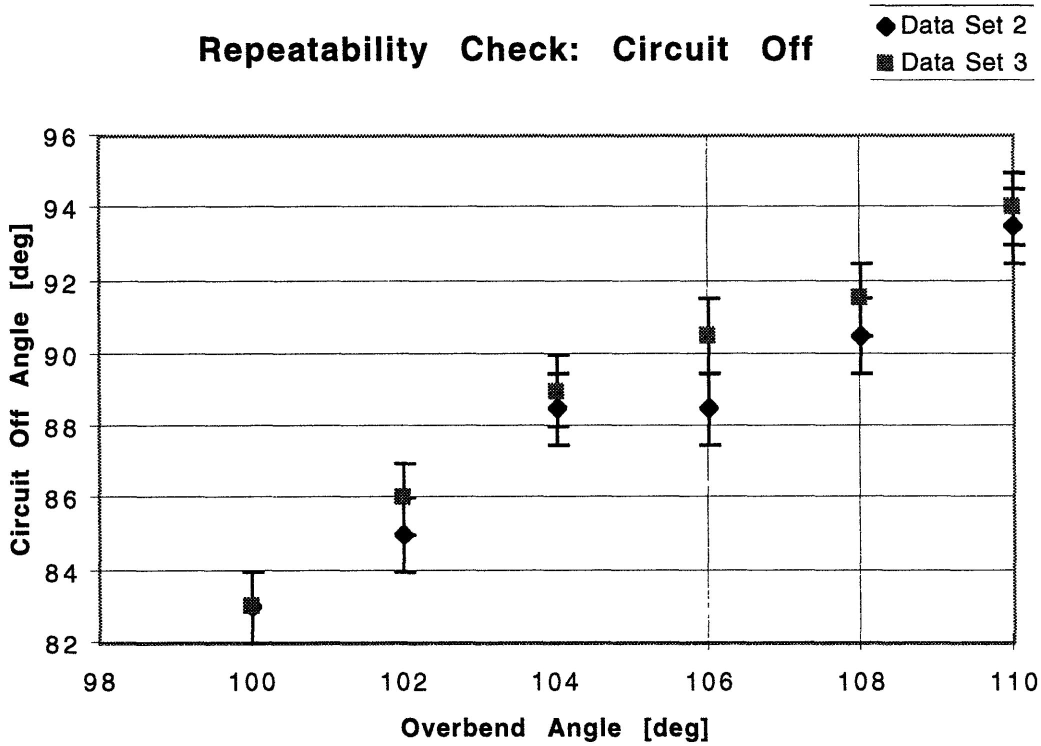

Figure 94: Repeatability check for circuit-off angle.

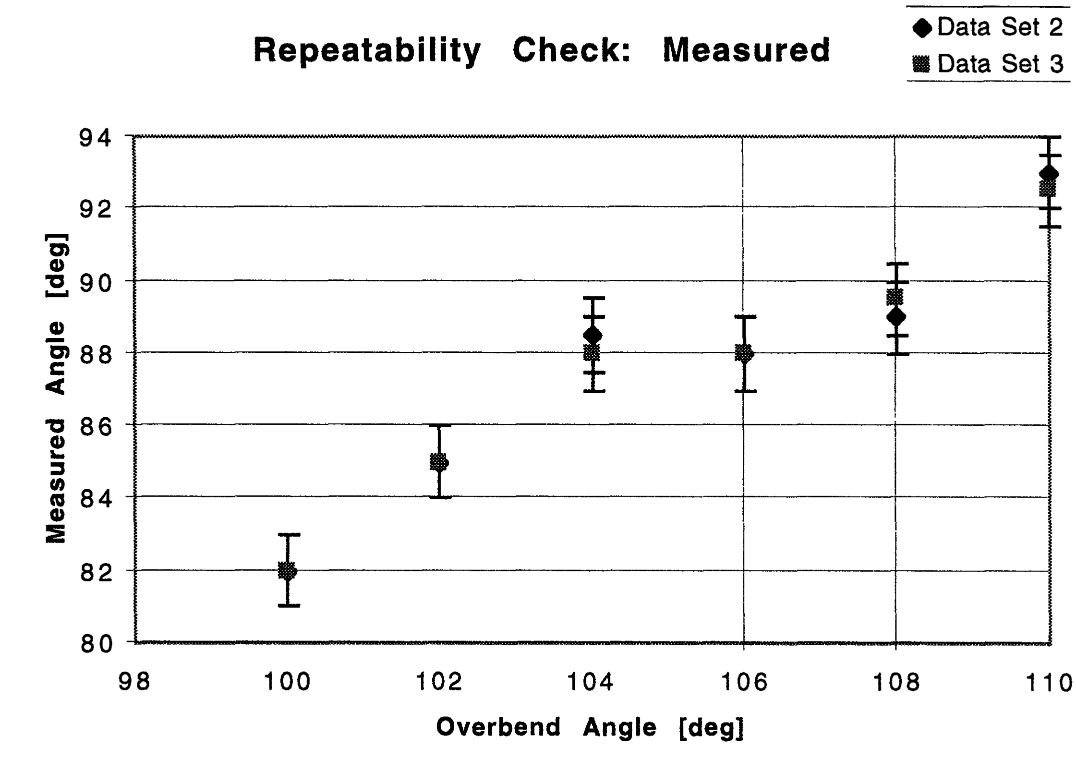

Figure 95: Repeatability check for measured angle.