

The Design of an Automated Fiber Optic Coil Winder

by

Brian E. Sonnichsen

B.S., Mechanical Engineering

University of California, San Diego

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

at the

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Signature of Author
Department of Mechanical Engineering
August 8, 1997

Certified by
Dr. Andre Sharon
Executive Officer of the Manufacturing Institute
Thesis Supervisor

Accepted by.....
Ain A. Sonin
Chairman, Department Committee on Graduate Students

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ABSTRACT

Fiber Optic Gyroscopes (FOGs) have become a real alternative to the mechanical variety in recent years. FOGs rely on light passing through an accurately wound coil of optical fiber to sense rotation. The winding of the coil is a difficult process as coils must be wound in an orthocyclic manner using a specific pattern and can contain very few errors in the winding. Such coils are currently wound on semiautomatic machines with the constant attention of an operator.

This document describes the development of an automated fiber optic coil winding machine was developed for an outside customer with the goal of reducing winding times and costs and increasing coil quality. From a set of specifications set forth by the customer, a test-bed machine was designed and fabricated. Many issues relating to the process of coil winding were researched using this test-bed. When testing was complete, an automated winder was designed, fabricated, and tested. A model of the main drive system was developed and a control algorithm created. The actual response of the main drive was compared against the response predicted by the model.

The machine described in this document could successfully wind a fiber optic sensing coil automatically with very little operator intervention. Preliminary results indicated that the coils wound on the machine were of higher quality than traditionally wound coils. Winding times on the machine were also much shorter than winding times on conventional systems.

Thesis Supervisor:
Dr. Andre Sharon, Executive Officer of the Manufacturing Institute

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Chapter 1 - Introduction

1.1.1 What is a gyroscope

A gyroscope (gyro) is a sensor that is used to detect rotation. It finds use in a wide range of applications but is most frequently used in navigation. From submarines to single-engine planes to satellites, gyroscopes make it possible for vehicles to accurately and reliably chart out their courses, either automatically or with the aid of an operator. Gyroscopes perform operations as simple as telling a plane which way is up or as complex as telling a satellite in near-zero gravity if it has deviated a thousandth of a degree off course.

One of the most recent applications of the gyroscope is associated with the Global Positioning System or GPS¹. Cars and other vehicles can use GPS to find their position on the globe. With the aid of a computer, the locational information from GPS can be used to track a car's progress and help a driver chart a course. Communication with satellites is not without problems: it is expensive and doesn't work well when obscured by obstacles, such as mountains or tall buildings. By using a gyroscope and a simple computer instead of satellite communication for locational information, a vehicle can navigate in almost any set of conditions. The problem to date has been that gyroscopes have been too expensive to find use in consumer automobiles. New technology has made it possible to reduce cost of gyroscopes, enabling their use in GPS. A gyro would sense any turns made by the vehicle and a computer would keep track of the distances and directions traveled. By referencing this

information to known maps and knowing a reference starting position, the computer can locate the vehicle on a map contained in memory.

Traditionally, gyroscopes have been mechanically based devices comprised of high precision parts that spin and then twist in response to rotations of the gyroscope². Exactly how these parts turn inside the sensor is well defined by modern dynamic theory. Essentially all mechanical gyroscopes operate on the same basic principle: conservation of angular momentum of a spinning mass. For a discussion of the dynamic theory of a basic mechanical gyroscope, see Crandall³. By measuring torques and forces generated as certain internal parts of a gyroscope move, the rotational motion of the entire gyroscope can be deduced. Different gyroscopes accomplish these measurements in different ways, depending on the application and accuracy grade of the sensor, but all give a measurement of the rotation of the sensor.

There are two significant disadvantages to using mechanical gyros. The first is that they are relatively expensive to produce, especially the highest accuracy varieties. The high precision components and accurately machined parts used inside these gyros are inherently expensive to produce². Though great effort has been expended to cost-reduce these parts, the parts are still difficult to manufacture and remain an impedance to cost-reducing mechanical gyros. In addition, the electronics and sensors in the gyros add expense due to their high accuracy and low noise requirements. All parties interested would like to see a less expensive gyroscope, but the current cost of the components of the mechanical sensor make this virtually impossible.

Another drawback of mechanical gyros is that they contain moving parts which are particularly subject to failure over time. Eventually, a bearing may wear out, a part of the structure may fatigue and break, or something could shift out of alignment. Either way, the gyroscope would become useless and would have to be replaced. On vehicles such as satellites, this is not easy and could lead to the destruction of the satellite. If moving parts could be eliminated altogether, these problems could obviously be avoided.

Optical solid state technology has been shown in the last few years to be a viable substitute to the traditional mechanical technology used in gyroscopes. These solid state gyros incorporate no moving parts, relying on light traveling through an optical pathway to detect rotation. There are several ways that the optical pathway can be created. Ring Laser Gyros (RLGs) and Fiber Optic Gyros (FOGs) use two of the possible methods. As suggested by its name, the RLG uses a set of accurate mirrors arranged in a ring around which the light can travel. The FOG uses optical fiber to define the light path. For reasons of greater possible accuracy and lower cost, the FOG seems to be the most promising alternative¹.

FOGs can reach, and possibly exceed the precision of current mechanical gyros and can operate under equal, or even harsher environmental conditions. The drawback to date has been the expense of building these solid state gyros. Only recently has the state of technology begun to progress enough to bring the cost of manufacturing the gyros down to competitive levels. Optical fiber has become less expensive and methods of interfacing the fiber with other

components in the gyro have become well defined. Likewise, the lasers, which act as the light source for the sensor, have recently become far more affordable.

The main factor currently driving the price of manufacturing solid state gyroscopes is assembly time⁴. It can take one person up to three weeks to assemble a working, three-axis gyroscope. If this assembly time can be reduced, through automation and other techniques, the cost of a solid state gyro can be brought significantly below that of a mechanical one. Given that they have no moving parts and could be made inexpensively, the solid state gyroscope could prove to be a superior alternative to its mechanical cousin.

1.1.2 How a fiber optic gyro works

The basic theory behind the FOG is fairly simple. Consider a long strand of optical fiber that is wrapped into a coil (Figure 1). The fiber is wound from its midpoint outwards so that half the fiber is wound counterclockwise around the coil and

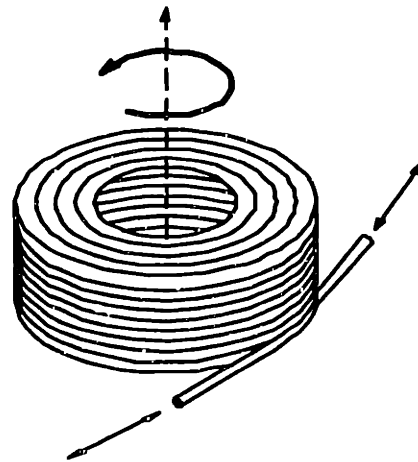


Figure 1. A representation of a FOG coil.

the other clockwise. The two ends of the fiber will then be located on the outside of the coil as shown in the figure. Light that is in phase is passed through both ends of the fiber coil. If the coil remains stationary, light coming out of the two ends will be in phase. However, if

the coil turns along its axis, as indicated in figure 1, light traveling through the coil in the direction of the rotation will take longer to travel the fiber length than the light traveling against the direction of rotation⁵.

A physical example may prove useful in understanding this concept. Consider two balls that are rolled along a plank of a given length from opposite ends, towards each other at equal speeds as shown in Figure 2. The balls are rolled along the two edges of the plank so that they don't collide. If the plank is held stationary, the balls will take the same amount of time to reach the ends of the plank. If, on the other hand, the plank is moved towards one of its ends while the balls are moving, the ball traveling in that direction will take longer to reach the end than the ball traveling in the opposing direction. The time difference between when the two balls reach the ends of the plank is related to the velocity of the plank.

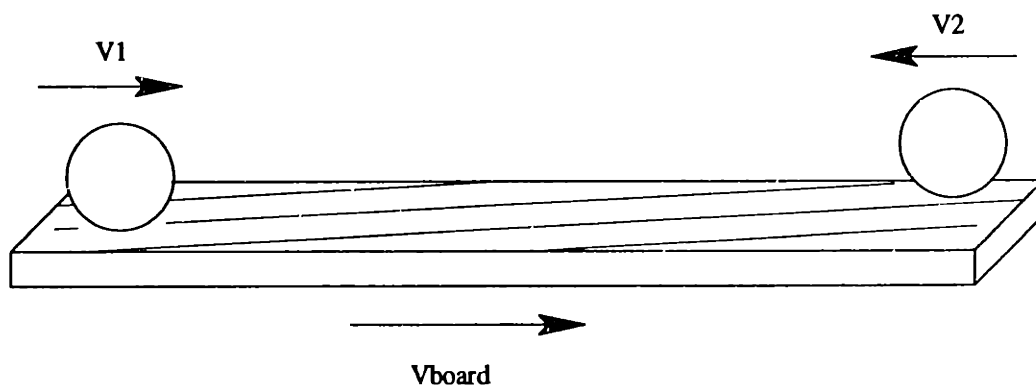


Figure 2. Balls rolling relative to a plank

This is a similar concept to that employed in the fiber optic gyroscope except that light takes the place of the balls and optical fiber takes the place of the plank. The primary conceptual difference is that the fiber is wound into a coil

so that rotation rather than translation is measured. The light emerging from opposite ends of the coil can be recombined and measured for phase shift. If the coil has experienced rotation, the light traveling in opposite direction will take different times to traverse the length of the coil and will be shifted in phase when they emerge. This shift is directly related to the rotation rate of the coil.

In 1914, a French researcher by the name of Sagnac first proposed using this concept of phase shift for measurement⁶. He created a simple device consisting of bright light, a 45° beam splitter, and a ring of mirrors. With this

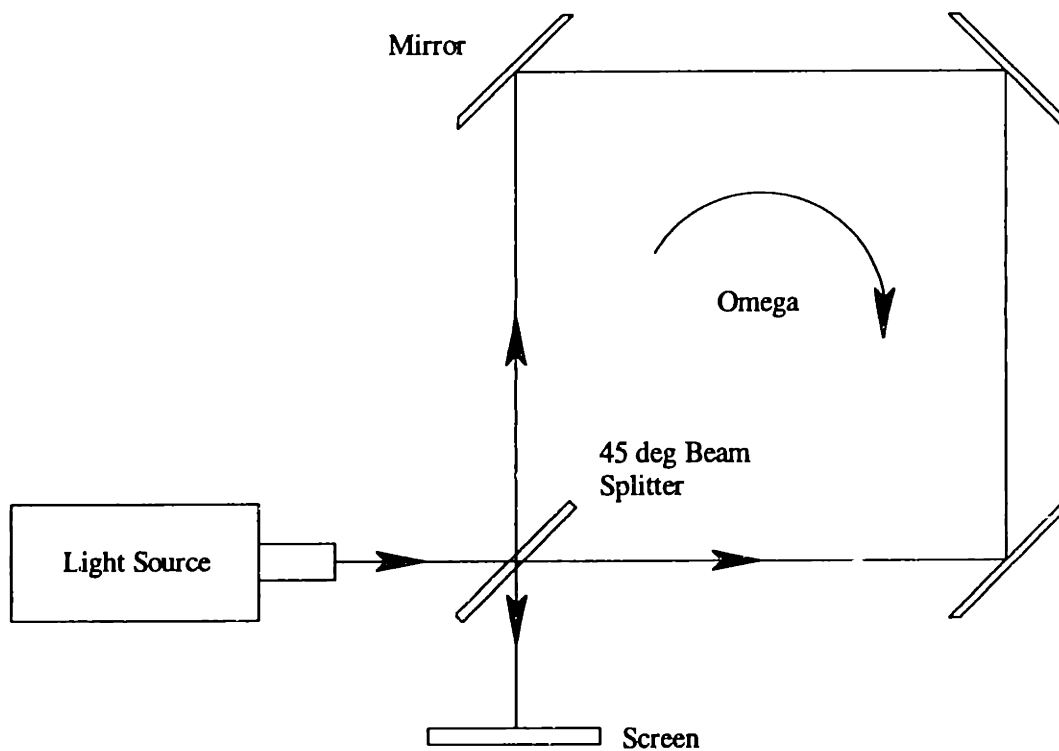


Figure 3. The Sagnac Interferometer

device, he was able to successfully prove the above described concept, which is now commonly known as the Sagnac Effect. His device, called a Sagnac interferometer is shown in Figure 3. The entire device was rotated at a velocity

Omega and the combined light was shown onto a screen. A fringe pattern was seen on the screen which corresponded directly to the rotation Omega. About sixty years later, in 1976, Vali and Shorthill proposed and implemented the first optical fiber gyroscope⁷. Since then, the technology has grown to the point where FOGs are as viable as their mechanical counterparts.

A few, simple mathematical relations are enough to entirely describe the Sagnac effect and relate the relative phase shift of the light to the rotation rate of the coil. When the light beams emerging from the coil are recombined, the beams will interfere, and there will be a power loss corresponding to the degree of phase shift between the beams. This relationship is described by the following equation:

$$P = \frac{1}{2}P_0(1 + \cos \Delta\phi) \quad (1)$$

where P is the detected output power, P₀ is the power input to the coil, and Δφ is the phase difference. By measuring the power output with a photodetector, and knowing the power input, the phase shift can be deduced. The rotation rate is related to the phase shift of the coil through the following relation:

$$\Omega = \frac{2\pi LD}{\lambda_0 c} \Delta\phi \quad (2)$$

where λ₀ and c are the free-space wavelength and velocity of light, respectively, L is the length of fiber, and D is the diameter of the coil. Δφ is the phase shift, as per equation 1, and Ω is the rotation rate of the coil. Thus, from a measurement of the power output from the recombined beams that have passed

through the sensing coil, a measure of the rate of rotation of the coil can be made. The majority of fiber optic gyroscopes use this basic concept in sensing rotation.

1.1.3 The components of a FOG

A diagram of a FOG is presented below.

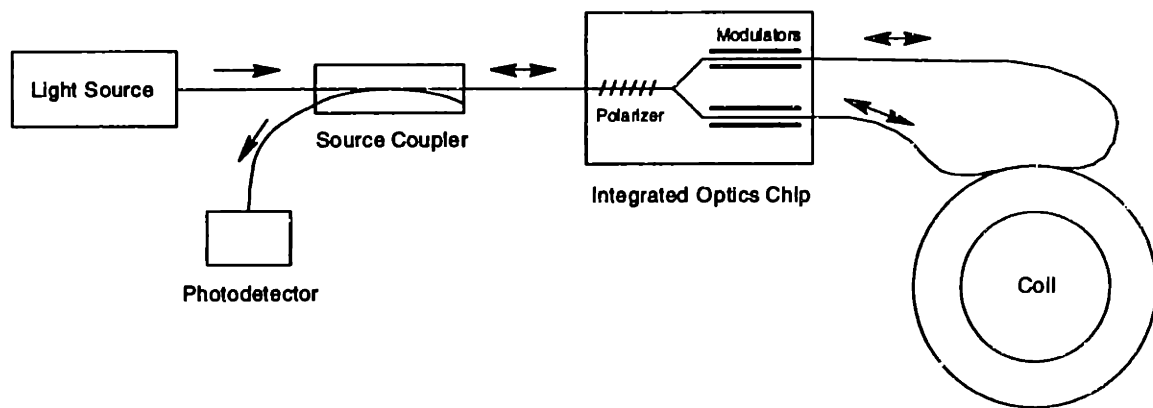


Figure 4. Diagram of Fiber Optic Gyroscope.

Each of the major components of the gyro can be seen in this illustration. These components are the light source, source coupler, integrated optics chip (IOC), coil, and photodetector². Each of these components will be discussed below.

1.1.4 The Light Source

The laser is the light source for the gyro. It provides a constant, coherent beam of light that drives the Sagnac Effect. Laser Diodes are commonly used in FOGs as they are small and exhibit adequate performance. To meet power

requirements, these diodes are coupled with a rare-earth doped fiber that acts to amplify the beam. Together, the diode and doped fiber act to generate the light beam for the gyro.

1.1.5 The Source Coupler

The source coupler serves the basic function of routing the returning light beam from the coil to the photodetector. The coupler also allows for light to travel from the light source towards the coil. In order to accomplish this, two fibers are fused together along their sides and are reinforced for durability. A fiber lead coming from the laser attaches to the end of one of the fibers and the lead from the coil is attached to the other end of the same fiber. The photodetector lead is attached to the end of the other fiber in the coupler. Finally, the last coupler fiber end is left unattached.

1.1.6 The IOC

In order to split the light beam as it enters the coil, a special beam splitter must be used. This splitter also serves the purpose of recombining the two light beams emerging from the coil. While there are various forms of splitters available, a special type called an Integrated Optics Chip is commonly used in gyroscopes. Light pathways are formed within a specially designed chip to efficiently guide the light between the single fiber and the two coil fiber ends. The IOC looks much like a rectangular piece of glass with the fibers attaching to the shorter ends of the rectangle. Where the IOC becomes unique from other beam splitters, is in the phase modulator contained within the IOC. The

modulator acts to modulate the frequency of the light beam which aids in the operation of the gyro and significantly increases their accuracy. The IOC also acts to polarize the light beams which also increases operating efficiency.¹¹

1.1.7 The Photodetector

In order to detect the phase shift in the light emerging from the coil, a two different types of photodetectors are commonly used. Both the PIN and Avalanche Photodiodes operate on the principle of photons of light creating free electrons upon contact with a specially designed surface. The surface is connected to an electrical circuit and the current is measured. By the nature of the interaction, current is proportional to light intensity; thus, providing a measure of the phase shift in the light beams. For various reasons, the PIN variety is the most common in FOGs.¹¹

1.1.8 The Coil

The fiber optic coil has the most bearing on this thesis of all the FOG components as the goal of the project was to build a coil winding machine. Therefore, the many unique features of the coil will be covered in much more detail than the other components. Many of the references cited above thoroughly discuss the other FOG components.

Among the many factors contributing to FOG performance, the quality of the sensing coil is one of the most important. The Sagnac Effect relies on the fact that the shift in the phase of the light beams is due solely to the rotation of the coil. However, there are other factors that can create this shift. These factors act to change the local index of refraction of the optical fiber. Changes in the index of refraction will locally change the velocity of light in the fiber. And, changes in the velocity of light could lead to an apparent shift in phase of the two propagating beams. Temperature variation and stress are the two most significant factors that can create these local changes in index of refraction. Coils are rigorously tested under thermal variations and severe vibration when determining the performance of the coils. A common measurement called drift, usually expressed in degrees per hour, corresponds to the rotation sensing error over time. The best coils have better than .0005 deg/hr of drift, termed strategic grade, the next best have .05 deg/hr, termed navigation grade, and the crudest coils have 1 deg/hr, termed tactical grade. With current technology, tactical grade coils are quite easy to wind². Navigation grade coils are commonly produced but at high costs and at low production rates. Strategic grade coils can be wound but require a great deal of care and are highly labor intensive. To be used for navigation applications, the gyros must be of navigation grade or better to avoid excess error in the sensor readings.

Different approaches can be taken to deal with these environmental variations. The coil could be carefully insulated and thus protected from thermal and vibration effects. This approach has been tested and works quite well. However, this type of insulation is bulky, and it is desirable to keep the gyroscope as small as possible in most applications. With this in mind, the problem then must be solved by altering the winding of the coil. The approach has been to create patterns of winding that achieve maximum symmetry within the coil. What is meant by symmetry in this case is that points in the fiber equidistant from its midpoint are located adjacent to each other within the coil. By doing this, the light traveling in opposite directions will see the same variations at the same time as it passes through the fiber. Hence, any change experienced by one of the beams should be virtually identical to that of the other, and when recombined, the two propagating light beams should only reflect phase shifts due to rotation.⁸ Figure 5 provides a clarification of this concept. Consider that the fiber is only folded over once, as shown, rather than wound

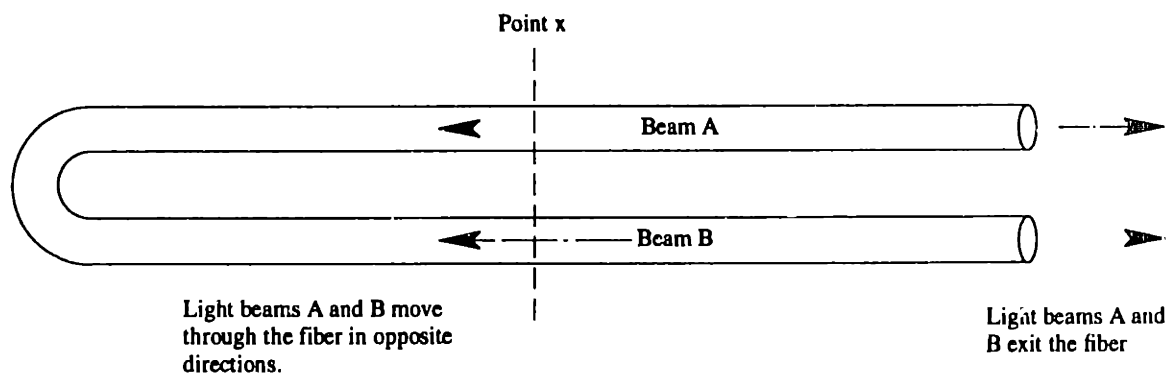


Figure 5. Demonstration of symmetry concept.

into a coil. If a temperature or vibration disturbance occurred at point x, light beams A and B traveling in both directions would experience this disturbance at the same time and at the same distance from each end of the fiber. Thus, the light beams emerging from the two ends of the fiber should exhibit the same phase shift caused by the disturbance. The two beams would therefore emerge in phase.

To wind the fiber with such symmetry, the coil must actually be wound from the inside out. More specifically, winding must start with the midpoint of the fiber and proceed outwards towards the fiber ends. Half the fiber will be wound clockwise around the coil and the other half counterclockwise. This will lead to a coil that has the fiber ends exiting the

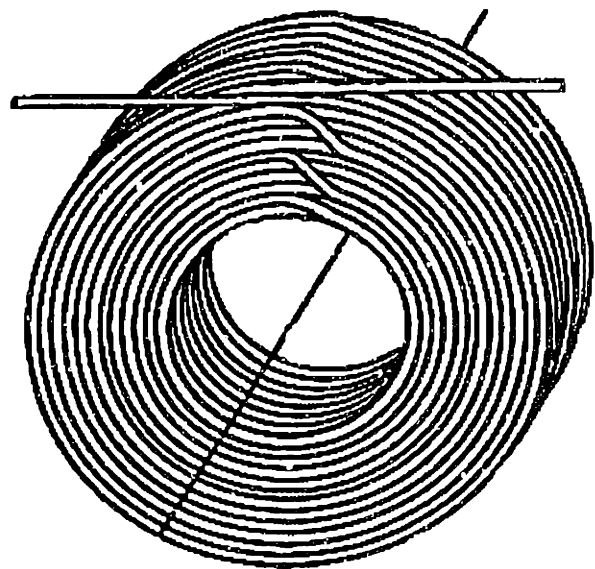
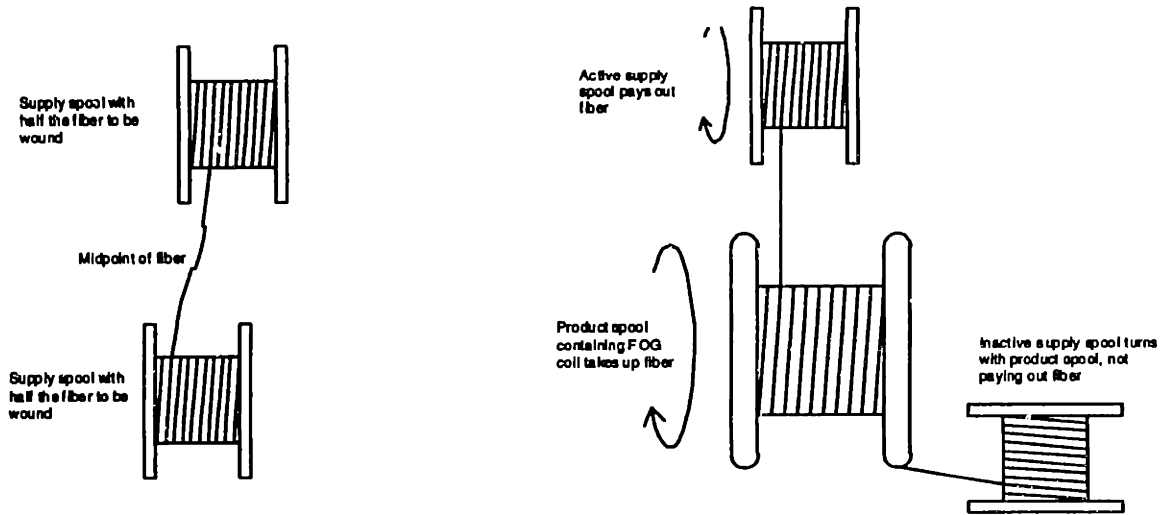


Figure 6. A FOG Coil.⁹

coil on the outside diameter and in opposite directions, as shown in Figure 6. To actually wind a coil this way, a length of fiber must first be wound half onto one supply spool and half on another, with the midpoint of the fiber between the supply spools.⁹ During winding, the supply spools will pay out fiber onto the product spool, which holds the coil. At any one time during a wind, one supply spool will be actively paying out fiber while the other is locked in place, rotating as the product spool rotates. An example sequence of winding is shown in

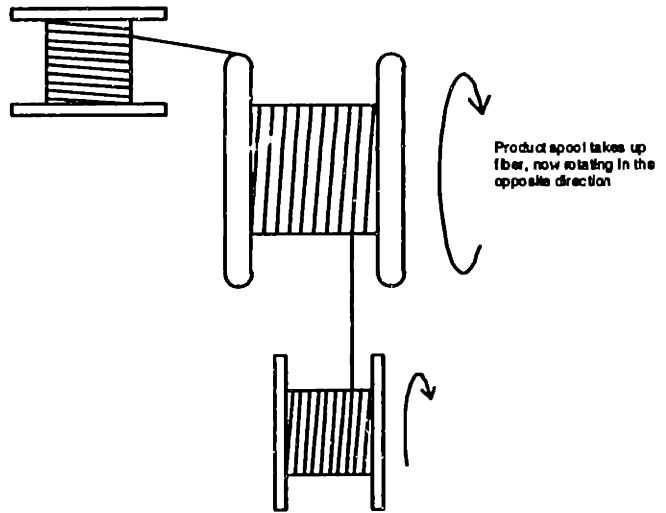
Figure 7. Note the two supply spools containing half the fiber and the inactive and active positions for each spool.

Keeping in mind that coils are wound in this inside-out manner, there are various patterns of fiber placement that can result in coil symmetry, some exhibiting better results than others. The most basic of these patterns is termed the “Quadrupole.” Figure 8 depicts a cross-section of a coil wound with this pattern. The white circles indicate fibers wound in one direction and the black indicate those fibers wound in the other. By examination, it can be seen that corresponding turns of the black and white fiber are in very close proximity within the fiber pack. Also note that fibers are placed into the grooves formed by the underlying layer of fiber. This helps in guiding the fiber and achieving a tighter pack. Variations on this basic pattern yield better symmetry which have been shown to improve coil performance. Coils that are wound without attention to this symmetry exhibit far inferior performance.



1. Fiber is divided between the two supply spools.

2. One supply spool pays out fiber while the other remains inactive.



4. The two supply spools alternate paying out fiber until the coil is completed.

3. The other supply spool becomes active and pays out fiber.

Figure 7. Winding Sequence

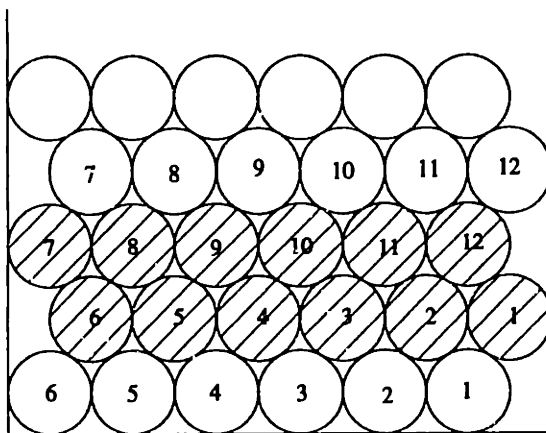


Figure 8. Quadrupole Cross Section

Regardless of what pattern is used in a coil, local areas of stress within the pack caused by poor winding can be quite detrimental to coil performance. Examples of poor winding would be lumps and voids within the fiber pack and severe variations in

tension. Essentially any characteristic of the coil wind that can cause varying pressures on the fibers can lead to variations in index of refraction. What this means is, to obtain optimum

performance, the coil must be wound as evenly as possible with minimal variations in tension. Not only must the tension be consistent, it must be low enough not to cause undue stress in the glass of the fiber. It has been shown that coils wound with low tensions yield superior performance¹⁰. These facts lay out the most basic of restraints for any coil winding machine: it must be able to

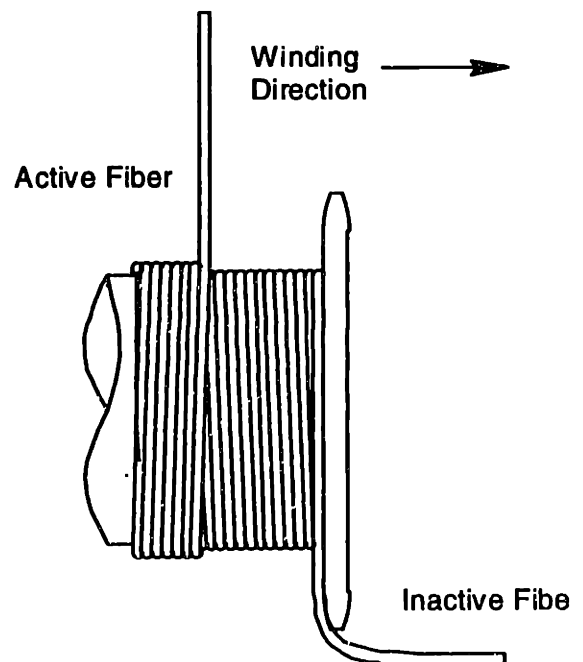
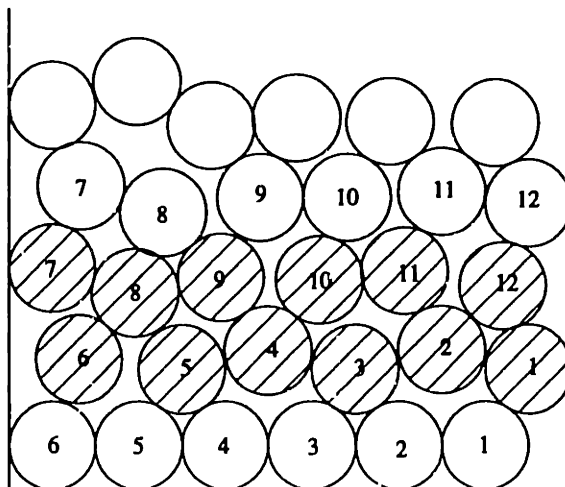


Figure 9. Helical Winding Technique

accurately place fibers during a wind and it must be able to accurately control tension.

To aid in achieving an even wind, a specific winding technique is commonly employed. The simplest way to wind is to place the fiber in a spiral, or helix, allowing it to traverse across the fiber pack naturally as it is wound onto the coil. Fiber follows the groove formed by the pervious layer when traversing in the direction of the helix and crosses those grooves when traversing against the helix as shown in Figure 9. The problem is that the fiber can jump to the next groove at any location around the circumference of the coil. The circumferential location of the can differ from turn to turn. It is easy to see that this phenomenon can cause inconsistency on the current layer, which can only get worse on progressing layers. An example of a cross-section of a coil wound in this way is shown in Figure 10. Dark circles correspond to fibers wound in the



clockwise direction and light circles correspond to fibers wound counter clockwise. Note the uneven placement of the fibers which can lead to localized stresses. And, these stresses can severely differ from fiber to fiber. Needless to say, coils wound in this way have poor performance.

Figure 10. Helical Wind Cross Section

One alternative is to wind coils in an Orthocyclic manner. This winding method was first developed by Phillips Industries in 1962 and is in common use today.¹¹ With this method, rather than winding in a helix, fibers are wound in concentric hoops and cross over to adjacent grooves formed by the underlying layer in well-defined zones. See

Figure 11 for an illustration of this type of wind. The zone where the fibers cross grooves is commonly termed the “jog zone” and is the only location in the coil where fibers deviate from a straight orientation. Many advantages are gleaned from Orthocyclic winding. Local stresses due to crossing over are limited to one small area of the coil.

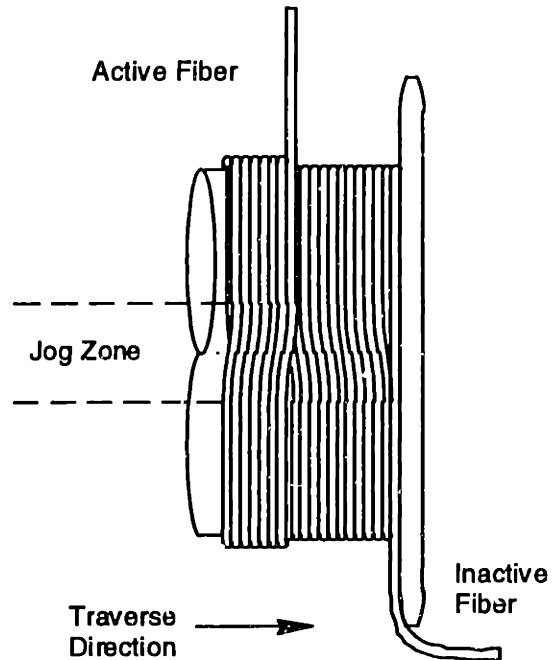


Figure 11. Orthocyclic Winding

Furthermore, adjacent fibers in this zone will experience similar stresses at the same locations, and because of the symmetry of the coil, this should not effect the relative phase of the light emerging from the coil. Another benefit is that the majority of the circumference of the coil is wound with an even, closely packed profile much like that shown in Figure 8. Any inconsistencies should then occur in the tight jog zone. Also, the close pack inherent to the Orthocyclic wind has been shown to improve thermal variation and vibration resistance of the coil.

1.3 The Drive to Automation

In review, for optimum performance, coils are wound in specific patterns in an Orthocyclic manner, starting from the center of a length of fiber and winding outwards to the ends. Fiber placement and tension control are known to be important factors in winding a high performance coil. Any machine that is created to wind coils must be able to deal with many variables. It must accommodate the different patterns and allow for Orthocyclic winding. It must be able to very accurately guide fiber and maintain a constant, low tension. To be successful, it also must allow for fiber diameter and hardware variations. One of the greatest challenges is to be able to wind an entire coil with few or no errors. An error is considered any place in the coil where there is a deviation from the pattern, such as a gap. These errors could degrade the performance of the coil and should be avoided.

The current state of production is generally semi-automatic, requiring the presence of an operator. The operator monitors the winding of a coil at all times and interferes when an error occurs. Generally, the operator would use a small tool to manipulate the fiber into the correct place. With the current machines and control, these errors actually happen quite often and greatly slow the winding times. In addition, with most machines the operator must switch the two supply spools manually between their active and inactive positions (see Figure 4), which takes time and adds risk. Any time the operator must interact with the fiber, there is the possibility of damaging, or even breaking, the fiber. Winding in

this manner has resulted in relatively slow winding times, up to a week in many cases, and also yields a significant rejection rate.

There is a definite opportunity in the industry for improvement of the coil winding process. If FOGs are to become competitive, the cost of coils must be reduced without sacrificing quality. To accomplish this goal, winding times must be decreased and rejection rates lowered. This implies that operator intervention must be removed from the process, or at least minimized. In the past, that wasn't a feasible possibility as the process of coil winding was poorly understood and it was difficult, if not impossible, to design a machine that could account for all the variables. Through experiments that have been completed in recent years, the process of coil winding has become somewhat better characterized and it has become easier to predict the behavior of the fiber during a wind. These experiments have also made it possible to place more accurate specifications on parameters that were previously loosely defined. Along with this improvement in process understanding, the state of technology has greatly improved, facilitating the development of machine automation.

Given that the process and technology are in place, and the opportunity is present, there is a definite drive to automation. It is now possible to design and build a machine that can automatically wind coils and, in the end, reduce the cost per coil. This thesis describes the design and development of such a machine. The additional refinement of the process necessary to make the automation possible will also be presented. In the next chapter, an overview of the entire design process will be given. Some aspects of the top level design will

also be described. Chapter 3 will discuss in detail the design and development of some of the subsystems of the machine. Chapter 4 contains a discussion of portions of the system characterization and testing. Chapter 5 will deal with some of the special issues relating to the coil winding process that were important to automation. Finally, a discussion of the success of the machine and future recommendations will be presented in chapter 6.

Chapter 2 - Overall System Design

This chapter will first cover the customer driven goals of the automated coil winding project. The design process followed in the development of the winder is then detailed. And, finally, a few key overall design choices are detailed and explained.

2.1.1 The Customer

Development of the automated coil winding machine was funded by an American company with aid from the federal government, to be referred to together hereafter as the Customer. In early 1995, the Customer approached the Manufacturing Institute at MIT (MIT/MI) with the automated coil winding project. The primary goal of the project was to cost reduce the winding of the fiber optic sensing coil used in FOGs. The current cost of FOGs derives heavily from the manual labor involved. Thus, the MIT/MI's main role in the project was to develop an automated coil winding machine. Two graduate students, Brian Sonnichsen and Stephen Lin, were placed onto the project and given responsibility over the entire design of the winding machine under the supervision of Dr. Andre Sharon, Director of the MIT/MI.

Specifications for the machine were drawn up early in the project. The details of these specifications cannot be revealed for reasons of confidentiality. The Customer desired a certain manufacturing throughput several times higher than conventional manual winding techniques. To achieve high quality coils winding errors were to be minimized and the tension on the fiber during winding

was to be under tight control. The machine also had to meet various levels of flexibility. These included flexibility in the size and length of fiber it could wind, in the nominal winding tension, in the size of mandrel wound onto, and in the particular pattern to be wound. This flexibility would allow the Customer to wind a variety of coils covering the range of performance grades to suit different applications.

2.1.2 The Design Process

To accomplish this task, MIT/MI followed a well defined design process. The process began with a review of existing technologies to learn the current state of the art in fiber optic coil winding. This review included reading through many articles written over the last decade to learn from the successes and failures of past experiences in coil winding. The review also included site trips to three existing fiber optic winding centers across the United States to see actual production and experimental winding machines in action and to speak with the engineers and technicians involved. In a meeting with the Customer immediately following these reviews, it was determined that the state of the art in coil winding was not a sufficient foundation to begin design of an automated winding machine. The MIT/MI proposed, and the Customer agreed, that a test-bed winding machine should be built on which to conduct winding experiments. The goal of the test bed would be to provide a flexible vehicle for proving out

designs for the final automated winder. It was believed that this test-bed would be essential for the success of the project.

The design of the test-bed itself began with brainstorming sessions both individually and as a group with periodic reviews with the Customer to insure the design was headed in an approved direction. Design constraints on the machine were taken from the specifications laid out by the customer as well as from information learned during the review of existing technologies. Design concepts were evaluated against the design constraints and each other and eliminated accordingly. Foam models followed detailed sketches of the more promising concepts. Following a final design review with the Customer on the test-bed, the detailed design of the winder began. A picture of the test-bed winder can be seen in Figure 12.

Once the test-bed was built, various experiments were run. The experiments served to prove out both hardware issues and process issues. For example, these experiments included determining the best mandrel and flange configurations for the product spool on which to wind the coil, how to wind fiber up against the flanges, how best to achieve tight tension control on the fiber while winding, how best to monitor the winding to detect errors, etc... Full coils of two different pattern types were wound on the test-bed, modified as deemed appropriate from the experiments, to prove out the overall concept for the final machine.

Design on the final machine began immediately after finishing experiments on the test-bed. Since the test-bed was very successful in its task, many of the issues in designing the final machine were related to automating those parts of the test-bed which had been performed manually. The final design also involved improving on the problem areas of the test-bed which were unable to be corrected at the time. For example, linear slides and swivels had to be replaced by motorize linear stages and rotary tables, and the mounting of the product spool had be to redesigned to eliminate runout while winding. The design process used during this design followed the same structure as when designing the test-bed. In some cases, working prototypes of competing ideas were constructed for evaluation.

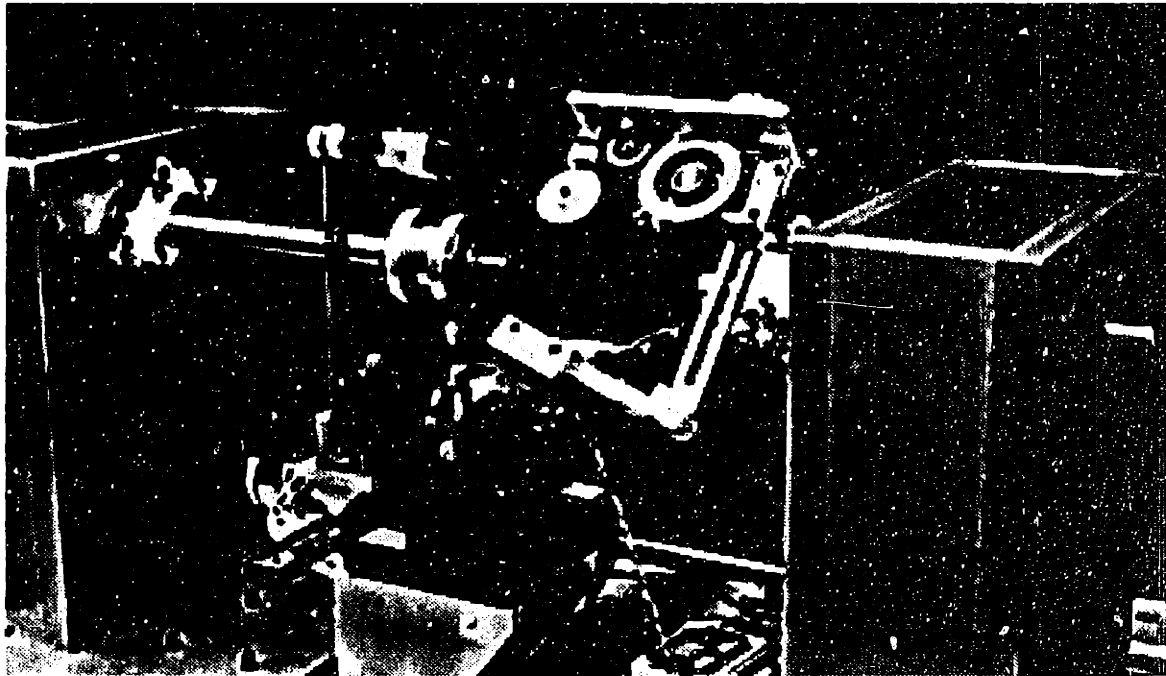


Figure 12. Test-Bed Winder

The actual design of the test-bed won't be covered in this thesis.

However, much of the design of the test-bed was used as a starting point for the design of the automated machine. Most of the process experiments discussed were performed on the prototype. This thesis will instead cover the design and development of the final automated winding machine. Many of the subsystem designs and various process development items will be covered in the thesis by Stephen Lin entitled "The Design and Development of an Automated Coil Winding Machine."

2.1.3 Basic Requirements For Winder Topology

In Chapter 1, the characteristics of a coil, including one example pattern (the quadrupole), are detailed. In order to determine what shape an automated coil winding machine must take, the process steps in winding a full coil must be carefully considered for each pattern to be wound. The choice of patterns to be wound greatly influences the topology of the machine. Many of the existing machines surveyed during the design process were constructed with the quadrupole pattern in mind. Figure 13 shows a basic machine layout for winding a quadrupole coil. In this configuration both supply spools lie along the main shaft either rotating with it or remaining stationary, depending on the state of two mechanical clutches. One supply spool remains clutched to the main shaft while the other pays out fiber onto the product spool as the product spool rotates. A guide pulley able to move laterally along the length of the product spool aims the fiber towards the proper location on the coil.

The quadrupole is considered one of the easier patterns to wind. One reason is that at all times during the wind, each supply unit remains on its respective side of the product spool. Figure 13 shows supply spool A as it nears the end of a layer winding from right to left. The following layer would also be wound from supply spool A as it traverses back to its side of the product spool. Subsequently, supply spool B would be used to wind two layers first from left to

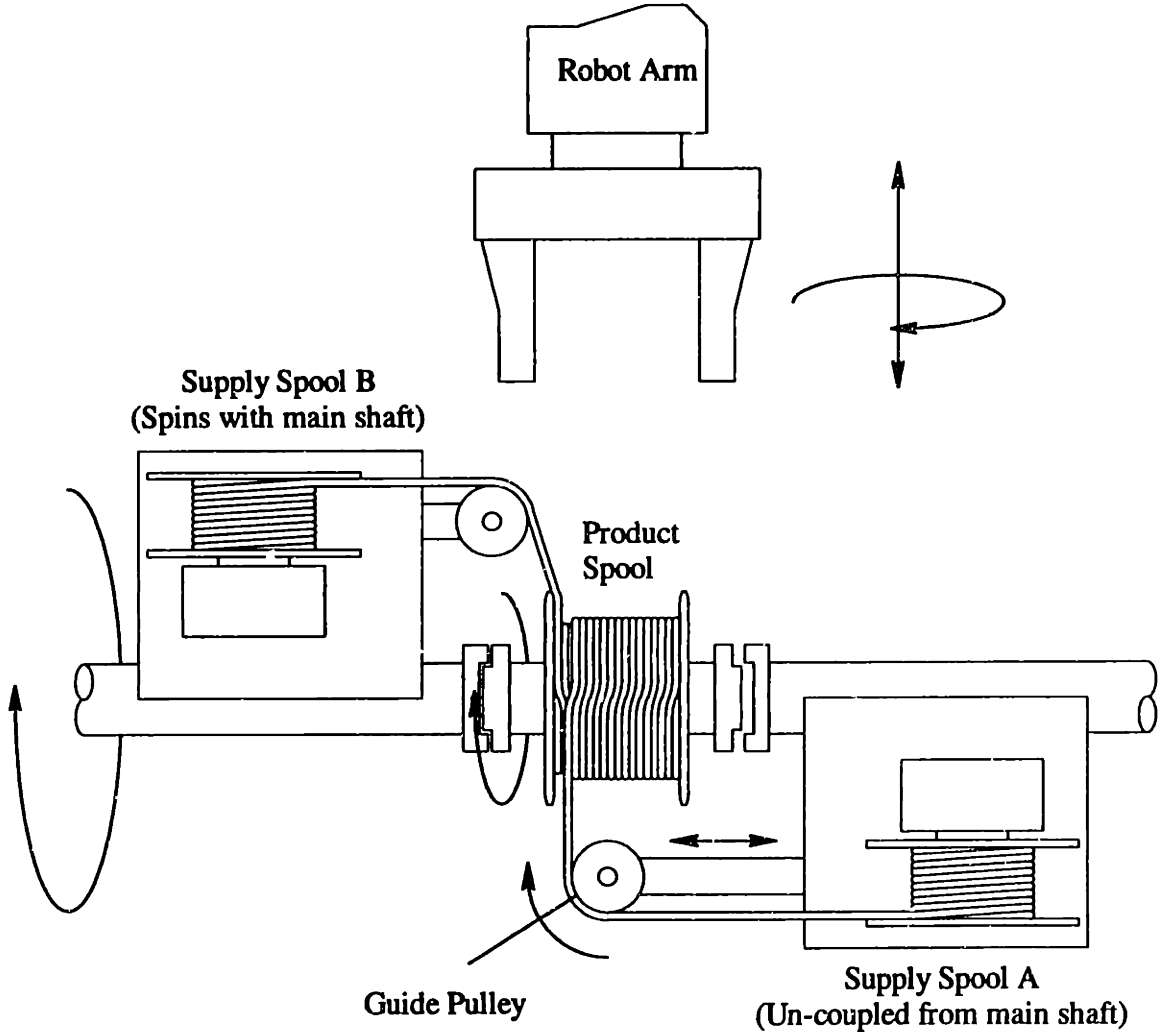


Figure 13. Basic layout for a machine design to wind a quadrupole coil

right and then right to left. This alteration continues until the coil is finished.

Thus, the fiber from the two spools never have to cross each other.

Now, imagine that at the point shown in Figure 13 that the next layer was to be wound using supply spool B. This would, in fact, be impossible as the fiber would not be able to wind onto the product spool without first colliding with the fiber from supply spool A. This crossing of fibers on the same layer is a feature

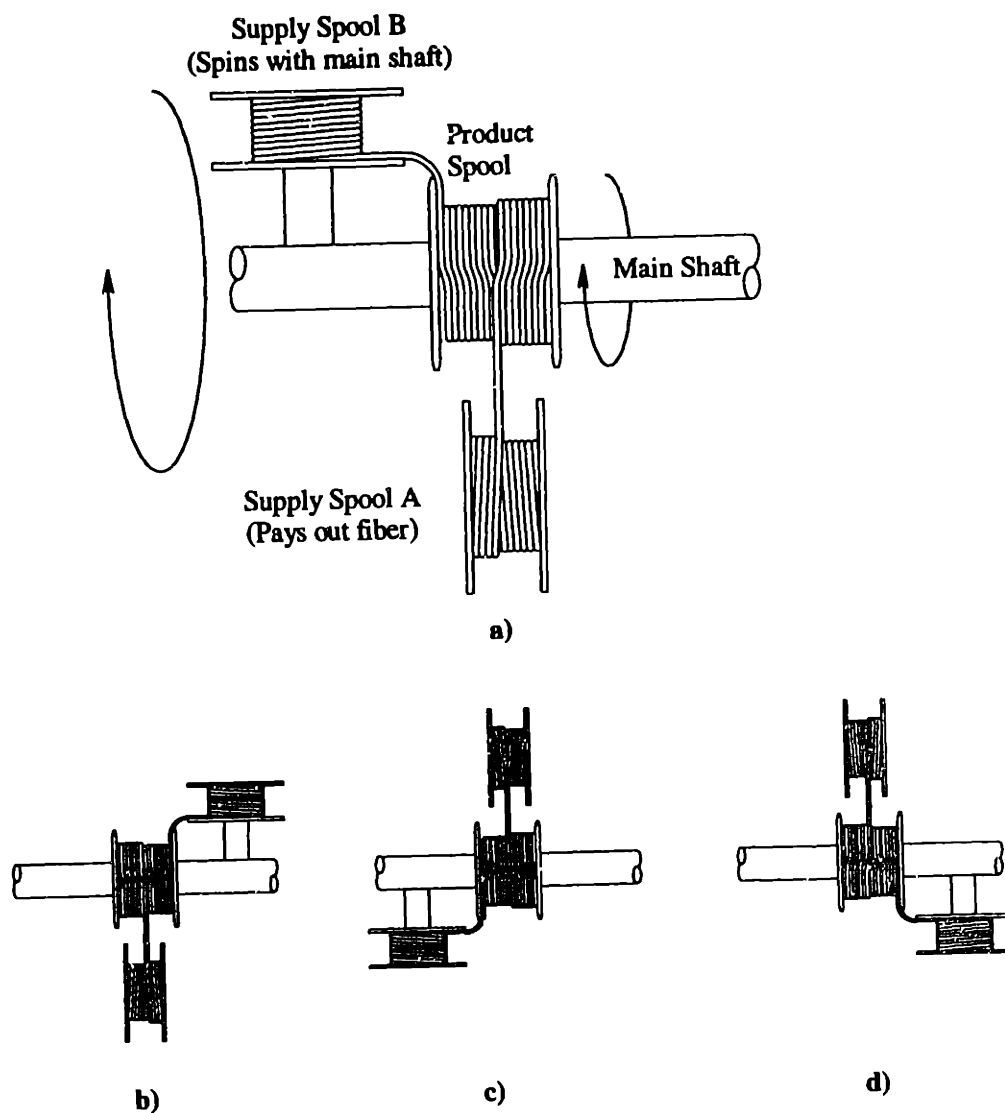


Figure 14. Four basic requirements for winding all the patterns desired by the Customer.

of a second winding pattern and also certain variations of a third pattern, both of which will not be named or described in detail for reasons of propriety.

After taking into consideration all the winding patterns desired by the Customer the basic requirements that the winder topology had to satisfy were determined. Each supply spool must be able to be placed on either side of the product spool and rotate with it while the other supply spool pays out fiber. Figure 14 illustrates these four basic requirements. Figure 14a shows supply A paying out fiber while supply B rides along on the left side of the product spool. Figure 14b shows the same only with supply B riding on the right side of the product spool. And, Figures 14c and 14d show supply B paying out fiber while supply A rides on either side of the product spool.

Looking again at the quadrupole winder in figure 13 it can be seen that the machine by itself can only satisfy two of the four basic requirements because it locks each supply spool to a particular side of the product spool. However, there is a way around this limitation. It is possible to effectively move each supply spool to the other side of the product spool by keeping the supply spools stationary and instead flipping over the product spool itself. This technique has been proven to work manually by a subcontractor to the Customer. Figure 13 shows a proposed robotic arm which could perform this task of flipping over the product spool. The difficulties of this method of achieving the two remaining requirements include the design of such a mechanism to grasp the product spool, de-couple it from the main shaft, flip it over, reattach it to the main shaft, and then retract so as not to interfere with the winding. In addition, the act of

performing this flip induces a half-twist into the fibers coming from both supply spools. Such twists can potentially degrade coil performance. Thus, this basic topology where both supply spools are tied to a particular side of the product spool was discarded as a possible basis for the automated winding machine.

The only remaining topology determined to satisfy all the basic requirements keeps the product spool fixed and instead moves the supply spools as shown in Figure 14. The difficulty here lies of course in actually moving the supply spools into these various locations without losing control of the fiber. The next section describes in more detail the design choices considered and ultimately the design chosen.

2.1.4 The Supply Unit Concept

Once it was decided that the product spool would remain fixed while the supply spools moved, an specific actuation method had to be chosen. There are two basic approaches to this task. The first approach seeks to minimize what is moved around, i.e., only the supply spool itself is moved from side to side. In the second approach everything necessary for paying out fiber from a supply spool and maintaining tension on it is moved from side to side.

During the survey of existing machines, an experimental winder was seen on site at a subcontractor to the Customer's facilities which satisfied the four basic requirements discussed and utilized the first approach described. However, this experimental apparatus was manually intensive. The operator was required to first secure the fiber to the product spool so that it would not

unravel. Then, he would detach the supply spool from the motor driving it and unthread the fiber from the various pulleys it was routed around. As he performed this, the extra fiber would be wound back onto the supply spool and finally secured to the spool before attaching it along the main shaft.

The main difficulty of this approach is apparent. Not only must some method be found to attach/detach the supply spool to and from its drive system and the main shaft, but another found must also be found to thread and unthread the fiber through its various pulleys. In addition, the fiber must be prevented from simply hanging free without control lest it unwind from the product spool and also become impossible to re-thread. Although, this approach seemed reasonable from the point of view of an operator, it was clear that automating it would prove very difficult.

There is no reasonable compromise between the two approaches, so the second approach, later to become known as the Supply Unit approach, was chosen.

The advantages of this approach are many. With all the mechanisms needed to payout fiber, guide it onto the product spool, and maintain tension on the fiber, located on one modular unit, suddenly there is only one task at hand. That task is to mechanically and electrically couple and de-couple the entire unit

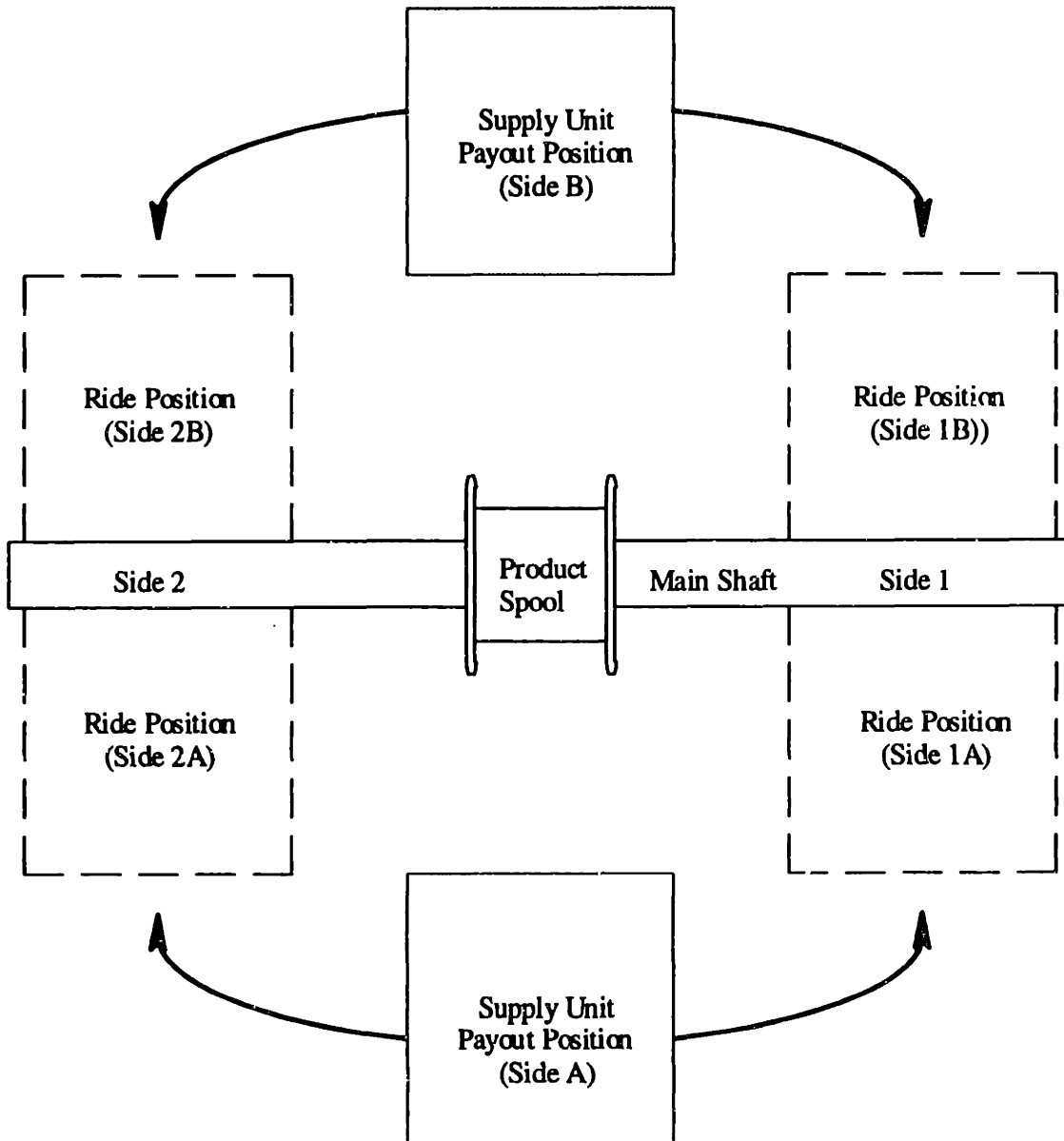


Figure 15. Ride and active positions for each supply unit

from either side of the main shaft and from its payout position. This task is commonly referred to as swapping and will be covered in more detail in Chapter 3. Figure 15 illustrates the three locations each supply unit (one for each supply spool) must assume to satisfy the four basic requirements.

2.1.5 Fiber guiding

Another issue affecting several areas of the machine design is the issue of fiber guiding. Fiber guiding is considered fundamental to winding a fiber optic coil, because without a good method of guiding the fiber into the proper place on the coil, winding errors are likely to occur. Any time a fiber is not placed where it is meant to be according to the pattern, a winding error is said to have occurred. Winding errors are known to degrade the ultimate performance of a fiber optic gyroscope.

Several methods were surveyed or designed for guiding the fiber onto the product spool, each with its own strengths, weaknesses, and consequences for the rest of the machine (the supply units in particular). The most rudimentary approach involves simply using a lead or lag angle and the previously wound turn to guide the fiber. Another approach uses a very thin guide wheel to place the fiber onto the coil. And a third approach involves the use of a fiber manipulator. This section investigates the choice of the guide wheel as the final choice for the automated winding machine.

2.1.6 Fiber Guiding Requirements

There are several situations that are encountered during the winding of any given pattern which require a slightly different form of fiber guiding. To be successful, the automated winding machine must provide the ability to guide the fiber satisfactorily in all situations. Figure 16 shows the three basic winding situations encountered during a wind. In 16a the next fiber turn is being placed adjacent to the previous fiber turn. In Figure 16b the next fiber turn is actually spaced out by one fiber from the previous turn. And in figure 16c the next fiber turn lies against the flange of the product spool. In this case, the fiber is one layer up, but this can also happen on the same layer.

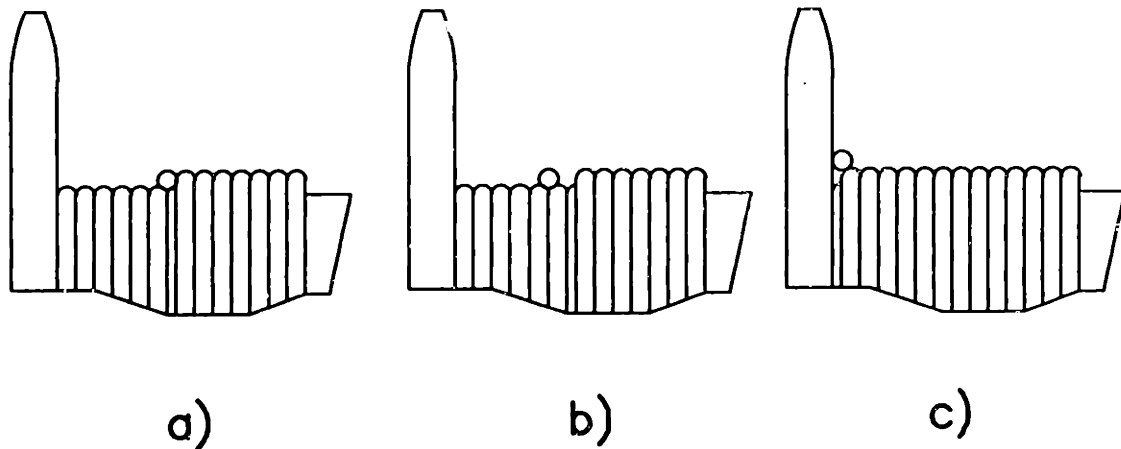


Figure 16. Three basic fiber guiding situations.

2.1.7 Lead and Lag Angles

Lead and lag angles offer the simplest method for fiber guiding. Figure 17 illustrates the use of both lead and lag angles. In this method, the last pulley routing the fiber from the supply spool onto the product spool is used to align the fiber. If the placement of the pulley is located behind the current turn, as shown

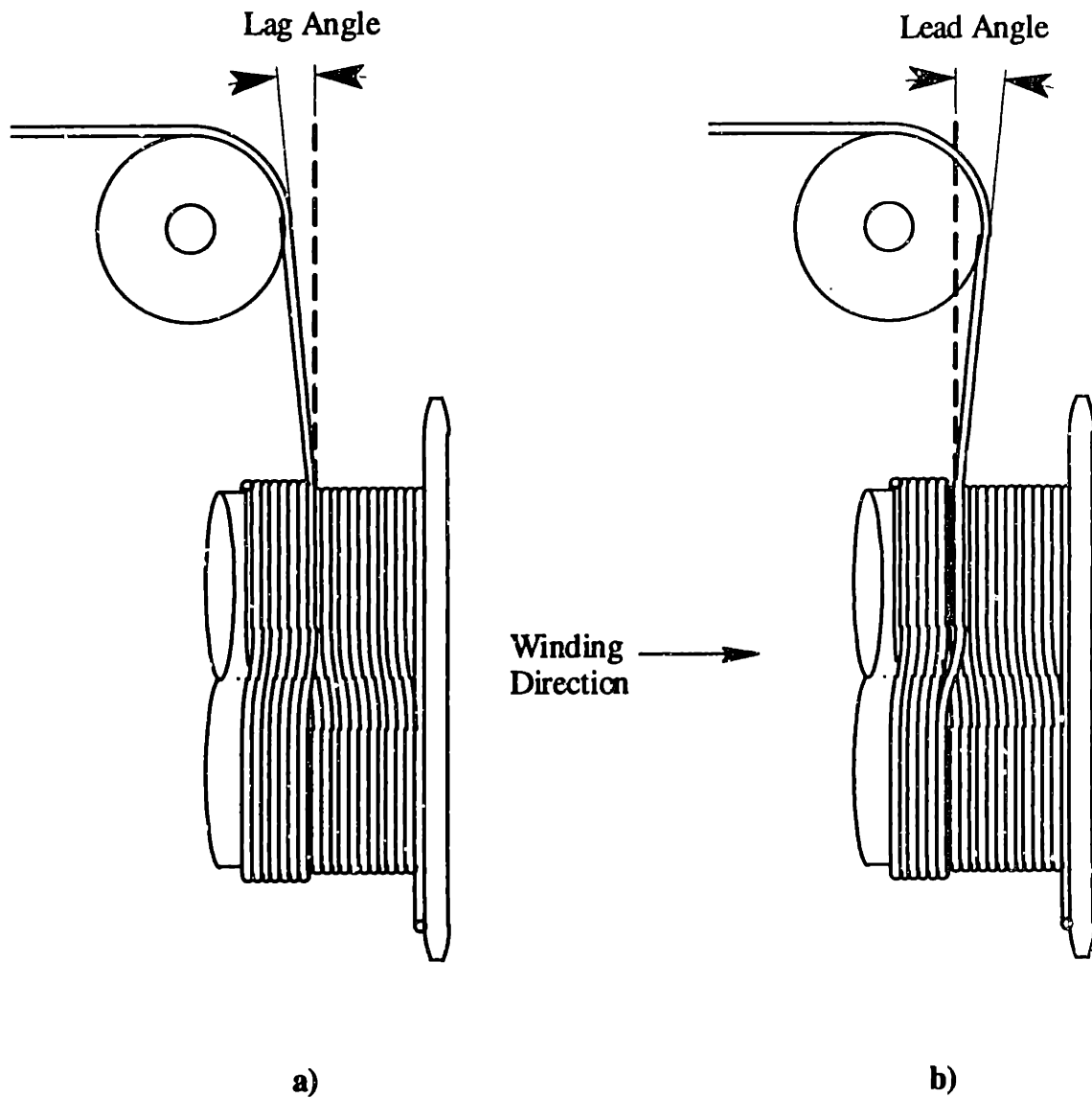


Figure 17. The use of lead and lag angles for guiding fiber onto the product spool

in figure 17a, then a lag angle is being used. Using a lag angle causes the fiber to push lightly against the previous turn thus using it as a guide for the next turn. If, on the other hand, the placement of the pulley is located ahead of the current turn, then a lead angle is being used. This would be useful in trying to achieve a spaced out fiber, such as the one shown in figure 17b, if required by the winding pattern.

The basic advantage of the purely lead/lag guiding scheme lies in its simplicity. By guiding the fiber from a distance there is no problem of interference of a guiding device with the flanges of the product spool. However, the simplicity of this method also limits its effectiveness. With such little control exercised over the fiber the use of lag angles can sometimes give rise to errors known as "climb" errors. This type of error occurs when a fiber lays down on top of the previous fiber turn rather than by its side. If no lead or lag angle is used, the nature of the orthocyclic wind will tend to cause another type of error known as a "gap" error. A gap error is where the fiber lays down with some space between it and the previous fiber turn rather than right next to it. A lead angle can be used to purposely induce such a spaced out fiber. However, the control over the placement over the fiber in a typical lead/lag guiding scheme is very limited and historically error prone.

2.1.8 Fiber Manipulators

Guiding the fiber from outside the flanges of the product spool is pretty much limited to the lead/lag scheme described previously. The next step in guiding is to attempt to place a fiber manipulator in-between the flanges to constrain the way the fiber lays down onto the coil.

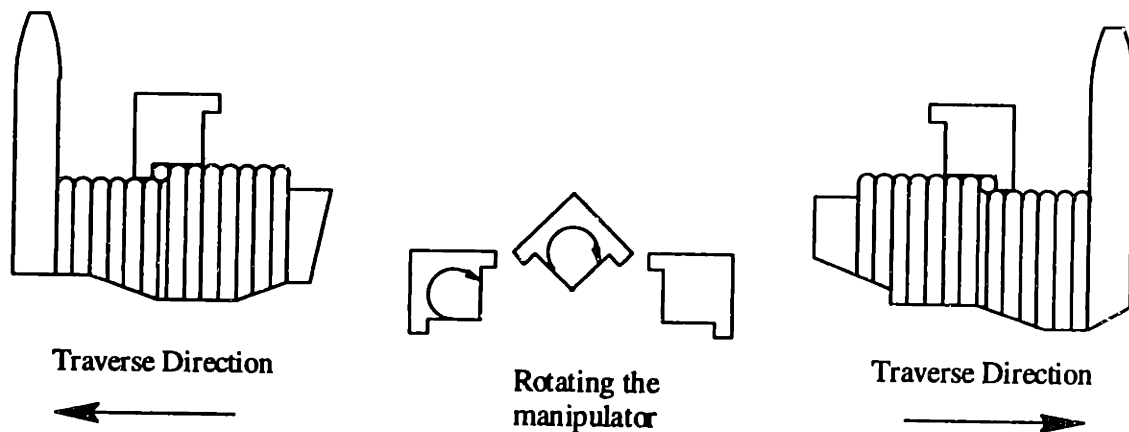


Figure 18. Fixed manipulator for constraining fiber placement.

Figure 18 shows one such concept taken from a subcontractor of the Customer where a tiny block of a special shape is used to constrain fiber placement to only one possible place. This particular manipulator is capable of constraining the fiber while winding in either direction, although, the manipulator must be rotated to achieve this. The limitations on this form of manipulator is its inability to constrain winding right next to the flange and its inability to create spaced out fibers in a controlled fashion.

The concept for the manipulator in Figure 19 was designed by the MIT/MI to address the limitations of the fixed block manipulator. As shown, by adjusting

the relative positions of the two finger-like mechanisms, this manipulator can be used to constrain the fiber in all three of the basic winding situations depicted in Figure 19.

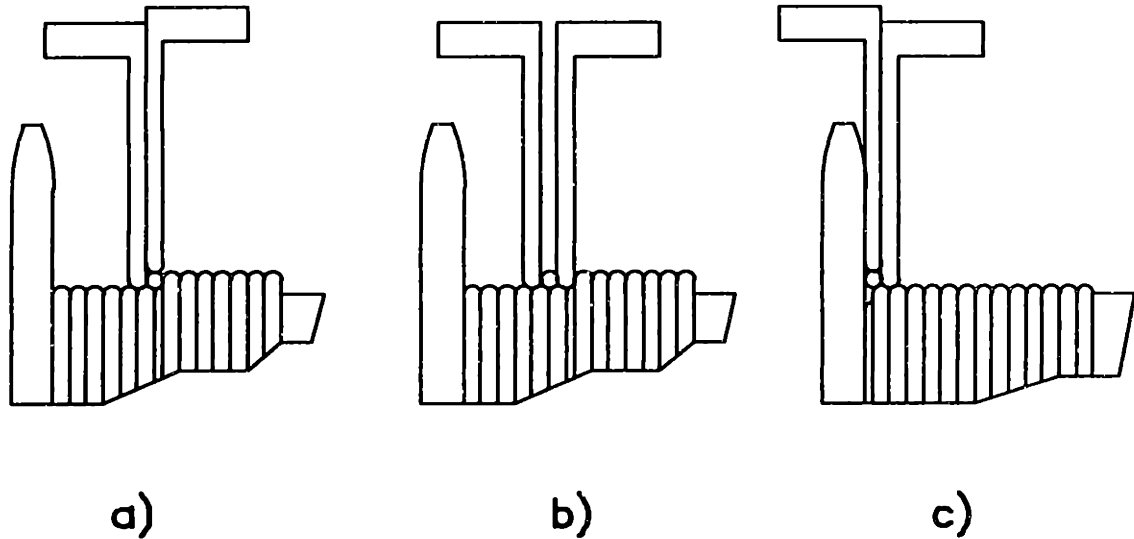


Figure 19. Adjustable finger-like manipulator for fiber guiding.

The idea of a manipulator is not without its difficulties. Apparent in the adjustable finger-like manipulator, any such device would have to be very thin to accommodate winding against either flange. Though they appear to constrain the fiber completely, it is unclear that such control would be maintained when switching from one winding situation to another, and it is equally unclear how control could be regained automatically should the fiber get free of the manipulator. The Customer also voiced concerns that the use of tools in continual sliding contact with the fiber might cause micro-cracks in the fiber and thus degrade the final coil's ultimate performance.

2.1.9 The Guide Wheel

Another form of fiber guiding was borrowed from another subcontractor for the Customer. This method involves the use of a guide wheel such as the one shown in Figure 20. The mechanism consists of the guide wheel itself which is a thin disc measuring no more than two fibers wide. It has a fine groove around its circumference in which the fiber can be placed and external hubs to support the guide wheel.

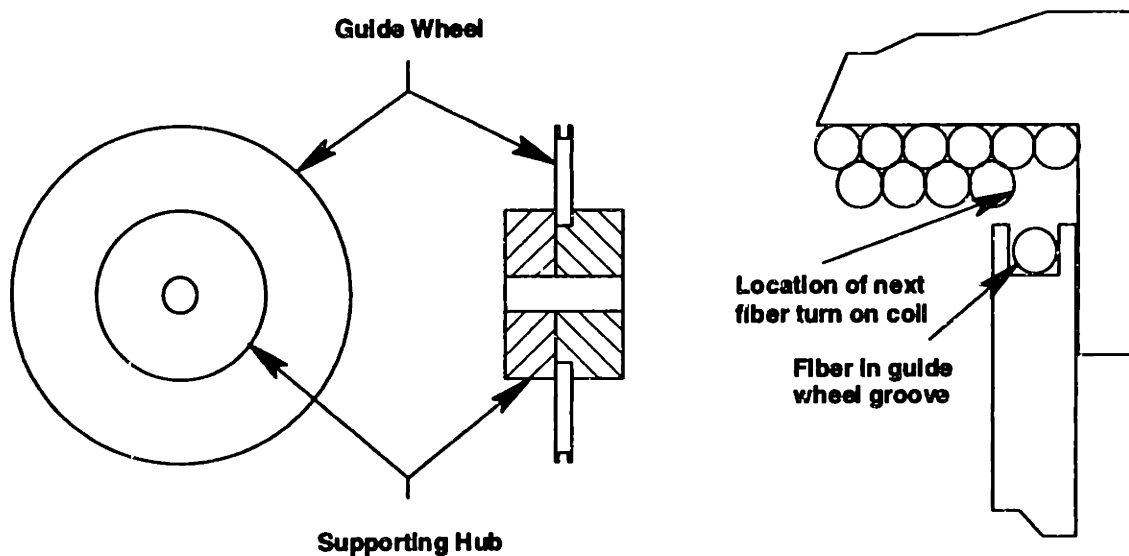


Figure 20. Guide wheel concept

The guide wheel's groove constrains the placement of the fiber onto the coil to only one possible place, assuming the gap between the guide wheel and the coil is kept small enough. The ultra-thin profile of the guide wheel allows it to accurately place fibers onto the coil across the entire coil except for fibers which lie directly against the flange. The contact between the fiber and the guide

wheel does not exhibit sliding contact as does the manipulators previously mentioned.

The difficulties associated with the guide wheel result from its manufacture. To insure proper placement of the fiber the guide wheel must exhibit very low axial runout. For instance, should a guide wheel have about .0015" of axial runout, this represents about 25% of a fiber diameter. Placement accuracy of only 75% would lead to many potential winding errors. Achieving low runout guide wheels was found by the mentioned subcontractor to the Customer to require a long and costly machining process.

2.1.10 Concept Selection

Ultimately the guide wheel was chosen as the method for fiber guiding on the automated winding machine. It was clear that a method so simple as the use of pure lead/lag angles to guide the fiber from a distance from the actual coil would not be enough. Although, the manipulator covers more potential winding situations with its ability to adjust its shape to conform to the needs of the situation, the issue of sliding contact with the fiber prevented the idea from getting beyond the drawing board. The guide wheel was a proven technology with only one real limitation, it's inability to wind a fiber right up against the flange. However, tests were performed on the test-bed winding machine to confirm the effectiveness of the guide wheel for positioning accuracy. These tests revealed that if the neighboring fiber turns are placed correctly, that the last

turn on a layer up against the flange will be self-guided. With these results from the test-bed winder the guide wheel concept was again selected for use on the final machine.

2.2 Overall Design Validation

As mentioned before, a test-bed winder was built prior to the final automated winding machine. The test-bed winder was built not only to perform experiments with the winding process itself but to validate the overall winder design concepts described earlier in this chapter. The test-bed was designed and built employing the supply unit concept, allowing the supply units to swap to either side of the product spool. Both supply units used the guide wheel concept to place fiber onto the coil. The test-bed winder was successfully used in several winding experiments to wind test coils anywhere from a few layers of fiber to the tens of layers of fiber which would be required for an actual gyroscope sensing coil. The tests included all three patterns mentioned and thus validated that the over all system design was flexible enough to meet the Customer requirements for the pattern types to be wound. While noting some problems with runout of the guide wheels and of the product spool on the main shaft, the guide wheels were validated as an effective means for accurately placing fiber onto the coil. Where guiding with the guide wheel was not possible, next to the flanges, it was discovered that a properly wound coil up to that point would provide a pathway for the next fiber turn against the shaft which would be naturally followed by the fiber.

2.3 Process Experiments

The other main purpose of the test-bed winder was to perform winding experiments to help characterize and define the winding process. When winding experts had previously been questioned about various issues in winding a FOG coil, different answers were sometimes received. The MIT/MI sought to learn as many of these answers as possible for itself. Many of the questions were answered with experiments on the test-bed winder, and remaining questions were completed with test winds on the final automated coil winding machine whose greater precision and controls allowed for more accurate and conclusive tests. The experiments performed included the following objectives: characterization of winding on the base layer, characterization of winding in the jog zone, characterization of winding the transition turns, and determining the affect of various tension levels on the quality of the wind. These topics are addressed in detail later in this paper and also in the thesis by Steve Lin mentioned previously.

Chapter 3 - Winder Design and Development

3.1 Topics of discussion

This chapter contains a discussion of the detail design and development of the main structure of the machine, the main drive system, the subsystem termed the ride clamp, and the product spool mount. Each of the following sections will contain the detail design process for each subsystem as well as a discussion of the fabrication and testing of each system.

3.2 The Main Structure

It was important to decide upon an exact structural layout for the winder at a very early stage in the design. Once the layout was accepted, it would be exceedingly difficult to change after the rest of the winder design had progressed. Needless to say, a considerable amount of thought and discussion went into the concept selection, and foam models were constructed before the design proceeded. The basic shape and dimensions were actually first incorporated into the design of the test-bed to determine if they would be acceptable. However, the structural design will be presented as though it were a one step process to simplify the discussion. In actuality, improvements and modifications were made on that original structure to create the final winder design.

3.2.1 Functional Requirements - Main Structure

There were several levels in the design to consider. On the broadest level were the locations of the main subsystems and their basic dimensions. After subsystem locations and sizes were chosen, the size and dimension of the main structure could be defined. Another level of the design involved the actual layout of the linear stages and main drive. For example, was the main shaft to move axially relative to the supply unit, or would the supply unit move in that direction? It had to be decided exactly how to place the stages and main drive to obtain optimum performance from the machine. Once these placements were set, the rest of the structure could be designed to support and accurately locate the various systems.

The first important issue to consider concerns stiffness and strength. While the machine would generally not be operating at high speeds or under strenuous loading conditions, it was relatively important to minimize all vibrations and to avoid structure deflections. To accurately place fiber, the machine had to be able to locate the guide wheel accurately, on the micron level, relative to the product spool. In addition, a wind error detection system, possibly involving a vision system, would be operating during winding and would require that its field of view have a low level of vibration to ensure proper operation. Thus, from two different standpoints, it would be important to consider vibration reduction techniques in the design.

Structural deflection also presented a problem. Deflection of the main shaft or of the guide wheel support system could result in misalignments that would lead to winding errors due to inaccurate fiber placement. While this misalignment could be mapped and stored in the control program, doing so would add a considerable amount of programming and fine-tuning. Given the time restraints of the overall project, such additional efforts were to be avoided if possible, which meant designing to reduce structural deflections to an acceptable level.

Both the deflection of the supporting structure and error motions between the guide wheel and product spool relate directly to the stiffness of the machine and of its components. No hard numbers were actually placed on the value of stiffness required, as a result of the time restraint. Instead, the approach was to consider the possible sources of loading and vibration and include a high enough factor of safety to ensure more than adequate stiffness. More importantly, the motion subsystems were chosen and positioned with the goal of reducing the possibility for vibrations and deflection.

Another requirement was accurate location of the guide wheel relative to the product spool. This made it difficult to put numbers to the allowable relative position error. It was decided to use proven techniques in the design that would reduce error stacking. Any error that would then result was predicted to be small enough to be dealt with through a one-time calibration of the machine.

Finally, the aesthetics of the machine was an important consideration. Given that the winder was a product being purchased by an outside customer, it

was a high priority that the machine look professional when finished. This would have direct bearing on the overall design as well as on many of the finer details.

- FR:
1. High stiffness and strength
 2. Precise location
 3. Aesthetics

3.2.2 Concept Generation and Selection - Main Structure

The real challenge in the design involved choosing locations of the linear motion actuators and of the main shaft drive which would maximize the stiffness and precision of the winder. While important to the success of the winder, the actual physical design of the supporting structure was straightforward and posed little conceptual challenge. There were some important issues relating to the functional requirements of the machine to be considered, however, and these issues will be presented in the next section. For now, the concept generation and selection of the actuator layout will be the topic of discussion.

Six different axes of motion were required to give the machine all the capabilities necessary to wind any coil pattern. The main drive would turn the main shaft, on which the product spool would be mounted (θ axis). The guide wheel would have to move axially along the width of the product spool to guide the fiber into place (x axis). As mentioned earlier, the guide wheel would guide from beneath the spool during winding. This means that there would need to be an axis to lower the guide wheel relative to the product spool as the coil

thickness grew as successive layers were wound (z axis). In addition, to allow for swapping between the two supply units, an axis would be required to move the guide wheels towards and away

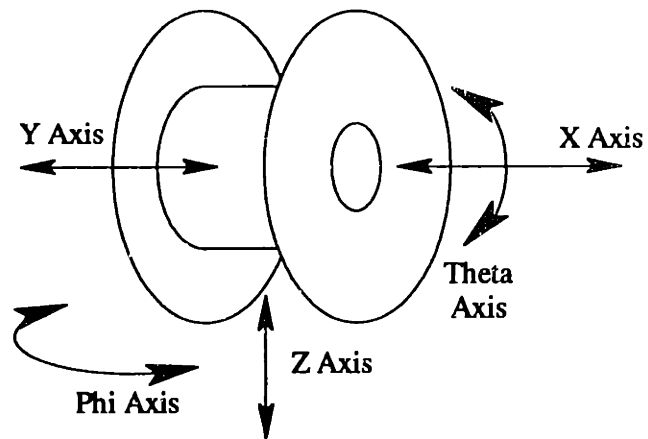


Figure 21. Axis Definitions

from the spool as required (y axis). Finally, given the winder topology selected, two axes would be required to swap the two units between the ride and active positions (phi a and phi b axes). Axis definitions are shown in Figure 21.

The challenge was to select the appropriate actuators and transmissions for all the axes and to locate them relative to each other on the machine so as to reduce the possibility for error motions and deflections. Two of the different arrangements considered for the final structural layout will be discussed here. Many more possibilities were actually considered, but the following two catered best to the chosen overall winder topology.

The first arrangement is shown in the conceptual sketch below (Figure 22). In this arrangement, the main shaft is both rotated and translated. The main shaft drive system would translate axially along the x direction, in effect moving the product spool along its width relative to the guide wheel, creating the necessary motion for fiber guiding. Assuming a lightweight drive system, this

would lead to a low inertia on that axis as the only weight to be moved is that of the drive system. Given that the x axis would directly affect the fiber guiding motion, it was particularly important to choose accurate and stiff components along this direction.

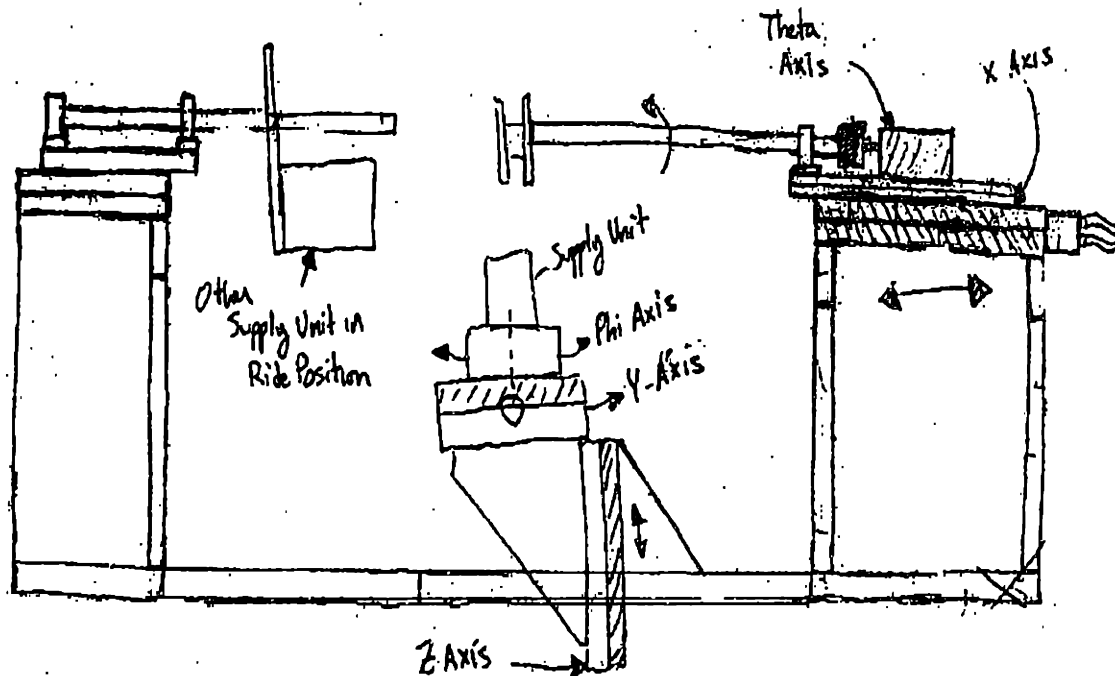


Figure 22. Sketch of Concept 1 for Winder Layout

Next, the y axis would mount in the center of the machine as shown above. It would need to be relatively accurate in placing the guide wheel, but more importantly, would need to be capable of fast movements. This axis is an integral part of swapping and would directly affect the total swapping time. It was a design goal to minimize swapping times so that useful winding time would be increased.

The z axis would be mounted beneath the y axis and would move both the x and theta axes vertically. During a wind, this axis only moves a few times, but still had to be repeatable in those motions.

Finally, both phi axes would sit atop the y axis. The supply units would be most directly in contact with these axes. These would both be rotary axes that would serve to move the supply units between the various swap positions. They too would need to be fast to minimize swapping time, but would also need to be very repeatable in rotary positioning. Errors on the order of arc minutes could translate to thousandths of an inch of misplacement of the guide wheel.

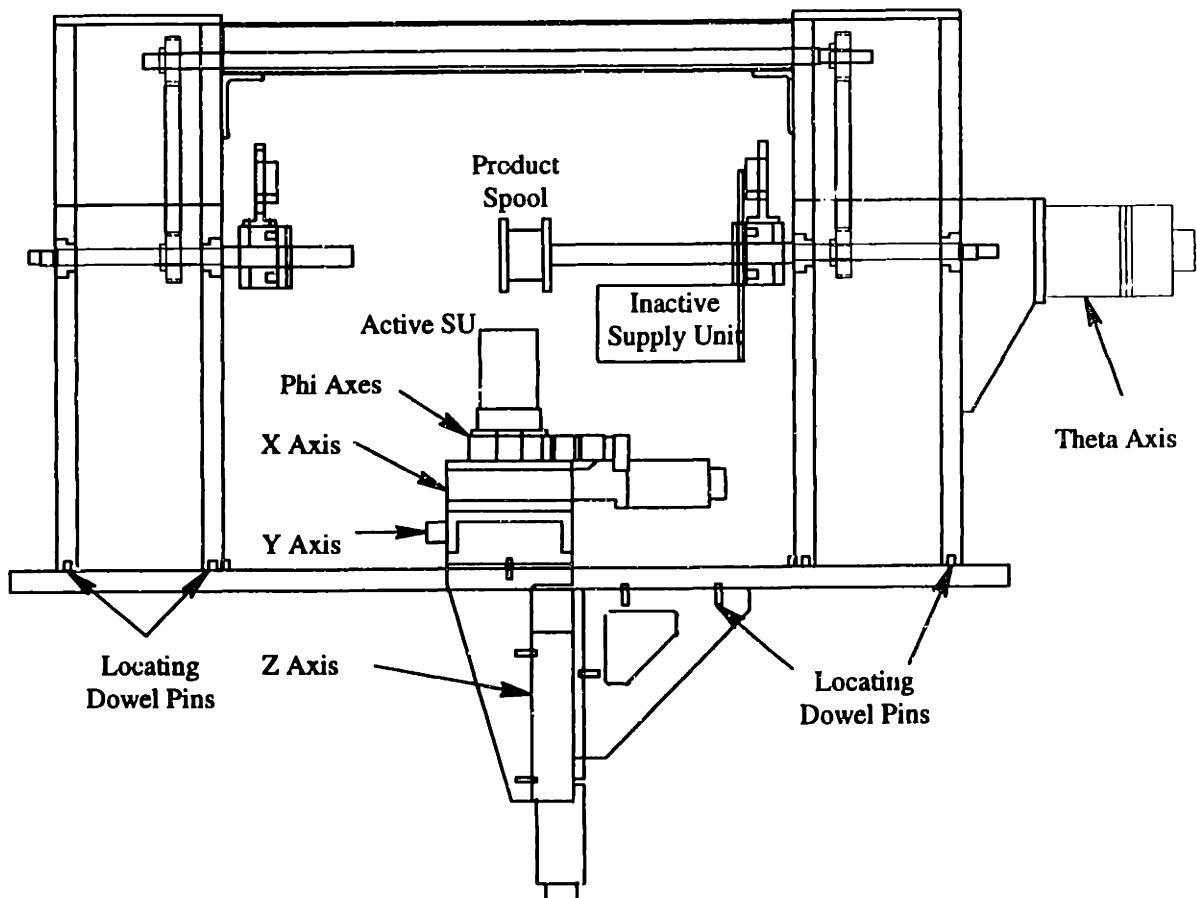


Figure 23. Assembly Drawing of Concept 2 for Winder Layout

The other alternative is shown in Figure 23. This is actually the chosen concept and the drawing is an actual assembly drawing of the machine. In this concept, the main shaft is held in a fixed position axially and is only free to rotate. All of the other axes are grouped together in the center of the machine. As for the characteristics of each of the remaining axes, the same concerns apply to those mentioned in the first concept. As the phi axes need to be in close proximity with the supply units, they are placed on the top of this grouping. The goal was to put as little weight on top of the x-axis as possible to reduce the inertia to be moved during fiber guiding. Hence, the x axis was placed immediately below the phi axes. As the y axis moved frequently and quickly, it was placed next in an effort to reduce the amount of weight that needed to be moved. Finally, the z axis is placed at the bottom of the arrangement as it neither needed to move quickly nor very often. Since accuracy was the only strict requirement on this axis, weight was not a significant concern.

In evaluating these two concepts, predicted stiffness and error motions were used to weigh the various characteristics in each design idea. In the end, the second concept was chosen as it seemed to give the greatest possibility for meeting these two design criteria.

There were a couple major reasons that the second concept was chosen. First, the isolation of the theta axis and product spool from the other axes of motion meant that error motions on the actual spool would be much lower than in the first concept. In addition, the weight surrounding this axis would not be a factor as it was never in motion. Hence, the support for the theta axis could be

made sufficiently strong and stiff to minimize deflection of the main shaft as well as lower vibrations transmitted to the shaft without having to worry about added weight.

A second reason for choosing the second concept is the placement of the x axis. In the first concept, the x-axis would be carrying all the weight of the theta axis. Upon consideration, it was decided that theta would not be a lightweight axis. This would require the x axis to move a large mass when attempting to guide fiber which would make the task more difficult than in the second concept. Given that coils are wound with an orthocyclic wind technique, the x axis would not be moving smoothly, but rather in quick jerks, or jogs, each time the jog zone passed by the guide wheel. These fast, short movements could make relative vibration between the guide wheel and the product spool a problem. The smaller and stiffer the mass being moved during jogging, the less likely such vibrations would arise. The second concept had the advantage that there was a much lower weight carried by the x axis and so that concept had a greater potential for reducing error motions.

The above two concerns were enough to push the decision in favor of the second concept. To help in visualizing the concept, a foam model was created. With this model dimensions of the different machine components were clarified and the relationships between the different axes could be clearly gauged. After working with the model, the basic structural layout concept was accepted with confidence and the detail design could progress.

3.2.3 Design Choices - Main Structure

Choosing the exact actuators and transmissions for the various axes was an important task. Given the requirements for the x axis described previously, it was imperative to choose the proper components. The axis was chosen to be a servo motor driven lead screw. It would be supported by linear, non-re-circulating ball bearings which are stiff, accurate, and smooth in operation. The servo motor was chosen with a high encoder count and the lead screw with a relatively high pitch so that the x axis could be accurately placed. This would lead to an placement resolution of about one micron in the x direction. A lead screw was chosen over a ball screw because lead screws can provide a smoother motion which was desirable on this axis. Another concern was backlash. This concern resulted in the selection of a direct drive motor and an anti-backlash lead screw nut. Much of the same type of reasoning went into the selection of the other axes components, and there is really no benefit in discussing those details here.

In the design of the actual structure that would support all of the axes, special locating techniques and machining instructions were employed in an effort to reduce error stacking. Referring to Figure 23, it can be seen that there are two large towers that support the main shafts on the machine. It was important that these shafts be as close to in line with each other as possible. The design involved in the location of the shafts will serve as an example of the types of special locating techniques and machining instructions used throughout the machine.

A very flat, large aluminum base plate was used as the reference and support surface for the entire machine. Dowel pins were accurately pressed into this surface and used to locate different structural members. The large towers were both located against the dowel pins in the base plate and bolted down. Locating in this manner ensured that the towers were placed as accurately as the dowels, which would be far superior to simply bolting the towers in place with no reference.

In addition to locating the towers, the shafts had to be located within the towers. Figure 24 shows one of the towers in which the shaft passes through

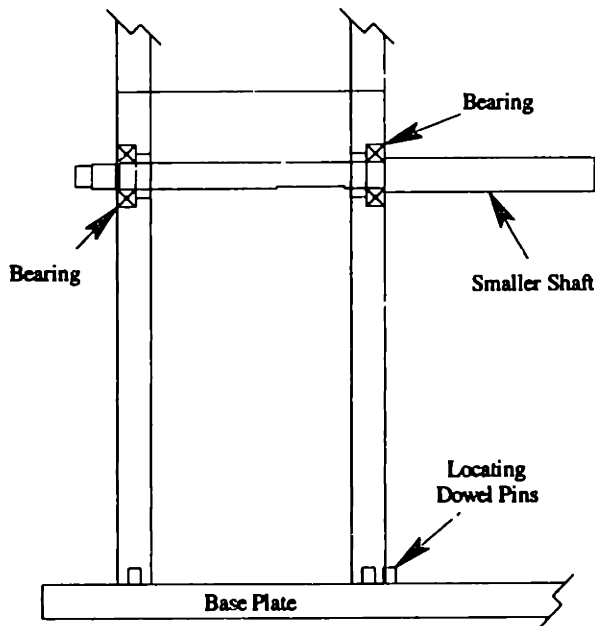


Figure 24. Smaller Shaft Located in Tower

bearings contained in the end plates of the tower. The challenge was to ensure that the two bearing holes were aligned relative to each other so that the shaft axis would be perpendicular to the tower. The machinist was instructed to machine the end plates of each tower together. More specifically, the outer profile of the plates and the bearing

hole locations would be cut simultaneously. Doing this would ensure that the bearing holes were aligned and that they were located in the same place relative to the bases of the end plates.

Another significant design choice was the size of the supporting structure. One inch plating was chosen for the base plate and the plates used in the towers. Thinner plating may have been sufficient, but there was no harm in using the extra material as it would add to the stiffness and strength of the structure. Also, an aluminum cross beam connected the tops of the towers. This beam would serve to create a closed design and would help to reduce relative motions between the two towers.

To help meet the functional requirement of aesthetics, the large plates were given a smooth finish and were black anodized. Also, the machine was supported by an industrial machine base that was purchased from an outside vendor. This base would also serve to provide a very stiff mounting surface for the rest of the machine. Finally, it was decided that the machine should have a obvious front and back as it was to be used by an operator in a production environment. This was accomplished by offsetting the main shafts towards one side of the machine and slanting the tops of the two towers. This would serve to allow easier access to one side than to the other, in effect, creating a front for the machine. These slants are visible in the machine end profile shown in Figure 25 above.

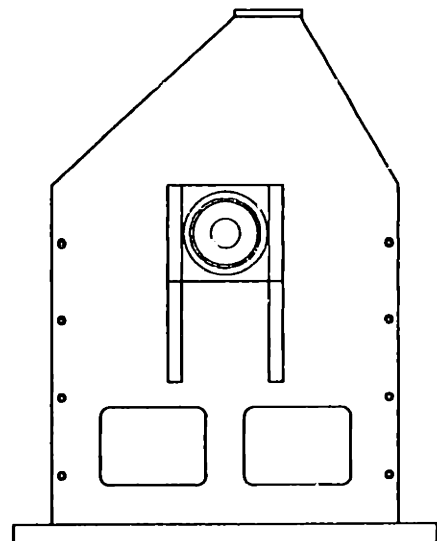


Figure 25. Machine Profile

3.2.4 Fabrication and Testing - Main Structure

The actual assembly of the machine was an arduous process because of the size of many of the components. However, the majority of the structure went together very smoothly. A few holes had to be re-drilled as the machine shop had made a few mistakes, but there were no major changes to be made. As far as performance, it was difficult to actually measure exact locations and deflections as the scale was so large. It was decided that actual use of the machine would reveal how well the structure actually performed. From measurements made with an accurate dial gauge, vibrations between the guide wheel and the product spool during winding were measured to be in the 5 micron range which would be more than adequate for good fiber placement.

While it was important that the two shafts rotate on the same axis, it was

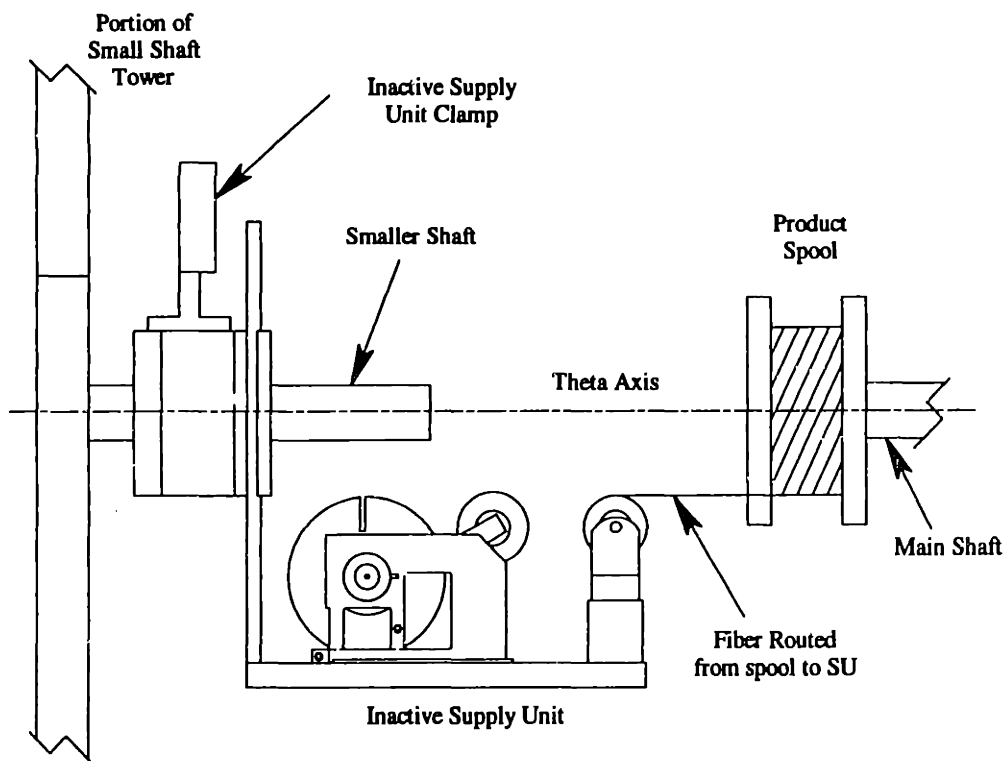


Figure 26. Fiber Routing from Inactive SU to Product Spool

not necessary make this measurement directly. Operation of the machine revealed that the two shafts were close enough to the same axis so as not to cause any problems in operation. More specifically, consider the case when one of the supply units rides on the smaller of the two shafts (Figure 26). The fiber would route from the guide wheel of this supply unit to the product spool, riding on the other shaft. If there was too much error in motion between the two shafts the fiber could fall off of the guide wheel, which would be disastrous to the current wind. Avoiding this problem was the real motivation behind the steps taken to ensure good shaft alignment. During operation, the fiber remained firmly in place on the guide wheel, revealing that shafts were more than adequately aligned to ensure successful operation of the machine.

In all respects, use of the machine showed that the structural design and axis placement functioned exceptionally well. Vibrations were very low and did not pose any problems. Also, structural deformation was hardly noticeable. There was no need to map out the motions of any of the axes as they were all aligned well enough to allow for accurate fiber placement on the product spool. A picture of the final winder is shown in Chapter 6 (Figure 74). The structure and axis placement are clearly seen in this picture.

3.3 The Main Drive System

During winding, the product spool is rotated by the main shaft which acts to draw fiber off of the supply spools and onto the product spool. The system

that actuates the rotation of the product spool is termed the main drive system and includes the motor and any necessary structure required to create the rotation. As explained earlier, the inactive supply unit (SU) has to turn as the product spool turns during winding. This means that the main drive system has to turn the inertia of a supply unit as well as that of the product spool. In addition, the layout of the drive system would have to accommodate all of the possible ride positions that the SUs could assume when in their inactive states. In other words, the drive system would have to rotate shafts on both sides of the product spool so that the inactive SU could spin with the product spool on either side of the spool. Without question, the drive system was one of the key components of the automated winder and would require careful thought to achieve a successful design.

3.3.1 Functional Requirements - Drive System

One of the most important requirements of the drive system was repeatability in positioning. Referring back to the orthocyclic wind, the jog zone occupies a specific angular position on the circumference of the coil. During winding, the guide wheel has to jog at the precise time to ensure the jog zone remains well defined on the circumference of the product spool. To accomplish this, the angular location of the product spool has to be well known. Furthermore, any time that rotation is stopped, for example during swapping, the product spool must be in a precise rotational location to ensure that any

operations being performed are done properly and do not damage the coil. For such reasons, the product spool must be able to move between positions repeatably.

In addition, the ride clamps, to be discussed in section 3.4, must also be positioned in a repeatable fashion. The coupling mechanisms that will move the SU between active position and inactive, or ride, position on the shaft will require that the ride position be within a known space each time the SUs are swapped. Any rotational misalignment of the ride positions could result in a failed coupling. While no exact number was specified for repeatability, a goal of less than 5 arc minutes of positioning error was established.

Such rotational precision also implied that there could be little or no backlash allowed in the drive system. The concept of backlash can be demonstrated with a simple example. Consider driving a car and steering by turning the steering wheel. It is possible to slightly turn the wheel back and forth without making the car turn. When the direction of turning the steering wheel is reversed, there is a mechanical delay in position of the wheels. The driver has to turn the wheel beyond this delay to get the wheels to turn. Such a mechanical delay in system output when input direction is reversed is termed backlash. While winding, the main shaft would rotate in both directions and would need to position repeatably regardless of the current winding direction. Backlash would hinder this process as it results in a degree of uncertainty of actual end point position and so it should be held to a minimum. A backlash of 1 arc minute was specified as a design goal.

Aside from repeatable positioning, the drive system must also meet the basic power requirements to turn the main shaft. On this shaft will be the product spool and the supply unit that is currently in the ride position. The system must be capable of maintaining a maximum rotational velocity of 3 revolutions per second (rps). The greatest strain on the system will be the acceleration of the shaft with an SU is in ride position. It was estimated that the SU could weigh up to 2 pounds. For the largest coil to be wound, the ride position could be as far off the shaft as 6 inches. Assuming a maximum acceleration of 10 rps², the system would have to be able to sustain a torque of about 12 in-lb. Given frictional effects and uncertainties in actual masses, a torque of 60 in-lb was actually targeted so that there would be plenty of room for error.

The main drive system design would also include the design of the shafting that would support the product spool and ride positions. It would have to be sturdy enough to resist the bending moments associated with the spinning mass of the inactive SU. Dynamic deflections of the shaft would need to be avoided. Such deflections could hinder guiding of the fiber onto the product spool. Similarly, runout of the shafting must also be held to a minimum. Any wobble seen at the location of the shaft that holds the product spool would cause a relative motion between the spool and the guide wheel. While this kind of motion could be mapped out and incorporated into the fiber guiding motion, the added complexity was undesirable.

- FRs
1. Meets basic power requirements
 2. Repeatable angular positioning and low backlash
 3. Accommodate all ride positions
 4. Low shaft runout

3.3.2 Concept Generation and Selection - Drive System

Sizing and choosing the actuator and the transmission for the drive system was not conceptually difficult. The design of the shafting posed some unique obstacles, but was also conceptually straightforward. The challenge was not in the concept selection but in the design which will be covered in the next section.

Choosing the actual layout of the drive system posed the real conceptual challenge. As discussed previously, the drive system had to be able to turn shafting on both sides of the product spool. The challenge was how to accomplish this while keeping it simple to load and unload the product spool from between the two

shafts. Figure 27

illustrates the problem.

The operator must have

easy access to the

product spool, but the

machine must be able to meet the functional requirements laid out above.

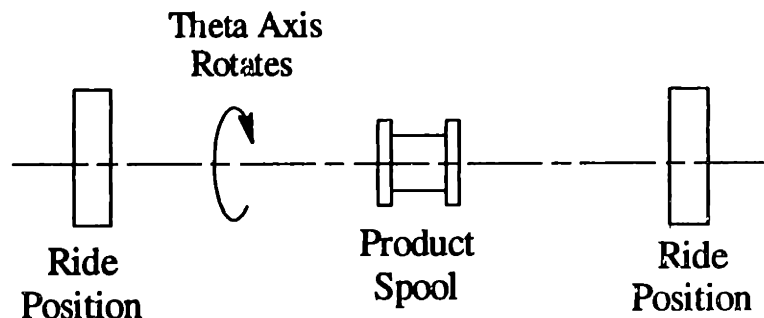


Figure 27. Product Spool is Located Between Ride Positions

Various arrangements were considered that could provide a solution to this design problem. Three of these arrangements concepts are discussed in this section.

The first arrangement involves one continuous shaft driven by a single motor (Figure 28). The difficulty in this type of layout is loading and unloading the product spool. In order that it be possible to remove the spool, there must be a coupling that allows the two shafts to be broken in the middle. Such a

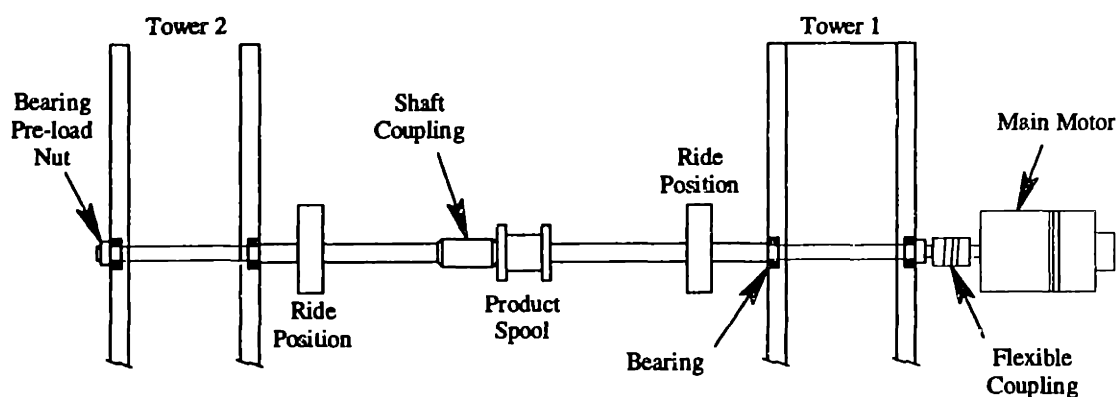


Figure 28. Continuous Shaft Concept

coupling would have to allow for misalignment and deflection of the two shafts and would need to be sturdy enough to transmit torque between the two halves of the shaft. Given the frequency of loading and unloading of product spools, the coupling would also need to be both durable and repeatable in joining the shafts together. Finally, it would have to be simple to engage and disengage to make access to the product spool simple for the operator.

Another idea incorporates a split shaft arrangement which is shown in Figure 29. A main shaft would carry the product spool and the ride positions on

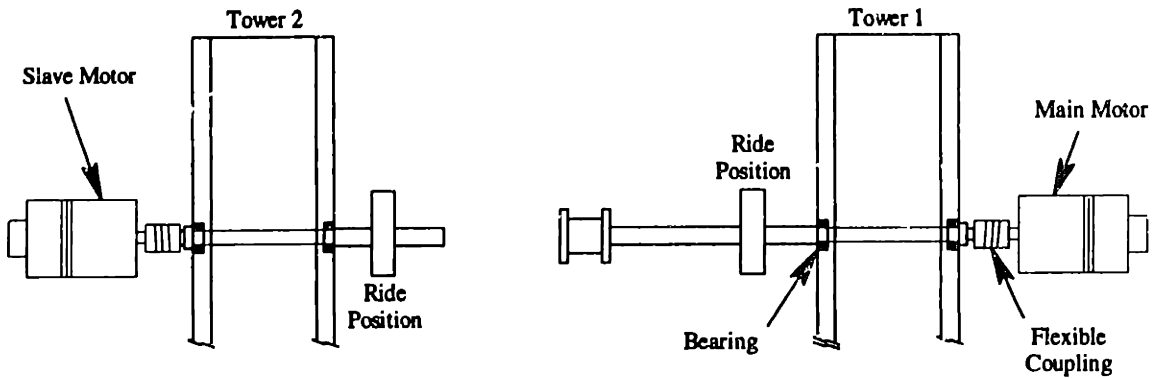


Figure 29. Split Shaft, Two Motors Concept

one side of the spool. Another shorter shaft, termed the ride shaft, would be located on the other side of the machine and would only serve to carry the other ride positions. Having the split in the shaft would allow for easy access and mounting of the product spool. Also, there would be no need to develop the kind of coupling required in the previous concept. A separate motor, one on either side of the machine, would drive each of the shafts. The motor on the ride shaft would be slaved to the one driving the product spool shaft to ensure that both shafts would always remain in the same position relative to each other.

The last concept is very similar to the previous one except that only one drive motor would be used. The main motor would drive the product spool, or main, shaft directly. A transmission would be required to transfer power to the ride shaft on the other side of the machine. Given that the distances are rather large between the two shafts, the transmission was chosen to be timing belts that would drive an intermediate, or transfer, shaft. This arrangement is shown

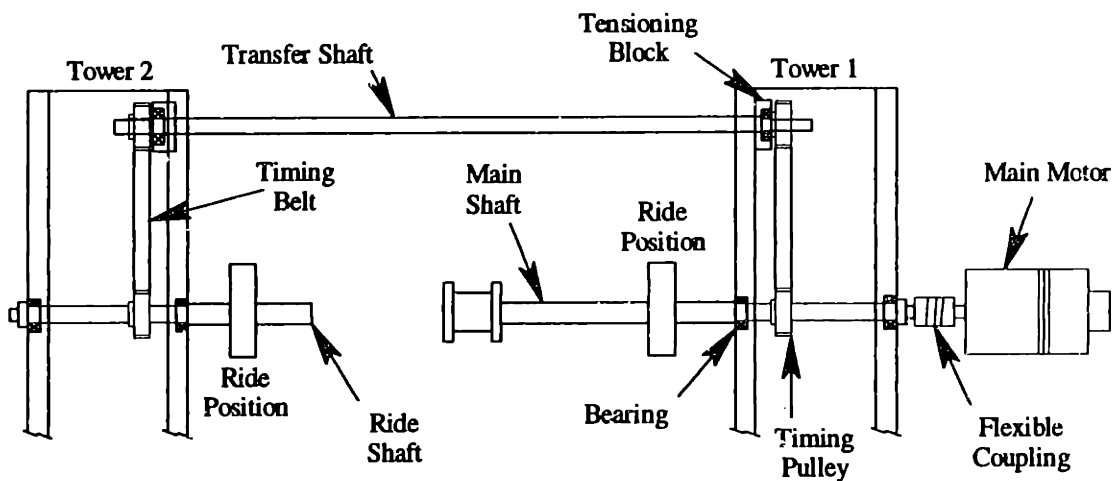


Figure 30. Split Shaft, Single Motor Concept

in Figure 30. Power would be transmitted from the main shaft to the transfer shaft through the belts. Timing belts would also serve to maintain a constant relative angular position between the ride and main shafts. A certain variety of timing belt, known as positive drive belts, would also result in an extremely low level of backlash between the two shafts.

Concept selection was rather simple in this case. Despite the simplicity of having one shaft and one motor, the possible complexity of the coupling in the single shaft concept was now welcomed. Given the many requirements that the

coupling would have to satisfy, it seemed to be exceptionally open to possible failure. Among the remaining two concepts, the single motor with the belted transmission was chosen because only one motor would be required. Having two motors would be more expensive and would involve more complex control and electronics. Also, there would always be the possibility that the slaved motor could become un-slaved or go out of control which would lead to breaking of the fiber if a supply unit was on that shaft at the time. A belted transmission would be simpler, cheaper, and would not be subject to slaving problems. However, provisions were made to allow for the attachment of a second slaved motor, as in concept 2, if the belting was found to be lacking.

3.3.3 Design Choices - Drive System

Having selected the layout of the drive system, the detail design of the shafting and transmission could progress. First, choosing the type of motor to use posed a problem. Either a direct drive or a gear motor could be chosen. A direct drive would have zero backlash but could be quite large and very expensive for the given loading requirements. On the other hand, a gear motor would be much more inexpensive and could be reasonably sized to meet the predicted loads. However, such a motor would include a gear head which have backlash by its very nature. A gear motor was found that had less than 1 arc minute of backlash in the gear head. It was decided that this level of error in positioning wouldn't affect winding. An additional benefit of a gear motor is that

it would better accommodate changes in inertia in the system. Thus, for reasons of cost and size, the gear motor drive was chosen over a direct drive.

As mentioned above, a belting system was to be used in the design. A belt would route from the main shaft to a transfer shaft. The transfer shaft would extend across the length of the machine and, on the other end, another belt would route to the ride shaft. Where exactly to place this transfer shaft needed to be decided. Referring back to the assembly drawing of the machine (Figure 23), note that a beam extends across the top of the machine. This is actually a length of square structural tubing. It was decided to place the transfer shaft inside this shaft to hide it from view.

Also, the type of belting to be used was an issue. Given that backlash was to be avoided, a “zero backlash” belt was chosen. A regular timing belt would work well for maintaining relative positioning between the shafts but can have up to several arc minutes of backlash due to the clearance of the belt teeth in the pulley grooves. A positive drive timing belt, however, has virtually no backlash, because it has specially shaped teeth that squeeze into mating grooves on the pulleys. The only apparent backlash will be caused by the belt's inherent flexibility.

Having decided on the belting to use, a method for tightening the belts had to be developed. An idler pulley could be used that pushes on the belt from the side (Figure 31). While such a method would work well, it would be difficult to access when placed inside the tower where the belting would be contained. Timing belts require frequent re-tensioning so it was desirable to make it easy to access the tensioning system.

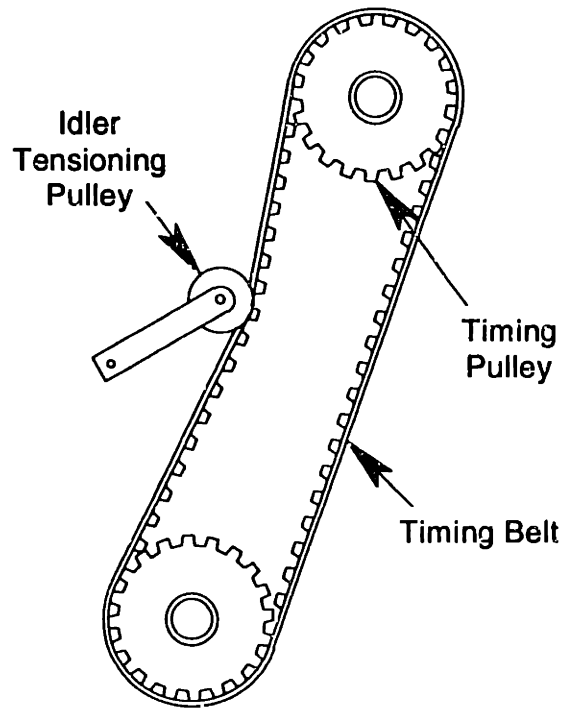


Figure 31. Idler Pulley

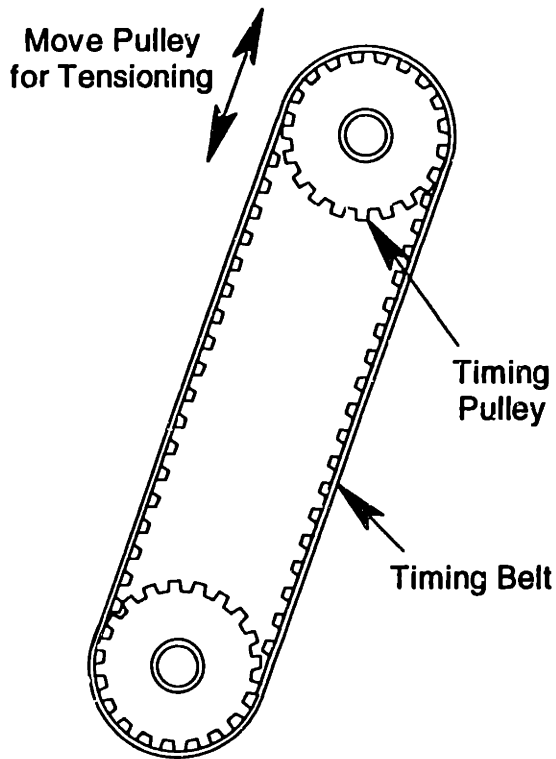


Figure 32. Center Distance Tensioning

Another tensioning alternative was to adjust the center distance between the pulleys by allowing one of the pulleys to be moved relative to the other (Figure 32). When incorporated into the towers, this method would actually allow for easy tensioning of the pulleys. A knob on the top of the machine could be turned by hand, which, through the action of a screw, would pull one of the pulleys away

from the other. The moving pulley was attached to the transfer shaft at the top of the machine. The bearing that would hold the transfer shaft was mounted in a moving block that could slide in a dovetail slot. Turning the knob move this block which would act to pull the attached pulley away from the other, tensioning the belt. This method was used on both sides of the transfer shaft to tension both belts.

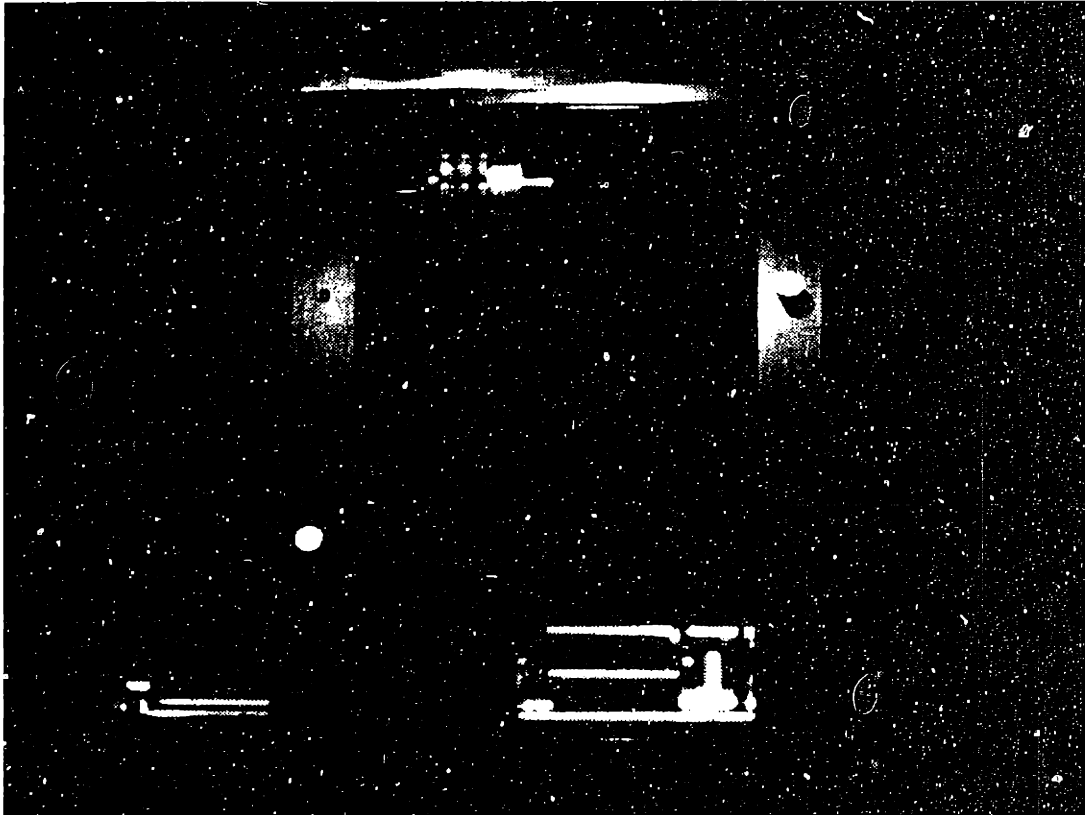


Figure 33. Belting and Tensioning Knob

Figure 33 shows a picture of the belting which illustrates the timing belt and pulleys. Also visible is the knob on top of the machine used to tighten the belt. The pistons that actuate the ride clamps, to be discussed in the next section, can also be seen within the tower.

The last important issue in the design was the main and ride shafts. Both shafts had to be able to support the rotating weights of the SUs in ride position without deflecting significantly. A simple model was developed on an Excel spread sheet for the main shaft. Worst-case loading conditions were included in the model and basic deflection equations were used to calculate bending under the load. Using the spreadsheet allowed for changing the shaft diameter to quickly determine the resulting shaft deflection. Through the use of the sheet, it was decided that a 1 inch diameter shaft could be used and would yield a maximum deflection of the main shaft of under 60 microns at the end. Given normal loading conditions, this value could be much lower. For the worst case, the deflection would not cause enough misalignment to be a problem. One question that arose was whether to use a hollow or solid shaft. The hollow shaft would have the advantage of deflecting less under its own weight, for an equal diameter. However, hollow shafts are expensive and involve special design considerations. Solid shafting was chosen as it could meet design requirements, it would be simpler to design, and it would be cheaper to implement.

The last issue to be considered in the design of the shafting was the specification of the machining instructions. For reasons of aesthetics and maintenance, stainless steel was chosen for all parts requiring steel as it doesn't rust and has an attractive finish. However, using this material led to some problems in the machining of the shafts. Of primary importance was the runout at the end of the main shaft which would result in error motions of the product

spool. If the shaft were not straight or the end face were not perpendicular, the product spool would wobble both axially and radially. Radial motions were not as important as the guide wheel distance to the product spool was not critical to good fiber guiding, meaning it could vary as much as several fiber diameters and still result in good guiding.

The real concern was axial motions of the product spool as this could lead to significant guiding errors. Special machining instructions were included on the machine drawings that specified, using geometric dimensioning and tolerancing, extremely tight tolerances on end face runout and total circular runout. The shops that were contacted to machine the shafts would only agree to meet certain tolerances given the difficulties in working with stainless steel. During machining, this material tends to warp due to thermal stresses which makes it difficult to meet straightness tolerances. The shop that was chosen agreed to meet a total circular runout specification of .002 inches and an end face runout of .0004 inches.

It was decided that these tolerances should be acceptable for winding, although the circular runout was thought slightly high. To address this concern, another special machining instruction was included on the machine drawings. As will be explained in section 3.5, the product spool is mounted to the shaft by inserting a dowel pin on the spool into a hole in the end of the shaft. This dowel pin serves to locate the center of the product spool to the center of the shaft. The shop was instructed to measure the circular runout of the shaft before boring this hole into the end of the shaft. They were then to adjust the hole

position by this measured runout. Doing this would effectively place the center of the product spool in line with the center of rotation of the shaft, rather than the center of the shaft. Thus, the product spool would not see the circular runout that is inherent to the shaft. Figure 34 illustrates this design approach.

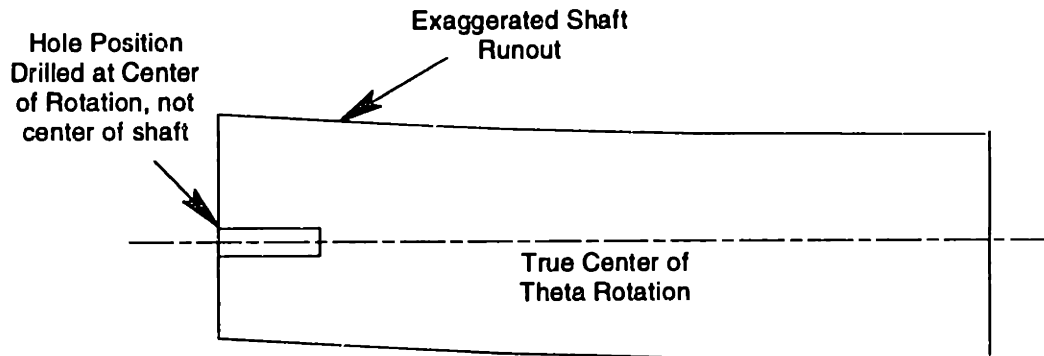


Figure 34. Adjusting Hole Position to Accommodate Runout

3.3.4 Fabrication and Testing - Drive System

Assembling the drive system on the machine actually went very smoothly. The arrangement of the shafting proved to work well. With the belts in place and tensioned, one shaft was held in place and a torque was applied to the other. As long as the belts were tensioned well enough, the relative movement between the two shafts, or the backlash, was smaller than could be measured with the available tools. The backlash in the gear motor was noticeable and was estimated be around 1 arc minute. Through actually winding and repositioning of the main shaft in rotational position, it was found that this backlash did not cause any problems in the winding or with the alignment of the ride clamps during swapping. As far as power transmission to the main shaft and to the ride shaft, the drive system was a success.

The runout on the shafting was measured using a dial gauge with .0001 inch resolution. Figure 35

shows the setup used to make this measurement. End face runout was measured to be .0004 inches and circular runout was .002 inches, both just within specification. With the product spool in place, the same setup was used to measure the axial and circular runout again. Axial

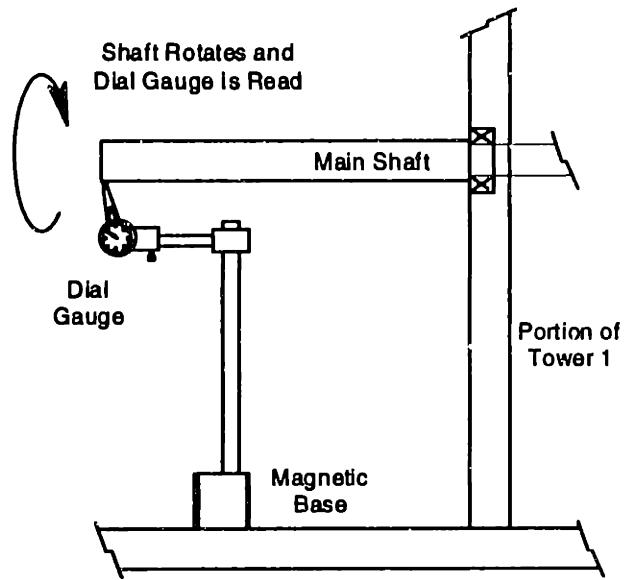


Figure 35. Circular Runout Measurement

runout increased slightly because of the runout in the product spool. However, the circular runout was measured to be under .001 inches. The circular runout of the spool itself was not detectable with the dial gauge used. Hence, it seemed that the adjustment of the hole location in the end of the main shaft did actually improve the total circular runout seen by the product spool.

By using a similar gauge setup at various positions along the length of the shaft, the deflection of the main shaft could be measured. The difference between the reading on the shaft at the end and near the support tower was about 30 microns. This would mean that during winding, there would be a static deflection of the shaft that would cause a slight angular misalignment of the product spool. Calculations showed that this would translate into less than .0003 inches of vertical misalignment along the width of the product spool.

Given that this vertical alignment is not critical to fiber guiding, this slight error would not even be noticeable during winding.

3.4 Ride Clamp

The ride clamp is the mechanism that secures the supply unit to the main shaft when the SU is in its inactive position. From a winding process point of view, the ride clamps would need to include some basic functionality. First, they would have to engage and disengage as necessary and rigidly secure the SU into place. With respect to the topology of the entire winder, there would need to be two such clamps, one on either side of the product spool so that all ride

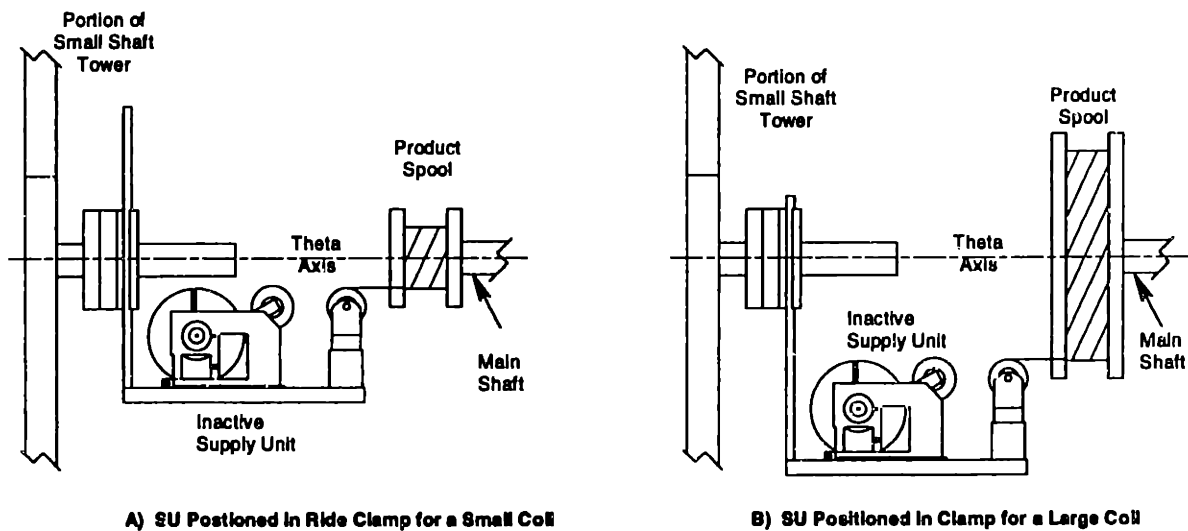


Figure 36. SU Positions in Clamp for Different Spool Sizes

positions could be accommodated. Also, the clamps would have to accept supply units from either side of the main shaft, depending on which one was swapping into place at the time. Finally, the clamps would have to allow for the

supply units to be clamped at different distances from the shaft. Figure 36 shows that the supply unit could be located at various distances from the shaft depending on the coil size and guide wheel position in the z direction within a given wind.

3.4.1 Functional Requirements - Ride Clamp

In addition to the basic functionality mentioned above, the design of the ride clamps must also meet several functional requirements. The first of these requirements is clamping force. The supply units would weigh approximately 2 pounds. Given the largest coil that would need to be wound, this two pound mass could be placed up to 6 inches off the main shaft. And, the shaft would be turned at a maximum velocity of 3 revolutions per second (rps). For safety concerns, it was actually decided to assume a maximum velocity of 5 rps. Simple calculations showed that the clamps would have to then resist a centripetal force of about 10 pounds. Also, assuming an appropriate acceleration, the clamp would need to resist the various torques that could cause the supply unit to twist out of position. The largest of these torques was calculated to be about 30 in·lb.

Not only would these forces need to be resisted during motion, but there could be very little shifting of the supply unit while in ride position. During swapping a coupling mechanism would grasp the SU when in ride position, the ride clamp would release, and the SU would be swung into active position. The

coupling mechanism would rely on the SU being in a certain position when the mechanism attempted to couple with the SU. Any shift of the SU in ride position could impair the action of the coupler. A large enough shift could result in the fiber falling off of the guide wheel on the inactive supply unit. Such an event would be catastrophic to the current wind. For similar reasons, the ride clamp had to be repeatable. That is, each time the SU was clamped into position, it would be located in the same position in the machine space, or as close to it as reasonably possible.

The importance of positioning implied that the ride clamp would have to have alignment features that would ensure the supply unit would not shift during rotation and would be in the same position each time it was clamped. The clamp was designed to minimize positioning error, and the coupling mechanisms were designed to accommodate a significant amount misalignment. The coupling mechanism design is discussed in the thesis by Steve Lin.

- FRs
1. Adequate clamping force
 2. Repeatable positioning
 3. Minimum shifting of the SU

3.4.2 Concept Generation and Selection - Ride Clamp

The first concept considered is illustrated in Figure 37. The idea involved using motors to provide the clamping force and friction between two surfaces to hold the SU in place. An arm would extend from the supply unit and a clamping pad attached to the motor would press the surface of this arm against a flat, high friction surface on the ride clamp. Alignment features on this surface would

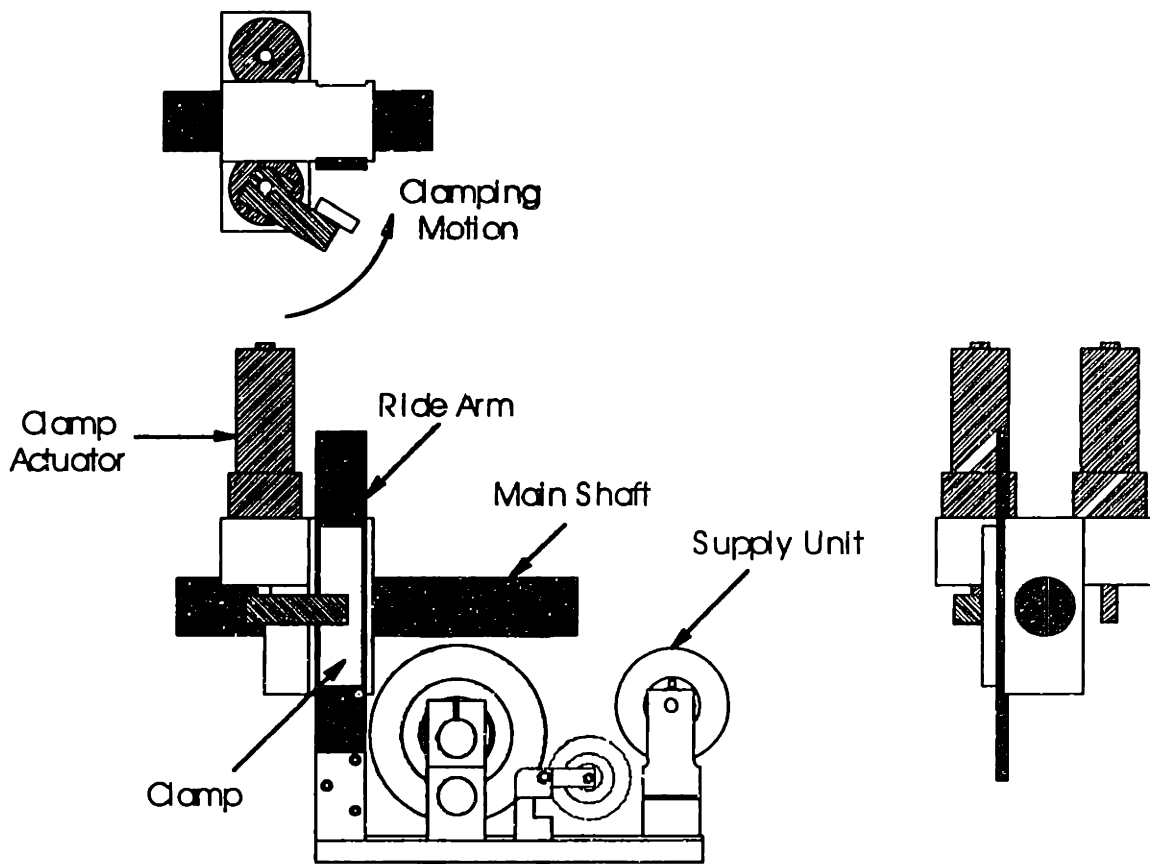


Figure 37. Motorized Ride Clamp Concept

locate the arm in a specific position on ride clamp and would aid in resisting twisting moments. The arm could be clamped into place anywhere along its length, allowing for the different clamping positions required.

Another concept was similar but relied on toggle clamps to provide the clamping force. Such clamps move between two positions through the motion of a four bar linkage. For a small actuation force, a large clamping force can be

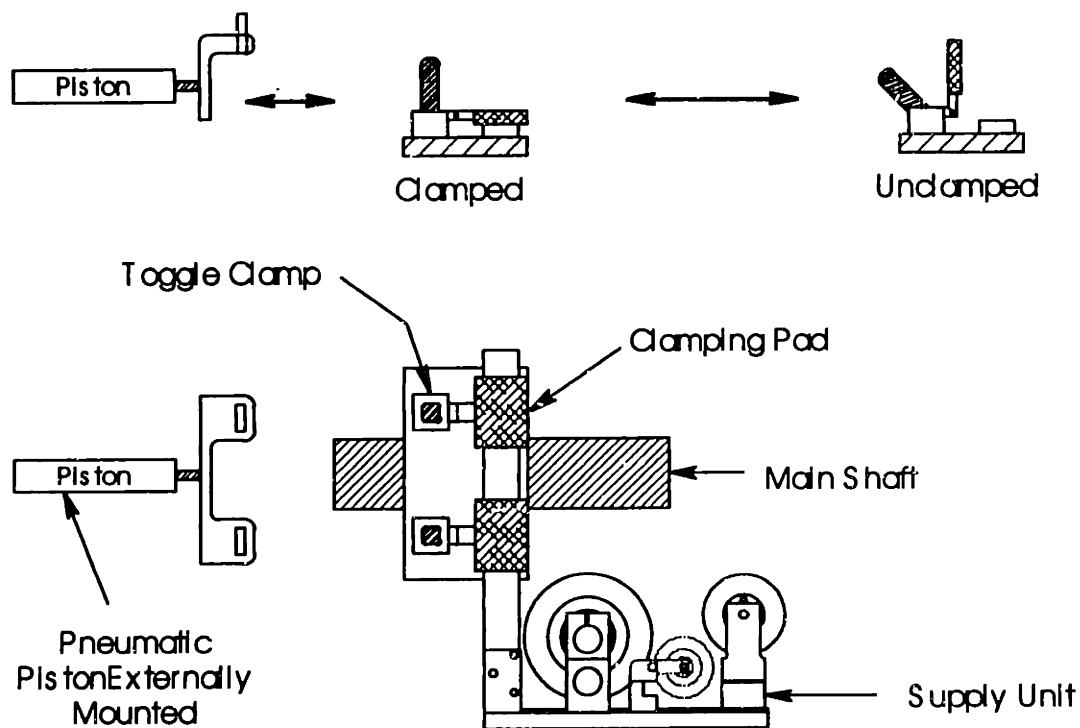


Figure 38. Toggle Clamp Concept

generated. The clamps would be actuated by pneumatic pistons that would be attached to the towers on either side of the machine. This arrangement is portrayed in Figure 38 above. The reason external actuation can work with the chosen topography is swapping always involves the same rotational location of the ride clamps. The pistons would extend from the faces of the towers and actuate the clamps as necessary. A separate, smaller actuator would ride on the end of the main piston. This second actuator would extend when then toggle

clamp had to be pulled open and would move out of the way when it was to be pushed closed.

The last concept considered was virtually the same as the toggle clamp idea except that springs rather than the action of a linkage would provide the clamping force. This concept is depicted in Figure 39. Such a mechanism

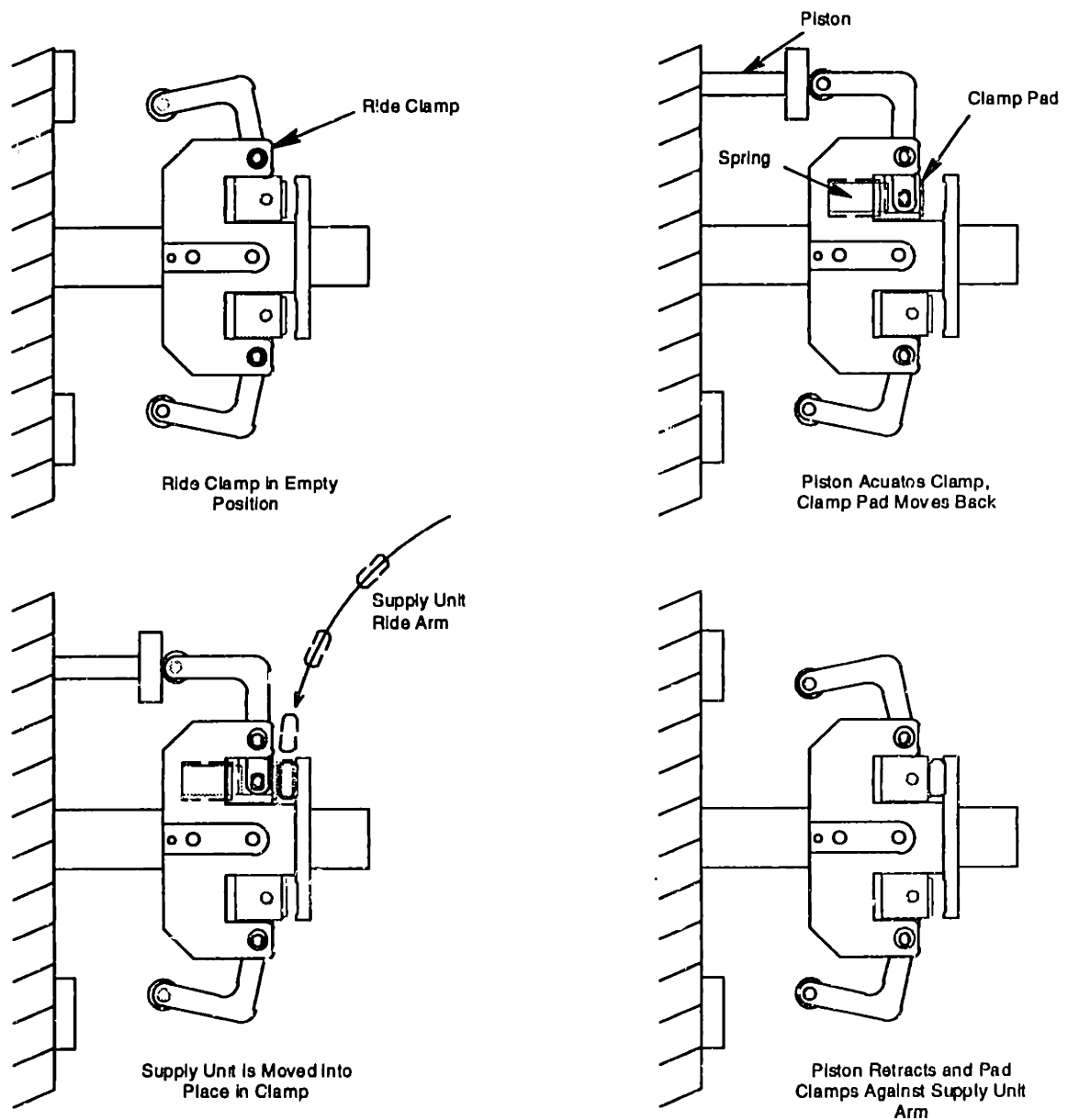


Figure 39. Externally Actuated Spring Concept

would involve springs that push against a clamping pad. The pad would press the arm extending from the SU against a flat, high friction surface, as with the previous two concepts. However, the default position of this clamp would be closed and the pistons in the towers would extend to open the clamp. Only one actuator would be required to open and close the clamp. The lever arm shown in the figure would allow for the use of high force springs for clamping and much lower force pistons to open the clamp. The same kind of alignment features would be involved as with the previous concepts.

Upon initial examination, it was difficult to gauge which concept had the greatest chance of success. To help in the selection process, working prototypes of the motor and toggle clamp concepts were built and tested. These prototypes are shown in Figure 40 and Figure 41. Exhaustive tests revealed no

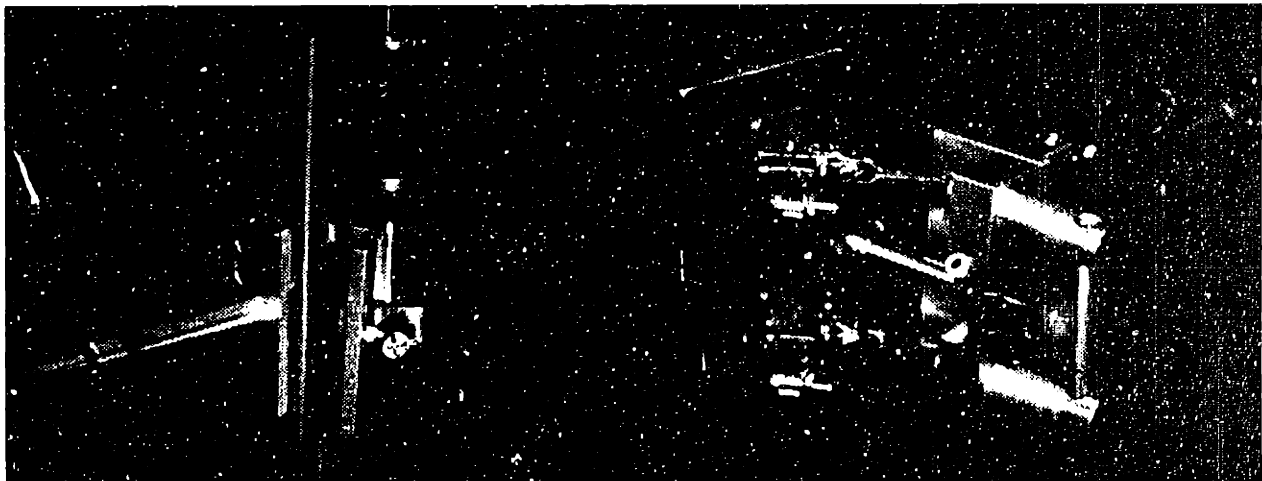


Figure 40. Motor Prototype

Figure 41. Toggle Clamp Prototype

significant functional advantage of either concept. Both seemed equally capable of meeting the basic functional requirements so additional selection parameters were considered. Cost and complexity became the deciding factor between the two concepts. It was decided that the externally actuated toggle clamp concept

would be simpler and cheaper to construct. The motor concept would require that electrical power be available on the spinning shafts. To supply electric power to a spinning shaft, electrical couplings must be used which would add significant cost and complexity to both the mechanical and electrical aspects of the machine. Also, the cost of the pneumatic actuators was predicted to be much cheaper than motorized ones.

A safety concern also came into question over the motorized concept. If power was to be lost for any reason, the motor would no longer provide clamping force and the supply units would be free to fall out of place. If the shaft was spinning at the time, this could pose a serious danger to the operator. Some quick calculations also revealed that even a highly geared motor, which are difficult to back-drive on power loss, would not be able maintain adequate clamping force.

The spring driven clamp concept was actually generated after the prototypes were tested. Using the toggle clamps was undesirable because two separate actuators would be required to open and close the toggle clamps. By using springs, only one actuator would be necessary. For this reason, the spring concept was actually chosen and detail design of that concept progressed.

3.4.3 Design Choices - Ride Clamp

Sizing the springs posed one of the greater design challenges in the ride clamp. Of primary concern was that the clamp could hold the supply unit given the 10 lb centripetal force at maximum shaft velocity. By the nature of the clamp, friction between the arm extending from the SU and the surface of the ride clamp would provide the restraining force. Assuming a coefficient of friction of .3, the springs were sized to give a holding force of 20 lbs. This would ensure that the SU would be safely secured under any possible condition. The range of the spring also had to be chosen. To allow the supply unit to move into place, the clamp had to open a certain amount. An opening distance of .25 inches was

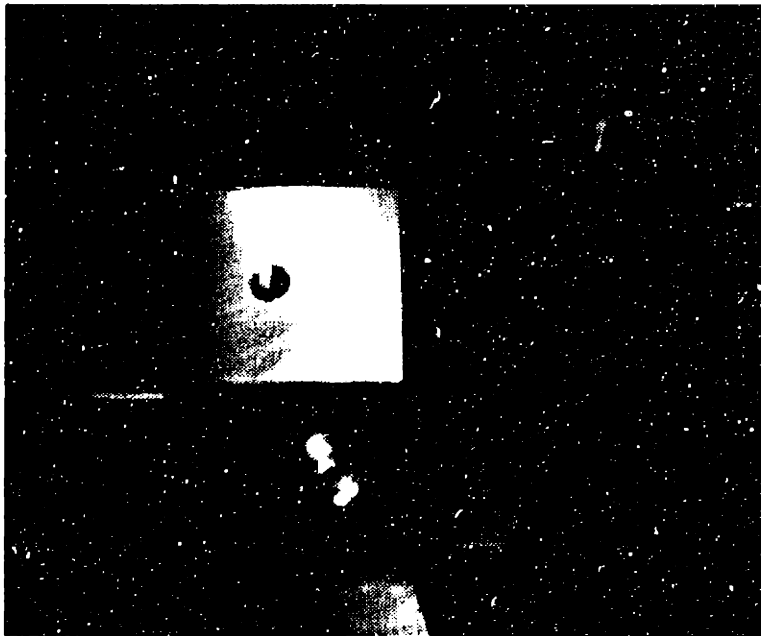


Figure 42. Spring and Clamp Pad in Ride Clamp

chosen to be adequate.

With that information, the springs were selected and fit into a supporting structure that would mount onto the shafts. Figure 42 shows the springs located in the ride clamp structure, the clamp structure, and the bronze clamping pads.

Note that the bronze pads make a close sliding fit with the ride clamp structure so that they are only free to move in the clamping direction. Restraining the clamps in this way prevents the pads from twisting or shifting which could lead to a shift in the supply unit arm position in the clamp.

To open the clamps, levers were designed into the structure that would be actuated by the pistons that were housed in the side towers. The levers were sized to give a two to one mechanical advantage to reduce the necessary size of the pistons. Having sized the springs and the lever, the pistons were then chosen. Rather than pistons, devices called linear thrusters were used. These devices are actuated by pistons and include additional support structure mounted on linear slides. The additional support prevents damage in the piston and greatly extends the actuator's life.

Locating features were included to align the SU arms during clamping and to help prevent the SU from shifting during rotation. These features would also aid in resisting twisting moments. This portion of the design actually went

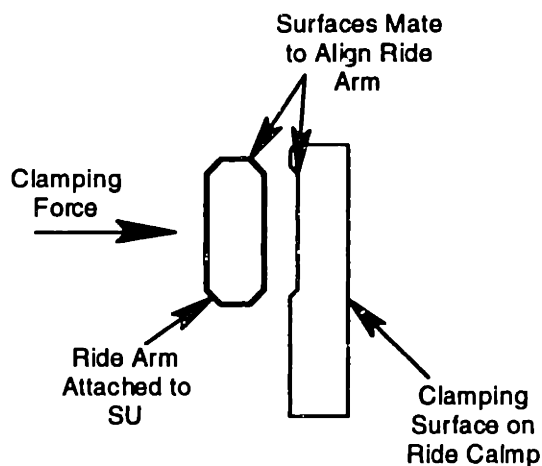


Figure 43. Mating Surfaces, First Concept

through two iterations. The first attempt involved mating a shaped ride arm to a shaped surface. The shapes would act to restrain and locate the arm when clamped into position. Figure 43 illustrates this concept.

While the design worked as expected in locating the ride arms repeatably, it was not entirely successful in resisting twisting moments. The clamp had been designed to resist a predicted twisting moment. However, the predicted moment was actually less than the actual moment. A factor of safety

had been included in the design, so it could resist the twisting to a degree, but the final clamp performance was not considered sturdy enough for actual machine operation.

A second iteration was designed and built that targeted resisting the twisting moments while retaining the locating ability. This design used a tongue and groove technique shown in Figure 44. Chamfers on both the tongue and groove would aid in aligning the SU to

the ride clamp during clamping. With the tongue and groove mated, the SU would be well aligned and the contact between the tongue and groove would serve to resist the twisting moments. In addition to this differing mating technique, the mating surfaces were widened by 50 percent

to increase the inherent moment resisting capability of the contacting surfaces.

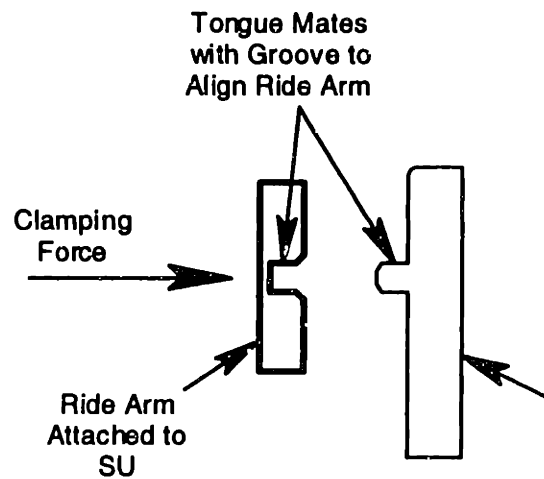


Figure 44. Tongue and Groove Mating Concept

3.4.4 Fabrication and Testing - Ride Clamp

The actual assembly of the ride clamp went smoothly. A few parts were machined out of specification and had to be re-machined, but that posed no real problem. The real test came when the supply unit was clamped into position and the holding power of the clamp could be tested. By hand, various forces were applied to the SUs in an attempt to generate a shift in the clamp (Figure

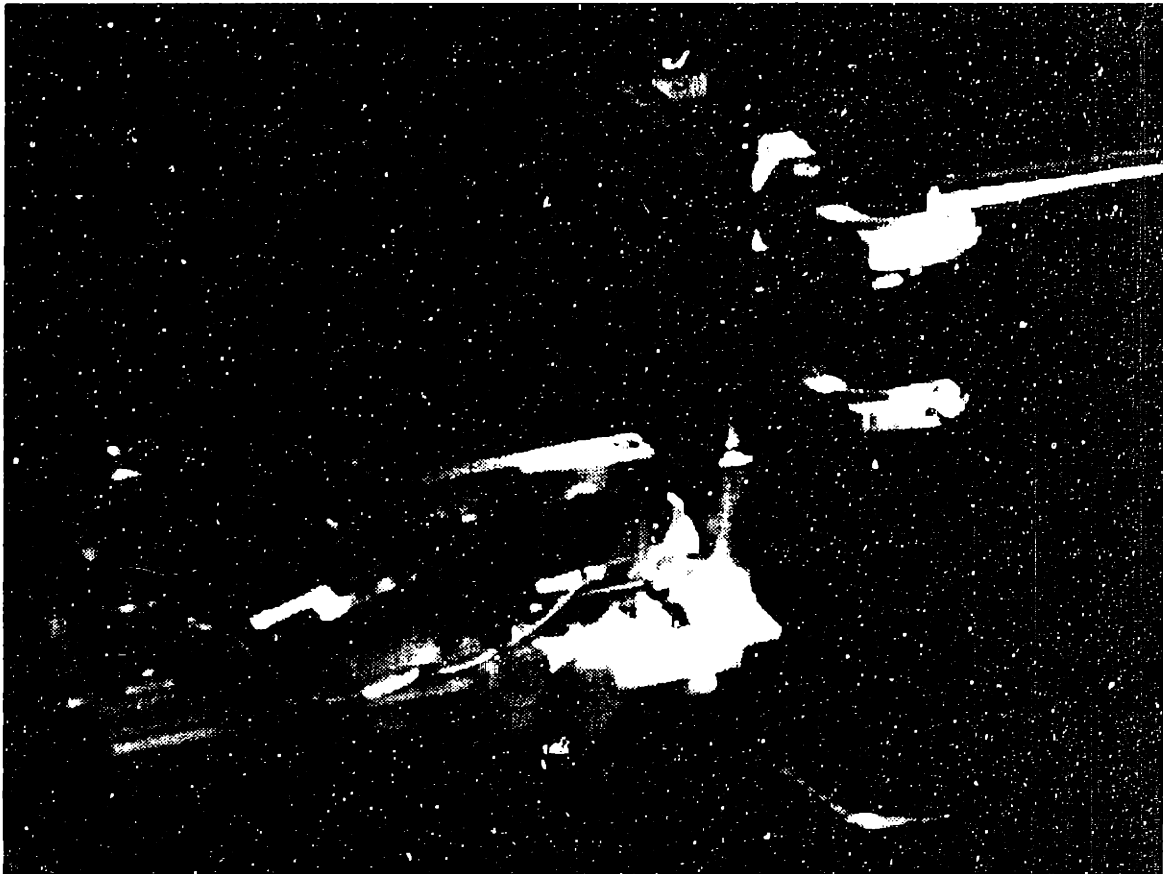


Figure 45. Testing Holding Power of Ride Clamps

45). This test revealed that the first iteration of the clamp was not sufficient for actual machine operation. When the second design iteration was incorporated, the ride clamp could more than adequately resist any twisting moments applied by hand.

The next test was to spin the shaft with the supply unit in ride position with fiber routed from the guide wheel to the product spool. The test would show if the SU shifted enough to cause the fiber to fall. This test would also show if the SU could be returned close enough to the starting position to allow for successful coupling with the retrieving coupling mechanism described earlier, termed the swing arm. With the SU in place, the shaft was spun at its maximum velocity and returned back to the starting position where it was coupled back to the retrieving mechanism. The clamp worked extremely well, holding the SU firmly in place under the most strenuous conditions that the machine could create. There was no measurable shifting of the SU in the clamp and it successfully coupled to the swing arm on each attempt. After extensive testing, it was concluded that the clamp functioned well within the design specifications.

As with the main structure, aesthetics were also important in this design

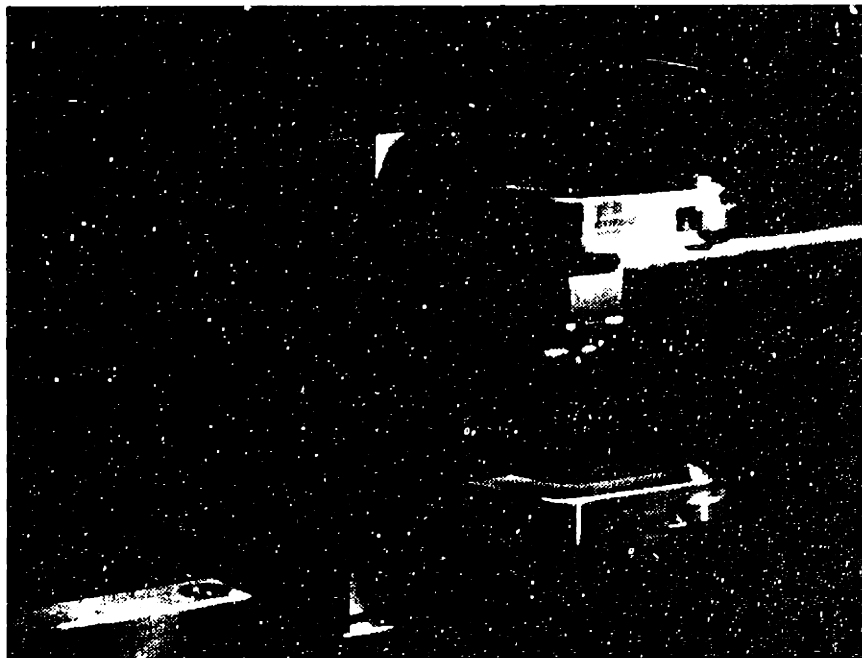


Figure 46. Ride Clamp on Machine with SU in Ride Position

so materials were chosen for their appearance as well as their strength properties. Figure 46 shows the final form and appearance of the ride clamp.

3.5 Product Spool Mount

While a somewhat minor item in the overall system, the method for mounting the product spool had significant bearing on the success of the final machine. If the product spool is not mounted repeatably or comes loose during operation, the coil being wound may be compromised. Also, if it takes too long to change coils, over the course of a thousand coils, significant operating time would be lost. And of course, if mounting or unmounting were so precarious that coil or fiber damage could result, that method would not be allowable.

The challenge was to develop a method for mounting the product spool that would simply and positively secure the gyro coil to the end of the main shaft. Given that this winder was to be used in a production environment and operated by production personnel, the mounting method had to be robust and easy to use. After an operator had wound hundreds of coils, it would be a simple matter for that operator to make a small mistake in the mounting of the spool. For this reason, the goal was to develop a method that involved simple smooth motions and that would guarantee repeatable mounting. However, given that this was a smaller system on the overall winder, too much time could not be spent on the design of the mount. In other words, it would eventually be decided to choose a method that worked well enough rather than one that worked ideally.

3.5.1 Functional Requirements - Product Spool Mount

From experience with the test-bed winder, it was determined exactly what basic functional requirements of the mounting system would allow for good winding. Angular position of the spool relative to the main shaft was important, but not critical. It was decided that a method that was repeatable to .2 degrees would be more than adequate. For the size of the largest size spool required, this would translate into about .005 inches around the coil circumference of error in fiber placement. Given the results of winding with the test-bed, this was decided to be more than adequate. Repeatability in the other degrees of freedom of the product spool also had to be considered. If the spool were to be cocked or placed incorrectly axially along the main shaft, the chances of winding inaccuracies would be increased. In addition to repeatability, holding power was also important. If the coil were to shift during winding, subsequent placement of fiber would no longer be correct relative to the coil and could result in winding errors. Thus, a method that did not allow for shifting would also be required. Finally, as mentioned above, the method of mounting had to be simple and consist of simple motions, involving a minimum number of steps.

- | | | |
|-----|----|----------------|
| FRs | 1. | Repeatability |
| | 2. | Rigid Mounting |
| | 3. | Simplicity |

3.5.2 Concept Generation and Selection - Product Spool Mount

The simplest concept involved screwing the product spool into the end of the shaft. Either a bolt or thumb screw could have been used to secure the spool and some locating feature, such as spheres in grooves, could ensure the angular location. An illustration of this concept is shown in Figure 47.

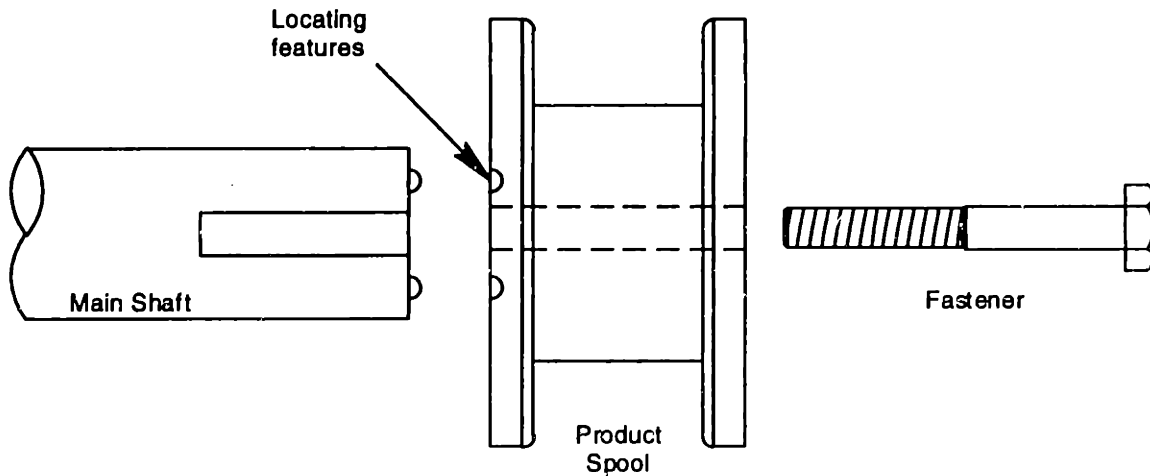


Figure 47. Screw Mounted Spool Concept

Another idea also included a screw for the tightening force and aligning features, however the concept differed in some important ways. A large dowel in the product spool would insert into a hole in the end of the shaft. A thumb screw in the shaft would be screwed against the dowel. This screw would never be removed from the machine and would involve only a few turns to secure the spool. The outer face of the product spool would be pulled against the face of the shaft by the action of the screw turning against a slant cut into the central dowel, as shown in Figure 48. A smaller, offset dowel would provide angular

location upon insertion into a mating hole on the spool. Friction would act to keep the spool held rigidly in place.

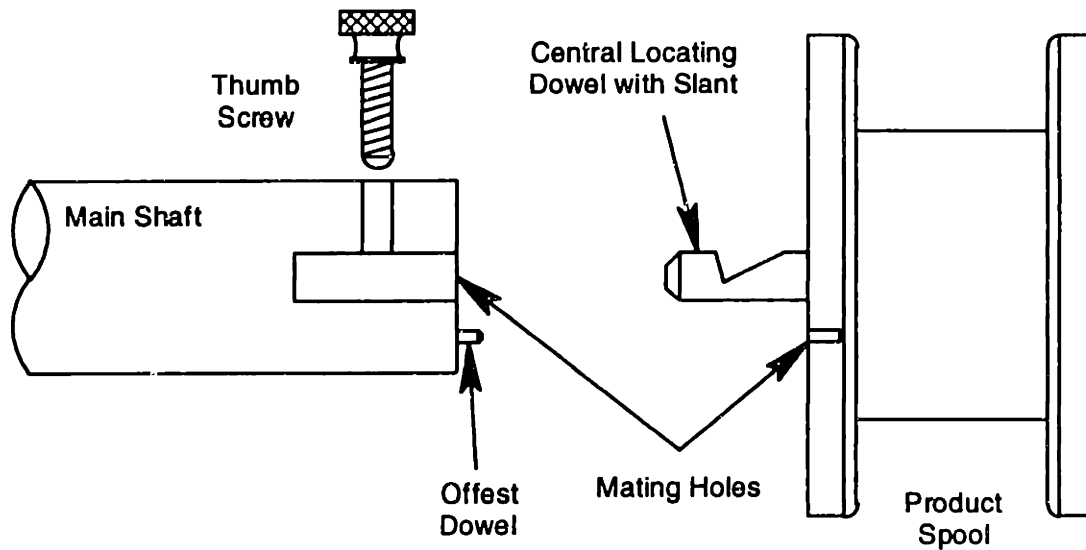


Figure 48. Thumb Screw on Dowel Concept

A third idea relied on magnets to provide the holding force for the spool. As in the previous idea, a central dowel would locate the spool on the shaft. A second, offset dowel would provide angular location. However, rather than tightening a screw, the product spool would simply be brought near the end of the flange and the magnets would snap it into place (Figure 49). Something like an ejector pin could be used in removing the spool.

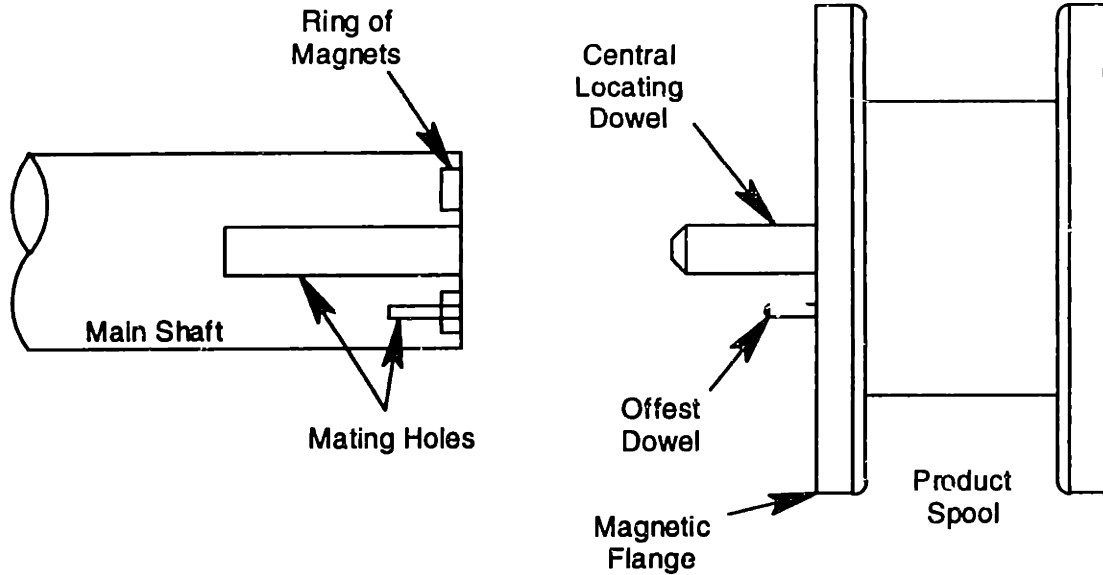


Figure 49. Magnet and Dowels Concept

Given the ease of use possible with the magnet concept, a quick mock-up of the idea was put built to demonstrate the concept (Figure 50). The idea of using dowels for location seemed to work quite well, but the magnets posed a problem. While it seemed that, with proper design, they could provide an adequate holding force, it was not possible to smoothly remove the spool from the shaft. With the possibility of damaging the coil, the idea of using magnets was decided against. The mock-up with the

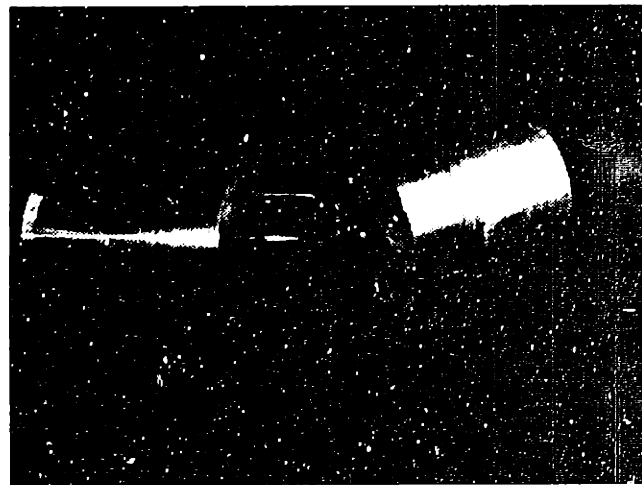


Figure 50. Magnet concept mock-up

magnets proved that using dowels for location would work well and dowels were kept in the design. Using the thumb screw against the slanted dowel seemed to be the most promising concept and was selected for the final design.

3.5.3 Design Choices - Product Spool Mount

In the design of the mount, several important issues had to be considered. The offset dowel pin had to be sized and located so that the functional requirement on repeatability could be met. With this in mind, the pin was placed about .375" from the center of the shaft and was sized to have a .001" clearance with the mating hole on the product spool. This would exceed the .2 degrees of repeatability specified and would provide for simple alignment of the spool to the end of the shaft.

The central dowel pin containing the slant actually posed a bit of a challenge. It was decided that a smooth, ball end thumb screw would be used so that the end of the screw would not dig into the dowel when the screw was turned. The exact slant had to be chosen so that it would provide adequate pulling force when the screw was tightened. Also, the slant had to be shallow enough to allow for the width of the screw, machining tolerances, and some additional clearance. In the end, a 30 degree slant was chosen as calculations revealed it would provide more than enough pulling force and would be shallow enough to allow adequate axial motion of the dowel in the shaft end.

Another issue dealt with a possible over-constrained situation with this design. It was decided that the face of the spool against the face of the shaft would be used for location in the two possible tilting directions and that the central dowel would be used to locate the center of the spool to the center of the shaft. This meant that the spool and shaft surfaces had to be machined to very tight flatness and runout tolerances to provide for accurate location. In addition, the pin and hole locations also had to be placed accurately. The problem involved the central dowel pin. If the dowel were to be rigidly fixed to the spool, by a press fit for example, and was not exactly perpendicular to the spool surface, the spool could cock when tightened against the shaft end. The design would be over-constrained as two separate mechanisms would be attempting to produce the same alignment. The dowel would be attempting to locate both the center position and the

angular location which was not meant to be the case.

The solution was to use a ball-end on the dowel as shown in Figure 51 which would provide center location but not tilting location. The dowel would insert into the

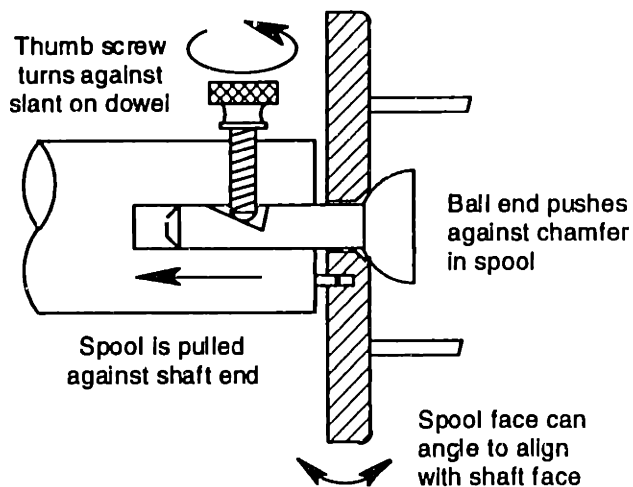


Figure 51. Ball End Dowel Actuation

hole in the shaft and be pulled in as the thumb screw was tightened. The ball at the end of the dowel would press against an accurately machined chamfer on

the inside of the flange of the product spool. By using the ball end, the spool face would be free to angle as necessary to press flat against the shaft end, but the force of the ball on the chamfer would locate the center of the spool to the center of the shaft. A retaining ring would keep the pin from falling out of the product spool when not in use. To keep the pin from rotating in the spool, a small pin in the spool flange would engage with the retaining ring on the pin (Figure 52).

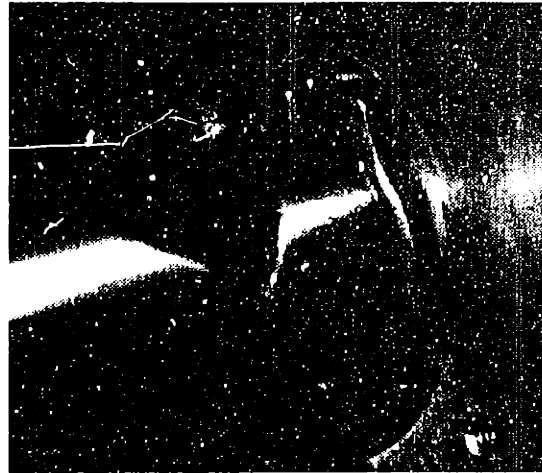


Figure 52. Slanted Dowel with Retaining Ring

3.6.3 Fabrication and Testing - Product Spool Mount

The assembly went smoothly, presenting no real problems. When tested with the main shaft, the concept worked quite well. With one hand, an operator could hold the product spool and insert the protruding dowel into the hole in the end of the main shaft. Simply by twisting the wrist, the angular alignment of the spool with the shaft could be adjusted until the small, offset dowel mated with the appropriate hole in the spool. With the other hand, the thumb screw could be turned which would draw the spool face against the shaft face. In terms of ease of use, the method worked quite well.

In terms of performance, the spool would be located to the same position as well as the sensor used (a dial gauge with .0001" resolution) could indicate in all

directions but rotation. In the rotational direction, the direction corresponding to rotating the spool relative to the end of the shaft, the placement was only as good as the locational ability of the offset dowel. It was measured to be repeatable to +/- .002" at a 2.5" product spool flange diameter. This corresponds to about .2 degrees of possible error, just barely within specification. It was noted, however, that the mating hole for the offset dowel was actually machined larger than specified which acted to increase the error in the design. Regardless, the mounting method proved to be adequately repeatable, secure, and simple to use. The following image shows the selected mounting method in action (Figure 53).

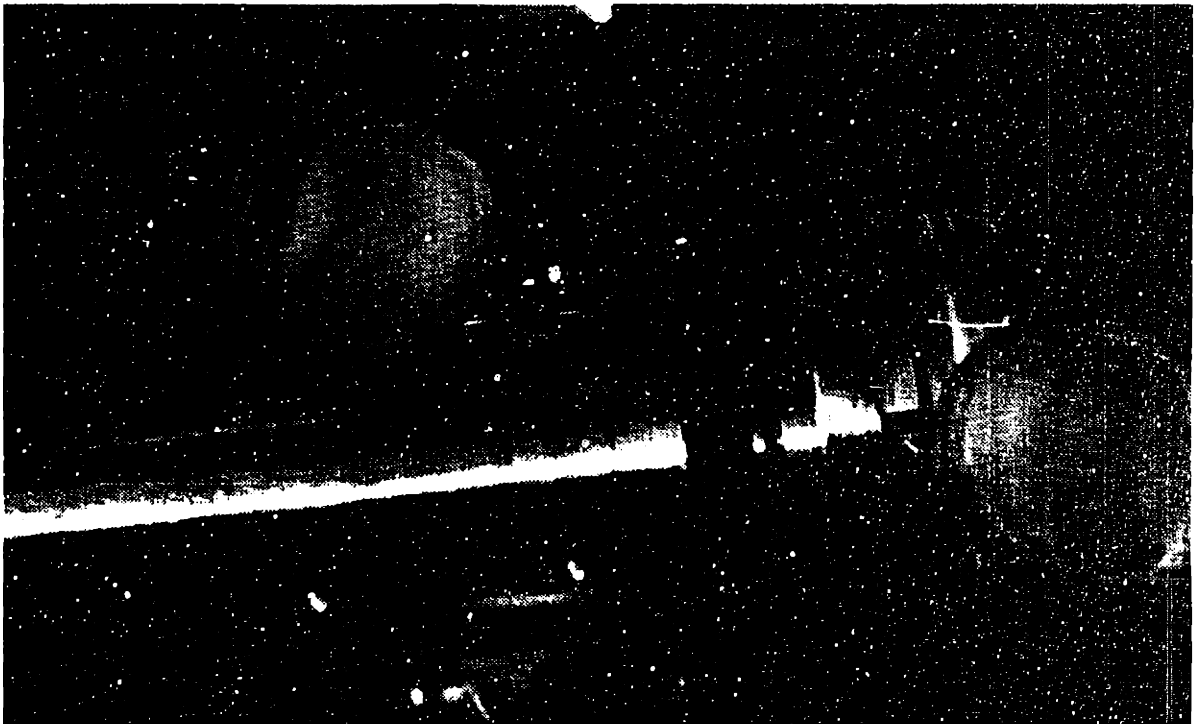


Figure 53. Mounting the Product Spool

Chapter 4 - Drive System Characterization

The main drive system on the automated winder presents an interesting control problem. A motor drives a shaft through a flexible coupling. The shaft transfers power through belting to a second shaft. That shaft transfers power to a third shaft through another belt. Both the belts and the coupling are flexible elements in the system. The torsional flexibility of the shafts is much higher than that of the belts and coupling and is therefore ignored. Each of the shafts are supported by pre-loaded bearings which provide a significant amount of damping. Both belts have a one to one transmission. If the system were perfectly rigid, when a command is sent to the motor to move to a position, all the shafts should move the same angular position. However, the real system contains flexibility and damping which complicate the dynamics between each of the shafts. With the system in motion, the angular position of the third shaft may not follow that commanded to the first shaft by the motor unless the system is

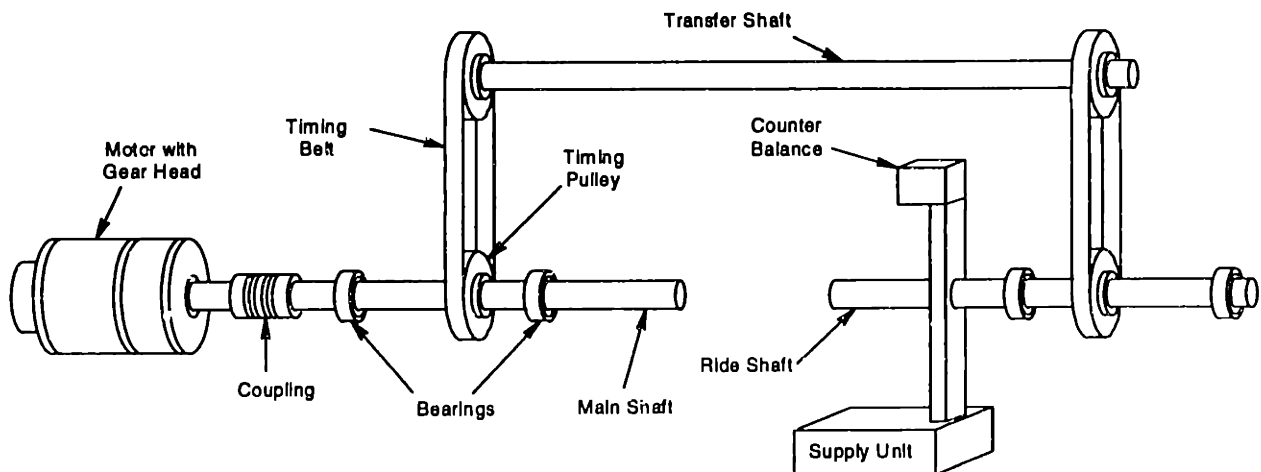


Figure 54. Physical Model of Drive System

properly controlled. Figure 54 shows a representation of the system under consideration.

The system is richly dynamic and provides a challenging control situation. During winding, accurate knowledge of the angular position of the product spool is critical. Also of importance is the relative motion between the main and ride shafts. If the shafts turn too much out of phase, the fiber can fall off the guide wheel of the supply unit riding on the ride shaft. While important, the positioning of the product spool is relatively simple to control and not entirely critical during normal winding. The more interesting problem is controlling the relative position between the two shafts. The goal here is to develop a mathematical model of the system and design a control algorithm that will minimize error motions between the two shafts. Responses will be simulated with the aid of a computer and then compared against actual machine responses.

4.1 Drive System Physical Model

The motor, shafts, belts, coupling, and bearings are indicated in the drawing of the system above (Figure 54). In order to move to a mathematical model of the drive system, each of the components must be reduced to their mathematical equivalents. The belts and coupling can be represented as flexible elements, or springs, that obey Hooke's law. As a torque is applied to the flexible element, that element twists linearly according to the equation¹²:

$$\tau_s = k\theta \quad (3)$$

where τ_s is the applied torque, θ is the rotational deflection of the element, and K is the torsional stiffness of the element. Next, damping is present in several places in the system. There is intrinsic damping in the motor and gearhead combination. Each of the bearings provide significant damping and there is also a good deal of loss as the belts rub against the pulleys. To simplify the analysis, Coulomb damping will be assumed. The damping force for each element will be considered proportional to the velocity of that element by a factor, b :

$$\tau_b = b \frac{d\theta}{dt} \quad (4)$$

where τ_b is the resulting damping torque, b is the damping constant, and $d/dt(\theta)$ is the angular velocity of the element.³ The shafts can be modeled as an inertia and represented by the symbol J . Finally, the motor will be represented by an applied torque in the system, τ , and a separate damping and inertia.

With all of the mathematical tools in place, the above physical model of

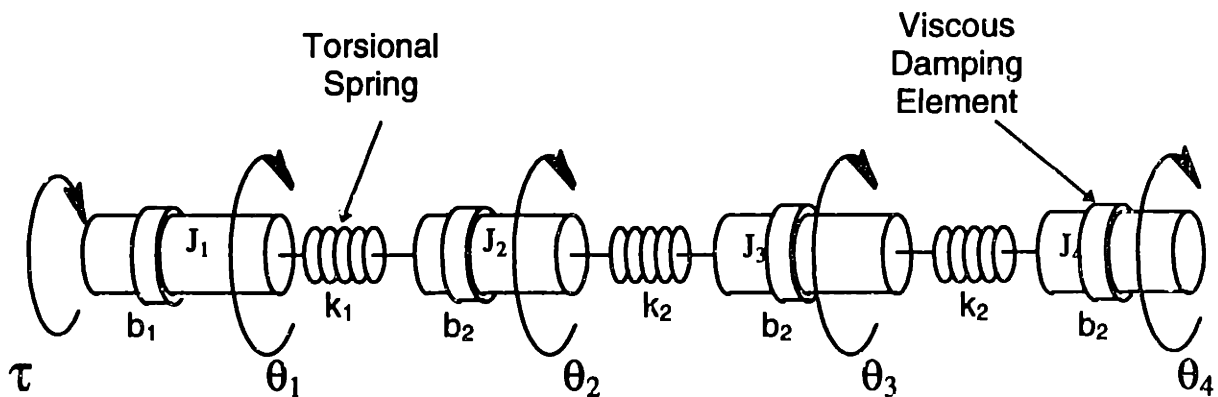


Figure 55. Mathematical Representation of System

the system can be reduced to a mathematical one, shown below (Figure 55).

In the model, J_1 , θ_1 , and b_1 correspond to the inertia, angular displacement, and damping of the motor in the drive system. K_1 is the torsional stiffness of the flexible coupling. θ_2 , b_2 , and J_2 represent the main shaft; θ_3 , b_2 , and J_3 correspond to the transfer shaft; and, θ_4 , b_2 , and J_4 represent the ride shaft and SU in ride position.

To simplify the analysis, several assumptions have been made regarding the sources of damping and the stiffness elements in the system. The different sources of damping on each of the shafts, meaning the bearings and the belt friction, have been lumped into one equivalent damping element. This damping is assumed to be approximately the same for each shaft and is termed b_2 . Also, Each of the two belts have been converted to equivalent torsional stiffnesses and are assumed to be of approximately the same magnitude, k_2 .

Having reduced the physical model to its mathematical equivalent, equations of motion can be developed for the system. Energy methods were applied to generate these equations. The first step was to determine the Lagrangian of the system:

$$L = T' - V \quad (5)$$

where L is the symbol for the Lagrangian, T' is the kinetic co-energy, and V is the potential energy in the system.³ To obtain the equation of motion, the variational indicator is used to derive the Lagrange equations (See Crandall):

$$\frac{d}{dt} \left(\frac{\partial L}{\partial \dot{\xi}_i} \right) - \frac{\partial L}{\partial \xi_i} = \Xi_i \quad (6)$$

Again, L stands for the Lagrangian. ξ represents the generalized coordinates, in this case, θ . And, Ξ represents both the non-conservative and the input forces in the system, in this case, the input torque, τ , and the damping, b . The lagrange equation actually is a shorthand notation for a set of i equations, one for each generalized equation in the system, θ_1 through θ_4 . When the lagrange equation is applied to the main drive system, the following equations of motion are obtained:

$$\begin{aligned}
 J_1\ddot{\theta}_1 + b_1\dot{\theta}_1 + k_1\theta_1 - k_1\theta_2 &= \tau \\
 J_2\ddot{\theta}_2 + b_2\dot{\theta}_2 + (k_2 + k_1)\theta_2 - k_1\theta_1 - k_2\theta_3 &= 0 \\
 J_3\ddot{\theta}_3 + b_2\dot{\theta}_3 + 2k_2\theta_3 - k_2\theta_2 - k_2\theta_4 &= 0 \\
 J_4\ddot{\theta}_4 + b_2\dot{\theta}_4 + k_2\theta_4 - k_2\theta_3 &= 0
 \end{aligned} \tag{7}$$

Using these equations, it would be possible to develop a controller for the system. However, expressing the equations in state-space format will greatly simplify the development of the controller. State-space equations take the general form:

$$\frac{d}{dt}(\vec{x}) = A\vec{x} + Bu \tag{8}$$

$$y = C\vec{x} + Du \tag{9}$$

where y is the output, u is the input, and x is the vector of system coordinates. A , B , C , and D are matrices that describe the state of the system.¹² When put into state-space form, the main drive system will be of eighth order. The A , B , C ,

and D matrices can be derived from the equations of motion and are shown below.

$$C = [0 \ 0 \ 0 \ 0 \ 0 \ 0 \ 1 \ 0] \quad B = [0 \ 1/J_1 \ 0 \ 0 \ 0 \ 0 \ 0 \ 0]^T$$

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ -\frac{k_1}{J_1} & -\frac{b_1}{J_1} & \frac{k_1}{J_1} & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ \frac{k_1}{J_2} & 0 & -\frac{(k_1+k_2)}{J_2} & -\frac{b_2}{J_2} & \frac{k_2}{J_2} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & \frac{k_2}{J_3} & 0 & -2\frac{k_2}{J_3} & -\frac{b_2}{J_3} & \frac{k_2}{J_3} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & 0 & \frac{k_2}{J_4} & 0 & -\frac{k_2}{J_4} & -\frac{b_2}{J_4} \end{bmatrix}$$

$$D = [0] \tag{10}$$

To aid in the development of the controller, MATLAB will be used to help simplify the process. A unique feature of MATLAB is that it can accept both state-space and conventional system representations. With this in mind, the system, referred to from this point forward as the plant, can remain in this state-space form as the controller is developed. However, the different aspects of the controller will be developed using the conventional, frequency domain representations. Frequency domain analysis greatly simplifies controller design.

The last step in the development of the plant model is the estimation of system parameters: $J_1 = 4 \times 10^{-4} \text{ kg}\cdot\text{m}^2$, $J_2 = .0025 \text{ kg}\cdot\text{m}^2$, $J_3 = .0014 \text{ kg}\cdot\text{m}^2$, $J_4 = .022 \text{ kg}\cdot\text{m}^2$, $k_1 = 2 \text{ N}\cdot\text{m}/\text{rad}$, $k_2 = 500 \text{ N}\cdot\text{m}/\text{rad}$, $b_1 = 250 \mu\text{N}\cdot\text{m}\cdot\text{sec}/\text{rad}$, and $b_2 = 500 \mu\text{N}\cdot\text{m}\cdot\text{sec}/\text{rad}$. The inertia values were obtained using basic physical equations for rotational inertia and knowledge of the shapes and masses of the

shafts. K_1 was obtained from the product information on the coupling. The value for the stiffness of the belts was determined by examining their construction. Timing belts of the type used in this design are constructed by encasing two steel cables in urethane. Using this construction allows for the creation of compliant teeth while maintaining a high belt stiffness. It was assumed, therefore, that the majority of the stiffness would be provided by the two steel cables in the belts. Knowing the cross-sectional area and Modulus of Elasticity for the cables, the following equation was used to determine the approximate belt stiffness value:¹²

$$K = \frac{EA}{L} \quad (11)$$

where K is the stiffness, A is the cross-sectional area of the cables, and L is the belt length between pulleys.

For the damping value for the motor and gearhead, b_1 , the product literature was consulted. A quick test was required to estimate b_2 as this is a lumped parameter including the dampings of several bearings and a belt over a timing pulley. By turning one of the shafts at constant velocity and measuring the required input torque, the damping constant could be estimated from the following equation:¹²

$$b = \frac{\tau}{\dot{\theta}} \quad (12)$$

where b is the damping constant, $\dot{\theta}$ is the rate of change of shaft angle, and τ is the applied torque to the shaft. With these parameter values for inertia

stiffness, and damping, the state equations and plant transfer function can be specifically defined.

For this particular system, the uncontrolled plant is unstable with respect to position for a given input torque from the motor. This is expected as an applied torque will cause the shafts to spin and, the angular position of the shafts will grow without bound from their starting positions. Such a system can be made stable by using the proper control algorithms with feedback on the position of the shafts. The development of the controller will be presented in the next section.

4.2 The Controller Structure

There are several different approaches to controlling the type of system under consideration. Lead/Lag, PID Pole Placement, and PID Optimization techniques are all possibilities.¹² The high complexity of the plant played a large role in determining which controller would be used. Pole placement becomes increasingly difficult with increasing system order and is quite difficult with the eighth order system discussed here. Of greater importance in choosing the controller was the presence of a restraint on which type of controller could actually be implemented in the system. With the electronics available in this machine, only a PID controller could be readily implemented. These electronics only allow for the entry of the three PID gains in the closed loop control. While it

would be possible to use a Lead/Lag controller with these electronics, it would be difficult to actually implement.

The most promising method would be to use PID Optimization techniques as such techniques would be simple to implement on the actual system (This concept will be described in detail in the following section). Moreover, the optimization could be simulated using SIMULINK in the MATLAB environment. By simulating the controller before implementing it on the machine, several benefits are apparent. First, an insight into the actual behavior of the system is gained. Before actually attempting to spin the shafts, one would have an clear idea on how they might behave. This would help to prevent accidents that could damage the machine or the operator. Second, there is a much greater freedom to optimize the controller using SIMULINK, because none of the physical limitations of using the machine would be present. Finally, the controller design process would proceed much faster than on the machine as parameters can be quickly varied in simulation. Varying those same parameters takes much longer on the actual machine. Once the optimization is complete, the PID would be implemented on the actual system. At that point, the PID gains could be further optimized to increase performance.

4.3 PID Optimization

The concept of PID Optimization is relatively simple, but highly effective. The technique involves varying the PID gains in a systematic way to achieve optimum system performance.¹² This technique can be implemented using

simulation software or using the actual system to be controlled. The later option is frequently used to develop controllers on many simple systems. As described above, simulation has many benefits and was used in the development of the controller for the winder.

Before discussing the optimization technique, it will be useful to review the concept of PID control. Consider the block diagram of the system shown below (Figure 56). Such a system consists of a plant, which is the system to be controlled, a sensor to detect the output of the system, and a controller that determines the input to the system based on the input and the feedback from the sensor. Such a system is said to be under “closed loop” control and uses the current output of the system to help define what the next input to the system should be. An “open loop” system takes inputs without reference to the current output.

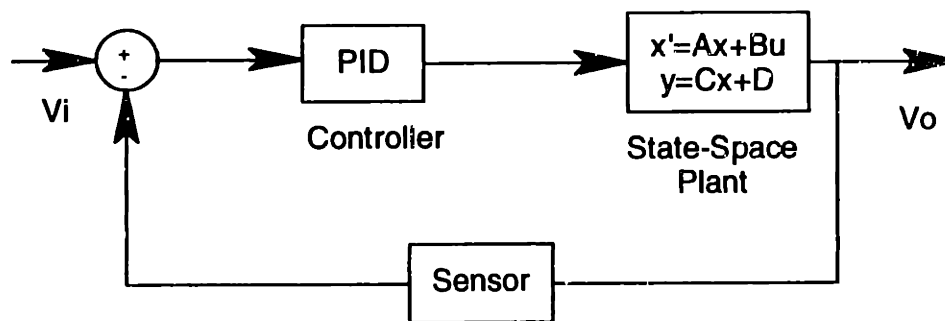


Figure 56. Closed Loop System Block Diagram

Suppose the system in the block diagram were a positional system, such as the one in this thesis, and the system was commanded to move to a certain position. An open loop system would take only a command input that would instruct the system where to move. Therefore, the system would have no

assurance as to if the desired position was reached, nor would there be any control over how the position was reached. Factors such as friction and backlash can cause the final output of the system to stop or exceed the final desired value. There would be no assurance that the system had reached its final goal or that the system hadn't gone out of control.

For the drive system on the winder, it has already been mentioned that it is unstable with open loop control; a desired angle would be the input to the system, but the winder would turn out of control. Using a closed loop, a great deal of control can be gained over the behavior of the system. By comparing the current output to the desired output, corrective action can be taken to ensure that the ideal output is achieved. Closed loop control allows the winder drive system to move to a commanded position, rather than go unstable, for a given input.

Comparison between the actual and desired output is made by taking the difference between the two. A sensor is commonly used to determine the actual output from the system. The current input to the system corresponds to the desired system output. By subtracting the actual from the desired output, an error signal is obtained. When the error signal goes to zero, the system has reached the desired output. Mathematical relations can be developed that use this error signal to determine what input should be given to the plant to get to the desired output. These mathematical relations are referred to as the "controller" which is one of the blocks in the block diagram above. By manipulating the form of the controller, different response profiles can be generated.

Another concern is any error that remains in the output regardless of the control action of a specific controller. Such error is commonly referred to a steady state (ss) error.¹² Various types of controllers allow for different amounts of ss error. In many cases, a faster system response time to an input can be achieved if a little ss error is allowed. For the winder drive system response speed is not as important as having zero steady state error..

The goal of designing the controller for the winder drive system is to obtain the optimum response profile while maintaining zero ss error. As mentioned earlier, the interesting aspect of the design is to allow for minimum lag between the ride and main shaft positions. A basic assumption was made to simplify the design. Rather than considering behavior of both shafts, the design was directed at optimizing the response of the ride shaft. It was assumed that because of the high damping in the system and the high belt stiffness, the relative motion between shafts would be small if the shafts were well behaved under control. In addition, it was assumed for the same reasons that if the ride shaft were well controlled, the main shaft would most likely be well controlled. In the end, this assumption proved to be true as seen by the response of the winder drive system.

The PID type of controller can be readily designed to meet the winder controller design goal. PIDs can be tuned to give excellent system response profiling as well as zero ss error. The mathematical form of the controller is as follows:¹²

$$C(s) = K_p + K_i \frac{1}{s} + K_d s \quad (13)$$

where $C(s)$ is the control transfer function. It is a ratio between the output signal to the plant and the error signal. S is the frequency domain equivalent of time in the time domain. Frequency domain representation simplifies the controller analysis. When the error signal, in frequency domain form, is multiplied by $C(s)$, a specialized output signal to the plant results. $C(s)$ is designed such that this output signal will optimize the response of the plant. The other terms in this equation will be described below.

PID is actually an acronym for Proportional Integral Derivative. The proportional term is the first term in the above equation. This term acts to amplify the error signal and is the mathematical analog of a spring. The proportional term in the controller results in a spring-like effect in the system output. The second term is the integral term which acts to integrate the error signal over time. Such a term creates a greater signal over time for a given error. Having the integral term ensures that there will be little or no ss error in the output. Finally, the last term is the derivative term. It amplifies the rate of change of the error signal and is the mathematical analog of the damping constant. This has the effect of providing damping in the output response. K_p , k_i , and k_d are the proportional, integral, and derivative gains, respectively, and can be varied to change the degree of action of each term in the PID controller. The challenge lies in choosing the gains to give the desired response for a given system.

Before continuing with this discussion, it is important to create an accurate block diagram of the entire winder drive system. On the machine, the inputs to the system will be in the form of a voltage corresponding to a desired output angle and the output will be in the form of a voltage corresponding to the angle of the ride shaft. A potentiometer sensor was attached to the ride shaft to provide the measure of output shaft position. The block diagram for the winder drive system then takes on the form shown in Figure 57. Note that this block diagram is similar to that presented earlier. The plant and controller blocks are present as before. The controller is the PID designed below and the plant is the mathematical model of the drive system described above. The input voltage, v_i , and the output voltage, v_o , can be seen in the diagram. Of particular interest is the three new elements. Two of these are gains that are inherently present in the system. One due to the gain in the motor amplifier and the second due to the gain in the sensor. The other block is a representation of the backlash in the gearhead on the motor. Mathematically, backlash can be represented as a pure delay in the system which is how it is represented in the block diagram. From actual use, the delay upon switching directions of turning the motor was estimated to be .1 seconds. With this block diagram, the design of the controller

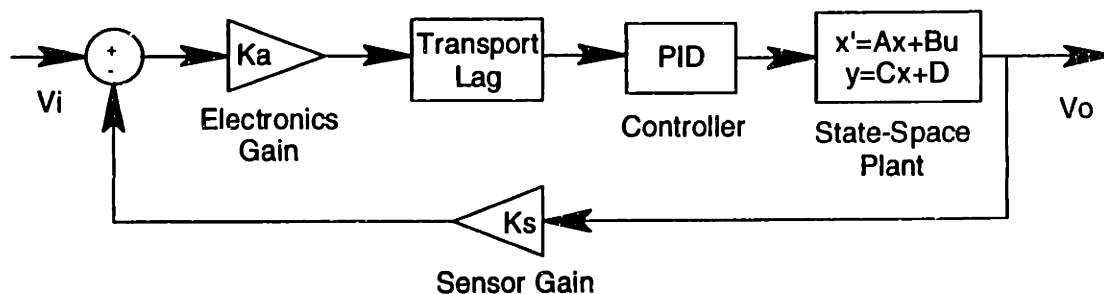


Figure 57. Drive System Block Diagram.

can proceed.

For the complex winder drive system, the technique of PID Optimization will be used to choose the gains for the PID controller. To begin with, some systems can generally operate with only proportional gain. While this usually results in poor performance with significant ss error, it is the simplest type of controller. For these reasons, the optimization technique begins by setting a PID controller such that $k_p = 1$, $k_d = 0$, and $k_i = 0$. This PID is inserted into the above block diagram in the “controller” block. The block diagram representing the closed loop system with the PID controller can be inputted into a software package such as SIMULINK. The program can then simulate the actual response of the system to various inputs. For this optimization technique, step inputs will be used as they generate a simple, yet revealing response from the system that can be quickly analyzed. The response to a step is termed the “step response.”

Having set up the block diagram with the simple PID and entered it into SIMULINK, the program is used to generate a step response. The next action in the optimization technique depends on the shape of the response. For this step of the process, the goal is to determine at what k_p does the system become unstable. An unstable system is one in which the step response grows without bound. If the system is unstable with $k_p = 1$, the gain is lowered until the system becomes stable. If the system is well behaved, the gain is increased until the system goes unstable. The gain at instability is noted. K_p is then set to half this value to be used in the next step.

Next, the derivative gain is manipulated in an attempt to speed up the response. This step involves a bit of finesse as increasing the derivative gain often leads to more overshoot in the response. Overshoot is defined as how far past the desired output the actual response first goes before tending back towards the desired output. Overshoot also tends to lengthen the amount of time it takes for the response to reach the desired output. This time is termed the settling time. Short settling times are generally the goal in positioning systems. On the other hand, increasing the derivative gain speeds up the initial reaction time of the system to the input. It is desirable to have fast reaction times, but also fast settling times. This is where the finesse comes in. By varying k_p and k_d , the overshoot and settling time can both be varied. The technique is to increase k_p and k_d until the system is as fast as possible, with as little overshoot as possible, before becoming unstable. When instability is reached, the gains are again noted. k_p and k_d are lowered until the system is well behaved and definitely stable.

The final step is to incorporate the integral gain into the controller. As mentioned above, this gain will have the effect of eliminating or reducing the ss error in the response. The problem with the integral term is that it tends to push the system towards instability as the integral gain is increased. Put simply, the reason for this is that the integrator in the term adds to the order of the system. The higher order system tends to be less stable, particularly when the magnitude of the integrator is increased. Again, there is a basic trade-off. The integral gain must be high enough to eliminate the ss error, but low enough as

not to cause instability. With regard to the optimization, an integral gain value of one tenth the proportional value is added into the PID controller. A step response is generated and the response evaluated. If it is unstable, the integral term is reduced until stability is reached. If there is no ss error, the system is well tuned. Otherwise, k_p and k_d must be adjusted to bring the system into stability, while maintaining the best performance as possible. If the system was stable to the initial integral gain, the gain can be increased if there is still steady state error.

At this point in the technique, the optimization can become somewhat of a game if one gets carried away. K_p , k_i , and k_d are each adjusted as necessary to improve the response. How long the game is played is up to the designer. However, the goal was to get the response good enough for the actual machine, but not necessarily perfect. As there may be inaccuracies in the system model and other sources of variation, it would be a waste of time to fully optimize the simulation. For the winder drive system, the optimization technique was followed until the response was well behaved to the point where it was stable, fast enough, and had virtually no ss error.

4.3 System Responses and Results

After performing the simulated optimization of the PID controller on the drive system plant, an acceptable step response to a 90 degree step was created for the system. This response is shown in Figure 58. Note that the

system exhibits a very well behaved response. The system settles to essentially 90 degrees in approximately 500 milliseconds. Also, there is no overshoot and virtually no ss error. The gains of the PID were modified within SIMULINK using the optimization techniques until the response behaved as shown. It was decided that this response would be adequate for use on the actual drive system. The gains determined for the simulated system were input into the control program for the actual main drive system.

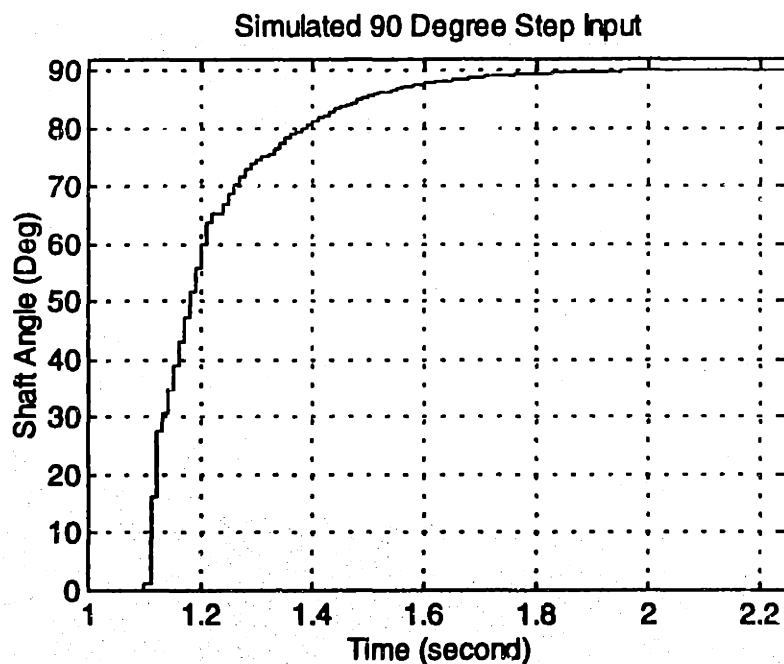


Figure 58. Step Response of the Simulated System

With a controller in place, a 90 degree step input was run on the system. As expected, the system did not behave exactly as the simulation predicted, although the response was reasonably close. In addition, the response was safe and surprisingly well behaved. Variations were attributed to differences in system parameters, such as stiffness and damping, as well as to the non-linearities in the system. The two greatest sources of non-linearity are the

backlash in the motor gearhead and the friction throughout the system. Both of these effects were approximated with the simulation, but neither could be exactly duplicated in the simulation.

Having observed the step response, the final step in the optimization could be performed. More specifically, the PID gains would be adjusted to obtain the best possible response with PID control. The same methods as were used in the simulated optimization were again applied. Only slight modification of the gains were actually required before an acceptable step response was generated. Figure 59 shows a plot of the step response. While similar in many ways, the effects of differences between the simulated and the actual plant are clearly visible. This response has virtually no overshoot and little steady state error, which are perhaps the most desirable aspects of the drive system

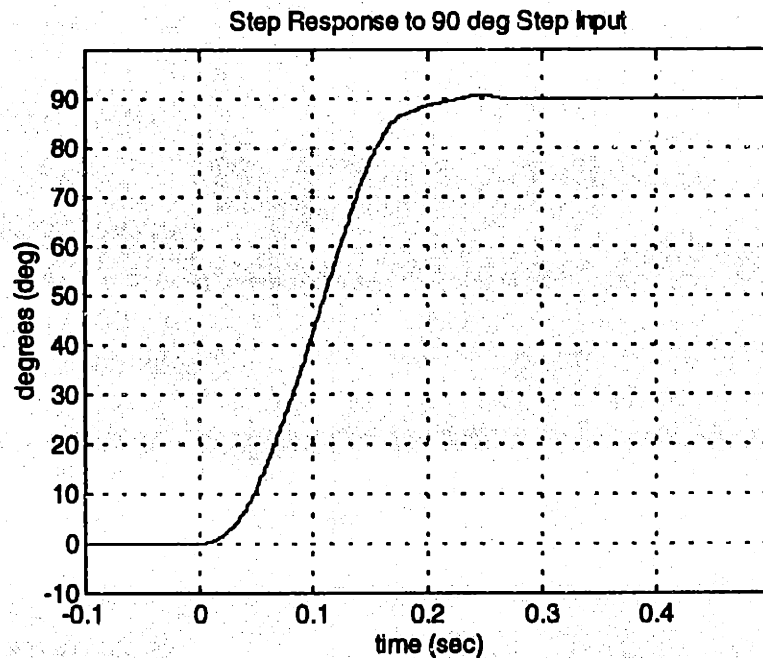


Figure 59. Measured Step Response of Drive System

behavior. Note however, that this response is much faster than the simulated one, achieving 90 degrees in about 300 milliseconds. By following through on the optimization of the actual plant, this faster response was achieved. The last significant difference between the response is the shape of the response curves. Notice how the actual response is less curved at first than simulated. Shape of the response is related to the order of the system and the exact parameters for stiffness, inertia, and damping. Given that the simulated response was only an approximation of these factors, the shape differences were to be expected. In actuality, this difference in response profile had no detrimental effects on the drive system and was more than acceptable for the drive system response.

The PID optimization technique proved to be an excellent design tool in creating the control system for the main drive system. It led to a deep understanding of the various factors affecting the behavior of the system. More importantly, the simulation greatly facilitated the selection of the PID gains that would lead to a desired response. This had the effect of ensuring a safe set of control parameters for the initial run of the controller on the actual drive system. Safety of the machine and, of course, the designers must always be first and foremost. In addition, the simulation resulted in a very rapid final gain selection on the actual machine. The design process is quite time consuming if the optimization must be run entirely on the actual machine to be controlled.

In the end, the drive system exhibited a response profile that was very acceptable and the design and implementation process went very smoothly. For systems similar to the main drive system on the winder, the PID optimization

technique is an excellent design tool. However, more complex or critical systems would be much better suited to more rigorous design techniques as they can lead to superior system control.

Chapter 5 - Special Issues

During the course of the project, it was necessary to define and characterize many aspects of winding fiber. A clear understanding of the behavior of fiber during winding was necessary before an automated machine could be built. Many of the issues surrounding orthocyclic winding were poorly understood and were typically avoided by other coil winding efforts. It was felt that these issues had to be well defined to wind a high performance coil. Several of these issues will be discussed here, including base layer winding, jog zone size, and fiber spacing.

5.1 The Orthocyclic Winding Process

The nature of the Orthocyclic wind was discussed earlier in the introduction. To review, the orthocyclic technique involves winding fiber in concentric loops around the spool and jogging the fiber from one loop to another in a specified zone. The zone in which the fiber crosses loops is commonly termed the jog zone and occupies a certain number of degrees of the product spool circumference. The alternative method involves winding the fiber in a continuous spiral across the width of the spool, termed a thread wind. For reasons discussed earlier, such a technique is not acceptable.

By forcing the fiber to jog in the well defined area of the jog zone, rather than anywhere around the spool as with the thread wind, certain problems arise. Fiber has a high stiffness and is relaxed in an unbent state. When the fiber is

made to jog over a fiber diameter from one loop to another in the orthocyclic wind, the fiber is being forced to take

on an unnatural state. It must bend

within the space of the jog zone

enough to move the fiber position over

by a diameter. Figure 60 illustrates

this concept. The shape described by

the fiber in the jog zone is actually much like that of a tangent curve.

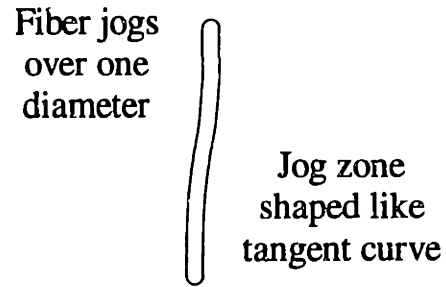


Figure 60. Shape of Jog Zone

While the shape of jogs fits well in the geometry of the coil, the physics involved cause some problems. Bending the fiber into the shape of the zone is

similar to deflecting a spring. If

fiber is bent by hand and then let

go, it will spring back to its

natural, straight position. The

stiffness of the fiber creates a

restoring force when the fiber is

subject to deflection from straight.

While the deflection of the fiber

and the accompanying force is

small, it is enough to create

difficulties in the winding of a coil. Consider a typical winding situation where

fibers are being wound onto a coil. Figure 61 shows the jog zone with several

fibers that have already been wound, each laid next to one another. The next

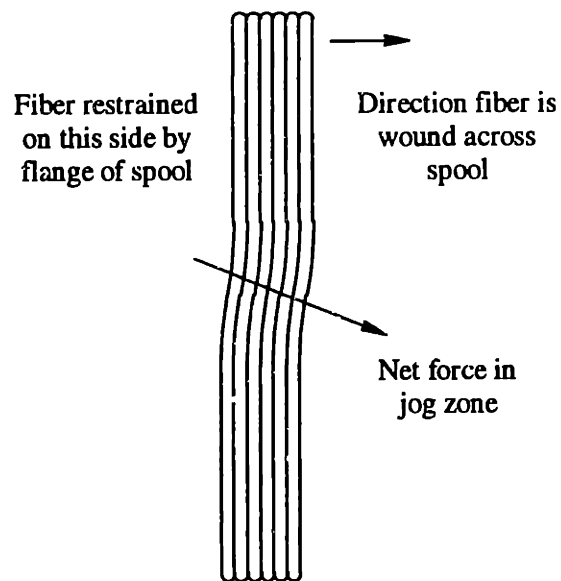


Figure 61. Jog Zone Forces

fiber will be wound to the right of those already laid down. Given that each fiber loop is inclined to restore itself to a straight orientation, there will be a resultant force throughout the jog zone which is indicated in the figure. This force will tend to cause the fibers to spread out in an attempt to alleviate the stored energy in the zone. Also, the zone will tend to grow in angle around the circumference of the coil so that the fibers are not bent in such a tight zone. Winding experiments involving the jog zone and the forces involved will be discussed in the following sections.

5.2 Base Layer Winding

The first layer of fiber wound onto the product spool is called the base layer. The quality of the wind of the base layer subsequently affects the quality of each additional layer wound.

Any errors or imperfections in this first layer can cause errors throughout the rest of the coil. For every layer beyond the base layer, fibers are wound into the grooves formed by the fibers on the underlying layers (Figure 62).

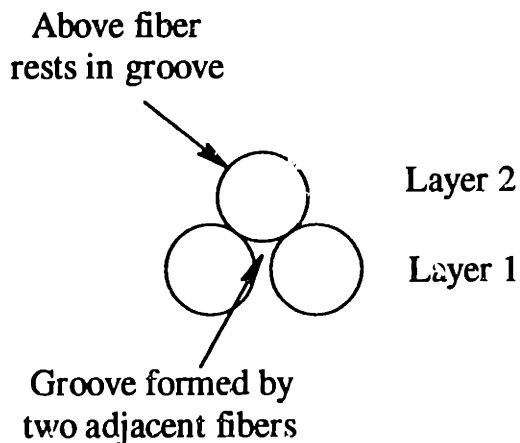


Figure 62. Fiber Wound into Groove

While these grooves help in guiding the fiber, any imperfections in the grooves can lead to errors in the current layer. Such imperfections could be caused by

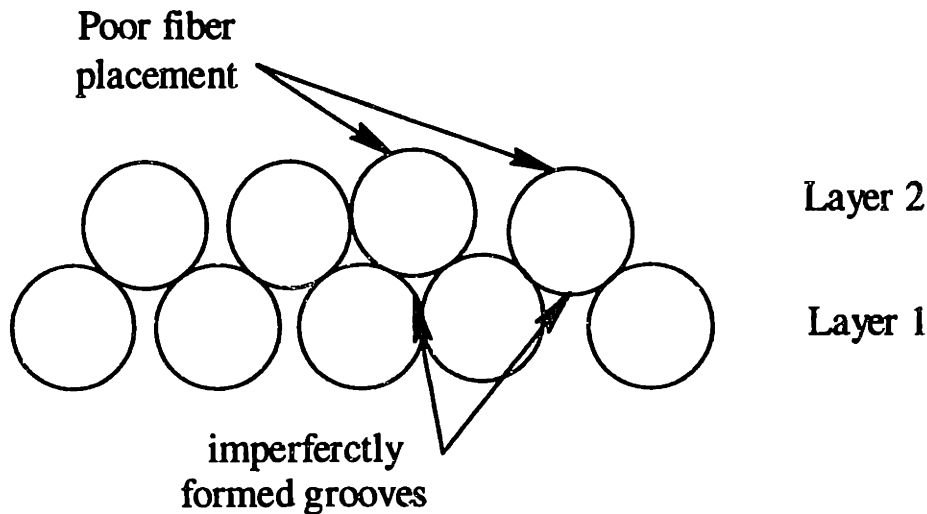


Figure 63. Errors in Fiber Spacing Due to Poorly Formed Grooves

gaps or overlapping of fibers on the underlying layer. Figure 63 shows an example of how fiber placement can be compromised by imperfect grooves.

The base layer plays a very important role in defining the quality of the grooves formed on all subsequent layers. Fiber turns on this layer must be well spaced with no gaps or squeezed areas. Also, the jog zone on this layer must be as perfect as possible for similar reasons. How to wind this layer was unclear at the onset of the project. To determine the best method, a series of experiments were performed. There were many factors that could be varied in winding the base layer. With the available facilities, it was only possible to vary tension profiling, fiber guiding, and mandrel, or base of the product spool, configuration. Tension could not be varied as one of the project requirements dictated that tension must be tightly controlled and must be able to varied to any level in a specified range. Also, the guiding technique was not really open to variation. From early in the design, the guide wheel was seen as the best option for guiding and so was chosen as the accepted guiding method. It was of

course possible to consider additional guiding methods, but this was to be avoided to keep the design as simple as possible. There were several choices on how to implement the guide wheel, but these could only offer limited improvements. The only real factor that could be significantly changed was the configuration of the product spool. The winding experiments therefore involved winding onto different types of product spools using the guide wheel to guide the fiber.

The differences in the spools was in the mandrel portion of the spool. The product spool consists of the flanges, or sides, and the mandrel, or the base of spool. Fiber is wound onto the mandrel and retained on the spool between the flanges. Three different mandrel configurations were considered as possibilities for winding fiber. For each configuration, 10 turns of fiber were wound onto the spool and the resultant wind quality examined. The wind was checked for fiber spacing, jog zone profile, and any major errors, such as a crossover where the fiber had wound back onto itself. It wasn't feasible to make specific measurements. Instead, the general quality of the wind was observed and rated on a relative scale against the results of winding on the other mandrels.

The first mandrel type considered was simply a smooth surfaced cylinder. Fiber would be guided onto the mandrel and would rely on friction with the surface to remain in place. Each subsequent fiber would be wound just touching the previous turn. Winding revealed that such a configuration would not promote a good base layer wind. First, gap and crossover type errors happened quite

frequently. Second, the jog zone tended to spread out. The zone is typically defined to occur within a specified number of degrees of the circumference of the mandrel. On the smooth mandrel, different zone sizes were attempted, but all tended to spread to much greater sizes.

Both of the observed deficiencies can be attributed to relaxation of the fiber. Friction with the surface did not seem to hold the fiber in place after it was guided onto the mandrel. Given the reaction forces in the jog zone, the fibers tended to spread out away from each other and the zone size tended to widen in angle. While the growth of the jog zone was undesirable, the spreading of the fibers was catastrophic. As the guide wheel progressed across the width of the mandrel, it would jog one fiber diameter each time the jog zone passed. After 10 turns, the guide wheel would be 10 fiber diameters from where it had started. On the other hand, the fiber had spread out and could be as much as 12 to 15

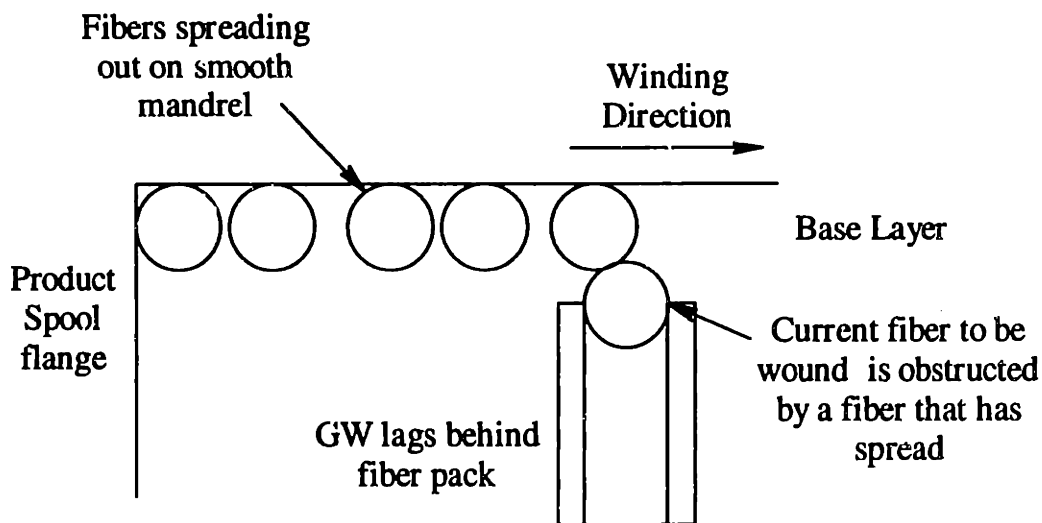


Figure 64. Guide Wheel Lagging Behind Fiber Pack as Fibers Spread

diameter from the starting position. Thus, the fiber would actually end up ahead of where the guide wheel was attempting to guide the fiber. This lead to

crossover errors which were not acceptable on the base layer. Figure 64 shows this problem.

Various techniques were attempted to achieve a good base layer wind on the smooth mandrel, but the basic problem of fiber spreading couldn't be solved. Another major problem that arose was that inconsistencies in the mandrel surface and dust particles could easily interfere with fiber placement on the smooth mandrel. Such inconsistencies often lead to errors. From the results of the winding with this configuration, it was decided that it would not be advisable to wind onto a smooth mandrel.

It seemed that the heart of the problem with smooth mandrel winding was a lack of restraint on the already wound fibers. Spreading out of the fibers severely impaired the machine's ability to guide the fiber. Also, the experience gained in the first experiment revealed another problem with the base layer. Variations in the fiber diameter and in the mandrel surface could lead to guiding errors. For example, if all the fibers are slightly oversized, the guide wheel could begin to lag behind the fiber pack during winding. To better understand this phenomenon, consider the following. The position of the guide wheel is based on calculations that use the nominal fiber diameter. However, if all the fibers are significantly oversized, the actual fibers will be spaced according to this oversized fiber width, but the guide wheel will be located according to the theoretical fiber location. With the guide wheel lagging the actual fiber pack in this manner, crossover errors are bound to occur.

Fiber variations could also lead to another much more basic problem. Consider Figure 65. Fibers could be as much as 3% over or under the specified diameter, and the width of the mandrel is generally sized to meet the nominal diameter, and the width of the mandrel is generally sized to meet the nominal

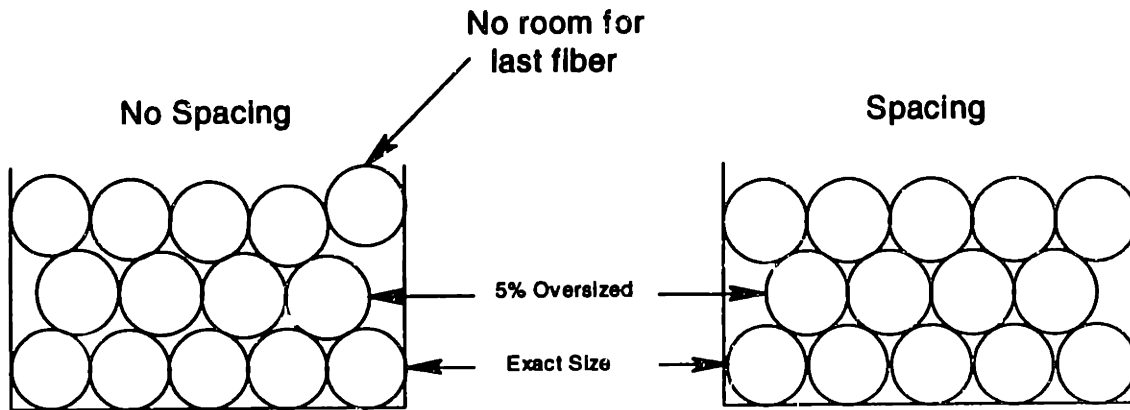


Figure 65. Problem with No Fiber Spacing

diameter. The figure shows the exaggerated case where all of the fibers are 3% oversized. The last fiber doesn't have room to fit between the flanges and is forced to the next layer. This would be considered a serious error in the wind. The solution is to allow extra space between the fibers to allow for these variations and to ensure that there was enough room for all of the fibers on a given layer. A similar problem would arise if the fibers were too small. Rather than a bulge, however, a sag would be seen in the wind where the small fiber could sink into the last gap against the flange. This phenomenon also severely degrades coil quality.

To address these concerns a smooth mandrel with a tacky surface was constructed. The tacky surface would hold the fiber in place, restraining the fiber from spreading and allowing for spacing between turns of fiber. Again 10 turns were wound, and the resultant wind quality inspected. It was found that the

tacky surface met requirements with limited success. Fibers were held in place as expected but the spacing between the fibers varied greatly. As discussed above, the spacing needed to be consistent so that above layers would see a good winding surface. Different levels of tackiness and fiber guiding profiles were used in attempt to even out the spacing, but the problem could not readily be solved. From observation of the fiber during winding, it appeared that the fiber would roll over slightly, possibly due to fiber or surface variations. This rolling seemed random in nature and lead to random spacing variations. Some fibers virtually touched and others were nearly two fibers apart. A simple solution to this problem did not present itself and it was concluded that, while it had potential, a tacky surfaced mandrel was not the way to go.

The last alternative was essentially the next logical step from the tacky mandrel. A mandrel was created that actually had grooves machined into the surface. The grooves would support the lower portion of the fibers and would serve to retain the fiber within the bounds of the groove. Such a configuration would eliminate spreading, allow for spacing, and allow for fiber diameter variations. The same set of experiments were run with the grooved mandrel and were deemed a total success. Guiding with the guide wheel along the width of the mandrel, every fiber wound perfectly



Figure 66. Fibers on Grooved Mandrel

into place and stayed within its designated groove. Figure 66 is a profile view of fiber being wound onto the base layer. The fibers are the dark circles and the grooves in the mandrel are the dark troughs.

From the results of the base layer winding experiments, it was clear that the grooved mandrel was the configuration of choice. While this method worked perfectly during winding, it did raise some questions concerning cost. The customer voiced concerns over the potential cost of machining the precisely cut grooves into the mandrel. This additional cost, however, would facilitate a much higher quality coil. With this in mind, the grooved mandrel was chosen as the base line configuration for the automated winder as it exhibited superior base layer winding capabilities.

5.3 Optimal Groove Spacing

Having decided on grooved mandrels, the optimum spacing for the

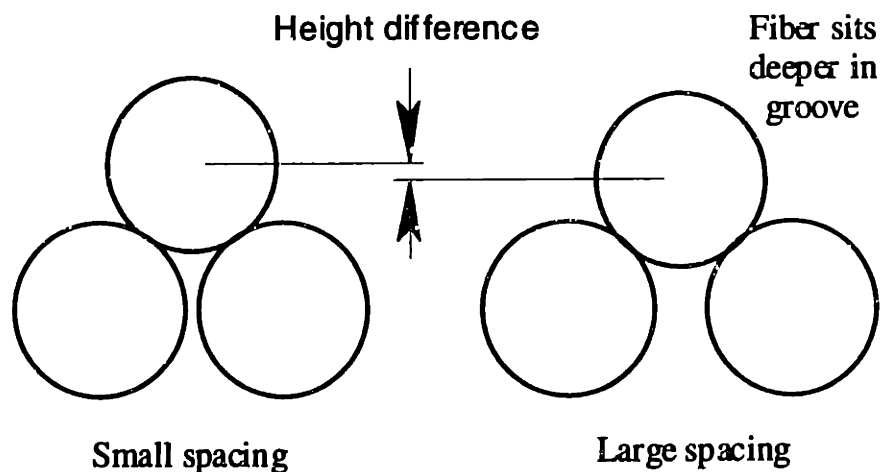


Figure 67. Fiber Groove Depth

grooves was left to be determined. It was a simple matter to choose a spacing that would ensure that the fibers were far enough apart to accommodate fiber variations. Determining when the fibers were too far apart was a more difficult task. It was actually desirable to space the fibers as far apart as possible as it would help in winding the next layer. Figure 67 shows that, to a point, the larger the spacing between adjacent fibers, the deeper the grooves will be.

While it would seem that deeper grooves would lead to higher quality coils, a secondary problem comes into play. Winding a fiber into the groove formed by two fibers in the layer below creates a spreading force on the two fibers. Given that fiber is wound with a certain tension, there will be a corresponding force causing the fibers below to spread apart.

These forces are illustrated in Figure 68. As the fiber spacing is increased

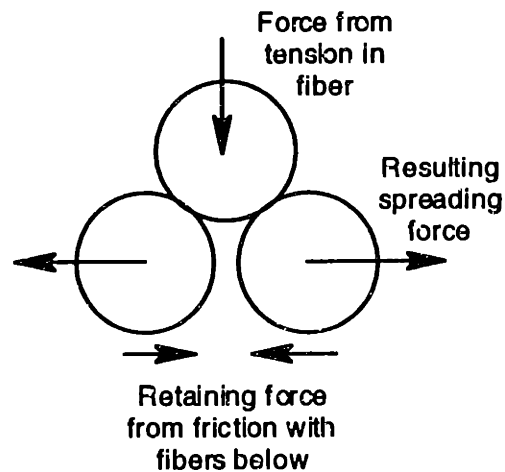


Figure 68. Forces on Fibers in Grooves

and the grooves become larger, the component of the force from fibers above encouraging spreading will also increase.

To a point, friction with the fibers below will keep the fibers from spreading. Once the fibers spread apart, the above fibers fall into the resulting gaps. This event is termed

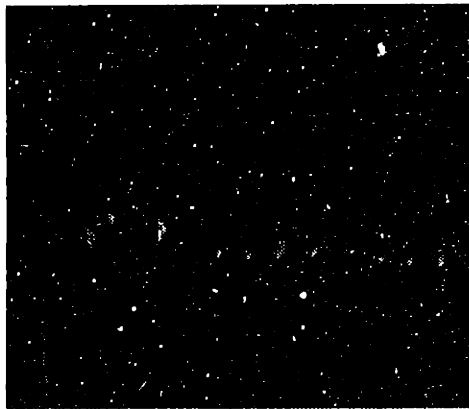


Figure 69. Sagging in Fiber Pack

“sagging” (Figure 69). Sagging effectively destroys the integrity of the current layer, and possibly several layers below, and is difficult to correct. Fiber spacing must then be chosen so that the diameter variations can be accommodated and resultant groove size maximized without encouraging sagging.

A series of experiments were performed to determine what spacing would be acceptable. Three different mandrels were created that had grooves cut at different spacings. As the grooves on the mandrel dictated the fiber position, the spacing of the grooves would therefore create the desired fiber spacing throughout the coil. Spacings of 3%, 5%, and 10% of the fiber diameter were attempted. For each mandrel, a base layer was wound, followed by several more layers. As there was no real quantitative measurement that could be made, the only measure was the general quality of the wind. Consistency of spacing and presence of sagging were used to compare the winds for each of the spacings.

Upon examination of the winds, sagging was found to have occurred on both the 5 and 10% winds. While not perfect, the 3% spacing seemed the best choice. There was still some variation in spacing consistency, but the general quality of the wind seemed acceptable. To use any smaller of a spacing would infringe upon the minimum space needed to avoid the problems associated with diameter variations. The larger grooves resulting from spacing did seem to aid in the placement of the fiber winding into the groove. Again, no specific measurement could be made, but there were much fewer alignment and fiber placement errors than occurred in coils with no spacing.

5.4 Optimal Jog Zone Size

The optimal jog zone size was also unclear at the onset of the project. Jog zone size refers to the number of degrees of the circumference of the mandrel that the zone occupies. As explained above, forcing the fiber to bend in the tight area of the jog zone leads to reaction forces that promote jog zone growth and fiber spreading. The tighter the jog zone, the greater this force. A 10 degree jog zone will have a greater amount of stored energy than a 30 degree zone and will be more likely to spread out. If this spreading were uniform, it would not be such a problem. However, along the width of the mandrel, due to fiber variations and inherent variations in the wind, the zone will not grow uniformly. The zone takes on a jagged profile with fibers starting and ending the jog in different places, as shown in Figure 70.

This jagged behavior in the zone adversely affects the above layers and leads to a poor quality wind. Larger jog zones do not tend to experience as significant of spreading problems and therefore do not exhibit such a degree of jaggedness.

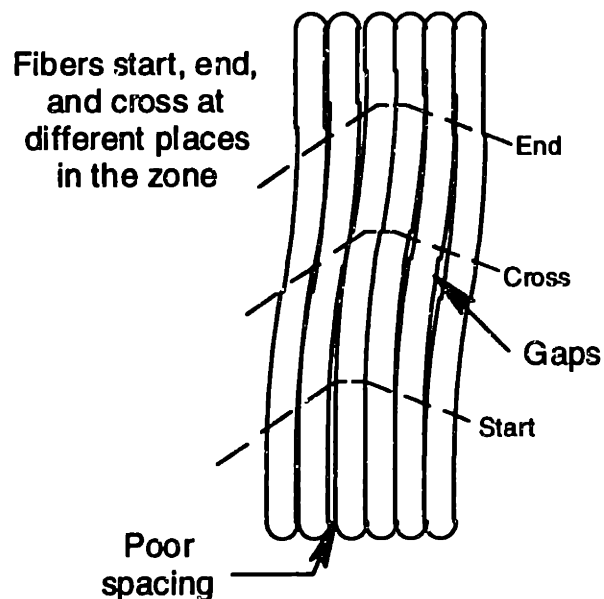


Figure 70. Example of a Poor Jog Zone Profile

Smaller jog zones, on the other hand, have the advantage of winding more consistently. By forcing the fiber to cross from one turn to another in a small space, the exact crossing point can be well controlled. Under ideal

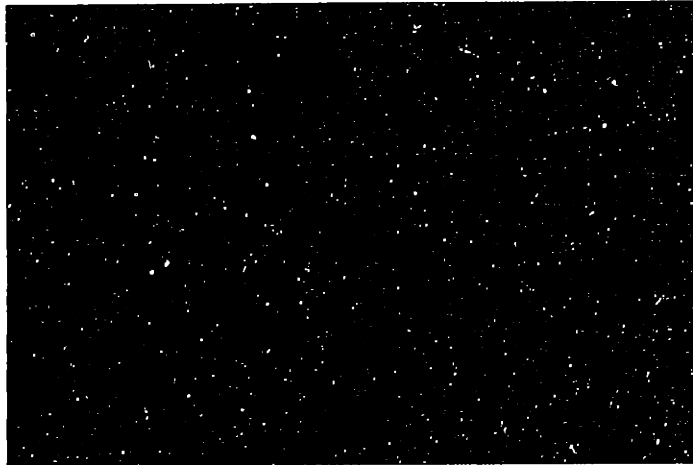


Figure 72. Gaps in Jog Zone

conditions, all the fibers would cross over at exactly the same angular position on the coil, but variations in the winding and in the fibers cause crossings in the zone to be inconsistent. For example, two adjacent fibers can cross over as much as 10 degrees apart. Figure 70 shows that this difference in crossing position can lead to gaps in the zone. Fibers in the above layers could sag into these gaps, leading to serious errors in the wind. Figure 72 is a picture of actual gaps in the jog zone due to variations in crossing over. By forcing the fiber to cross in a smaller zone, the chances that fibers cross at the same angular location increases. Hence, smaller zones tend to lead to cleaner jog zones.

Thus, there is a basic trade-off in the jog zone size. Larger zones tend to resist

conditions, all the fibers would cross over at exactly the same angular position on the coil, but variations in the winding and in the fibers cause crossings in the zone to be inconsistent.

For example, two adjacent

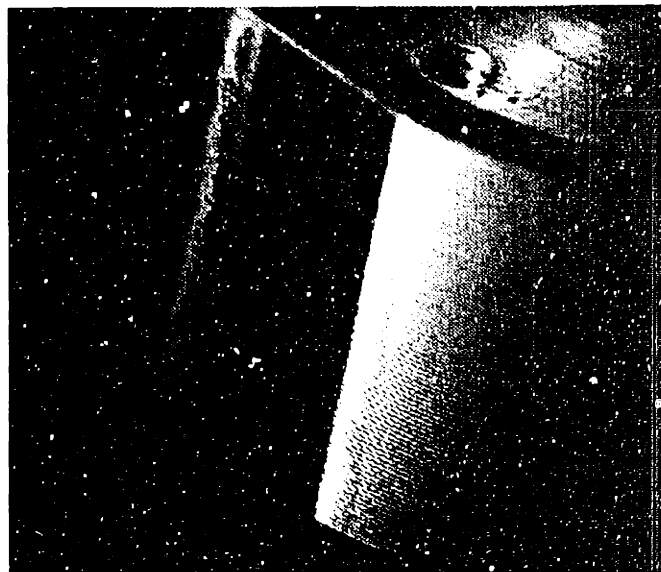


Figure 71. Flat on Grooved Mandrel

spreading and remain straighter along the width of the mandrel. Smaller zones have better controlled cross over locations and remain “cleaner.” A set of experiments were performed to determine the optimum zone size.

A feature of the grooved mandrels not previously mentioned is a smooth area on the circumference where the jog zone occurs. This smooth area of the mandrel is depicted in Figure 71. On the base layer, the fibers cross from one groove to another in this smooth area and thus define the jog zone for the entire coil. Individual grooves are difficult to see in the picture; the grooved portion is the lighter area on the mandrel.

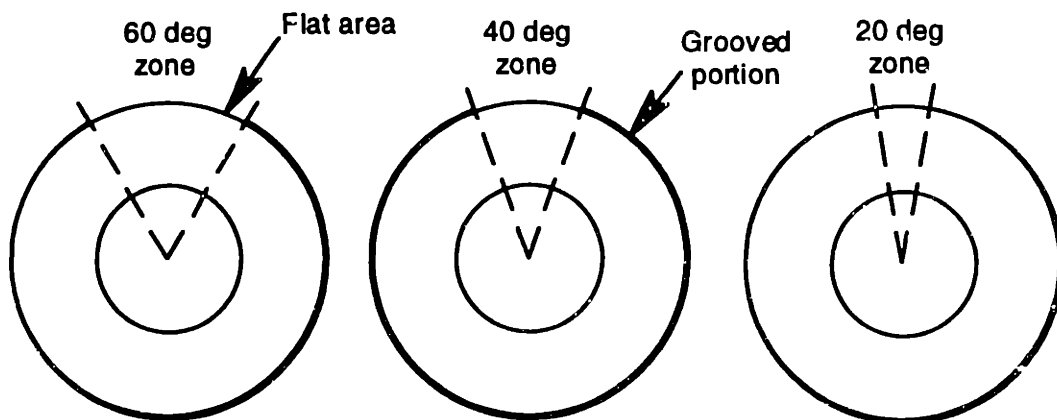


Figure 73. Grooved Mandrel with Different Zone Sizes

To determine the optimum zone size, several mandrels were created with different zone sizes and fiber was wound onto them. Figure 73 shows a profile of the coil with varying zone sizes from 10 to 60 degrees. As with the spacing experiments, the only real graduation between the winds with different zone sizes was the general wind quality. Upon examination of each of the winds, it was determined that the coil with the 20 degree zone exhibited the cleanest jog zone with minimum spreading. In actuality, there was no sharp delineation

between the different winds. It appeared that the 20 degree zone would lead to the best coils based on the winding experience of the design team. As a result of the experiments, the 20 degree mandrel was chosen to be the base line for the automated winding machine.

Chapter 6 - Conclusions and Recommendations

The assembly of the overall winding machine involved the installation of the hardware of the various subsystems, the integration of the electronics necessary to drive each subsystem, and the development of the control and interface software. Each of the subsystems were individually developed, integrated into the machine structure, and adjusted as necessary. This development of the subsystems has been discussed in this thesis and in the thesis written by Steve Lin.

Once all of the hardware was installed and functioning within acceptable parameters, the electronics were laid out within the machine base and wired to the appropriate items in the hardware. Among the electronics were the amplifiers and power supplies that would drive all the motors, the various sensors installed throughout the machine, and the data acquisition and motion control circuit boards installed into the controlling computer. Much attention was paid to the organization of the wiring as it was quite extensive and had the potential to become quite unmanageable. Predetermined electrical connectors were used throughout the machine and wires were carefully bundled and color-coded according to which physical element in the machine they corresponded.

Development of the software for the machine was an extensive process. All of the software was designed and written in C++ by the Steve Lin and the author. Much of the groundwork was completed in the development of the test-bed as that machine required a relatively high degree of computer control.

Making the final winder fully automated, however, required a much higher degree of sophistication of the software. There are over 11 axes to be controlled on the machine, in addition to the many actuators that require on/off outputs from the control software. Moreover, many of the axes had to be moved simultaneously during winding and tension control had to be maintained at all times. A great deal of design and development was required to bring all of the control software on-line.

In addition to the control aspects, the interface with the operator also required a good deal of work. Given that the machine was to be used in a production environment, the control program had to be robust and simple to use. In contrast, the actual process of winding a coil automatically is extremely complicated. The interface software served to remove much of the complexity through canned functions that the user could access through a command list. With this approach, the operator would only need to write instructions from a list of available alternatives into a command file. The software would interpret these commands into sets of specific machine instructions and execute them as necessary. A lot of effort went into defining the available user commands and relating these to specific machine operations that would accomplish the desired command. An example of a command would be "Swap." By writing this command into the command list, the machine would know to execute the set of operations necessary to perform the swapping operation. The software and electronic design details could not be presented for reasons of confidentiality and so are not included in this thesis.

In the end, the automated fiber optic coil winding machine successfully met all of the specifications laid out by the customer and was fully capable of automatically winding the gyroscope sensing coils. Each of the subsystems, the electronics, and the software were successfully integrated and applied to the process of coil winding. The investigation into the different process related issues proved to be essential to the success of the machine. The research put into developing the grooved mandrel, the fiber spacing, and the optimal zone size, as well as into other issues discussed by Steve Lin, resulted in a marked simplification in the process of coil winding. By making the process simpler, the machine was better able to wind high quality coils, having fewer unknowns to deal with.

The design methodology followed in the design of the machine greatly facilitated the development of the winder. Team brainstorming and concept selection proved invaluable. It was the development of models and prototypes that made the greatest difference. For each of the subsystems, these prototypes quickly revealed any design deficiencies and demonstrated the functionality of the concept. They helped to quickly choose between competing concepts and reveal possible improvements. Of greatest importance to the success of the design was the creation of the test-bed winder. This machine served both as a testing ground for the design concepts and as a research tool for defining the winding process. Without this machine prototype, successfully building the final automated winder would have been extremely difficult. As it turns out, the prototype itself is actually superior to most existing coil winders and will continue

to be used for semi-automated winding. A photograph of the final winder is included below (Figure 74). The machine, the machine base, and the control computer are all visible. This picture has been purposely degraded to maintain confidentiality.

For future research projects of this kind, it is highly advisable to create

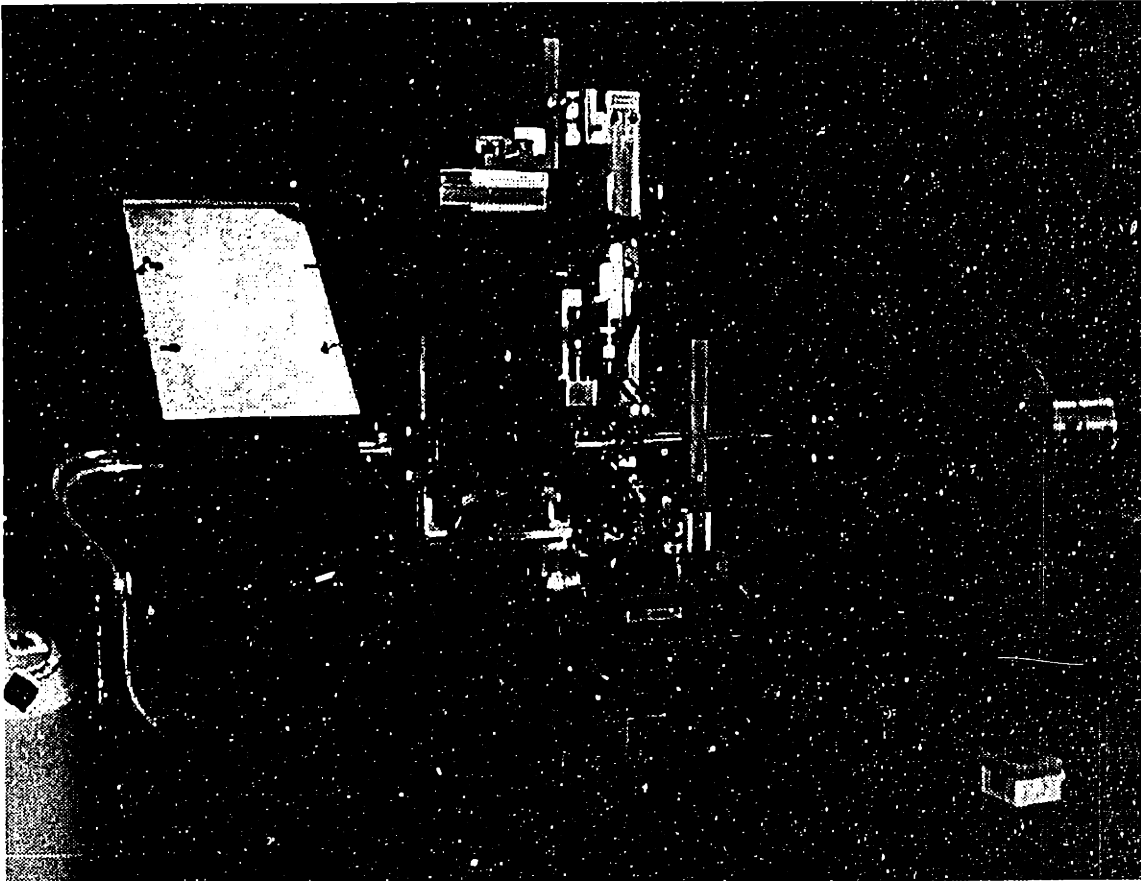


Figure 74. Automated Winding Machine

prototypes, and, if the process is not clearly defined, to use these prototypes to aid in the process definition. While this will take time and effort up front, it will greatly aid in the success of the end product. Also, if there is time available, computer-based solid models would also be useful. Such models are useful in determining interferences and defining dimension in the machine space. If the

right software were used, vibrational and stress analyses could also be used as an aid in the design. Due to restrictions laid out by the customer, there wasn't time to create these models during the design process. Determination of stress and vibration problems was left to the experience of the designers.

In conclusion, the design and development of the automated fiber optic coil winder was a successful venture. Both the customer and the Manufacturing Institute at MIT were pleased with the final outcome. The winder was put to use as an actual production machine soon after the completion of the project.

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