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Frank Speckhart, Major Professor

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THE DESIGN AND PROTOTYPING OF A VESTIBULAR STIMULATION DEVICE

A Thesis

Presented for the

Master of Science

Degree

The University of Tennessee, Knoxville

E. J. Asbury III

August 1990

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ABSTRACT

Stimulation of the vestibular system (inner-ear balance center) has been proposed as a major therapeutic tool for the treatment of children and adults with neurological disabilities. Numerous studies have attempted to establish its value and, in practice, different forms of the therapy are being used by occupational and physical therapists in the treatment of individuals with these disabilities. There have however, been only limited systematic investigations of the value of different forms of vestibular movements and their clinical results. A major drawback to such research is the lack of a well designed research tool. This thesis discusses the design, construction and testing of such a device and is presented as a masters thesis in Mechanical Engineering, Machine Design emphasis.

The device may rotate the patient on three independent axes. By a rotation of one of these axes the patient may be pitched forward and back, rolled from side to side or spun about a vertical axis. The patient is rotated about the center of gravity of the patient/machine system. This situation results in very low torque requirements at each of the device joints. By simultaneous rotations of the three machine axes the patient may be rotated about any desired axis. Computer controlled

stepper motors drive each of the three axes. This drive system allows for precise and programmable control of the motion. Each joint is driven by the stepper motor through a custom designed worm gear reduction. Aluminum castings were designed and fabricated to house the worm gears selected. The patient is seated in a TUMBLE FORMS brand carry seat that provides positive patient positioning and restraint. A fiberglass enclosure prevents the patient from any possible contact with the machine joints or foundation. The machine joints and foundation are fabricated from extruded aluminum tube.

This thesis discusses the methodology of the design of the device. A design overview, prototype design and construction, assembly instructions, performance testing and cost analysis of the prototype, and recommendations for future models are presented.

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CHAPTER 1

INTRODUCTION

1.1 Therapy Explanation

The vestibular system is a person's inner ear balance center. The primary purposes of the vestibular system are to detect motion, detect and respond to the earths gravitational pull, and influence motor co-ordination, posture, and muscle tone. A young child with poor vestibular system development will typically exhibit delays in motor skill, poor balance and co-ordination, and distractibility. If these problems remain undetected until the school age years problems in reading, writing and other skills may emerge as a result of the poor vestibular function and sensory integration. The basic theory of the therapy is that if a child with poor vestibular function is given movement therapy, the neurological deficit will be repaired to some degree. The most beneficial types of motions have not been identified and there is no measured data on their effectiveness. Motion identification will be the primary goal of the research done with the prototype. This type of motion therapy also holds promise for a wide range of other disorders such as down-syndrome and cerebral palsy.

1.2 Project Initialization

The idea for a vestibular stimulator was brought to light in the summer of 1987. Dr Andrew Zimmerman, a child neurologist at Fort Sanders Hospital in Knoxville recognized the need for a device to assist in the therapy being done in vestibular stimulation. He also desired a research device that was capable of moving a patient in a variety of ways so the effect of various types of motion could be investigated. Dr. Zimmerman related his need to Dr. Frank Speckhart professor of mechanical engineering at the University of Tennessee, who agreed to investigate if some sort of device were possible. In the fall of 1987 a senior design class developed two possible conceptual designs. The complete design and prototyping of a vestibular stimulation device was then selected as a masters thesis.

1.3 Design Goals and Requirements

The proposed device is intended as a medical research and therapy tool. The target group for the initial research is children of ages 1 to 4 years. The research tool must be able to precisely rotate these children through a variety of motions. Repeated sinusoidal motion about various axes has been presented as the most likely form of initial testing. This requirement implies the device must provide precise motion over many cycles

and the axis and speed of rotation must be variable and easily selected by the researcher. The premier design requirement of the device is the complete safety of the child. The patient must be held securely in position and protected from all moving parts. It is with these broad criteria, and an emphasis on low overall cost and ease of manufacture and assembly, that the design is based.

1.4 Thesis Scope

This thesis discusses the design and prototyping of a medical research and therapy device. This work includes an interpretation of the requirements of the device. Since no such device exists, the design goals and requirements were interpreted from discussion with industry personnel as part of the project. This thesis covers all mechanical design of the device. The requirements of the controlling computer program were also interpreted as part of the project. The actual control program was written by Dr. James Euler and is not presented as part of this thesis. The design of the device is consistent with limited production on a job-shop basis. The project also included the design and fabrication of part molds required for the prototype.

This thesis discusses the methodology of the design of the device. A design overview, prototype design and construction, assembly instructions, performance testing and cost analysis of the prototype, and recommendations for future models are presented.

CHAPTER 2

DESIGN OVERVIEW

2.1 General Description

Three perpendicular axes of rotation are used. The patient may be pitched forward and back, rolled from side to side and spun about a vertical axis. The patient is rotated about the center of gravity of the patient / machine. This layout results in very low torque requirements at each of the joints. By simultaneous rotations of the three machine axes the patient may be rotated about any desired axis. Computer controlled stepper motors drive each of the three axes. This configuration allows for precise and programmable control of the motion. Each joint is driven by the stepper motor through a worm gear reduction. The worm gears provide for a high gear reduction in a relatively small space and are non-backdrivable. The inability of the gears to backdrive means that no amount of torque on a joint axis can result in an undesired rotation. This condition provides an important self-locking feature to the machine when it is at idle and in the event of a power failure. Due to the expense of existing worm gear drives on the market and the low torque / speed requirements of each joint, a custom gearbox was designed for the device.

Aluminum castings were designed and fabricated to house the stock BOSTON GEAR worm gears selected. The patient is seated in a TUMBLE FORMS brand carry seat that provides positive patient positioning and restraint. A fiberglass enclosure prevents the patient from any possible contact with the machine joints or foundation. The machine joints and foundation are fabricated from extruded aluminum tube. The complete device is shown in Figure 2.1.

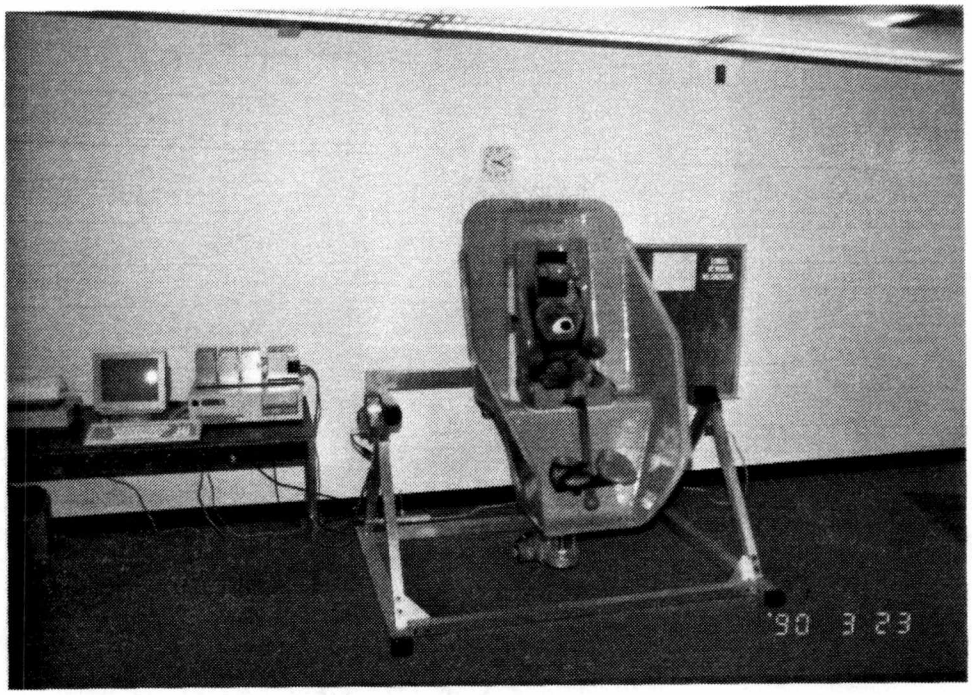


Figure 2.1 Vestibular Stimulation Device.

2.2 Motion Description

The machine uses three axes of rotation. These axes basically coincide to the Euler angles of pitch, roll and yaw. The first axis in the chain is the pitch axis. It is mounted to the foundation and rotates the roll and yaw axis. The roll axis is mounted to the pitch frame and rotates the yaw axis. The yaw axis is mounted to the roll frame. A rotation of the yaw axis does not rotate any of the other axes. The pitch and roll axes each rotate through +/- 30 degrees. The yaw axis may rotate through 360 degrees as in a continuous spinning motion or may rotate sinusoidally through any angle equal to or less than 30 degrees. The axes of rotation of the device are presented in Figure 2.2.

By coordinating the rotations of the three axis the patient may be placed in any position and orientation within a certain operating space. The patient may also be moved about this operating space in an infinite variety of paths. In order to visualize the operating space, imagine the patient seated in a chair with his torso and head reclined 10 degrees from the vertical. Now imagine an inverted circular cone with an included angle of 60 degrees placed about the patient. The cone is fixed in space with its apex located in the patients lower abdomen. This is called the operating space cone. A positional reference axis is now defined for the patient. The axis

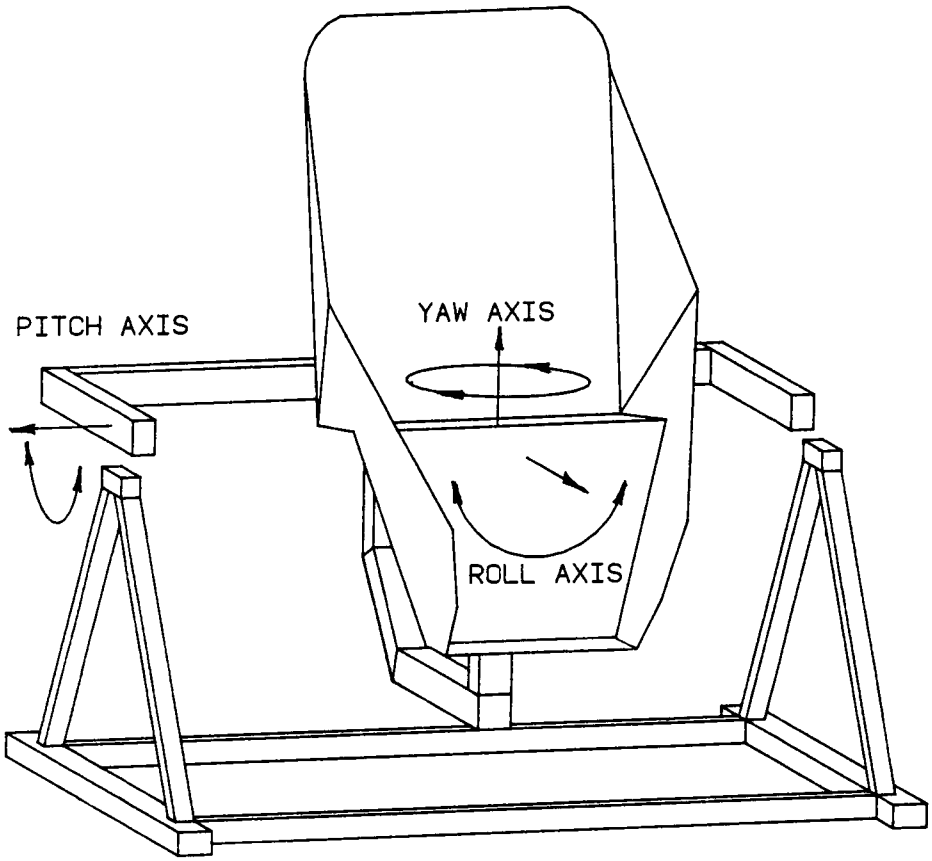


Figure 2.2 Device Axes of Rotation.

passes up vertically through the patients torso and head. This reference axis always passes through the apex of the operating space cone and initially coincides with the space cones centerline. By the use of the above references the possible patient motions will now be described. Photos of the device executing a variety of motions are provided in Appendix A.

Through rotations of the pitch and roll axes the patients reference axis, and of course the patient with it, may be rotated to any location within the space cone.

This rotation will occur about the apex of the operation space cone. From an inertial reference the patient will appear to be rotating about a point within his lower abdomen. Through continuous rotations of the pitch and roll axis the position of the patients reference axis may be continuously changed within the operating space cone. In this manner the reference axis may be made to follow a path within the space cone. A rotation of the yaw axis spins the patient about the patient reference axis. By rotations of the yaw axis the patient may be made to always face the center or outside of the space cone even while the patient reference axis is following a path.

2.3 Major Components

The machine may basically be divided into four major components: frames and foundation, mechanical drives, enclosure, and tumble forms chair.

2.3.1 Frames and Foundation

The frames comprise the major stressed portion of the machine. They are made of 6063-T5 square aluminum tubing with a 1/8 inch wall thickness and are TIG welded to shape. The pitch frame is fabricated from 3 X 3 inch square tube and 1.75 X 3 inch rectangular tube. The roll frame is fabricated entirely from 3 X 3 inch square tube. The foundation serves as the supporting base of the

machine and also serves to partially enclose the internal moving parts. The foundation is fabricated from 1.75 X 1.75 inch square tube and 1.75 X 3 inch rectangular tube. The pitch, roll and foundation frames are presented in Figure 2.3, Figure 2.4, and Figure 2.5 respectively.

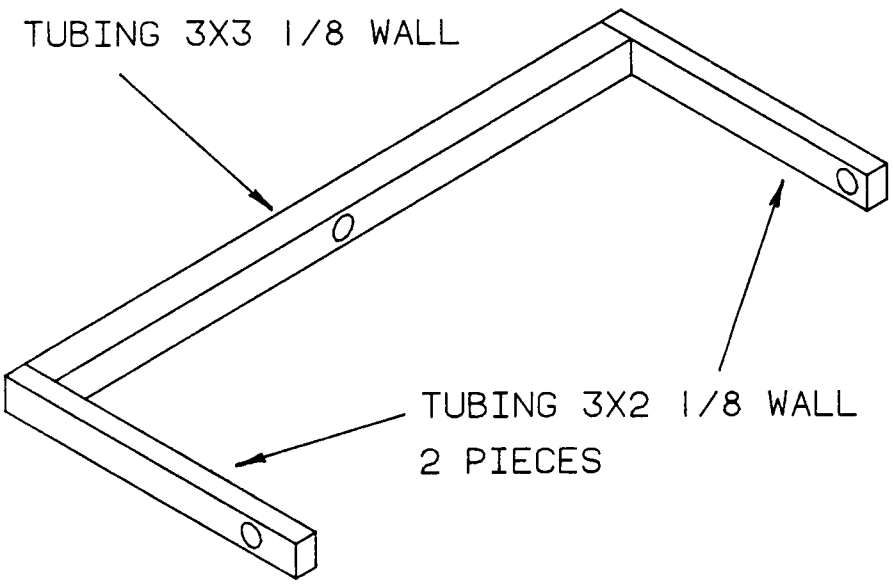


Figure 2.3 Pitch Frame.

In order to facilitate manufacture, all frames are planer in layout. This allows the frames to be clamped to a suitable flat surface for welding thus eliminating the need for costly holding fixtures. Consideration was also given to joint design between the components of each

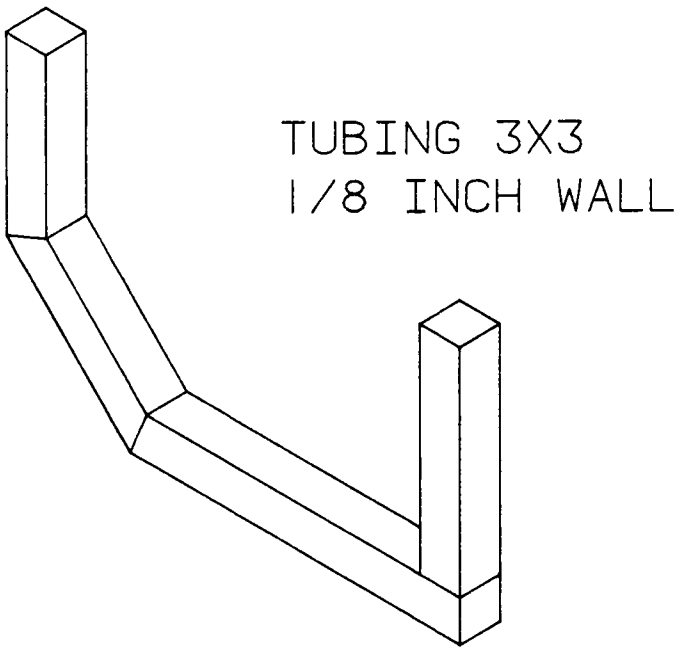


Figure 2.4 Roll Frame.

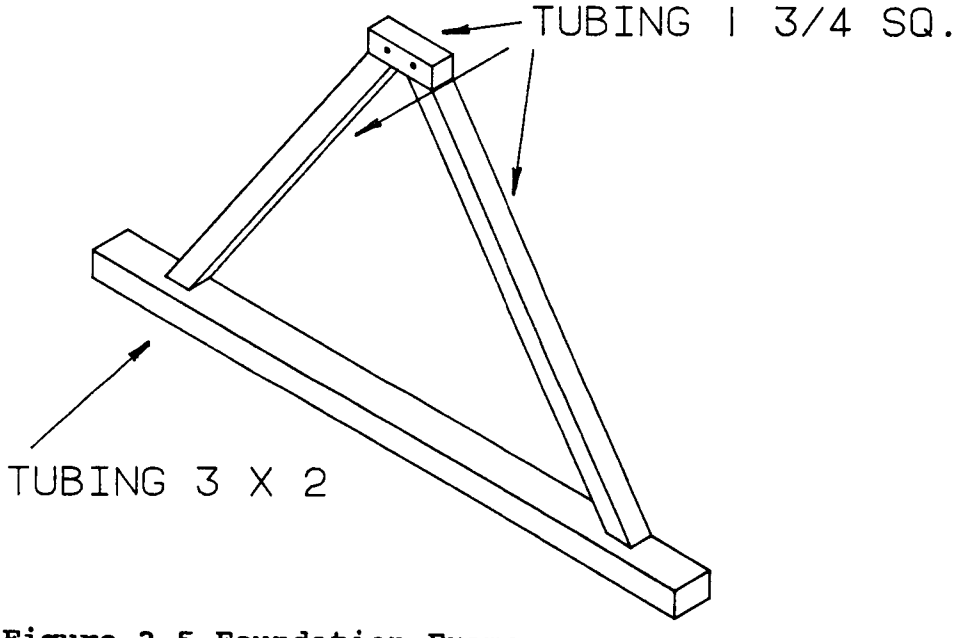


Figure 2.5 Foundation Frame.

frame. The majority of joints use straight cuts through the mating tubes at 90 degrees. All welds are full penetration square groove butt welds. This type of joint when used with the relatively thin 1/8 inch wall tubing requires little or no joint preparation prior to welding.

2.3.2 Mechanical Drives

The device uses worm gear reductions driven by stepper motors. The gearboxes were custom designed for the device. They utilize stock Boston Gear components operating in a cast aluminum housing. A single gearbox and motor are used at each axis. The pitch and roll gearboxes are identical with the exception of the length of their output shafts. The yaw gearbox uses a double threaded worm and wheel to achieve less mechanical reduction while still utilizing the same aluminum housings as are used in the pitch and roll axes. The reduction for the Pitch and Roll axes is 60:1. The gear reduction of the Yaw axis is 30:1. All gearboxes utilize identical motor mount plates, couplings, worm shafts and spacers. The aluminum housings used at all three joints are identical and may be assembled into either a 60:1 or 30:1 gearbox.

Electronic D.C. stepping motors are used to drive each axis. The roll and yaw axis are driven by 300 oz*in stall torque motors. The pitch axis is driven by a 450 oz*in stall torque motor. The motors have 200 steps per

revolution or 1.8 degrees per step. All motors have identical mounting flanges and output shafts. This allows the use of a common motor mounting plate and coupling within each gearbox.

2.3.3 Enclosure

The enclosure is mounted atop the yaw axis on a pedestal and is the final driven member in the kinematic chain. The prototype enclosure is presented in Figure 2.6. A rotation of any of the 3 axes results in a rotation of the enclosure. The enclosure serves to partially surround the patient seated in the device. This condition results in a sense of security for the child. The enclosure also serves as a mounting base for the various size Tumble Forms chairs. It is sized and shaped to prevent a child from reaching outside its boundaries when strapped in the device. The enclosure shape is contoured to provide positive retention of the chairs. The prototype enclosure was fabricated from fiberglass using a mold designed and built for the project. If additional devices are built their enclosures may be vacuum formed from plastic sheet using the same mold.

2.3.4 Tumble Forms Chairs

The Tumble Forms chairs are specifically designed for use in the care of disabled children. They provide

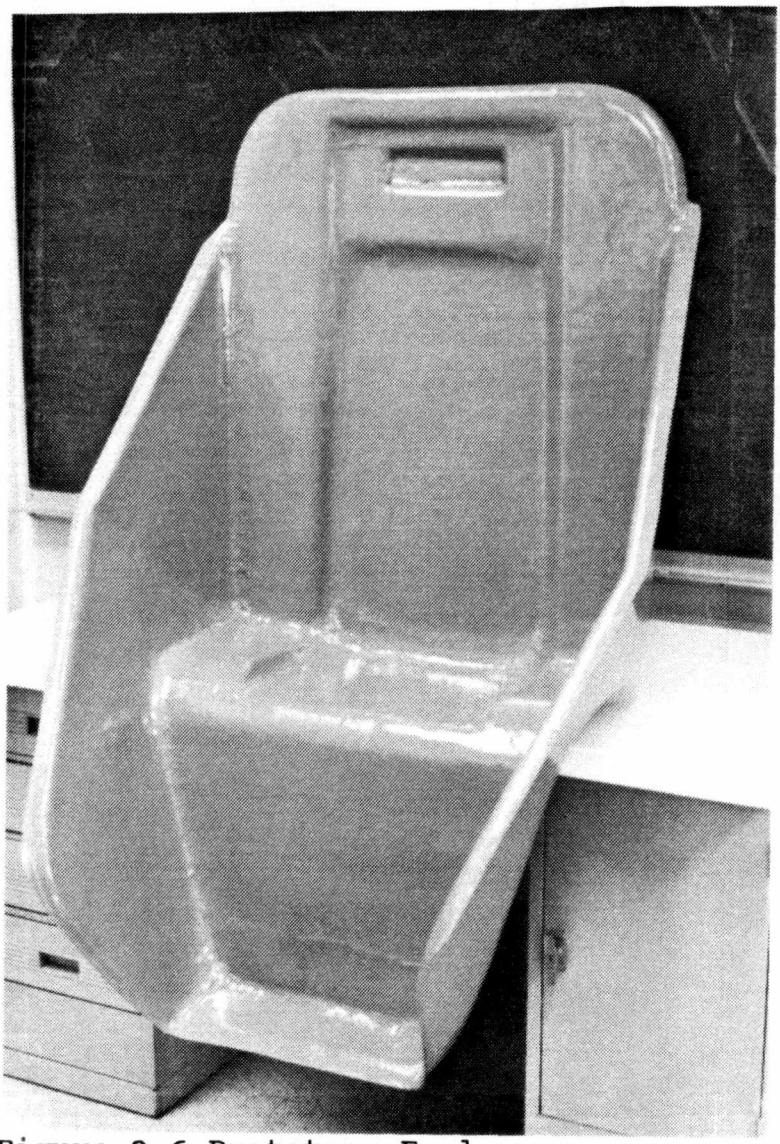


Figure 2.6 Prototype Enclosure.

comfortable, positive positioning of a child. The chairs are molded to provide support to the child's torso, head and hips. An adjustable foot rest provides leg support. The chairs are equipped with an adjustable lap belt, shoulder straps and foot straps. The use of the chairs ensures a fixed position of the child within the device.

The chairs are firmly mounted within the enclosure via two adjustable straps that are provided at the back of each chair. The chairs are available in three different sizes to accommodate from a 30 inch to a 58 inch tall child. This height implies a weight range of 20 to 100 lbs. The appropriate size chair may be easily mounted to the device for any child undergoing therapy.

2.4 Control

The device is controlled using a personal computer of 250 K memory or higher. Chopper circuits are used to control the pulsing of the stepper motor windings. The chopper circuits are controlled using the parallel port of the computer. The device is controlled "open loop" with no feedback. This is possible because of the precise nature of the stepping motors and robust operating space of the system. The controlling computer program was written by Dr. James Euler of the Mechanical Engineering department. The program is menu driven and allows a therapist to easily program any motion with a few simple commands. The main portion of the program is written in Quick Basic. An assembly language subroutine controls the actual pulsing of the chopper circuits. The assembly language routine was necessary in order to achieve the speed required to simultaneously control three axes with a digital computer.

2.5 Performance

The device is designed to accommodate up to a 70 lb child for any motion. The device may also be operated with no child within the chair. The weight of the child is used to counter-balance the weight of the frame and gearbox of pitch and roll axes. The ideal child weight is approximately 35 lbs. At this weight the center of gravity of the machine/child system aligns with the centerline of each axis. This results in very low torques required of the motors since the weight of the machine/child must not be lifted during the motion. Under these conditions the motor load is generated solely by the inertial of the machine / child. If a child that is heavier than 35 lbs is placed in the device the center of gravity of the system moves off the centerline of the pitch and roll axes. The torque required of the motor driving through it's gear reduction is now equal to the product of the distance from the C. G. to the centerline of the axis and the total weight of the machine / child. With the sinusoidal motion of the pitch and roll axis the motor must supply this torque during the portion of motion that the center of gravity of the system is lifted and will be driven to an extent by this torque when the C.G. is being lowered. The magnitude of this torque is much greater than the inertial torques generated by accelerating the machine and patient. When a child that is lighter than 35 lbs is placed in the

chair or when the device is operated with no child the center of gravity shifts in the opposite direction of that for the heavier child and torque loads are generated as before. The above descriptions are true only for the pitch and roll axes. The center of gravity of the system for any child weight always remains along the yaw axis due to symmetry of the loading. For this reason the yaw axis must only overcome the inertia of the patient, chair and enclosure during any rotation.

The pitch, roll and yaw axis may be moved sinusoidally a maximum of 30 degrees in either direction from their center position for a total range of motion of 60 degrees. The yaw axis may also be continuously rotated for a full 360 degrees of travel. All axis may be moved any increment of their maximum motions. The sinusoidal period of the pitch and roll axes for a 30 degree amplitude must be equal to or greater than 10 seconds. The period of motion for the yaw axis at a 30 degree amplitude may be 5 seconds or greater. The yaw axis may be continuously rotated at a maximum rate of 1 revolution per 10 seconds. The above maximums apply to any child within the 70 lb range. By using the menu-driven program the axes may be moved separately or in combination. The total time the device executes the desired motion may be any value and will be rounded to the nearest half period during execution.

CHAPTER 3

DESIGN METHODOLOGY

3.1 Consumer Feedback

The initial discussion about what the device needed to accomplish was held with Dr. Andrew Zimmerman. The design requirements were not rigid. Since no device existed for comparison the performance requirements of the device were not readily known. Safety of the patient was defined as the premier design requirement. The motions that would be most beneficial to the patient were also unclear. Rotational motions were preferred over straight line translations. The most obvious rotations were pitching the patient forward and back, rolling the patient from side to side and spinning the patient about a vertical axis. Dr. Zimmerman emphasized there had been very little investigation as to which motions were the most beneficial therefore a device that could rotate the patient on a variety of axes, through varying angles and at different speeds would be useful. The stimulation desired was the rotation of the inner ear in a gravity field, not any violent accelerations of the inner ear. For this reason smooth harmonic motions were desired. This type of motion would also be less frightening to the young patients involved. It was also desirable that the patient

could be spun continuously about the vertical axis. Rotation about the vertical axis was sighted as one of the more promising motions. This was seen as the most difficult design requirement.

Two local child rehabilitative therapy centers were visited. At Fort Sanders Rehabilitation Center some of the present methods of administering vestibular stimulation were demonstrated. A child was laid over a large beach ball and then slowly rolled from side to side, and forward to back. The angle and rate of rotation were not precise, and the movement required unreasonable physical effort by the therapist. Swing sets were also utilized to swing the children forward and back, and also to spin about a vertical axis. Talking with the therapy center personnel about the effect of the vestibular stimulation was extremely encouraging. The therapist had no doubt that vestibular stimulation had a positive effect on their patients. Their belief was so strong that they were willing to go to such physical extremes to administer it. The benefits of the therapy may not be widely accepted in the medical literature, but therapists who worked with disabled children on a daily basis strongly believed in its results.

3.2 Design Goals

After discussions with Dr. Zimmerman and the physical therapists, the general performance guidelines for the device began to emerge. A maximum child weight of 70 lbs was determined. This weight range would easily accommodate the range of children for which the therapy is most promising. Rotations of the child forward to back and side to side would be limited to a maximum of 30 degrees. This range of motion was seen as sufficient for the therapy to be effective while still maintaining positive child retention and comfort. It was also noted a larger angle might be frightening to the child. The rate at which the patient could be pitched forward and back and rolled from side to side was limited to a maximum of 6 cycles per minute. The patient could be rotated 30 degrees in one direction, reverse direction and rotate 60 degrees in the other, reverse direction again and rotate 30 degrees back to the starting point, all in 10 seconds. This meant the patient could rotate through 120 degrees total, with 2 direction changes in a minimum period of 10 seconds. For the spinning motion a faster rate of rotation was desired. For a 30 degree amplitude of oscillation the patient would be rotated at a rate of 12 cycles per minute, or 5 seconds per cycle. When the patient was being continuously spun the device would be able to complete 1 full rotation, 360 degrees, in 10 seconds. These maximum rates of rotation

were determined to be adequate for the therapy while limiting the amount of power the device required.

The consumer appeal of the device was also a major requirement. In discussions with the neurologist and therapists, a fear began to emerge. Their foremost concern was the safety of their young patients when in the grip of such a perceived "mechanical monster". This was noted time and again. Their second and more subtle fear was of the "programming" of the device. Therapists and neurologist are non-technical professionals. When the word programming was used, they became very uneasy. It was obvious the device motions must be easily selected by the operator using simple, straightforward commands. A concern over the potential loss of human interaction in the administering of the therapy to the young patients was also noted. For safety considerations, it was required that the therapist always be present during the operation of the device. This also helped to alleviate the human interaction concern.

The device is intended for use in professional therapy centers. Its aesthetic appearance was the first key to consumer appeal. It must be noted there is fierce competition for patients in the rehabilitation field. The professional appearance of the centers and the equipment they use is a strong influence on the number of patients they retain. The device had to appear attractive, safe and professional to the neurologists, therapists and patient's

families that were involved. It was not enough for the device to be safe based on engineering analysis, the device had to appear to be extremely safe to a non-technical consumer. This consideration implied a generously proportioned, robust device even to the point of exceeding the engineering requirement for the size and strength of its components.

3.3 Conceptual Development Theoretical Considerations

The conceptual development of the device had been ongoing since the beginning of the project. As the design requirements were identified and considered, broad theoretical considerations were exposed that would serve as guidelines for the conceptual design.

3.3.1 Maximum Flexibility of Motion

In discussions with Dr. Zimmerman, it was emphasized that the motions that were most useful in the therapy had not yet been determined or quantified by anyone in the neurological field. This investigation was to be one of the major goals of the research conducted with the device. From a design engineering standpoint this meant the device needed the most flexible variety of motions that could be attained. Rotations of the child had been identified as the desired form of stimulation. From rigid body dynamics it is shown that to rotate a body to any arbitrary

orientation in three dimensional space requires three consecutive rotations about three independent axes.

Each of the three axes must be capable of infinite positioning within their rotating range. In order for the body to be dynamically rotated through any motion, the axes must be able to move simultaneously and in the proper relation to each other.

3.3.2 Rotation About the Center of Gravity

In order to minimize the power required at the individual axes of the device it was found necessary to rotate about the center of gravity of the system. A senior design class in the mechanical engineering department that initially examined the possibility of a vestibular stimulator noted the lower torque requirement of rotating about the patient's center of gravity. Upon further analysis it was realized that the device had to rotate not about the center of gravity of the patient, but rather about the center of gravity of the entire patient and device system. The weight of the patient could be used to counterbalance the weight of the supporting structure and mechanical drives at each joint.

3.3.3 Center of Gravity Dilation

In using the weight of the patient to counterbalance the weight of the structure the center of gravity of the

patient must be set some distance off the centerline of the axis in question. The weight of the structure creates a moment about the axis in one direction. The weight of the patient is used to create a moment in the opposite direction, thus the two moments about the joint cancel. The distance the patients center of gravity must be placed from the centerline of the axis is dependent on the weight of the patient and the weight of the structure. The weight of the structure for any given design is a fixed quantity, hence the moment it will create about each axis is a constant magnitude. The weight of the patient is allowed to vary from a maximum of 70 lbs to a minimum of 0 lbs, or no child riding within the device. The case of operating the device with no child is important in that it allows a therapist to view a motion before administering it to a patient. The distance of the patient center of gravity from the rotational axis centerline will be a fixed value once the structure is assembled. It was considered impractical to provide a means of adjusting this distance for each individual patient weight. For this reason each axis of the device will be balanced for only one specific child weight.

For a child weight other than the ideal weight, the center of gravity of the system moves or "dilates" off the centerline of each axis. Figure 3.1 presents the location of the system center of gravity for various patient

weights as viewed from the side of the device along the pitch axis. The center of gravity shift will create a torque about the axis equal to the horizontal dilation distance multiplied by the weight of the system. A torque generated about a device axis in this manner will be called a dilation torque. The mechanical drive at each axis must be able to overcome this torque.

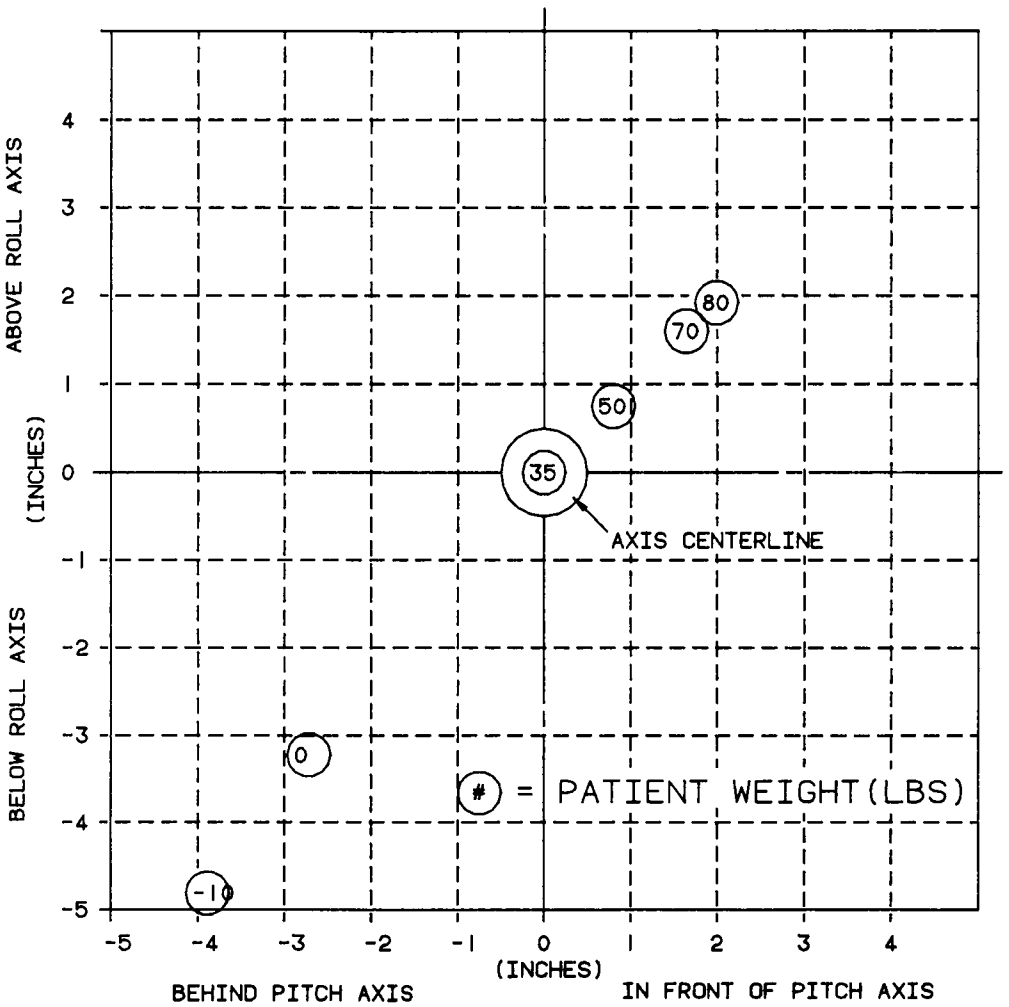


Figure 3.1 Center of Gravity Dilation for Various Patient Weights.

In order to minimize the power required to drive the device, it is necessary to minimize the dilation torque. The distance between the C. G. of the patient and the center of each axis must be adjusted such that the dilation torque for a 70 lb child is of equal magnitude but of opposite sign to the dilation torque that exists with no child. This adjustment is extremely important in that it centers the torque requirements of the device within the available operating space of the system. This results in an equal factor of safety on power for both a 70 lb and 0 lb child.

To minimize the magnitude of torque required at each of these cases it is necessary to minimize the distance between the patient C. G. and the centerline of each axis. This action minimizes the moment arm any excess patient weight has to generate torques about each axis. This analysis is also true for a patient that weighs less than the ideal weight. At each joint this distance is governed by the magnitude of the moment of the structure that must be counterbalanced by the patient weight. In order to minimize the maximum torque required of the mechanical drives, the moment of the structure must be minimized. The structural components must be placed as close to each axis centerline as possible and the weight of the individual components must be as light as possible. This also serves to reduce the inertia of the system about each axis.

Appendix B demonstrates the impact of the magnitude of the structural moment on the magnitude of the dilation torque.

In summary the design of the structure of the device should consider the following points:

- The structure and mechanical drives for the device should be made as lightweight as possible.
- The structural elements should be positioned as close to each axis centerline as possible.
- The layout of the structure must be such that the patient weight may be used to counterbalance the weight of the structure about each axis.

3.3.4 Order of Rotations

As stated in section 3.3.1 to rotate the child to any arbitrary orientation requires three independent axes. The maximum angle the patient could be pitched forward and back and rolled from side to side had been limited to 30 degrees. It had also been stated that the machine be capable of continuous rotations of the patient about a vertical axis. The continuous spinning motion was seen as the most difficult to achieve. To supply power to any axis that came after the continuously spinning axis in the kinematic chain would require the use of a special power connection. This connection would transmit power to the rotating mechanical drive from the stationary axis

directly before it in the kinematic chain. Connections of this type are commonly called slip rings. To supply power to the mechanical drive of the spinning axis would be relatively simple in that the drive would be mounted to the previous axis in the kinematic chain. To supply to an axis following either the pitching or rolling axes would only require sufficient excess cable to allow the joint to rotate freely. For these reasons it was chosen to place the spin axis last in the kinematic chain.

It was desired that the spin axis be faster than the pitching or rolling axes. This goal also supported placing the spin axis last in the chain since it would not be necessary to move any of the other axes with a spinning motion. This placement results in the least possible inertia to be overcome by the spin axis.

The last consideration is somewhat more subtle. If the spin axis were the last axis in the kinematic chain, the other two axes would be supporting it. This situation creates the opportunity to surround the spinning portion of the device with the structural frames of the other two axes. In this manner the continuously rotating portion of the device is partially enclosed or bounded by the structure of the other axes. The foundation supporting the device could also serve to bound the outer limits of the machine. This order of rotations could add to the safety of the device for both the patient and the operator.

3.3.5 Control

An important design goal was that the device be easily programed by a non-technical operator such as a neurologist or therapist. The operator should be able to select the axis or axes of rotation, the angular range the patient would be rotated through and the speed of rotation. The controlling interface must interpret these commands and pulse the individual stepping motors in the proper order. A menu-driven program running on a personal computer was seen to satisfy these requirements. The program would pose pertinent questions to the operator as a desired motion was programmed. Customer feedback from the initial prototype could be incorporated into the program. A personal computer would also provide a means of storing exact data on the therapy each patient received. In the medical research and therapy environment the ability to easily document the treatment that each patient receives would be a valuable asset.

3.3.6 Safety

The safety of the patient was the most important design requirement. The patient must not be able to come into contact with any potentially harmful part of the device. This requirement dictates the patient be securely and comfortably held in position. It was also determined necessary to enclose the patient in some type of cavity

within the device that prevents the child from any contact with the rotating members. The use of such an enclosure would define a volume within the device that the patient could safely occupy. This volume would add to the sense of security of the patient. The young children involved might envision themselves in the cockpit of some exotic spacecraft. The value of having the device appeal to the young patients should not be underestimated. The children would be much easier to deal with by the therapist if they enjoyed the treatment sessions.

The operator should not be in any danger during the device operation even with direct contact of the moving members. The maximum angular rotation and speed of rotation of each axis must be limited. The computer program may be used to control the device but should not be used as a fail safe to over-rotations or excessive speed. Joints should be mechanically limited to prevent a rotation beyond their intended maximums. Some provision should also be incorporated so a joint may not rotate substantially faster than intended. This would be termed a 'runaway condition'.

3.4 Axis Configuration Selection

As the theoretical guidelines were defined various axis configurations were being considered. It was difficult to develop a kinematic chain that would allow for continuous rotation of the vertical spin axis while still following the other design guidelines. The solution was found in the configuration of a common gyroscope. The gyroscope possesses three axes of rotation. The last axis is the continuously spinning axis of the flywheel. The patient would occupy the position of the flywheel and could thus be continuously rotated about one axis. The gimbals of the gyroscope would represent the pitching and rolling axes. The first axis in the kinematic chain would be the pitch axis. The second axis would be the roll. The final axis would be the continuously spinning axis that would be called the yaw axis. The initial concept of the device is shown in Figure 3.2.

This initial concept did satisfy the dynamic requirements for the motion but did not satisfy other consumer appeal considerations. The ease with which a patient could be placed into and removed from the device was an important consideration. The encompassing pitch and roll joints made access to the patient location difficult. After basic stress calculations were done, it was found the roll frame could be cantilevered from one side of the pitch frame. The portion of the roll frame extending from

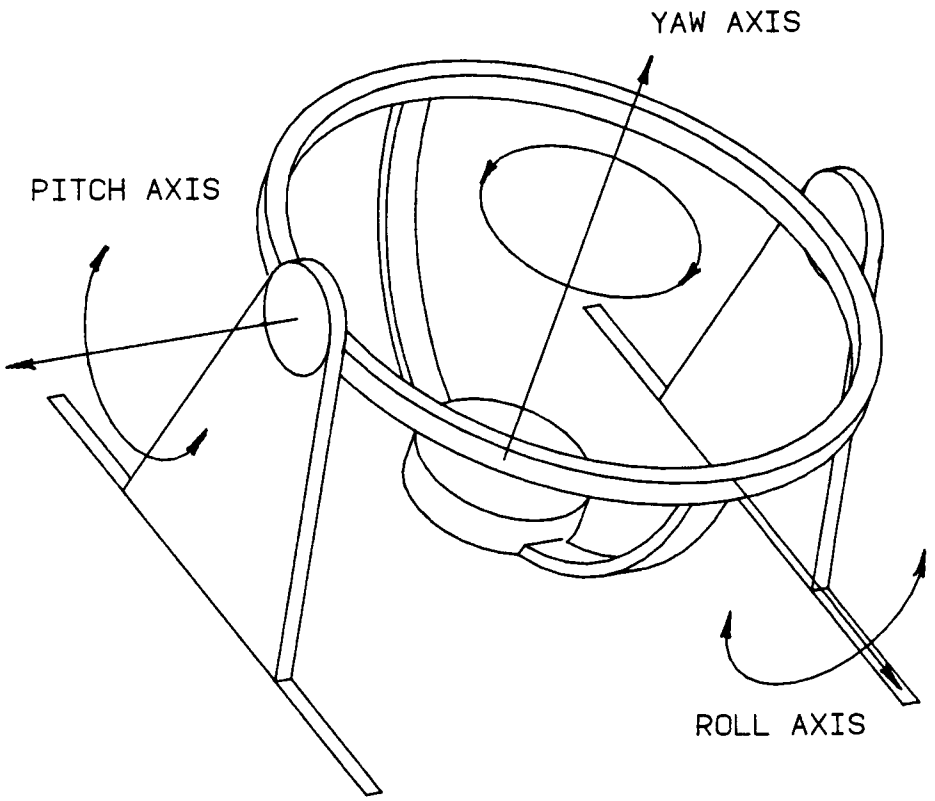


Figure 3.2 Concept 1.

the yaw axis to one side of the pitch frame could be removed. The same side of the pitch frame could also be removed to almost the centerline of the pitch axis. This modification would provide access for the patient and therapist to the interior of the device where the patient would be seated. The modified axis configuration is shown in Figure 3.3.

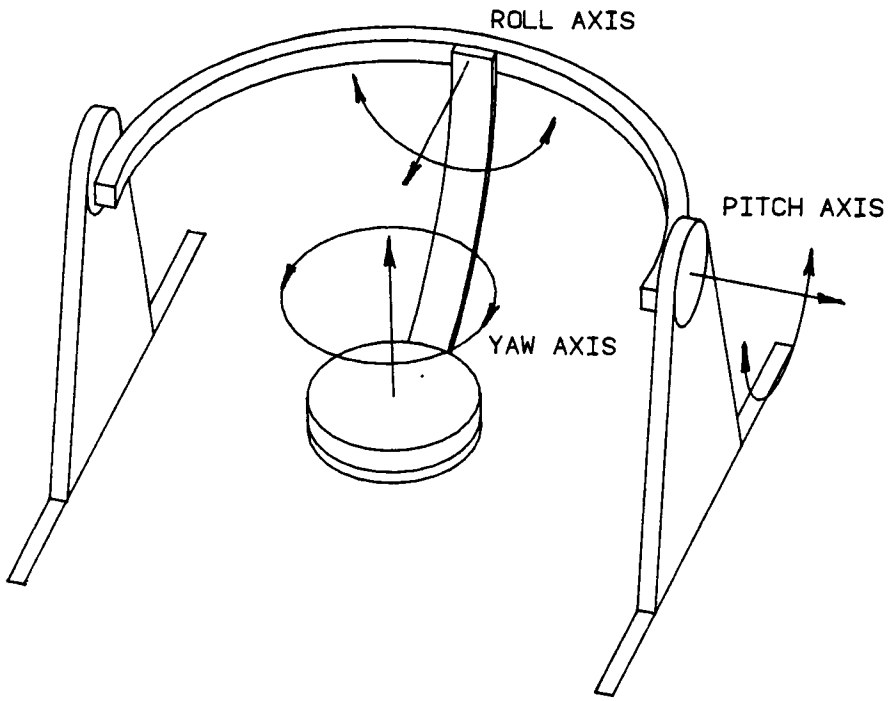


Figure 3.3 Concept 2.

3.5 Patient Restraint and Enclosure

The primary factors governing the safety of the device are the restraint and enclosure of the patient. The patient must be held firmly yet comfortably in position. The learning-disabled children for which the device is intended often have poor muscle control and posture. The seat within the device must provide gentle support to ensure the child remains in the proper "prone" position.

3.5.1 Chair Selection

Dr. Andrew Zimmerman suggested the use of a Tumble Forms brand Carrie Seat. Tumble Forms produces a seating system designed especially for disabled children. The seats are available in three sizes to accommodate a range of children. These seats are already in common use in therapy centers and private homes. They are equipped with a lap belt, shoulder straps and an adjustable leg support. These features would ensure the child is firmly restrained within the device. The Tumble Forms seats could be mounted to the device using two adjustable straps that are provided at the back of each. The seats are molded to a child's contours providing gentle support to the torso, head and hips. Since the chairs are popular in the rehabilitative field the therapists and children involved are already familiar with their use. The use of the Tumble Forms seats would also add to the professional appearance of the device. Vender data on the Tumble Forms chairs is provided in Appendix C.

3.5.2 Enclosure Shape and Size Development

To increase the safety of the patient, the Tumble Forms chair is mounted within a protective enclosure. The use of some type of enclosure would also add to the sense of security of the patient and to the aesthetic appeal of the device. The initial shape for the enclosure was

discussed using a clay model. A scale model of the Tumble Forms seat was constructed to use as a reference within the clay enclosure. It was determined that the enclosure should be sized so as not to allow the patient to extend an arm outside it's protective volume. Dr. Zimmerman requested that the enclosure not obstruct a patients peripheral view when seated within the device. It was also noted the patient should be able to place their arms comfortably at their sides and that a portion of the enclosure should extend above the head of the patient. The basic shape that was derived is shown in Figure 3.4.

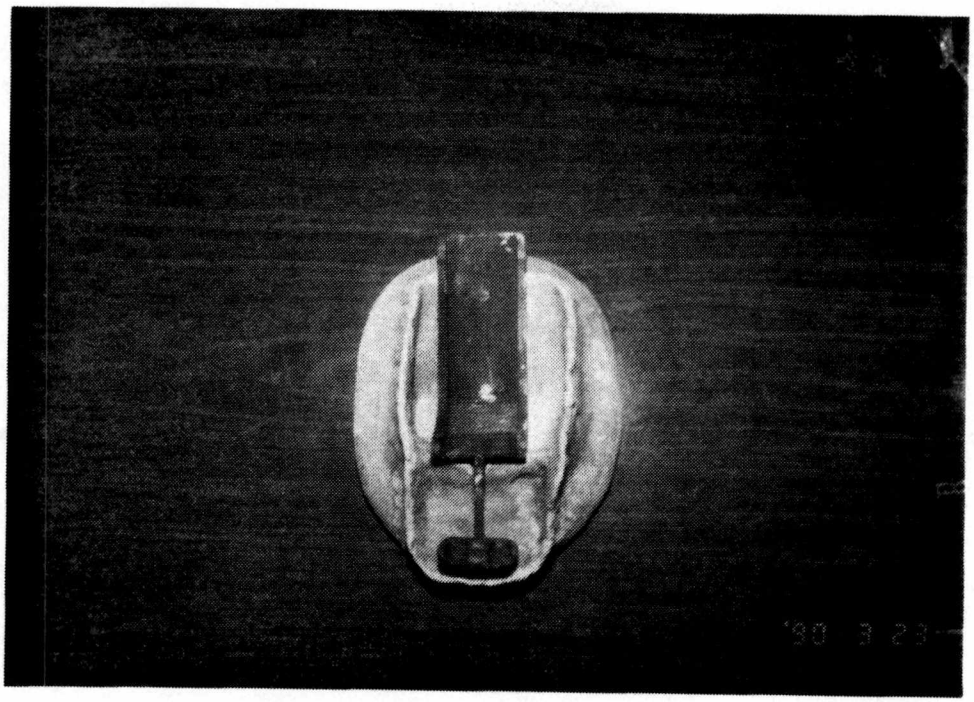


Figure 3.4 Enclosure Clay Model.

With the basic shape modeled in clay the actual size of the enclosure could be determined. A full-sized model was carved from a block of styrofoam using the largest size Tumble Forms seat as a reference. This initial enclosure prototype and the Tumble Forms seat were taken to Dr. Zimmermans office and to Childrens Rehabilitation Center where actual patients were studied seated within it. Small modifications were noted and the final enclosure size and shape were determined. A CADAM schematic of the basic enclosure shape is presented in Figure 3.5.

3.5.3 Method of Manufacture

Two methods of manufacture for the enclosure part were considered; fiber-glassing and vacuum forming. Fiber-glassing would result in an extremely strong and lightweight component. The laminate thickness could be varied as needed to accommodate the stress at each location within the part. The major disadvantage of fiberglass is the magnitude of labor involved in hand lay-up of the laminate. The vacuum-forming process may be used to produce parts rapidly in production. Plastic sheet can be purchased in a variety of colors and thicknesses and drawn into shape using a mold. Once the part is drawn, it needs only to be trimmed of the excess plastic flash. The major disadvantage of the process is the expense of the robust mold required to take the stress generated during

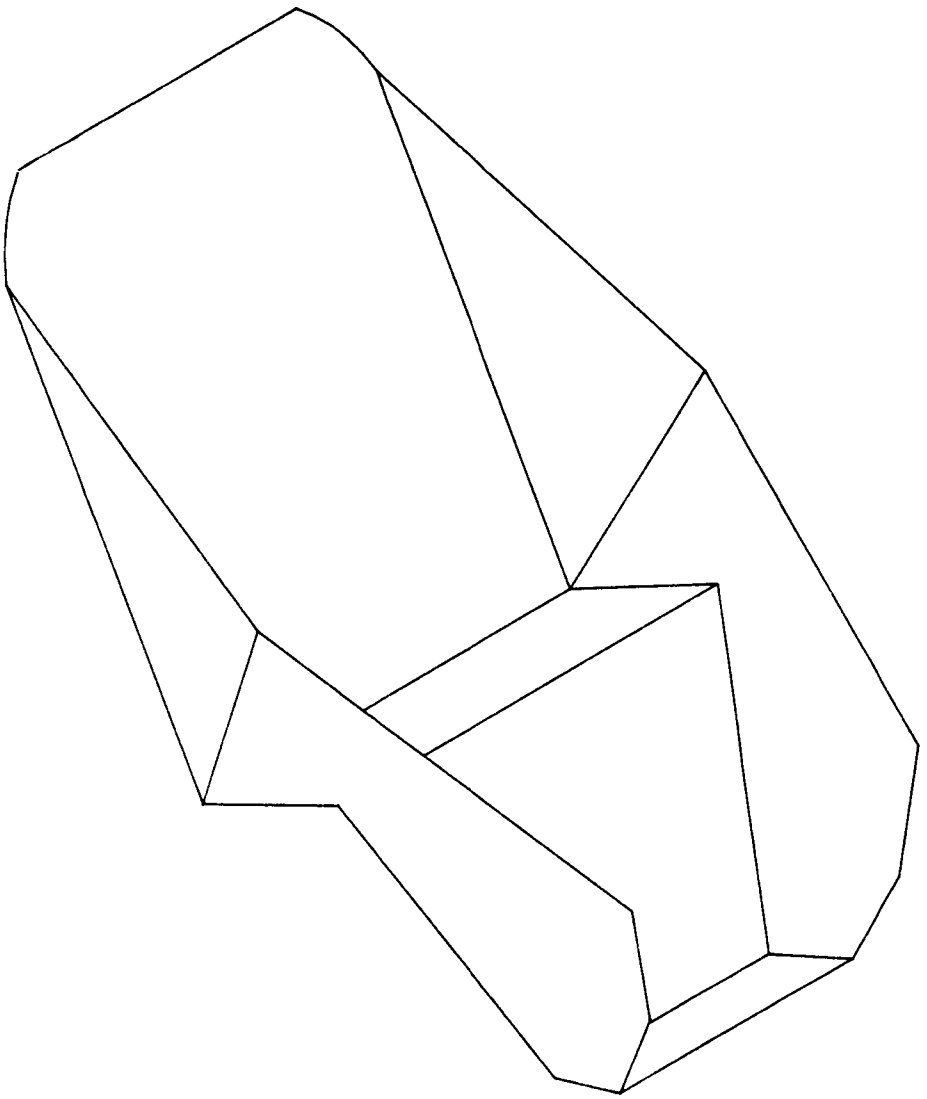


Figure 3.5 CADAM Schematic of Enclosure.

the drawing of the part. After consideration, the vacuum forming process was chosen. Mr. Joel Lenoir, a graduate student in the mechanical engineering department was commissioned to build the enclosure mold.

3.6 Frames and Foundation Development

3.6.1 Material Selection

The two principal materials considered for the device structural frames were aluminum and steel. In order to minimize the C.G. dilation and inertial torques at each axis, the frame components needed to be as lightweight as possible. The deflection of the structure under load was also an important consideration. The research into the effect of different types of motions could not be pervaded by excess compliance in the device structure. Perceptible deflection of the structure would also appear unsafe to the consumer. For this reason the required sizes of the frame structural elements were determined using a deflection analysis. After each frame component had been sized, a stress analysis was performed. Square tubing was selected for the main structural components of the device. The use of tubing would provide a lightweight structure with a high section modulus. The square section would facilitate drive component attachment to the frames. This would also simplify the design of the mechanical joints between frame member components.

Steel and aluminum tubing were compared using the deflection and weight of the resulting structure as guidelines. A steel vs. an aluminum tube was modeled using the second-order deflection equation of a beam. Each tube

was required to have equal deflection under equal load. It was found that the section modulus of the aluminum tube was required to be 288% that of the steel tube. Using the densities of steel and aluminum, the weight per unit length of each tube was considered. It was found that the steel tube would weigh approximately 165% that of the aluminum tube. This demonstrated that an aluminum structure would weigh much less than an equivalent deflection steel structure even with the larger aluminum tubing sizes required. The above analysis is presented in Appendix D.

There are other considerations in the material selection. An aluminum structure would provide an attractive surface with no secondary operations. If needed, the bare aluminum could be easily polished and coated with an automotive wax. A bare aluminum structure was also perceived to provide a more professional, high tech. appearance. Steel tubing would require priming and painting after fabrication to avoid oxidation. The only disadvantage of aluminum noted was a higher cost when compared to steel. With the technical advantages sighted and the demand in the medical industry for the most professional equipment possible, the cost of aluminum was easily justified. After review of the stock aluminum tubes manufactured, 6063-T52 extruded tube was selected. This tubing is available with a .125 inch wall thickness in a

range of sizes for both square and rectangular shapes.

3.6.2 Frame Layout

With the axis configuration and the enclosure size determined, the basic layout of each frame member could be developed. Curved frame members as shown in the initial conceptual drawings were considered impractical to fabricate. The aluminum tubing selected would tend to buckle when bent to a curved shape. Welded assemblies of straight sections of tube would be used in their place. Using a minimum number of sections the pitch and roll joint layouts were developed. The frame members were sized to allow approximately 5 inches of clearance between the member and the patient enclosure during any motion. The frames were continuously being modified as the enclosure shape and size was developed. The angle that the enclosure would be laid back within the device was also the subject of much debate. The lay-back angle of the enclosure governs the angle that the patient would be reclined when seated within the device. This angle is critical to the maximum stimulation of the patient during a rotation. The initial lay-back angle was 30 degrees, upon further review of the literature by Dr. Zimmerman and Karen Goss the angle was changed to 10 degrees. The use of CADAM made this development process much easier. The frame and enclosure drawings could be overlaid and their clearance

measured directly. Using the group function the patient enclosure could be rotated relative to the frame components in any of the three standard views to simulate a rotation of the device axes. A CADAM drawing of the device structural frames, foundation and enclosure is presented in Figure 3.6.

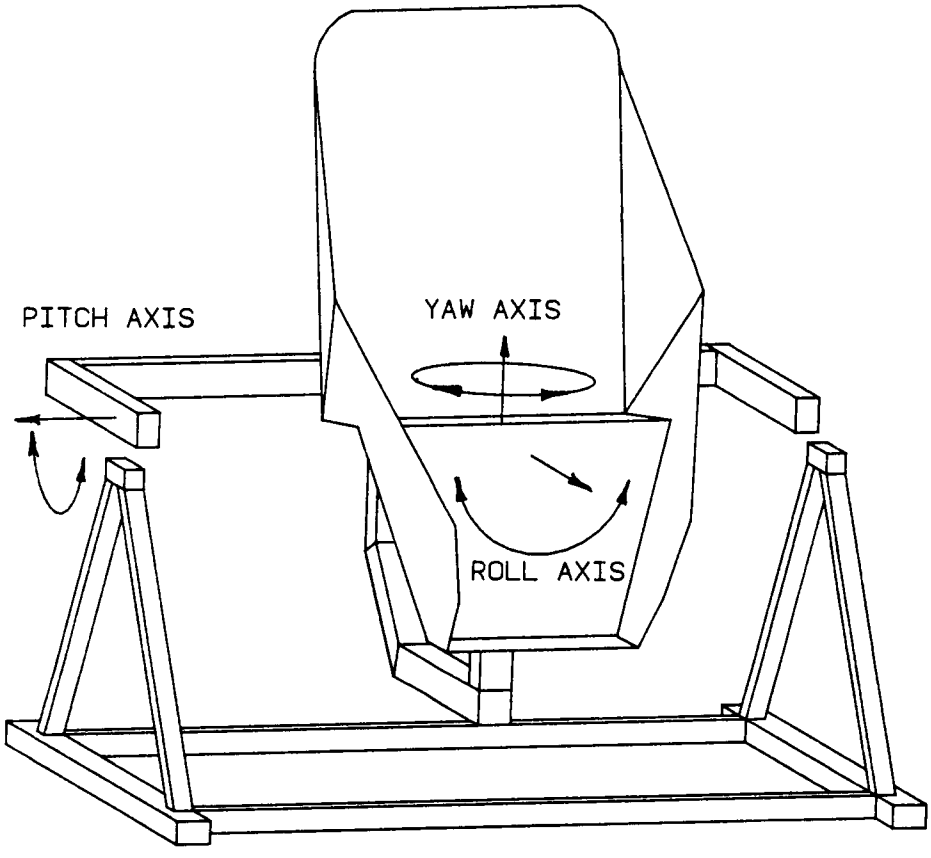


Figure 3.6 CADAM Drawing of Device Frames, Foundation and Enclosure.

3.6.3 Member Sizing Based on Deflection

The required size of the device structural frame elements was determined using a deflection analysis. The aluminum tubing selected was readily available in a range of rectangular and square sizes with a .125 inch wall thickness. The basic layout of the frame elements had been developed. The device frames were modeled using frame elements of various cross-sections by a structural analysis program. The program is entitled Static Structural Analysis on Microcomputers or SSAM.

SSAM may be used to analyze 2 and 3 dimensional linear elastic structures consisting of truss and frame elements. The program is based on the direct stiffness technique. This technique uses a system of linear equations to represent the structural model and its applied loads. The equations are represented by the following matrix form:

$$[K]\ [X] = [F]$$

the [K] is the stiffness matrix of the structure, the [X] matrix are the degrees of freedom of each node, and the [F] is the applied loads. Loads may be concentrated forces and moments at nodes or specified translations and rotations of nodes. These equations are then solved using the Gauss-Jordon method. The program output gives the translation and rotation of each node and the forces and moments at each node.

The actual size of the structure had not been fully determined at this point. The modeled size of the structure was 60 inches wide and 30 inches deep at the pitch axis. The end of the roll axis was placed 20 inches below and 30 inches away from the pitch axis. A concentrated load was applied to the end of the roll frame. The load represented the weight of the child, enclosure, and Tumble Forms chair and was estimated at 120 lbs. The structure was completely restrained from any translation or rotation at one side of the pitch axis. This side would be supported by the pitch gearbox. The opposite side of the pitch axis was restrained from any translation but was allowed to rotate in all 3 planes. The rotational freedom was done to simulate a spherical roller bearing supporting the idle side of the axis.

The frame model was constructed using elements of various cross sections and the resulting deflection of the structure at each node was found. By examining the individual nodal deflections, the members that required the highest area moment of inertia were identified. The backbone of the pitch axis required a high polar moment of inertial due to the torsional loading imposed on it by the cantilevered roll axis. A 3 inch by 3 inch square tube was found to result in acceptable torsional deflection at this location. The cross-section height of the two pitch arms was set at 3 inches in order to match the pitch backbone.

Various widths of the pitch arms were then modeled. A 1.75 inch by 3 inch arm was found to result in acceptable member deflection.

The roll arm section height is set to 3 inches. This dimension simplifies the design of the dynamic joint between the pitch and roll axes and would provide a high moment of inertia in the plane of the roll frame that experiences the greatest bending stress. The roll axis also receives a side load during a rotation of the roll joint. This load is equal to 60 lbs for a 30 degree rotation angle. This side load was imposed on the end of the roll joint in the structure modeling. A 1.75 inch by 3 inch section was found to result in acceptable deflections in both planes of the roll frame. Selected SSAM models and results are presented in Appendix E.

The prototype roll frame member is fabricated using 3 inch by 3 inch square tube. This was done for a combination of reasons. The larger dimension tube gives the device a more robust appearance. This increases the consumer appeal of the device. The use of a 3 inch width tube also simplifies the mounting of the yaw axis to the roll frame. And the 3 inch square tube is already used in the pitch backbone. The calculated vertical deflection of the final prototype structure for a 70 lb patient was .38 inch. The prototype model and result is presented in Appendix E.

3.6.4 Member Stress and Fatigue Analysis

A stress analysis was performed on the frame components using the basic equations for bending and torsion. The material strength used in the analysis was as published by the Aluminum Association. The published values for welded 6063-T5 aluminum are as follows:

Tensile Yield = 11 ksi

Tensile Ultimate = 17 ksi

Shear Yield = 6.5 ksi

Shear Ultimate = 11 ksi

Compression Yield = 11 ksi

These values apply within 1 inch of the welded joint. With the rapid TIG welding process used in the prototype the aluminum at the location of the joints should not completely soften to the annealed strengths shown. The conservative assumption that the aluminum is in the fully softened state will result in an added factor of safety. The fatigue of the structure is analyzed based on the Aluminum Association literature. With the type of welded joints involved in the prototype, the allowable varying stress range for a design life of over 2,000,000 cycles is 4 ksi. For a design life of 100,000 to 500,000 cycles the allowable varying stress is 8.2 ksi. In operation it is predicted the device will only receive 5,000 cycles per

year. The frame stress and fatigue calculations are presented in Appendix F.

3.6.5 Foundation Development

The foundation serves to support the moving frames of the device. Each foundation side frame supports one side of the pitch axis. The two foundation side frames are held together by two cross members. The height of the foundation was adjusted to provide approximately 3 inches of clearance between the bottom of the roll axis and the floor during rotation. The foundation frames and cross members also serve to bound the outer limits of the device. This enhances the safety of the operator. Each foundation side is placed outside the pitch axis and thus partially encloses or bounds the sides of the device. The bottom of each side frame was extended so that the majority of the patient enclosure lies within its limits when viewed from the side. This gives the tangible indication of when an observer is entering the motion range of the device. The sizing of the frames in this manner results in a wide stable device foundation.

The major load on the foundation frames is a compressive load on the angled vertical members. For this reason, even a small cross-section member may easily take the applied load. The size of the foundation members was chosen based on consumer appeal considerations. The

foundation had to appear strong and stable in the eyes of a non-technical consumer. Aluminum tubing measuring 1.75 inch square is used for the vertical and top members of each frame. The bottom of each frame is 3 inch by 2 inch tubing. The 3 inch dimension is placed parallel to the floor and gives the indication of a strong base member for the entire device. The aluminum cross members that tie the foundation frames together are made of 3 inch by 2 inch tubing.

3.7 Power Selection and Development

3.7.1 Required Axis Torque Analysis

The torque required at the device axes has two components. A dilation torque exists at each axis due to the center of gravity of the system lying off the centerline of the axis. The other component of the torque is due to the acceleration of the mass of the device during an axis rotation. In the analysis of the relative magnitudes of the two torques, the torque due to C. G. dilation is found to be much greater than the inertial torque.

The magnitude of the inertial torque was investigated using a dynamic analysis and the established performance guidelines of the device. We assume sinusoidal motion of the device axes. The pitch and roll axes would have a

maximum amplitude of 30 degrees and a minimum period of 10 seconds. The yaw axis would have a minimum period of 5 seconds for a 30 degree amplitude. Using these values the maximum angular velocity and acceleration of each axes was found to be:

Pitch and Roll Axes

Max. Vel. = .329 radians/sec

Max. Acc. = .207 radians/sec^2

Yaw Axis

Max. Vel. = .658 radians/sec

Max. Acc. = .827 radians/sec^2

The rotational inertial of the device about each axis was calculated. The maximum child weight of 70 lbs was used in the analysis. The enclosure weight was estimated at 30 lbs. The total weight of the child/ enclosure/ Tumble Forms chair body was estimated to be 125 lbs. A mass distribution for this assembly was modeled as a rectangular solid. The rotational inertia of the solid about each appropriate axis was calculated. The rotational inertial of the device aluminum frames were calculated about each joint axis. The mechanical drives were modeled as point masses and their inertia about each axis calculated. The total rotational inertial about each axis was found to be:

Pitch Axis

I = 132 in lb sec^2

Roll Axis

I = 110 in lb sec^2

Yaw Axis

I = 50 in lb sec^2

The maximum torque required about each axis may now be calculated using the Euler equations of motion. The Euler equations consider both the torque due to the angular acceleration of the body and any gyroscopic torque that may be developed about the axis in question. In the gyroscopic calculations, the maximum angular velocity of each axis is used even though both axes may not reach peak angular velocity at the same time as the axis in question is undergoing peak angular acceleration. The maximum torques found about each axis are as follows:

Pitch Axis

Dynamic Torque = 40.3 in lb

Roll Axis

Dynamic Torque = 40.5 in lb

Yaw Axis

Dynamic Torque = 43.7 in lb

It must be noted that these values are an absolute maximum. The inertia estimate about each axis is quite generous and the gyroscopic torques are given the maximum value possible even though no practical motion of the device may result in all the velocities and accelerations maximizing at the same time. The complete dynamic analysis calculations are presented in Appendix G.

The possible dilation torque about the device axes was investigated using a spread sheet. The weight of each device component was estimated as closely as practical. The moment arm each structural component had about a given axis was easily found using CADAM. The weight of the patient, Tumble Forms chair, and enclosure assembly were combined into one mass point. The location of the mass point was varied to counter-balance the structural weight. This analysis is carried out on the pitch axis in the horizontal plane. The roll axis is balanced in the vertical plane. The device posses mass symmetry about the centerline of the yaw axis and therefore need not be balanced on this axis. The two perpendicular pitch and roll balance calculations establish the position that the patient must be placed within the device.

The balancing of each axis is carried out in the following manner:

The magnitude of the structural moment is calculated using an estimated weight of each component and its distance

from the center of rotation. This moment is a fixed quantity in the analysis. The location of the combined patient/ enclosure/ chair weight is varied. The patient weights of 70 and 0 lbs are considered. The location of the combined weight is varied until the dilation torque of the 70 lb child is of equal magnitude but opposite sign to the dilation torque of the 0 lb child. This centers the torque load of the system within the available operating space of the system. The location of the combined weight is now fixed. The weight of child that results in no torque about the device axis may now be found. This weight will be called the ideal child weight.

During the assembly of the prototype, the location of the patient/enclosure/chair assembly was verified empirically. Weights were positioned on the roll axis to simulate the weight of the assembly. The torque required to move the pitch and roll axes could be directly felt by rotating the shaft of the stepper motor for each axis. The magnitude and location of the counterbalancing weight was varied until the torque required to move the axis was minimized. In this manner the proper location and height of the yaw axis could be determined. The complete torque balancing spreadsheets for the device in its final form are presented in Appendix H. Figure 3.7 shows the device during the manual balancing of the axes.



Figure 3.7 Balancing of Device Axes.

The magnitude of the horizontal dilation torque for the 0 and 70 lb child weights was found to be 282 in lb. The maximum possible inertial torque was found to be 44 in lb. This shows the maximum magnitude of the inertial torque is only 16% of the maximum horizontal dilation torque. For this reason the torque analysis of the device is based on the dilation torque.

3.7.2 Worms Gears Driven by Stepper Motors

The device axes were required to have a high angular resolution within their operating range. Each axis would be constantly varying velocity and direction. It was also desirable to control the device open loop. The open loop control means the personal computer controlling the device would receive no feedback as to the location of the axes. It was these requirements that dictated the use of direct current stepping motors.

A typical D. C. stepping motor has 200 steps per revolution or 1.8 degrees per step. When the device is driven through a mechanical reduction, the joint axis resolution would increase proportionally. A stepping motor may be run at continuously varying velocities by varying the time between the individual steps of the armature. The velocity resolution within the performance range of the individual motor is infinite. A stepping motor also requires no feedback. When the motor windings are pulsed in the proper sequence, the motor shaft will rotate through precisely one step. By keeping track of the number of steps the motor had taken in relation to an initial reference, the position of the device axis would always be known. The next step was to select the type of mechanical reductions used to drive the device axes.

The mechanical reduction was necessary to multiply the torque of the stepping motors. After considerable

thought, worm gear reductions were selected. A worm gear drive would provide a high gear reduction in a compact space. A worm gear drive could also provide an important self-locking feature to the device. Depending on the lead angle of the worm, these drives can be insensitive to torque applied to their output shafts. This insensitivity meant that no amount of torque applied to the device axes would result in a rotation of the axes. This feature would provide an extremely reliable mechanical brake to the device when at idle. This brake would act even with no power being supplied to the device and at any rotational position of the joint axes. In the event of a power failure during the execution of a programed motion, the device would simply stop in place. The use of a worm gear reduction would also mean the complete enclosure of all rotating mechanical drive components with no special guarding. This feature would augment the safety of the device. The housing of the gear drives could be directly bolted to the device frames. In this manner the gear drives would not only provide a torque, but would also serve as the main bearing members of each axis. A more subtle benefit of the worm gear drives would be a 90 degree power transfer between the motor and the device axes. The gearbox and attached stepping motor would lie in the same plane as the driven axes. With the axis layout selected, this feature results in the physical volume of

the drive units blending with the rest of the device in a complimentary manner. The only disadvantage noted of a worm gear reduction was a relatively low efficiency. With a single-threaded worm, the efficiency of this type of drive is typically 60%. With the advantages noted this was considered acceptable.

3.7.3 Motor Torque and Gear Reduction Analysis

A BASIC program was written to explore the possible gear reduction and motor combinations. After some calculations a torque of 400 in lbs was assumed as the torque required from a mechanical drive under dynamic conditions. The drive would be required to supply this torque at the maximum velocity of the device axis. The program iterates on gear reduction. For each reduction, the smallest gear pitch that will take the imposed dynamic load is found. The dynamic load is calculated from the required torque and the pitch line velocity of the gear. The strength of the gear is calculated using the Lewis equation. The strength of the bronze used in the gear is assumed to be 24 ksi and the pressure angle is assumed to be 14.5 degrees. These values correspond to the mass-produced Boston Gear worm gears. The program assumes a single-threaded worm and a contact ratio of 1. The efficiency of each acceptable gear-set is calculated using the published Boston Gear equation and an average value of

the friction coefficient. The program output includes the required step motor torque and speed for each reduction / pitch combination. The program and results are presented in Appendix I.

In making the motor selection it is important to note the torque verses speed characteristics of stepping motors. Stepping motors are rated by a maximum holding torque. This value is the torque the motor is able to supply at 0 shaft RPM. The stepping motors also have a maximum RPM. At this speed, the torque the motors are able to supply is said to fall to 0. The actual shape of the torque speed curve is both a function of the motor characteristics and the electronics used to drive it. In the absence of a performance curve for the particular combination used, a linear plot of torque vs. speed may be assumed. Thus the available torque will decrease in direct proportion to the percent of maximum RPM at which the motor is operated. This is a conservative assumption for most motor/driver combinations.

The gear reduction and motor selection were made after an analysis of the program data and a review of the available stepping motors. A series of stepping motors were available that have a holding torque of 300 or 450 oz in and a maximum speed of 300 RPM. The motors in this series have identical output shafts and mounting flanges.

After torque and speed calculations the following selections were made for each axis:

Pitch Axis

450 oz in motor

60 : 1 gear reduction

Roll Axis

300 oz in motor

60 : 1 gear reduction

Yaw Axis

300 oz in motor

30 : 1 gear reduction

It was noted that the 30 : 1 gear reduction of the yaw axis could be achieved by utilizing a double threaded worm. This would allow the use of the same physical size gears in all gearboxes.

3.7.4 Theoretical Operating Space Analysis

The torque each worm gear / motor combination is capable of supplying may now be calculated. The efficiency of the 60 : 1 gear reductions was found to be 62.2 %. The efficiency of the 30 : 1 reduction was found to be 76.3 %. The motors of each axis would be required to run at a maximum of 189 RPM to achieve the design performance goals. The maximum speed of each motor was 300 RPM. By use

of the linear torque-speed approximation the torque supplied by each motor was found to be 37 % of their individual maximum holding torques. The torque available at each axis at maximum velocity may now be calculated by the following formula:

Torque = Efficiency * .37 * Stall Torque * Reduction

For each axis this results in the following:

Pitch Axis Torque = 392 in lbs

Roll Axis Torque = 261 in lbs

Yaw Axis Torque = 160 in lbs

A theoretical operating space for each of the device axes may now be calculated. It is important to note the worst case dynamic condition that exists with each axes. Due to the center of gravity of the device/patient system dilating off the centerline of the pitch axis in both the horizontal and vertical planes, this axis will experience a maximum dilation torque at some angular position other than the pitch axis horizontal position. This torque is due to the vertical C. G. dilation exerting a force about the pitch axis as the system is rotated. Due to the sinusoidal motion the angular speed of the pitch axis is maximum at the pitch axis horizontal position. As the axis

is rotated, the speed of the motor decreases and, therefore, the available torque increases. However, the magnitude of the imposed dilation torque also increases. Initially the rate at which the load torque increases will be greater than the rate at which the available torque increases. At some angular rotation of the axis the rate at which the available torque increases will begin to exceed the rate at which the load torque increases.

To correctly determine the theoretical operating space of the pitch axis a BASICA program was written. The program iterates on patient weight and rotational angle of the axis. At each angle the dilation torque and available drive torque are found. The dilation torque is calculated using the known dimensions and weight of the structure. The dilation distance of the system C. G. is calculated in both the horizontal and vertical directions. The moment of the system about the pitch axis may then be calculated for each rotational angle of the axis. The available drive torque is calculated using the linear torque / speed assumption, the stall torque of the motor, and the gear reduction and efficiency. For each patient weight the minimum period of a 30 degree oscillation for which the available drive torque exceeds the dilation torque for all angles of rotation is found. The program and results are presented in Appendix J. The theoretical operating space is presented in Figure 3.8.

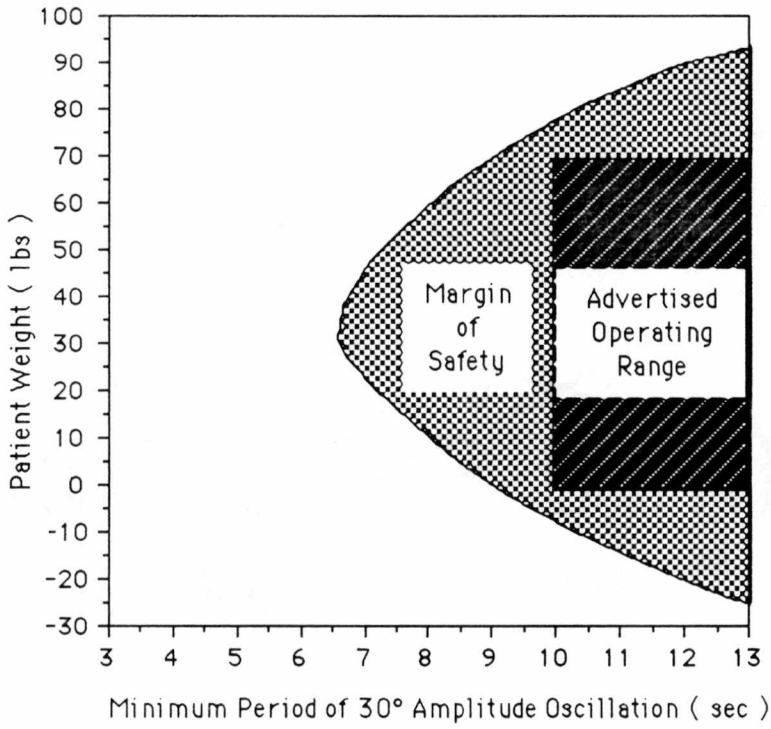


Figure 3.8 Pitch Axis Theoretical Operating Space.

The roll axis experiences a torque due only to the vertical C. G. dilation and the inertial torque. The dilation torque is a maximum when the roll axis is at maximum angular amplitude and minimum velocity. The dilation torque is theoretically 0 when the roll axis is vertical but, this is the point of maximum axis velocity. In the same manner as the pitch axis, a program was written to determine the theoretical operating space of the roll axis. The program finds the minimum period that the roll axis may execute a 30 degree amplitude

oscillation for a range of patient weights. The program and results are presented in Appendix J. The theoretical operating space is presented in Figure 3.9.

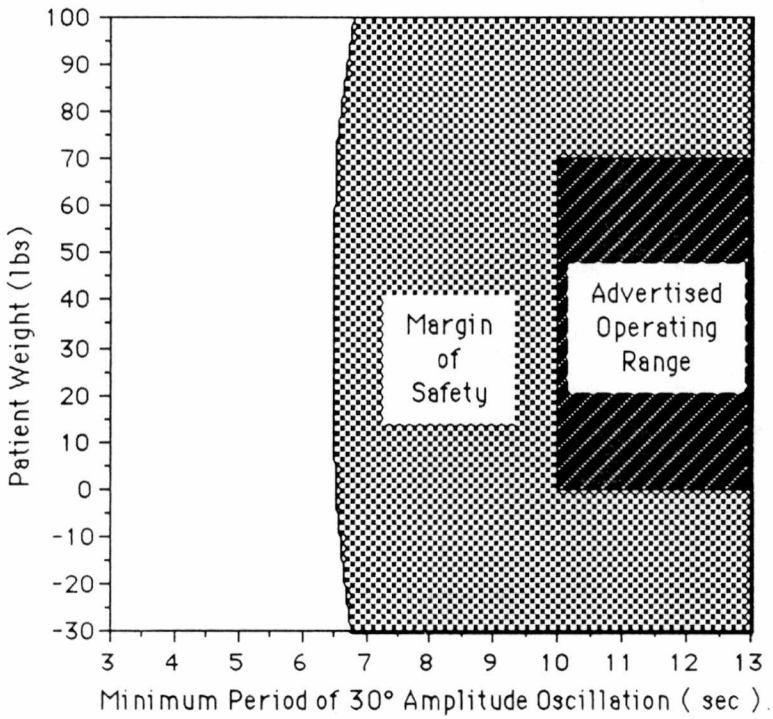


Figure 3.9 Roll Axis Theoretical Operating Space.

The operating space of the yaw axis is extremely robust. The only torque requirement on the axis is to overcome the inertial torque. The maximum inertial torque of the yaw axis was previously shown to be 43 in lb. The calculated torque supplied by the mechanical drive at maximum RPM is 160 in lb. This available torque results in a factor of safety of almost 4.

3.7.5 Design of a Custom Unit

The requirements on the mechanical drive were somewhat unique. The drive was required to be as lightweight as possible. The gear reduction would also be the main load bearing member of each joint. The reduction needed to be easily mounted in a variety of positions. The worm gear reductions available on the market are quite heavy and somewhat over-built for this application. The units are also very expensive. These facts and the desire to tailor the mechanical drive to the device requirements dictated the design of a custom unit. The worm gear drive could use stock Boston Gear components in a cast aluminum housing. The design of this custom unit was both challenging and rewarding. The prototype gearbox design is presented in section 4.2.

CHAPTER 4

PROTOTYPE DESIGN AND CONSTRUCTION

4.1 Frame and Foundation

4.1.1 Joint Design

Each axis joint was designed to use the minimum number of parts and take maximum advantage of the custom designed worm gear reductions. Each joint utilizes the gearbox as the major stressed member.

In the pitch axis joint the gearbox attaches directly to the right foundation frame. The gearbox is bolted to the outside of the horizontal top member of the foundation. The right side of the pitch frame, with respect to a seated patient, is pinned directly to the output shaft of the gear drive. On the opposite side of the pitch frame, a pillow block bearing is bolted to the top of the left side foundation. The left side of the pitch frame is supported by a short idle shaft which rides within this bearing. The bearing has a spherical outer race that simplifies its alignment with the idle pitch axis shaft. The right and left pitch axis joints are presented in Figure 4.1 and Figure 4.2.

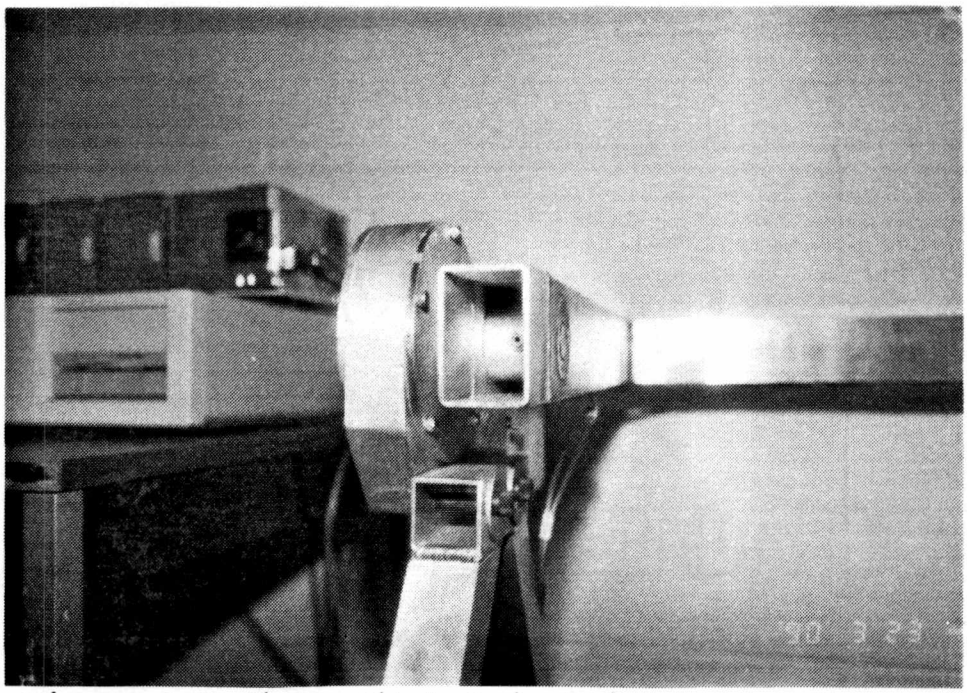


Figure 4.1 Right Pitch Axis Joint.

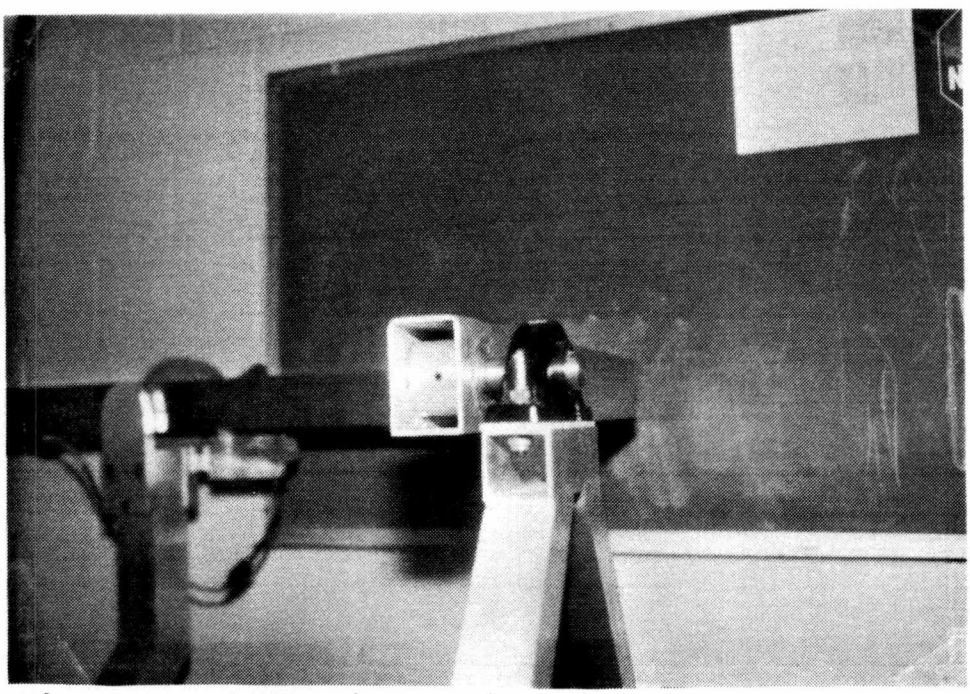


Figure 4.2 Left Pitch Axis Joint.

The pitch joint is mechanically limited from a rotational amplitude greater than 30 degrees. This feature may be seen in Figure 4.1. If a greater rotation is attempted, the right side of the pitch frame will come into contact with the top horizontal member of the right foundation frame. The permissible angle of rotation is controlled by the length of the horizontal frame member.

The roll joint experiences the greatest stress of the device joints due to the cantilevered roll axis. The roll axis joint is presented in Figure 4.3 and Plate 27. The roll gearbox and an opposing bearing bracket form the major stressed members of the joint. The joint experiences a severe bending moment perpendicular to the axis of rotation. The roll gearbox is positioned on one side of the pitch frame backbone and experiences a compressive stress due to this load. The bearing bracket is positioned on the other side of the pitch frame and experiences a tensile stress due to the load. The output shaft of the gearbox passes through and is pinned to the pitch frame backbone. The shaft extends beyond the opposite side of the pitch frame and engages a bearing within the bearing bracket. In a rotation of the roll axis the output shaft and gear of the roll gearbox remain stationary in relation to the pitch frame. The gearbox worm and housing rotate about the stationary gear and shaft. This allows the gearbox to be used in the mounting of the roll frame.

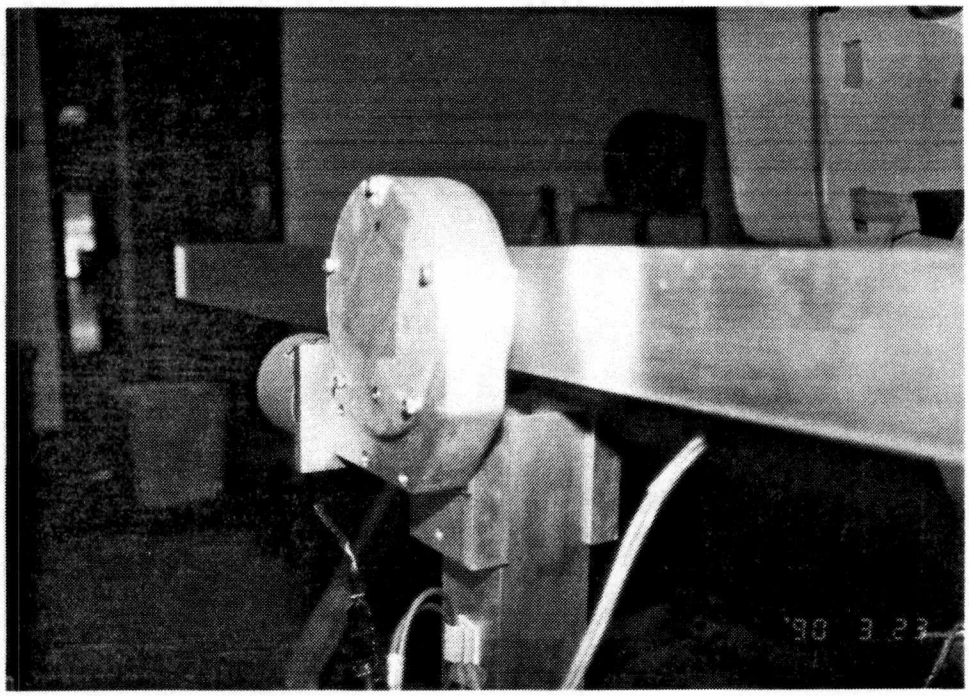


Figure 4.3 Roll Axis Joint.

The top of the roll frame is sandwiched between the roll gearbox on one side and the bearing bracket on the other. The top of the roll frame with mounted gearbox is presented in Figure 4.4. This sandwich construction serves to distribute the stress imposed on the roll axis tubing by the joint. The gearbox is mounted to the roll frame using an adapter plate. The gearbox mounting plate and bearing bracket are both recessed to engage the roll axis tubing positively. Two socket-head cap screws pass through the bearing bracket, roll axis tubing, and gearbox mount. The screws engage the tapped mounting holes of the gearbox housing. These screws serve a multi-function in that they

mount the mechanical drive, mount the bearing bracket, and firmly retain the roll axis within the assembly. A third cap screw is utilized below the first two to increase the rigidity of the sandwich about the axis of the imposed bending load. This screw passes through the bearing mount and roll tubing and engages the gearbox mount via a tapped hole. This lower screw also facilitates assembly of the roll axis joint in that the bearing support and gearbox mount may be mounted to the roll frame prior to the gearbox attachment.

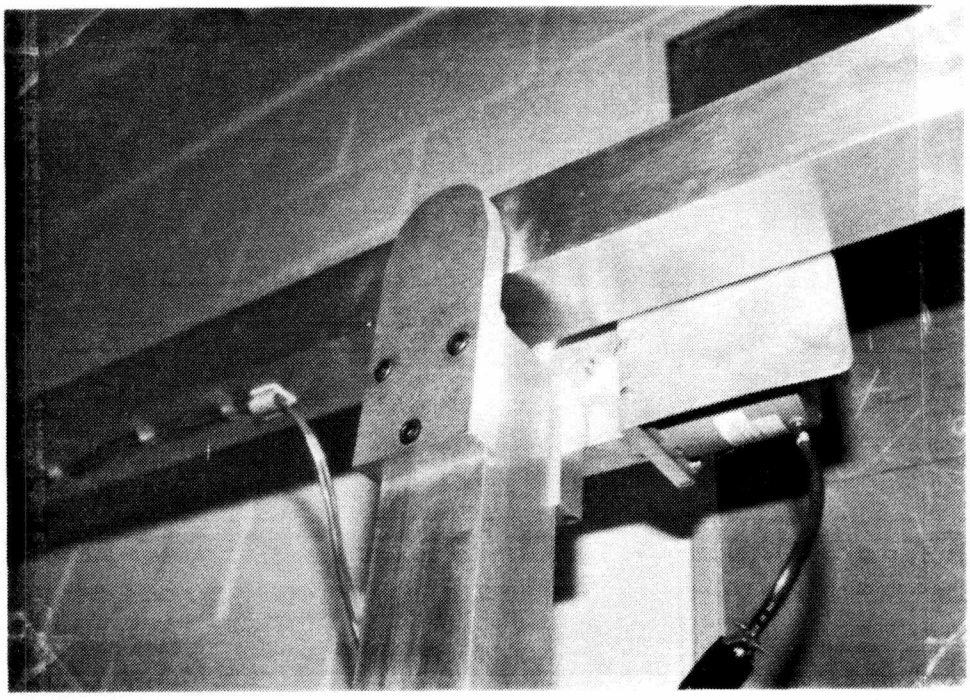


Figure 4.4 Top of Roll Frame.

The roll axis is mechanically limited to a maximum rotational amplitude of 30 degrees. If a greater rotation is attempted, a stepped portion of the bearing bracket will contact the backbone of the pitch frame. The maximum permissible angle of rotation is controlled by the location of the step on the bearing bracket. This feature may be seen in Figure 4.4.

The yaw axis is the last in the kinematic chain. The yaw axis construction is presented in Figure 4.5 and Figure 4.6 respectively. The yaw gearbox bolts to the bottom of the roll frame. Cylindrical aluminum spacers are used between the frame and gearbox to allow clearance for the gearbox end-plate. The output shaft of the gearbox extends upwards through a vertical tube of the roll frame. The shaft passes through a flanged bearing at the top of the frame. This bearing is bolted to a plate welded to the end of the vertical tube. The bearing serves to stabilize the end of the yaw axis shaft. The enclosure pedestal is pinned to the top of the output shaft. In this arrangement the major joint stress is a compressive axial load on the gearbox output shaft. Since continuous rotation of the yaw axis is desired the joint is not mechanically limited.

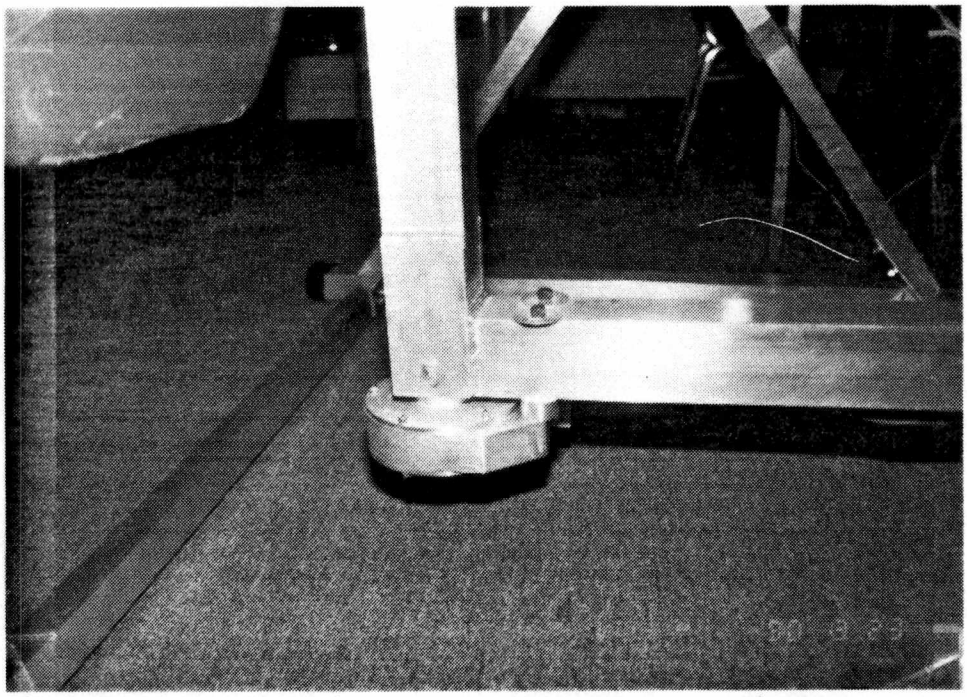


Figure 4.5 Yaw Axis Gearbox to Frame Joint.

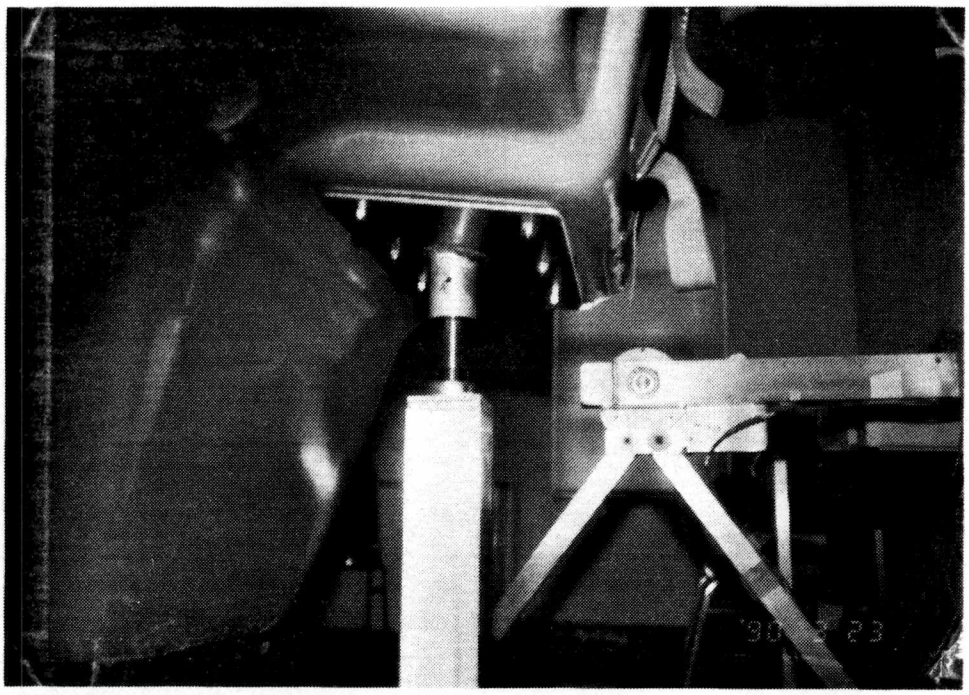


Figure 4.6 Top of Yaw Axis.

4.1.2 Frame Inserts

A high torsional force must be transmitted to the square aluminum frame tubing at both the pitch and roll axes. This torque will be supplied by the output shafts of each gearbox. Cylindrical aluminum frame inserts (Plate 3) are welded within the aluminum tubing to accomplish this. The inserts have an outer diameter of 2.0 inches and a through hole the same size as the gearbox output shafts. The shafts are inserted into the frame inserts and secured by a .25 inch roll pin. The use of a roll pin provides positive torque transfer between the shaft and the frame, simple joint assembly and disassembly, and no backlash in the system due to the axis joints. An additional frame insert is welded in place at the left side of the pitch frame to accommodate the supporting idle shaft. The frame insert stress calculations are presented in Appendix F.

4.1.3 Manufacturing Considerations

The frame members are welded together from individual pieces of straight tubing. Gas tungsten arc welding (TIG) was selected due to its speed, strength and good as-welded bead surface. A square butt weld geometry is used with SWAC power and helium-argon mixture shielding gas. Alternating current power was selected for its combined cleaning action and good joint penetration. A 4043 filler metal was selected over the stronger 5356 filler due to

its lower cracking tendency. The weld bead is ground flush with the surface after welding to further reduce any stress concentrations and possible fatigue failure. The rapid TIG welding process results in minimum time at temperature which means the heat treated aluminum alloy does not completely soften to the annealed state and the size of the weld affected zone is minimized. The as welded strength of 6063-T5 aluminum is given in the Aluminum Association literature as 11,000 psi.

All frame members are planer in layout. This geometry means no special welding jig is required for fabrication. All joints in the square aluminum tubing are perpendicular to the plane of the frame member. No compound angle cuts are required. The pitch, roll and foundation pieces may simply be clamped to a flat welding table during fabrication. This feature enhances the ease with which the device may be manufactured.

The prototype aluminum frames were polished using steel wool. Sections of the aluminum tubing purchased for the prototype were severely oxidized. These sections required excessive polishing time. The surface finish of tubing purchased for future prototypes should be specified and inspected at delivery. After polishing each frame member was given a protective coating of automotive wax.

The tolerances on the individual frame members are not restrictive. All linear tolerances are +/- .125 inch.

The frames are designed so that small misalignments may either be compensated for or will not will not adversely affect the performance of the device. For example, the pitch axis uses an idle shaft that rides within a pillow block bearing on the right side frame. This assembly is shown in Figure 4.1. The shaft locks within the bearing by use of a locking collar and set-screws. The axial position of the shaft within the bearing is not critical. When the pitch axis is mounted to the foundation, the shaft is inserted within the bearing. After final mounting of the pitch gearbox, the locking collar is secured. In this manner, the right side of the pitch axis controls the axial position, the position of the left side is allowed to float as needed. This feature makes the tolerance for the throat opening of both the pitch axis and mating foundation non-critical.

4.2 Gearboxes

4.2.1 Design Overview

The worm gear reductions utilize stock Boston Gear components operating in a cast aluminum housing. Plate 26 is an overlay drawing of the gearbox assembly. The center portion of the housing contains the worm, worm shaft, bearings, oil seal and motor to gearbox coupling. These components mount within a bored hole in the housing side.

A motor mounting plate bolts to the side of the housing over the bored hole. The worm and bearings are controlled from any axial movement by means of steel spacers placed between the worm, bearings and motor plate. The assembly is axially pre-loaded by the compressive force of the motor plate bolts. This pre-loading ensures no backlash within the gearbox due to translation of the worm within the housing.

An aluminum end-plate bolts to either side of the center housing. Each end-plate houses an output shaft bearing. The worm gear is positioned between the two end-plates within the center housing.The worm gear is fixed to the output shaft by a roll pin passing through the gear hub. The wheel is restrained from any axial movement by direct contact between it and the inner races of each end-plate bearing. The gearbox assembly is mounted to the device frames using two threaded holes positioned at the base of the center housing.

4.2.2 Shafting Selection

The worm shaft (Plate 19) is identical in each of the axes gearboxes. The shaft is .5 inches in diameter to match the stock bore of the Boston Gear worms. The worm is torsionally restrained on the shaft by a .125 inch square key. The output shaft size of each gearbox was chosen considering the imposed torsional and shear stress at each

joint. The output shaft of the pitch and roll axes (Plates 20,21) is 1.0 inch in diameter. The output shaft of the yaw axis (Plate 22) is .75 inch in diameter. All shafts are fabricated from SAE #1018 steel drill rod. The drill rod was selected due to its high strength (90 ksi tensile yield) and close diametrical tolerance (+/- .0005 inch). The shafting stress calculations are presented in Appendix K.

4.2.3 Bearing Selection

The worm bearings of each gearbox are required to have an inner diameter of .500 inch to match the bore of the worm. The bearings are also required to have an outer diameter greater than the outer diameter of the worm. These dimensions would allow all components to fit within the same bore. Ball bearings having an outer diameter of 1.375 inches and a width of .438 inch were selected.

The output shaft bearings of each gearbox were selected to accommodate the 1.0 and .75 inch diameter output shafts. For the pitch and roll gearboxes, a ball bearing was selected having an inner diameter of 1.0 inch and outer diameter of 2.0 inches. For the yaw gearbox, a ball bearing was selected having an inner diameter of .75 inch and outer diameter of 2.0 inches. Each bearing has a width of .563 inch. The common outer diameter and width of the two bearings allow the use of identical gearbox end-

plates for the pitch, roll and yaw axes. All bearings are Boston Gear 1600 series ground radial ball bearings. Each bearings load rating was compared against the worst case radial and thrust load. The bearing load calculations are presented in Appendix K.

4.2.4 Housing and End-plate Design

The gearbox housing (Plate 12) and end-plates (Plates 14,15) were designed around the gears and bearings that had been selected. In the design of the center housing, the gear and worm were laid out at the appropriate center distances. The worm bearings and coupling were also positioned. The part then took shape around the components it would house. Consideration was given to the mounting of the motor to the gearbox and the mounting of the gearbox to the device frames.

The end-plate diameters were based on the center housing. There is a front and back end-plate design. The front end-plate has the bearing boss facing the interior of the center housing. The back end-plate has the boss facing away from the center housing and a hole for the output shaft. The end-plates are turned in this manner due to the worm gear having a hub on one side and the desire to have the end-plates control the axial position of the gear. The gear face is axially centered within the housing thickness. The front end-plate boss projects into the

housing against the side of the gear without the hub. The gear hub faces the back end-plate. In this manner the inner races of the output shaft bearings contact the gear on both sides, thus restraining it from any axial play. The outer diameter and bearing boss size is the same in both the front and back end-plates. This feature allows the use of a common casting from which either end-plate may be machined.

4.2.5 Mold Design and Fabrication

The housing and end-plates were poured from alloy #535 aluminum. The molds are oversized to account for the shrinkage of the molten aluminum as it cools. The shrinkage rate used was .016 inches / inch. This rate was as recommended by Southern Castings. The dimensions of each part are increased by this magnitude and by an addition amount for critically machined dimensions / surfaces such as the thickness of the center housing. All edges are tapered at least 2 degrees to insure the release of the mold from the compressed sand during the ramming of the part shape. The actual molds were made of honduras mahogany. The end-plate mold was turned on a lathe. The center housing was cut using a numerically controlled mill and 2 degree tapered cutter. The parts are cast utilizing a match-plate backing. To facilitate production two end-plate molds were fabricated. The housing and two end-

plate molds are mounted to the same match-plate. In this manner the castings for one complete gearbox are poured simultaneously. The mahogany gearbox molds are presented in Figure 4.7. A set of machined aluminum gearbox castings are presented in Figure 4.8.



Figure 4.7 Mahogany Gearbox Molds.

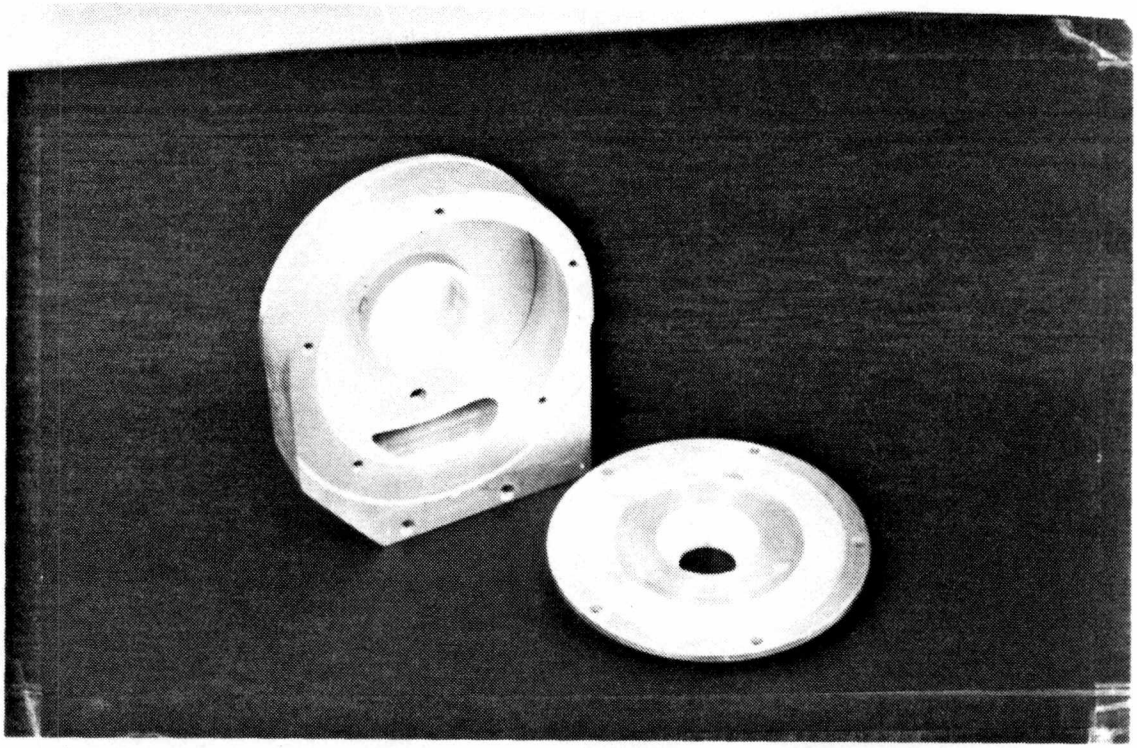


Figure 4.8 Machined Aluminum Gearbox Castings.

4.2.6 Critical Tolerances

The most critical dimension of the gearbox assembly is the center-distance between the worm and gear. This distance governs the internal backlash the gearbox will have. The backlash occurs due to the teeth of the worm not fully engaging the gear teeth. The final tolerance on the center distance is the sum of tolerances associated with the boring of the worm hole and the diameter of the mating gearbox and end-plate bores. In the gearbox center housing the distance between the worm bore centerline and the gear bore centerline is difficult to measure. The reason for

this difficulty is that the bore axes are at 90 degrees to each other. This distance is given a manufacturing tolerance of +.002/-.000 . The diametrical tolerance on the housing gear bore is +.003/-.000 . The diametrical tolerance of the end-plate counter-bore is +.000/-.003 . This gives a centerline tolerance of the completed housing / end-plate assembly of +.005/-.003 .

The tolerance between the bearing inner races and gearbox shafting is also an important consideration. To facilitate assembly, a sliding fit is desired. However, the fit may not be allowed to loose or the shafting may rotate within the bearing bore. The inner bore tolerance of the bearings used is +.0000/-.0005 . The shafting tolerance was set at -.0005/-.0008 . This results in a shaft to bearing fit of +.0008/-.0000 . These tolerances resulted in a tight sliding fit in the prototype. To further increase the reliability of the assembly, Loctite adhesive was applied to each bearing/shafting joint. The Loctite product selected was # 242. This is a general purpose locking compound with a shear strength of 800 psi. Joints made with the compound were found to be quite strong, but were removable when required by a sharp, direct blow. It is recommended that Loctite be used in any future gearbox assemblies.

The last tolerancing consideration of the gearboxes is between the bearing outer race and the end-plate

bearing bores. The outer race of the bearing is toleranced to +.0000/-.0005. The end-plate bores are toleranced to +.002/-.002 . This results in a bearing to bore fit of +.0025/-.0020 . The bearings are desired to have a light press fit within their bores. The aluminum of the end-plate also tends to gall with the steel of the bearing outer race thus increasing the friction of the joint. If an especially loose fit is experienced Loctite could be applied to this joint.

4.2.7 Machining Operations

The basic machining steps of the gearbox center housing and end-plates will now be presented. The gearbox center housing is first chucked on a lathe using the as-cast inside diameter of the gear opening. The face of the housing is then milled to a surface finish of RMS 200 or finer. The inner bore to engage the front end-plate is then cut. The housing is flipped over and chucked using this new trued inner bore as a mounting surface. The opposite side of the housing is then faced until a total thickness of 1.500 inches results. The inner bore to engage the back end-plate is then cut. The housing is then removed from the lathe and positioned on a mill. The side of the casting that will contact the motor mounting plate is milled flat, perpendicular to the trued housing faces and parallel to the as cast surface. The worm bore is then

drilled perpendicular to the milled flat and at the proper center distance. The worm bore is finished using a ream. To complete the part the mounting holes for the end-plates, motor plate and gearbox mounting bolts are drilled and tapped.

The critical machining of each gearbox end-plate is done on a lathe. The end-plate is first chucked using the as cast bearing boss. The outer surface of the end-plate is then cut to the proper dimension. The end-plate is now chucked to this trued outer diameter. The outer face of the end-plate is then given a step to engage the inner bore of each housing. The bearing seat is then cut to the proper depth in relation to the outer step. It is important to note the end-plate is not repositioned on the lathe between the cutting of the step and bearing seat. Machining in this order ensures excellent concentricity between the bearing seat and step. This is important in controlling the tolerance of the center-distance between the worm and gear. The output shaft hole, if applicable, may now be bored. The end-plate is finished on a mill by drilling mounting holes around the periphery of the part.

4.2.8 Shaft Seals

Each gearbox is lubricated with high viscosity gear oil. The gearbox is filled to a level that immerses approximately one half of the worm. A lip type shaft seal

is installed within each worm bore. The seal is positioned outboard of the outer worm bearing and inboard of the motor to worm shaft coupling. The seal O. D. is a press fit within the worm bore. The internal rubber lip is designed to seal against the .5 inch worm shaft. The output shafts of each gearbox do not require a special seal because the oil level is not high enough to contact these joints. Each output shaft bearing uses integral nylon seals and is grease packed for lubrication. These bearings should not require any further lubrication for the service life of the device. The nylon bearing seals also serve to exclude any foreign matter from entering the gearbox.

4.2.9 Coupling Selection

A flexible coupling is required between the worm and motor shafts. The coupling has an overall size restraint in that it is required to fit within the worm bore of the center gearbox housing. A Helical flexible coupling was selected. The coupling engages both the .500 inch worm shaft and the .375 inch motor shaft. Two set-screws lock each shaft within the coupling. The coupling is torsionally rigid and has a maximum torque rating of 28 in lb. Surprisingly, the Helical coupling was also the least expensive of all couplings priced.

4.2.10 Assembly

The assembly of each gear reduction unit must be accomplished in the proper sequence. Refer to Plates 26 and 27. Numbers appearing as (#) are the part numbers of the individual components.

-The worm (22,23), worm key (33), shaft spacers (36), and bearings (30) are assembled on the worm shaft. The bearings are installed on the shaft with a single drop of the Loctite adhesive. The assembly is positioned axially on the shaft such that the inner bearing is flush with the end of the shaft and all other components are driven tightly towards this end. The assembly is then placed aside for a minimum of 20 minutes to allow the locking compound to cure.

-The gear (20,21) is pinned to the output shaft (25,26,27). The gear must be positioned on the shaft such that the gear hub faces the proper end of the shaft. The output shaft bearings (28,29) are pressed into their seats within each end-plate. One end-plate (18) is temporarily mounted onto the central housing using its mounting screws. The gear and remaining end-plate (19) are placed in position but not secured. The axial position of the gear within the center housing is now adjusted by sighting through the worm bore. The position of the gear is altered as needed by placing shims between the gear and bearing on the appropriate side. After the position of any required

shims is determined the end-plates and gear are removed from the housing.

-The shaft seal (35) is placed over the end of the worm shaft with the lip of the seal towards the worm. The seal must be carefully rotated on the shaft during installation to insure the seal lip is not damaged. The flexible coupling (34) is then installed on the end of the shaft.

-The entire worm assembly is now inserted into the worm bore of the gearbox housing. Loctite may be applied between the outer race of the bearing and worm bore if required. The assembly is driven into position using the spacer (37) between the outer bearing and motor mounting plate (31). Ensure that the coupling set-screw access hole in the spacer aligns with the access hole in the gearbox housing. The assembly and motor mounting plate are then secured in position using the two motor plate cap screws. The motor plate screws must only be tightened sufficient to remove all play from the system. The screws must not be torqued to more than 6 in lb.

-The gear may now be positioned within the housing and one end-plate installed. The side of the housing is coated with a thin film of silicone sealant. The end-plate is then slid over the output shaft and into position adjacent to the gear. A drop of Loctite is applied to the shaft at the location of the end-plate bearing. The gear/shaft assembly is now inserted within the housing and the end-

plate mated to the housing side. The gear must be in position engaging the worm before the end-plate bolts are tightened.

-The gear lubricant is now added to the interior of the housing. The second end-plate is then installed over the output shaft with silicone sealant on the housing face and Loctite at the end-plate bearing. This completes the assembly of the gearbox.

The stepping motor (32,33) may now be mounted to the gearbox. A round hub on the front of the motor engages a recess in the motor mounting plate. This ensures concentricity between the motor and worm shaft. The motor is attached to the plate using four screws. The set screws of the coupling may then be tightened through the access hole in the side of gearbox housing. This completes the assembly of an axis mechanical drive. The time required for the complete assembly of a mechanical drive, including the gearbox assembly is approximately 30 minutes.

4.3 Enclosure

4.3.1 Mold Design

The enclosure mold (Plate 5) was designed to tolerate the stresses generated during the vacuum forming process. Approximately 25 inches of mercury vacuum would be applied to the interior of the mold to draw the heated plastic

sheet into the shaped part cavity. The high vacuum is necessary to draw the softened sheet plastic quickly into the cavity before the plastic has time to cool and harden. With the large surface areas involved, this internal vacuum creates tremendous compressive forces within the mold.

Formall Plastic of Knoxville provided the general mold design guidelines based on their extensive vacuum forming experience. The most important guidelines are as follows. In order to allow adequate plastic volume to be drawn into the part cavity, the throat opening of the cavity should be 1.5 to 2 times the depth of the cavity. This ensures the plastic sheet will not be stretched too thin when the part is drawn. The mold must also have a positive draft of at least 3 degrees at all locations to allow removal of the part after it has been drawn and cooled. The enclosure shape that had been previously developed satisfied both of these requirements.

Molds of this size are generally constructed using sheet plywood. The molds are internally braced using .75 inch sheet ribs placed on 6 inch centers. The surfaces of the part cavity are formed by at least two layers of .75 inch plywood bonded together and attached to the top surfaces of the ribs. All corners are then given a generous radius using automotive body filler. To complete the mold the surface of the cavity is drilled with .047

inch diameter holes, one per each square inch of surface area.

4.3.2 Mold Construction

The mold was constructed by Mr. Joel Lenoir, a graduate student in mechanical engineering. The mold was made primarily of .75 inch thickness birch plywood. All joints were generously glued and fastened. The construction of the mold is chronicled in Appendix L.

4.3.3 Fiberglass Prototype

The cost of vacuum forming the prototype enclosure was found excessive due to economies of scale associated with the process. Vacuum forming would be cost effective if several parts were to be produced at one time, but was inefficient for the production of a single part. The high cost was mainly due to a large initial set-up charge associated with the vacuum forming equipment. For this reason the prototype enclosure was made of hand-laid fiberglass using the existing mold. The laminate schedule was derived by considering existing fiberglass parts of the same size. The main body of the enclosure was made of 1.5 ounce chop mat sandwiched between one layer of 1 ounce cloth on each side. Additional reinforcement is provided at the foot rest by one layer of 1.5 ounce chop mat and 1 layer of 1 ounce cloth. After curing the part was sanded

smooth and given a durable urethane finish. The fiberglass enclosure prototype is presented in Figure 4.9.

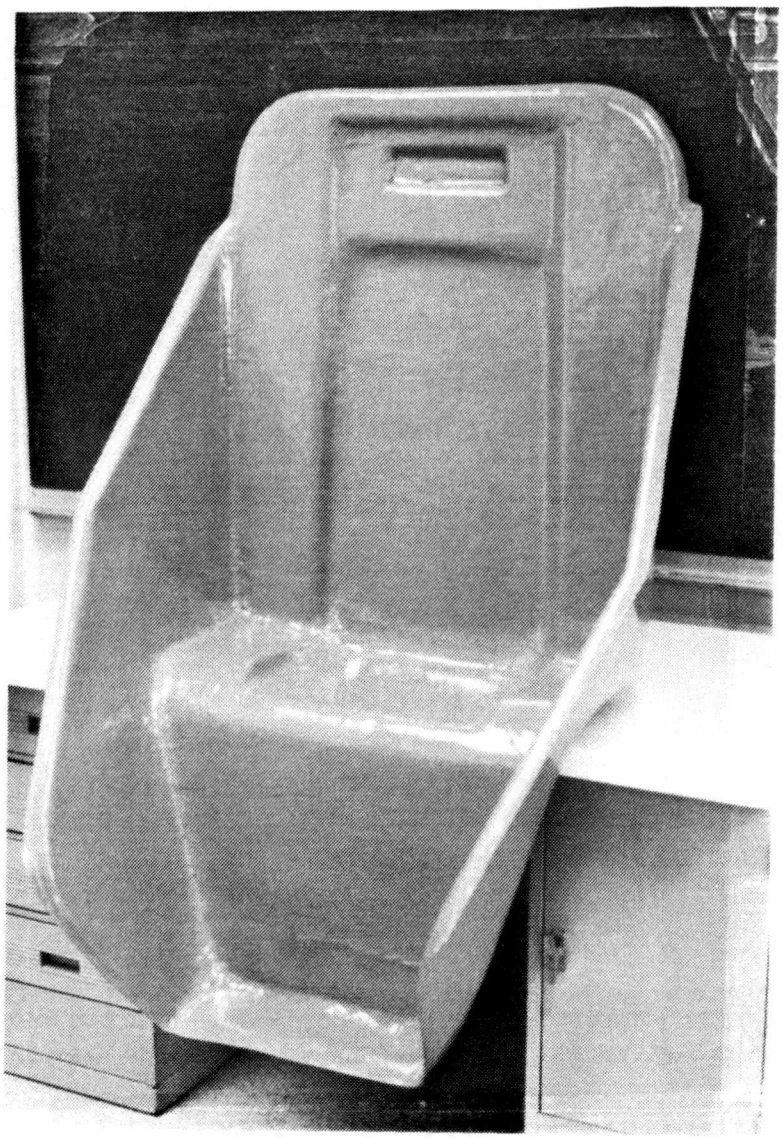


Figure 4.9 Prototype Fiberglass Enclosure.

4.3.4 Pedestal Mount

The enclosure is mounted to the top of the yaw axis output shaft using a pedestal mount. This assembly is shown in Figure 4.6. The pedestal mount (Plate 8) is composed of an aluminum plate with a boss welded to one side. The plate is 12 * 9 inches and is .20 inches thick. The boss is 2 inches in diameter and 3 inches long. A hole is bored thru the boss the same size as the yaw axis shaft. The boss is welded to the plate using a .25 inch bead, full circumference weld. The yaw axis is inserted into the boss and secured with a roll-pin. The enclosure is secured on the pedestal with 6 button head cap screws. A square plate is used within the enclosure to act as a washer between the screws and fiberglass. The internal square plate distributes the stress imposed on the fiberglass enclosure by the clamping action of the cap screws.

4.4 Assembly

The device is initially assembled into component parts prior to shipment to the customer facility. Each mechanical drive is assembled as described in section 4.2.10 . The output shaft of the pitch axis mechanical drive (25) is inserted into right side insert of the pitch frame (1) and secured with a roll pin (39). The output shaft of the roll axis mechanical drive (26) is inserted

into the center insert of the pitch frame and secured with a roll pin. The pitch axis idle shaft (12) is inserted into the left side insert of the pitch frame and secured with a roll pin. These steps complete the preassembly of the pitch axis.

The roll axis bearing (25) is pressed into the bearing support plate (11). The bearing support and mechanical drive mounting plate (10) of the roll axis are assembled onto either side of the roll frame (2) and secured with their lower mounting bolt. The yaw axis mechanical drive is bolted in position at the bottom of the roll frame. A flanged bearing (15) is placed over the yaw axis output shaft and bolted to the top of the roll frame. The enclosure pedestal mount (9) is then pinned to the top of the yaw axis output shaft. The pillow block bearing (14) which supports the idle side of the pitch axis is bolted to the top of the left foundation frame. The foundation brackets (8) are bolted to each end of the two foundation cross-braces (7). These steps complete the preassembly of the device components.

The major device components as delivered to the customer site are the pitch frame assembly, roll frame assembly, left and right foundation sides and cross-braces, and enclosure. These components are presented in Figure 4.10.

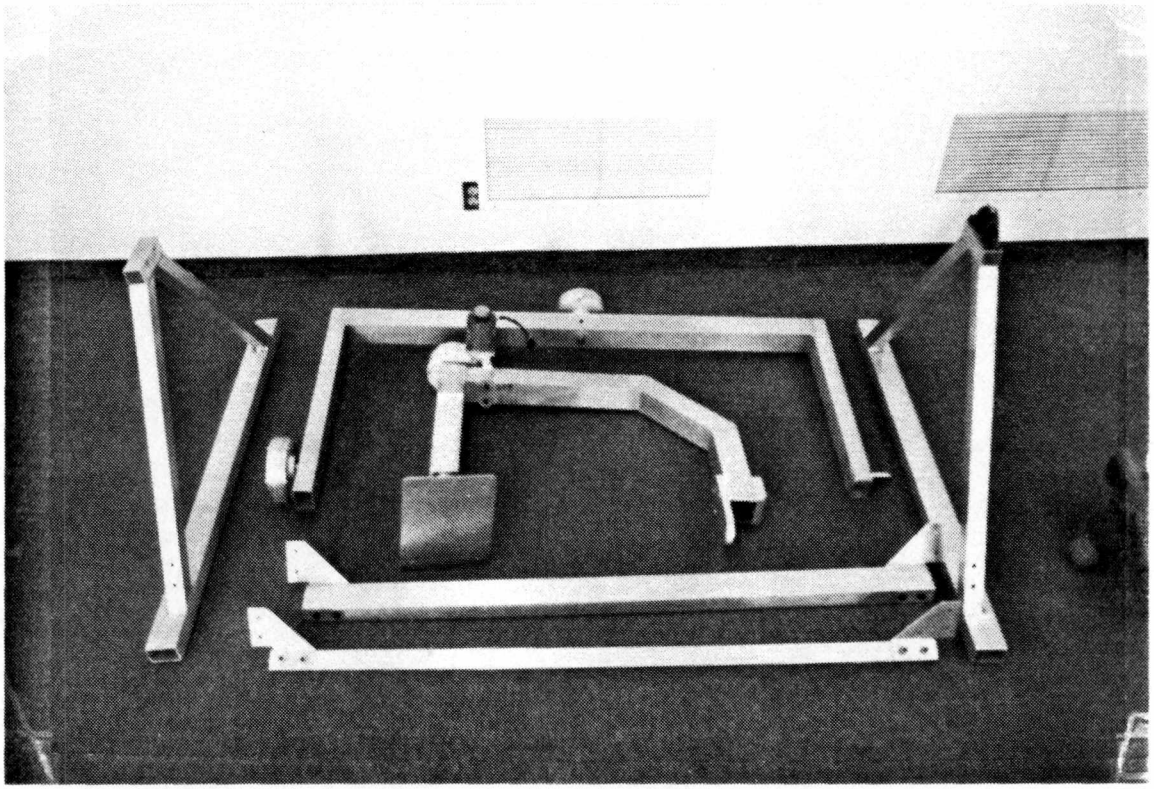


Figure 4.10 Major Device Components.

The device final assembly is accomplished at the customer facility. The foundation is assembled by bolting the left and right foundation sides (3,4) to the frame cross-members (7). Prior to final tightening of the fasteners each foundation side is adjusted to be vertical using a spirit level. The assembled foundation is presented in Figure 4.11.

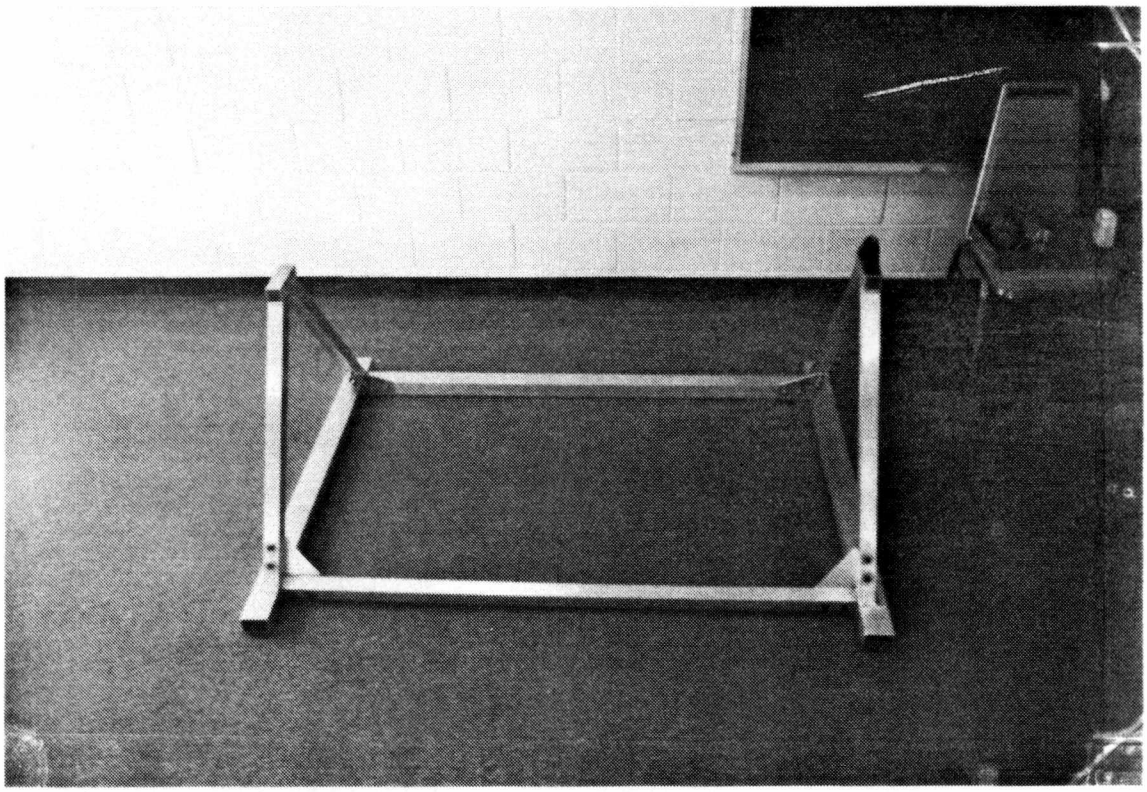


Figure 4.11 Foundation Assembly.

The pitch frame assembly is now positioned on the floor with the roll axis output shaft facing upward. The roll frame assembly is position over the pitch frame. The roll assembly is then lowered onto the pitch frame such that the roll shaft is inserted into its bearing and the roll gearbox aligns with its mounting holes in the roll frame. The assembly will independently remain in this position while the two joint cap screws are installed. The screws pass thru the bearing bracket, roll frame, and gearbox mount and engage two threaded holes within the gearbox housing. The roll joint screws are now given their final assembly torque.

The pitch / roll assembly is now installed onto the foundation. The assembly is held in position between the foundation sides. The pitch axis idle shaft is inserted into the pillow block bearing on the foundation left side. The bearing will now support the left side of the pitch/roll assembly. The right side of the assembly is now translated/rotated to align a mounting hole of the pitch gearbox with its corresponding hole in the right side foundation frame. A cap screw is then installed thru the foundation frame and engages the threaded hole in the gearbox housing. The pitch/roll assembly may now be rotated to allow installation of the second frame to gearbox screw. Both screws are now given their final assembly torque. The foundation with mounted pitch and roll assemblies is presented in Figure 4.12.

The enclosure may now be mounted to the top of the yaw pedestal. A foam gasket is positioned atop the pedestal to act as bedding material. The enclosure may now be placed atop the pedestal and will independently remain there until the joint fasteners may be installed. A square plate of thin metal is placed inside the enclosure to act as a washer for the six mounting screws. Each button-head cap screw is then inserted down thru the plate, enclosure, and pedestal. The screws are secured using nylon insert lock-nuts installed below the pedestal. One of the three Tumble Forms chairs may now be strapped within the

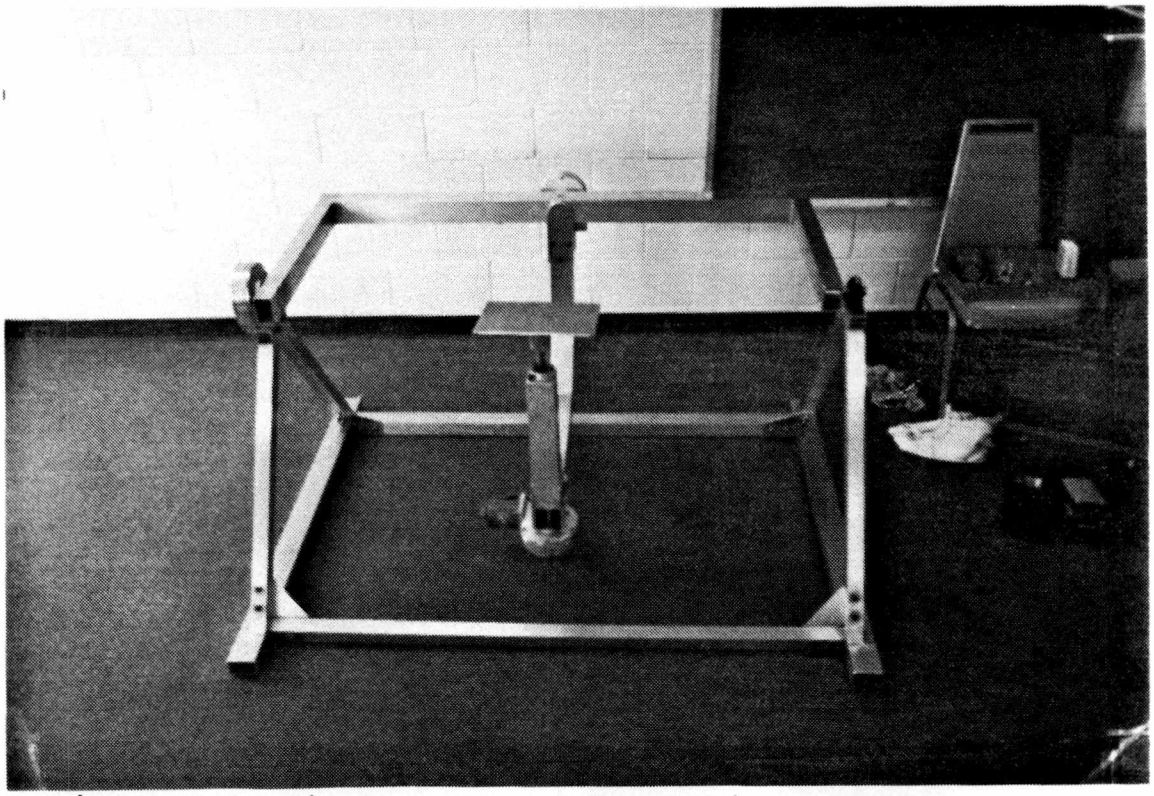


Figure 4.12 Pitch, Roll and Foundation Assembly.

enclosure. This completes the mechanical assembly of the device. The assembled device is presented in Figure 4.13.

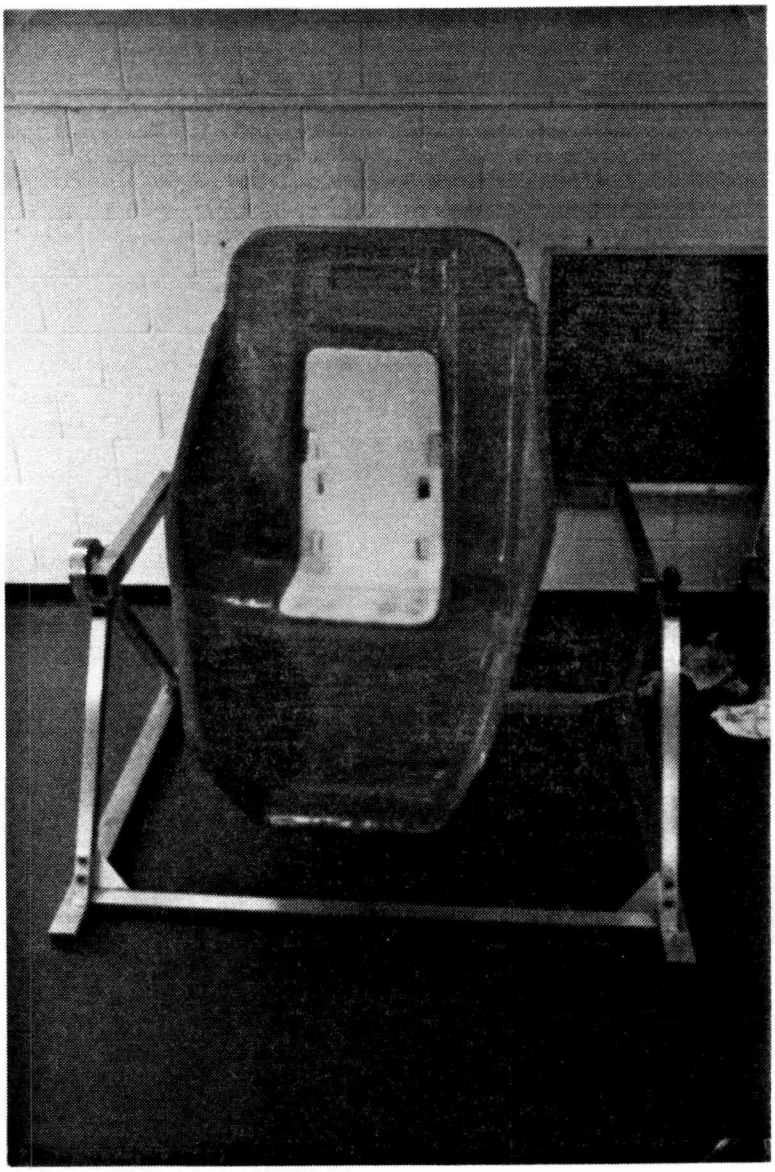


Figure 4.13 Assembled Prototype.

CHAPTER 5

PERFORMANCE TESTING AND EVALUATION

5.1 Gearbox and Chopper Initial Test

Each mechanical drive was assembled and pretested using the IMS choppers pulsed by an oscilloscope. To test the effectiveness of the worm shaft oil seal and housing to end-plate seals an extended-operation test was performed. The outer housing of the mechanical drive was carefully de-greased. The drive was then continuously operated for approximately 8 hours. A close inspection of all joints showed no signs of lubricant leakage. No accurate method could be found for measuring the torque on the output shafts. The mechanical drive performance would be evaluated using the assembled device. Severe natural frequency bands were observed at very low speeds. It was expected that the natural frequencies would change position and magnitude once the mechanical drive was fastened to the device structure therefore they were not addressed at this time.

5.2 Natural Frequency Evaluation and Resolution

After the assembly of the mechanical drives to the device structure the natural frequencies were not dampened to an acceptable level. A severe natural frequency was

found to be centered at approximately 100 pulses per second. At or near this frequency the stepping motor produced no usable torque and was actually observed to oscillate from one direction of rotation to another. A second natural frequency was observed at approximately 200 pulses per second. The natural frequency bands of all three axes were the same as could best be determined. The stiffness of the structure at each mechanical drive however varies. The pitch drive is attached to the relatively solid foundation frame while the yaw drive is cantilevered on the end of the roll frame. This difference in structure implies the natural frequency band is strongly a function of the stepping motor and gearbox assembly and not strongly related to the stiffness of the structure to which the mechanical drives are attached.

It is important to note the driving force behind the stepping motor vibration. The forcing function is not due to any rotating imbalance in the motor assembly, but rather the actual pulsing of the individual motor steps. Depending on the order with which the windings of the motor are pulsed the shaft may take a full, half or quarter step per pulse. Therefore the natural frequency of vibration may not be expressed in terms of RPM at the motor shaft, but must be expressed in terms of pulses per second at the motor windings.

Various methods were attempted to alleviate the natural frequencies, the chosen solution is now presented. The first frequency occurs very low in the speed range of the axis. Due to the sinusoidal motion profile the axes are undergoing their highest accelerations at this location. This high acceleration means the pulsing frequency will quickly pass through the problem range. This frequency is completely avoided by a dead band about the problem range. The stepping motors are not pulsed at a rate between approximately 22 and 220 pulses per second. This results in a step change in the velocity profile. However, due to the low speed involved and the narrow width of the dead band this discrete change in velocity is undetectable.

The second natural frequency is avoided by a full step, half step manipulation. At approximately 138 pulses per second the motor is pulsed in half step at a rate of 275 pulses per second. The motor continues to be pulsed in half step through the velocity profile until the rate of 550 pulses per second is reached. At this point the motor is returned to full step and the pulsing rate is reduced to 275 pulses per second. The motor shaft RPM is a smooth curve throughout these manipulations. The dead band and half step / full step range are demonstrated graphically in Figure 5.1.

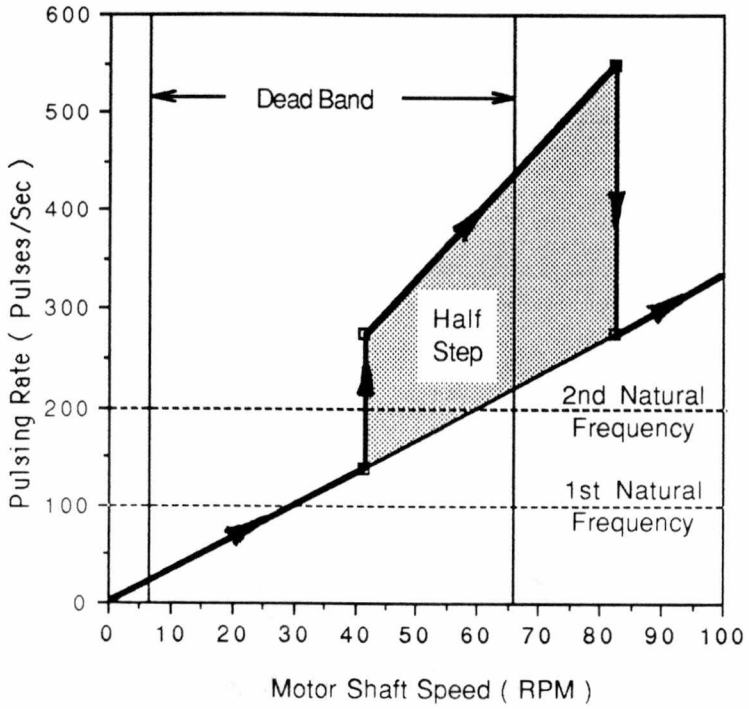


Figure 5.1 Dead Band and Half Step / Full Step Manipulations.

5.3 Operating Space Evaluation

The performance testing of the device was a pleasant surprise. The stepping motors will run at a speed greater than the advertised 300 RPM. If the motor could run at a maximum speed of 300 RPM the pitch and roll axes would execute a 30 degree amplitude motion in a period of 6.25 seconds. The pitch axis when balanced by the weight of a 43 lb child will execute a 30 degree amplitude motion in 5 seconds. The roll axis will execute the same motion in a

period of 4 seconds. These periods compute to a motor RPM of 377 and 471 respectively. The operation of the device at these speeds has been fully verified. With the linear torque speed assumption this maximum RPM has the effect of increasing the available torque throughout the speed range, with the greatest increase in the upper speed range.

The performance of each device axis was tested using actual patients at the Childrens Rehabilitation Center. The performance was also tested using weights to simulate patient weights that were not available. The theoretical and as-tested operating spaces of the pitch axis are presented in Figure 5.2 and Figure 5.3 respectively. The theoretical and as-tested operating spaces of the roll axis are presented in Figure 5.4 and Figure 5.5 respectively. For each patient weight, the minimum period in which the device could repeatably execute a 30 degree amplitude oscillation was found. The yaw axis was capable of a 3 second period, 30 degree amplitude oscillation when laden with a 43 lb child. When tested with no child, the yaw axis exhibited the same performance capability.

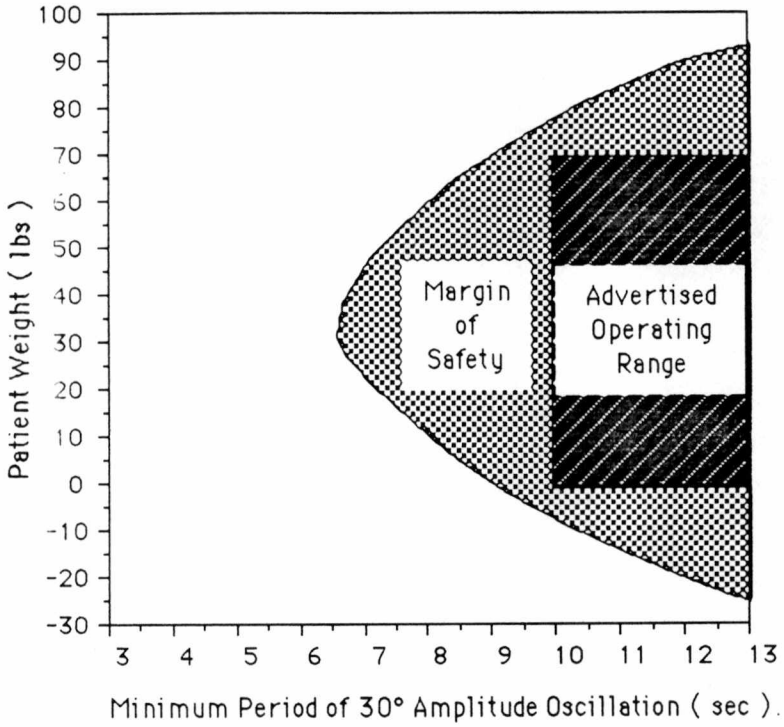


Figure 5.2 Theoretical Pitch Axis Operating Space.

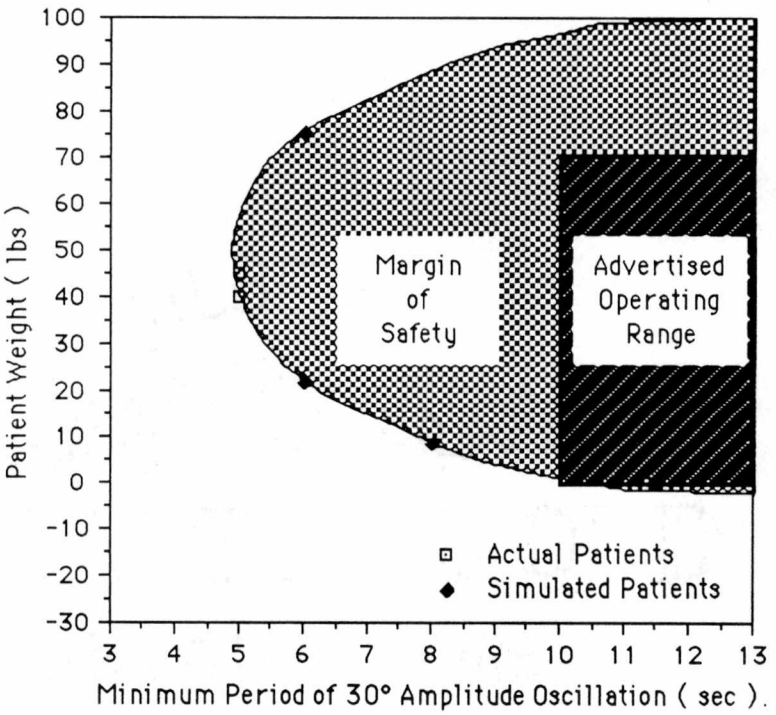


Figure 5.3 Prototype Pitch Axis Operating Space.

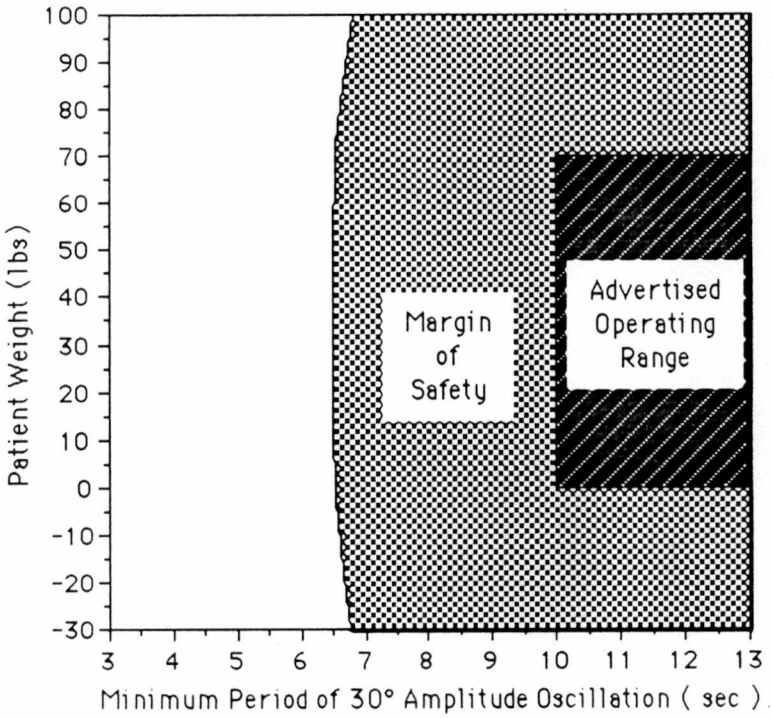


Figure 5.4 Theoretical Roll Axis Operating Space.

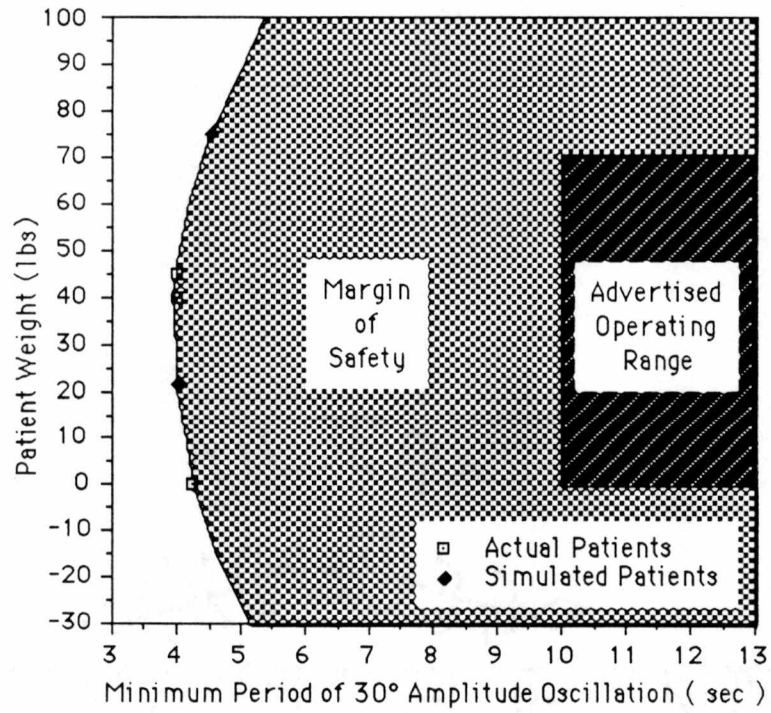


Figure 5.5 Prototype Roll Axis Operating Space.

The operating space of the pitch axis does not fully cover the advertised operating range of the axis. The operating space intersects the advertised operating range when the device is operated with no child. This intersection is due to the location of the patient within the structure. The pitch axis is balanced for a somewhat heavier patient than intended. In order to center the operating space, the location of the yaw axis should be moved farther from the pitch axis. In the prototype, this dimension is 8 inches.

The prototype operation has been satisfactory however even with this result. When a motion is to be executed with no patient, the period of the pitch axis must be 12 seconds or more. The prototype operating space is better suited to the weights of patients used in the research. The minimum possible weight of an actual patient is approximately 15 lbs. The device will easily meet the advertised performance specifications for this weight.

5.4 Failure Modes

Initially the pitch axis was powered by a 300 oz*in stepping motor as used in the roll and yaw axes. The use of this motor was attempted due to availability and with the tacit understanding that a more powerful 450 oz*in motor could be installed if required. The mounting flanges and shafts of both motors are identical thus allowing the

installation of either unit with no modifications. The 300 oz*in unit's performance was unacceptable, consequently, the more powerful 450 oz*in motor was purchased and installed.

In the initial prototype the compression strength of the aluminum used for the frame inserts was overestimated. The published value is equal to the tensile yield strength of 11 ksi. Calculations done for the roll pin hole within the driven pitch axis insert using this information demonstrated a possible failure condition. After some months of operation at the University with severe loading, the device was disassembled for transportation to Childrens Rehabilitation Facility. At this time a careful inspection was made of the roll pin hole. The hole was slightly enlarged in the direction of the compressive load.

The problem was corrected by the addition of a steel sleeve within the insert at the location of the roll pin hole. The steel sleeve provided a greater contact area for the compressive load to act on the aluminum of the frame insert. It was also noted during disassembly that the steel roll pin had a tendency to gall within the soft aluminum insert. The steel sleeve is permanently secured within the insert by a press fit. Upon assembly and disassembly of the joint the steel roll pin is driven within the steel sleeve and thus the galling problem was

eliminated. In future models to facilitate assembly, all inserts should incorporate the internal steel sleeve.

In the prototype the mounting holes thru the foundation for the pitch axis mechanical drive were drilled .438 inch. Two .375 inch bolts were utilized in the mounting of the drive. Due to the high torsional load on the joint, the bolts were migrating about the limits of their holes. To correct the problem, the pitch gearbox housing was drilled and tapped to accept two .438 inch bolts. These fasteners are diametrically tight within the foundation holes. The prototype has shown no sign of play at this joint since the correction was made. In future models the foundation mounting holes should be drilled .375 inch.

Serious pinch points exist in the prototype. One is formed between the stepper motor housing of the pitch axis and the rotating pitch frame. The other is between the stepper motor housing of the roll axis and the pitch frame. The problems were alleviated to some extent by the addition of guards attached to the motor mounting plate of each drive. These guards prevent the insertion of a finger or hand into the area of danger. The guards, however, do not guarantee that an injury cannot occur at the device joints. They are also quite unattractive. Other pinch points exist between the rotating pitch frame and the foundation, and between the rotating roll frame and pitch

backbone. These are due to mechanically limiting the rotation of the axes by contacting some part of the frame. No way of effectively guarding these locations has been found. The ideal way to eliminate the pinch points is to mechanically limit the axes internally within each gearbox. It may be feasible to limit the possible rotation of the gear within the housing. If future devices are made this possibility should be investigated.

CHAPTER 6

CONCLUSIONS

The device was delivered to Childrens Rehabilitation Center of Knoxville, Tennessee in December of 1989. It has functioned there since that time with no failures of any kind as of this writing. Dr. Zimmerman and Karen Goss appear to be quite pleased with the engineering results of the project. The most complimentary result of the project is the following:

Dr. Zimmerman and Karen Goss had difficulty identifying which of the possible device motions to select for the initial research. In other words, the customer has such a wide variety of motions available to him, he is having difficulty selecting how to start. Dr. James Euler deserves a great deal of the credit for this result. His controlling computer program makes it possible for the therapist to select any motion they desire with ease. The device should satisfy the research requirements for some time to come.

The children involved in the research have been very receptive to the use of the device. After an initial introduction to both Karen Goss and the device, they have been enthusiastic. Each child receives approximately 30 minutes of therapy with the device twice a week. Mrs. Goss

and Mrs. Mary Boyd, an occupational therapist at Childrens Rehabilitation, initially tested each child in the study to benchmark their level of sensory integration. All children in the initial study will receive the same series of selected motions. A control group is being utilized that will receive their usual treatment without the device. In approximately 6 months, both groups of children will be re-evaluated. It is interesting to note that each individual patient evaluation is being taped. The tapes will be reviewed by an independent third party. This independent party will not know which children received treatment with the device and which were in the control group. This method of evaluation eliminates any possible bias in the conclusions drawn.

CHAPTER 7

RECOMMENDATIONS

The performance of the device can be increased by the counterbalancing of the pitch axis. The magnitude of the structural moment of the prototype about the pitch axis is 775 in*lbs. If a 16 lb weight were placed 1 foot from the pitch axis the structural moment will be reduced to 583 in*lbs. These values represent a 25% reduction in the required torque. The position of the yaw axis can then be moved closer to the centerline of the pitch axis. As noted in Appendix B, the magnitude of the dilation torque is directly proportional to the magnitude of the structural torque. Since the dilation torque is the main load on the pitch axis, it is reasonable that the operating space of the pitch axis will increase by almost 25%. This increase in the operating space means the weight of patient that may now be rotated with a given amplitude and period could be increased by the same percentage. This change would add more of a performance factor of safety to the pitch axis. In view of the robust operating spaces of the roll and yaw axes this would bring the performance of the pitch axis more on par with the other axes.

The counterbalancing weight can be incorporated into the device by an extension of the pitch frame. The side

arms of the frame may be extended approximately 12 inches out from the centerline of the axis. The counterbalancing weight would then be placed within the aluminum tubing. Calculations show the volume of steel required for a 10 lb weight would fit within the dimensions of the aluminum tube used. If both sides of the pitch frame incorporated this weight, the required moment would be produced.

The major disadvantage of counterbalancing is that it may reduce the aesthetic appeal of the device. The extended pitch arms might appear out of place. However, when the prototype is viewed from the side, the enclosure assembly appears to be off in space in that its centerline is 8 inches in front of the pitch axis. The extension of the pitch arms would lesson this impression.

The foundation cross braces and corner brackets may be improved. It may be possible to eliminate the corner bracket entirely. The cross brace tubing is 3 inches by 2 inches with a .125 inch wall thickness. This thickness means the inner dimensions of the tubing is 1.75 by 2.75 inches. The foundation tubing to which the corner brackets fasten is 1.75 inches square. The two 2 inch sides of the cross bracing tubing can be removed at each end of the brace. This modification is shown in Figure 7.1. To remove the sides would require only 4 straight saw cuts on each end of the tube. The foundation frame would then be inserted into the resulting opening and secured with two

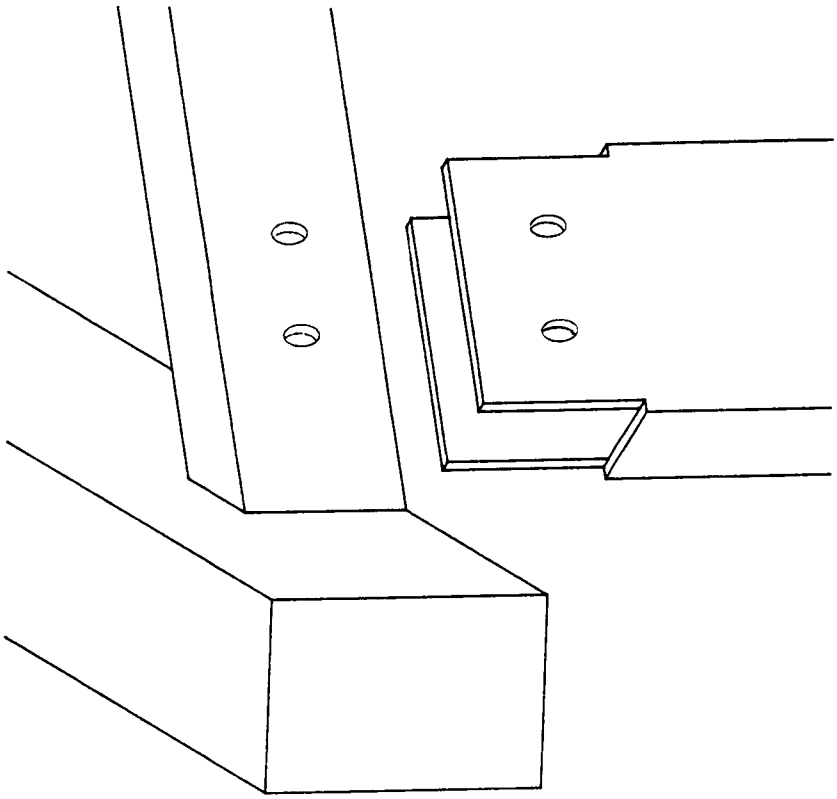


Figure 7.1 Foundation Cross-brace Modification

fasteners. This modification would mean the elimination of four corner brackets and 8 fasteners from each device. The resulting foundation will also be more laterally rigid in that effectively the new design will have 8 braces instead of the 4 used now. The design will be much more torsionally rigid since the foundation member is gripped on both sides by the cross brace. The prototype design may be more aesthetically appealing in that the cross braces now lay flat against the floor. The proposed modification would mean the braces would have their long dimension angled upward.

The mounting point of the pitch axis mechanical drive to the foundation frame may require additional reinforcement. The foundation tubing receives a compressive load from the joint fasteners and a high torsional load from the pitch axis. The fasteners transmit the torsional load from the pitch axis to the foundation tubing as a compressive load on the inside diameter of their mounting holes. As stated in section 5.4 the compressive strength of the aluminum used is equal to the tensile yield strength of 11 ksi. Due to the small wall thickness of the aluminum tube, the fastener to tube contact stress at this location is high. Frame inserts could be welded within the foundation tubing to increase the contact area of the fasteners. The two fasteners which attach the mechanical drive would pass through the inserts. It is not certain that this change is required. The prototype was modified at this location due to an unrelated error. As of this writing, the prototype has not developed any problems with this joint.

By inspection of the device operating space curves, the stepper motors appear to be operating very high in their RPM ranges during the minimum period motions. The performance of the device may be improved by decreasing the gear reduction of the mechanical drives. This change would allow the motors to operate at a lower RPM during any device motion. With the torque speed characteristics

of the stepping motor this could result in more available torque at the axes. This decision cannot be made, however, until a true torque speed curve for the motor and driver used is obtained.

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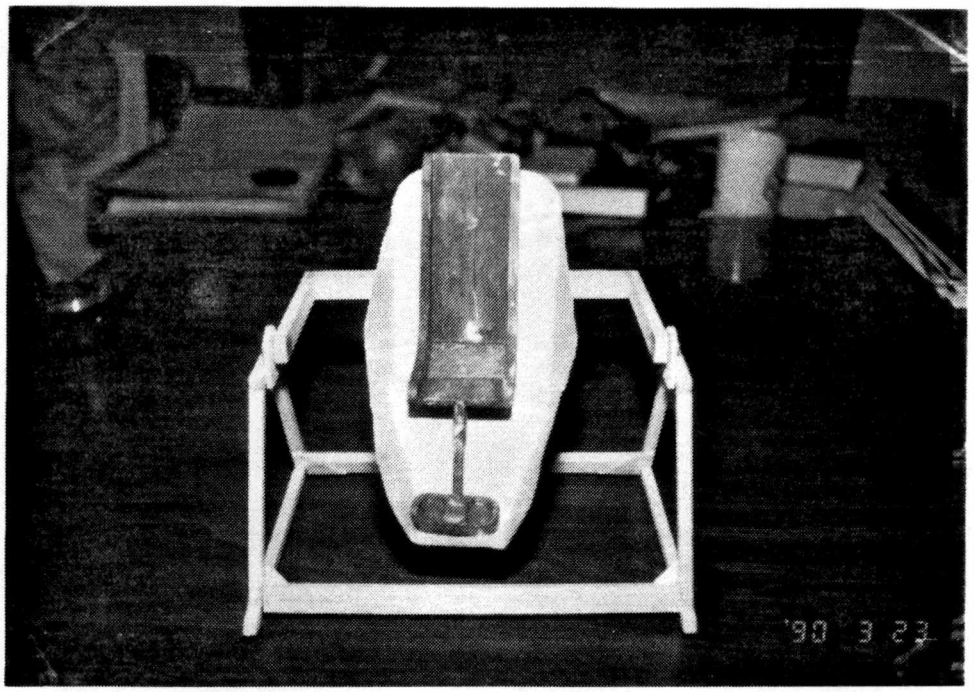
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APPENDICES

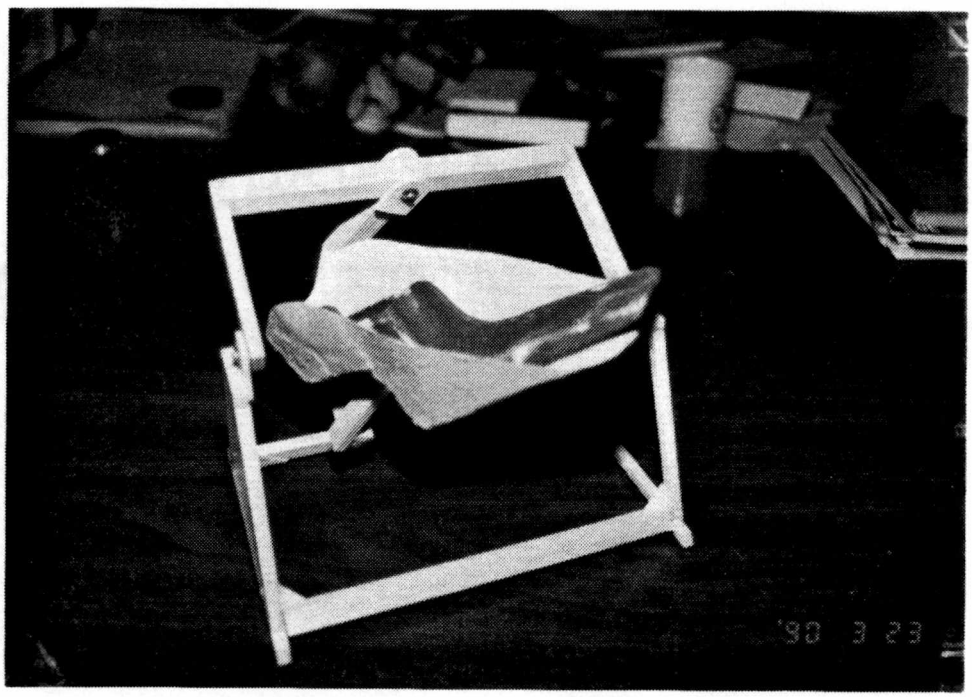
APPENDIX A

Prototype Motion Photos

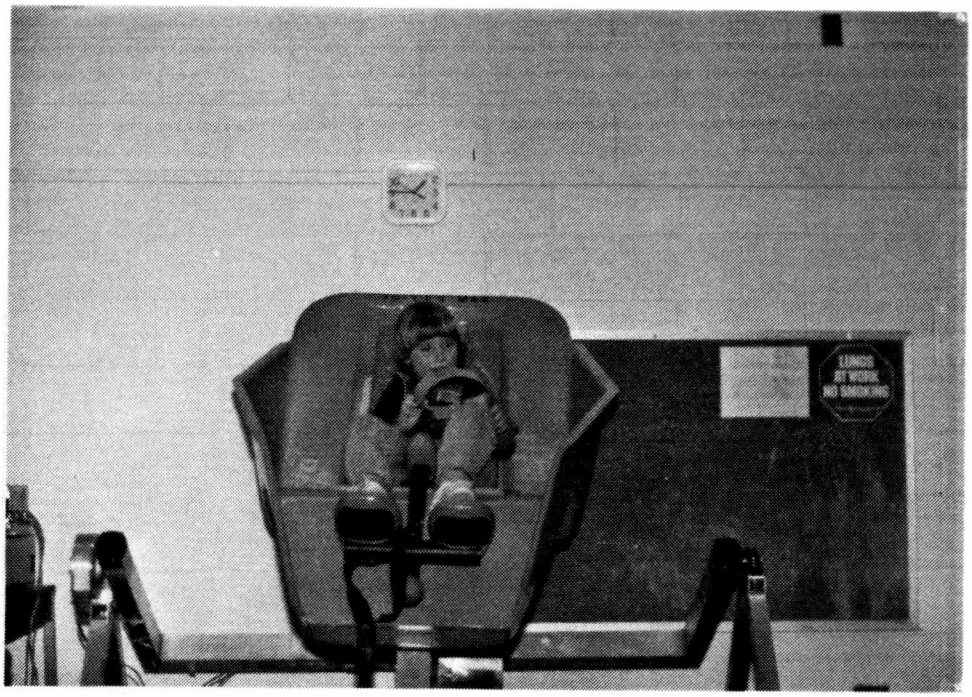
- Model of prototype built to discuss possible device motions with Dr. Zimmerman and therapists.

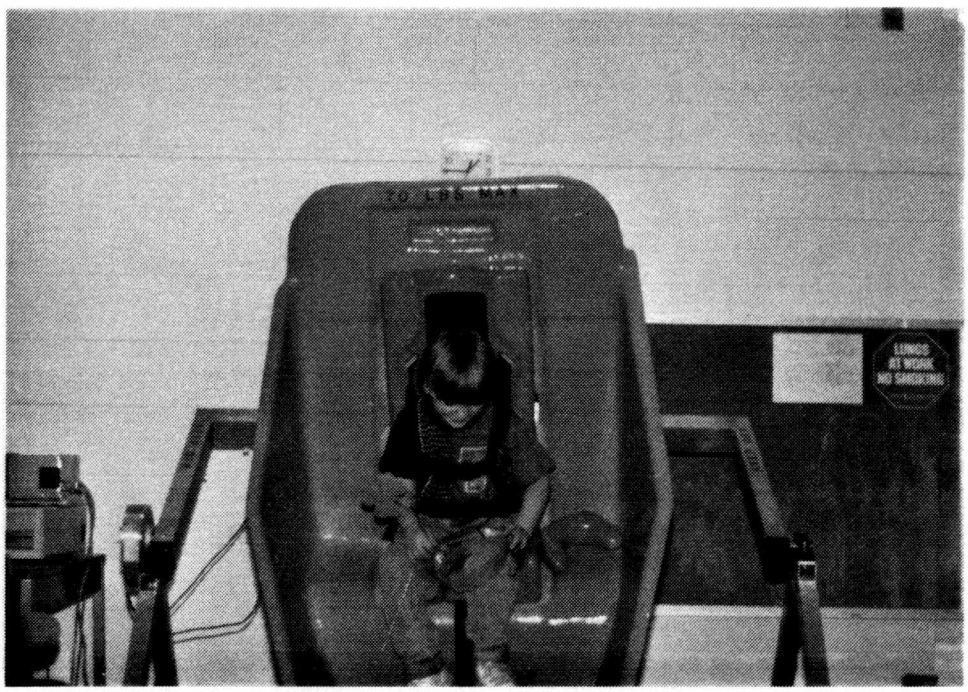


- Model executing a simultaneous pitch, roll and yaw rotation.

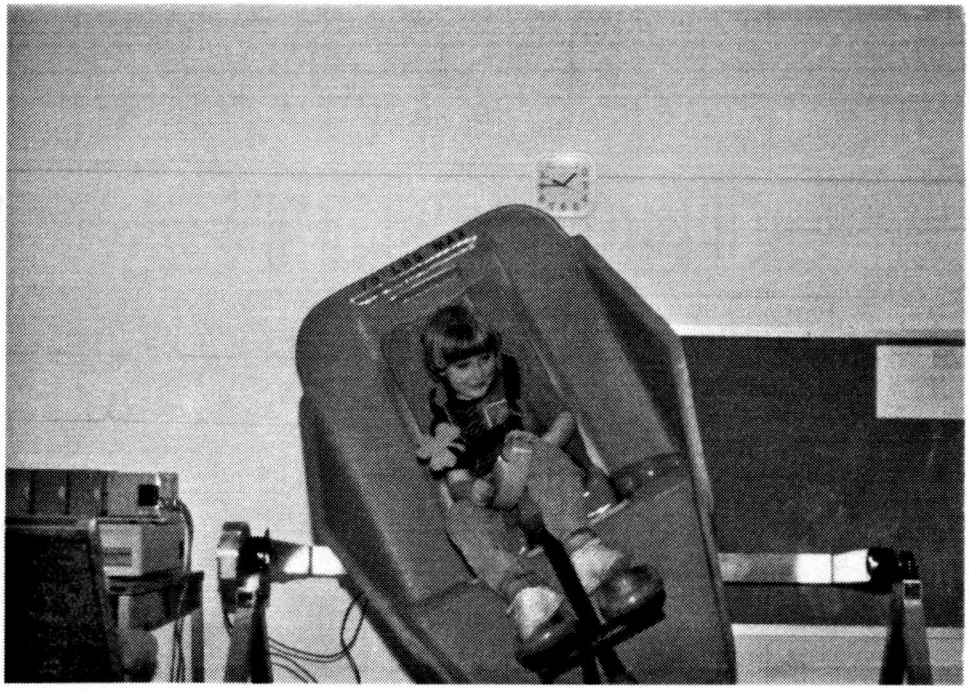


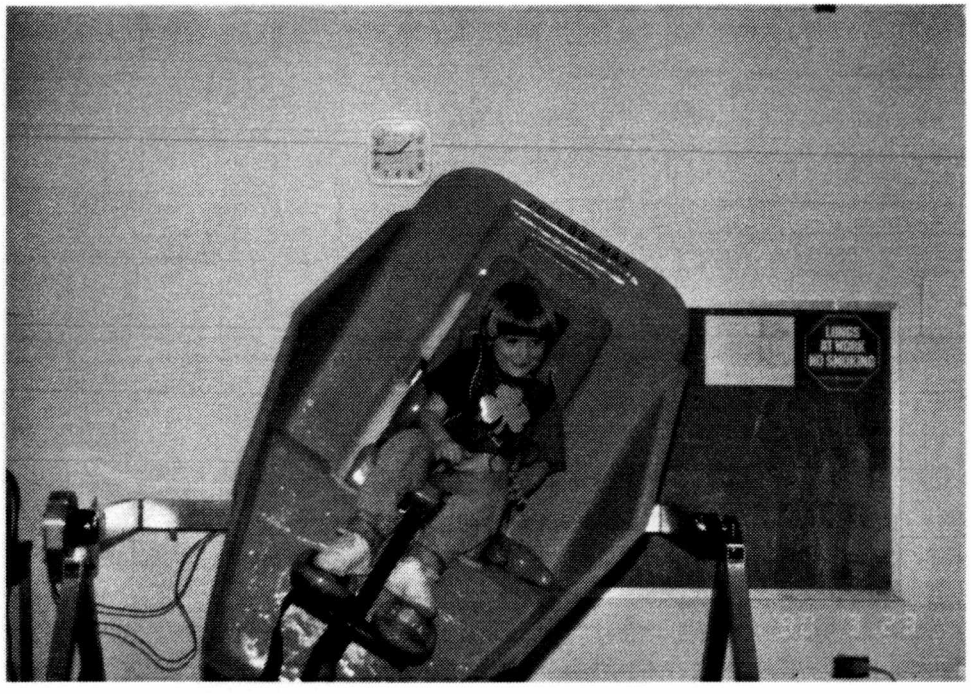
- Pitch axis rotation with a 30 degree amplitude.



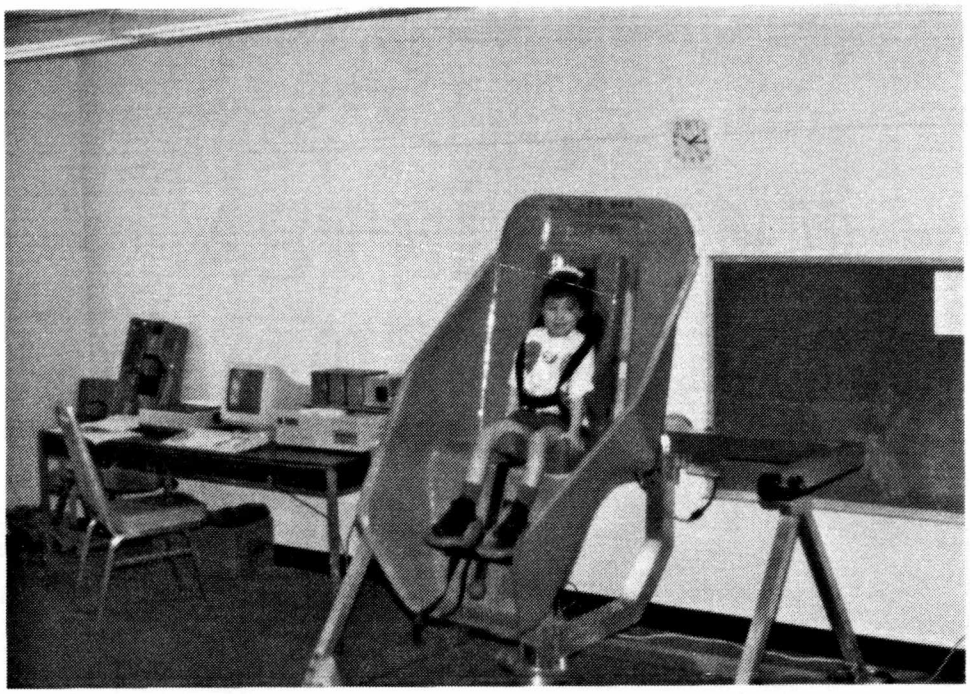


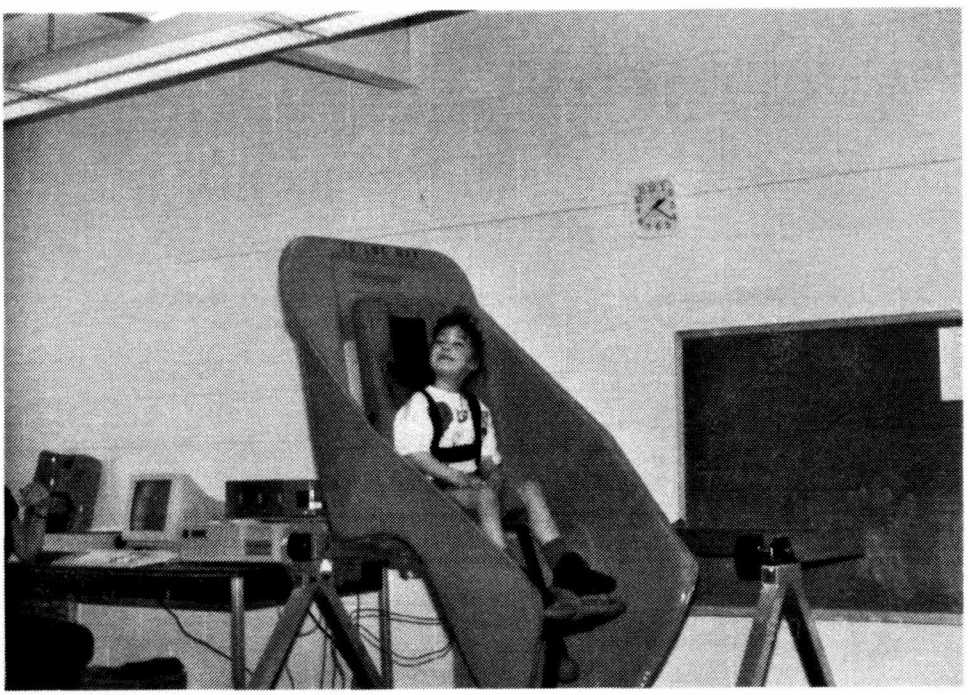
- Roll axis rotation with a 30 degree amplitude.



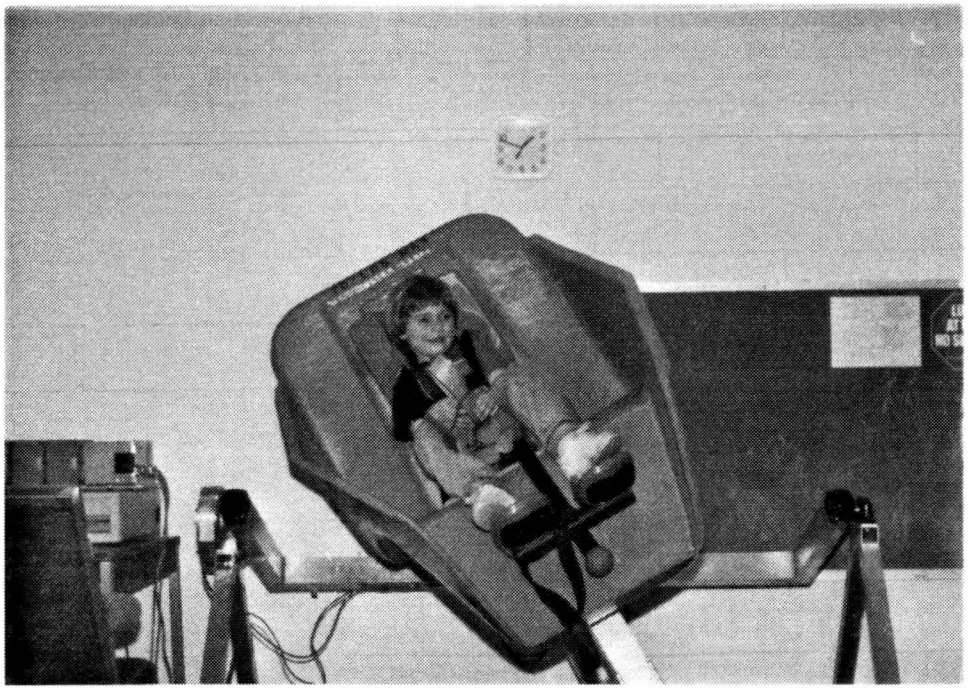


- Yaw axis rotation 360 degree rotation, shown initially at 0 degrees and at approximately 90 degrees.



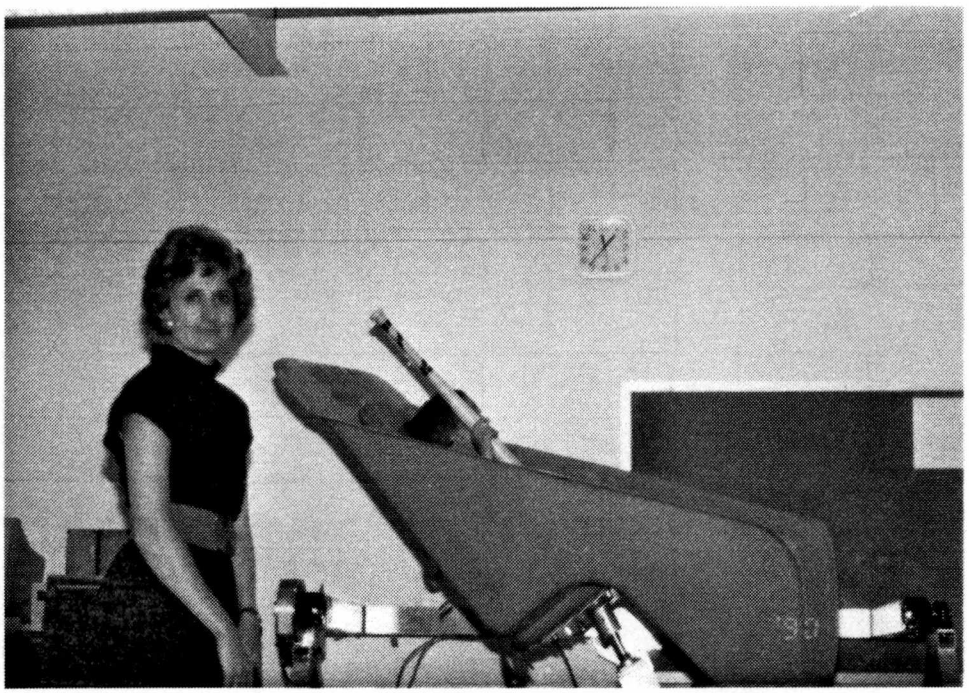


- Pitch and Roll combination motion. This is referred to as a diagonal motion.





- Pitch, Roll, and Yaw combination motion. This is referred to as a rotary motion.





APPENDIX B

Center of Gravity Dilation Investigation

In the following analysis, we investigate the effect of the moment of the device structure on the dilation torque generated about a device axis. We will estimate a value for the structural moment and calculate the resulting dilation torques. We will then assume a new structural moment and calculate the new dilation torques. The weight of the patient that will perfectly balance the structure will be set at 35 lbs. The dilation torque will be calculated based on a 70 lb patient.

If we assume the weight of the structure that must be counterbalanced is 40 lbs and the average moment arm of this weight is 25 inches, then the magnitude of the structural torque is:

Structural Torque = 25 * 40 = 1000 in lbs

The weight that balances this torque is the sum of the weight of the child, Tumble Forms chair, enclosure, yaw gearbox and drive hardware. We estimate the total weight of the Tumble Forms chair, enclosure, yaw gearbox and drive hardware assembly at 60 lbs. The distance this assembly and the 35 lb child must be placed from the device axis is:

D = 1000 / (60 + 35) = 10.5 inches

This distance is a fixed quantity once the structure is assembled. If a 70 lb child is placed in the device the true center of gravity of the system dilates of the axis centerline by the amount:

Delta = 10.5 - (1000 / (60 + 70)) = 2.81 inches

This dilation will create a torque about the axis of:

Dilation Torque = 2.83 * (60 + 70) = 368 in lb

This value represents the dilation torque the mechanical drive must overcome.

If we repeat the analysis for a structural weight of 50 lbs we get:

Structural Torque = 25 * 50 = 1250 in lb

D = 1250 / (60 + 35) = 13.2 inches

For a 70 lb child, the center of gravity would dilate:

Delta = 13.2 - (1250 / (60 + 70)) = 3.54 inches

And the dilation torque equals:

Dilation Torque = 3.54 * (60 + 70) = 461 in lb

The analysis shows that for a 10 lb increase in the structural weight of the device frames the torque required to drive the device axis increased by approximately 25%. We know derive a relationship between the dilation torque and structural torque.

We define the following:

DT = Dilation Torque

ST = Structural Torque

D = Distance from centerline of axis to C. G. of patient

Delta = Dilation of system C. G. off axis centerline

WP = Weight of patient

35 lbs = ideal patient weight

60 lbs = weight of enclosure/chair assembly

We know that the distance D is equal to the structural torque divided by the weight of the ideal patient and assembly weight:

$$D = ST / (60 + 35)$$

We may write a torque equation about the device axis as:

$$DT = ST - ((WP + 60) * D)$$

Substituting for D we have:

$$DT = ST - [(WP + 60) * ST] / 95$$

We reduce to:

$$DT = [1 - ((WP + 60) / 95)] * ST$$

This analysis shows that the magnitude of the dilation torque is directly proportional to the magnitude of the structural torque about each axis.

APPENDIX C

Tumble Forms Chair Information

Support and Protection That Works...

... and Plays



Up to this moment, daily seating requirements for people with disabilities have been met with custom seating. Now, Tumble Forms has designed the seating system that solves the positioning problems in day-to-day living.

The Carrie Seat comes complete with footrest, headrest, adjustable tray, and F.M.V.S.S. 213 approved harness and tether strap.

- Anti-thrust seat stabilizes the pelvis.
- Seat back articulates 20° to accommodate abnormal trunk tone.
- Foot rest has a number of adjustments to maintain hip, knee and ankle flexion.
- Height adjustable head rest and extended lateral head supports keep the head in mid-line and neutral.
- Molded seat shape follows the child's contours, providing lateral trunk support and shoulder protraction.
- Tray adjusts in and out and tilts to various angles. Once in place the tray assists upper trunk control.
- Vertical guides adjust the height of the shoulder straps.
- **Meets Federal Motor Vehicle Safety Standard 213.**

Custom seating is designed to accommodate only limited growth. Tumble Forms has made the Carrie Seat available in four "Growth Stage" sizes. Simply by choosing the correct size, the caregiver can individually fit each child.

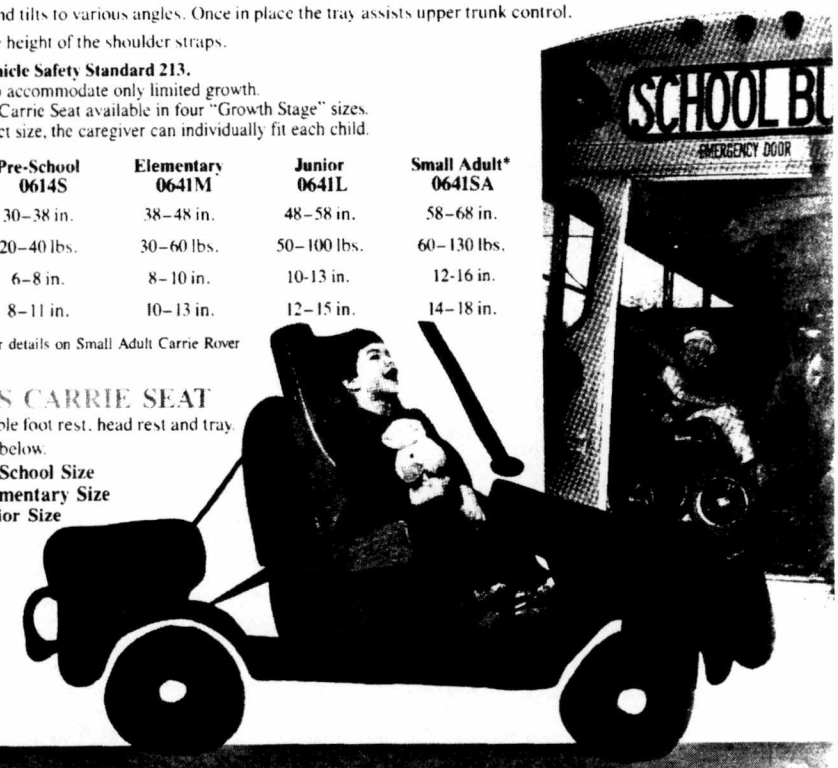
	Pre-School 0614S	Elementary 0641M	Junior 0641L	Small Adult* 0641SA
Standing Height	30–38 in.	38–48 in.	48–58 in.	58–68 in.
Weight	20–40 lbs.	30–60 lbs.	50–100 lbs.	60–130 lbs.
Sitting Hip Width	6–8 in.	8–10 in.	10-13 in.	12-16 in.
Thigh Depth	8–11 in.	10–13 in.	12–15 in.	14–18 in.

*See Page 27 for details on Small Adult Carrie Rover

TUMBLE FORMS CARRIE SEAT

Carrie Seat includes adjustable foot rest, head rest and tray. Available in four sizes listed below.

0641S Carrie Seat, Pre-School Size
0641M Carrie Seat, Elementary Size
0641L Carrie Seat, Junior Size
0641SA Carrie Seat, Small Adult Size



TEC THERAPEUTIC EQUIPMENT CORPORATION

APPENDIX D

Frame Material Analysis

We will now compare the use of aluminum or steel for the device frames. The pertinent properties of the two materials are their elasticity and density.

Elasticity:

Aluminum = 10.4 Mpsi

Steel = 30 Mpsi

Density:

Aluminum = .10 lb/in^3

Steel = .28 lb/in^3

We now consider the second order deflection of a beam:

$$dV^2/dX^2 = -M/EI$$

We consider two solid beams having equal deflection. One beam is made of aluminum (Al) and the other steel (St). The beams will be subjected to the same load hence M will be equal in each. Substituting into the deflection equation we obtain the relation:

$$10.4\ (I\ Al) = 30\ (I\ St)$$

$$I\ Al\ /\ I\ St = 2.88$$

If we assume a square tube then the Moment of Inertia (I) is proportional to the height (H) of the tube raised to the 4th power. Therefore we may find the required ratio of the height of the tubes:

$$H\ Al\ /\ H\ St = 1.303$$

The weight (W) of each tube will be proportional to the area of the tubing material multiplied by the density of the material. The area of the tubing is proportional to

H^2. Using the ratio of the densities of the material we find:

$$(H Al / H St)^2 * (den Al / den St) = 1.303^2 * .357$$

$$W AL / W St = .606$$

This analysis demonstrates that for equivalent deflections the weight of the steel structure will be approximately 165% that of the aluminum structure.

APPENDIX E

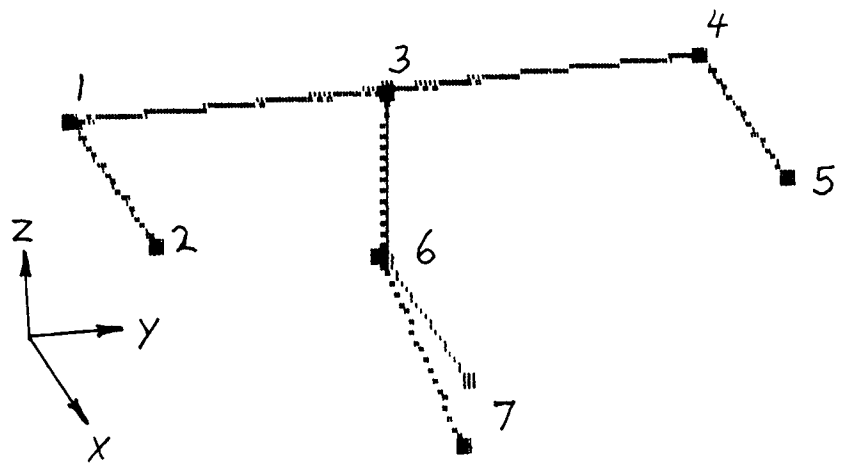
Frame Stiffness Analysis

The device frame was modeled using various sizes of tubing for each member. By inspecting the deflections of each node, the optimal size tubing for each frame component was determined. In the model results the deflection of the structure at the patient location is boxed. The sectional properties for the various sizes of tubing are presented below:

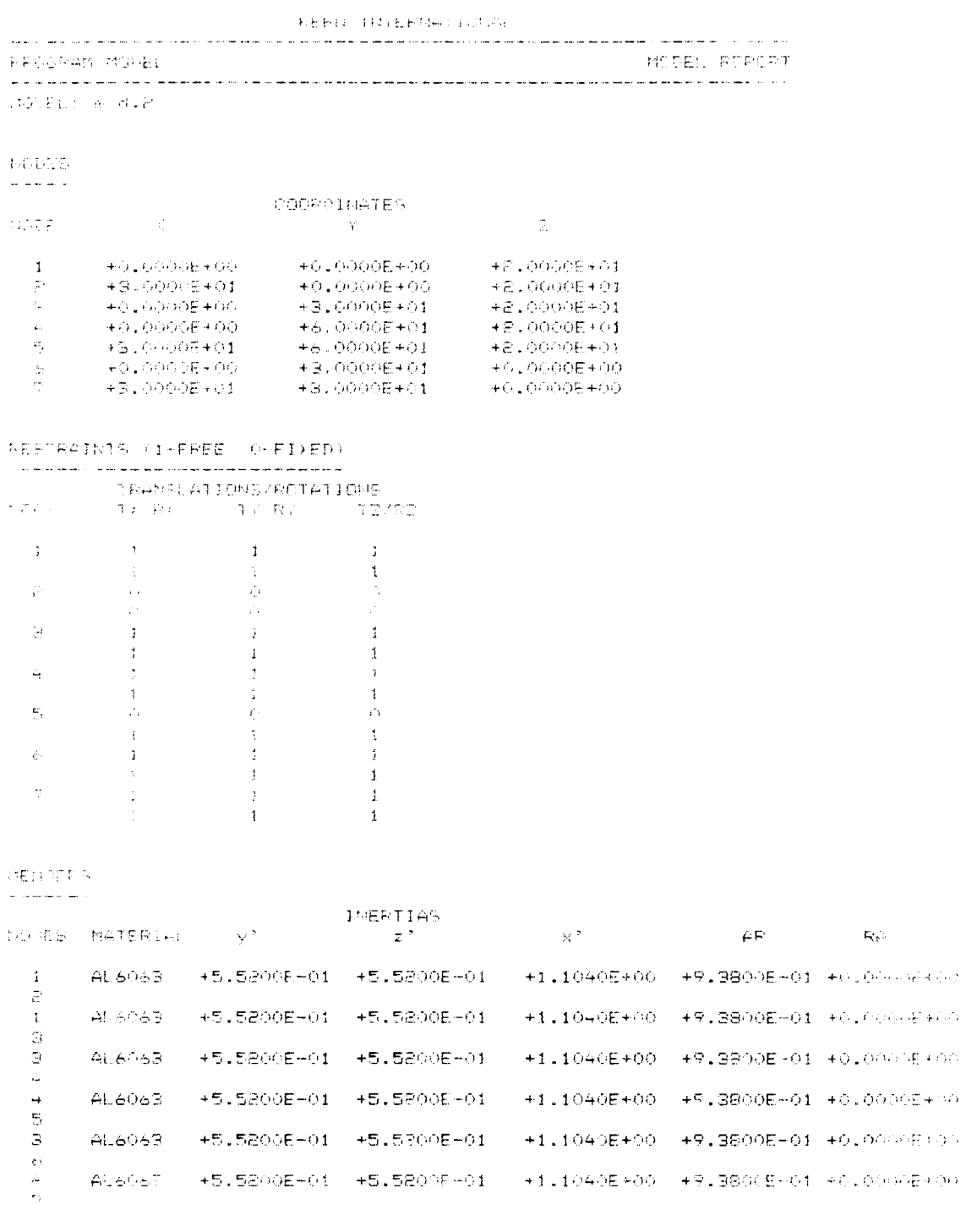
SECTION PROPERTIES CALCULATIONS

HEIGHT	WIDTH	THICKNESS	I(X)	I(Y)	I(POLAR)	AREA
1.000	1.000	.125	.057	.057	.114	.438
1.500	1.500	.125	.218	.218	.437	.688
2.000	2.000	.125	.552	.552	1.104	.938
2.500	2.500	.125	1.119	1.119	2.239	1.188
3.000	3.000	.125	1.984	1.984	3.968	1.438
3.500	3.500	.125	3.208	3.208	6.416	1.688
4.000	4.000	.125	4.854	4.854	9.708	1.938
2.000	1.000	.125	.332	.105	.437	.688
2.000	1.500	.125	.442	.278	.719	.813
2.000	1.750	.125	.497	.401	.898	.875
3.000	1.000	.125	.950	.153	1.104	.938
3.000	1.500	.125	1.209	.396	1.605	1.063
3.000	1.750	.125	1.338	.566	1.904	1.125
3.000	2.000	.125	1.467	.772	2.239	1.188

The device structure as modeled by SSAM is shown below. The structure nodes are labeled and the axis direction convention is shown.



- Model of the structure using 2 inch square tubing for all components.



PROGRAM MODEL — MODEL REPORT

MODEL A.M.P

NODES

NODE	COORDINATES X	Y	Z
1	+0.0000E+00	+0.0000E+00	+2.0000E+01
2	+3.0000E+01	+0.0000E+00	+2.0000E+01
3	+0.0000E+00	+3.0000E+01	+2.0000E+01
4	+0.0000E+00	+6.0000E+01	+2.0000E+01
5	+3.0000E+01	+6.0000E+01	+2.0000E+01
6	+0.0000E+00	+3.0000E+01	+0.0000E+00
7	+3.0000E+01	+3.0000E+01	+0.0000E+00

RESTRAINTS (1=FREE 0=FIXED)

NODE	TRANSLATIONS/ROTATIONS Tx Rx	Ty Ry	Tz/Rz
1	1	1	1
	1	1	1
2	0	0	0
	0	0	0
3	1	1	1
	1	1	1
4	1	1	1
	1	1	1
5	0	0	0
	1	1	1
6	1	1	1
	1	1	1
7	1	1	1
	1	1	1

MEMBERS

MEMBER	MATERIAL	INERTIAS y'	z'	x'	AR	RR
1	AL6063	+5.5200E-01	+5.5200E-01	+1.1040E+00	+9.3800E-01	+0.0000E+00
2						
1	AL6063	+5.5200E-01	+5.5200E-01	+1.1040E+00	+9.3800E-01	+0.0000E+00
3						
3	AL6063	+5.5200E-01	+5.5200E-01	+1.1040E+00	+9.3800E-01	+0.0000E+00
4						
4	AL6063	+5.5200E-01	+5.5200E-01	+1.1040E+00	+9.3800E-01	+0.0000E+00
5						
3	AL6063	+5.5200E-01	+5.5200E-01	+1.1040E+00	+9.3800E-01	+0.0000E+00
6						
6	AL6063	+5.5200E-01	+5.5200E-01	+1.1040E+00	+9.3800E-01	+0.0000E+00
7						

- Results of the structure using 2 inch square tubing for all components.

PERN INTERNATIONAL

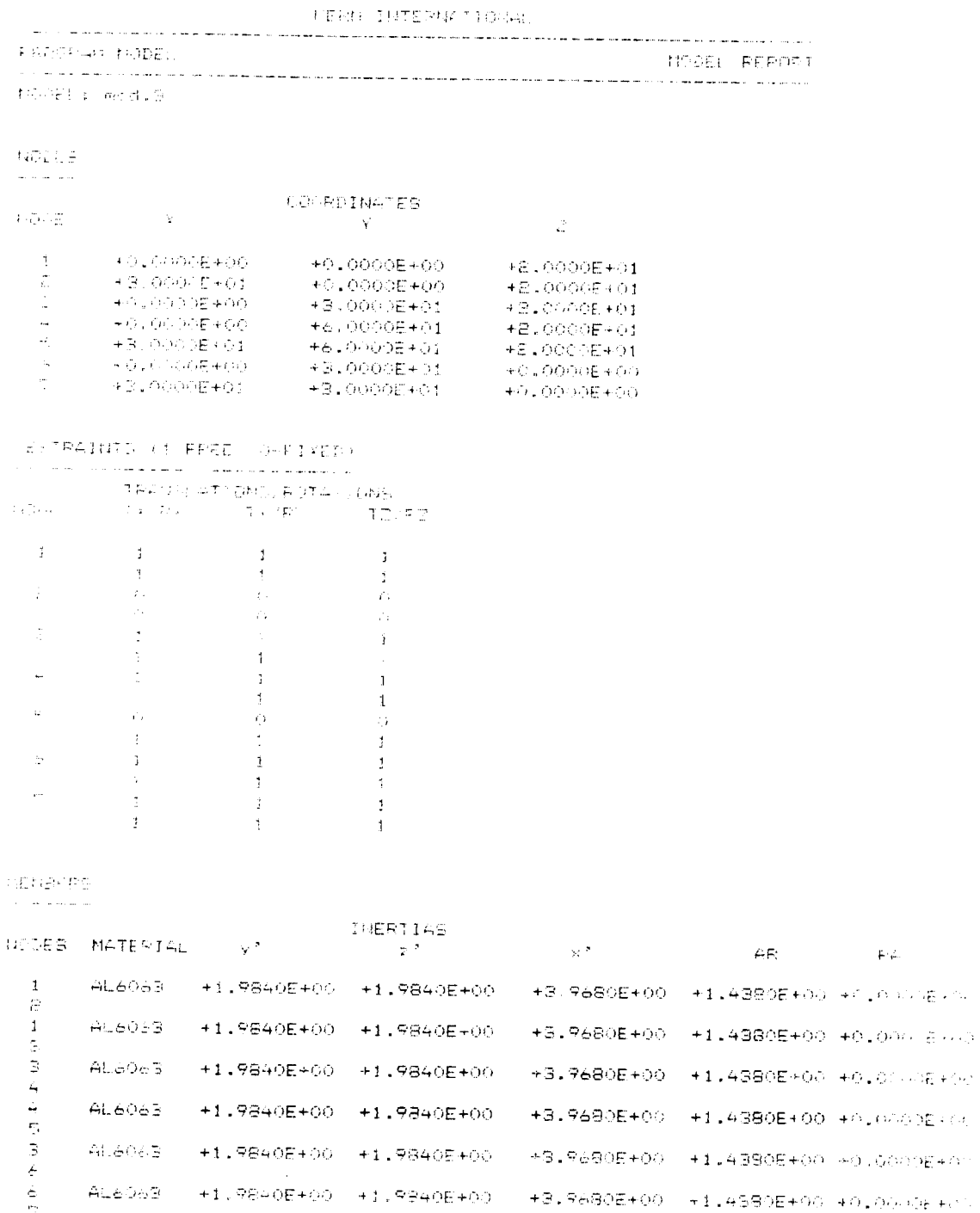
PROGRAM PRINT

RESULT FILE: rmod.2 SCALE FACTOR: 1

LOCAL DISPLACEMENTS

NODE	TRANSLATIONS/ROTATIONS X/XX	Y/YY	Z/ZZ
1	-2.2592E-03	-1.1742E-06	+5.3373E-02
	-2.2749E-03	+5.3365E-03	+1.6114E-07
2	+0.0000E+00	+0.0000E+00	+0.0000E+00
	+0.0000E+00	+0.0000E+00	+0.0000E+00
3	-9.2185E-06	-1.1645E-06	+1.1882E-02
	-1.8045E-03	+1.9018E-02	-7.5264E-09
4	-2.4456E-03	-1.1544E-06	+1.5945E-01
	+5.7044E-03	+7.6150E-03	-1.3032E-07
5	+0.0000E+00	+0.0000E+00	+0.0000E+00
	+5.7044E-03	+8.1652E-03	+1.2241E-07
6	-5.1165E-01	+2.5729E-02	+1.1626E-02
	+1.2465E-03	+8.2062E-02	-6.4736E-09
7	-5.1165E-01	+2.5729E-02	-1.1459E+00
	+1.2665E-03	+4.1844E-02	-7.3534E-09

- Model of the structure using 3 inch square tubing for all components.



[illegible] INTERNATIONAL

[illegible] MODEL [illegible] MODEL REPORT

MODEL : model

NODES

NODE	COORDINATES X	Y	Z
1	+0.0000E+00	+0.0000E+00	+2.0000E+01
2	+3.0000E+01	+0.0000E+00	+2.0000E+01
3	+0.0000E+00	+3.0000E+01	+2.0000E+01
4	+0.0000E+00	+6.0000E+01	+2.0000E+01
5	+3.0000E+01	+6.0000E+01	+2.0000E+01
6	+0.0000E+00	+3.0000E+01	+0.0000E+00
7	+3.0000E+01	+3.0000E+01	+0.0000E+00

RESTRAINTS (1 PAGE SKIPPED)

NODE	RESTRAINTS ROTATIONS	TRANS	TRANS
1	1	1	1
	1	1	1
2	0	0	0
	0	0	0
3	1	1	1
	1	1	1
4	1	1	1
	1	1	1
5	0	0	0
	1	1	1
6	1	1	1
	1	1	1
7	1	1	1
	1	1	1

MEMBERS

MEMBERS	MATERIAL	INERTIAS y'	z'	x'	AR	FA
1 2	AL6063	+1.9840E+00	+1.9840E+00	+3.9680E+00	+1.4380E+00	+0.0000E+00
1 3	AL6063	+1.9840E+00	+1.9840E+00	+3.9680E+00	+1.4380E+00	+0.0000E+00
3 4	AL6063	+1.9840E+00	+1.9840E+00	+3.9680E+00	+1.4380E+00	+0.0000E+00
4 5	AL6063	+1.9840E+00	+1.9840E+00	+3.9680E+00	+1.4380E+00	+0.0000E+00
3 6	AL6063	+1.9840E+00	+1.9840E+00	+3.9680E+00	+1.4380E+00	+0.0000E+00
6 7	AL6063	+1.9840E+00	+1.9840E+00	+3.9680E+00	+1.4380E+00	+0.0000E+00

- Results of the structure using 3 inch square tubing for all components.

KERN INTERNATIONAL

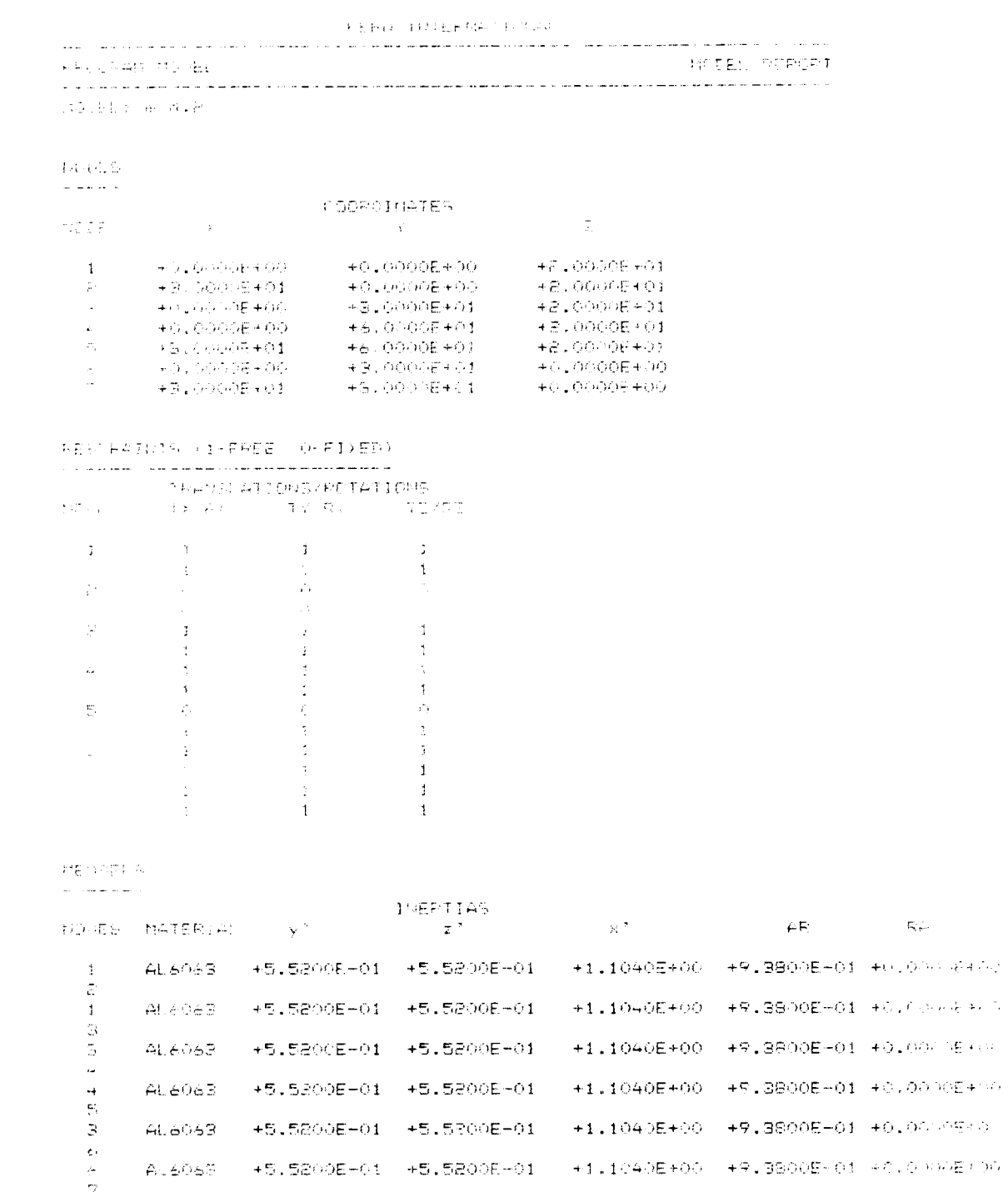
PROGRAM PRINT

RESULT FILE: PROG.3 SCALE FACTOR: 1

LOCAL DISPLACEMENTS

NODE	TRANSLATIONS/ROTATIONS X/XX	Y/YY	Z/ZZ
1	-1.9659E-09	-5.2888E-08	+1.4844E-02
	-6.3294E-04	+1.4844E-03	+6.2735E-09
2	+0.0000E+00	+0.0000E+00	+0.0000E+00
	+0.0000E+00	+0.0000E+00	+0.0000E+00
3	-1.9418E-07	-5.2130E-08	+3.2935E-03
	+3.5747E-04	+5.2904E-03	-5.8335E-10
4	-2.2385E-09	-5.1332E-08	+3.8773E-02
	+1.5551E-02	+3.1174E-03	-4.1465E-09
5	+0.0000E+00	+0.0000E+00	+0.0000E+00
	+1.5951E-03	+8.7994E-04	+4.4636E-09
6	-1.4710E-01	+7.1534E-03	+8.1266E-03
	+3.5726E-04	+8.9194E-03	-1.4295E-09
7	-1.4210E-01	+7.1533E-03	-3.1869E-01
	+3.5767E-04	+1.1641E-02	-1.7488E-09

- Model of the prototype structure with as built frame dimensions and tubing sizes. Note the additional node point used to model the roll axis.



NODE COORDINATES

NODE	X	Y	Z
1	+0.0000E+00	+0.0000E+00	+7.0000E+01
2	+3.0000E+01	+0.0000E+00	+2.0000E+01
3	+0.0000E+00	+3.0000E+01	+2.0000E+01
4	+0.0000E+00	+5.0000E+01	+2.0000E+01
5	+5.0000E+01	+6.0000E+01	+2.0000E+01
6	+0.0000E+00	+3.0000E+01	+0.0000E+00
7	+3.0000E+01	+5.0000E+01	+0.0000E+00

RESTRAINTS (1-FREE 0-FIXED)

NODE	TRANSLATIONS/ROTATIONS X	Y	Z
1	1	1	1
2	1	1	1
3	0	0	0
4	1	1	1
5	0	0	0
6	1	1	1
7	1	1	1

MEMBERS

MEMBER	MATERIAL	INERTIAS y'	z'	x'	AR	SH
1	AL6063	+5.5200E-01	+5.5200E-01	+1.1040E+00	+9.3800E-01	+0.0000E+00
2						
1	AL6063	+5.5200E-01	+5.5200E-01	+1.1040E+00	+9.3800E-01	+0.0000E+00
3						
5	AL6063	+5.5200E-01	+5.5200E-01	+1.1040E+00	+9.3800E-01	+0.0000E+00
6						
4	AL6063	+5.5200E-01	+5.5200E-01	+1.1040E+00	+9.3800E-01	+0.0000E+00
5						
3	AL6063	+5.5200E-01	+5.5200E-01	+1.1040E+00	+9.3800E-01	+0.0000E+00
6						
6	AL6063	+5.5200E-01	+5.5200E-01	+1.1040E+00	+9.3800E-01	+0.0000E+00
7						

- Results of the prototype structure.

VERN INTERNATIONAL

PROGRAM PRINT

RESULT FILE:PMODFRE.PRO SCALE FACTOR: 1

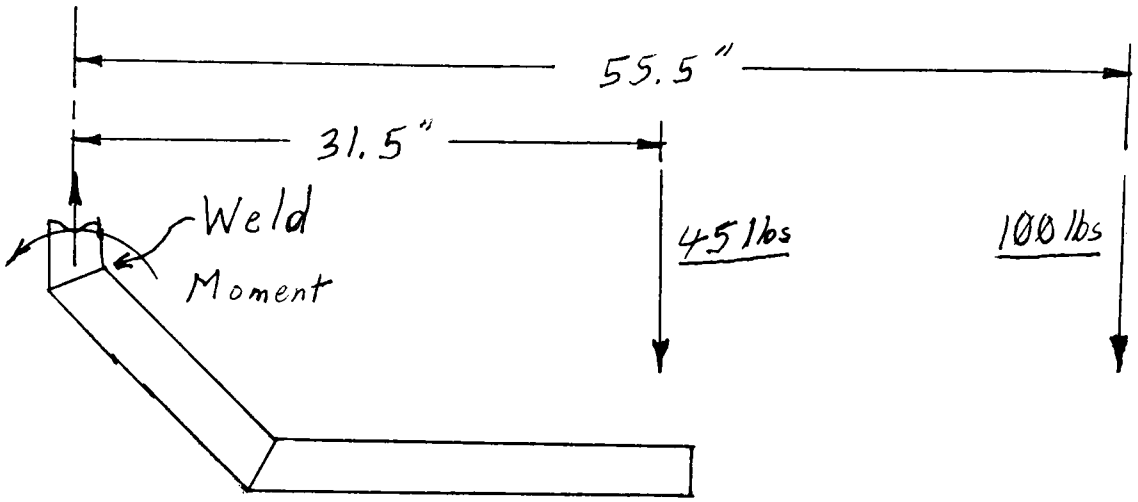
LOCAL DISPLACEMENTS

NODE	TRANSLATIONS/ROTATIONS X/XX	Y/YY	Z/ZZ
1	-8.1797E-06	-4.2210E+03	+2.5167E-02
	-1.0036E-03	+2.4177E-03	-1.6078E-05
2	+0.0000E+00	-4.2210E+03	+0.0000E+00
	-1.0034E-03	+0.0000E+00	-9.2735E-05
3	-5.9086E-04	-4.2210E+03	+3.7466E-03
	-1.5277E-04	+6.2256E-03	+2.4375E-05
4	+6.2775E-04	-4.2210E+03	+3.5432E-02
	+1.6719E-03	+2.5415E-03	-8.9520E-05
5	+0.0000E+00	-4.2210E+03	+0.0000E+00
	+1.6715E-03	+1.0388E-03	-2.5234E-04
6	-1.1837E-01	-4.2210E+03	+3.6190E-03
	-4.7715E-04	+9.8193E-03	+2.7611E-05
7	-1.9746E-01	-4.2210E+03	-7.5561E-02
	-4.7690E-04	+1.1117E-02	+4.1687E-05
8	-1.9746E-01	-4.2210E+03	-3.8127E-01
	-4.7850E-04	+1.2969E-02	+4.1687E-05

APPENDIX F

Frame Stress Calculations

The first stress case considered is the bending stress on the roll axis. The stress will be considered at the last weld of the assembly. This weld receives the maximum bending stress seen in the structure.



The worst case load imposed on the roll structure is made of 2 components. The weight of the enclosure / yaw gearbox / Tumble Forms chair and associated components is approximately 45 lbs and is 31.5 inches from the joint in question. The child is considered stepping into the device assisted by a therapist. The total weight imposed is considered at 100 lbs. The location of this load will be a the step incorporated at the base of the enclosure. The distance to the weld is 55.5 inches. We know use the standard bending stress equation:

Stress = Moment * Y / Area Moment of Inertia (I)

Where:

M = (45 * 31.5) + (100 * 55.5) = 7000 in * lb

I = ((3 * (3.25)^3) - (2.75 * (3)^3)) / 12 = 2.39 in^4

Y = 1.625 in

Stress = 4759 psi

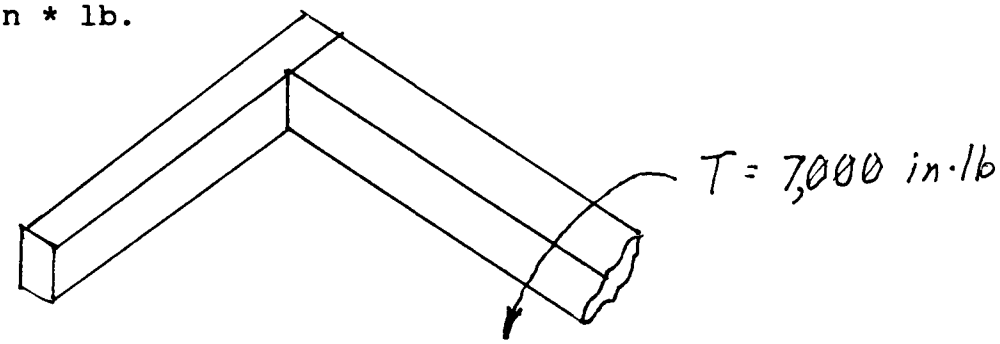
Their is an additional stress on the structure due to the axial load on the tubing. This stress has a magnitude of 86 psi. The tensile yield of the aluminum in the as welded state is 11,000 psi. The factor of safety is:

F.S. = 11000 / (4759 + 86)

F. S. = 2.25

The Aluminum Association Recommends a Factor of Safety of 1.85 for this type of structure.

The second stress case considered is the torsional load imposed on the backbone of the pitch axis by the cantilevered roll axis. The torsional load (T) is equal to 7000 in * lb.



The formula for the torsional stress of a square cross-section tube is:

$$Tau = T / (2 * Sec * t)$$

where:

Sec = the sectorial area of the cross-section

Sec = (3 - .125)^2 = 8.27 in^2

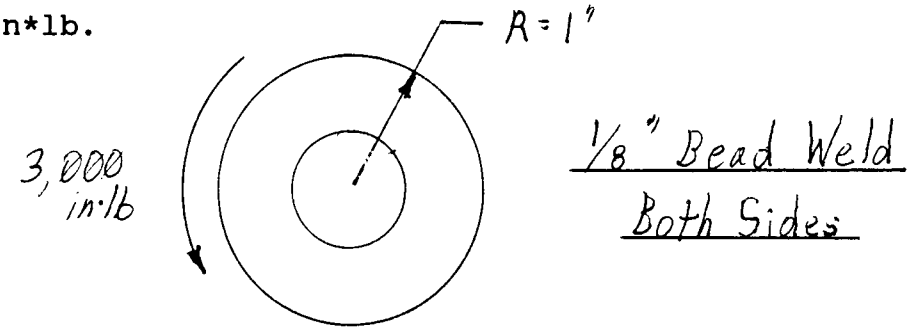
t = the wall thickness of the section = .125

$$Tau = 2420 \text{ psi}$$

The as welded shear strength of 6063-T5 is 6.5 ksi. The factor of safety is:

$$F. S. = 6000 / 2420 = 2.5$$

The third stress case considered is the torsional shear stress on the frame insert on the driven side of the pitch axis. This insert transmits all of the torque imposed on the pitch axis to the pitch axis mechanical drive. The radius of the insert is 1 inch and it is welded to both sides of the tube in which it is inserted. The maximum moment about the joint is approximately 3000 in*lb.



The weld throat length is:

WThroat = .125 * .707 = .088 in

The area of material in shear is:

Area = 2 * Pi * r * 2 * WThroat = 1.11 in^2

The shear stress is:

Tau = (Moment / r) / Area

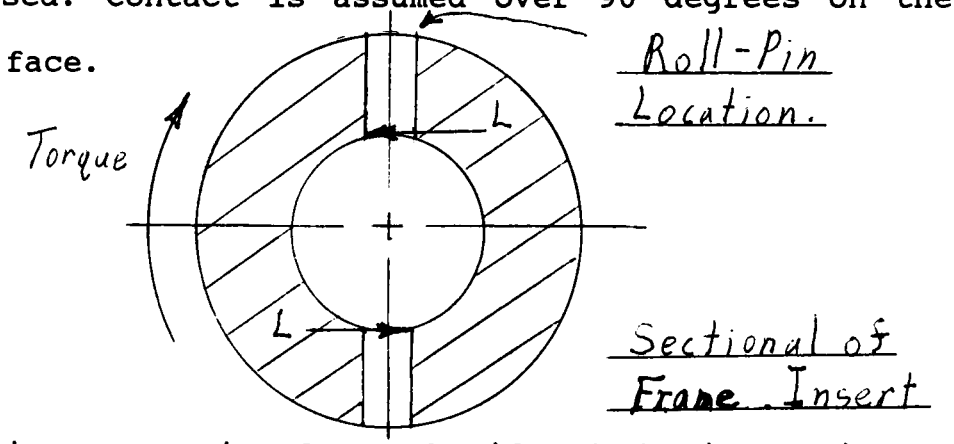
Tau = 2700 psi

The shear strength of the material is 6500 psi

The factor of safety is;

F. S. = 2.4

The frame inserts are analyzed based on a compressive stress analysis at the roll-pin hole. The compressive yield strength of 6063-T5 Aluminum is 11,000 psi. The inserts have a 2 inch O. D. and a 1 inch bore. A .25 roll pin is used. Contact is assumed over 90 degrees on the roll pin face.



The Area in compression for each side of the insert is:

Area = 2 * .25 * (.5) * .707 = .177 in^2

The load is placed at the worst case radius of .5 inches. With these assumptions the allowable torque is:

Torque = .177 in^2 * .5 * 2 * 11,000 psi = 1,770 in*lb

The pitch axis frame insert experiences a higher torque than this and thus required the addition of a steel insert as discussed in section 5.4 .

The frame is analyzed for fatigue using Aluminum Association published S/N information for aluminum alloy and the type of welded joints as used in the prototype. This information is presented below:

Design Stress Cycles	Allowable Stress Range (ksi)
up to 100,000	12.0
100,000 - 500,000	8.2
500,000 - 2,000,000	5.9
over 2,000,000	4.0

The # of cycles the device may experience is:

2 Cycles per Patient

8 Patients per Day

260 Days per Year

This results in only 4,160 cycles per year.

The maximum stress imposed on the structure is less than 5,000 psi. The magnitude of the varying stress is less.

APPENDIX G

Dynamic Analysis

The maximum velocity and acceleration of the pitch roll and yaw axes will now be calculated. We differentiate the sinusoidal motion equation to obtain expressions for the velocity and acceleration. In the calculations, the period of motion is T, the circular frequency is W, the amplitude of oscillation is A, and PI = 3.1415.

```
ANG = A * SIN( W * t )
VEL = A * W * COS( W * t )
ACC = -A * W^2 * SIN( W * t )
```

where W = 2 * PI / T and A = PI / 6

The period of motion for the pitch and roll axes is 10 seconds. The period for the yaw axis is 5 seconds. Using these values the maximum velocity and acceleration of each axis is calculated:

Pitch and Roll Axes

VEL = .329 rad / sec

ACC = .207 rad / sec

Yaw Axis

VEL = .658 rad / sec

ACC = .827 rad / sec

The rotational inertia of the device about each axis will now be estimated. The total weight of the patient, enclosure and Tumble Forms chair was estimated at 125 lbs. The mass distribution was modeled as a rectangular solid of dimensions 50 * 30 * 30 inches. The moment of inertia (I) of this solid was calculated about each of its axes by:

$$I = Mass * (height^2 + width^2) / 12$$

The inertia of all other rotating components was calculated about each axis using the parallel axis theorem. The frame components of each axis were modeled as slender rods. The gearboxes were modeled as point masses. The total rotating inertia about each axis was found to be:

Pitch Axis = 132 in lb sec^2

Roll Axis = 110 in lb sec^2

Yaw Axis = 50 in lb sec^2

The maximum torque about each axis is calculated using the Euler equations of motion. The following notations are used: Subscripts 1, 2, and 3 refer to the pitch, roll and yaw axes respectively, I is the mass moment of inertia about the axis, W is the maximum angular velocity of rotation, and DW is the maximum angular acceleration about each axis. The Euler equations are:

$$N(1) = I(1) * DW(1) + (I(3) - I(2)) * W(2) * W(3)$$

$$N(2) = I(2) * DW(2) + (I(1) - I(3)) * W(1) * W(3)$$

$$N(3) = I(3) * DW(3) + (I(2) - I(1)) * W(1) * W(2)$$

Substituting the values presented, we obtain the maximum dynamic torques about each axis as:

Pitch Axis = 40.3 in lb

Roll Axis = 40.5 in lb

Yaw Axis = 43.7 in lb

APPENDIX H

Axes Torque Balancing Spreadsheets

- Pitch axis with a 70 lb child.

MEMBER	LENGTH (IN)	WEIGHT (LB)	DISTANCE (IN)	MOMENT (IN LB)
3*3 PITCH AXIS	58.25	8.39	23	192.97
1.75*3 PITCH AXIS (2)	49	5.5	12	66
3*3 ROLL #1	19.24	2.77	23	63.71
3*3 ROLL #2	13.09	1.88	16.15	30.362
3*3 ROLL #3	22.74	3.27	1.75	5.7225
GEARBOX ROLL		18	27	486
TOTAL WEIGHT (LBS) =	39.81			
TOTAL MOMENT (IN LB) =	775.097			
CHILD		70		
ENCLOSURE		20		
TFORMS CHAIR		20		
TOWER		2.16		
YAW GEARBOX		13		
SHAFT/BEARING/PEDESTAL		7		
TOTAL WEGHT (LBS)		132.16		
DISTANCE PITCH AXIS (IN)		-8		
MOMENT (IN LB)		-1057.28		
TRUE CG LOCATION (IN)	-1.64089			
REQ. DRIVE TORQUE (IN LB)	-282.183			

- Pitch axis with no child.

MEMBER	LENGTH (IN)	WEIGHT (LB)	DISTANCE (IN)	MOMENT (IN LB)
3*3 PITCH AXIS	58.25	8.39	23	192.97
1.75*3 PITCH AXIS (2)	49	5.5	12	66
3*3 ROLL #1	19.24	2.77	23	63.71
3*3 ROLL #2	13.09	1.88	16.15	30.362
3*3 ROLL #3	22.74	3.27	1.75	5.7225
GEARBOX ROLL		18	27	486
TOTAL WEIGHT (LBS) =	39.81			
TOTAL MOMENT (IN LB) =	775.097			
CHILD		0		
ENCLOSURE		20		
TFORMS CHAIR		20		
TOWER		2.16		
YAW GEARBOX		13		
SHAFT/BEARING/PEDESTAL		7		
TOTAL WEGHT (LBS)		62.16		
DISTANCE PITCH AXIS (IN)		-8		
MOMENT (IN LB)		-497.28		
TRUE CG LOCATION (IN)	2.724497			
REQ. DRIVE TORQUE (IN LB)	277.817			

- Roll axis with a 70 lb child

MEMBER	WEIGHT (LB)	DISTANCE (IN)	MOMENT (IN LB)
3*3 ROLL #1	2.77	6	16.62
3*3 ROLL #2	1.88	18	33.84
3*3 ROLL #3	3.27	23	75.21
3*3 TOWER	2.16	10	21.6
YAW GEARBOX	13	26.5	344.5
SHAFT	3	10	30
TOTALS	26.08		521.77

CG LOCATION OF MEMBERS	20.00652
CHILD	70
ENCLOSURE	20
CHAIR	20
PEDESTAL/BEARING	5
TOTAL WEIGHT (LBS)	115
DISTANCE ROLL AXIS (IN)	6.5
MOMENT (IN LB)	747.5
TRUE CG LOCATION (IN)	1.600014
REQ. DRIVE TORQUE (IN LB)	112.865

- Roll axis with no child.

MEMBER	WEIGHT (LB)	DISTANCE (IN)	MOMENT (IN LB)
3*3 ROLL #1	2.77	6	16.62
3*3 ROLL #2	1.88	18	33.84
3*3 ROLL #3	3.27	23	75.21
3*3 TOWER	2.16	10	21.6
YAW GEARBOX	13	26.5	344.5
SHAFT	3	10	30
TOTALS	26.08		521.77

CG LOCATION OF MEMBERS	20.00652
CHILD	0
ENCLOSURE	20
CHAIR	20
PEDESTAL/BEARING	5
TOTAL WEIGHT (LBS)	45
DISTANCE ROLL AXIS (IN)	6.5
MOMENT (IN LB)	292.5
TRUE CG LOCATION (IN)	-3.22552
REQ. DRIVE TORQUE (IN LB)	-114.635

- Pitch axis with 35 lb child.

MEMBER	LENGTH (IN)	WEIGHT (LB)	DISTANCE (IN)	MOMENT (IN LB)
3*3 PITCH AXIS	58.25	8.39	23	192.97
1.75*3 PITCH AXIS (2)	49	5.5	12	66
3*3 ROLL #1	19.24	2.77	23	63.71
3*3 ROLL #2	13.09	1.88	16.15	30.362
3*3 ROLL #3	22.74	3.27	1.75	5.7225
GEARBOX ROLL		18	27	486
TOTAL WEIGHT (LBS) =	39.81			
TOTAL MOMENT (IN LB) =	775.097			
CHILD		35		
ENCLOSURE		20		
TFORMS CHAIR		20		
TOWER		2.16		
YAW GEARBOX		13		
SHAFT/BEARING/PEDESTAL		7		

TOTAL WEGHT (LBS)		97.16		
DISTANCE PITCH AXIS (IN)		-8		
MOMENT (IN LB)		-777.28		
TRUE CG LOCATION (IN)	-.015938			
REQ. DRIVE TORQUE (IN LB)	-2.183			

- Roll axis with 35 lb child.

MEMBER	WEIGHT (LB)	DISTANCE (IN)	MOMENT (IN LB)
3*3 ROLL #1	2.77	6	16.62
3*3 ROLL #2	1.88	18	33.84
3*3 ROLL #3	3.27	23	75.21
3*3 TOWER	2.16	10	21.6
YAW GEARBOX	13	26.5	344.5
SHAFT	3	10	30
	---		---
TOTALS	26.08		521.77
CG LOCATION OF MEMBERS	20.00652		
CHILD	35		
ENCLOSURE	20		
CHAIR	20		
PEDESTAL/BEARING	5		

TOTAL WEIGHT (LBS)	80		
DISTANCE ROLL AXIS (IN)	6.5		
MOMENT (IN LB)	520		
TRUE CG LOCATION (IN)	-.016686		
REQ. DRIVE TORQUE (IN LB)	-.885		

APPENDIX I

Mechanical Drive Analysis

- BASICA program for gear reduction and pitch combinations.

```
10 REM GEAR 2
20 REM
30 REM UPDATED VERSION OF GEAR
40 REM
50 REM PROGRAM TO COMPUTE THE WORM GEAR REDUCTION
60 REM
70 REM E. J. ASBURY
80 REM
90 REM THE TORQUE REQUIRED OF THE GEARBOW IS T ( IN LB )
100 T = 400
110 REM
120 REM THE REQUIRED SPEED IS FOR A PERIOD OF 10 SECONDS
130 REM WITH A AMPLITUDE OF 30 SECONDS
140 REM
150 REM ITERATE ON GEAR REDUCTION
160 FOR RED = 50 TO 100 STEP 10
170 REM
180 REM CALCULATE SIZE OF GEAR USING THIS REDUCTION
190 REM ASSUMING ONE TOOTH ON THE WORM AND ITERATING ON
200 REM DIAMETRICAL PITCH UNTIL THE STRENGTH IS ACCEPTABLE
210 REM THE FIRST TWO POSSIBLE PITCHES WILL BE PRINTED
220 REM
230 LPRINT"  "
250 LPRINT"  "
260 LPRINT"  "
270 LPRINT USING"    GEAR REDUCTION  ###.#";RED
290 LPRINT"  "
300 REM ITERATE ON PITCH
310 REM P = PITCH
320 FOR P = 24 TO 8 STEP -2
330 REM CP = CIRCULAR PITCH
340 CP = 3.1416/P
350 REM DG = PITCH DIAMETER OF GEAR
360 DG = RED*CP/3.1416
370 REM FT = FORCE TRANSMITTED TO GEAR TOOTH (LB)
380 FT = T/(DG/2)
390 REM VG = PITCH LINE VELOCITY OF GEAR (FT/SEC)
400 VG = (3.1416^2/30)*DG/2/12
410 REM FD = DYNAMIC FORCE GENERATED
420 FD = FT*((1200+VG)/1200)
430 REM SN = FATIQUE STRENGTH OF BRONZE
440 SN = 24000
450 REM Y = LEWIS FORM FACTOR FOR 14.5 DEGREE PRESSURE ANGLE
460 Y = .1
470 REM DW = PITCH DIAMETER OF WORM BY CURVE FIT OF BOSTON GEARS
480 DW = 1.7 + .142*P - .032*P^2 + .0015*P^3 - .00002*P^4
490 REM B = FACE WIDTH OF GEAR
500 B = DW/2
510 REM FS = GEAR STRENGTH    FS MUST BE GREATER THAN FD
520 FS = SN*B*CP*Y
530 REM CHECK THAT STRENGTH EXCEEDS LOAD IF NOT THEN NEXT PITCH
540 IF FD > FS THEN 720
550 REM LA = LEAD ANGLE OF WORM
560 LA = ATN(CP/(3.1416*DW))
570 REM EFF = EFFICIENCY OF GEARSET
580 EFF = (TAN(LA)*(1-.04*TAN(LA)))/(.04+TAN(LA))
590 REM MS = REQUIRED MOTOR SPEED (RPM)
600 MS = (3.14159^2/30)*RED*(60/(2*3.14159))
610 REM MT = REQUIRED MOTOR TORQUE (IN LB)
620 MT = T/(EFF*RED)
625 LPRINT"  "
630 LPRINT USING "    RED = ##     PITCH = ##    MOTOR TORQUE = ###.##";
      RED,P,MT
640 LPRINT"  "
650 LPRINT USING "    EFF = .###    GEAR DIA = ###.#   WORM DIA = ##.##";
```

```
      EFF,DG,DW
660 LPRINT"  "
670 LPRINT USING "    REQUIRED MOTOR SPEED = ####.#    ";MS
680 LPRINT"  "
690 LPRINT USING "    STRENGTH = #####.#     LOAD = #####.#";FS,FD
700 LET COUNT = COUNT + 1
710 IF COUNT = 2 THEN 730
720 NEXT P
730 COUNT = 0
740 NEXT RED
750 END
```

- Results of Mechanical Drive Analysis

```
GEAR REDUCTION   50.0

RED = 50     PITCH = 12    MOTOR TORQUE =  11.78
EFF = .679    GEAR DIA =   4.2   WORM DIA =  0.97
REQUIRED MOTOR SPEED =  157.1
STRENGTH =   305.8     LOAD =   192.0
RED = 50     PITCH = 10    MOTOR TORQUE =  11.94
EFF = .670    GEAR DIA =   5.0   WORM DIA =  1.22
REQUIRED MOTOR SPEED =  157.1
STRENGTH =   459.9     LOAD =   160.0


GEAR REDUCTION   60.0

RED = 60     PITCH = 14    MOTOR TORQUE =   9.55
EFF = .698    GEAR DIA =   4.3   WORM DIA =  0.76
REQUIRED MOTOR SPEED =  188.5
STRENGTH =   205.6     LOAD =   186.7
RED = 60     PITCH = 12    MOTOR TORQUE =   9.81
EFF = .679    GEAR DIA =   5.0   WORM DIA =  0.97
REQUIRED MOTOR SPEED =  188.5
STRENGTH =   305.8     LOAD =   160.0


GEAR REDUCTION   70.0

RED = 70     PITCH = 14    MOTOR TORQUE =   8.19
EFF = .698    GEAR DIA =   5.0   WORM DIA =  0.76
REQUIRED MOTOR SPEED =  219.9
```

STRENGTH = 205.6 LOAD = 160.0

RED = 70 PITCH = 12 MOTOR TORQUE = 8.41

EFF = .679 GEAR DIA = 5.8 WORM DIA = 0.97

REQUIRED MOTOR SPEED = 219.9

STRENGTH = 305.8 LOAD = 137.2

GEAR REDUCTION 80.0

RED = 80 PITCH = 14 MOTOR TORQUE = 7.17

EFF = .698 GEAR DIA = 5.7 WORM DIA = 0.76

REQUIRED MOTOR SPEED = 251.3

STRENGTH = 205.6 LOAD = 140.0

RED = 80 PITCH = 12 MOTOR TORQUE = 7.36

EFF = .679 GEAR DIA = 6.7 WORM DIA = 0.97

REQUIRED MOTOR SPEED = 251.3

STRENGTH = 305.8 LOAD = 120.0

GEAR REDUCTION 90.0

RED = 90 PITCH = 16 MOTOR TORQUE = 6.21

EFF = .715 GEAR DIA = 5.6 WORM DIA = 0.61

REQUIRED MOTOR SPEED = 282.7

STRENGTH = 144.5 LOAD = 142.2

RED = 90 PITCH = 14 MOTOR TORQUE = 6.37

EFF = .698 GEAR DIA = 6.4 WORM DIA = 0.76

REQUIRED MOTOR SPEED = 282.7

STRENGTH = 205.6 LOAD = 124.5

GEAR REDUCTION 100.0

RED = %100 PITCH = 16 MOTOR TORQUE = 5.59

EFF = .715 GEAR DIA = 6.2 WORM DIA = 0.61

REQUIRED MOTOR SPEED = 314.2

STRENGTH = 144.5 LOAD = 128.0

RED = %100 PITCH = 14 MOTOR TORQUE = 5.73

EFF = .698 GEAR DIA = 7.1 WORM DIA = 0.76

REQUIRED MOTOR SPEED = 314.2

STRENGTH = 205.6 LOAD = 112.0

APPENDIX J

Theoretical Operating Space Analysis

- BASICA program for pitch axis operating space.

```
10 REM E. J. ASBURY
20 REM V. S. CHAIR
30 REM
40 REM THIS PROGRAM COMPUTES THE THE AVAILABLE MOTOR TORQUE
50 REM AND LOAD TORQUE OF THE PITCH AXIS FOR VARIOUS ANGLES
60 REM THE PROGRAM USES SINUSOIDAL MOTION
70 REM THE WEIGHT OF THE PATIENT IS ITTERATED
80 REM AT EACH PATIENT WEIGHT THE PERIOD OF OSCILLATION
90 REM IS ITTERATED ON UNTIL THE MINIMUM POSIBLE PERIOD
100 REM IS FOUND
110 REM
120 REM
130 PI = 3.14159
140 REM
150 REM FORMAT OUTPUT
160 REM
170 LPRINT"PATIENT  X-C.G. Y-C.G. PERIOD  ANG TOR.LOAD TOR.DRIVE"
180 LPRINT" WEIGHT   DIL.   DIL.   (SEC) (DEG)  IN*LB    IN*LB"
190 LPRINT"-------------------------------------------------------"
200 LPRINT
210 REM PW = WEIGHT OF PATIENT
220 FOR PW = 90 TO -20 STEP -10
230 REM X = DILATION DISTANCE OF THE SYSTEM CG
240 REM BET = DILATION ANGLE
250 REM
260 REM MPS AND MRS ARE THE MOMENTS OF THE PITCH AND
270 REM ROLL STRUCTURES ABOUT THEIR RESPECTIVE AXES.
280 MPS = 775.1
290 MRS = 521.8
300 REM
310 REM THE DISTANCE OF THE PATIENT C. G. FROM THE PITCH AND
320 REM ROLL AXES IS XP AND XR
330 XP = 8
340 XR = 6.5
350 REM
360 REM THE WEIGHT OF THE ENCLOSURE/CHAIR ASSEMBLY FOR EACH
370 REM AXIS IS WECP AND WECR
380 WECP = 62.2
390 WECR = 45
400 REM
410 REM DX = HORIZONTAL DILATION DISTANCE
420 REM DY = VERTICLE DILATION DISTANCE
430 DX = (MPS-((WECP+PW)*XP))/(39.8+WECP+PW)
440 DY = (MRS-((WECR+PW)*XR))/(26.1+WECR+PW)
450 REM
460 X = SQR(DX^2+DY^2)
470 BET = ATN(DY/DX)
480 REM LOOP FOR ITTERATING ON PERIOD
490 FOR TAU = 12 TO 5 STEP -.1
500 REM LOOP FOR ITTERATING ON ROTATION ANGLE
510 FOR THT = 0 TO PI/18 STEP PI/180
520 REM TL = TORQUE LOAD
530 TL = X*(PW+WECP+39.8)*COS((BET)-THT)
540 REM
550 REM TS = STALL TORQUE OF THE 450 OZ IN STEPPING MOTOR
570 TS = 450/16
580 REM OMEG = CIRCULAR FREQUENCY (RAD/SEC)
590 OMEG = PI*2/TAU
600 REM AM = AMPLITUDE OF OSCILLATION
610 AM = PI/6
620 REM TIM = TIME AT ANGLE
630 TIM = ATN((THT/AM)/SQR((-1*THT*THT)/(AM*AM)+1))/OMEG
640 REM DTHT = ANGULAR VELOCITY
650 DTHT = AM*OMEG*COS(OMEG*TIM)
660 REM TM = TORQUE SUPPLIED BY MOTOR
```

```
670 TM = (1-(DTHT/.523))*TS
680 REM TA = TORGUE AVAILABLE AFTER GEAR REDUCTION
690 TA = TM*.622*60
700 ANG = THT*180/PI
710 IF (TA-TL) < 24 THEN 740
720 NEXT THT
730 NEXT TAU
740 LPRINT USING
        " ###.#    ##.##   ##.##    ##.#   ##.#    ###.#     ###.# ";
           PW,DX,DY,TAU,ANG,TL,TA
750 LPRINT
760 NEXT PW
770 END
```

- Results of the pitch axis analysis.

PATIENT WEIGHT	X-C.G. DIL.	Y-C.G. DIL.	PERIOD (SEC)	ANG (DEG)	TOR.LOAD IN*LB	TOR.DRIVE IN*LB
90.0	-2.30	-2.21	12.0	7.0	490.9	514.6
80.0	-1.99	-1.92	10.3	7.0	402.5	426.3
70.0	-1.64	-1.60	9.0	3.0	296.5	319.7
60.0	-1.25	-1.23	8.0	0.0	202.5	224.3
50.0	-0.81	-0.79	7.3	0.0	122.5	145.2
40.0	-0.30	-0.28	6.7	0.0	42.5	64.2
30.0	0.28	0.34	6.6	0.0	37.5	49.2
20.0	0.96	1.09	7.2	0.0	117.5	132.6
10.0	1.76	2.03	8.0	1.0	201.4	224.8
0.0	2.72	3.23	9.0	4.0	299.8	322.6
-10.0	3.89	4.82	10.4	7.0	408.8	432.3
-20.0	5.34	7.03	12.0	5.0	486.1	507.1

- BASICA program for roll axis operating space.

```
10 REM E. J. ASBURY
20 REM V. S. CHAIR
30 REM
40 REM THIS PROGRAM COMPUTES THE THE AVAILABLE MOTOR TORQUE
50 REM AND LOAD TORQUE OF THE ROLL AXIS FOR VARIOUS ANGLES
60 REM THE PROGRAM USES SINUSOIDAL MOTION
70 REM THE WEIGHT OF THE PATIENT IS ITTERATED
80 REM AT EACH PATIENT WEIGHT THE PERIOD OF OSCILLATION
90 REM IS ITTERATED ON UNTIL THE MINIMUM POSIBLE PERIOD
100 REM IS FOUND
110 REM
120 REM
130 PI = 3.14159
140 REM
150 REM FORMAT OUTPUT
160 REM
170 LPRINT"PATIENT  Y-C.G. PERIOD  ANG TOR.LOAD TOR.DRIVE"
180 LPRINT" WEIGHT    DIL   (SEC) (DEG)  IN*LB    IN*LB"
190 LPRINT"-----------------------------------------------"
200 LPRINT
210 REM PW = WEIGHT OF PATIENT
220 FOR PW = 150 TO -40 STEP -10
250 REM
260 REM MRS IS THE MOMENT OF THE ROLL
270 REM STRUCTURE ABOUT ITS AXIS.
290 MRS = 521.8
300 REM
310 REM THE DISTANCE OF THE PATIENT C. G. FROM THE
320 REM ROLL AXIS IS XR
340 XR = 6.5
350 REM
360 REM THE WEIGHT OF THE ENCLOSURE/CHAIR ASSEMBLY FOR
370 REM THE ROLL AXIS IS WECR
390 WECR = 45
400 REM
420 REM DY = VERTICLE DILATION DISTANCE
440 DY = ABS((MRS-((WECR+PW)*XR))/(26.1+WECR+PW))
450 REM
480 REM LOOP FOR ITTERATING ON PERIOD
490 FOR TAU = 12 TO 3 STEP -.1
500 REM LOOP FOR ITTERATING ON ROTATION ANGLE
510 FOR THT = 0 TO PI/6 STEP PI/180
520 REM TL = TORQUE LOAD
530 TL = DY*(PW+WECR+26.1)*SIN(THT)
540 REM
550 REM TS = STALL TORQUE OF THE 300 OZ IN STEPPING MOTOR
570 TS = 300/16
580 REM OMEG = CIRCULAR FREQUENCY (RAD/SEC)
590 OMEG = PI*2/TAU
600 REM AM = AMPLITUDE OF OSCILLATION
610 AM = PI/6
620 REM TIM = TIME AT ANGLE
630 TIM = ATN((THT/AM)/SQR((-1*THT*THT)/(AM*AM)+1))/OMEG
640 REM DTHT = ANGULAR VELOCITY
650 DTHT = AM*OMEG*COS(OMEG*TIM)
660 REM TM = TORQUE SUPPLIED BY MOTOR
670 TM = (1-(DTHT/.523))*TS
680 REM TA = TORGUE AVAILABLE AFTER GEAR REDUCTION
690 TA = TM*.622*60
700 ANG = THT*180/PI
710 IF (TA-TL) < 24 THEN 740
720 NEXT THT
730 NEXT TAU
740 LPRINT USING
        " ###.#    ##.##    ##.#   ##.#    ###.#     ###.# ";
```

```
          PW,DY,TAU,ANG,TL,TA
750 LPRINT
760 NEXT PW
770 END
```

- Results of the roll axis analysis.

PATIENT WEIGHT	Y-C.G. DIL	PERIOD (SEC)	ANG (DEG)	TOR.LOAD IN*LB	TOR.DRIVE IN*LB
150.0	3.37	7.9	16.0	205.5	228.4
140.0	3.22	7.6	14.0	164.7	187.5
130.0	3.06	7.3	11.0	117.5	138.8
120.0	2.88	7.1	9.0	86.1	108.3
110.0	2.68	7.0	9.0	76.0	99.9
100.0	2.46	6.8	6.0	44.0	65.5
90.0	2.21	6.7	4.0	24.8	48.6
80.0	1.92	6.6	3.0	15.2	36.2
70.0	1.60	6.6	4.0	15.7	38.8
60.0	1.23	6.5	0.0	0.0	22.6
50.0	0.79	6.5	0.0	0.0	22.6
40.0	0.28	6.5	0.0	0.0	22.6
30.0	0.34	6.5	0.0	0.0	22.6
20.0	1.09	6.5	0.0	0.0	22.6
10.0	2.03	6.5	0.0	0.0	22.6
0.0	3.23	6.6	4.0	16.0	38.8
-10.0	4.82	6.6	3.0	15.4	36.2
-20.0	7.03	6.7	4.0	25.1	48.6
-30.0	10.32	6.8	6.0	44.4	65.5
-40.0	15.73	7.0	9.0	76.5	99.9

APPENDIX K

Gearbox Stress Calculations

Bearing Load Calculations

1/2 inch Worm Shaft Bearing:

The 1/2 inch worm shaft bearings of the pitch axis receive a high thrust load. The pitch axis has a maximum moment of 300 in*lb under dynamic operating conditions. This moment is transmitted to the worm shaft as an axial load. The radius of the gear is 2.5 inches. The magnitude of the axial load is 120 lbs. The thrust load rating of the bearing is 200 lbs.

1 inch Roll Shaft Bearing:

The roll shaft bearings receive a high radial load due to the cantilevered roll axis. The worst case static moment about the roll axis is 7,000 in*lb. This exists only during static conditions of patient loading and unloading. The bearings of the roll axis joint are placed 2.5 inches to either side of the roll joint centerline. The bearing load is analyzed as a force couple. The calculated radial load is:

$$\text{Radial Load} = 7{,}000 / (2 * 2.5) = 1{,}400 \text{ lbs}$$

The static radial load rating of the 1 inch bearing is 1,400 lbs. It must be noted the calculation is an absolute worst case. The dynamic moment imposed on the roll axis is 3,650 in*lb. The bearing radial load for this operating condition is 730 lbs. The 100 RPM load rating of the bearing is 905 lbs.

3/4 inch Yaw Shaft Bearing:

The yaw shaft bearings receive a thrust load due to the weight of the patient / chair / enclosure assembly. The maximum weight of the assembly is 110 lbs. The thrust load rating of the bearing is 280 lbs. This is a factor of safety of 2.5 .

Roll Pin Shear Torque

The roll-pins used throughout the device are 1/4 inch diameter by 2 inches long. Roll-pins of this size have a minimum double shear strength of 7,700 lbs. For a 1 inch diameter shaft this makes the torque required to shear the joint:

Torque = 7,700 lbs * .5 inches = 3,350 in * lb

The pitch axis joint receives the highest torsional force of approximately 2,000 in * lb during patient loading. This is a factor of safety of 1.68 .

Gear Stress Analysis

Torque of gear at failure of tooth profile is recommended by Boston Gear to equal 3 * the 100 RPM Torque rating.

100 RPM torque = 1,660 in * lb

Torque at failure = 5,000 in * lb

Maximum torque imposed on any axis is 2,000 in * lb

This is a factor of safety of 2.5 .

Shaft Stress Analysis

The most severly loaded shaft is the pitch axis. This shaft recieves a torsional load during patient loading of approximately 2,900 in*lb. The basic stress equation is:

$$Tau = 16 * Torque / (3.14159 * (diameter)^3)$$

The shaft has a 1/4 inch hole for the connecting rollpin which generates a stress concentration of 1.45. The shear stress on the shaft is therefore:

$$Tau = (16 * 2,900 / (3.14159 * (1)^3)) * 1.45$$

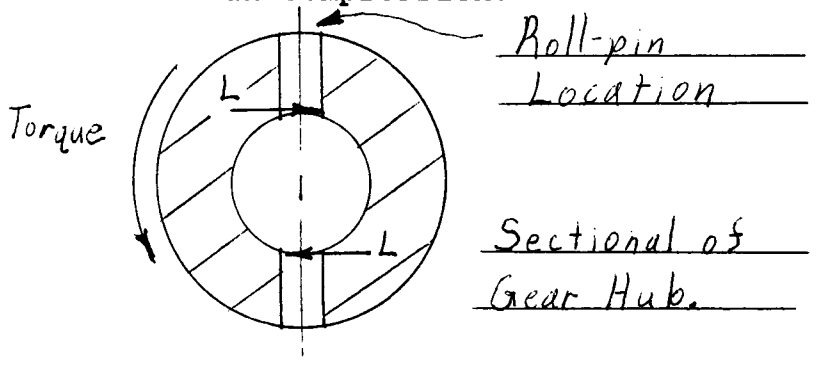
$$Tau = 21,000 \text{ psi}$$

The tensile yield of the shaft material is 90,000 psi.

The Factor of Safety is 90,000 / 2 / 21,000 = 2.14

Gear Hub Compression at Rollpin

The gear transmits torque to the gearbox output shafts by means of a rollpin connection. The Gear bore is 1 inch, the rollpin is 1/4 inch in diameter and the outer diameter of the gear hub is 2 inches. A contact area of 90 degrees as shown in the fiqure will be assumed. We first calculate the area of bronze in compression:



Area = 2 * .5 * cos(45) * .25 = .177 in^2

We know consider the load as acting on the inner radius of the gear hub. The compression strength of the bronze used is 20,000. This makes the allowable torque the joint may carry:

Torque = Area * Stress * Radius

Torque = .177 * 20,000 * .5 = 1,770 in*lb

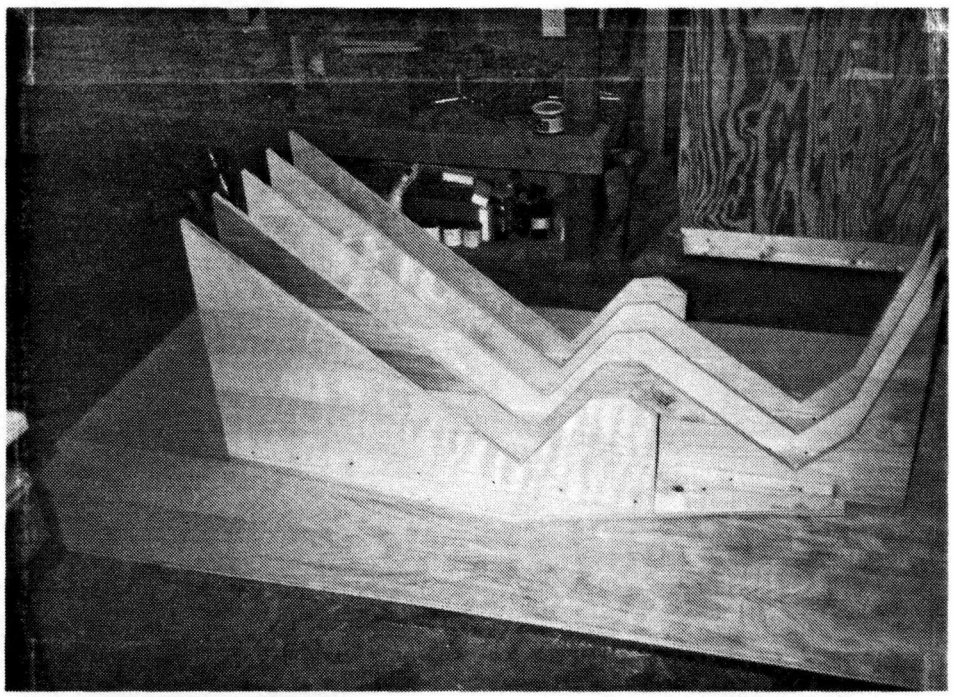
In this conservative analysis the gear bore may carry 1,770 in*lbs. Under severe patient loading conditions the torque about the joint may be 2,000 in*lbs. The Gear hubs will need to incorporate a steel insert as noted in section 5.4 .

APPENDIX L

Enclosure Mold Construction

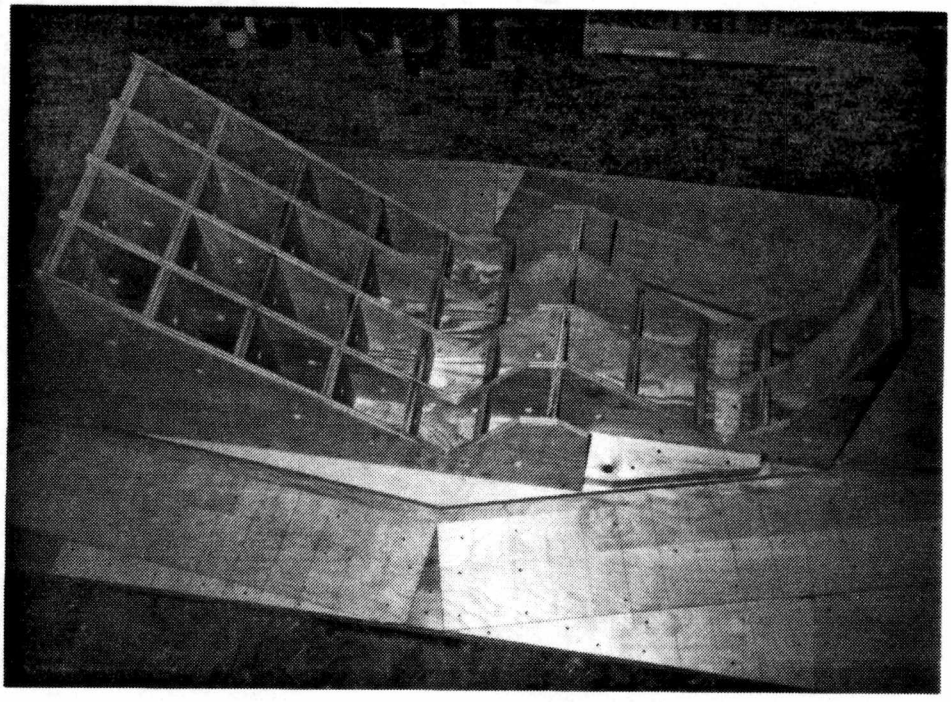
The following chronicles the major steps in the mold construction:

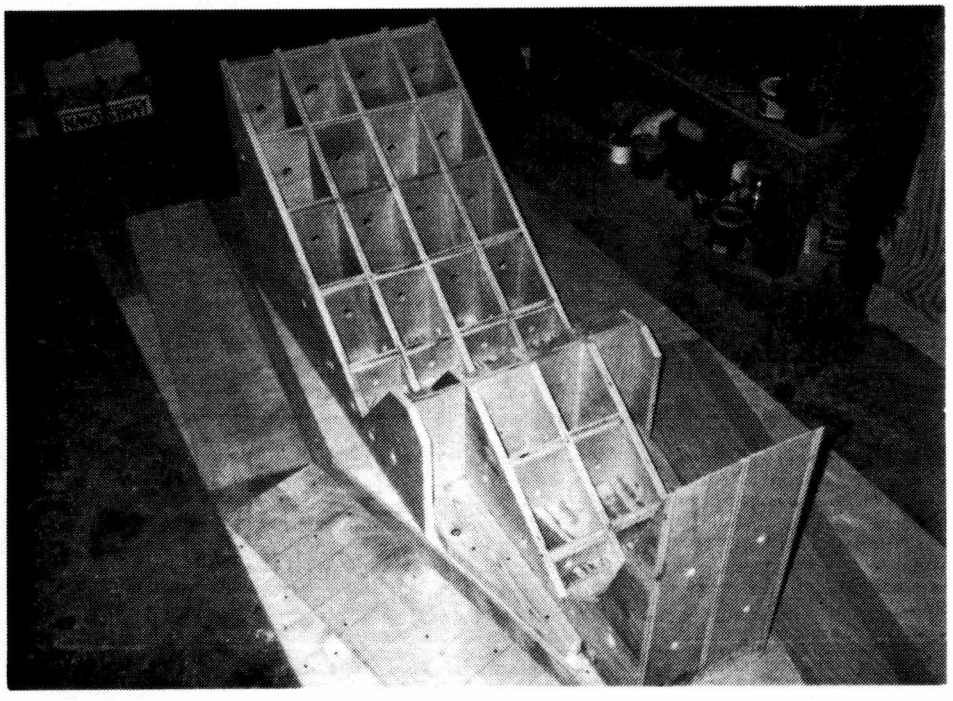
- The main structural ribs were positioned and braced on a sheet plywood base. The ribs were placed on 6 inch centers. The base was centered on a 4 * 8 feet sheet.



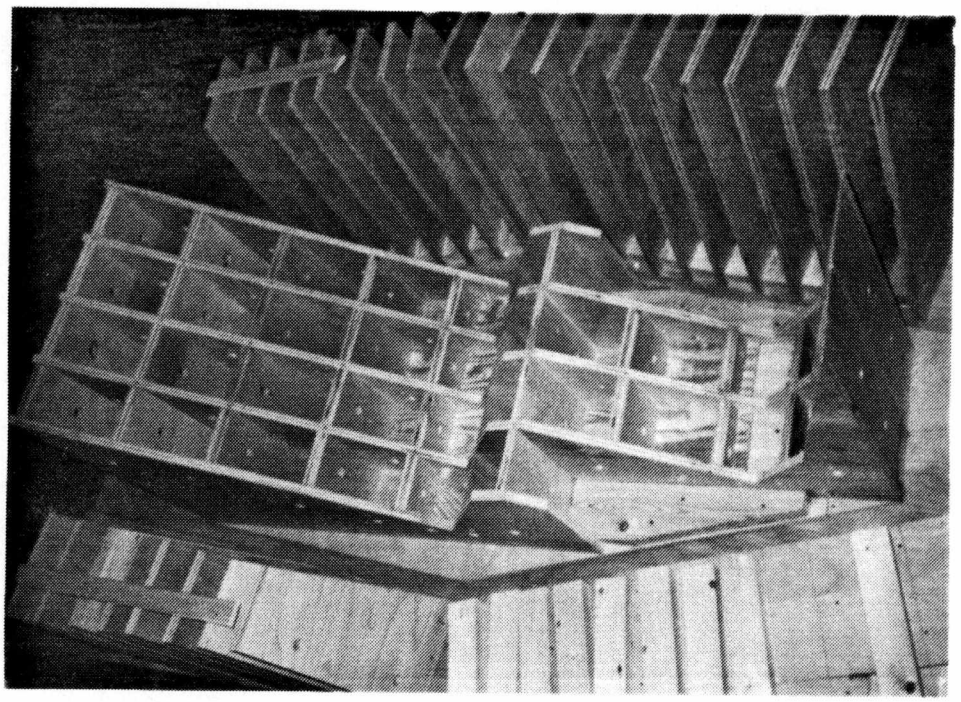


- The main ribs were internally braced using plywood spacers on 6 inch centers. A second layer of plywood has been added to the base sheet around the periphery of the base plate.



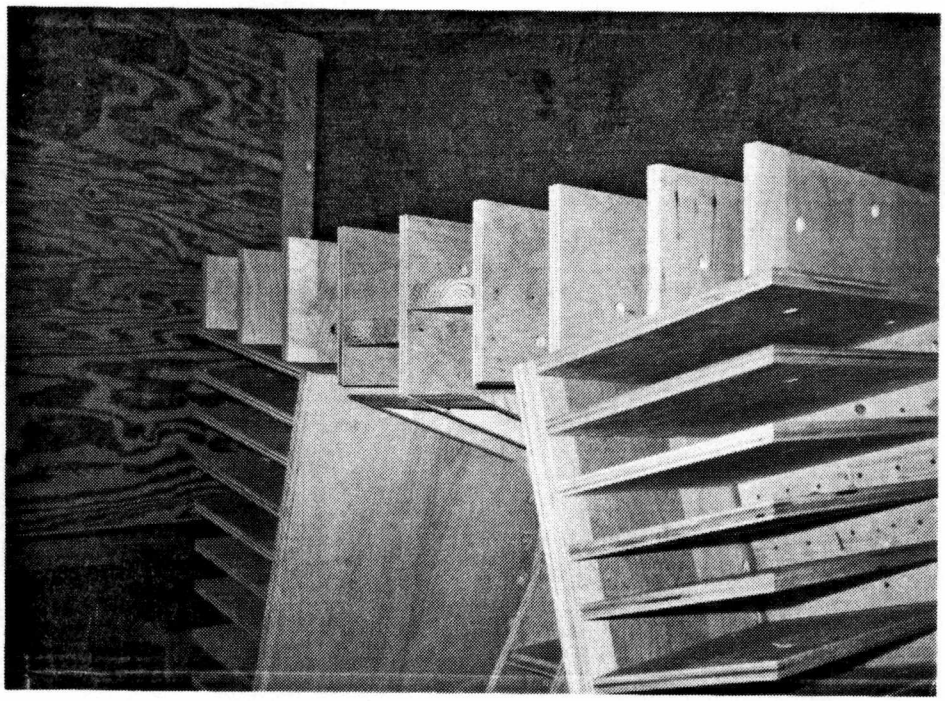


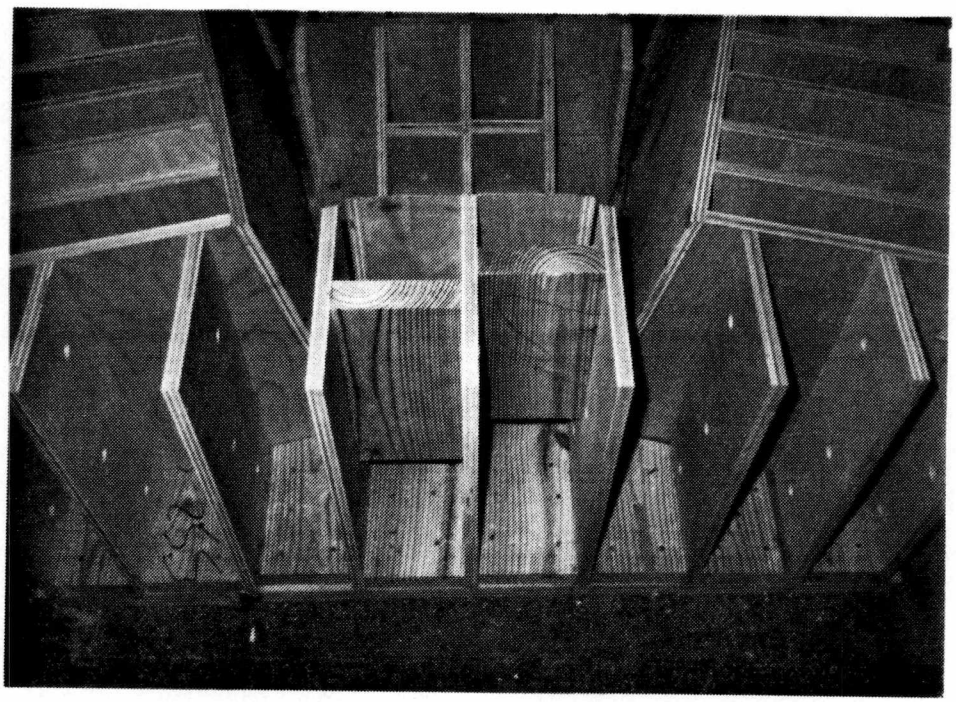
- Side ribs were positioned and braced about the main cavity. Each rib is unique and was individually measured and cut for its position. The ribs are placed on 3 inch centers.



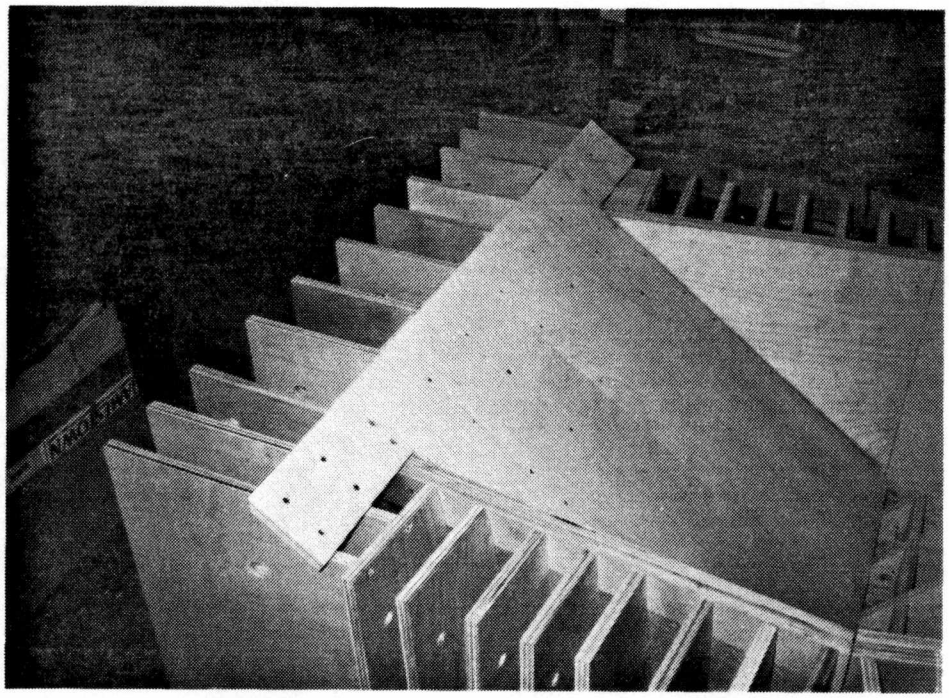


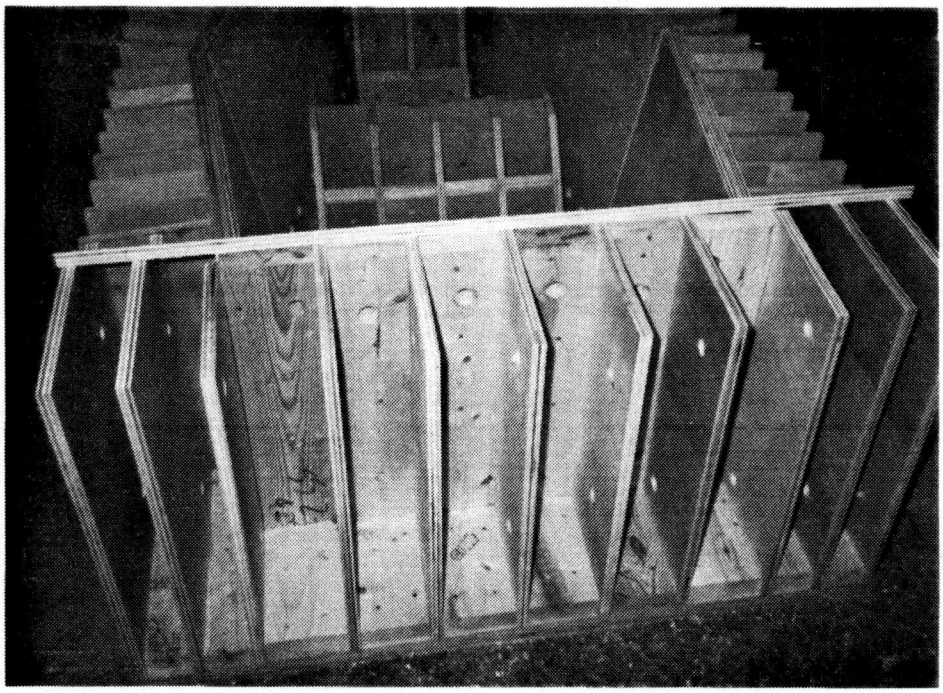
- Ribs were placed on 6 inch centers at the foot of the mold. Plywood sheet that forms the sides of the cavity is shown mounted to the side rib faces.



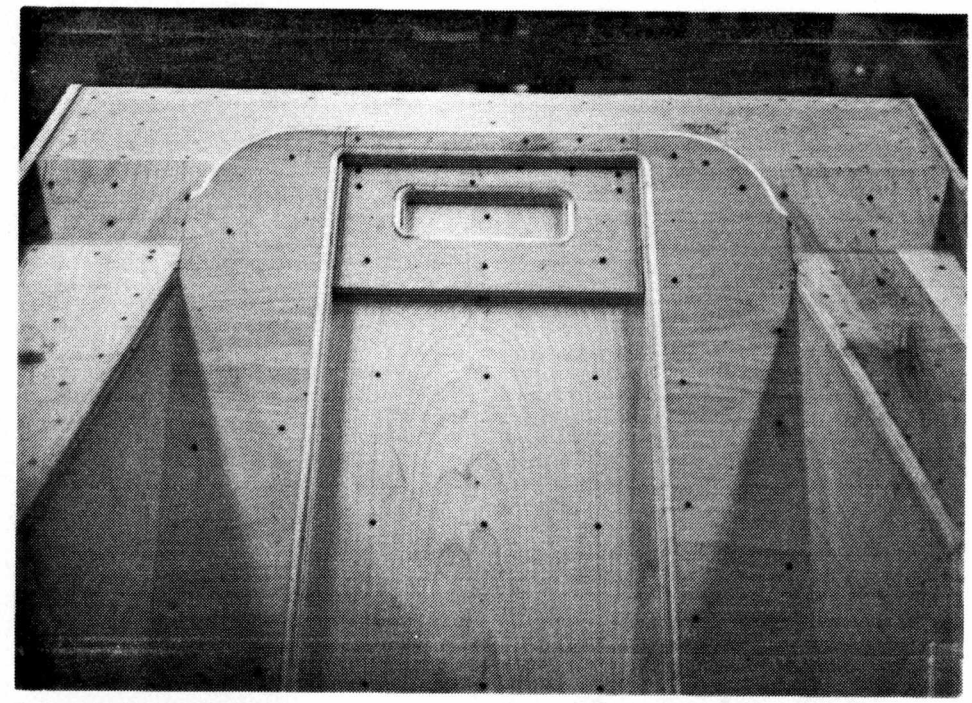


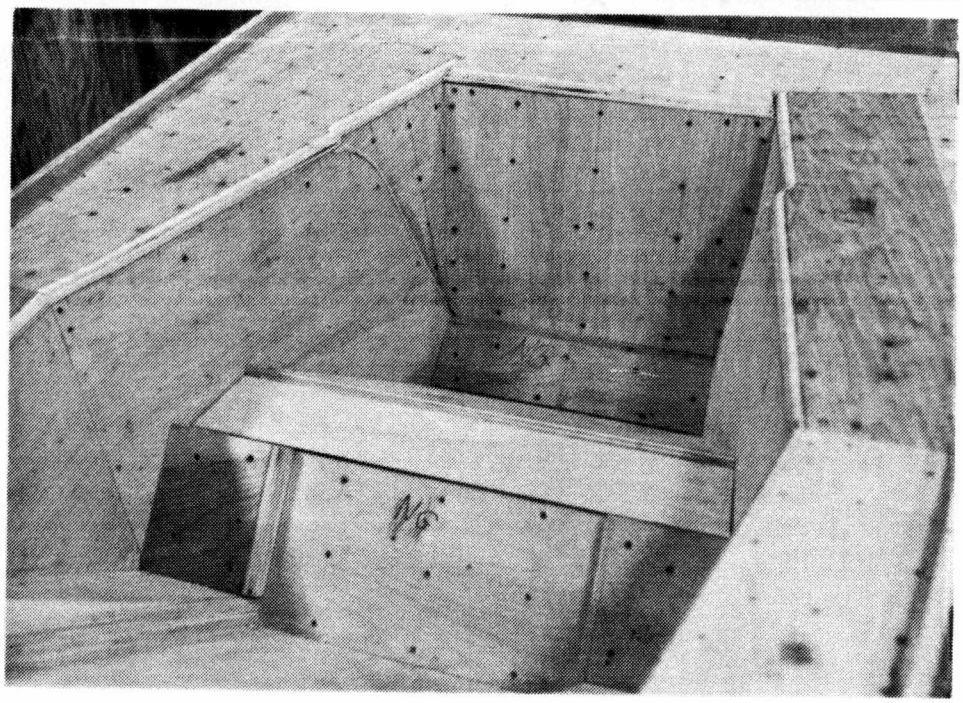
- Ribs were placed on 6 inch centers at the top of the mold. The sides of the mold cavity are shown receiving a double thickness of plywood sheet. The first layer of the backrest plate has been fastened to the rib faces and



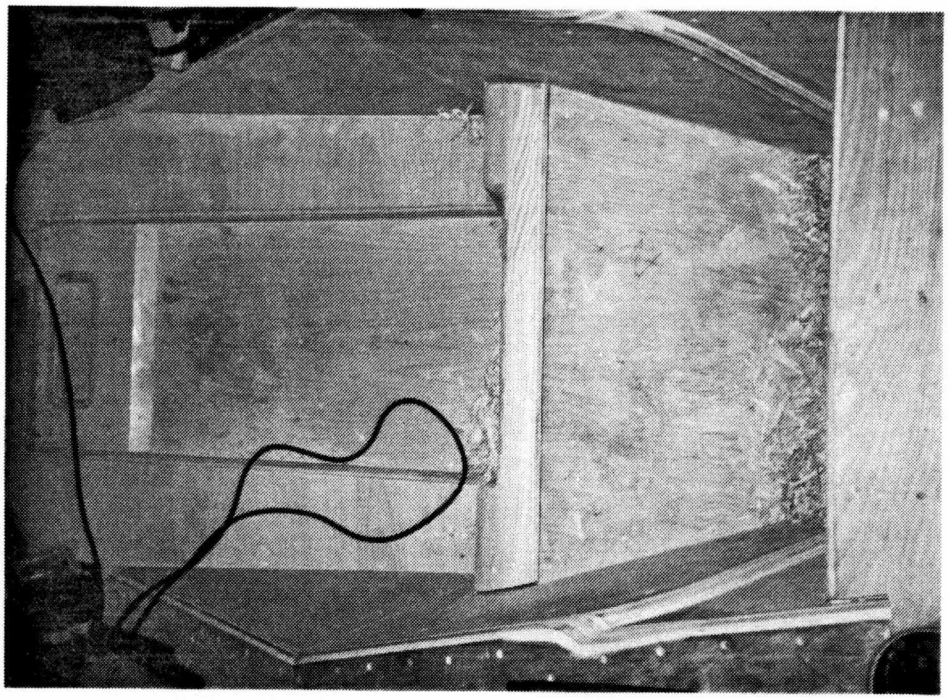


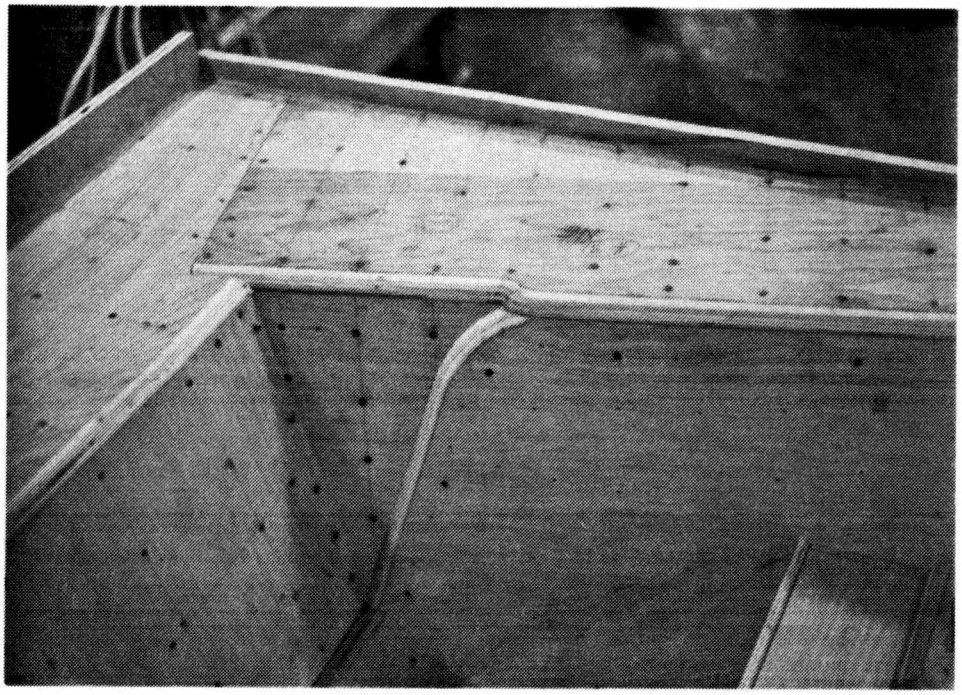
- The interior detail of the cavity was built-up from sheet plywood and pine lumber. One layer of plywood decking was added to the tops of the ribs about the cavity. Sheet plywood sides were also added to the exterior of the mold.





- To complete the mold all corners were given a generous radius.





APPENDIX M

Parts List and Cost Analysis

Device Parts List

Part #	Part	Source
1	Pitch Frame	Plate 1, 3
2	Roll Frame	Plate 2
3	Left Foundation Frame	Plate 4A
4	Right Foundation Frame	Plate 4B
5	Enclosure	Plate 5
6	Tumble Forms Chair: Small Medium Large	Therapeutic Equipment Corp. Part # 0641S Part # 0641M Part # 0641L
7	Foundation Cross Braces	Plate 6
8	Corner Brackets	Plate 7
9	Yaw Pedestal	Plate 8
10	Roll Gearbox Mount	Plate 9
11	Roll Bearing Mount	Plate 10
12	Pitch Axis Idle Shaft	Plate 20
13	Yaw Axis Spacers	Plate 11
14	Pitch Axis Pillow Block Bearing	Fafnir Part # RAK 1
15	Yaw Axis Flanged Bearing	Fafnir Part # VFMST 1
16	Vinyl Tubing End Caps	StockCap SC 3.000-1-1/2

Part #	Part	Source
Gearboxes		
17	Housing	Plate 12, 13
18	Front End-plate	Plate 14, 16
19	Back End-plate	Plate 15, 16
20	Gear Single Thread	Plate 17 Boston Gear Part # GB1260A
21	Gear Double Thread	Plate 18 Boston Gear Part # DB1260A
22	Worm Single Thread	Boston Gear Part # H1056RH
23	Worm Double Thread	Boston Gear Part # H1407RH
24	Worm Shaft	Plate 19
25	Pitch Axis Shaft	Plate 20
26	Roll Axis Shaft	Plate 21
27	Yaw Axis Shaft	Plate 22
28	1 inch Bearing	Boston Gear Part # 1641DC
29	3/4 inch Bearing	Boston Gear Part # 1638DC
30	1/2 inch Bearing	Boston Gear Part # 1621DC
31	Motor Mount Plate	Plate 23
32	Motor 300 oz in	Superior Electric Part # M092
33	Motor 450 oz in	Superior Electric Part # M093

Part #	Part	Source
34	Coupling	Helical Products Co. Part # AR100-16-12
35	Worm Shaft Seal	C R Industries Part #
36	Worm Shaft Spacer	Plate 24
37	Motor Spacer	Plate 25
38	Worm Shaft Key	1/8 square, 1 inch long
39	Roll Pin	1/4 dia., 2 inch long
Fasteners		
40	Screw, 1/4 NC Socket Head Cap, 3/4 Long (End-plate to Housing, Motor Mount to Housing)	
41	Screw, 3/8 NC Socket Head Cap, 1.0 Long (Pitch Axis Bearing to Foundation Frame)	
42	Screw, 3/8 NC Socket Head Cap, 2.5 Long (Foundation to Cross-braces, Pitch Gearbox to Foundation Frame)	
43	Screw, 3/8 NC Socket Head Cap, 4.0 Long (Roll Axis Joint)	

Part #	Part	Source
44	Screw, 3/8 NC Socket Head Cap, 5.0 Long (Yaw Gearbox to Frame)	
45	Screw, 3/8 NC Socket Head Cap, 6.0 Long (Roll Gearbox to Frame)	
46	Screw, 3/8 NC Button Head Cap (Enclosure to Yaw Pedestal)	
47	Nut, 3/8 NC Nylon Insert (Foundation to Cross-braces, Enclosure to Yaw Pedestal)	

Device Costs

Part #	Part	Individual Cost	Qty.	Cost per Chair
1	Pitch Frame	140.00	1	140.00
2	Roll Frame	88.00	1	88.00
3	Left Foundation Frame	120.00	1	120.00
4	Right Foundation Frame	120.00	1	120.00
5	Enclosure	800.00	1	800.00
6	Tumble Forms Chair:			480.00
	Small	472.50	1	
	Medium	479.50	1	
	Large	549.50	1	
7	Foundation Cross Braces	86.00	2	86.00
8	Corner Brackets	84.00	4	84.00
9	Yaw Pedestal	42.00	1	42.00
10	Roll Gearbox Mount	45.00	1	45.00
11	Roll Bearing Mount	47.50	1	47.50
12	Pitch Axis Idle Shaft	10.00	1	10.00
13	Yaw Axis Spacers	14.00	2	14.00
14	Pitch Axis Pillow Block Bearing	20.95	1	20.95
15	Yaw Axis Flanged Bearing	14.02	1	14.02
16	Vinyl Tubing End Caps	0.35	6	2.10

Part #	Part	Individual Cost	Qty.	Cost per Chair
Gearboxes				
17	Housing	75.50	3	226.50
18	Front Endplate	40.70	3	122.10
19	Back Endplate	40.70	3	122.10
20	Gear Single Thread	75.00	2	150.00
21	Gear Double Thread	73.50	1	73.50
22	Worm Single Thread	23.50	2	27.00
23	Worm Double Thread	25.07	1	25.07
24	Worm Shaft	8.50	1	8.50
25	Pitch Axis Shaft	10.00	2	20.00
26	Roll Axis Shaft	10.50	1	10.50
27	Yaw Axis Shaft	15.50	1	15.50
28	1 inch Bearing	9.24	5	46.50
29	3/4 inch Bearing	9.57	2	19.14
30	1/2 inch Bearing	6.31	6	37.86
31	Motor Mount Plate	35.00	3	105.00
32	Motor 300 oz in	180.00	2	360.00
33	Motor 450 oz in	210.00	1	210.00
34	Coupling	11.77	3	35.31
35	Worm Shaft Seal	3.25	3	9.75
36	Worm Shaft Spacer	6.00	6	36.00
37	Motor Spacers	9.50	3	28.50
38	Worm Shaft Key	0.15	3	0.45
39	Roll Pin	0.20	6	1.20

Part #	Part	Individual Cost	Qty.	Cost per Chair
Fasteners				
40	Screw, 1/4 NC Socket Head Cap, 3/4 Long	9.60 /100	36	3.45
41	Screw, 3/8 NC Socket Head Cap, 1.0 Long	27.20 /100	2	0.45
42	Screw, 3/8 NC Socket Head Cap, 2.5 Long	52.36 /100	18	9.45
43	Screw, 3/8 NC Socket Head Cap, 4.0 Long	98.00 /100	1	0.98
44	Screw, 3/8 NC Socket Head Cap, 5.0 Long	137.0 /100	2	2.74
45	Screw, 3/8 NC Socket Head Cap, 6.0 Long	209.8 /100	2	4.20
46	Screw, 3/8 NC Button Head Cap	121.1 /100	6	7.27
47	Nut, 3/8 NC Nylon Insert	5.48 /100	24	1.32

Total Mechanical Device Cost (less Computer, Computer Program, Power Supplies, and Electronic Motor Drivers) **3,833.76**

- Device priced as supplied with 1 Tumble Forms Chair, additional seats extra.

Machining Time Estimates and Material Cost of Device Components

Part #	Part	Material Cost $	Mach. Time (hrs)	Cost per Part $
1	Pitch Frame	7.18 /ft.	2.0	140.00
2	Roll Frame	7.18 /ft.	1.5	88.00
3	Left Foundation Frame	5.52 /ft.	2.0	120.00
4	Right Foundation Frame	5.52 /ft.	2.0	120.00
7	Foundation Cross Braces	5.52 /ft.	0.5	43.00
8	Corner Brackets	6.00	0.5	21.00
9	Yaw Pedestal	12.00	1.0	42.00
10	Roll Gearbox Mount	15.00	1.0	45.00
11	Roll Bearing Mount	10.00	1.25	47.50
12	Pitch Axis Idle Shaft	5.74 /ft.	0.25	10.00
13	Yaw Axis Spacers	1.00	0.20	7.00
Gearboxes				
17	Housing	14.50	2.0	74.50
18	Front Endplate	10.70	1.0	40.70
19	Back Endplate	10.70	1.0	40.70
20	Gear Single Thread	59.39	0.5	75.00
21	Gear Double Thread	66.00	0.25	73.50
24	Worm Shaft	1.65 /ft.	0.25	8.50
25	Pitch Axis Shaft	5.74 /ft.	0.25	10.00

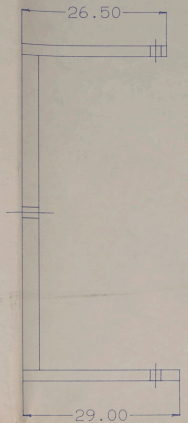
Part #	Part	Material Cost $	Mach. Time (hrs)	Cost per Part $
26	Roll Axis Shaft	5.74 /ft.	0.25	10.50
27	Yaw Axis Shaft	3.29 /ft.	0.25	15.50
31	Motor Mount Plate	5.00	1.0	35.50
36	Worm Shaft Spacer	2.68	0.10	6.00
37	Motor Spacer	3.50	0.20	9.50

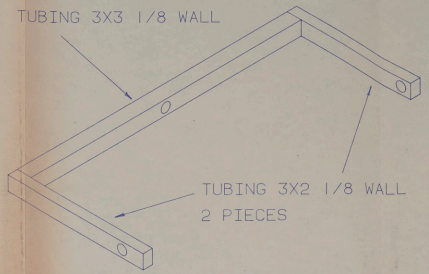
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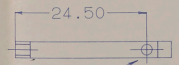
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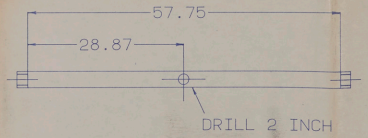
E. J. Asbury III was born on October 7, 1964 in Knoxville, Tennessee. He graduated from Karns High School in Knoxville in 1982 and enrolled at the University of Tennessee in the Fall of 1982. A participant in the Cooperative Engineering Program at Newport News Shipbuilding, in Newport News, Va where he was involved in the overhaul of Nimitzs class aircraft carriers. He received a Bachelor of Science degree in Mechanical Engineering in the Summer of 1987. In the Fall of 1987 he entered graduate school in Mechanical Engineering with an emphasis in product design and development. As a graduate teaching assistant for the Engineering Science and Mechanics Department he was an instructor of engineering graphics, and for the Mechanical Engineering Department he was an instructor of an experimental methods laboratory. His graduate course work included advanced courses in Design, Dynamics, Vibrations, Strength of Materials, Manufacturing, and Finite Element Analysis. His masters work was done as an entrepreneurial partnership with Doctorates of Engineering, Frank Speckhart and James Euler and with Doctor of Neurology, Andrew Zimmerman.

He received his Master of Science in Mechanical Engineering in August 1990. Mr. Asbury's interests include volleyball, radio control models, and antique restoration.









PITCH AXIS FRAME

6063 ALUMINUM TUBE 1/8 WALL

T. I. G. WELDED USING 4043 ROD

V GROOVE WELD AS POSSIBLE

GRIND ALL WELDING BEADS FLUSH

ALL LINEAR TOLERANCES +/- 1/8 INCH

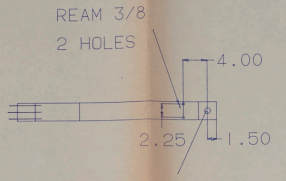
E. J. ASBURY

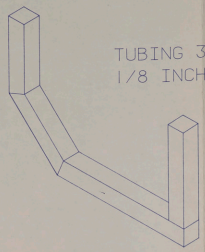
V. S. CHAIR

PN= PART #1 PN= PLATE 1

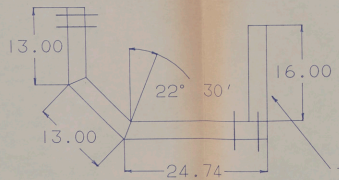
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DRAWN BY: E. J. ASBURY III
SECTION: PLATE 1
SCALE: XXXXXXXXXX DATE:5/10/90
PRODUCED ON CADAM

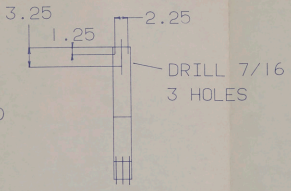
Library
The University of Tennessee
Knoxville





ROLL AXIS FRAME
6063 ALUMINUM TUBING
T. I. G. WELDED USING 4043 ROD
V GROOVE WELD AS POSSIBLE
GRIND ALL WELD BEADS FLUSH
LENGTH OF TUBE TOLERANCE +/- 1/8 INCH
HOLE SPACING TOLERANCE +/- 1/16
E. J. ASBURY
V. S. CHAIR





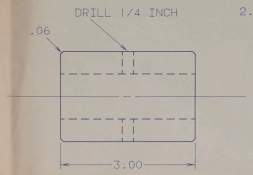
THIS TUBE NOT SHOWN IN RIGHT SIDE VIEW FOR CLARITY

ALL DIMENSIONS IN INCHES

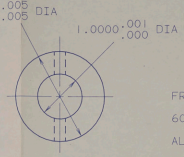
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DRAWN BY: E. J. ASBURY III
SECTION: PLATE 2
SCALE: XXXXXXXXX DATE:5/10/90
PRODUCED ON CADAM

Library
The University of Tennessee
Knoxville



ONE REQUIRED



FRAME INSERTS

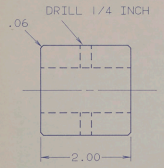
6061 OR 6063 ALUMINUM

ALL TOLERANCES +/- 1/32 UNLESS

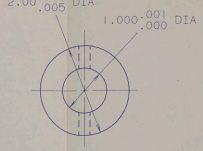
OTHERWISE NOTED

E. J. ASBURY

V. S. CHAIR



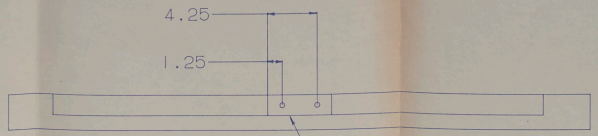
TWO REQUIRED

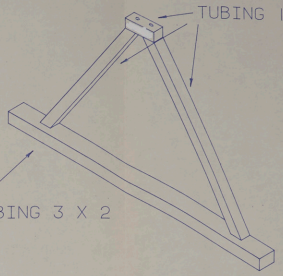


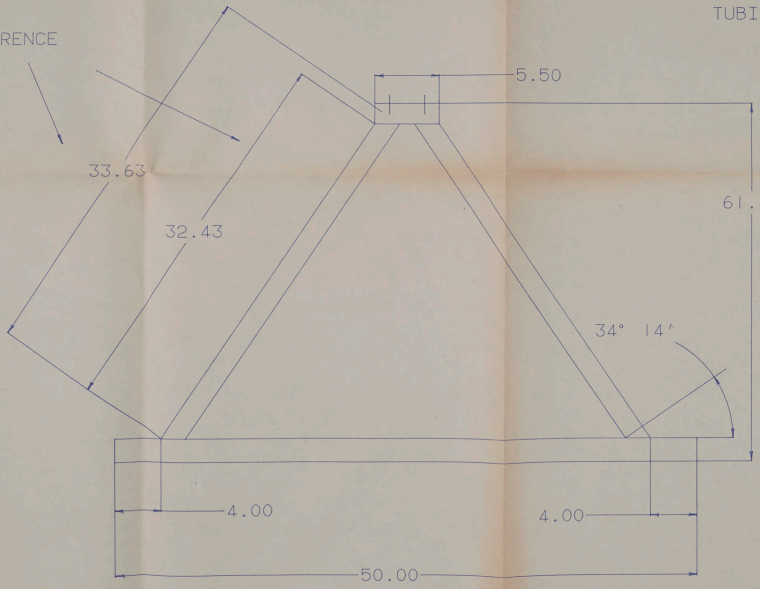
UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: FRAME INSERTS, PART# 1
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 3
SCALE: XXXXXXXXX DATE:5/10/90
PRODUCED ON CADAM

PN# PART# 1 PN# PLATE 3

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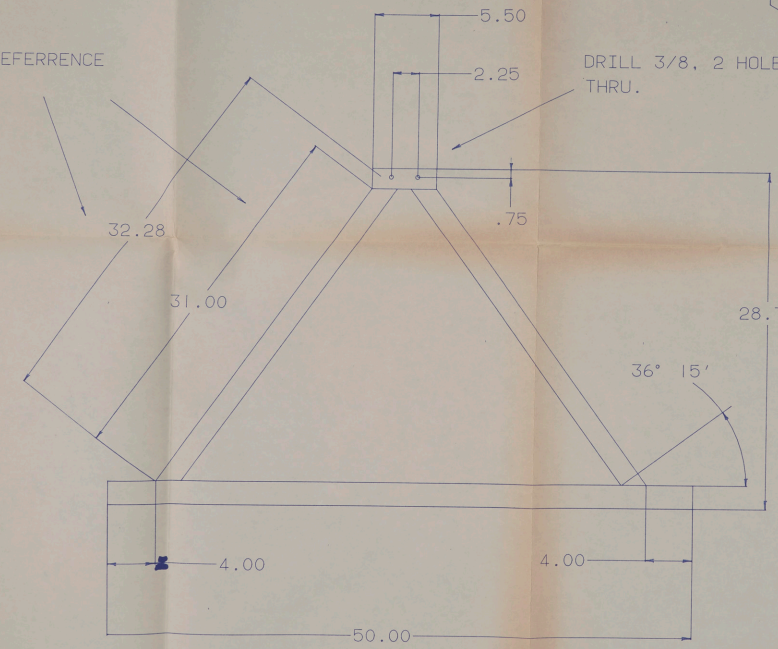


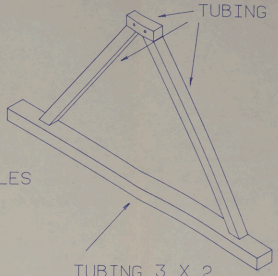
LEFT FOUNDATION FRAME
6063 ALUMINUM TUBING 1/8 WALL
1 REQUIRED
T. I. G. WELDED USING 4043 ROD
V GROOVE WELD AS POSSIBLE
ALL WELDS FULL CERCUMFERRENCE
E. J. ASBURY
V. S. CHAIR

UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: LEFT FOUNDATION FRAME, PART# 3
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 4A
SCALE: XXXXXXXXX DATE:5/10/90
PRODUCED ON CADAM

PN= PART# 3 PN= PLATE# 4A

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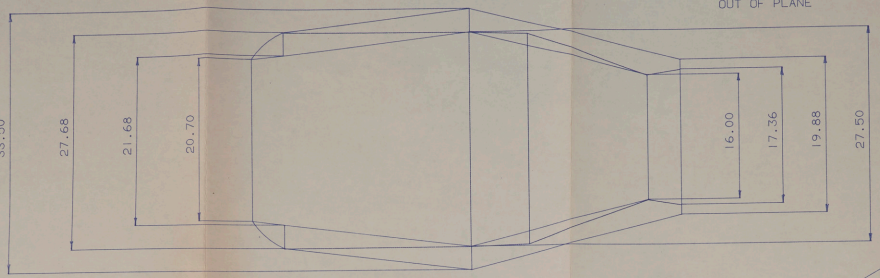
RIGHT FOUNDATION FRAME
6063 ALUMINUM TUBING 1/8 WALL
1 REQUIRED
T. I. G. WELDED USING 4043 ROD
V GROOVE WELD AS POSSIBLE
ALL WELDS FULL CERCUMFERRENCE
E. J. ASBURY
V. S. CHAIR

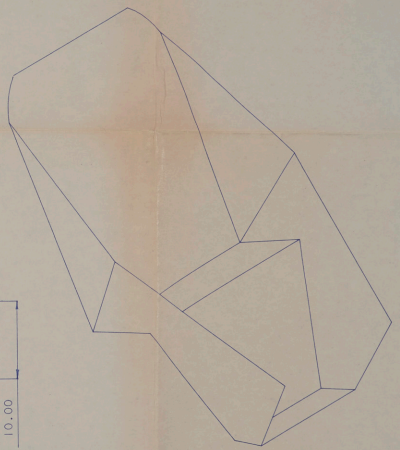
UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE:RIGHT FOUNDATION FRAME, PART# 4
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 4B
SCALE: XXXXXXXXX DATE:5/10/90
PRODUCED ON CADAM

FN= PART# 4 FN= PLATE# 4B

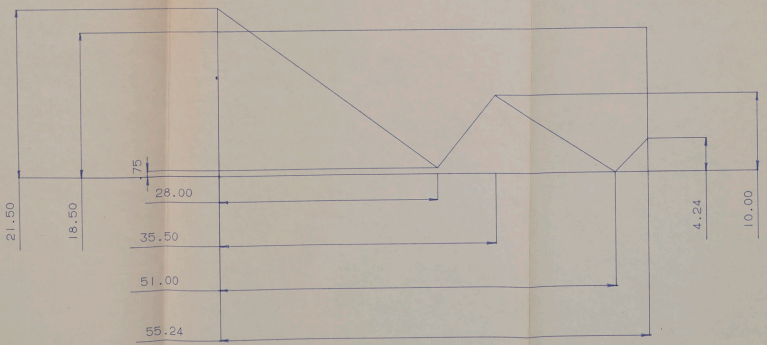
Library
The University of Tennessee
Knoxville

DASHED LINES ARE
OUT OF PLANE





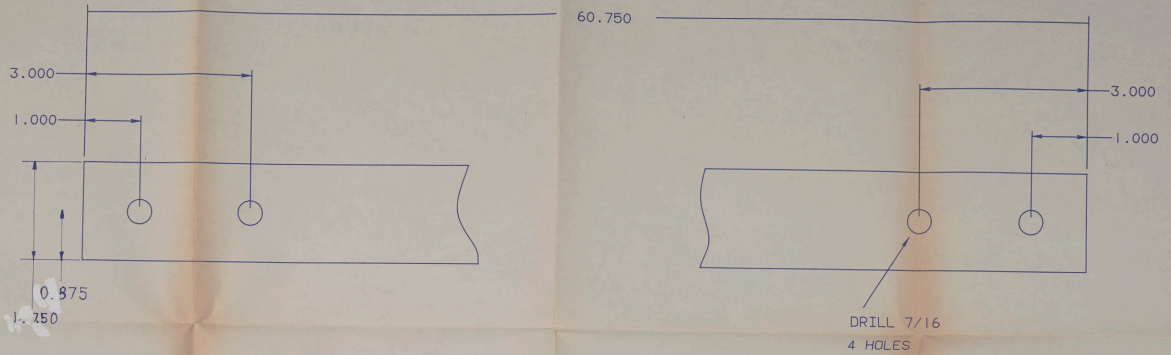
ISOMETRIC NOT TO SCALE



PN= PART# 5 PN= PLATE 5

UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: ENCLOSURE MOLD DIMEN., PART# 5
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 5
SCALE: XXXXXXXXX DATE:5/10/90
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FOUNDATION CROSS BRACES

6063 ALUMINUM TUBE

1.75 * 3.0 INCH TUBE

2 REQUIRED

ALL TOLERANCES +/- 1/16

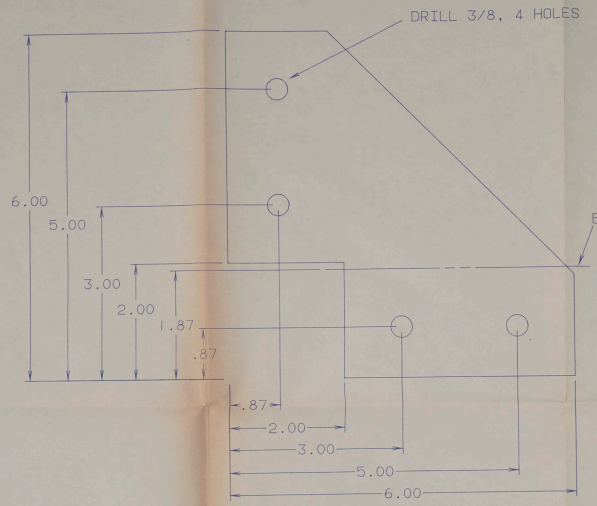
E. J. ASBURY

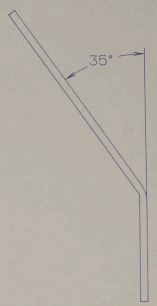
V. S. CHAIR

FN= PART# 7 FN= PLATE 6

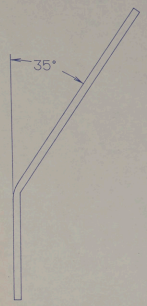
UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: FOUNDATION CROSS BRACES, PART# 7
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 6
SCALE: XXXXXXXXX DATE: 5/10/90
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LEFT BRACKET



RIGHT BRACKET

FOUNDATION CORNER BRACKETS

1/8 INCH ALUMINUM PLATE

4 BRACKETS REQUIRED, 2 RIGHT HAND, AND 2 LEFT HAND

CUT TO THE DIMENSIONS SHOWN AND THEN BEND TO 35 DEG ANGLE

BEND ARC NOT TO EXCEED 1/8 INCH TO EITHER SIDE OF BEND LINE

FILE ALL SHARP EDGES SMOOTH

ALL LINEAR TOLERANCE +/- 1/16

ALL ANGULAR TOLERANCES +/- 2 DEGREES

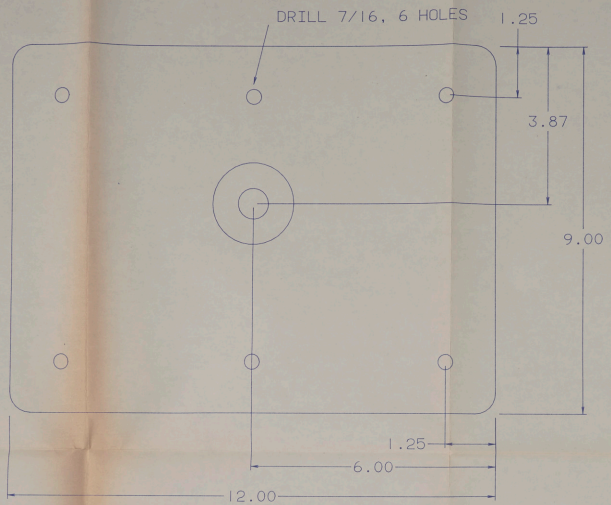
E. J. ASBURY

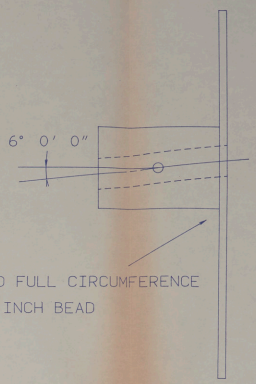
V. S. CHAIR

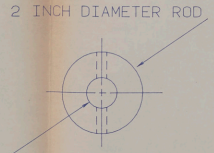
PN= PART# 8 FN= PLATE 7

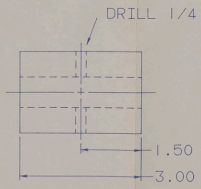
UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: CORNER BRACKETS, PART# 8 DRAWN BY: E. J. ASBURY III SECTION: PLATE 7 SCALE: XXXXXXXXXX DATE:5/10/90
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ALL MATERIAL 6063 ALUMINUM
RADIUS ALL SHARP CORNERS
WELD USING 4043 ROD
1 REQUIRED

YAW AXIS PEDESTAL MOUNT
E. J. ASBURY
V. S. CHAIR

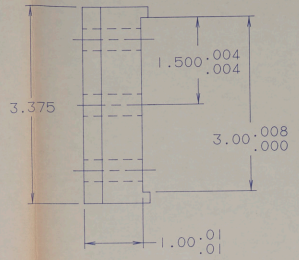
NOTE: CENTERLINE OF 3/4 INCH HOLE MUST PASS THROGH
CENTERLINE OF 1/4 INCH HOLE.

PN= PART# 9

PN= PLATE 8

UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: YAW PEDESTAL, PART# 9
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 8
SCALE: XXXXXXXXX DATE:5/10/90
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ROLL GEARBOX MOUNT

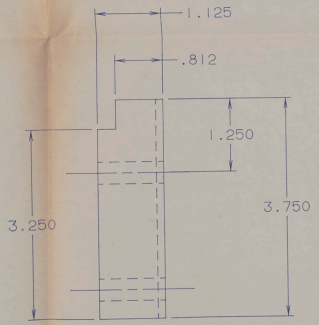
1 INCH ALUMINUM PLATE, 6063

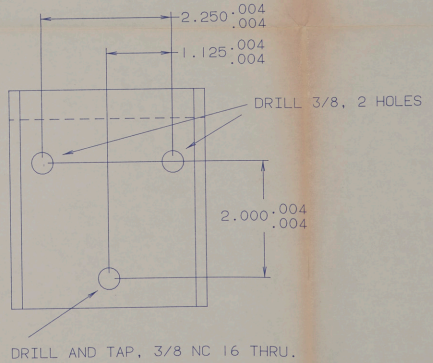
1 REQUIRED

ALL TOLERANCES +/-.05 UNLESS OTHERWISE NOTED

E. J. ASBURY

V. S. CHAIR

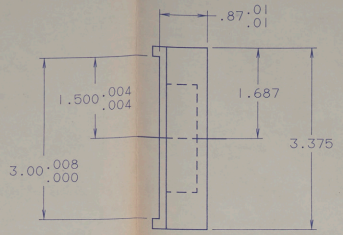




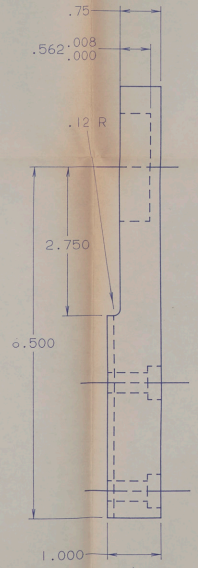
PN= PART# 10 FN= PLATE 9

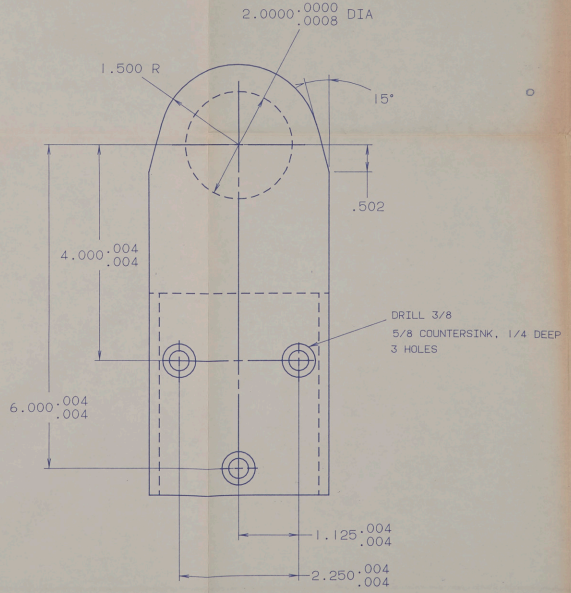
UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: ROLL GEARBOX MOUNT, PART# 10
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 9
SCALE: XXXXXXXXX DATE:5/10/90
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BEARING MOUNT ROLL AXIS
1 INCH ALUMINUM PLATE, 6063
1 REQUIRED
ALL TOLERANCES +/-.05 UNLESS OTHERWISE NOTED
E. J. ASBURY
V. S. CHAIR



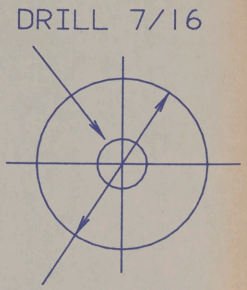


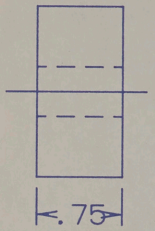
PN= PART# 11 FN= PLATE 10

UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.	
TITLE: ROLL BEARING MOUNT, PART# 11	
DRAWN BY: E. J. ASBURY III	
SECTION: PLATE 10	
SCALE: XXXXXXXXX	DATE: 5/10/90
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YAW AXIS SPACERS

ALUMINUM

2 REQUIRED

LINEAR TOLERANCE +/- 1/16

DIAMETRICAL TOLERANCE +/- 1/16

ALL DIMENSIONS INCHES

E. J. ASBURY

V. S. CHAIR

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UNIVERSITY OF TENNESSEE
CENTER FOR C.I.E.M.

TITLE:YAW AXIS SPACERS, PART# 13

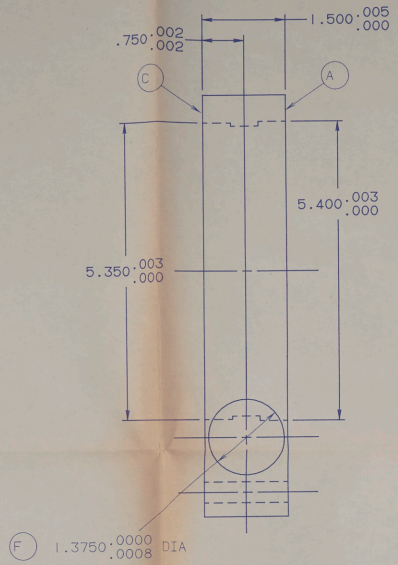
DRAWN BY:E. J. ASBURY III

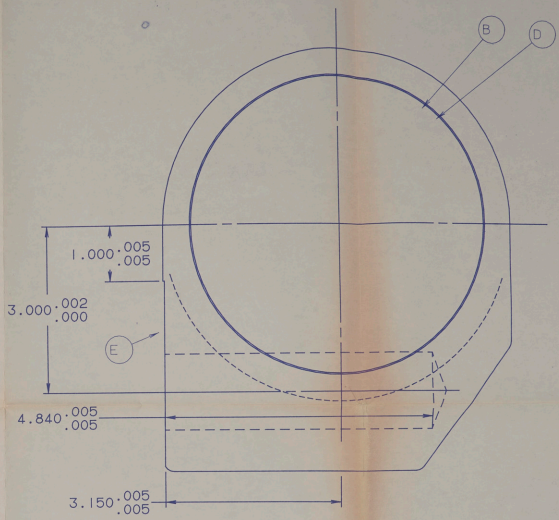
SECTION: PLATE 11

SCALE: XXXXXXXXXX DATE:5/10/90

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PN=PART# 13 FN=PLATE 11





MACHINING INSTRUCTIONS:

A -FACE ENDPLATE SEALING SURFACE RMS 200 OR FINER

B -CUT INNER BORE DIA. 5.400, +.003-.000 TO A DEPTH .500 +/-.01

C -FACE ENDPLATE SEALING SURFACE PARRALLEL TO SURFACE "A"
USING SURFACE "B" AS TRUE SURFACE UNTIL 1.500 +.005 -.000
SURFACE TO SURFACE CLEARANCE RESULTS

D -CUT INNER BORE DIA. 5.350, +.003 -.000 TO A DEPTH .500 +/-.01

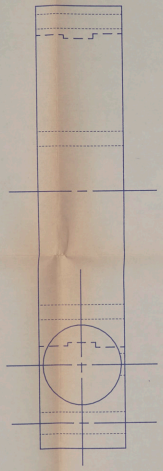
E -CUT SIDE OF HOUSING PERPENDICULAR TO SEALING
SURFACES "A AND C" AND PARRELLEL WITH SIDE OF CASTING

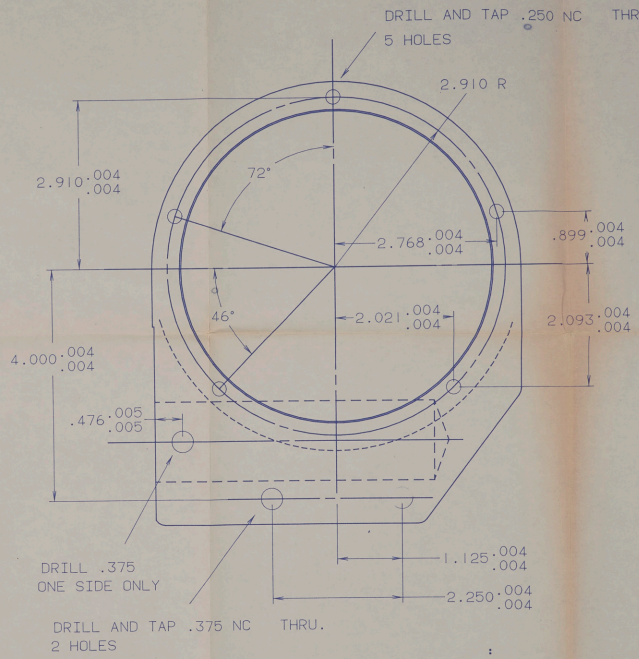
F -REAM HOLE 1.375 PERPENDICULAR TO FACED HOUSING SIDE
TO A DEPTH OF 4.840 +.005 -.005, FROM FACED SURFACE "E"

PN= PART# 17 FN= PLATE 12

UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: GEARBOX HOUSING, PART# 17
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 12
SCALE: XXXXXXXXXX DATE:5/10/90
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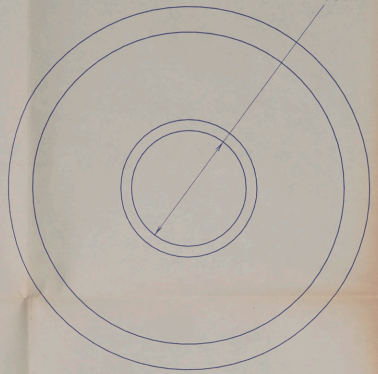


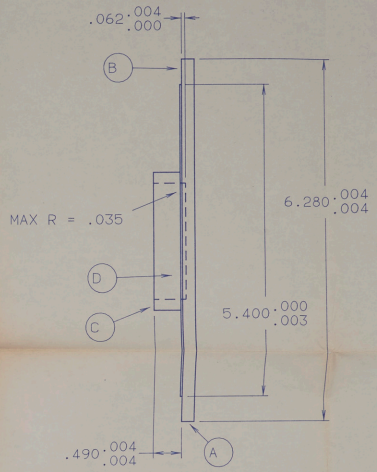


PN= PART# 17 FN= PLATE 13

UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: GEARBOX HOUSING, PART# 17
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 13
SCALE: XXXXXXXXX DATE:5/10/90
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E. J. ASBURY

V. S. CHAIR

FRONT ENDPLATE 1 INCH SHAFT

2 REQUIRED

MACHINING INSTRUCTIONS:

A -CUT OUTER SURFACE TO DIA 6.280

B -FACE .030 DEEP TO DIA 5.400

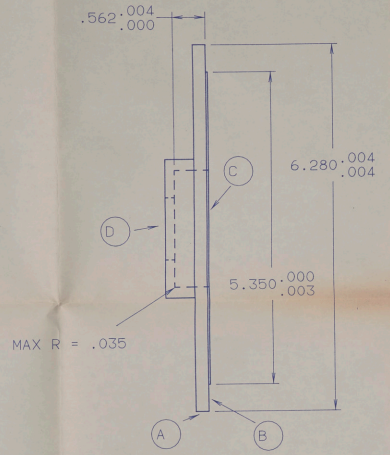
C -BORE CONCENTRIC TO DEPTH .062
FROM SURFACE "B"

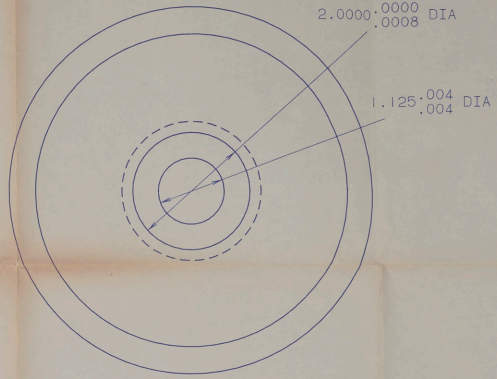
D -FACE TO .490 FROM SURFACE "B"

UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: FRONT END-PLATE, PART# 18
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 14
SCALE: XXXXXXXXX DATE:5/10/90
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PN= PART# 18 PN= PLATE 14

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E. J. ASBURY

V. S. CHAIR

BACK ENDPLATE 1 INCH SHAFT

3 REQUIRED

MACHINING INSTRUCTIONS:

A -CUT OUTER SURFACE TO DIA 6.280

B -FACE .030 DEEP TO DIA 5.350

C -BORE CONCENTRIC TO DEPTH .562
FROM SURFACE "B"

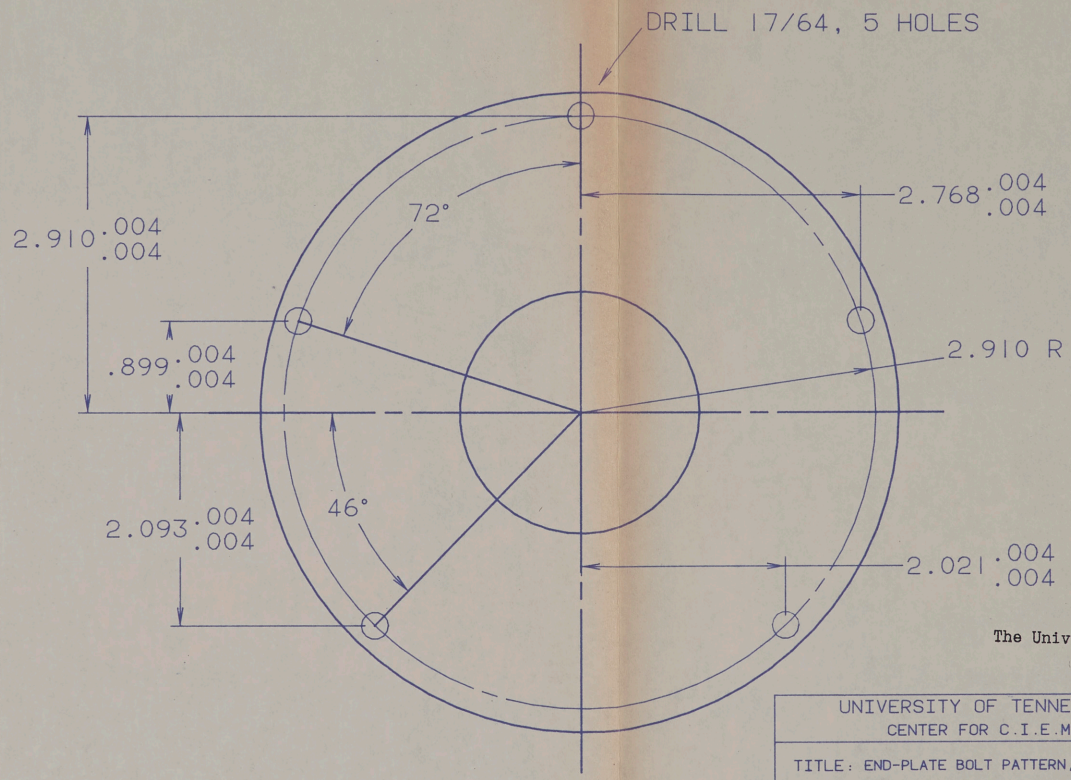
D -BORE 1.125

UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: BACK END-PLATE, PART# 19
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 15
SCALE: XXXXXXXXX DATE:5/10/90
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PN= PART# 19 FN= PLATE 15

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BOLT PATTERN ALL ENDPLATES
6 REQUIRED

PN= PART# 18,19 FN= PLATE 16

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TITLE: END-PLATE BOLT PATTERN, PART# 18,19

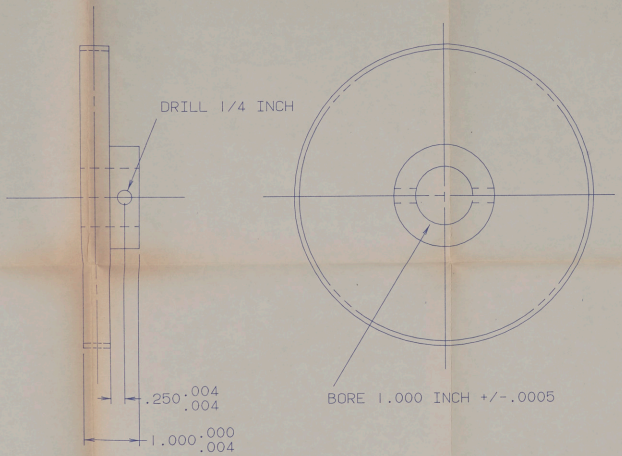
DRAWN BY: E. J. ASBURY III

SECTION: PLATE 16

SCALE: XXXXXXXXXX DATE: 5/10/90

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CAUTION: THESE INSTRUCTIONS FOR THE SINGLE THREAD GEARS ONLY



E. J. ASBURY

V. S. CHAIR

2 GEARS REQUIRED

BOSTON GEAR PART# GB1260A

MACHINING INSTRUCIONS:

-FACE THE GEAR HUB TO THE DIMENSION SHOWN

-REBORE GEAR TO DIMENSION SHOWN ANGULAR AND RADIALLY CONCENTRIC TO THE EXISTING BORE

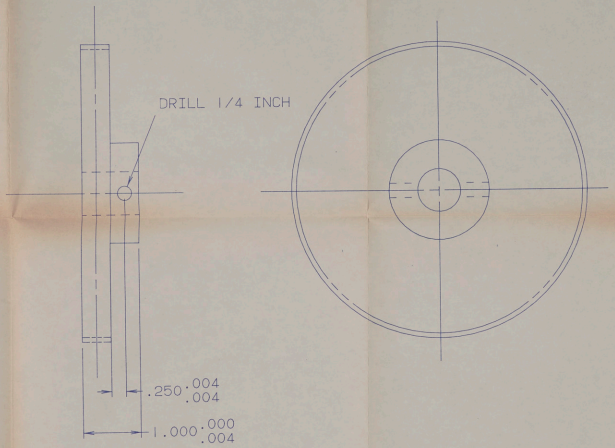
-DRILL 1/4 INCH HOLE THROUGH BORE C.L.

PN= PART# 20 PN= PLATE 17

UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: GEAR SINGLE THREAD, PART# 20
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 17
SCALE: XXXXXXXXXX DATE:5/10/90
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CAUTION: THESE INSTRUCTIONS FOR THE DOUBLE THREADED GEAR ONLY



E. J. ASBURY

V. S. CHAIR

1 REQUIRED

BOSTON GEAR PART# DB1260A

MACHINING INSTRUCIONS:

-FACE THE GEAR HUB TO THE DIMENSION SHOWN

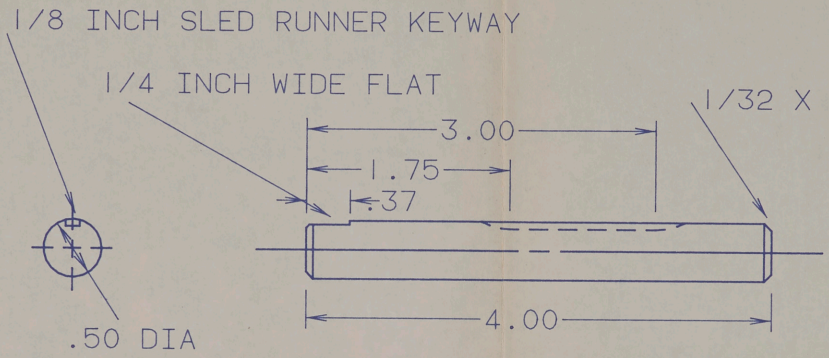
-DRILL 1/4 INCH HOLE THROUGH BORE C.L.

UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: GEAR DOUBLE THREAD, PART# 21
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 18
SCALE: XXXXXXXXX DATE:5/10/90
PRODUCED ON CADAM

PN= PART# 21 FN= PLATE 18

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Pocket #1
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1/2 INCH WORM WHEEL SHAFT

POLISH SHAFT .4997 INCH, +.0000/-.0004

3 REQUIRED

1018 DRILL ROD

E. J. ASBURY

V. S. CHAIR

PN=PART# 24 FN=PLATE 19

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CENTER FOR C.I.E.M.

TITLE:WORM SHAFT, PART# 24

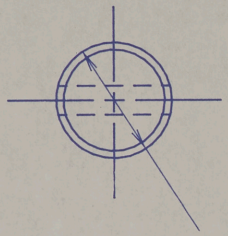
DRAWN BY:E. J. ASBURY III

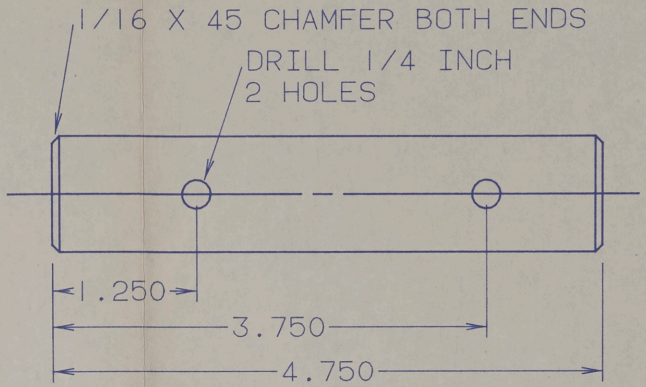
SECTION: PLATE 19

SCALE: XXXXXXXXXX DATE:5/10/90

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90
.A837
Pocket pt
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PITCH AXIS SHAFT

2 REQUIRED

1018 DRILL ROD

POLISH TO DIA. .9997 +.0000/-.0003

ALL LINEAR TOLERANCES +/- .01

E. J. ASBURY

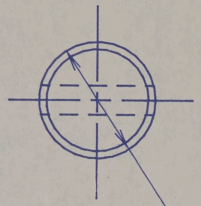
V. S. CHAIR

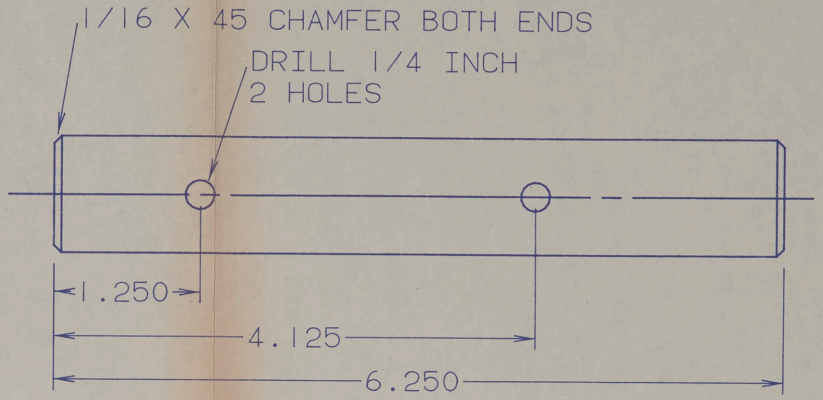
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TITLE:PITCH AXIS SHAFT, PART# 12,25
DRAWN BY:E. J. ASBURY III
SECTION: PLATE 20
SCALE: XXXXXXXXXX DATE:5/10/90
PRODUCED ON CADAM

PN=PART# 12,25 FN=PLATE 20

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.A887
pocket pt.
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ROLL AXIS SHAFT

1 REQUIRED

1018 DRILL ROD

POLISH TO DIA. .9997 +.0000/-.0003

ALL LINEAR TOLERANCES +/- .01

E. J. ASBURY

V. S. CHAIR

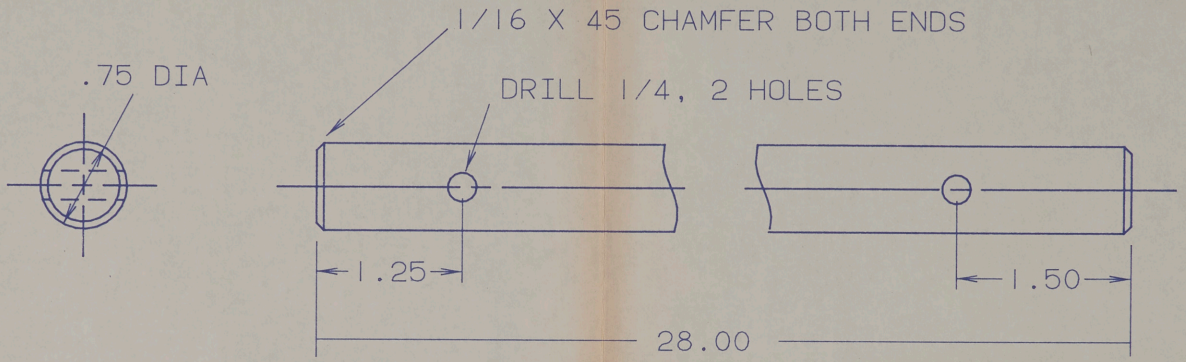
PN=PART# 26 FN=PLATE 21

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UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE:ROLL AXIS SHAFT, PART# 26
DRAWN BY:E. J. ASBURY III
SECTION: PLATE 21
SCALE: XXXXXXXXXX DATE:5/10/90
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.A827
Packet Pt.
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NOT TO SCALE



YAW AXIS SHAFT

1 REQUIRED

1018 DRILL ROD

POLISH FIRST 6 INCHES

BOTH ENDS, .9997 +0000/-.0003

ALL LINEAR TOLERANCES +/- .01

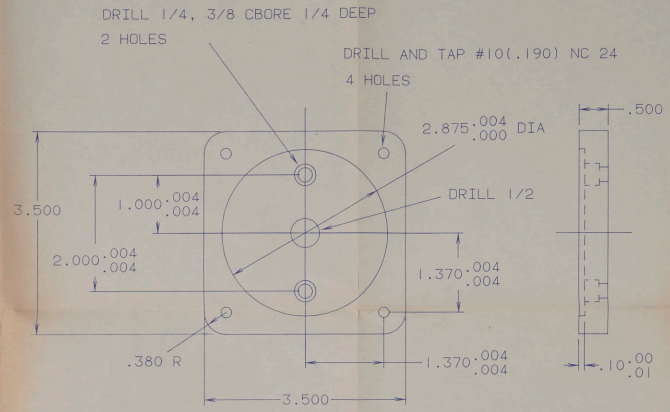
E. J. ASBURY

V. S. CHAIR

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UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: YAW AXIS SHAFT, PART# 27
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 22
SCALE: XXXXXXXXXX DATE: 5/10/90
PRODUCED ON CADAM

PN=PART# 27 FN=PLATE 22



1/2 INCH ALUMINUM PLATE

ALL TOLERANCES +/-.05 UNLESS OTHERWISE NOTED

STEPPER MOTOR MOUNTING PLATE

3 REQUIRED

E. J. ASBURY

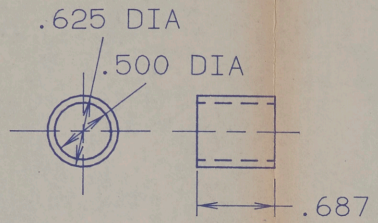
V. S. CHAIR

UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: MOTOR MOUNT PLATE, PART# 31
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 23
SCALE: XXXXXXXXX DATE:5/10/90
PRODUCED ON CADAM

PN= PART# 31 FN= PLATE 23

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GEARBOX SPACERS

6 REQUIRED

MILD STEEL

ALL TOL. +.03/-.00

E. J. ASBURY

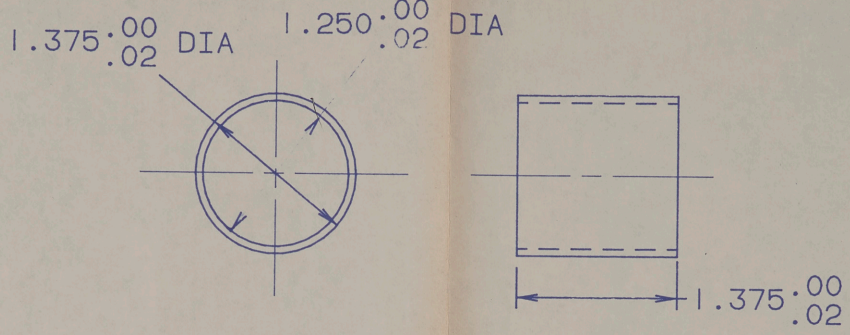
V. S. CHAIR

PN=PART# 36 FN=PLATE 24

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UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE: WORM SHAFT SPACERS, PART# 36
DRAWN BY: E. J. ASBURY III
SECTION: PLATE 24
SCALE: XXXXXXXXXX DATE: 5/10/90
PRODUCED ON CADAM

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26 of 28

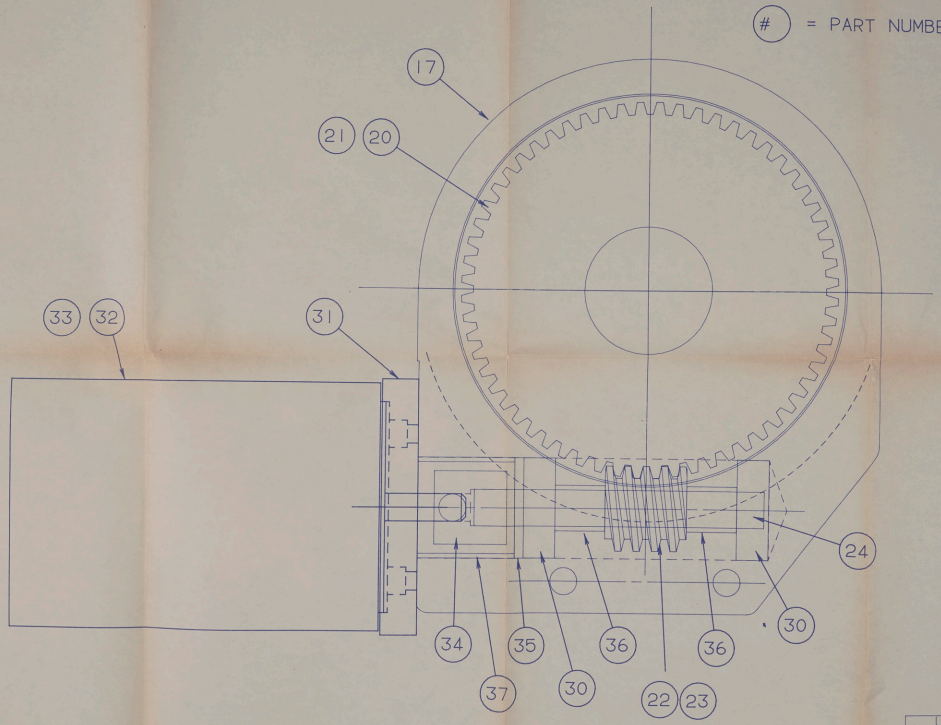


MOTOR SPACER
3 REQUIRED
MILD STEEL
E. J. ASBURY
V. S. CHAIR

PN=PART# 37 FN=PLATE 25

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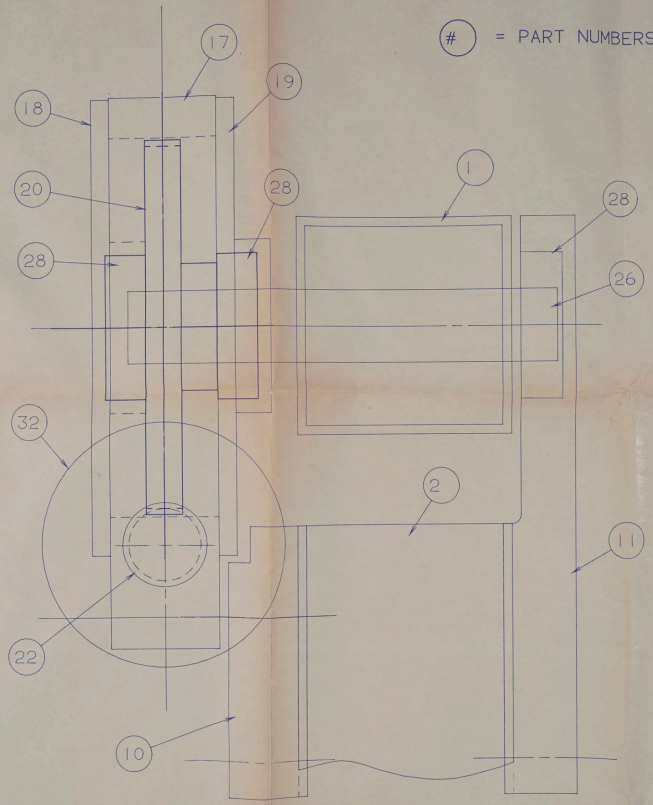
UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.	
TITLE:MOTOR SPACER, PART# 37	
DRAWN BY:E. J. ASBURY III	
SECTION: PLATE 25	
SCALE: XXXXXXXXXX	DATE:5/10/90
PRODUCED ON CADAM	



PN=XXXXXXXXXXX FN=PLATE 26

UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.
TITLE:GEARBOX OVERLAY
DRAWN BY:E. J. ASBURY III
SECTION:PLATE 26
SCALE:XXXXXXXXXX DATE:5/10/90
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PN= XXXXXXXXXX PN= PLATE 27

UNIVERSITY OF TENNESSEE CENTER FOR C.I.E.M.	
TITLE: ROLL JOINT OVERLAY	
DRAWN BY: E. J. ASBURY III	
SECTION: PLATE 27	
SCALE: XXXXXXXXX	DATE: 5/10/90
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