NASA-CR-192015

NASW-4435

IN-54-CR

141647

P-158

Design of a Reusable Kinetic Energy Absorber for an Astronaut Safety Tether to be Used During Extravehicular Activities on the Space Station

Submitted To:

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
UNIVERSITIES SPACE RESEARCH ASSOCIATION
Johnson Space Center
Houston, Texas
Mr. Gary Krch
Specialist Engineer

Prepared by:

Dawn E. Borthwick
Daniel F. Cronch
Glen R. Nixon, Team Leader

Mechanical Engineering Department
THE UNIVERSITY OF TEXAS AT AUSTIN

Fall 1991

(NASA-CR-192015) DESIGN OF A REUSABLE KINETIC ENERGY ABSORBER FOR AN ASTRONAUT SAFETY TETHER TO BE USED DURING EXTRAVEHICULAR ACTIVITIES ON THE SPACE STATION Final Report (Texas Univ.) 158 p

N93-17973

Unclas

G3/54 0141647

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ACKNOWLEDGEMENTS

The design team members thank the National Aeronautics and Space Administration and the Universities Space Research Association (NASA/USRA) for sponsoring this project. We thank Mr. Gary Krch, our NASA contact engineer, for specifying our design problem and providing assistance.

At The University of Texas at Austin, we thank Dr. Alfred Traver, our faculty advisor, for his support and technical expertise. We also thank Mr. Wendell Deen for critiquing our design drawings, Mr. Bert Herigstad for providing materials and advice, and Mr. Hank Franklin for his assistance in the machine shop. We also express our appreciation to Mr. Hank Kleespies, our teaching assistant, for his guidance and unending support throughout the semester.

Finally, we would like to thank Dr. Steven Nichols for overseeing the Mechanical Engineering Design Projects Program and providing valuable information on practical aspects of engineering.

TABLE OF CONTENTS

TABLE OF CONTENTS (continued)

TABLE OF CONTENTS (continued)

LIST OF FIGURES

LIST OF TABLES

EXECUTIVE SUMMARY

On space shuttle missions, the kinetic energy of an astronaut drifting away is absorbed by a nonreusable safety tether. Safety tethers limit the tension in the tether line to prevent damage to the astronaut's space suit or to the structure of the spacecraft. For use on the space station, NASA engineers desire a reusable safety device. This report discusses the problem statement and a proposed plan, alternate designs, the design solution, and recommendations for design improvements.

NASA/USRA sponsored a Mechanical Engineering Design Project team at The University of Texas to design a reusable kinetic energy absorber for an astronaut safety tether to be used during extravehicular activity on the space station. The device must limit the tension in the tether line, absorb the kinetic energy of a drifting astronaut, signal the astronaut that the safety device has been deployed, and allow resetting after use.

A brainstorming session led to a large number of design concepts using hydraulic, pneumatic, magnetic, electrical, and mechanical methods. Six of the mechanical concepts were chosen as feasible and developed further. Basic calculations and a preliminary analysis were performed on each of the design alternatives, and a decision matrix was used the determine the most feasible candidate for a design solution.

The design team selected a constant force spring alternative for further embodiment. The design consists of a pair of constant force springs assembled back to back to limit the tension in the tether line. The springs uncoil to absorb the kinetic energy of the astronaut and a grip roller mechanism prevents recoil of the springs and allows resetting.

The design team completed detailed design drawings and constructed a prototype for testing. Testing confirmed that the device can absorb the kinetic energy and limit the force as required. Recommendations for design improvements were made after testing the prototype.

I. INTRODUCTION

The goal of this project is to design a reusable safety device for a waist tether which will absorb the kinetic energy of an astronaut drifting away from the space station. The safety device must limit the tension on the tether line in order to prevent damage to the astronaut's space suit or to the structure of the spacecraft. The tether currently used on shuttle missions must be replaced after the safety feature has been enacted. A reusable tether for the space station would eliminate the need for replacement tethers, conserving space and mass. This report presents background information, scope and limitations, methods of research and development, alternative designs, a final design solution and its evaluation, and recommendations for further work.

1.1 Statement of the Problem

1.1.1 Sponsor

The National Aeronautics and Space Administration (NASA) is an independent government agency charged with conducting America's space flight programs. The Johnson Space Center (JSC), in Houston, Texas, is one of the many research centers that NASA operates across the United States. The main purposes of JSC include the design, development, and testing of spacecraft for manned space flight; the selection and training of astronauts, payload and mission specialists; and the planning and conducting of manned space flight missions [1]. The Universities Space Research Association (USRA), also headquartered in Houston, is funded by NASA to encourage joint projects between universities and the space industry.

1.1.2 Background

NASA/USRA is sponsoring several mechanical engineering design project teams at The University of Texas at Austin. Our project is to design a reusable safety device for an extravehicular activity (EVA) waist tether. Waist tethers are used during EVA to attach the astronaut to the spacecraft and prevent him or her from drifting away, for example, in the event of a disabled maneuvering unit [2]. The astronaut may drift at a high enough velocity to damage the space suit when the astronaut reaches the end of the tether, placing the astronaut's life in jeopardy. Less catastrophically, the spacecraft structure may be damaged, or the hooks at the ends of the tether may fail.

To prevent damage to the space suit or other equipment, a safety feature is designed into the waist tether. The waist tether has hooks that attach to tether points on the astronaut's space suit. The safety device currently used on space shuttle missions is a one-inch wide Nomex webbing strap that is doubled over and stitched together (see Figure 1) [3]. This safety tether has five parallel rows of stitching of "B" size polyester thread. This stitching begins to rip apart when the load exceeds 75 pounds [4]. Each row of stitching tears 30 inches to absorb the kinetic energy of the astronaut. This prevents overloading and damage to the space suit, the spacecraft, or the tether hooks. The stitching is contained in a Teflon fabric sleeve to retain the broken threads that will result if the safety feature is ever employed. After breakaway has occurred, the extended tether can withstand a load of 585 pounds (2602 Newtons) before failing which is much greater than the force that the tether hooks can withstand. Therefore, the hooks will break before the space suit's force limit is reached. Once the safety stitching has broken away, the safety tether must be replaced because there is no way of resetting it.

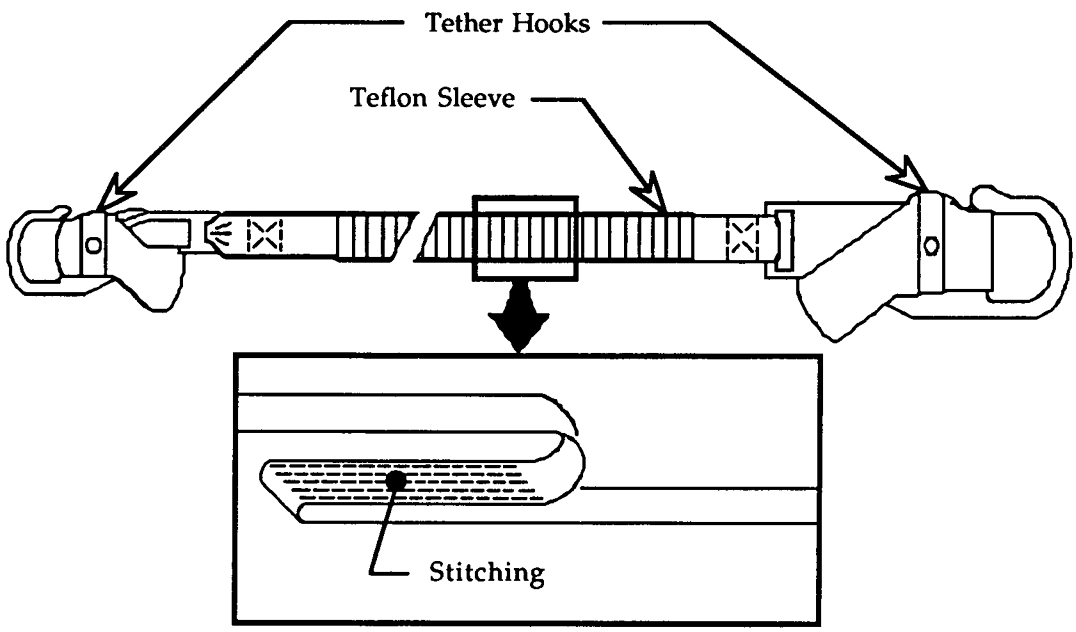


Figure 1. Webbing strap used for safety tether on shuttle missions. Five rows of stitching break to absorb kinetic energy [3].

1.1.3 Purpose

The purpose of this project is to design a safety device for the EVA waist tether that has an energy damping capacity which can be reset by the astronaut after it has been used. NASA engineers are designing a permanently manned space station assuming a reusable safety tether will be available [4]. However, there is no such device in existence. A safety tether that could be reset by the astronaut engaged in EVA will allow him or her to complete a mission rather than returning to the spacecraft to replace the tether. The design may reduce weight and space as spare tethers will not be required.

1.1.4 Criteria

NASA engineers have given the following criteria for the device:

1. Limit the force in the tether line. The space suit will be damaged if a force of 200 pounds (890 Newtons) is reached. To provide a factor of safety, the force in the line must be limited to under 100 pounds (444.5 Newtons).

2. Absorb the kinetic energy of an astronaut drifting away from the spacecraft. A space station crewperson in a space suit with tools can weigh as much as 700 pounds (317.4 kilograms) on earth [4]. The design team has been asked to design for a translation rate of four feet per second (1.2 meters per second) giving the astronaut a kinetic energy of 174 foot-pounds (236 Newton-meters). Appendix A contains calculations for the determination of the kinetic energy.

3. Maximize crewperson safety. Use of the safety device must not introduce any additional risk to the astronaut. For example, the energy absorbing device should not recoil and propel the crewperson back towards the spacecraft.

4. The device must be resettable after deployment. Approximately one EVA mission is planned per week. Current EVA equipment is designed to be used for 100 missions. It is anticipated that the safety device will be deployed once every four to five missions. Therefore, the life of the safety device should be at least 25 cycles. The device is to be reset by an astronaut with a gloved hand, so it must be within

his reach and easy to use. All safety devices for NASA must operate with two simultaneous motions performed with a single hand.

5. Minimize mass and size. The mass and size of the device must be minimized to reduce the weight of the payload which must be lifted into orbit and to conserve space in the space station.

6. Use materials which are suitable to the space environment. Materials and components must withstand an environment which lacks atmosphere, has extreme temperature variation and high solar and other radiation. The safety device must be durable enough to be reused many times and retain reliability.

7. Signal the astronaut. The device must signal the astronaut that the device has been deployed and needs to be reset.

1.1.5 Results Required

The design team was asked to accomplish the following tasks:

1. Formulate several design alternatives for a reusable safety device for the EVA waist tether.
2. Select a feasible design from the alternatives for development.
3. Complete embodiment design of the selected alternative.
4. Construct and test a prototype model of the device.

1.2 Proposed Plan

1.2.1 Scope and Limitations

Several design concepts for the tether safety device have been developed. The design team evaluated each concept and then chose one for further development. The team prepared complete working drawings and specifications and constructed a prototype of the device. The prototype was then tested under loads approximating that of an astronaut during EVA. Appendix B contains a Gantt chart showing the schedule for completing these activities.

Due to the time and facilities constraints placed on a student design team, there were several limitations to this project. Computer modeling, including finite element analysis and dynamic simulation, were not performed. The design team did not perform environmental tests, such as temperature cycling, radiation exposure, and weightless conditions.

1.2.2 Methods

The design process began with research on energy absorbing devices, as well as research on the effect of the space environment on materials. This study was based on available literature including books, journals and patents. In addition to the literature search, the Pahl and Bietz design methodology was used for developing alternate solutions [5].

This method began with the clarification of the task as presented by NASA and a detailed specifications list. The specifications list included the requirements for functioning, materials, ergonomics and safety, production and quality control, reliability, maintenance, cost, and schedule (see Appendix C for the specification list). The specification list was used to derive a function structure diagram, which is included in Appendix D. Through the use of the function structure, several different design concepts were formulated that satisfied the various functions. The concepts were evaluated through preliminary calculations and on their ability to meet the required specifications.

The design team ranked the concepts using a decision matrix to determine the most feasible candidate for further development. Once a concept was chosen, the team proceeded to complete the embodiment design. The embodiment design phase began with choosing working principles which fulfilled the required functions. Next, a scale layout of the form design was prepared, followed by selection of appropriate materials. The design was evaluated for strength and durability, considering the space environment and service life required. Safety, ergonomics, operation, and maintenance of the device were also important criteria that were assessed. Once the final form design was determined, detailed drafting was done. A prototype was constructed according to the working drawings and tested to determine its ability to fulfill NASA's requirements.

II. ALTERNATE DESIGNS

There are four main tasks that the EVA waist safety device must accomplish as determined by the function structure: limit the force in the tether, absorb the kinetic energy of a drifting astronaut, signal the astronaut that the safety device has been deployed, and allow resetting after use. The design team examined various methods of accomplishing these tasks including hydraulic, pneumatic, magnetic, electrical, and mechanical methods. A brainstorming session led to a large number of design concepts covering all of these areas. However, an initial investigation revealed that many of these ideas were not feasible. An idea was considered unfeasible if it did not meet all of the criteria that were listed previously. The hydraulic, pneumatic, magnetic, and electrical methods were all eliminated for reasons that are given below. Of the mechanical concepts, six were chosen as feasible designs. A tabulation of 51 design ideas from the brainstorming activity is included in Appendix E.

The hydraulic and pneumatic concepts are not feasible because of a possible problem with leakage [6]. In the case of pneumatic or gas systems, the operating fluid may leak out, rendering the device useless without the knowledge of the astronauts. Leakage of hydraulic fluids could prove hazardous aboard the spacecraft because the fluids could contaminate the cabin and damage sensitive equipment. Also, viscous fluids needed in the hydraulic designs are not suited to the large range of operating temperatures required in space.

The design team considered using magnetic means to absorb the kinetic energy. Magnetic particle, hysteresis, and eddy current brakes are all common energy absorbing devices in which braking torque is derived from electromagnetic reactions rather than mechanical friction [7]. These devices require electronic controls and electrical power for their operation. The team eliminated these for much the same reasons that pneumatics were eliminated because it may be hard to tell when a battery is discharged or when electric power is cut, making the device useless. Another negative consideration is the possibility of the magnetic device interfering with sensitive instruments or radio communications.

The team decided that any system that required electrical power would be less attractive than a self-contained mechanical system. The electrical systems we considered were "active" controlled servo-mechanisms and so-called "electric" brakes. Electric brakes are actually friction brakes that are electromagnetically activated [8]. The design team believes that the problem can be solved with a simple mechanical device. A complex electronic system is not necessary and may not be reliable.

The mechanical concepts were divided into three main categories: plastic deformation, friction, and springs (or elastic deformation). Plastic deformation includes the bending and unbending of a metal and the extrusion of a viscous substance. The friction category includes clutches and brakes, Velcro, snaps, ropes, and various high friction materials. Finally, the spring concepts include extension, compression, and torsional springs, constant force springs, and elastic materials.

Basic calculations and a preliminary analysis were performed on each mechanical concept to reduce the list to six designs for further development.

One of the designs is in the plastic deformation category, four use friction, and one design is in the spring category. The following sections describe the six design alternatives.

2.1 Plastic Deformation and Extrusion

All solid materials can be deformed when subjected to an external load [9]. Up to a certain load, a solid recovers its original dimensions when the load is removed. This recovery is known as elastic behavior. The load beyond which the material no longer behaves elastically is the elastic limit and occurs at the yield strength of the material. The yield strength is the stress which will produce a small amount of permanent deformation. Once the elastic limit has been exceeded, the solid is permanently deformed when the load is removed. This is known as plastic deformation or yielding.

Only ductile materials exhibit the ability to undergo plastic deformation to a great degree. Brittle materials rupture before any appreciable elongation occurs. A ductile material absorbs the energy required to plastically deform it. Since the resulting deformation is permanent, the absorbed energy dissipates and will not be returned to the system. The force needed to deform a solid material depends on the volume of material to be deformed and its yield stress.

Extrusion is the process of squeezing a material past its compressive yield strength by forcing it to flow rapidly through an orifice [10]. A viscous material will flow rapidly but cannot withstand the temperature range of outer space. A metal can withstand the temperature range, but the force needed to extrude a metal is greater than the force available. Therefore, extrusion was ruled out.

2.1.1 Bending Force Alternative

2.1.1.1 Description. The design concept involving the yielding of a material is based on a patent already owned by NASA (see Appendix F) [11]. A long tape made from a material that tends to yield and retain its shape is wound from its center on the shaft of a reel so that it can be unwound from two directions (see Figure 2). The tape winds around the shaft so that there is sufficient clearance for the tape to be unwound uniformly. One end of the tape can be attached to the astronaut and the other to the tether line. The whole device is enclosed in a container with slits to guide the extending metal. The extended metal is rewound by a handle or a crank that is attached to the reel.

The metal needs to extend for 30 inches (0.762 meters) to absorb 174 foot-pounds (236 Newton-meters) of kinetic energy. To exert a constant force, the diameter of the coil must remain essentially the same. This can be accomplished by winding the tape around a shaft with a larger inner diameter; for example, two inches.

The design disclosed in the NASA patent uses aluminum as the yielding material. Aluminum is soft and ductile in its high-purity form [12]. The 1000 series of aluminum is 99 percent or higher in purity. These compositions have excellent corrosion resistance and workability as well as the ability to withstand a large temperature range.

Appendix G shows some initial calculations for this design using a one-inch wide aluminum tape. To bend the aluminum with a force of 75 pounds (333.6 Newtons), the thickness of the tape needs to be 0.036 inches (0.09 centimeters). Thirty inches (0.762 meters) of tape will coil around a two-inch shaft approximately five times, for a total diameter less than 2.5 inches (0.0635 meters). The calculations were performed at hot and cold temperature extremes. The very cold aluminum has a higher yield strength, but still limits the force to just over 100 pounds (445 Newtons). The ultimate tensile strength of this aluminum is at least double its yield strength so there is little worry of the extended tape failing. These initial calculations indicate that this design is a feasible one. Aluminum was used in the NASA patent and the design team's initial calculations, however, other materials may be sufficient or even more suitable .

Strain hardening moderately increases the strength of high purity aluminum, and this must be taken into account in the design. Fatigue will also occur as a result of cyclic stresses. In the first stage of fatigue, crack growth occurs at a very low rate. The life of this design is based on the endurance limit of the metal and will be limited to the first stage of fatigue.

2.1.1.2 Evaluation.

Advantages:

1. The device is small and lightweight.
2. The yielding material is a metal having a large temperature range and resistance to corrosion.
3. The strength of the extended tape depends on the ultimate strength of the metal used, which can be well above the yield strength.

4. The device can be easily reset by the astronaut using a power wrench or a crank.

Disadvantages:

1. Strain hardening that will occur after repeated bending cycles must be taken into account
2. The fatigue life, or endurance, of the material will affect the number of cycles that this design can withstand.
3. The metal tape may have sharp edges.

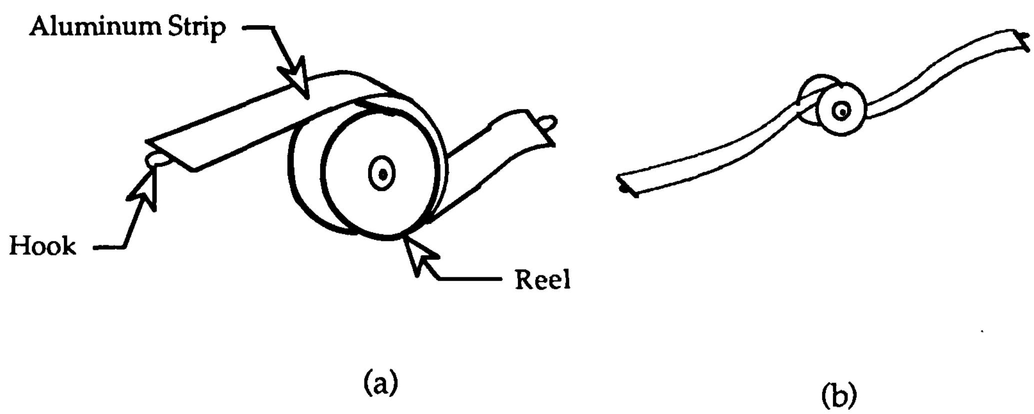


Figure 2. Bending force alternative design.
(a) The material is spooled around a reel.
(b) The material is unwound.

2.1.1.3 Requirements Satisfied. The bending force design limits the tension in the line by plastically deforming when 75 pounds (333.6 Newtons) of force is applied. The force will remain essentially constant as the material yields. All the kinetic energy of the astronaut will be absorbed if 30 inches (0.76 meters) of the material is unwound from the reel. Due to its yielding, the tape will remain extended until the astronaut rewinds it, eliminating the problem of recoil. The astronaut is signaled by the extension of the tape, which can be made clearly visible with brightly colored warning labels. To reset the safety device, the astronaut can rewind the tape with a power wrench or other tool. Because of its spooled design, the system has low mass and volume.

2.2 Friction

When two solid surfaces slide over each other, the resistance force they encounter is called friction [13]. The friction force is proportional to the normal force between the two surfaces. By controlling the normal force, the friction force can be held constant to limit the tension in the tether line. Kinetic energy can be transformed to heat energy by the friction. The following alternatives using friction include a brake, two concepts using Velcro, and a ball pulled through a constriction.

2.2.1 Brake Alternative

2.2.1.1 Description. Brakes seem to be well suited to our purpose. The function of a brake is to absorb kinetic energy, convert it to heat by friction, and dissipate this resulting heat [14]. There are four main classifications of friction brakes: disk brakes, cone brakes, drum brakes, and band brakes. These classifications are based on the geometry of the brake. The brake alternative consists of a spool of safety line connected to a friction brake which operates in much the same way as the drag mechanism of a fishing reel. The brake torque, or drag, can be set such that the safety line will unreel when the tension in the tether reaches 75 pounds (333.6 Newtons). The torque is set by adjusting the force that actuates the brake. The safety device can attach between the astronaut and the tether line. After the safety line has extended, the astronaut can release the brake, then reel in the line to reset the device. For our preliminary design concept we chose the cone brake (see Figure 3), although other types of brakes could be used. The cone brake is very efficient due to the wedging action of the brake shoe into the cone [15]. The required actuating force of a cone brake is only about one-fifth of that of a corresponding disk brake with one friction surface.

In preliminary calculations, (see Appendix H), assuming some approximate dimensions, it was determined that an actuating force of about 50 pounds (222.4 Newtons) is required to absorb the kinetic energy of an astronaut. Given a more detailed design analysis, we believe the required actuating force can be lowered. A lower actuating force will allow the use of lighter components and require less force to release the brake. From these general calculations, we can conclude that this concept is feasible.

2.2.1.2 Evaluation.

Advantages:

1. The fact that similar devices already exist reinforces the feasibility of this design.

2. Since this device will not be used repeatedly, wear on the surfaces in contact is not of great concern, and it should be able to withstand the anticipated number of cycles.

3. The brake release mechanism will allow easy rewinding of the safety line.

Disadvantages:

1. It may be difficult to accurately set the drag. If the drag is set too low, the device will activate at a lower force and will not absorb enough energy. If the drag is set too high, the device will not limit the force at a safe level [16].

2. This is a fairly complex design with a number of parts.

3. The safety line may have to be wound evenly across the reel to prevent tangling.

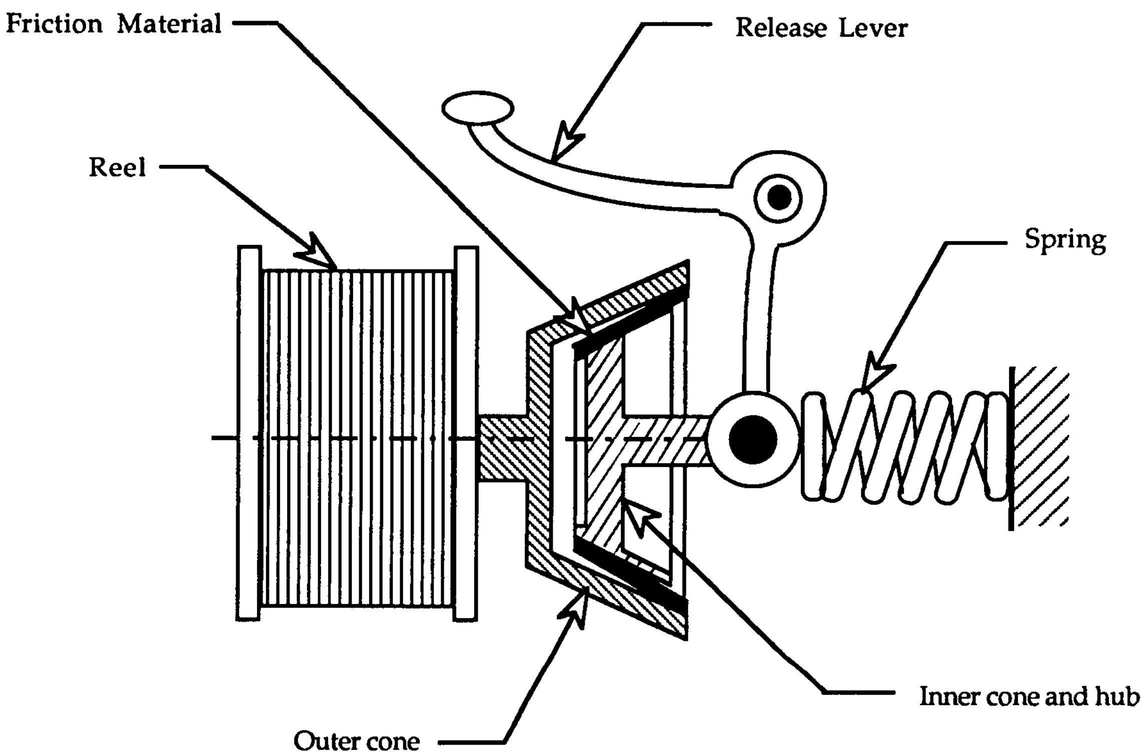


Figure 3. Friction cone brake alternative design.

2.2.1.3 <u>Requirements Satisfied.</u> The friction brake alternative limits the force to 75 pounds and will dissipate the kinetic energy by converting it to heat. A line will unwind from a reel as the energy is absorbed, signaling the astronaut that the device has been activated. A brake release mechanism will allow the line to be easily wound when resetting the safety device.

2.2.2 Velcro Reel Alternative

2.2.2.1 Description. Velcro is a fastener that consists of two woven strips. One strip is covered with tiny, stiff hooks and the other with pliable loops [17]. When the strips are pressed together, the hooks and loops fasten and can be opened by pulling the strips apart. There are several varieties of Velcro, including a flame resistant type made of Nomex that can withstand a wide temperature range.

Velcro gets its strength from the area of engagement as well as the closure pressure. Side to side movement or vibration can increase closure performance. Also affecting the strength of a Velcro closure is the way that it is pulled apart. Figure 4 shows Velcro being separated in shear, in overlap shear around a curved surface, in dynamic tension, and lengthwise peel and also gives the minimum strength for each. The surface area required to attain a force of 75 pounds (333.6 Newtons) is calculated in Appendix I.

The design team thought of several different design ideas using Velcro. After a few calculations, most of these ideas were eliminated. The peel strength of Velcro is very low, only 0.5 pounds per inch of width. To support the required force of 75 pounds (333.6 Newtons) by peeling, a width of 150 inches (3.81 meters) would be needed. Shearing Velcro provides significantly more strength. A simple design would be to merely overlap ten square inches (64.5 square centimeters) of Velcro and pull them apart in shear.

However, the strength reduces as the area of contact decreases. For this reason, a design that has a constant area of contact is required. The shear strength of Velcro is doubled if the shearing takes place around a circular surface. This configuration is used in the following design alternative.

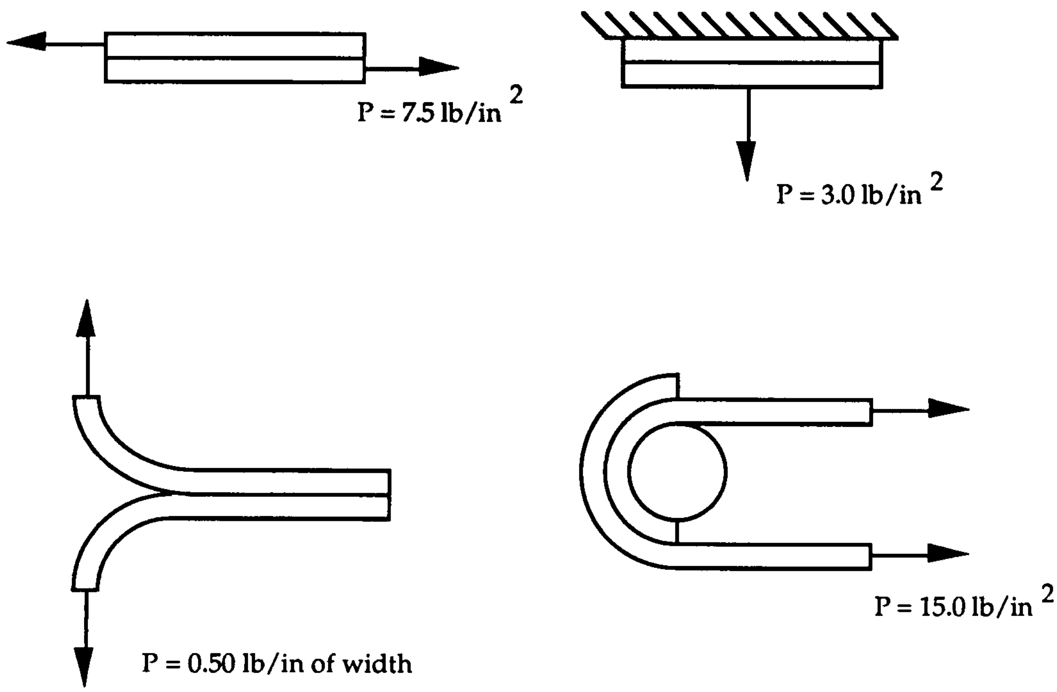


Figure 4. Strength of Velcro fastener in various configurations.

The Velcro reel alternative is shown in Figure 5. It consists of a coil of loop Velcro and a 3.2 inch (8.1 centimeter) diameter cylinder wrapped in hook Velcro, or vice-versa. This provides five square inches of contact area. The location of the hooks and loops may depend on which configuration gives the best durability. The free end of the coil wraps around the cylinder, providing the maximum shear strength of the Velcro and then attaches to the tether line. A ratchet locks the cylinder when the tether line is in tension and allows the cylinder to spin when reeling in the extended Velcro. This design will limit the force in the tether to 75 pounds (333.6 Newtons) and extend the reel of Velcro to absorb the kinetic energy. If the strength of the extended line of Velcro is not enough to prevent failure, the Velcro can be sewed on a stronger material such as the Nomex webbing used in the current safety device.

2.2.2.2 Evaluation.

Advantages:

1. This design is small, lightweight, and has a simple configuration.

2. Velcro can be used at temperatures as high as 300^{o}F.

3. The area of contact between the Velcro hooks and loops is fixed, providing a constant force.

4. The ratchet mechanism allows easy resetting of the device.

Disadvantages:

1. Since the area of contact is relatively small, there may be a high rate of wear on the material around the cylinder.

2. Velcro has a limited extreme cold temperature range of -70^{o}F.

3. The life of this device depends on the durability of the Velcro in shear.

4. Because of the lack of information on the durability of Velcro in shear, it must be determined experimentally.

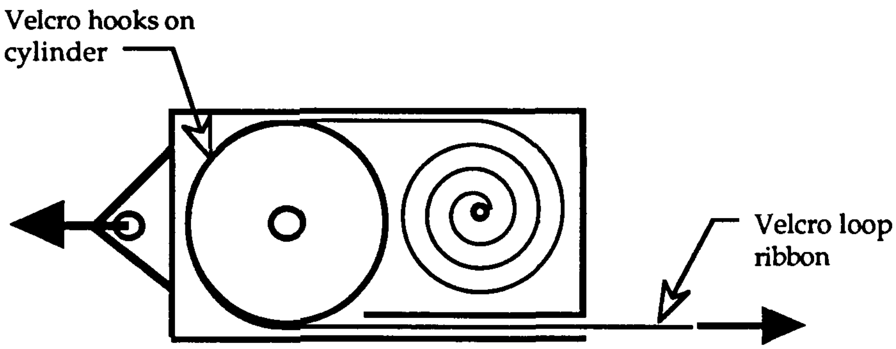


Figure 5. Velcro reel alternative design.

2.2.2.3 Requirements Satisfied. Because the five-square-inch area of contact is fixed, the tension in the tether will be limited to 75 pounds (333.6 Newtons). The reel of Velcro will extend for 30 inches (0.76 meters) to absorb the 174 foot-pounds (236 Newton-meters) of kinetic energy of the astronaut. The extension of the Velcro line will signal the astronaut that the device has been deployed. Olive green is the standard color of this Nomex composite Velcro, but, ideally, it could be brightly colored to catch the astronaut's eye. The Velcro covered cylinder can rotate freely when unlocked to allow resetting of the device.

2.2.3 Layered Velcro Alternative

2.2.3.1 Description. The layered Velcro design uses the hook and loop fastener in tension as shown in Figure 6. The tensile strength of Velcro is 3.0

pounds per square inch (2.07 Newtons per square centimeter). Therefore, a five-by-five inch panel of Velcro will support up to 75 pounds (333.6 Newtons) of force in tension. In this design concept, a series of these panels connected together like an accordion, pull apart one at a time to absorb the kinetic energy of the astronaut. The device will be reset by pressing the layers back together. A feature will be included to keep the panels in alignment during resetting, such as a nesting design of the panels.

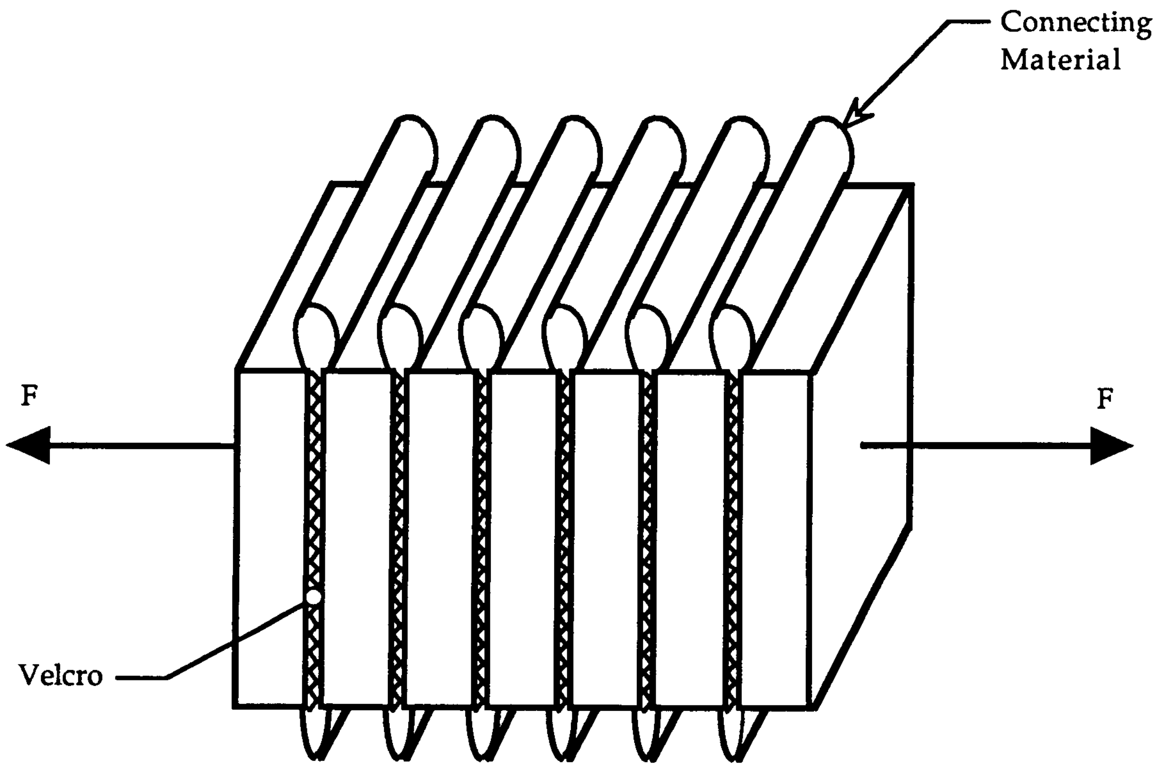


Figure 6. Layered Velcro alternative design.

2.2.3.2 Evaluation.

Advantages:

1. This is a simple design that will limit the force in the tether to 75 pounds (333.6 Newtons).

2. Velcro in tension has better endurance than Velcro in shear.

Disadvantages:

1. A large number of layers may be needed to absorb the required energy, making this design large and bulky.

2. Since the force provided will not be constant, a greater length of line may be needed.

3. The shock from one layer of Velcro breaking away may cause the next layer of Velcro to pull apart sooner.

4. There is a possible problem with a weakness where the Velcro layers are attached together, lowering the strength of the extended layers..

5. A device may be needed to press the Velcro firmly together for maximum force.

2.2.3.3 Requirements Satisfied. Twenty-five square inches (1.6 square meters) of Velcro pulled in tension will withstand 75 pounds (333.6 Newtons) of force. After the Velcro has separated, the tension in the tether quickly drops until the next layer catches, and the process repeats until all the energy is absorbed. This causes a sawtooth shaped energy curve. Because the force is not constant, the length over which the energy is absorbed must be increased. The astronaut will be signaled by the shocks and vibrations transmitted through the tether line as well as the extension of the device. Pressing the layers together resets the system

2.2.4 Ball Through Tube Alternative

2.2.4.1 Description. In this design concept, a ball attached to a line is pulled through a tube of smaller diameter than the ball (see Figure 7). For our preliminary calculations, the ball is made of steel, and the tube is made of thin aluminum. The tube will yield as the ball passes through it, and a friction force will resist the movement of the ball. The kinetic energy is absorbed by pulling the ball from one end of the tube to the other. When the ball has reached the end of the tube, the tube may be turned around to be reused, or the ball may be pulled completely through and then be returned to the entry end. This design will require a stop to prevent the ball from pulling completely out of the device. The force required to pull the ball must be fairly constant over a wide temperature range.

If the force is limited to 75 pounds (333.6 Newtons), the length of the tube must be at least 30 inches (0.762 meters) in order to absorb enough kinetic energy. The design team decided that the overall length should be as short as possible. Therefore, the force limit was increased to 100 pounds (445 Newtons). At this force, the tube need only be 18 inches (0.547 meters) to absorb enough kinetic energy. Assuming an aluminum tube with a one inch (0.0254 meters) inside diameter and a .050 inch (0.0013 meters) wall thickness, a ball of 1.00024 inch (0.025406 meter) diameter is required (see Appendix J). Therefore, meeting tolerances is essential.

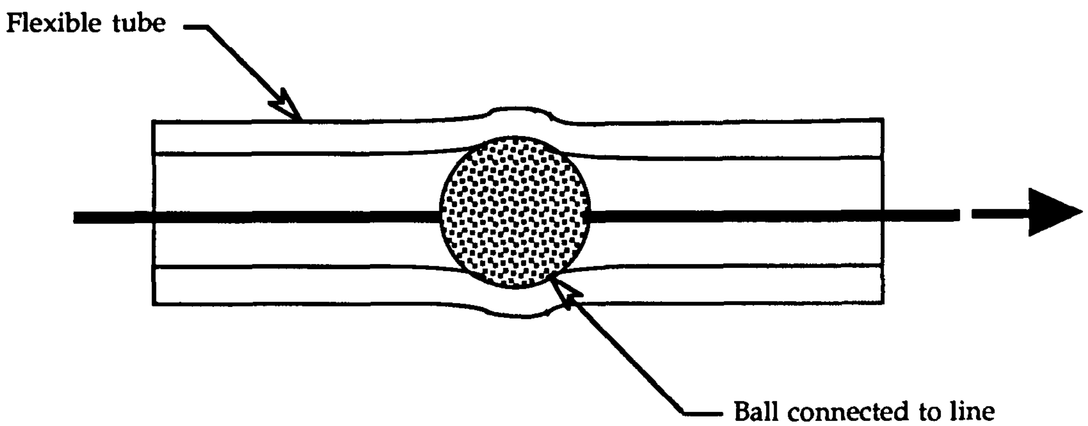


Figure 7. Ball through tube alternative design.

2.2.4.2 Evaluation.

Advantages:

1. The concept is a simple design, with few parts.
2. A constant force is supplied due to the surface contact.

Disadvantages:

1. The ball and tube may wear due to the surface friction. This will affect the tolerances that are essential in maintaining this design.
2. Resetting the device is difficult. The ball may have to be reloaded or the device turned around for reuse.
3. At 18 inches (0.547 meters), the device is too long and cumbersome.

2.2.4.3 Requirements Satisfied. The ball through tube alternative will limit the force to 100 pounds (445 Newtons). In order to absorb 174 foot-pounds (236 Newton-meters) of kinetic energy, the tube will be about 18 inches long (0.547 meters). The astronaut will have to see the extension of the line from the tube in order to know that the device has actuated. Again, the line could be brightly colored for high visibility. Resetting the device is possible, but it will be somewhat difficult.

2.3 Springs

Springs are elastic devices that exert forces or torques and absorb energy which can be stored and later released [18]. Springs are usually made of metal but can be made of plastic if the load is light. Rubber blocks can constitute springs as well as pneumatic springs using compressed gases and hydraulic springs using compressed liquids. A torsion bar is a simple form of a spring. Helical tension or compression springs can be thought of as a torsion bar wound into a helix.

Calculations that were performed on a compression spring with a typical wire diameter and spring diameter are included in Appendix K. These calculations indicate that for a compression spring to limit the force between 75 and 100 pounds (333.6 and 445 Newtons), the free length must be over 19 feet (5.79 meters) and, it needs to be compressed 6 feet (1.82 meters) to preload it to 75 pounds (333.6 Newtons). It will compress two feet (0.61 meters) further to absorb the required kinetic energy. Obviously, a device that used such a spring would be very large and heavy. From these results, a helical compression spring is not feasible for this design.

2.3.1 Constant Force Spring Alternative

2.3.1.1 Description. Constant force springs were originally patented under the name "Neg'ator" [19]. A strip of prestressed spring material is formed into a coil with a virtually constant radius and mounted on a freely rotating shaft [20]. Because of the nearly constant radius, the coil resists unwinding with a force that remains constant throughout any extension. The force is determined by the thickness and width of the material and the diameter of the coil. These springs have a long extension capability and have very little intercoil friction.

A design idea for the safety device uses two constant force springs as shown in Figure 8. The springs are available as a stock catalog item from Hunter Spring Products (see Appendix L). Each spring is 2.7 inches (0.069 meters) in diameter and withstands a load of 40 pounds (178 Newtons) for a total resistant force of 80 pounds (356 Newtons) [21]. The springs can extend 45 inches (1.14 meters) to absorb the kinetic energy of the astronaut, and they have a fatigue life of 2500 cycles.

The two springs will be housed in a box with a slit to guide the extension of the springs. The free end of the springs will attach to the tether line, and the opposite end of the device will hook to the astronaut. A ratchet device must be mounted to the device to prevent sudden recoil of the astronaut. The device would automatically reset when the load is removed, and the ratchet is released.

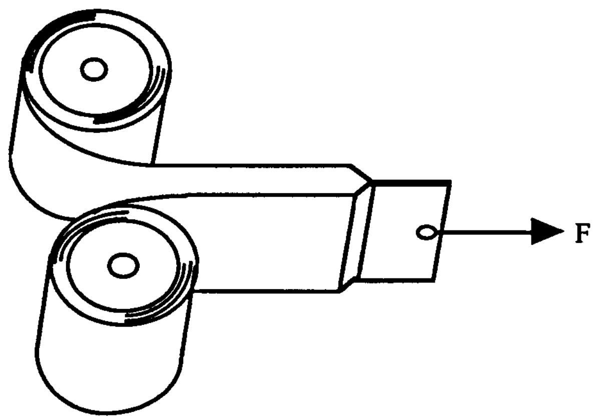


Figure 8. Constant force spring alternative design.

2.3.1.2 Evaluation.

Advantages:

1. The constant force spring alternative is a simple but effective design.

2. The springs have a small diameter and a long extension for absorbing the kinetic energy.

3. The springs are made of 301 stainless steel which has a temperature range well beyond the range required for use in space.

4. With a rated fatigue life of 2500 cycles, the constant force spring will withstand repeated use.

Disadvantages:

1. A device is needed to prevent recoil of the springs after they have been extended.

2. A mechanism to slow the spring return when resetting will have to be included in the design.

2.3.1.3 Requirements Satisfied. The two constant force springs as described will limit the tension in the tether to 80 pounds (356 Newtons). Above 80 pounds (356 Newtons), the springs will extend up to 45 inches (1.14 meters) to absorb the kinetic energy while providing a constant force. The astronaut is signalled that the device has been activated by the extension of the springs, which can be labeled with bright warning labels. Release of the ratchet will allow the springs to rewind to reset the safety device.

III. DESIGN SOLUTION

The design team selected the constant force spring alternative for further embodiment. This section presents the final design that was developed as well as a discussion of the decision process the design team used to choose a design solution. An overall description of the device is given, and the construction of a prototype is discussed. Finally, a detailed discussion of each design component is presented.

3.1 Evaluation of Alternate Designs

3.1.1 Decision Criteria

The design team decided on eight design parameters to be considered in determining a choice for the final design solution. Presented below is a brief description of these eight parameters.

1. Number of cycles. This is the number of times that the safety device can be reset and perform accurately. A life of 25 cycles was considered the minimum acceptable level.
2. Ease of resetting. The device must be reset by an astronaut with a gloved hand. NASA requires all safety devices to be operable using only one hand. A simultaneous release mechanism is required so the device does not accidentally activate.
3. Space materials. The material used on the device must be compatible with the harsh space environment which has extreme temperature variations and high solar and other radiation.

4. Extended strength. This is the strength that the device has once it has been fully extended. The design should be stronger than the tether hooks or space suit.
5. Size and weight. A design that is small and lightweight is optimal for taking into space.
6. Signal astronaut. This is the ability of the device to alert the astronaut that the safety feature has been employed and needs resetting.
7. Simplicity. The simplicity of the design affects production time and cost.
8. Reliability. This represents the accuracy of the device once it has been reset.

3.1.2 Decision Matrix

The design team constructed a decision matrix to evaluate the alternate designs and determine the final solution. A decision matrix uses "weighting factors" to indicate the relative importance of each design parameter [21]. The "Method of Pairs" procedure was used to assign weighting factor values to each of the design considerations. In the Method of Pairs, every possible combination of design parameter pairs is considered. The parameter that is deemed to be of greater importance in each pair is given a tally mark. The criteria with the greatest number of tally marks will be the most important factor in the selection of an alternate design while the criteria with the second largest number of tally marks will be the next important factor and so on.

Weighting factors were determined by dividing the number of tally marks that each parameter had by the total number of tally marks (see Appendix M).

Next, each alternative was evaluated according to the eight design parameters and given a score from one to ten. A score of ten is superior, and a score of one is poor. The scores were multiplied by the weighting factors and summed to give the total number of points for each alternate. Appendix N contains the decision matrix performed on the six alternative designs for the safety device. The constant force spring alternative ranked highest in this decision matrix and has been further developed and modified.

3.2 Design Configuration

The configuration of the final design is shown in Figure 9. The design consists of a pair of constant force springs assembled back to back to make the sum of their forces available at a single point. The free end of the springs attach to the tether and limit the tension in the line by uncoiling when the force reaches 80 pounds (356 Newtons). The springs are coiled around spools that are mounted in an aluminum housing. The housing provides structure for a tether attachment ring and a grip roller mechanism that prevents recoil of the springs. The device will mount on a waist tether that attaches to the tether points on a space suit.

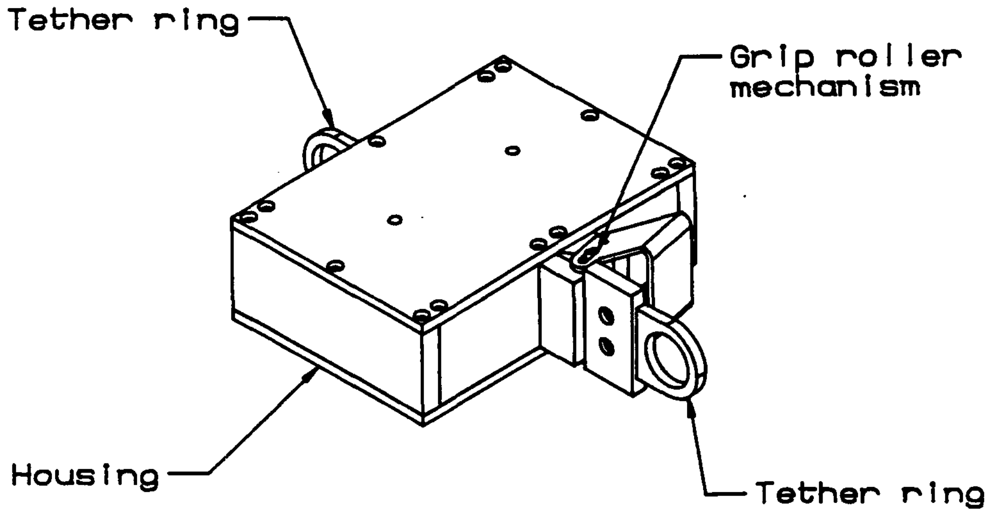


Figure 9. Final design configuration.

3.2.1 Constant Force Springs

A constant force spring is a strip of prestressed spring material which is formed into a coil with a nearly constant radius [22]. When the spring is deflected as shown in Figure 10, a resisting force, P, results which does not increase with increasing extension as with a conventional spring. Figure 11 compares the force versus deflection curves for conventional extension springs and constant force springs. The material is straightened from its original curvature over a short distance, x. The straightened material stores energy but does not add to the resisting force.

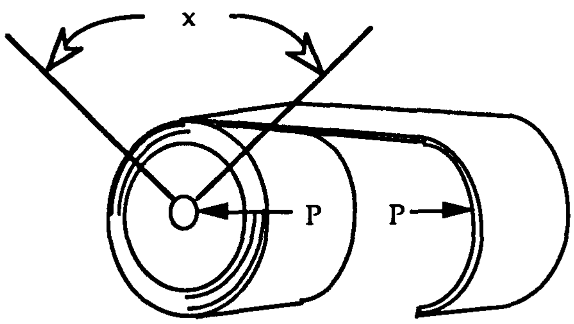


Figure 10. Deflection in a constant force spring.

The constant force springs are made of flat strips of 301 stainless steel that are coiled tightly around Delrin spools. The stainless steel strips are 0.031 inches (0.8 millimeters) thick, two inches (5.08 centimeters) wide, and 58 inches (1.47 meters) long. The springs are connected to each other at their free ends where they are riveted to a tether attachment ring. The center spools are 2.71 inches (6.88 centimeters) in diameter. As long as one turn or more of the material remains on the spool, the springs hold on by a natural gripping action. Each spring provides 40 pounds (178 Newtons) of force +/- 10 percent. When mounted back to back, the springs provide a total force of 80 pounds (356 Newtons) +/- 10 percent. With this force, the springs will extend 26 inches (0.66 meters) to absorb the kinetic energy of the astronaut. This leaves approximately three turns of spring material around the spools. However, as a safety precaution, the springs can be attached to the spools at the inner end.

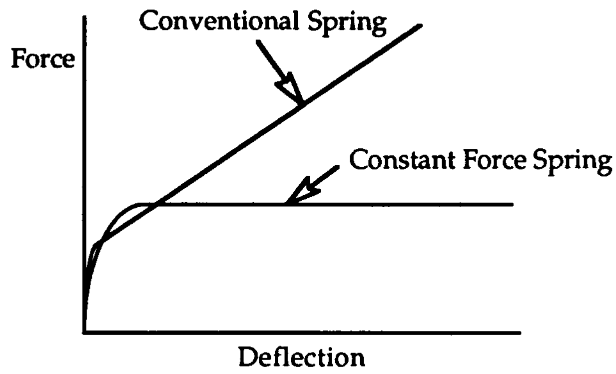


Figure 11. Force vs. deflection curves comparing constant force and conventional springs.

3.2.2 Mounting Structure

The spools are mounted inside of an aluminum housing. The sides of the housing fit closely to the sides of the spools in order to guide the springs and prevent misalignment during rewinding. The free end of the springs will exit the housing through a slit on one side. Adjacent to this slit there is a grip roller mechanism to prevent the spring from recoiling when the device is deployed. On the opposite side of the device is a ring for attaching a tether.

3.2.3 Grip Roller Mechanism

To prevent the spring from recoiling after the kinetic energy is absorbed, a grip roller mechanism is provided. The mechanism consists of a spring loaded roller in contact with the side of the constant force spring. The roller is situated inside a groove with angled sides. When the spring is pulled out of the housing, the roller is pushed to the wide end of the angled groove.

The roller is free to rotate when it is in this position, and the spring is allowed to extend without resistance. When the spring begins to return into the housing, the roller is forced into the narrow side of the angled groove. The roller becomes wedged in the groove and provides a friction force on the spring to keep it from recoiling.

After the device has been deployed, the grip roller mechanism will allow easy resetting of the device. To reset the device, the crewman will depress a lever to manually pull the grip roller out of its wedged position, and the spring will return automatically. By depressing the lever slowly, the astronaut can control the amount of friction applied and slow the rate of return of the spring. To prevent depressing the lever accidentally, a locking device can be included. For example, a button may have to be pushed before the lever can be moved.

3.2.4 Warning Feature

The feature to warn the astronaut that the safety device has been deployed will be high visibility warning labels attached to the sides of the springs. The extension of the springs will expose the warning labels and thus signal the astronaut.

3.3 Prototype Design and Assembly

As part of this project, the design team chose to design and build a prototype of the constant force spring device. A set of detail drawings and an assembly drawing are included in Appendix O. The prototype was designed to be easily produced using common machine shop equipment. For example, the housing is made of built-up plates which can be produced on an end mill. A cast housing could be used in the final production device. A cast housing would have thinner and more uniform wall sections, and the walls could easily be contoured to the shape of the springs. Also, a cast housing would eliminate the many machine screws required to hold the plates together. These changes would make the device smaller and lighter than the prototype is.

The spools that the springs wind around were turned on a lathe out of solid Delrin material. Delrin was used because it was available at a local plastics vendor. However, we recommend polyethylene teraphthalate (PET) for this component. PET has similar characteristics to Delrin but can be higher temperatures. In a production device, the spools could be made of an injection molded plastic or die cast metal. Again, the production spools would have thin wall sections to conserve mass. The axles of the spools are made of stainless steel dowel pins that are press fit into the Delrin material.

In the prototype, the constant force springs are held on the spools simply by the tension of the springs gripping the spools that are of a larger diameter than the free diameter of the springs. Therefore, the springs are prestressed when they are wound around the spools. In a production device to be used on space missions, the springs should be affixed to the spools for safety. The springs should not be able to come free of the device in case a crewman has enough kinetic energy to extend the springs past their full length.

The grip roller mechanism consists of an aluminum roller mounted in a lever mechanism made of sheet metal. The lever mechanism is spring loaded with torsion springs to bias the roller into the wedge. In the prototype there is no latching mechanism to prevent inadvertent pressing of the lever. Such a device must be included in the final product.

The free ends of the constant force springs are connected back to back to each other and attached to a tether ring end fitting. To allow easy assembly and disassembly of the prototype, the springs are connected to the end fitting with screws and nuts. In the final product for use in space, the springs will be riveted to the end fitting to insure that the fasteners cannot work loose.

3.4 Materials

Materials selection and qualification for use in the space environment consists of four main steps:

1. Identify requirements for the material
2. Select candidate material
3. Analyze environmental capability of the material
4. Qualify the material [23].

The design team limited the scope of the material selection process in this design project. The team identified the material requirements and selected candidate materials. In the selection of the candidate materials, the designers considered environmental capabilities published in the literature. Information on the effects of the space environment on materials has accumulated over the years of space flight activity. Therefore, there is little need to determine material capabilities experimentally. Likewise, due to the tight schedule and lack of facilities to simulate the space environment, qualification testing of the materials under those conditions cannot be performed by the design team.

3.4.1 Material Requirements

The primary requirements, or design functions, of the materials in the energy absorber are mechanical in nature [24]. They are the mechanical properties which can be measured or derived, such as density or tensile strength. In addition to the design functions, requirements and limitations that are specific to the mission were considered. These requirements include mission life, reliability, weight, cost, and schedule. Environmental requirements are governed by the mission requirements. Since the device is intended for extravehicular use, the materials must withstand the harsh environment of outer space. The space environment is characterized by intense solar radiation, extreme temperatures, vacuum, particulate radiation, and the presence of micrometeoroids.

All of these environmental elements can have detrimental effects on materials. Ultraviolet solar radiation can cause lattice defects in crystalline materials and cross-linking of polymers. Solar radiation can also cause thermal stresses in materials. Temperature extremes can cause softening or embrittlement, chemical degradation, and can accelerate other environmental effects. Charged particle radiation includes solar winds, solar event particles, and cosmic rays. This type of radiation can cause lattice defects in crystalline materials and cross-linking of polymers, as ultraviolet radiation does, and can also cause further radiation damage. The lack of atmosphere, or vacuum, of space causes out-gassing of volatiles, diffusion, and vacuum welding. Finally, micrometeoroids can cause fractures or punctures depending on their size and velocity.

3.4.2 Selection of Candidate Materials

By taking the various material requirements into account, the design team selected candidate materials for components of the kinetic energy absorbing device. In general, inorganic structural materials are not severely affected by the space environment. The selection of these materials is based on strength-to-weight ratios and mechanical properties required [25]. The material selections for the main components are given in the following sections, including some advantages and disadvantages of each.

3.4.2.1 Constant Force Springs: 301 Stainless Steel.

Advantages:

1. Off-the-shelf component, short lead time
2. Constant properties over wide temperature range
3. High strength
4. Low cost

Disadvantages:

1. Low strength to weight ratio

3.4.2.2 Spring Spools: Polyethylene Teraphthalate (PET) .

Advantages:

1. Dimensional stability and toughness
2. High strength and wear resistance
3. Maximum working temperature = +230° F
4. Low coefficient of friction
5. Good machinability

Disadvantages:

1. Low resistance to ultraviolet [26]
2. Relatively high cost

3.4.2.3 Housing: Aluminum Alloy 6061.

Advantages:

1. Constant properties over wide temperature range
2. Insignificant vacuum sublimation [27]
3. Resistant to radiation
4. High strength to weight ratio

Disadvantages:

1. No major disadvantages

3.4.2.4 Grip Roller: Aluminum.

Advantages:

1. Relatively high coefficient of friction
2. Similar advantages as listed for housing above

Disadvantages:

1. Low wear resistance
2. Tendency of galling or welding to itself

3.4.2.5 Release Lever: Aluminum.

Advantages:

1. High strength-to-weight ratio
2. Unaffected by radiation, vacuum, or temperature

Disadvantages:

1. May strain harden
2. Relatively low strength—might bend

3.4.2.6 Dowel Pins and Torsion Springs: Stainless Steel.

Advantages:

1. Off-the-shelf components, short lead time
2. Constant properties over wide temperature range
3. High strength
4. Low cost

Disadvantages:

1. Low strength-to-weight ratio

3.5 Cost Analysis

Table 1 gives approximate costs to produce all of the machined parts of the final design solution. The material costs are from McMaster-Carr Supply Company 1991 Catalog [28]. The machining times and machining costs were calculated based on machining rates and engineering costs provided by Dr. Grady Rylander [29] (see Appendix P). The prices for all stock items are shown in Table 2. The approximate cost to produce a single constant force spring safety device is $171.71.

Table 1

Material and Machining Costs

Part	Material	Material Cost ($)	Material Removed (in^3)	Machine Time (min.)	Machine Cost ($)	Total Cost ($)
Spools	Delrin	14.50	3.65	8.11	10.14	26.03
Top/Bot. Panels	Aluminum	1.34	3.78	12.61	20.76	22.10
Side Panel (2)	Aluminum	0.69	0.20	7.98	9.98	10.67
T-shape	Aluminum	1.61	20.84	9.20	7.75	13.11
L-shape (2)	Aluminum	0.81	6.24	4.11	5.14	5.95
End Fitting	Aluminum	0.16	0.51	6.39	7.99	8.15
Roller	Aluminum	0.01	0.012	2.00	2.52	2.53
Handle	Aluminum	0.10	15 in	7.00	8.85	8.95

Total $97.49

Table 2

Cost of Stock Parts

PART	QUANTITY	MATERIAL	COST/UNIT ($)	TOTAL ($)
Spools	2	Delrin	8.60	17.28
Springs	2	Steel	22.50	45.00
1/4" Dowels	2	Steel	0.29	0.58
1/8" Dowels	8	Steel	0.11	0.88
Flat hd screws	26	Steel	0.12	3.12
Rivets	2	Aluminum	0.07	0.14
Torsion spring	2	Steel	2.99	5.98
3/16" Dowels	2	Steel	0.62	1.24

Total $74.22

IV. EVALUATION OF THE DESIGN SOLUTION

The first step in the evaluation of the design was calculating the strength of each of the components used in the device. Calculations of the strength of the tether attachment rings, the screws and rivets, and the spool axles are included in Appendix Q. All of these components have very high factors of safety considering the relatively low forces that the device will encounter in use.

The design team assembled a prototype for mechanical testing. The ability of the device to limit the force was evaluated by performing tension tests. The energy absorption capability of the device was tested by using it to stop a rider on a bicycle translating on level ground. The endurance of the device was tested by repeatedly dropping a weight suspended from the device so that the springs extended to the maximum working deflection.

4.1 <u>Test Results</u>

4.1.1 <u>Tension Testing</u>

Tension testing was performed using an Instron testing machine. The Instron machine is commonly used to measure the tensile or compressive strength of materials. The Instron machine used in our tests featured a crosshead speed control calibrated in metric units and a 100 kiloNewton (22481 pounds) load cell. The prototype was mounted in the Instron and pulled at crosshead speeds of 50, 100, 200, and 500 millimeters per minute (3.9, 7.8, and 19.7 inches per minute). The force versus deflection curves from these tests are included in Appendix R. The test was repeated at different speeds because the literature on constant force springs noted that the performance of this type of spring may vary at high deflection velocity [30]. However, there was no apparent difference in the results at each of the different crosshead speeds. Each curve increased rapidly to a maximum force of about 440 Newtons (99 pounds). Then, the force fell slowly and finally leveled at about 385 Newtons (86.5 pounds). From the results of this test, the team concluded that the device performed as expected and limited the force to less than 100 pounds (445 Newtons). The actual measured force is higher than the 80 pounds (356 Newtons) predicted from the spring specifications. The team credited this discrepancy to the tolerance of +/- 10 percent given for the spring force in the catalog and to friction between the spool axles and the housing.

4.1.2 Energy Absorption Testing

In order to determine the kinetic energy absorption capabilities of the device, the team improvised a simple test. The fixed end of the energy absorber was attached to a stationary bicycle rack. One end of a long rope was attached to the movable end of the energy absorber while the other end was tied to a bicycle. One team member rode the bicycle away from the device. When the rider reached the end of the rope, he was stopped by the energy absorption device. To estimate of the rider's velocity, a second team member measured the time it took the bicycle to pass two chalk marks on the pavement. The third team member measured the distance that the bike traveled past the point where the rope became taut which represented the amount the springs extended. From the data recorded in this experiment and knowing the weight of the rider and bicycle, the kinetic energy of the rider was derived. Using this derived value of the kinetic energy, the deflection of the spring required to absorb the kinetic energy was calculated. The team compared this predicted value to the actual values measured in the test. The results of this test are included in Appendix S. On average, the actual values of the deflection were 14 percent longer than the predicted values.

The team feels that there are many possible reasons for the difference between the actual and predicted deflections. There were several possible sources of error and variability. One source of error could have been the starting and stopping of the stopwatch, since there is an element of human reaction time and visual parallax. The measurement of the deflection also included visual parallax and human error of marking the stopping point. The deflection may have been longer because the spring force does not reach its rated value until the spring deflects about four inches. The calculation of the deflection was done assuming the force was a constant 85 pounds and did not account for the initial deflection. Finally, the rope that the bicycle was attached to probably stretched a significant amount. In order to eliminate this source of error, the deflection should be measured where the spring extends out of the device, not at the front of the bicycle at the end of the rope. Considering that this test was quickly improvised and that there are many sources of variability, the design team feels that being within 14 percent of the predicted value is a good result. However, the team recommends further testing under more controlled conditions using the mass and velocity given in the design specifications.

4.1.3 Endurance Testing

To test whether the device could withstand repeated use, the design team devised a test in which a weight attached to the energy absorber was dropped from a height to develop kinetic energy. The team used a dumbbell with 70 pounds (31.7 kilograms) of weight on the bar to perform this test. The weights were suspended from the energy absorber at an initial height of 40 inches (1.02 meters) above the ground. The weights were lifted 18 inches (0.46 meters) and released. The device stopped the weights just before they hit the ground. This procedure was repeated 25 times. In addition to these 25 repetitions, the device was cycled eight or ten times with 80 to 90 pounds (36.3 to 40.8 kilograms) dropped from various heights to find the right combination of weight and height to prevent hitting ground. After 25 cycles the device was disassembled and inspected for signs of failure or wear.

During disassembly, it was difficult to remove the axle of one of the spools from the hole in the aluminum side plate. However, it seemed to rotate freely in the hole. Close inspection of the bearing surface showed signs of wear, which was to be expected. The bearing was not oiled in these tests. If the prototype is used for further testing, lubrication is recommended to reduce the amount of wear at the axles.

With the side plate removed, grooves could be seen worn in the side plate where the spring rubbed against it. During the test, the team observed that the device was hanging at a slight angle, causing the spring to contact the sides during extension. Because the prototype was not completely finished according to the prints before testing, the device was suspended with rope around a flat temporary cross member. The circular tether attachment point of the final design would allow the device to be positioned to prevent contact between the springs and the housing. In addition to proper positioning, the problem may be aided by the addition of sides to the spring spools to keep the springs aligned on the spools.

The team concluded that the amount of wear was reasonable for this prototype. There were no signs of serious failure. Therefore, we believe a minimum life of 25 cycles can be achieved.

4.2 Design Performance

The performance of the kinetic energy absorber can be evaluated on the basis of its ability to meet the specified design criteria. The kinetic energy absorber was tested to determine if it would limit the force in the tether line, absorb the kinetic energy, and endure the required design life. The results of these tests indicated that the device will limit the force to less than 100 pounds (445 Newtons) and will endure at least 25 cycles. Preliminary results of the kinetic energy tests indicate that the device will absorb the required amount of kinetic energy.

Because the prototype did not include the grip roller mechanism at the time of testing, the performance of this device could not be determined experimentally. This feature is needed to insure that the crewperson does not recoil and to allow controlled resetting of the device.

The weight of the prototype, at 6 pounds (2.72 kilograms), is relatively high for lifting the mass into space. The major contributor to the weight is the stainless steel springs. Since they were purchased off the shelf on the basis of the mechanical properties desired, the springs cannot be modified to reduce their weight. The housing, which is constructed of several thick aluminum plates, could be made lighter. All but the structurally necessary material could be removed leaving a skeletal frame of aluminum. A thin casing of plastic or other material could then be used to enclose the device.

Aluminum was used extensively on the prototype because it is well suited to the space environment. An aluminum housing protects the polymer components inside the device from radiation exposure. The effects of the space environment must be considered when choosing lightweight replacement materials.

The extension of the spring acts as a warning to the astronaut that the safety device has been enacted. Bright warning labels could be attached to the springs as an added warning feature.

V. CONCLUSIONS AND RECOMMENDATIONS

5.1 Further Testing

Due to time constraints and lack of appropriate test facilities, an abbreviated amount of testing was performed on the prototype. To confirm that the device will work properly under the specified conditions, the force exerted by the springs when pulled by a mass corresponding to 700 pounds (317 kilograms) at a rate of four feet per second (1.22 meters per second) should be measured. From this test, the extension of the spring required to absorb the kinetic energy can be found. To be acceptable, the length of this extension must be less than the maximum working deflection of the spring.

Tests should be performed to determine the effects of the space environment on the materials used in the device. Since sufficient material specifications and design data may be found in literature on the materials, material qualification may be accomplished as a part of the prototype performance testing [31]. Suitable tests may include temperature cycling, prolonged exposure to ultraviolet and other radiation and extreme temperatures in a vacuum.

5.2 Design Improvements

The most critical shortcoming of the prototype is its excessive weight. The existing springs are the heaviest components of the device. One possible way to reduce the weight of the device is to use springs made of composite materials. Composite springs have been successfully used in such applications as automobile suspensions [32] and aircraft door counterbalance mechanisms [33]. The design team determined that a high strength graphite/epoxy unidirectional fiber composite spring with the same energy storage capability as the stainless steel may be as much as six times lighter (see Appendix T).

The environmental factors to consider for a composite material used in the space environment are the effects of temperature, ultraviolet radiation, and vacuum. There are several temperature effects that should be addressed. Creep may occur in one or both of the components at elevated temperatures [34]. Chemical degradation of the matrix resins is another problem. Also, residual stresses may develop due to differences in thermal expansion of the two components. Ultraviolet radiation can also cause degradation of the polymer matrix material. Finally, volatiles may evaporate from the polymers in the vacuum environment.

In addition to trimming weight from the springs, considerable weight can be reduced by eliminating all but the structurally necessary material from the housing. The remainder of the housing is only used as covering to prevent foreign objects from entering the device. However, the aluminum housing provides good shielding from radiation for the internal polymer components. This shielding may be even more important if composite springs are used. The thickness of the aluminum may be reduced in areas that serve only as covering. A small aluminum structural frame may be used, covered by a thin shell of aluminum or plastic if a suitable plastic for the environment can be identified.

In conclusion, the design team believes that the constant force spring alternative is a viable solution to the problem of absorbing the energy of a drifting astronaut. Testing of the prototype confirmed that the design will satisfy the specified requirements. Further development is necessary to finalize the requirements of preventing recoil of the astronaut and of reducing the size and mass of the device. The team is confident that these requirements can be met.

REFERENCES

1. Maps and General Information, brochure, (Houston: National Aeronautical and Space Administration, Johnson Space Center, 1991).

2. A. I. Sibila, "Flight Safety Aspects of Astronaut Tether Dynamics," Space Safety and Rescue, 1979-1981, (San Diego: American Astronautical Society, 1983), Vol. 54, p. 399.

3. R. C. Trevino and R. K. Fullerton, EVA Catalog: Tools and Equipment, (Houston: National Aeronautical and Space Administration, Johnson Space Center), Rev. A, pp. W-1 - W-2.

4. Gary Krch, (Houston: National Aeronautical and Space Administration, Johnson Space Center), Personal communication, 9 September 1991.

5. G. Pahl and W. Bietz, Engineering Design: A Systematic Approach, (London: The Design Council, 1991), p. 169.

6. F. D. Yeaple, Fluid Power Design Handbook-2nd Edition, (New York: Marcel Dekker, Inc., 1990), pp. 8-15.

7. W. C. Orthwein, Clutches and Brakes: Design and Selection, (New York: Marcel Dekker, Inc., 1986), pp. 65-67.

8. Orthwein, p. 126.

9. E. A. Avallone and T. Baumeister III, eds., Marks' Standard Handbook for Mechanical Engineers, (New York: McGraw-Hill Book Co., 1978), pp. 5-18 - 5-21.

10. Avallone and Baumeister III, pp. 13-21--13-22.

11. Hoffman and Vranas, U.S. Patent No. 3,347,004, (August 1967).

12. Aluminum Standards and Data-1984, (Washington, D.C.: The Aluminum Association, Inc., 1984), pp. 10-33.

13. Avallone and Baumeister III, pp. 3-24.

14. R. C. Juvinall and K. M. Marshek, Fundamentals of Machine Component Design, (New York: John Wiley and Sons, 1991), p. 674.

15. Juvinall and Marshek, pp. 683, 684.

16. Velcro Brand Fastening Products and Systems, brochure, (Velcro USA, Inc.), pp. 3961-3990.

17. Juvinall and Marshek, p. 427.

18. Juvinall and Marshek, p. 462.

19. H. E. Nankonen, "7 Applications for the Constant-Force Spring," Mechanisms, Linkages, and Mechanical Controls, Ed. Nicholas P. Chironis, (New York: McGraw-Hill, 1965), pp. 210-211.

20. "Neg'ator constant force extension springs," brochure, (Sellersville, PA: Ametek, Hunter Spring Products, U. S. Gauge Div., 1990), pp. 4-8.

21. Pahl and Bietz, p. 121.

22. Design and Application of Small Standardized Components, Data Book 757, (New York: Stock Drive Products, 1983), pp. 418-419.

23. W. A. Symonds, "Design, Mission and Environmental Considerations in the Selection of Materials for Space", The Effects of the Space Environment on Materials, Society of Aerospace Material and Process Engineers, (North Hollywood, CA: Western Periodicals Company, 1967), pp. 44.

24. Symonds, pp. 44, 46-47.

25. John B. Rittenhouse and J. B. Singletary, "Selection of Materials for Optimum Performance", Space Materials Handbook - Third Edition, Lockheed Missiles and Space Company, (Springfield VA: Clearinghouse, 1968), p. 629.

26. A. S. Dunbar, "Materials for Antenna Systems", Space Materials Handbook - Third Edition, Lockheed Missiles and Space Company, (Springfield VA: Clearinghouse, 1968), p. 391.

27. Rittenhouse and Singletary, p. 628.

28. McMaster-Carr Supply Company Catalog 97, (Elmhurst, Illinois: McMaster-Carr Supply Company, 1991).

29. Dr. Grady Rylander, (Mechanical Engineering Department, The University of Texas at Austin), Personal communication.

30. Design and Application of Small Standardized Components, p. 423.

31. Symonds, p. 53.

32. Scott Beck, "Effective Use of Carbon Fiber in the Automotive Industry," How to Apply Advanced Composites Technology, (Dearborn, MI: ASM International, 1988), p. 474.

33. John A. Ennes, "Composite Material Spring for 767 Entry and Service Door Counterbalance Mechanism," Advanced Composites: The Latest Developments, (Dearborn, MI: ASM International, 1986), p. 5.

34. Bryan Harris, Engineering Composite Materials, (London: Print Power Ltd., 1986), p. 107.

APPENDICES

APPENDIX A

Kinetic Energy Calculation

APPENDIX A

KINETIC ENERGY CALCULATION

$$F = Mg \Rightarrow M = \frac{F}{g} = \frac{(700\,lb)(4.448\,kg\cdot m/s^2/lb)}{9.81\,m/s^2}$$

$$M = 317.4\,kg$$

$$V = (4\,FT/s)(12\,IN/FT)(.0254\,m/IN) = 1.219\,m/s$$

KINETIC ENERGY, $KE = \frac{1}{2} m V^2$

$$KE = \frac{1}{2}(317.4\,kg)(1.219\,m/s) = 235.9\,N\cdot m$$

$$\Rightarrow KE = 173.99\,FT\text{-}lb$$

$$KE = Fd \Rightarrow d = \frac{KE}{F}$$

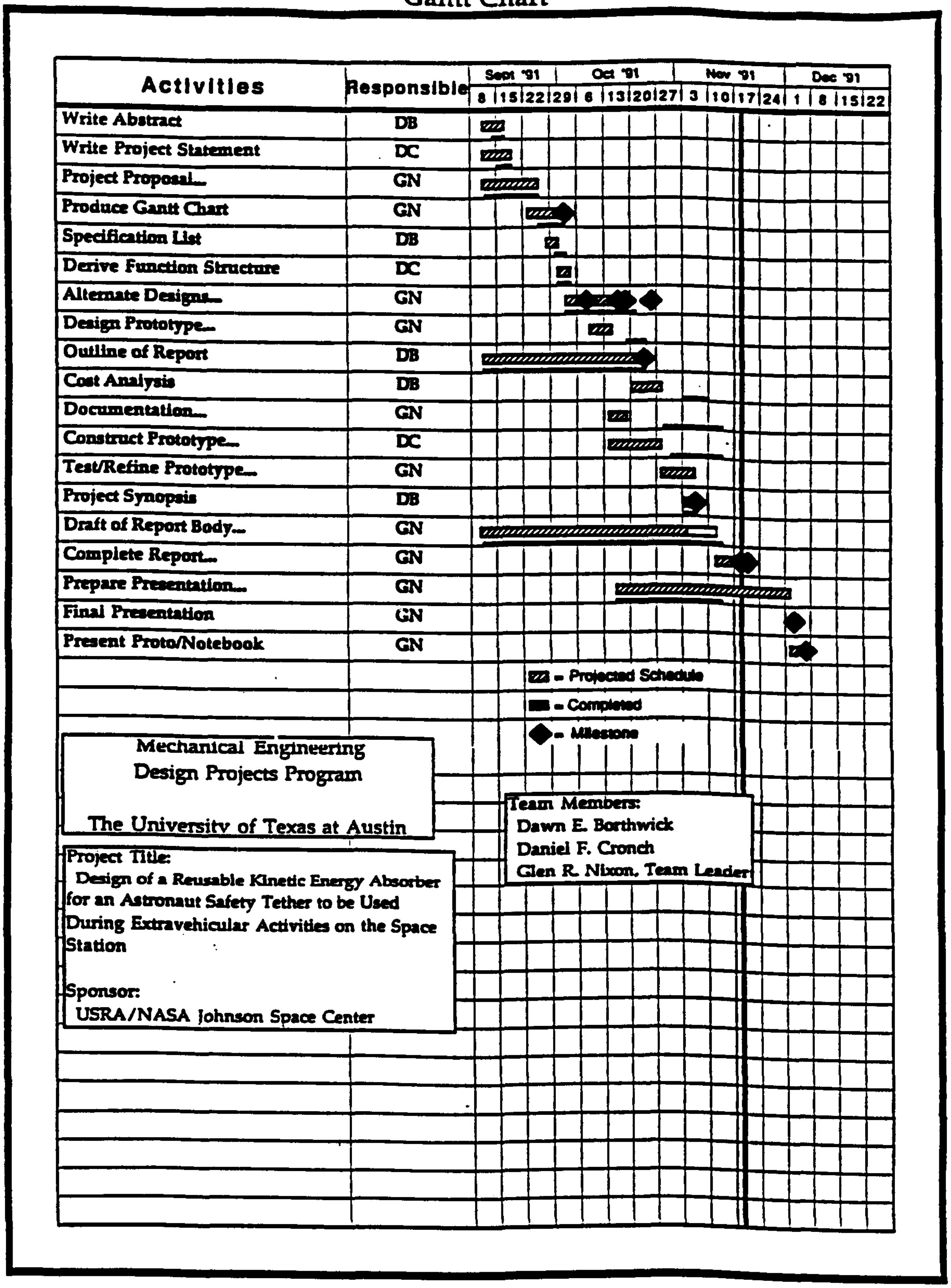
$$d = \frac{174\,FT\text{-}lb}{75\,lb} = 2.32\,FT = 27.8\,IN$$

APPROXIMATELY 30 IN NEEDED TO ABSORB THE REQUIRED KINETIC ENERGY

APPENDIX B

Gantt Chart

APPENDIX B
Gantt Chart



APPENDIX C

Specifications List

APPENDIX C

Specification List

Change	D/W	Requirements	Resp.
		Functional	
	D	1. attaches to tether or suit	
	D	2. limits tension in line to less than 100 lbs	
	D	3. activates at ~ 75 lbs	
	D	4. absorbs kinetic energy	
	D	5. reusable/resettable - # of uses t.b.a.	
	W	6. resettable during EVA	
	D	7. total length of tether 25-55 ft.	
	D	8. applied loads: suit and crewman ~ 700lbs translation rate = 4 ft/sec	
	D	9. extended tether strength at least 200 lbs	
		Materials	
	D	1. operating temp. range: -200° to 250° F	
	D	2. resistant to radiation	
	W	3. light weight	
	D	4. handle stress/fatigue	
	D	5. noncorrosive	
		Ergonomics and Safety	
	W	1. minimize size	
	D	2. prevent astronaut from recoiling	
	W	3. ease of use	
	D	4. minimize hazards during use or failure	
	W	5. speed of resetting	
	D	6. able to use with gloved hand	
	W	7. stowability	
	D	8. astronaut must notice device has been activated	
	W	9. visible to astronaut (accessable)	
		Production/Quality Control	
	W	1. simplicity of design	
	W	2. ease of manufacturing	
		Reliability	
	W	1. life	
	D	2. maintain tension specifications	
	D	3. must reset to proper condition	

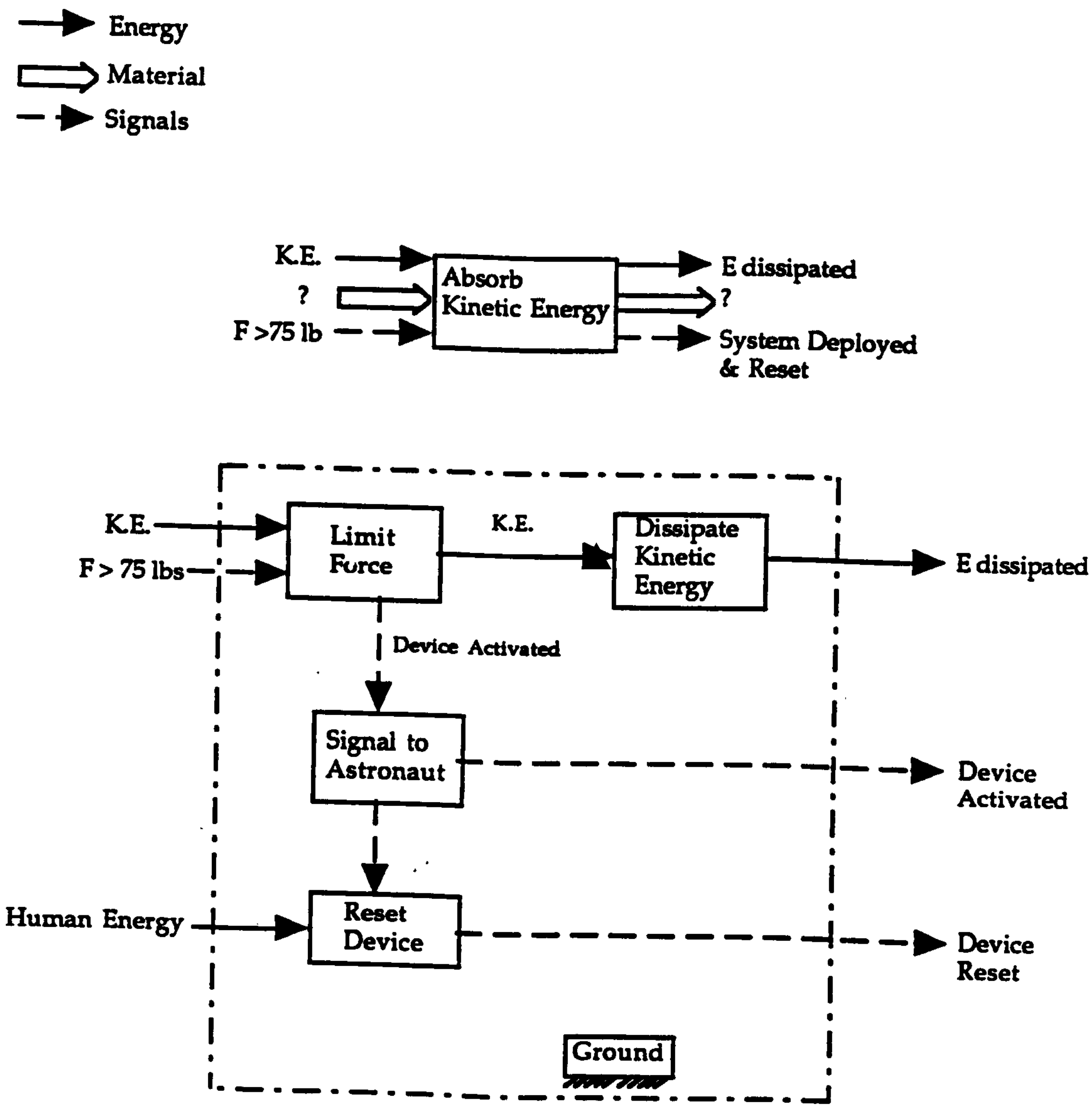
Change	D/W	Requirements	Resp.
		Maintenance	
	W	1. minimize	
	W	2. calibration	
	W	3. self maintaining	
		Cost	
	D	1. analysis required	
		Schedule	
		1. see Gaant chart	
		2. presention Dec. 2	

APPENDIX D

Function Structure Diagram

APPENDIX D

Function Structure Diagram



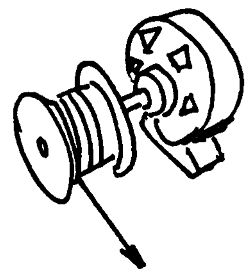
APPENDIX E

Brainstorming Design Ideas

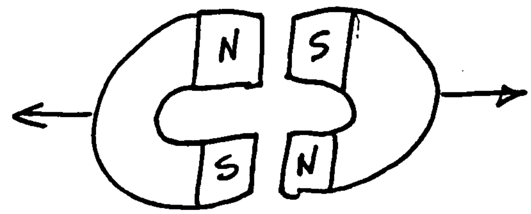
APPENDIX E

BRAINSTORMING DESIGN IDEAS

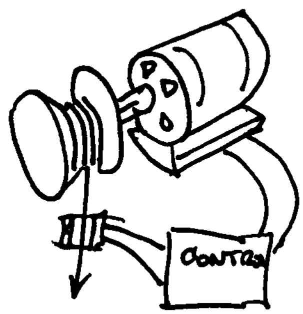
EDDY CURRENT BRAKE

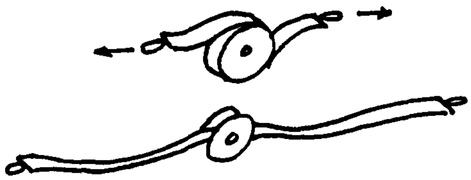
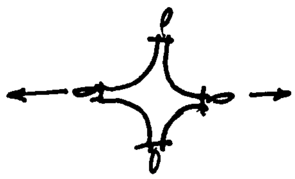
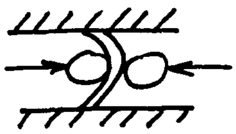
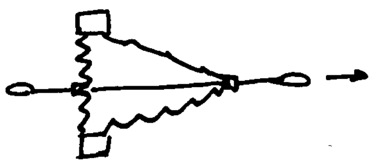
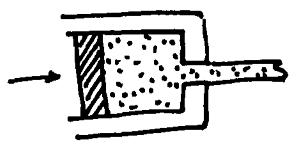
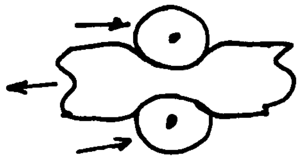


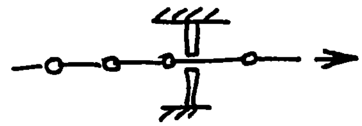
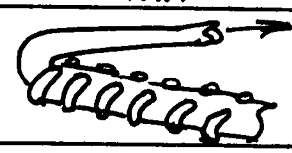
MAGNET



ACTIVE FEEDBACK CONTROL



REEL OF METAL THAT ABSORBS ENERGY WHEN YIELDING	
YIELDING METAL ACCORDIAN STYLE	
YIELDS ONE WAY THEN THE OTHER	
COIL OF METAL UNWINDS	
METAL DIAPHRAGM YIELDS	
METAL COIL YIELDS	
EXTRUSION	
ROLLING TO EXTRUDE	

BALL PULLED THROUGH TUBE	
BALL PULLED THROUGH COLLET	
BALLS POPPED THROUGH HOLE	
CORK PULLED THROUGH TUBE	
FLEXIBLE HOLDS THAT GIVE	
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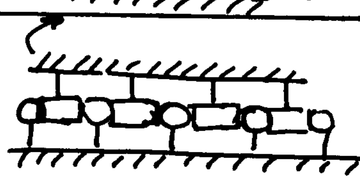
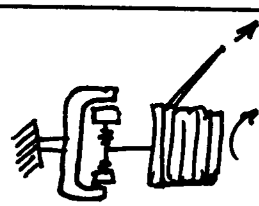
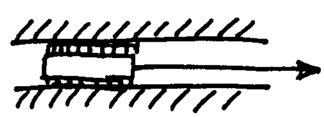
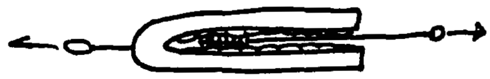
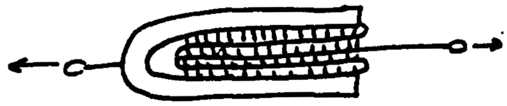
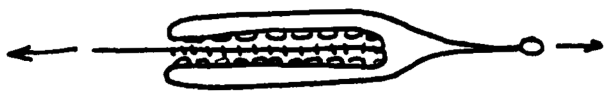
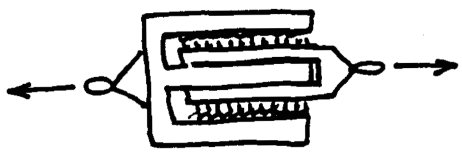
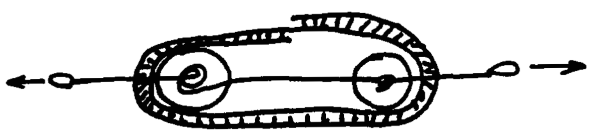
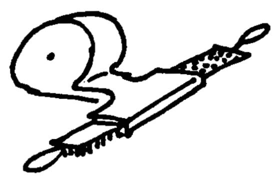
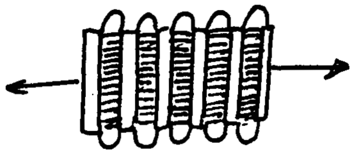
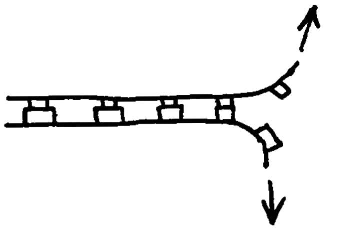
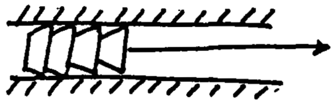
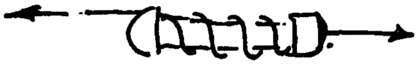
BALLS SNAP BETWEEN CYLINDERS	
BEAD SLIDES ON STRINGS TO CREATE RESISTANCE	
CENTRIFUGAL CLUTCH	
ABRASIVE SURFACE FRICTION	

RIGID SIDES WITH VELCRO LINING - VELCRO STRIP PULLS THROUGH.	
DOUBLED OVER LAYERS OF VELCRO	
VELCRO WRAPPED AROUND A CYLINDER	
FLEXIBLE SIDES LINED WITH VELCRO	
CYLINDER LINED WITH VELCRO	
REELS COVERED WITH VELCRO	
VELCRO FORCED TO SHEAR APART	
VELCRO REELS AROUND A CURVED SURFACE	

LAYERS OF VELCRO IN TENSION	
SNAPS	
RUBBER STOPPER THROUGH TUBE	
"SKY GENIE"	
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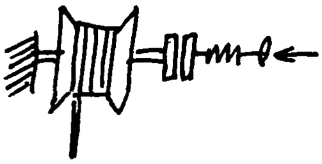
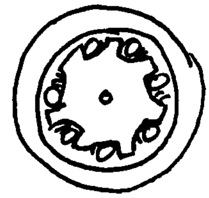
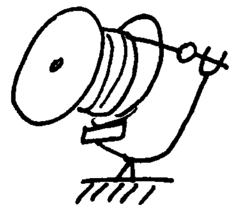
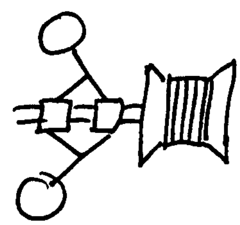
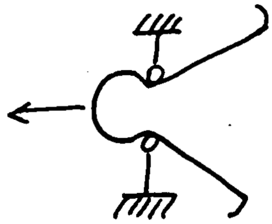
CLUTCH MECHANISM	
ONE-WAY CLUTCH	
RATCHET	

SPRAG (ONE-WAY) CLUTCH	
SAFETY BRAKE	
FLY BALL GOVERNOR	
CONSTANT FORCE SPRING	
COMPRESSION SPRING	
TENSION SPRING	
TORSION SPRING E6	

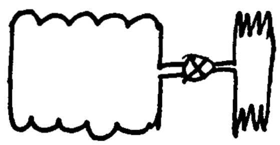
SPRINGY MATERIAL PULLED
SMALL OPENING



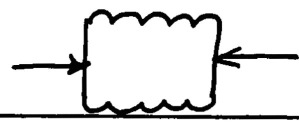
STRETCHING AN ELASTIC
MATERIAL

PNEUMATIC

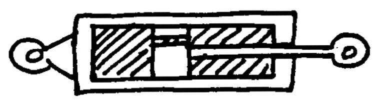
BELLOWS FILLED WITH
AIR



AIR SPRING

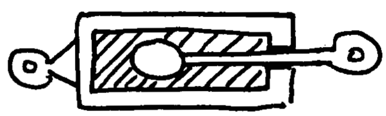


HYDRAULIC
FLUID SWITCHES
CHAMBERS



PADDLE WHEEL

BALL MOVES THROUGH
FLUID



APPENDIX F

Impact Energy Absorber Patent

APPENDIX F

Aug. 22, 1967 I S HOFFMAN ETAL 3,337,004

IMPACT ENERGY ABSORBER

Filed Dec. 22, 1964 2 Sheets-Sheet 1

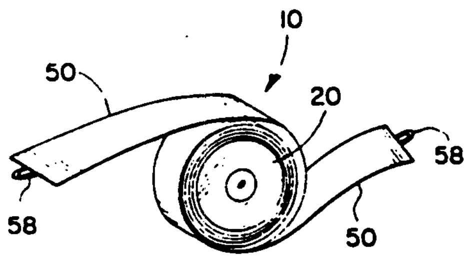


FIG. 1

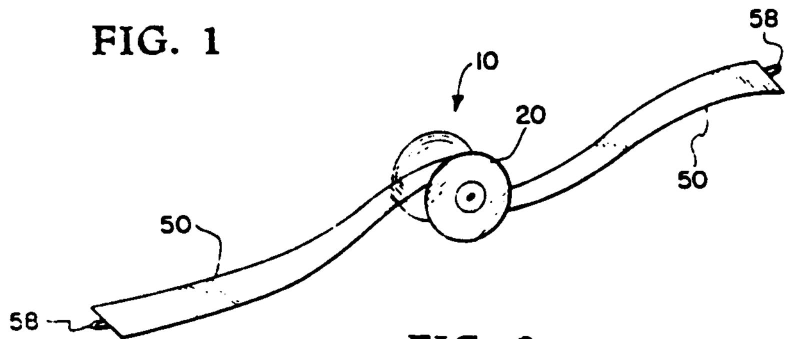


FIG. 2

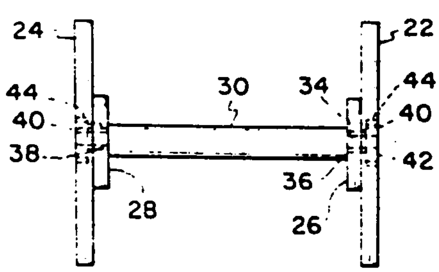


FIG. 3

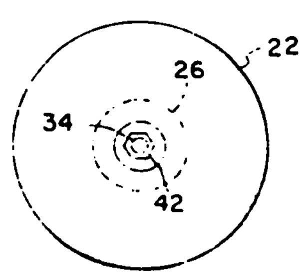


FIG. 4

IRA S. HOFFMAN
THOMAS VRANA

Aug. 22, 1967 I. S. HOFFMAN ETAL 3,337,004

IMPACT ENERGY ABSORBER

Filed Dec. 22, 1964 2 Sheets-Sheet 2

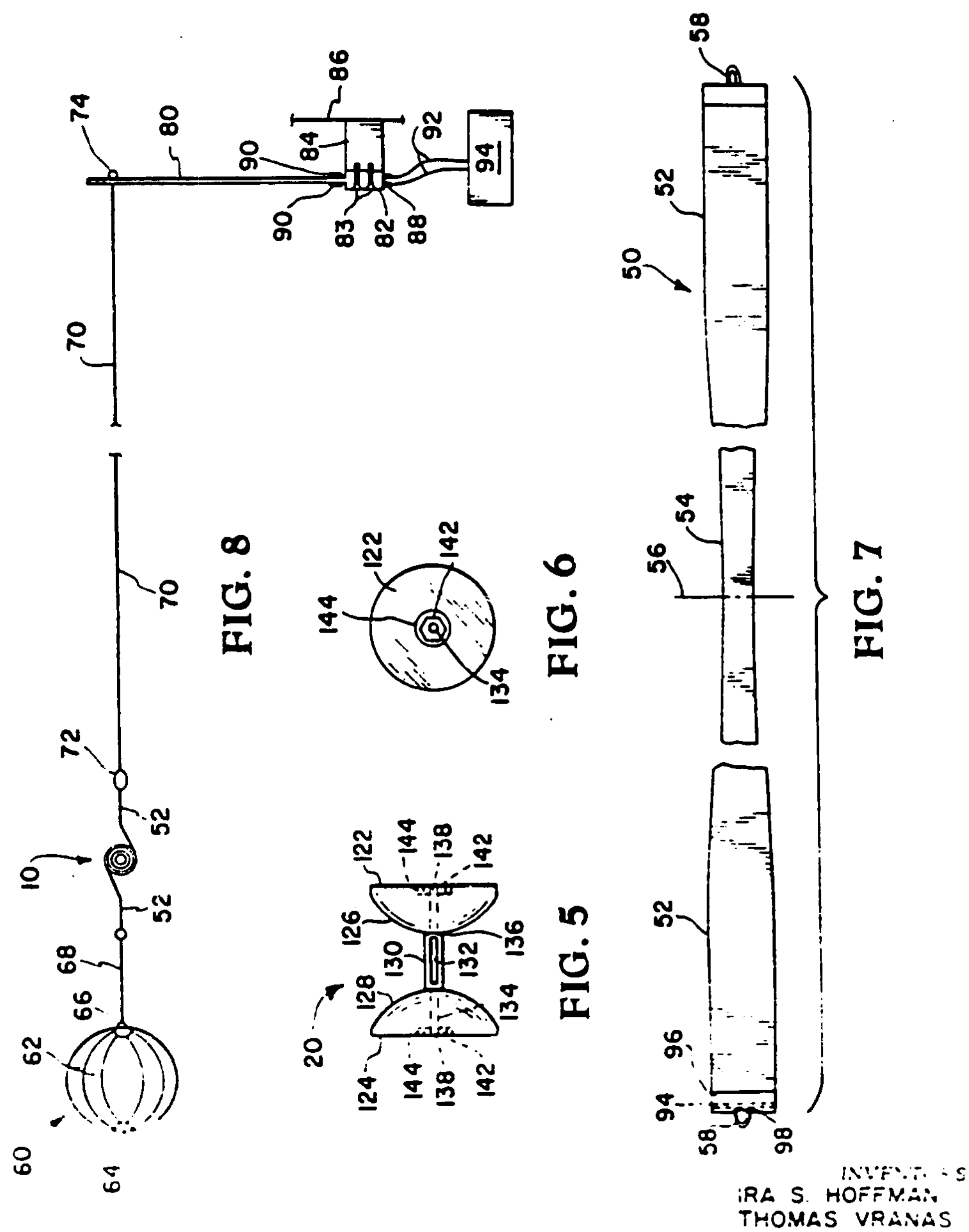


FIG. 8

FIG. 6

FIG. 5

FIG. 7

INVENTORS
IRA S. HOFFMAN
THOMAS VRANAS

F2

IMPACT ENERGY ABSORBER

Ira S. Hollman, Newport News, and Thomas Vranas, Hampton, Va., assignors to the United States of America as represented by the Administrator of the National Aeronautics and Space Administration

Filed Dec. 22, 1964, Ser. No. 420,466
16 Claims. (Cl. 188—1)

ABSTRACT OF THE DISCLOSURE

An impact energy absorber for a color visibility and drag measuring system having an energy absorber adjacent a drag member with one end secured thereto and the other end secured to a cable. The end of the cable is attached to a cantilever beam to which are applied strain measuring devices. The energy absorber includes a tape gradually widening from the center toward the ends thereof and made from a material tending to yield and retain its rolled shape. The tape is folded upon itself about its centerline and, starting from the narrow portion, is wound about the shaft of a reel which has the inner faces on the end plates thereof spaced to provide sufficient clearance for the tape to be unwound uniformly. Energy is absorbed at a decreasing rate by the unrolling of the tape, as well as the rotation of the reel.

The invention described herein may be manufactured and used by or for the Government of the United States of America for governmental purposes without the payment of any royalties thereon or therefor.

This invention relates generally to energy absorption and more particularly to an impact energy absorber having a decreasing absorption rate.

As space exploration continues, many studies, experiments, and tests must be conducted to provide information as to the operability of devices in the vacuum of outer space. For example, it is necessary to obtain data regarding the effects of the space environment upon bodies moving therethrough. If these traveling bodies are to be manned and controlled by a human pilot, the visual ability of that person becomes an essential factor. The instant invention involves a single system device that permits obtaining both characteristics of movement through space and visual characteristics of bodies near the traveling vehicle. That is, a member may be deployed that is of a color to determine visual characteristics as well as providing data relating to the air density by measuring the drag of that member as it moves or is drawn through orbit at high altitudes. However, if a member is deployed from a moving vehicle, when that member reaches the end of the cable attaching it to the vehicle, there is a sudden impact which, depending upon the circumstances, would cause the member to reverse direction and overtake the vehicle from which it was deployed. In order to prevent this bounce effect, some device must be provided to either slow down the deployed member or to absorb the energy created by the drag forces. Such a device must be miniaturized and yet withstand maximum vibrational effects to operate in a space environment as well as covering a wide range of impact energy magnitudes while not being affected by several weeks of dormancy prior to use. It must also possess the feature of diminishing absorption rate versus time during absorber operation. It is readily apparent that the spring balances and recoil springs, for example those used in counterbalances for windows and maintaining light fixtures at a predetermined position, are incapable of meeting the requirements enumerated hereinabove. Such devices are cumbersome, susceptible to damage from periods of dormancy and the intensive vibrations during launch, as well as being limited in range
and varying gravitational effect, Further known devices are unable to absorb deployment forces while permitting maintenance of an inelastic connection after complete deployment.

The present invention has all of the required features which are provided by utilizing a tape that narrows from each end toward the center where it is folded and starting at the center of the tape is wrapped about a reel.

It is an object of this invention to provide an energy absorber for minimizing impact forces.

Another object of this invention is to provide an impact energy absorber for application where moving masses are suddenly stopped by arresting cables or ropes.

A further object of this invention is to provide a reel and tape energy absorption device having a decreasing rate of absorption.

A still further object of this invention is to provide a color visibility study system utilizing an energy absorber for absorption of impact energy upon deployment of the system.

Still another object of the instant invention is to provide a drag measuring system for use in a low vacuum environment and which utilizes a novel impact energy absorber.

A further object of this invention is to provide a method of forming an energy absorber.

Another object of the instant invention is to provide a method for visual color study and drag measurement in a low vacuum environment.

Generally, the foregoing and other objects are accomplished by locating an energy absorber adjacent a drag member and having one end secured thereto and the other end secured to a cable. The other end of the cable is attached to a cantilever beam to which are applied strain measuring devices. The impact energy absorber includes a tape which gradually widens from the center toward the ends thereof and which is folded upon itself about its centerline. Starting with the narrow portion, the tape is wound about the shaft of a reel which has the inner faces on the end plates thereof spaced to provide sufficient clearance for the tape to be unwound. Energy is absorbed at a decreasing rate by the unrolling of the tape as well as the rotation of the reel.

A more complete appreciation of the invention and many of the attendant advantages thereof will be readily apparent as the same becomes better understood by reference to the following description when considered in connection with the accompanying drawings, wherein:

FIG. 1 is an isometric of the energy absorber of the present invention;

FIG. 2 is an isometric diagrammatic view of the absorber of FIG. 1 after partial operation;

FIG. 3 is a front view of a reel of one of the embodiments of the absorber of the instant invention;

FIG. 4 is an end view of FIG. 3;

FIG. 5 is a front view of an alternative embodiment of the reel of the instant absorber;

FIG. 6 is an end view of FIG. 5;

FIG. 7 is a plan view of the tape with portions omitted for clarity; and

FIG. 8 is a diagrammatic view of the visual study and drag measuring system in which the instant invention is utilized.

Referring now to the drawing wherein like reference numerals designate identical or corresponding parts throughout the several views and more particularly to FIG. 1 wherein an energy absorber, generally designated by reference numeral 10, is shown as having reel 20 and tape 50. Reel or spool 20 is comprised of elongated shaft 30 and disk-shaped end plates 22 and 24 and inner plates 26 and 28. The opposite ends of shaft 30 are reduced in diameter at 34 to form stop [illegible] 36 which prevents end plates 22 and 24 and disk-shaped inner plates 26 and 28 from moving inwardly. Sections 34, of reduced diameter, on shaft 30 are threaded at 38. The outer faces of end plates 22 and 24 are provided with counterbores or rabbets 44 surrounding a central bore 40 for receiving nut 42 which is threaded upon reduced diameter sections 34 of shaft 30.

The alternative embodiment of spool 20, FIGS. 5 and 6, is similar to that shown in FIGS. 3 and 4. However, shaft 130 has an elongated radial slot 132 centrally located between end plates 122 and 124 that receives a small portion of the narrow section of tape 50. End plates 122 and 124 have convex hemispherical inner faces 126 and 128, respectively, prevented from moving inwardly by abutment 136 formed by reduced diameter end sections 134 of shaft 130. End sections 134 have threads 138 for receiving nuts 142 in counterbores 144 extending about central bore 140 in end plates 122 and 124. Although the precise degree of absorption ability is difficult, if not impossible, to compute, convex inner faces 126 and 128 retain tape 50 in a neatly wound package which may be more desirable under certain conditions.

As more clearly shown in FIG. 7, tape 50 has wide end portions 52 and gradually decreases in width to narrow central portion 54 at centerline 56. Wide end portions 52 are provided with attachment members 58 for attaching the opposite ends of tape 50 to cables or similar energy transmitting elements. It has been found that an operable and expediently formed attachment member can be constructed by folding a small portion 96 of ends 52 of tape 50 over a rod 94 to which loop or eye 58 is secured and then welding folded portion 96 to the original end 52. Since aperture 98 is provided at the fold for eye 58, it is free to swivel.

Referring now to FIG. 8 wherein is shown a novel visual study system and drag measuring mechanism incorporating absorber 10 of the instant invention. Drag member 60 is shown as an inflatable sphere made of gores 62 and end panels 64. Cable 68 of minimum length is attached at one end to absorber 10 and at the other end to sphere 60 at 66. The opposite end of absorber 10 is attached to connector 72 which is secured to cable 70. The other end of cable 70 is attached to hook 74 mounted on beam 80. Cable 70 is preferably a braided nylon line which has been found to operate quite successfully under the conditions for the tests to be made with the instant invention.

The drag measurement system is shown in FIG. 8 to include cantilever beam 80 which is machined from a single piece of material to have built-up end section 82 which is attached to support 84 by bolts 83. Support 84 is rigidly secured, as by welding, at 86 to the vehicle from which sphere 60 is to be dragged. Strain gages 90 are mounted on beam 80 immediately adjacent end section 82 and are connected by terminals 88 and leads 92 to instrumentation 94 for determining the strains in cantilever beam 80 and therefore the drag force caused by sphere 60. The instrumentation to which leads 92 are connected is of conventional design and within the purview of the art.

Operation

Drag member 60 is preferably a folded, inflatable, aluminum Mylar sphere that is packaged to be released by explosive bolts and ejected jack-in-the-box fashion. The drag of the folded sphere will gradually increase as [illegible] sphere 60 to float

the reel once it is fully deployed. Since reel is not secured to either the vehicle or cable 70, it is free to fall away from the deployed system without becoming entangled therewith. In the absence of absorber 10, when sphere 60 has floated rearward the full extent of cable 70 there is an impact and attendant rebound of sphere 60 which could overtake the aerospace vehicle and possibly become entangled therewith. Accordingly, energy absorber 10 is positioned near sphere 60 to absorb the energy resulting from the rearward flotation thereof. An ideal absorber 10 would be completely nonelastic and absorb energy over a sufficiently long period of time to prevent excessive buildup of forces within cable 70. In order to accomplish the ideal situation, absorber 10 is provided with a diminishing absorption rate.

The diminishing rate of absorption of absorber 10 is accomplished by designing tape 50 with relatively wide end sections that gradually taper to narrower central portion 54. Tape 50 is folded about centerline 56 to position ends 52 over one another. Should the alternative embodiment of spool 20 be utilized, a small portion of central section 54 is fitted into slot 32 in shaft 30. Tape 50 is then wound about shaft 30 and as the ends 52 are approached, tape 50 is held neatly coiled by inner plates 26 and 28 or convex faces 126 and 128 or end plates 22 and 24 or 122 and 124, respectively.

In view of the above-identified operation, it is seen that as the tensile forces between sphere 60 and cable 70 increase, tape 50 is unwound from spool 20. Tape 50 has been found to perform efficiently when made from soft aluminum sheet that tends to yield and retain its rolled shape. The material then has to yield again while being unrolled and most of the velocity or impact energy is expended in this way. However, some energy is expended in angularly accelerating the rolled mass and an additional amount is expended in linear acceleration of the tape. As can be seen from FIGS. 2 and 7, the tape gradually tapers to a smaller width as it unwinds as well as the configuration of the diameter of the rolled up absorber decreasing as tape 50 is unwound. This provides the diminishing absorption rate desired. It is apparent that the amount of energy to be absorbed will dictate the dimensions for tape 50 which may be readily calculated by the skilled artisan. For example, a different material may be used or the length, width or thickness of the tape may be varied. In fact, if the exact impact velocity is known, tape 50 could be designed to absorb energy at a constant rate.

The instant invention contemplates a one-shot type mechanism. That is, absorber 10 will function and operate only once and then must be rewound for a subsequent use. Once sphere 60 has been fully deployed with cable 70 fully extended and the energy absorbed by absorber 10, the visual study and drag measurement system is in operative position. Accordingly, the aerodynamic drag induced upon sphere 60 and the aerospace vehicle is transmitted to cantilever beam 80. Strain gages 92 pick up the strain from beam 80 and transmit them through terminals 88 and leads 90 to the required instrumentation 94. It has been found that painting or coloring sphere 60 provides the best means for the color or visual study. However, the use of aluminum for tape 50 in absorber 10 has a tendency to reflect light rays which cause a glare that hinders an accurate visual study. In order to overcome the glare caused by the aluminum, a coating, such as Alodine-100, is applied to provide tape 50 with a dull [illegible] reduced diameter sections 34 of shaft 30.

The alternative embodiment of spool 20, FIGS. 5 and 6, is similar to that shown in FIGS. 3 and 4. However, shaft 130 has an elongated radial slot 132 centrally located between end plates 122 and 124 that receives a small portion of the narrow section of tape 50. End plates 122 and 124 have convex hemispherical inner faces 126 and 128, respectively, prevented from moving inwardly by abutment 136 formed by reduced diameter end sections 134 of shaft 130. End sections 134 have threads 138 for receiving nuts 142 in counterbores 144 extending about central bore 140 in end plates 122 and 124. Although the precise degree of absorption ability is difficult, if not impossible, to compute, convex inner faces 126 and 128 retain tape 50 in a neatly wound package which may be more desirable under certain conditions.

As more clearly shown in FIG. 7, tape 50 has wide end portions 52 and gradually decreases in width to narrow central portion 54 at centerline 56. Wide end portions 52 are provided with attachment members 58 for attaching the opposite ends of tape 50 to cables or similar energy transmitting elements. It has been found that an operable and expediently formed attachment member can be constructed by folding a small portion 96 of ends 52 of tape 50 over a rod 94 to which loop or eye 58 is secured and then welding folded portion 96 to the original end 52. Since aperture 98 is provided at the fold for eye 58, it is free to swivel.

Referring now to FIG. 8 wherein is shown a novel visual study system and drag measuring mechanism incorporating absorber 10 of the instant invention. Drag member 60 is shown as an inflatable sphere made of gores 62 and end panels 64. Cable 68 of minimum length is attached at one end to absorber 10 and at the other end to sphere 60 at 66. The opposite end of absorber 10 is attached to connector 72 which is secured to cable 70. The other end of cable 70 is attached to hook 74 mounted on beam 80. Cable 70 is preferably a braided nylon line which has been found to operate quite successfully under the conditions for the tests to be made with the instant invention.

The drag measurement system is shown in FIG. 8 to include cantilever beam 80 which is machined from a single piece of material to have built-up end section 82 which is attached to support 84 by bolts 83. Support 84 is rigidly secured, as by welding, at 86 to the vehicle from which sphere 60 is to be dragged. Strain gages 90 are mounted on beam 80 immediately adjacent end section 82 and are connected by terminals 88 and leads 92 to instrumentation 94 for determining the strains in cantilever beam 80 and therefore the drag force caused by sphere 60. The instrumentation to which leads 92 are connected is of conventional design and within the purview of the art.

Operation

Drag member 60 is preferably a folded, inflatable, aluminum Mylar sphere that is packaged to be released by explosive bolts and ejected jack-in-the-box fashion. The drag of the folded sphere will gradually increase as it becomes inflated. The drag causes sphere 60 to float back and probably somewhat downward from the aerospace vehicle from which it is deployed. Energy absorber 10 is secured to sphere 60 at 66 and is deployed therewith as well as cable 70 which is unwound from a reel, such for example as the commercially available spin fishing type. In order to permit accurate measurement of the drag force, cable 70 would be wound about the reel in a fashion to permit cable 70 to be completely free from could overtake the aerospace vehicle and possibly become entangled therewith. Accordingly, energy absorber 10 is positioned near sphere 60 to absorb the energy resulting from the rearward flotation thereof. An ideal absorber 10 would be completely nonelastic and absorb energy over a sufficiently long period of time to prevent excessive buildup of forces within cable 70. In order to accomplish the ideal situation, absorber 10 is provided with a diminishing absorption rate.

The diminishing rate of absorption of absorber 10 is accomplished by designing tape 50 with relatively wide end sections that gradually taper to narrower central portion 54. Tape 50 is folded about centerline 56 to position ends 52 over one another. Should the alternative embodiment of spool 20 be utilized, a small portion of central section 54 is fitted into slot 32 in shaft 30. Tape 50 is then wound about shaft 30 and as the ends 52 are approached, tape 50 is held neatly coiled by inner plates 26 and 28 or convex faces 126 and 128 or end plates 22 and 24 or 122 and 124, respectively.

In view of the above-identified operation, it is seen that as the tensile forces between sphere 60 and cable 70 increase, tape 50 is unwound from spool 20. Tape 50 has been found to perform efficiently when made from soft aluminum sheet that tends to yield and retain its rolled shape. The material then has to yield again while being unrolled and most of the velocity or impact energy is expended in this way. However, some energy is expended in angularly accelerating the rolled mass and an additional amount is expended in linear acceleration of the tape. As can be seen from FIGS. 2 and 7, the tape gradually tapers to a smaller width as it unwinds as well as the configuration of the diameter of the rolled up absorber decreasing as tape 50 is unwound. This provides the diminishing absorption rate desired. It is apparent that the amount of energy to be absorbed will dictate the dimensions for tape 50 which may be readily calculated by the skilled artisan. For example, a different material may be used or the length, width or thickness of the tape may be varied. In fact, if the exact impact velocity is known, tape 50 could be designed to absorb energy at a constant rate.

The instant invention contemplates a one-shot type mechanism. That is, absorber 10 will function and operate only once and then must be rewound for a subsequent use. Once sphere 60 has been fully deployed with cable 70 fully extended and the energy absorbed by absorber 10, the visual study and drag measurement system is in operative position. Accordingly, the aerodynamic drag induced upon sphere 60 and the aerospace vehicle is transmitted to cantilever beam 80. Strain gages 92 pick up the strain from beam 80 and transmit them through terminals 88 and leads 90 to the required instrumentation 94. It has been found that painting or coloring sphere 60 provides the best means for the color or visual study. However, the use of aluminum for tape 50 in absorber 10 has a tendency to reflect light rays which cause a glare that hinders an accurate visual study. In order to overcome the glare caused by the aluminum, a coating, such as Alodine-100, is applied to provide tape 50 with a dull finish.

As indicated by the description of the invention hereinabove, it is readily apparent that the instant energy absorber will occupy a very small volume and withstand maximum vibration as well as being capable of operation in a space environment. Further, the instant absorber is not affected by several weeks dormancy prior to actual operation and possesses the feature of a diminishing absorption rate versus time during absorber operation over a wide range of impact energy magnitudes.

Obviously many modifications and variations of the subject invention are possible in the light of the above teachings. It is therefore to be understood that within the scope of the appended claims the invention may be practiced otherwise than as specifically described.

What is claimed as new and desired to be secured by Letters Patent of the United States is:

1. An impact energy absorber for preventing destruction of masses joined by a cable comprising: reel means for absorbing a portion of the energy as the masses are separated; tape means wound upon said reel means for absorbing the remainder of the energy causing separation of the masses; and said tape means being wider at the ends than the center thereof and made of a material tending to yield and retain its rolled shape, whereby there is a decreasing absorption rate for energy during operation of the absorber.

2. An energy absorber comprising: spool means including a shaft extending between end plates; an elongated tape gradually increasing in width from the center toward the ends thereof; said tape being of a material tending to yield and retain a rolled shape and folded at the center thereof; said center portion of said tape positioned adjacent said shaft; and said tape wrapped about said shaft whereby forces at the ends of said tape which tend to unwrap said tape are absorbed by the unwrapping of said tape and rotation of said spool means.

3. An energy absorber comprising: reel means including a shaft extending between end plates having convex inner faces; a slot through said shaft and located substantially centrally between said convex inner faces; an elongated tape gradually increasing in width from the center toward the ends thereof and being of a material tending to yield and retain a rolled shape; said tape folded at the center thereof; said center portion of said tape inserted into said slot; and said tape wrapped about said shaft whereby forces at the ends of said tape which tend to unwrap said tape are absorbed by the unwrapping of the tape and rotation of the reel.

4. An energy absorber comprising: reel means having a pair of end plates joined by a shaft; tape means of gradually greater width from the center to the ends thereof and being of a material tending to yield and retain its rolled shape; the narrower central portion of said tape means positioned adjacent said shaft; and said tape means disposed about said shaft between said end plates whereby forces tending to pull the ends of the tape means away from one anothe. are absorbed at a decreasing rate.

5. The energy absorber of claim 4 wherein said tape means is of one-piece soft aluminum.

6. The energy absorber of claim 4 wherein said tape means is of substantially greater dimension in length than in width.

7. The energy absorber of claim 6 wherein a centrally located radial slot extends through said shaft; and the narrower central portion of said tape means positioned in said slot.

8. The energy absorber of claim 6 wherein said tape means is of one-piece aluminum; and inner plates are in juxtaposition to the interior face of said end plates.

9. The energy absorber of claim 6 wherein said tape means is of one-piece aluminum; and said end plates having convexly curved surfaces on the shaft side thereof.

10. The energy absorber of claim 9 wherein said tape means includes attachment means on the ends thereof.

11. An energy absorption device comprising: a shaft having an elongated radial slot through the center thereof [illegible]

end plates each having a substantially flat face and a convex face; a bore extending through the center of each of said end plates for receiving said shaft therethrough; counterbores in said flat faces about said bores for nuts threaded upon the ends of said shaft to form a reel; an elongated energy absorber tape having a narrow central portion and gradually widening toward the ends thereof; said tape being of a material tending to yield and retain a rolled shape; loop elements secured to the ends of said tape for attaching cords thereto; said tape folded at the center thereof; the center of said tape inserted in said shaft slot; and said tape rolled around said shaft between said convexly shaped faces of said end plates whereby tensile forces applied to the ends of said tape are absorbed at a decreasing rate as said tape is unrolled from said shaft.

12. The energy absorption device of claim 11 wherein said tape is made from thin, flexible aluminum material.

13. An energy absorption device comprising: a shaft having threads on the ends thereof; a pair of disk-shaped inner plates; a pair of disk-shaped end plates of greater diameter than said inner plates; each of said plates having a central bore extending therethrough for receiving said shaft; counterbores in the outer surface of said end plates about said bores for nuts threaded upon the ends of said shaft to form a reel; an elongated energy absorber tape having a narrow central portion and gradually widening toward the ends thereof and being of a material tending to yield and retain its rolled shape; loop elements secured to the ends of said tape for attaching cords thereto; said tape folded at the center thereof; and said tape rolled around said shaft between said inner plates with the narrow central portion adjacent to said shaft whereby tensile forces applied to the ends of said tape are absorbed at a decreasing rate as said tape is unrolled from said shaft.

14. The energy absorption device of claim 13 wherein the threaded ends of said shaft are of less diameter than the central portion thereof to form abutments for preventing said plates from moving inwardly toward the center of said shaft.

15. The energy absorption device of claim 14 wherein said tape comprises thin, flexible aluminum material.

16. In the method of forming an energy absorber the steps of: folding at its center a tape having a narrow central portion gradually widening toward the ends thereof and being of a material tending to yield and retain its rolled shape; wrapping said tape around the shaft of a spool starting with the narrow central portion; and connecting two elements to be separated by attaching the ends of said tape thereto, whereby the energy of forces tending to separate the two elements are absorbed without destruction of the connection.

References Cited

UNITED STATES PATENTS

290,840	12/1883	Blakeslee	73—184 X
2,483,655	10/1949	Schultz.	
2,709,054	5/1955	Roth	73—143 X
2,768,068	10/1956	Juve et al.	73—15.6 X
2,785,775	3/1957	Stevinson.	
3,098,630	7/1963	Connors	244—113
3,126,072	3/1964	Johansson	188—1
3,217,838	11/1965	Peterson et al.	188—1

JAMES J. GILL, *Acting Primary Examiner.*

RICHARD C. QUEISSER, *Examiner.*

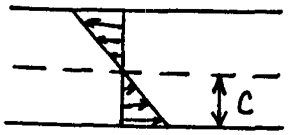
C. A. RUEHL, *Assistant Examiner.*

APPENDIX G

Aluminum Tape Design Calculations

APPENDIX G

ALUMINUM TAPE DESIGN CALCULATIONS



ASSUME ALUMINUM 1100-0

YIELD STRENGTH, σ_Y: AT 300°F $= 4.2 \times 10^3 \frac{lb}{IN^2}$

AT -320°F $= 6 \times 10^3 \frac{lb}{IN^2}$

ASSUME W = 1 INCH, 300°F

$F = \sigma_Y CW \Rightarrow C = \frac{F}{\sigma_Y W}$

$$C = \frac{75\ lb}{(4.2 \times 10^3 \frac{lb}{IN^2})(1\ IN)} = 0.018\ IN$$

$$t \approx 2C = 2(0.018\ IN) \Rightarrow t = 0.036\ IN$$

AT -320°F

$$F = (6 \times 10^3 \frac{lb}{IN^2})(0.018\ IN)(1\ IN) = 108\ lb$$

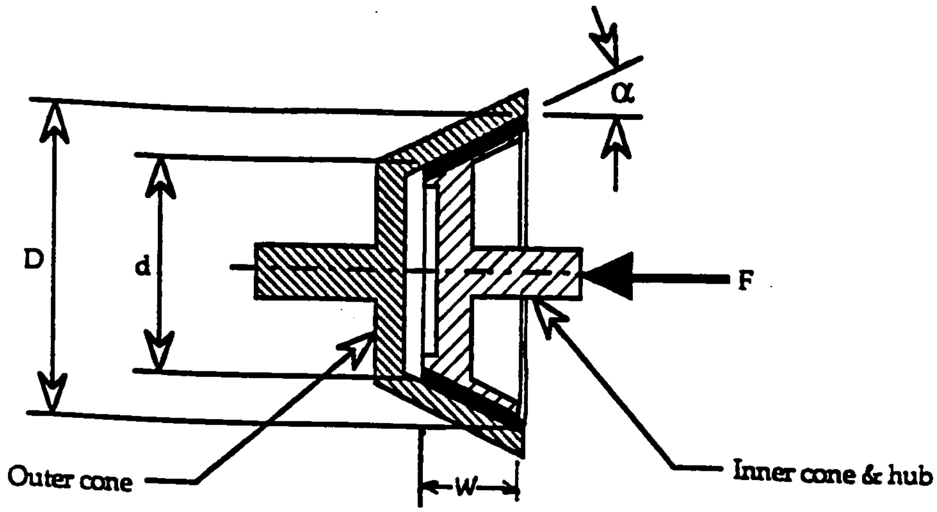
THEREFORE, A 1 IN WIDE, 0.036 IN THICK STRIP OF ALUMINUM WILL DEFORM WHEN A FORCE OF 75 - 108 lb IS APPLIED. THESE CALCULATIONS DO NOT ACCOUNT FOR STRAIN HARDENING OR FATIGUE THAT WILL OCCUR WITH REPEATED USE.

APPENDIX H

Cone Clutch Calculations

APPENDIX H

CONE CLUTCH CALCULATIONS



UNIFORM WEAR – FROM MARKS [1]

$$F = \frac{\pi P_A d}{2}(D-d)$$

$$T = \frac{\pi \mu P_A d}{8 \sin\alpha}(D^2-d^2)$$

WHERE P_A = MAXIMUM PRESSURE OCCURING AT $\frac{d}{2}$

ASSUMING $D = 2$ IN, $\alpha = 12^\circ$, $W = \frac{1}{2}$ IN,

$T = 75$ IN-lb, $d = 2 - 2(.106) = 1.787$ IN

$$P_A = \frac{8 \sin\alpha\, T}{\pi \mu d(D^2-d)} = \frac{8 \sin 12^\circ (75 \text{ IN-lb})}{\pi(.3)(1.787 \text{ IN})(2^2 - 1.787^2)}$$

$$P_A = 91.82 \text{ lb/IN}^2$$

$$F = \frac{\pi}{2}(91.82\ \text{lb/IN}^2)(1.787\ \text{IN})(2 - 1.787)\text{IN}$$

$$\Rightarrow F = 54.90\text{lb}$$

FROM JUVINALL [2]:

$$F = \frac{T \sin\alpha}{\mu\left(\frac{r_o + r_i}{2}\right)} = \frac{(75\ \text{IN}\cdot\text{lb})\sin 12^\circ}{(0.3)\left(\frac{2 + 1.787}{4}\right)\text{IN}}$$

$$\Rightarrow F = 54.90\ \text{lb}$$

APPENDIX H: References

1. Eugene A. Avallone and Theodore Baumeister III, eds. Marks'Standard Handbook for Mechanical Engineers. (New York: McGraw-Hill Book Co., 1978), p. 3-24.

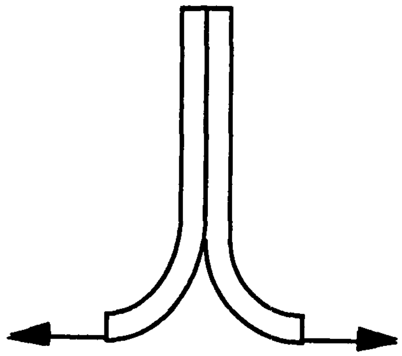
2. R. C. Juvinall and K. M. Marshek, Fundamentals of Machine Component Design, (New York: John Wiley and Sons, 1991), p. 674.

APPENDIX I

Velcro Calculations

APPENDIX I

VELCRO CALCULATIONS



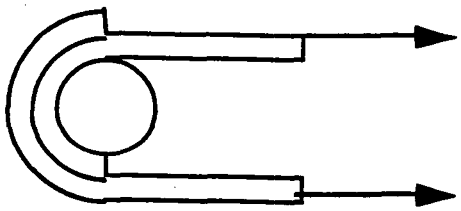
PEELING VELCRO

PEEL STRENGTH = $0.5 \frac{lb}{IN\ WIDTH}$

AMOUNT OF VELCRO TO HOLD 75 lb

$$WIDTH = \frac{75\ lb}{0.5 \frac{lb}{IN}} = 150\ IN$$

THEREFORE, $150\ IN^2$ OF VELCRO IS NEEDED TO HOLD 75 lb. THIS IS AN UNACCEPTABLE AMOUNT.



SHEARING VELCRO AROUND A CYLINDER

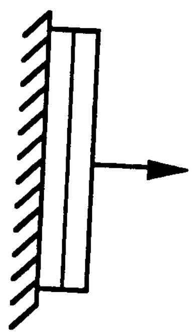
$$\frac{\pi D}{2} = 5\ IN \Rightarrow D = \frac{2(5\ IN)}{\pi}$$

$$D = 3.2\ IN$$

SHEAR STRENGTH = $15 \frac{lb}{IN^2}$

$$AREA = \frac{75\ lb}{15 \frac{lb}{IN^2}} = 5\ IN^2$$

A 1 INCH WIDE STRIP OF VELCRO NEEDS TO OVERLAP AROUND A CYLINDER WITH A 3.2 IN DIAMETER TO HOLD 75 lb.

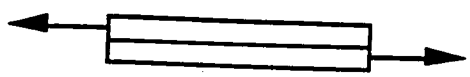


VELCRO IN TENSION

TENSION STRENGTH $= 3 \frac{lb}{IN^2}$

$$AREA = \frac{75\,lb}{3\,\frac{lb}{IN^2}} = 25\,IN^2$$

A 5 IN × 5 IN PLATE OF VELCRO IS NEEDED TO HOLD 75 lb.



SHEARING VELCRO

SHEAR STRENGTH $= 7.5 \frac{lb}{IN^2}$

$$AREA = \frac{75\,lb}{7.5\,\frac{lb}{IN^2}} = 10\,IN^2$$

APPENDIX I: References

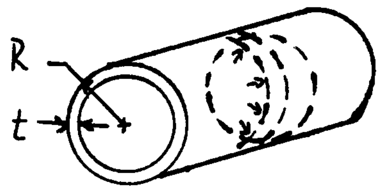
1. Velcro Brand Fastening Products and Systems, brochure, (Velcro USA, Inc.), p. 3963.

APPENDIX J

Ball Through Tube Calculations

APPENDIX J

BALL THROUGH TUBE CALCULATIONS



ASSUME: STEEL BALL:
(IGNORE DEFORMATION)

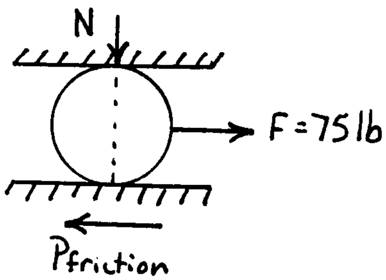
ALUMINUM TUBE:

$E = 10 \times 10^6 \ lb/IN^2$ $\quad$ $R = .55 \ IN$

$\nu = .33$ $\quad$ $t = .05 \ IN$

$$\lambda = \left[\frac{3(1-\nu^2)}{R^2 t^2}\right]^{1/4} = \left[\frac{3(1-(.33)^2)}{(.55)^2(.05)^2}\right]^{1/4} = 7.71 \ IN^{-1}$$

$$D = \frac{Et^3}{12(1-\nu^2)} = \frac{(10 \times 10^6)(.05)^3}{12(1-(.33)^2)} = 116.9 \ IN \cdot lb$$



COEFFICIENT OF FRICTION, ALUMINUM ON MILD STEEL:

$\mu_{STATIC} = .61$

$\mu_{DYNAMIC} = .47$

CIRCUMFERENCE $= \pi D = \pi(1) = \pi \ IN$

$$P_{FRICTION} = \frac{75 \ lb}{\pi \ IN} = 23.87 \ lb/IN$$

$$N_{DYNAMIC} = \frac{23.87 \ lb/IN}{.47} = 50.8 \ lb/IN = -p, \text{ RADIAL LOAD}$$

RADIAL DEFLECTION :

$$y = \frac{-p}{8 D \lambda^3} = \frac{-(50.8 \text{ lb/IN})}{8(116.9 \text{ IN} \cdot \text{lb})(7.71 \text{ IN}^{-1})^3}$$

$$y = 0.00012 \text{ IN}$$

THEREFORE, THE DIAMETER OF THE BALL, d,

$$d = 1.00024 \text{ IN}$$

APPENDIX J: References

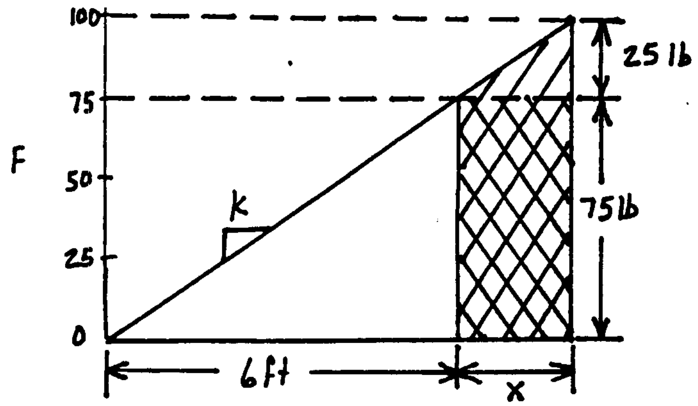
1. R. J. Roark, Formulas for Stress and Strain, (New York: McGraw-Hill, Inc. 1975), p. 459.

APPENDIX K

Compression Spring Calculations

APPENDIX K

COMPRESSION SPRING CALCULATIONS

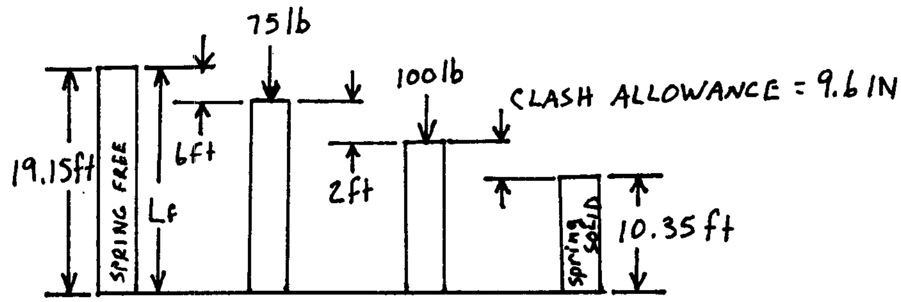


ENERGY ABSORBED = AREA UNDER THE CURVE

$$75X + \frac{1}{2}(25)X = 175 \text{ ft-lb}$$

$$87.5\,X = 175 \Rightarrow X = 2\text{ ft}$$

$$kX = 25\text{ lb} \Rightarrow k = \frac{25\text{ lb}}{2\text{ ft}} = 12.5\,\frac{\text{lb}}{\text{ft}} = 1.04\,\frac{\text{lb}}{\text{IN}}$$



ASSUME: 302 SS SPRING WIRE, $S_y = 75\times10^3\ \frac{\text{lb}}{\text{IN}^2}$

$$S_{sy} = .585 S_y = 43.5\times10^3\ \frac{\text{lb}}{\text{IN}^2}$$

$$c = \frac{D}{d} = 12, \quad d = 0.08\text{ IN}, \quad D = 12(.08) = .96\text{ IN}$$

$$\text{CLASH ALLOWANCE} = 0.1\left(\frac{100\text{ lb}}{12.5\text{ lb/ft}}\right) = 0.8\text{ft} = 9.6\text{ IN}$$

$$F_{SOLID} = 100\ lb + 12.5\ \frac{lb}{ft}(0.8\ ft) = 110\ lb$$

$K_s \approx 1.04$ = STRESS CORRECTION FACTOR [1]

$$\tau_{SOLID} = 0.45 S_u = 0.45(250\ ksi) = 112.5\ ksi$$

$$\tau_{SOLID} = \frac{8 F_{SOLID} D}{\pi d^3} K_s \Rightarrow d^3 = \frac{8 F_{SOLID} D K_s}{\pi \tau_{SOLID}} \quad [2]$$

$$d^3 = \frac{8(110\ lb)(1\ IN)(1.04)}{(122.5 \times 10^3\ \frac{lb}{IN^2})\pi} \Rightarrow d = 0.1373\ IN$$

CHOOSE $d = .125\ IN$

$$\tau_{SOLID} = \frac{8 F_{SOLID} C K_s}{\pi d^2} \Rightarrow C K_s = \frac{\tau_{SOLID} \pi d^2}{8 F_{SOLID}}$$

$$C K_s = \frac{(112.5 \times 10^3\ \frac{lb}{IN^2})(.125\ IN)^2 \pi}{8(110\ lb)} = 6.27$$

$$C = \frac{D}{d} = 5.5 \Rightarrow D = (.125)(5.5) = .6875\ IN$$

$$D + d = .6875 + .125 = .8125$$

$$k = \frac{d^4 G}{8 D^3 N} \Rightarrow N = \frac{d^4 G}{8 D^3 k} = \frac{(.125\ IN)^4 (11 \times 10^6\ \frac{lb}{IN^2})}{8(.6875\ IN)(1.04\ lb/IN)}$$

$N = 991.74$ = NUMBER OF ACTIVE TURNS

$N_T = N + 2 = 993.74$ = TOTAL NUMBER OF TURNS

$$L_s = N_t d = (994)(.125\ IN) = 124.25\ IN = 10.35\ ft$$

$$\frac{110\ lb}{12.5\ lb/ft} = 8.8\ ft \qquad L_f = 10.35 + 8.8 = 19.15\ ft$$

THEREFORE, A COMPRESSION SPRING IS NOT SUITABLE

APPENDIX K: References

1. R. C. Juvinall and K. M. Marshek, Fundamentals of Machine Component Design, (New York: John Wiley and Sons, 1991), p. 462.

2. E. A. Avallone and T. Baumeister III, eds., Marks' Standard Handbook for Mechanical Engineers. (New York: McGraw-Hill Book Co., 1978), p. 8-74.

APPENDIX L

Hunter Spring Catalog

APPENDIX L

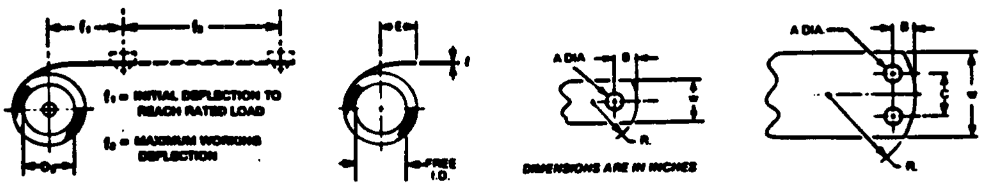
Stock spring products

Neg'ator extension springs

The NEG'ATOR constant force extension springs listed below are available from stock for immediate shipment from established distributors or directly from the factory (see price list SP-2 for ordering information). These springs can be used by designers to assist in the development of product prototypes or supplied in small production quantities.

Application of these springs is detailed on page 4 of this brochure. All stock extension springs are supplied in Type 301 stainless steel with hole punches in the free ends (see schematic below for fastening). The wide variety of sizes and load capacities available permit you to closely match a spring to your application at a considerable savings over a custom-designed spring (minor modifications can be made when required).

EXTENSION SPRING SELECTION TABLE



PART NUMBER	LOAD LBS ±10%	DRUM* DIA (D2)	FREE I.D. (REF)	f_1	f_2	(t) THICKNESS	(W) WIDTH	BAND LENGTH	NO OF HOLES	R.	A DIA.	B	C	E
FATIGUE LIFE—2,500 CYCLES MINIMUM														
SH3C10	.37	.281	.210	.30	8	.003	.187	10	1	5/32	.095	.098	–	5/32
SH4C14	.40	.340	.291	.525	12	.004	.187	14	1	5/32	.095	.098	–	5/32
SH4D14	.66	.340	.291	.525	12	.004	.250	14	1	7/32	.131	.187	–	7/32
SH4E14	.83	.340	.291	.525	12	.004	.312	14	1	7/32	.131	.187	–	7/32
SH5E15	1.03	.438	.383	.600	12	.005	.312	15	1	7/32	.131	.187	–	7/32
SH6F21	1.40	.523	.438	.786	16	.006	.375	21	1	1/4	.191	.344	–	1/4
SH6G21	1.87	.523	.438	.786	16	.006	.500	21	1	1/4	.191	.344	–	1/4
SH8G28	2.83	.697	.581	1.05	24	.008	.500	28	1	1/4	.191	.344	–	3/8
SH8J28	3.29	.697	.581	1.05	24	.008	.625	28	1	1/4	.191	.344	–	3/8
SH10J29	4.12	.873	.727	1.31	24	.010	.625	29	1	1/4	.191	.344	–	7/16
SH10K29	4.86	.873	.727	1.31	24	.010	.750	29	1	1/4	.191	.344	–	7/16
SH12K35	5.94	1.050	.873	1.56	30	.012	.750	35	1	1/4	.191	.344	–	1/2
SH12P38	7.92	1.050	.873	1.56	30	.012	1.000	38	1	1 5/32	.193	.375	–	1/2
SH16P38	10.60	1.400	1.160	2.1	30	.016	1.000	38	1	1 5/32	.193	.375	–	5/8
SH20R47	16.50	1.750	1.450	2.6	36	.020	1.250	47	1	1 5/32	.193	.375	–	3/4
SH20S47	19.80	1.750	1.450	2.62	36	.020	1.500	47	2	1 5/32	.250	.625	.700	3/4
SH25S48	24.80	2.180	1.820	3.27	36	.025	1.500	48	2	1 5/32	.250	.625	.700	1 1/8
SH25U48	33.00	2.180	1.820	3.27	36	.025	2.000	48	2	1 5/32	.250	.625	.700	1 1/8
SH31U58	40.90	2.710	2.260	4.0	42	.031	2.000	58	2	1 5/32	.250	.625	.700	1 1/8
FATIGUE LIFE—13,000 CYCLES MINIMUM														
SK4D15	.32	.533	.444	.80	12	.004	.250	15	1	7/32	.131	.187	–	1/4
SK4E15	.40	.533	.444	.80	12	.004	.312	15	1	7/32	.131	.187	–	1/4
SK5E16	.40	.666	.5[illegible]	1.00	12	.005	.312	16	1	7/32	.131	.187	–	5/16
SK6F23	.71	.796	.666	1.20	18	.006	.375	23	1	1/4	.191	.344	–	3/8
SK6G23	.95	.798	.6[illegible]	1.20	18	.006	.500	23	1	1/4	.191	.344	–	3/8
SK8G30	1.26	1.06	.888	1.59	24	.008	.500	30	1	1/4	.191	.344	–	1/2
SK8J30	1.58	1.06	.888	1.59	24	.008	.625	30	1	1/4	.191	.344	–	1/2
SK10J32	1.98	1.33	1.11	2.00	24	.010	.625	32	1	1/4	.191	.344	–	5/8
SK10K32	2.37	1.33	1.11	2.00	24	.010	.750	32	1	1/4	.191	.344	–	5/8
SK12K40	2.84	1.59	1.33	2.38	30	.012	.750	40	1	1/4	.191	.344	–	3/4
SK12P40	3.79	1.59	1.33	2.38	30	.012	1.000	40	1	1 1/32	.193	.375	–	3/4
SK15P42	4.74	1.99	1.66	2.98	30	.015	1.000	42	1	1 1/32	.193	.375	–	1
SK16P47	5.00	2.40	2.00	3.60	32	.016	1.000	47	1	1 1/32	.193	.375	–	1 1/8
SK20S52	9.48	2.66	2.22	3.97	36	.020	1.500	52	2	1 1/32	.250	.625	.700	1 1/4
SK25S57	11.9	3.33	2.78	5.00	36	.025	1.500	57	2	1 1/32	.250	.625	.700	1 1/2
SK25U57	16.8	3.33	2.78	5.00	36	.025	2.000	57	2	1 1/32	.250	.625	.700	1 1/2
SK31U66	19.8	4.12	3.44	6.10	42	.031	2.000	66	2	1 1/32	.250	.625	.700	2

APPENDIX M

Weighting Factors

APPENDIX M

Weighting Factors

Decision Criteria	Tally Marks	Weighting Factor
Number of cycles	•••••	0.139
Ease of resetting	•••	0.083
Space materials	••••••••	0.222
Extended strength	••••••	0.167
Size and weight	•	0.028
Signal astronaut	••••	0.111
Simplicity	••	0.056
Reliability	•••••••	0.194
Total	**36**	**1.000**

APPENDIX N

Decision Matrix

APPENDIX N
Decision Matrix

Design Considerations / Weighting Factors	Design Parameters: Space Resistant	Reliability	Tether Strength	Number of Uses	Warning Capability	Ease of Use/Reset	Simplicity of Design	Minimize Vol./Mass	Sum of Products
	0.222	0.194	0.167	0.139	0.111	0.083	0.056	0.028	1.000
Neg'ator Spring	8 / 1.778	8 / 1.556	9 / 1.500	9 / 1.250	7 / 0.777	8 / 0.667	5 / 0.278	7 / 0.194	**8.00**
Cone Brake	8 / 1.778	7 / 1.361	7 / 1.167	8 / 1.111	7 / 0.777	8 / 0.667	5 / 0.278	7 / 0.194	**7.33**
Velcro Reel	7 / 1.556	7 / 1.361	9 / 1.500	6 / 0.833	7 / 0.777	8 / 0.667	7 / 0.389	8 / 0.222	**7.30**
Bending Mat'l	8 / 1.778	4 / 0.778	6 / 1.000	7 / 0.972	7 / 0.777	7 / 0.583	8 / 0.444	8 / 0.222	**6.55**
Layered Velcro	6 / 1.333	6 / 1.167	6 / 1.000	8 / 1.111	5 / 0.555	4 / 0.333	5 / 0.278	4 / 0.111	**5.88**
Ball Thru Tube	5 / 1.111	4 / 0.778	6 / 1.000	6 / 0.833	5 / 0.555	2 / 0.167	6 / 0.333	4 / 0.111	**4.88**

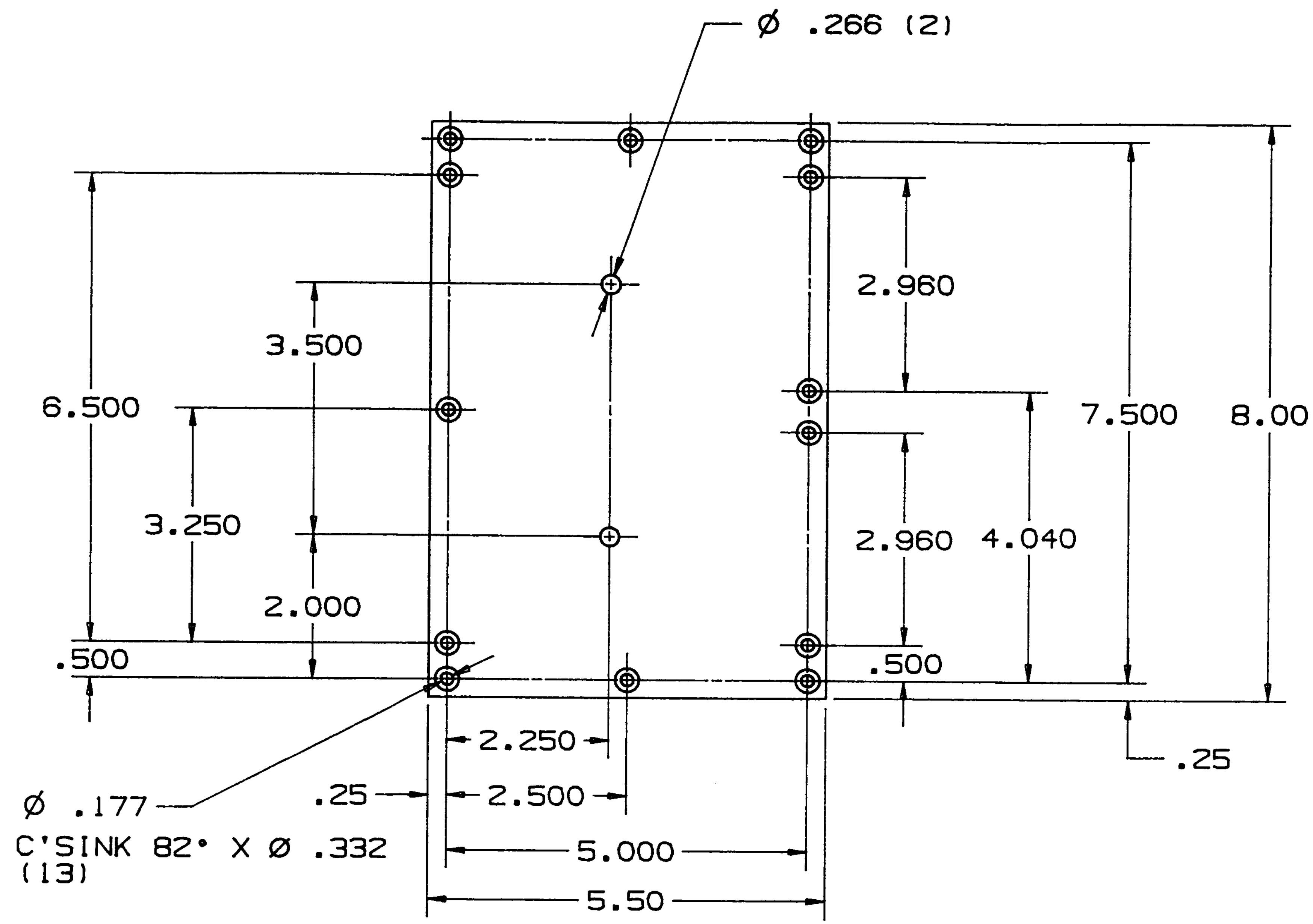
APPENDIX O

Detail Drawings

FOLDOUT FRAME 1.

FOLDOUT FRAME 2.

REV	ECO	DESCRIPTION	DRFT	CHKD
1		PROTOTYPE REVISED ON 10-NOV-91 AT 19: 9	DFC	



DESIGN REF

NEXT ASSY

DETACHED LISTS

MM(INCH)
25.4MM=1 INCH

TOLERANCES EXCEPT AS NOTED

MM	INCHES
0 ±	.0 ±
.0 ±	.00 ± 0.02
.00 ±	.000± 0.005

ANGLES ± 1°

MATERIAL .250 THK
6061 T6 ALUMINUM

FINISH

DFTG D. F. CRONCH	DATE 11-10-91	
CHKD	DATE	
	DATE	
MFG	DATE	
APPVL	DATE	

INTERPRET PER ANSI Y14.5M-1982

THIRD ANGLE PROJECTION

MECHANICAL ENGINEERING DESIGN PROJECTS
The University of Texas at Austin

TITLE

COVER PLATE

FSCM NO. | SIZE B | DRAWING NO. USRA-TETHER-01 | REV 1

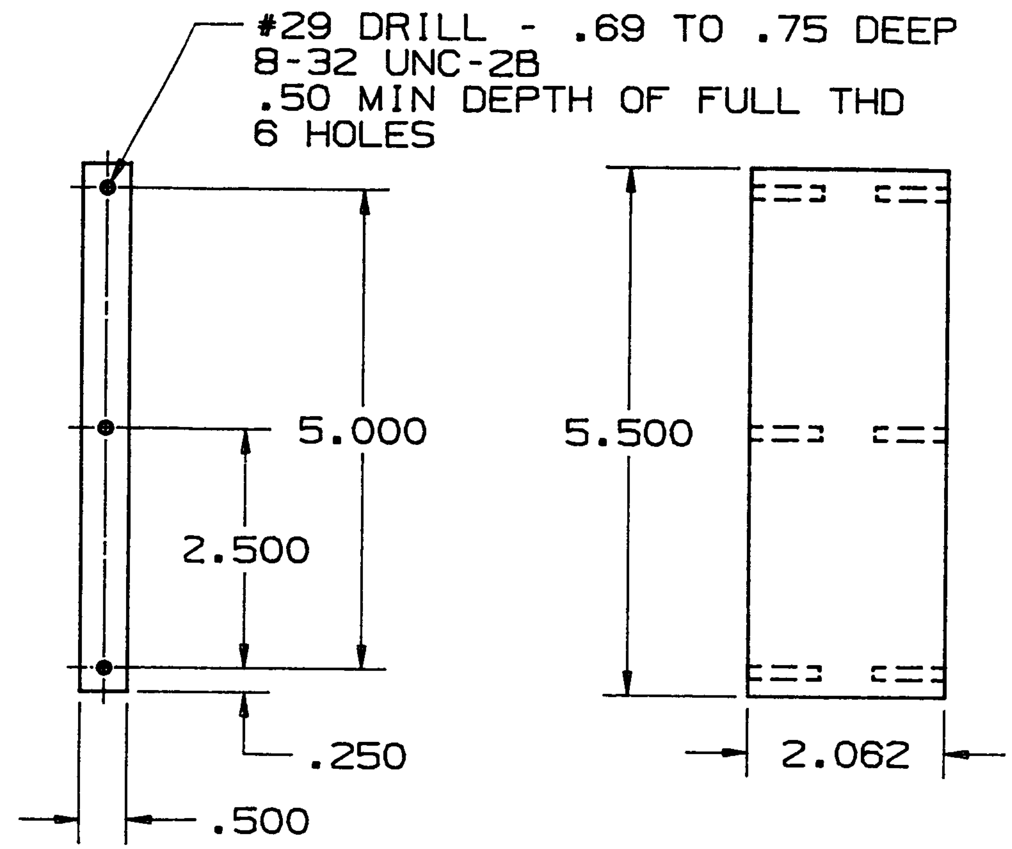
DO NOT SCALE DRAWING | DET LISTS ☐ YES ☐ NO | SHT 1 OF 1

DRAWING NO.
USRA-TETHER-01

FOLDOUT FRAME 1.

FOLDOUT FRAME 2.

REVISION RECORD				
REV	ECO	DESCRIPTION	DRFT	CHKD
1		PROTOTYPE REVISED ON 10-NOV-91 AT 19: 9	DFC	



DESIGN REF

NEXT ASSY

DETACHED LISTS

MM(INCH)
25.4MM=1 INCH

TOLERANCES EXCEPT AS NOTED

MM	INCHES
0 ±	.0 ±
.0 ±	.00 ± 0.02
.00 ±	.000± 0.005

ANGLES ± 1°

MATERIAL
6061 T6 ALUMINUM

FINISH

DFTG D. F. CRONCH	DATE 11-10-91
CHKD	DATE
	DATE
MFG	DATE
APPVL	DATE

INTERPRET PER ANSI Y14.5M-1982

THIRD ANGLE PROJECTION

MECHANICAL ENGINEERING DESIGN PROJECTS
The University of Texas at Austin

TITLE

SIDE PLATE

FSCM NO.	SIZE	DRAWING NO.	REV
	B	USRA-TETHER-02	1

DO NOT SCALE DRAWING | DET LISTS ☐ YES ☐ NO | SHT 1 OF 1

DRAWING NO. USRA-TETHER-02

02

FOLDOUT FRAME 1.

FOLDOUT FRAME 2.

REVISION RECORD

REV	ECO	DESCRIPTION	DRFT	CHKD
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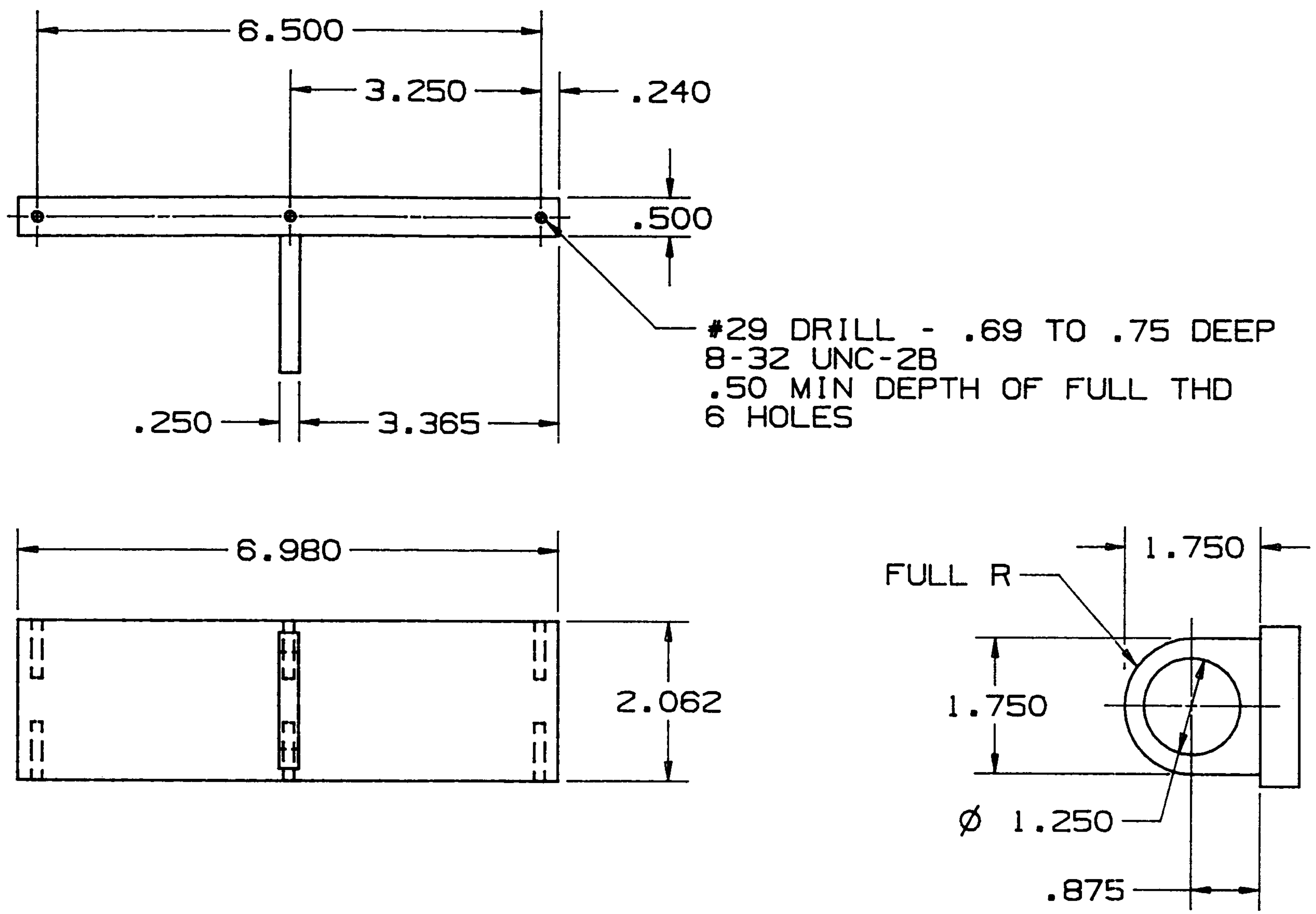

DESIGN REF

NEXT ASSY

MM (INCH) 25.4MM=1 INCH	DFTG D. F. CRONCH	DATE 11-14-91	MECHANICAL ENGINEERING DESIGN PROJECTS The University of Texas at Austin
TOLERANCES EXCEPT AS NOTED	CHKD	DATE	
MM: 0 ±, .0 ±, .00 ± / INCHES: .0 ±, .00 ± 0.02, .000± 0.005		DATE	TITLE TETHER POINT PLATE
ANGLES ± 1°	MFG	DATE	
	APPVL	DATE	
MATERIAL 6061 T6 ALUMINUM	INTERPRET PER ANSI Y14.5M-1982		FSCM NO. / SIZE B / DRAWING NO. USRA-TETHER-03 / REV 1
FINISH	THIRD ANGLE PROJECTION		DO NOT SCALE DRAWING / DET LISTS ☐ YES ☐ NO / SHT 1 OF 1

DETACHED LISTS

DRAWING NO. USRA-TETHER-03

03

FOLDOUT FRAME

FOLDOUT FRAME 2.

DRAWING NO. USRA-TETHER-04

REVISION RECORD				
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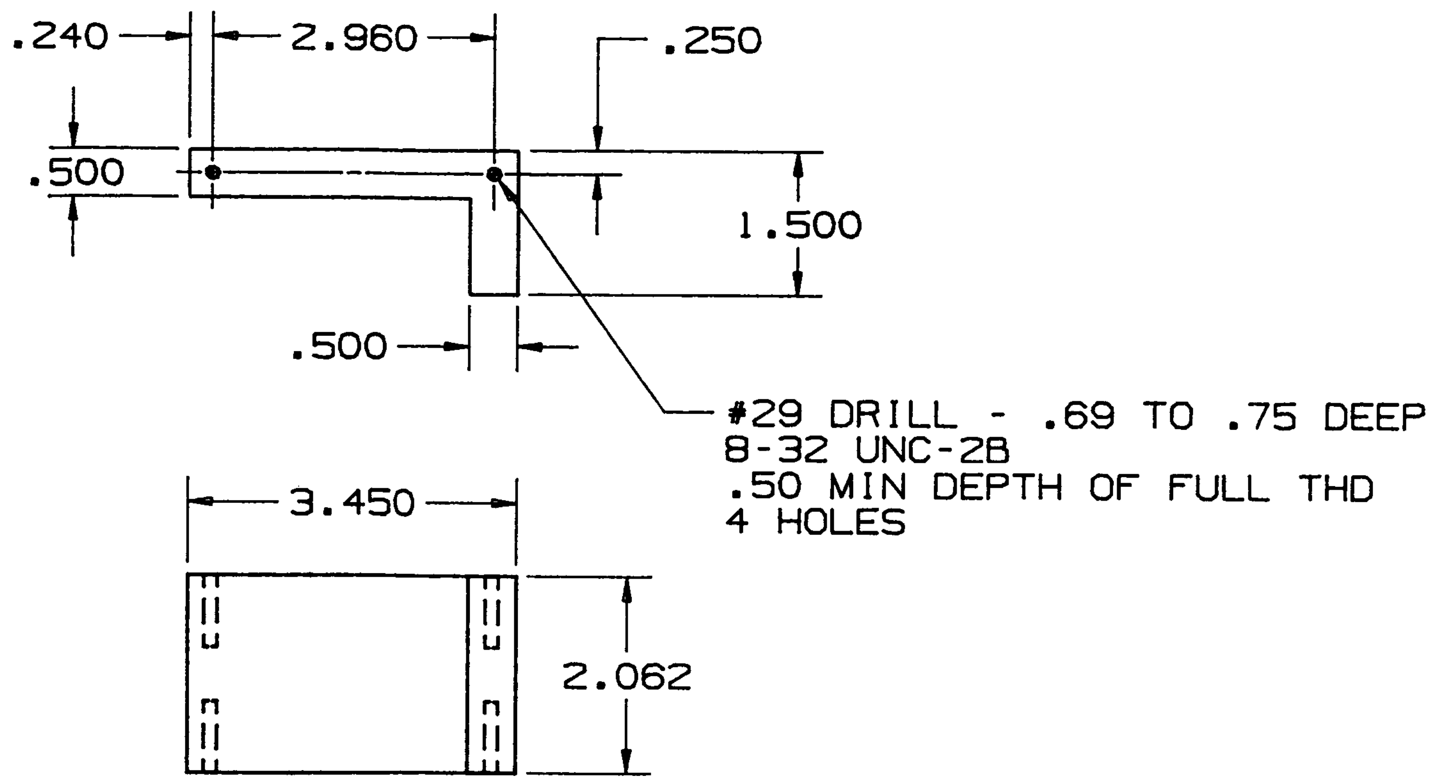


DESIGN REF

NEXT ASSY

DETACHED LISTS

MM (INCH)
25.4MM=1 INCH

TOLERANCES EXCEPT AS NOTED

MM	INCHES
0 ±	.0 ±
.0 ±	.00 ± 0.02
.00 ±	.000± 0.005

ANGLES ± 1°

MATERIAL
6061 T6 ALUMINUM

FINISH

DFTG D. F. CRONCH	DATE 11-14-91
CHKD	DATE
	DATE
MFG	DATE
APPVL	DATE

INTERPRET PER ANSI Y14.5M-1982

THIRD ANGLE PROJECTION

MECHANICAL ENGINEERING DESIGN PROJECTS
The University of Texas at Austin

TITLE
GRIP ROLLER L-PLATE

FSCM NO. | SIZE B | DRAWING NO. USRA-TETHER-04 | REV 1

DO NOT SCALE DRAWING | DET LISTS ☐ YES ☐ NO | SHT 1 OF 1

04

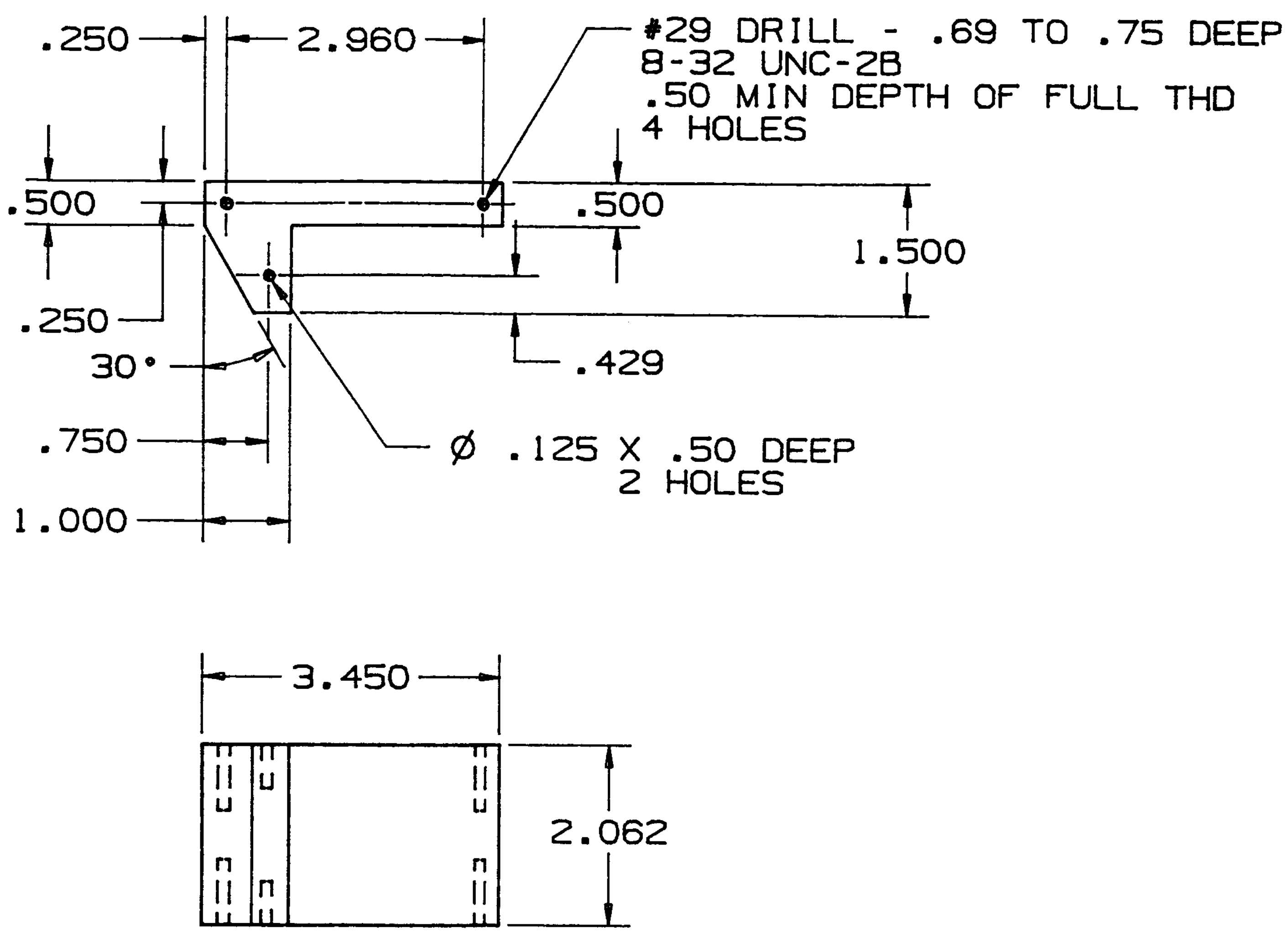
FOLDOUT FRAME 1

FOLDOUT FRAME 2

DRAWING NO. USRA-TETHER-05

REVISION RECORD

REV	ECO	DESCRIPTION	DRFT	CHKD
1		PROTOTYPE REVISED ON 14-NOV-91 AT 22:37	DFC	



DESIGN REF

NEXT ASSY

DETACHED LISTS

MM (INCH)
25.4MM=1 INCH

TOLERANCES EXCEPT AS NOTED

MM	INCHES
0 ±	.0 ±
.0 ±	.00 ± 0.02
.00 ±	.000± 0.005

ANGLES ± 1°

MATERIAL
6061 T6 ALUMINUM

FINISH

DFTG	D. F. CRONCH	DATE	11-14-91
CHKD		DATE	
		DATE	
MFG		DATE	
APPVL		DATE	

INTERPRET PER ANSI Y14.5M-1982

THIRD ANGLE PROJECTION

MECHANICAL ENGINEERING DESIGN PROJECTS
The University of Texas at Austin

TITLE

GRIP ROLLER MOUNTING PLATE

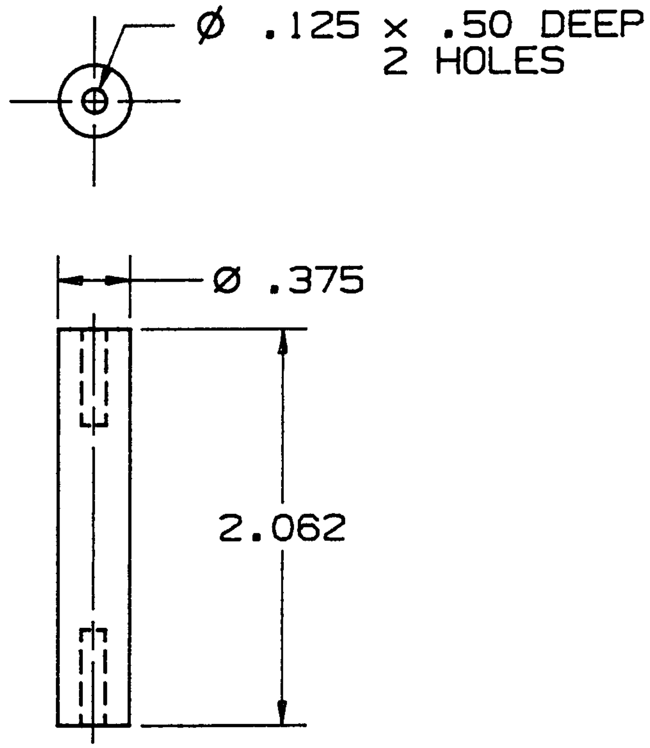
FSCM NO. | SIZE B | DRAWING NO. USRA-TETHER-05 | REV 1

DO NOT SCALE DRAWING | DET LISTS ☐ YES ☐ NO | SHT 1 OF 1

05

REVISION RECORD				
REV	ECO	DESCRIPTION	DRFT	CHKD
1		PROTOTYPE REVISED ON 15-NOV-91 AT 9:38	DFC	

DRAWING NO. USRA-TETHER-06



NOTES:

1. MAT'L: 6061 T6 ALUMINUM

2. TOLERANCES: .00 ±0.02
.000±0.005

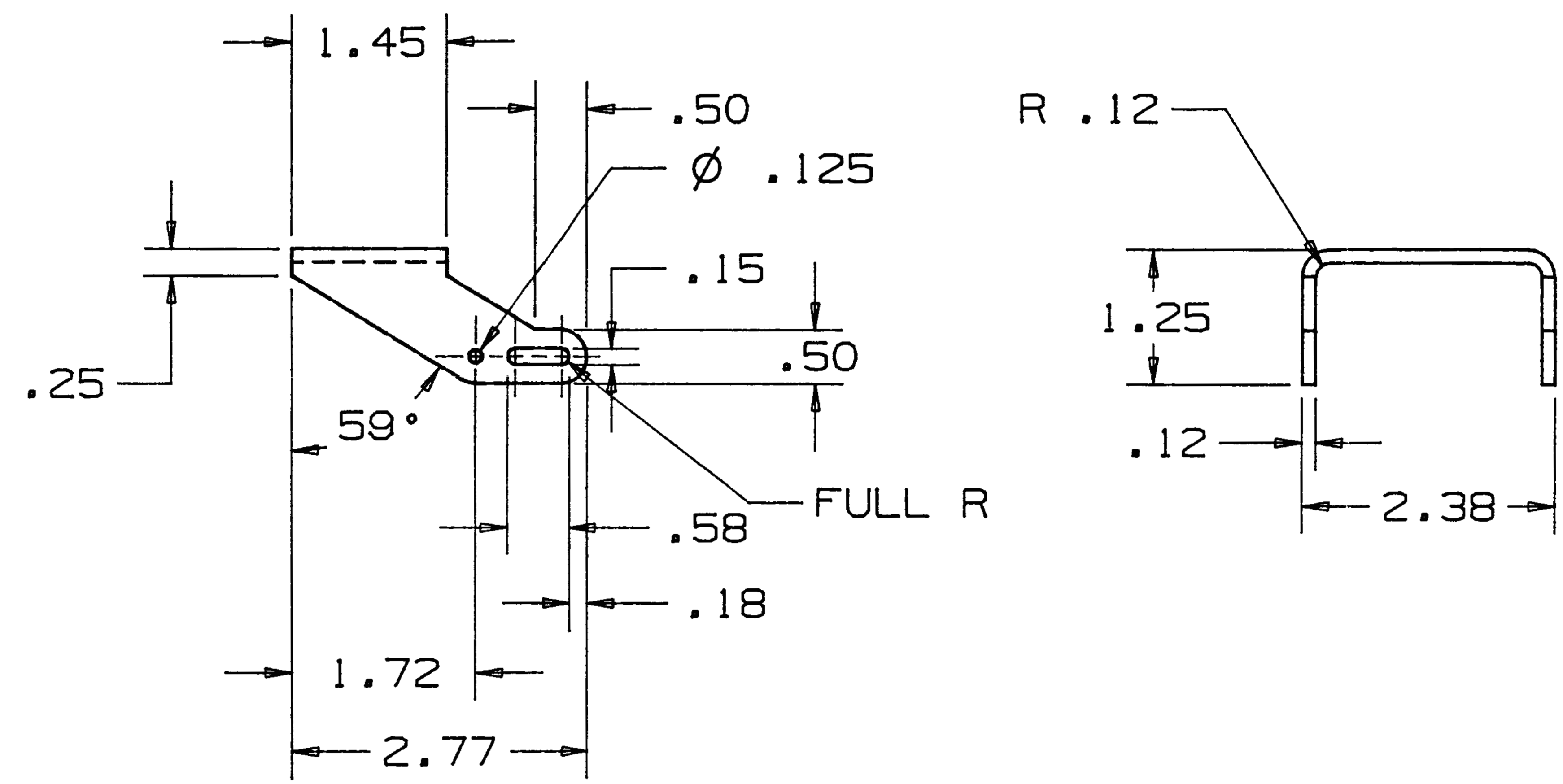
DFTG D. F. CRONCH	DATE 11-15-91	MECHANICAL ENGINEERING DESIGN PROJECTS The University of Texas at Austin	
CHKD	DATE		
	DATE	TITLE GRIP ROLLER	
MFG	DATE		
APPVL	DATE		
INTERPRET PER ANSI Y14.5M-1982		FSCM NO. / SIZE A / DRAWING NO. USRA-TETHER-06	REV 1
THIRD ANGLE PROJECTION		DO NOT SCALE DRAWING / DET LISTS ☐ YES ☐ NO	SHT 1 OF 1

FOLDOUT FRAME 1.

FOLDOUT FRAME 2.

REV	ECO	DESCRIPTION	DRFT	CHKD
1		PROTOTYPE REVISED ON 16-NOV-91 AT 20:14	DFC	

REVISION RECORD



DRAWING NO. USRA-TETHER-07

DESIGN REF

NEXT ASSY

DETACHED LISTS

MM (INCH)
25.4MM=1 INCH

TOLERANCES EXCEPT AS NOTED

MM	INCHES
0 ±	.0 ±
.0 ±	.00 ± 0.02
.00 ±	.000± 0.005

ANGLES ± 2°

MATERIAL .12 THK
6061 T6 ALUMINUM

FINISH

DFTG D. F. CRONCH	DATE 11-16-91
CHKD	DATE
	DATE
MFG	DATE
APPVL	DATE

INTERPRET PER ANSI Y14.5M-1982

THIRD ANGLE PROJECTION

MECHANICAL ENGINEERING DESIGN PROJECTS
The University of Texas at Austin

TITLE

GRIP ROLLER LEVER

FSCM NO.	SIZE	DRAWING NO.	REV
	B	USRA-TETHER-07	1

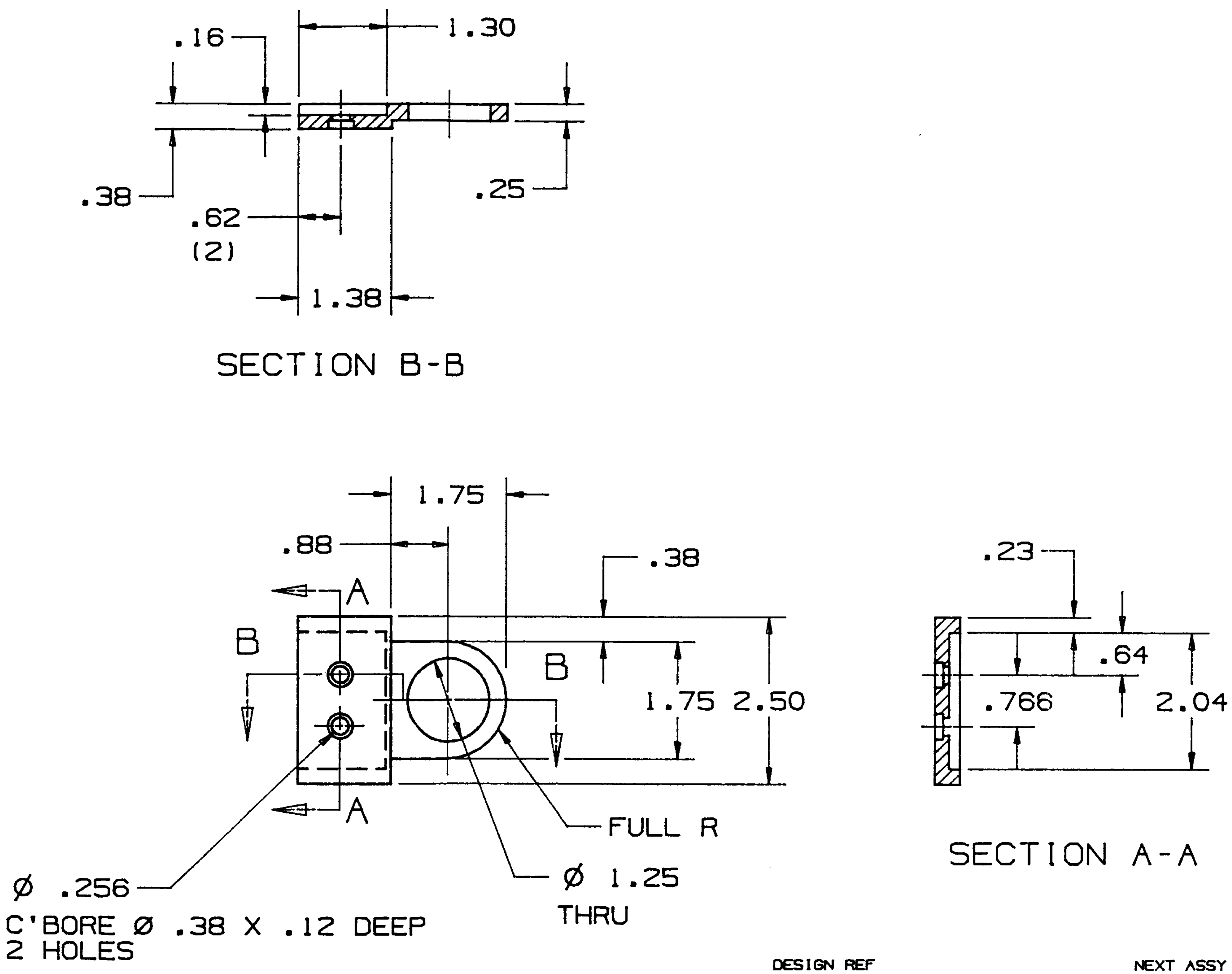
DO NOT SCALE DRAWING | DET LISTS ☐ YES ☐ NO | SHT 1 OF 1

FOLDOUT FRAME 1.

FOLDOUT FRAME 2.

DRAWING NO. USRA-TETHER-08

REVISION RECORD				
REV	ECO	DESCRIPTION	DRFT	CHKD
1		PROTOTYPE REVISED ON 17-NOV-91 AT 12:57	DFC	



DESIGN REF

NEXT ASSY

DETACHED LISTS

MM(INCH)
25.4MM=1 INCH

TOLERANCES EXCEPT AS NOTED

MM	INCHES
0 ±	.0 ±
.0 ±	.00 ± 0.02
.00 ±	.000± 0.005

ANGLES ± 1°

MATERIAL
6061 T6 ALUMINUM

FINISH

DFTG D. F. CRONCH	DATE 11-17-91
CHKD	DATE
	DATE
MFG	DATE
APPVL	DATE

INTERPRET PER ANSI Y14.5M-1982

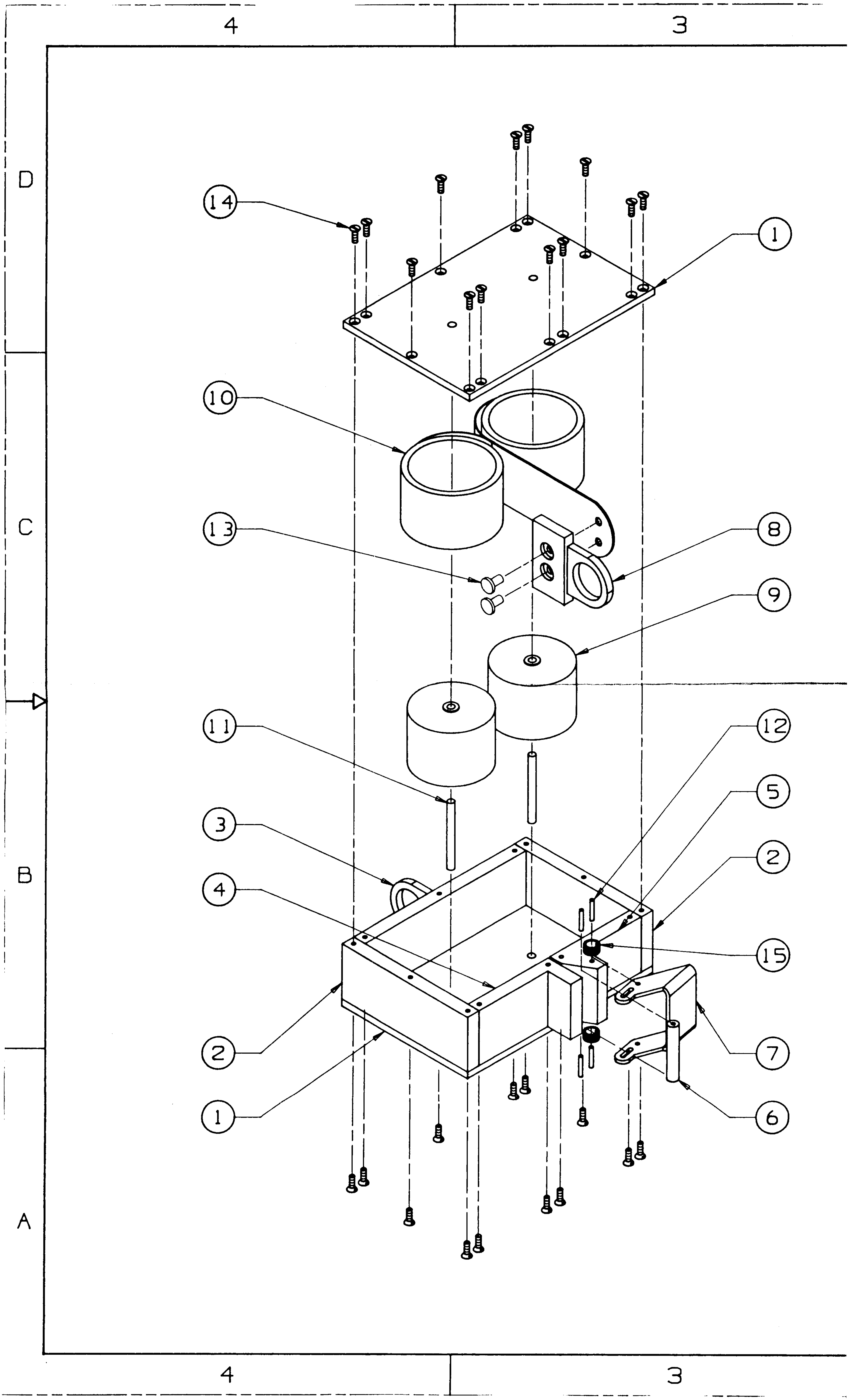
THIRD ANGLE PROJECTION

MECHANICAL ENGINEERING DESIGN PROJECTS
The University of Texas at Austin

TITLE
TETHER POINT
SPRING ENDPIECE

FSCM NO.	SIZE	DRAWING NO.	REV
	B	USRA-TETHER-08	1

DO NOT SCALE DRAWING | DET LISTS ☐ YES ☐ NO | SHT 1 OF 1


4
3
D
C
B
A
14
1
10
13
8
9
11
12
5
3
2
4
15
7
2
6
1
4
3

REVISION RECORD				
REV	ECO	DESCRIPTION	DRFT	CHKD
1		PROTOTYPE REVISED ON 19-NOV-91 AT 10: 7	DFC	

ITEM	NAME	QTY	PART NUMBER
15	TORSION SPRING	2	
14	8-32 UNC X .5 FL HD CAP SC	26	
13	RIVET, Ø.25 X .50	2	
12	DOWEL PIN, Ø.125 X .75	4	
11	DOWEL PIN, Ø.250 X 2.50	2	
10	CONSTANT FORCE SPRING	2	HUNTER SPRING SH31U56
9	SPRING SPOOL	2	USRA-TETHER-09
8	TETHER POINT SPRING ENDPIECE	1	USRA-TETHER-08
7	GRIP ROLLER LEVER	1	USRA-TETHER-07
6	GRIP ROLLER	1	USRA-TETHER-06
5	GRIP ROLLER MOUNTING PLATE	1	USRA-TETHER-05
4	GRIP ROLLER L-PLATE	1	USRA-TETHER-04
3	TETHER POINT PLATE	1	USRA-TETHER-03
2	SIDE PLATE	2	USRA-TETHER-02
1	COVER PLATE	2	USRA-TETHER-01

DETACHED LISTS

MM (INCH)
25.4MM=1 INCH
TOLERANCES EXCEPT AS NOTED
MM: 0 ± / .0 ± / .00 ±
INCHES: .0 ± / .00 ± / .000±
ANGLES ±
MATERIAL
FINISH

DFTG	D. F. CRONCH	DATE 11-19-91
CHKD		DATE
		DATE
MFG		DATE
APPVL		DATE

INTERPRET PER ANSI Y14.5M-1982
THIRD ANGLE PROJECTION

MECANICAL ENGINEERING DESIGN PROJECTS
The University of Texas at Austin

TITLE
ASSEMBLY - KINETIC ENERGY ABSORBER

FSCM NO. | SIZE C | DRAWING NO. USRA-TETHER-10 | REV 1
DO NOT SCALE DRAWING | DET LISTS ☐ YES ☐ NO | SHT 1 OF 1

DRAWING NO. USRA-TETHER-10

APPENDIX P

Machining Costs and Rates

APPENDIX P

MACHINING COSTS AND RATES

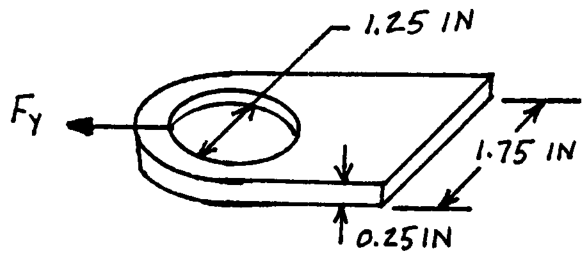
DRILLING RATE	=	$1 IN^3/MIN$
LATHE RATE	=	$3.5 IN^3/MIN$
TOOL CHANGE	=	1 CHANGE /MIN
ALUMINUM	=	\$0.57 / lb
DELRIN	=	\$ 40 / ft (3 INCH DIAMETER)

APPENDIX Q

Strength of Components

APPENDIX Q

STRENGTH OF TETHER POINT



THE FORCE REQUIRED TO YIELD THE TETHER POINT:

$$F_y = \frac{S_y A}{K_t} \quad [1]$$

WHERE: S_y = YIELD STRENGTH

A = CROSS SECTIONAL AREA MINUS HOLE DIAMETER

K_t = STRESS CONCENTRATION FACTOR

FOR 6061 T6 ALUMINUM, $S_y = 40 \times 10^3$ lb/IN^2 [2]

$A = (1.75\,IN - 1.25\,IN)(0.25\,IN) = 0.125\,IN^2$

$K_t \approx 2.3$ (SEE FIGURE Q-1)

$$\Rightarrow F_y = \frac{(40 \times 10^3\,lb/IN^2)(0.125\,IN^2)}{(2.3)} = 2174\,lb$$

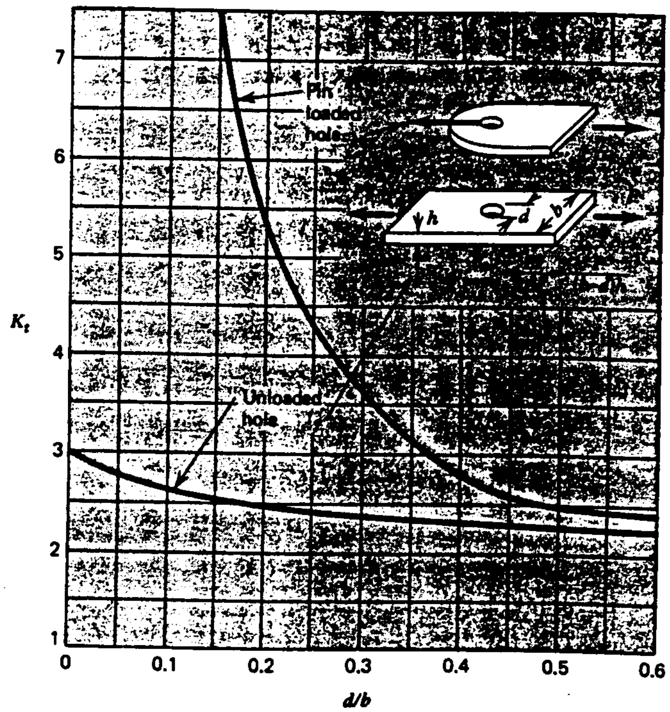
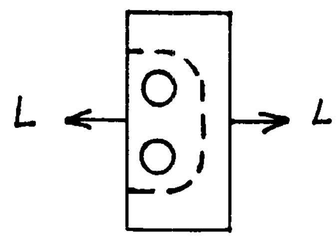


FIGURE Q-1. STRESS CONCENTATION FACTORS.

STRENGTH OF RIVETS



2 RIVETS $\varnothing$ 0.25 IN X $\frac{3}{8}$ IN LONG

RIVET MATERIAL = 6061 T3 ALUMINUM

SAFE TENSILE LOAD BASED ON DOUBLE SHEAR OF TWO RIVETS:

$$L = 2 \times \frac{2\pi d^2}{4} S_y \qquad [3]$$

WHERE : L = LOAD

d = RIVET DIAMETER

S_y = YIELD STRENGTH

$$L = 2 \times \frac{2\pi\,(0.25\ \mathrm{IN})^2}{4} \left(20 \times 10^3\ \frac{\mathrm{lb}}{\mathrm{IN}^2}\right)$$

$\Rightarrow L = 3927$ lb

STRENGTH OF SPOOL AXLE IN SHEAR

4 AXLES IN SHEAR

$L = 4AS_y$

WHERE : L = LOAD

A = CROSS SECTIONAL AREA OF AXLE

S_y = YIELD STRENGTH

$$L = 4\left[\frac{\pi(0.25\ \text{IN})^2}{4}\right](50\times10^3\ \text{lb/IN}^2)$$

$$\Rightarrow \quad L = 9817\ \text{lb}$$

STRENGTH OF SCREWS

6 SCREWS IN SHEAR   8-32 X 0.5 IN SCREWS

AREA = 0.01196 IN^2 [4]

$L = 6\,A\,S_y$

WHERE: L = LOAD

A = CROSS SECTIONAL AREA OF A SCREW

S_y = YIELD STRENGTH

$L = 6(0.01196\ IN^2)(50 \times 10^3\ lb/IN^2)$

$\Rightarrow\ L = 3588\ lb$

APPENDIX Q: References

1. R. C. Juvinall and K. M. Marshek, Fundamentals of Machine Component Design, (New York: John Wiley and Sons, 1991), p. 98.

2. E. A. Avallone and T. Baumeister III, eds., Marks' Standard Handbook for Mechanical Engineers, (New York: McGraw-Hill Book Co., 1978), p. 5-3.

3. Juvinall, p. 104.

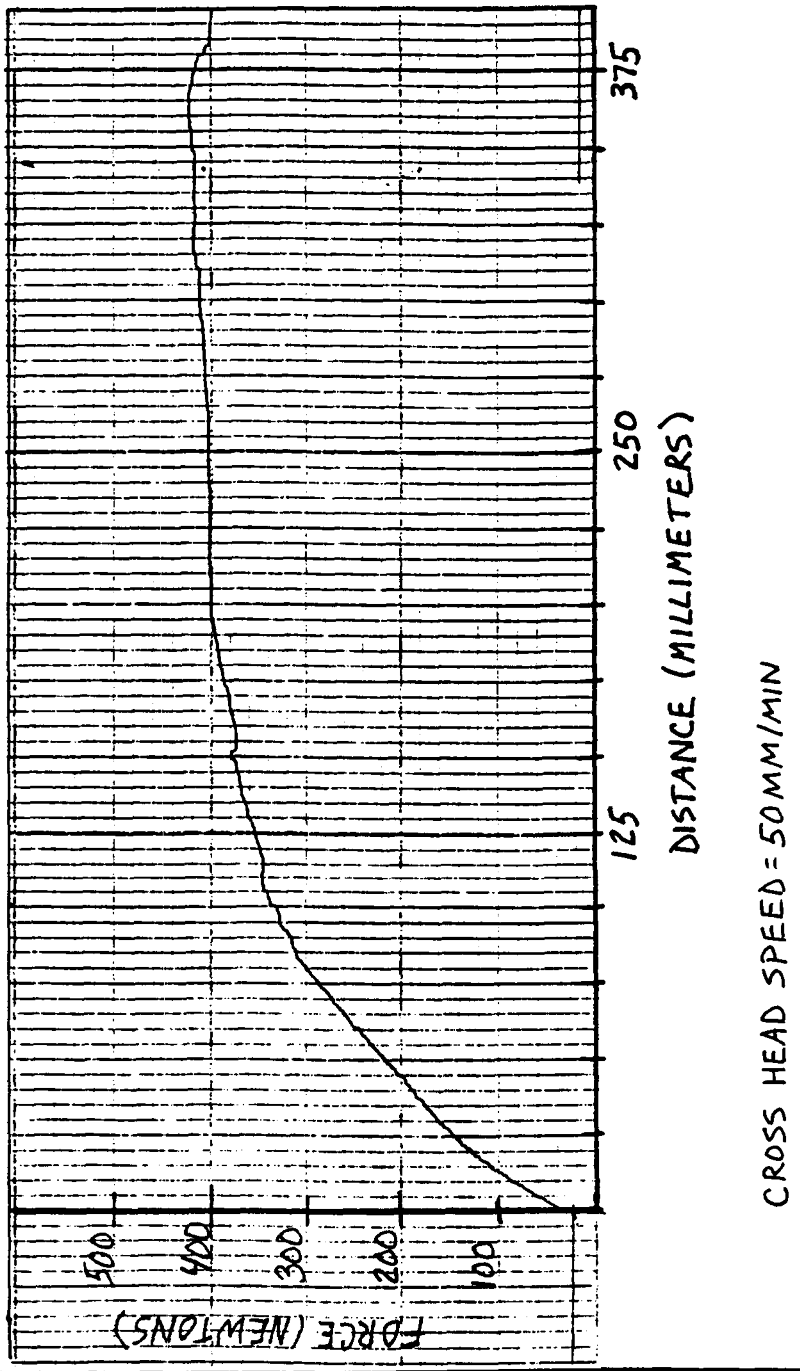
4. Avallone and Baumeister, p. 1261-1262.

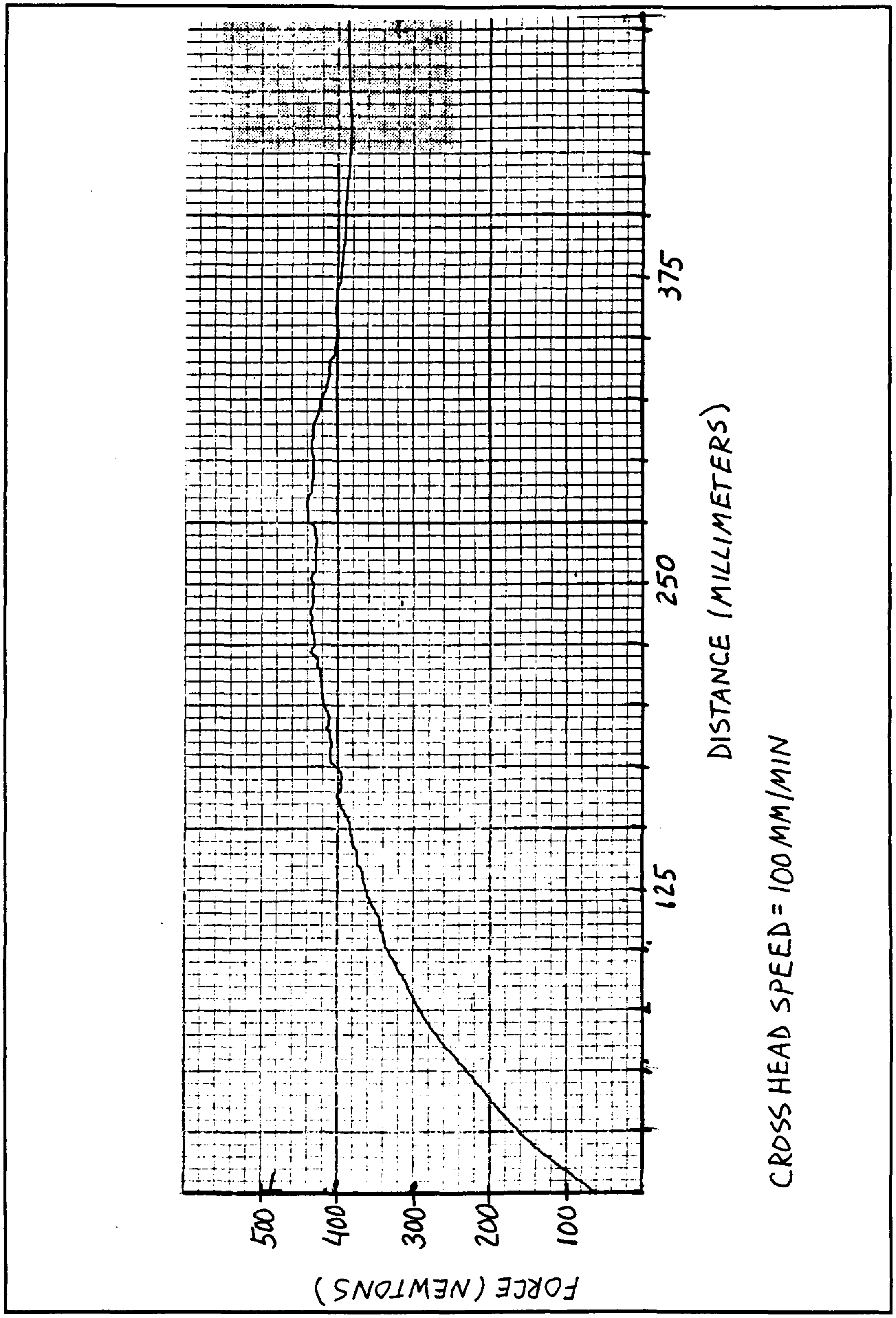
APPENDIX R

Force Versus Deflection Curve for Prototype

APPENDIX R

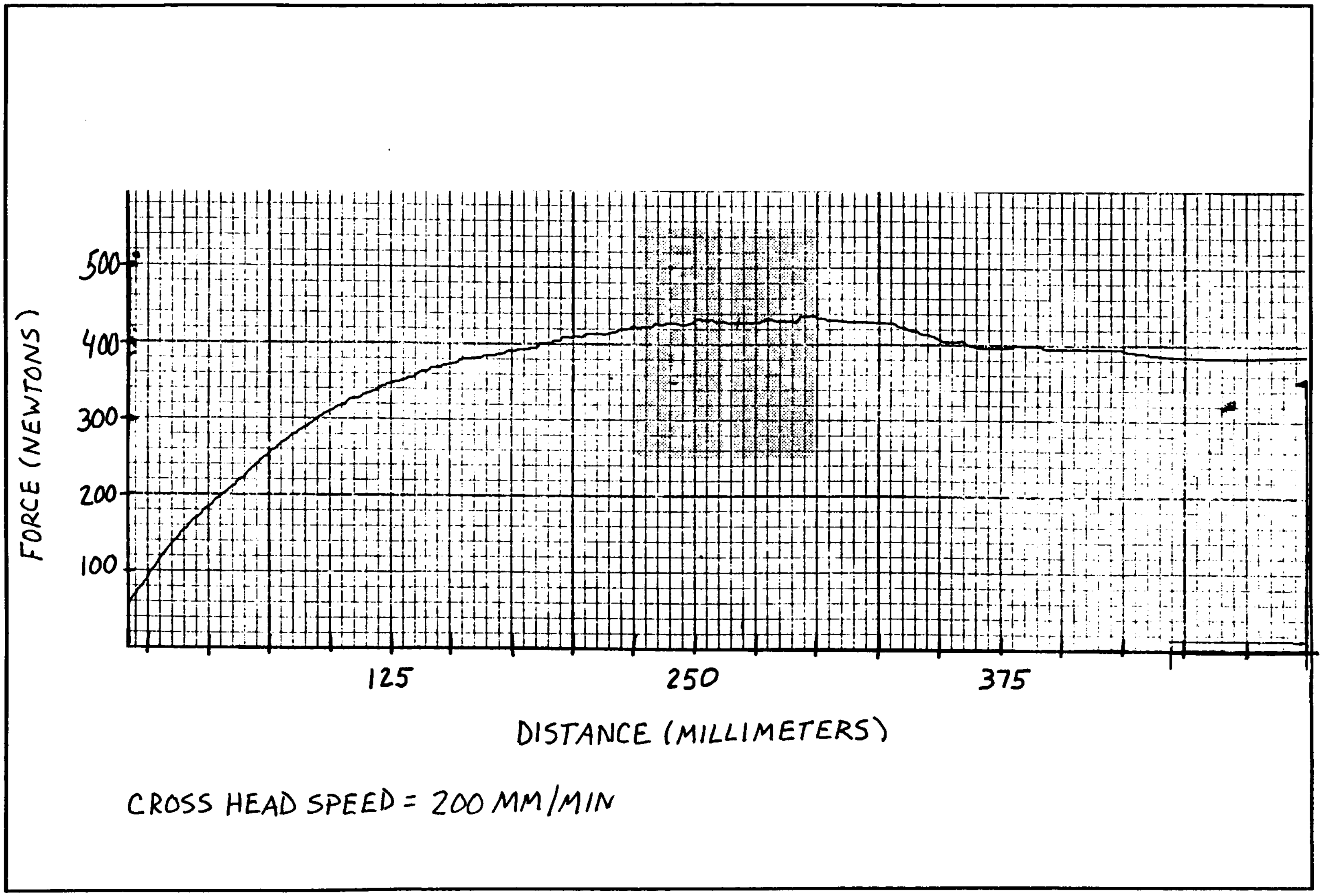
FORCE VS. DEFLECTION CURVE FOR PROTOTYPE





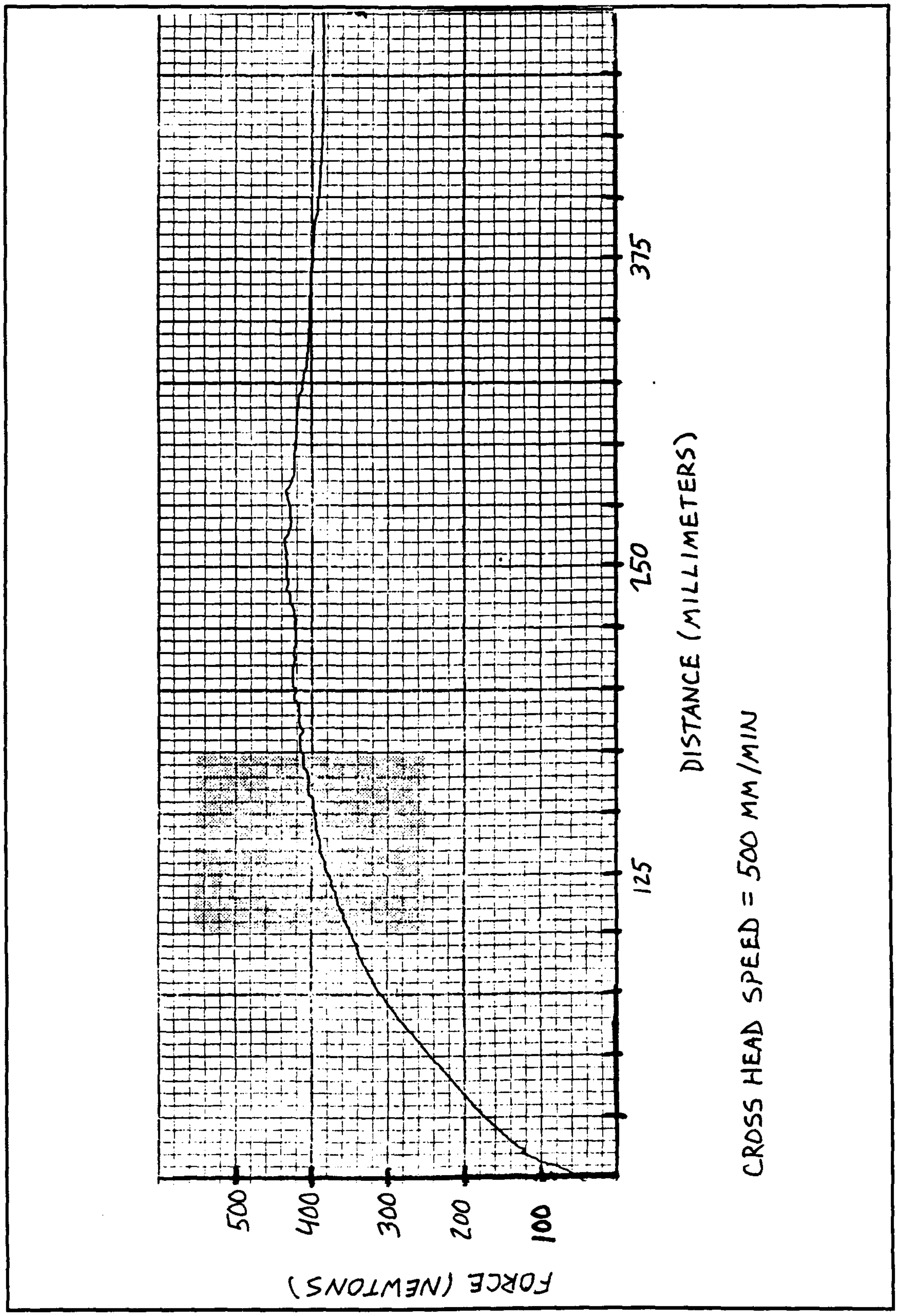
FORCE (NEWTONS)
500
400
300
200
100
125
250
375
DISTANCE (MILLIMETERS)
CROSS HEAD SPEED = 100 MM/MIN

R2



500
400
300
200
100
FORCE (NEWTONS)
125
250
375
DISTANCE (MILLIMETERS)
CROSS HEAD SPEED = 200 MM/MIN

R3



FORCE (NEWTONS)
500
400
300
200
100
125
250
375
DISTANCE (MILLIMETERS)
CROSS HEAD SPEED = 500 MM/MIN

R4

FROM GRAPH ON PAGE R1:

KINETIC ENERGY, KE = AREA UNDER THE CURVE

$$KE \cong [5(25 \text{ mm})(400 N)]\, 2.5 = 125{,}000 \text{ Nmm}$$

$$KE = 125 \text{ N-m} = 92.2 \text{ ft-lb}$$

ASSUMING A CONSTANT FORCE AND EXTENDING THE CURVE TO 2 FEET (609.6 mm):

$$KE \approx 125{,}000 \text{ Nmm} + [5(25 \text{mm})(3.872 \text{ mm})(400 N)]$$

$$\Rightarrow \quad KE = 318.6 \text{ N-m} = 235 \text{ ft-lb}$$

APPENDIX S

Results of Kinetic Energy Test

APPENDIX 5

RESULTS OF KINETIC ENERGY TEST

F_{bike} = 33 lb (146.8N)

F_{Dan} = 200 lb (

T_{TOT} = 233 lb (1036.4 N)

$$MASS = \frac{F}{g} = \frac{1036.4\,N}{9.81\,m/s} = 105.7\ kg$$

$$KE = \frac{1}{2} MV^2$$

$$d = \frac{KE}{F}$$

WHERE: d = DISTANCE THE SPRING EXTENDED

V = VELOCITY OF BIKE

KE = KINETIC ENERGY

THE EXPECTED EXTENSION OF THE SPRING BASED ON THE MASS AND VELOCITY OF THE BIKE AND RIDER WAS COMPARED TO THE ACTUAL EXTENSION OF THE SPRING. AN AVERAGE ERROR OF 14% WAS CALCULATED (SEE TABLE 51)

Table S1

Comparison of the Actual Extension of the Spring to the Expected Extension of the Spring

Velocity (ft/s)	Extension (in) actual	Extension (in) expected	Error (percent)
4.255	13.0	9.3	40.04
4.938	13.5	12.5	8.30
5.333	14.5	14.5	0.30
4.706	15.0	11.3	32.50
4.348	10.0	9.7	3.50
3.810	9.0	7.4	21.30
4.762	10.5	11.6	9.40
4.000	9.5	8.2	16.10
4.040	9.0	8.3	7.80
3.922	9.5	7.9	20.80
4.124	9.25	8.7	6.40

APPENDIX T

Comparison of the Energy Storage Capacity Between a Stainless Steel Spring and a Composite Spring

APPENDIX T

COMPARISON OF THE ENERGY STORAGE CAPACITY BETWEEN A STAINLESS STEEL SPRING AND A COMPOSITE SPRING

ENERGY STORAGE / UNIT WEIGHT $\equiv \mu'$

$$\mu' = \frac{C_F \sigma_Y^2}{2E\rho} \quad \text{(FOR E TYPE SPRINGS) [1]}$$

WHERE : C_F = FORM COEFFICIENT

σ_Y = YIELD STRENGTH

E = MODULUS OF ELASTICITY

ρ = DENSITY

FOR STAINLESS STEEL, AISI 301 :

$\sigma_Y = 140 \times 10^3$ psi (COLD WORKED) [2]

$E = 28 \times 10^6$ psi

$\rho = 0.2833$ lb IN^3

ASSUME $C_F = 0.33$ FOR A CONSTANT FORCE SPRING SINCE IT IS A RECTANGULAR E SPRING [3]

$$\mu' = \frac{(0.33)(140 \times 10^3 \text{psi})^2}{2(28 \times 10^6 \text{psi})(0.2833 \text{ lb/IN}^3)} = 407.7 \frac{\text{IN-lb}}{\text{lbm}}$$

FOR HIGH STRENGTH Gr/EPOXY UNIDIRECTIONAL AS-4 FIBER (SQUARE WIRE), $\mu' = 2551.5 \frac{\text{IN-lb}}{\text{lbm}}$ [4]

$\Rightarrow$ # OF TIMES LIGHTER $= \frac{2551.5}{407.7} = 6.26$

APPENDIX T: References

1. John A. Ennes, "Composite Material Spring for 767 Entry and Service Door Counterbalance Mechanism," Advanced Composites: The Latest Developments, (Dearborn, MI: ASM International, 1986), p. 7.

2. "Materials Selection Charts, Irons and Steels," Materials Engineering, (December 1989), p. 48.

3. Ennes, p. 5.

4. Ennes, p. 8.