ALTERNATIVE DESIGN FOR THE COMPOUND ARCHERY BOW

by

Deanna Lee Griffith

SUBMITTED TO THE DEPARTMENT OF MECHANICAL ENGINEERING IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF

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Submitted to the Department of Mechanical Engineering on December 13, 1991 in partial fulfillment of the requirements for the degree of Bachelor of Science.

Abstract

It was determined that in order to improve the aesthetic qualities of the compound bow, an alternative to the pulley system must be found. A linkage and spring mechanism that fits in the handle section of the bow was designed. Appropriate parts were chosen or designed to meet energy storage and stress requirements. Detailed drawings were made for the machining of a prototype. Initial analysis showed that the spring mechanism is a feasible replacement for the pulley system. A prototype must be built and analyzed to determine the practicality of the mechanism and the effect of unmodeled factors on the force-draw characteristic.

Thesis Supervisor: Carl R. Peterson
Title: Professor of Mechanical Engineering

Dedication

To my parents, Tom and Diane Griffith,
Thank You.

Table of Contents

List of Figures

List of Tables

1 Design Problem Background

Archery is an ancient sport characterized by the beauty and simplicity of the archery bow. Because of the tremendous skill and physical strength required to use a bow, archery is a popular modern-day sport. The simplicity of the archery bow continues to challenge archery enthusiasts and engineers to push the limits of the design. One major development in archery equipment is the compound bow (figure 1.1). The compound bow is fitted with eccentric pulleys or cams on the end of each limb, with the string looped around to produce a distinctive performance curve (figure 1.2). The curve is distinguished by its parabolic rise to peak draw force at the middle of the draw, followed by a parabolic drop to a holding force of about half the peak.

The distinctive force-draw characteristic of the compound bow has two advantages over the more traditional recurve bow (figure 1.3). One advantage is ease of use. The performance curve for the recurve bow is shown in figure 1.4. A recurve bow of the same maximum draw force has a holding weight twice that of the compound bow. The lower holding weight of the compound bow causes less fatigue to the archer, making it easier to aim. The second advantage is the energy storage of the compound bow. Integration under the performance curves shows that the compound bow stores approximately 1/3 more energy than the recurve of the same draw weight.

Although the compound bow has great advantages in performance over the recurve bow, the pulley system greatly detracts from the simplicity and aesthetic quality that are the essence of the sport. The recurve bow has a simple, graceful line from the tip of one

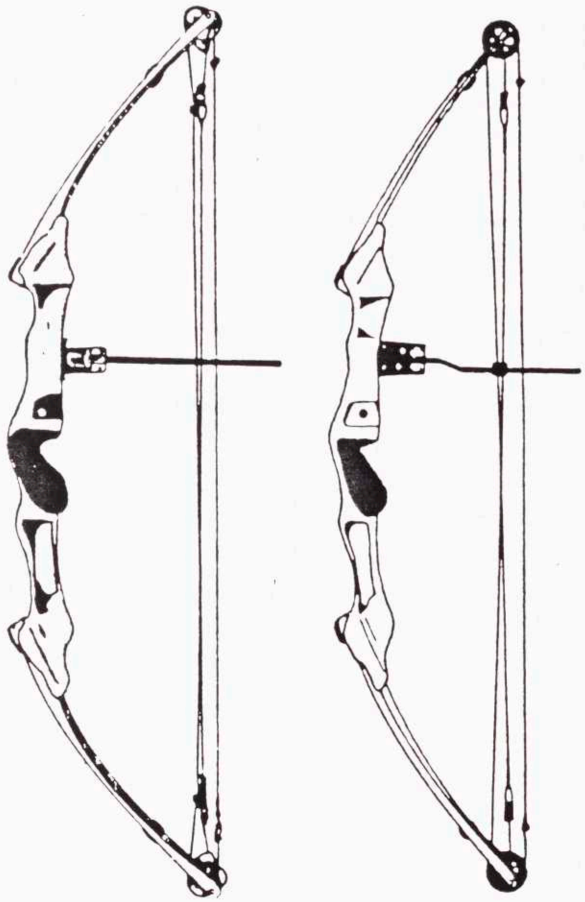


Figure 1.1 Compound Archery Bow [Annis 86]

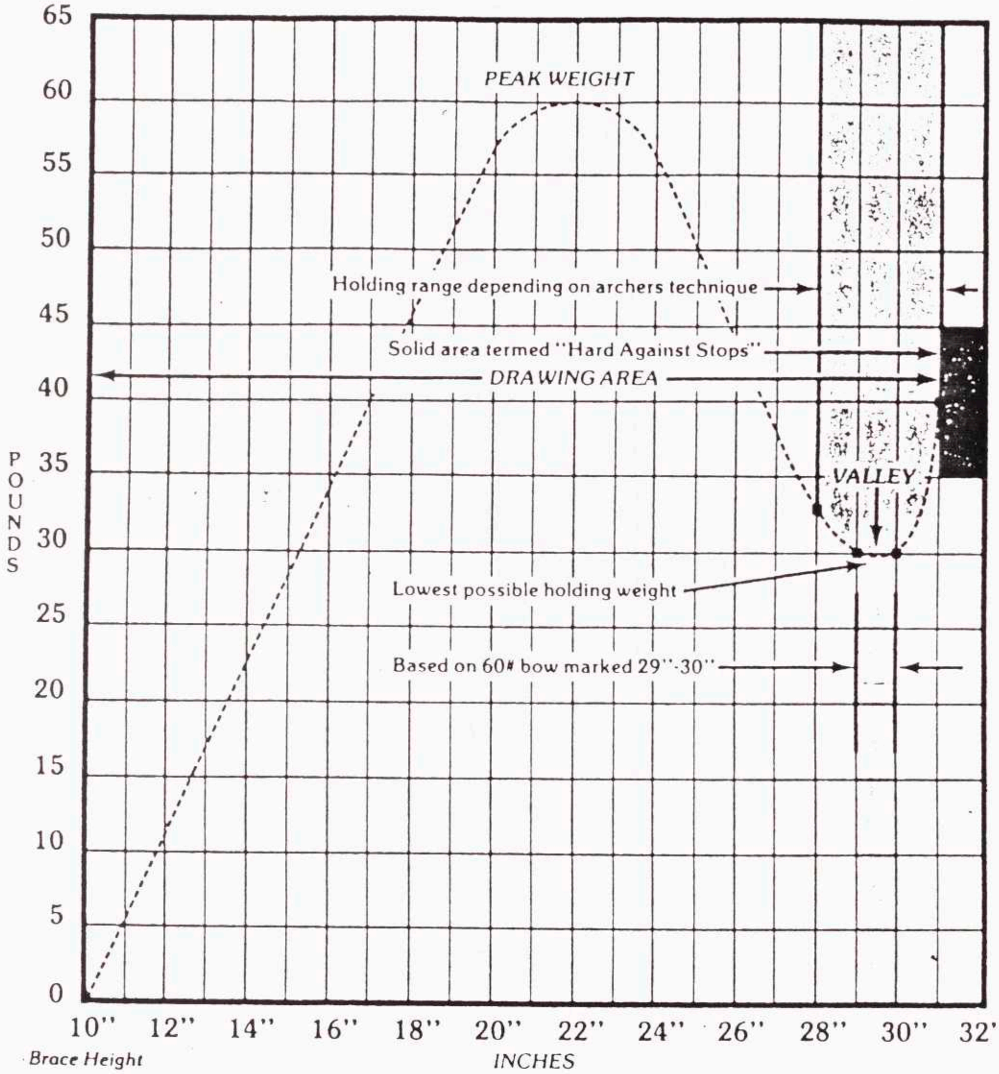


Figure 1.2 Performance Curve of Compound Archery Bow [Annis 86]

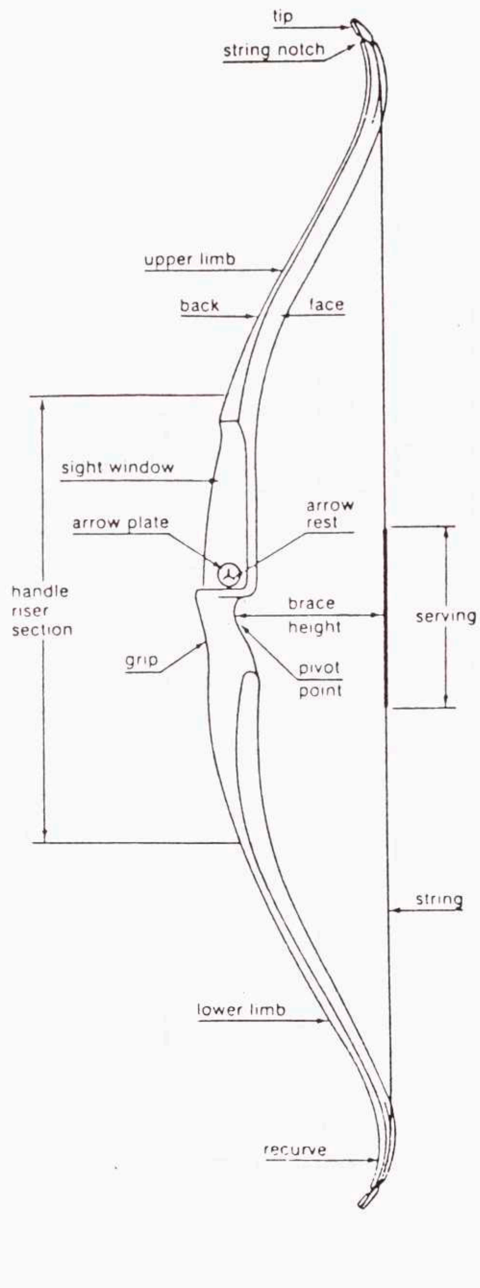


Figure 1.3 Recurve Archery Bow [Annis 86]

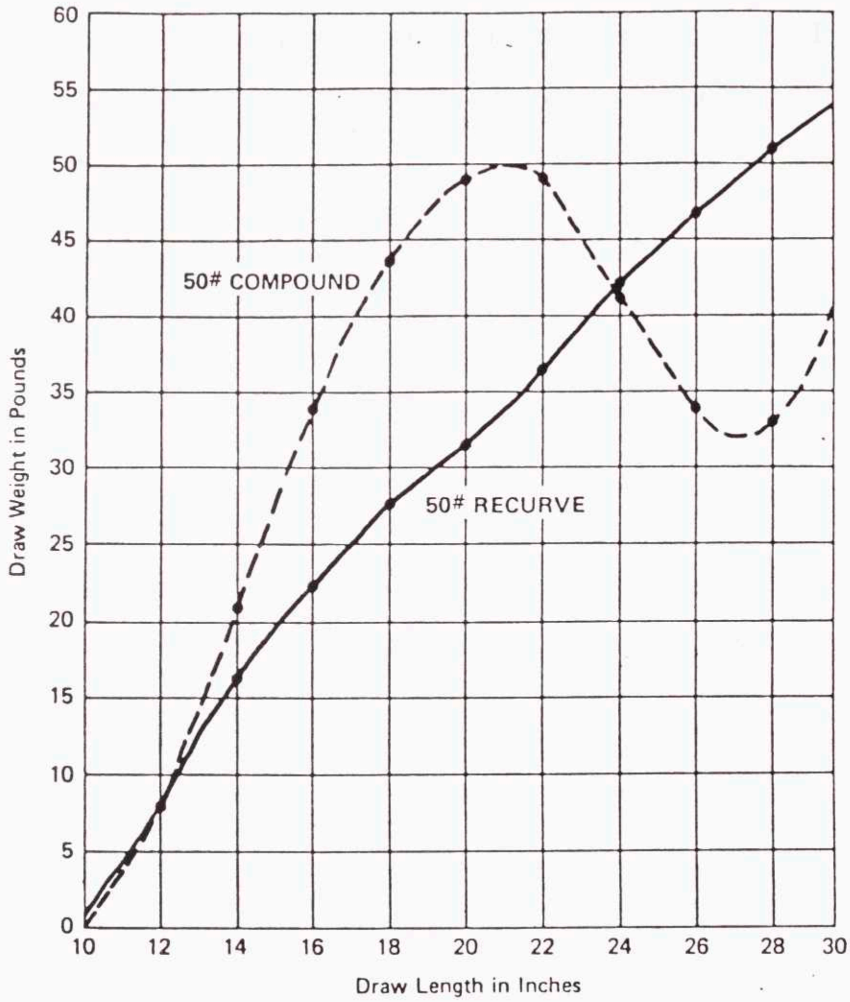


Figure 1.4 Performance Curve of Recurve Archery Bow [Annis 86]

limb to the other, whereas on the compound bow, the line of the limbs ends in pulleys and a maze of strings. Unfortunately, the potential for improving the aesthetics of the compound bow using the pulley system is limited. The pulley location and size is not very flexible, and the string must be looped for the mechanism to work. If the aesthetic quality of the compound bow is to be improved, an alternative to the pulley mechanism needs to be found.

2 Preliminary Design Solution

The proposed design uses an over-center spring in place of the pulley mechanism to achieve the same parabolic force-draw characteristic. The system uses a separate energy storage element (spring) and linkage which can be concealed in the handle section of the bow. The limbs are used to apply leverage to the spring, and, ideally, do not bend at all.

A free-body diagram of the mechanism is shown in figure 2.1. As the bow is drawn, the limbs pivot, shortening the spring section of the linkage, and reducing the angle between the spring and the limb. As the mechanism flattens, the moment of the spring force on the limb approaches zero. In addition, as the limbs rotate, the string angle to the limb increases, and the moment of the string tension about the pivot increases until it's normal to the limb, then decreases. To meet equilibrium requirements, these two moments must be equal and opposite. With the appropriate choice of linkage parameters, the force should increase with draw, to a peak, and then fall toward zero. From the characteristics of the over-center spring, it follows that the force characteristics of the bow should be roughly parabolic.

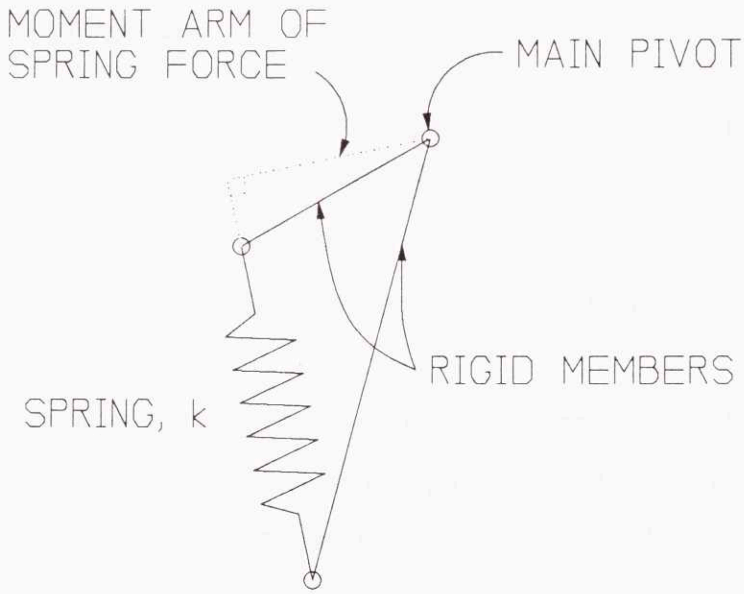


Figure 2.1 Free-body Diagram Of Over-center Spring Mechanism

3 Design Parameters

3.1 Overall bow dimensions

Before designing the bow, some basic dimensions and constraints must be set. These dimensions include the size of the bow, and the maximum holding weight. Some constraints exist in the basic construction and operation of an archery bow, such as the location of the handle relative to the pivot, and the location of the window for the arrow rest.

The average height of a hunting bow was found to be approximately 56 inches. Of this 56 inches, the handle and riser section is approximately 15 inches. The arrow rest is not included in the prototype, however, room is left to install it. The bow is designed to have a 45 pound maximum draw weight. The primary use of compound bows is for hunting. The minimum draw weight for a hunting bow is 42 pounds, so the 45 pound draw weight is consistent with existing compound bow designs. The draw length ranges from approximately 10 inches at rest, to a holding length of 28 to 31 inches. The holding length used for calculations is 30 inches. These dimensions are summarized in figure 3.1.

In the operation of a bow, it is important that the draw force be in line with the pivot point to avoid creating a moment on the system. The dimensions for this area, where the hand wraps around the handle, need to be fairly small. A reasonable size is less than 2.5 inches effective diameter. This is purely an estimate based on items that are comfortable to hold. This dimension is important, but not critical, since when the bow

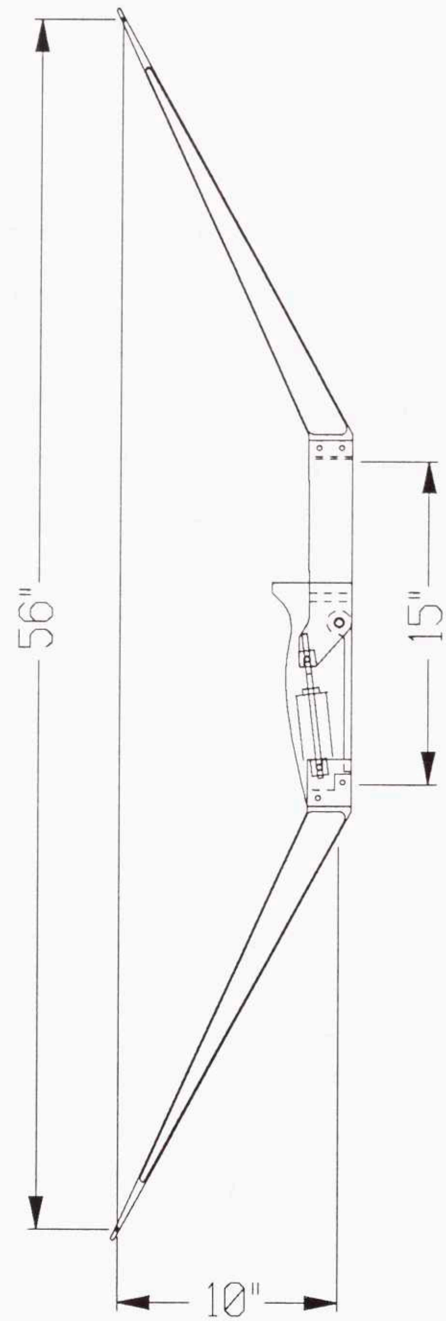


Figure 3.1 Bow Dimensions

is drawn, the handle is not so much gripped, as it is cradled between the thumb and forefinger.

3.2 Spring choice

The most important element in the mechanism design is the spring. It is vital to the design to find an efficient, economical, and light weight storage element. The spring chosen is a urethane tube die spring. It can be assumed from its application that it is engineered for low hysteresis, and therefore, high efficiency. It must also take very high forces, and recover very quickly. This makes it an ideal choice. Also, the energy, 560 inch- pounds, can be stored in a rather small spring, which is very important since the spring must fit in the handle section.

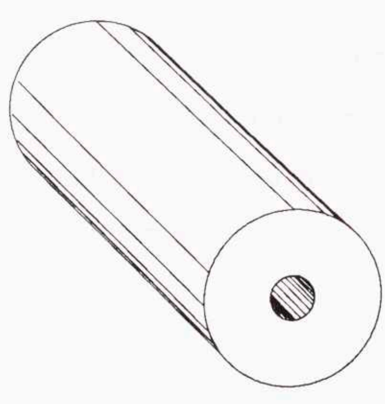


Figure 3.2 Urethane Polymer Tube Spring

4 Bow Kinematics

To establish the force-draw characteristics, the bow is modeled by the moment arms from the pivot to the points of action of the string and spring forces (figure 4.1). The system is modeled as "ideal," i.e. all elements are massless, and rigid, and all rotating joints are frictionless. These simplifications are justified, in that these calculations are used to determine the force-draw characteristics. The mass of the elements effects mostly the dynamics of the system, i.e. come into play when the string is released, not drawn. The friction has the effect of making the bow slightly more difficult to draw, but again, mostly comes into play when the bow is released. The most significant simplification is the assumption that the limbs are rigid. In the final design, it will be desirable to make the limbs as light as possible. As this is done, some rigidity will be sacrificed. The bending of the limbs, however slight, stores some energy and serves to shift the force-draw characteristic. Some attempt has been made to account for this, however, the amount was rather arbitrary. The full effect of the limb bending will need to be determined experimentally from the prototype.

The angles of the bow are given by the law of cosines. From figure 4.1:

$$\gamma = \mathrm{acos}[(b^2 + c^2 - a^2)/2bc] \qquad \text{(eq 4.1)}$$

$$\beta = \mathrm{acos}[(a^2 + b^2 - c^2)/2ab] \qquad \text{(eq 4.2)}$$

where a and b are fixed bow dimensions, and c is the draw length, ranging from starting length to holding length.

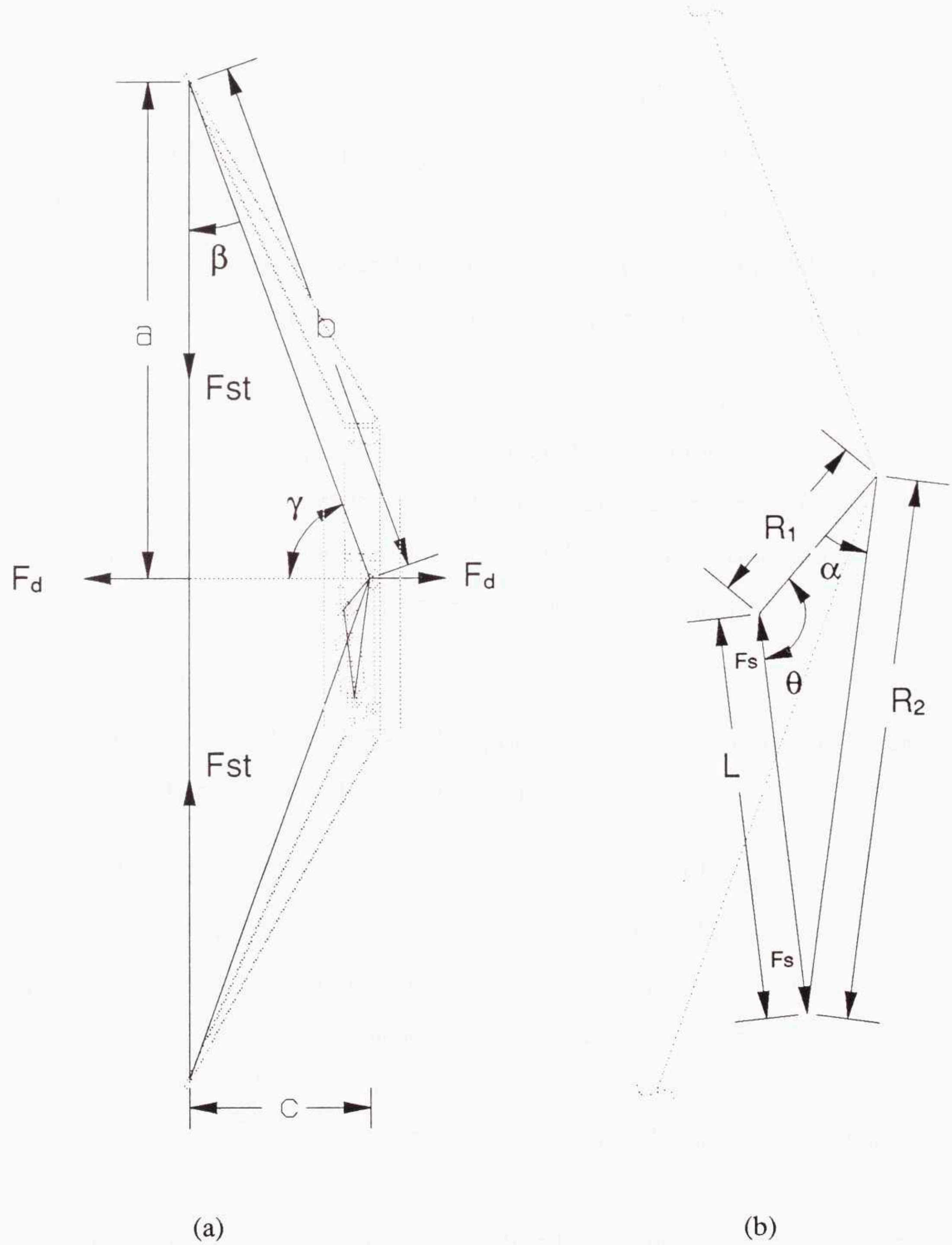


Figure 4.1 Free-Body Diagram of (a) Bow and (b) Mechanism

The dimensions of the mechanism links are related by the law of cosines. From figure 4.1b:

$$\alpha_o = \mathrm{acos}[(R_2^2 + R_1^2 - L_o^2)/2R_1R_2] \qquad \text{(eq 4.3)}$$

$$\theta = \mathrm{acos}[(L^2 + R_1^2 - R_2^2)/2LR_1] \qquad \text{(eq 4.4)}$$

$$L = (R_2^2 + R_1^2 - 2R_1R_2\cos\alpha)^{1/2} \qquad \text{(eq 4.5)}$$

where α_o is the initial angle between R_1 and R_2, and L_o is the initial length of the spring member. The change in γ equals half the change in α, therefore, α is given by

$$\alpha = \alpha_o - 2\gamma_o + 2\gamma \qquad \text{(eq 4.6)}$$

To find the forces in the system, the bow is modeled in equilibrium at successive intervals throughout the draw length. Equilibrium requires the moments of the spring and string forces about the pivot to be equal. Since the system is symmetrical, the equations are concerned with forces on one limb only. The spring force is given by

$$F_S = F_{So} + (L_o - L)k \qquad \text{(eq 4.7)}$$

where F_{So} is the spring preload, and k is the spring constant. The spring, string, and draw forces are related by the following equations:

$$F_{st} = R_1F_s\sin\theta/b\sin\beta \qquad \text{(eq 4.8)}$$

$$F_d = -2F_{st}\cos(\beta + \gamma) \qquad \text{(eq 4.9)}$$

Equations 4.1-4.9 completely describe the force-draw relationship. It can be seen in the above equations that the mechanism can be at any angle to the limbs without effecting the force-draw characteristic. Equilibrium is concerned only with the moment of the spring force about the pivot, which is independent of the mechanism's angle to the limbs.

The energy of the system is intended to be stored solely in the spring. The total energy in the spring is given by

$$E_s = 0.5k[F_{So}/k + (L_o - L)]^2 \qquad \text{(eq 4.10)}$$

The initial energy in the spring (due to preload) is given by

$$E_o = 0.5k(F_{So}/k)^2 = 0.5F_{So}^2/k \qquad \text{(eq 4.11)}$$

Combining equations 4.10 and 4.11, the potential energy available in the spring is

$$E = E_S - E_o \qquad \text{(eq 4.12)}$$

To determine what dimensions will yield the desired force-draw characteristics, the equations were solved for each inch of draw using a computer spreadsheet. From section 3, the bow height, 2a, is 56 inches; the draw length, c, ranges from 10 inches to 31 inches; and limb length, b, is given by the pythagorean theorem,

$$b = (a^2 + c_o^2)^{1/2} \qquad \text{(eq 4.13)}$$

where c_o is the initial draw length. This yields a "limb" length of ≈ 26 inches.

The mechanism dimensions are determined by trial and error using the spreadsheet, to obtain the desired force-draw characteristics. Several limiting factors were considered. First, the spring member must be long enough to accommodate an adequate length spring, as well as the supporting hardware. The spring type chosen in section 3 has a large range of spring constants in springs 3" and under. Allowing a minimum of 1" for hardware, the minimum spring member length is 4." Also, the spring type has a maximum deflection of 30%, so the maximum change in length of the spring member is ≈ 0.75" (some deflection occurs in the preload). The maximum mechanism dimensions are limited by the handle size. The handle-riser section in a bow is on the order of 15" long. It is desirable to make the handle approximately half that length. Also, since the hand must be in line with the pivot, the mechanism should be as thin as possible (small R_1), to make it easy to grip.

Using these criteria, a set of dimensions yielding the desired force-draw relationship was found. The performance curve is shown in figure 4.2. These dimensions include a spring preload of 250 lbs. This gives an initial string tension of ≈ 40 lbs, low enough that the bow is not difficult to string, but high enough to give some initial rigidity to the

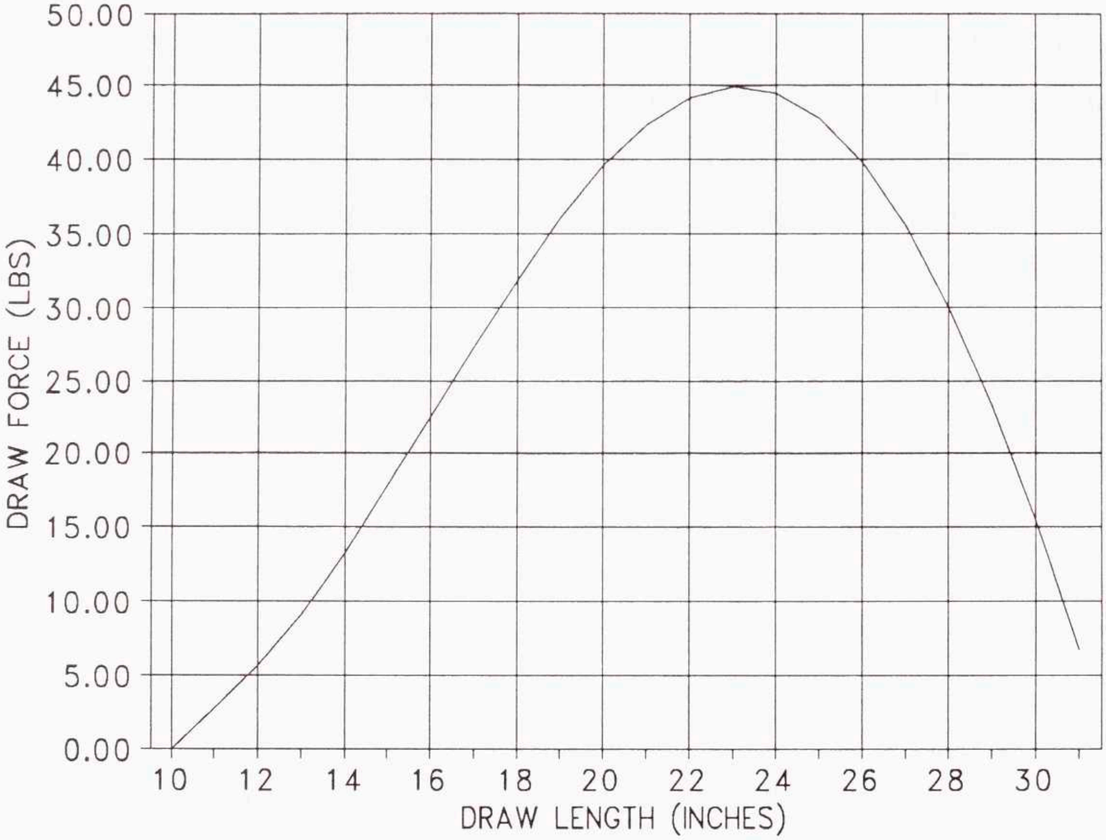


Figure 4.2 Performance Characteristic for Bow Using Over-Center Spring Mechanism

system. The maximum draw force is ≈ 45 lbs, at a draw length of 23," falling to ≈ 15 lbs at 30." The calculated holding weight is intentionally below the target weight of 23 lbs. This is because the limbs, modeled as rigid, are expected to bend slightly. As this occurs, the draw length will increase, without a resulting deflection of the spring. For this reason, draw length is expected to reach 30" before the draw force falls to 15 lbs. The spring chosen has a spring constant of 3600 in/lb. The corresponding urethane tube spring is 2" long, 1.125" in diameter, for a 5/16" shaft. It deflects ≈ 0.57." having a maximum diameter of < 1.355" at full draw length. The energy stored in the spring is 560 in-lbs. The maximum spring load is 2040 lbs. The complete results of the spreadsheet calculations are in appendix A.

5 Prototype Design

5.1 Goals and limitations

The purpose of the prototype is to help determine if the over-center spring mechanism is a viable alternative to the pulley system on the compound bow. The specific aspect of the over-center spring mechanism being tested is the force-draw characteristic. Other factors, such as aesthetics, ergonomics, and dynamics, are of concern, but are freely compromised in the prototype in the interest of cost and ease of manufacture.

The main constraints in the design are cost and manufacturability. The prototype must be made from inexpensive, commercially available materials, that are easily machined or assembled in a conventional machine shop. Because of the high stresses taken by the materials as outlined in the rest of this section, and ease of machining, the materials used are metals. Because weight is a major concern, aluminum is used wherever possible, and steel, only where stresses demand the use of the stronger material. The greatest strength to weight ratio comes in hardened alloys. The aluminum used is 6061-T6, a widely available heat-treated aluminum alloy with a yield strength of 40,000 psi. There are many high strength steels available. It is recommended because of the high stresses taken by the steel parts, that a steel with a yield strength in excess of 120,000 psi be used.

5.2 Mechanism

5.2.1 Spring parameters

The spring corresponding to the desired spring constant, k, of 3600 lbs/in, is 2" free length, 1.125" outer diameter, and 5/16" inner diameter. It has a recommended maximum deflection of 25-30% of its free length. The deflection required for this application is ≈ 0.57," which is within the acceptable range.

Because the spring is a solid elastomer tube, the diameter increases as the length decreases. It is necessary to calculate the maximum diameter of the deflected spring in order to determine the size of the handle section. The volume of the urethane tube spring is given by:

$$V = L\pi(R_2^2-R_1^2) \qquad \text{(eq 5.1)}$$

where R_2 is the outer radius, R_1 the inner radius, and L, the free length. It is assumed, because of the shaft through the spring center, that the inner radius does not change. The new radius becomes:

$$R = [(L/L_2)(R_2^2-R_1^2) + R_1^2]^{1/2} \qquad \text{(eq 5.2)}$$

$$R = 0.658"$$

It is important to note that this is only an estimate. The calculation assumes the spring maintains a uniform diameter as it deflects. Because of sticking friction at both ends of the spring, the majority of the deflection in diameter will occur at the center of the spring. Therefore, the maximum deflected diameter will actually exceed the value predicted by equation 5.2.

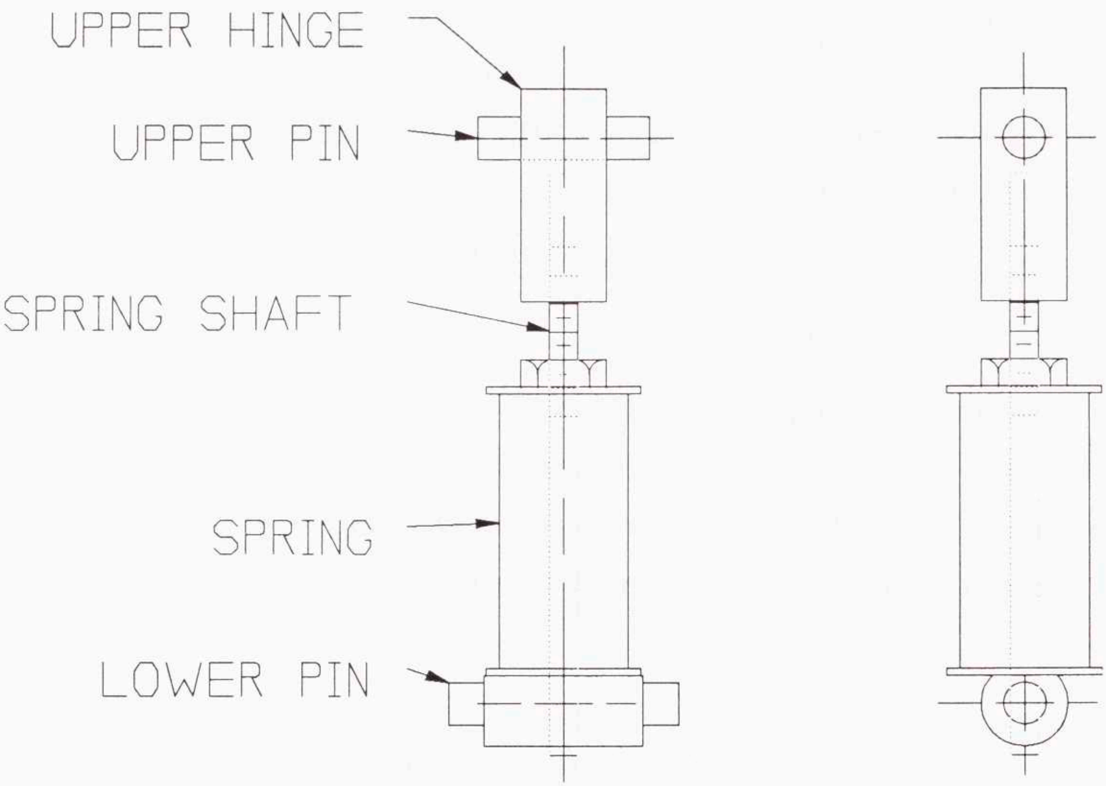


Figure 5.1 Mechanism Design for Prototype

5.2.2 Spring member

The main section of the spring member is the slender rod of the spring shaft. Since the system is loaded in compression, the main concern is the possibility of buckling in the thin section of the spring shaft.

The spring itself requires a 0.25" shaft. In the interest of cost and ease of machining, it is desirable to make the threaded portion of the shaft 0.25" nominal diameter as well. The thread is 1/4-28 UNF, with an effective area of 0.0364 in^2. The shaft pivots at a length of 5," with the thin section a maximum length of 1". The critical load, P_{cr}, for buckling of a round column is given by:

$$P_{cr} = n\pi^2 EA/(l/r)^2 \qquad \text{(eq 5.3)}$$

where n=1 for pinned supports at each end, and $E \approx 28 \times 10^6$ psi for tempered steel. Using the full 5" length for a conservative estimate, the critical load given by eq. 5.3 is 6300 lbs. The load on the column is the spring force, 2000 lbs. This is an acceptable design, given that the 6300 lbs itself is an extremely conservative estimate of the buckling load.

5.2.3 Mechanism pins and support structure

The spring member applies a force of approximately 2000 lbs on each of the pins in the upper and lower limbs. To determine the size of the pins, bearings, and supporting hardware, the stress from shear and bending in each pin, and the stress in the aluminum support structure must be found.

The upper limb pin can be modeled as a beam with a distributed load of 2000 lbs over a distance of 0.75" with pinned supports on each side. The modeled system has a maximum bending moment (M_b) in the center of the pin, and a maximum shear at each support.

$M_{bmax} = Wl^2/8$ (eq 5.4)

$V_{max} = Wl/2$ (eq 5.5)

The stresses due to bending (σ_b) and shear (σ_s) are:

$\sigma_b = -M_b y_{max}/I_{zz}$ (eq 5.6)

$\sigma_s = 4V/3\pi r^2$ (eq 5.7)

where I_{zz} is the moment of inertia of the cross section:

$I_{zz} = \pi r^4/2$ (eq 5.8)

For weight and frictional concerns it is desirable to make the pin as small a diameter as possible. The length dimensions are 1.5" total with 3/8" of support on each end. For a 3/8" diameter pin, the maximum stresses are:

$\sigma_b = 54{,}300$ psi

$\sigma_s = 9100$ psi

Because of the very high stresses, steel with a minimum yield stress of 110,000 psi is recommended. The safety factor of two is to account for uncertainty due to simplifications in modeling. One of the major sources of concern is the assumption of evenly distributed loading. Unevenly distributed loading could occur because of the geometry of the upper limb section, the possibility of moments applied by the operator, or slight eccentricities in the mechanism.

Using the projected area of the pin on the support structure, the compressive stress on the bearing is ≈ 7100 psi. The bearing chosen, a 0.375" nominal bore Garlock DU bearing, will take stresses up to 20,000 psi for the type of loading in this application. The stress on the aluminum support structure is essentially the same as that on the bearing (slightly less, due to the larger area). The tensile yield strength of the material, 6061-T6, is 40,000 psi. The stress on the aluminum is not a major design concern.

The lower limb pin is a large shaft with small diameter extensions at each end, a clearance hole through the center for the spring shaft, and a milled flat to support the spring. The hole is 0.25." The center portion of the pin is 0.75" diameter with a milled flat perpendicular to the hole. The pin has a total length of 2." This geometry has a significant stress concentration around the hole in the center of the pin. This is of major concern, as it occurs at the location of maximum stress due to bending. The theoretical stress at the center of the pin (σ_o) is:

$$\sigma_o = M_b/[(\pi D^3/32) - (dD^2/6)] \qquad \text{(eq 5.9)}$$

where D is the pin diameter, and d is the hole diameter. The bending moment at this point is given by eq. 5.4. For a d/D ratio of ~ 0.3, the theoretical stress concentration factor, K_t, is ~1.9. This gives a stress of 59,400 psi. This is on the same order as the stress in the upper pin, and the same material recommendations apply.

The small extensions from the lower pin are the same diameter as the upper pin, 0.375," but are supported by two 0.25" wide sections, rather than 0.375" thick sections. This gives a bearing and support load of approximately 10,700 psi. This value is more in line with target safety factor of 2 than the upper support design. The upper support is thicker because the extra material is necessary in the main pivot, and a constant thickness is easier to machine.

5.3 Main Pivot

5.3.1 Pin force

The main pivot provides the reaction forces to the string tension force, spring force, and draw force on each limb. To determine the force on the pin, the normal ($\hat{n}$) and tangential ($\hat{s}$) components of each of the above forces, F_{st}, F_s, and F_d, are calculated and summed. The magnitude of this vector is the pin force. The forces on the limb are (fig 5.2):

$$\vec{F}_{st} = -F_{st}(\sin\beta)\hat{n} + F_{st}(\cos\beta)\hat{s} \qquad \text{(eq 5.10)}$$

$$\vec{F}_s = F_s\sin(\theta-\psi)\hat{n} - F_s\cos(\theta-\psi)\hat{s} \qquad \text{(eq 5.11)}$$

$$\vec{F}_d = F_d\sin\gamma\hat{n} + F_d\cos\gamma\hat{s} \qquad \text{(eq 5.12)}$$

The total force on the limb is given by

$$\vec{F}_t = \underbrace{(-F_{st}\sin\beta + F_s\sin(\theta-\psi) + F_d\sin\gamma)}_{N}\hat{n} + \underbrace{(F_{st}\cos\beta - F_s\cos(\theta-\psi) + F_d\cos\gamma)}_{S}\hat{s} \qquad \text{(eq 5.13)}$$

The magnitude of the reaction force is

$$|F_{pin}| = |F_t| = (N^2 + S^2)^{1/2} \qquad \text{(eq 5.14)}$$

where N is the normal component and S the tangential component, and the direction relative to the moment arm

$$\text{angle } F_{pin} = \text{atan } N/S \qquad \text{(eq 5.15).}$$

The angle of the mechanism to the limb moment arm, ψ, is 114.35°, or 1.996 rad. The equations were added to the spreadsheet simulation, and the results are shown in figures 5.3. The maximum pin force is approximately 2000 lbs.

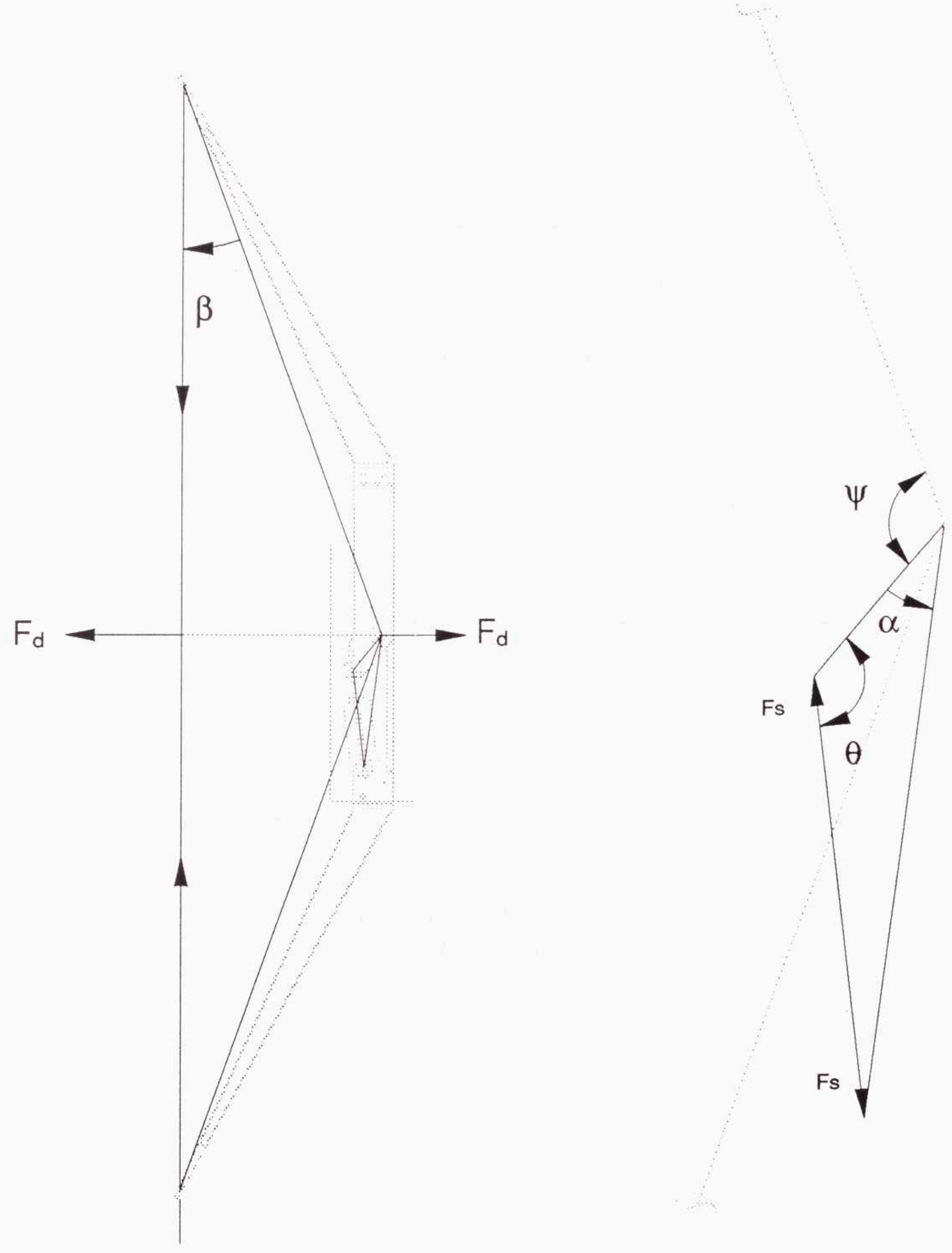


Figure 5.2 Forces on Main Pivot

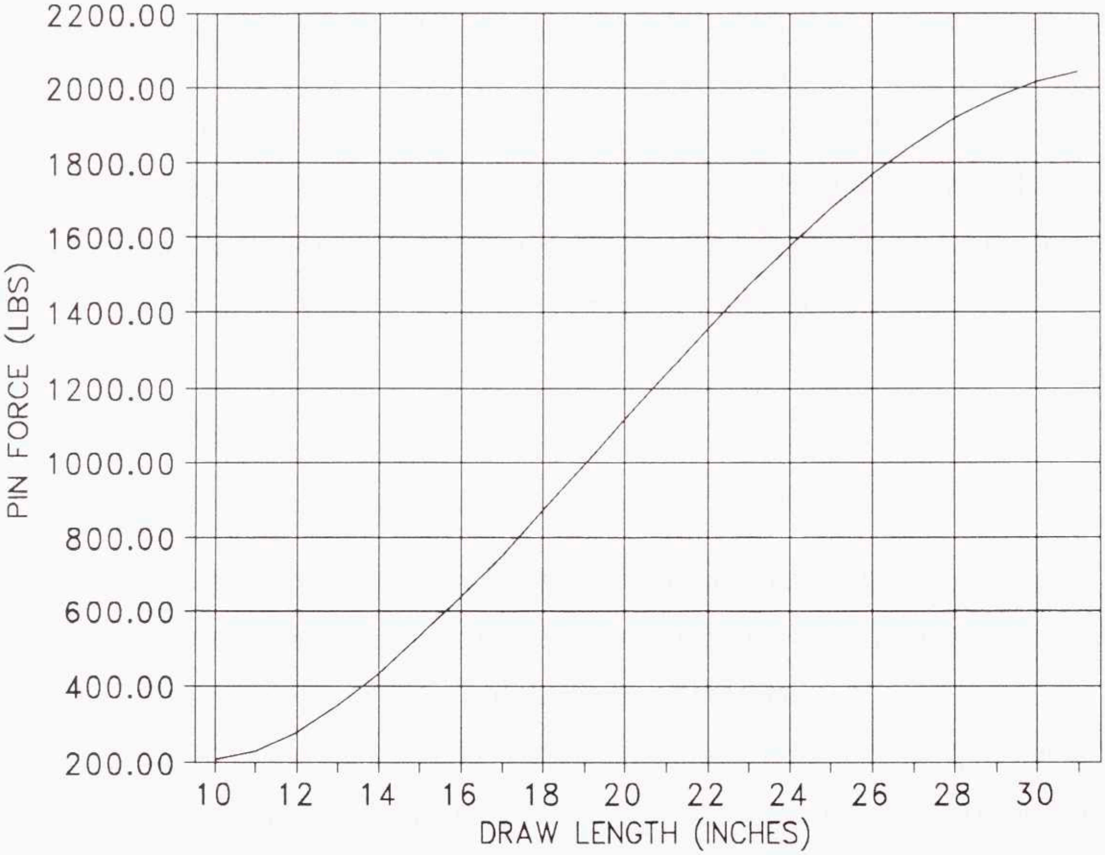


Figure 5.3 Force on Main Pivot

5.3.2 Stress Calculations

The main pivot support can be modeled as a plate loaded in tension by a pin where the width (w) is the hinge minimum diameter, 1.125", the thickness (t) is 0.75", and the pin diameter (d) is 0.375". The theoretical stress (σ_o) on the hinge is

$$\sigma_o = F/(w-d)t \qquad \text{(eq 5.16)}$$

This geometry has a stress concentration factor of approximately 4. For a load of 2000 lbs, the maximum stress in the hinge is $\approx$14,000 psi. The yield strength for the material, 6061-T6, is 40,000 psi, making the design consistent with a safety factor of 2.

5.4 Limb Design

5.4.1 Goals and limitations

The limb design is a compromise between ease of machining, cost, rigidity, and weight. Ease of machining and cost are complementary, however, rigidity and weight are each, for the most part, attained at the expense of the other.

As mentioned in section 5.1, 6061-T6 aluminum is used for the main portion of the bow. It is commercially available, easy to machine, and relatively strong (yield strength of 40,000 psi). It is a lightweight metal, but still heavy for this application. This is a concern, however, as explained in section 5.1, weight is a secondary concern for this prototype. One of the main concerns in designing for the use of 6061-T6, is that the material strength comes from the heat treating process. It is important not to use any manufacturing process that ruins the integrity of the material, such as welding. The geometry of the design must be able to be entirely machined and/or assembled.

The structure of the limb is an I-beam cross-section with a linear taper from the large ends of the handle and riser sections, to the points where the string attaches. This geometry offers the greatest rigidity for the least material (weight).

The limbs and handle and riser sections could conceivably be machined from two solid pieces of aluminum. This would maintain the integrity of the material, but would require a great deal of machining. The other option is to assemble the bow from smaller machined pieces. This option is practical and cost-effective. The bow is therefore made of four separate sections of aluminum, with the limbs attached to the handle riser sections by steel dowel pins.

5.4.2 Bending moment in limb

To determine the dimensions of the limb, it is necessary to derive a model to give an estimate of the stresses in the material. The loading and geometries are complex, but the goal is to derive a simple model to give a rough estimate of stresses in the limb to act as a guideline for determining limb dimensions.

The majority of the stress in the limb is assumed to be from bending. This is true for angles β around 90°. For smaller angles, the limb is under compression, which, although a storage of applied energy, not a likely material failure mode. The angle β does exceed 90°, placing the limb in tension. However, this is at large draw lengths, when the draw force has begun to decline. Even in this case, the major component of the force is still normal to the limb rather than tangential. The loading is modeled as an end load on a cantilever beam (fig 5.4). The load is the component of the string tension force normal to the limb. The interface between the limb and handle or riser section is modeled as a rigid support. This is justified, because the handle and riser sections are considerably more rigid at the interface than the limb sections. The bending moment in the system of fig. 5.4 is

$$M_b = Fx \qquad \text{(eq 5.17)}$$

where

$$F = F_{st}\sin\beta \qquad \text{(eq 5.18)}$$

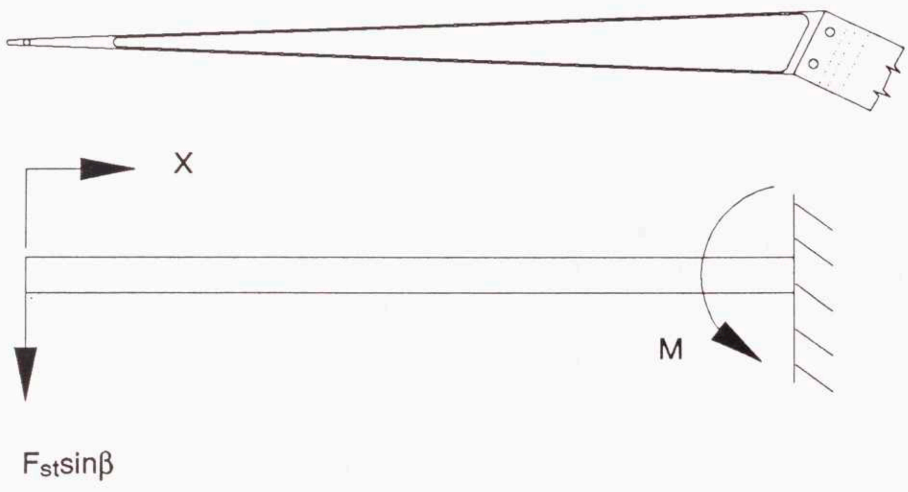


Figure 5.4 Modeling Forces on Limb

and x is the distance from the point of action of the load. The actual angle of F_{st} to the physical limb is slightly larger than β, but contributes little error to the calculation.

5.3.3 Stress calculations

The I-beam dimensions for the interface and end sections are shown in fig. 5.5. The maximum stress due to bending is given by eq 5.6

$$\sigma_b = -M_b y_{max}/I_{zz} \qquad \text{(eq 5.6)}$$

where I_{zz} is the moment of inertia of the cross section. I_{zz} for this section is given by:

$$I_{zz} = th^3/12 + wt^3/6 + (wt/2)(h+t)^2 \qquad \text{(eq 5.19)}$$

The limbs have a linear taper

$$h = x(h_i-h_f)/L_l + h_f \qquad \text{(eq 5.20)}$$

$$w = x(w_i-w_f)/L_l + w_f \qquad \text{(eq 5.21)}$$

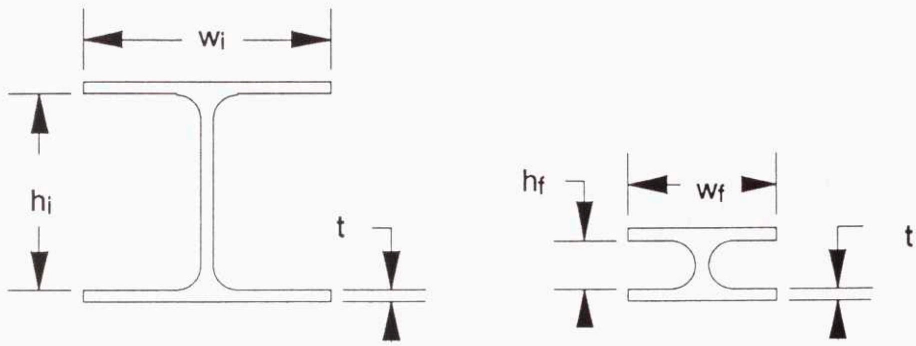


Figure 5.5 Limb I-beam Cross-section Dimensions

L_l is the length of the limb

$$L_l = [(a-h_l)^2 + c_o^2)^{1/2} \qquad \text{(eq 5.22)}$$

where a is 1/2 the string length, h_l is the length of the handle and riser sections, and c_o is the initial draw length. The physical system has the following parameters:

$h_i = 2"$, $h_f = 0.25"$, $w_i = 2"$, $w_f = 0.5"$, $t = 0.0625"$

These equations and values were added to the spreadsheet simulation. The results are shown in fig 5.6.

According to the model, the maximum stress occurs approximately 8.7" from the point of action of the string force. The stress at this location vs. draw length is shown in fig. 5.7. The maximum stress predicted by the model is well below the yield stress of the material, 40,000 psi.

Considering the low predicted stress, the size of the limb could be reduced, however, any reduction in size occurs at the expense of rigidity. Also, a web thickness of less than 0.0625" would require more accurate machining, which, in general, is less cost effective.

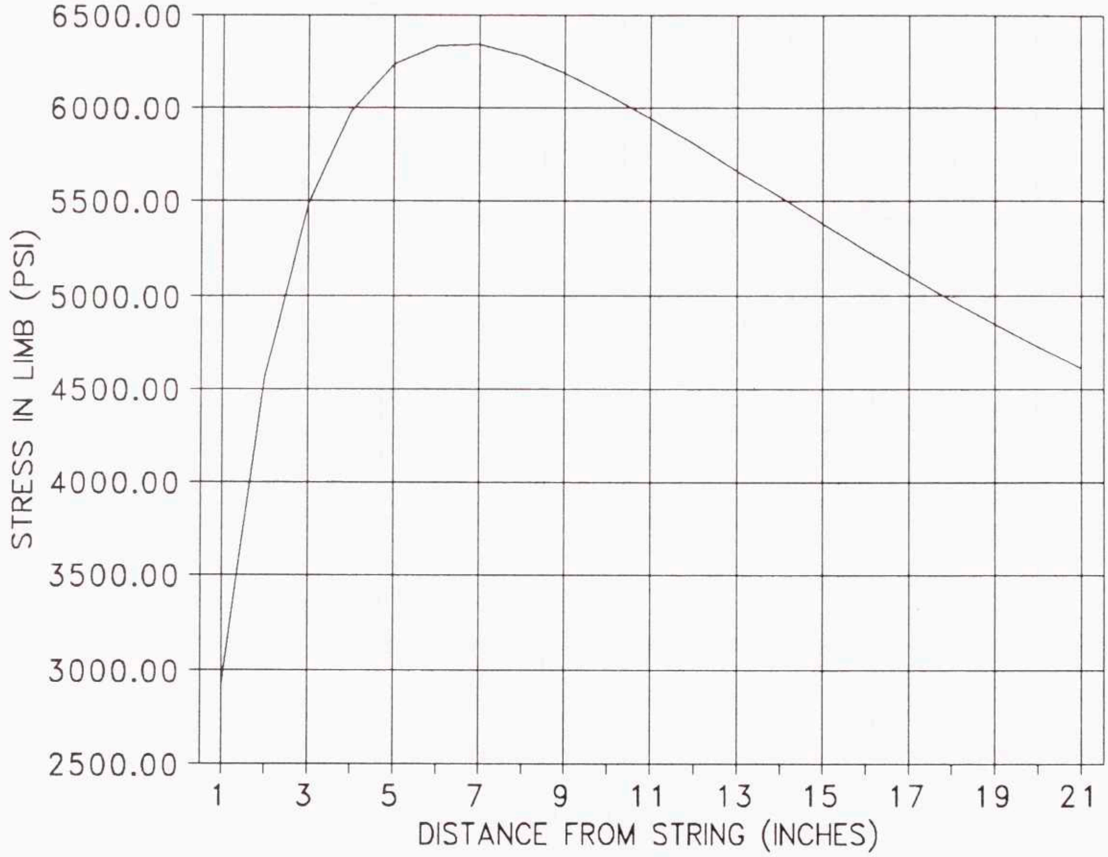


Figure 5.6 Stress Due to Bending Along Length of Limb

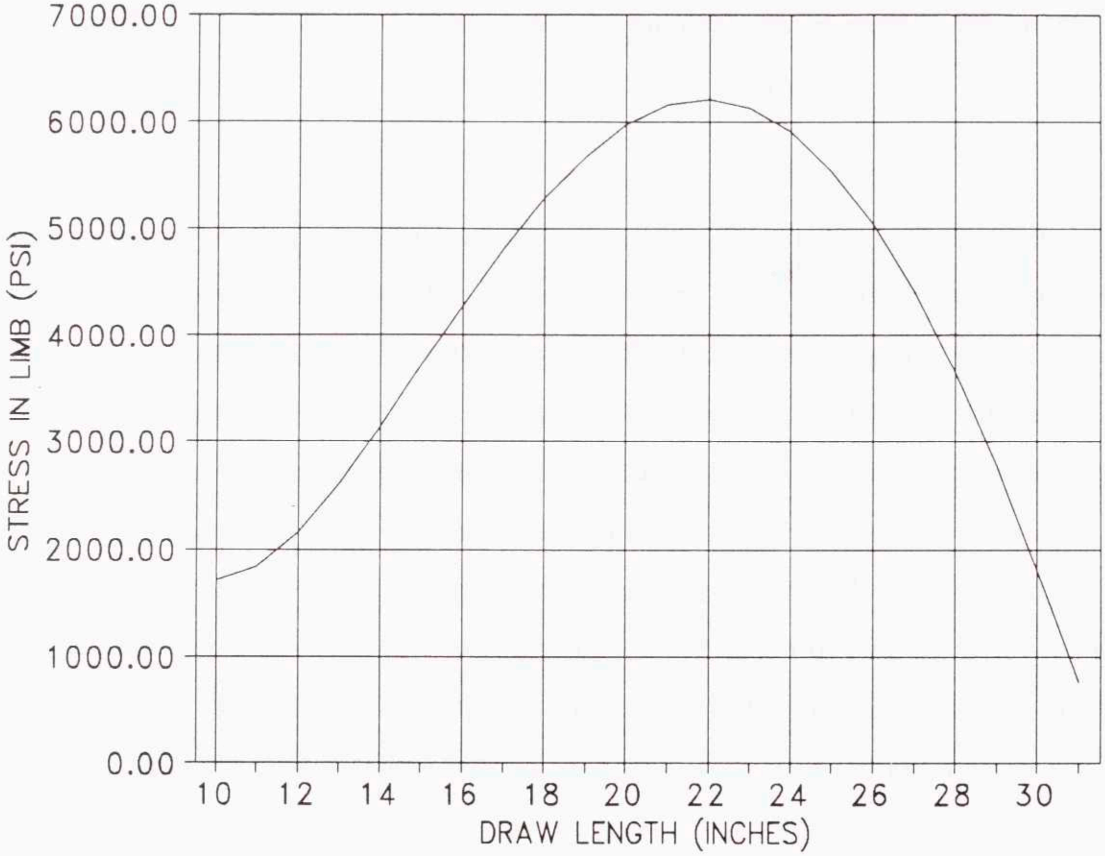


Figure 5.7 Stress in Limb at Maximum Stress Point, 8.7" From String.

5.4.3 Assembly pin design

The limbs are rigidly attached to the handle and riser sections with steel dowel pins press fit in the aluminum (fig 5.7, item 9). The maximum moment on the limb at the attachment location is 1045 lb-in (from the spreadsheet simulation, appendix A). Balancing the moments about the center of the joint, the force on each of the pins is given by

$$M_b = Fd \qquad \text{(eq 5.23)}$$

where Mb is the bending moment from the string force, F is the force on each pin, and d is the distance between the pins. The distance between the pins is 1," giving a load of 1000 lbs on each pin. This gives a shear of 500 lbs, with a resulting stress for a 1/4" diameter pin (eq 5.7) of 13,600 psi. This stress is far below the yield stress for steel.

The aluminum around the steel pin can be modeled as a plate loaded in tension by a pin where the width (w) is 1/2 the width of the limb, 1", and the thickness (t) is 3/4." The pin diameter used is 1/4." Using equation 5.16, the theoretical stress on the joint is 1780 psi. The stress concentration factor, related to the height of aluminum above the pin, 3/8," is approximately 5.3. This gives a stress of 9400 psi in the aluminum. This is well below the yield stress for the material. A diameter for the assembly pins of 1/4" is satisfactory for the design.

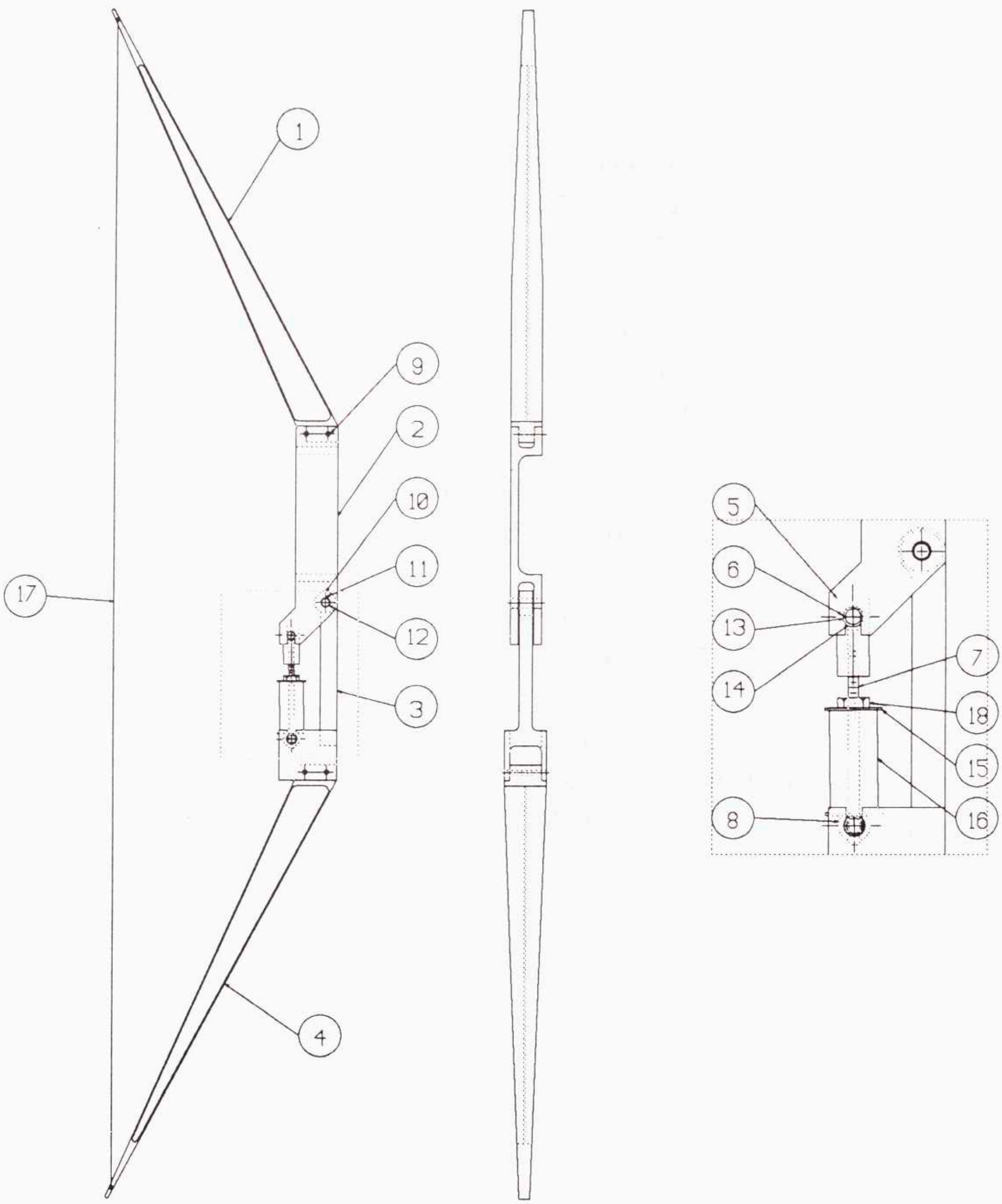


Figure 5.7 Prototype Design for Compound Bow Using Over-center Spring Mechanism

Table 5.1 Prototype Design for Compound Bow

REF NO.	DESCRIPTION
1	UPPER LIMB
2	RISER SECTION
3	HANDLE SECTION
4	LOWER LIMB
5	UPPER HINGE
6	UPPER PIN
7	SPRING SHAFT
8	LOWER PIN
9	STEEL DOWEL PIN (1/4" DIA)
10	TEFLON WASHER
11	BEARING
12	MAIN PIVOT PIN
13	TEFLON WASHER
14	BEARING
15	STEEL WASHER
16	URETHANE POLYMER SPRING
17	BOW STRING
18	NUT

5.5 Machining and assembly

The parts list and detailed drawings for all parts are in appendix B. Most machining and assembly notes are incorporated in the drawings, and will not be dealt with in detail here. This section contains a brief summary of the guidelines used for determining part geometries for machining and assembly. It also outlines which dimensions can be altered without effecting the performance characteristics.

5.5.1 Limbs

Both limbs and handle and riser sections are machined from 6061-T6 aluminum. Most tolerances are very large, with the limiting factors being the spring size, and handle size. The dimensions which must be held very close are the mechanism and main pivot hole locations. These must be located precisely to attain the desired force-draw characteristic.

The upper and lower limbs are identical with one exception, the width of the upper limb is trimmed to 1.5" to match the smaller size of the riser section. The width is the least significant variable in the stress calculations, so it was not necessary to redo the calculations for this alteration. The two limbs can be made in the same set-up, then the upper limb can be trimmed. This should be more cost-effective since a major portion of machining time for unique parts is in set-up.

As described in section 5.4.1, The limbs are attached to the handle and riser sections by hardened steel dowel pins. Instructions for assembly are in appendix B. Prior to assembly, it is recommended that the parts be laid out and measured. The important measurements are the distance from the pivot to the point at which the string attaches, and the angle at which it occurs. These dimensions are given in section 4. Once these dimensions have been checked, assemble as described in the drawings, then measure again, and machine the string notches. It is very important that all edges be rounded in and around the string notches to prevent damage to the string.

5.5.2 Mechanism

The mechanism spring member is designed to be adjustable in length in order to change the preload, or allow for slight variations in spring size. The length is changed by either moving the nuts that secure the upper spring support, or moving the entire shaft in or out of the upper hinge. The shaft has a slot on the lower end to allow the use of a screwdriver to change the length, however, in the prototype, this is not accessible. If this slot proves difficult or expensive to machine, it can be eliminated.

The bearing holes in the handle-riser sections are designed to take partial bearings inserted from the side. The upper and lower pivots slide into place through slots in the aluminum and bearings. Currently, there are no retaining rings to prevent the bearings from slipping out. It is recommended either retaining rings be added, or a handle be fitted that will serve this function.

5.6 Handle

There are no detailed drawings included for a handle. The limiting dimensions are determined by the volume needed by the spring, and the range of motion of the limbs. These minimum interior dimensions for the handle are shown in fig 5.8. The only limitation on the exterior dimensions is that the point of action of the draw force be roughly in line with the main pivot through the full range of motion. A suitable handle can be machined from wood or plastic, and secured to the lower section of the bow.

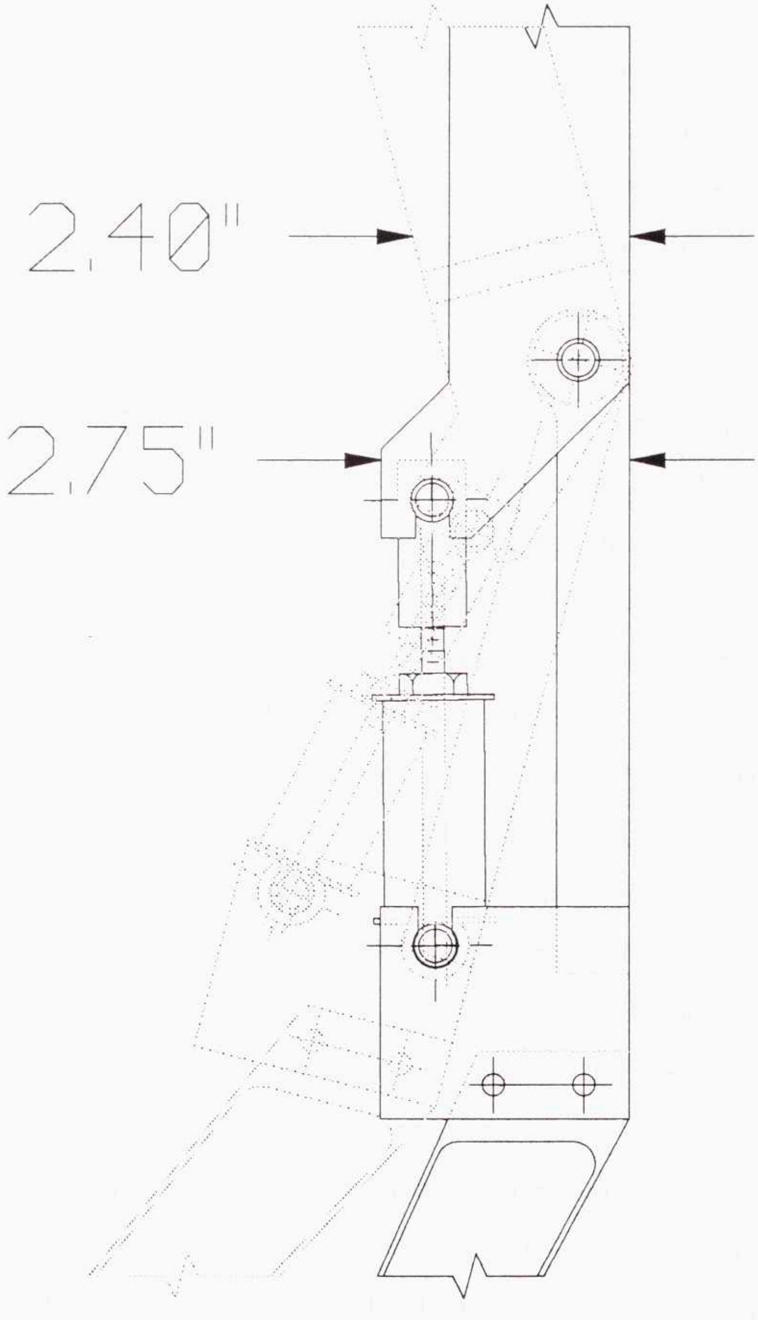


Figure 5.8 Minimum Handle Interior Dimensions

5.7 Safety

Due to the large forces involved in the operation of the bow, there exists a potential for injury to the user, unless proper precautions are taken. The major safety concerns center around the mechanism, because this location has the highest forces and highest stresses in the system. All parts have been designed to take stresses much greater than occur in regular operation of the bow, so failure is not a major concern, however precautions should be taken so if something does fail, it does not endanger the operator.

One of the main safety problems in using the over-center spring is the potential for drawing the bow to zero draw force. When this occurs, the spring releases the large amount of energy stored during draw, into the limbs. This can be prevented by installing a 'bumper' in the front of the handle section in line with the upper pivot. The bumper, if in the form of a spring, would also provide the parabolic rise in draw force that occurs in the existing compound bow after a draw of 30."

Another concern is the location of the mechanism. The mechanism is located in the center of the bow, with the energy storage element aimed directly at the user. In the case something in the mechanism or the main pivot fails during draw, or the limbs are not kept under control during stringing or unstringing, parts of the mechanism are propelled at high speeds, not only toward the user, but toward vital areas. It is therefore necessary that some sort of handle be securely placed around the mechanism, not only before testing or use, but before stringing the bow. The handle should not be removed before unstringing. Unfortunately, in the prototype, this means the bow must be unstrung to make adjustments in the mechanism. The screwdriver slot in the mechanism shaft will be made accessible without removing the handle in future designs.

Another safety concern is "pinch points" around the handle. It is recommended that any areas where this possibility exists, i.e. between the top of the handle and the upper limb, that a soft material, such as foam-rubber, be inserted, so as to not interfere with the motion of the limbs, but inhibit fingers from entering and getting pinched.

The other dangers exist in releasing the bow. No dynamic calculations have been done on this bow. The limbs are heavy, and will presumably have a great deal of momentum when released. The operator needs to be careful to stay clear of these limbs after releasing the bow. As with any bow, the bow should not be fired without an arrow in it, or the system must absorb the energy. This energy will be absorbed in the form of flailing, heavy, aluminum limbs, adding to the likelihood of the user being hit. Also, because no dynamic calculations have been done, and the rather unorthodox rotating handle and arrow rest, when firing an arrow, there is no guarantee it will travel straight, in fact, it is unlikely that it will. The bow has the draw weight of a hunting bow, so an uncontrolled arrow is very dangerous.

My final safety recommendation is that the prototype be used only for what it was designed for, testing the force-draw characteristic to determine if the mechanism is a viable alternative to the existing pulley system.

6 Conclusions and Recommendations

The spreadsheet simulation indicates that the over-center spring mechanism will give nearly the same force-draw characteristic as the existing compound bow. The use of an elastomer tube spring as the energy storage element allows the mechanism to be made small enough to be concealed in the handle section. The prototype design is thorough, and should provide a bow suitable for testing.

The most important element in this implementation of the over-center spring mechanism is the elastomer spring. It is the key that makes this particular implementation practical. The calculations assume no friction along the spring shaft, or upper and lower supports. This is clearly not the case, however the exact magnitude of its impact is not known. Before the design is ruled out due to inefficiency, attempts should be made to reduce the friction in this area. One possibility is to coat the shaft with teflon, so the sliding is rubber on teflon, another is to line the spring with teflon, so the sliding is teflon on steel. Either way, the material wants to expand into the shaft area, causing large forces. Ways should by investigated to use the spring material in a shape that eliminates this problem.

The limbs in the prototype are relatively rigid. It should be determined how the bending, how ever small, of the limbs effects the force-draw characteristic. When dynamics come into consideration in the final design, weight will be a major concern, and it will be beneficial to be able to optimize the system for all factors.

The handle and arrow rest in the prototype rotate with the limbs. For a usable bow, a handle-riser section needs to be designed that keeps the handle and riser perpendicular to the pivot. The rotating handle produces a large moment on the wrist, making the user want tilt the bow upward. This makes the bow difficult to hold, and more difficult to aim. In addition, because the arrow rest rotates with the upper limb, the arrow travels along an arc as it's released. This further complicates aiming.

The major deficiency in this design is that it deals only with statics. The function of a bow is to fire an arrow, which would indicate the overwhelming concern is dynamics. The prototype bow is designed with concern primarily for the energy storage, or draw characteristics of the bow. Still lacking, is an analysis of the dynamics of the system, including energy lost in the movement of the bow, the behavior of the arrow, and the effects of dynamic loading.

The design is still far from complete, however, I am confident that this prototype will give valuable insight into the practicality of the over-center spring mechanism, that will allow movement into the next steps in the design process.

Appendix A Spreadsheet Simulation Results

STRING 1/2 LENGTH (a)	28	28	28	28	28	28	28
LIMB LENGTH (b)	29.73214						
DRAW (c)	10	11	12	13	14	15	16
SPRING PRELOAD (Fso)	250						
SPRING CONSTANT (k)	3600						
SMALL RADIUS (R1)	2.25						
LARGE RADIUS (R2)	6.75						
SPRING LENGTH (L)	5	4.994714	4.980708	4.960216	4.934927	4.906149	4.874925
GAMMA	1.227772	1.226149	1.221814	1.215382	1.207288	1.197846	1.18729
BETA	0.343024	0.378745	0.414506	0.450337	0.486266	0.522315	0.558505
ALPHA INITIAL	0.566804						
ALPHA	0.566804	0.563556	0.554886	0.542023	0.525835	0.506951	0.485839
SPRING LOAD (Fs)	250	269.03	319.45	393.2207	484.2629	587.8622	700.2696
THETA	2.330751	2.335009	2.346401	2.363376	2.384862	2.410098	2.43853
STRING TENSION (Fst)	40.77377	39.74996	42.85821	47.99309	53.83775	59.56545	64.65752
DRAW FORCE (Fd)	-1.3E-14	2.710224	5.612385	9.097591	13.18476	17.72781	22.51467
SPRING ENERGY (Es)	8.680556	10.05238	14.17337	21.47535	32.57091	47.9975	68.10798
ENERGY RELEASED (Ebr)	0	1.371823	5.492819	12.7948	23.89035	39.31694	59.42742

WINDOW LENGTH	7	7	7	7	7	7	7
I-BEAM LENGTH	21.23676						
Al THICKNESS	0.0625	0.0625	0.0625	0.0625	0.0625	0.0625	0.0625
LARGE HEIGHT	2	2	2	2	2	2	2
SMALL HEIGHT	0.25	0.25	0.25	0.25	0.25	0.25	0.25
LARGE WIDTH	1.5	1.5	1.5	1.5	1.5	1.5	1.5
SMALL WIDTH	0.5	0.5	0.5	0.5	0.5	0.5	0.5
WIDTH	0.547088	0.594176	0.641264	0.688353	0.735441	0.782529	0.829617
MOMENT OF INERTIA	0.002826	0.004527	0.006778	0.009651	0.01322	0.017558	0.022738
HEIGHT	0.332404	0.414809	0.497213	0.579617	0.662021	0.744426	0.82683
LOCATION	1	2	3	4	5	6	7
STRESS FROM Mb	2927.284	4561.168	5477.817	5979.374	6232.082	6331.7	6335.567
BENDING MOMENT (Mb)	49.78001	99.56001	149.34	199.12	248.9	298.68	348.46

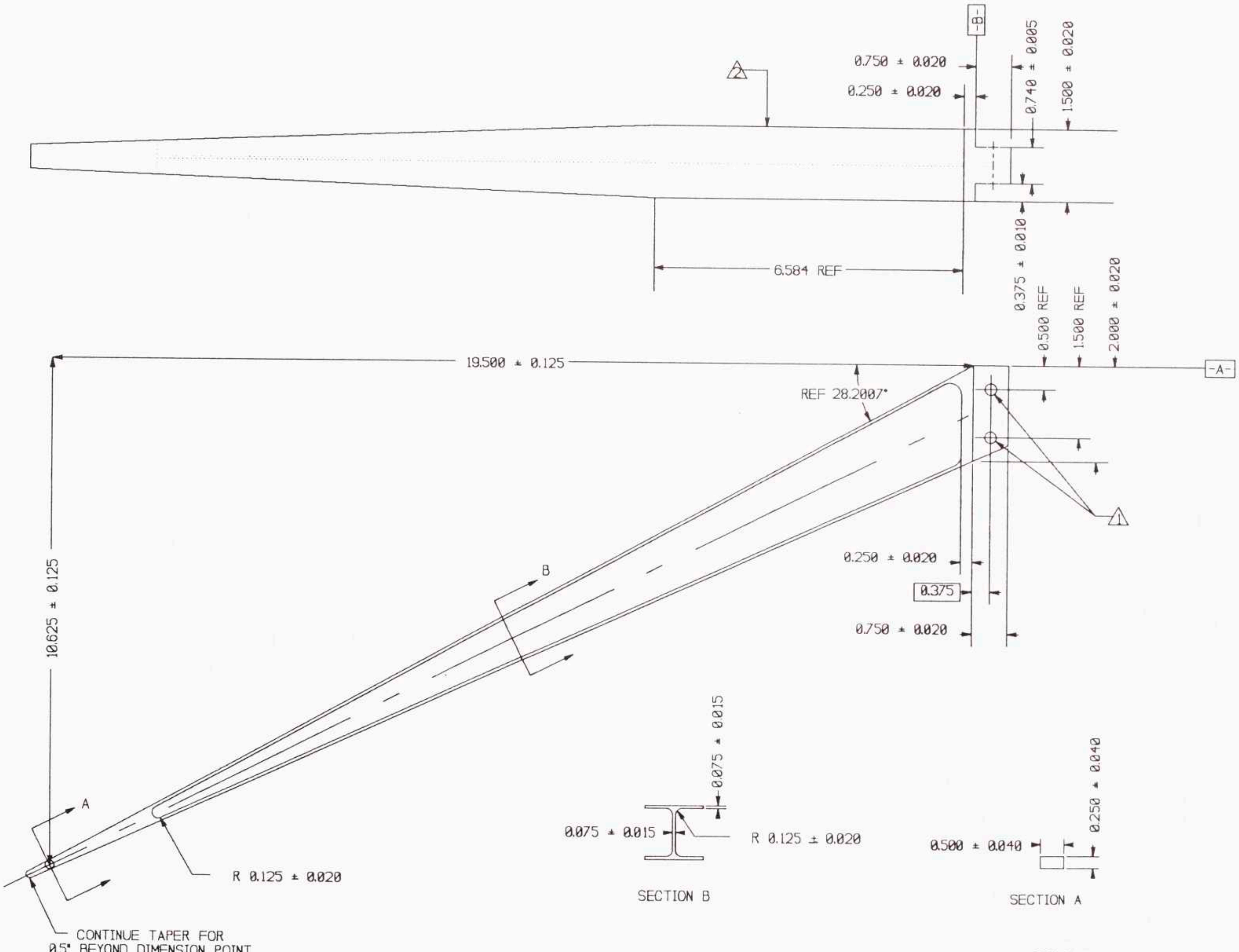
WIDTH	0.909667						
MOMENT OF INERTIA	0.033683						
HEIGHT	0.966917						
LOCATION	8.7						
STRESS FROM Mb	1712.458	1835.338	2155.368	2608.565	3141.764	3710.755	4278.523
BENDING MOMENT (Mb)	119.3092	127.8704	150.1673	181.7421	218.8908	258.5331	298.0903
PIN FORCE	209.2278	229.2992	276.7033	345.5407	431.0897	529.4973	637.5544
MECH ANGLE	1.995784	1.995784	1.995784	1.995784	1.995784	1.995784	1.995784
ANGLE OF PIN FORCE	-0.3334	-0.34419	-0.36101	-0.38244	-0.40754	-0.43564	-0.46629
ANGLE IN DEGREES	-19.1023	-19.7209	-20.6844	-21.9122	-23.3501	-24.9605	-26.7163
STRING 1/2 LENGTH (a)	28	28	28	28	28	28	28
DRAW (c)	17	18	19	20	21	22	23
SPRING LENGTH (L)	4.842103	4.808394	4.774403	4.740663	4.707648	4.67579	4.645488
GAMMA	1.175796	1.163499	1.150502	1.136888	1.122719	1.108047	1.09291
BETA	0.594857	0.631391	0.668124	0.705077	0.742268	0.779719	0.817449
ALPHA	0.462851	0.438256	0.412263	0.385034	0.356697	0.327353	0.297079
SPRING LOAD (Fs)	818.4277	939.7822	1062.148	1183.612	1302.467	1417.158	1526.244
THETA	2.469746	2.503435	2.539361	2.577339	2.617221	2.658891	2.702254
STRING TENSION (Fst)	68.79203	71.77511	73.49813	73.91091	73.00492	70.80281	67.35191
DRAW FORCE (Fd)	27.31448	31.90005	36.05829	39.59513	42.33789	44.13682	44.86641
SPRING ENERGY (Es)	93.0311	122.6654	156.6885	194.5746	235.6139	278.9355	323.5308
ENERGY RELEASED (Ebr)	84.35055	113.9848	148.008	185.894	226.9333	270.2549	314.8502
WINDOW LENGTH	7	7	7	7	7	7	7
Al THICKNESS	0.0625	0.0625	0.0625	0.0625	0.0625	0.0625	0.0625
LARGE HEIGHT	2	2	2	2	2	2	2
SMALL HEIGHT	0.25	0.25	0.25	0.25	0.25	0.25	0.25
LARGE WIDTH	1.5	1.5	1.5	1.5	1.5	1.5	1.5
SMALL WIDTH	0.5	0.5	0.5	0.5	0.5	0.5	0.5
WIDTH	0.876705	0.923793	0.970882	1.01797	1.065058	1.112146	1.159234
MOMENT OF INERTIA	0.028833	0.035916	0.044061	0.053339	0.063825	0.075591	0.088711

HEIGHT	0.909234	0.991639	1.074043	1.156447	1.238851	1.321256	1.40366
LOCATION	8	9	10	11	12	13	14
STRESS FROM Mb	6279.129	6184.863	6067.317	5936.053	5797.422	5655.669	5513.635
BENDING MOMENT (Mb)	398.24	448.02	497.8001	547.5801	597.3601	647.1401	696.9201
STRESS FROM Mb	4813.87	5290.399	5685.82	5981.509	6162.25	6216.132	6134.551
BENDING MOMENT (Mb)	335.3886	368.589	396.1385	416.7395	429.332	433.086	427.4022
PIN FORCE	752.5306	872.0521	994.0104	1116.494	1237.735	1356.072	1469.92
MECH ANGLE	1.995784	1.995784	1.995784	1.995784	1.995784	1.995784	1.995784
ANGLE OF PIN FORCE	-0.49912	-0.53389	-0.5704	-0.60848	-0.64802	-0.68893	-0.73113
ANGLE IN DEGREES	-28.5977	-30.5898	-32.6813	-34.8633	-37.129	-39.4729	-41.8905
STRING 1/2 LENGTH (a)	28	28	28	28	28	28	28
DRAW (c)	24	25	26	27	28	29	30
SPRING LENGTH (L)	4.617118	4.591038	4.56759	4.547107	4.52991	4.516316	4.506633
GAMMA	1.077339	1.061357	1.044982	1.028226	1.011097	0.9936	0.975734
BETA	0.85548	0.893836	0.93254	0.971618	1.011097	1.051008	1.09138
ALPHA	0.265937	0.233973	0.201223	0.167712	0.133454	0.098459	0.062726
SPRING LOAD (Fs)	1628.375	1722.263	1806.674	1880.415	1942.323	1991.262	2026.123
THETA	2.74723	2.79375	2.841754	2.891188	2.942001	2.994142	3.047565
STRING TENSION (Fst)	62.72046	56.99542	50.28141	42.70017	34.39043	25.50789	16.22506
DRAW FORCE (Fd)	44.42699	42.74657	39.78309	35.52699	30.0039	23.27752	15.45244
SPRING ENERGY (Es)	368.2783	411.9707	453.3434	491.1057	523.9746	550.7119	570.163
ENERGY RELEASED (Ebr)	359.5978	403.2902	444.6629	482.4251	515.2941	542.0314	561.4824
WIDTH	1.206322	1.253411	1.300499	1.347587	1.394675	1.441763	1.488851
MOMENT OF INERTIA	0.103257	0.119303	0.136921	0.156185	0.177168	0.199943	0.224583
HEIGHT	1.486064	1.568469	1.650873	1.733277	1.815681	1.898086	1.98049
LOCATION	15	16	17	18	19	20	21
STRESS FROM Mb	5373.212	5235.641	5101.718	4971.928	4846.542	4725.678	4609.355
BENDING MOMENT (Mb)	746.7001	796.4801	846.2601	896.0401	945.8201	995.6001	1045.38
STRESS FROM Mb	5912.292	5547.669	5042.684	4403.21	3639.143	2764.532	1797.648

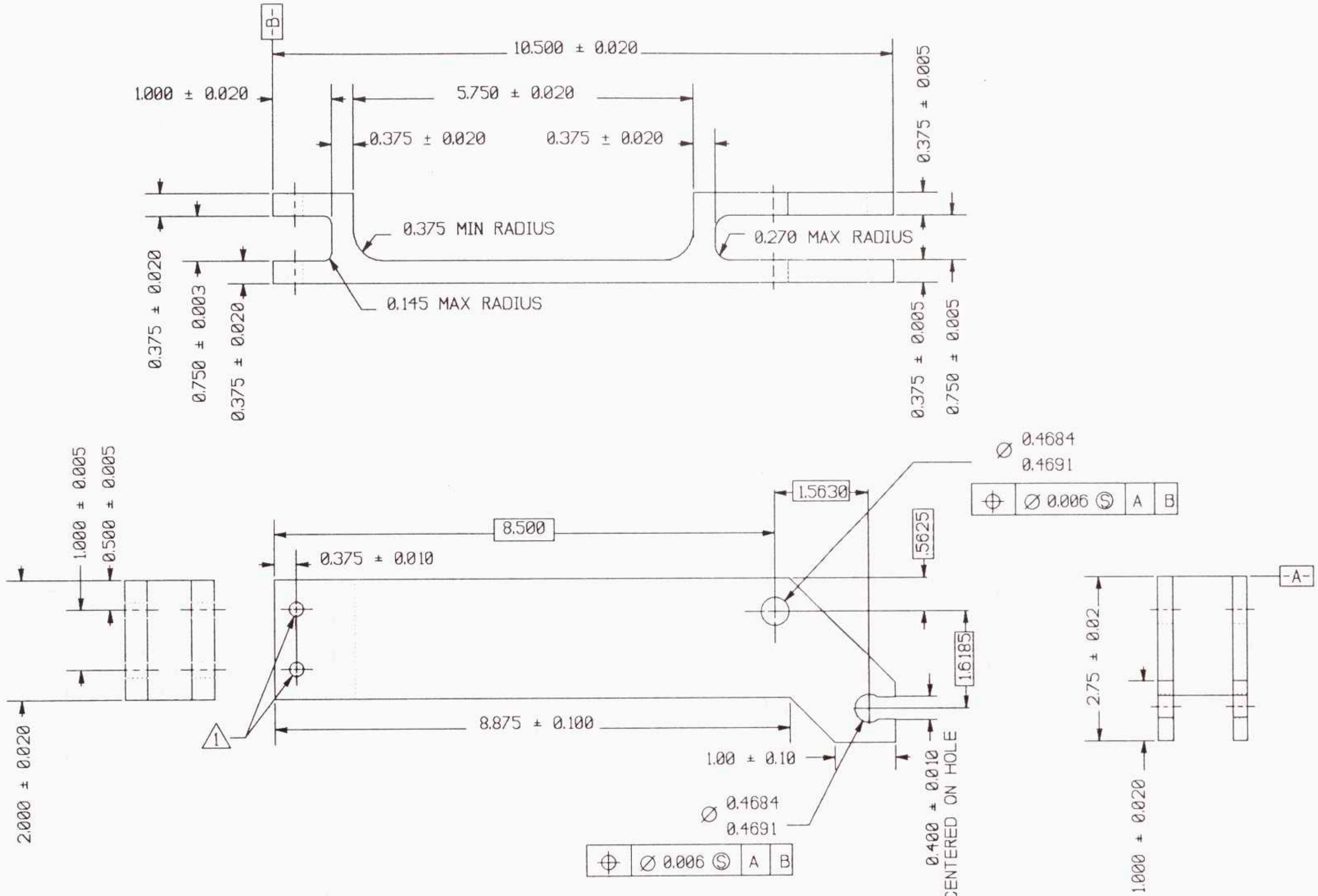
BENDING MOMENT (Mb)	411.9171	386.5133	351.3304	306.7774	253.5438	192.6085	125.2445
PIN FORCE	1577.749	1678.068	1769.417	1850.36	1919.485	1975.406	2016.768
MECH ANGLE	1.995784	1.995784	1.995784	1.995784	1.995784	1.995784	1.995784
ANGLE OF PIN FORCE	-0.77455	-0.81914	-0.86486	-0.91167	-0.95953	-1.00843	-1.05834
ANGLE IN DEGREES	-44.3782	-46.9331	-49.5526	-52.2346	-54.9772	-57.7789	-60.6383

Appendix B Parts List and Detailed Drawings

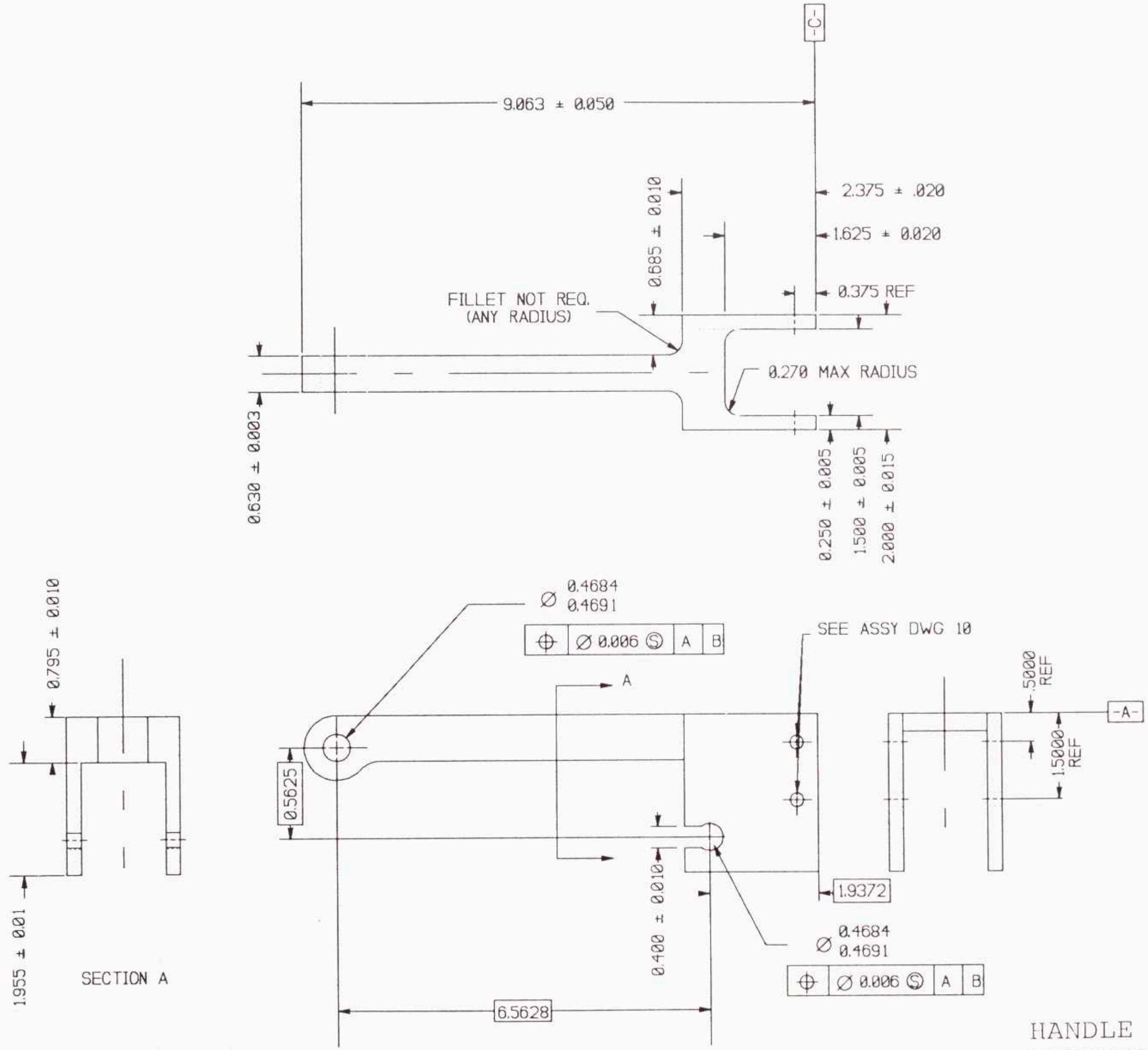
REF #	DWG #	QTY	DESCRIPTION	VENDOR	P/N
1	01	1	UPPER LIMB, 6061-T6 ALUMINUM		
2	02	1	RISER SECTION, 6061-T6 ALUMINUM		
3	03	1	HANDLE SECTION, 6061-T6 ALUMINUM		
4	04	1	LOWER LIMB, 6061-T6 ALUMINUM		
5	05	1	UPPER HINGE, 4340 OR 4130 HEAT TREATED STEEL		
6	05	1	UPPER PIN, STEEL, MAX 32 FINISH FOR BEARING JOURNAL	BERG	
7	06	1	SPRING SHAFT, STEEL		
8	07	1	LOWER PIN, STEEL, MAX 32 FINISH FOR BEARING JOURNAL	BERG	
9		4	STEEL DOWEL PIN (1/4" NOM. DIA.) FOR ASSEMBLING LIMBS	BERG	
10		4	TEFLON WASHER, 3/8" ID, 3/4" OD, FOR UPPER & LOWER PIVOTS		
11		2	UPPER PIVOT BEARING, 3/8" DIA. X 3/8" LONG, SLOTTED TO CLEAR 3/8" SHAFT	GARLOCK	06DU06
11b		2	LOWER PIVOT BEARING, 3/8" DIA. X 1/4" LONG, SLOTTED TO CLEAR 3/8" SHAFT	GARLOCK	06DU04
12	12	1	MAIN PIVOT PIN, STEEL	BERG	
13		2	TEFLON WASHER, 3/8" ID, 1 1/8" OD FOR MAIN PIVOT		
14		2	MAIN PIVOT BEARING, 3/8" DIA. X 3/8" LONG	GARLOCK	06DU06
15		2	STEEL WASHER, 1/4" NOM. ID, 1 1/4" OD FOR SPRING SHAFT		
16		1	URETHANE POLYMER SPRING	LEMPCO	713-3112-200
17		1	BOW STRING		
18		2	NUT, 1/4-28 UNF, SPRING SHAFT		
19		2	E-RING, FOR 3/8" NOM. DIA SHAFT, MAIN PIVOT	WALDES TRUARC	5133-37



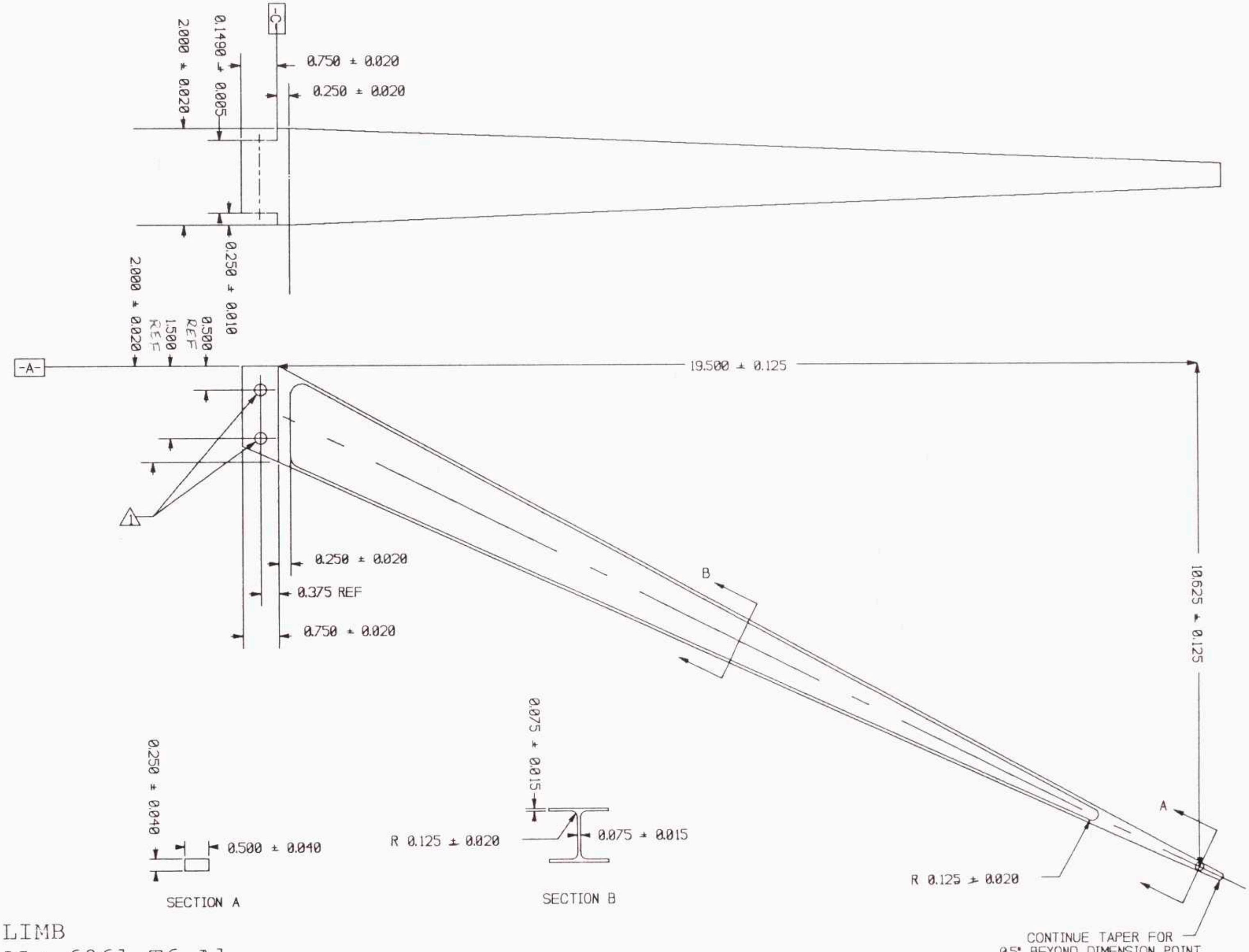
UPPER LIMB
MATERIAL: 6061-T6 Al
DWG NO.: 01



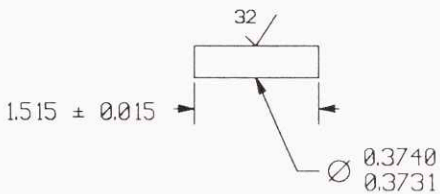
RISER SECTION
MATERIAL: 6061-T6 Al
DWG NO.: 02



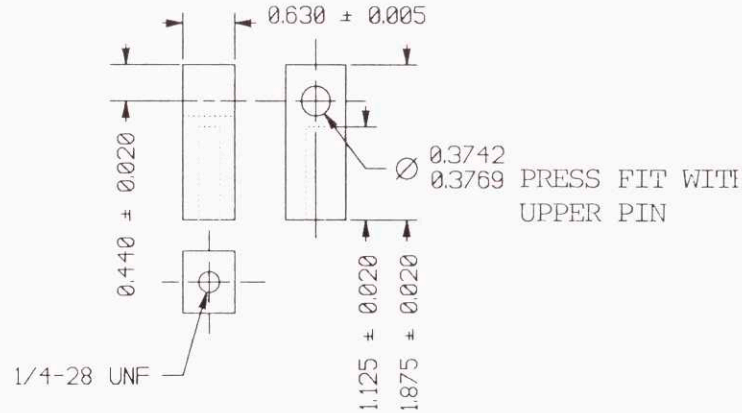
HANDLE SECTION
MATERIAL: 6061-T6 Al
DWG NO.: 03



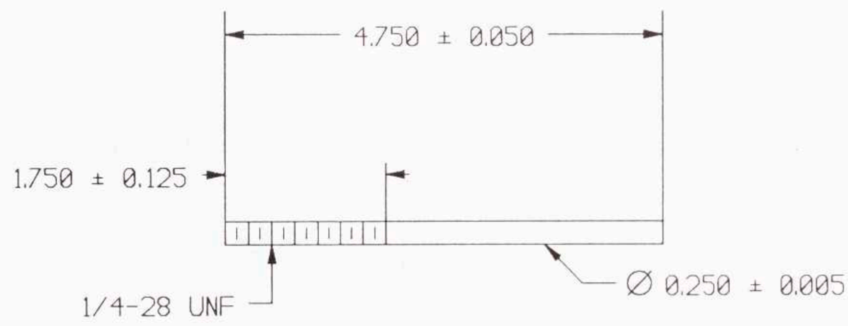
LOWER LIMB
MATERIAL: 6061-T6 Al
DWG NO.: 04



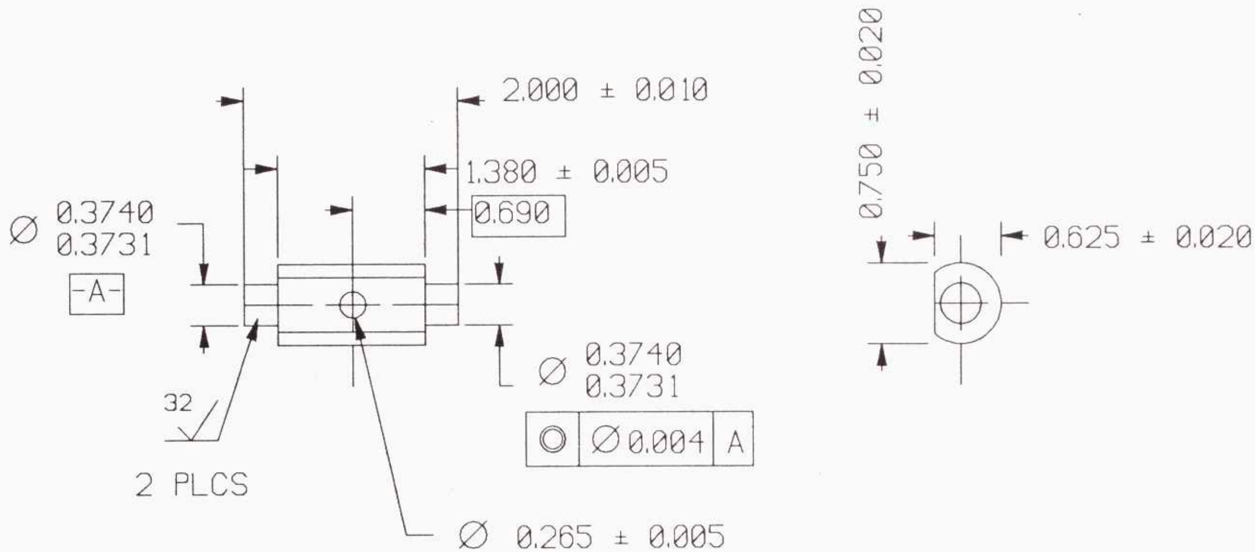
UPPER PIN



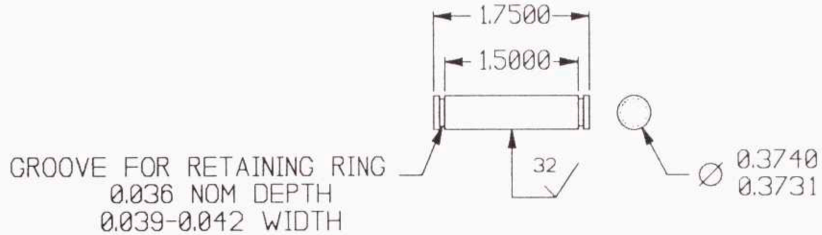
UPPER HINGE AND PIN
MATERIAL: STEEL
DWG NO.: 05



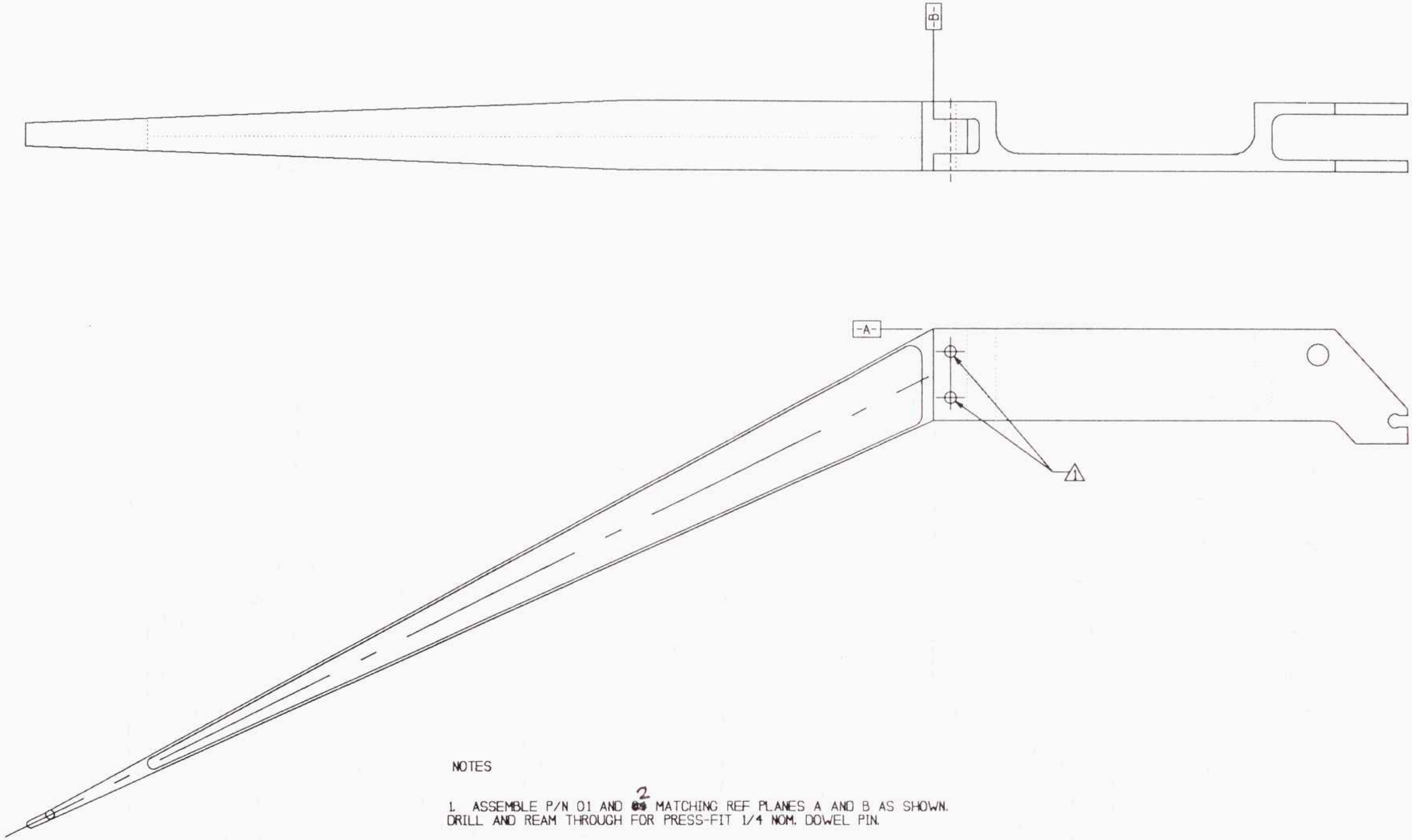
SPRING SHAFT
MATERIAL: STEEL
DWG NO.: 06



LOWER PIN
MATERIAL: STEEL
DWG NO.: 07



MAIN PIVOT
MATERIAL: STEEL
DWG NO.: 12

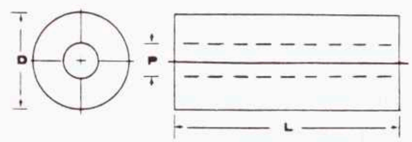


NOTES

1. ASSEMBLE P/N 01 AND 2 MATCHING REF PLANES A AND B AS SHOWN.
DRILL AND REAM THROUGH FOR PRESS-FIT 1/4 NOM. DOWEL PIN.

Appendix C Urethane Polymer Tube Die Spring Catalog Information

Table C.1 Heavy Pressure Tube Springs [Lempco 77]



P Punch Body Diameter	D Outside Diameter	L Free Length in.	L Free Length mm	Catalog Number	Pressure to Deflect Springs ⅛"/3.2 mm lb.	Pressure to Deflect Springs ⅛"/3.2 mm kg.	Pressure to Deflect Springs 3/16"/4.8 mm lb.	Pressure to Deflect Springs 3/16"/4.8 mm kg.	Pressure to Deflect Springs ¼"/6.4 mm lb.	Pressure to Deflect Springs ¼"/6.4 mm kg.
¼ 6.4 mm	1 25.4 mm	1	25.4	713-2510-100	625	283.5	940	426.3	1250	566.9
		1¼	31.8	713-2510-125	500	226.8	750	340.1	1000	453.5
		1½	38.1	713-2510-150	350	158.7	525	238.1	700	317.5
		1¾	44.5	713-2510-175	300	136.0	450	204.1	600	272.1
		12½	317.5	713-2510-000	–	–	–	–	–	–
5/16 7.9 mm	1⅛ 28.6 mm	1	25.4	713-3112-100	800	362.8	1200	544.3	1600	725.7
		1¼	31.8	713-3112-125	625	283.5	940	426.3	1250	566.9
		1½	38.1	713-3112-150	525	238.1	790	358.3	1050	476.2
		1¾	44.5	713-3112-175	500	226.8	750	340.1	1000	453.5
		2	50.8	713-3112-200	450	204.1	675	306.1	900	408.2
		12½	317.5	713-3112-000	–	–	–	–	–	–
⅜ 9.5 mm	1⅛ 28.6 mm	1	25.4	713-3812-100	725	328.8	1090	494.4	1450	657.7
		1¼	31.8	713-3812-125	550	249.4	825	374.2	1100	498.9
		1½	38.1	713-3812-150	475	215.4	715	324.3	950	430.9
		1¾	44.5	713-3812-175	400	181.4	600	272.1	800	362.8
		2	50.8	713-3812-200	300	136.0	450	204.1	600	272.1
		12½	317.5	713-3812-000	–	–	–	–	–	–
½ 12.8 mm	1¼ 31.8 mm	1	25.4	713-5025-100	775	351.5	1165	528.4	1550	703.0
		1¼	31.8	713-5025-125	600	272.1	900	408.2	1200	544.3
		1½	38.1	713-5025-150	500	226.8	750	340.1	1000	453.5
		1¾	44.5	713-5025-175	425	192.7	640	290.3	850	385.5
		2	50.8	713-5025-200	400	181.4	600	272.1	800	362.8
		12½	317.5	713-5025-000	–	–	–	–	–	–
⅝ 15.9 mm	1⅜ 34.9 mm	1	25.4	713-6238-100	875	396.8	1315	596.4	1750	793.7
		1¼	31.8	713-6238-125	700	317.5	1050	476.2	1400	635.0
		1½	38.1	713-6238-150	575	260.8	865	392.3	1150	521.6
		1¾	44.5	713-6238-175	500	226.8	750	340.1	1000	453.5
		2	50.8	713-6238-200	425	192.7	640	290.3	850	385.5
		12½	317.5	713-6238-000	–	–	–	–	–	–
¾ 19.1 mm	1¾ 44.5 mm	1¼	31.8	713-7575-125	1250	566.9	1875	850.4	2500	1133.9
		1½	38.1	713-7575-150	1075	487.6	1615	732.5	2150	975.2
		1¾	44.5	713-7575-175	900	408.2	1350	612.3	1800	816.4
		2	50.8	713-7575-200	800	362.8	1200	544.3	1600	725.7
		2¼	57.1	713-7575-225	725	328.8	1090	494.4	1450	657.7
		2½	63.5	713-7575-250	625	283.5	940	426.3	1250	566.9
		2¾	69.8	713-7575-275	575	260.8	865	392.3	1150	521.6
		12½	317.5	713-7575-000	–	–	–	–	–	–
1 25.4 mm	2½ 63.5 mm	1¼	31.8	713-1025-125	2575	1168.0	3865	1753.1	•	•
		1½	38.1	713-1025-150	2100	952.5	3150	1428.8	•	•
		1¾	44.5	713-1025-175	1800	816.4	2700	1224.7	3600	1632.9
		2	50.8	713-1025-200	1525	691.7	2290	1038.7	3050	1383.4
		2¼	57.1	713-1025-225	1300	589.6	1950	884.5	2600	1179.3
		2½	63.5	713-1025-250	1200	544.3	1800	816.4	2400	1088.6
		2¾	69.8	713-1025-275	1100	498.9	1650	748.4	2200	997.9
		3	76.2	713-1025-300	1000	453.5	1500	680.3	2000	907.1
		4	101.6	713-1025-400	750	340.1	1125	510.2	1500	680.3
		12½	317.5	713-1025-000	–	–	–	–	–	–

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