Torque-Limiting Belt Tensioner

by

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TORQUE-LIMITING BELT TENSIONER

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ABSTRACT

A rise in warranty claims due to rear bearing failures has been a rising trend for SETCO belt driven spindles. Traditional belt tensioning methods for these type spindles currently have no way to prevent over-tensioning the belt which in turn leads to rear spindle bearing failure. To prevent this, a new method of tensioning utilizing engineering design processes was designed.

The most important features of the design improvement based on survey results were the ability to limit to a target tension, increased spindle bearing life, and repeatability of tension. Combining the voice of the customer with careful engineering analysis produced two potentially effective designs to limit belt tension. These design concepts were then analyzed utilizing engineering design tools. The Two-Piece Nut Assembly was selected as the most viable design concept for overall functionality and cost.

The required belt tension was calculated based on the belt manufacturer’s specification for design horsepower and speed. Adding twice the vertical component of belt tension to the total weight to lift, resulted in a total load to lift. The torque to lift was calculated using a simple power screw formula. The normal force at the time of torque-limiting was calculated by dividing that torque by the radius. The vertical component of this force was then divided by the positional change of the ball to determine the proper limiting spring rate needed.

The components were designed to target the customer needs. Limiting the tension with repeatable performance prevents overloading the rear spindle bearings. The tolerances between moving parts were minimized to prevent harmful contaminants from entering the assembly. Proper selection of a lithium based grease guards against the possibility of washout from machine coolant and/or wash down maintenance. The components were also designed to be assembled using only standard available tools. Materials were selected to prevent wear and vital surfaces were black oxide treated to provide durability in a machine tool environment. The assembly was designed to retrofit any standard SETCO belt driven spindle in less than 5 minutes, and at a cost no greater than 5% of the spindle drive package.
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INTRODUCTION

BACKGROUND

For over 90 years, SETCO has designed and manufactured precision spindles and is recognized worldwide as a technical leader in the design and manufacture of precision spindles. SETCO products are used worldwide in a large variety of industries; including automotive, aerospace, construction, die/mold, cabling and winding, plastics, woodworking, stone cutting and general metalworking industries. Spindles are used for many types of manufacturing processes such as milling, boring, grinding, honing, and drilling. The main components comprising a spindle include a stationary housing, a rotating shaft supported by bearings at both ends, and seals to protect the labyrinth areas between the rotating and non-rotating components.

Spindle components such as housings and shafts are manufactured to extremely tight tolerances. Shaft diameters that mount bearings are held cylindrical to within 0.0001” and held concentric front to rear within 0.0002”. Precision angular contact ball bearings, held to ABEC 7 or better bearing tolerances, are mounted to this shaft and then assembled into the housing. Critical diameters on the shaft are held to runout tolerances below 0.0002” for most spindles. The ball bearings at the front of the spindle typically handle axial and radial loads induced from the tool and cutting operation. The rear bearings are generally allowed to float axially and primarily offset the radial forces generated from tensioning the belt. The drive belt spans two pulleys, one mounted on the shaft and the other mounted to a motor. This drive belt tension for a belt driven spindle is factory set to a predetermined value prior to shipment. Belts however, are consumable components and should be replaced at regular maintenance intervals. As these belts are replaced, they should be tensioned to the same specifications as originally set from the factory. Unfortunately, many times these belts are assembled incorrectly in the field, resulting in either an under-tensioned or over-tensioned belt. An under-tensioned belt may prevent full power transmission, sometimes even leading to belt slippage. An over-tensioned belt may transfer the power, but could cause a spindle failure due to rear bearing failure. In either case, the spindle is compromised for reasons that could otherwise be avoided.

The focus of this design project is to re-design the tensioning components so that the intended tension is inherent to the design. Eliminating the possibility of human error in tensioning would effectively eliminate one of the leading causes of premature spindle failure. When a spindle performs the way it is designed to perform, the customer remains satisfied as production continues and SETCO benefits from lower warranty costs. Meeting or exceeding the customer needs and delivering cutting edge technological features allows SETCO to remain the industry choice for their precision spindle needs.

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1 The ABEC scale is a system of rating the manufacturing tolerance of precision bearings. The system was developed by the Annular Bearing Engineering Council (ABEC), a division of the American Bearing Manufacturers' Association (ABMA). (13)
There are several different methods of tensioning a spindle drive belt. The most common practice in the machine tool industry is to mount the motor and drive pulley to a slider plate as shown in Figure 1 (1) and Figure 2 (1). The large upright rectangular bracket mounted to the back of the spindle is the belt guard. This belt guard houses the motor pulley, spindle pulley, and belt. With this type of assembly, an overhead crane must be used to hold the weight of the motor, motor plate, drive pulley, threaded rod mounting block, and adjusting rod when replacing the belt. After the replacement belt is in place, the upper hex jam nut is used to pull the sliding assembly upward, thus tensioning the belt. This assembly is inexpensive, it requires ordinary tools, and is relatively easy for any experienced maintenance person to handle alone. The downside to this design is that it permits the assembler to tension the belt to any value. Under-tensioning may lead to the belt slipping or possibly even breaking. Over-tensioning can cause excessive shaft deflection, leading to decreased bearing life or bearing failure.

Figure 1 - Current SETCO Tensioning Method
Figure 2 is a close-up look at the tensioning assembly for the spindle shown in Figure 1. Tension is either increased or decreased by turning the upper hex jam nut. Slots in the motor plate allow for the moving assembly to travel up and down for belt replacement and re-tensioning.

![Figure 2 – Slider Plate and Tensioning Assembly](image)

**ALTERNATE MOVING MOTOR DESIGNS**

Other spindle manufacturers use slightly different variations of the moving motor design. As shown in Figure 3 and Figure 4 (2), the vertical adjustment of the motor is provided as sprockets turn four separate threaded rods simultaneously by means of a chain. This is a safe but expensive and time consuming method to tension a belt.

![Figure 3 – Alternate Moving Motor Design](image)  ![Figure 4 - Close-up of Chain](image)
UNSAFE BELT TENSIONING DESIGN

The tensioning design shown in Figure 5 (3) is a crude design not fit for machine tool standards. The motor swings through an arc in order to tension the belt, but the method to do so is unsafe. The belts themselves are not guarded, exposing the operator and other nearby workers to unsafe working conditions. The motor is supported only by the adjusting screw on the one side. This could result in excessive spindle vibrations which can adversely affect part quality.

![Figure 5 - Unsafe Belt Tensioning Design](image)

TYPICAL MOVING PULLEY DESIGN

The assembly in Figure 6 (4) is another moving pulley design similar to the one in Figure 5. The pulley here is supported well, reducing vibration and providing a safe method of adjustment. This would be a typical tensioning method for an application having very light power transmission.

![Figure 6 - Typical Moving Pulley Belt Tensioner](image)
**FIXED CENTERS WITH OUTSIDE IDLER DESIGN**

Another tensioning method makes use of an idler to take up belt slack. Both pulleys are mounted fixed and the belt and idler are assembled last. Figure 7 (5) shows how the Snapidle® tensioner mounts to the outside of the belt. This design allows for ease in belt replacement but is reserved for relatively low speed drive applications.

![Figure 7 - Fixed Centers with Idler](image)

**FIXED CENTERS WITH ARM-TYPE IDLER DESIGN**

The arm-type idler shown in Figure 8 (6) mounts to a spring-loaded shaft. The roller houses a small radial bearing that offsets the load imposed on the arm from the belt. This design is typically used to tension on the outside of the belt for toothed belts. For flat belts it may be used on either the inside or outside of the belt. As the belt wears and stretches, the spring in the arm housing compensates to keep constant tension on the belt.

![Figure 8 - Fixed Centers with Arm-Type Idler Design](image)
**MECHANICAL IDLER**

Another idler shown in Figure 9 (7) is a mechanical idler. It is similar in design to the arm-type idler, but it also incorporates a mechanism to dampen the system. Changes in belt load are offset as a dampener hub integral with the driven pulley works to absorb fluctuations in the driven sprocket.

![Figure 9 - Mechanical Idler](image)

**ELECTRONICALLY ENERGIZED IDLER**

Idler designs exist for many different applications. Shown in Figure 10 (8) is a highly specialized idler assembly that can be remotely energized to tension the belt. The piston rod is held by an electro-mechanical braking system that can release to allow for the tensioning piston to actuate.

![Figure 10 - Electronically Energized Idler](image)

For more information on the many types of tensioning devices and methods, refer to Appendix A.
CUSTOMER FEEDBACK, FEATURES, AND OBJECTIVES

SURVEY ANALYSIS

Twenty-five of the thirty customer surveys distributed were returned and analyzed to determine the customer needs. The surveys were completed by spindle assemblers (8), engineers (7), service technicians (7), upper management personnel (2), and purchasing agents (1) to get a wide range of customer feedback (See Appendix B for full survey and customer results). The survey included a wide range of product features arranged in categories of operation/function, maintenance, and cost.

The first half of the survey focused on customer importance as they were asked to rate the importance of features on a scale from 1-5 with 5 being the most important. The results in Table 1 have been sorted from most important to least important.

<table>
<thead>
<tr>
<th>Rank</th>
<th>Question surveyed</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Ability to automatically limit to a target tension</td>
<td>4.68</td>
</tr>
<tr>
<td>2</td>
<td>Increased spindle bearing life</td>
<td>4.64</td>
</tr>
<tr>
<td>3</td>
<td>Durability to machine tool conditions</td>
<td>4.52</td>
</tr>
<tr>
<td>4</td>
<td>Repeatability of tension</td>
<td>4.48</td>
</tr>
<tr>
<td>5</td>
<td>Cost</td>
<td>4.32</td>
</tr>
<tr>
<td>6</td>
<td>Compactness of design</td>
<td>4.04</td>
</tr>
<tr>
<td>7</td>
<td>Maintenance-free operation</td>
<td>3.92</td>
</tr>
<tr>
<td>8</td>
<td>Ease of installation</td>
<td>3.76</td>
</tr>
<tr>
<td>9</td>
<td>Ability to retrofit into existing drives</td>
<td>3.60</td>
</tr>
<tr>
<td>10</td>
<td>Versatility to pulley sizes</td>
<td>3.32</td>
</tr>
<tr>
<td>11</td>
<td>Serviceability for belt replacement</td>
<td>3.28</td>
</tr>
<tr>
<td>12</td>
<td>Versatility to belt type</td>
<td>3.12</td>
</tr>
<tr>
<td>13</td>
<td>Ability to assemble without special tools</td>
<td>2.48</td>
</tr>
<tr>
<td>14</td>
<td>Ability to reduce belt slippage</td>
<td>1.48</td>
</tr>
<tr>
<td>15</td>
<td>Ability to reduce belt &amp; pulley wear</td>
<td>1.45</td>
</tr>
</tbody>
</table>

Customer Importance
Rank 1 to 15 is Most Important to Least Important

The second half of the survey listed the same questions, but surveyed the customer for satisfaction with the current. Their satisfaction was ranked on a scale from 1-5 with 5 being the most satisfied. The results in Table 2 are ranked from least satisfied to most satisfied.

<table>
<thead>
<tr>
<th>Rank</th>
<th>Question surveyed</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Repeatability of tension</td>
<td>1.88</td>
</tr>
<tr>
<td>2</td>
<td>Ability to automatically limit to a target tension</td>
<td>2.48</td>
</tr>
<tr>
<td>3</td>
<td>Cost</td>
<td>3.12</td>
</tr>
<tr>
<td>4</td>
<td>Increased spindle bearing life</td>
<td>3.16</td>
</tr>
<tr>
<td>5</td>
<td>Versatility to pulley sizes</td>
<td>3.28</td>
</tr>
<tr>
<td>6</td>
<td>Serviceability for belt replacement</td>
<td>3.32</td>
</tr>
<tr>
<td>7</td>
<td>Versatility to belt type</td>
<td>3.32</td>
</tr>
<tr>
<td>8</td>
<td>Ability to retrofit into existing drives</td>
<td>3.41</td>
</tr>
<tr>
<td>9</td>
<td>Ease of installation</td>
<td>3.60</td>
</tr>
<tr>
<td>10</td>
<td>Maintenance-free operation</td>
<td>3.80</td>
</tr>
<tr>
<td>11</td>
<td>Ability to reduce belt slippage</td>
<td>3.92</td>
</tr>
<tr>
<td>12</td>
<td>Durability to machine tool conditions</td>
<td>4.04</td>
</tr>
<tr>
<td>13</td>
<td>Ability to reduce belt &amp; pulley wear</td>
<td>4.32</td>
</tr>
<tr>
<td>14</td>
<td>Compactness of design</td>
<td>4.44</td>
</tr>
<tr>
<td>15</td>
<td>Ability to assemble without special tools</td>
<td>4.48</td>
</tr>
</tbody>
</table>

Customer Satisfaction
Rank 1 to 15 is Least Satisfied to Most Satisfied
The survey results show that the ability to automatically limit to a target tension, increased spindle bearing life, durability to machine tool conditions, and repeatability of tension were the most important features to the customer. Three of these four features were also ranked among the top four features the customer is least satisfied with. With this evidence, new design concepts should address these desirable features as the voice of the customer.

**PRODUCT FEATURES AND OBJECTIVES**

The product objectives were developed from the list of customer features as determined from the survey. These customer features were cross-referenced with engineering characteristics and rated for a strong to weak correlation. Ranked below are the features in order of customer importance followed in parentheses by the average response in the customer survey. Below each feature is the engineering characteristic followed by the method or objective that will be used for the prototype design.

The project goal was that these product objectives be delivered from a product designed to limit the belt tension, as specified by the belt manufacturer, when tightening the motor drive belt of a standard SETCO Sentry belt drive. These product features and objectives are:

1. **Ability to automatically limit to a target tension** (4.68)
   - The assembly will be designed to limit the belt tension to a factory set value suitable for the following drive package:
     a. The assembly will be mounted to a SETCO standard size B070 Sentry boring spindle driven by a 3 HP at 3450 rpm (145TC frame) AC induction motor. The sprockets will be Goodyear Eagle Pd™ White series (8mm pitch) (9).
     b. The belt will be tensioned for a spindle whose operating speed will be 6469 rpm through a speed up sprocket ratio of 1.875:1.
       i. Motor sprocket: 75 tooth, Eagle Pd™ White series
       ii. Spindle sprocket: 40 tooth, Eagle Pd™ White Series
       iii. Belt: Eagle Pd™ White series, 1000mm long
       iv. Center distance: 10.49 inches
     c. The torque limiting value will be in accordance with the corresponding tension value as published by the belt manufacturer within +/- 15%. Goodyear recommends a belt strand tension for the above described drive to be 63 lbs. Measurement will be verified using a Goodyear type RSM2000 Belt Frequency Meter (10).

2. **Increased spindle bearing life** (4.64)
   - Limiting the tension will prevent over-loading the rear spindle bearings. Spindle shaft displacement due to belt tensioning will be monitored for conformance to acceptable levels as determined by the SETCO engineering department. Pulley shaft deflection will be measured on the periphery at the far end of the sprocket shaft and will not deflect more than .0005” due to belt tension forces.
3. **Durability to machine tool conditions (4.52)**
   
a. The assembly will be designed to operate in a machine tool environment and perform for the life of the spindle: properly sealed, and subject to vibration.
   
b. Parts will be hardened as necessary to prevent wear that could otherwise affect the performance of the product.

4. **Repeatability of tension (4.48)**
   
a. Belt tension will return to the allowable tension range after every belt replacement and re-tensioning (assuming the use of new belts).
   
b. Repeatability testing will require that re-tension values fall within +/- 15% of the manufacturers specifications.
   
c. Conformance to repeatability will be determined as follows:
      
      i. Tension the belt by turning the outer nut.
      
      ii. When the device acts to limit the tension, lock the tensioning assembly in place with a standard hex jam nut.
      
      iii. Verify that the belt tension conforms to manufacturers requirements by measuring the tension with the Goodyear RSM2000 Belt Frequency Meter.
      
      iv. Loosen the belt by backing off the standard hex jam nut and releasing the 2-Piece Nut Assembly from the tensioned state. Continue turning the outer nut to allow for belt removal.
      
      v. Disengage the teeth in belt from sprocket and engage them in another random set of sprocket teeth.
      
      vi. Re-tension the belt.
      
      vii. Verify the belt tension.
      
      viii. Repeat 5 times.

5. **Cost (4.32)**
   
a. The cost of adding this feature to the spindle will not exceed 5% of the sell price for the belt drive package. With the drive package selected for this design project, the tensioning package should not exceed $250.

6. **Compactness of design (4.04)**
   
a. The design will be such that existing SETCO standard belt drive parts can be used without modification.

7. **Maintenance-free operation (3.92)**
   
a. Parts will be designed to hold lubrication inside the assembly eliminating the need for re-lubrication.

8. **Ease of installation (3.76)**
   
a. Part design will limit the amount of time to install and require no more than five minutes to do so. Time required to tension is defined to be a single process of repeatability test and will be recorded at that time (refer to objective 4).
9. **Ability to retrofit into existing designs (3.60)**
   a. Standard SETCO parts will be used wherever possible, thus allowing existing spindles to be retrofit with the 2-Piece Nut Assembly.

10. **Versatility to pulley sizes (3.32)**
    a. Flexibility of design will be considered as many belt drive combinations are offered as standard packages by SETCO. Due to time limitations for this project, only one standard drive offering will be tested as specified in objective 1. Interchanging of springs will accommodate other tensions for drives not tested.

11. **Serviceability for belt replacement (3.28)**
    a. The product will be designed to accommodate field service where possible (refer to objectives 8 and 13).

12. **Versatility to belt type (3.12)**
    a. The design will accommodate flat belts, v-belts, and synchronous type belts. The scope of this project however will be limited to testing only the Goodyear Eagle Pd™ white series belt with the drive components as specified in objective 1.

13. **Ability to assembly without special tools (2.46)**
    a. Standard, readily available fasteners will be used allowing for assembly to the spindle with no more than 2 standard tools.

**CONCEPT GENERATION AND SELECTION**

**2-PIECE NUT DESIGN**

The first concept, shown in Figure 11 is referred to as the 2-Piece Nut Design. As a torque is applied to the outer nut by means of a wrench, the spring force on the balls transfer the torque to the inner nut. Because the outer nut is not allowed to move downward against the fixed adjusting plate, it forces the threaded adjust rod and the components attached to it to move upward. The threaded adjust rod is fixed to the moving components including the motor, motor plate, and threaded rod mounting block (refer to Figure 2). The upward movement of these components engage the motor sprocket with the belt to tension it. The current design of these components however does not limit the torque. The more the upper hex jam nut is turned, the more the belt tension is increased. The 2-Piece Nut Design however limits the torque when the opposing force of the inner nut pushes the ball far enough up the ramp to allow it to pass under the ball. When this occurs, the belt is properly tensioned and a standard hex jam nut and washer can be used to cover the top of the outer nut. At this point the system is factory set for the optimal belt tension and the screws on the motor can be tightened down to hold tension. In the event that the belt needs to be changed or disassembled, reversing this process will loosen the belt. The inner nut is contoured to grab the balls to reverse the direction of the inner
nut. With this design, it would be important to hold tight tolerances to keep the balls in relative like positions at all times; however, the simplicity of the design would allow it to be integrated immediately into the current SETCO belt drive standard assembly.

Figure 11 - 2-Piece Nut Design

**THRUST NUT DESIGN**

The second concept shown in Figure 12 is the Thrust Nut Design. The concept of the force of the inner nut on the ball limiting the tension is the same, but the adjust rod is turned rather than the outer nut. When the preset tension is reached the ball will be forced from the detent in the periphery of the inner nut, thus allowing the inner nut and adjust rod to spin freely without increasing tension. The needle thrust bearing transfers the lifting force from the inner nut upward to lift the block which is in turn mounted to the motor slide plate. A possible flaw with this design is whether or not the nut would release when attempting to loosen the belt. This assembly also would require more space to mount on the motor plate which would require a change to adapt to standard belt drive parts.
The two concepts were evaluated using a weighted decision matrix, graded on a five-point scale, with scores ranging from zero being inadequate, to four being excellent. Only the design criteria that were different between the two concepts were scored using relative weights taken from the QFD (See Appendix C). The weighting factors from the five criteria as shown in Table 3 were adjusted from the QFD to total 1.00. The scores given for each criteria were multiplied by the weight factors and then added together to determine the total rating for each concept. The 2-Piece Nut Design was the obvious choice to develop as it scored higher in all of the criteria.

Table 3- Weighted Decision Matrix

![Figure 12 - Thrust Nut Design](image-url)
CALCULATIONS

CALCULATING THE TORQUE REQUIRED TO LIFT AND TENSION

Initial calculations were used to determine a theoretical spring force required to tension the belt. With the calculated force and known displacement of the ball, a theoretical spring rate was targeted. By selecting a spring with an outside diameter that had many available spring rates, flexibility was added to the design allowing for further optimization without changing the geometry of the manufactured parts. In order to accurately estimate the torque required to lift and tension, the following theory was used to model the system.

The 2-Piece Nut Assembly should limit the torque applied to the outer nut by allowing the inner nut to slip at the predetermined limiting force on the balls. The inner and outer nuts should then turn together to lift the load and tension the belt until the force on the balls exerted from the inner nut resisting to turn would cause the balls to ride over the top of the inner nut. When this occurs, the assembly will have limited the tension of the belt to the manufacturer’s specified value. Lastly, a standard hex jam nut and hardened washer should be used to seal off and lock the assembly in place.

A simple power screw formula was used to calculate the torque required to lift the load.

\[ T_{LIFT} = \left( \frac{(P \cdot d_m)}{2} \right) \left( L + \left( \frac{\pi \cdot \mu \cdot d_m \cdot \sec \alpha}{(\pi \cdot d_m) - (\mu \cdot L \cdot \sec \alpha)} \right) \right) \]

The total load \((P)\) for this calculation was determined as the summation of all the moving components added to the required belt tension force, or \((2 \times T_Y)\) as shown in Figure 13. See Appendix F for the complete calculation of the total load \((P)\).

![Figure 13 - Belt Layout](image)
The adjusting rod used is a commercially available 1/2-13 UNC threaded rod. For a v-thread such as this, the normal thread load is inclined to the axis by an amount $\alpha$, which is equal to half of the included thread angle. The effect of this is to increase the frictional force by the wedging action of the threads. The frictional terms therefore must be divided by $\cos \alpha$. The coefficient of friction is published for screw-thread friction applications such as this to be .15 (9). The screw thread data are published values for standard coarse-thread unified screw threads (10). The equation for the torque required to lift the load ($T_{LIFT}$) in in*lbs is:

\[
P = \text{Total load to lift} = 203.9 \text{ lbs} \\
\alpha = \frac{1}{2} \text{ the included angle (degrees)} = 30^\circ \\
\text{Pitch} = \text{Pitch of screw} = 13 \text{ (threads per inch)} \\
L = \text{Lead of screw (in)} = 1/\text{Pitch} = 1/13 \text{ t.p.i} = 0.077 \text{ in} \\
d_m = \text{Mean diameter of threaded rod (in)} = 0.450 \text{ in} \\
\mu = \text{Coefficient of friction} = 0.15
\]

\[
T_{LIFT} = \left( \frac{203.9 \text{ lbs}}{2} \right) \left[ \frac{0.077 \text{ in} + (\pi)(0.15)(0.450 \text{ in})(\sec 30^\circ)}{(\pi)(0.450 \text{ in}) - (0.15)(0.077 \text{ in})(\sec 30^\circ)} \right]
\]

\[
T_{LIFT} = 10.5 \text{ in*lbs}
\]

**CALCULATING THE FORCE ON THE BALL**

To translate the torque to lift and tension into an equivalent force acting in the radial direction of the screw axis, $T_{LIFT}$ was divided by the radius at which the force was applied. Figure 14 shows how the radial distance of the force was determined graphically. The ball is shown in the position at the instant just before it slips to limit tension.

![Figure 14 - Graphical Force Analysis](image-url)
At this instant, the inner nut will stop rotating and allow the balls to push up into the holes that house the springs. The force $F_n$ shown is the greatest value for which the ball remains on the ramp. Force $F_n$ passes through the center of the ball and acts perpendicular to the ramp at this instant in time. Dividing the torque previously calculated by the radius at which force $F_n$ acts yields the total force required to set the set the springs. $F_n$ was calculated given the following:

\[
F_n = \frac{T_{\text{LIFT}}}{n r_t} = \frac{10.5 \text{ in*lbs}}{(3)(0.380 \text{ in})} = 9.2 \text{ lbs}
\]

The vertical component of this force $F_s$ is the spring force required. It can be calculated using the cosine function of the angle $\theta$ which is determined graphically.

\[
F_s = \text{Spring force required per spring (lbs)}
\]

\[
\theta = 41.0^\circ
\]

\[
F_s = (\cos \theta)(F_n) = (\cos 41^\circ)(9.2 \text{ lbs}) = 7.0 \text{ lbs}
\]

By measuring the position of the ball in the extreme positions, the total deflection $d$ was determined. The radial position to the center of the ball when seated in the curved hook of the inner nut is .438”. When the assembly passes the limiting torque to release the balls they increase to a radial position of .625”. The target spring rate $K_t$ thus becomes the spring force $F_s$ divided by the positional difference of the ball.

\[
F_s = \text{Spring Force (lbf)} = 7.0 \text{ lbs}
\]

\[
p_1 = \text{position 1 (in)} = 0.437 \text{ in}
\]

\[
p_2 = \text{position 2 (in)} = 0.625 \text{ in}
\]

\[
\Delta p = \text{change in position (in)} = p_2 - p_1 (\text{in}) = 0.188 \text{ in}
\]

\[
K_t = \frac{F_s}{\Delta p} = \frac{7.0 \text{ lbs}}{0.188 \text{ in}} = 37.2 \text{ lbs/in}
\]
THE COMPONENT DESIGN FEATURES

The components of the assembly as shown in Figure 15 were designed to perform in a machine tool environment. Tight clearances were held between the moving parts to prevent harmful contaminants from entering the assembly. The spring and ball pockets were greased to reduce the friction between moving and stationary parts. The inner nut and outer nut were black oxide finished for corrosion resistance and aesthetic purposes. With all these features, the simplicity of this design required only a single wrench to accurately and consistently tension the belt.

![Figure 15 - Top View](image)

The outer nut was designed with a standard hex machined integral to the nut body. This allows for tightening with a standard open, box-end, or adjustable wrench. These tools should be readily available to any assembler or maintenance worker.

All outer surfaces were black oxide treated to prevent corrosion. Most spindle environments subject these components to splash coolant and/or water from routine wash down maintenance.

Figure 15 - Top View

Standard and commercially available purchased parts were used wherever possible. The balls are standard parts used in readily available ball bearings. The springs selected for this design have an outer diameter that allows for hundreds of interchangeable springs having variations in spring rate and length. This allows SETCO to manufacture the main components in large quantities and still have the ability to preset the assembly for any number of limiting tensions.

![Figure 16 - Bottom View](image)

The use of a readily available standard snap ring is a simple and economical way to hold the assembly together (16).

Changing the limiting tension is as simple as removing the set screws and springs and replacing them with another pre-determined set. The pre-determined sets can be stocked by SETCO to allow for any number of pulley, belt, speed, and horsepower combinations.

Figure 16 - Bottom View
The cross section views in Figure 17 illustrate how the balls are captivated and forced to move along the axis of the spring as tension is increased. Thrust needle roller bearings are assembled on both sides of the rotating inner nut to minimize the friction between it and the non-rotating parts. On either side of the thrust needle roller bearings are hardened washers. These hardened washers prevent wear induced by the rolling elements in the needle bearings. Equally as important, the washers and bearings are extremely cost-effective and readily available from almost any bearing distributor for only a few dollars each. This eliminates the need for SETCO to manufacture and stock other special, more expensive parts for this assembly. Controlled part tolerances on the manufactured parts guarantee form, fit, and function as parts fit tightly together but still allow for free-turning movement.

![Figure 17 - Cross Section Views](image)

The inner nut shown in Figure 18 was through-hardened to a hardness of Rc 28-32 to reduce wear due to sliding friction of the balls.

![Figure 18 - The Inner Nut](image)

All edges where the balls track have been rounded to remove sharp corners. This will reduce the stress concentrations as the balls roll from one surface to another.

The inner hole is threaded to match the threads of the adjust rod. These threads provide the lifting force that tensions the belt.

The width of the pockets were tolerated to a dimensional width .010” wider than the ball diameter. This prevents the balls from moving vertically and keeps the force against the spring along the axis of compression.
**Figure 19** shows how the 2-Piece Nut Assembly mounts to the standard SETCO spindle. No changes to existing parts are necessary, making it field retrofittable for existing machinery.

![Figure 19 - The Assembly to the Spindle](image)

A standard hex jam nut and hardened washer prevent harmful contaminants from entering from the top.

The 2-Piece Nut Assembly can easily be installed in only a few minutes. It considerably reduces the time required to replace a belt because it eliminates the time otherwise needed to properly set the tension with a belt tension meter or other tensioning device.

**DRAWINGS**

Outline, assembly, and detail drawings provide complete specifications for the product including material selection, geometric dimensioning and tolerancing, assembly instructions, and finishing treatments required. Refer to **Appendix F** for the component part and assembly drawings.

**LUBRICATION**

The proper selection and placement of lubrication will ensure that the device performs as originally intended for the life of the spindle. Most spindle environments include some exposure to water or machining coolant, so the selection of a water resistant grease is crucial to the longevity and performance of the product. PENNZOIL premium wheel bearing grease type 707L is a lithium base grease which contains high quality petroleum base oils (See **Appendix H**). Special additives in 707L prevent rust and corrosion, reduce wear, and resist water washout making it an ideal selection for this application.

**SURFACE TREATMENT OF PARTS**

The inner and outer nuts for the prototype were black oxide treated after finish machining. These parts going forward in production will be treated in batch quantities yielding low treatment costs per part. The advantages of this treatment are:
• **No dimensional changes:** The as-machined dimensions do not change with this treatment. The treatment is a coloring of the base metal, no metal is removed or deposited.

• **Dark black color:** The dark black color is aesthetically attractive.

• **The finish will not chip, peel, flake, or rub off:** The finish will add to the durability of these parts and allow them to withstand the hazardous conditions it may encounter.

• **Improved lubrication characteristics:** The improved lubricity and anti-galling characteristics due to the after-finish result in smoother running mating parts.

**ASSEMBLING THE PARTS**

The simplicity of the parts allow for a quick and easy assembly requiring only four tools: snap ring pliers to set the retaining ring, adjustable wrench to turn the outer nut, an allen wrench to position the set screws, and a grease gun to inject lubricant. **Figure 20** shows the sequence of assembly. Once assembled to the spindle, the device requires only a single wrench to loosen or tension the belt.

![Figure 20 - The Sequence of Assembly](image)

Each spring and ball set is grease lubricated with a lithium based wheel bearing grease PENNZOIL type 707L.

Each needle thrust bearing is backed on either side by a hardened thrust washer. The needle bearings are also grease lubricated with 707L grease.

**TESTING PROCEDURE**

Testing was used to prove the product objectives were met: the ability to automatically and repeatedly limit to a target tension, ease of setting tension and doing so in a timely fashion, and conformance to an acceptable shaft deflection. The 2-Piece Nut Assembly was assembled to a standard SETCO type B070B belt driven spindle as illustrated in **Figure 21**, having the following drive parameters:
- Drive motor: Baldor AC induction motor, 3 HP @ 3450 rpm, 145TC frame
- Drive sprocket: Goodyear Eagle Pd White series, 75 tooth
- Spindle sprocket: Goodyear Eagle Pd White series, 40 tooth
- Belt: Goodyear Eagle Pd White series, 1000mm long
- Center distance: 10.49 inches

**Figure 21 - Test Spindle Setup**

**METHOD TO MEASURE TENSION**

The product objectives for the torque limiting value were required to fall within +/- 15% of the recommended belt tension and take no longer than 5 minutes to do so. The calculations suggest a target belt tension mean of 63.3 lbs., yielding an acceptable tension range from 54-73 lbs..

Tension was measured with a programmable tension meter Goodyear type RSM2000. Time to tension was measured with a stop watch. Prior to starting the procedure, all components were assembled with the belt un-tensioned. The following procedure was used to determine belt tension and time to tension.

**Procedure to Measure Tension and Time to Tension**

a. Start the stop watch.

b. Align the loose belt so that the belt teeth are fully engaged with teeth on the spindle sprocket.
c. Begin the tensioning process by turning the outer nut clockwise to initiate the upward movement of the motor assembly (see Figure 22).

d. Rotate the belt so that its teeth mesh properly with the teeth of the motor sprocket (see Figure 23).

e. Continue rotating the outer nut clockwise until the 2-Piece Nut Assembly reaches the limiting tension value (As tension increases, the balls will ride up and over the ramp on the inner nut, then snap back into the hooked portion on the other side. When this occurs the torque required to turn the outer nut will lessen, followed by a sharp sound of the balls rapidly returning inward against the inner nut. The belt is now tensioned to the limiting value.)

f. Cover the top of the assembly with a hardened washer and lock the assembly in place with a standard hex jam nut.

g. Stop the stop watch and record the time required to tension the belt.
   (Record the shaft deflection at this time)

h. Measure the tension of the belt with the RSM2000 belt tension meter (refer to Appendix A9 for operating instructions).

i. Loosen the belt by turning the standard hex jam nut and 2-Piece Nut Assembly counter-clockwise. Continue turning the outer nut to allow for belt removal.
   (Zero the shaft deflection indicator)

\[\text{Figure 22 - Tensioning the Belt} \quad \text{Figure 23 - Aligning the Belt Teeth}\]

**METHOD TO VERIFY REPEATABILITY**

Conformance to repeatability was measured by repeating the “Procedure to Measure Tension and Time to Tension” five times. It was required that each tension reading fall within the acceptable tension range from 54 - 73 pounds to meet the product objective for repeatability.

**METHOD TO MEASURE SHAFT DEFLECTION**

The shaft deflection was measured using a dial indicator having .000050” graduations. Figure 24 shows how the indicator was used to measure the deflection at the far end of the spindle pulley shaft. Prior to tensioning the belt, the indicator was firmly mounted to the spindle housing with the indicator needle placed at the 12 o’clock position when viewing the spindle rear. The shaft deflection was recorded after each tension measurement. The indicator was zeroed before each tension test with the belt slack.
TEST RESULTS

The results of the tension, repeatability, and shaft deflection tests are recorded in Table 4.

<table>
<thead>
<tr>
<th>Test #</th>
<th>Tension Force</th>
<th>Indicator mounting base</th>
<th>Spindle pulley</th>
<th>12 o’clock position</th>
<th>Dial indicator</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>262 lb</td>
<td>1.25</td>
<td>1.10</td>
<td>1.00</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>265 lb</td>
<td>1.25</td>
<td>1.10</td>
<td>1.00</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>270 lb</td>
<td>1.25</td>
<td>1.10</td>
<td>1.00</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>257 lb</td>
<td>1.25</td>
<td>1.10</td>
<td>1.00</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>273 lb</td>
<td>1.25</td>
<td>1.10</td>
<td>1.00</td>
<td></td>
</tr>
</tbody>
</table>

DISCUSSION OF RESULTS

The time required to tension the 2-Piece Nut Assembly was well below the maximum allowable time of five minutes. The average time to tension was just over three minutes. The time required to initially tension the assembly versus the current method was virtually the same, however the advantage of the 2-Piece Nut Assembly is that it eliminated the time necessary to check tension and/or re-tension.

Shaft deflections were well below the maximum allowable deflection of .0005”, ranging from .00010” to .00015”. Shaft deflections for toothed belt drives such as this are typically much lower than the friction drives that the specification was originally created for, so the low values were expected.

The measured tension forces were all lower than the target force of 63.3 pounds but did fall within the allowable range of 54-73 pounds. Average tension for the five tests was 60 pounds. The low tension values could have been influenced by several factors. Friction of the rolling elements
was neglected in the calculations and could be adding to overall load to lift. Imperfections in the
thread used to lift the load could also have translated into a less efficient power screw. A less
efficient screw would require more torque to turn the inner nut, so the force translated back through
the balls and springs would be more than originally accounted for. The assembly would then reach
the limiting torque sooner than expected, resulting in a lower belt tension. Part tolerances could have
also affected the tension. Friction between the inner and outer nut caused by misalignment could
have increased the load. The calculations also assumed that all balls released at the limiting torque at
the exact moment in time. Part tolerances were used to control these critical features, but variations
from intended geometry should be expected. The 2-Piece Nut Assembly was designed with features
to compensate for these factors. The initial spring heights can be increased or decreased for slight
adjustments, or springs with different spring rates can be inserted to drastically change the limiting
tension value.

PROJECT MANAGEMENT

PROJECT SCHEDULE

The project schedule began January 9, 2008 with the completion of a weighted objective
method and proof of design statement. The timeline covered twenty-three weeks ending on June 9,
2008 with the presentation of the final report. The project remained on schedule until delays in
manufacturing pushed back testing a week. Completion of testing in less than the allotted three
weeks allowed the project to get back on schedule. The time management of the project is further
outlined in Appendix D.

<table>
<thead>
<tr>
<th>Key milestone</th>
<th>Due date</th>
<th>Completion date</th>
</tr>
</thead>
<tbody>
<tr>
<td>Proof of Design Agreement</td>
<td>Jan-10-08</td>
<td>Jan-09-08</td>
</tr>
<tr>
<td>Design Freeze</td>
<td>Feb-20-08</td>
<td>Feb-20-08</td>
</tr>
<tr>
<td>Oral Design Presentation</td>
<td>Mar-12-08</td>
<td>Mar-12-08</td>
</tr>
<tr>
<td>Design Report Due</td>
<td>Mar-17-08</td>
<td>Mar-17-08</td>
</tr>
<tr>
<td>Demo/Proof of Design</td>
<td>May-05-08</td>
<td>May-14-08</td>
</tr>
<tr>
<td>Tech Expo</td>
<td>May-22-08</td>
<td>May-22-08</td>
</tr>
<tr>
<td>Oral Presentation</td>
<td>Jun-02-08</td>
<td>May-28-08</td>
</tr>
<tr>
<td>Final Report Due</td>
<td>Jun-09-08</td>
<td>Jun-09-08</td>
</tr>
</tbody>
</table>

PROJECT BUDGET

A budget of expenses was created to estimate the costs associated with the project. All
expenditures were to be provided by SETCO Sales Company. Actual cost of the prototype project
was $320, exceeding the original budget of $250 by $70. This overage was attributed to the cost of
having to outsource the manufacturing of both the inner and outer nut. Initial plans were for SETCO
to provide these and do so at a lower cost, but backlog for SETCO manufacturing prevented this.
Manufacturing these parts in batch quantities for production runs will drastically reduced the cost per
piece and allow the production assembly to be within budget. Quotes for the production run
manufacturing were obtained from a local vendor, Carter Manufacturing, reducing the cost of the
manufactured parts from $300 to $205 per assembly. The specifics of the forecasted, prototype, and
production budgets are detailed in Table 5.
CONCLUSION

In today’s global business climate, it is particularly important for a company to establish an edge over its competition to be successful. For SETCO, its edge has been the ability to provide super precision products that are unmatched in quality and performance. As a worldwide technical leader, SETCO must continue to set the pace and continue to provide solutions that produce better, more reliable products. The development of this tension limiting device is an effort to meet and exceed the customer needs and provide a product that is superior to any other on the market. This design addresses all of the objectives set forth during product realization. The final product is a versatile, compact, tensioning assembly that delivers extended spindle life at an affordable cost.
REFERENCES

APPENDIX A: BELT TENSIONING REFERENCES

SETCO SALES CO. CURRENT MOVING MOTOR DESIGN

This photo is taken from the SETCO shipping dock. This is the latest design currently being manufactured by SETCO Sales Co. Motor and plate slide to tension belt as hex nut on tension rod is turned. When belt is tensioned properly, a second hex nut from under the adjust plate is turned in to lock down both sides. Then (4) socket head cap screws are tightened to hold entire motor assembly in place. Motor plate is slotted to allow for many combinations of drive ratios.

Photos taken by Brian Schloemer on 9/28/07
Interview with spindle assembler Paul Brighton
Standard SETCO belt drive assembly

Fairly inexpensive
Versatile for many drive ratios

Results below are from interview
Requires overhead crane to tension
Good strength
Does not prevent over-tensioning
Allows for front and rear mounting of drive motor
Photos taken by Brian Schloemer on 9/28/07
Interview with OEM engineer Bill Smith
Alternate Belt Tensioning Assembly Using a Chain Drive Mechanism

Chain drives all (4) sprockets which in turn thread all four hex nuts at once. This raises the motor plate and motor vertically to tension the belt. Separate idler shown in lower photo removes slack from chain.

Comparatively expensive

Results below are from interview
Requires lubrication
Robust design
Dampened design
<table>
<thead>
<tr>
<th><strong>UNSAFE BELT TENSIONING DESIGN</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image_url" alt="Image" /></td>
</tr>
<tr>
<td>This is a picture of an internal grinder headstock spindle being driven with 2 flat belts. The tensioning assembly is a crude form of a swing plate. Shown in the foreground is a hex head screw used to swing the motor to tension the belts. This type of design would not be a viable design for any machine tool product used for precision machining such as a cylinder boring operation.</td>
</tr>
</tbody>
</table>
| Unsafe method of tensioning  
Aesthetically unappealing  
Results below are from interview  
Not precise – may lead to premature belt wear or failure  
Unsafe – should be guarded  
Belts can be replaced easily  |

Photos taken by Brian Schloemer on 9/28/07  
Interview with machine operator Chad Stevens  
Unacceptable Swing Plate Design
**TYPICAL MOVING PULLEY DESIGN**

<table>
<thead>
<tr>
<th>Stainless steel resists corrosion</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inexpensive</td>
</tr>
<tr>
<td>Easy to set up</td>
</tr>
<tr>
<td>Simple</td>
</tr>
<tr>
<td>Convenient</td>
</tr>
<tr>
<td>Adjustable</td>
</tr>
</tbody>
</table>

The existing bolt that locks the alternator in position relative to the adjusting arm is removed. A longer bolt is installed and then the Indigo Belt Tensioning Device is installed on this new bolt such that the short leg of the device is above the upper end of the adjusting arm. The new nut and lockwasher are screwed onto the new longer bolt but not fully tightened. The adjusting bolt in the Tensioning Device is then screwed in until it contacts the top of the adjusting arm. Further turning of the adjusting bolt will thus lift the alternator relative to the adjusting arm. The alternator is then lifted to properly tension the belt. Once the belt is properly tensioned, the new nut and lockwasher are fully tightened to lock the alternator in place.

http://www.atomic4.com/ADJ.htm
9/28/07
Atomic 4 Belt Tensioning Device
Indigo Electronics
**FIXED CENTERS WITH OUTSIDE IDLER DESIGN**

<table>
<thead>
<tr>
<th>Feature</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Free-floating</td>
<td></td>
</tr>
<tr>
<td>Reduces belt slippage</td>
<td></td>
</tr>
<tr>
<td>Tensions v-belt &amp; timing belts</td>
<td></td>
</tr>
<tr>
<td>Dampens vibration</td>
<td></td>
</tr>
<tr>
<td>Easy to install</td>
<td></td>
</tr>
<tr>
<td>Reduces belt &amp; pulley wear</td>
<td></td>
</tr>
<tr>
<td>Requires no special tools</td>
<td></td>
</tr>
<tr>
<td>Speed limited</td>
<td></td>
</tr>
<tr>
<td>May require lubrication</td>
<td></td>
</tr>
</tbody>
</table>

**Snapidle® Belt Tensioner** is a new approach to an age-old problem: "How to tension a belt drive?" Typically maintenance personnel over-tension belt drives at start up. The resulting wear on the belt and pulleys causes slippage. Past solutions have been to add maintenance, replace pulleys and belts, add some sort of mounted tensioning system, or replace the belt drive with another type drive system.

[http://www.snapidle.com/belt_tensioners.htm](http://www.snapidle.com/belt_tensioners.htm)

9/28/07
Snapidle® Belt Tensioner
Snapidle
A split in the base of this idler arm, along with a locking screw, permits easy movement and tight clamping on the shaft. The shaft mounted tensioner when used in pairs, makes a practical snubber for head and tail pulleys.

http://www.brewertensioner.com/shaftmount.html
9/30/07
Shaft Mounted Tensioners
Brewer Machine & Gear Co.
A belt drive device is constructed by looping a timing belt (14) about a driving sprocket (9), a cam sprocket (11), and a driven sprocket (12) for transmitting a driving force from the driving sprocket (9) to the sprockets (11), (12). A dynamic damper (15), comprising a ring mass (153) and an elastic ring (152) for holding the ring mass (153), is mounted on the driven sprocket (12) that has the largest amplitude of torque fluctuation among the sprockets (11), (12). The dynamic damper (15) absorbs revolution fluctuations of the driven sprocket (12) that has the largest amplitude of torque fluctuation. Thus, at the time of resonance during which the belt load on the timing belt (15) sharply increases, the sharp increase in the timing belt load can be minimized, and maximum load on the timing belt (15) can be decreased, whereby the durability of the timing belt (15) can be secured.

Absorbs fluctuations is shock
Sharp increase in the timing belt load can be minimized
Durability
ELECTRONICALLY ENERGIZED IDLER

A belt drive including a first belt sheave having a center, the first belt sheave being pivotally mounted on a member, and a motor base pivotally connected at one end to the member. The belt drive further includes a motor mounted on the other end of the motor base, and a second belt sheave rotatable by the motor. A belt is trained around the first and the second belt sheaves. The belt also includes an adjusting mechanism extending between the motor base and the member for fixing the position of the motor base relative to the member and for fixing the center to center distance between the first and the second belt sheaves. The adjusting mechanism comprises a cylinder assembly including a cylinder housing and a piston rod extending from the housing and being extendable from and retractable into the housing, with the housing being connected to the member and the piston rod being connected to the motor base. The adjusting mechanism further comprises a rod brake mounted on the piston rod, adjacent the cylinder housing with the piston rod extending through the rod brake, for releasably securing the piston rod to the cylinder housing to prevent the retraction of the piston rod into the cylinder.
A major difficulty in installing transmission belts is in achieving the correct belt tension. Lifetime support and reliability of bearings and transmission belts largely depends on optimally adjusted belt tension. In the case of linear actuators, only a correctly adjusted belt tension provides the best possible positional accuracy. The belt tensioning device RSM2000 allows the measurement of the actual belt tension. The RSM2000 can be used with all industrial transmission belts such as V belts, timing belts, poly V belts, etc. The microcontroller-based RSM2000 measures vibrations through a highly sensitive sensor and calculates the tension by numerically filtering the Eigen frequency of the tensioned belt. Given the specific mass of a belt and its free vibrating length, the RSM2000 provides the correct belt tension in Newtons. The RSM2000 comes with several useful features, such as a language switch, user-programmable standard values for specific mass and belt length and an 'in-sensor body-integrated' trigger button. Optionally, an RS232 port for data output of the previous 50 measurements is available.

Operating Instructions for the RSM2000

1. Turn the power on.
2. Input the specific mass value of the belt (supplied by Goodyear).
3. Input the free vibrating length of the belt in centimeters (obtained from measurement).
4. Direct the microphone portion of the meter near the mid-span point and hit START on the keypad.
5. Provide a slight impact to the belt at the approximate mid-span point.
6. Read and record the belt strand tension in Newtons.

http://www.newenglandbelting.com/bt_method1.asp
9/28/07
Frequency Gauge, RSM2000
Goodyear

Extremely accurate
Repeatable
Interchangeable units
Easily programmable
Can output data
Compact
Relatively expensive
Excellent choice for testing
**ECONOMICAL TENSION METERING DEVICE**

The 'Kriket' devices are based on the deflection principle and provide an easy, economical method to directly measure belt strand tension. However, since the gauge calibration is dependent on belt construction, the accuracy of the tension readings is not very high.

(10)

9/28/07
'Kriket' Deflection Devices
Goodyear

Economical
Quick and easy to use
Not extremely accurate
Small size
HELICAL TOOTH SPROCKET DESIGN

| Designed for high speeds |
| Tooth design allows for high power with low noise |
| Kevlar reinforced belts |
| Many stock sizes and widths |
| Require more-intensive alignment procedure |
| Relatively expensive |
| Tooth profile reduces heat |

Is a patented drive system well suited for many high-end, high horsepower drive applications.

Is now available in 3500mm, 3136mm and 3304mm lengths, well-suited for cooling tower applications and other long-center applications.

Has the capacity to reduce noise by 17 – 19db and lessen vibration when compared to other straight tooth synchronous belts.

Can reduce drive size and weight when replacing many traditional v-belt drives.

Has a wide range of operating temperatures for greater versatility.

Is capable of improved efficiencies to optimize your energy dollar.

(11)

[http://www.goodyearep.com/Goodyear/uploadedFiles/Products/Power_Transmission/Synchronous_Belts/eagle_user.pdf](http://www.goodyearep.com/Goodyear/uploadedFiles/Products/Power_Transmission/Synchronous_Belts/eagle_user.pdf)

Eagle Pd™ Synchronous Drive System
Goodyear
TYPICAL MACHINE BELT GUARD

Belt guard cover standard products are available from this source. This is a good source not only for commercially available in stock covers, but also may be a good reference to conform to guard safety standards.

(12)

http://www.machineguard.com/reasons/
9/28/07
Reasons To Buy ABS Plastic Guards
Machine Guard & Cover Co.

Strong
Scratch resistant
Impact absorbent
Rust-proof
Seamless and without rivets
No corners to snag materials
Temperature resistant
Sound deadening
APPENDIX B: CUSTOMER SURVEY AND RESULTS
## APPENDIX C: QUALITY FUNCTION DEPLOYMENT ANALYSIS

### Operation/Function

<table>
<thead>
<tr>
<th></th>
<th>Compactness of Design</th>
<th>Ease of Installation</th>
<th>Repeatability of Tension</th>
<th>Durability to Machine Tool Conditions</th>
<th>Increased Spindle Bearing Life</th>
<th>Versatility to Belt Type</th>
<th>Versatility to Pulley Sizes</th>
<th>Ability to Retrofit into Existing Drives</th>
<th>Ability to Automatically Limit to a Target Tension</th>
</tr>
</thead>
<tbody>
<tr>
<td>Absolute Importance</td>
<td>3.7</td>
<td>4.9</td>
<td>2.4</td>
<td>2.2</td>
<td>3.5</td>
<td>1.8</td>
<td>1.5</td>
<td>2.4</td>
<td>4.6</td>
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<tr>
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### Tensioner now on market

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# APPENDIX D: SCHEDULE

## TORQUE-LIMITING BELT TENSIONER SCHEDULE

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**Winter Quarter**
- Week 5: Setup for testing
- Week 6: Assemble Components
- Week 7: Testing
- Week 8: Tuning
- Week 9: Final Data Collection
- Week 10: Demo/Proof of Design
- Week 11: Prepare for Tech Expo
- Week 12: Tech Expo
- Week 13: Complete Oral Presentation
- Week 14: Oral Presentation
- Week 15: Complete Final Report
- Week 16: Present Final Report

**Spring Quarter**
- Week 1: Complete Oral Presentation
- Week 2: Oral Presentation
- Week 3: Complete Final Report
- Week 4: Present Final Report
## APPENDIX E: BUDGET

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APPENDIX F: CALCULATIONS

CALCULATING THE TORQUE REQUIRED TO LIFT AND TENSION

Initial calculations will be used to determine a theoretical spring force required to tension the belt. With this calculated force and known displacement of the ball, a theoretical spring rate can be targeted. Selecting a spring outside diameter that has many available spring rates will add flexibility to the design and allow for optimization without changing the manufactured parts geometry.

The 2-Piece Nut Assembly will limit the torque applied to the outer nut by allowing the inner nut to slip at a predetermined limiting force on the balls. The inner and outer nuts will turn together to lift the load and tension the belt until the force on the balls exerted from the inner nut resisting to turn causes the balls to ride over the top of the inner nut. When this occurs the assembly will have limited the tension of the belt to the manufacturer specified values and a standard hex jam nut will be used to lock the assembly in place.

A simple power screw formula can be used to calculate the torque required to lift the load. The total load (P) for this calculation is the summation of all the moving components added to the belt tension force as specified from Goodyear. The load P is calculated given the following:

\[ P = W_M + W_P + W_{MP} + W_B + W_R + F_B \]

where:

- \( W_M \) = Weight of the motor
- \( W_P \) = Weight of the motor sprocket and bushing
- \( W_{MP} \) = Weight of the motor plate
- \( W_B \) = Weight of the threaded rod mounting block
- \( W_R \) = Weight of the adjusting rod
- \( F_B \) = Belt tension force required

The force acting on the system due to belt tension \( F_B \) is twice the vertical component of \( T \) or \( T_Y \) as shown in Figure 15. The tight side tension \( T_1 \) and slack side tension \( T_2 \) were used to calculate \( T \).

Figure 15 – Belt Layout
The installation tension $T$ is also dependent on the speed and horsepower of the drive. The effective belt tension and centrifugal belt tension were calculated as follows:

$$S = \text{Belt speed (ft/min)}$$
$$\text{RPM} = \text{Moor speed under load (rpm)} = 3450 \text{ rpm}$$
$$\text{Pd} = \text{Motor sprocket pitch diameter (in)} = 7.519 \text{ in}$$

$$S = \frac{\text{Pd}(\pi)(\text{RPM})}{12} = \frac{7.519 \text{ in}(\pi)(3450 \text{ rpm})}{12} = 6791 \text{ ft/min}$$

$$T_E = \text{Effective belt tension (lbs)}$$
$$\text{HP} = \text{Motor power (HP)} = 3.0 \text{ HP}$$

$$T_E = \frac{\text{HP}(33,000)}{S} = \frac{3.0 \text{ HP}(33,000)}{6791 \text{ ft/min}} = 14.58 \text{ lbs}$$

$$T_C = \text{Centrifugal tension (lbs)}$$
$$W = \text{Belt weight per foot (lbs) from Goodyear} = .092 \text{ lbs}$$

$$T_C = \frac{W(S)^2}{32.2} = \frac{(0.092 \text{ lbs})(6791 \text{ ft/min})^2}{32.2} = 36.60 \text{ lbs}$$

$$T_1 = \text{Tight side tension (lbs)}$$
$$\text{AR} = \text{Ratio factor from Goodyear} = 1.09$$

$$T_1 = (T_E)(\text{AR}) + T_C = (14.58 \text{ lbs})(1.09) + 36.60 \text{ lbs} = 52.5 \text{ lbs}$$

$$T_2 = \text{Slack side tension (lbs)}$$

$$T_2 = (T_E)(\text{AR}-1) + T_C = (14.58 \text{ lbs})(1.09-1) + 36.60 \text{ lbs} = 37.9 \text{ lbs}$$

$$T = \text{Installation tension for new belt (lbs)}$$

$$T = (0.7)(T_1 + T_2) = (0.7)(52.5 \text{ lbs} + 37.9 \text{ lbs}) = 63.3 \text{ lbs}$$

The vertical component $T_Y$ can then be calculated by resolving $T$ acting along the line of the belt into its components. The angle of wrap $\beta$ is solved for graphically:

$$T_Y = \text{Vertical component of T (lbs)}$$
$$\beta = \text{Angle of belt wrap (°)} = 9.6°$$

$$T_Y = (T)(\cos \beta) = (63.3 \text{ lbs})(\cos 9.6°) = 62.4 \text{ lbs}$$

$F_B$ thus becomes twice the vertical component of the installation tension:

$$F_B = (2)(T_Y) = (2)(62.4 \text{ lbs}) = 124.8 \text{ lbs}$$

$W_M = \text{Weight of the motor} = 45.0 \text{ lbs}$
$W_p = \text{Weight of the motor sprocket and bushing} = 18.0 \text{ lbs}$
$W_{MP} = \text{Weight of the motor plate} = 15.2 \text{ lbs}$. 
WB = Weight of the threaded rod mounting block = 0.6 lbs
WR = Weight of the adjusting rod = 0.3 lbs
FB = Belt tension force = 124.2 lbs

Solving for the total load to lift:

\[ P = W_M + W_P + W_{MP} + W_B + W_R + F_B \]

\[ P = 45.0 \text{ lbs } + 18.0 \text{ lbs } + 15.2 \text{ lbs } + 0.6 \text{ lbs } + 0.3 \text{ lbs } + 124.8 \text{ lbs } = 203.9 \text{ lbs} \]

The adjusting rod used was commercially available 1/2-13 UNC threaded rod. For a v-thread, the normal thread load is inclined to the axis by an amount \( \alpha \), which is equal to half of the included thread angle. The effect of this is to increase the frictional force by the wedging action of the threads. The frictional terms therefore must be divided by \( \cos \alpha \). The coefficient of friction is published for screw-thread friction applications such as this to be .15 (9). The following equation was used to calculate the torque required to lift the load (\( T_{LIFT} \)) in in*lbs:

\[
T_{LIFT} = \frac{\left( \frac{P_d_m}{2} \right)}{\left[ \frac{L + (\pi \mu d_m \sec \alpha)}{\pi d_m} - (\mu L \sec \alpha) \right]}
\]

The screw thread data are published values for standard coarse-thread unified screw threads (10).

\[ \alpha = \frac{1}{2} \text{ the included angle (degrees) } = 30^\circ \]
\[ \text{Pitch} = \text{Pitch of screw} = 13 \text{ (threads per inch)} \]
\[ L = \text{Lead of screw (in)} = 1/\text{Pitch} = 1/13 \text{ t.p.i } = 0.077 \text{ in} \]
\[ d_m = \text{Mean diameter of threaded rod (in)} = 0.450 \text{ in} \]
\[ \mu = \text{Coefficient of friction} = 0.15 \]

\[
T_{LIFT} = \frac{(203.9 \text{ lbs})(0.450 \text{ in})}{2} \left[ \frac{0.077 \text{ in} + (\pi)(0.15)(0.450 \text{ in})(\sec 30)}{(\pi)(0.450 \text{ in}) - (0.15)(0.077 \text{ in})(\sec 30)} \right]
\]

\[ T_{LIFT} = 10.5 \text{ in}*\text{lbs} \]

**CALCULATING THE FORCE ON THE BALL**

To translate the torque to lift and tension into an equivalent force acting in the radial direction of the screw axis, it is necessary to divide that value by the radius at which the force is applied. Figure 16 shows how the radial distance of the force is determined graphically.
The ball is shown at the point just before the tension reaches the limiting value. The inner nut will stop rotating and allow the balls to push up into the holes that house the springs. The force $F_n$ shown is the greatest value for which the ball remains on the ramp. Force $F_n$ passes through the center of the ball and acts perpendicular to the ramp at this instant in time. Dividing the torque previously calculated by the radius at which force $F_n$ acts yields the total force required to set the set the springs. $F_n$ is calculated given:

$$T_{\text{LIFT}} = \text{Torque to lift (in*lbs)} = 10.5 \text{ in*lbs}$$
$$r_t = \text{Radius at which } F_n \text{ acts (in)} = 0.380 \text{ in}$$
$$n = \text{Number of balls/springs} = 3$$

$$F_n = \frac{T_{\text{LIFT}}}{n \cdot r_t} = \frac{10.5 \text{ in*lbs}}{(3)(0.380 \text{ in})} = 9.2 \text{ lbs}$$

The vertical component of this force $F_s$ is the spring force required. It is calculated using the cosine function of the angle $\theta$ which is determined graphically.

$$F_s = \text{Spring force required per spring (lbs)}$$
$$\theta = 41.0^\circ$$

$$F_s = (\cos\theta)(F_n) = (\cos41^\circ)(9.2 \text{ lbs}) = 7.0 \text{ lbs}$$

By measuring the position of the ball in the extreme positions the total deflection $d$ can be determined. The radial position to the center of the ball when seated in the curved hook of the inner nut is .438”. When the assembly passes the limiting torque to release the balls they increase to a radial position of .625”. The target spring rate $K_t$ thus becomes the spring force $F_s$ divided by the positional difference of the ball.
Fs = Spring Force (lbf) = 7.0 lbs
p1 = position 1 (in) = 0.437 in
p2 = position 2 (in) = 0.625 in
Δp = change in position (in) = p2 – p1 (in) = 0.188 in

\[ K_t = \frac{F_s}{\Delta p} = \frac{7.0 \text{ lbs}}{0.188 \text{ in}} = 37.2 \frac{\text{lbs}}{\text{in}} \]
APPENDIX G: DRAWINGS
ASSEMBLY DRAWINGS

SECTION B-B

BOTTOM VIEW

TOP VIEW

SECTION

NOTE:
- IMPERIAL DRAWING
- UNLESS OTHERWISE SPECIFIED
- ALL DIMENSIONS ARE IN INCHES

APPENDIX G: DRAWINGS
ASSEMBLY DRAWINGS

SECTION B-B

BOTTOM VIEW

TOP VIEW

SECTION

NOTE:
- IMPERIAL DRAWING
- UNLESS OTHERWISE SPECIFIED
- ALL DIMENSIONS ARE IN INCHES
THE OUTER NUT DETAIL

- 3 flats
- Diameter: 3.00
- Flats Width: .65
- Flats Length: 30°
- Hole Diameter: .354
- Hole Diameter: .357
- Hole Depth: .046
- Hole Depth: .049
- Bolt Diameter: .734
- Taper Diameter: .61
- Radius: .01
- Nut Width: 1.219
- Nut Height: 1.222
- Nut Diameter: .001

SECTION A-A

UNLESS OTHERWISE SPECIFIED:
- METRIC: Mm, mm
- IMPERIAL: Inch, in

G-3
APPENDIX H: PURCHASED COMPONENTS

COMPRESSSION SPRING \#LC 035C 06 (LEE SPRING CO.)

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<th>LEE STOCK NUMBER</th>
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<th>WIRE DIAMETER</th>
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<th>FREE LENGTH</th>
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PRECISION CHROME STEEL BALL, ¼” DIA. (THOMSON)

Material: 52100 chrome steel
Outside Diameter: .250 inch
**SNAP RING #N5000-102 (WALDES TRU-ARC)**

![Diagram of SNAP RING #N5000-102 (WALDES TRU-ARC)](image)

Sizes -25 thru -250 are available in tape-wrapped Rol-Pak® cartridges for an extra charge.

<table>
<thead>
<tr>
<th>HOUSING DIA.</th>
<th>TRUARC RING DIMENSIONS</th>
<th>GROOVE DIMENSIONS</th>
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<td><strong>S</strong></td>
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NEEDLE THRUST BEARINGS AND THRUST RACES

Actual part # used: #NTA-916

Actual part # used: #TRB-916
LUBRICATION

PREMIUM WHEEL BEARING 707L RED GREASE

Multi-Purpose, High Temperature Wheel Bearing Grease

PRODUCT DESCRIPTION

PENNZOIL® PREMIUM WHEEL BEARING 707L RED GREASE is a high temperature, NLGI Grade 2, lithium complex grease which contains high quality petroleum base oils. The primary application for 707L is in high temperature operations such as wheel bearings on vehicles equipped with conventional, disc and antilock braking systems. These high temperature applications demand the use of 707L to resist melting of the grease and the resulting leakage from the bearings. 707L may be used with confidence at continuous operating temperatures up to 325°F and in intermittent use at temperatures up to 450°F. Special additives in 707L prevent rust and corrosion, reduce wear, and provide extreme pressure protection. 707L has excellent resistance to softening or hardening in service, resists water washout, and flows readily at cold temperatures. 707L's special thickeners resist wheel bearing leakage. Precision manufacturing reduces oil bleeding in long term storage and produces a consistent color and texture. Its extreme pressure properties insure its performance under heavy or shock loading. PENNZOIL® PREMIUM WHEEL BEARING 707L RED GREASE exceeds the requirements of the NLGI GC-LB specification, which is the highest performance rating available. This means that it is satisfactory for use as both a severe duty chassis and wheel bearing lubricant, resisting oxidation, evaporation and consistency degradation while protecting the components from corrosion and wear.
## APPENDIX I: BILL OF MATERIALS

SETCO Type: TENSIONER.73849  
SETCO Assembly Drawing: S331930

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<td>* TENSIONING DEVICE USED TO LIMIT THE BELT TENSION</td>
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<td>TO A RECOMMENDED VALUE</td>
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