Real Mech: Precision Tennis Ball Launcher
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Abstract:

The purpose of this document is to provide the design procedure for a ball launcher intended for use by children at the COSI museum. The idea behind the ball launcher is to provide a fun and interactive way for younger members of our society to learn about physical and mechanical properties, as well as projectile motion. The launcher consists of four major machine elements that the team designed for: gears, bearing, spring, and wire. The machine was designed to be in an exhibit of 10’ by 10’ by 10’ in size. Further, the launcher was designed to sustain a minimum of 2448 hours of usage without maintenance. The team also considered other factors for design: Intuitive, easy to operate, and strong and robust to accommodate less than gentle users. These objectives were achieved by using the machine element design principles outlined throughout the Machine Elements II course and the engineering methodology from Collins, 2009.

Acknowledgments:

We would like to acknowledge Professor Alok Sutradhar for his support. He helped us to think the project more creatively by suggesting which things we should consider or apply for the mechanical structure, and his advice made us participate in project more eagerly.

Also, we would like to acknowledge Jaejong Park, the teaching assistance of mechanical element II, since he suggested many ways to develop our research and analysis.

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Introduction:

The purpose of this design is to incorporate at least three distinct machine elements into an educational and intuitive mechanical demonstration. The goal of the design is to teach how changing the angle of a launched projectile impacts the distance it travels. This is accomplished by using a system of goals and targets along with a variable pitch and azimuth ball launcher. Users will be able to see that using the same launch velocity (i.e. the same spring and spring compression), the ball can be made to hit nearly any target within a specified radius if the angle of launch is changed. This is how an important concept can be made fun and intuitive.

Specifications:

There are several specifications of the design both as an exhibit size limitation, and desire performance parameters. These specifications are as follows:

- Size for the design must be limited to a 10'x10'x10' space. This includes the distance that may be covered from flying projectiles. If necessary, a net will be placed around the exhibit to prevent stray projectiles from leaving this area.
- Spring must be capable of propelling the ball (and platform it rests on) to the desired launch velocity
- Spring must also be an appropriate stiffness that a child can compress it to the required displacement to ensure proper launch velocity
- Design must function for 51 weeks, 6 days a week, and 8 hours a day. All components of the design must meet the fatigue requirements for this life span
- Design must also be able to withstand unforeseen stresses (i.e. a child treating the apparatus less gently than intended)

- The whole design should be relatively light and easy to set up.

Design Procedure

Initial Design Considerations:

Several initial designs of the ball launcher were considered. The first design included a crank attached to a gear train to pull down the spring along with a variable angle tube for launching. A second design was similar to this, but involved two springs with different spring constants and tubes as to compare the projectile motion from different launch angles and forces imparted on them. The final design considered was a single spring and tube with a variable launch angle, a rotating platform, and lever to pull back the spring. This third design was chosen over the others for ease of design and use, as well as more freedom when aiming the launcher. A sketch of this design is on the next page in Figure 1.

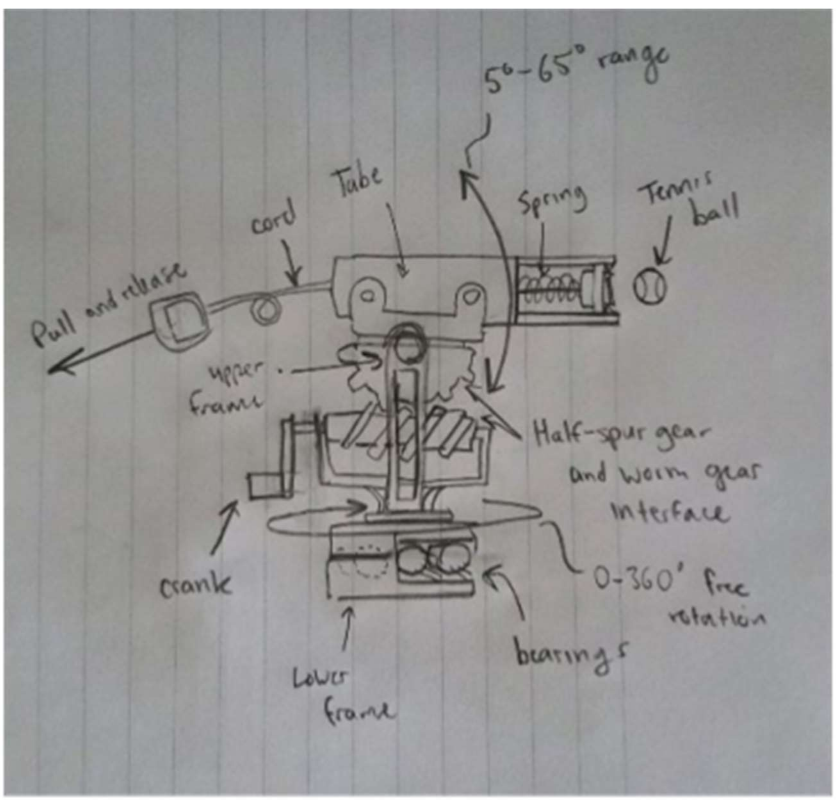


Figure 1: Sketch of the chosen ball launcher design

Assumptions:

In order to start creating the machine to launch a ball, some initial assumptions needed to be made. First, it was assumed that the machine will fail at the machine components analyzed. Second, the exhibit would be encased in a net with the ball launcher in one of the corners of the 10’ by 10’ by 10’ space. Movable targets within the area would be place such that the ball would be able to land inside of them. Third, all machine components are securely fastened together with welds, bearings or bolts where needed and these components do not fail.

Spring:

In order to select a spring to use for the launcher, the projectile motion of the ball needed to be analyzed at the angle of launch for max distance (45°). Projectile motion for an initial launch height of one meter and max distance of three meters was determined.

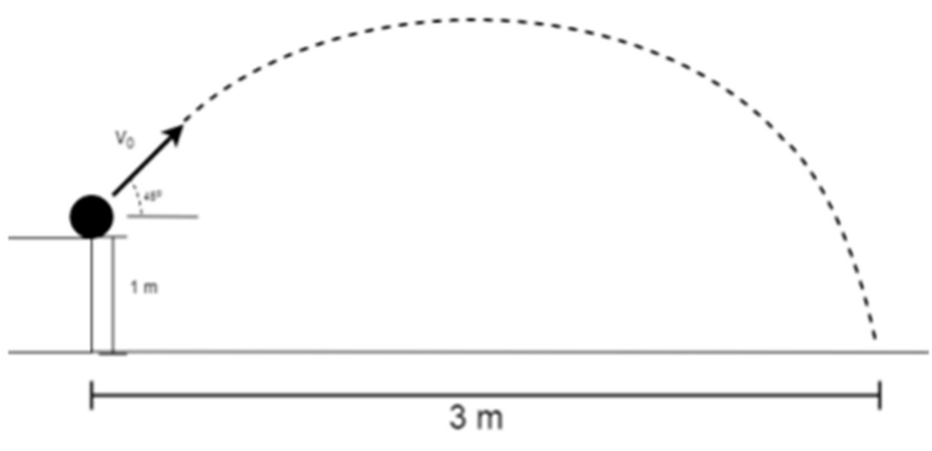


Figure 2: Desired max projectile motion of the baseball

The equations to solve for initial velocity are below.

$$x = V_0 sin\,(45)\,t \tag{1}$$

$$y = V_0 sin(45)t + \frac{1}{2}gt^2 \quad (2)$$

Where *x* and *y* are horizontal and vertical distance in meters, V_0 is initial velocity in meters per second, *t* is time in seconds, and *g* is acceleration due to gravity in meters per second squared. Solving the equations yielded an initial velocity of 4.7 m/s to achieve the desired motion.

Next, several springs from McMaster Carr were selected based on their rate. Springs between 5 and 8 inches long, with outer diameters smaller than that of a baseball and fairly small rates were selected. In order to calculate the distance and force needed to propel the ball at 4.7 m/s, the energy equation and Hooke's law for springs were used.

$$\Sigma E_i = \Sigma E_f \rightarrow \frac{1}{2}kx^2 = \frac{1}{2}mv^2 \quad (3)$$

$$F = kx \quad (4)$$

Where E_i and E_f are the initial and final energies of the spring, *k* is spring rate, *x* is the distance the spring is deflected, *m* is the mass of the baseball and ball cup, *v* is desired velocity, and *F* is the force required to compress the spring. The final potential energy of the ball is neglected as it adds a negligible energy value. Equation # was first evaluated for *x*, then equation # for *F*. A summary of the results can be seen below in the table.

Table 1: Summary of selected springs with deflection and force required for each

McMaster Carr Spring Number	Spring Rate (lb/in)	Energy Required (J)	Spring Deflection (in)	Force Needed (lb)
9657K469	22.25	22.38	2.98	66.38
96485K135	38.2	22.38	2.28	86.98
96485K156	32.3	22.38	2.48	79.98

While each of these forces is likely too high for a child to pull back by themselves, a lever would be used to amplify the pullback force. Each of these springs were then checked for failure. Due to the need to be operated for a total of 2440 hours per year, it was conservatively estimated that the spring would be deflected 10^6 times per year, and fatigue strength would be the limiting failure mechanism. The equations for max compression spring shear stress can be seen below.

$$\tau_w = K_w \frac{8FD}{\pi d^3} \tag{5}$$

$$K_w = \frac{4c-1}{4c-4} + \frac{0.615}{c} \tag{6}$$

$$c = \frac{D}{d} \tag{7}$$

Where τ_w is max shear stress, K_w is the Wahl factor, and c is the spring index. The fatigue failure limits were then calculated using the equations below.

$$\tau_f = 0.4 S_{ut} \tag{8}$$

$$S_{ut} = Bd^a \tag{9}$$

Where τ_f is the shear stress fatigue limit, S_{ut} is the ultimate tensile strength, and B and a are empirically found coefficients for the spring materials from Table 14.1 in the textbook [1]. The factor of safety for each of the springs was then found using the equation below.

$$n = \frac{\tau_f}{\tau_w} \qquad (10)$$

Table 2 below summarizes the results of the fatigue analysis.

Table 2: Summary of selected springs for analysis

McMaster Carr Spring Number	Max Shear Stress (ksi)	Fatigue Shear Stress Limit (ksi)	Factor of Safety
9657K469	110.53	103.52	0.94
96485K135	63.40	79.46	1.25
96485K156	52.39	78.37	1.50

The analysis assumed each spring was unpeened and has a much higher life cycle than needed giving a conservative factor of safety. From each of these springs, spring 94685K156 was chosen as it gives the largest factor of safety for this application, and is greater than the design requirement of 1.25. Buckling and static yielding were not of concern as the spring would be in a guiding tube and the max load for this spring is 176 lbs. - well over the pullback force required for launch.

Gears:

For this machine, a gear and a worm gear were used to adjust the angle of launch of the launcher. A crank is rotating the worm gear which then moves the spur gear. The gears are assumed to rotate at slow velocity, for calculation purpose it was assumed to be 60 rpm. The gears dimensions were calculated based on the change in the angle of launch per rotation of the worm gear. The specifications can be found in table 3:

Table 3: Gears characteristics

Characteristic	Worm gear	Spur gear
Pitch diameter (in)	1.25	5.51
Diametral pitch	13.1	13.1
Number of teeth	2	72
Face width (in)	-	1.0
Pressure angle (°)	20	20

The force analysis for the gears was done using the AGMA Refined Approach. The gears were checked for surface fatigue and bending. The tangential force was found using Eq. (11), then the radial force was found using the tangential force and the pressure angle in Eq. (12). There was also an axial force found for the worm gear using Eq. (13). The forces also had to be converted between the worm gear and the spur gear using Eq.(14), Eq.(15), Eq. (16)). Results are shown in table 4

$$F_t = \frac{T}{r} \tag{11}$$

$$F_r = F_t * tan\varphi \tag{12}$$

$$F_{wa} = F_{wt} * \frac{cos\varphi cos\lambda - \mu sin\lambda}{cos\varphi sin\lambda + \mu sin\lambda} \tag{13}$$

$$F_{gt} = F_{wa} \tag{14}$$

$$F_{ga} = F_{wt} \tag{15}$$

$$F_{gr} = F_{wr} \tag{16}$$

Table 4: Forces acting on gears

Type of force	Worm gear	Spur gear
Tangential force (lb)	20	49.80
Radial force (lb.)	18.88	18.88
Axial force (lb.)	49.80	20

For the stress analysis, it was achieved with the AGMA approach. All the values were found from the textbook [1]. For the bending and surface fatigue stress, the elastic modulus of steel was used, the ultimate bending stress was taken for Grade 1 steel, a factor of application for light shock, a mounting factor for less rigid mounting, the idler factor for one-way bending, the geometry factor for a 72 teeth gear, a geometry factor from our gear ratio, and the life span of the gear used in the calculation was of 10^7 cycles at a 99% reliability. The values are presented in table 5:

Table 5: Stress Analysis Factors

K_a	1.5
K_m	1.6
K_I	1.0
Geometry factor, J	0.48
Geometry factor, I	0.1061
Reliability factor, Rg	1
Y_N	1
Z_N	1
S_{bf}	45 Ksi
S_{tbf}	170 Ksi

The maximum bending stress was found using Eq. (17). The Cp factor for surface bending was found using Eq. (18), the surface fatigue was then found using Eq. (19). Then the factors of safety for each failure type were calculated using Eq. (20).

$$\sigma_b = \frac{F_t * P_d}{b * J} * K_a * K_m * K_I \tag{17}$$

$$Cp = \sqrt{\frac{1}{\pi(\frac{1-v^2}{E} + \frac{1-v^2}{E})}} \tag{18}$$

$$\sigma_{sf} = C_p \sqrt{\frac{F_t}{b * d * I} * K_a * K_m * K_l} \tag{19}$$

$$n_d = \frac{Maximum\ Allowed\ Stress}{Max\ Stress\ Occuring} \tag{20}$$

Table 6: Maximum stress and factor of safety of gears

	Worm gear	Spur Gear
Max bending stress (ksi)	1.437	3.579
Max bending stress allowed (ksi)	45	45
Bending factor of safety	20.9	8.4
Max surface bending stress (ksi)	1.561	2.464
Max surface bending stress allowed (ksi)	170	170
Surface bending factor of safety	7.3	4.6

It can be seen from the value in the table 6 that the gear design failure mode is governed by the surface bending of the spur gear, however since the factor of safety is 4.6 the design is acceptable since it is way above the minimum factor of safety required of 1.25.

The last thing to verify was to check if the worm gear was self-locking, this way once the position of the launcher was set the worm gear wouldn't rotate without an input from the crank. For this Eq. (21) had to be respected. Since the result was positive, the worm gear is self-locking.

$$-\cos(\varphi) * \sin(\lambda) + \mu * \cos(\lambda) \geq 0 \tag{21}$$

Wire:

For the wire calculation, it was assumed that the wire would bend about 100 times per day. The machine was estimated to be used 5 times per week for 2 years. This gave a total of 52,000 bends for the life of the wire:

$$\frac{100\ Bends}{1\ day} * \frac{5\ Days}{1\ week} * \frac{52\ Weeks}{1\ year} * 2\ years = 52000\ total\ bends \tag{22}$$

Also, a static factor of safety of 3 was desired and a fatigue factor of safety of 1.5 was desired. The team decided on a 6X19 wire. A summary of the assumptions is in the table below:

Table 7: Wire Assumptions

Assumptions:	
Tension Force T (N):	387
Bends/day:	100
Years of use:	2
Days/week:	5
n_static:	3
n_fatigue:	1.5
Su (MPa):	1379
Wire type:	6 x 19

The load on the wire would be the pull of the spring which is about 87 lbs or 387 N. The equation for 6X19 area is used for the calculation of the diameter.

$$A = 0.404 * d_r^2 \tag{23}$$

The design stress was calculated using a factor of static safety of 3.

$$\sigma_d = S_u / n_{static} \tag{24}$$

The static diameter required was calculated using this value and setting it equal to tension stress and solving for d_r.

$$\sigma_t = \frac{T}{A} \tag{25}$$

This gives a static diameter of 1.444 mm. Fatigue was calculated as follows. Using figure 17.17 and the total bends an (Rn)f of 0.006 was determined (Collins, 2009). The (p)Nf was calculated by multiplying the (Rn)f by the ultimate tensile strength.

$$(p)_{Nf} = (R_n)_f * S_u \tag{26}$$

This was then divided by the fatigue safety factor of 1.5 to get the (pd)fatigue. The fatigue diameter was calculated to be 2.032 mm.

$$(d_r)_{fatigue} = \frac{2*T}{d_s(p_d)_{fatigue}} \text{ and } d_s = 34 * d_r \tag{27}$$

Lastly to calculate the wear, table 17-10 was used to get $p_{max} = 6.205\ MPa$ for cast carbon steel (Collins, 2009). This was used along with the equation below to get the wear diameter of 1.915 mm.

$$(d_r)_{wear} = \frac{2*T}{d_s(p_d)_{wear}} \tag{28}$$

To conclude the wire failure is controlled by fatigue as it requires the largest diameter of 2.032 mm. A summary of results can be seen in the table below.

Table 8: Wire Calculation Summary

Calculations:	
Static:	
σd (MPa):	459.667
(dr)static (mm):	1.444
Fatigue:	
Total Bends:	52000
(Rn)f (from figure 17.17):	0.006
(p)Nf:	8.274
(pd)fatigue (MPa):	5.516
(dr)fatigue (mm) (17-33):	2.032
Wear:	
(pd)wear (MPa) (carbon steel):	6.205
(dr)wear (mm):	1.915
Failure is controlled by Fatigue:	2.032

The wire that was chosen from McMaster Carr is a 6X19 steel wire with a diameter of 3/16" diameter(~4.76mm). The specifications of the wire can be seen in the appendix below. This wire has a greater diameter that is required, however, it is the smallest available. This will be more than adequate for the launcher operation and safety of the user.

Bearing

To make a rotational movement of structure, the team designed a bearing should be installed between two bottom plates. Since the bottom plate is placed on the ground, it was assumed that a user can step on in or exert force as much as their weight at the angle of 45˚. The average weight of ten-year-old kids which is 72lb was used with 10kg of structure weight, to calculate overall axial and radial load.

The axial and radial loads are

$$F_x = mgsin\theta$$
$$= 32.66kg * \frac{9.81m}{s^2} * \sin(45°) \tag{29}$$
$$= 226.55N$$

$$F_y = m_{user}gcos\theta + m_{mach}g$$
$$= 32.66kg * \frac{9.81m}{s^2} * \cos(45°) + 10kg * \frac{9.81m}{s^2} \tag{30}$$
$$= 324.65N$$

The designed life of gear was estimated from design criteria and the team assumed each people make ten lateral rotations with three minutes of use.

$$L_d = \frac{200rev}{hr}\frac{8hr}{day}\frac{6day}{week}\ 51week \tag{31}$$
$$= 489600\ rev$$

The reliable factor and impact factor to calculate the basic dynamic load rate were assumed 99% of reliability and heavy for each to accommodate safety of its user.

$$K_r(99\%) = 0.21$$
$$IF = 3.0$$

For the type of bearing, since the machine is expected to receive both of axial and radial loads, angular ball contact ball bearing was chosen.

To calculate a dynamic equivalent radial load P_e, below factors from table 11.4 (Collins) were used.

Table 9: Approximate Radial Load Factors

	X_{d1}	Y_{d1}	X_{d2}	Y_{d2}
Single-row Angular contact ball bearing (steep angle)	1	0	0.45	1.2

Therefore, the dynamic equivalent radial load is

$$P_e = X_d F_r + Y_d F_a \tag{32}$$

$$P_{e1} = X_{d1} F_r + Y_{d1} F_a$$

$$= 1 * 226.55$$

$$= 226.55N$$

$$P_{e2} = X_{d2} F_r + Y_{d2} F_a$$

$$= 0.45 * 226.55 + 324.65 * 1.2$$

$$= 491.53\text{N}$$

Since P_{e2} has bigger value than P_{e1}, 491.53N was used for assuming the basic dynamic load rating and for a, value of 3 was used for calculation because of the roller bearing.

$$[C_d(R)] = \left[\frac{L_d}{K_R(10^6)}\right]^{\frac{1}{a}} (\text{IF})P_e \tag{33}$$

$$= \left[\frac{489600}{0.21 * 10^6}\right]^{\frac{1}{3}} * 3 * 491.53$$

$$= 1955.3\text{N}$$

$$= 1.955\text{kN}$$

As a result, 71900 AC bearing was chosen, and its specifications are listed in below table.

Table 10: Specification of 71900 AC bearing

Bearing Type	Bore Dia. (d, mm)	Outer Dia. (D, mm)	Width (B, mm)	Dynamic Load Rating (Cd, N)	Static Load Rating (Cs, N)
25 deg contact angle	10.0	22.0	6.0	2700	1700

Since Dynamic Load Rating of the bearing is 2.7 kN, it satisfies the design criteria.

Summary and Recommendations:

To summarize this project, the launcher is made with a spring 94685K156 from McMaster, a 6X19 carbon steel wire, a set of custom made spur and worm gears and a 71900 AC bearing from McMaster. The frame and support elements along with the gear sets are going to be made by Columbus Machine Works at a cost of $60/hr., manufacturing all those pieces should be about 10 hours so $600 plus the $117.04 from McMaster for the standard pieces. Therefore, the total price is $717.04 which is not too expensive for an exhibit that is supposed to last multiple years.

Table 11: Standard Item

Item	**Price**	**Quantity**	**Total Price**
71900 AC Bearing	$10.05	1	$10.05
96485K156 (McMaster Carr)	$11.62	1	$11.62
3440T55 6X19 Wire (McMaster Carr)	$13.40	1	$13.40
Total Cost for Standard Items			**$35.07**

Non-Standard Item

-Spur and worm gears

-Frame

-Ball holder platform

-Launch tube - machined from 1 ft of steel rod - McMaster Carr 8920K88 - $81.97

-Targets

-Crank

Total hours required: around 10 hr.

The strength of this design would be the simplicity of it, it makes it really easy to use the machine and it would be easily fixed if anything would break due to unforeseen circumstances, there are a few things that were a bit weaker and might be designed differently if it had to be done again, such as the worm and spur gear system, it would have been easier to simply use two spur gears which would have made the design much more simple, also it would have been a good idea to include something to lock the bearing once the angle of launch was deemed acceptable so it stops moving.

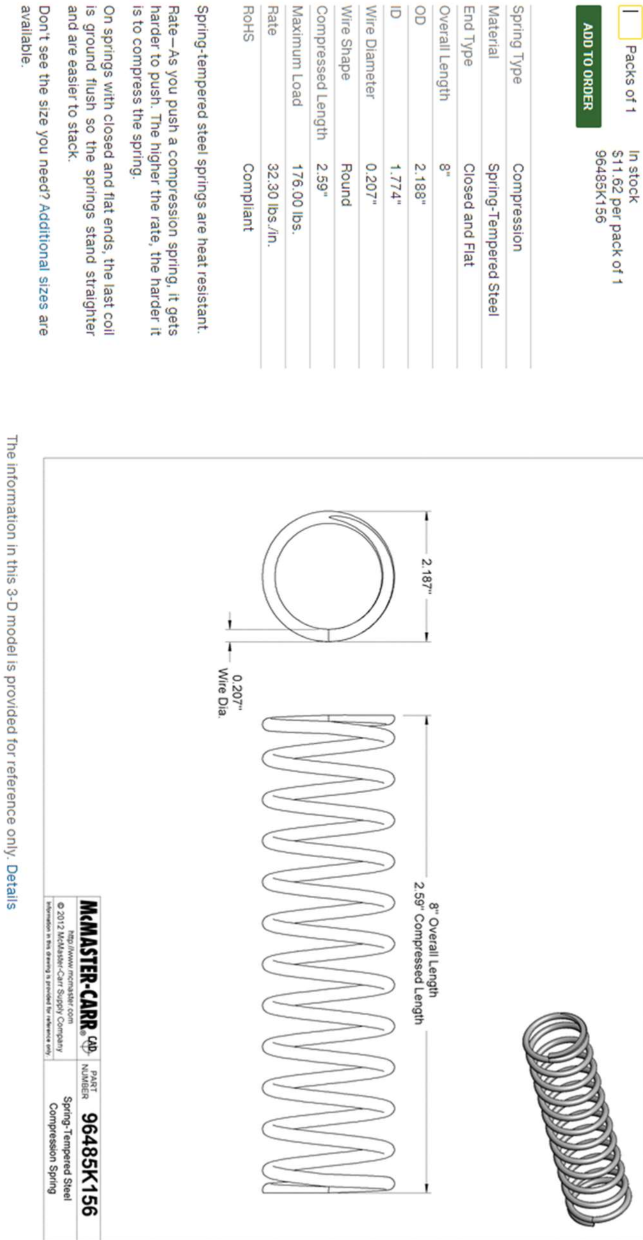
References:

[1] J. A. Collins, H. Bushby, G. Staab. “Mechanical Design of Machine Elements and Machines: A Failure Prevention Perspective, 2nd Edition”. New York, NY: Wiley, 2009.

The Average Height and Weight by Age. (2018, March 26). Retrieved April 21,2018 from https://www.livestrong.com/article/328220-the-average-height-and-weight-by-age/

Angular Contact Ball Bearing(n.d.). Retrieved April 22, 2018 from https://www.astbearings.com/catalog.html?cid=ang_cont_bearing

Appendix


Packs of 1

In stock
$11.62 per pack of 1
96485K156

ADD TO ORDER

Spring Type	Compression
Material	Spring-Tempered Steel
End Type	Closed and Flat
Overall Length	8"
OD	2.188"
ID	1.774"
Wire Diameter	0.207"
Wire Shape	Round
Compressed Length	2.59"
Maximum Load	176.00 lbs.
Rate	32.30 lbs./in.
RoHS	Compliant

Spring-tempered steel springs are heat resistant.

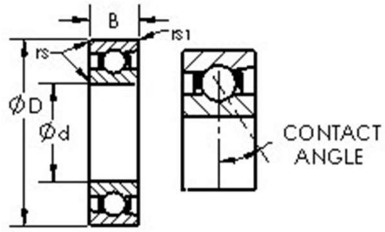
Rate—As you push a compression spring, it gets harder to push. The higher the rate, the harder it is to compress the spring.

On springs with closed and flat ends, the last coil is ground flush so the springs stand straighter and are easier to stack.

Don't see the size you need? Additional sizes are available.

The information in this 3-D model is provided for reference only. Details

Figure 3: Spring Specification Sheet



1. 25 degree contact angle, relieved outer ring
2. Also available with metal shields (ZZ) or rubber seals (2RS)
3. The desired preload must be specified.

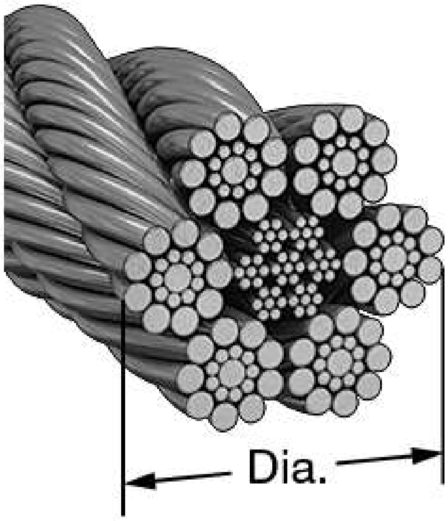
Product Details

	Specifications
Bearing Type	25 deg contact angle
Bore Dia (d)	10.000
Outer Dia (D)	22.000
Width (B)	6.000
Radius (min) (rs)	0.30
Dynamic Load Rating (Cr)	2,700
Static Load Rating (Cor)	1,700
Max Speed (Grease)	67,000
Max Speed (Oil)	100,000
Radius (min) (rs1)	0.10
Shaft (Fw)	0.10
Material	52100 Chrome steel (or equivalent)

Figure 4: Bearing Specification Sheet

Wire Rope - for Lifting

3/16" Diameter



Length, ft.
✓ 10

Each

ADD TO ORDER

In stock
$13.40 Each
3440T55

Application	For Lifting
Diameter	3/16"
Capacity	750 lbs.
Fits Pulley/Drum Diameter	6 3/4"
Material	Steel
Finish	Unfinished
Construction	6 × 19
Core Type	IWRC
Lubrication	Lubricated
Preformed	Yes
Specifications Met	ASTM A1023
Attachment Type	Plain
Length	10 ft.
RoHS	Compliant

This 6×19 IWRC wire rope has a good balance of abrasion resistance and flexibility. It is preformed to prevent it from unraveling when cut, and it's lubricated to reduce wear.

Note: The safety factor for capacity is 5:1. Warning: Never use to lift people or items over people.

Figure 5: Wire Specifications

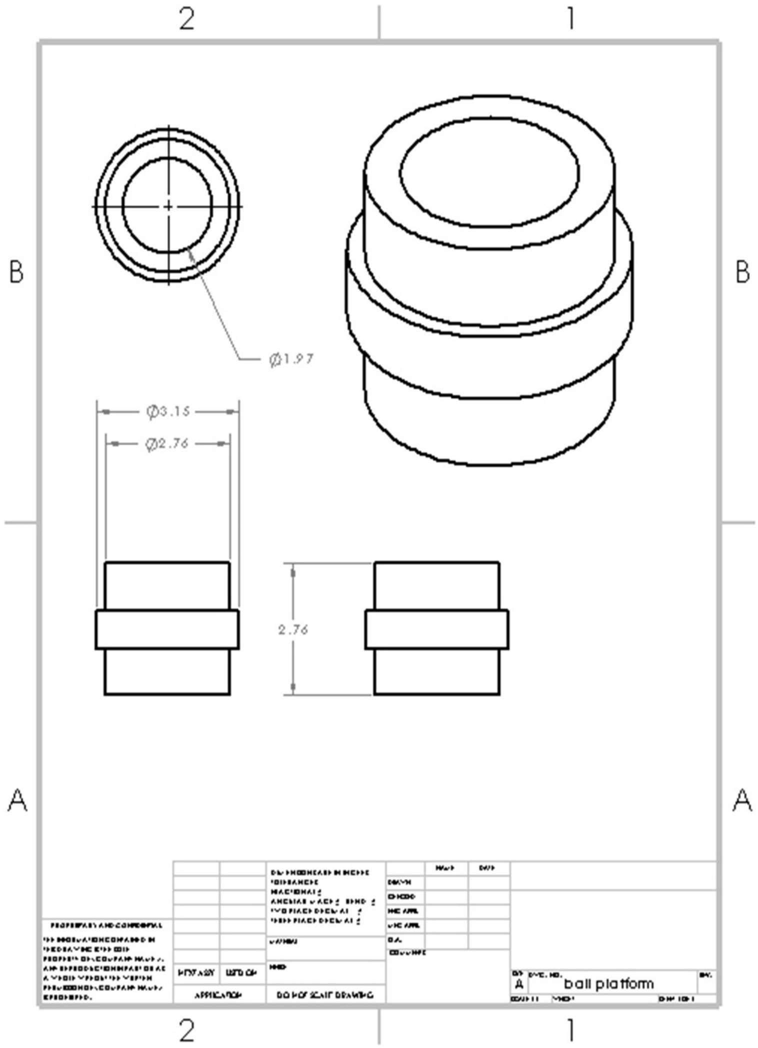


Figure 6: Drawing of the Ball Platform

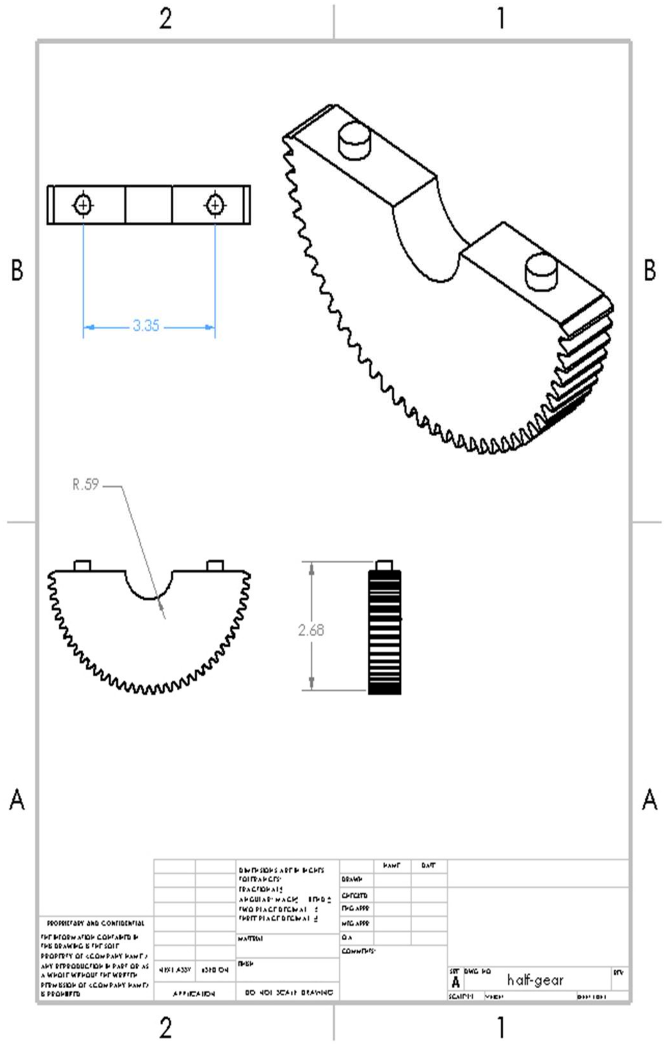


Figure 7: Drawing of the Half-gear

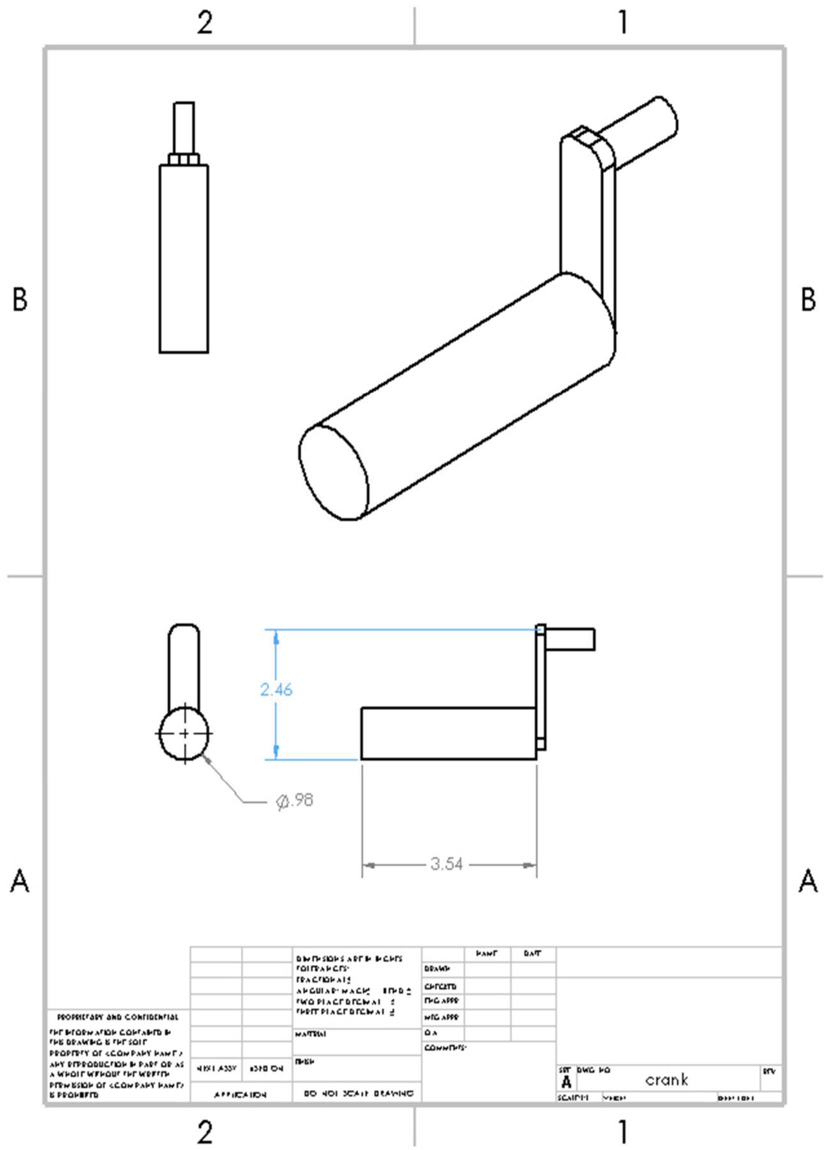


Figure 8: Drawing of the Crank

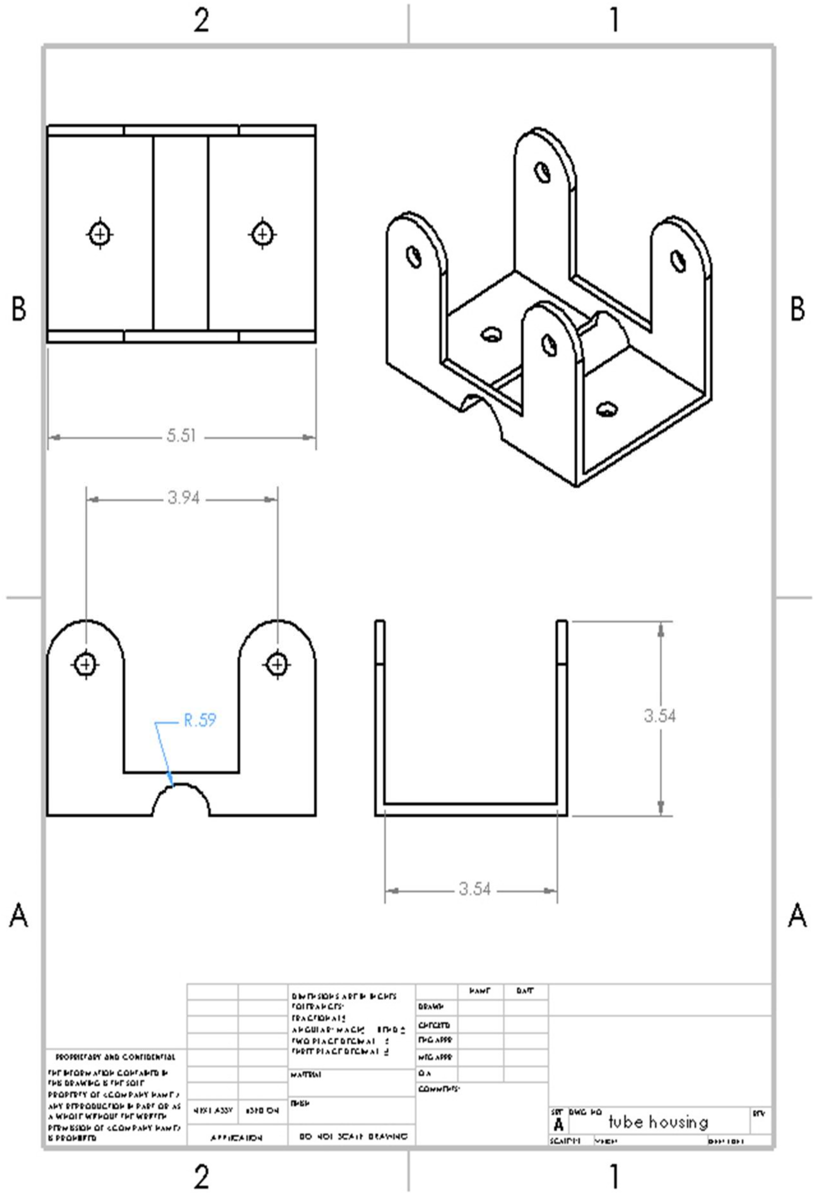


Figure 9: Drawing of the Tube Housing

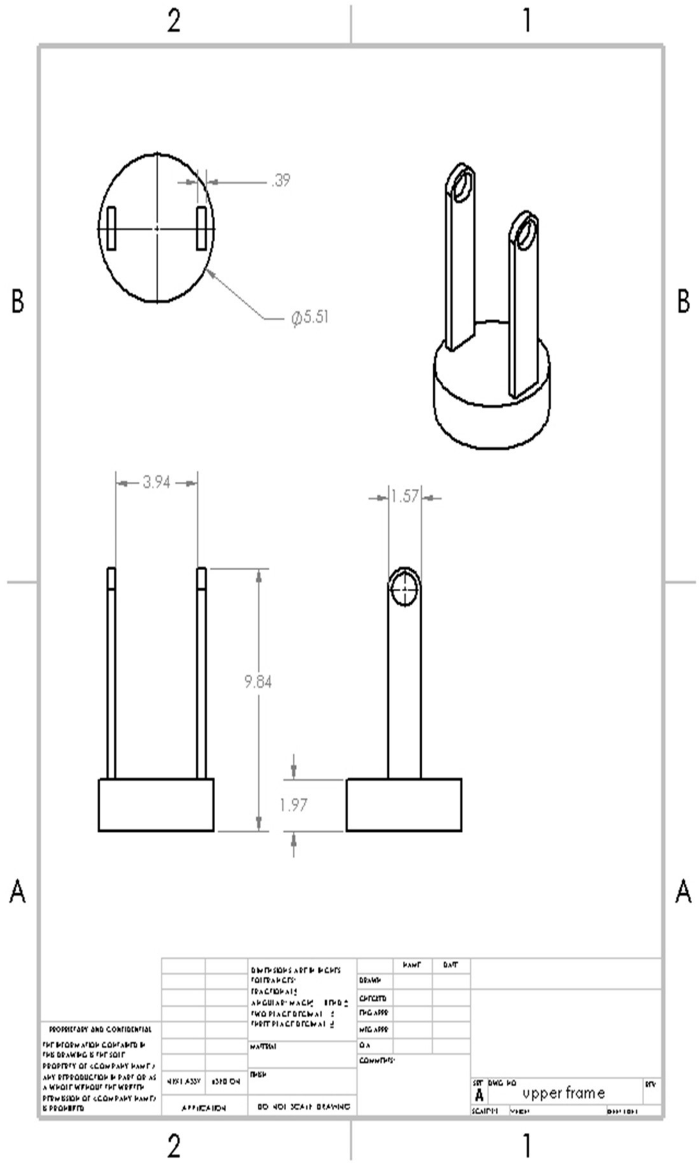


Figure 10: Drawing of the Upper Frame

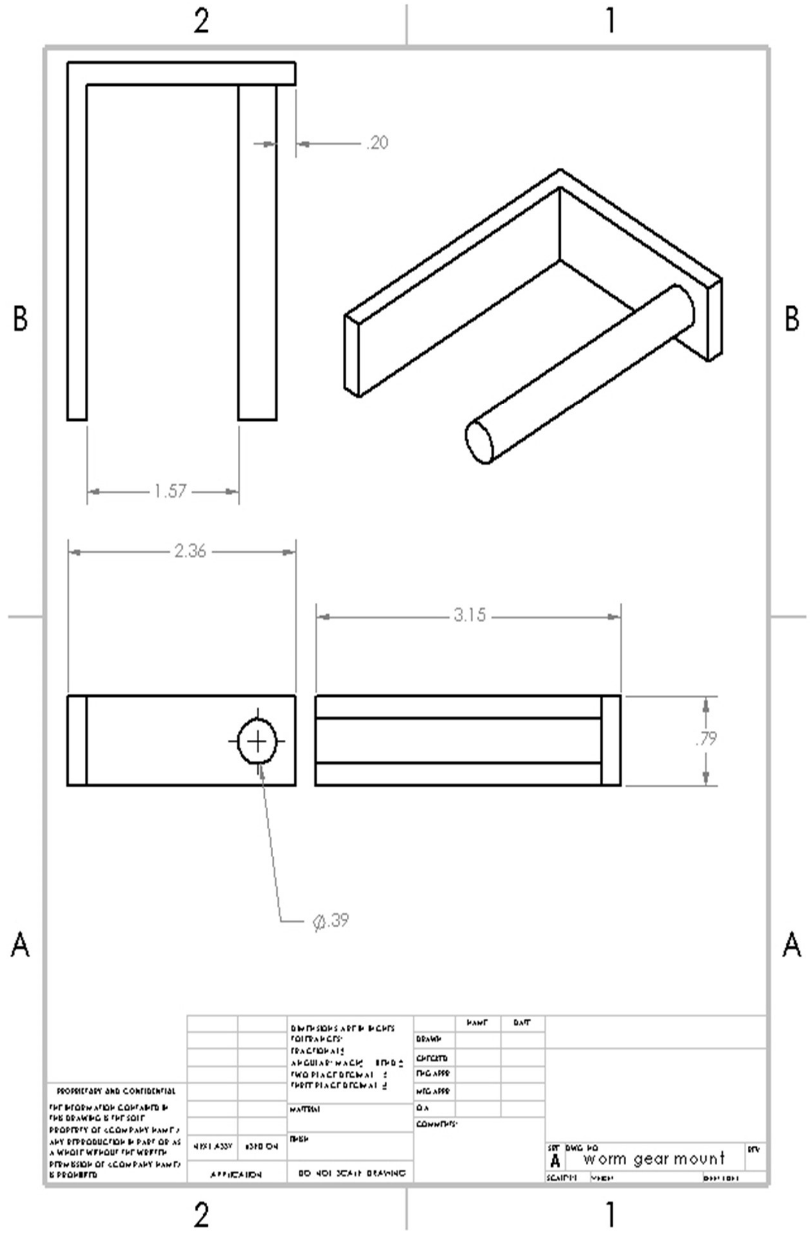


Figure 11: Drawing of the Worm Gear Mount

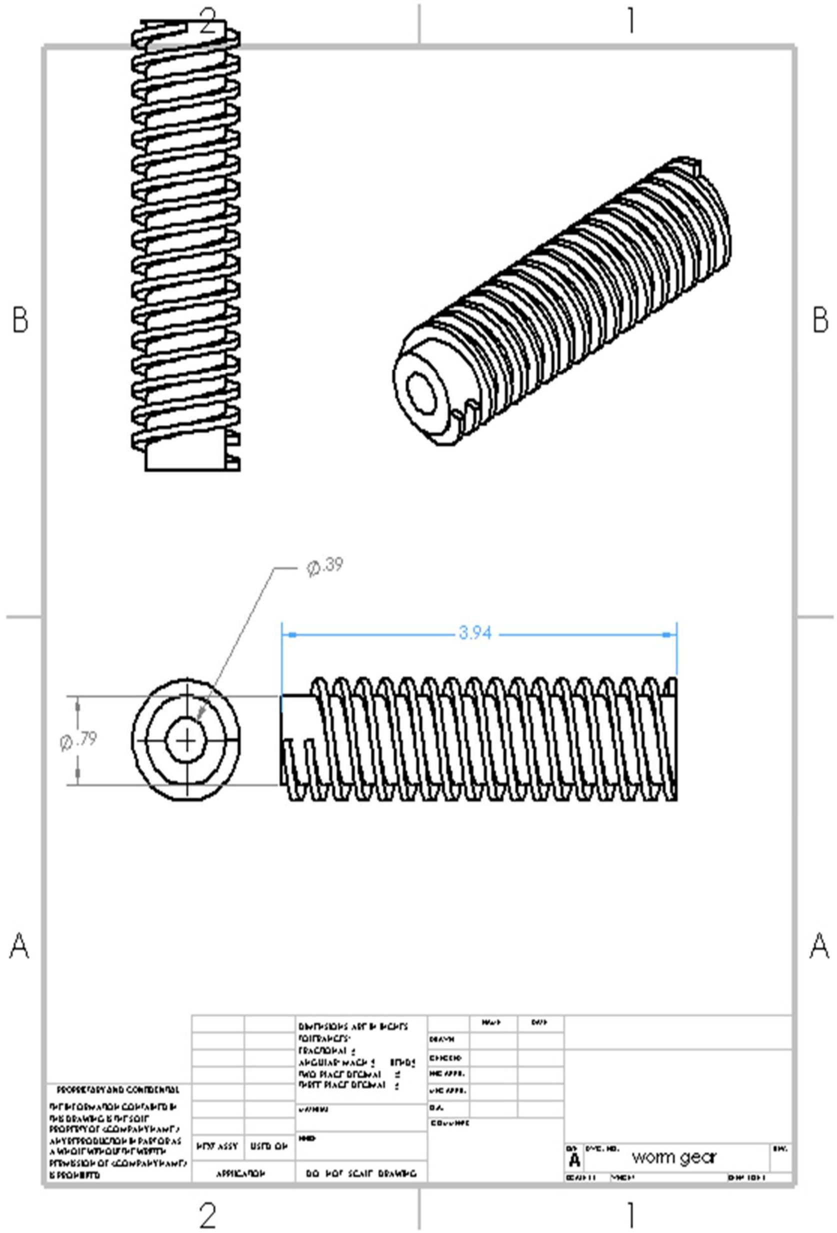


Figure 12: Drawing of the Worm Gear

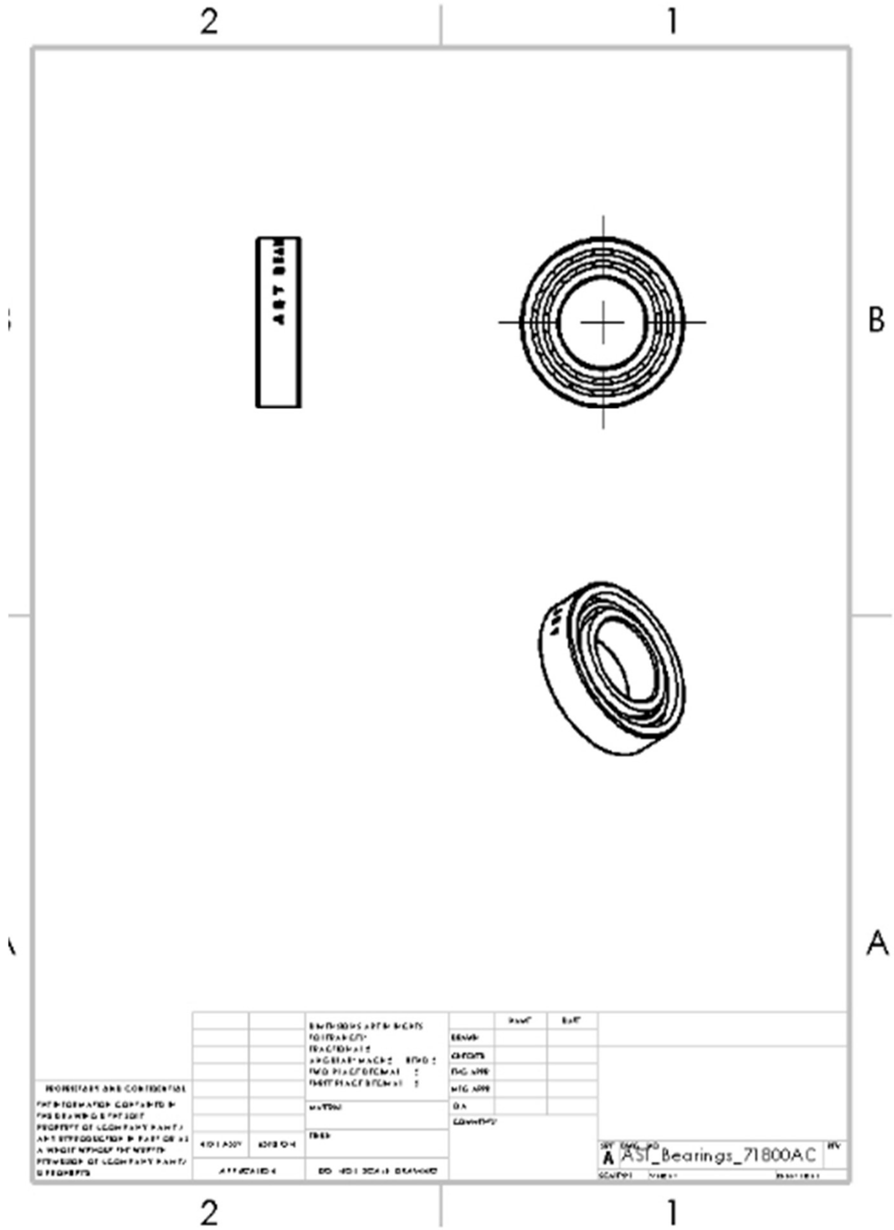


Figure 13: Drawing of the Bearing

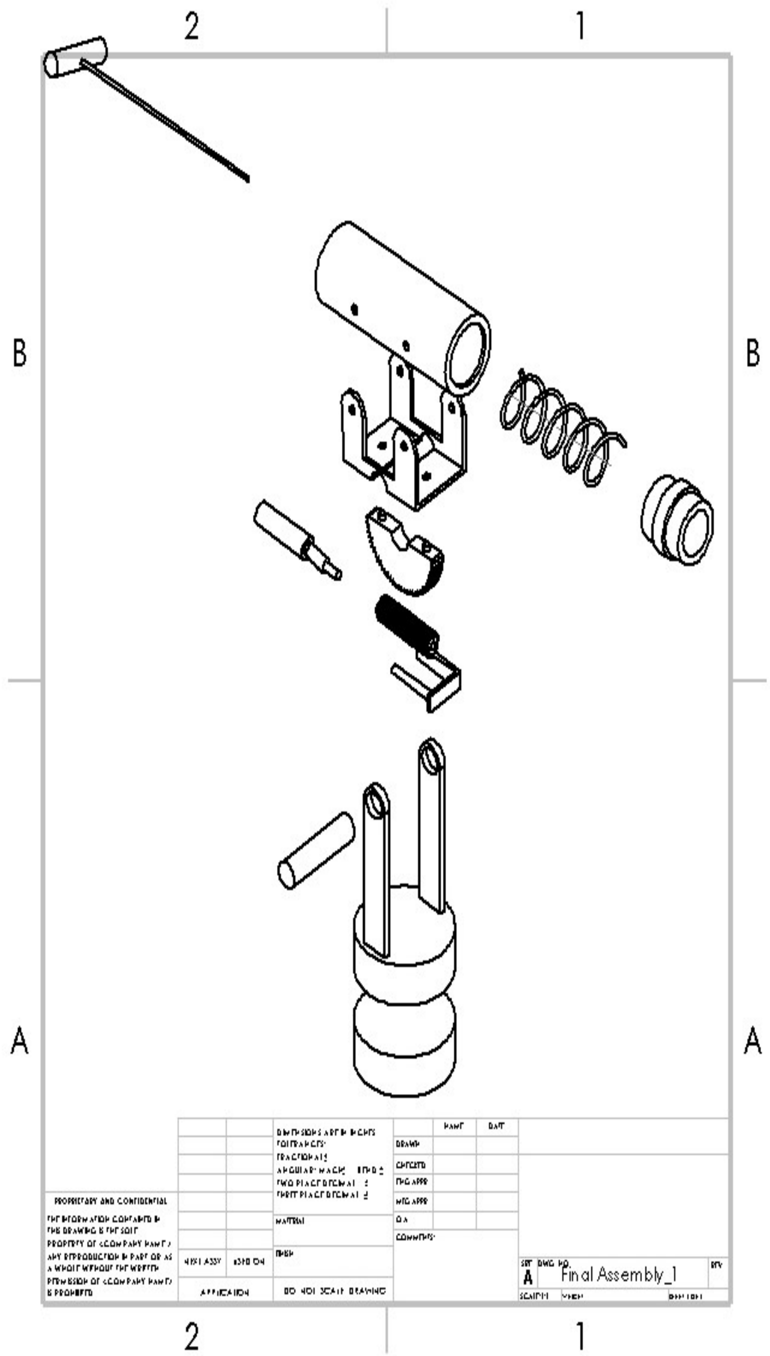


Figure 14: Drawing of the Final Assembly

Spring Calculation Code

```
%% Spring Sample Calculations
% for spring that was chosen - 96485K156 from McMaster Carr

%dimensions of spring
Do = 2.188;         %in
Di = 1.774;         %in
d = 0.207;          %in
D = (Do+Di)/2;  %in
c = D/d;            %index
Kw = ((4*c-1)/(4*c-4)+0.615/c); %wahl factor

syms V t    %projectile motion equations solved
[solV, solt] = solve(V*sin(45*pi()/180)*t==3, V*sin(45*pi()/180)*t+0.5*(-
1)*9.81*t^2==-1);  %solving parabolic motion
v = double(solV(2));    %initial velocity required

m = 1.013;  %kg mass of baseball and baseball cup
k = 32.3;   %spring rate - lb/in

kM = k/0.0057101471627692;  %convertion of spring rate to N/m

x = (m*v^2/kM)^0.5/2.54*100;  %distance to be pulled back in inches
F = k*x;    %force to pull back required

%Stress Calculations
%max shear stress spring experiences
tw = Kw*8*F*D/(d^3*pi());

%fatigue failure limit for tempered steel
Sut = 146.8*10^3*d^-0.1833;
tf = 0.4*Sut;

%factor of safety
n = tf/tw;
```

Gear calculation code

```
clc
clear

%GEAR CALCULATIONS
```

Design Characteristics

```
Ng=360/5;
dg = 5.51; %Inch

Pd = Ng/dg;
m =25.4/Pd;

b = 1; %inch

Px = pi()/Pd;
Nw = 2;
Lw = Nw*Px;

dw = 1.25;
lambdaw = atan(Lw/pi/dw)/pi*180;


C = (dg+dw)/2;

dow = dw +2/Pd;
drw = dw - 2.314/Pd;

drg = dg - 2.314/Pd;
dog = dg + 3/Pd;
```

Force Analysis

```
mu = 0.25;

e= -cosd(20)*sind(lambdaw)+mu*cosd(lambdaw);

Ssf = 170*10^3;

%Assume maximum torque applied on the crank of 100 lb.*in and the crank to
be
%5in long
T = 100;
r = 5;
Fwt = T/r;
Fwa = Fwt*(cosd(20)*cosd(lambdaw)-
mu*sind(lambdaw))/(cosd(20)*sind(lambdaw)+mu*cosd(lambdaw));
Fwr = Fwt*sind(20)/(cosd(20)*sind(lambdaw)+mu*cosd(lambdaw));

Fgt = Fwa;
Fga = Fwt;
Fgr = Fwr;
```

```
%Bending force spur gear
J = 0.48; Ka = 1.50; Kv = 1.1; Km = 1.6; Kl = 1.0;

sigmabspur = Fgt*Pd/b/J*Ka*Kv*Km*Kl;
sigmabworm = Fwt*Pd/b/J*Ka*Kv*Km*Kl;

Stbf1 = 45000;
Rg = 1.0;
Yn = 1.0; %Assume to have 10^7 cycle
Stbf = Stbf1*Rg*Yn;
nd = 1.5;
sigmabmax = Stbf/nd;

FactorSafetyBendingSpur = sigmabmax/sigmabspur;
FactorSafetyBendingWorm = sigmabmax/sigmabworm;
%Surface fatigue  spur gear
v = 0.3; Ep = 30*10^6;
Cp = sqrt(1/(pi*((1-v^2)/Ep+(1-v^2)/Ep)));

I = sind(20)*cosd(20)/2*(m/(m+1));

sigmasfspur = Cp*sqrt(Fgt/b/dg/I/m*Ka*Kv*Km);
sigmasfworm = Cp*sqrt(Fwt/b/dg/I/m*Ka*Kv*Km);

Zn = 1.0; %Life cycle of 10^7
Ssf1 = Ssf*Zn*Rg;
sigmasfmax = Ssf/nd;

FactorSafetySfSpur = sigmasfmax/sigmasfspur;
FactorSafetySfWorm = sigmasfmax/sigmasfworm;
```
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Bearing calculation code

```
clc
clear all

g=9.81;
theta=45;
rot=200;
hr=8;
day=6;
wk=51;

%Axial & Radial Loads
m_user=32.66;
m_mach=10;

F_x=m_user*g*sind(theta);
F_y=m_user*g*cosd(theta)+m_mach*g;

%Designed Life
L_d=rot*hr*day*wk;

%Safety Factors
K_r=0.21;
IF=3.0;

%Radial Load Factors
Xd1=1;
Yd1=0;
Xd2=0.45;
Yd2=1.2;

%Dynamic Equivalent Radial Load
Pe1=Xd1*F_x+Yd1*F_y;
Pe2=Xd2*F_x+Yd2*F_y;

if Pe1>Pe2
      Pe=Pe1;
else
      Pe=Pe2;
end

%Basic Dynamic Load Rating
Cd=(L_d/(K_r*10^6))^(1/3)*IF*Pe;
```