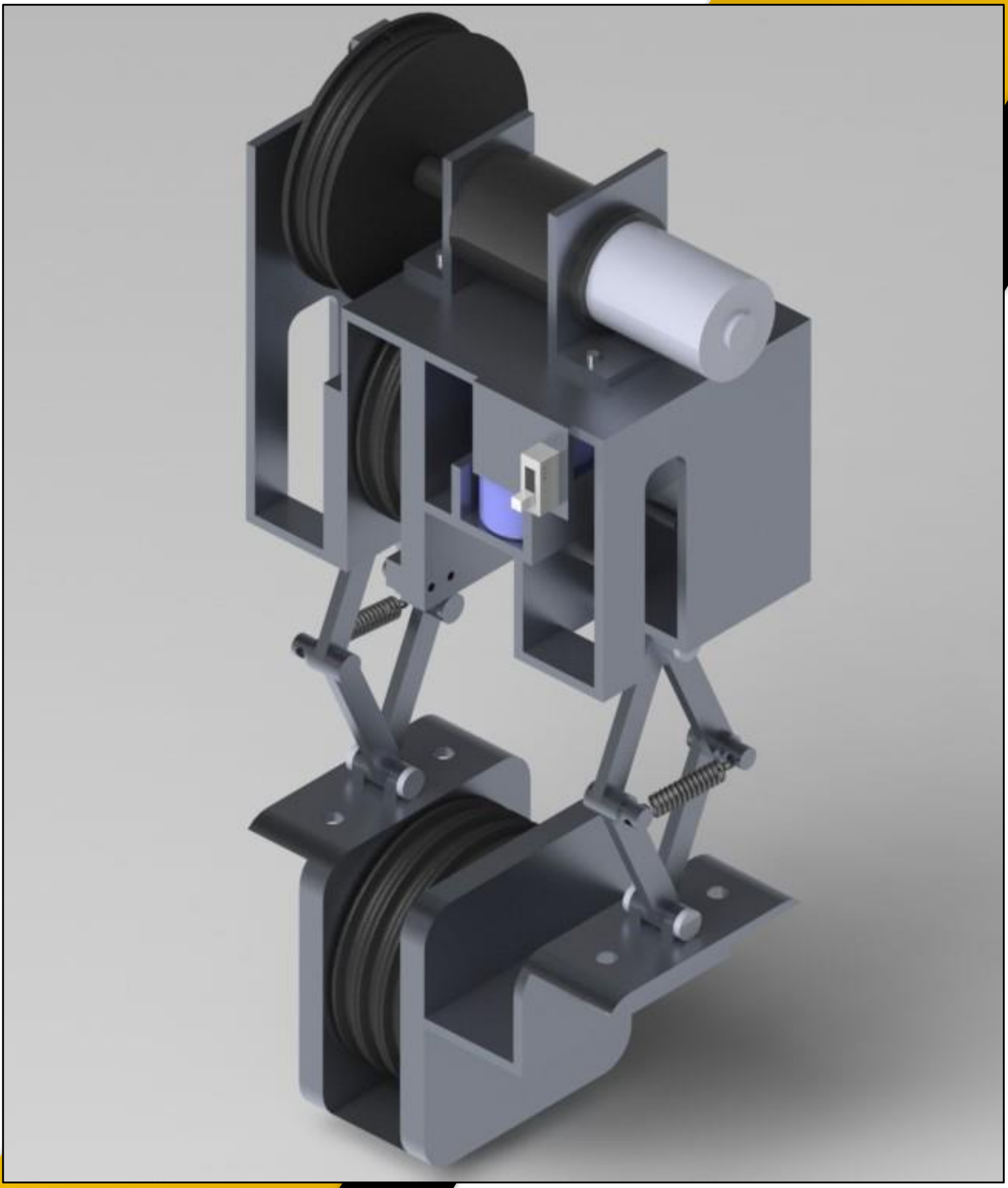
Design of Machines and Mechanical Systems

PULLEY DRIVEN CRIMPING TOOL ATTACHMENT



Design Team

Manav Mepani - M1810029
Tanish Mishra - M1810034
Jay Naik - M1810037
Vedant Patil - M1810043

Problem Statement

To provide engineering consultancy to a company that manufactures crimping tools by proposing a machine design that circumvents the need for the application of large forces on the handle of their crimping tool by workers. Repeated application of such forces manually leads to carpal tunnel syndrome which must be avoided.

Objective

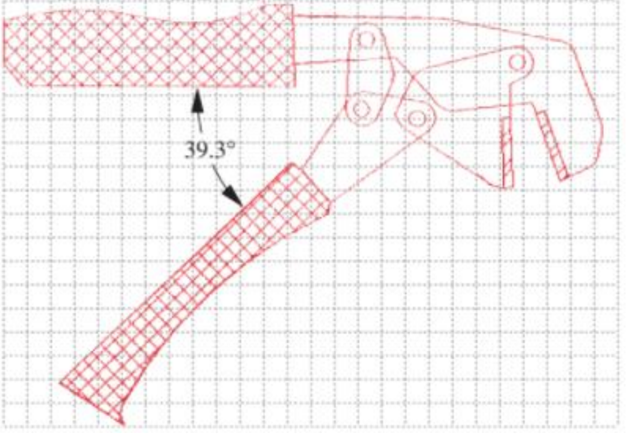
The design must be lightweight, inexpensive, easy to use and must minimize human effort.

Client Requirements

1) The device must be no larger than 300 mm x 100 mm x 150 mm.
2) Must be self-contained, including a portable energy supply.
3) Weigh less than 5 kg.
4) Require only one- or two-handed operation by one person.
5) Perform a crimp in less than 10 second.
6) Accommodate maximum handle angle of 40° and handle splay (spread) of 175mm.

Angle Between Handles, Deg.	Torque, lb_f-in	Distance Between Handles, in	Force, lb_f
39.3	0	2.89	0
24.4	49.17	1.89	11.84
22.3	73.27	1.74	17.36
20.9	97.53	1.63	22.89
19.7	122.01	1.54	28.42
18.7	146.62	1.46	33.95
17.8	171.38	1.40	39.47
16.5	220.91	1.30	50.52
15.1	271.10	1.19	61.57
13.7	321.77	1.08	72.63
12.6	372.41	1.00	83.68
11.2	423.77	0.89	94.73
9.5	475.78	0.75	105.79
8.0	502.66	0.64	111.32
7.6	528.12	0.60	116.84
5.5	580.52	0.43	127.89
4.8	606.27	0.39	133.42
0.8	653.34	0.06	143.29

Table of force (at r=115 mm) and torque variation with phase of crimp



Crimping tool on a 1cm grid

TABLE OF CONTENTS

Section 1 : Design Overview

1.1 Introduction 3
1.2 Working principle 4
1.3 Explanation of Design Decisions 5

Section 2 : Rope Drive

2.1 Selection of Rope 13
2.2 Rope Drum Design 15
2.3 Rope Sheave 16
2.4 Sheave Axle Design 17

Section 3 : Bearings

3.1 Bearing Requirements 19
3.2 Bearing calculation 20

Section 4 : Design Considerations

4.1 Considerations 22

Section 5 : Sheets & Models

5.1 All Views of model 25
5.2 Exploded View with BOM 29
5.3 General Drawing Assembly 30
5.4 Detailed Drawing for important components 31

1. Design Overview

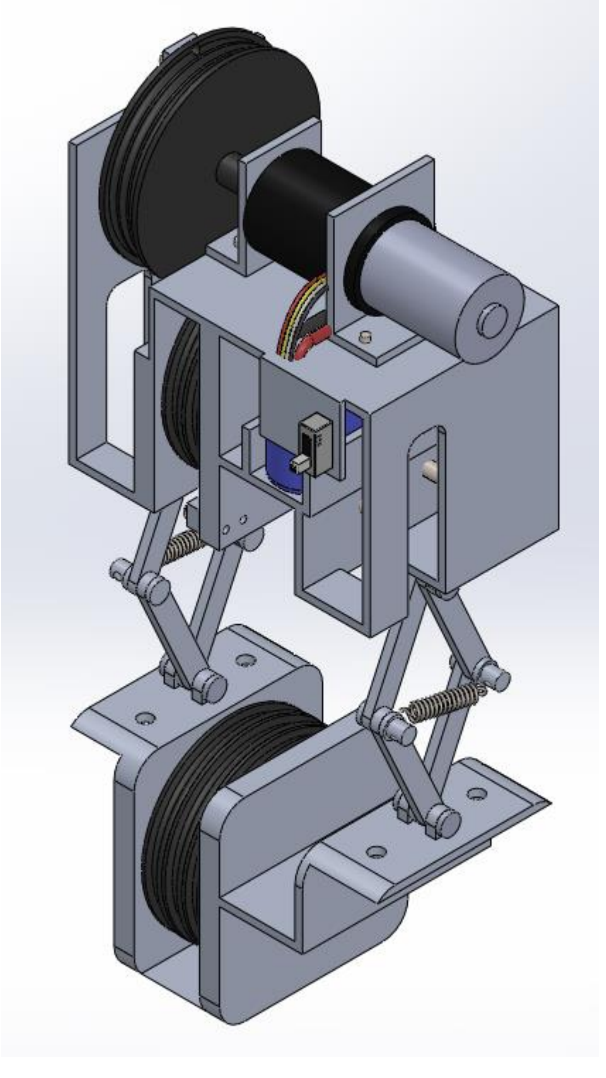
1.1 Introduction
1.2 Working Principle
1.3 Explanation of Design Decisions
    1.3.1 Selection of DC Motors and Number of Pulleys
    1.3.2 Return Mechanism and the Spring
    1.3.3 The Cell and the Switch

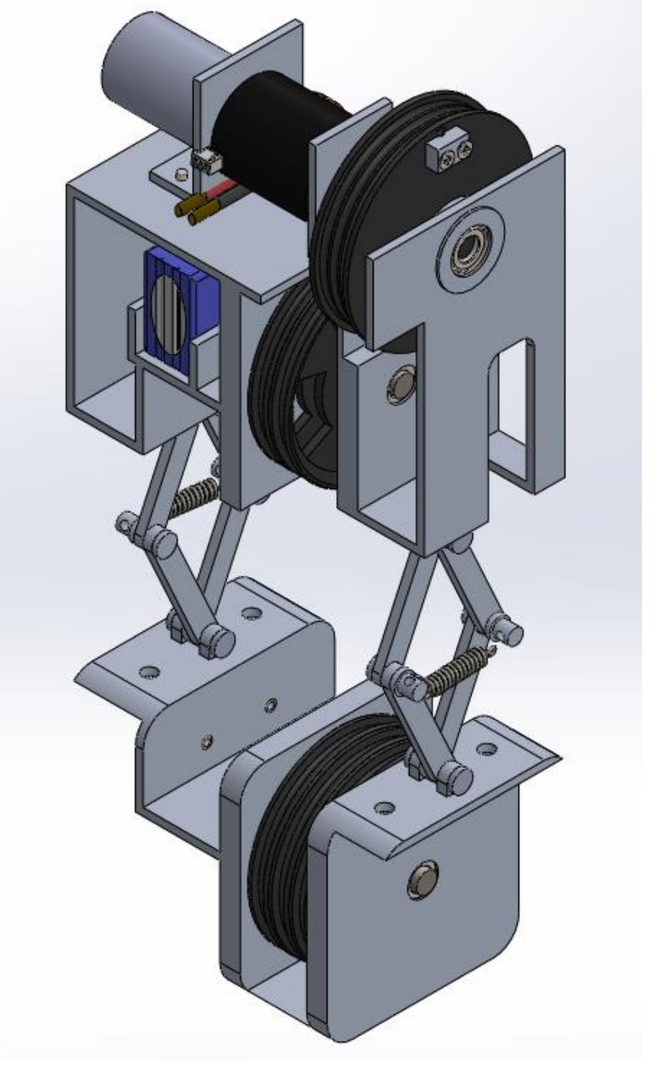
1.1 Introduction

To overcome the issues mentioned in the problem statement, the proposed design aims to eliminate the work done by the workers almost entirely. This design contains a 5 pulley system which provides a mechanical advantage of 6 and is actuated by a motor powered by a 12V Li-Po battery

The device is expected to be operated in a handheld position, through the use of the gripping slots provided on either side of the housing as shown in the pictures. A conveniently placed 3 position sliding switch is used to either start the crimping action, reverse it or to switch off the device; with the thumb.

Appropriately sized clamps with grub screws have been incorporated in the design to allow for convenient attachment and detachment of the crimping tool by its handle.
The two halves of the mechanism are connected by a parallelogram linkage

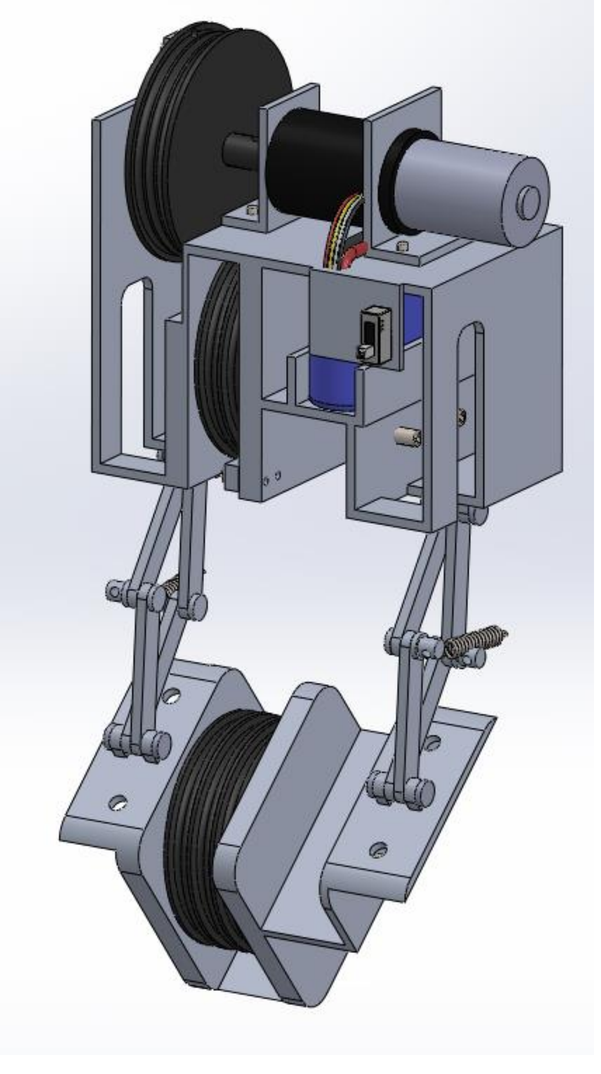


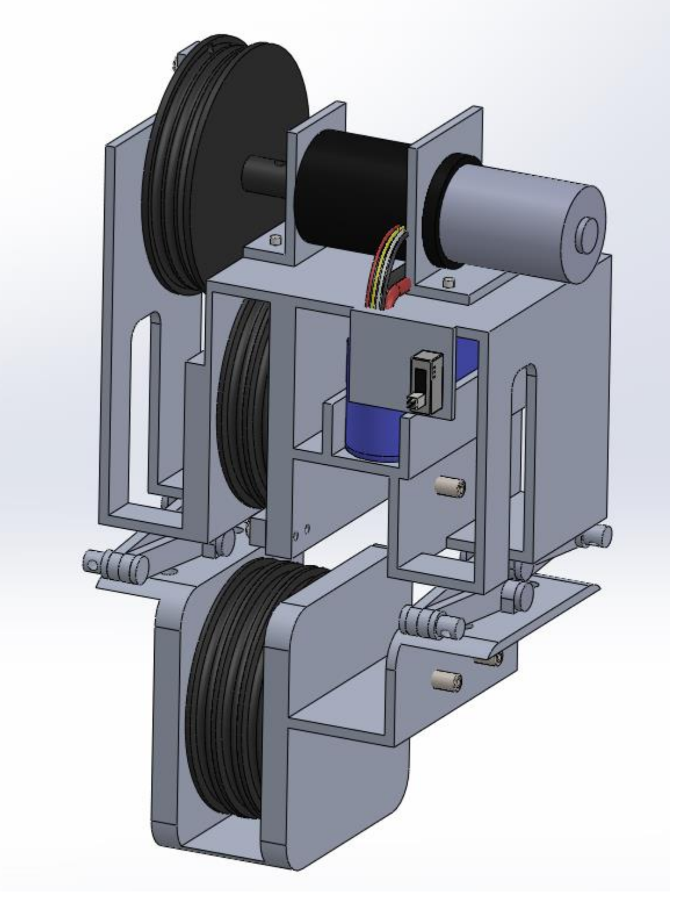


1.2 Working Principle

Given the substantial forces and torques that need to be generated despite the relatively small dimension constraints, direct actuation of the crimping action becomes infeasible due to the correspondingly large size of any appropriately powerful motor.
Hence, some sort of mechanical advantage is necessary.

Pulley arrangement was chosen for the same due to its high efficiency and minimal maintenance requirements.
When the motor rotates, the rope is wound around the drum, tending to bring the two halves of the device closer together, along with the handles of the crimping tool which are clamped in the space provided.





Open position of crimping tool

Closed position of crimping tool

1.3 Explanation of Design Decisions

1.3.1. Selection of DC Motor and Number of Pulleys

Calculations were initiated on the basis of client requirements.
Pulley was assumed to be attached at radius 115mm since data was available directly at that point.

The following assumptions were made to start the calculations with:

1) Cable diameter must be the smallest possible, found to be 3mm (*Source:*https://www.steelwirerope.com/WireRopes/Galvanised/6x19-fibre-core-wire-rope.html)
2) Mechanical Advantage of 6 was taken considering size constraints with respect to the rope diameter

Now, given that the max dist between the 2 handles of the tool is 2.89" (73.406mm) at 115mm, the motor must wind 6*73.406mm of the rope to complete the crimp operation.
To perform this within 10 sec, Linear Velocity of the rope must be

$$\frac{(6*73.406)}{10} = 44.0436\, mm/s = 0.044\, m/s$$

As is explained in Section 2.1, ratio of sheave and drum diameter to rope diameter is 23.
Hence, diameter of drum=23*3=69mm

Therefore, min RPM of motor is (assuming constant value throughout the crimping operation) is:

$$\frac{Velocity*60}{(2*\pi*(\frac{(Drum\ Diameter + Rope\ Diameter)}{2}))} = \frac{0.044*60}{(2*\pi*(\frac{(0.069+0.003)}{2}))} = 11.68\, RPM$$

Maximum force output required on the crimping tool (using the data table) is 645N. Let Let this be rounded up to 1000N to allow for a factor of safety in the design
The velocity of the point on the handle at 115mm is

$$\frac{0.044}{6} m/s = 0.00733 m/s$$

Therefore max output power at handle =

$$1000\,N * 0.0073406\,m/s = 7.3406\,W$$

But,
Output power=Input power= Torque of motor*Angular velocity of motor

Hence, torque of motor =

$$\frac{7.3406}{\left(\frac{11.68 * 2\pi}{60}\right)} = 6.01\,Nm = 600.1\,Ncm$$

Based on these specifications (Torque>=600.15Ncm, RPM>=12.17), the following motor was found to be feasible:
Rhino IG32 12V Planetary Geared DC Motor 30 RPM and 686.7 Ncm
(Picture can be seen in the CAD model)

Note that the rated RPM of this motor is 30 which means that the crimp operation can be completed in 3.9 sec

1.3.2. Return Mechanism and the Spring

The pulley mechanism is driven by an open loop cable. Hence, the unwinding operation cannot be carried out by the motor alone, considering the fact that the cable can only transfer power in tension.
The device is therefore designed to unwind under the action of self weight in combination with the spring provided across the 4 bar parallelogram linkage.

The minimum force that needs to be applied to initiate and carry out the unwinding is the force required to overcome friction in the bearings (and some additional force to overcome the friction in the crimping tool, the data for which is unavailable).

As explained in Section 3.2, the frictional torque due to each bearing is 9.63 Nmm. Since there are 4 bearings (2 on either shaft), the net frictional torque to be overcome

$$=4*9.63$$

$$=38.52\,Nmm$$

Hence, net difference in tension required in cable sections on either side of the sheave is

$$\frac{38.52}{(Radius\ of\ Sheave + Radius\ of\ Cable)}$$

$$=\frac{38.52}{72}=0.535\,N$$

When the cable tends to become slack (as the drum unwinds when the motor is rotated in reverse), the tension on one side drops to 0. Hence, 0.535 can be taken as the sum of tensions on one side of each fall of the cable=downward force required to cause the unwinding.

Weight of the lower assembly= 5.1N>>0.535N. Hence, even under the action of self weight only, unwinding action should occur. With the addition of the spring force, the robustness of the unwinding is further expanded.

The presence of the 4 bar linkage ensures that the spring acts under tension rather than compression that could have otherwise led to buckling.
Further, the shape of the 4 bar linkage is such that the effect of spring force in the vertical direction is maximum when the tool is completely open and minimum when the tool is closed (explained subsequently). This is complementary to the force required for the crimping action which is maximum as the angle between the handles falls to zero (acc. to the data table). This means that the maximum force of the spring will never act at the same time as the maximum force for crimping, helping to mitigate the negative consequences (additional load) of adding the spring.

The specifications of the spring (found here, chosen based on ease of availability) are as follows:

Specifications

Brand	RS Pro	Material	Steel
Free Length	41.4mm	Maximum Extended Length	125.70mm
Outside Diameter	6mm	Wire Diameter	0.55mm
Initial Tension	9.5N	Spring Rate	0.1N/mm
Country of Origin	GB		

Analysis of the force characteristics of the 4 bar Linkage can be carried out as follows:
The linkage and the spring can be considered as a truss.
The picture on the right shows the equivalent truss structure and the forces in each member.
Considering the total length of spring at any time to be ‘x’, The following equations can be derived-
Taking Equilibrium about Point A:

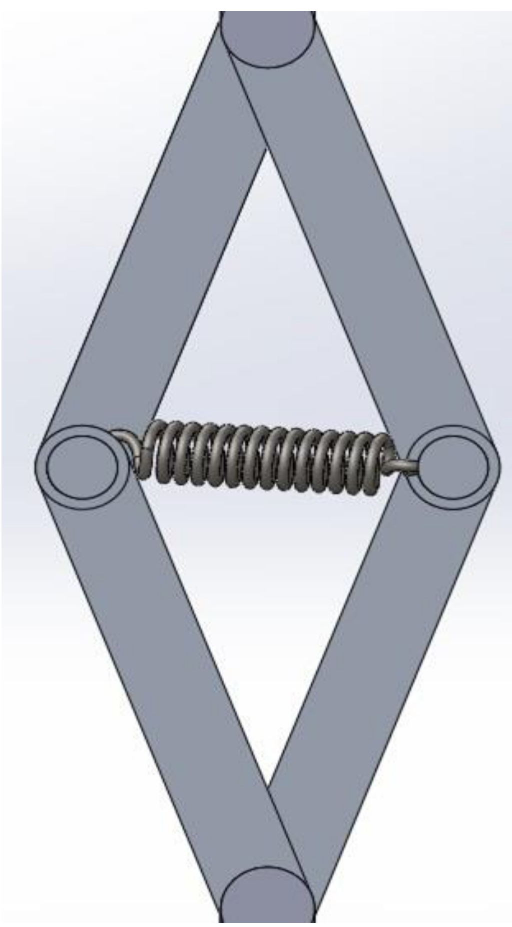
$$2T*\cos(\theta)=S=k(x-x_0)+P$$

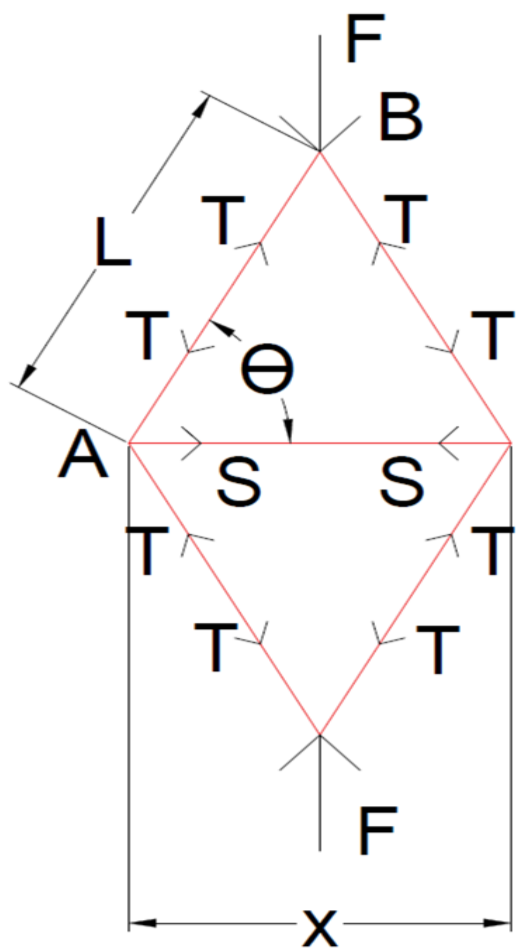
Where, k=Spring Rate=0.1N/mm; P=Initial Tension=9.5 N; x_0=Initial Length of spring
Next, Taking Equilibrium about B:

$$2T*\sin(\theta)=F$$

Further, $\frac{x}{2}=l*\cos(\theta)$ and $\frac{x_0}{2}=l*\cos(\theta_0)$

Solving, we get: $F=2*(k*l*\sin(\theta)-k*l*\cos(\theta_0)*\tan(\theta))+P*\tan(\theta)$

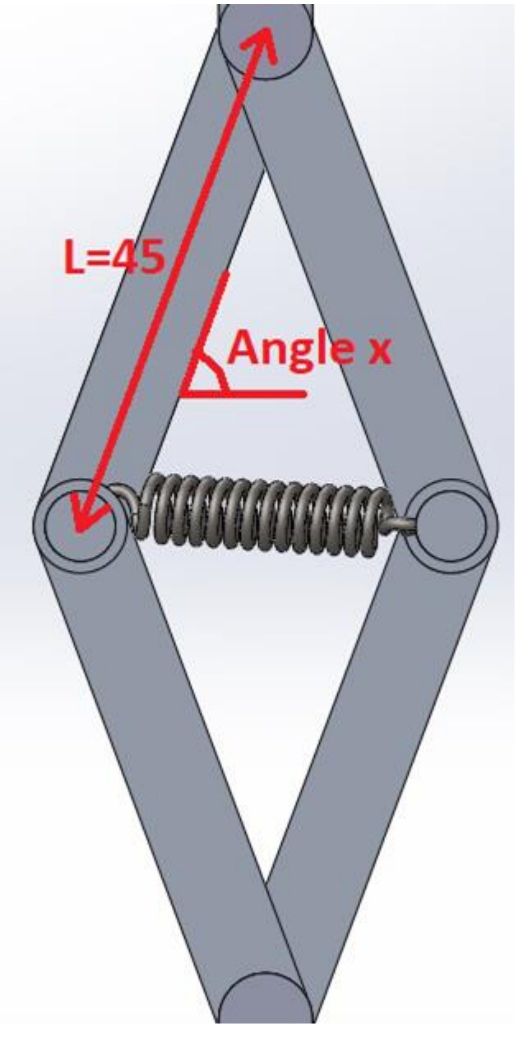


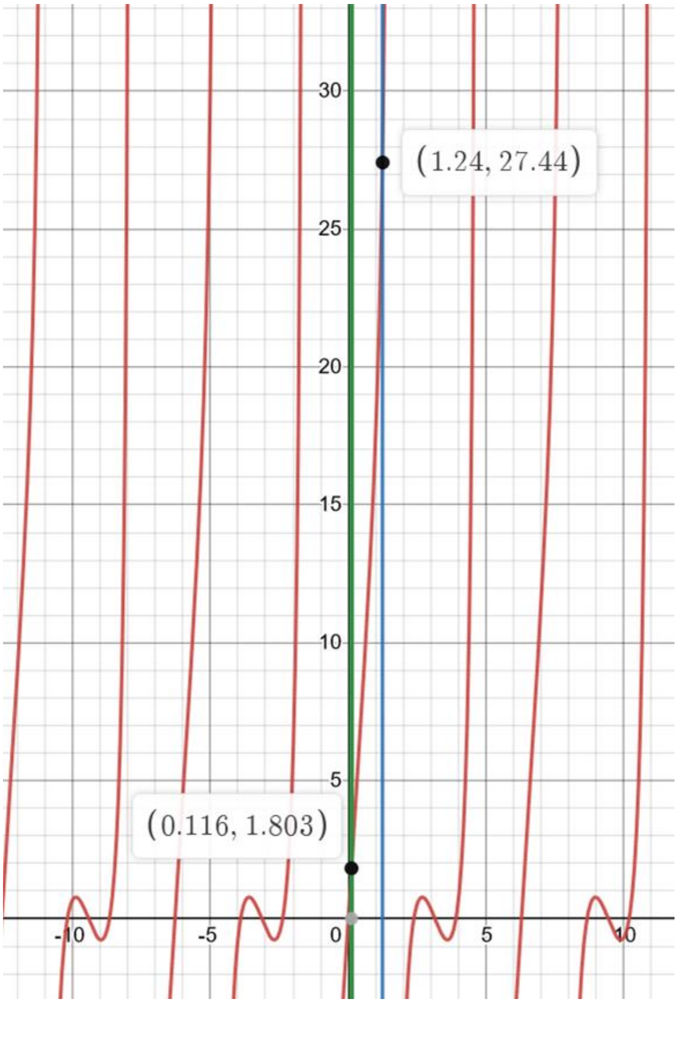


Using these calculations, a plot (shown on the picture on the right) of the force against angle 'x' (shown in the picture on the left) can be made.
In the open position, x is 1.24 rad (=x_0) and in the closed position, x is 0.116 rad (=x_1)(found using the CAD model)

$$\text{Spring Force in vertical Direction } (y) = 2*(k*l*\sin(x) - k*l*\cos(x_0)*\tan(x)) + P*\tan(x)$$

By plotting the graph, max force=27.44N (Shown below) and occurs at the start of the crimping action (i.e. when x is maximum)





1.3.3. The Cell and the Switch

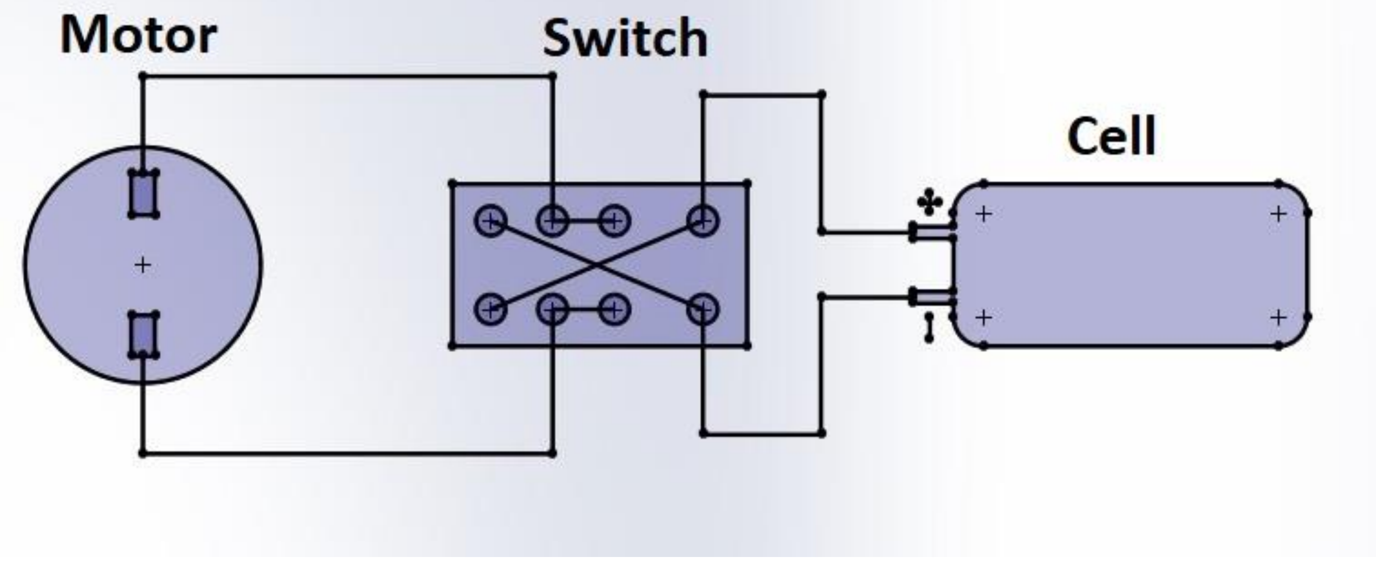
The most popular choices of batteries for applications like these are Li-Ion, Li Polymer cells and Lead Acid. Of these, Li-Po is the safest (no potential leakage of electrolyte), has the least weight, highest energy density and can undergo the most charge cycles and is therefore the power source of choice. The specific model chosen can be found here

A 3-Position 8 pin DPDT sliding switch is used to control the motor since it must be able to reach 3 possible states (On, Off, Reverse).
We know that the motor is expected to generate 7.3406W (Section 1.3.1)
Rounding it up to 10W and considering that the voltage is 12V,

$$Expected\,current = \left(\frac{Power}{Voltage}\right) = \left(\frac{10}{12}\right) = 0.833\,A$$

Based on this current requirement, the switch found here is chosen (Rated for 4A)

The wiring and circuit diagram for the same can be found below:



2. Rope Drive

2.1 Selection of Rope
2.2 Rope Drum Design
2.3 Rope Sheave
2.4 Sheave Axle Design

2.1 Selection of Rope

- **Procedure:**

➢ As the wire rope bends multiple times during crimping, wire rope of construction 6 X 19 was selected due to its higher flexibility over wire rope of construction 6 X 7.
➢ The calculation of minimum rope diameter was done on the basis of long life of the rope.
➢ The wire rope has long fatigue life if the ratio (p/Sut) is less than or equal to 0.0015.

$$p = \frac{2P}{d_r D}$$

where,
p = pressure between rope and sheave (N/mm^2)
P = tension in the rope (N)
d_r = nominal diameter or wire rope (mm)
D = sheave diameter (mm)
S_{ut} = ultimate tensile strength of wire (N/mm^2)

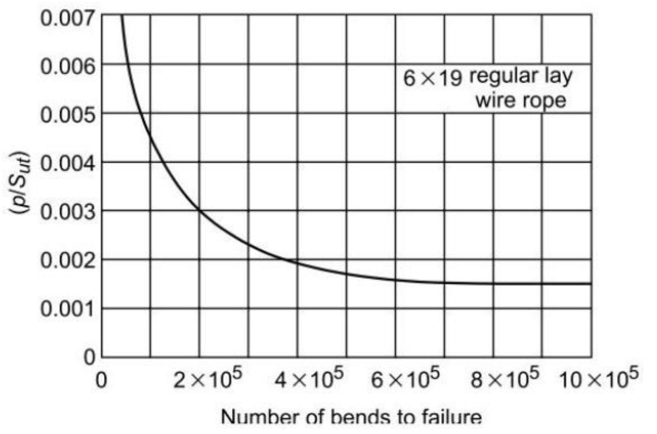
➢ The wire rope of Tensile Designation 1570, 1770 & 1960 (N/mm^2) were considered and respective pressures were calculated.
➢ The minimum recommended value of 30 was considered for D/d_r.
➢ Total force:

- F = Max. Crimping load + Weight of lower housing assembly + Frictional load (bearing losses) + Vertical component of the spring force
  = 645 N + 5.10 N + 1.11 N + 27.55 N = 678.76 N
  The calculation for bearing losses are presented in the bearing section.
- A higher force of 1000 N was considered which gives tension on individual fall as 1000/6 = 166.66 N

➢ The Tensile Designation of 1960 yielded the lowest value of rope diameter of 2 mm for P = 166.66 N Excel.
➢ But minimum rope diameter was considered to be 3 mm due to unavailability of rope of diameter 2 mm link.
➢ D/d_r was changed to 23 as this application doesn't involve high forces as those in cranes or hoists.
➢ The final calculations considering the bending load on rope are provided in this Excel.

$$P_b = \frac{AE_r d_w}{D}$$

where,
P_b = Equivalent bending load (N)
A = metallic area of rope (mm^2)
d_w = diameter of wire (mm)
D = sheave diameter (mm)
E_r = modulus of elasticity of rope (N/mm^2)



Fatigue diagram for 6 X 19 regular lay
Courtesy: Design of Machine Elements, V B Bhandari

6 X 19 Fiber Core Wire Rope

- **The final parameters for the wire rope are as follows:**

Wire Rope	
Construction	6 X 19 regular lay
Core	fiber
Modulus of Elasticity (E_r) (N/mm^2)	83, 000
Nominal diameter of wire rope (d_r) (mm)	3
Diameter of wire (d_w) (mm)	0.189
Metallic area of rope (A) (mm^2)	3.6
Sheave diameter (D) (mm)	69
Breaking load (kN)	5.4
Static factor of safety (FOS)	5.48
No. of bends to fatigue failure	90,000
Number of Crimps to rope failure	7500

2.2 Rope Drum Design

- **Procedure:**

➢ The rope drum was designed by referring to IS 3177.

➢ According to the standard, the combined stress generated in the drum must be less than the allowable stress which is calculated on the basis of material UTS and various design factors.

➢ The calculations for the same are provided in this Excel.

- **Formulae:**

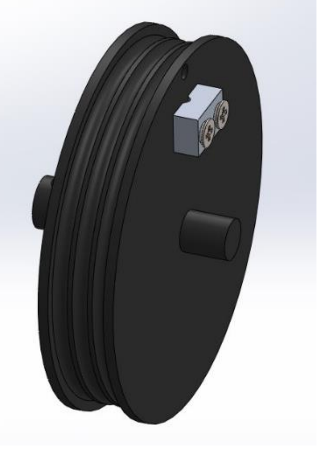
1) Allowable stress:

$$F_a = \frac{UTS}{C_{df} C_{bf} C_{sf}}$$

where,
F_a = Allowable stress (N/mm^2)
UTS = ultimate tensile strength (N/mm^2)
C_{df} = impact/duty factor
C_{bf} = basic stress factor
C_{sf} = safety factor

$$\sqrt{(f_c + f_{bc})^2 + 3f_s^2} < F_a$$

where,
f_c = compressive stress (N/mm^2)
f_{bc} = bending stress (N/mm^2)
f_s = shear stress (N/mm^2)



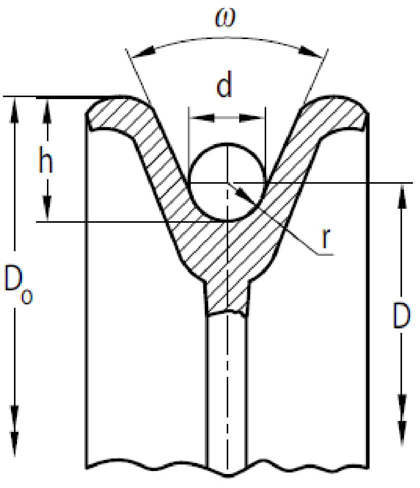
Rope Drum

- **The final parameters for the rope drum are as follows:**

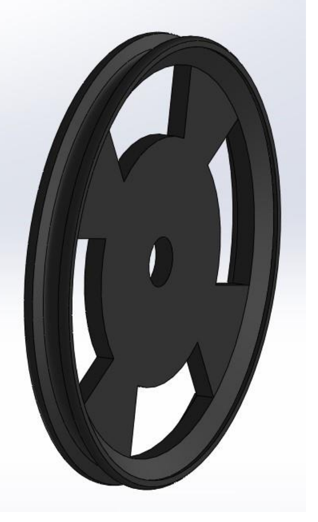
Rope Drum	
Material of rope drum	FG 200
Length of rope drum (L) (mm)	12
PCD of rope drum (D) (mm)	69
Radius of groove in the drum (r) (mm)	3.15
Pitch of ropes (p) (mm)	4
Allowable stress (F_a) (N/mm^2)	152.38
Combined stress < F_a	Pass

2.3 Rope Sheave

- The proportions for the sheave were taken from Design Data Book Table No. 27.15 without direct consideration of the stresses.
- The sheaves are press fitted on the axle and no gap was kept between each sheave.



Proportions of sheave
Courtesy: Design Data Book, V B Bhandari



Rope Sheave

- **The dimensions of the sheave are as follows:**

Sheave	
Material of rope sheave	FG 200
PCD of rope sheave (D) (mm)	69
Outside diameter of sheave (D_o) (mm)	72
Depth of sheave groove (h) (mm)	3
Angle between sides of sheave (ω) (˚)	45
Radius of groove in the sheave (r) (mm)	3.15

2.4 Sheave Axle design

- **Procedure:**

➢ The axle was designed on the basis of Maximum Shear Stress Theory of Failure.
➢ Considering the thickness of the sheave and the required spacing between the sheave and the bearing the minimum length of the axle was determined to be 26 mm.
➢ The maximum bending moment acting on the axle was calculated due to the forces on each sheave and the minimum required standard diameter of the axle was calculated to be 8 mm Excel.

- **Formulae:**

$$\tau_{\max} = \frac{16}{\pi d^3}\sqrt{(M_b)^2 + (M_t)^2}$$

$$S_{sy} = 0.5\, S_{yt}$$

$$\tau_{\max} = \frac{S_{sy}}{(fs)} = \frac{0.5 S_{yt}}{(fs)}$$

where,
τ_{max} = maximum shear stress (N/mm^2)
M_b = bending moment acting on hollow shaft (N-mm)
M_t = torsional moment acting on hollow shaft (N-mm)
d = diameter of shaft (mm)
S_{yt} = yield strength of shaft material (N/mm^2)
S_{sy} = yield strength of shaft material in shear (N/mm^2)
fs = factor of safety

- **The final parameters for the axle are as follows:**

Sheave Axle	
Material	45C8
Yield Strength (S_{yt}) (N/mm^2)	380
Factor of safety (fs)	3
Length of shaft (L) (mm)	26
Diameter of shaft (d) (mm)	8
Bending moment (M_b) (N-mm)	3833.33

3. Bearings

3.1 Bearing Requirements
3.2 Bearing Calculations

3.1 Bearing requirements

Design conditions:

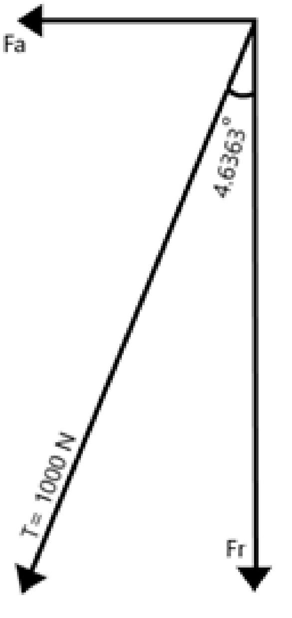
1. The bearing must be compact and lightweight.
2. The bearing must have a good life.
3. The bearing must be able to sustain high radial loads along with some minor axial loads.
4. Power dissipation due to friction in the bearing should be minimum.
5. The bearing must be economical.
6. The bearing should generate less noise.

Based upon design conditions, deep groove ball bearing was selected. DGBB has the following advantages:

1. DGBB have high load carrying capacity.
2. DGBB can take both radial as well as axial loads.
3. DGBB generate less heat and frictional losses because of point contact between the balls and races.
4. DGBB generate less noise due to point contact.

For all axles, two bearings of same configuration are used in order to sustain the axial forces in both directions considering the bidirectional movement of sheaves.

Grease Lubrication was selected for bearings as it allows easy maintenance and eliminates fluid spillage.
SKF LGGB2 Grease was used, which is biodegradable.



Total tension to be applied = 1000 N

Fa = Horizontal component = 80.83 N

Fr= Vertical component/2 = 996.727868/2 = 498.363934 N

3.2 Bearing calculations

Based upon shaft diameter of 8mm, bearing designations were considered. Initially, SKF 608 bearing was considered.

Bearing Specifications:

Designation	d (mm)	D (mm)	B (mm)	C (kN)	C0 (kN)	Mass (kg)
608	8	22	7	3.45	1.37	0.012

According to DDBT 15.2, Equivalent dynamic load = P = X Fr + Y Fa
From DDBT 15.9, we get X=1 & Y=0.
Hence the axial force is negligible and the equivalent dynamic load is equal to the radial load.
P= Fr = 498.363934 N = 0.498363934 kN
n= 18 rpm

Bearing life calculations:

$$L_{10} = \left(\frac{C}{P}\right)^3 = \left(\frac{3.45}{0 \cdot 498363934}\right)^3 = 331.7549939\, mr$$

$$L_{10h} = \frac{L_{10} \times 10^6}{60 \times n} = \frac{331.7549939 \times 10^6}{60 \times 18} = 307180.5499\, hours$$

Bearing Friction calculations:

Friction *M : 9.63 Nmm*

	Frictional moment		Friction sources				Power loss
Designation	Total	At start 20-30°C and zero speed	Rolling	Sliding	Seals	Drag loss	
	M (Nmm)	M_{start} (Nmm)	M_{rr} (Nmm)	M_{sl} (Nmm)	M_{seal} (Nmm)	M_{drag} (Nmm)	P_{loss} (W)
☆ ■ 608	9.63	11.8	0.16	9.47	0	0	0.018

Bearing friction moments were obtained from SKF Bearing select tool.
Bearing friction Force = 4*Frictional Moment/Sheave Radius = 1.1165 N

The relubrication interval is 12300 hrs as calculated according to SKF Bearing select

Thus **SKF 608** bearing was selected as it fulfilled the requirements.

4. Design Considerations

Economic
Environmental and Sustainability
Social and Cultural
Legal and Ethical
Safety and Health
Manufacturability
Ergonomics

4.1 Considerations

- **Economic**

- Each part's dimension, material and other specifications were chosen to satisfy customer requirements along with minimal cost.
- Most of the parts used are Off-the-shelf/Standard items for easy availability and replacement. Many variations of these were compared and most economic options were chosen.
- Least wastage of material was ensured by the selected manufacturing process.
- Over design was avoided and in turn unnecessary expenditure was reduced.

- **Environmental and Sustainability**

- Usage of fluids was avoided in the design, hence spillage issues and usage of toxic fluids is eliminated.
- A biodegradable grease is chosen for bearing lubrication.
- Most of the metals used are recyclable.
- The LiPo battery is disposed by following standard procedures.

- **Social and Cultural**

- To support Make in India moment many of the parts were designed so that they can be manufactured indigenously instead of importing them.
- Employment opportunities for local workers can be created in rural areas due to the indigenous nature of manufacturing.
- Training programs to create skilled labor are encouraged as the manufacturing processes require some skill.

- **Legal and Ethical**

- All the practices followed in the design & manufacturing are in conformity with legal guidelines provided by Government.
- As specified by Engineering code of Ethics "Hold paramount the safety, health, and welfare of the public", we have designed our crimping tool mechanism keeping the health and safety of the user and handler in mind.
- Products used have a Certificate of Compliance for the EU Directive 2011/65/EU
- All the data provided in this report is verified and true.

Safety and Health

- Usage of toxic materials has been avoided.
- The design is made such that it avoids any harm due to spillage of fluids.
- The design ensures safety of fingers from the wire rope and hence avoids any bruising or cutting.
- All individual parts are provided with sufficient margin for error to avoid any kind of sudden failure and harm occurring due to it.
- It was ensured that the noise was minimum.
- The LiPo battery is protected by a hard casing in order to avoid rupturing/ puncture and explosion of the battery.
- All wiring is well insulated and chances of electrocution are completely eliminated.

Manufacturability

- It is taken care that all materials used in designing are readily available and easy to machine.
- The processes that are used for manufacturing are the standard processes used in industry, to ensure low cost of parts and being able to be manufactured readily.
- Selection of recommended parts from manufacturer's catalogue were chosen to reduce the need for custom made parts.

Ergonomics

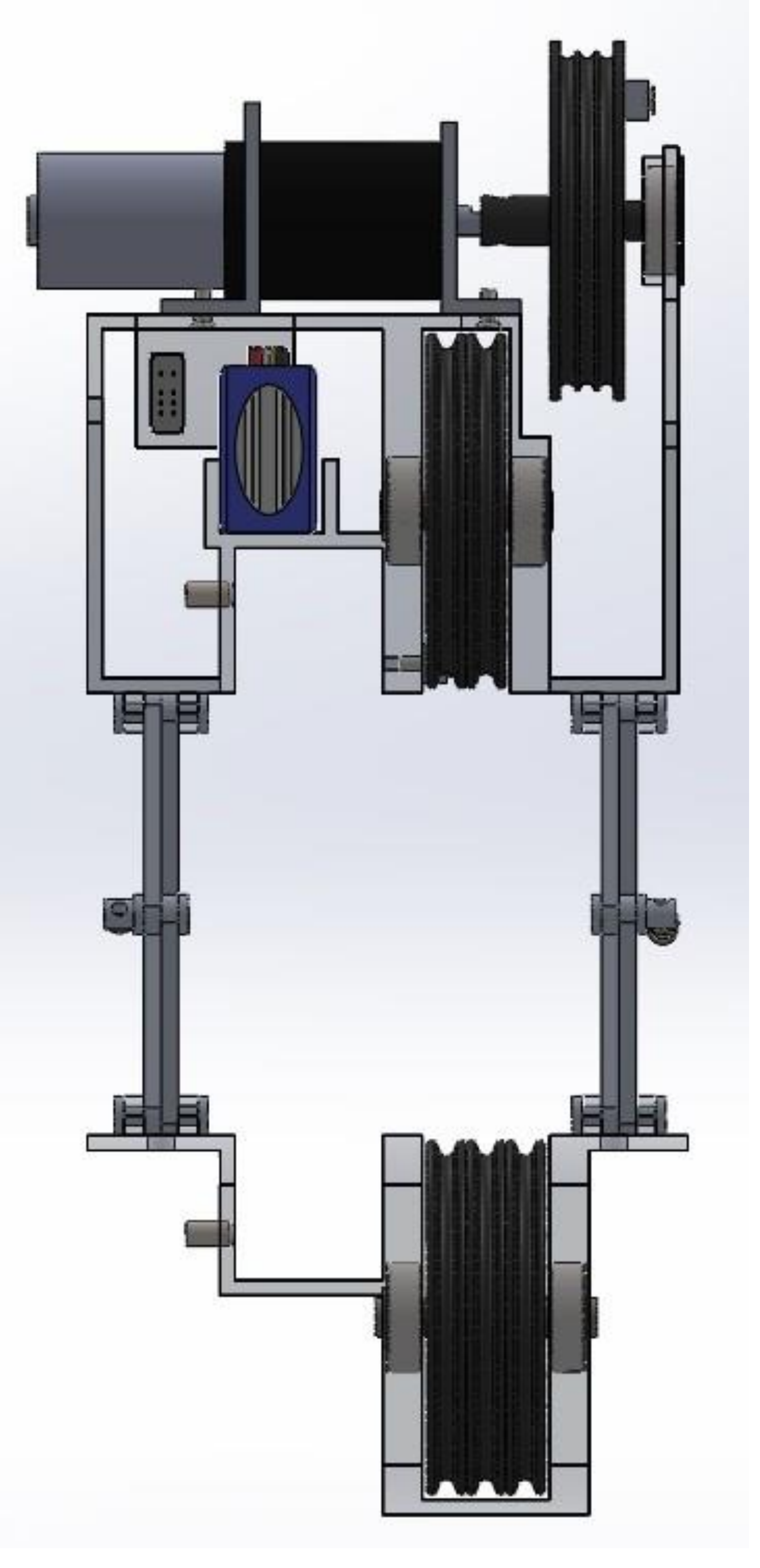
- Proper gripping slot is provided for continuous comfortable use of the tool for long time.
- Due to low friction, self weight, spring force, no effort is needed for resetting the tool after each crimping action.
- The switch is easily accessible with the thumb and smooth actuation is possible.
- During design procedure, attention has been paid to ensure easy dismantling and replacement of different replaceable parts without damaging any other component.

5. Sheets & Models

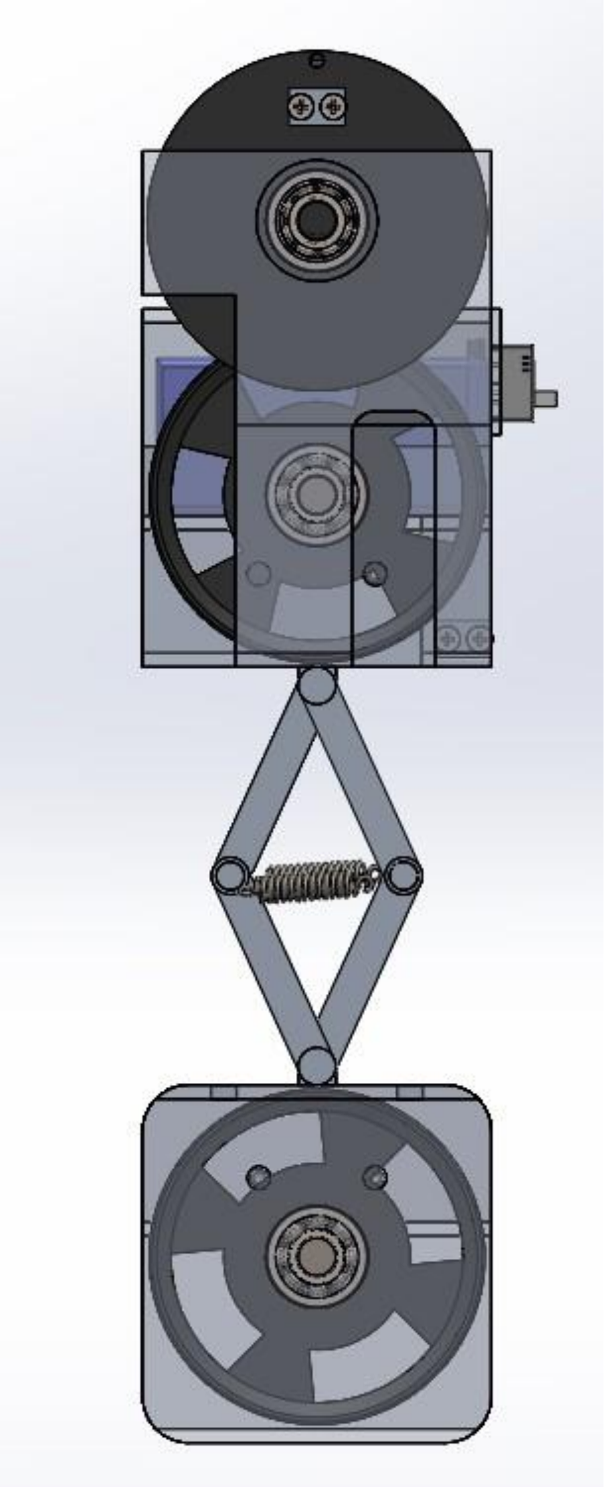
5.1 All Views of model
5.2 Exploded View with BOM
5.3 General Drawing Assembly
5.4 Detailed Drawing for important components

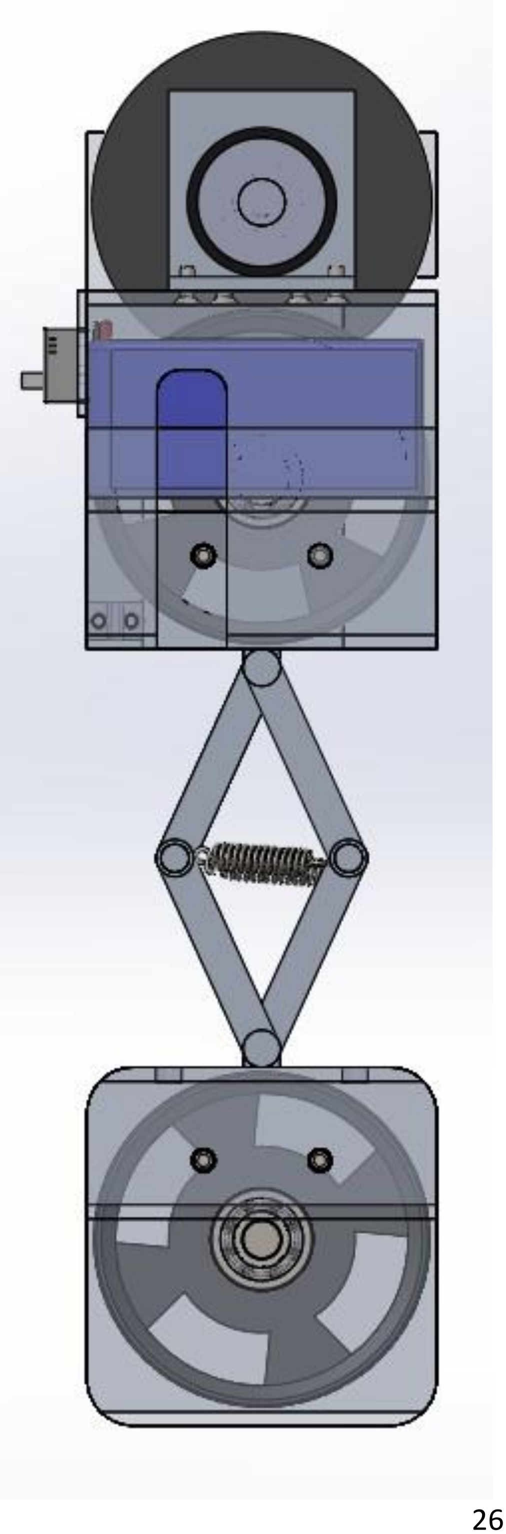
5.1 All Views

5.1.1 Front View

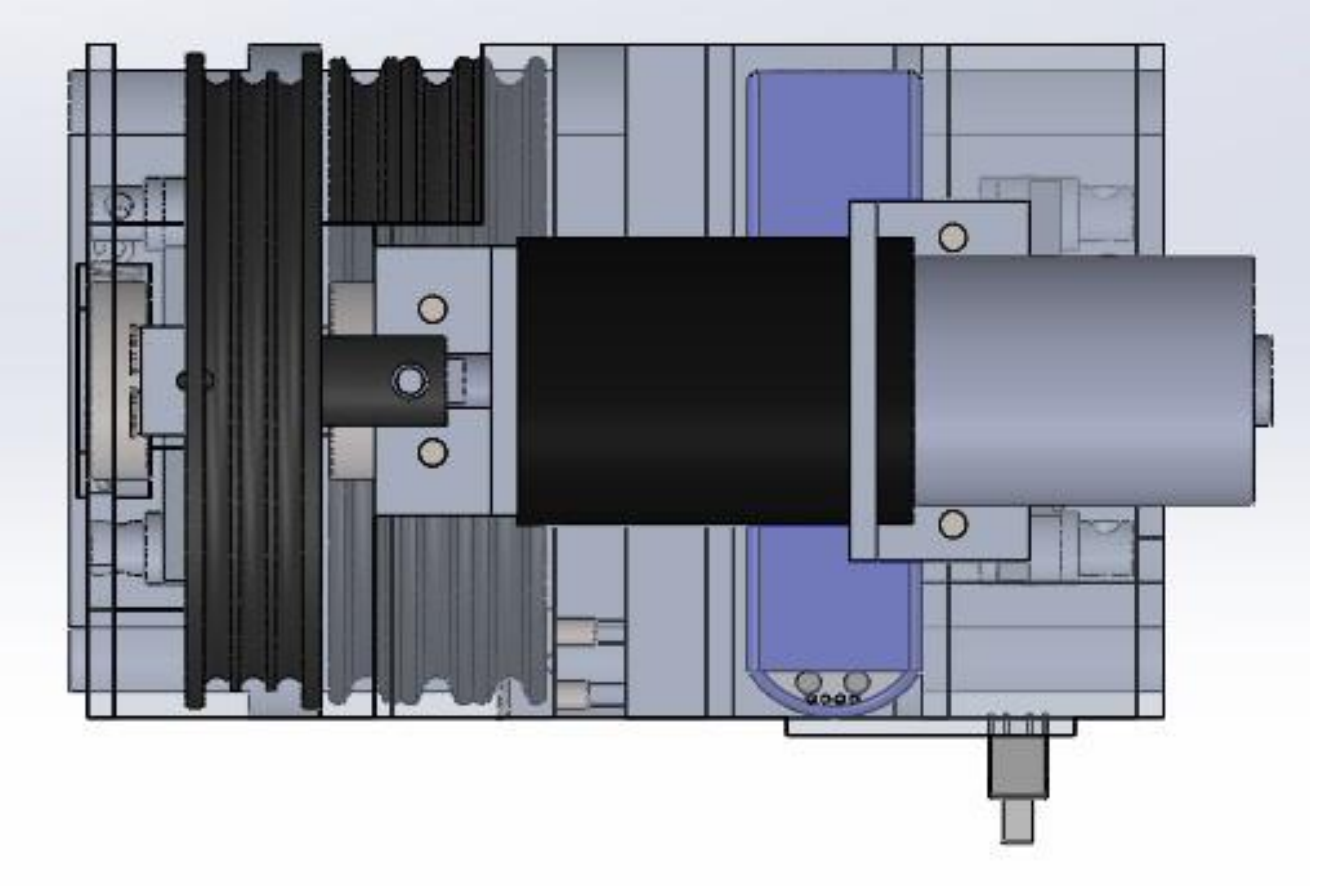


5.1.2 Side Views

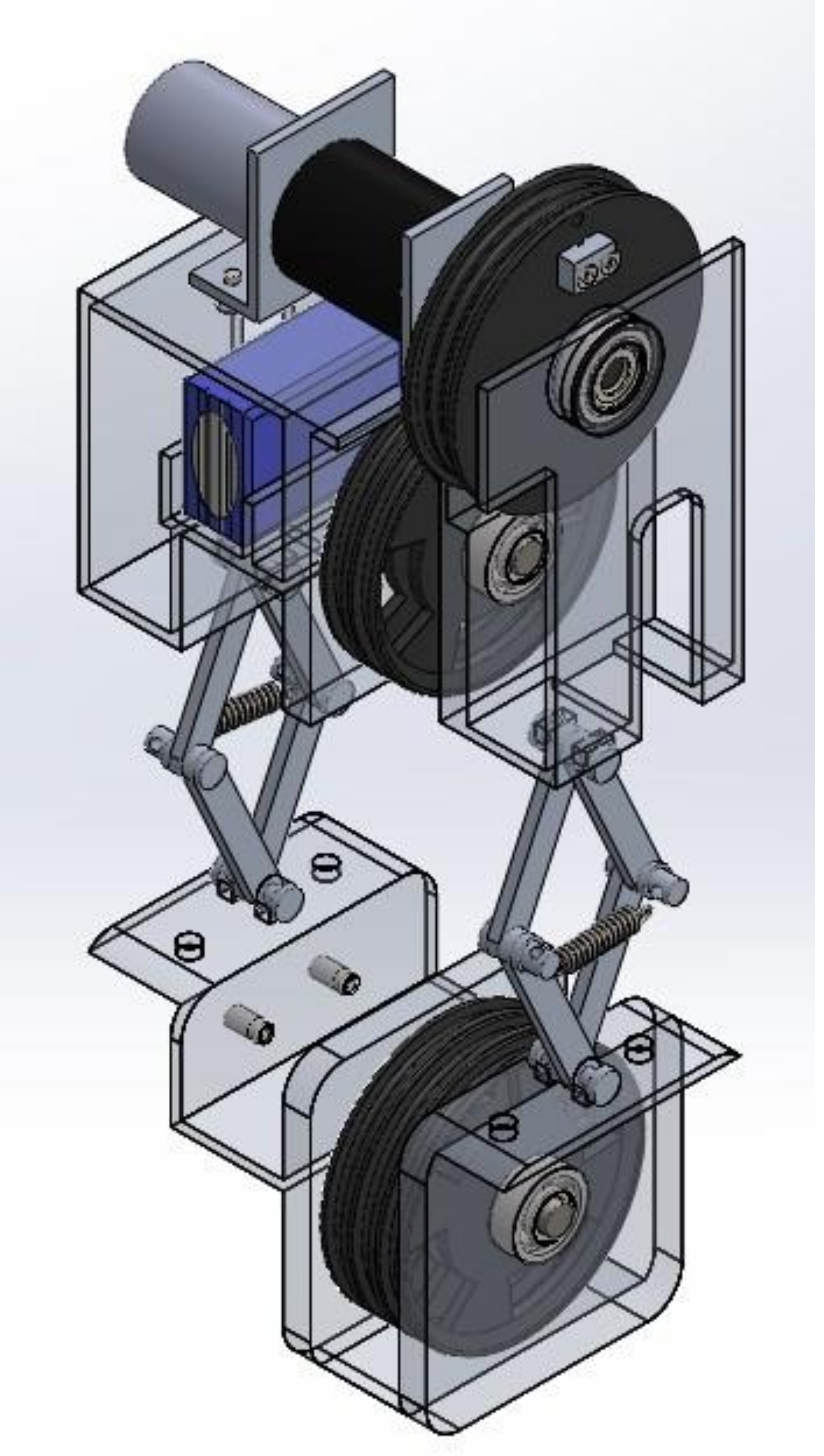




5.1.3 Top View

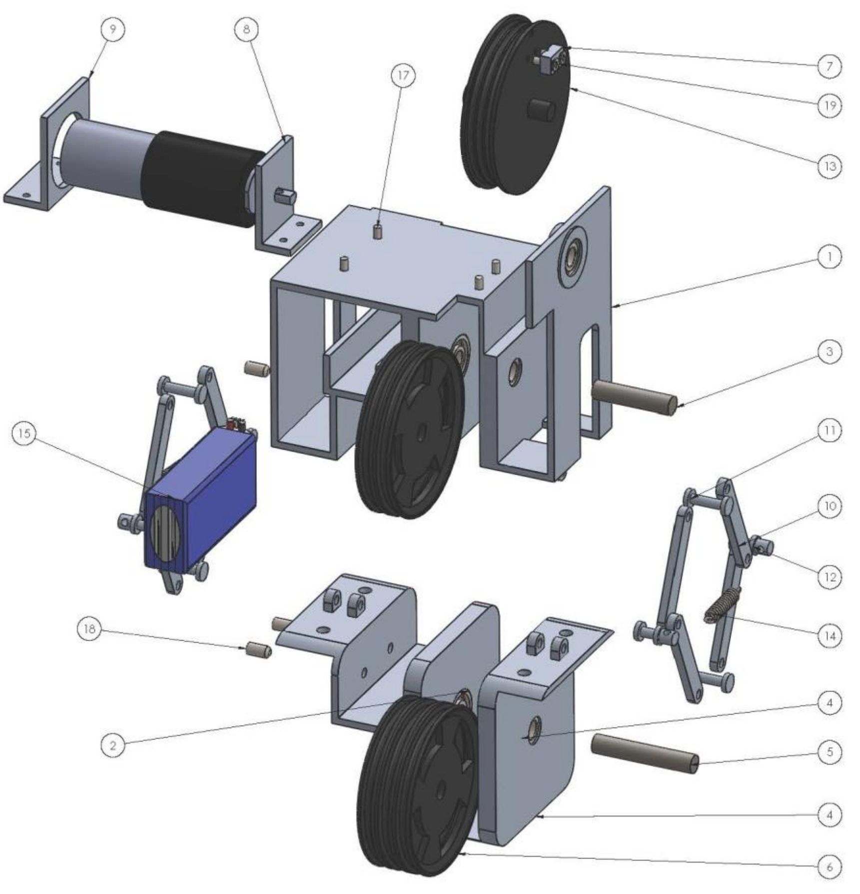


5.1.4 Isometric View



5.2 Exploded View with BOM

- [Exploded view PDF.]
- [Exploded View animation.]
- [STEP File for Assembly]



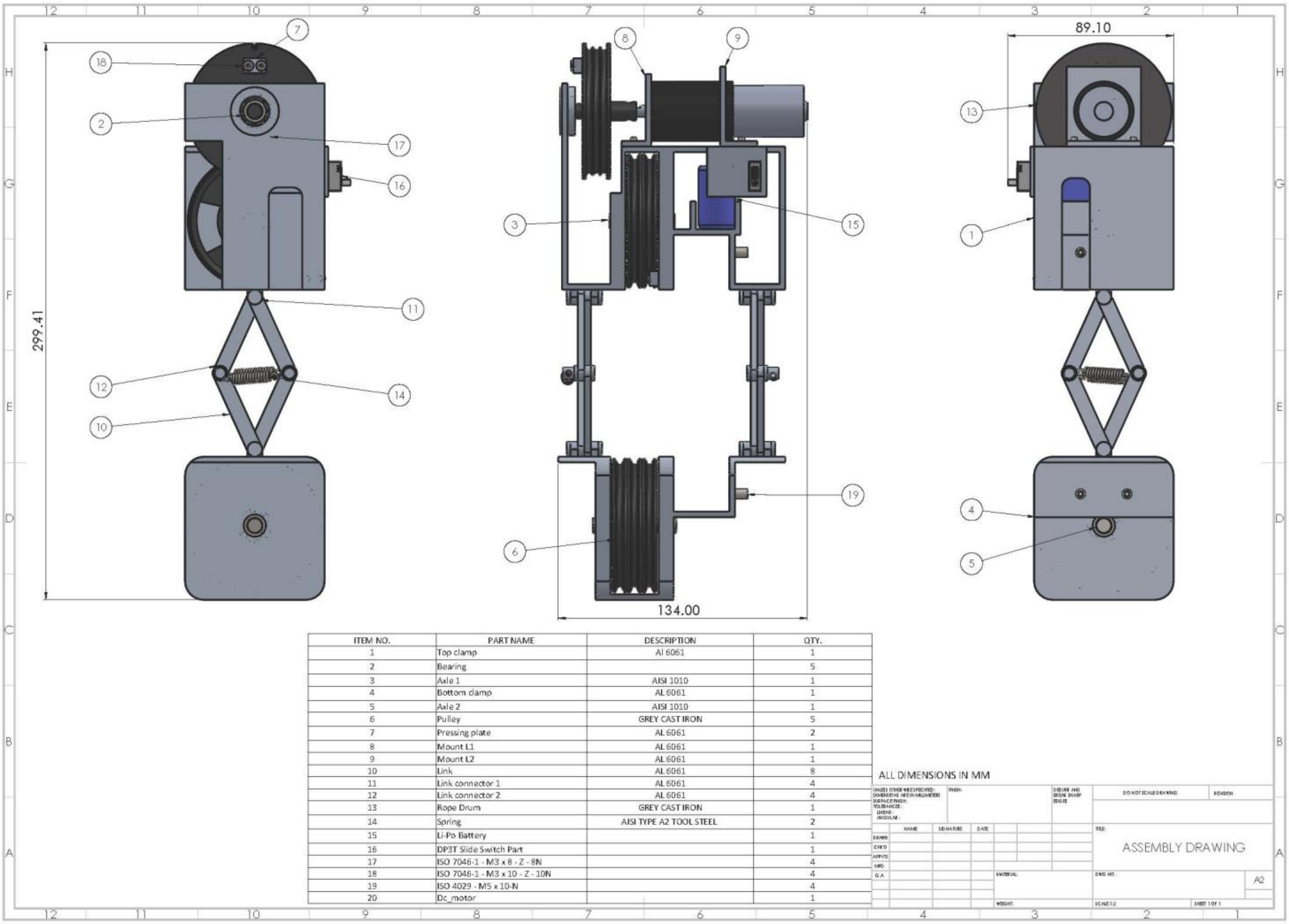
ITEM NO.	PART NUMBER	Material	QTY.
1	Top clamp	Al 6061	1
2	Bearing		5
3	Axle 1	AISI 1010	1
4	Bottom clamp	AL 6061	1
5	Axle 2	AISI 1010	1
6	Pulley	GREY CAST IRON	5
7	Pressing plate	AL 6061	2
8	Mount L1	AL 6061	1
9	Mount L2	AL 6061	1
10	Link	AL 6061	8
11	Link connector 1	AL 6061	4
12	Link connector 2	AL 6061	4
13	Rope Drum	GREY CAST IRON	1
14	Spring	AISI TYPE A2 TOOL STEEL	2
15	Li-Po Battery		1
16	DP3T Slide Switch Part		1
17	ISO 7046-1 - M3 x 8 - Z - 8N		4
18	ISO 4029 - M5 x 10-N		4
19	ISO 7046-1 - M3 x 10 - Z - 10N		4
20	Dc_motor		1

Complete assembly
Exploded view and BOM

A2

5.3 General Drawing Assembly

- [General Assembly Drawing PDF.]()

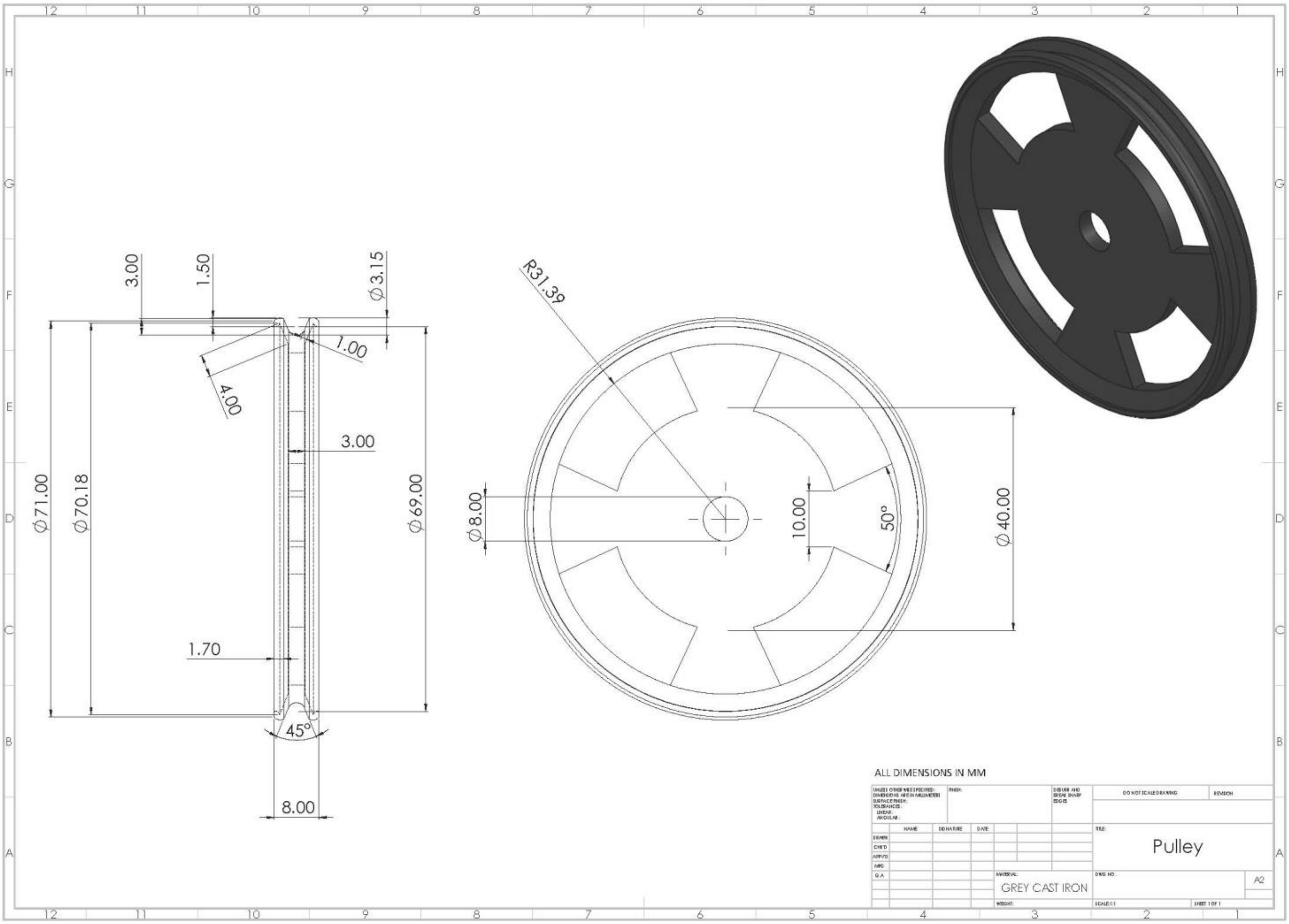


ITEM NO.	PART NAME	DESCRIPTION	QTY.
1	Top clamp	Al 6061	1
2	Bearing		5
3	Axle 1	AISI 1010	1
4	Bottom clamp	AL 6061	1
5	Axle 2	AISI 1010	1
6	Pulley	GREY CAST IRON	5
7	Pressing plate	AL 6061	2
8	Mount L1	AL 6061	1
9	Mount L2	AL 6061	1
10	Link	AL 6061	8
11	Link connector 1	AL 6061	4
12	Link connector 2	AL 6061	4
13	Rope Drum	GREY CAST IRON	1
14	Spring	AISI TYPE A2 TOOL STEEL	2
15	Li-Po Battery		1
16	DP3T Slide Switch Part		1
17	ISO 7046-1 - M3 x 8 - Z - 8N		4
18	ISO 7046-1 - M3 x 10 - Z - 10N		4
19	ISO 4029 - M5 x 10-N		4
20	Dc_motor		1

5.4 Detailed Drawing for important components

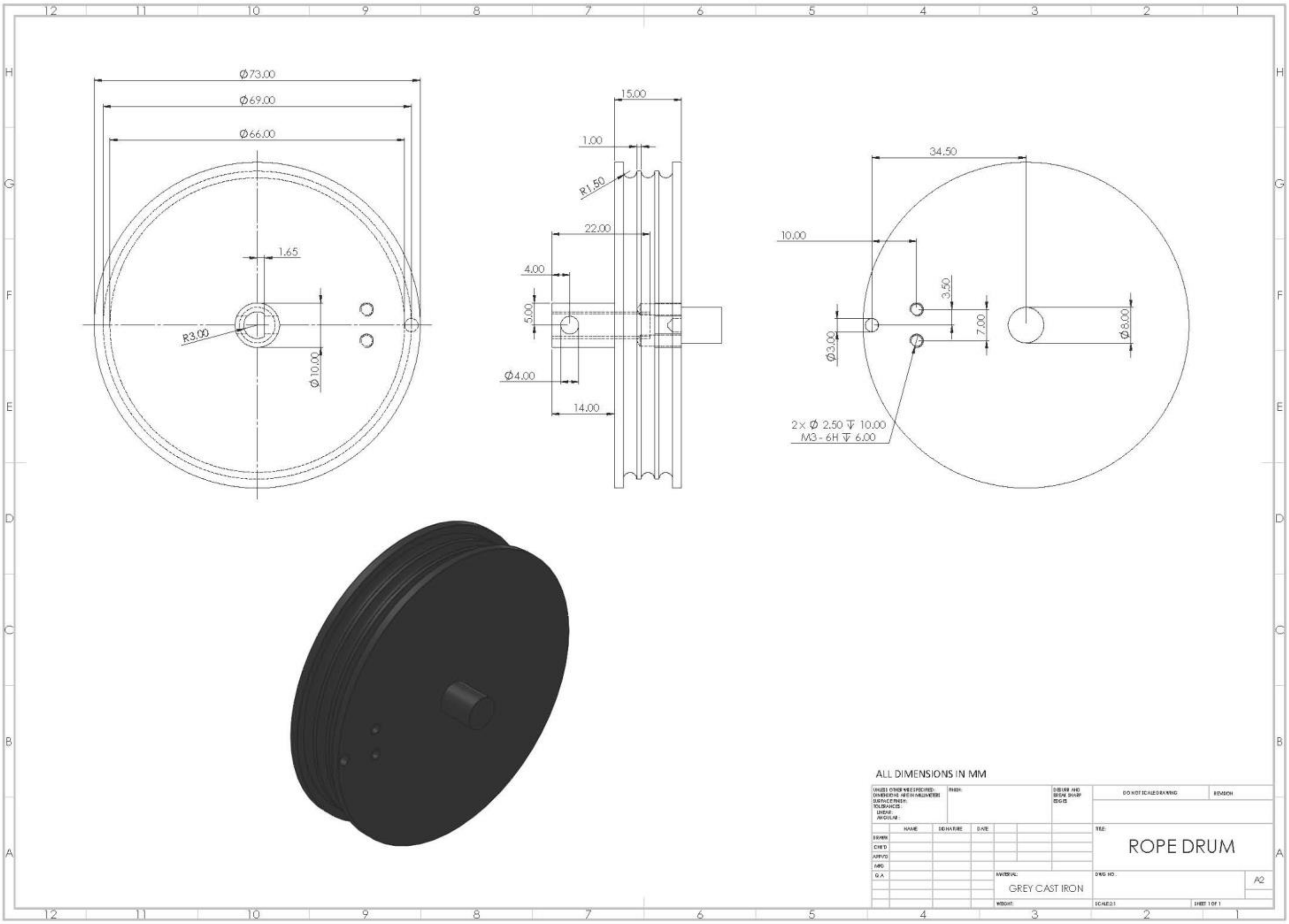
5.4.1 Pulley (Sheave)

- Pulley Drawing PDF.



5.4.2 Rope Drum

- [Rope Drum Drawing PDF.]



5.4.3 Top Clamp

- [Top Clamp Drawing PDF.]