Degree Thesis

3D Printing of Automobile Power Transmission System Using Tough PLA

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Degree programme: Materials Processing Technology

Date: 16/04/2020

DEGREE THESIS	
Arcada	
Degree Programme:	Materials Processing Technology
Identification Number:	**18993**
Author:	Suman Thapa
Title:	3D Printing of Automobile Power Transmission System Using Tough PLA
Supervisor (Arcada):	Mathew Vihtonen
Commissioned By:	Rene Herrmann

Abstract:

The objective of this thesis project is to design and 3D print a functional automobile power transmission system using a relatively new material called Tough PLA. For simplicity, only spur gears and bevel gears are used in the design. All the parts are designed using SolidWorks 2019 and printed using MakerBot Replicator 5th generation printers. SolidWorks animation and motion analysis are used to create animation and analyse the the torque and angular velocity of each gears. For energy source, a DC motor is used that transfers the rotational motion to all the gears. The results obtained form the motion analysis are very accurate to theoretical calculations. But, the experimental angular velocities are a bit lower. This might be due to uneven cooling of the printed parts that led to lack of clearance and friction between some gears. Experimental verification of angular velocies are done using iPhone slow motion camera.

For testing of material properties, tensile test and flexural test are conducted. The tensile test result shows, the ultimate tensile strength of normal PLA and tough PLA are 55 MPa and 30 MPa respectively and the Young's modulus are 1.69 GPa and 1.23 GPa respectively. The flexural test result shows, the ultimate flexural strength of normal PLA and tough PLA are 95 MPa and 52 MPa respectively and the flexural modulus are 2.97 GPa and 2.07 GPa respectively. It is clear that normal PLA can handle large amount of load before break compared to tough PLA. But when it fails, it breaks completely, without significant plastic deformation, showing characteristic brittle nature. Whereas, tough PLA undergoes a very long plastic deformation before it breaks. Both the tests suggest that tough PLA has suitable combination of ultimate tensile strength and ductility and can absorb much more amount energy before failure. So, tough PLA can be a material of choice for printing gears.

Keywords:	Fundamental Law of Gearing, Spur Gear, Bevel Gear, Angular Velocity, Manual Transmission System, Differential, DC Motor, Tensile Strength, Flexural Strength.
Number of Pages:	
Language:	English
Date of Acceptance:	

Table of Contents

List of Tables

List of Symbols

ω	Angular Velocity
m_V	Angular Velocity Ratio
r	Pitch Radius
D	Pitch Circle Diameters
T	Number of Teeth
P_C	Circular Pitch of Gear
P_d	Diametral Pitch
m	Module
m_V'	Transmission Ratio
W_N	Force Acted by Driving Pinion on Driven Gear
M	Maximum Bending Moment at the Critical Section of Teeth
t	Thickness of Teeth
y	Half of Thickness of Teeth at Critical Section
b	Width of Teeth
I	Moment if Inertia
σ_W	Maximum Value of Bending Stress occurring at the Critical Section of Teeth
γ	Pitch Angle of a Bevel Gear
A_0	Cone Distance of a Bevel Gear
α	Addendum Angle
δ	Dedendum Angle
r_b	Back Cone Radius of a Bevel Gear
P_t	Tangential Component of Force P
P_r	Radial Component of Force P
P_a	Axial or Thrust Component of Force P
τ	Torque Transferred by a Gear
θ	Angular Displacement
N	Number of Revolutions Per Minute
B	Magnetic Field
I	Current

V	Applied Voltage
ϕ	Magnetic Flux Per Pole
R_a	Armature Resistance
I_a	Armature Current
ρ	Resistivity of Conductor
Y	Young’s Modulus

FOREWORD

I would like to express my immense gratitude to my supervisor Mr. Mathew Vihtonen for his excellent guidance throughout this thesis. I am also grateful for the support provided by Mr. Silas Gebrehiwot and Harri Anukka for their technical guidance in material testing. I truly appreciate the advices and support provided by Mr. Tobias Jansson who has been very much supportive throughout the lab experiments.

Finally, I am thankful to the entire Materials Processing Technology department of Arcada, my friends and my family members for supporting me throughout my studies and during the completion of my thesis.

1. INTRODUCTION

1.1. Background

The paradigm of manufacturing is changing. With the integration of new processes, new materials and new innovations, the pace at which the modern manufacturing is changing our world is magnificent. Amongst all these, “3D Printing”, is one of the most astonishing example of technological advancement that mankind has ever achieved and can be considered as a pioneer of modern manufacturing. This technology has already started to change the way we used to produce objects and has been emerging as a new tool that could be used for shaping products. 3D printing can be used to print tools and toys, clothing’s, body parts and complex structures of almost any shape or geometry that any other processes fails to manufacture. Moreover, 3D printing is not just limited to plastics, already, we can print glass, metals, semi-solid foods, organs, etc.

Gears are complicated and surprisingly useful piece of tools that work by meshing mechanism. Every day we tackle many objects like vehicles, machineries or household utilities that uses some form of gears. Gears form the heart of power transmission systems. They use meshing teeth for the propagation of rotational energy. Gears are fascinating things to 3D print. Though plastic gears may not be durable and might not be seen in high performance devices, but they might be an excellent alternative for making prototypes. One can use those printed gears to test, plan and reinvent the design.

1.2. Objectives

This thesis aims to investigate various mechanical properties and calculations associated with 3D printed gears. The study will be supported by using Tough PLA as a raw material for 3D printing and comparing the mechanical properties of tough PLA and normal PLA. The main objectives of this thesis are:

- Designing of automobile power transmission system using SolidWorks software.
- 3D printing of gears using MakerBot Replicator printers.
- Analysing the feasibility of gear system by connecting the gears with DC motor and meshing the gears.

- Verifying the fundamental law of gearing and calculating various parameters like gear ratio, angular velocities, torque ratio or mechanical advantage and power transferred by the gears.
- Comparing the tensile and flexural strength of tough PLA and normal PLA.

1.3. Compliance with the Degree Programme Theme

Material science engineering is study of everything like metals, composites, polymers, biomaterials, etc. at the atomic as well as macroscopic level. Material science helps in making materials lighter, stronger, faster and easier to process and manufacture. It is a field that is always changing.

Themes such as 3D printing of plastic parts and comparing of their mechanical properties are very closely related to Materials Processing. The gears are designed using SolidWorks software, which is one of the most important course studied at Arcada UAS. Understanding the principles behind the gear operations requires in-depth knowledge of Machine Design, Mechanics, Strength of Materials and Manufacturing Processes.

1.4. Relevance to the Existing Knowledge

Studies in this thesis are not just related to 3D printing but also surrounds wide range of mechanical knowledge about gears, manual transmission system, working principles of DC motors and potentiometers and material strengths. In present era, it is very rare to find some mechanical devices or machines that don’t use some form of gears. So, there are lots of research papers published which focuses on working mechanism of gears or 3D printing of gears.

A similar thesis was published by Mingchuan (2014) regarding Automobile Transmission design wherein the author design’s five-speed transmission using SolidWorks. Here the author mainly focuses and explains in detail about the gear changing mechanism using locking ring synchronizers.

In one of the studies carried out by Vaibhav S.Jadhav and Santosh R. Wankhade (2017), authors have compared the feasibility regarding the cost of 3D printed gears with that of injection molded gears. The results showed that 3D printing method was much more cost efficient and time efficient than injection molding. Similarly, in a research carried out to study the effects of FDM processing parameters in printing of ABS plastics (N.Mohammed Raffic, Dr. K.Ganesh babu, 2017), it was found that parameters like Filament Thickness, Layer Thickness, Infill Density, Raster Width, Part orientations, etc. have direct effect on the printing cost, print time and strength of the part.

By reviewing the above studies, this thesis tries to connect the ideas of gear manufacturing, power transmission and 3D printing in unification. It also explains the workig principle behind DC motors and potentiometers. Moreover, the gears are printed using tough PLA, significantly a new material for 3D printing industries. The findings of this thesis will help students to understand the working principle behind automobile power transmission system and also help production engineers to make a wise choice between normal and tough PLA depending upon the product requirement.

2. LITERATURE REVIEW

2.1. Gear Design

“Gears are defined as toothed wheels or multilobed cams, which transmit power and motion from one shaft to another by the means of successive engagement of teeth.” (V B Bhandari , 2006).

During transfer of motion between two tangentially connected smooth shafts, slipping usually takes place. This reduces the velocity ratio and power transmission between the shafts. Addition of meshing teeth helps to overcome slip between belt and shaft thus maintaining positive drive in the system. “The motion and power transmitted between gears can be considered kinematically equivalent to that transmitted between frictional wheels.” (R.S. Khurmi, 2005).

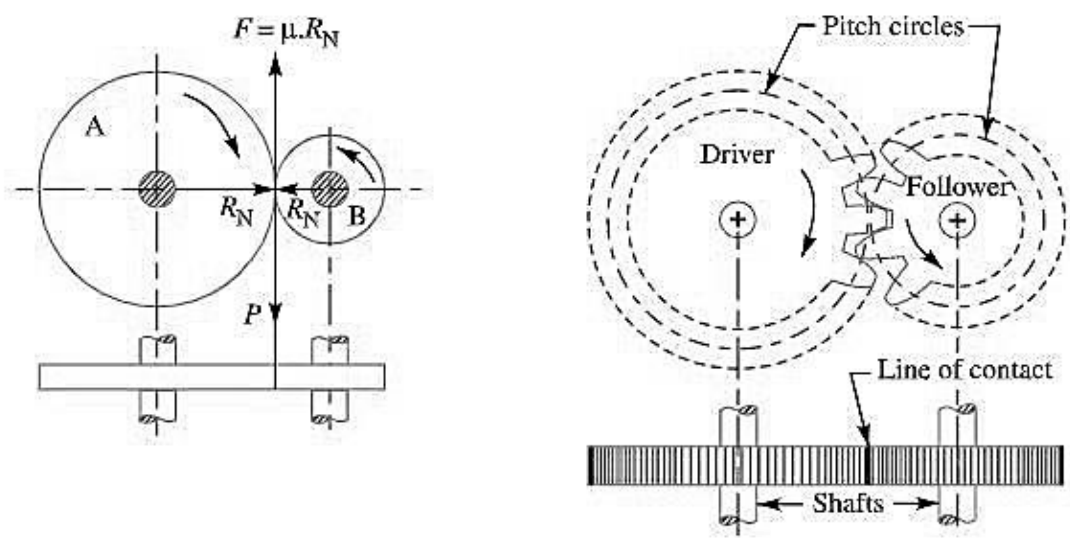

Figure 1, Friction Wheels and Gear or Tooth Wheels (R.S. Khurmi, 2005, p. 1022)

Consider two wheels (A and B) with sufficient surface roughness as shown in Figure. 1. Both the wheels are mounted on shafts such that they press against each other. When the wheel A is rotated, it rotates wheel B but in opposite direction. “The rotation between wheels are maintained until the tangential force (P) exerted by wheel A does not exceeds the maximum frictional resistance (F) between the two wheels.” (R.S. Khurmi, 2005). But the slipping occurs if tangential force is gearter than the frictional resistance.

One of the great ideas to overcome problem of slipping was to introduce number of teeth like projections on the boundary of those wheels. They then become gears. Conventionally, smaller of these two is called as pinion and the other is simply called as gear.

Gear drives offer following advantages over chain, rope or belt drives (V B Bhandari , 2006):

- It maintains constant velocity ratio in the system.
- It allows to transfer much larger amount of power compared to belt or chain drives.
- Motion at very low speed can be achieved.
- "The efficiency of the gear drives is very high, even up to 99 percent in case of spur gear." (V B Bhandari , 2006).
- Gear shifting is possible by using different arrangements of gears. This helps to achieve different combinations of velocity ratio.

2.2. Classification of Gears

Gears can be classified as follows:

1. ***According to the position of axes of the shafts:*** On the basis of position of axes of the shaft, gears can be classified as:

   (a) Parallel, (b) Intersecting, and (c) Non-intersecting and non-parallel

   "When the gears have teeth parallel to the axis of the wheel, these gears are called spur gear. In spur gearing, two parallel and co-planer shafts are connected by gears." (R.S. Khurmi, 2005). The modification of spur gear with its teeth inclined or at an angle to the axis is called as helical gear. Helical gears produce low noise and are more efficient than spur gear (R.S. Khurmi, 2005).

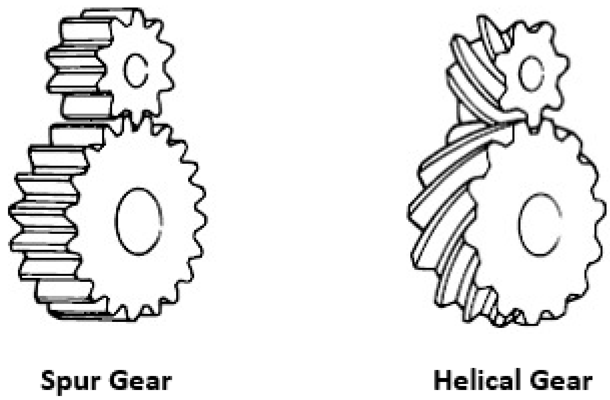

Figure 2, Spur Gear and Helical Gear (Kohara Gear Industry Co.,Ltd., 2015)

“When two intersecting, but co-planer shafts are connected by gears, these gears are called bevel gears.” (R.S. Khurmi, 2005). The size of gear tooth, including the thickness and height, decreases towards the apex of the cone. Bevel gears are normally used for the shafts which are right angle to each other. “Bevel gears with their teeth inclined to the face of the bevel are called as helical bevel gears.” (R.S. Khurmi, 2005)

Figure 3, Bevel Gear (Kohara Gear Industry Co.,Ltd., 2015)

“When two non-intersecting and non-parallel shafts are connected by gears, these gears are called spiral gears or worm gears. It consists of a worm and a worm wheel.” (R.S. Khurmi, 2005). The worm is in the form of threaded screw, which meshes with the matching toothed wheel.

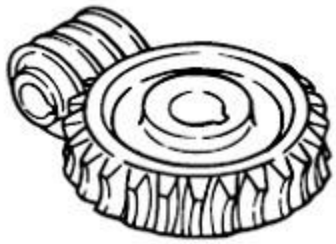

Figure 4, Worm or Spiral Gear (Kohara Gear Industry Co.,Ltd., 2015)

2. ***According to type of gearing:*** On the basis of type of gearing, gears can be classified as;

   (a) External gearing, (b) Internal gearing, and (c) Rack and pinion

   "When the gears of the two shafts mesh externally with each other, the gears are supposed to be in external gearing and when the gears of two shafts mesh internally with each other, the gears are supposed to be in internal gearing." (R.S. Khurmi, 2005).

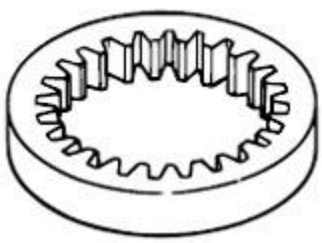

Figure 5, Internal Gear (Kohara Gear Industry Co.,Ltd., 2015)

"In some cases, the shaft meshes externally with the gear in a straight line. Such type of gear is called rack and pinion. The straight-lined gear is called rack and the circular wheel is called pinion." (R.S. Khurmi, 2005)

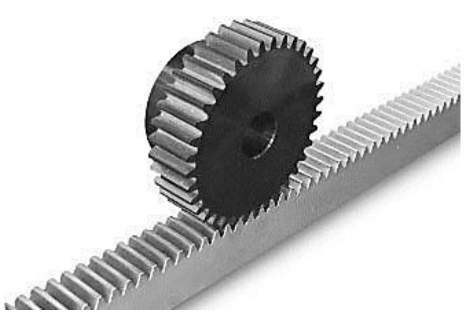

Figure 6, Rack and Pinion (Kohara Gear Industry Co.,Ltd., 2015)

Selection of proper type of gear is always a vital decision while designing a particular gear for a given application. Various factors like general layout of shafts, velocity ratio, input speed, speed reduction, power to be transmitted, costs, etc. should be considered during gear selection process. In general, spur and helical gears are used for parallel shafts. For shafts intersecting at right angle, bevel gears can be used. For perpendicular and non-intersecting shafts, worm gears can be used. (V B Bhandari , 2006)

Although modern transmission system might consist of various complicated systems of gears, this thesis is mainly focused on designing of a simple transmission system using spur and bevel gears.

2.3. Fundamental Law of Gearing

It is very important for two meshing gears to have constant angular velocity ratio throughout the overall motion of the gears. Fundamental law of gearing states that, “In order to obtain a constant velocity ratio, the common normal to the tooth profile at a point of contact should always pass through a fixed point called the pitch point.” (V B Bhandari , 2006).

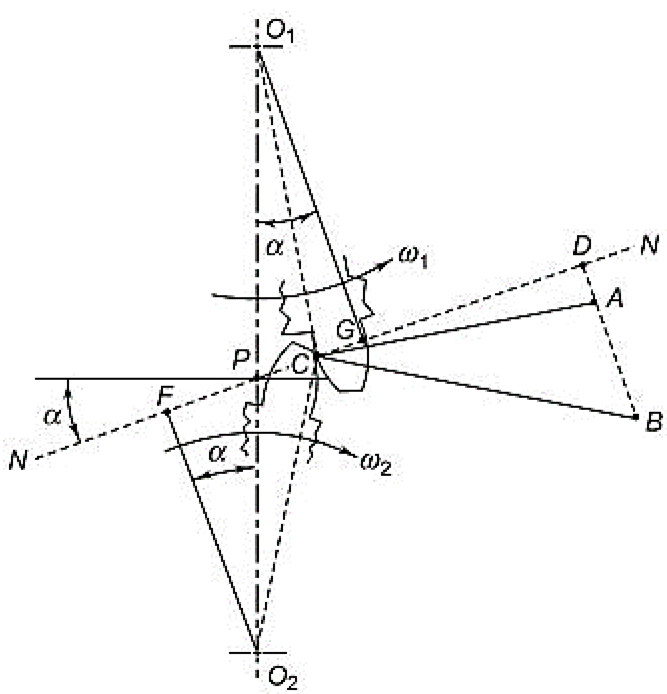

Figure 7, Law of Gearing (V B Bhandari , 2006, p. 649)

Consider the section of the two meshing teeth as shown in the figure above. Let, C be the point of contact between two teeth and the wheels are rotating in the direction as shown in the figure. Consider a line NN which is a common normal at the point of contact C. O_1 and O_2 are the center of two gears rotating with the angular velocities ω_1 and ω_2 respectively.

Let, $\overrightarrow{CA}$ and $\overrightarrow{CB}$ be the velocities of the point C on wheel 1 and 2 respectively.

Also, $O_1C \perp CA$ and $O_2C \perp CB$

Also, for the teeth to remain in mesh, $\overrightarrow{CD}$ which is the projection of two vectors $\overrightarrow{CA}$ and $\overrightarrow{CB}$ along the common normal NN must be equal.

$$\overrightarrow{CA} = \omega_1 \times O_1C$$

$$\overrightarrow{CB} = \omega_2 \times O_2C$$

$$\frac{\omega_1}{\omega_2} = \frac{O_2C}{O_1C} = \frac{\overrightarrow{CA}}{\overrightarrow{CB}} \tag{1}$$

From similar triangle, ΔO_1CG and ΔCAD,

$$\frac{O_1C}{CA} = \frac{O_1G}{CD} \tag{2}$$

From similar triangle, ΔO_2FC and ΔCBD,

$$\frac{O_2C}{CB} = \frac{O_2F}{CD} \tag{3}$$

From equations 2 and 3,

$$\frac{CA}{CB} = \frac{O_1C}{O_2C} \times \frac{O_2F}{O_1G} \tag{4}$$

From equations 1 and 4,

$$\frac{\omega_1}{\omega_2} = \frac{O_2F}{O_1G} \tag{5}$$

Similarly, from similar triangle, ΔO_2FP and ΔO_1GP,

$$\frac{O_2F}{O_1G} = \frac{O_2P}{O_1P} \tag{6}$$

From equations 5 and 6,

$$\frac{\omega_1}{\omega_2} = \frac{O_2P}{O_1P} \tag{7}$$

Also,

$$O_1P + O_2P = O_1O_2 = constant \tag{8}$$

Equation 7 shows that, the angular velocity ratio is inversely proportional to the ratio of distance of P from the center O_1 and O_2 i.e. pitch radius. This point P is called pitch point. Also, "the angular velocity ratio m_V is equal to the ratio of the pitch radius of the input gear to that of the output gear." (V B Bhandari , 2006)

$$m_V = \frac{\omega_{out}}{\omega_{in}} = \pm\frac{r_{in}}{r_{out}} \tag{9}$$

Now, if two gears are considered with pitch circle diameters D_1 and D_2 and number of teeth T_1 and T_2 respectively, then the velocity ratio can be calculated as,

$$\frac{\omega_2}{\omega_1} = \frac{D_1}{D_2} = \frac{T_1}{T_2} \qquad \text{(V B Bhandari , 2006)} \tag{10}$$

2.4. Spur Gear

2.4.1. Terminology of Spur Gear

Terms used in gears includes many peculiar definitions. The picture below indicates some of the parametric terms that are used to define the properties associated with spur gears.

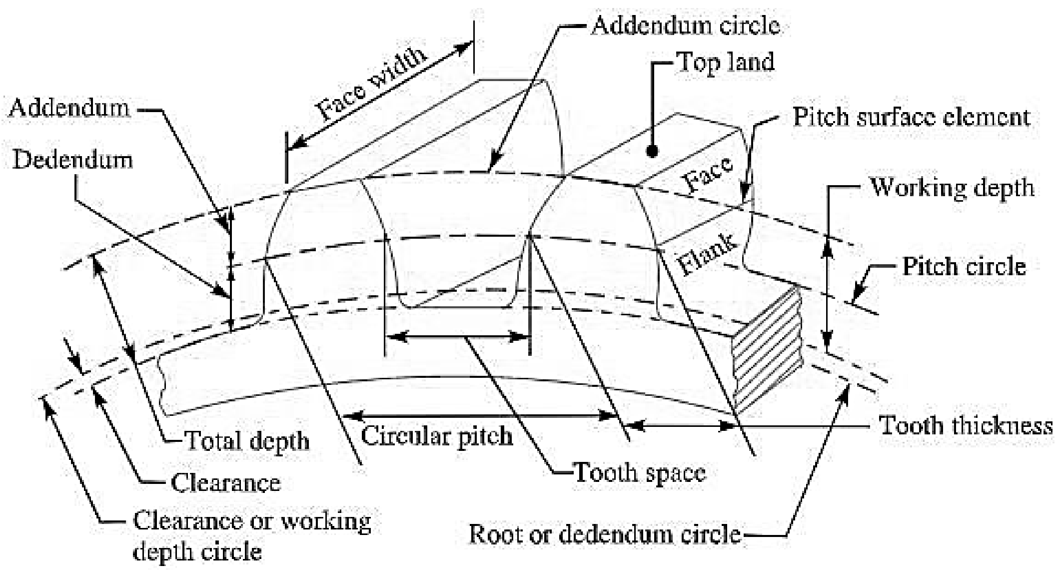

Figure 8, Terminology of Spur Gear (R.S. Khurmi, 2005, p. 1025)

Some of the terminology associated with spur gears are mentioned below:

1. Pitch Circle: It is an imaginary circle that rolls to represent the motion of actual gear. It is tangent with the pitch circle of a mating gear (R.S. Khurmi, 2005).
2. Pitch Circle Diameter (D): “The pitch circle diameter or pitch diameter is the diameter of a pitch circle. The size of a gear is usually specified by pitch diameter.” (R.S. Khurmi, 2005)
3. Addendum: “It is the radial distance of a tooth from the pitch circle to the top of the tooth.” (R.S. Khurmi, 2005).
4. Dedendum: “It is the radial distance of a tooth from the pitch circle to the bottom of the tooth.” (R.S. Khurmi, 2005).

5. Circular Pitch (P_C): "The circular pitch is the arc length along the pitch circle circumference measured from a point on one tooth to the same point on the next. It defines the tooth size." (R.S. Khurmi, 2005).

$$P_C = \frac{\pi D}{T} \qquad \text{(R. S. Khurmi, 2005)}$$

where, $D = pitch\ diameter,\ T = number\ of\ teeth$

6. Diametral Pitch (P_d): "It is the ratio of number of teeth to the pitch circle diameter in millimeters." (R.S. Khurmi, 2005)

$$P_d = \frac{T}{D} = \frac{\pi}{P_C} \qquad \text{(R. S. Khurmi, 2005)}$$

7. Module (m): "A module is inverse of diametral pitch or the ratio of pitch circle diameter in millimeters to the number of teeth." (R.S. Khurmi, 2005). Basically, a module indicates how big or small the gear is. For two gears to mesh, they must have same module.

$$m = \frac{D}{T} \qquad \text{(R. S. Khurmi, 2005)}$$

8. Velocity ratio or speed ratio (m_V): "It is the ratio of angular velocity of the driving gear to the angular velocity of the driven gear." (V B Bhandari , 2006)

$$m_V = \frac{\omega_{out}}{\omega_{in}} = \pm \frac{r_{in}}{r_{out}} \qquad \text{(V B Bhandari , 2006)}$$

9. Transmission ratio (m'_V): "It is the ratio of angular velocity of the first driving gear to the angular velocity of last driven gear in the gear train." (V B Bhandari , 2006)

$$m'_V = \frac{\omega_{first}}{\omega_{last}} \qquad \text{(V B Bhandari , 2006)}$$

10. Clearance: "It is the amount by which the dedendum of a given gear exceeds the addendum of its mating tooth." (V B Bhandari , 2006).

2.4.2. Force Analysis of Spur Gear Teeth – Lewis Equation

During the transmission of powers between the gears, force is exerted by the tooth of driving gear on the meshing tooth of the driven gear. According to Wilfred Lewis (1892), "when the load is being transmitted between the set of gears, it is all given and taken by a single tooth." So, every single tooth should have ability to transmit full load. Figure below shows a driven gear which is acted upon by a force W_N by the driving pinion. According to the fundamental law of gearing, this resultant force W_N should always act along the pressure line at pitch point A. On resolving W_N

into its two components, tangential component W_T acts in horizontal direction and radial component W_R acts vertically downward. “The tangential component induces a bending stress which tends to break the tooth while the radial component induces a compressive stress of relatively small magnitude.” (R.S. Khurmi, 2005). The effect of radial component is very small and can be usually neglected. But the tangential component is useful as it determines the magnitude of torque and eventually the power transmitted by the gear. (R.S. Khurmi, 2005)

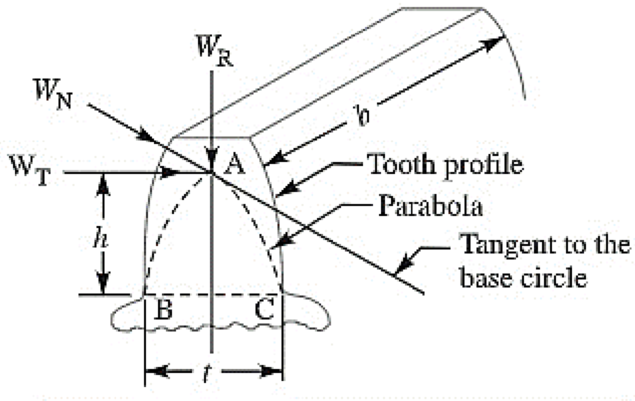

Figure 9, Force Analysis of Tooth of Spur Gear (R.S. Khurmi, 2005, p. 1037)

To calculate the maximum bending stress, a parabola is drawn through pitch point A tangential to the tooth curves at B and C. The stress at all the sections of parabola is the same. “Since, the tooth is larger than the parabola at every section except section BC. Therefore, section BC is considered as the section of maximum stress or the critical section.” (R.S. Khurmi, 2005). The maximum value of the bending stress at this section BC is given by,

$$\sigma_W = \frac{My}{I} \qquad \text{(R. S. Khurmi, 2005)} \qquad (11)$$

Where, σ_W = Maximum value of bending stress at the section BC

M= Maximum bending moment at the critical section $BC = W_T \times h$

y = Half the thickness of the tooth at critical section= $t/2$

I = Moment of inertia about the center line of the tooth= $bt^3/12$

b = width of gear tooth

Substituting the values for M, y and I in equation 11 gives,

$$\sigma_W = \frac{6(W_T \times h)}{bt^2} \qquad (\text{R. S. Khurmi, 2005})$$

$$W_T = \frac{\sigma_W bt^2}{6h} \qquad (12)$$

The above equation 12 is used to calculate the tangential load acting on the tooth when the maximum value of bending stress is provided. This tangential load is also called as the beam strength of the tooth.

2.5. Bevel Gears

2.5.1. Terminology of Bevel Gears

The picture below indicates some of the parametric terms that are used to define the properties associated with bevel gears.

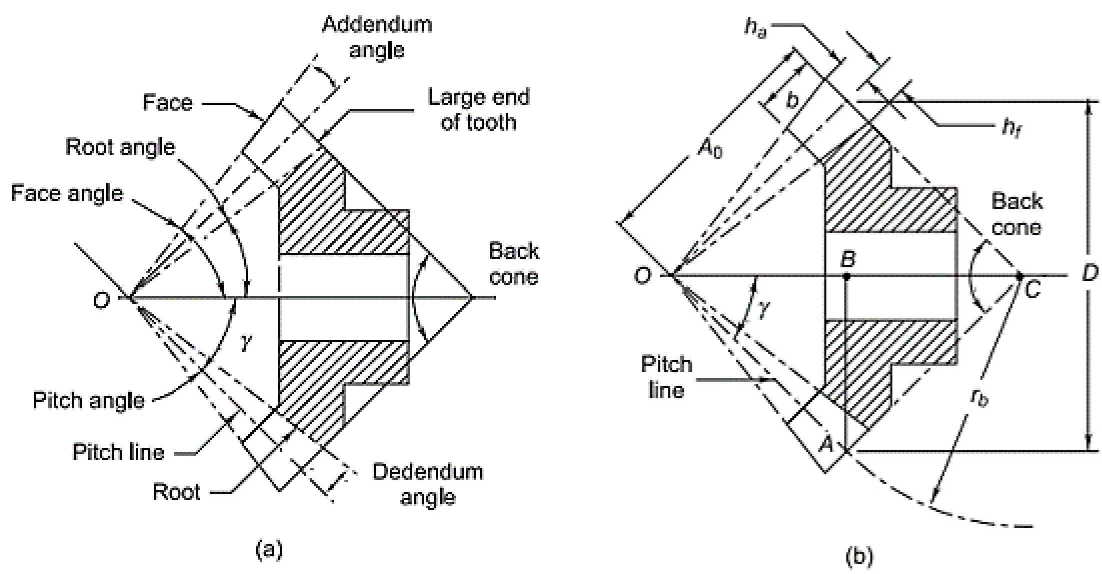

Figure 10, Terminology of Bevel Gears (V B Bhandari , 2006, p. 713)

Following terms are important in the terminology of bevel gears:

1. Pitch Cone: “It is an imaginary cone, the surface of which contains pitch lines of all the teeth in the bevel gear.” (V B Bhandari , 2006).

2. Cone Center: “It is the apex of the pitch cone where the axis of two mating gears intersect with each other.” (R.S. Khurmi, 2005).
3. Pitch Angle: “The angle that the pitch line makes with the axis of the gear is called pitch angle. It is represented by γ in the above figure.” (V B Bhandari , 2006)
4. Cone Distance: “It is the length of the pitch cone element. It is also called as a pitch cone radius.” (V B Bhandari , 2006). It is represented by the length A_0 in the above figure.
5. Addendum angle: “It is the angle subtended by addendum at the cone center. It is denoted by α.” (V B Bhandari , 2006).
6. Dedendum angle: “It is the angle subtended by dedendum of the tooth at the cone center. It is denoted by δ.” (V B Bhandari , 2006).
7. Face Angle: “It is the angle subtended by the face of the tooth at the cone center”. (V B Bhandari , 2006).

$$Face\ angle = (pitch\ angle + addendum\ angle) = (\gamma + \alpha)$$

8. Root angle: “It is the angle subtended by the root of the tooth at the cone center.” (V B Bhandari , 2006).

$$Root\ angle = (pitch\ angle - dedendum\ angle) = (\gamma - \delta)$$

9. Back Cone: “It is an imaginary cone, perpendicular to the pitch cone at the end of the tooth.” (V B Bhandari , 2006)
10. Back Cone Distance or Back Cone Radius: “It is the length of the back cone. It is represented by r_b in the above figure.” (V B Bhandari , 2006).

2.5.2. Force Analysis of Bevel Gear Teeth

"In force analysis of bevel gear, the resultant force between pair of two meshing teeth is supposed to be concentrated at the midpoint along the face width of the tooth." (V B Bhandari , 2006)

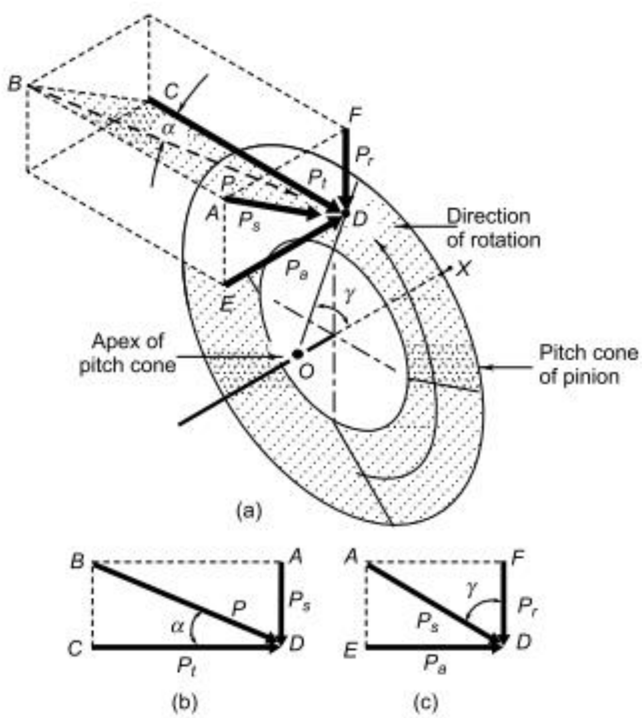

Figure 11, Components of Tooth Face of Bevel Gear (V B Bhandari , 2006, p. 716)

Consider a figure as shown above, where the resultant force P acts at the midpoint D of the face width of the pinion. Also, γ be the pitch angle of the pinion. The resultant force has three components:

$$P_t = tangential\ componnt\ (N)$$

$$P_r = radial\ component\ (N)$$

$$P_a = axial\ or\ thrust\ component\ (N)$$

To calculate the resultant force at D, a shaded plane $ABCD$ as shown in the figure. 11(a) is considered. The same plane is represented in figure. 11(b). Now, from the triangle BCD,

$$\tan\alpha = \frac{BC}{CD} = \frac{P_s}{P_t} \qquad \text{(V B Bhandari , 2006)}$$

$$P_s = P_t \tan\alpha \tag{13}$$

Where, $P_S = Separating\ component\ (N)$ and $\alpha = Pressure\ angle\ in\ degrees$

Now, from the geometry of figure. 11(a), $AD \perp OD$ and $FD \perp OX$. These are two pairs of perpendicular lines. Thus, the angle between OD and OX must be equal to the angle between AD and FD. Therefore, $\angle ADF = \angle XOD = \gamma$.

Consider the plane $DEAF$ as shown in figure. 11(c). Now, from triangle ADF,

$$P_r = P_s \cos\gamma$$

$$P_a = P_s \sin\gamma$$

Substituting the value of P_s from equation 13, above equations become,

$$P_r = P_t \tan\alpha \cos\gamma \tag{14}$$

$$P_a = P_t \tan\alpha \sin\gamma \tag{15}$$

Also, Torque transmitted by the gears, $\tau_t(Nmm) = P_t \times r_m$

Where, $r_m = radius\ of\ the\ pinion\ at\ midpoint\ along\ the\ face\ width\ (mm)$

$$P_t = \frac{\tau_t}{r_m} \qquad \text{(V B Bhandari , 2006)} \tag{16}$$

Equations 14, 15 and 16 are used to determine the components of tooth force on the pinion. The component of tooth force acting on the gear can be determined by considering actions and reactions as equal and opposite.

To calculate r_m, a two-dimensional representation of force components is illustrated below.

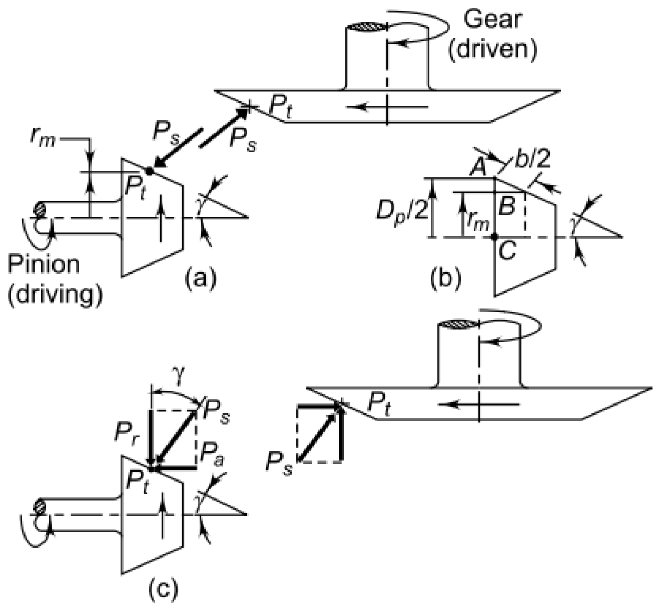

Figure 12, Force Analysis of Bevel Gear (V B Bhandari , 2006, p. 717)

In the above figure, the force that is perpendicular to the plane of paper and towards the observer is represented by $(\bullet)$ sign and the force that is perpendicular to the plane of paper and away from the observer is represented by $(\times)$ sign.

From, figure. 12(b), the mean radius r_m where the resultant force acts, is given by,

$$r_m = AC - AB \tag{17}$$

$$AC = \frac{D_P}{2} \tag{18}$$

$$AB = \sin\gamma \times \frac{b}{2} \tag{19}$$

From equations 17, 18 and 19,

$$r_m = \frac{D_P}{2} - \frac{b\sin\gamma}{2} \qquad \text{(V B Bhandari , 2006)} \tag{20}$$

Where, $b = face\ width\ of\ tooth\ in\ mm$

Also, it is clear from figure. 12(c), the radial component on the gear and the axial component P_a on the pinion are equal. Similarly, the axial components on the gear and the radial component P_r on the pinion are equal.

2.6. Parameters Associated with Gears

2.6.1. Gear Ratio

"Gears ratio is simply the ratio of numbers of teeth on two meshing gears". (V B Bhandari , 2006). If the number of teeth on the driver wheel is T_1 and that on driven wheel is T_2, the gear ratio is given by:

$$Gear\ ratio = \frac{T_2}{T_1} \qquad \text{(V B Bhandari , 2006)} \tag{21}$$

2.6.2. Angular Velocity Ratio

Angular velocity ratio is the ratio of turning speed of the input gear to that of output gear.

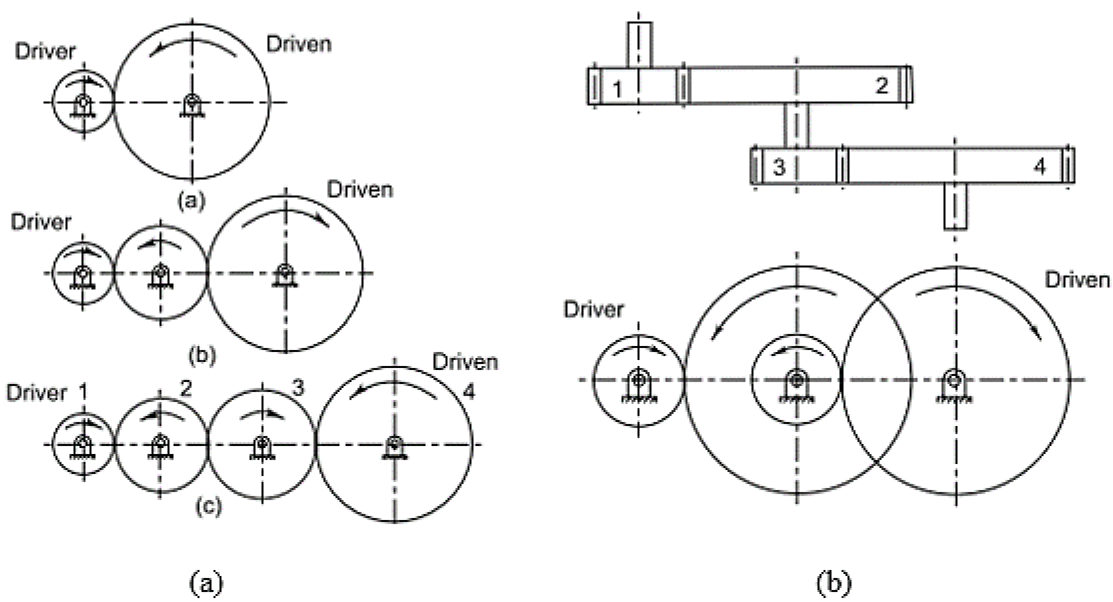

Figure 13, Gear Trains (V B Bhandari , 2006, p. 656)

For a simple gear train as shown in figure. 13(a), "the velocity ratio is equal to the ratio of number of teeth on last driven gear to the number of teeth on the first driving gear." (V B Bhandari , 2006). The velocity ratio for the gear train in figure. 13(a)(a), is given by,

$$\frac{\omega_1}{\omega_2} = \frac{T_2}{T_1} \tag{22}$$

Similarly, The velocity ratio for gear train in figure. 13(a)(c), is given by,

$$\frac{\omega_1}{\omega_4} = \frac{T_4}{T_1}$$

In figure. 13(a), the gears in the middle, other than the driving and driven gears are called idlers. "The main function of the idler gears is to change the direction of the last driven gear relative to the first driving gear." (V B Bhandari , 2006). They have no effect on the over gear ratio, regardless of how many teeth they have.

A compound gear is shown in figure. 13(b). It consists of an intermediate shaft with two gears. One of the gear meshes with the gear on the driving shaft and other gear meshes with the gear on the driven shaft. The angular velocity of two gears mounted on the intermediate shaft are the same. Thus, the velocity ratio is given by,

$$\frac{\omega_1}{\omega_4} = \left(\frac{T_2}{T_1}\right)\left(\frac{T_4}{T_3}\right) \qquad \text{(V B Bhandari , 2006)} \tag{23}$$

2.6.3. Power and Torque Transferred by a Gear Pair

Power and torque are two important variables while considering the performance of gears or rotary machines. "Power is simply defined as the rate of doing work or energy per unit time." (Bhatia, 2019).

$$P = \frac{W}{\Delta t} \tag{24}$$

But, for rotary motion (Bhatia, 2019), $W = \tau \times \Delta\theta$

Torque τ and the angular displacement θ play the same role in circular motion as played by the force F and the distance d in linear motion. So, equation 24, becomes,

$$P = \frac{\tau\Delta\theta}{\Delta t}$$

$$or, P = \tau \times \omega \qquad \text{(Bhatia, 2019)} \qquad (25)$$

Where, $\omega = \frac{\Delta\theta}{\Delta t}$ is the angular velocity of the rotating body.

“Torque is a measure of force that causes an object to rotate about an axis.” (Bhatia, 2019). It is the rotational equivalent of linear force. In order to produce torque, the force must act at some distance from the axis or pivot point. Consider a force F is applied tangentially at a tooth of the gear with radius r and shaft is presented at the center of the gear:

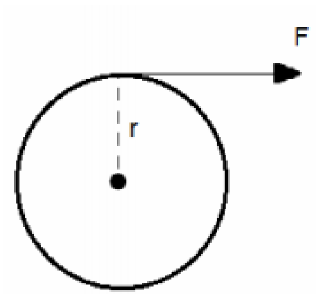

Figure 14, Torque Calculation (Bhatia, 2019)

Now, the amount of torque about the axle or shaft is given by,

$$\tau = F \times r \qquad \text{(Bhatia, 2019)}$$

Substituting the value of τ in equation 25 gives,

$$P = F \times r \times \omega \qquad (26)$$

But angular velocity is often referred to as rotational velocity and measured in complete revolutions per minute (rpm). So, equation 26 becomes,

$$P = F \times r \times \frac{2\pi N}{60} \qquad (\text{Bhatia}, 2019) \tag{27}$$

Where, N represents number of revolutions per minute.

Now, consider two gears in mesh rotating in opposite direction. Let, ω_1 and ω_2 be the angular velocities of the driving and the driven gear respectively. Also, let, τ_1 and τ_2 be the torque acting on the shaft of the driving and the driven gear respectively.

Using, equation 25, power transferred by the driving gear is calculated as,

$$P_{in} = \tau_1 \times \omega_1$$

And, power transferred to the driven gear is calculated as,

$$P_{out} = \tau_2 \times \omega_2$$

Assuming no friction losses, the input and output power can be considered equal to each other.

$$P_{in} = P_{out}$$

$$or, \tau_1 \times \omega_1 = \tau_2 \times \omega_2$$

$$or, \frac{\omega_1}{\omega_2} = \frac{\tau_2}{\tau_1} \tag{28}$$

Since, $\omega = \frac{2\pi N}{60}$, equation 28 can be written as,

$$\frac{\omega_1}{\omega_2} = \frac{\tau_2}{\tau_1} = \frac{N_1}{N_2} = \frac{T_2}{T_1} \qquad (\text{Bhatia}, 2019) \tag{29}$$

Equation 29 shows that the torque and speed are inversely proportional to each other. If high torque is required, the speed must be lowered and if the speed increases, the torque decreases proportionally.

2.7. Power Transmission System in Automobile

The power transmission system in automobile consists of two parts; transmission gearbox and differential. The power is produced in the engine by the reciprocating motion of piston. The power thus produced is transferred to the transmission gear box via rotary motion of the crank shaft. To transfer or disengage the power between engine and the transmission gearbox, a clutch system is located between them. Various range of motion is obtained at the gearbox output shaft through different gearing arrangements. This motion of the output shaft is then transferred to the differential. Finally, the power from differential is transmitted to both the road wheels according to the speed requirement.

2.7.1. Manual Transmission Gearbox

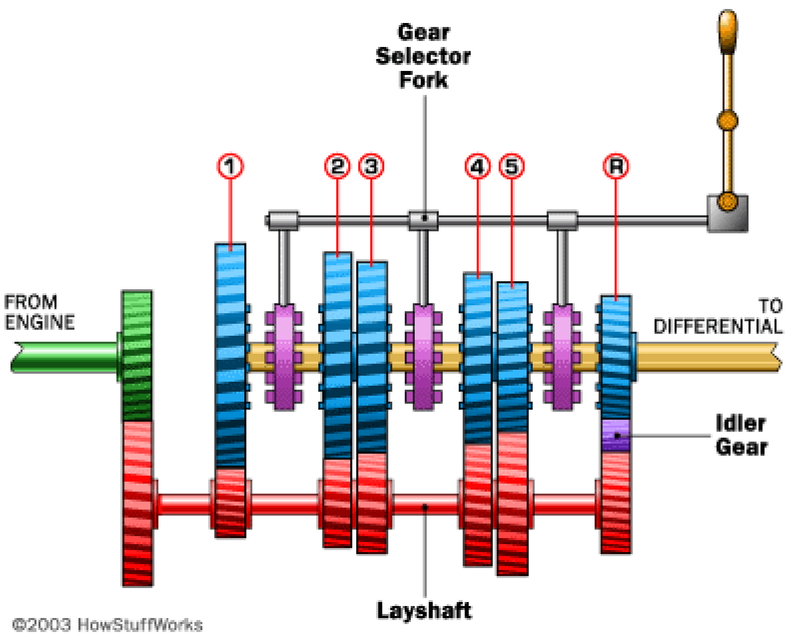

Figure 15, Manual Transmission (Marshall Brain, 2019)

The basic function of the transmission is to control the speed and torque available to the road wheels for different driving conditions. For example, to climb a hill, a vehicle needs more torque. This torque can be obtained by reducing the speed at the transmission. This is clearly seen in equation 25. If the angular speed of the gear is decreased, it generates more torque for the same power input. Similarly, if the torque demand is low, the angular speed of the gear must increase.

The working of manual transmission is based on the principle of gearing and the gear ratio. As shown in figure. 15, the input shaft (green shaft) from the engine and the output shaft (yellow shaft) from the differential are connected through a layshaft (red shaft). The gears are always in mesh with each other, but the output gears are loosely connected to the output shaft. To obtain a particular speed at a time, only one gear should be attached to the output shaft at that particular instance. This is achieved by using selectors as shown in the figure above. Sliding the selector and meshing it with any particular gear engages that gear (Marshall Brain, 2019).

2.7.2. Differential

Differential plays an important role in motion of automobiles. “The main function of the differential is to allow the rare wheels to rotate at different angular speed.” (TECHTRIXINFO, 2018). Consider the two rare wheels of an automobile which are taking a turn as shown in figure. 16. It is clear that the distance two wheels have to travel are different. This means, one wheel has to rotate at a higher speed than the other wheel. This is not possible, since the wheels are connected with a solid shaft. This problem can be solved using a differential gearbox which allows the right and left wheel to rotate at different angular speed.

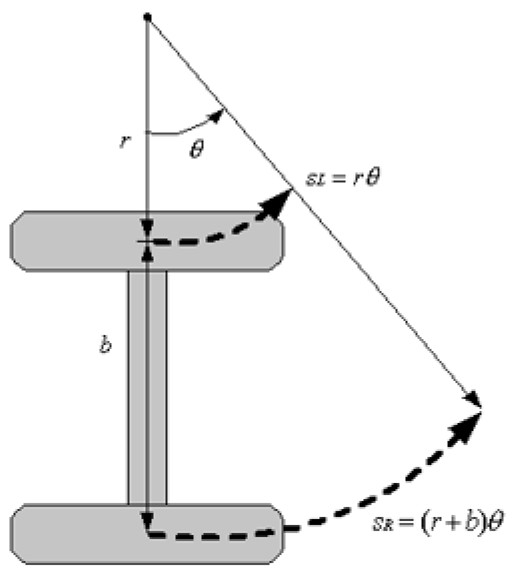

Figure 16, Rare Wheel Taking Turn (Lucas, 2001)

A differential as shown in figure. 17, consists of a pinion gear, a ring gear, a set of spider gear and two side gears. A pinion gear transfers the power coming from the manual transmission to the ring gear. The ring gear has two parallel spider gears attached to it. Spider is free to rotate about its own axis and also along with the ring gear. Spider gears are in mesh with two side gears. The side gears are connected to the shaft containing left and right wheels. Therefore, the power from the transmission flows from the pinion to the left and right wheels via ring gear and spider gear. (TECHTRIXINFO, 2018).

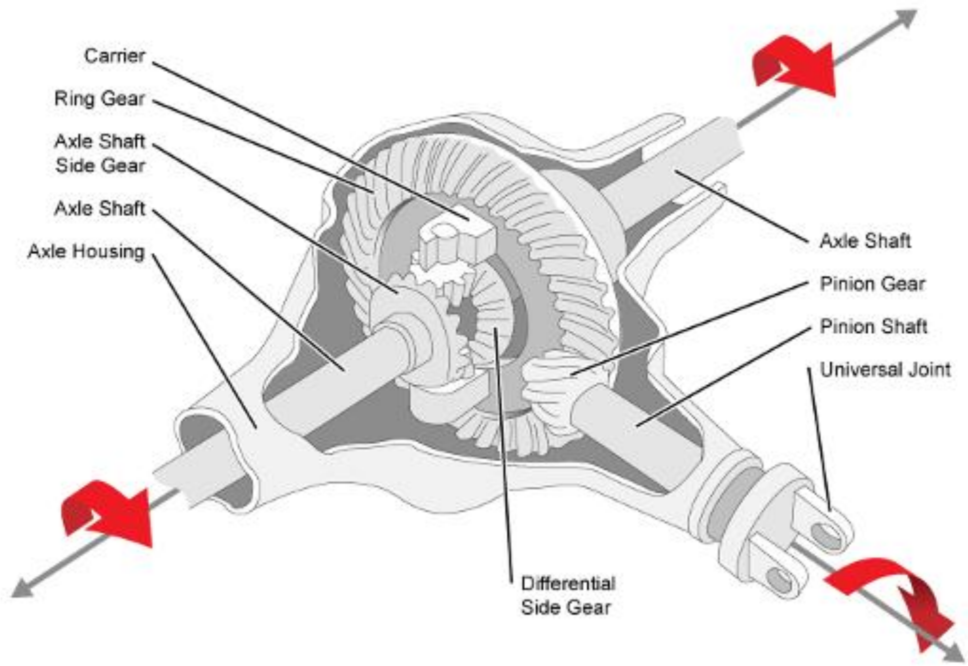

Figure 17, Differential Gear (The MathWorks, Inc., 2019)

"When the vehicle is moving in a straight line, the spider gear rotates along with the ring gear but does not rotates on its own axis. In this case, spider gear will push and make the side gears turn at the same the same speed." (TECHTRIXINFO, 2018).

$$N_{right} = N_{left}$$

But when the vehicle is taking a turn as shown in figure. 16, the spider gear rotates along with the ring gear and also on its own axis. "When the spider gear is rotating as well as spinning, peripheral velocity of one side gear is the sum of spinning and rotational velocity and that of other side gear is the difference of spinning and rotational velocity." (TECHTRIXINFO, 2018). This allows the wheels to rotate at different angular velocities.

$$N_{right} \neq N_{left}$$

2.8. Speed Control System for Gears

In this thesis project, speed or rpm of the gears are controlled using a dc motor. A dc motor itself is also connected with the potentiometer and a rheostat to control the voltage across the dc motor.

2.8.1. DC Motor

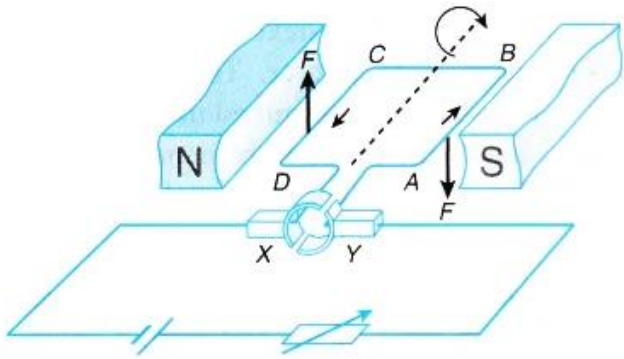

Figure 18, Rectangular Wire Loop Placed in a Magnetic Field (Veerendra, 2017)

A DC motor is an electrical rotary machine that converts electrical energy into mechanical energy. It consists of loops of wire attached to a vertically rotating shaft placed between the magnetic field as shown in figure. 18. The working of a DC motor is based upon the principle that, "when a charge carrying conductor is placed in a magnetic field, it experiences a mechanical force or torque." The direction of the force is given in accordance with the Flemings left hand rule that states, "if the thumb, fore finger and the middle finger of the left hand are stretched perpendicular to each other and if the fore finger represents the direction of magnetic field, the middle finger represents the direction of current, then the thumb represents the direction of the force." (Daware, 2016)

A dc motor basically consists of large number of current loops in a magnetic field. To derive the expression for torque experienced by a current loop in a magnetic field, consider a rectangular loop $ABCD$ of length l and breadth b placed in a uniform magnetic field B. The loop is carrying current I in the direction along $ABCD$ as shown in the figure. 19(a).

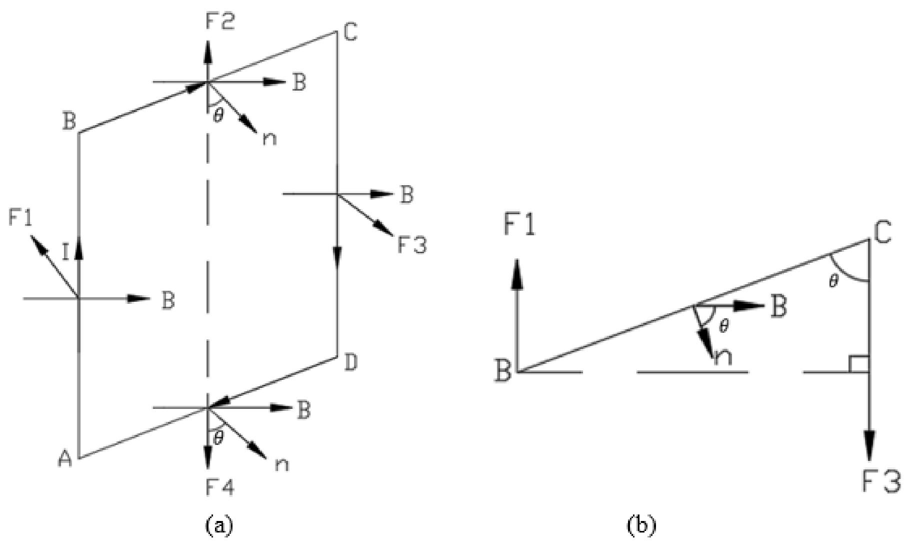

Figure 19, Diagram Representing Torque on a Rectangular Current Loop (Drawn Using AutoCAD)

Now, force on arm AB, $\overrightarrow{F_1} = \overrightarrow{Il} \times \vec{B}$ (EduPoint, 2017)

Since the angle between the current element $\overrightarrow{Il}$ and the direction of magnetic field $\vec{B}$ is 90°,

So, $F_1 = IlB \sin 90° = IlB$

This force F_1 is perpendicular to plane of the paper and inwards.

Similarly, force on arm CD, $\overrightarrow{F_3} = \overrightarrow{Il} \times \vec{B} = IlB \sin 90° = IlB$

This force F_3 is perpendicular to plane of the paper and outwards.

These forces F_1 and F_3 are equal in magnitude, opposite in direction and consists different lines of action. Hence, they constitute a couple. (EduPoint, 2017)

To calculate the F_2 and F_4, let θ be the angle between the normal to the plane of the loop and the direction of the magnetic field.

Then, force on arm BC, $\overrightarrow{F_2} = \overrightarrow{Ib} \times \vec{B} = IbB \sin(90 - \theta) = IbB \cos\theta$ (EduPoint, 2017)

Similarly, force on arm DA, $\overrightarrow{F_4} = \overrightarrow{Ib} \times \vec{B} = IbB \sin(90 + \theta) = IbB \cos\theta$

These forces F_2 and F_4 are equal in magnitude, opposite in direction and have same lines of action. Hence, they cancel out each other and have no resultant effect on the loop. (EduPoint, 2017)

To calculate the torque, take the vertical projection of the arm BC as shown in the figure. 19(b).

Since, torque is the product of any one of the force and perpendicular distance between then,

Hence, Torque $\tau = IlB \times b \sin\theta = IB \sin\theta \times A = IAB \sin\theta$ (EduPoint, 2017)

If the coil contains n turns, then,

$$\tau = nIAB \sin\theta \tag{30}$$

The torque thus obtained causes the drive shaft to rotate.

Speed in rpm N of a dc motor is generally given by (Alnaib, 2019),

$$N = \frac{(V - I_a R_a) \times 60A}{PZ\phi} \tag{31}$$

Where, $V = applied\ voltage$

$I_a = armature\ current$

$R_a = armature\ resistance$

$P = number\ of\ poles$

$Z = number\ of\ armature\ conductors$

$\phi = flux\ per\ pole$

$A = parallel\ paths$

But, for a dc motor, A, P and Z are constant. I_a and R_a can also be kept constant using rheostat. So, equation 31 can be written as;

$$N \propto \frac{V}{\phi}$$

This implies that keeping the flux per pole constant, if the voltage applied to a dc motor increases, the speed in rpm N of a dc motor also increases and vice versa. A potentiometer can be used to control the voltage of a dc motor.

2.8.2. Potentiometer

To control the speed of the motor, a device called potentiometer is used. “A potentiometer is a manually adjustable variable resistor with 3 terminals.” (LEETS Academy, 2018). It consists of three terminals and a circular resistive element inside it. Two outer terminals are connected to both the ends of resistive element and the middle terminal consists of a wiper that mover over the resistive element. “The output voltage of the potentiometer is determined by position of the wiper.” (LEETS Academy, 2018)

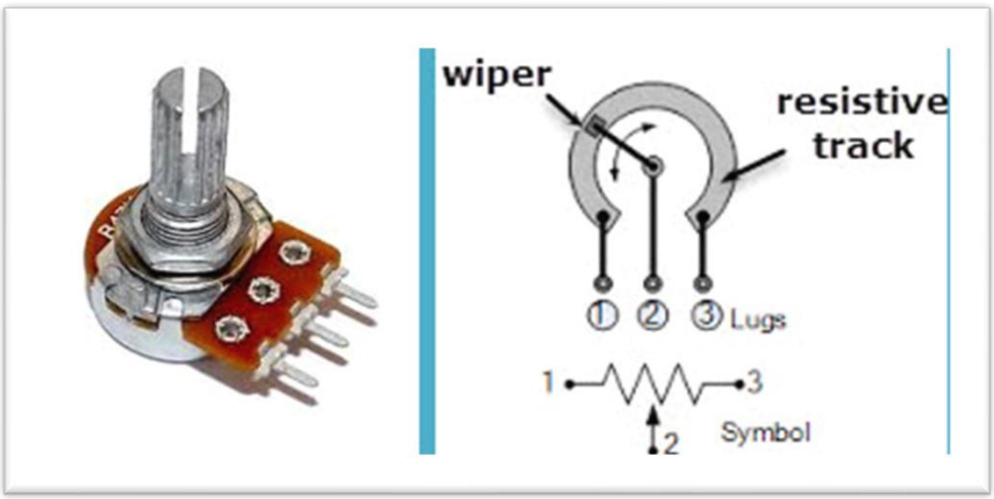

Figure 20, Potentiometer (Elprocus, 2019)

Consider a potentiometer with total resistance of R and I be the current passing through it. Then by Ohm’s law, voltage across the two points of the resistors is given by,

$$V = IR \qquad \text{(Elprocus, 2019)} \tag{32}$$

But, the electrical resistance of a wire with length L and cross-sectional A is given by,

$$R = \rho \frac{L}{A} \qquad \text{(Elprocus, 2019)}$$

Where, ρ is a constant called the resistivity of the conductor.

Putting the value of resistance in equation 32, gives,

$$V = \frac{I\rho L}{A} \tag{33}$$

Since, ρ and A are constant for a particular resistor and I is also kept constant using a rheostat.

Equation 33 implies that the voltage across the resistor is directly proportional to the effective length of the resistor wire i.e. $V \propto L$. So, if the effective length of a resistor wire increases the voltage across the resistance also increases and vice versa. (Elprocus, 2019)

2.9. Power Flow Diagram of the System

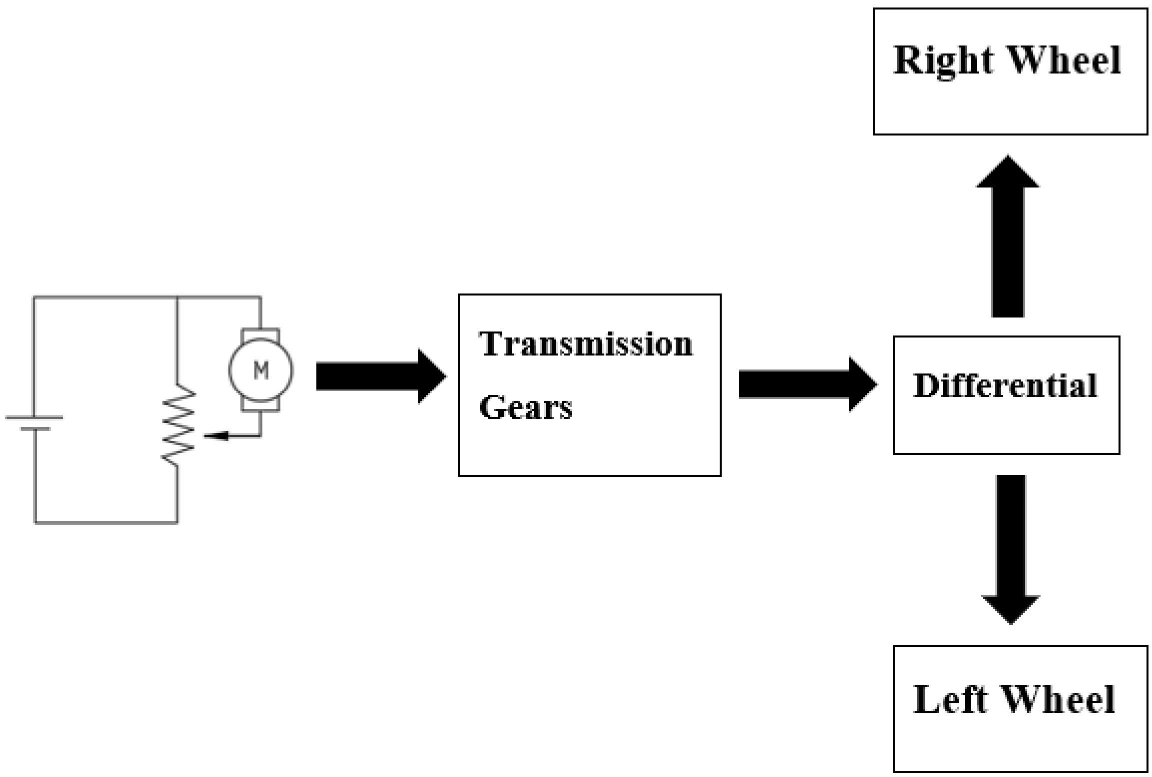

Figure 21, Schematic Representation of Power Flow of the System (Drawn Using AutoCAD and Microsoft Word)

2.10. 3D Printing

As the name suggests, 3D printing or additive manufacturing is a process of making three dimensional solid objects from a digital file using an additive process. “In this process an object is created layer by layer, by laying down successive layers of material on top of each other. Each of these layers can be seen as a thinly sliced horizontal cross-section of the eventual object” (3dprinting.com, 2019).

The first step in 3D printing is to create a design file. Modelling software’s like AutoCAD, SolidWorks, Blender, MakerBot can be used to create own designs. The design file is then saved as a STL file, most common file type that is used for 3D printing. STL file stores information about 3D models. Then the STL file is feed to the printer. Objects with complex geometries might require support structures that are drawn automatically by MakerBot along with the design model. Sometimes post processing or cleaning are required to be done manually. Although there are various kinds of 3D printing methods, the particular method this thesis deals with is Fused Deposition Modelling.

Fused deposition modelling is one of the most widely used technique for printing of thermoplastics. In this process layers of raw materials are deposited by spraying or squeezing melted paste through the nozzle. In this process the design file saved in STL format is feed into the printer. Then, the slicing software converts the model into number of tiny slices or layers. After the file is uploaded, it prints the object layer by layer. “Based on the generated tool paths, the head of the printer moves in XY plane and the platform holding the part moves in Z plane depositing a new layer on top of the previous one.” (K.S. Bopari, 2017). Due to high dimensional accuracy, simplicity and ability to work with various CAD software, FDM has gained its popularity amongst homes, schools and offices.

2.11. Modelling Software and Tool for Designing and Printing

2.11.1. SolidWorks

SOLIDWORKS is a 3D designing software acquired by DASSAULT SYSTEMES, France in 1997 and is aimed to allow users to create, simulate, publish and manage their data. SOLIDWORKS products are easy to learn and use, and work together thus helping design products better, faster, and more cost-effectively (Dassault Systemes, 2002-2019).

SolidWorks is a powerful tool that is easy to use and one can reuse the model for many other applications like 3D printing, Mastercam, etc. It helps users to transform their imagination into design and finally into product. In this thesis project, SolidWorks 2018 is used for making the design files and then the file is transferred to MakerBot Replicator for printing.

2.11.2. MakerBot Replicator

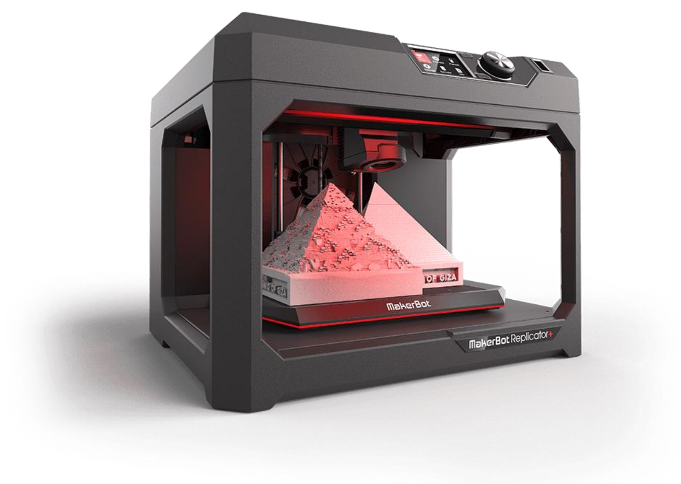

Figure 22, MakerBot Replicator (MakerBot Industries, 2009-2019)

MakerBot 3D printers are some of the most popular and affordable printers. These printers utilize fused deposition technology. It’s the method that pushes plastic filament through a heated nozzle. We can connect the printer directly to the computer or use the USB port to directly transfer the design file. There is also a small LCD panel to navigate through the menus.

2.12. Materials Used for Printing

There are quite few different materials options available for FDM 3D printing. There material properties range from durability, flexibility, to ability to print and price. Some of these materials include ABS (Acrylonitrile Butadiene Styrene), PLA (Polylactic Acid), HIPS (High Impact Polystyrene), Nylon, Carbon fiber, PP (Polypropylene), Polycarbonate, Metal filled and Wood filled, etc. Each of these materials have their own unique properties and some flaws associated with them. Among them, the two most commonly used material are ABS and PLA. This thesis project studies and compares the material and mechanical properties two different kinds of PLA; Normal PLA and Tough PLA.

2.12.1. Normal PLA

PLA is the most popular material for FDM 3D printing. It is made from corn starch or sugar cane. PLA is easy to print and works for wide variety of 3D printing applications. PLA can be printed at lower temperature as operating temperature of PLA is $190℃ - 230℃$. PLA is much less sensitive to temperature changes compared to ABS. So, it does not need a heated platform or enclosed chamber. The printed parts have better aesthetic quality and more flexibility with printing conditions. Since PLA material can be easily pigmented, PLA filaments come in so many colors. PLA can be infused with various materials like wood, algae, metal, carbon fiber, etc. to give different new properties to the print. Unlike other thermoplastics which might take thousands of years to decompose, PLA being biodegradable takes about 3 to 6 months to decompose. However, there are also some draw bags associated 3D printing of PLA. PLA filaments have lower resistance to heat and tend to melt or deform when heat is applied. The mechanical properties or durability of PLA is much lower than ABS. The printed parts are brittle and are not suitable for wear and tear. (All3DP, 2019).

2.12.2. Tough PLA

Although, PLA was easy to use and was flexible with the printing conditions but it lacked mechanical properties and durability like ABS. To overcome the defects of traditional PLA, MakerBot in 2016 introduced Tough PLA as a powerful alternative to ABS. It is one of the interesting printing material that is engineered for strength and durability but with the reliability and ease of use for which PLA is known. The impact strength of Tough PLA is significantly more than PLA and twice compared to that of ABS. It has operating temperature of $190℃ - 230℃$ as normal PLA (Spectrum Group Sp., 2019) but heat resistance nearly equal to that of ABS, this overcomes common flaws of ABS printing like warping or curling.

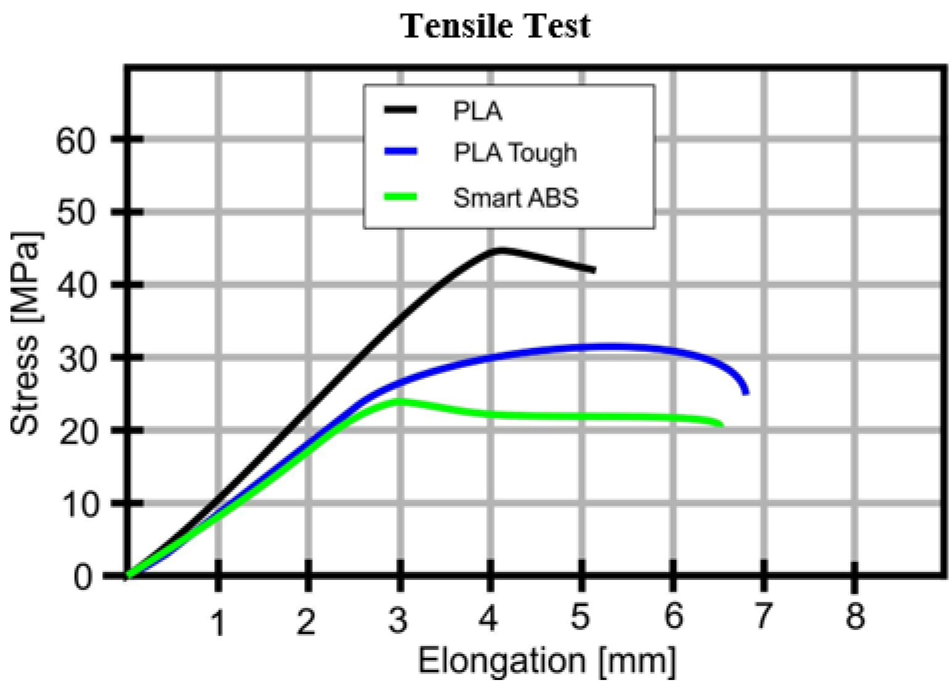

Figure 23, Tensile Test Comparison (Spectrum Group Sp., 2019)

The figure above shows the tensile test results comparison between normal PLA, tough PLA and ABS according to test method ASTM D638. The samples have thickness of 3.2 mm, length of 165 mm, width of 13 mm and gauge length of 50 mm. The tensile strength of Tough PLA is less than that of normal PLA but deforms much more before breaking. It also has plastic deformation range greater than that of ABS which overcomes the brittle nature of normal PLA. Because of these mechanical properties, Tough PLA has proved itself to be more versatile then normal PLA and ABS and suitable for many functional features like hinges, interlocking parts, thread features and for machining (MakerBot Industries, 2016).

3. DESIGNING

The whole power transmission system is designed part by part using SolidWorks. The primary components are the gears and the secondary are the shafts, synchronizing hub, sliding sleeve, bearings and support blocks. After all the parts are first designed, assembled and finally simulated by connecting motor.

3.1. Spur Gear

All the gears are designed by using SolidWorks toolbox library. The gears can be found under Design Library, Toolbox, ANSI Metric, Power Transmission. To design any gear, in this case spur gear, right click on the spur gear and select Create Part. This brings a gear with configuration table. The required gear is obtained by placing all the parameters like Module, Number of Teeth, Pressure Angle, Face Width and Nominal Shaft Diameter.

3.1.1. Input Gear and Layshaft Gear

The table below shows various parametric values to obtain each type of input and layshaft gear.

Table 1, Parameters for Input and Layshaft Gears

	Module	Number of Teetth	Pressure Angle	Face Width	Nominal Shaft Diameter
Input Gear	1.375	28	14.5	8	9
Input Layshaft Gear	1.375	42	14.5	8	17
Layshaft Gear 1	1.5	24	14.5	8	19
Layshaft Gear 2	1.5	28	14.5	8	19
Layshaft Gear 3	1.5	32	14.5	8	28
Layshaft Gear 4	1.5	36	14.5	8	32
Layshaft Gear 5	1.5	40	14.5	8	36
Layshaft Reverse Gear	1.5	18	14.5	8	11
Idler	1.5	13	14.5	8	9

Depending upon the provided parameters all the input and layshaft gears are obtained. After the gears are ready, a cut is made at the center of each input and layshaft gears. This will help provide a mating site between gears and shaft.

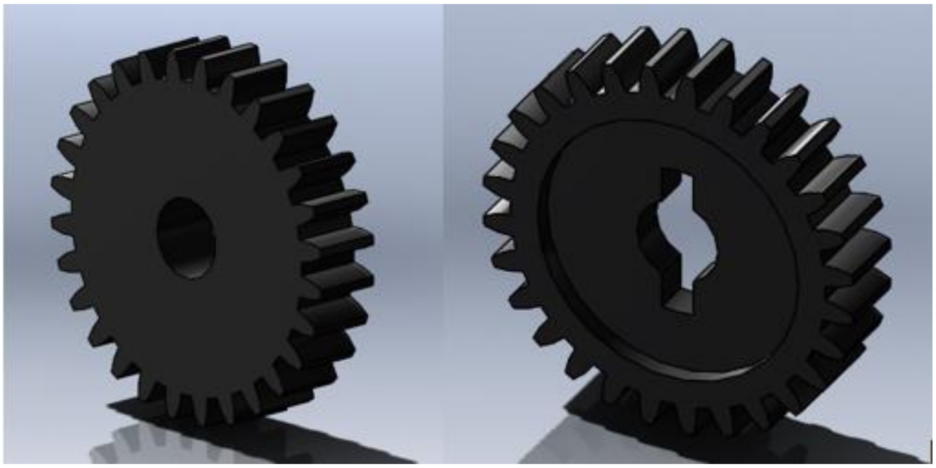

Figure 24, Input Gear (without cut and with cut)

3.1.2. Output Gear

The table below shows various parametric values to obtain each type of output gear.

Table 2, Parameters for Output Gears

	Module	Number of Teetth	Pressure Angle	Face Width	Nominal Shaft Diameter
Output Gear 1	1.5	42	14.5	8	28
Output Gear 2	1.5	38	14.5	8	28
Output Gear 3	1.5	34	14.5	8	28
Output Gear 4	1.5	30	14.5	8	28
Output Gear 5	1.5	26	14.5	8	28
Output Reverse Gear	1.5	22	14.5	8	26

Depending upon the provided parameters all the output gears are obtained.

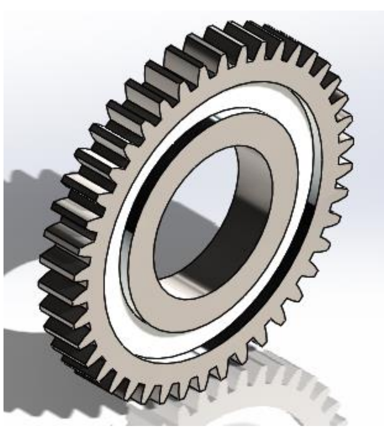

Figure 25, Output Gear without Synchronizer Cone

Each output gears have a synchronizer cone attached to them. These cone mesh with the synchronizing hub using a sliding sleeve or gear selector fork.

To create a synchronizer cone, a sketch as shown below is drawn in the Front Plane and extruded to 4 mm. Then, a circular cut of 6 mm diameter is made. Finally, total eight cuts are made using a circular pattern.

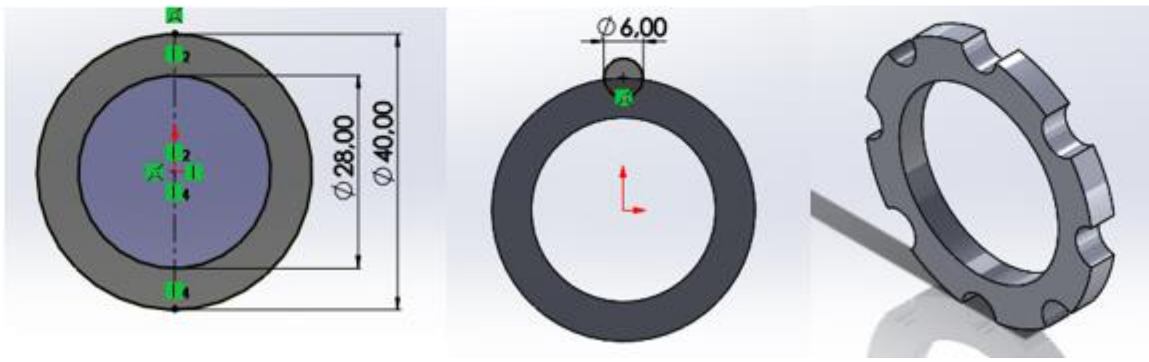

Figure 26, Creating a Synchronizer step by step

Now the synchronizer is attached to each output gears using SolidWorks Combine feature.

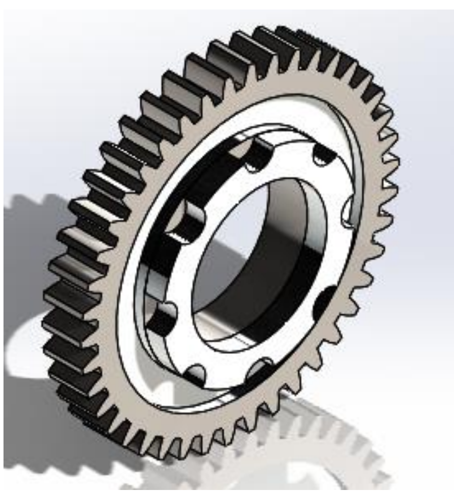

Figure 27, Output Gear with Synchronizer Cone

3.2. Bevel Gear

Bevel gears are created in similar way as spur gears from the design library. To create a bevel gear, right click on the bevel gear and select Create Part. This brings a bevel gear with configuration table. The required gear is obtained by placing all the parameters like Module, Number of Teeth, Pinion's No. of Teeth, Pressure Angle, Face Width, Hub Diameter, Mounting Distance and Nominal Shaft Diameter.

A Differential constitutes of a pinion gear, a ring gear with two spider gears and two side gears.

3.2.1. Pinion Gear

It is a gear that transfer the rotational motion from the output gears to the ring gear. The table below shows various parametric values to obtain a pinion gear.

Table 3, Pinion Gear Parameters

Parameters	Values
Module	1.5
Number of Teeth	32
Pinion's No. of Teeth	44
Pressure Angle	14.5
Face Width	14
Hub Diameter	20
Mounting Distance	42
Nominal Shaft Diameter	12

Depending upon the provided parameters, a pinion gear is obtained. The hub can be extruded upon requirement. A rectangular cut is made at the back of the pinion gear to allow the transmission shaft to mate with the gear.

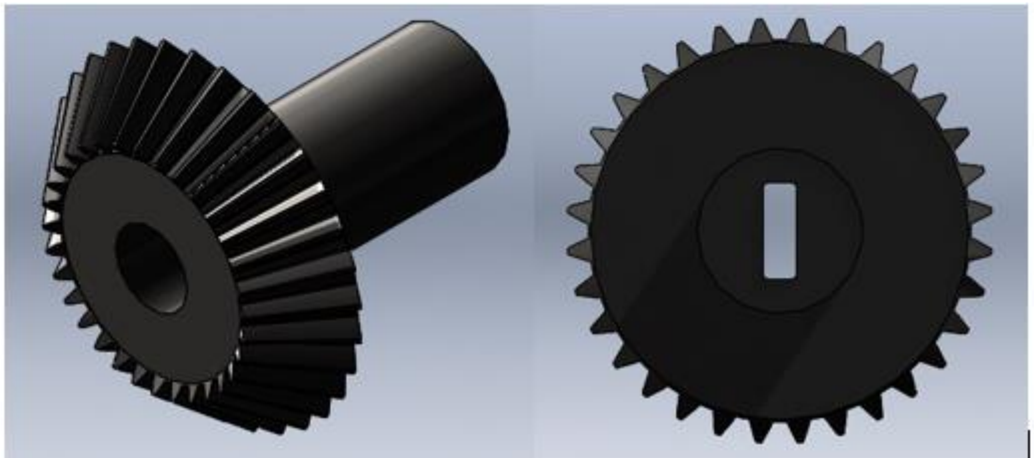

Figure 28, Pinion Gear

3.2.2. Ring Gear

It is the biggest gear that receives the motion from the pinion gear and transfers the motion to the two side gears through a pair of spider gears. The table below shows various parametric values to obtain a ring gear.

Table 4, Ring Gear Parameters

Parameters	Values
Module	1.5
Number of Teeth	44
Pinion's No. of Teeth	32
Pressure Angle	14.5
Face Width	14
Hub Diameter	20
Mounting Distance	42
Nominal Shaft Diameter	14

Depending upon the provided parameters, a ring gear is obtained.

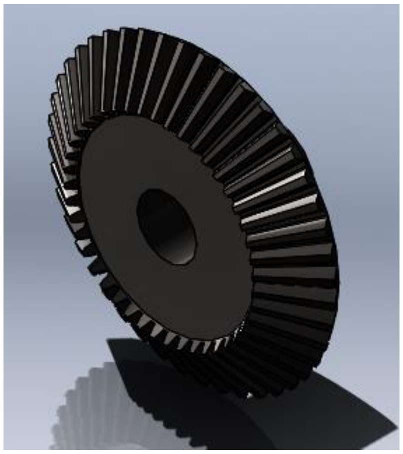

Figure 29, Ring Gear

A circular cut of 28.4 mm and depth of 8 mm is made at the center the center of the ring gear for the fitting of 28 mm diameter bearing. The ring gear also consists of two carrier plates containing spider gear. So, a circular cut of 13 mm is made on the carrier plates for the fitting of spider gears.

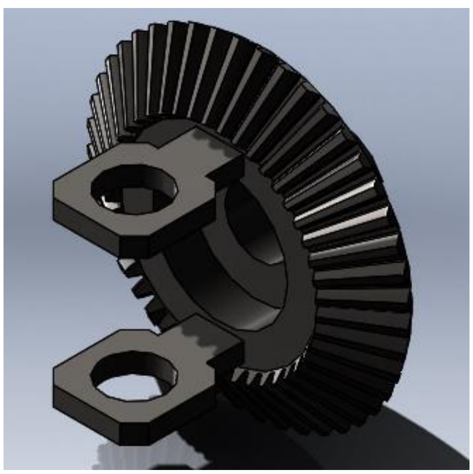

Figure 30, Ring Gear with Carrier Plates and Cuts

3.2.3. Spider Gears

Spider gears are two small gears which are fixed on the two carrier plates of the ring gear. It is free to rotate about its own axis and also along with the ring gear. The table below shows various parametric values to obtain a spider gear.

Table 5, Spider Gear Parameters

Parameters	**Values**
Module	1.5
Number of Teeth	14
Pinion's No. of Teeth	14
Pressure Angle	14.5
Face Width	8
Hub Diameter	12
Mounting Distance	20
Nominal Shaft Diameter	4

Depending upon the provided parameters, two spider gears are obtained.

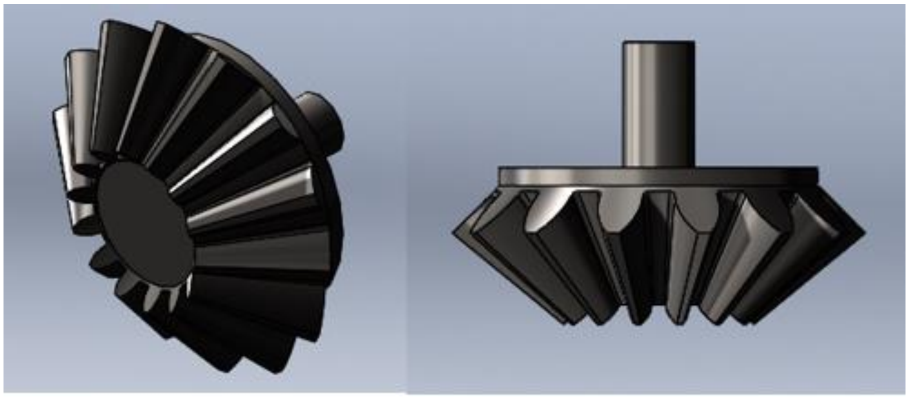

Figure 31, Spider Gear

3.2.4. Side Gears

The side gears are two gears connected to the shaft containing left and right wheels. Side gears are always in mesh with the spider gears. The table below shows various parametric values to obtain a side gear.

Table 6, Side Gear Parameters

Parameters	Values
Module	1.5
Number of Teeth	14
Pinion’s No. of Teeth	14
Pressure Angle	14.5
Face Width	8
Hub Diameter	12
Mounting Distance	20
Nominal Shaft Diameter	12

Depending upon the provided parameters, two side gears are obtained.

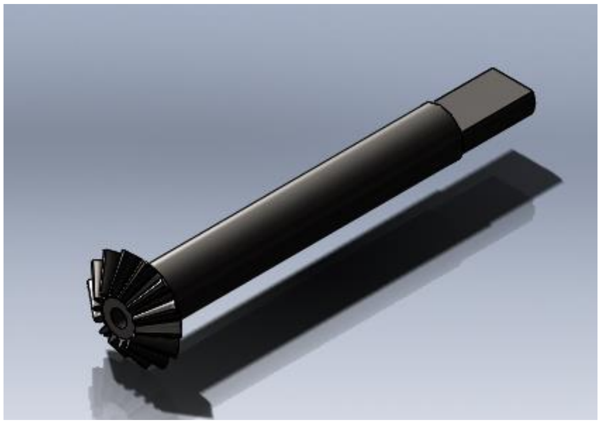

Figure 32, Side Gear

3.3. Synchronizing Hub

Synchronizing hub play a vital role in manual transmission system. A synchronizing hub, when synchronized with the synchronizer cone of gear receives the power from the output gear and transfers it to the output shaft.

To create synchronizing hub, a sketch shown below is drawn on the front plane and is extruded to 7 mm.

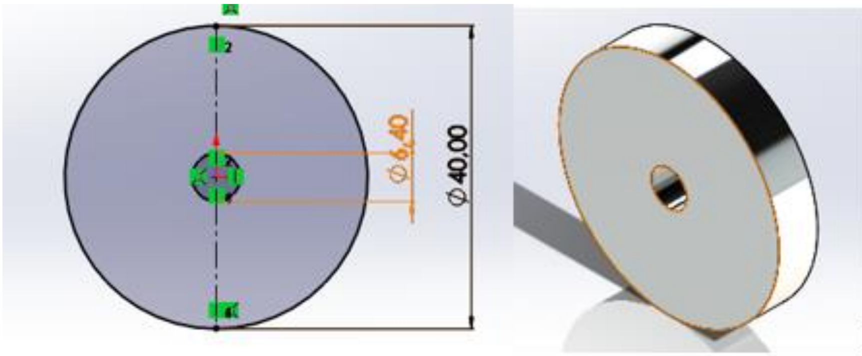

Figure 33, Designing of Synchronizing Hub

After this, the required cuts are made on the center and outer circumference to allow the output shaft and sliding sleeve slide through and over it.

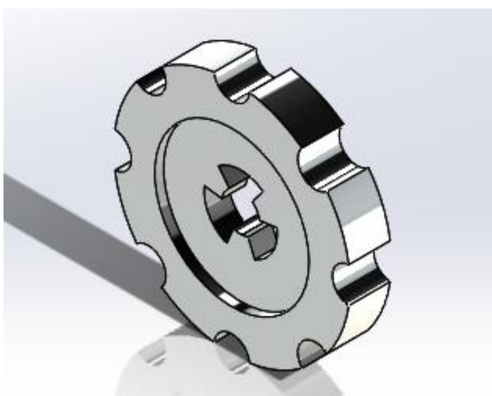

Figure 34, Designing of Synchronizing Hub

3.4. Sliding Sleeve or Selector Fork

To create selecting fork, a sketch shown below is drawn on the front plane and is extruded to 8 mm.

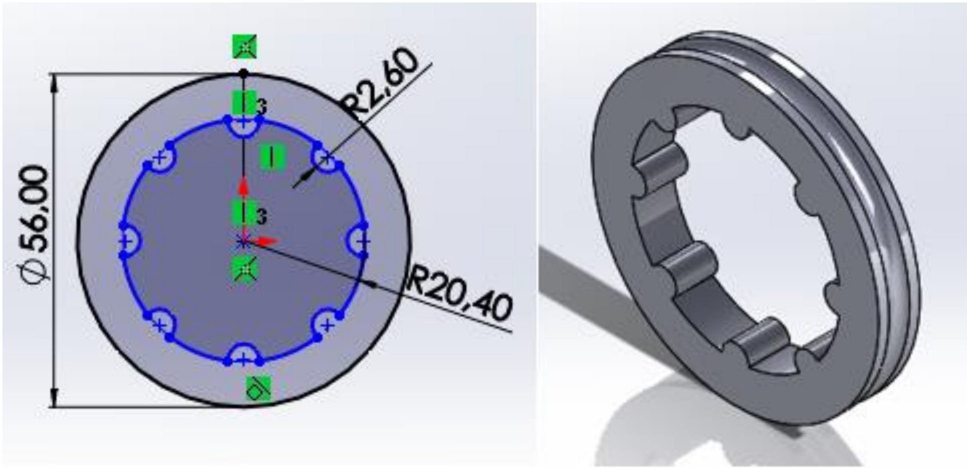

Figure 35, Designing of Selecting Fork

3.5. Motor Shaft

It is the shaft that transfers the rotary motion from the motor to the input gear. To create a motor shaft, a sketch shown below is drawn on the front plane and converted into solid shaft using Revolved Boss feature.

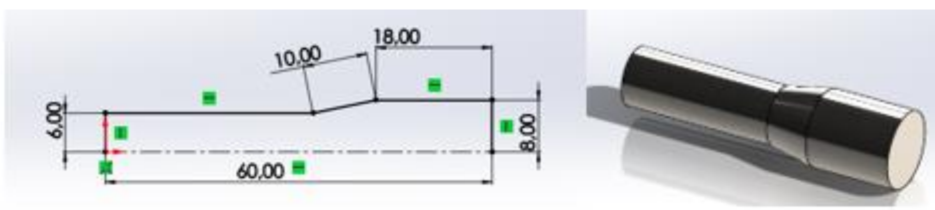

Figure 36, Designing Motor Shaft

Now, A extruded cut is made on the front face of the shaft by creating a sketch shown below.

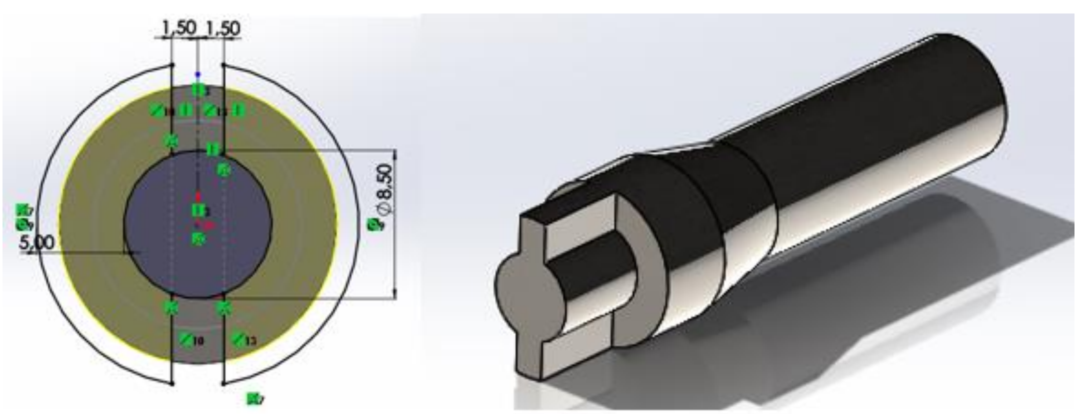

Figure 37, Designing Motor Shaft

Finally, six semicircular cuts are made on the back face of the shaft.

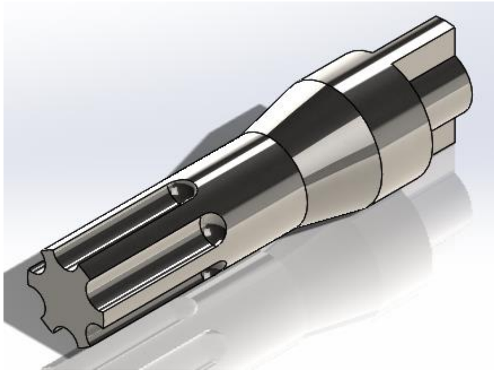

Figure 38, Designing Motor Shaft

3.6. Layshaft

It is the shaft helps to transfer the rotary motion from the input shaft to the output shaft.

To create the layshaft, a sketch shown below is drawn on the front plane and converted into solid shaft using Revolved Boss feature.

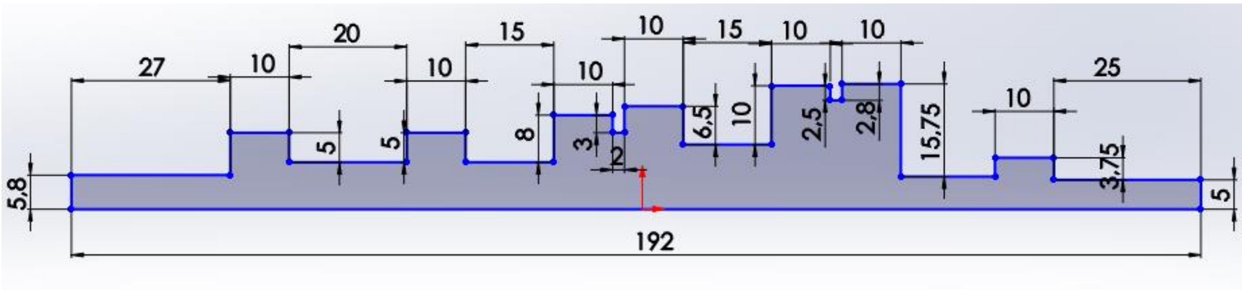

Figure 39, Designing of Layshaft

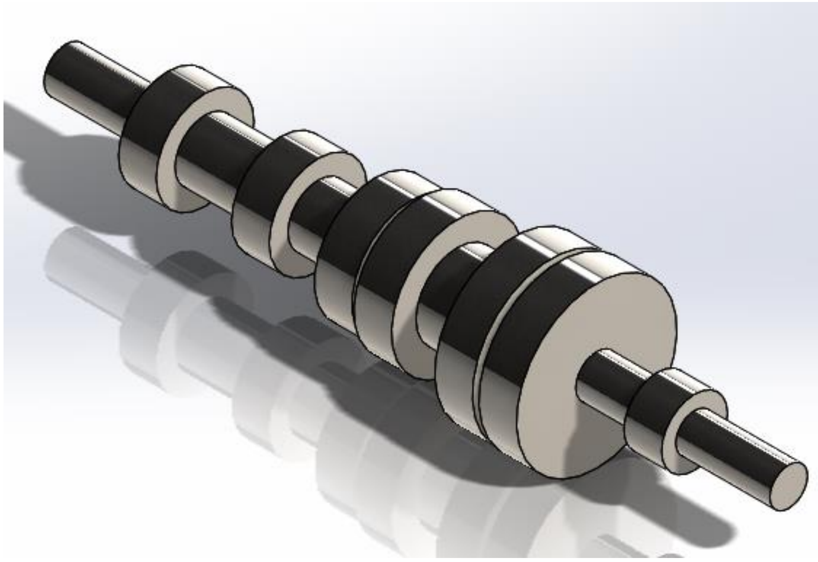

Figure 40, Designing of Layshaft

Depending upon the respective gear that will mate to layshaft, a cut is made on each circular block. Then the layshaft is ready.

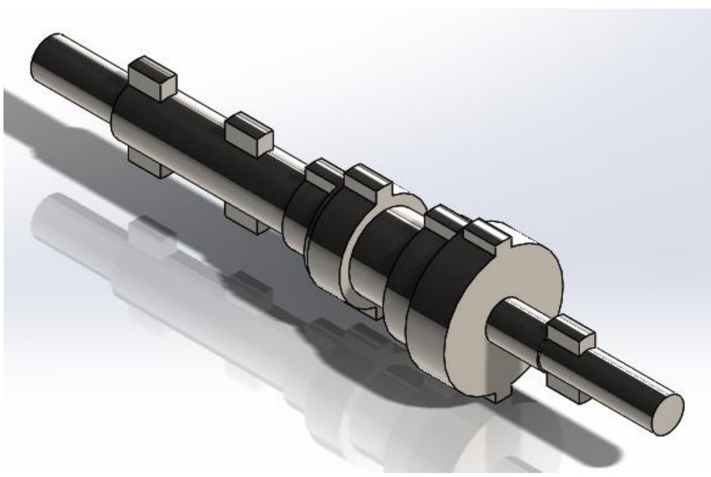

Figure 41, Designing of Layshaft

3.7. Output Shaft

It is the shaft receives the power from the layshaft and transfer it to the differential. Output shaft consists of series of output gears, synchronizing hub and sliding sleeve.

To create the output shaft, a sketch shown below is drawn on the front plane and converted into solid shaft using Revolved Boss feature.

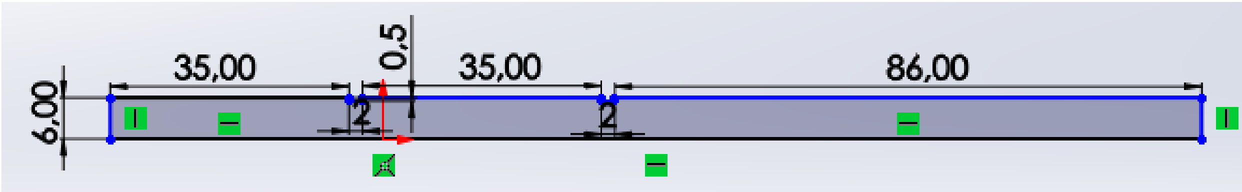

Figure 42, Designing of Output Shaft

Three similar and equidistant cuts are made to allow the synchronizing hub to slide over the shaft.

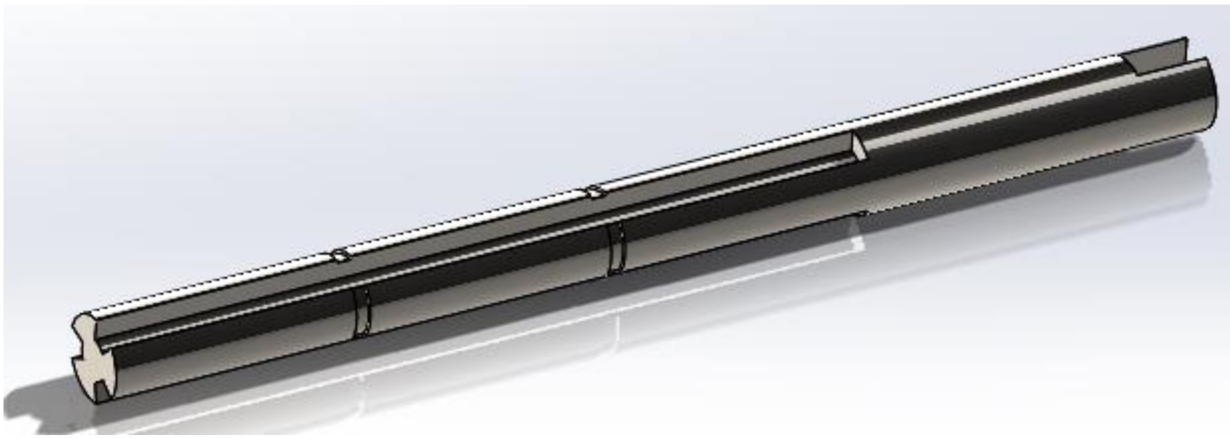

Figure 43, Designing of Output Shaft

3.8. Transmission Shaft

It is a shaft that connects output shaft to the differential. To create the transmission shaft, a sketch shown below is drawn on the front plane and converted into solid shaft using Revolved Boss feature. Then, cuts are made on the two ends of the shaft.

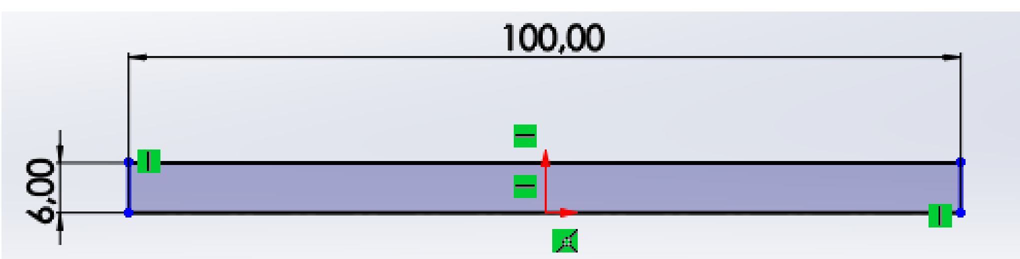

Figure 44, Designing of Transmission Shaft

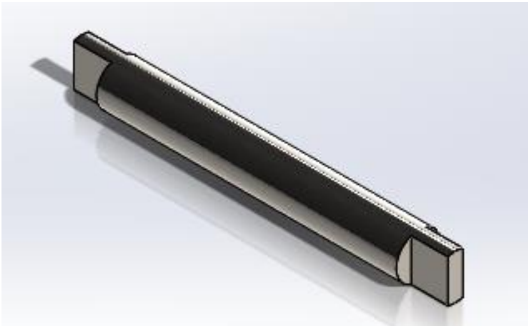

Figure 45, Designing of Transmission Shaft

3.9. Bearings

Bearings can be found inside SolidWorks Design Library, Toolbox, ANSI Metric, Ball Bearings, Radial Ball Bearings. The required bearing is obtained by placing all the parameters like Size, Bore, Outer Diameter and thickness. Three sizes of bearings are used in this project.

Table 7, Bearings Parameters

	Size	Bore	Outer Diameter	Thickness
Bearing 12,28,8	10-12	12	28	8
Bearing 20,42,12	10-20	20	42	12
Bearing 4,13,5	02-4	4	13	5

Depending upon the provided parameters, three bearings are obtained.

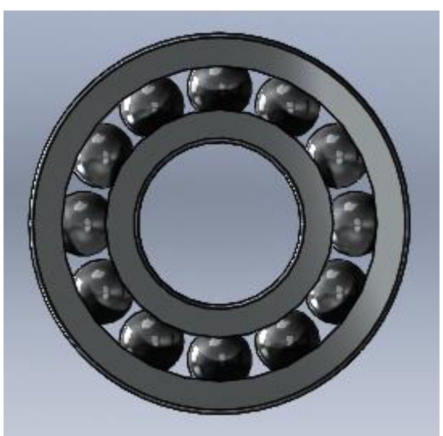

Figure 46, Bearing

3.10. Wheels

To design a wheel a sketch shown below is created in the Front Plane and revolved around the central axis using Revolved Boss feature.

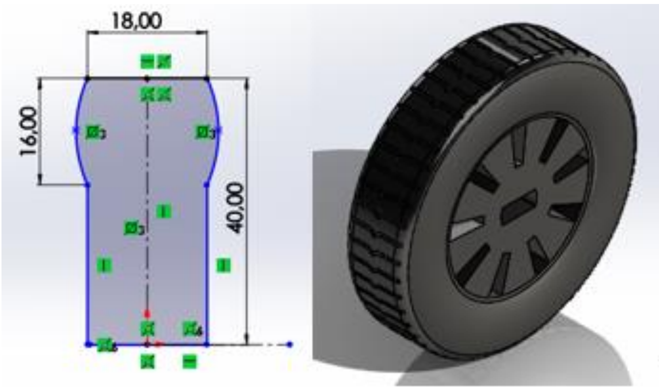

Figure 47, Designing of Wheel

3.11. Gear Cases

Gear cases are supportive structures that hold the gears in position. Depending upon the total length width and height of the gear assembly, two very simple gear cases are designed. Some circular and semicircular cuts are made to fit bearings.

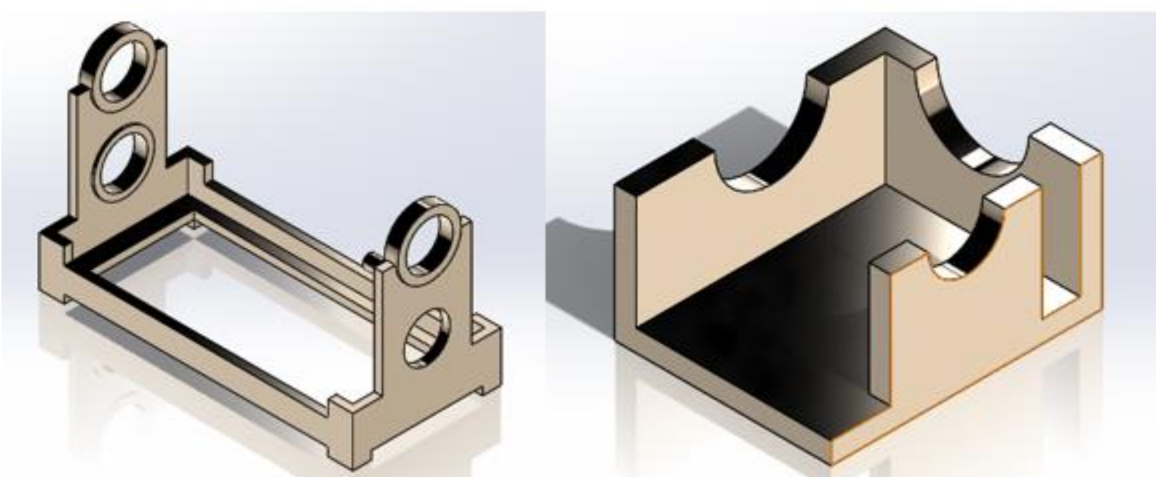

Figure 48, Gear Cases (manual transmission case on the left and differential case on the right)

4. GEARS ASSEMBLY

Since, all the individual parts are now created, they need to be assembled into several sub-assemblies and then these sub-assemblies will be used to make a final assembly. The gears will be assembled into two blocks, manual transmission and differential block.

4.1. Motor Shaft and Input Gear Sub-assembly

Motor shaft and input gear is assembled using suitable concentric, parallel and coincident mate.

Figure 49, Motor Shaft and Input Gear Assembly

4.2. Layshaft and Layshaft Gears Sub-assembly

Layshaft and layshaft gears are assembled using suitable concentric, parallel and coincident mate. The farthest left gear or the first gear in the series is the input layshaft gear, the second one is the layshaft gear 1, the third is layshaft gear 2, the fourth is layshaft gear 3, the fifth is layshaft gear 4, the sixth is layshaft gear 5 and the smallest gear at the last is the layshaft reverse gear. Here, the number of teeth or pitch diameter goes on increasing from layshaft gear 1 to layshaft gear 5.

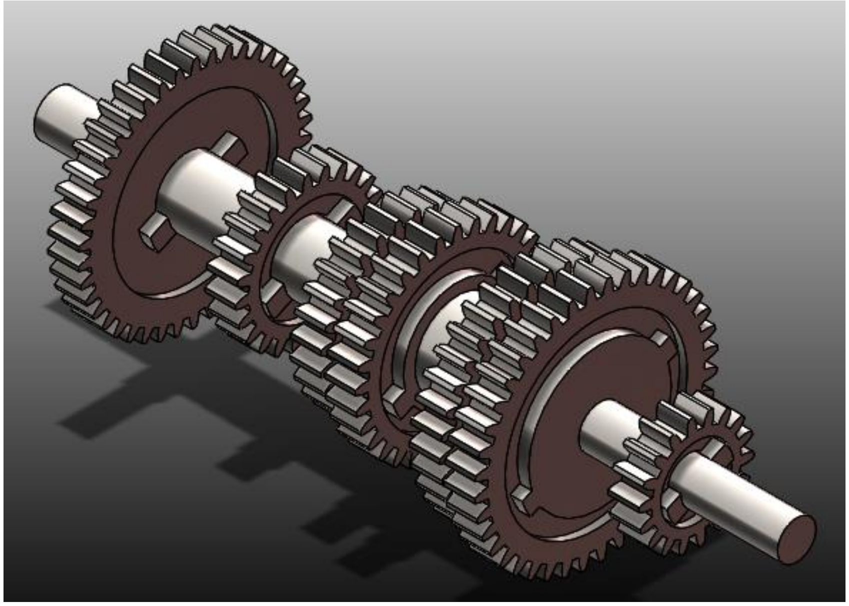

Figure 50, Layshaft and Layshaft Gears Sub-assembly

It is very important to note the distances between each layshaft gears because same distance should be maintained between the consecutive gears in the output shaft.

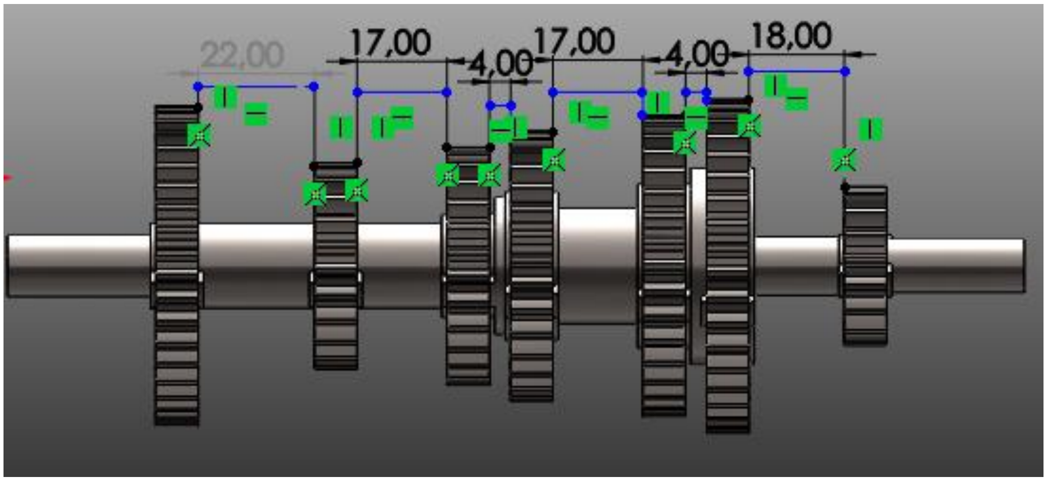

Figure 51, Distance Between Consecutive Layshaft Gears

4.3. Output Shaft and Output Gears Sub-assembly

Output shaft and output gears are assembled using suitable concentric, parallel and coincident mate. The bearing of size 10-12 is fitted to the center of each gear. The first gear in the series is output gear 1, the second is output gear 2, the third is output gear 3, the fourth is output gear 4, the fifth is output gear 5, the sixth is output reverse gear. Here, the number of teeth or pitch diameter goes on decreasing from output gear 1 to output reverse gear. The distance between consecutive output gears should always be equal to the distance between two consecutive layshaft gears because the corresponding layshaft gear and output gears should always be in mesh.

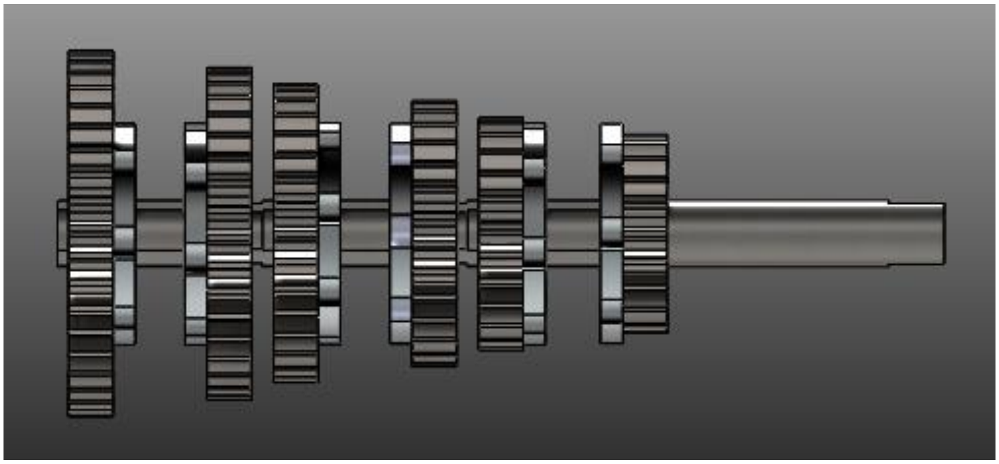

Figure 52, Output Shaft and Output Gears Sub-assembly

This sub-assembly also contains synchronizing hub and sliding sleeve for the changing of gears. To fit the synchronizing hub, except gear 1 all the gears are to be hided. Then the synchronizing hub mates to the output shaft.

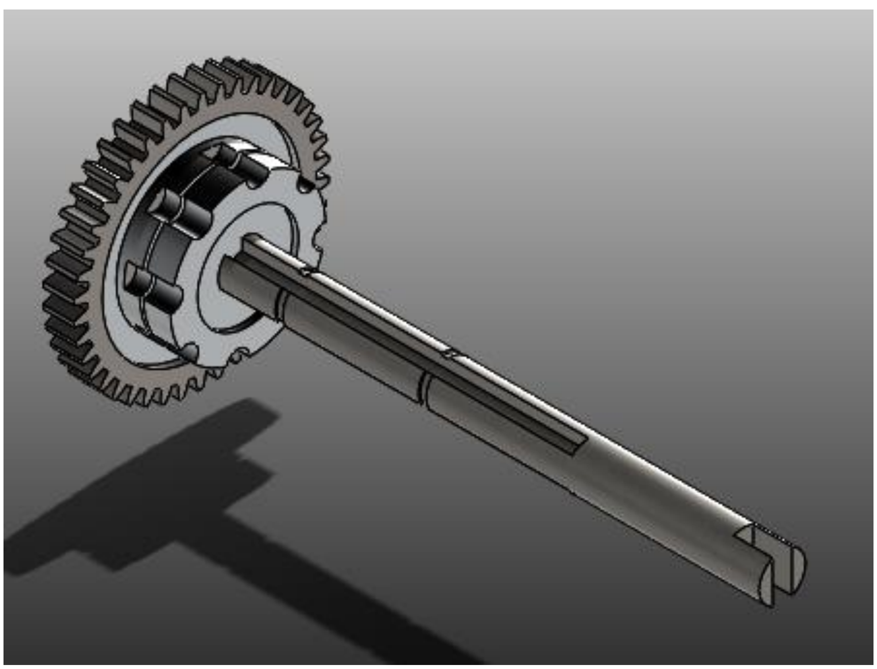

Figure 53, Output Shaft and Synchronizing Hub Sub-assembly

Now, a sliding sleeve is added over the synchronizing hub. This sleeve slides over synchronizing hub, output gear 1 and output gear 2. This sliding of sleeve causes changing of gears.

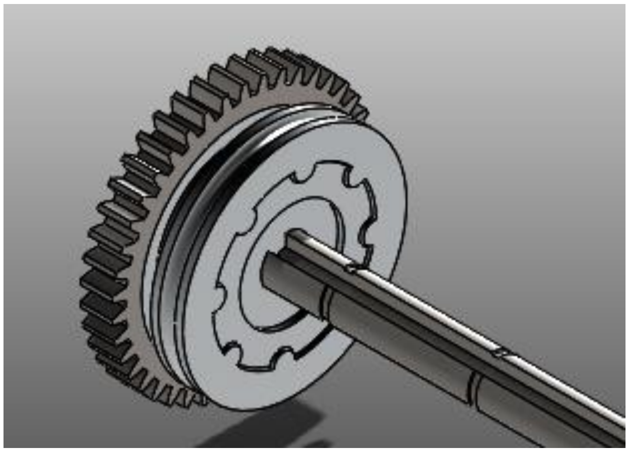

Figure 54, Output Shaft, Synchronizing Hub and Sliding Sleeve Sub-assembly

Similarly, the other two synchronizing hub and sliding sleeve are also added between other consecutive gears. Finally, the assembly is ready.

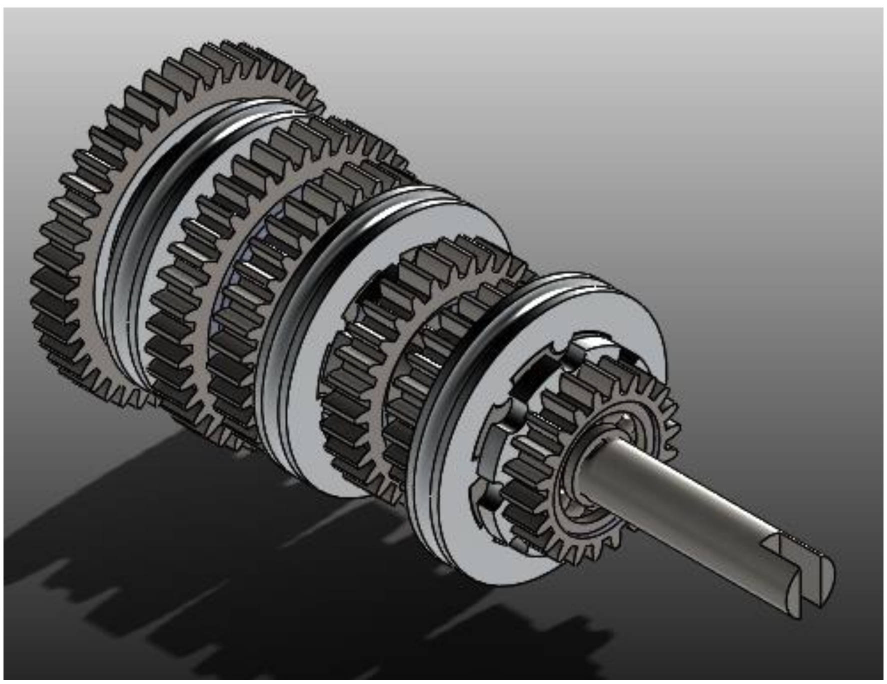

Figure 55, Output Shaft and Output Gears Sub-assembly

4.4. Manual Transmission Assembly

It is very important to know the pitch diameter of each gear. For meshing gears, the pitch circle of one gear should be in tangent with the pitch circle of corresponding gear. To begin with assembly, a sketch shown below is drawn on the right plane.

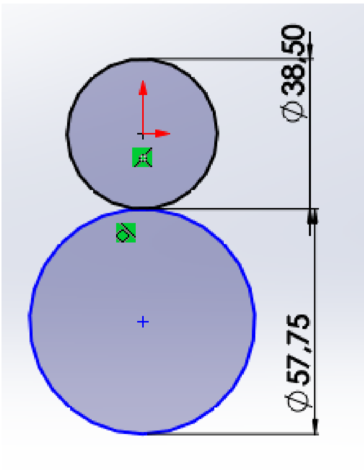

Figure 56, Sketch on Right Plane

Here the upper circle is 38.5 mm and lower circle is 57.75 mm because the pitch diameter of input gear is 38.5 mm and that of input layshaft gear is 57.75 mm.

Now, go to insert component, select motor shaft and input gear sub-assembly and layshaft and layshaft gear sub-assembly. Then, select concentric and coincident mates between the circles and the gears as shown below.

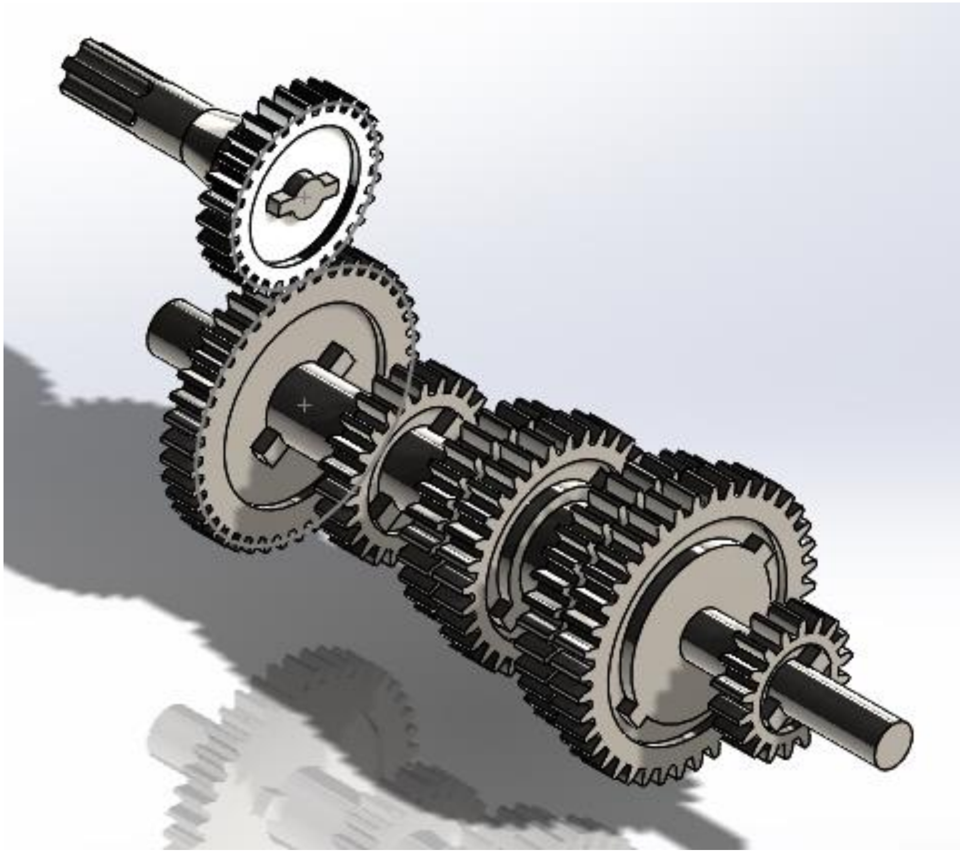

Figure 57, Manual Transmission Assembly

Next step is to insert the output block into the assembly. For meshing of output gears with the layshaft gears, the pitch circle of each layshaft gears should be in tangent with the pitch circle of corresponding output gears. It is known that the layshaft gear 1 has a pitch diameter of 36 mm and that of output gear 1 is 63 mm. So, a sketch shown below is drawn on the left planer face of layshaft gear 1.

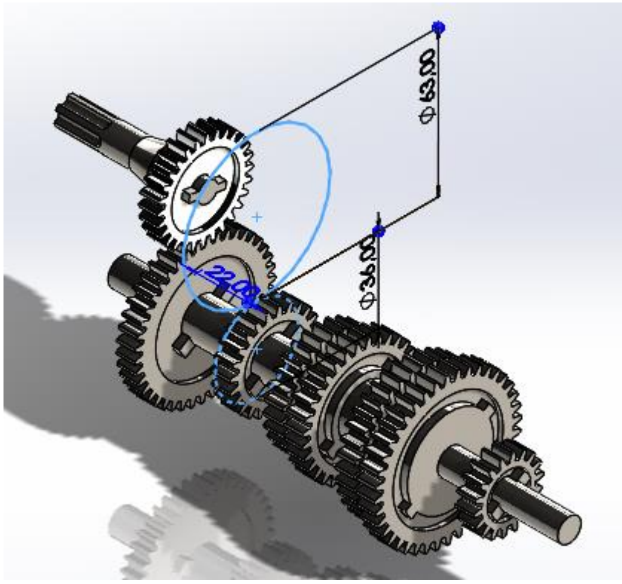

Figure 58, Manual Transmission Assembly

Now, insert the output block and output gears sub-assembly into the assembly. Then, create a concentric and coincident mates between the circle of diameter 63 mm and the output gear 1.

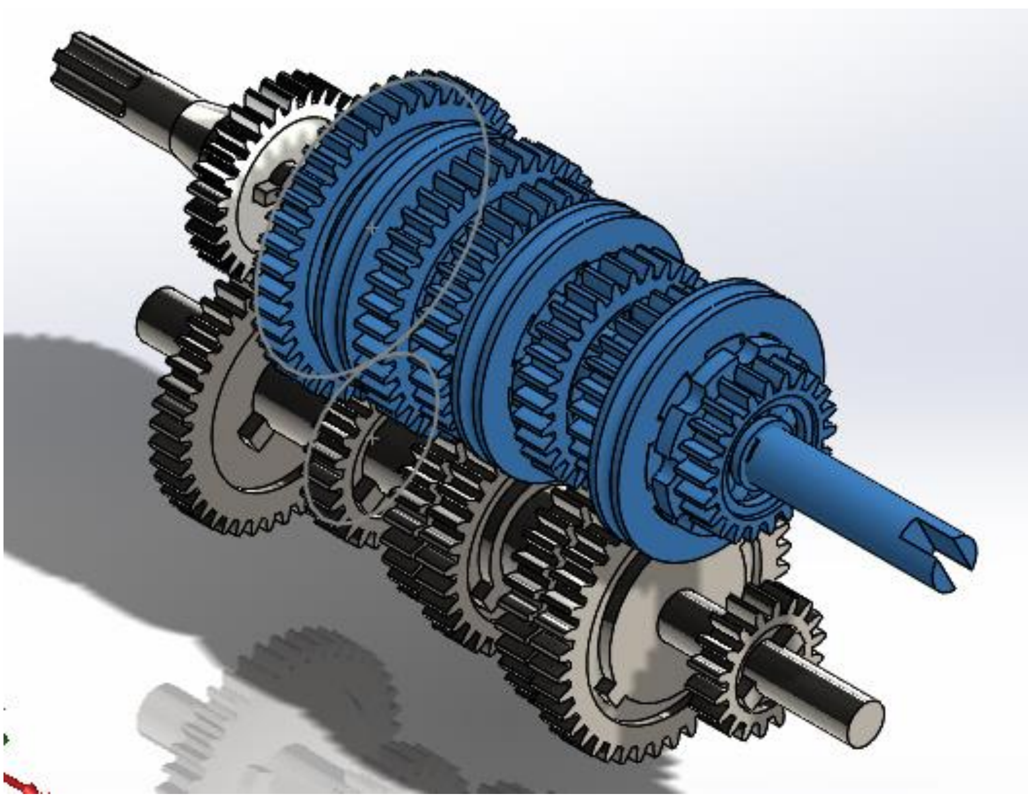

Figure 59, Manual Transmission Assembly

Next step is to insert an idler gear between the two reverse gears. Idlers are very important because they help to change the direction of rotation of the output shaft. To insert idler gear, a sketch shown below is drawn on the right planer face of the reverse gear.

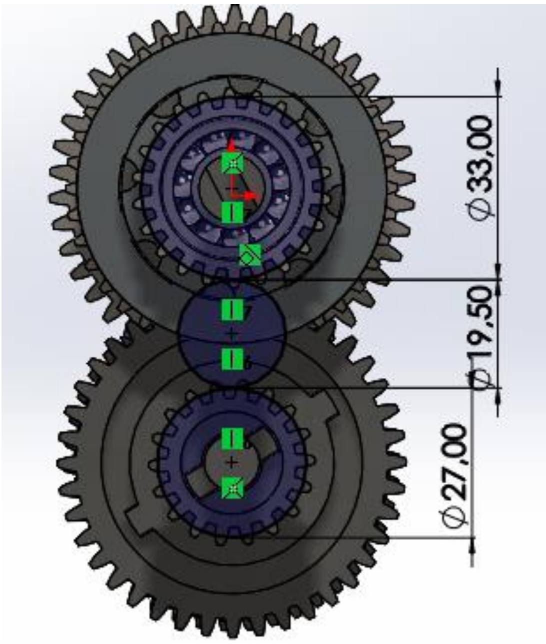

Figure 60, Manual Transmission Assembly

Now, insert the idler gear and create a concentric and coincident mates between the circle of diameter 19.5 mm and idler gear.

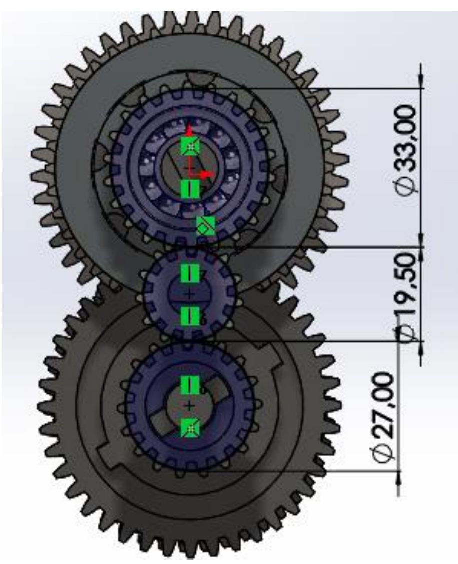

Figure 61, Manual Transmission Assembly

Since, the distance between consecutive output gears was designed to be equal to the distance between two consecutive layshaft gears, all the corresponding layshaft gears and output gears are now aligned.

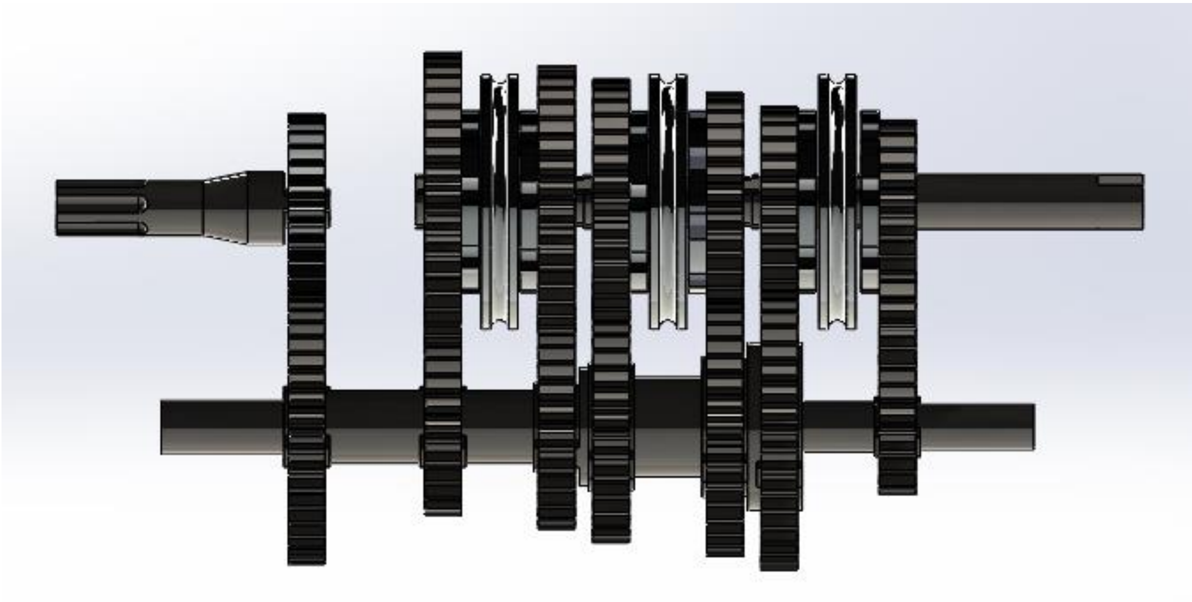

Figure 62, Manual Transmission Assembly

Until now, though the gears are aligned, the rotation of one gear does not affect the rotation of its corresponding gear. This is because, there is no mate relation established between the gears.

To mate the corresponding gears, orient the assembly to front view and zoom if required. Make sure that the gear teeth are not intersecting or touching.

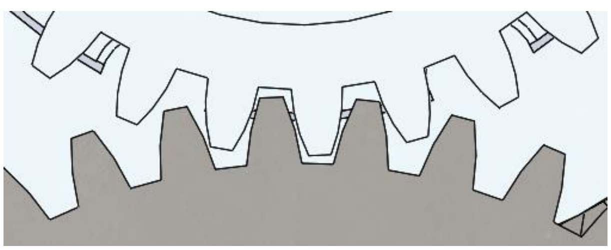

Figure 63, Manual Transmission Assembly

Now, to add mechanical mates between two gears, go to Mechanical Mate, select the two corresponding faces of gears and input the pitch diameter value in the Teeth/Diameter box. Now the two gears are in mesh.

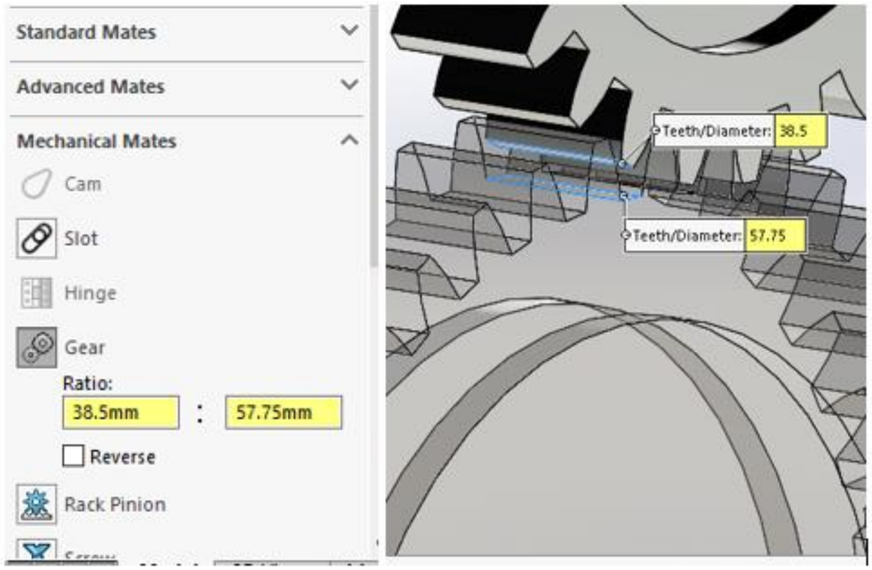

Figure 64, Manual Transmission Assembly

Similarly, mechanical mates for all the remaining gears are added. The manual transmission gear block is now ready.

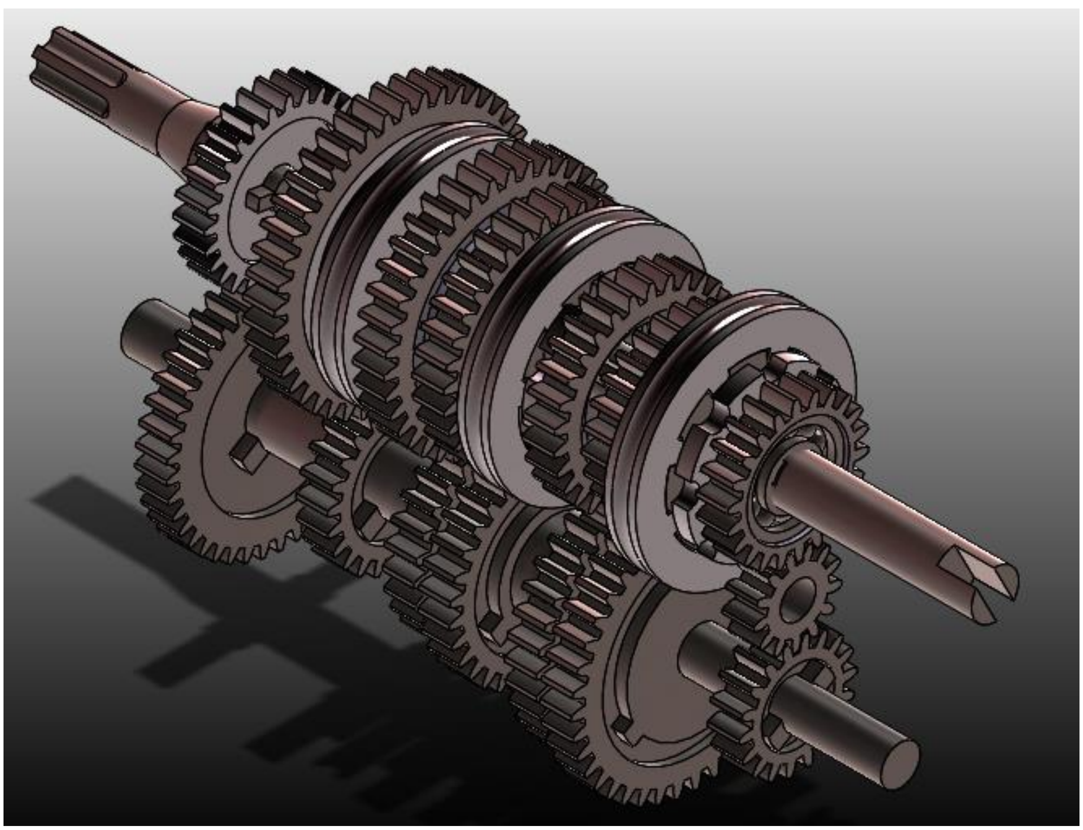

Figure 65, Manual Transmission Assembly

Finally, the gear block is fitted into the gear case using suitable mates and 10-12 sized bearings.

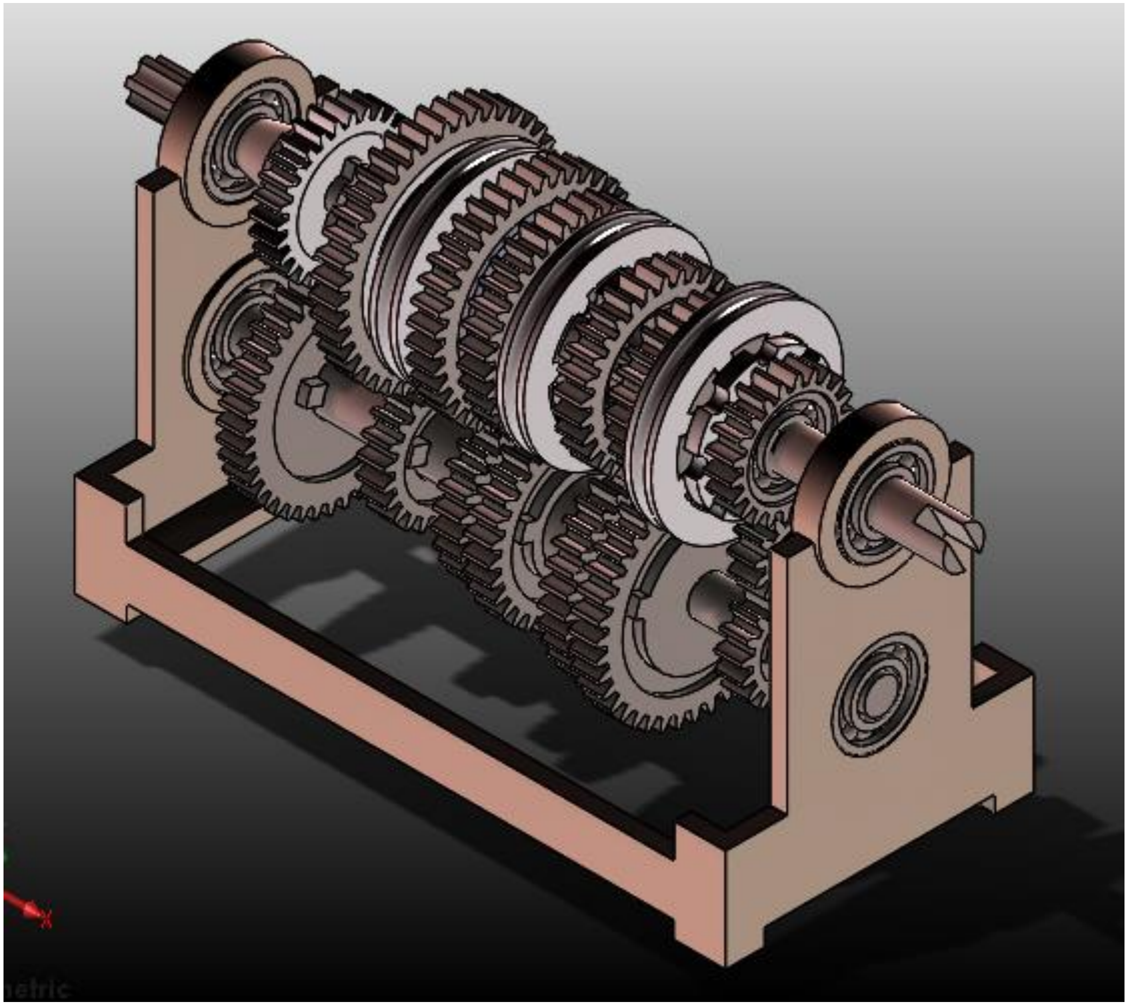

Figure 66, Manual Transmission Assembly

4.5. Differential Block Assembly

To begin with differential block assembly, create a new assembly and insert differential gear case, pinion gear and ring gear into the assembly. Then, create a concentric mate between the hub of each gear and semicircular cut of the gear case as shown below.

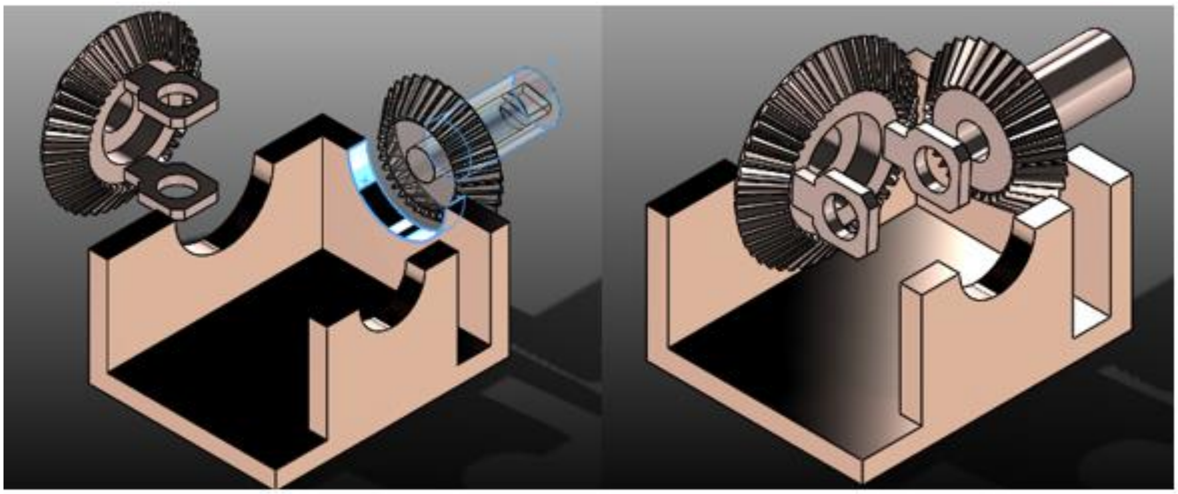

Figure 67, Differential Assembly

To mate the corresponding pinion and ring gears, orient the assembly to isometric view and zoom if required. Bring the gears close to each other but make sure that the gear teeth are not intersecting or touching. Then insert the mechanical mate between the gears as before by using corresponding pitch diameters.

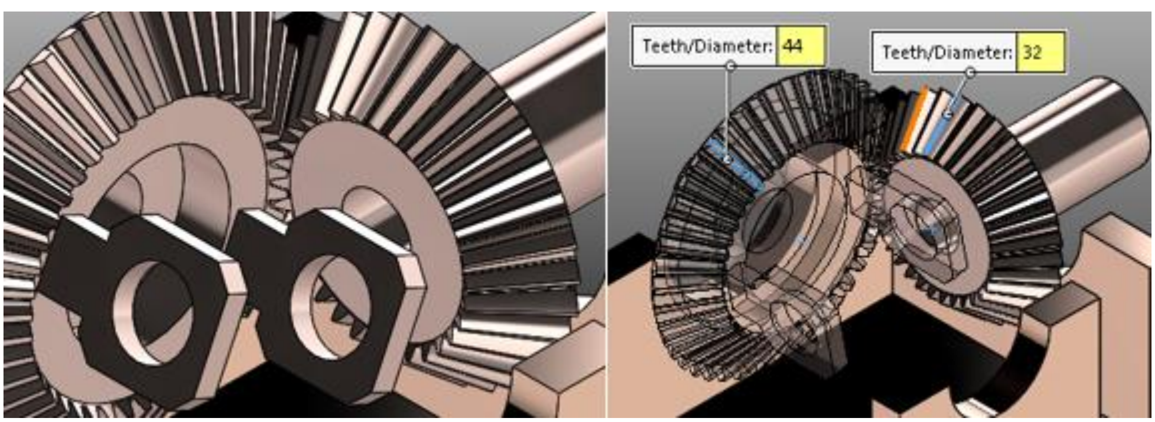

Figure 68, Differential Assembly

Now, insert two 10-20 sized bearings in the hub of pinion and ring gear and one 10-12 sized bearing at the center of ring gear as shown below.

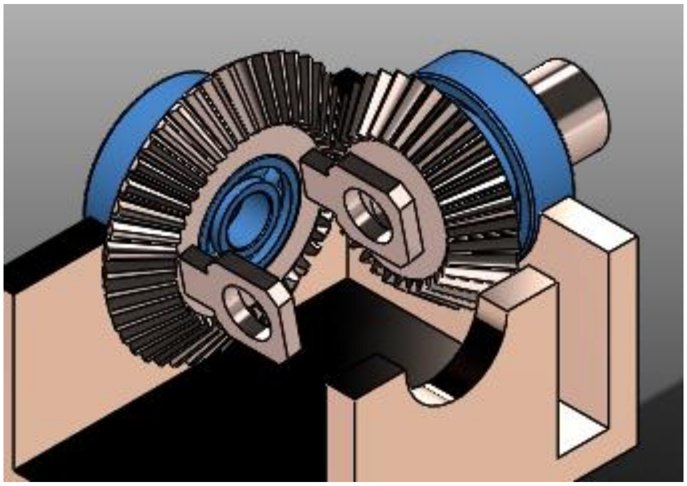

Figure 69, Differential Assembly

Now, insert two spider gear and two side gears to the assembly and mate with each other as done for pinion and ring gear. Then insert 0-2 sized bearing to hub of each spider gear and 10-12 sized bearing to the hub of side gear.

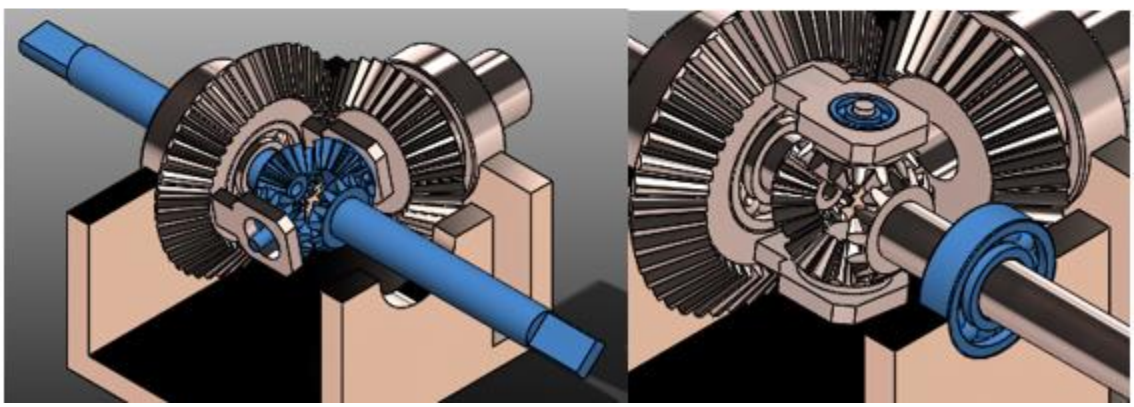

Figure 70, Differential Assembly

Finally, insert two wheels at the end of each side gears, then the differential assembly is ready.

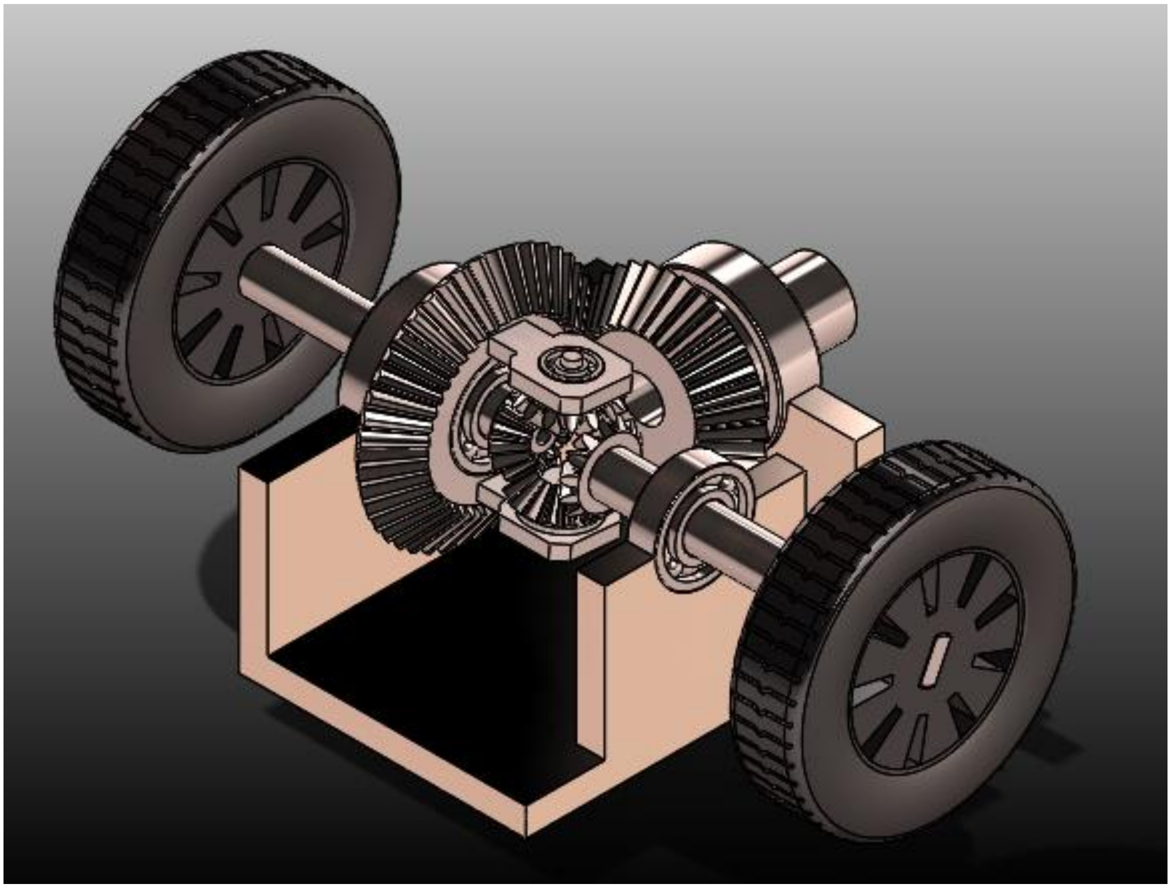

Figure 71, Differential Assembly

4.6. Final Assembly

Now, since the manual transmission block and the differential block are ready, the whole transmission is system is made ready by connecting these two blocks by a transmission shaft.

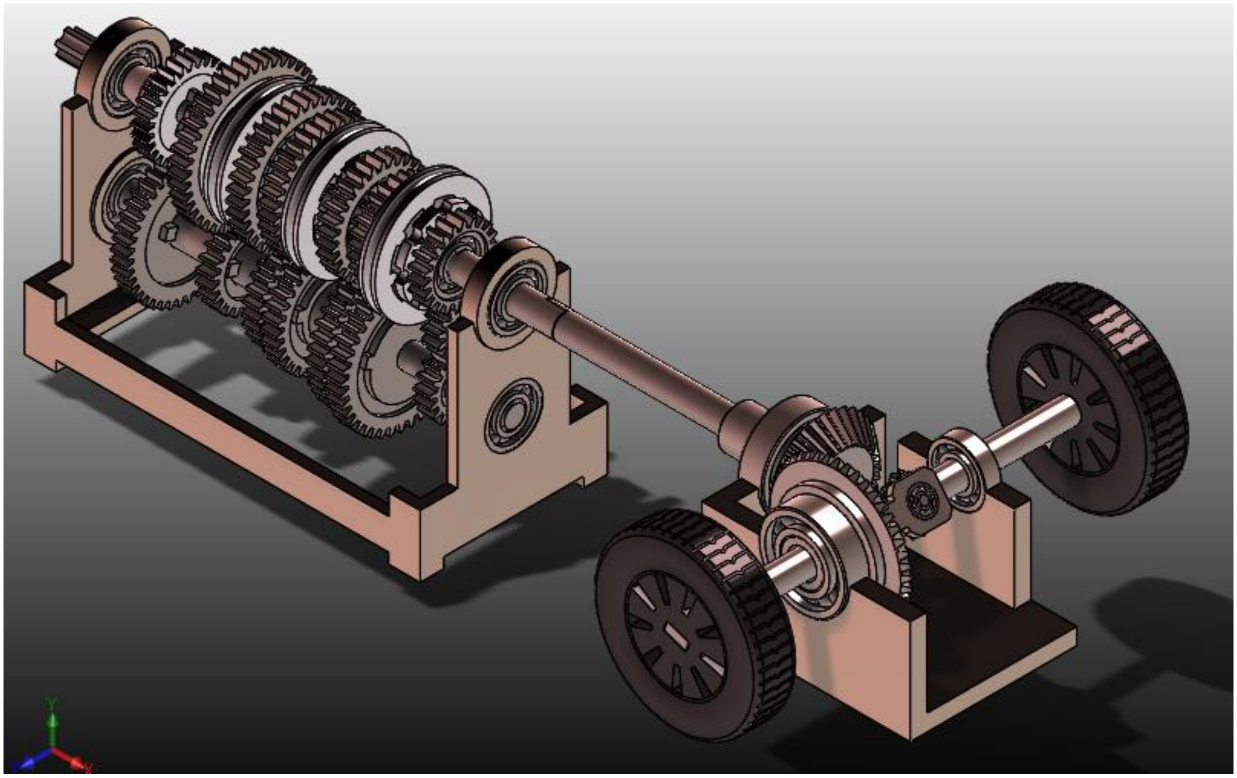

Figure 72, Final Assembly

5. DRAWING FROM ASSEMBLY

Below is the engineering drawing of the whole assembly showing each parts in exploded view.

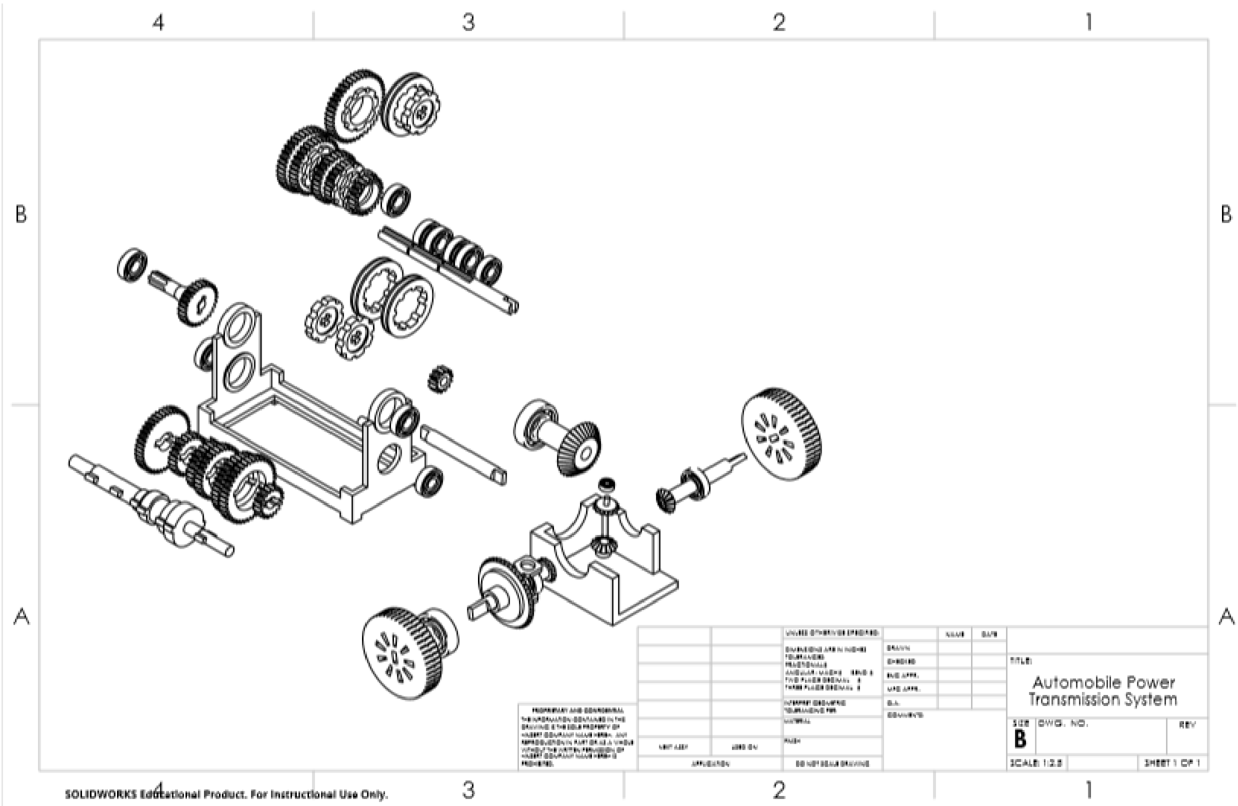

Figure 73, Drawing of Assembly

6. 3D PRINTING

MakerBot Print is fairly simple software that allows user to add design file and print a 3D model using the design file. The model is analyzed in virtual representation of build plate before printing for calculating the printing time and checking preview. Suitable orientation that needs less support material and less time to be printed is selected for printing. Now, select “Replicator 5th Gen” for printer, select “Export” and save the file. The design consisted of 56 parts in total. All the gears are printed using tough PLA and all the shafts and support structures are printed using normal PLA. Bearings are attached to the shafts which allows the gears to rotate freely. Below is the image of 3D printed model.

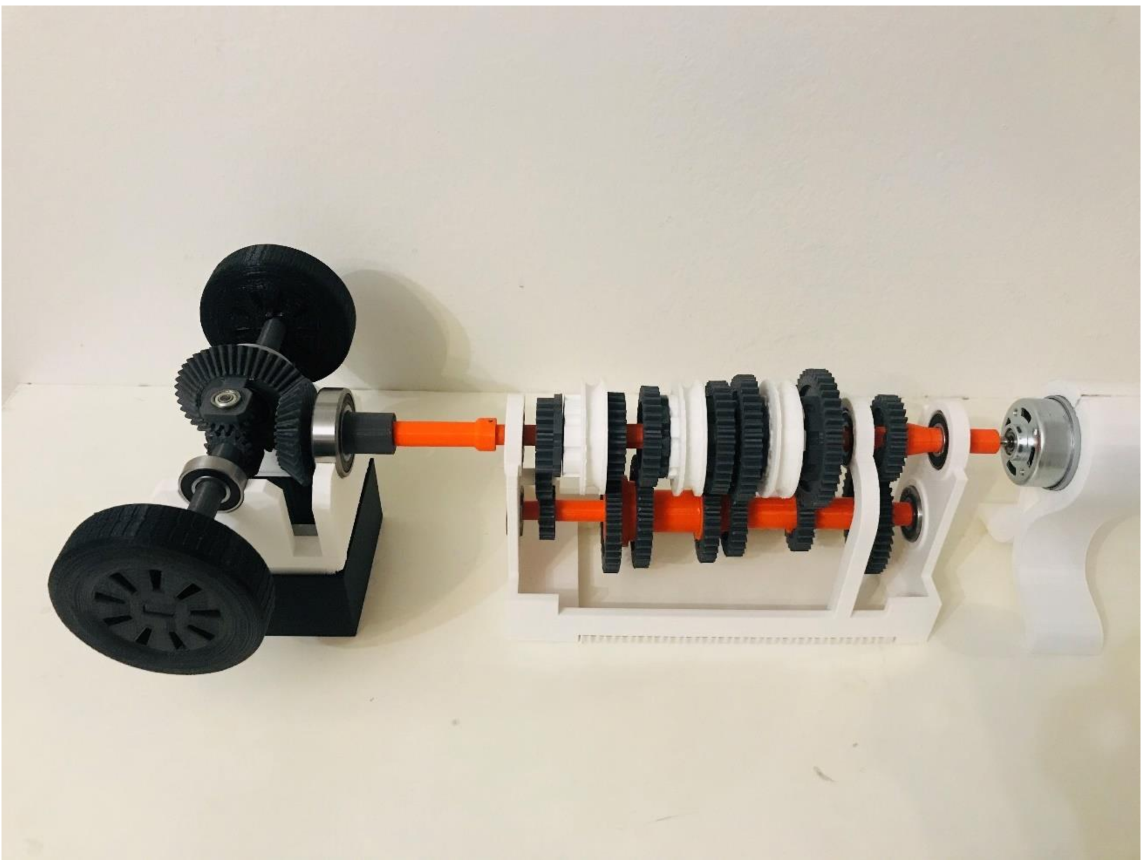

Figure 74, 3D Printed Manual Transmission System

7. MOTION ANALYSIS USING SOLIDWORKS

SolidWorks motion analysis is a powerful tool that helps to simulate a moving or dynamic system. It is a virtual simulation tool that provides the designers with the ability to get key performance parameters regarding a design. Depending upon the output required, Motion analysis can be used to analyze the motion of individual parts or whole system. This helps engineers to assure that each members are functioning correctly and thus helps to minimize the production cost of the product without compromising with the required strength of the part. Various parameters that can be calculated are forces, acceleration, torques, angular velocities, power, etc.

To begin with the motion analysis of gears, open the manual transmission assembly and make sure all the gears are mated correctly. Go to Add-ins and turn on SolidWorks Motion. Then, go to motion study 1 on the bottom left of the graphics area or create a new one by right click, create new motion study. From the pull-down menu switch to motion analysis.

Now, go to motor on the toolbar. Under motor type select rotary motor. Then, select motor shaft for the Component/Direction.

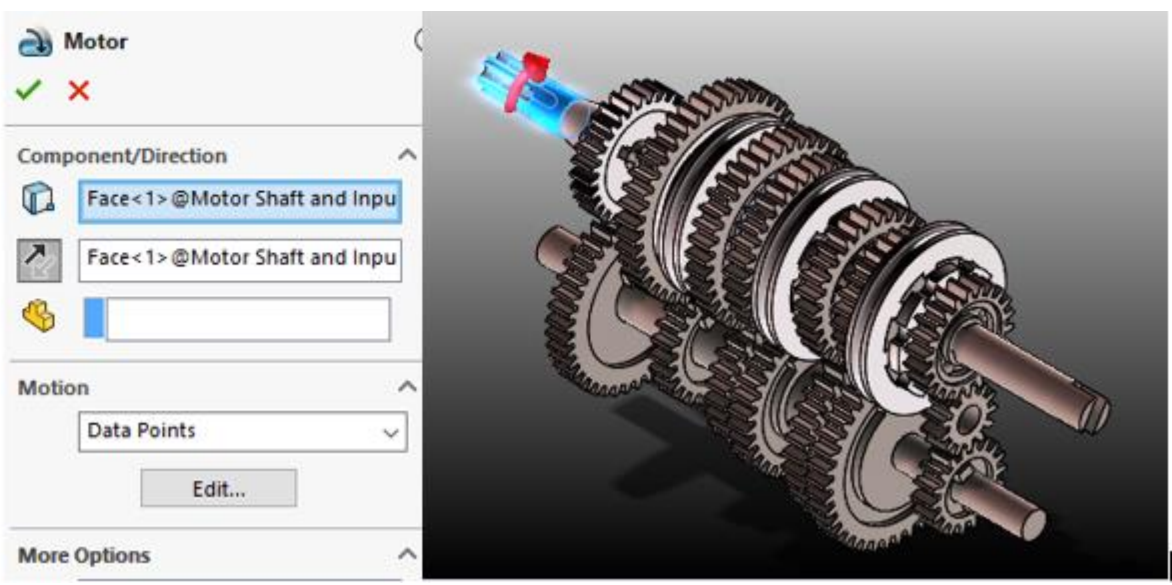

Figure 75, Motion Analysis

To define the motion parameters, scroll down to motion, select data points. This brings a function builder tab on the screen.

- Select Velocity (deg/s) for y-axis, Time (s) for x-axis and set the Interpolation type to Linear.
- Input the values to the Time and Value table.
- Since the motor starts from rest, at zero second, the velocity should be zero degrees. To input the next values, click to add row.
- It is assumed that the motor accelerates and reaches a speed of 583 rpm (3500 deg/s) in 4 seconds. So, at 4 seconds the velocity will be 3500 deg/s. Then the motor runs at a constant speed of 3500 deg/s for 10 sec.

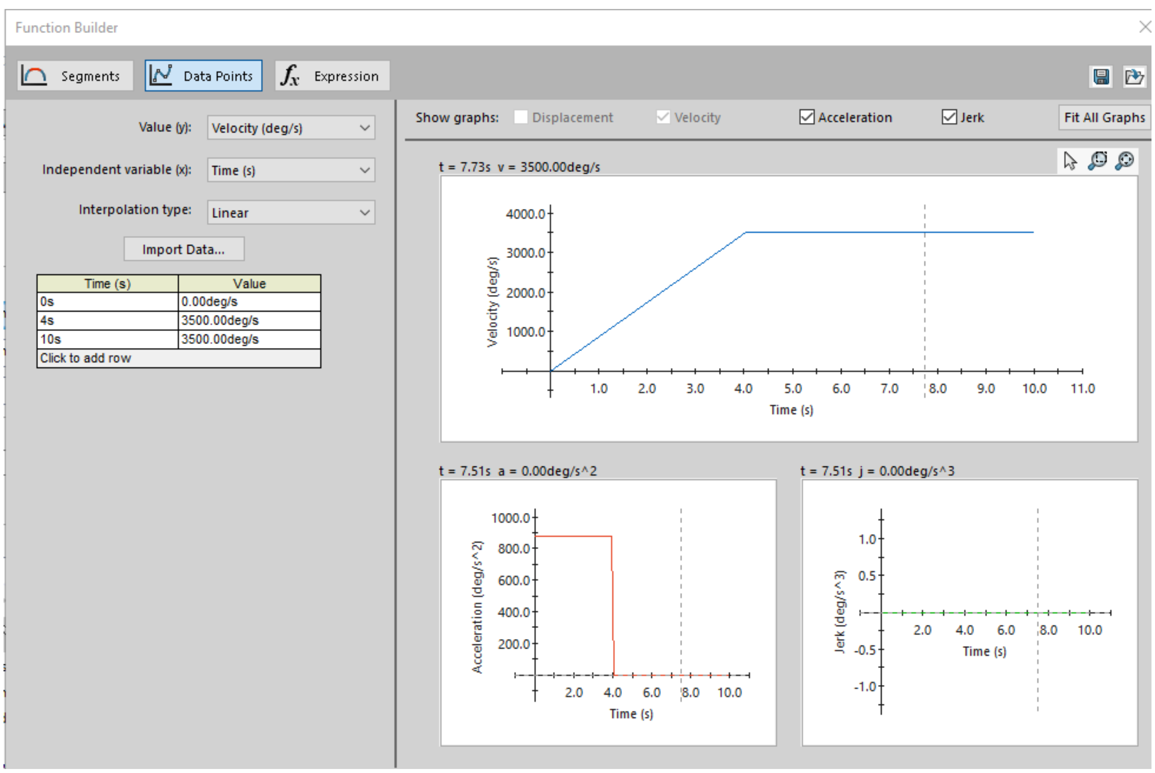

Figure 76, Motion Analysis

- Now, go to Contact on to simulate the contact between the selected components. Under components, select motor the component as shown below in blue color.

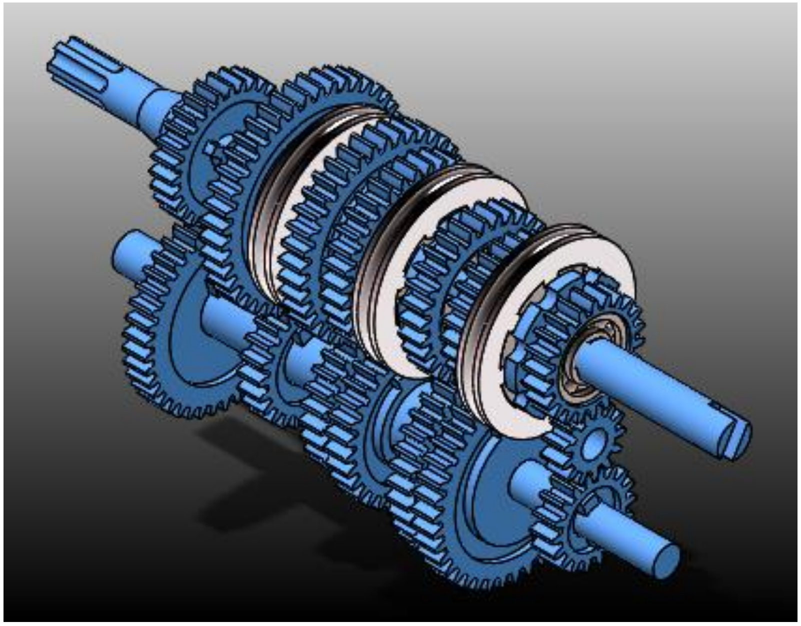

Figure 77, Motion Analysis

- Now, drag the key properties cursor to 10 seconds and click Calculate.

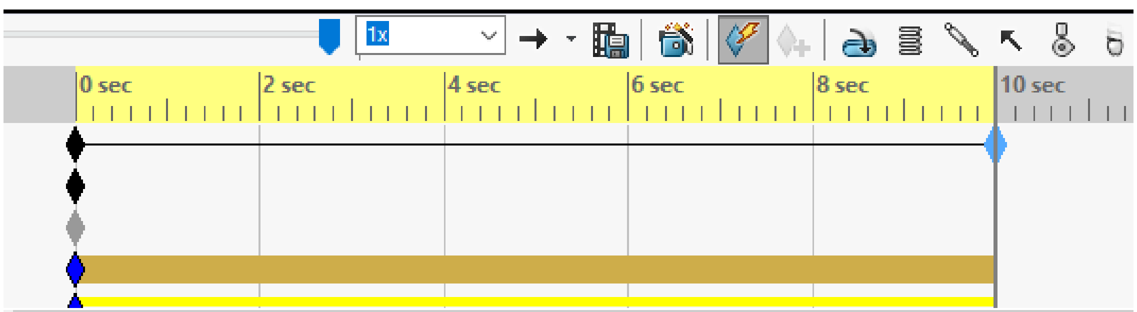

Figure 78, Motion Analysis

- To get the output data for a gear, click on Results and Plots. Under category select Displacement/Velocity/Acceleration, under sub-category select Angular Velocity, under result component select Magnitude, and select any planer face of the required gear.

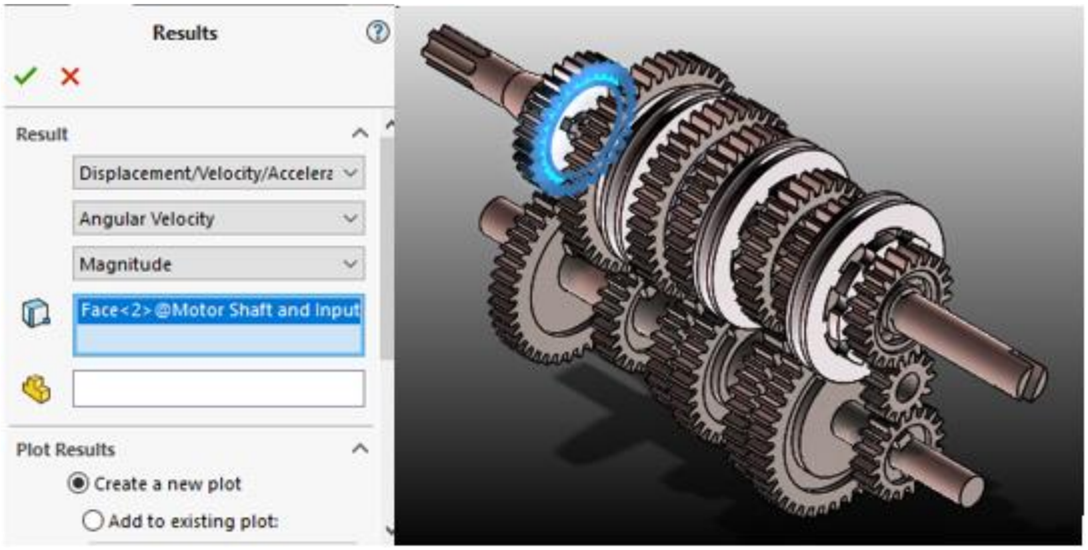

Figure 79, Motion Analysis

Now, the whole gear assembly is connected a power supply. The current is set to 5 amps and the voltage is set to 18 volts.

Therefore, $Power\ (P) = I \times V = 5 \times 18 = 90\ watts$

Also, the angular velocity of motor is set to 3500 deg/s (583 rpm),

$$\omega_1 = \frac{583 \times 2\pi}{60} = 61.07\ rad/s$$

Therefore, Motor Torque, $\tau_1 = \frac{P}{\omega_1} = \frac{90}{61.07} = 1.47\ Nm = 1473.71\ Nmm.$

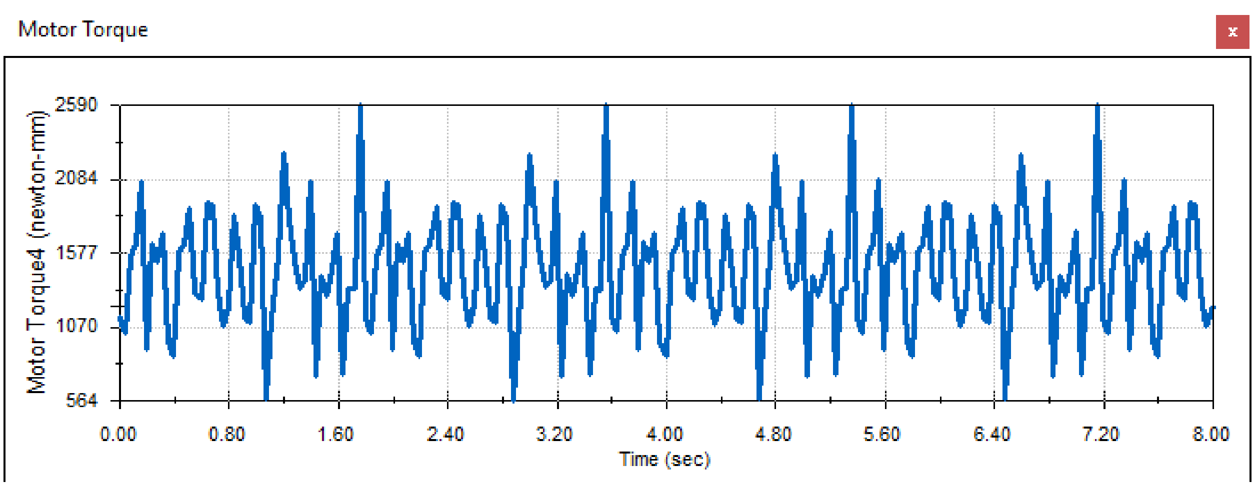

Figure 80, Motor Torque using Motion Analysis

7.1. Angular Velocity of Output Gear 1

Since the motor has an angular velocity of 3500 deg/s (583 rpm), the input gear should also have an angular velocity of 3500 deg/s. Using equation 23 for output gear 1;

$$\frac{\omega_{input}}{\omega_{output}} = \left(\frac{T_2}{T_1}\right)\left(\frac{T_4}{T_3}\right)$$

$$or, \frac{3500}{\omega_{output}} = \left(\frac{42}{28}\right)\left(\frac{42}{24}\right)$$

$$\therefore \omega_{output} = 1333.33$$

So, the angular velocity of output gear 1 is 1333.33 deg/s (222.22 rpm) This value should show very close resemblance to the value obtained by the motion analysis.

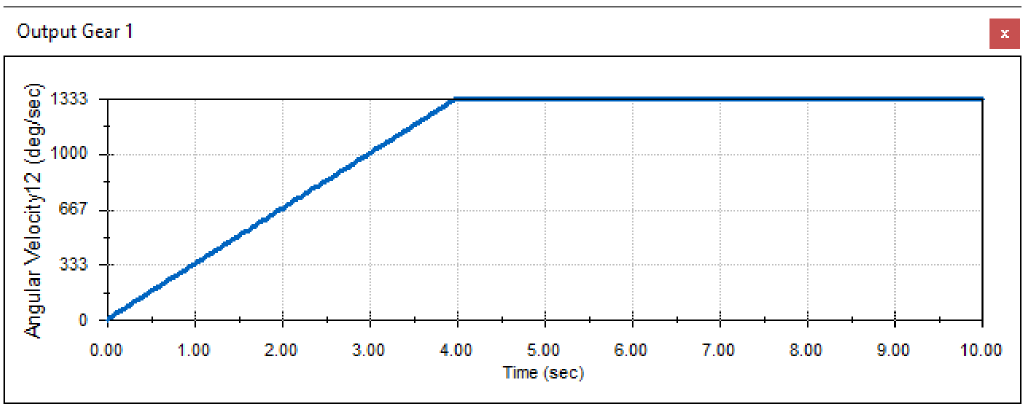

Figure 81, Angular Velocity of Output Gear 1 using SolidWorks Motion Analysis

7.2. Torque Transferred by Output Gear 1

Using equation 29 for output gear 1,

$$\frac{\tau_{output}}{\tau_{input}} = \left(\frac{T_2}{T_1}\right)\left(\frac{T_4}{T_3}\right)$$

$$or, \frac{\tau_{output}}{1.47} = \left(\frac{42}{28}\right)\left(\frac{42}{24}\right)$$

$$\therefore \tau_{output} = 3.85\ Nm$$

So, the torque transferred by output gear 1 is 3.85 Nm.

7.3. Angular Velocity of Output Gear 2

Similarly, for output gear 2,

$$\frac{\omega_{input}}{\omega_{output}} = \left(\frac{T_2}{T_1}\right)\left(\frac{T_4}{T_3}\right)$$

$$or, \frac{3500}{\omega_{output}} = \left(\frac{42}{28}\right)\left(\frac{38}{28}\right)$$

$$\therefore \omega_{output} = 1719.29$$

So, the angular velocity of output gear 2 is 1719.29 deg/s (286.54 rpm).

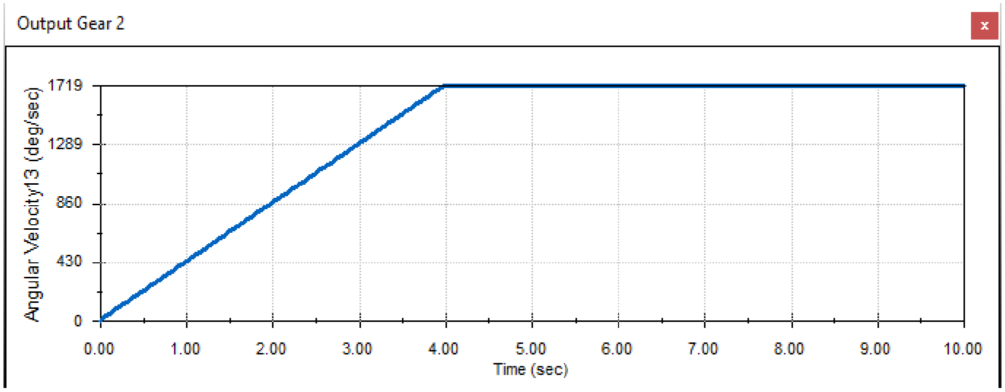

Figure 82, Angular Velocity of Output Gear 2 using SolidWorks Motion Analysis

7.4. Torque Transferred by Output gear 2

For output gear 2,

$$\frac{\tau_{output}}{\tau_{input}} = \left(\frac{T_2}{T_1}\right)\left(\frac{T_4}{T_3}\right)$$

$$or, \frac{\tau_{output}}{1.47} = \left(\frac{42}{28}\right)\left(\frac{38}{28}\right)$$

$$\therefore \tau_{output} = 2.99\ Nm$$

So, the torque transferred by output gear 2 is 2.99 Nm.

7.5. Angular Velocity of Output Gear 3

For output gear 3,

$$\frac{\omega_{input}}{\omega_{output}} = \left(\frac{T_2}{T_1}\right)\left(\frac{T_4}{T_3}\right)$$

$$or, \frac{3500}{\omega_{output}} = \left(\frac{42}{28}\right)\left(\frac{34}{32}\right)$$

$$\therefore \omega_{output} = 2196.07$$

So, the angular velocity of output gear 3 is $2196.07\ deg/s$ $(366\ rpm)$.

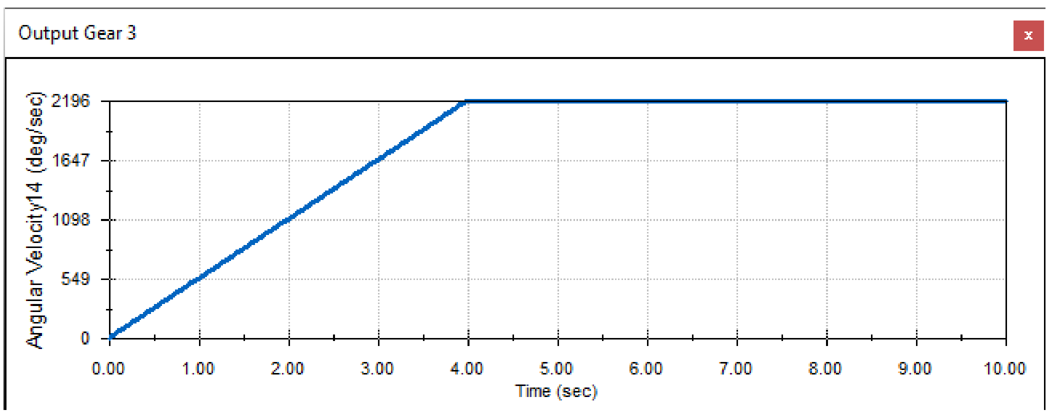

Figure 83, Angular Velocity of Output Gear 3 using SolidWorks Motion Analysis

7.6. Torque Transferred by Output gear 3

For output gear 3,

$$\frac{\tau_{output}}{\tau_{input}} = \left(\frac{T_2}{T_1}\right)\left(\frac{T_4}{T_3}\right)$$

$$or, \frac{\tau_{output}}{1.47} = \left(\frac{42}{28}\right)\left(\frac{34}{32}\right)$$

$$\therefore \tau_{output} = 2.34\ Nm$$

So, the torque transferred by output gear 3 is $2.34\ Nm$.

7.7. Angular Velocity of Output Gear 4

For output gear 4,

$$\frac{\omega_{input}}{\omega_{output}} = \left(\frac{T_2}{T_1}\right)\left(\frac{T_4}{T_3}\right)$$

$$or, \frac{3500}{\omega_{output}} = \left(\frac{42}{28}\right)\left(\frac{30}{36}\right)$$

$$\therefore \omega_{output} = 2800$$

So, the angular velocity of output gear 4 is $2800\ deg/s$ $(466.66\ rpm)$.

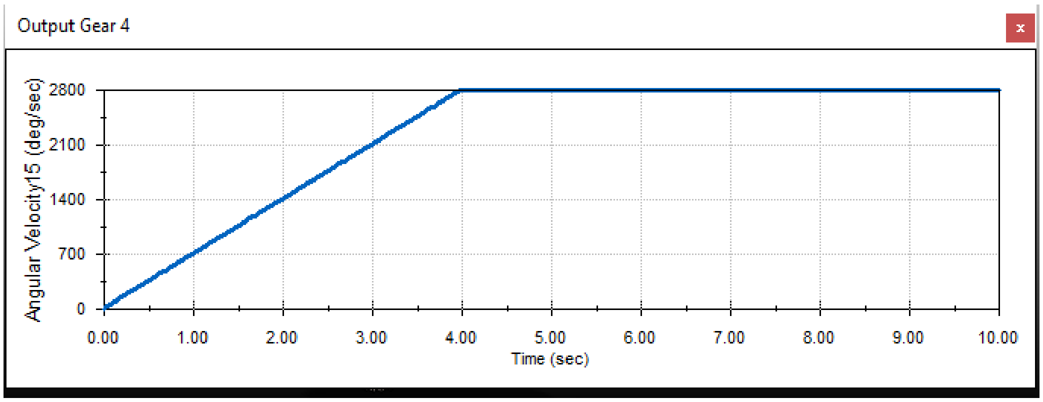

Figure 84, Angular Velocity of Output Gear 4 using SolidWorks Motion Analysis

7.8. Torque Transferred by Output gear 4

For output gear 4,

$$\frac{\tau_{output}}{\tau_{input}} = \left(\frac{T_2}{T_1}\right)\left(\frac{T_4}{T_3}\right)$$

$$or, \frac{\tau_{output}}{1.47} = \left(\frac{42}{28}\right)\left(\frac{30}{36}\right)$$

$$\therefore \tau_{output} = 1.83\ Nm$$

So, the torque transferred by output gear 4 is $1.83\ Nm$.

7.9. Angular Velocity of Output Gear 5

For output gear 5,

$$\frac{\omega_{input}}{\omega_{output}} = \left(\frac{T_2}{T_1}\right)\left(\frac{T_4}{T_3}\right)$$

$$or, \frac{3500}{\omega_{output}} = \left(\frac{42}{28}\right)\left(\frac{26}{40}\right)$$

$$\therefore \omega_{output} = 3589.74$$

So, the angular velocity of output gear 5 is $3589.74\ deg/s$ ($598.29\ rpm$).

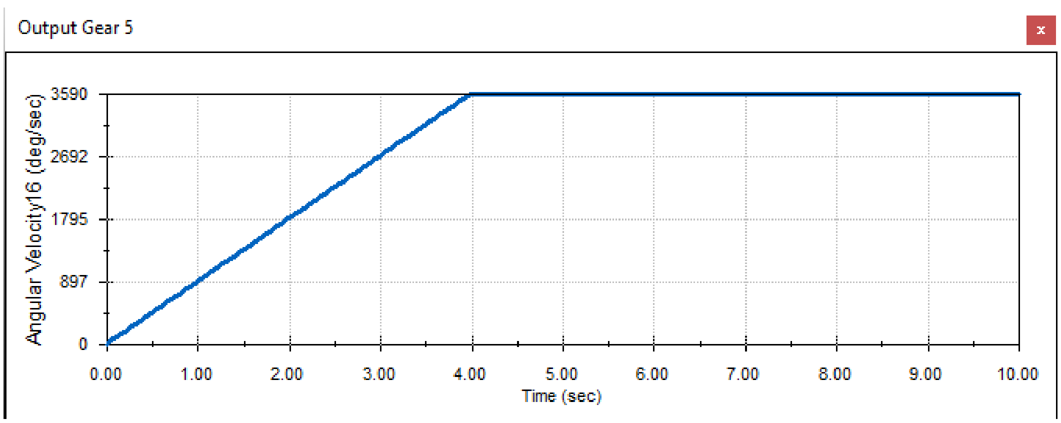

Figure 85, Angular Velocity of Output Gear 5 using SolidWorks Motion Analysis

7.10. Torque Transferred by Output gear 5

For output gear 5,

$$\frac{\tau_{output}}{\tau_{input}} = \left(\frac{T_2}{T_1}\right)\left(\frac{T_4}{T_3}\right)$$

$$or, \frac{\tau_{output}}{1.47} = \left(\frac{42}{28}\right)\left(\frac{26}{40}\right)$$

$$\therefore \tau_{output} = 1.43\ Nm$$

So, the torque transferred by output gear 5 is $1.43\ Nm$.

8. EXPERIMENTAL MOTION ANALYSIS

For experimental measurement of angular velocity, iPhone slow motion camera was used. The camera was set to 240 frames per second (fps). Its means it takes 240 images per seconds.

So, $time\ for\ one\ frame = \frac{1}{240}\ s$

If, $f\ be\ the\ number\ of\ frames\ in\ 1\ revolution,$

Then, $time\ in\ seconds\ for\ 1\ rev\ (t) = \frac{1}{240} \times f$

Conversely, $total\ revolutions\ in\ 60\ s\ (RPM) = 60/t$

The graph below shows the number of revolutions per minute made by the wheels upon the engagement of respected gears 1,2,3,4 and 5.

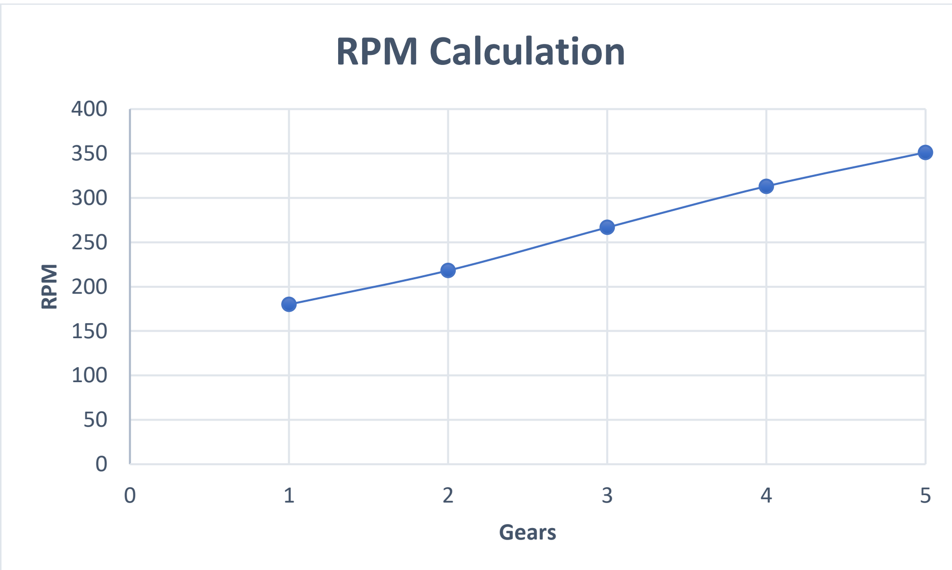

Figure 86, Experimental Calculation of Wheel Speed

The graph above shows that the angular velocity of wheels increases as the gear increases.

9. TESTING MATERIAL PROPERTIES

Since this thesis uses a relatively new material, Tough PLA, it was obvious to investigate about its material properties and compare that to a normal PLA. For this, two kinds of tests were conducted, Tensile Test and 3Point Bending Test. The test was conducted using Testometric M 350=5CT in B225 lab at Arcada UAS.

9.1. Tensile Test

The tensile test piece was designed and tested based on Finnish Standards Association SFS, ISO 157, type 1A and testing speed of 50 mm per minute.

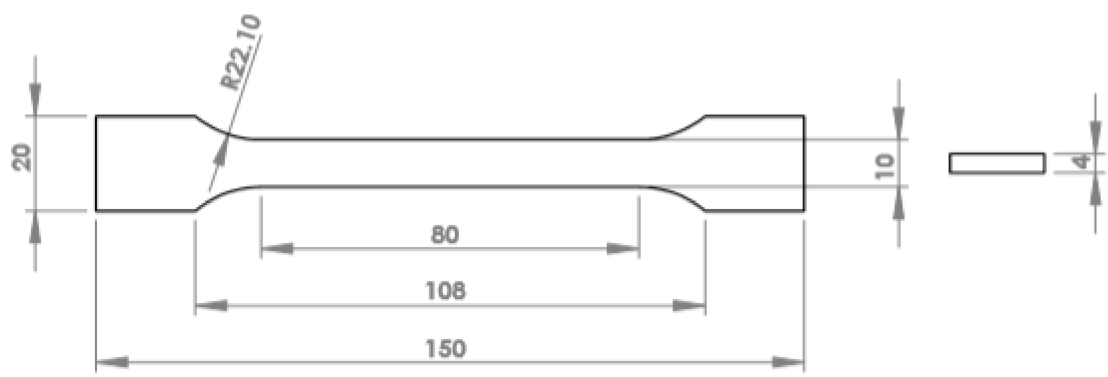

Figure 87, Tensile Test Specimen

Five different samples of tough PLA and normal PLA in the form of dog bone were tested. The Testometric device was connected to data acquisition software. The data were gathered in the form of Excel file and was then plotted on engineering stress-strain curves to compare the samples. The data was used to calculate and compare young’s modulus, elongation at break and ultimate tensile stress.

9.1.1. Results

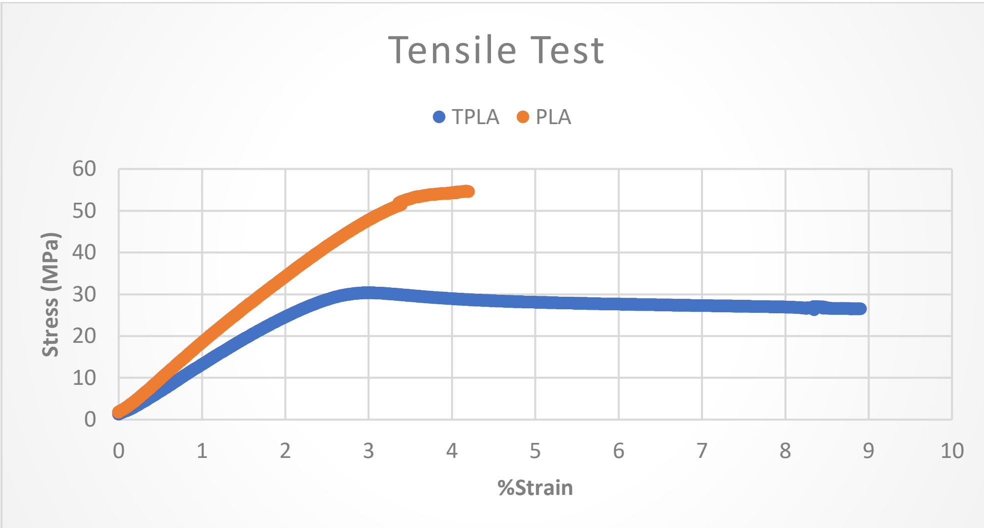

Figure 88, Tensile Strength Comparison Between Tough PLA and Normal PLA

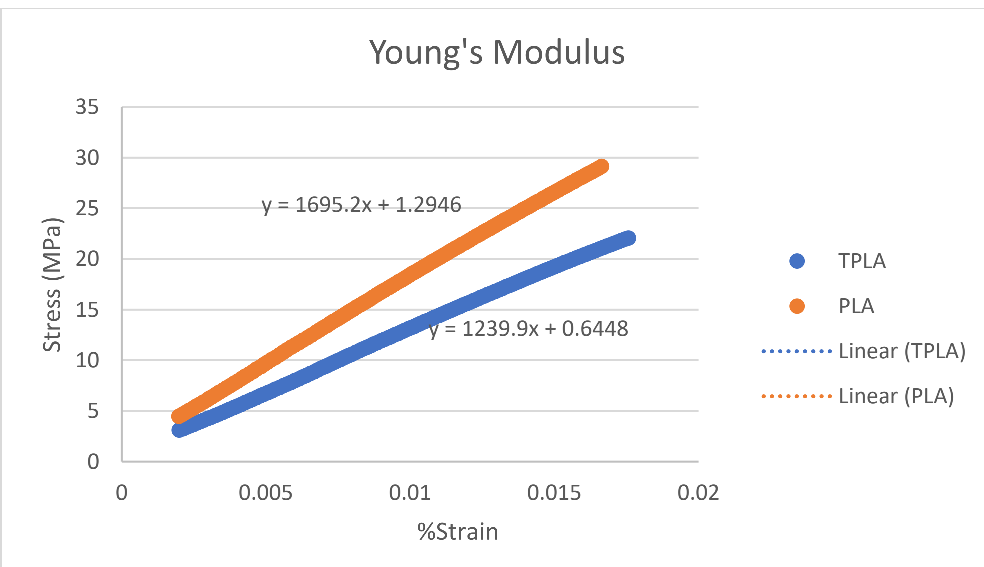

Figure 89, Young's Modulus Comparison

From the graphs, it is seen that the ultimate tensile strength of normal PLA is nearly 55 MPa and that of tough PLA is nearly 30 MPa. The Young's Modulus for normal PLA is 1.69 GPa and that of tough PLA is 1.23 GPa. But it can also be seen that tough PLA is more ductile than normal PLA as it undergoes significant amount of plastic deformation before it breaks. Also, Tough PLA absorbs much more amount of energy before it breaks than normal PLA.

9.2. Flexural Test

Flexural test piece was designed and tested based on Finnish Standards Association SFS, EN ISO 178:2019 type 1A and testing speed of 100 mm per minute. Flexural test is done to test the mechanical properties of brittle material.

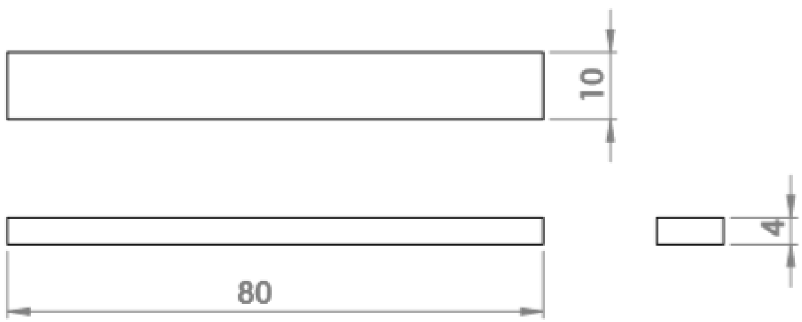

Figure 90, Flexural Test Specimen

Figure 91, Flexural test of tough PLA

Samples for flexural test are usually rectangular beams. These beams are placed over two supports and the load is applied at the middle of the beam. This load creates a tensile stress on the convex side of the beam and compressive stress on the concave side. Because of this load a moment is created opposite to the direction of applied load. This bending moment tends to break the material but the normal stress developed between the cross-section of the beam try to resist the load. So, the flexural strength is the measure of material's ability to resist deformation under load. It is measured as the highest stress value developed within a material while it ruptures.

Figure 92, Flexural Test Showing Brittle Nature of Normal PLA and Ductile Nature of Tough PLA

The data from 5 different samples of each normal PLA and tough PLA were gathered and plotted using Excel to obtain stress-strain curves. The data was used to calculate and compare flexural modulus, elongation at break and brittleness of a material.

9.2.1. Results

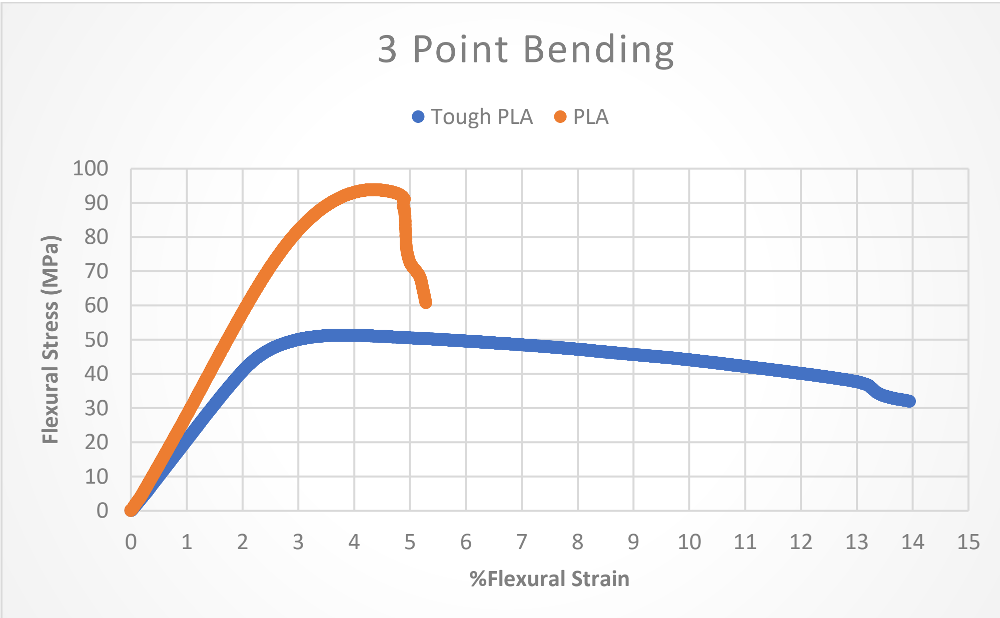

Figure 93, Flexural Strength Comparison Between Tough PLA and Normal PLA

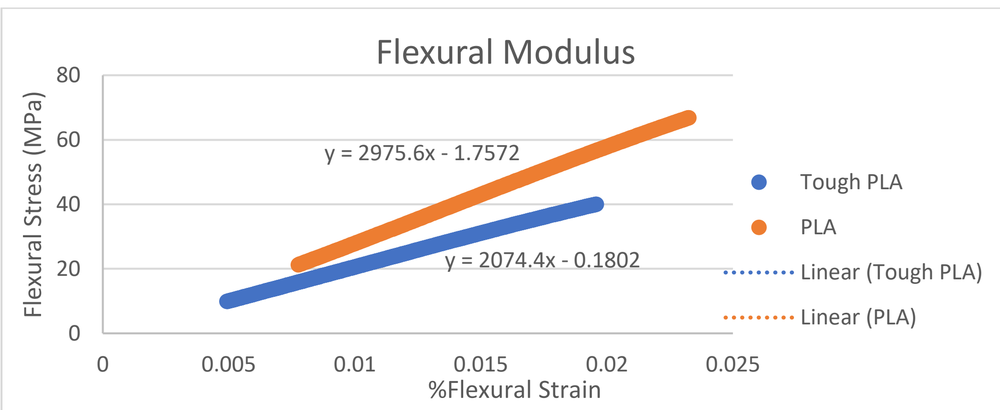

Figure 94, Flexural Modulus Comparison

From the graphs, it is clear that tough PLA bends or flexes much more before break compared to normal PLA. The ultimate flexural strength of normal PLA is nearly 95 MPa and that of tough PLA is nearly 52 MPa. Flexural modulus of normal PLA is 2.97 GPa and that of tough PLA is 2.07 GPa. It can also be seen that the normal PLA undergoes very little plastic deformation under stress as compare to tough PLA.

10. DISCUSSION

10.1. Designing and 3D Printing of Gears

The designing of gears was first stage in this thesis. Once again, SolidWorks proved itself to be very versatile for designing and motion analysis of gears. The simulation of gears motion was very realistic. The results obtained from the motion analysis were very precise to the results obtained via theoretic calculations. However, the experimental angular velocities calculated using slow motion camera were a bit lower. This might be due to uneven cooling of the printed parts that led to lack of clearance and caused friction between some gears.

Various problems were faced while 3D printing process. Since the overall assembly consists of two types of material, the parts showed irregular expansions and mating problems. Even though, suitable clearance was maintained during the designing process, the printed parts didn't fit. To solve this a clearance of 0.2 mm was added to the design of mating parts and printed again. Finally, all the parts were assembled and functioned as they were supposed.

10.2. Material Properties

From the result obtained via tensile and flexural tests, it is clear that the ultimate tensile strength and flexural strength of normal PLA are nearly double to that of tough PLA. Also, the Young's Modulus and flexural modulus for normal PLA are fairly higher than tough PLA. So, it is clear that normal PLA can handle large amount of load before break compared to tough PLA. But when it fails, it breaks completely, without significant plastic deformation, showing characteristic brittle nature. Whereas, tough PLA undergoes a very long plastic deformation before it breaks. Both the tests suggest that tough PLA can absorb much more amount energy before failure.

Tough PLA has lower ultimate tensile strength compared to normal PLA, but it is strong enough to be used as 3D printing material. The plastic deformation range of tough PLA is significantly greater than that of normal PLA and ABS. It has suitable combination of ultimate tensile strength and ductility that leads to its high toughness. Tough PLA can be a material of choice for designs that require high flexibility and undergo torsion like hinges and interlocking parts.

Failure of gear teeth occurs due to cyclic stresses developed at the point of contact between two meshing teeth. Since the surface hardness of both tough and normal PLA are nearly the same, wearing and tearing due to other environmental factors can be neglected under similar testing conditions. From force analysis of spur gear teeth or Lewis equation, it is known that tangential force between the gear pairs determines the magnitude of torque transferred by the gear. This tangential force tries to deform and break the teeth apart from the gear. Since, tough PLA can handle significant amount of torsion and deformation than normal PLA, it is more suitable to use tough PLA instead of normal PLA as gears.

11. CONCLUSION

The overall goal of the thesis is achieved and is fairly satisfying. The 3D printed power transmission system is similar to its parent design and functions very well. The printed gears meshed very well and the results for angular velocities of gears are very satisfying i.e. the angular velocity of wheels increases as the gear is increased from 1^{st} to 5^{th}. Moreover, it was a great experience working with SolidWorks animation and motion analysis. Significant correlation is seen between the results obtained by motion analysis and theoretical calculations. Both tensile and flexural test shows that tough PLA can handle significant amount of torsion and deformation than normal PLA. This thesis also supports the viability of 3D printing for making real goods or prototypes. Finally, this design can also be used to explain the principle behind gear manufacturing and working of manual transmission system.

For further studies, one can study effects of printing parameters like infill density, filament thickness, part orientation, etc. on the print time and strength of tough PLA. Since, tough PLA can withstand more deformation and torsion, it would also be very interesting to explore the feasibility of tough PLA gears for machinery uses.

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