Design, Analysis and Prototype Manufacturing of a Sequential Transmission

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June, 2016

National University of Sciences & Technology

FINAL YEAR PROJECT REPORT

We hereby recommend that the dissertation prepared under our supervision by: Fajar Sohail Latif (NUST201200522BSMME11112F), Khuldoon Usman (NUST201200387BSMME11112F), Momin Jawaid Ali (NUST201201370BSMME11112F) and Mukaram Ali (NUST201200818BSMME11112F) Titled: Design, Analysis and Prototype Manufacturing of a Sequential Transmission be accepted in partial fulfillment of the requirements for the award of Bachelors of Engineering in Mechanical Engineering degree with (_____ grade)

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Declaration

We certify that this research work titled "*Design, Analysis and Prototype Manufacturing of a Sequential Transmission"* is our own work. The work has not been presented elsewhere for assessment. The material that has been used from other sources has been properly acknowledged / referred.

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Copyright Statement

Acknowledgments

First of all, we would thank Allah Almighty, who gave us the knowledge and dedication to be able to complete this project. We are extremely grateful to our supervisor, Dr. Samiur Rahman Shah, for giving us the guidance required to complete this endeavor of ours. Our department's Design and Manufacturing Resource Centre also assisted us greatly. We would like to extend our gratitude to our entire department, its faculty and staff, for all their assistance in our project.

Abstract

The purpose of this project was to design a sequential transmission for our department's formula student car, and also to analyze that design. We successfully performed both, using Solidworks and Ansys for the respective tasks. We also produced a 3D-printed prototype of the transmission. The latter part of project, though, involved preparing the prototype vehicle for the formula student competition, of which some components were already available, while the rest we had to manufacture

Keywords: Design, Sequential transmission, Analysis, Formula Student, Prototype

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Chapter 1
Introduction

1.1 Background

Less than nearly four decades ago, a competition was started in which teams of university students would design and construct racing vehicles and compete against eachother. Nowadays this competition is known as Formula SAE. Three years ago, the School of Mechanical and Manufacturing Engineering (SMME), NUST commenced work on a vehicle to participate in this competition in the near future. Since then, its students, under the supervision of the department's automotive expert Dr. Samiur Rahman Shah, have been involved in the construction of this vehicle. The design and fabrication of various components of this vehicle have been undertaken as final year projects by groups of students. This year, one of those components was the transmission of the vehicle.

1.2 Aim and Objectives

The aim of this project was to design a sequential transmission for the formula student vehicle. The required design would need to satisfy the following criteria:

- Be compact, in order to easily fit into the designed chassis.
- Have short shifting times, to be an effective racing transmission
- Be lighter in weight

Our objectives for the project were as follows:

- Design of the input and output shafts and gears.
- Stress analysis of the above components
- Manufacturing a prototype of the transmission assembly
- Design of a selector drum and shift forks
- Design and manufacture of the shifting mechanism

Another major deliverable was to complete the Formula Student vehicle. This deliverable included the following objectives.

- Design of Upright components and their integration with the chassis and the suspension system of the vehicle.
- Designing and constructing the suspension components along with chassis mountings.
- Integrating the current engine with the chassis
- Steering behavior and suspension analysis for camber, caster and scrub radius setup.
- Engine tuning, testing and chassis fitting.

1.3 Research Methodology

The literature study was carried out by technical books reading, video lectures and tutorials, seeking professional design help and through physical examination of similar components.

1.4 Thesis Structure

The brief description of the contents of the remaining chapters in thesis is described below.

Chapter 2 Literature Review: This chapter outlines the details of the literature consulted and other aides used for the initial design concept of the transmission. This includes, besides deign, the study carried out for making modifications to the stock transmission available and suspension design setup.

Error! Reference source not found. **Transmission Design and Analysis:** This chapter will discuss the basic rules and criteria defined for designing the transmission. This includes all details from gear ratios, gear sizing, mesh type and sizing of the shafts. The FEM modeling of all the parts is also part of this.

Error! Reference source not found. **Vehicle Construction:** This chapter will discuss the design and manufacturing limitations faced during the process; including FSAE rules modification limitations on available components. Gearbox alterations and iterative failures to adhere to the chassis’ dimensional limits.

Chapter 5 Conclusions and Future Work: This chapter presents the conclusion of the conducted research along with the proposed future work.

Chapter 2
Literature Review

2.1 Type of Transmission

This chapter discusses the type of transmission that we undertook from designing. Categories include constant mesh and synchromesh transmissions, gear types, speeds for transmission, transverse or longitudinally mounted transmission etc. The book "Automotive Transmissions Fundamentals, Selection, Design and Application" (1) was consulted to decide this.

Parameter	**Configurations (commercial vehicles)**				
Engine and gearbox configuration	Front – longi-tudinal	Front – transverse	Rear – longi-tudinal	Rear – transverse	Underfloor
Number of drive axles	One	Two	Three	Four	————
Type of axle	Drive axle with/without drive-through	Steering axle	Driven steering axle	Trailing axle	Leading axle
Drive-through to second axle	Yes	No	————	————	————
Final drive	Bevel gears – helical bevel drive	Bevel gears – hypoid bevel drive	Worm gears	Double bevel gears	Spur gears
Reduction in centre drive	Single	Spur wheel reduction, shiftable	Spur wheel reduction, not shiftable	Planetary reduction, shiftable	Planetary reduction, not shiftable
Differential gear	Spur gears	Bevel gears	Helical gears	Worm gears	————
Differential locking	Unlocked	Self-locking	Manual locking	Electronic locking	————
Hub drive, see Figure 6.75	Without hub drive	Spur gear reduction, external toothed	Spur gear reduction, internal toothed	Spur gear planetary reduction	Bevel gear planetary reduction

Table 2.1 Matrix of the configurations of commercial vehicle powertrains

The table above is a guide to different types of gear configurations that could be used for designing the transmission.

A figure that defines different types of engine-transmission configuration is given below

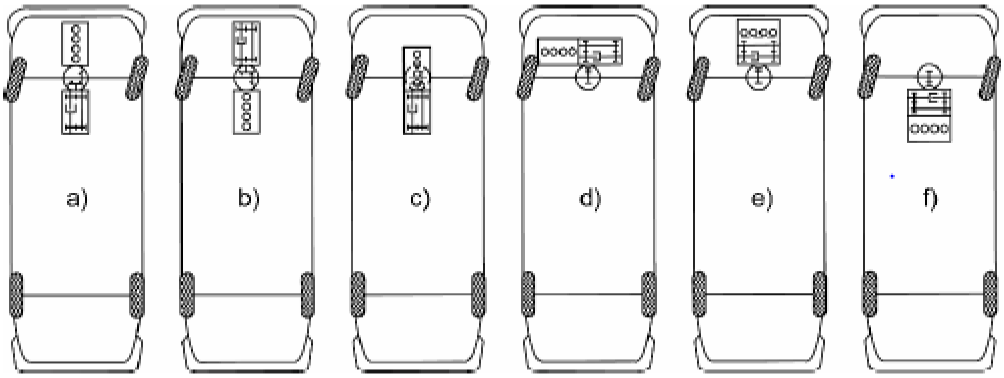

Figure 2.1 Front wheel drive. a) *Longitudinal engine in front of axle, longitudinal gearbox;* b) *longitudinal engine behind axle, longitudinal gearbox;* c) *longitudinal engine above axle, longitudinal gearbox;* d) *transverse engine beside the gearbox;* e) *transverse engine above the gearbox;* f) *transverse engine behind the gearbox*

The transmission that was finalized was an **all spur gear transversely-mounted constant-mesh 4-speed gearbox.**

2.2 Gear Calculations, Material and Related Data

The Formula Student rule book 2016 was used to decide on design and machining of any components. This includes maximum allowed tracks speeds, different types of differentials allowed and manufacturing materials allowed.

The formulas for gear design were consulted, and so were the stress and bending formulas for analysis (6).

Equation 2.1

$$\sigma_F = \frac{F_t}{b\, m_n} Y_{Fa}\, Y_{Sa}\, Y_\varepsilon\, Y_\beta\, K_A\, K_V\, K_{F\beta}\, K_{F\alpha}\,,$$

where:

F_t	nominal circumferential force at the reference circle in N,
b	face width in mm,
m_a	standard module in mm,
Y_{Fa}	form factor,
Y_{Sa}	stress correction value,
Y_ε	contact ratio,
Y_β	helical overlap,
K_A	application factor,
K_V	dynamic load factor,
$K_{F\beta}$	longitudinal load distribution factor, and
$K_{F\alpha}$	transverse factor.

The permissible root strength is determined as,

Equation 2.2

$$\sigma_{FG} = \sigma_{F,lim}\, Y_{ST}\, Y_{NT}\, Y_{\delta,relT}\, Y_X\,,$$

where:

$\sigma_{F,\,lim}$	tooth endurance strength value,
Y_{ST}	stress correction factor,
Y_{NT}	service life factor,
$Y_{\delta,relT}$	relative support figure,
Y_X	tooth root size factor.

The safety factor can then be calculated as a quotient of the two values calculated above,

Equation 2.3

$$S_F = \frac{\sigma_{FG}}{\sigma_F}\,.$$

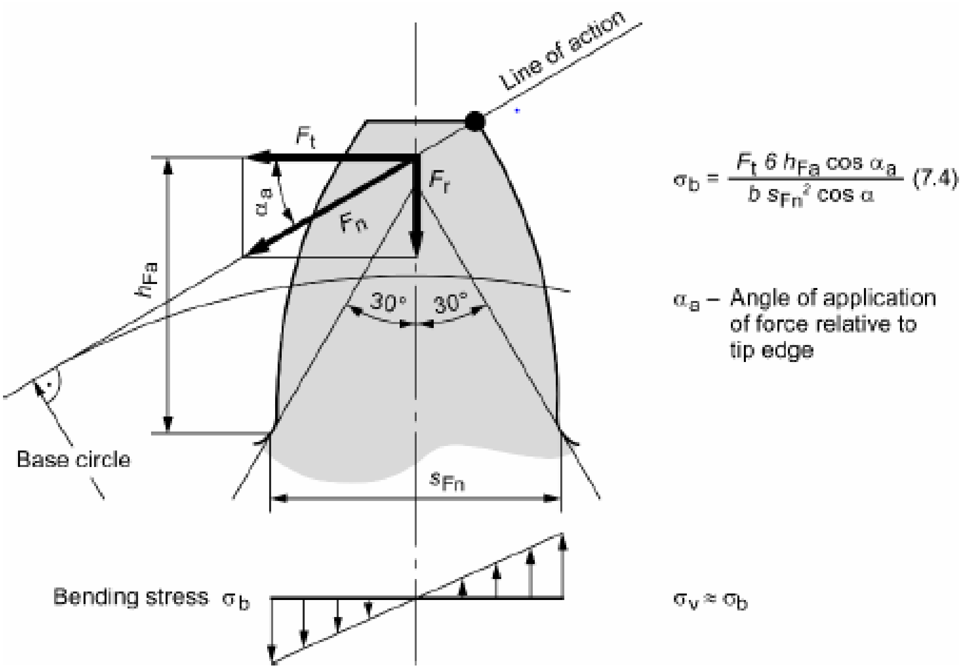

Figure 2.2 Bending stress at the tooth root with force acting on the tip

2.2 Shaft Design

2.3.1 Methodologies

Following are the three approaches when designing shafts for transmissions:

1. Initially specifying the diameters of the shafts: This involves an initial estimate of the shaft diameters required for a given loading.

2. Designing shafts for endurance strength: Using the maximum engine torque to work out the maximum expected force, the transmission is designed such that is able to sustain that load without succumbing to fatigue. An over-engineered transmission is one which is entirely fatigue resistant, even under maximum load.
3. Specifying the operational fatigue strength of the shafts: Having decided upon a service life for the shaft, it is designed based on existing load profiles. These profiles are built base on the load each gear carries, and the mileage sun by each gear (5).

2.3.2 Problems

Two problems to which transmission shafts are particularly vulnerable to are deflection and vibrations.

Deflection: In vehicle transmissions, there are large distances between shaft bearings and the shaft loading is generally unbalanced. This results in large deflections *f* and large bending angles φ (Figure 8.3). Additionally, the teeth become inclined, reducing the contact area between gears in a pair, increasing stress. Shaft deflection, thus, has to be taken into account alongside strength calculations.

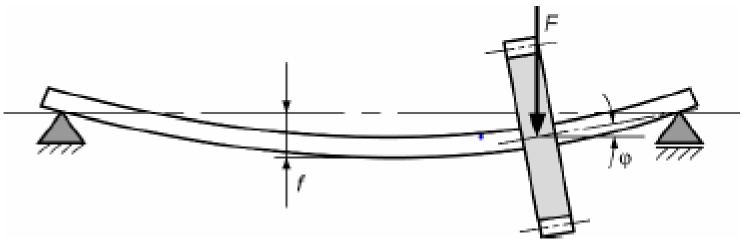

Figure 2.3 Deflection f and bending angle φ in shafts with large distances between bearings and asymmetrical loading.

Vibration: Vibrations, in essence, cause the transmission to become more rigid while it undergoes its rotary motion. This additional stiffness varies with speed and gear position, resulting in stress peaks at certain points. It is vital to study, and thus, nullify these peaks for the transmission to serve its desired life. Vibration analysis can be performed on either test benches or via computer simulation. There are numerous mathematical models available.

2.4 Selection of Gear Type

The use of spurs gears is solely based on their efficiency rating and easy availability and machine-ability in the current market.

Type of gearbox		η (%)
Gear pair	Spur gear	99.0–99.8
	Bevel gear	90–93
Mechanical transmission with splash lubrication	Passenger car	92–97
	Commercial vehicle	90–97
Automatic transmission with various gear ratios (AT, DCT)		90–95
Mechanical continuously variable transmission		87–93
Hydrostatic continuously variable transmission without power-split and mechanical part		80–86

Table 2.2 Efficiency ranges of different gearbox types

2.5 Gearwheel Performance Limitations

Gears are designed based on their performance limits, i.e. their causes of failure. These limits basically consist of 4 different types of gear failure:

- tooth failure,
- macropitting and micropitting,
- gear scuffing (hot scuffing) and
- wear (cold scuffing).

These damage types limit the load capacity of the gearwheels (Figure 2.4). The major factors affecting the performance limits indicated above are:

- operating conditions (load type, angular speed, temperature),
- the material,
- gear geometry,
- manufacturing accuracy,
- surface treatment and surface roughness and
- lubricant (chemical and physical characteristics). (3)

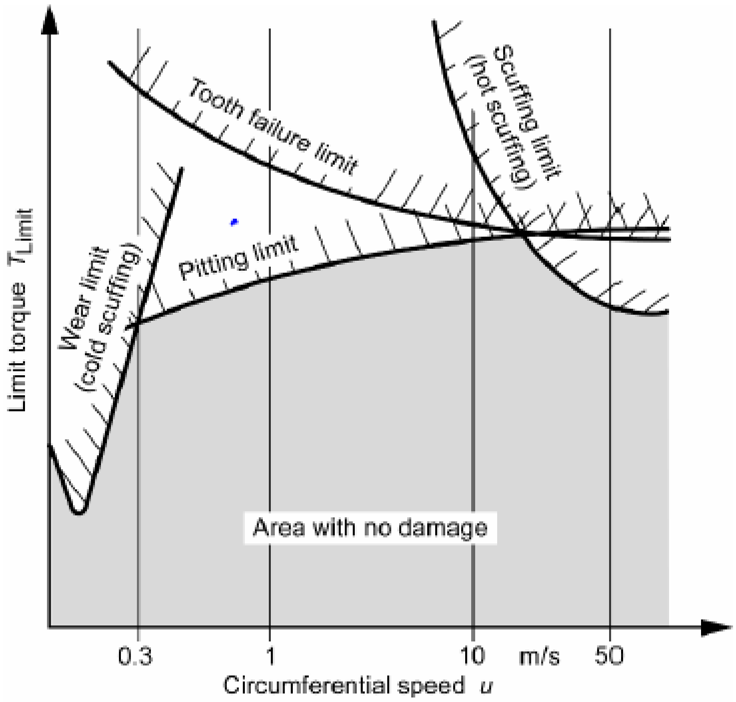

Figure 2.4 Torque limits of case-hardened, lubricated gearwheels plotted against tangential speed.

Out of the 4 above-mentioned damage types, the two more prominent ones, tooth failure and macropitting, are discussed below.

<u>Tooth failure</u>: it is defined as failure where the tooth breaks off, partly or in its entirety. This is further classified into overload failure and vibration fatigue failure. The former occurs when the gear pair is briefly burdened with an intense overload. For the latter to occur, the stress level should at least intermittently be greater than the vibrational resistance of the gear. This resistance is largely determined by factors such as tooth root design, surface and heat treatments, etc. Figure 7.3 shows vibration fatigue failure in a straight spur gear (4).

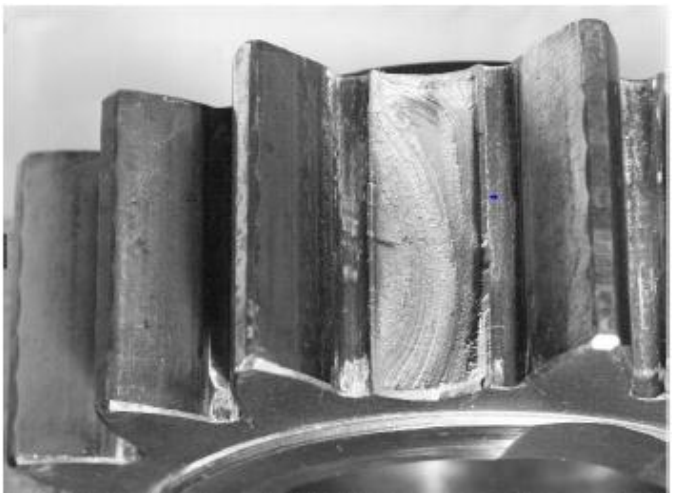

Figure 2.5 Vibration fatigue failure in a spur gear

Macropitting: Characterized by formations such as pin holes or pits on the flanks of gear teeth, usually below the pitch circle. It indicates fatigue at the tooth flanks. Pitting load capacity is calculated using Hertzian stress, which forms the basis for computing surface stress. However, it should be notes that the corresponding subsurface stress is as responsible for pitting as surface stress.

Figure 2.6 Macropitting due to flank surface fatigue

Pitting can only be found in lubricated transmissions. Pitting resistance is determined by several factors including material type, oil viscosity and temperature, and flank surface properties. The macropitting phenomenon is shown in Figure 2.6.

2.6 Lubrication System

Under consideration were splash lubrication vs forced lubrication systems for the transmission. The conventional splash lubrication system was selected keeping in view its simplicity over forced lubrication (2), its application in our 4-speed sequential transmission and RPM range.

Chapter 3
Transmission Design and Analysis

3.1 Design Concept

The basic design of the transmission was based on a motorcycle transmission - a sequential type of transmission - with gears and gear ratio calculations defined by parameters including but not limited to top speed, acceleration, mass of transmission, maximum torque and horsepower of the engine.

We had also acquired a Daihatsu Mira transmission, which we dismantled and examined – each component individually – not only to understand its working, but also to take measurements from, which would serve as the initial iterations of our gears' parameters. These parameters were modified to suit both standards and our requirements.

The table below details all data relevant to gearbox:

	Parameters	Input Shaft	Output Shaft		Ratio	
	Shaft Dia(mm)	20				
	ID(mm)	23.75				
	OD(mm)	35.05				
	Pitch Dia(mm)	30	105			
	Module(mm)	2.5	2.5			
Gear 1	Number of Teeth	12	42		3.5	
	ID(mm)	39.37				
	OD(mm)	49.55				
	Pitch Dia(mm)	45	90			
	Module(mm)	2.25	2.25			
Gear 2	Number of Teeth	20	40		2	
	ID(mm)	55				
	OD(mm)	64				
	Pitch Dia(mm)	60	74			
	Module(mm)	2	2			
Gear 3	Number of Teeth	30	37		1.233333	
	ID(mm)	65.62				
	OD(mm)	73.55				
	Pitch Dia(mm)	70	64.75			
	Module(mm)	1.75	1.75			
Gear 4	Number of Teeth	40	37		0.925	

Table 3.1 Gear Parameters

The design is based on metric standards of gears and shafts, and the designing was carried out in Solidworks 2015.

3.2 Vehicle Speeds, Acceleration and Forces

The table below shows the calculations needed for the vehicle:

HP@Peak RPM	75				
Torque(Nm)	118				
RPM Limit	7200		**Max Speed(km/h)**	**Wind Resistance(N)**	**First Gear Torque(Nm)**
Gear 1		3.5	37.91931895	11.68439655	1934.2855
Gear 2		2.1	63.19886492	32.45665708	1160.5713
Gear 3		1.233	107.6379694	94.14911093	681.421149
Gear 4		0.925	143.4785041	167.2857358	511.204025
Final Drive		4.93			
Tire Radius(m)(Hoosier 41100)		0.229			
Weight(kg)		350			
Rolling Resistance	44.6355	N			
Frontal Area(approximated)	0.595	m^2			
Transmission Efficiency	0.95				
Air Density	1.18	kg/m^3			
Drag Coefficient	0.3				

Traction Force at Wheels(N)	**Net Force(N)**	**Net Acceleration(m/s^2)**	**Time to Max Speed(s)**
8446.661572	8390.341676	23.97240479	0.439386213
5067.996943	4990.904786	14.25972796	0.931828714
2975.638205	2836.853594	8.105295984	2.45480773
2232.331987	2020.410751	5.772602146	4.179455268

Table 3.2 Speed, force and acceleration calculations for each gear

3.3 First Draft

The initial draft was totally based on Daihatsu Mira 5-speed gearbox with a top speed of approximately 200 km/h with a time to top speed of around 11 seconds.

The pictures below show the first draft.

3.3.1 Individual Transmission Components

Figure 3.1 Input shaft (with the first gear)

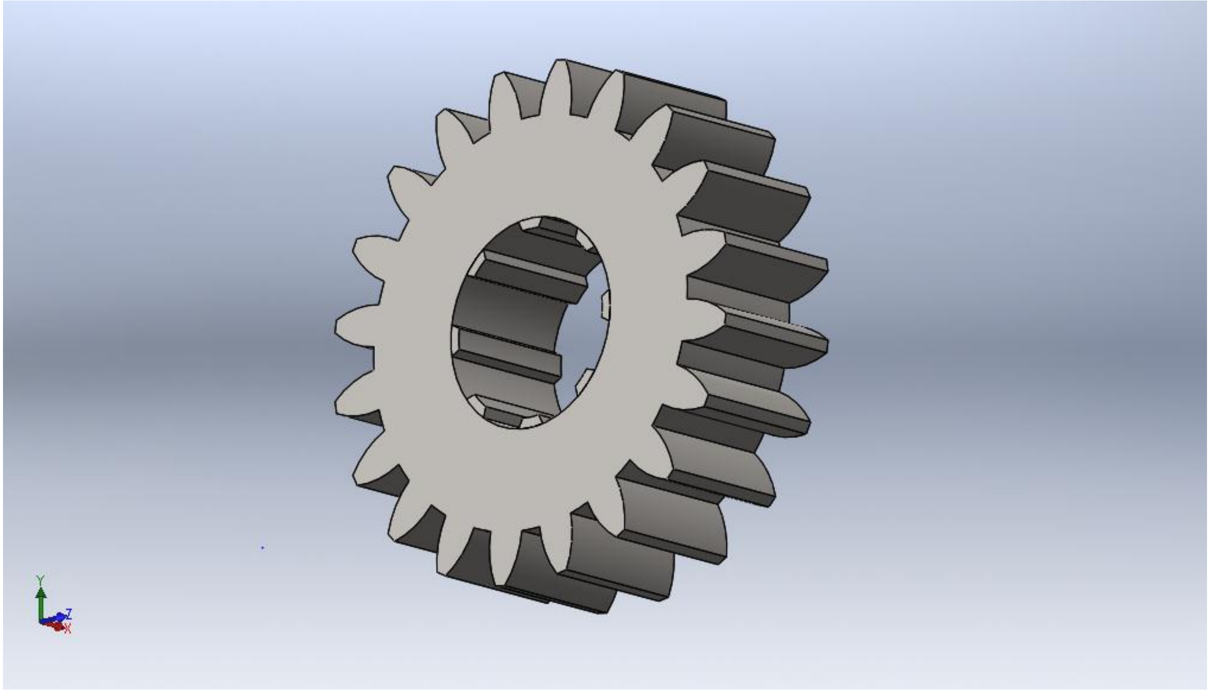

Figure 3.2 2nd gear – input shaft

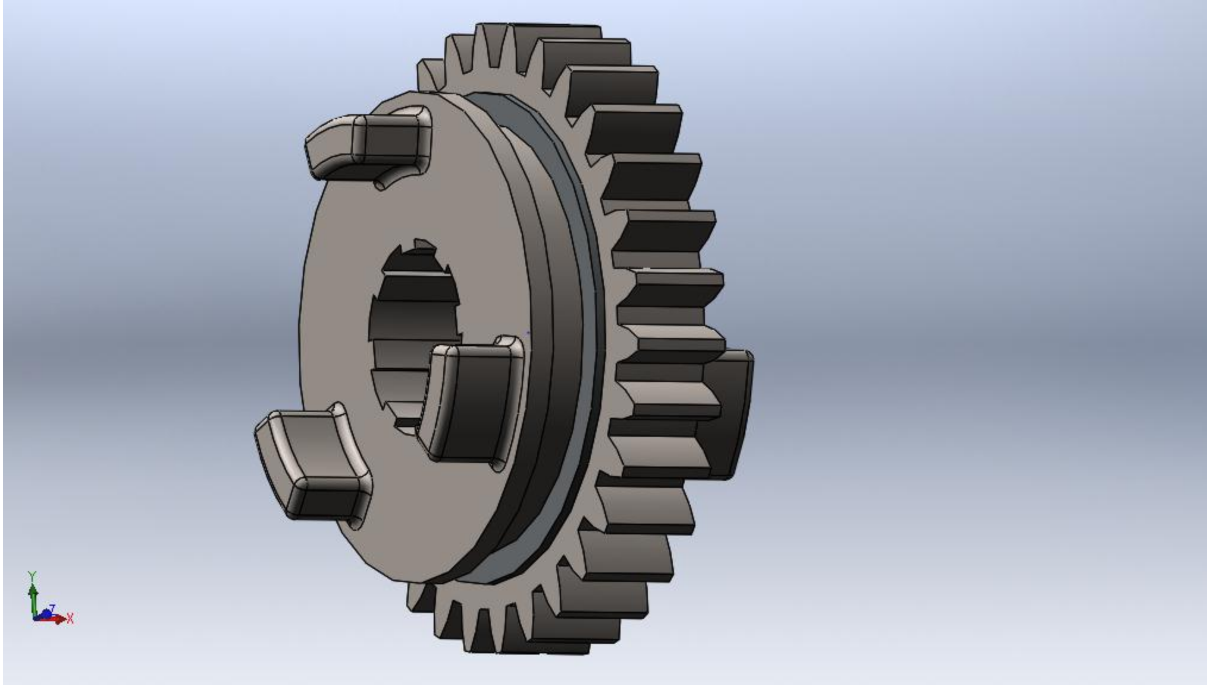

Figure 3.3 3rd gear – input shaft

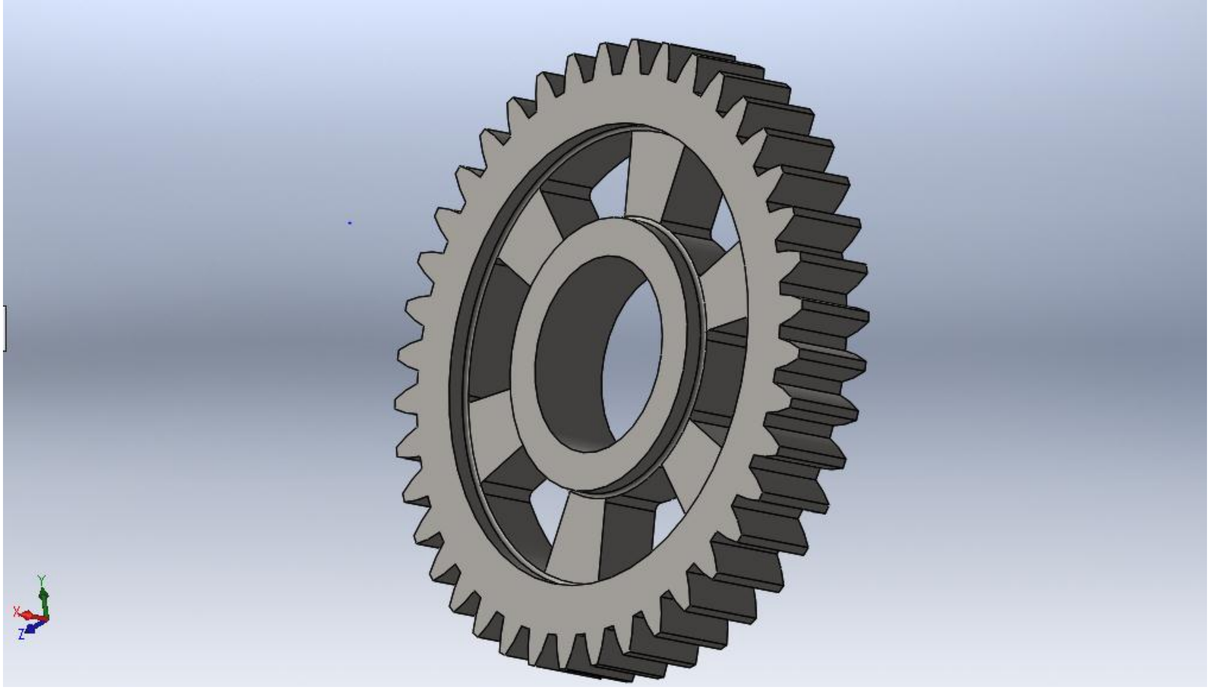

Figure 3.4 4th gear – input shaft

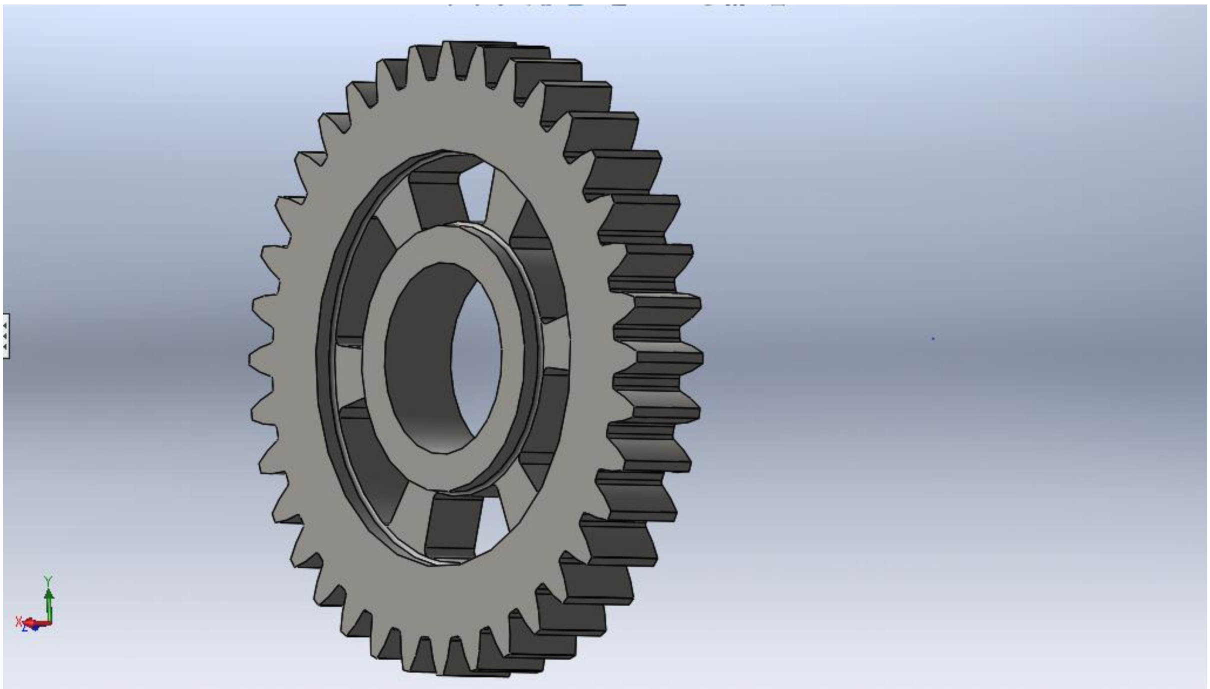

Figure 3.5 5th gear – input shaft

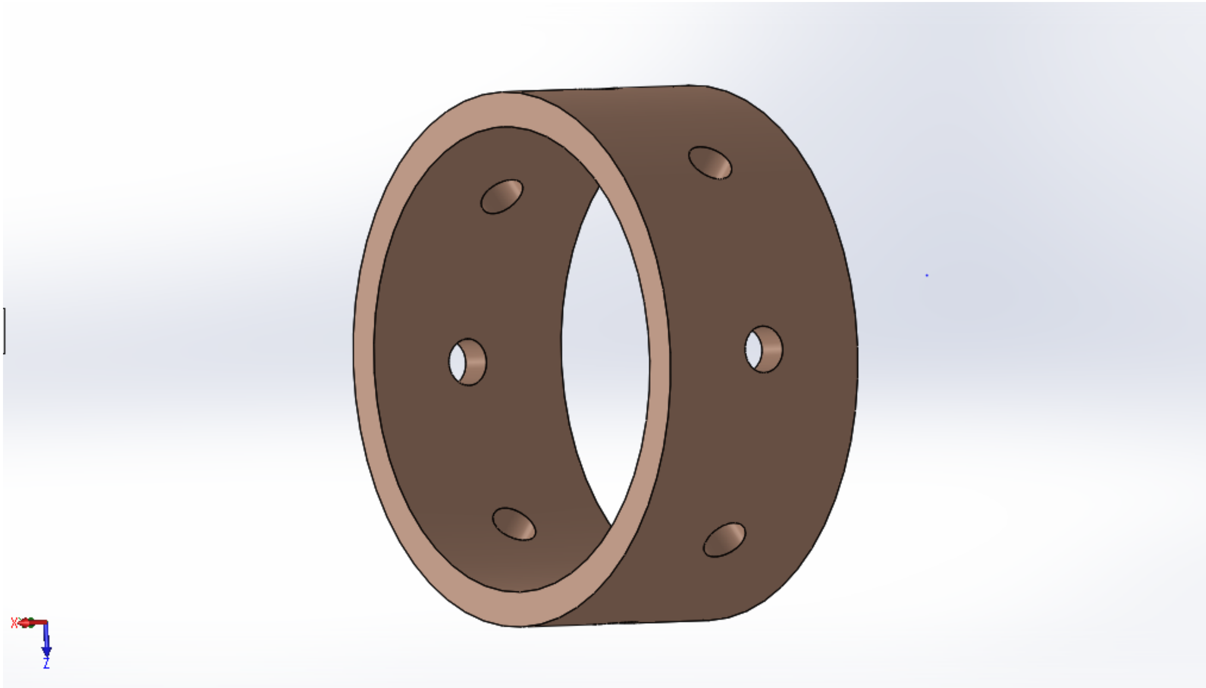

Figure 3.6 Bushing

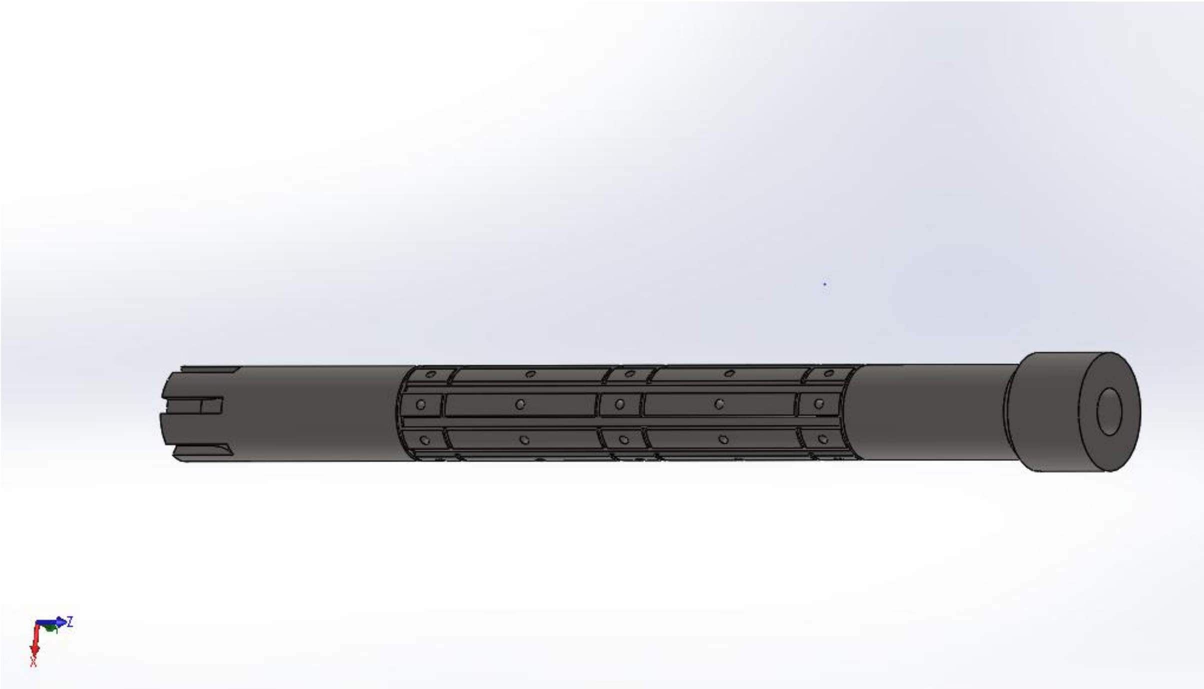

Figure 3.7 Output shaft

Figure 3.8 1st gear – output shaft

2nd and 3rd gears are similar to 1st gear, just with different gear ratios.

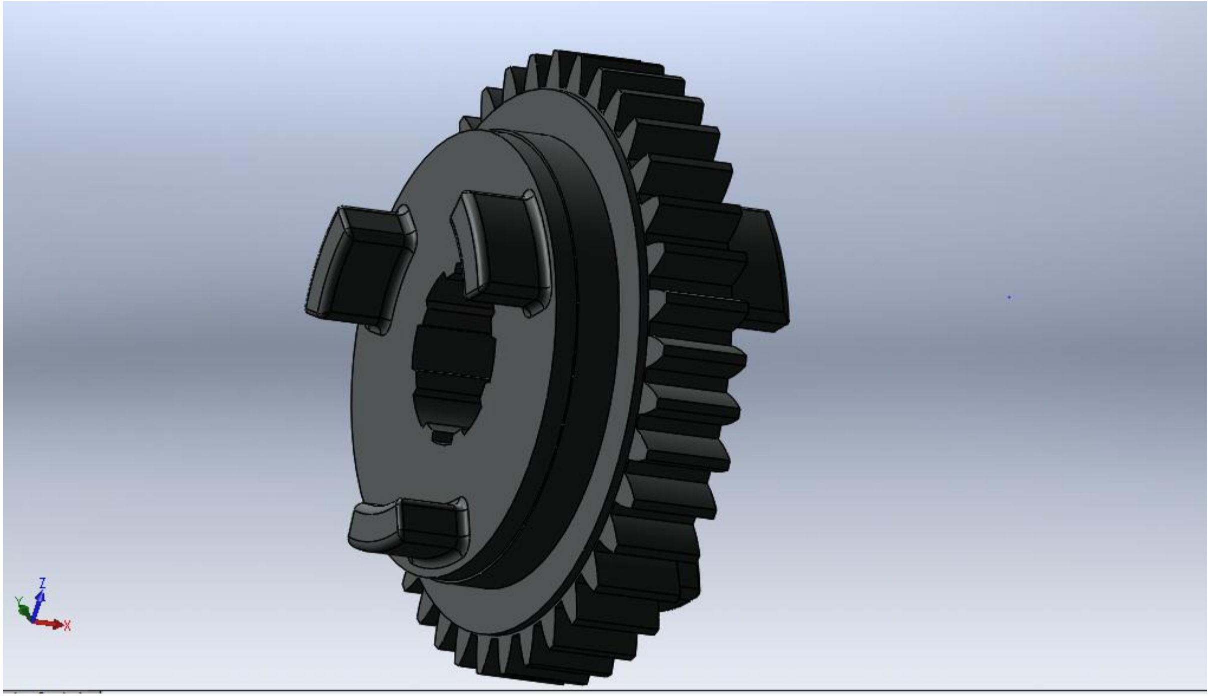

Figure 3.9 4th gear – output shaft

5th gear is similar to 4th with only a different gear ratio.

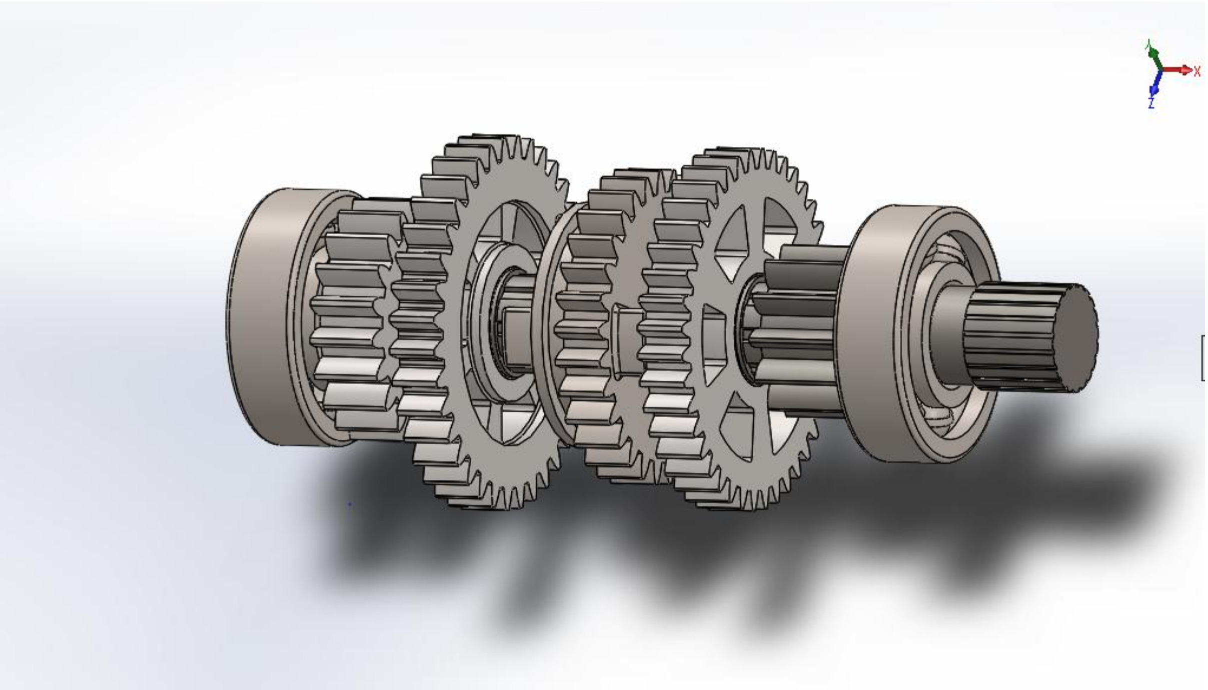

Figure 3.10 Input shaft assembly

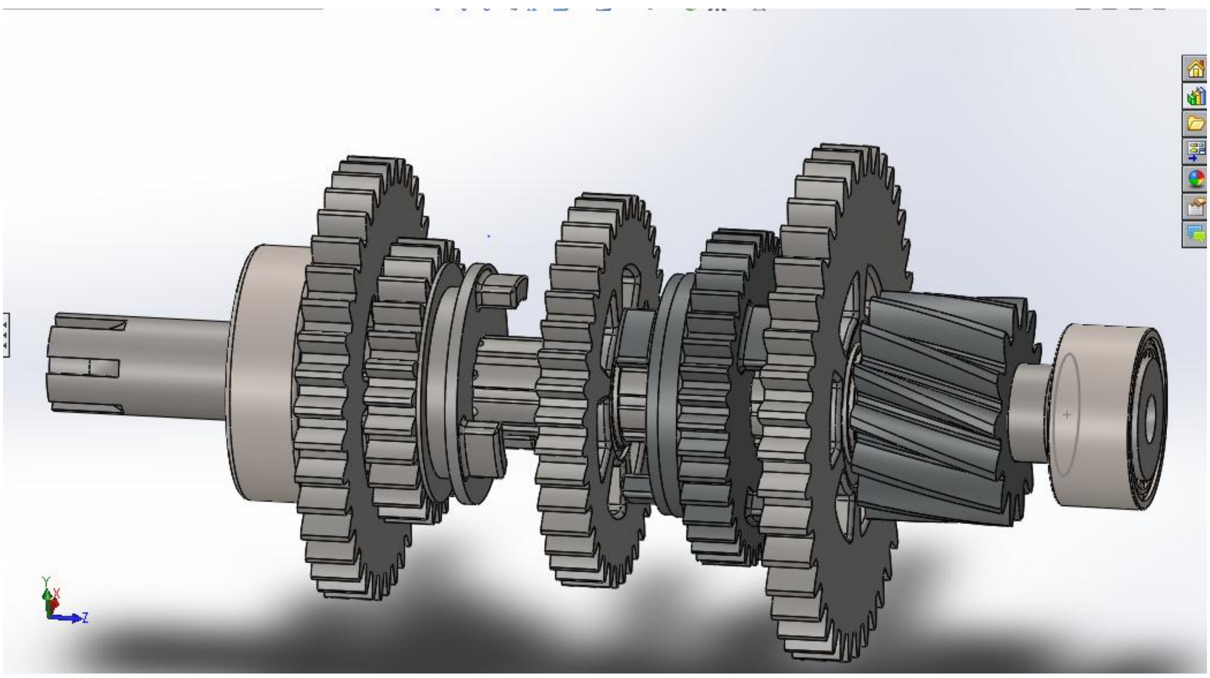

Figure 3.11 Output shaft assembly

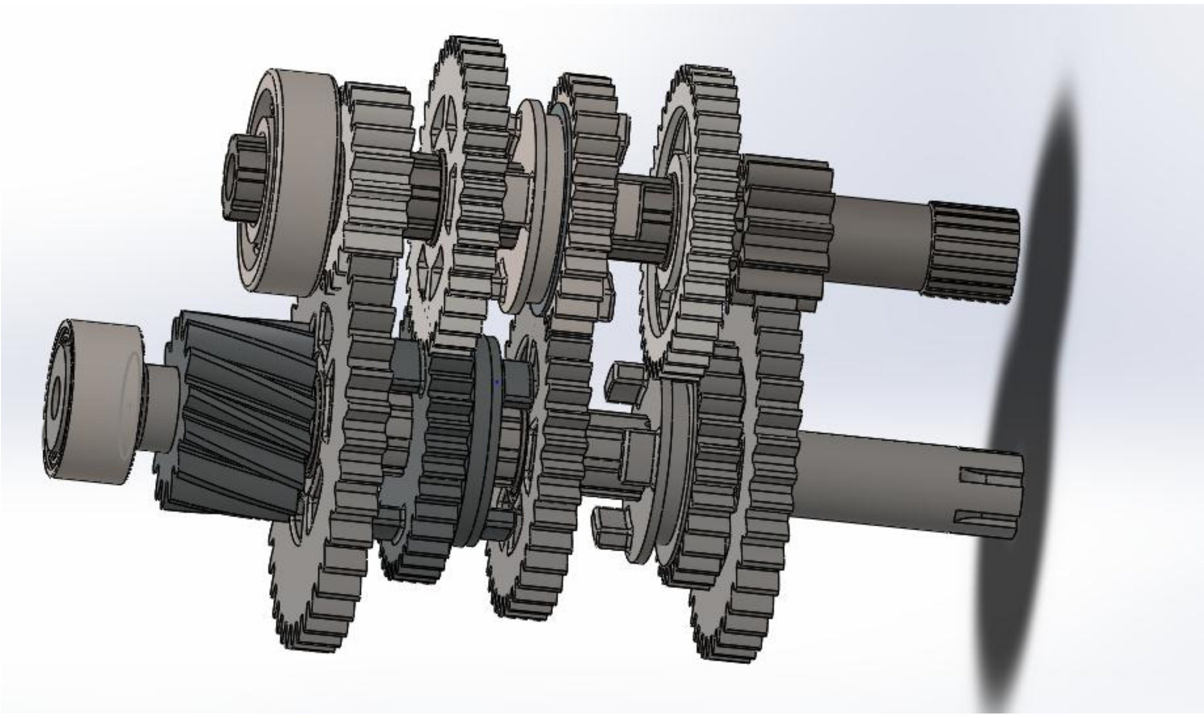

Figure 3.12 Gearbox assembly

3.4 Final Draft

The initial draft comprised of a 5-speed transmission. After further research and consideration, a final draft with a 4-speed transmission was designed. Major factors under consideration were top speed, weight and moment of inertia.

3.4.1 Transmission Components

All the gear ratios were kept the same. The fifth gear configuration was removed, shafts adjusted accordingly and the dogs altered to suit the shifting mechanism.

The final shafts and assembly along with the pinion gear for the differential are shown below.

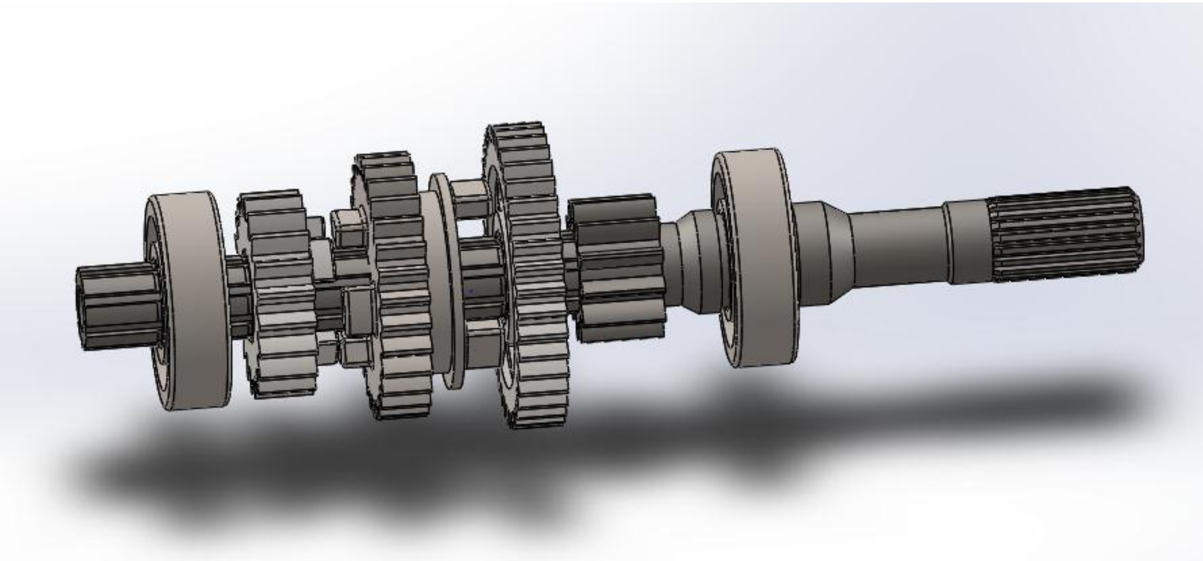

Figure 3.13 Final input shaft assembly

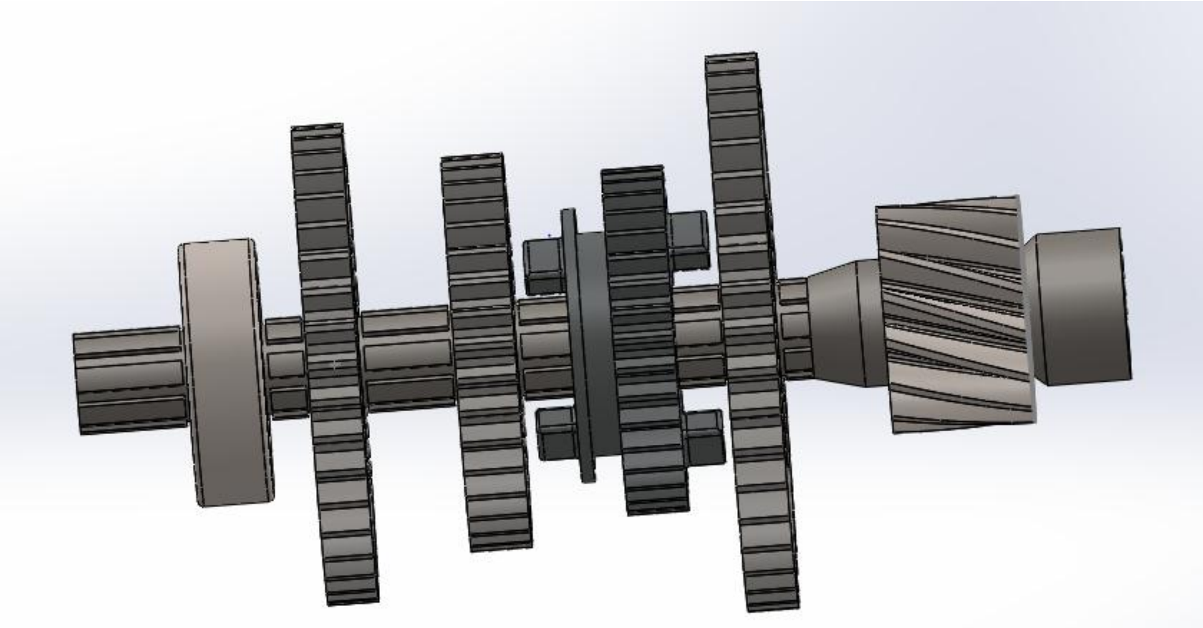

Figure 3.14 Final output shaft assembly

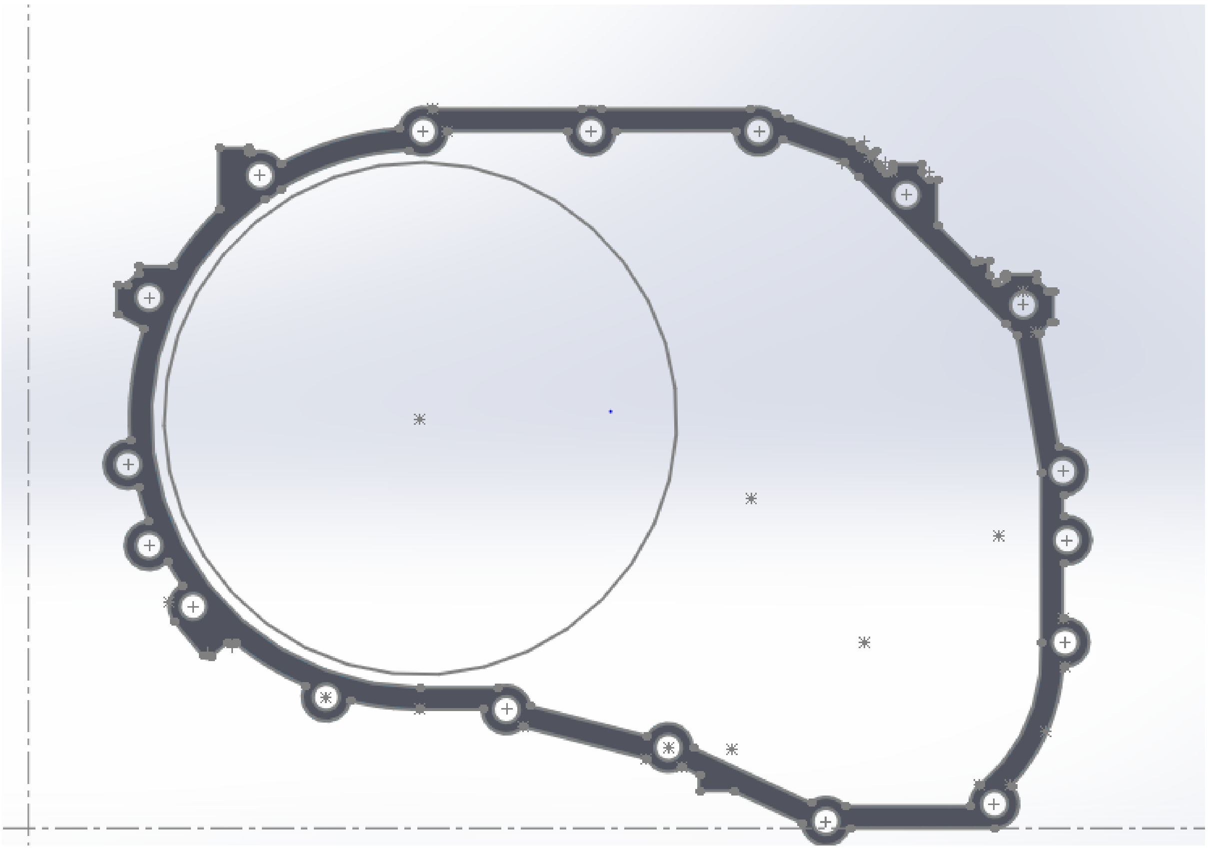

Figure 3.15 Casing cross-section

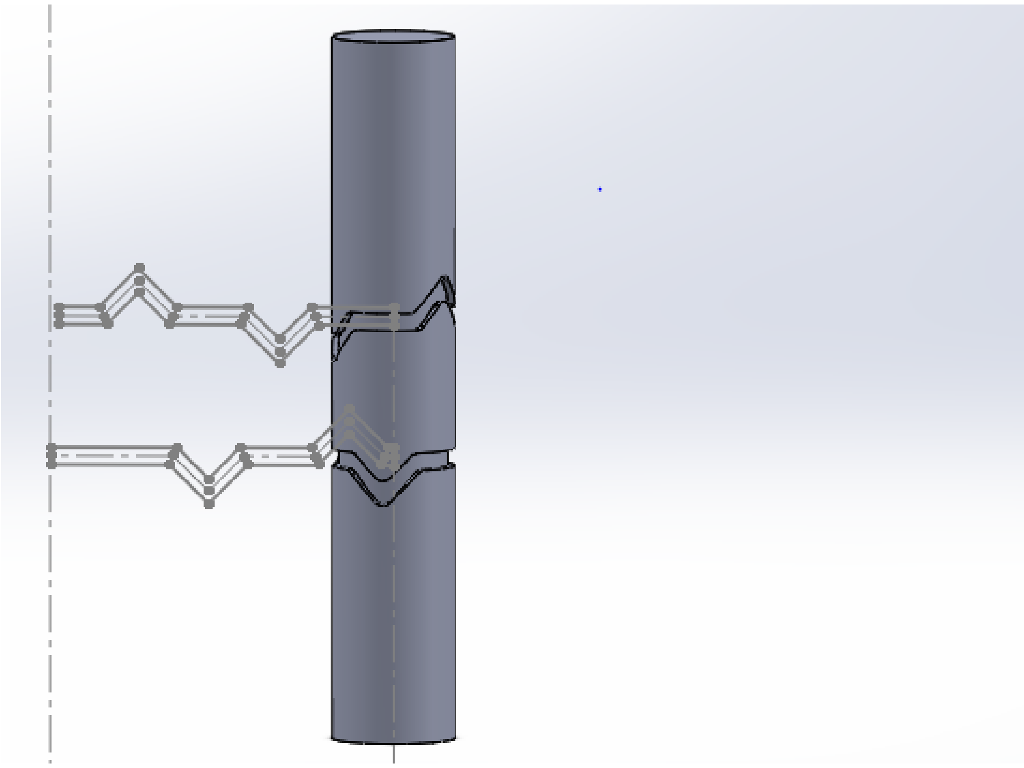

Figure 3.16 Gear Selector Cam Drum (Barrel)

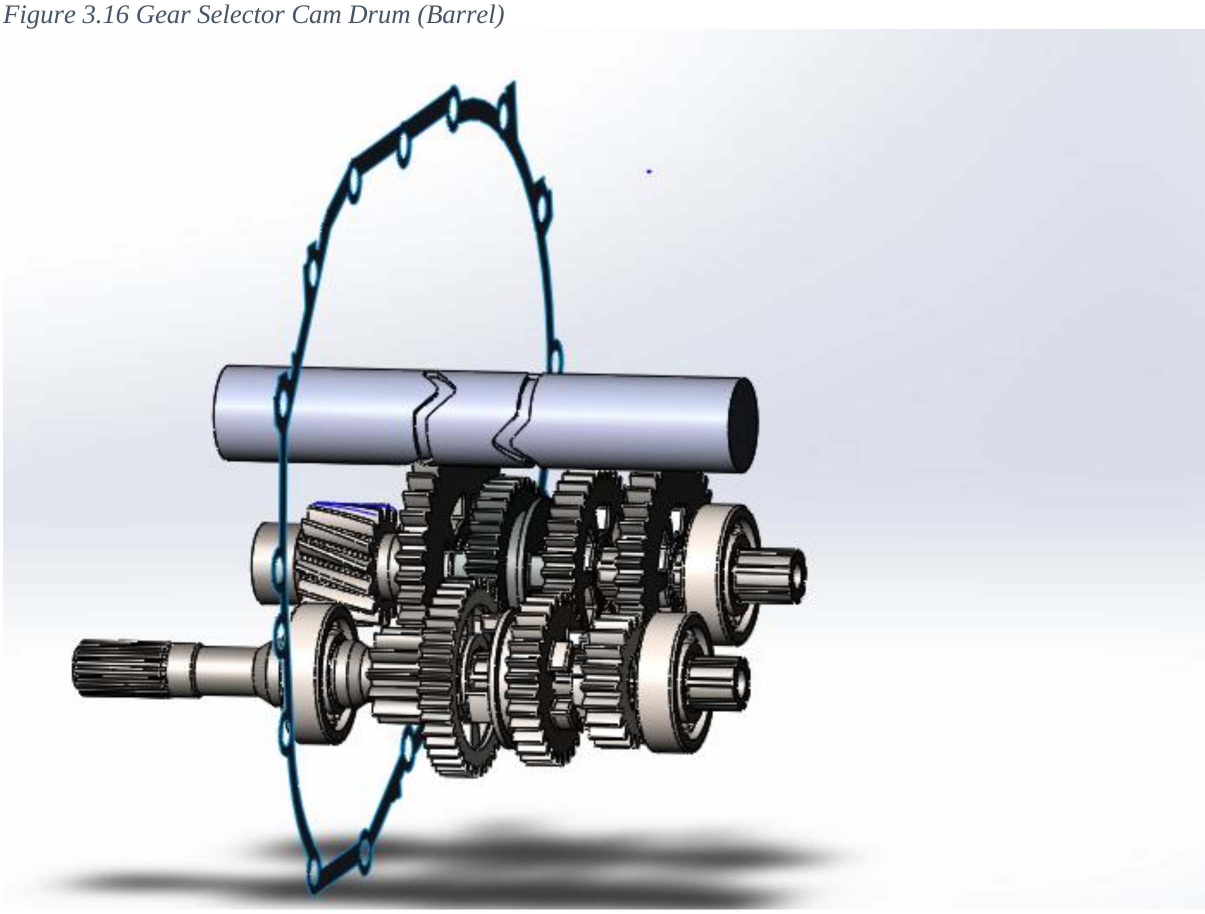

Figure 3.17 Final Gearbox assembly

3.4.2 Transmission Analysis

Transmission Analysis was carried out on ANSYS. Two different materials were used for the stress and overall analysis of the transmission, namely, grey cast iron and 4340 steel.

Figures 3.18 to 3.21 depict the effect (deflection) of each gear torque on the input shaft of the transmission.

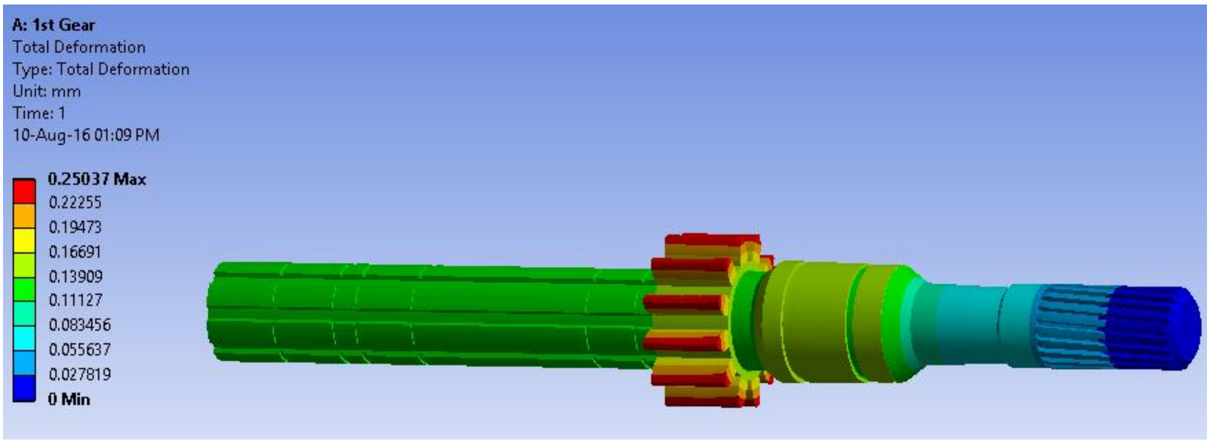

Figure 3.18 1st gear torque

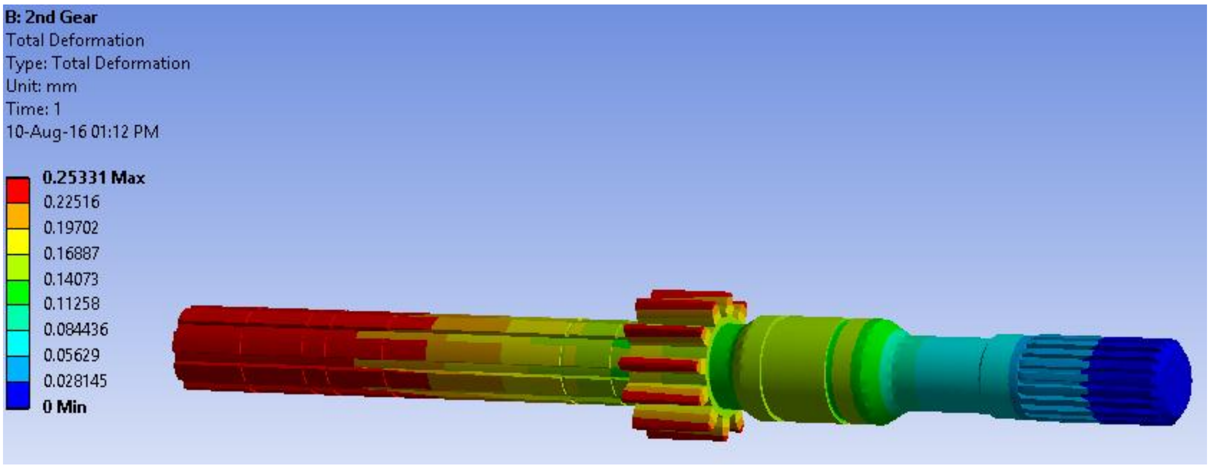

Figure 3.19 2nd gear torque

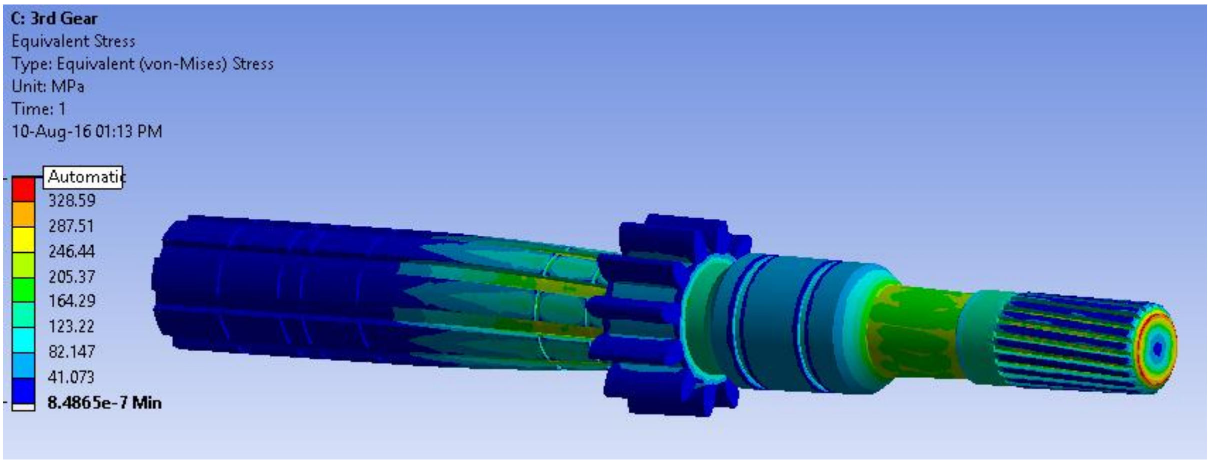

Figure 3.20 3rd gear torque

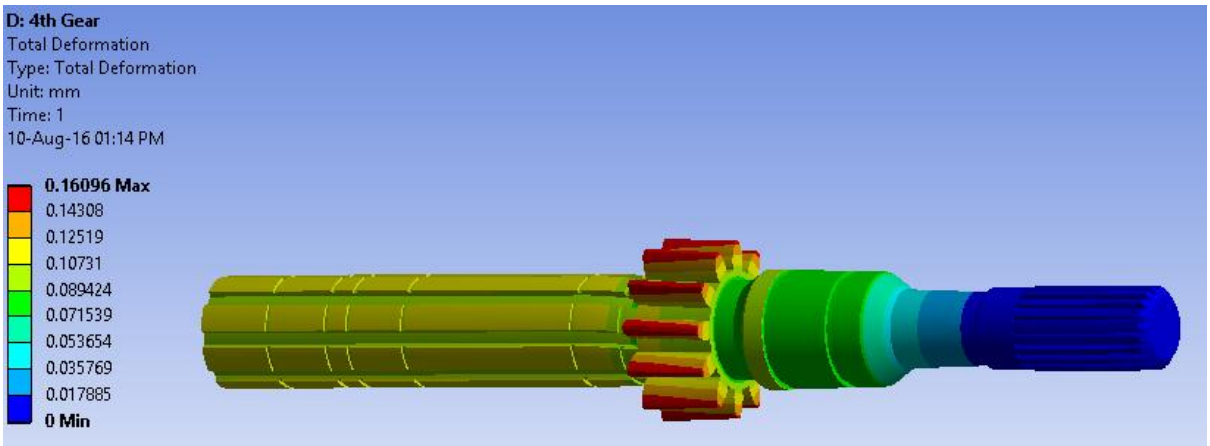

Figure 3.21 4th gear torque

The next three figures display the gear-tooth analysis of each gear pair.

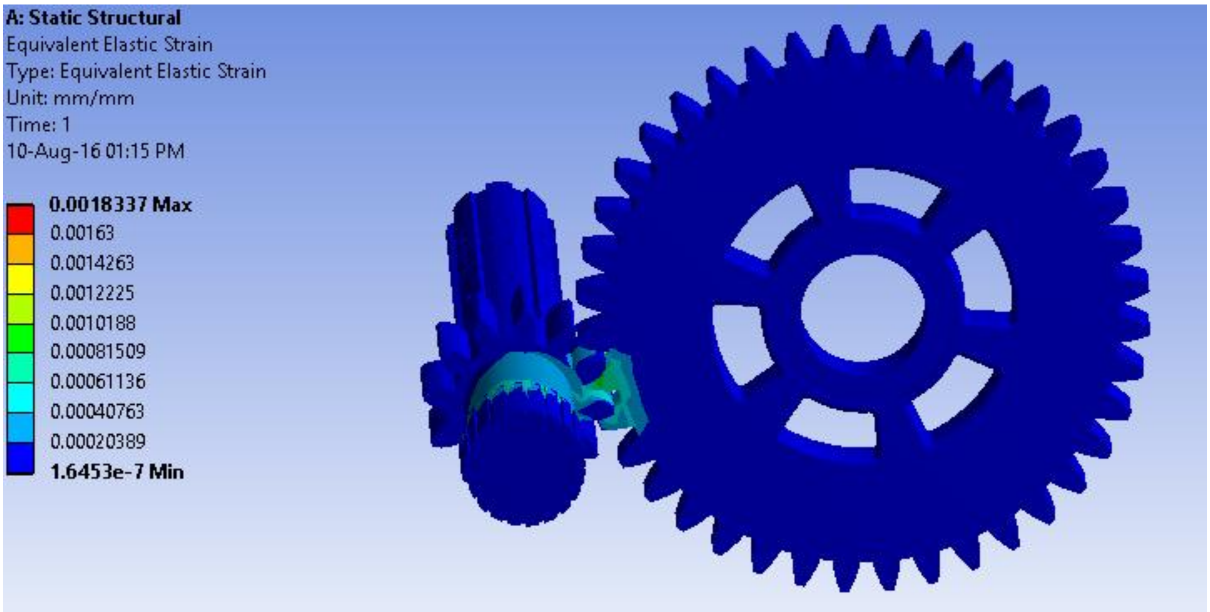

Figure 3.22 1st gear pair analysis

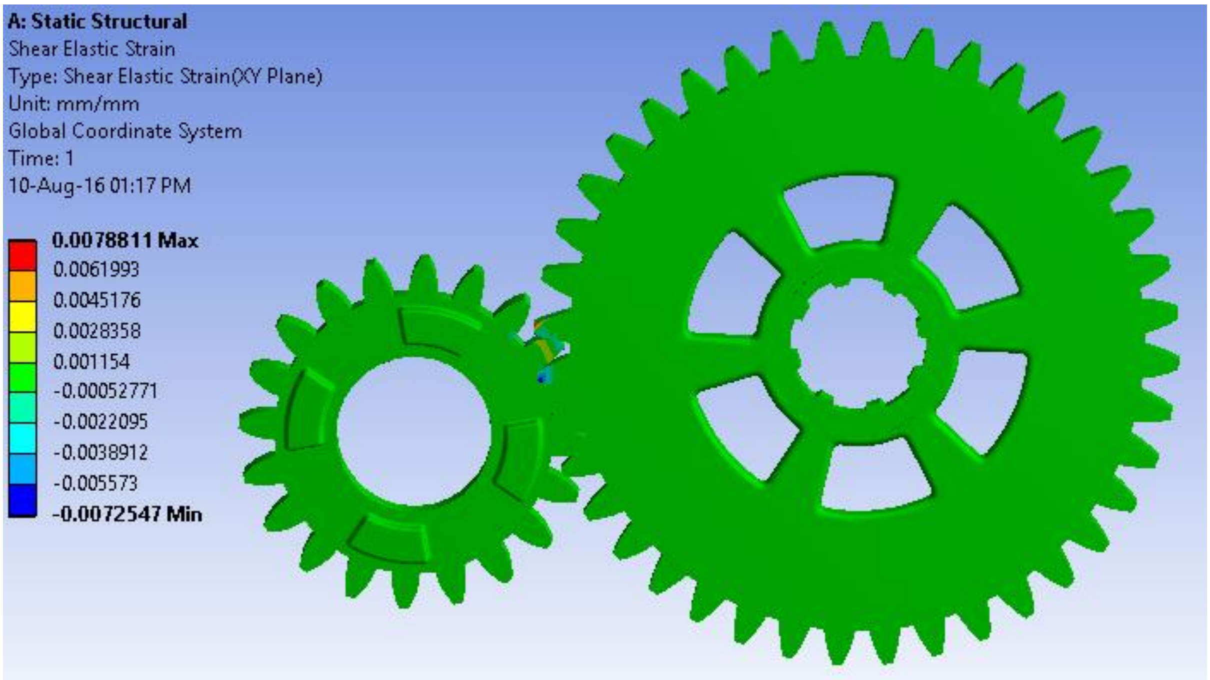

Figure 3.23 2nd gear pair analysis

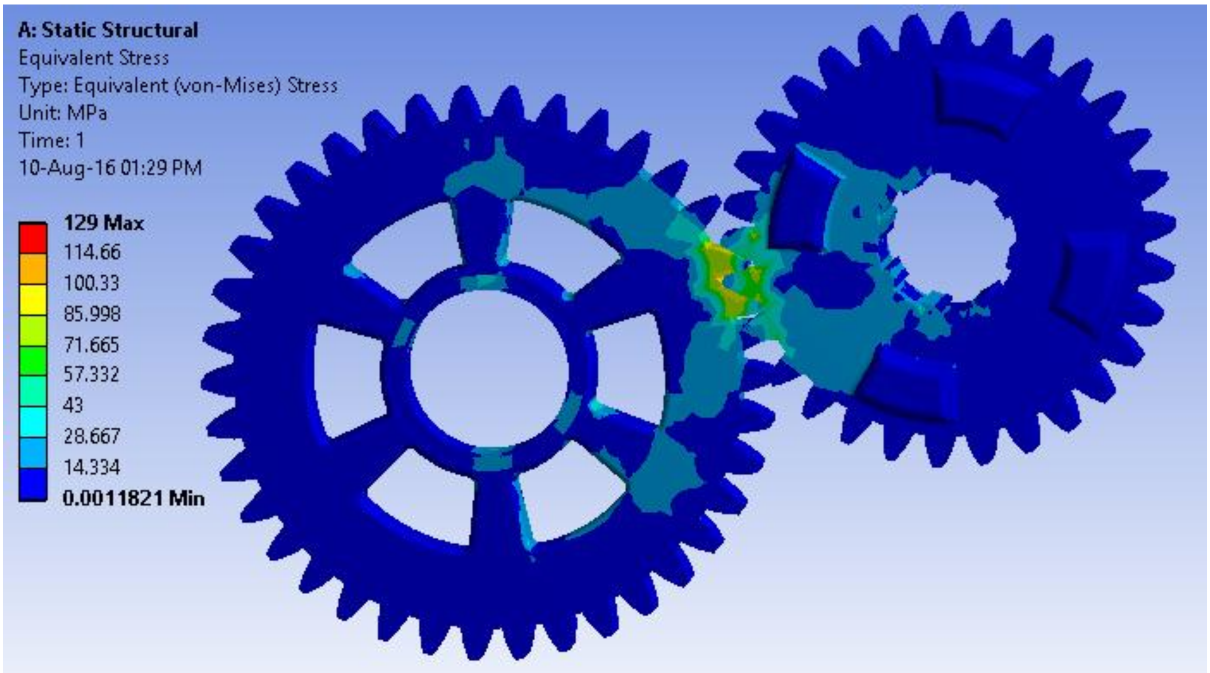

Figure 3.24 3rd gear pair analysis

The analysis behavior of the output shaft and 4th gear assembly are similar because of their given diameter and relation to the other gears and input shaft. The stresses and strains in each simulation were within the elastic limits of the materials.

3.5 3D printed prototype

Lack of metal manufacturing facilities led us to manufacture a 3D printed prototype of the transmission. ABS material was used for the 3D printed model.

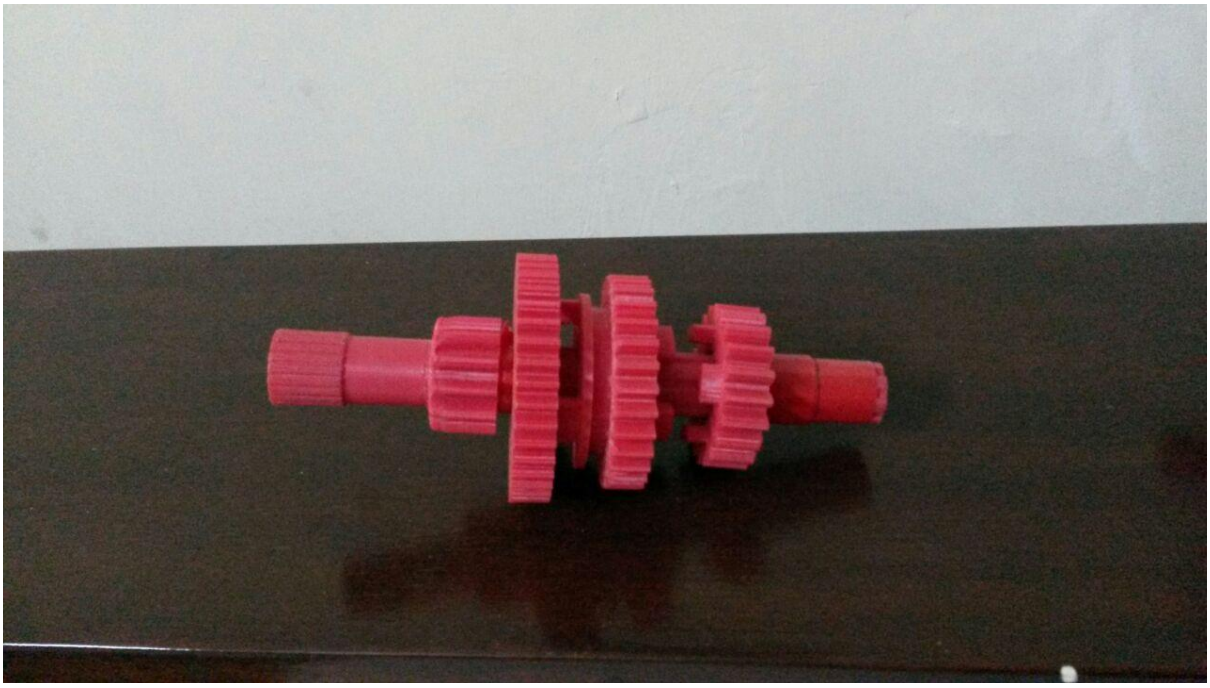

Figure 3.25 3D-printed input shaft assembly

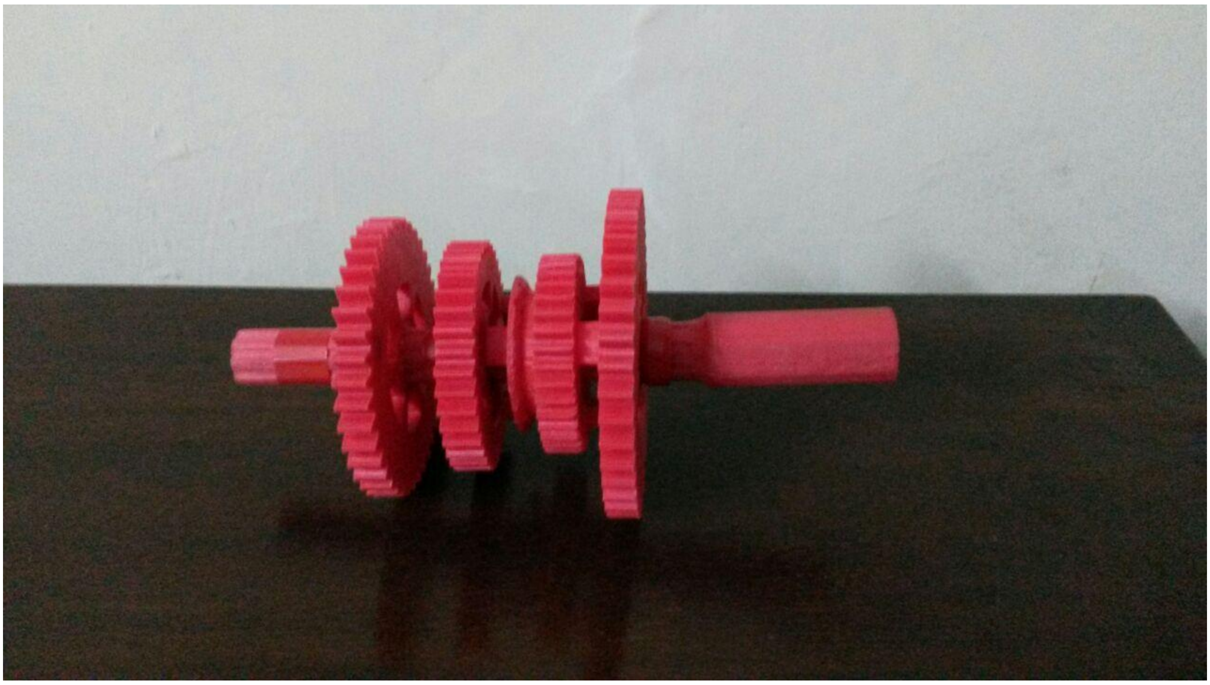

Figure 3.26 3D-printed input shaft assembly

Chapter 4
Vehicle Construction

4.1 Components

The following components were already available to us at the start of our project:

- A Daihatsu Mira 660cc, 3-cylinder engine
- 4x stock Daihatsu Mira uprights
- 4x 155/65 R13 tires
- 4x pairs of suspension a-arms
- The chassis
- The steering system
- The pedal assembly

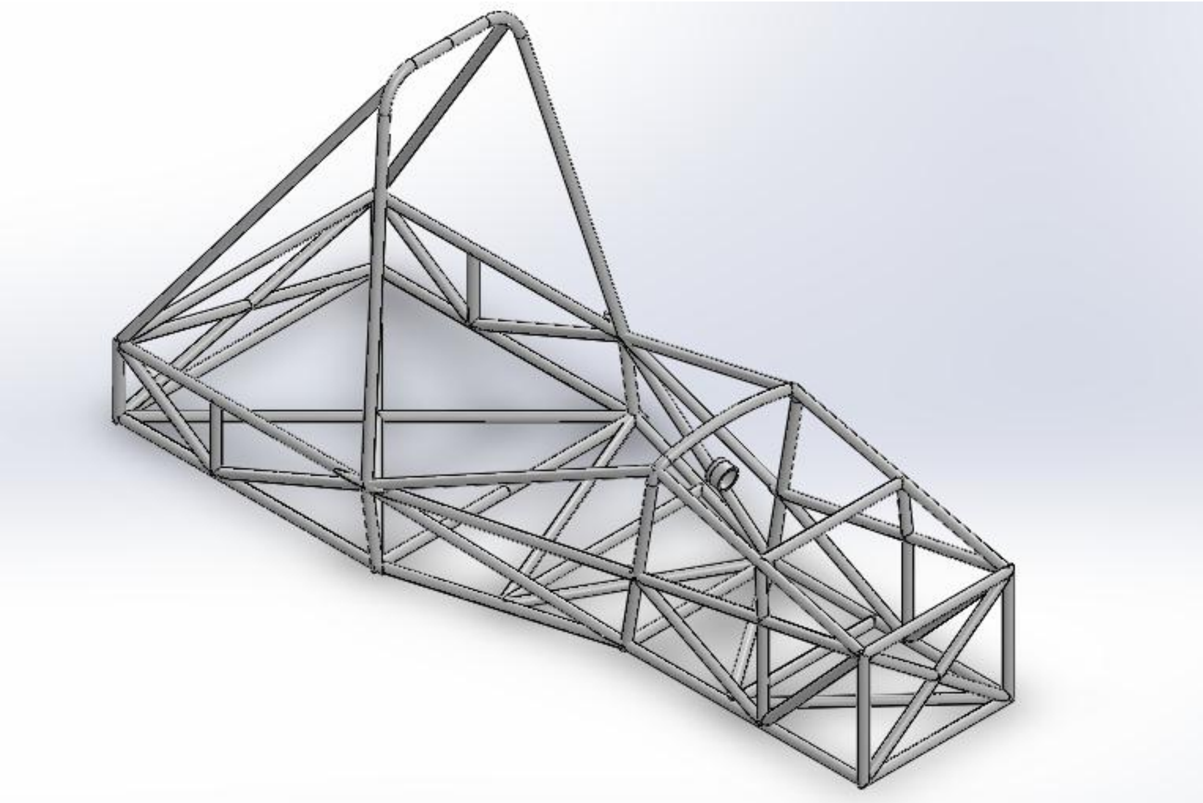

Figure 4.1 CAD model of the chassis of our vehicle

Following were the parts we had to manufacture and assemble:

- Modified suspension mountings for each upright
- A-arm mountings for the chassis
- Press-fit bushings for the a-arms, to house the rod-ends
- Engine mountings
- Attachment of the steering system to the chassis
- Making the pedal assembly operational
- Elongation of the axles
- Attachment of the a-arms to the chassis
- Attachment of the hubs to the a-arms
- Attachment of the shocks to the suspension and chassis
- Mounting the tires onto the hubs

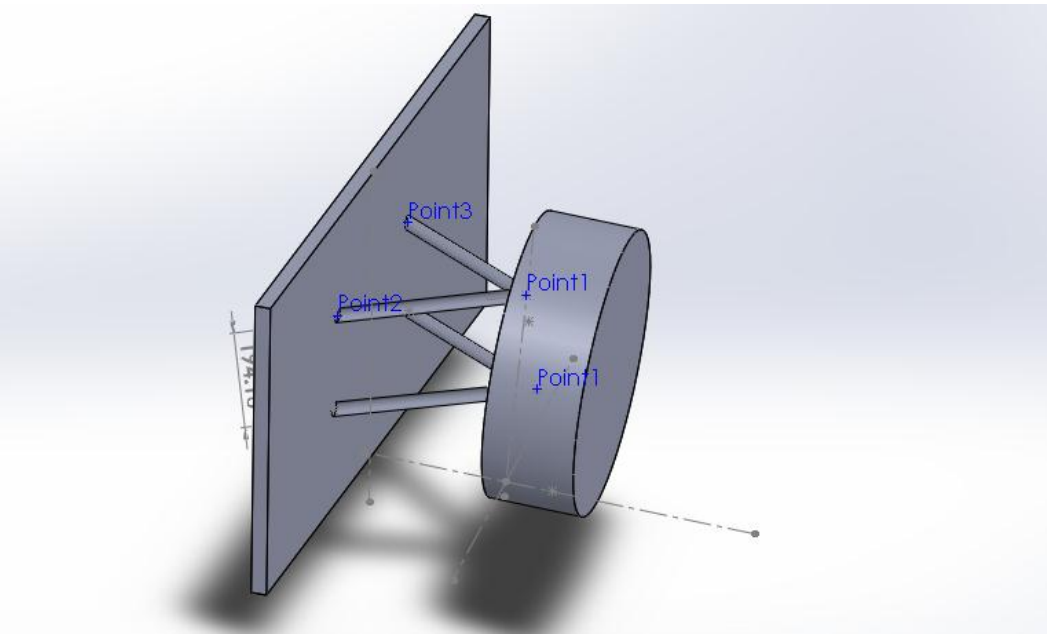

Figure 4.2 Uprights' steering behavior analysis

The unequal A-arms were used in conjunction with the uprights and Honda CG125 suspension springs to make up the suspension system. The caster factor was introduced for optimum steer feel and behavior. This was simulated through a model in Solidworks (Fig. 4.2.). Limited adjustability of around 8 degrees allowed for camber settings.


Figure 4.3 Modified hub and wishbone assembly

Figure 4.4 Rod ends

A custom built steering system was used in conjunction with quad bike tie-rods for length adjustments.

The axles had to be elongated due to the modifications in the hub and suspension. Furthermore, custom mountings had to be constructed for the engine.

Figure 4.5 Axle elongation

Figure 4.6 Left engine mounting

Figure 4.7 Right engine mounting

Figure 4.8 Mounted engine

4.2 Facts and Figures

Important calculated and measured figures are stated below:

- Engine weight = 103.5 ±2.5kg
- Chassis weight = 61.3 ±2.5kg
- Stiffness Constant Calculated (theoretical) = 20000 N/m
- Stiffness Constant Measured (practical) = 22000 N/m
- Sag in suspension under car weight = 5 cm
- Total suspension travel (y-axis) = 13 cm
- Caster = 12 degrees
- Camber (Adjustable) = -4 to 4 degrees
- Wheel size = 13 inches
- Scrub Radius = 4 mm
- Axle elongation (short) = 10.2 cm
- Axle elongation (long) = 11.4 cm

Other figures and calculations are available in solidworks files.

List of procedures and standards used is given below

- TIG welding is used throughout the chassis and components' connections
- Standard SS bolts were used. Majorly 10mm with certain exceptions e.g. Uprights bolts
- Standard engineering practices e.g. balancing methods, assembly procedures and manufacturing were carried out.
- All construction and manufacturing were carried out keeping into consideration FSAE rules.

4.3 Final Vehicle

Figure 4.9 Mounting the suspension and tires onto the frame

Figure 4.10 The prototype vehicle for the subsequent batches

Chapter 5
Conclusions and Future Work

5.1 Conclusions

In summary, we were able to design a sequential transmission for the Formula Student vehicle of our department. Furthermore, we conducted successful stress analyses on the gearwheels and shafts of our transmission, and even produced a 3D-printed prototype. On the other hand, we also assembled the prototype vehicle to near-completion, putting together the previously available parts with newly manufactured ones.

5.2 Future Work

Following are the list of tasks to be performed by the Formula Student team in the future:

a) Vibrational analysis of the designed gearbox.

b) Manufacturing of the sequential transmission and attaching it to the Daihatsu Mira engine of the actual racing vehicle for the competition.

c) Completing the prototype vehicle with the wiring of the engine, installation of the brakes, and attachment of body kits, seats and other external elements. With this vehicle up and running, the team will be able to secure sponsors for manufacturing the actual vehicle for the competition. This vehicle will also serve as a test vehicle on which the team will be able to perform trial and error.

d) Completing the competition vehicle which will feature a new chassis, suspension, braking and steering systems and the manufactured transmission as well.

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