

# Design and Fabrication of Two-Stage Reduction Gearbox

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**ABSTRACT:** All-terrain Vehicles considered as sport-utility vehicles have always had their share of craze. The performance of these vehicles compelled the younger generation to learn their functioning, while competitions like SAE BAJA challenge engineering students to design, fabricate and build ATVs of their own from scratch. This paper aims at developing a Two-stage reduction gearbox, a prime component of the drivetrain of an ATV. The name Two-stage reduction gearbox suggests the gearbox designed to produce two outputs using two sets of gears. In our case, a high torque-low velocity and high speed-low torque output, achieved by varying gear ratios presented in SAE Baja 2018, India manufactured to our best efforts considering the design goals, rules, and regulations laid down by SAE BAJA.

**KEYWORDS:** Two-stage reduction Gearbox, Briggs and Stratton, Baja, Integrated Shaft, CVT, Finite Element Analysis, Goodman's Equation.

## I. INTRODUCTION

In this project, various design goals were set considering Vehicle Dynamics, Durability, and driving Fatigue. This paper is a quick guide for students participating in national and international SAE BAJA competitions. It explains all the aspects of gearbox design like sizing calculations, analysis, material selection, and manufacturing processes. The paper also highlights the advantages of using a two-stage gearbox over a conventional off-the-shelf gearbox.

## II. PROBLEM STATEMENT

There are four principal shortcomings identified in the conventional gearbox design.

- 1)Weight- Too high
- 2)Compactness- Too low
- 3)Poor compatibility with the Briggs and Stratton, engine used at Baja event.
- 4) Low Torque output

## III. OBJECTIVES

This study focuses on developing a gearbox suitable for dynamic events like SAE BAJA with primary objectives as follows:

- 1)High Toque
- 2)Maximum Acceleration
- 3)Minimum Weight and High Reliability
- 4) Cost-Effectiveness

## IV. GEARBOX METHODOLOGY

### A. Selection of Gears using an evaluation matrix

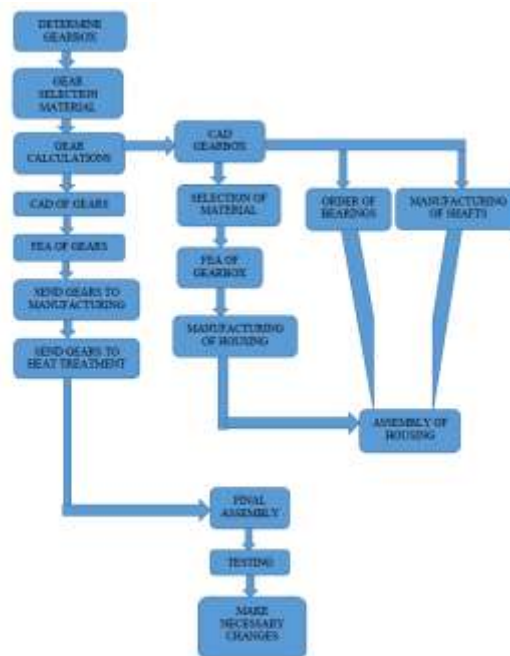
There are various arrangements of gears that can be employed to build a Two-stage gearbox. We used a matrix-based evaluation approach to effectively boil down to the optimal system, considering key parameters delineated in the first column of the following table. We presented the weightage of each key parameter in column II. Each gear type is assigned a score ranging from -2 to +2 (Positive score indicating favourability and vice-versa). The cumulative score for each gear type is then calculated by multiplying the score corresponding to a particular parameter with the latter's weightage and summing it all.

TABLE -1: Evaluation Matrix

PARAMETER	POINTS	SPUR	BEVEL	HELICAL	PLANETARY
Cost	20	2	-1	-1	-2
Weight	20	2	-1	2	-1

Manufacturing	10	2	2	2	-2
Ease of Assembly	5	2	-1	2	-2
Load Bearing Capacity	2	1	0	2	2
Advantages	1	0	-1	1	2
Maintenance	2	2	0	2	-2
Amount of Raw Material	2	2	-1	2	-2
Design Complexity	2	0	-2	-1	-2
<b>TOTAL</b>		120	-32	19	-96

### B. Flowchart for Designing and Manufacturing of Gears



### C. Classical Calculations for Design of Gears and Shafts

Module 2.5mm  
N 2600rpm  
Face width 20mm=0.7874"  
Diametral pitch 10.16  
Pinion dia 1.717"  
Gear dia 6.84"

#### 1<sup>ST</sup> STAGE:

- Torque(T)= $63000 \times \frac{6.84}{2600} = 165.731\text{f}$   
= $165.73 \times 4.448 = 737.16 \text{ N-m}$
- Transmitted load( $w_t$ )= $\frac{165.73 \times 2}{1.717} = 193.046\text{lbs}$
- Pitch line velocity(v)= $\frac{1.717}{2} \times \frac{1}{12} \times 2600 \times 2\pi = 1168.72\text{ft/min}$
- Overload factor( $k_o$ )=1.5

- Dynamic factor( $k_v$ ):  
 $B=0.25 \times (12-8)^{2/3} = 0.629$   
 $A=50+56 \times (1-0.629) = 70.77$   
 $k_v = \left( \frac{70.77 + \sqrt{1168.72}}{70.77} \right)^{0.629} = 1.28$
- Size factor ( $k_s$ ):  
 $(k_s)_{\text{pinion}} = 1.192 \left( \frac{0.7874 \sqrt{0.309}}{10.16} \right)^{0.0535} = 1.007$   
 $(k_s)_{\text{gear}} = 1.192 \left( \frac{0.7874 \sqrt{0.415}}{10.16} \right)^{0.0535} = 1.015$
- $k_m = 1 + c_{pf} + c_{ma}$   
 $c_{pf} = \frac{0.7874}{10 \times 1.717} - 0.025 = 0.02$   
 $c_{ma} = 0.127 + (0.0158 \times 0.7874) + (-0.930 \times 10^{-4} \times 0.7874^2) = 0.139$   
 $k_m = 1 + 0.02 + 0.139 = 1.159$
- Geometric factor(J) (from graph):  
 $J_{\text{pinion}} = 0.32$

$$J_{\text{gear}} = 0.41$$

**Bending stress:**

$$(\sigma_b)_{\text{pinion}} = \frac{193.046 * 1.5 * 1.28 * 1.007 * 10.16 * 1.159 * 1}{0.7874 * 0.34}$$

$$= 16417.01 \text{ psi}$$

$$(\sigma_b)_{\text{gear}} = \frac{193.046 * 1.5 * 1.28 * 1.015 * 10.16 * 1.159 * 1}{0.7874 * 0.41} = 13722.26 \text{ psi}$$

**Contact stress ( $\sigma_c$ ):**

Elastic coefficient ( $c_p$ ):

$$c_p = \left( \frac{1}{\pi * \left( \frac{1 - (0.29)^2}{29700 * 10^3} + \frac{1 - (0.29)^2}{29700 * 10^3} \right)} \right)^{0.5}$$

$$= 2776 \sqrt{N/mm}$$

$$I = \frac{\cos 20 * \sin 20}{2 * 1} * \frac{3}{3 + 1} = 0.12$$

$$(\sigma_c)_{\text{pinion}} = 2776 * \sqrt{\left( 193.046 * \frac{1.5 * 1.28 * 1.007 * 1.159}{1.717 * 0.7874} * \frac{1}{0.12} \right)}$$

$$= 143345.2998 \text{ psi}$$

$$(\sigma_c)_{\text{gear}} = 2776 * \sqrt{\left( 193.046 * \frac{1.5 * 1.28 * 1.015 * 1.159}{1.717 * 0.7874} * \frac{1}{0.12} \right)}$$

$$= 143913.56 \text{ psi}$$

**Strengths:**

• **Bending strength:**

$$S_t' = \frac{Y_N}{K_T * K_R} * S_t$$

$$Y_N = 1.3558 * N^{-0.0178}$$

$$(Y_N)_{\text{pinion}} = 1.3558 * (10^7)^{-0.0178} = 1.0176$$

$$(Y_N)_{\text{gear}} = 1.3558 * \left( \frac{10^7}{3} \right)^{-0.0178} = 1.0377$$

$$S_t = 102 * H_B + 16000 \text{ psi}$$

$$S_t = 102 * 576 + 16000 = 75152 \text{ psi}$$

$$(S_t)_{\text{pinion}} = \frac{1.0176}{1 * 1} * 75152 = 76474.67 \text{ psi}$$

$$(S_t)_{\text{gear}} = \frac{1.0377}{1 * 1} * 75152 = 77985.2304 \text{ psi}$$

• **Contact strength:**

$$S_c' = \frac{Z_N * C_H}{K_T * K_R} * S_C$$

$$Z_N = 2.466 * N^{-0.056}$$

$$(Z_N)_{\text{pinion}} = 2.466 * 10^7^{-0.056} = 0.99$$

$$(Z_N)_{\text{gear}} = 2.466 * \frac{10^7}{3}^{-0.056} = 1.06$$

$$S_C = 349 * H_B + 34000 \text{ psi}$$

$$S_C = 349 * 576 + 34000 = 235324$$

$$S_c' = \frac{0.99 * 1}{1 * 1} * 235324 = 232970.76 \text{ psi}$$

$$S_c' = \frac{1.06 * 1}{1 * 1} * 235324 = 249443.44 \text{ psi}$$

• **Factor of safety:**

**Bending:**

$$FS = \frac{S_t'}{\sigma_b}$$

$$(FS)_{\text{pinion}} = \frac{76474.67}{16417.01} = 4.65$$

$$(FS)_{\text{gear}} = \frac{77985.2304}{13722.26} = 5.68$$

**Contact:**

$$FS = \frac{S_c'}{\sigma_c}$$

$$(FS)_{\text{pinion}} = \frac{232970.76}{143345.2998} = 1.62$$

$$(FS)_{\text{gear}} = \frac{249443.44}{143913.56} = 1.73$$

Module 2.5mm  
N 2600rpm  
Face width 30mm=1.1811”  
Diametral pitch 10.16  
Pinion dia 1.717”  
Gear dia 6.2008”

**2<sup>ND</sup> STAGE:**

- Torque(T) =  $63000 * \frac{6.84}{866.6} = 497.25 \text{ lbf}$   
 $= 497.25 * 4.448 = 2211.768 \text{ N-m}$
- Transmitted load( $w_t$ ) =  $\frac{497.25 * 2}{1.717} = 579.2 \text{ lbs}$
- Pitch line velocity(v) =  $\frac{1.717 * \frac{1}{12} * 866.6}{2} * 2\pi = 389.54 \text{ ft/min}$

- Overload factor( $k_o$ ) = 1.5
- Dynamic factor( $k_v$ ):

$$B = 0.25 * (12 - 8)^{2/3} = 0.629$$

$$A = 50 + 56 * (1 - 0.629) = 70.77$$

$$k_v = \left( \frac{70.77 + \sqrt{389.54}}{70.77} \right)^{0.629} = 1.1673$$

- Size factor ( $k_s$ ):

$$(k_s)_{\text{pinion}} = 1.192 * \left( \frac{1.1811 * \sqrt{0.309}}{10.16} \right)^{0.0535} = 1.1294$$

$$(k_s)_{\text{gear}} = 1.192 * \left( \frac{1.1811 * \sqrt{0.426}}{10.16} \right)^{0.0535} = 1.1470$$

- $k_m = 1 + c_{pf} + c_{ma}$   
 $c_{pf} = \frac{F}{10 * d} - 0.0375 + 0.0125F$   
for pinion = 0.046  
for gear = -0.0036  
 $c_{ma} = 0.127 + (0.0158 * 1.1811) + (-0.930 * 10^{-4} * 1.1811^2) = 0.145$   
 $(k_m)_{\text{pinion}} = 1 + 0.046 + 0.145 = 1.191$   
 $(k_m)_{\text{gear}} = 1 - 0.0036 + 0.145 = 1.1414$

- Geometric factor(J) (from graph):

$$J_{\text{pinion}} = 0.34$$

$$J_{\text{gear}} = 0.34$$

**Bending stress:**

$$(\sigma_b)_{\text{pinion}} = \frac{579.2 * 1.5 * 1.1699 * 1.1294 * 10.16 * 1.191 * 1}{1.1811 * 0.34} = 34590.4 \text{ psi}$$

$$(\sigma_b)_{gear} = \frac{579.2 * 1.5 * 1.1699 * 1.1470 * 10.16 * 1.414 * 1}{1.1811 * 0.34}$$

$$= 33666.499 \text{ psi}$$

**Contact stress ( $\sigma_c$ ):**

Elastic coefficient ( $c_p$ ):

$$c_p = \left( \frac{1}{\pi * \left( \frac{1 - (0.29)^2}{29700 * 10^3} + \frac{1 - (0.29)^2}{29700 * 10^3} \right)} \right)^{0.5}$$

$$= 2276 \sqrt{N/mm}$$

$$I = \frac{\cos 20 * \sin 20}{2 * 1} * \frac{3}{3 + 1} = 0.12$$

$$(\sigma_c)_{pinion} = \frac{2276 * \sqrt{(579.2 * \frac{1.5 * 1.1699 * 1.1294 * 1.191}{1.717 * 1.1811}) * \frac{1}{0.12}}}{0.12}$$

$$= 170595.3548 \text{ psi}$$

$$(\sigma_c)_{gear} =$$

$$\frac{2276 * \sqrt{(579.2 * \frac{1.5 * 1.1699 * 1.1470 * 1.1414}{6.2008 * 1.1811}) * \frac{1}{0.12}}}{0.12}$$

$$= 88562.39 \text{ psi}$$

**Strengths:**

• **Bending strength:**

$$S_t = \frac{Y_N}{K_T * K_R} * S_t$$

$$Y_N = 1.3558 * N^{-0.0178}$$

$$(Y_N)_{pinion} = 1.3558 * (10^7)^{-0.0178} = 1.0176$$

$$(Y_N)_{gear} = 1.3558 * \left(\frac{10^7}{3}\right)^{-0.0178} = 1.0377$$

$$S_t = 102 * H_B + 16000 \text{ psi}$$

$$S_t = 102 * 576 + 16000 = 75152 \text{ psi}$$

$$(s'_t)_{pinion} = \frac{1.0176}{1 * 1} * 75152 = 76474.67 \text{ psi}$$

$$(s'_t)_{gear} = \frac{1.0377}{1 * 1} * 75152 = 77985.2304 \text{ psi}$$

• **Contact strength:**

$$S'_c = \frac{Z_N * C_H * S_C}{K_T * K_R}$$

$$Z_N = 2.466 * N^{-0.056}$$

$$(Z_N)_{pinion} = 2.466 * 10^{7-0.056} = 0.99$$

$$(Z_N)_{gear} = 2.466 * \frac{10^{7-0.056}}{3} = 1.06$$

$$S_C = 349 * H_B + 34000 \text{ psi}$$

$$S_C = 349 * 576 + 34000 = 235324$$

$$S'_c = \frac{0.99 * 1}{1 * 1} * 235324 = 232970.76 \text{ psi}$$

$$S'_c = \frac{1.06 * 1}{1 * 1} * 235324 = 249443.44 \text{ psi}$$

• **Factor of safety:**

**Bending:**

$$FS = \frac{S_t}{\sigma_b}$$

$$(FS)_{pinion} = 2.17$$

$$(FS)_{gear} = 2.23$$

**Contact:**

$$FS = \frac{S'_c}{\sigma_c}$$

$$(FS)_{pinion} = 1.40$$

$$(FS)_{gear} = 2.70$$



**D. RESULTS:**

**TABLE 2- STAGE 1:**

Torque	165.73 lbf
Transmitted load	193.06 lbs
$(FS)_{pinion}$	1.62
$(FS)_{gear}$	1.73

**TABLE 3- STAGE 2:**

Torque	497.25 lbf
Transmitted load	579.2 lbs
$(FS)_{pinion}$	1.40

$(FS)_{gear}$	2.70
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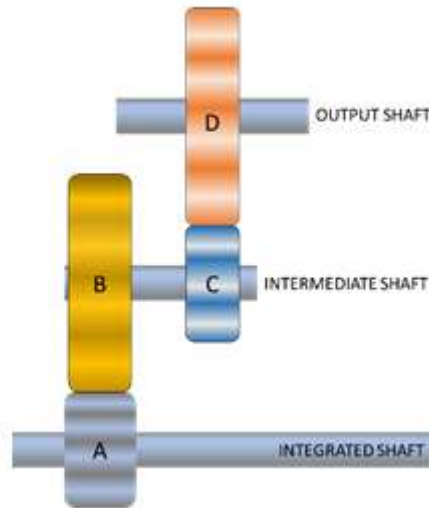
**TABLE 4- STAGE REDUCTION:**

STAGE 1	3:1
STAGE 2	3.5:1

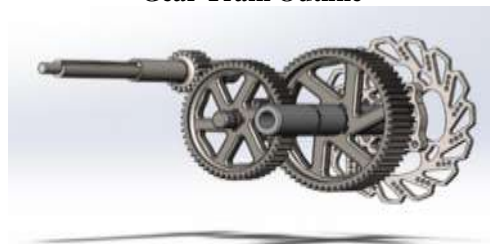
**E. Design Phase:**

We have designed the main elements- spur gears, shafts, and bearings using SOLIDWORKS design software following the design goals, CVT, and Engine specifications. The design is updated till it appears to us as there is no need to upgrade further. This design phase of ours is an imitation of

the realistic industrial challenges faced during the design of a gearbox considering all requirements for making it a market-ready product that is durable, smoother, and efficient that helps in quicker assembly and disassembly of the gearbox from the vehicle. The layout of the gear box is shown with the help of the following schematic.



**Gear Train Outline**



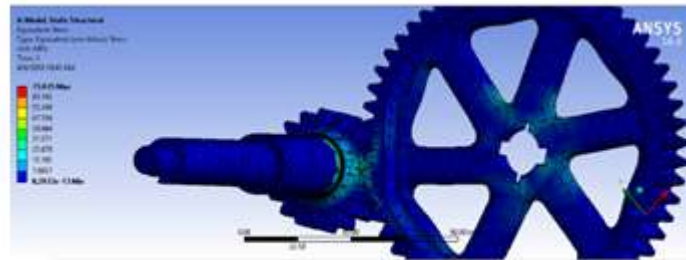
**Assembly of Gears**



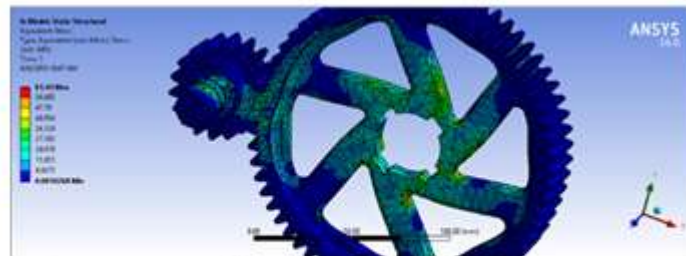
**Exploded View Of Gearbox**

### F. Analysis Phase:

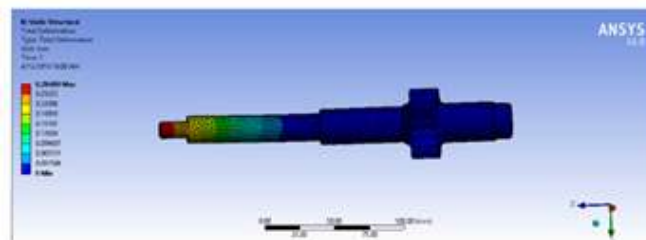
After Designing the Gearbox, we used a commercial Multiphysics modeling package called ANSYS for performing Finite-Element Analysis on the various elements such as gears, shafts, and the casing. This analysis serves the purpose of validating the results derived from Classical Calculations. We considered Forces from the Bearing and CVT for testing the analysis and obtaining the Casing FOS (Factor of Safety).



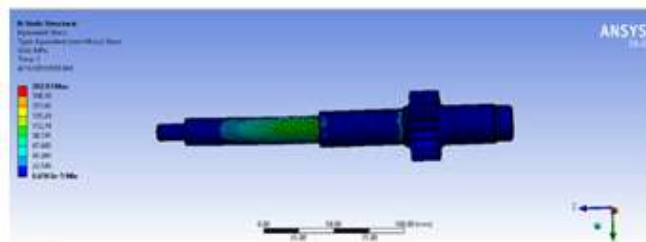
Stage one contact stress  
Result: 71.035 MPa



Stage two contact stress  
Result: 61.43 MPa

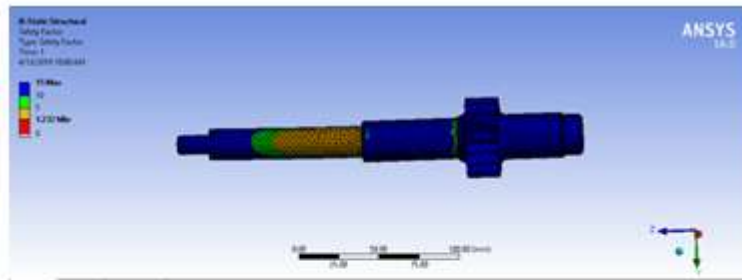


Input shaft deformation  
Result: 0.28409 mm

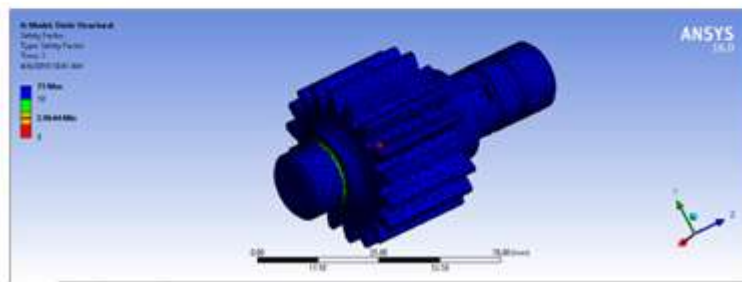


Input shaft stress  
Result: 202.93 MPa

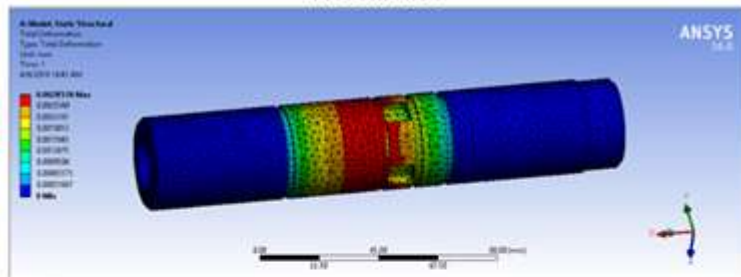




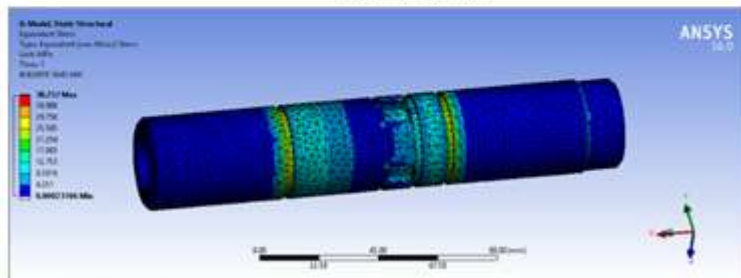
Input shaft factor of safety  
 Result: 1.232



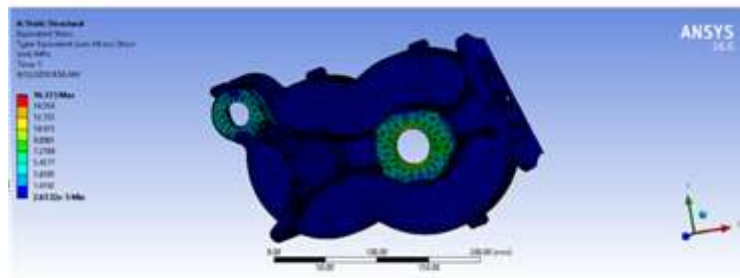
Intermediate shaft factor of safety  
 Result: 2.96



Output shaft deformation  
 Result: 0.0028



Output shaft stress  
 Result: 38.57 MPa



Casing stress  
 Result: 16.37 MPa

**G. Material Selection:**

For gears and shafts, AISI4340 is used. It has very good fatigue resistance. This alloy, 4340, may be heat treated to high strength levels while maintaining good toughness, wear-resistance, and fatigue strength levels, combined with good atmospheric corrosion resistance, and strength and it's affordable.

For Gearbox casing AI6061 T6 is used. Generally, 6061 aluminium is solution heat-treated, then aged. T4 temper aluminium is naturally aged, and T6 temper aluminium is artificially aged for maximum strength. The key reason to choose AI6061 over AI7075 is that AI6061 is highly machinable and easily weldable.

**Mechanical Properties of AISI 4340.**

Mechanical Properties	Values
Tensile Strength, Ultimate	1110 MPa
Tensile Strength, Yield	710 MPa
Modulus of Elasticity	205GPa
Bulk Modulus	140GPa
Poisson's Ratio	0.29
Machinability	50%
Shear Modulus	80GPa

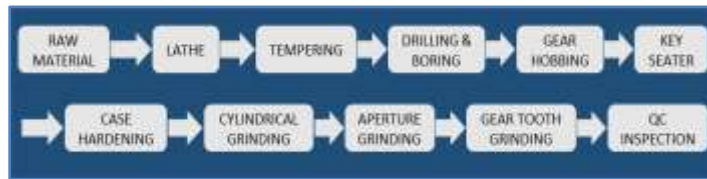
**Mechanical Properties of AI 6061 T6.**

Mechanical Properties	Values
Tensile Strength, Ultimate	310 MPa
Tensile Strength, Yield	276 MPa
Modulus of Elasticity	68.9GPa
Poisson's Ratio	0.33
Machinability	50%
Shear Modulus	26GPa

**H. Manufacturing:**

We have concluded the theoretical aspects of the project. The next stage is the Manufacturing of the Gearbox. We developed a flow chart containing a meticulous record of all the steps involved, such as raw materials selection, the Manufacturing Process, and Quality Inspection of the Gearbox.





## V. CONCLUSION

We arrived upon the initial sizing of the various elements of the Gearbox using Classical Methods. We then modeled all the components using SOLIDWORKS and prepared the 3D Assembly. Ultimately, we used ANSYS to validate the Classical Calculations using the Finite-Element Method.

In conclusion, we chose to use and analyze the design of the sequential transmission for the Baja Vehicle due to its superiority over the manual transmission. After selecting which transmission system to implement into the vehicle, we calculated the forces acting against it in the Hill Climb Challenge. We then calculated the gear ratio required to ascend the hill in the least possible time. We assumed an ascent rate of 100 feet in around 4 seconds for the Acceleration Test, letting the vehicle's top speed hit roughly 35 miles per hour. The gear ratios again were calculated to take these values into account. Eventually, We compared the gear ratios needed for the Hill Climb and Acceleration Tests, respectively, and arrived at the final gearbox Design.

- a. Mechanical engineering design by shigley
- b. Automotive transmission by Herald naunheimer
- c. Web:Gears educational systems; gearseds.com

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## VII. Hand books: