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 Torque
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>
<thead>
<tr>
<th>PARAMETER</th>
<th>POINTS</th>
<th>SPUR</th>
<th>BEVEL</th>
<th>HELICAL</th>
<th>PLANETARY</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cost</td>
<td>20</td>
<td>2</td>
<td>-1</td>
<td>-1</td>
<td>-2</td>
</tr>
<tr>
<td>Weight</td>
<td>20</td>
<td>2</td>
<td>-1</td>
<td>2</td>
<td>-1</td>
</tr>
</tbody>
</table>

TABLE - I: Evaluation Matrix
B. Flowchart for Designing and Manufacturing of Gears

C. Classical Calculations for Design of Gears and Shafts

<table>
<thead>
<tr>
<th>Manufacturing</th>
<th>Ease of Assembly</th>
<th>Load Bearing Capacity</th>
<th>Advantages</th>
<th>Maintenance</th>
<th>Amount of Raw Material</th>
<th>Design Complexity</th>
<th>TOTAL</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>120</td>
</tr>
</tbody>
</table>

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

1ST STAGE:
- Torque(T) = 63000 * 6.04/2600 = 165.731f
  = 165.73 * 4.448 = 737.16 N-m
- Transmitted load(w_i) = 165.73 * 2 = 193.046lbs
- Pitch line velocity (v) = \( \frac{1.717 \times 2 \times 2600 \times 2 \pi}{12} = 1168.72 \text{ft/min} \)
- Overload factor \( k_o \) = 1.5

- Dynamic factor \( k_d \): B = 0.25 * (12 - 8) \( \frac{6}{3} \) = 0.629
- A = 50 + 56 * (1 - 0.629) = 70.77
- \( k_d = \left( \frac{70.77 + \sqrt{1168.72 \times 0.629}}{70.77} \right) = 1.28 \)
- Size factor \( k_s \):
  \[ k_{s\text{pinion}} = 1.192 \left( \frac{0.7874 \times 0.309}{10.16} \right) = 1.007 \]
  \[ k_{s\text{gear}} = 1.192 \left( \frac{0.7874\sqrt{0.15}}{10.16} \right) = 1.015 \]

- Geometric factor \( J \) (from graph):
  \[ J_{\text{pinion}} = 0.32 \]

Bending stress:

\[
(\sigma_b)_{\text{pinion}} = \frac{193.046+1.5 \times 1.28 \times 1.007 \times 10.16 \times 1.159^1}{0.7874+0.34} = 16417.01 \text{ psi}
\]

\[
(\sigma_b)_{\text{gear}} = \frac{193.046+1.5 \times 1.28 \times 1.015 \times 10.16 \times 1.159^1}{0.7874+0.41} = 13722.26 \text{ psi}
\]

Contact stress (\(\sigma_c\)):

\[
\sigma_c = \frac{1}{2}(1-0.29^2)^{1/2} \left( \frac{1}{29700+18^2} - \frac{1}{29700+10^2} \right) = 2776N/mm
\]

\[
I = \frac{\pi}{3} \times 20 \times 20 \times 20 \times 3 = 0.12
\]

\[
(\sigma_c)_{\text{pinion}} = 2776 \sqrt{(193.046+1.5 \times 1.28 \times 1.007 \times 1.159 \times 1.717 \times 0.7874)} \times \frac{1}{0.12} = 143345.2998 \text{ psi}
\]

\[
(\sigma_c)_{\text{gear}} = 2776 \sqrt{(193.046+1.5 \times 1.28 \times 1.015 \times 1.159 \times 1.717 \times 0.7874)} \times \frac{1}{0.12} = 143913.56 \text{ psi}
\]

Strengths:

- **Bending strength**:

  \[
  S_c = \frac{Y_N}{K_T+K_R} \times S_t
  \]

  \[
  Y_N = 1.3558 \times N^{0.0178}
  \]

  \[
  (Y_N)_{\text{pinion}} = 1.3558 \times (10^7)^{0.0178} = 1.0176
  \]

  \[
  (Y_N)_{\text{gear}} = 1.3558 \times (10^7)^{0.0178} = 1.0377
  \]

- **Contact stress**:

  \[
  S_c = \frac{Z_N \times C_H \times S_t}{K_T \times K_R}
  \]

  \[
  Z_N = 2.466 \times N^{-0.056}
  \]

  \[
  (Z_N)_{\text{pinion}} = 2.466 \times 10^{-7} \times 10^{-0.056} = 0.99
  \]

  \[
  (Z_N)_{\text{gear}} = 2.466 \times 10^{-7} = 1.06
  \]

- **Factor of safety**:

  **Bending**:

  \[
  \text{FS} = \frac{S_t}{\sigma_b}
  \]

  \[
  (FS)_{\text{pinion}} = \frac{26474.67}{1617.01} = 4.65
  \]

  \[
  (FS)_{\text{gear}} = \frac{24944.44}{143913.56} = 5.68
  \]

**Contact**:

\[
\text{FS} = \frac{S_c}{\sigma_c}
\]

\[
(\text{FS})_{\text{pinion}} = \frac{13348.2598}{1617.01} = 8.32
\]

\[
(\text{FS})_{\text{gear}} = \frac{14944.44}{143913.56} = 1.03
\]

**2ND STAGE**:

- Torque(T)=63000 \times 6.84 = 497.25lbf

  \[
  = 497.25 \times 4.448 = 2211.768 \text{ N-m}
  \]

- Transmitted load(w) = \frac{497.25 \times 2}{1.171} = 579.2lbs

- Pitch line velocity(v) = \frac{193345}{70.77} = 2304 \text{ m/min}

- *2\pi=389.54 \text{ ft/min}

- Overload factor(k_o) = 1.5

- Dynamic factor(k_d):

  \[
  B = 0.25*(12-8)^{\frac{1}{3}} = 0.629
  \]

  \[
  A = 50+56*(1-0.629) = 70.77
  \]

  \[
  k_o = \frac{70.77 + \sqrt{389.54}}{70.77} = 0.629
  \]

  \[
  k_d = \frac{10.16}{0.145} = 1.1294
  \]

  \[
  k_o = \frac{1.1294}{10.16} = 0.117
  \]

- Geometric factor(J) (from graph):

  \[
  J_{\text{pinion}} = 0.34
  \]

  \[
  J_{\text{gear}} = 0.34
  \]

**Bending stress**:

\[
(\sigma_b)_{\text{pinion}} = \frac{5792.1+1.5 \times 1.1699 \times 1.1294+0.16 \times 1.191^1}{11811+0.34} = 34590.4
\]

\[
506 \text{ psi}
\]
\[ \sigma_{bg} = \frac{579.2 \times 1.5 \times 1.1699 \times 1.1470 \times 10.16 \times 1.4141}{1.1811 + 0.34} \]

= 33666.499 psi

Contact stress \((\sigma_c)\): 

Elastic coefficient \((c_p)\):

\[ c_p = \left( \frac{1}{\pi \times \left( \frac{1}{29700 + 10^6} \right)} \right)^{0.5} \]

= 2276√\(N/mm\)

\[ I = \cos \frac{20 \times \sin 20}{2+1} \times \frac{3}{3+1} = 0.12 \]

\[ (\sigma_c)_{pinion} = 2276 \times \sqrt{(579.2 \times \frac{1.5 \times 1.1699 \times 1.1470 \times 1.1414}{6.2000 \times 1.1811} \times \frac{1}{0.12})} \]

= 88562.39 psi

\[ (\sigma_c)_{gear} = 2276 \times \sqrt{(579.2 \times \frac{1.5 \times 1.1699 \times 1.1470 \times 1.1414}{6.2000 \times 1.1811} \times \frac{1}{0.12})} \]

= 88562.39 psi

Strengths:

- **Bending strength:**

  \[ S_t = K_T \times S_i \]

  \[ Y_N = 1.3558 \times N \times 0.0178 \]

  \[ (Y_N)_{pinion} = 1.3558 \times (10^7)^{0.0178} = 1.0176 \]

  \[ (Y_N)_{gear} = 1.3558 \times \left( \frac{10^7}{3} \right)^{0.0178} = 1.0377 \]

  \[ S_i = 102 \times H_p + 16000 \text{psi} \]

  \[ S_c = 102 \times 576 + 16000 = 75152 \text{ psi} \]

- **Contact strength:**

  \[ S_c = Z_N \times H_p + 34000 \text{psi} \]

  \[ (Z_N)_{pinion} = 2.466 \times 10^7 - 0.056 = 0.99 \]

  \[ (Z_N)_{gear} = 2.466 \times 10^7 - 0.056 = 1.06 \]

- **Factor of safety:**

  **Bending:**

  \[ FS = S_T / \sigma_b \]

  \[ (FS)_{pinion} = 2.17 \]

  \[ (FS)_{gear} = 2.23 \]

  **Contact:**

  \[ FS = \sigma_c / S_c \]

  \[ (FS)_{pinion} = 1.40 \]

  \[ (FS)_{gear} = 2.70 \]

D. RESULTS:

**TABLE 2- STAGE 1:**

<table>
<thead>
<tr>
<th>Torque</th>
<th>165.73 lbf</th>
</tr>
</thead>
<tbody>
<tr>
<td>Transmitted load</td>
<td>193.06 lbs</td>
</tr>
<tr>
<td>(FS)\text{pinion}</td>
<td>1.62</td>
</tr>
<tr>
<td>(FS)\text{gear}</td>
<td>1.73</td>
</tr>
</tbody>
</table>

**TABLE 3- STAGE 2:**

<table>
<thead>
<tr>
<th>Torque</th>
<th>497.25 lbf</th>
</tr>
</thead>
<tbody>
<tr>
<td>Transmitted load</td>
<td>579.2 lbs</td>
</tr>
<tr>
<td>(FS)\text{pinion}</td>
<td>1.40</td>
</tr>
</tbody>
</table>
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.

<table>
<thead>
<tr>
<th>TABLE 4- STAGE REDUCTION:</th>
</tr>
</thead>
<tbody>
<tr>
<td>STAGE 1</td>
</tr>
<tr>
<td>STAGE 2</td>
</tr>
</tbody>
</table>

(FS)_{gear} 2.70

[Images of gear train outline, assembly of gears, and 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).
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
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, thenged. 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.

<table>
<thead>
<tr>
<th>Mechanical Properties</th>
<th>Values</th>
<th>Mechanical Properties</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tensile Strength, Ultimate</td>
<td>1110 MPa</td>
<td>Tensile Strength, Ultimate</td>
<td>810 MPa</td>
</tr>
<tr>
<td>Tensile Strength, Yield</td>
<td>710 MPa</td>
<td>Tensile Strength, Yield</td>
<td>276 MPa</td>
</tr>
<tr>
<td>Modulus of Elasticity</td>
<td>205GPa</td>
<td>Modulus of Elasticity</td>
<td>68.9GPa</td>
</tr>
<tr>
<td>Bulk Modulus</td>
<td>140GPa</td>
<td>Poisson's Ratio</td>
<td>0.33</td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
<td>0.29</td>
<td>Machinability</td>
<td>50%</td>
</tr>
<tr>
<td>Machinability</td>
<td>50%</td>
<td>Shear Modulus</td>
<td>26GPa</td>
</tr>
</tbody>
</table>

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.

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