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Design and Topology Optimization of two-stage Gearbox for All-Terrain Vehicles

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ABSTRACT: Engineering design is an iterative process that requires to be dealt with all feasible design solutions in order to arrive at a desired objective. With the advent of technologies, engineering design is reaching its new heights thereby incorporating solutions with compact designs, lighter weight and cost efficiency. The main purpose of this assignment is to design and analyse a two-stage reduction automotive gearbox used in an All-Terrain Vehicle competing in BAJA SAE INDIA. The gearbox is coupled to a Continuously-Variable Transmission which is powered by a 7.5KW- four-stroke gasoline engine. The first stage includes strength calculations to obtain dimensions of the gearbox and its components. The second stage involves design modelling of the components in CATIA V5R21. In next stage, Finite-Element analysis is performed to validate the design. The final stage involves testing of the gearbox on diverse and harsh terrains.

KEYWORDS: Reduction gearbox; strength calculations; Finite-Element Analysis

I. INTRODUCTION

A gearbox is defined as a mechanical system used to transmit power in certain distance, usually involving a change in speed and torque. Gearbox consists of components like gears, shafts, keys, bearings and snap rings. Gears are defined as toothed wheels or multi-lobed cams, which transmit power and motion from one shaft to another by means of successive engagement of its teeth. Gears are mainly typed as spur gears, helical gears, double helical gears, bevel gears, crown gears, hypoid gears, worm gears, rack and pinion etc. The goal of this project is to develop a lightweight, compact gearbox which will increase the efficiency, performance, and durability of the vehicle.

This paper presents the design, calculation and analysis of vehicle acceleration, spur gears and casing. The material used for gears is SAE 8620 and casing is made of 6061-T6 Aluminium Alloy. We have used AGMA equations for stress calculation of gears and calculated bending moment of the shaft for minimum allowable diameter. The speed reduction of the entire unit was less than 10:1 and hence spur gears were best suited for the application. From cost considerations, spur gears are the cheapest since their manufacturing is easy and faster as compared to helical gears. Usually, in commercial vehicles a differential unit is mounted within the gearbox which allows the outer drive wheel to rotate faster than the inner drive wheel during a turn. It also works as a reduction gearing. However, the racing competition of BAJA SAE INDIA consists of harsh terrains including bumps, ditches and mud pools. Since the differential allows both driven wheels to spin at different angular velocities; if one wheel gets stuck in such terrain, the differential sends power to the other wheel and thus it is difficult for the vehicle to manoeuvre through it. In addition to this, the weight of the differential itself is 1.5- 2.0 kg which is a significant contribution to a 150kg racing ATV. Considering these circumstances, the differential unit is being replaced by a Locked Axle, which evenly splits the power in 1:1 ratio between the driven wheels irrespective of the terrain.

For further weight reduction, Topology Optimization is being performed on the gears using Hypermesh v13.0 thereby reducing the weight by 14%. The final drive gear assembly provides the final stage of gear reduction in order to

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decrease the RPM and increase the rotational torque and thus, the wheels never spin as fast as the engine (in almost all applications) even when the CVT is in overdrive condition.

II. LITERATURE REVIEW

In the book, ‘Fundamentals of Vehicle Dynamics’, the author Thomas D. Gillespie has prescribed principles of vehicle dynamics including acceleration, braking and cornering demonstrating the forces and moments generated by the vehicle at the ground. The vehicle and its components are studied to determine what forces will be produced by the dynamic behaviour, and how the vehicle will respond to these forces. The author has stated a methodology for calculating the maximum performance of a motor vehicle in longitudinal direction along with the comparison of performance characteristics of gasoline and diesel engines through speed-power-torque-specific fuel consumption curves. Though case studies, procedure to calculate tractive force and acceleration is explained which is being used in this paper.

In the book, ‘Design of Machine Elements’, the author V.B. Bhandari has stated applications of scientific principles, technical information and innovative ideas for the development of a new or improved machine. The book focuses on Indian Standards for material selections, screw threads, springs, gears, shafts and pressure vessels along with selection of a machine component from manufacturer’s catalogue. The design procedures required for designing the components of a gearbox are described and the maximum distortion energy theory is identified to be the major criterion of failure of a mechanical component loaded in elastic limit.

In the book, ‘Manual Gearbox Design’, the author Alec Stokes has provided guidance to the amount of work involved in the design of a manually operated automotive gearbox and the problems that can be encountered both during the design stages and in operation. The comparison between different types of gear is made and recommendations are suggested based on application. The book focuses on the design of racing-type rear engine mounted gearbox. The design procedures for the internal gear pack are discussed, along with the arrangements of the various shafts. The author further emphasizes on the problems that can be encountered with the lubrication system and various methods used to cope with the high speeds and heavy tooth loads.

In the book, ‘Practical Finite Element analysis’, the author, Nitin S. Gokhale has introduced the domains of FEA including Topology Optimization. For geometries having considerable lengths in all the three dimensions, Hexahedrons are proved to provide most appropriate simulation results. A pure HEX mesh can be obtained through solid mapped meshing. The author has given values for maintaining element quality which includes Aspect ratio, Skewness, Jacobian; to obtain best possible results of simulation.

III. DESIGN TARGETS

1. To maximize the tractive power in accordance with the terrain and manoeuvre fastest through it.
2. To provide an optimum reduction ratio so that the resultant torque will enhance the performance of the ATV in acceleration event (a 45.72m straight patch of loose soil which is to be covered in minimum possible time), sled pull (a 1.5 tonne trolley which is to be pulled on a 35m straight patch of loose soil).
3. Design for manufacturing and Assembly (DFMA) for reduction in costs and complexity in assembling the gearbox.

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IV. DESIGN CALCULATIONS

A. Input Characteristics

Engine Model	Briggs and Stratton OHV Intek
Engine Displacement	305 CC
Max Power	9.11 HP @ 3800 Rpm
Max Torque	18.65 N-m @ 2600 Rpm
Compression ratio	8.1 to 1
Fuel type	Gasoline 87 Octane
CVT Overdrive ratio	0.98:1
CVT Low ratio	3.89:1

Table 4.1:- Input characteristics

As per the rules and regulations of BAJA SAE INDIA, all the All-Terrains Vehicles must use the same engine prescribed by the organising committee. The specifications of the engine are mentioned in the above table. The engine is couple to a Gaged GX-9 Continuously-Variable Transmission (CVT).

B. Material Selection-

The gears are manufactured in integration with the shafts in order to avoid the complexity of keys and snap rings.

Material Specification	Value
Surface Hardness	62 HRC
Ultimate Tensile Strength	1150 MPa
Permissible Bending Stress (σ_b)	384 MPa
Yield Strength	880 MPa
Material Density	7850 kg/m ³
Young's Modulus	210 GPa

Table 4.2:- Material specifications

The main aim of material selection is to minimize the cost while meeting the product performance goals. SAE 8620 is being used for gears and shafts because of its excellent fatigue characteristics and easy availability at a low cost. Other materials available for gears are 20MnCr5, however the above UTM test results proved us that SAE 8620 was best suited for the application.

C. Calculations for Vehicle Speed and Torque-

1. Mass of the vehicle with driver = 250 kg
2. Effective tyre radius (R)=11.5 inch (D= 0.5842 m)
3. Primary pulley CVT rpm = Max. Engine rpm i.e. 3800
4. Secondary pulley rpm = Max Primary RPM/CVT Reduction= 565.5rpm
5. Secondary Pulley torque = Engine Torque*CVT reduction= 71.38 Nm
6. Gearbox Reduction = 7.4 (Selected by various iterations)
7. Gearbox Input rpm = Secondary pulley rpm
8. Gearbox Output rpm = Gearbox Input rpm/ Gearbox reduction= 76.42 rpm

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9. Total Reduction= CVT Reduction*Gearbox Reduction = 28.78
10. Wheel Torque = Torque at secondary pulley*Gearbox Reduction= 528.21 N-m

An iterative design process was used to calculate gearbox reduction such that a perfect balance between torque & speed was maintained keeping a focus on the different terrains. The performance parameters of the transmission system are-

- i. **Torque- 528.21 N-m @ 2400 rpm**
- ii. **Speed- 59.86 km/hr @ 3800 rpm**
- iii. **Acceleration- 5.26 m/s²**

D. Spur Gear Design

1. *Following are the Input parameters for 1st stage-*

Reduction ratio- 2.68:1; Torque- 71.38 N-m; N=565.55rpm; $\alpha= 20^\circ$; min number of teeth= 18

$$V = \frac{\pi * D * N}{60} = 16.62 \text{ m/s} \sim 59.86 \text{ km/hr}$$

Hence to achieve a maximum speed of 59.86 km/hr, a gearbox reduction of 7.4:1 is required.

2. *Module estimation on the basis of beam strength-*

$$m = \sqrt[3]{\left[\frac{60 * 10^6}{\pi} \left\{ \frac{(kw) * C_s * f_s}{z^2 * N * C_v * \frac{b}{m} * \frac{S_{ut}}{m} * Y} \right\} \right]} \quad \dots \text{ where, } C_s=1.5; f_s=1.5; Y= 0.308; C_v = \frac{3}{3+v} = 3/8 \text{ (assuming } v=5\text{m/s)}$$

$$m = 1.89 \text{ mm}$$

3. *Module estimation on wear strength*

$$m = \sqrt[3]{\left[\frac{60 * 10^6}{\pi} \left\{ \frac{(kw) * C_s * f_s}{z^2 * N * C_v * \frac{b}{m} * Q * K} \right\} \right]} \quad \dots \text{ where, } Q = \frac{2 * Z_g}{z_g - z_p} = 3.44; K = 0.156 \left[\frac{BHN}{100} \right]^2 = 6.75$$

$$m = 2.32 \text{ mm}$$

Based on the above 2 equations **module = 2mm**

4. *Gear & Pinion Parameters*

1. Pinion Diameter(D_p)=m*Z_p=36mm
2. Gear Diameter(D_g)=m*Z_g=86mm
3. Beam Strength S_b= $\sigma_b * b * m * Y = 3548.16 \text{ N}$

5. *Check for Design*

1. Moment (M_t)= $\frac{60 * 10^6 * Power}{2 * \pi * N_p} = 126648.49 \text{ N-mm}$
2. Tangential Force (P_t)= $\frac{2 * M_t}{m * Z_p} = 7036.02 \text{ N}$
4. Service factor (C_s)= 1

5. *Effective Load (P_{eff})*

1. Velocity of Pinion (V_p)= $\frac{\pi * D_p * N_p}{60000} = 0.9451 \text{ m/s}$
2. C_v= $\frac{3}{3+v} = 0.7604$
3. P_{eff}= $\frac{C_s * P_t}{C_v} = 9253.05 \text{ N}$

6. *Factor of Safety f_s*

$$f_s = \frac{S_b}{P_{eff}} = 1.65$$

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7. Factor of safety based on dynamic load by Buckingham's Equation
 1. For Pinion
 2. For Gear,
 3. For Grade 8; $e_p=16+ 1.25 \varphi= 20.37\mu\text{m}$...where, $\varphi= m+ 0.25\sqrt{Dp}= 3.5$
 $e_g=16+ 1.25 \varphi= 21.39\mu\text{m}$
 4. Error between centre distance= $e= e_p+ e_g=41.76*10^{-3}$ mm
 5. Deformation factor= 11400 N/mm^2
 6. Dynamic Load (P_d)= $\frac{21V(\text{beC}+Pt_{\text{max}})}{21V+\sqrt{\text{beC}+Pt_{\text{max}}}}= 119.68 \text{ N}$
 7. $P_{\text{eff}}= P_t+ P_d= 7155.702 \text{ N}$
 8. $f_s= \frac{S_b}{P_{\text{eff}}}= 1.06$

Thus, the design is satisfactory.

8. Following are the Input parameters for 2nd stage-

Reduction ratio- 2.676:1; Torque- 191.30 N-m; $N=76.42\text{rpm}$; $\alpha= 20^\circ$; min number of teeth= 19

Using similar formulas the Factor of Safety was being calculated to produce a satisfactory design.

E. Design of Gearbox Casing

1. The gearbox casing is designed by maintaining adequate clearances between the casing and gears
2. Proper rigid mounting is provided for gearbox and engine which ensures the sturdy transmission system.
3. Thickness = $(0.03A_0+3.797) = 5 \text{ mm}$

V. 3D MODELLING

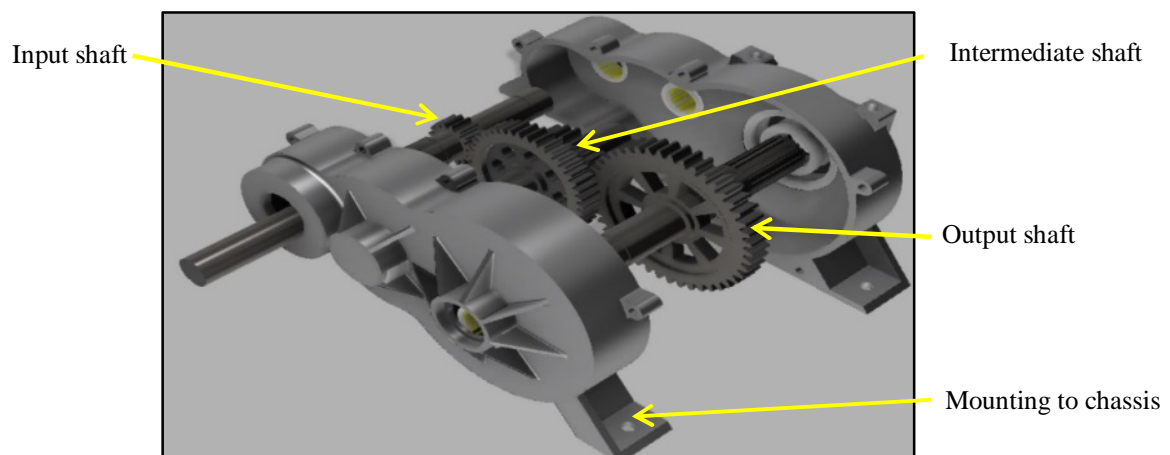


Fig 5.1:- Exploded view of Gearbox Assembly

Catia V5 R21 was being used for 3D modelling of the components of gearbox. Extensive use of ANSYS Fluent was done to ensure that external contour of the gearbox is such, that oil is circulated to all gears and bearings even if the gearbox is mounted at an angle on the chassis. The input shaft is connected to the secondary pulley of CVT through a key. Input shaft meshes with the intermediate shaft which provides change in direction of rotation. At the final stage, it is connected to the output shaft. The driveshafts are connected to the output shaft through splines.

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VI. FINITE ELEMENT ANALYSIS

Entire Analysis of the gearbox and its components was done using Hypermesh 13.0 and ANSYS Workbench 16.0

A. Input Shaft-

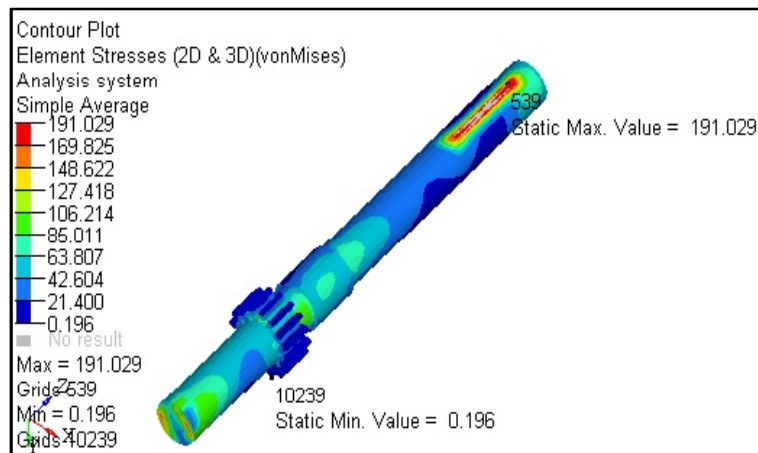


Fig 6.1: - Input shaft Von-Mises stresses

Type of Elements- Mixed elements with tetrahedrons < 10 %; Element size- 4 mm; Material SAE 8620. Rigid elements were created to apply torsional force exerted by the CVT and shear force by weight of CVT. Tooth load was applied as per Buckingham's law of tooth loading. The shaft was constrained at the location of bearings.

B. Intermediate Shaft

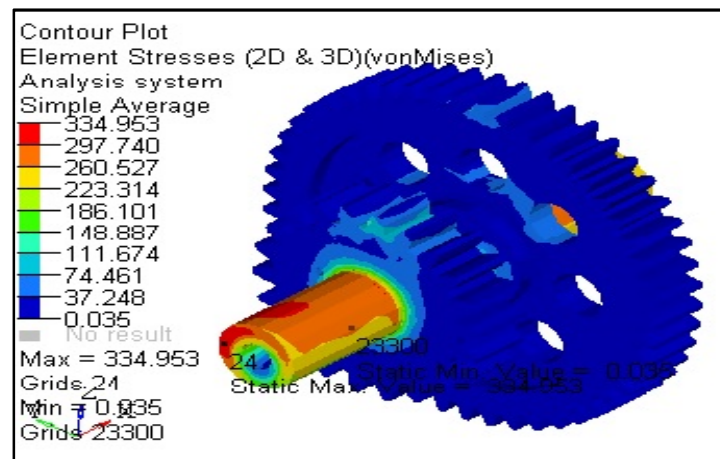


Fig 6.2:- Intermediate shaft Von-Mises stresses

Type of Elements- Mixed elements with tetrahedrons < 10 %; Element size- 4.5 mm; Material SAE 8620. Since the stress generated on the face gears was less, through optimization the diameters holes drilled was obtained. However, during initial runs a considerable stress concentration was seen at the location where gear is integrated to the shaft which is a critical area. To compensate for this, a hole was drilled in the shaft which shifted the localization of stress from that location.

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C. Topology Optimization of Output shaft

1. Optimization can be defined as automatic process to make a system or component as good as possible based on an objective function and a subject to certain design constraints. There are many different methods to organize a certain structure
2. We have used Optistruct solver for optimizing our components
3. Topology optimization is mathematical technique that produces an optimized shape and material distribution for a structure within a given package space. It removes the material from the designed space and creates voids.

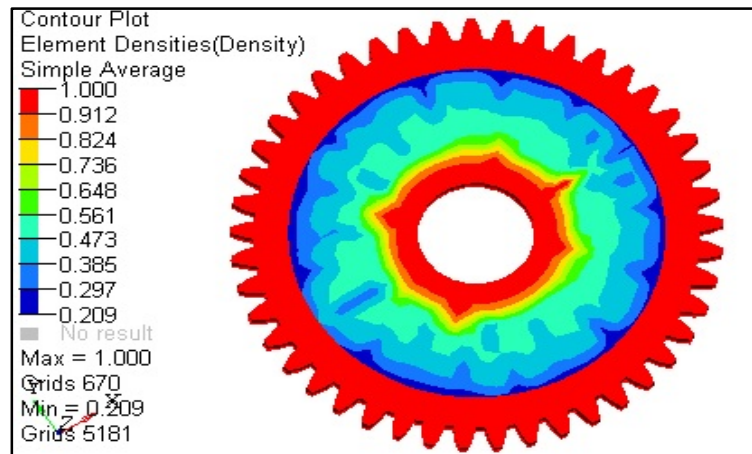


Fig 6.3:- Topology Optimization of Output Gear

Type of Elements- Mixed elements with tetrahedrons < 10 %; Element size- 4.5 mm; Material SAE 8620. Topology Optimization was done using Optistruct solver. The red area represents non-design area, i.e. the area specified to the solver from where it should not remove the material. For effective tooth engagement and for manufacturing feasibility it is advisable that there should be no voids in such area. Rest area is specified as design area. Following constraints were specified to the solver-

- a. Maintain a FOS of 1.5 at the highly stressed location.
- b. Reduce up to 30% of the total volume (30 % of un-optimized body).
- c. Create voids wherever possible.

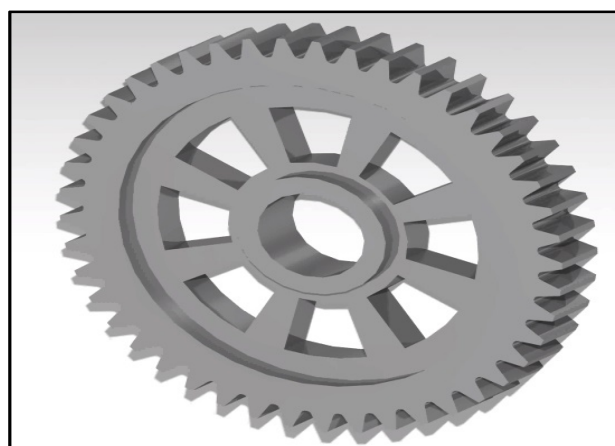


Fig 6.4:- Refined CAD Model of Output Gear

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The results obtained were needed to be interpreted focusing on manufacturing feasibility, since Optistruct solver removes material and creates voids wherever the given constraints are satisfied. However, it is practically impossible to drill holes are change the contour at random locations. Thus by refining the obtained results a smooth CAD is generated in CATIA. The resulted model is analysed in ANSYS Workbench for further design validation.

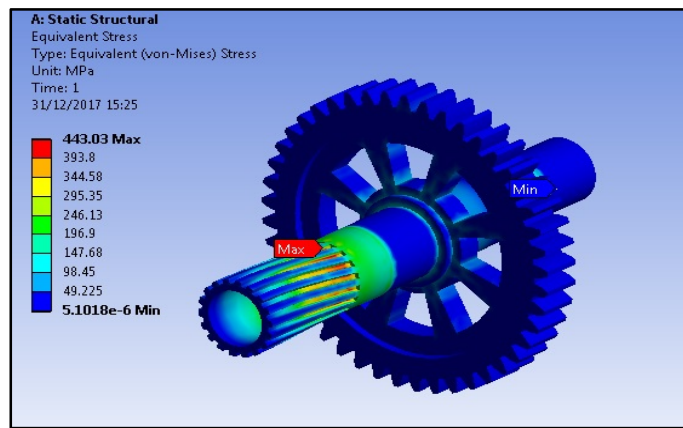


Fig 6.7:- Output Gear Von-Mises stresses

The optimized gear was further analysed in Static Structural for the validation design and topology optimization. A total weight reduction of 16.3% in output gear itself was obtained through the entire simulation process. Since in-board braking was used for the ATV, the brake disc was mounted on the splines of the gear. Thus a braking torque was also applied on the splines.

D. Gearbox Casing-

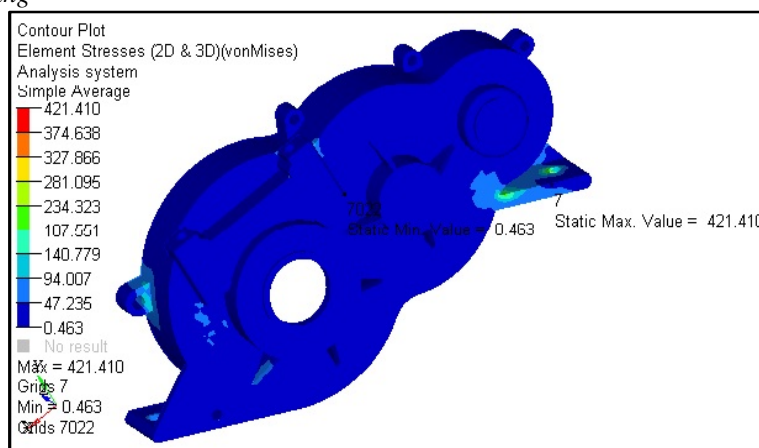


Fig 6.9:- Gearbox Casing Von-Mises stresses

Type of Elements- Mixed elements with tetrahedrons < 20 %; Element size- 3 mm; Material- 6061-T6 Al. Bearing reactions were applied at bearing locations. If the bearing fails, the entire torque will be transmitted into the casing. Considering this extreme situation, a moment on all the nodes was applied in Optistruct solver.

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VII.RESULTS

Component	Results		
	Equivalent Von-Mises Stress	Total Deformation	Life (min load cycles before fatigue failure)
Input Gear	191.02 MPa	0.139 mm	3.496E+05
Intermediate Gear	334.95 MPa	0.442 mm	7.122E+04
Output Gear	443.03 MPa	0.119 mm	6.596E+04
Gearbox Casing	421.41 MPa	0.688 mm	-

Table 7.1:- Results of Finite Element Analysis

The values of deformation were also obtained in simulations which were well within elastic limits. Using the Goodman theory of fatigue failure, the life of the components was calculated in terms of load cycles. The estimated running of the vehicle was around 700 km. The life of the components satisfies this phenomenon

VIII. CONCLUSION

The finite element method is best suitable for evaluating and verifying the design. It bridges the gap between design and manufacturing thereby reducing costs and provides an optimum solution. The performance of the gearbox was tested on the ATV at real time running conditions. The acceleration timings were measured using Arduino-based Data Acquisition System. The topology optimization has significantly contributed in reducing the overall weight of the vehicle thereby improving the acceleration by 15%.

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