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DESIGN AND ANALYSIS OF GEARBOX WITH INTEGRATED CV JOINTS

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ABSTRACT

The gearbox of an automatic transmission of an All-Terrain vehicle is the third element of the power train. The engine is coupled with a continuous variable transmission (CVT) which is further coupled to the gearbox. It is used to change the speed and torque of the vehicle according to variation of road and load conditions. For better acceleration of the vehicle as well as efficient hill climbing the weight of the vehicle should be low without compromising with the strength of the vehicle. A standard two stage reduction gear box consists of an input shaft, idler shaft and output shaft which finally transmits the power to the wheels through CV joints coupled with the output shaft. The disadvantage of a standard gearbox is, it adds up an extra rotational mass to the output shaft and a larger centre to centre distance between input and output shaft. The result of the report deals with the designing and analysis of a lighter and a more compact gearbox with integrated CV joints without jeopardising its performance specifications.

Key words: gearbox, All- terrain vehicle, integrated, CV joint, output shaft, CVT.

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1. INTRODUCTION

Gearbox is the heart of the transmission system which varies the torque flow towards the wheels of the vehicle. Therefore, gearbox designing is a critical and intricate process and involves bit by bit calculations of each and every component. After the calculations are done, the designing of gear train is done in a CAD software. The designing of the gear train gives us the outline of the gearbox casing .To optimise casing by reducing its weight without compromising the load flow path, we carry out the topology optimisation analysis in a CAE software. This gives us the shape of the casing through which we design the whole assembly of the gearbox and then we go through with analysing the final design.

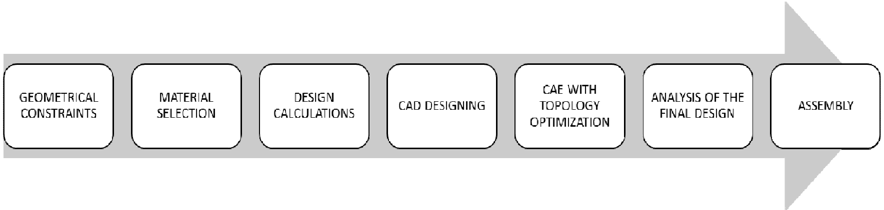


Figure 1 Flowchart depicting entire process of Designing

The main intent of integrating the CV joints with the output shaft was to increase the compactness and decrease the overall mass of the gearbox. The data for the design calculations is based on a CVT driven 200 kg Baja ATV participating in Baja SAE competitions. The designing is done with the constraints of reducing the size of the gearbox to the extent where the gearbox does not hinder with other powertrain components. The assembly of the gearbox consists of a compound gear train comprising of an output gear, input gear and two idler gears. In a standard gearbox, the output gear is connected to the output shaft which is further connected to an OEM CV joint as shown in **Figure 2**, but in this peculiar design the output shaft does the job of mounting of the output gear on it, and it plays the role of CV joints as well, thus integrating the output shaft and the CV joints.

Figure 2 Conventional gear box with OEM CV joints

2. MATERIAL SELECTION

- For decreasing the weight without compromising with the strength, material selection plays a critical role.
- For gears, a case hardened material with high tensile strength as well as high endurance strength is used to withstand static as well as dynamic loads.
- For shafts, a through hardened material with high torsional strength is selected which can withstand bending loads and can transfer the torque at higher rpm.
- For gearbox casing, material with high strength to weight ratio is needed which can withstand radial loads coming from gears.

From above pre-requisites, following materials are selected:-

Table 1 Material and their properties

GEARS	SHAFTS	CASING
SAE 9310, mock carburized, 790°C reheat, 150°C temper	SAE 4340 normalized	Aluminium 7050-T7451 Annealing Temperature 410°C
MECHANICAL PROPERTIES		
Tensile Strength - 1286 Mpa Yield Strength – 980 Mpa Hardness - 40 HRC Shear Modulus - 80 GPa	Tensile Strength - 1110 MPa Yield Strength - 710 MPa Shear Modulus- 80 GPa Hardness – 35 HRC	1. Tensile Strength 524 MPa 2. Yield Strength - 469 MPa 3. Shear Modulus - 26.9 GPa

3. SPECIFICATIONS

Max. Torque = 19.6 Nm @ 2600 rpm

Max. Power= 10 hp @ 3800 rpm

Engine Capacity= 305 cc

Max speed selected = 50 km/h

$$\text{Speed} = \frac{\pi * D * N}{60 \text{ x gear ratio x cvt low ratio}}$$

CVT lower Ratio= 3.5

CVT Higher Ratio= 0.9

D= Tire diameter = 21 inch = 0.533m

N= engine rpm = 3600 rpm (max)

Gear ratio = 7.1

Total reduction = gear ratio * CVT ratio

= 7.1*3.5 = 24.85

4. CALCULATIONS FOR MODULE AND DIAMETER OF GEARS

For First stage gear A and B:

Factor of safety = 1.5

Service factor = 1

No. of teeth on A are 20 i.e. Z_a=20, b=8m, Y= 0.320 for Z_a=20 teeth

Using the formula Module is given by,

$$m = \left\{ \frac{60 * 10^6}{\pi} \left(\frac{(P)(cs)(fs)}{z * n * C * \frac{b}{m} * \frac{uts}{s} * y} \right) \right\}^{\frac{1}{3}}$$

By solving,

m= 2 mm

Now,

Diameter of gear = d_a = z_a.m

= 2*20

= 40 mm

Also, Module for gear B = 2 mm.

First stage reduction = 2.55

No. of teeth in gear B = 51

Diameter of gear B = 51*2 = 102 mm

Similarly, for 2nd stage gears

Second stage reduction = 2.77

N = 1411 rpm

For Second stage gear C and D:

Factor of safety = 1.5

Service factor = 1

No. of teeth on A are 20 i.e. Z_c=22, b=8m, Y= 0.330 for Z_c=22 teeth

Assuming pitch line velocity as 3 m/s

$$\text{Velocity factor, } C_v = \frac{3}{3+3} = 0.5$$

Module is given by,

$$m = \left\{ \frac{60*10^6}{\pi} \left(\frac{(P)(cs)(fs)}{z*n*C*\frac{b}{m}*\frac{uts}{3}*y} \right) \right\}^{\frac{1}{3}}$$

By solving, we get

m= 2.25 mm

Now,

Diameter of gear = $d_c = z_c.m$

= 2.25*22

= 47.25 mm

Also, Module for gear D = 2.25 mm

Diameter of gear = $d_d = z_d.m$

= 2.25*61

= 137.25 mm

Calculations for Static load on gear:

Torque transmission (x) is given by,

$M_t = \frac{60*10^6*P}{2*\pi*N} = 19600$ Nmm

Net Torque = M_t * Cvt low ratio = 19600 * 3.5 = 68600 Nmm

Tangential Load,

$$P_t = \frac{2*Mt}{da} = 3430\text{N}$$

Similarly, for Gear B = 3430 N

Gear C = 7100 N

Gear D = 7100 N

5. DESIGN OF IDLER SHAFT

Idler shaft is designed by the following formula:-

Maximum shear stress is given by,

$$\tau_{max} = \frac{16}{\pi*d^3} * \sqrt{(Kb*Mb)^2 + (Kt*Mt)^2}$$

For uniform load application,

K_b= combined shock and fatigue factor applied to B.M = 1.5

K_t = combined shock and fatigue factor applied to T.M = 1

Solving this,

d ≅ 18 mm

Similarly, for input shaft, d = 20 mm

6. DESIGN OF OUTPUT HOLLOW SHAFT AS WE ARE ONSIDERING AN OEM RZEPPA JOINT

∴, Inner diameter = 52 mm, Outer Diameter = 65 mm

Torque transmitted by shaft, Mt = $\frac{P*60}{2*\pi*N}$

= 494 Nm

Maximum shear stress theory for hollow shaft is,

$$\tau_{max} = \frac{16}{\pi*do^3(1-k^4)} * \sqrt{(Kb*Mb)^2+(Kt*Mt)^2}$$

$$K = \frac{do}{di} = \frac{65}{52} = 0.8$$

By solving,

τ_{max} = 15.84 N/mm^2

Allowable shear stress for EN 24 is 210 Mpa.

∴ hence, design is safe.

7. SELECTION OF BEARING

As we know that only radial loads come in spur gears, so we have selected Machined Needle roller bearings which will meet the necessary requirements and will support radial loads.

For idler shaft:-

The bearing life for the gearbox which will run in the competition as well as for the testing purposes is estimated in between 150-200 hrs.

L_{10h} = 150 hrs

Radial load, Fr = 4000 N

$$L_{10} = \frac{60*n*l10h}{10^6}$$

$$= \frac{60*1411*150}{10^6}$$

= 12.7 mrev

Dynamic load capacity,

$$C = P(L10)^{\frac{1}{3}}$$

$$C = 4000*(12.7)^{\frac{1}{3}}$$

C = 9255 N

From SKF catalogue, TAFI 172916 bearing no. is selected which has Dynamic load capacity of 14700 N.

Similarly, all shaft bearings are selected through the same procedure.

8. STATIC ANALYSIS ON ANSYS WORKBENCH 16

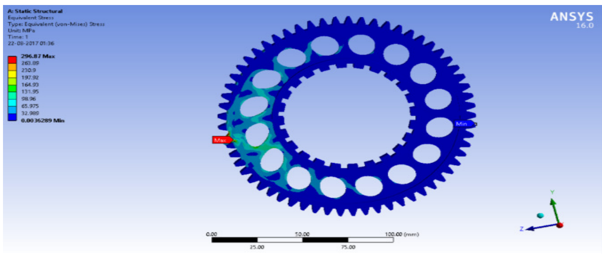


Figure 3 Stress distribution in 2nd stage output gear

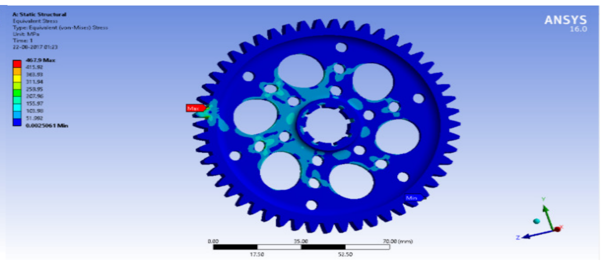


Figure 4 Stress distribution in 1st stage idler gear

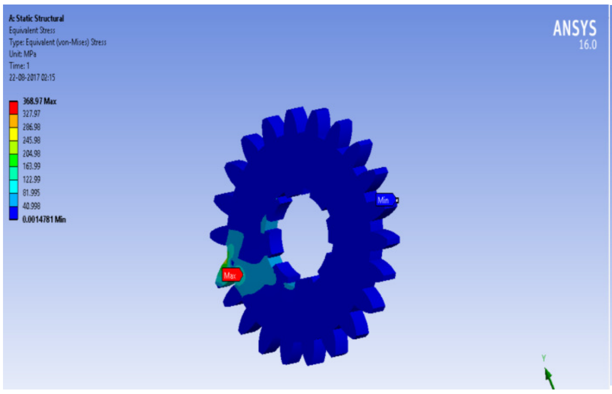


Figure 5 Stress distributions in 2nd stage idler gear

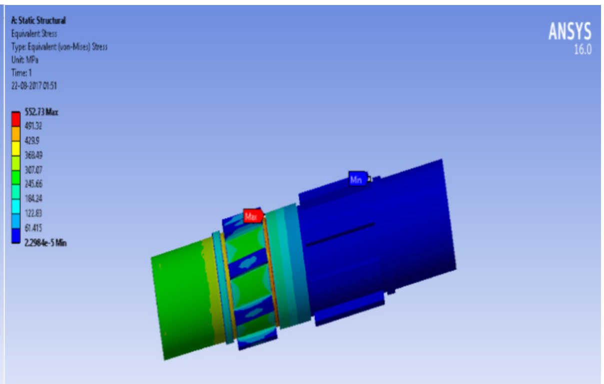


Figure 6 Stress distribution in idler shaft

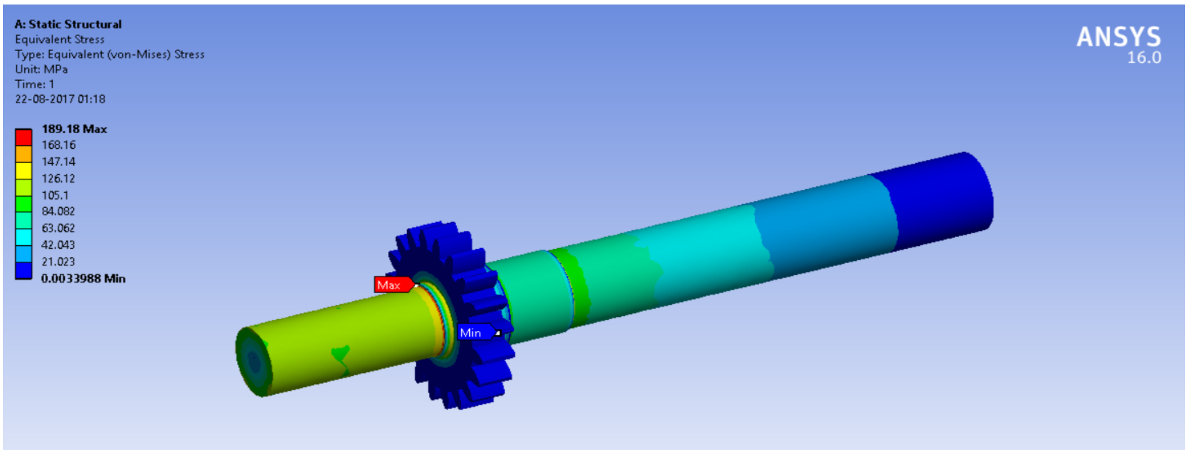


Figure 7 Stress distributions in input shaft with integrated gear

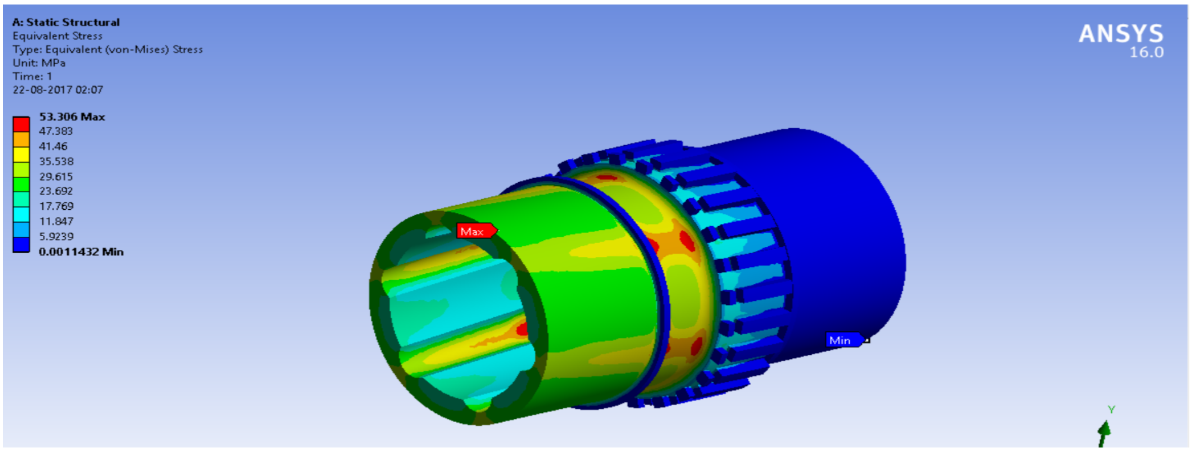
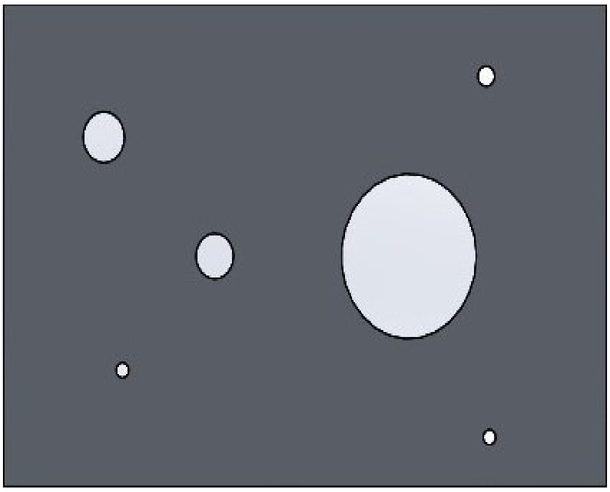


Figure 8 Stress distributions in output shaft with integrated CV joints

9. ANALYSIS AND OPTIMIZATION OF CASING IN HYPERWORKS

After the culmination of the analysis of the gears, we get a rough shape of the casing of the gear train. The rough shape of the casing is then modelled in a CAD software. After which the topology optimization of the casing block is done using Hyperworks software. The constraints being the displacement and the main objective of the study being the minimization of the volume. This study provides us with the cardinal information about the stress flow throughout the Casing of the gear train.



The minimum material which is required for the casing, shown in **Figure 10**. Although this shape will take the load without breaking, there are various constraints such as machinability and the ease of assembly which compels us in putting more material to it and designing the casing which is aesthetically pleasing and complies with the aforementioned constraints. In the figure, the blue region indicates that the material can be added to improve the machinability and packaging, whereas the red region in the shape has to be maintained so as to sustain the loads coming.

The final iteration of the gearbox can be seen in **Figure 11**. Necessary cavities that were also input in the Hyperworks (**Figure 9)** software for the topology optimisation are created to accommodate different components such as bearings and shafts for the incorporation of the gearbox in the power train unit. According to topology optimization study, another cavity is created as material was not needed in that region. But as the gearbox needs to be completely sealed, incorporation of Carbon Fibre is done to reduce the weight of the casing as much as possible.

10. RESULTS

After carrying out the structural stress analysis, the above stress plots were obtained. The maximum stresses that were analysed were within the permissible stress limits i.e. lesser than the yield strength of the materials of respective components. The shafts are found to have an average factor of safety of 2, the gears having a factor of safety of 1.5 and the gearbox casing is found with a factor of safety of 3. The output shaft with the integrated CV joints has a factor of safety 5.The gearbox in the cad software weighs only 4.1 kg which is way lesser than the weight of the conventional gearbox i.e. about 20% without compromising with the efficiency.

10.1 Weight Comparison

Table 2 Weight Comparison of Conventional and modified design

Component	Conventional	Modified design
Casing	1.6 kg	1.1 kg
Gears	2 kg	1.4 kg
Shafts	0.5 kg	0.4 kg
Output shaft + CV joint separate	2 kg	-
Output shaft with integrated CV joint	-	1.2 kg
Total Weight	5.1 kg	4.1 kg

11. CONCLUSIONS AND FURTHER SCOPE OF WORK

Nowadays the popularity of ATVs is increasing day by day so the companies can implement this design in future to decrease the overall weight and cost of the ATV. Furthermore, vehicle with a CVT based transmission connected to a gearbox.

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