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"Design and Optimization of Two Stage Transaxle for an ATV"

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ABSTRACT: The drive-train, the propelling system of the vehicle is one of the essential system in any automobile whose function is to transfer the required torque and power generated by the engine to the wheels as and when required by the driver. The main aim of this paper is to increase the overall performance of the transmission system by reducing the rotating inertial mass through customizing two stage reduction Transaxle having Cage-less Differential so as to attain the maximum speed and torque during the dynamic event in BAJA off road motorsport event. This research includes all design considerations, calculations and implementation of various components used in the designing of transaxle. The objective of this work is to design and develop a transaxle which is reliable, safe and cost effective. The modelling part was carried out in Solid works and analysis was done using ANSYS Workbench 15.0. A sequential approach of gear design, bearing design was adopted. Its use on an All-Terrain Vehicle was seen to be effective.

Keywords: Inertial losses, Gear analysis, Transaxle, Tractive Effort, Von misses stress, Solidworks and ANSYS 15.0

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I. INTRODUCTION

The transmission system is one of the most important systems to consider when designing a car. A transaxle performs both the gear-changing function of a transmission and the power-splitting ability of an axle differential in one integrated unit. In the automotive field, a transaxle is a major mechanical component that combines the functionality of the transmission, the differential, and associated components of the driven axle into one integrated assembly. In addition to allowing the driver to change gear ratios (the function of the transmission), it also has to be able to distribute torque to the drive wheels.

Since a transaxle consists of a transmission, differential, and other final drive components, there are a lot of potential failure points to consider. The differential of Transaxle designed in this study comes without the housing inside the crown gear. This results in weight ultimately reducing inertial losses, besides space reduction and makes the differential compact. This Transaxle was designed for an ATV made for a competition called BAJA. BAJA is an annual motorsport event held by Society of Automotive Engineers (SAE). In order to achieve a better acceleration, it is very important to keep the weight as low as possible. The strength, reliability and weight should be optimized to the best possible outcome. Inertial forces or transient forces are primarily comprised of acceleration related forces where a change in velocity is required. These are Rotational Inertia requirement and the Translation mass. If rotational mass is added to a translating vehicle, it adds not only rotational inertia but also translation inertia.

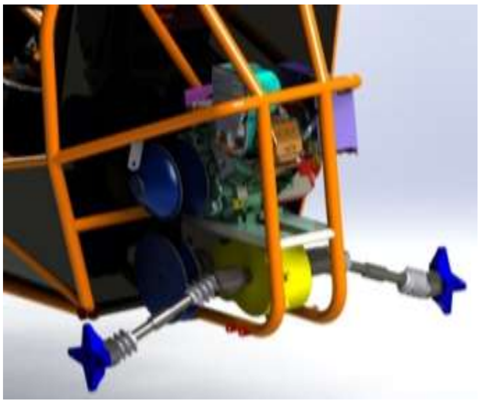


Fig. 1 Layout of Transaxle

II. PRE-REQUISITES

The different parameters used for design and analysis of differential were selected considering an off road vehicle powered by 10 hp engine, 305 cc capacity.

2.1 Design Methodology:

1. Identifying the system key requirements.
2. Selection of the transmission.
3. Selection of gear reduction.
4. Checking the feasibility of various parts.
5. Designing and analyzing of customized parts.

6. Deciding the implementation and layout.

2.2 Design Considerations:

1. To reduce power-train assembly weight.
2. To minimize inertial mass for reducing power loss.
3. To optimize gear reduction.
4. To damp maximum vibrations generated by the power train assembly.
5. To increase serviceability of each part.

Also, it is considered that the differential is coupled with a gearbox which effectively provides a gear ratio of 36. The various other important parameters are stated below.

1 Gross Vehicle Weight = 240 kgs
2 Weight Distribution = Front – 96 kgs, Rear – 144 kgs
3 Wheel Track = 49 inch
4 Engine Power = 10 hp
5 Engine Torque = 19.2 Nm @ 2700 rpm
6 Engine Maximum RPM = 3800 rpm
7 Tyre Size = 23 x 7 x 10 inch
8 Gear Material = 20MnCr5, Yield Strength: 850MPa and Ultimate Strength: 1300 MPa (Post Harden)
9 Material for Transaxle Casing = Al 7075-T6

III. METHODOLOGY

The research methodology is subdivided into three phases namely design of transaxle(gearbox and centrally suspended cageless differential), geometric modelling and FEM analysis. The adopted methodologies are described in brief below:

The preliminary design of the customized transaxle will be done in accordance to calculations based on the data relevant for the design of gears and forces acting on them.

3.1 Designing of Gears

3.1.1 Force Analysis of the Spur Gear

First of all we find the module of the gears according to the beam strength of the gear by using ultimate tensile strength of 20MnCr5 and Factor of safety we need for the gear.

By using the formula from here, we get

$$m = \sqrt[3]{60X10^6\{\frac{(Kw)(Cs)(F.O.S)}{(Z)(n)(Cv)\left(\frac{b}{m}\right)\left(\frac{Sut}{3}\right)Y\pi}}$$

m = 2.5 for spur gears
m = 4 for bevel gears

The resultant force P acting on the tooth of the spur gear is resolved into two components

That is as follows:-

- Pt, Pr
- Pt = Tangential component (N)
- Pr = radial component (N)

S.No	No. of teeth	Module	Face Width (mm)	Dia (mm)	Speed
1	14	2.5	20	35	7037
2	42	2.5	20	105	2345
3	14	2.5	20	35	2345
4	62	2.5	20	155	530

Table 1: Gear selection Criteria.

Amount Of Torque Transmission

As according to the engine – power curve the maximum torque in given by the engine at 2600 rpm

Torque transmitted by the 1st gear

$Mt_1 = 60\times10^6 \times Kw/2\pi n_1$

$Mt_1 = 10119.18$ N-mm

Torque transmitted by the gear 2nd

$Mt_2 = 60\times10^6 \times Kw/2\pi n_2$

$Mt_2 = 30357.56$ N-mm

Torque transmitted by the gear 3rd

$Mt_3 = 60\times10^6 \times Kw/2\pi n_3$ $Mt_3 = 30357.56$ N-mm

Torque transmitted by the gear 4th

$Mt_4 = 60\times10^6 \times Kw/2\pi n_4$ $Mt_4 = 13456.8$ N-mm

3.1.2 Tangential forces on the gears

Tangential load on 1st gear

$Pt_1 = 2\times Mt_1 /D$

$Pt_1 = 578.238$ N

Tangential force on 2nd gear

$Pt_2 = 2\times Mt_2 /D$

$Pt_2 = 578.238$ N

Tangential force on 3rd gear

$Pt_3 = 2\times Mt_3 /D$

$Pt_3 = 1734.71$ N

Tangential force on 4th gear

$Pt_4 = 2\times Mt_4 /D$

$Pt_4 = 1733.63$ N

3.1.3 Velocity of each gear is

$v = (\pi dn/60\times10^3)$

$v_1 = 12.896$ m/s

$v_2 = 12.896$ m/s

$v_3 = 4.298$ m/s

$v_4 = 4.298$ m/s

3.1.4 Deformation factor of gears

Taking safety factor of 1.75 for the medium and moderate shock as generated by the driving and driven parts respectively.

As deformation factor (C) is calculated by the Young's modulus of 20MnCr5 , which is 1.9×10^5, by using the formula

$C = \frac{k}{\frac{1}{Eg\,1}+\frac{1}{Eg\,2}}$ where k=0.111 for full depth 20^0 gears

$C = 11400\ N/mm^2$

3.1.5 Dynamic Load And Effective Load

The dynamic load will be maximum at 3800 rpm (maximum rpm) , as at that time the gears would have the maximum pitch line velocity.

So, by using the formula

$$P_d = \frac{21v(Ceb+Pt)}{21v+\sqrt{(Ceb+Pt)}}$$

We get,

$P_{d1} = 2151.356N$

$P_{d2} = 2151.356\ N$

$P_{d3} = 2229.62\ N$

$P_{d4} = 2229.11\ N$

After this here is the effective load which is calculated by the EARLE BUNKINGHAM equation which is,

$P_{eff} = C_s P_t + P_d$

$P_{eff\,1} = 3163.2725\ N$

$P_{eff\,2} = 3163.2725\ N$

$P_{eff\,3} = 5265.36\ N$

$P_{eff\,4} = 5262.96\ N$

from here we can see that, $p_{eff} <$ beam strength

Therefore the design of spur gears and related parameters is safe.

3.1.6 Design Of Sun And Bevel Gear Of Differential Assembly

As the ring gear rotates it will rotate the whole differential assembly too. So, the torque on the crown gear will be equal to the torque on the pinion. Then,

Pitch cone angle of the gear of a sun gear

tanA = N/n

Where,

N = number of the teeth on the gear (sun)

n = no of teeth on pinion (planet)

A = 60.94

Sun module - 4

$Mt_p = 1229389.614$ N-mm

$Rm = Dp/2 - b\times \frac{\sin(\gamma)}{2}$

$Rm = 72/2 - 10\times \frac{\sin(60.94)}{2}$

Rm = 36 – 8.17

Rm = 27.83 mm

Dm = 55.66 mm

3.1.7 Procedure Adapted For Analysis Of Bevel Gears

While designing bevel gears flowing flow is adopted

$Ptp = 2Mt_p/D$

Where,

Pt = Tangential load on the pinion (N)

Mt = torque transmitted by pinion N-mm

$Pt = 2\times \frac{1229389.614}{72}$

Pt = 87813.5 N

$A_0 = \sqrt{(Dp/2)^2 + (Dg/2)^2}$

$A_0 = \sqrt{(36/2)^2 + (72/2)^2}$

$A_0 = \sqrt{(18)^2 + (36)^2}$

$A_0 = 40.25$ mm

And , $1 - \frac{b}{A} = 0.5$, known as the bevel factor

The Sun bevel gear has a pitch line velocity of 1.5 m/s

The dynamic load acting on it is $P_d = 4994.6$ N

And the effective load is $P_{eff} = 14545.8$ N

From here we see that the $P_{eff} <$ BEAM STRENGTH

3.1.8 Shaft Diameter

Calculation of the shafts which being used in the transaxle. Shafts are designed on the basis of ASME code, to find the diameter of the shaft using the formula

$$d^3 = \frac{16}{\pi X\tau} \times \sqrt{(Kb \times Mb)^2 + (Kt \times Mt)^2}$$

We get d = 25 mm

3.1.9 Bearing Selection

Bearing selection is based on 90% reliability, for the 8 hours operation per day life = 16000 hrs

Bearing on the input shaft termed as bearing A and B

$$P = \frac{load}{number of bearing used for as\ \ haft}$$

Where P = equivalent load

$$Load = \sqrt{Tangentialload^2 + Axialload^2}$$

We get , $P_1 = 2.867$ kN

$P_2 = P_3 = 17.199$ kN

f_n= speed factor ,used instead of speed to determine the index of dynamic stressing $f_L$by the formula

$$f_n = \sqrt[p]{\frac{33\frac{1}{3}}{n}}$$

p = 3, for ball bearings

$p = \frac{10}{3}$, for roller bearings

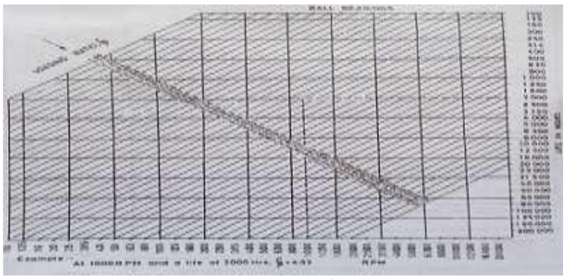
So we get $f_{n1} = 0.158$ for 7037 rpm

$f_{n2} = 0.24$ for 2345 rpm

$f_{n3} = 0.396$ for 533 rpm

Where,

F_L is called the index of dynamic stressing. It is convenient to express the value recommended for dimensioning not in hours. It is directly calculated from the graph of loading ratio for ball bearing.



Graph 1: Loading ratio for ball bearing

f_{L1} = 19.6 for input shaft
f_{L2} = 12.40 for the second shaft
f_{L3} = 8.43 for the bevel gears
From here we calculate the dynamic load C for the bearing, by using the formula

$$C = \left(\frac{fL}{\text{fn}}\right)P$$

We get C_1 = 36.291 kgf
C_2 = 45.33 kgf
C_3 = 37.36 kgf
By the value of C we select the bearing of SKF 6005, having
ID = 25 mm
OD = 47 mm
WIDTH = 12 mm and maximum permissible rpm of 16000.

3.2 Geometry modelling of transaxle

The modelling of the transaxle is done using geometric modelling software SOLIDWORKS 2015.

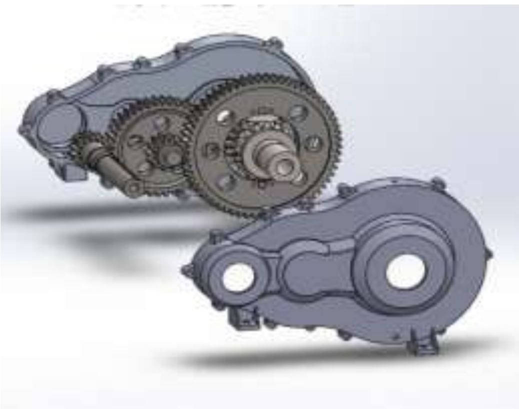


Fig 2: Exploded View Of Transaxle

3.2.1 Customization of Transaxle:

Steps involved in the customization of transaxle-

1. Determination of the vehicle torque requirement.
2. Calculation of the overall gear reduction.
3. Deciding the number of speed reduction stages, selection of gear types, its parameters and operations.
4. Calculation of the diameter of shafts and selection of bearing type.
5. Selection of appropriate fit and tolerance for mating parts like shafts, gears, bearings.
6. Designing of transaxle in SOLIDWORKS 2015 and analyzing in ANSYS 15.0

3.2.2 CAD Models

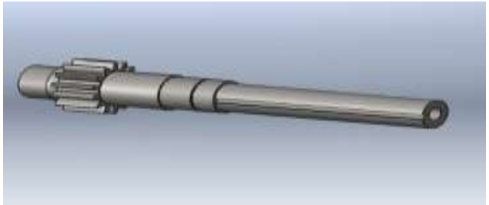


Fig 3: Input shaft with 14 toothed gear

Fig 4 :Ring Gear

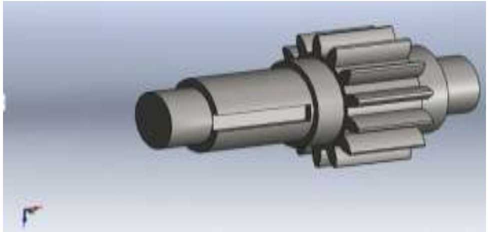


Fig 5: Intermediate Gear

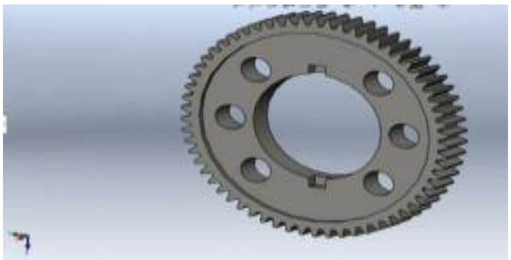


Fig 6: Differential Ring Gear

Fig: 7 Bevel Gears.

Fig 8 : Fabricated Gearbox Casing

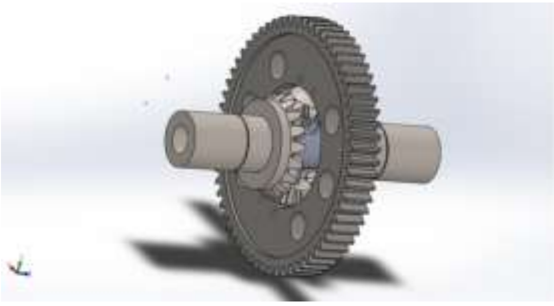


Fig 9: Cage less differential assembly

3.3 Analysis of gear

3.3.1 Static Structural analysis

A simple structural analysis was performed as the first step to see if components were structurally strong. If a component failed with the loadings, then no need to continue stress or fatigue analysis since the component is not strong enough to be used. The analysis of the various components of the differential was done in ANSYS 15.0 WORKBENCH for meshing as well as solving.

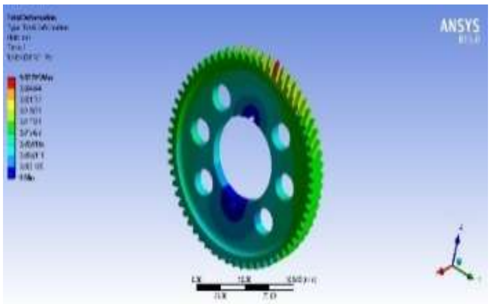


Fig 10: Def of differential gear

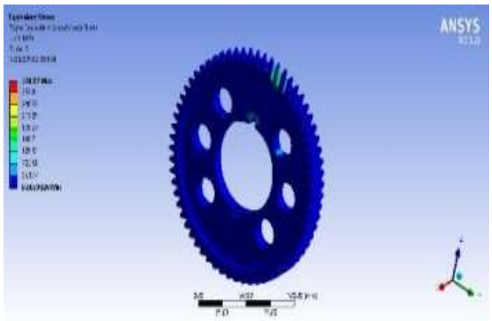


fig11: FOS of differential gear

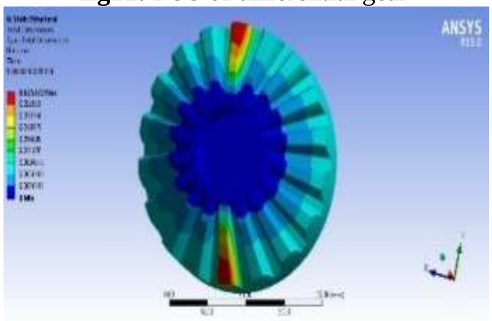


Fig12: Def of spider gear

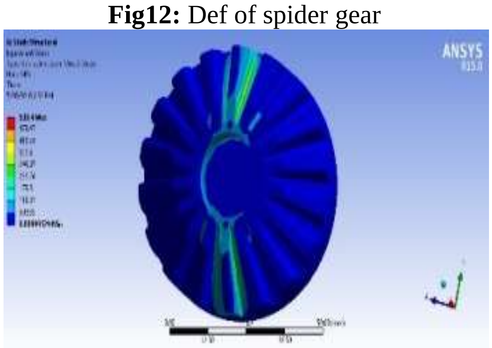


Fig 13:FOS of spider gear

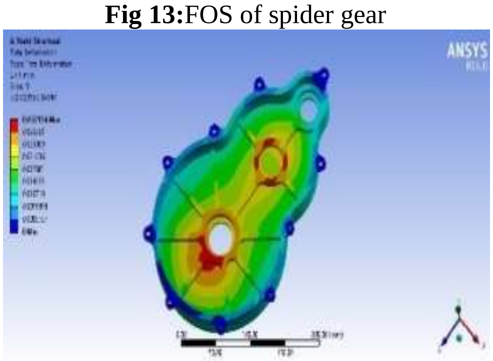


Fig 14: Def of transaxle casing

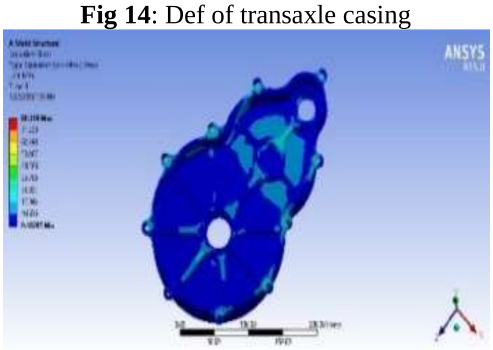


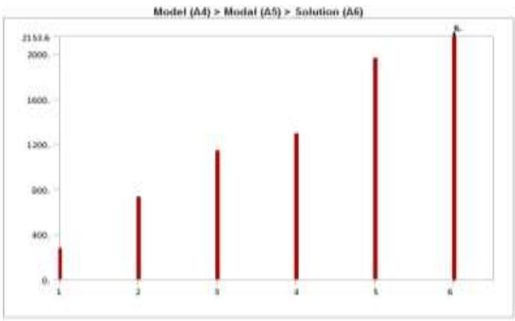
Fig 15:FOS of transaxle casing

S.no.	GEAR	DEF (mm)	F.O.S
1	14 teeth	0.013	2.73
2	42 teeth	0.017	4.17
3	14 teeth	0.026	1.60
4	Planet	0.056	2.93
5	Spider	0.031	1.52

Table2 :Gears And Casing Analysis Results.

3.3.2 Modal Analysis

Modal analysis or vibrational analysis has been done to ensure that the natural frequency of the transaxle casing does not match the frequency of vibration of the gears.



Graph 2: Vibration Modal analysis

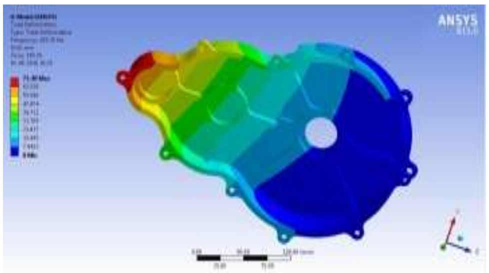


Fig 16: Casing results

S.no	Mode	Frequency (Hz)
1	1	280.35
2	2	730.38
3	3	1144.7
4	4	1291.7
5	5	1962.6
6	6	2153.6

Table 3: Vibration Modal analysis Result

IV. CONCLUSION

This paper has illustrated the entire design methodology of a transaxle. After customizing the above transaxle there is a commendable increase in the performance of the vehicle without any failure. Our goal is to design a lightweight cage less open differential with a gearbox it is much more compact and better in strength perspectives. The fatigue life of the components seems to be effectively high. This gearbox is easy to assemble and disassemble. This is used in the motorsport event BAJA SAE INDIA 2017 vehicle by the Team Benign Beaders.

Fig 17 : Practical implementation of transaxle in SAE BAJA 2017 ATV

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