DESIGNING OF THE RACK AND PINION GEARBOX FOR ALL TERRAIN VEHICLE FOR THE COMPETITION BAJA SAE INDIA AND ENDURO STUDENT INDIA

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Abstract

Preliminary aim is to design a Rack and Pinion Gearbox (RPG) which has desired steering ratio, zero play in the RPG and sensitive steering.The design of rack and pinion has been done by using CATIA software. Computer aided analysis (CAE) of RPG casing and clevis joint has been done so as to make sure they are safe. The newly designed RPG is compared withthe original equipment manufacturer (OEM) rack and pinion gearbox on the basis of cost, weight and the steeringgear ratio parameter. The results of the design shows that weight of new rack and pinion gearbox is 20% less than the OEM RPG with nearly 40% reduction in the cost achieving the desired steering ratio.

***Keywords:** -Rack and Pinion Gearbox (RPG)**,** CATIA, CAE, ATV, Baja, arm crossover.*

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1. INTRODUCTION

The function of the steering system is to provide directional control to the vehicle. For this a gearbox is used which converts rotational motion of steering wheel into translational motion of tie rod which in turn rotates the tires. For the steering system there are different types of gear box like Rack and Pinion, recirculating ball type, worm and sector etc. Among these, the rack and pinion gearbox consists of fewer moving parts and is easy to manufacture. As a result, most of the modern vehicle use rack and pinion gearbox. Therefore the objective of this paper is to design the Rack and pinion gearbox for an All-Terrain Vehicle (ATV) for the competition BAJASAE INDIA and ENDURO STUDENT INDIA. The BAJA SAE Series is an event for the undergraduate engineering students, organized globally by the Society of Automotive Engineers (SAE), USA (United States of America). The BAJA SAE tasks the students to design, fabricate and validate a single seated four-wheeled off road vehicle which is raced for 4 hours on the hard off-road terrain. The competition started in India in 2009 by the BAJA SAE INDIA. Enduro Student India is an exclusive student design competition which has been conceived and conducted exclusively by Delta Inc. The competition focuses on safety, good engineering practice, engineering design, education and creating opportunities for participating students.

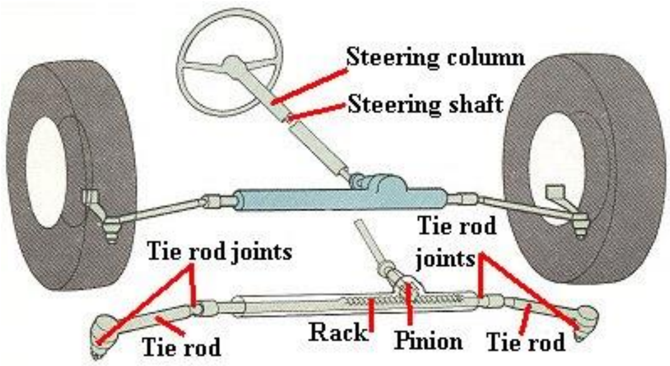

Fig- 1: Basic rack and pinion steering mechanism [1]

2. RACK AND PINION GEARBOX (RPG)

A rack and pinion is a type of linear actuator that comprises of a pair of gears which convert rotational motion into linear motion. A circular gear called "the pinion" engages teeth on a linear "gear" bar called "the rack"; rotational motion applied to the pinion causes the rack to move relative to the pinion, thereby translating the rotational motion of the pinion into linear motion. A Rack and Pinion Gearbox (RPG) has following components:-pinion gear with integrated shaft, rack, casing, bearing, and bronze bush.

2.1 Aim to Design (RPG) Gearbox

1. Integrated pinion shaft – Generally conventional manufacturing consists of pinion and shaft as a separate part and are coupled by means of keyways. This increases possibility of play in steering system as well as cost of manufacturing.

2. Desired steering ratio - Since the ATV Baja is generally designed with very compact space for driver. So driver's ergonomics must be considered. The original equipment manufacturer (OEM) RPG has standard steering ratio of 4:1(Around 270 deg lock to lock) at the steering wheel. However this results in arm crossover which can affect driver's performance in actual racing.

3. Reduction in weight – The OEM RPG weighed 1.5 kg (The custom RPG weighs 1.2 kg)

4. Reduction in cost of product – The OEM being imported from the US cost 12000 INR including custom charges.The cost of designed rack and pinion gearbox is 6000 INR

3. RACK AMD PINION DESIGN

3.1 Selection of Material

For the designing of any machine part, selection of material is the first step. Improper material selection can lead to its failure. Material selection process considers the factors such as strength, hardness, cost, and availability.Since the pinion is in rotational motion the tooth of the pinion gear is subjected to the fluctuating bending stress because of the tangential component (Pt) which arises due to rated torque (Mt).[2]The tangential force (Pt) causes the bending moment about the base of the tooth of pinion gearwhen it comes in contact withthe tooth of the rack gear. Since the teeth are subjected to the fluctuating stresses, endurance limit stress (Se) is the criterion of design. The endurance limit stress of gear tooth or bending strength is approximately one-third of the ultimate tensile strength of the material. [2]Another possibility of failure is wear. Wear is an erosion of material from the surface by the action of another surface. To avoid above failure modesand considering the above mentioned material selection factors the material for gear is chosen to be SAE 9310 with Yield strength 896 Mpa and Ultimate tensile strength 1500 Mpa.

3.2 Pinion Design

In the design of gears, it is required to decide the number of teeth on the pinion and gear. There is a limiting value to the minimum number of teeth on the pinion. While decreasing the number of teeth, a point is reached when there is interference. Interference is non-conjugate action and results in excessive wear, vibrations and jamming. To avoid interference, minimum number of teeth for 20° full depth systemrequired is 17 [2]. Using Lewis equation, the module is calculated. The module and minimum no of teeth were cross-checked with KISSsoft: softwaredeveloped by 'KISSsoft AG'for gear design. It predicted the chances of teeth interference. So, number of teeth on gear is increased to 21. And then again designed using Lewis equation and Buckingham equation and cross-verified with Kiss Soft. The mathematics of gear design is discussed below.

3.3 Inputs [2]

Maximum torque at the steering wheel (Mt) =60 N-m =60000 N-mm[4]
Material for rack and pinion= SAE 9310
Ultimate tensile strength of material (post hardening) (Sut) = 1500 Mpa
Bending strength=Sut/3= 500Mpa … [2]
Number of teeth on pinion (Z) = 21
Pressure angle=20 degree
Lewis form factor for pinion (Yp) = 0.544
Lewis form factor for rack (Yr) = 0.823

3.4 Calculations [2]

Factor of Safety*(fs)*

The recommended factor of safety is from 1.5 to 2. [2]

In preliminary design of pinion gear factor of safety of 1.5 is selected.

Estimation of the module based on the Beam Strength:-
In order to avoid failure of tooth due to bending Sb (bending strength) should be greater than Peff (effective load on the pinion)

I.e. Sb >Peff

Therefore Sb = Peff * *fs*

$$\text{Peff} = \frac{\text{Cs}}{\text{Cv}}\,\text{Pt}$$

Cs = 1.05

Cv = 0.9836 [2]

The tangential force onthe gear (Pt)

$$=\frac{2*\text{maximum torque}}{\text{no of teeth}*\text{module}}$$

$$=\frac{2*\text{Mt}}{\text{Z}*\text{m}}$$

$$=\frac{2*60000}{21*m}$$

$$=\frac{5714.28}{m}$$

Beam strength of gear (Sb)

= Module*face width*maximum bending load*Lewis form factor.

This equation is known as Lewis equation.

$$\text{Sb} = \text{m*b*}\frac{\text{Sut}}{3}\text{*Yp}$$

In the preliminary stages of gear design, the face Width is assumed as ten times of module.

Therefore b = 10 m.
fs = 1.5

Sut = 1500 Mpa
Yp = 0.544

Therefore

$$m = \frac{\frac{Cs}{Cv}\,\text{Pt} * fs}{b * \frac{Sut}{3} * \text{Yp}}$$

From this we getpreliminary module (m) = 1.4983

Modification in dimension of pinion gear

Since the forces acting on the pinion gear cannot be of exact values as it varies according to torque transmitting efficiency of steering system. If the torque transmitted from steering wheel to pinion gear is only 90% then the actual force on pinion gear will be less than the assumed force.

Therefore to decrease the module factor of safety is assumed to be 1.4 and face width value changes to 16 *m

New module

$$m = \frac{\frac{1.05}{0.9836} * \frac{5714.28}{m} * 1.4}{16 * m * 500 * 0.544}$$

$$m = 1.2519$$

Therefore (Dp) pitch circle diameter of pinon = m * Z
1.25 *21
= 26.25
b (face width) = 16 *m
= 20 mm

Check for the design

The effective load on the gear

$$= \frac{2 * \text{maximum torque}}{\text{no of teeth} * \text{module}} * \frac{\text{Cs}}{\text{Cv}}$$

$$= \frac{2 * 60000}{21 * 1.25} * 1.0675$$

$$= 4879.99 \text{ N}$$

$$\text{Sb} = m * b * \frac{Sut}{3} * \text{Yp}$$

$$= 1.25 * 20 * 500 * 0.544$$

$$= 6800 \text{ N}$$

$$fs = \frac{\text{Sb}}{\text{Peff}}$$

$$= \frac{6800}{4879.99}$$

$$= 1.3934$$

Hence design is satisfactory and module should be 1.25 mm

Procedure to calculate steering ratio: - lotus shark software gives values of inside angle (in degree), outside angle (in degree), turning radius and percentage Ackermann for a particular rack travel (in Millimeter). Since the designed rack and pinion gearbox has maximum rack travel of 24 mm for the lock angle of 95 degree of steering wheel. We noted the inside angle for 24 mm rack shift from the results of lotus shark software.

Lotus shark software: -the Lotus Suspension Analysis SHARK module is a suspension geometric and kinematic modelling tool, with a user- friendly interface which makes it easy to apply changes to proposed geometry and instantaneously assess their impact through graphical results. Lotus shark software gives instantaneous results of inside angle (in degree) for a particular rack shift.

Rack shift or rack travel per rotation is the horizontal distance travelled by the rack gear when the pinion gear completes one rotation.

Table -1: Results obtained from lotus shark software

Rack travel (mm)	Inside angle(degree)
24	40.38

Steering ratio calculation:-

$$\text{Steering ratio} = \frac{\text{Turn of a steering wheel in degree}}{\text{Turn of the wheel}}$$

For the lock angle 95 degree of steering wheel the maximum rack travel is 24 mm and inside angle in degrees is 40.38

Therefore

$$\text{Steering ratio} = \frac{95}{40.38}$$

$$= 2.3526$$

From the above results steering geometry selected has steering ratio of 2.3526 and lock angle of 95 degree of steering wheel which helps to eliminate arm crossover.

4. SHAFT DESIGN AND BEARING SELECTION

4.1 Shaft Design

The design of shaft is based on torsional and bending considerations so as to avoid failure in either. ASME Code is used for designing shaft.

Length of shaft = 44 mm
Mt = Torque on shaft = 60000 N-mm
The permissible shear stress for shaft material
τ = Max. Shear stress for steel = 0.18* Sut [2]
= 0.18 *1500=270N/mm^2
Mb = net bending moment of shaft
Mb = Bending moment due to radial load (Mb1) +
Bending moment due to tangential force (Mb2)

Mb1 =Radial load (Pr) * distance of the load from Centre of bearing
Radial load (Pr) =Pt tan (pressure angle)
= 4571.4 tan (20)
= 1663.85 N.
Therefore Mb1 =1663.84 * 22
=36604.7 N-mm

(Mb2) = Pt * distance of the load from Centre of bearing
= 4571.42* 22
= 100570.8 N-mm
Net bending moment (Mb) $=\sqrt{Mb1^{2}+Mb2^{2}}$
= 107025.18 N-mm

Therefore,

$$d^3=\frac{16*\sqrt{Mt^{\wedge}2+Mb^{\wedge}2}}{3.142*\tau}[2]$$

Where,
d= diameter of shaft

Therefore diameter of shaft (d) =13.22mm ≈ 15 mm

4.2 Bearing Selection

Forces on Bearing:
Radial loadon bearing
Total radial load on the pinion gear = Pt tan (pressure angle)
= 4571.4 tan (20)
= 1663.85 N.

Since there are two bearings, the effective load on each bearing will be half of the total load i.e. 831.92 N
Therefore radial load (Pr) = 831.92 N

Tangential load on each bearing (P)

$$=\frac{\text{total tangential load on pinion gear}}{2}$$

$$=\frac{4571.4}{2}$$

P = 2285.7 N

Considering bearing loads and Table 15.5 (Dimensions and static and dynamic load capacities of single-row deep groove ball bearings) from 'Design of Machine Elements' by Dr. V. B. Bhandari, SKF Bearing No. 6002 was used, the dimensions for bearing are as follows-
Inner diameter (d) = 15 mm
Outer diameter (D) = 32 mm
Axial width (B) = 9 mm

5. BRONZE BUSH

In order to achieve free movement of rack inside the casing, bronze bushes have been added on both ends. The bushes are fixed to the casing via grub screws to ensure positive locking.

6. CLEVIS JOINT

It is a joint used to connect tie rods with rack Clevis joint consists of a heim joint and C shaped cup which partially houses the heim joint. A bolt passes through the C shaped structure and the heim joint.

6.1 Design of Clevis

Clevis was designed for the following constraints:
Distance between two (which) faces = 23 mm
Hole on each face = 3/8" = 9.525 mm

A general clevis has an extended threaded surface to enter the rack. This creates a cantilever beam and subjected to bending failure. To avoid this, a countersunk bolt of grade 10.9 is used whose countersunk part enters the clevis thus eliminating cantilever beam.

6.2 CAE of clevis

Maximum Steering wheel torque = 60 Nm [4]
Pinion radius = 0.018238 m
Steering wheel radius = 0.15 m
Force on steering wheel = steering wheel torque/radius of steering wheel [3]
Pinion torque = steering wheel torque (considering 100%efficiency)
Force along pinion =pinion torque *pinion radius
Force along rack =force along pinion
Force along clevis =force along rack
Thus force on clevis=force along rack =3289.83N

6.2.1 Mesh parameters and load steps

While doing meshing main aim is to obtain maximum accuracy with less computational time. Ideally results obtained from a finite element analysis get more accurate with increase number of element but it is also important to find balance between the accuracy of results and process time required to run the analysis.

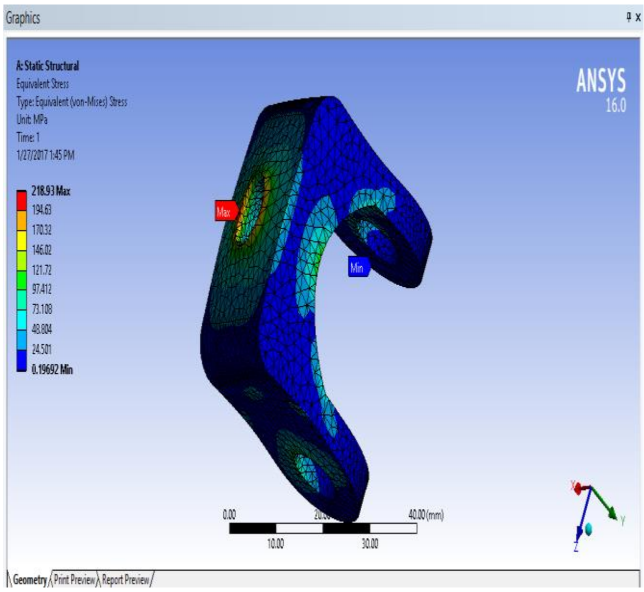

Fig-2: Equivalent stress of clevis = 218.93 Mpa

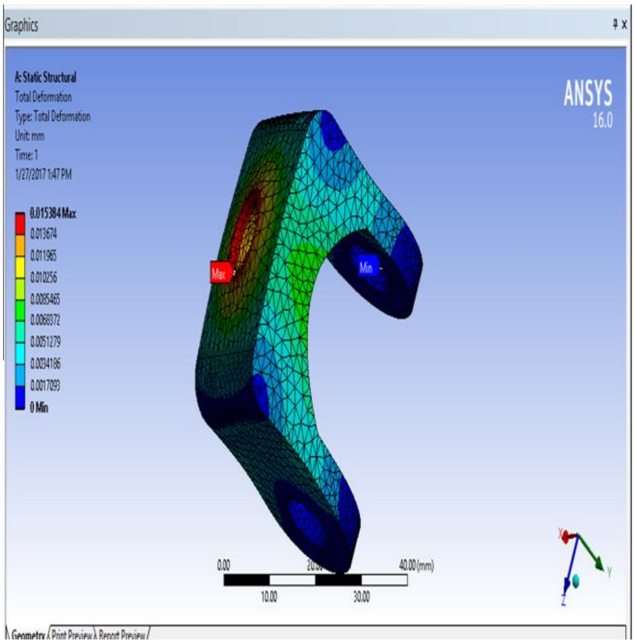

Fig-3: Total deformation of clevis = 0.015384 mm

Yield strength of Aluminum 6062–T6 alloy is 280 Mpa. Equivalent (von-mises) stress generated in clevis is 218 Mpa. As equivalent stress generated is less than yield strength the design is safe.

7. CASING DESIGN

7.1 Description

Casing is important component in rack and pinion gear box hence we must know that how much stress is generated inside casing. In casing, stress isgenerated due to compressive or radial forces. Forces on casing are same as that of forces on bearing.

7.2 Material

Failure of casing is due to the compressive forces which results in deformation of casing. Hence elastic modulus is the selection criteria for material. Hence aluminum 6082–T6 material is selected with elastic modulus of 69 GPa.
Fx = total radial force on bearing = 831.92 N
Fz = total tangential force on bearing = 2285.7 N

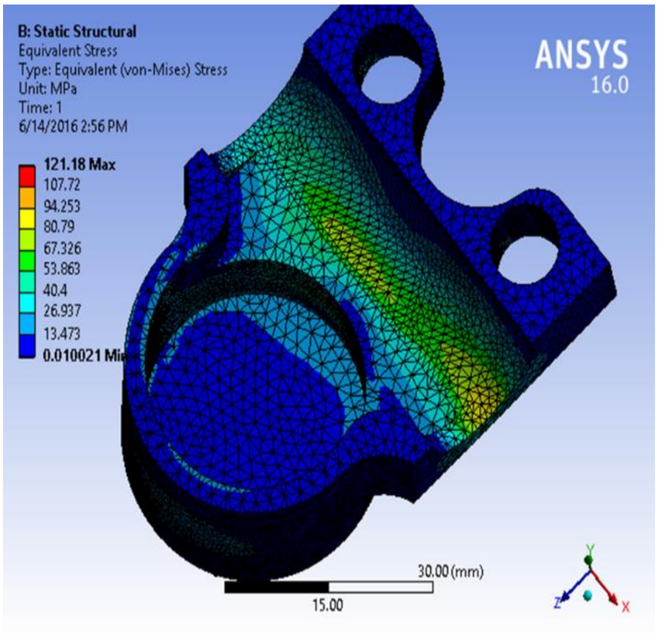

Fig-4: Equivalent stress of casing = 121.18 Mpa

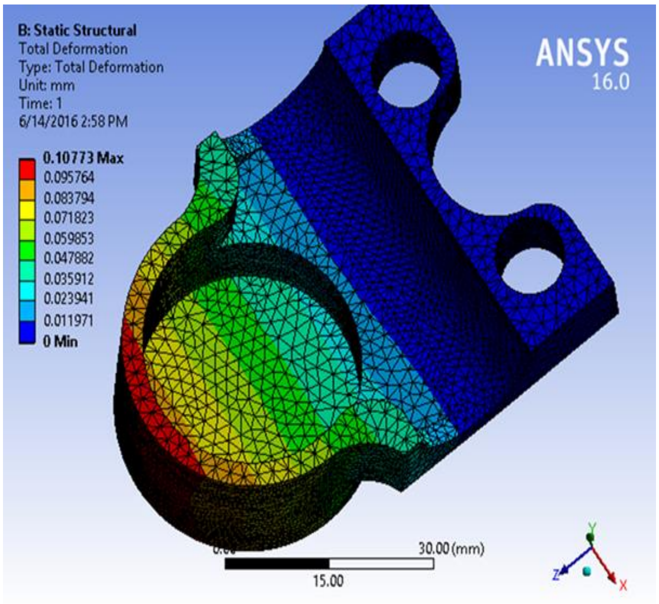

Fig- 5: Total deformation of casing = 0.10773 mm

Yield strength of Aluminum 6062–T6 alloy is 280 Mpa. Equivalent (von-mises) stress generated in casing is 121.18 Mpa. As equivalent stress generated is less than yield strength the design is safe.

8. KEY FEATURES OF DESIGNED RACK AND PINION GEARBOX

1. Integrated pinion shaft –Manufacturing of both pinion and shaft as one part. Therefore cost of two different manufacturing parts has been reduced and design becomes simple and full proof.

2. Weight reduction – we designed our casing with minimum thickness and modification in design so that weight is reduced but strength remains the same.

3. Sensitive steering – designed steering gear ratio (2.35:1) helps to achieve sharper turn with less turn of steering wheel and eliminates arm crossover

Table- 2: Comparison of designed RPG with OEM RPG

Parameter	weight	cost	Steering gear ratio	Lock to lock angle
OEM RPG 11''	1.5 kg	12000 INR (including custom charges)	4:1	270 degree
Custom made RPG	1.2 kg	6000 INR	2.37:1	190 degree

9. RESULT AND CONCLUSION

Design of custom made RPG gearbox has done with 20% reduction in weight and in less cost than OEM.

By using designed RPG gearbox steering geometry selected should produce lock to lock of 48 mm rack shift and thus desired steering angle.

All the necessary analysis of component hasbeen done to ensure that design is safe.

To reduce equivalent stresses edge fillet should be given where stress concentration is more which reduces weight with increase in strength.

From the structural analysis results we can conclude that to avoid failure of material Deformation should be as small as possible.

Thus RPG gearbox is designed with reduction in cost as well as Weight and desired steering ratio can be used for ATV Baja

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BIOGRAPHIES

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