Design and Analysis of Gearbox with Integrated CV Joint

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Abstract— This is a detailed report about the gearbox and CV joint of the ATV designed for SAE BAJA India. The powertrain is a driving force of an ATV whose function is to transfer required torque and power generated by engine to the driving wheels. To do so engine is coupled with CVT which is further coupled with two stage gearbox. CVT has a variable driving ratio which is multiplied with two stage gearbox with constant ratio to satisfy the torque and power requirements according to variable driving conditions and loads. Designing of two stage gearbox is vital task to minimize its rotational mass and overall weight.it is advantageous to integrate CV joints in output shaft of the gearbox which also decreases inclination angle of half shaft and give an extension in overall performance of an ATV.
Keywords: gearbox, CVT, CV-joints, rotational loads, weight

I. Introduction

Powertrain contains most critical components of an ATV which encompassed with engine, CVT, gearbox, driveshaft and universal joint Out of which engine is provided by Briggs & Stratton a 305 cc power unit which produces 19.6NM of torque. This torque is transferred with the help of CVT to gearbox and finally to drive shaft via universal joint. The main objective of integrating the constant velocity joints with the output shaft was to increase the Compactness and reduce the overall mass of the transmission. CVJ are important to transfer velocity but OEM joints have loss in power transmission. Thus, we designed a gearbox with integrated CVJ. With Improved efficiency of gearbox, result in better acceleration and gradeability of the vehicle. Designing of gearbox is done with the aid of CAD and CAE software along with hand calculations.

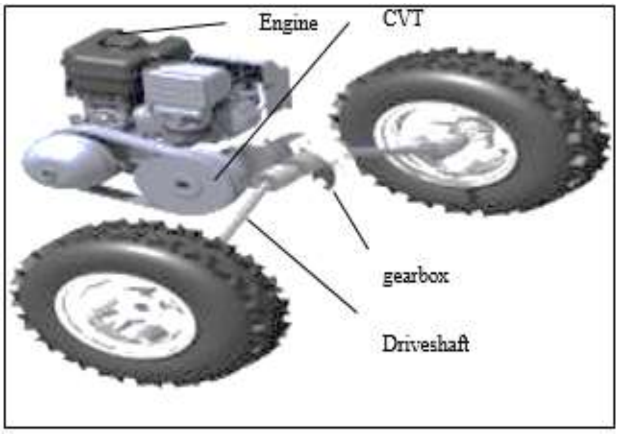

Fig. 1: powertrain system of an ATV

A. *Engine specification,*

Briggs and Stratton single cylinder engine is used which is stock engine for SAE BAJA India competition.

Engine Displacement	305cc.
Torque	19.6NM
No of Cylinders	Single
Power	10HP
Weight	28kg
RPM	3800

Table 1: Engine specification.

B. *CVT Specification*

CVTech CVT is used.

Over drive ratio	0.43.
Under drive ratio	3.1
Center to center distance	214mm
Weight	10kg.
Engagement RPM	1800

Table 2: CVT specification.

II. Design Targets

- Reduction in final weight and cost.
- Increase acceleration of vehicle.
- Maximization of efficiency of powertrain.
- Minimize vibration of system.
- Lowering the rotational mass.
- Low maintenance and serviceability.
- Speed within 60km/hr.

A. *Design consideration*

- Weight of vehicle -253kg.
- Tyre Size = 23 ″-7 ″-10 ″.
- Coefficient of friction between road and tyre=0.65
- Wheel base =50 ″
- C.G height = 17 ″
- Weight distribution =60% at rear and 40% at front.

III. Vehicle Dynamics Calculations

Required tractive force = $F_{xmax} = \dfrac{\mu \frac{W b}{L}}{1 - \frac{h}{L}\mu}$

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=1242.5N.

- Required torque = tractive force× tyre radius = 362.93N-M.
- Velocity $= \dfrac{\pi \times \text{diameter of tyre} \times \text{engine rpm}}{60 \times \text{reduction}}$

=59km/hr.

- Developed tractive force $= \dfrac{T \times \eta \times \text{reduction}}{\text{Rolling radius}}$

=2013.2N

- Aerodynamic drag = 0.5×Cd×A×ρ×V2 =122.33
- Rolling resistance = μmg =248.19
- Developed tractive including looses =1642.73
- Acceleration $= \dfrac{\text{developed tractive force}}{\text{mass of the vehicle}}$

$=6.4 m/s^2$

- Grade angle = mg×sinθ

Θ = 41.29

– Gradeability = 87.8%
μ= coefficient of friction
b= distance of c.g from front axel
h= height of c.g
L= wheel base
T= engine torque
η= transmission efficiency
C_d= Coefficient of Drag.
ρ= Density of air = 1.225kg/m^3

IV. DESIGN CALCULATIONS

A. Material selection

The first step in gearbox design is the selection of the material. A material must be selected by intensively researching the properties of the different materials. A material must be selected taking into account the various parameters such as strength, weight, durability, cost and other parameters in order to construct the gear. 20MnCr5 case hardening steel is selected as the gear material due to its better mechanical properties.

Material	20MnCr5
Yield strength	950MPa
Ultimate tensile strength	1200MPa
Hardness	60 HRC(case hardened)

Table 3: Gear material specification.

B. Gearbox Design Calculation

– Number of teeth in each stage should be odd to increase wear cycle.
– For two stage gear box velocity ratio for single stage should not exceed 6:1.
– To determine the speed reduction at each stage the ratio of angular velocity of last driven gear to the first driving gea r is given by taking root of overall gear ratio.

$$i = \sqrt{i}$$

– For optimum range face width is taken 10 time of module.

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.

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

M=2.5mm

C. First stage reduction

Pinion speed = 900RPM
σ_{bp}=S_{ut}/3 = 1200/3
= 400N/mm^2
Gear Ratio (i_1) = 3.3
σ_{bg}= S_{ut} /3 = 1100/3
=400N/mm^2
Lewis Form Factor (Y) = 0.484-(3.36/Z)
=0.484-(3.36/19)
=0.306
Yg=0.484-(3.36/Zg)
=0.484-(3.36/64)
=0.4315
σ bp × Yp < σ bg × Yg
As Pinion is weaker than gear it is necessary to design pinion.
Diameter of pinion (dp) = m × Zp
= 2.5×19
Face width (b) = 20mm
Ratio Factor (Q) = 2 × Zg / (Zg + Zp)
= 2 × 64/ (64+19)
= 1.542
Load Stress Factor (K) = 0.16 × $(BHN/100)^2$
= 0.16 × (650/100)2
= 6.76 N/mm^2
Beam Strength: Fb= σ_{bp}× b × m× Y_p
= 400×20×2.5×0.306
= 6120N
Wear Strength (Fw) = dp × b × Q × K
= 47.5×20×1.5×6.76
= 9633N

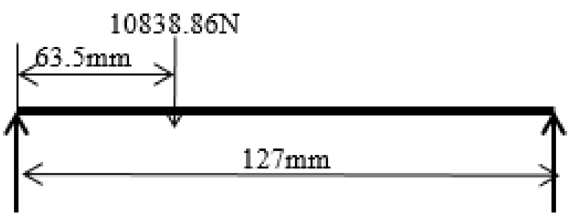
As, Fb < Fw, Gear Pair is weak in bending hence it is designed for safety against bending.

Torque at pinion = T_1=64.68N-M
Pitch line velocity at first stage pinion (V1)
V_1= π × dp × Np / 60 ×1000
= π × 47.5/60×1000
= 2.23m/sec
Tangential Force at tooth.
(Ft)= σ× C_v× b× d_p× y_p =15116.4N.
Velocity Factor (Kv) = 6/(6+V_1)
= 6/(6+2.23)
= 0.722
Service Factor (K_m) = 1.3
F_{eff} = {K_m × F_t} / K_v
= {1.3 × 15116.4} / (0.722)
= 27217.89
Diameter of pinion in first stage (dp1) = m × Zp
= 2.5×19
dp1= 47.5 mm
Diameter of Gear of first stage (dg1) = m × Zg
= 2.5 × 64
dg1 = 160 mm

D. Shaft calculations.

Shaft design on the basis of strength is to ensure that the stress at any location of the shaft does not exceed the material yield stress.
Shaft subjected to bending and torsional.

1) Input shaft



$$P = \frac{F_t \times v}{Cs} \quad (Cs=1.54)$$

P = 22.8KW

$$\text{Torque (Nm)} = \frac{60 \times \text{Power (kW)}}{2\pi \times \text{Speed (Rpm)}}$$

T = 241.9N-M
Tangential force = 2T/D
= 10185.2N
Normal force(F_n) = F_t /cosα
=10838.86
Bending moment M=W×L/4
=344.13N-M
Twisting moment $T_e=\sqrt{M^2+T^2}$
=420.6N-M
Shear strength of the material
= 0.577*1200 = 692.4 MPa
$T=\pi/16\times\tau\times d^3$
d=17mm.
for input shaft d=20mm.
for intermediate shaft d=25mm.

2) Output hollow sahft

Internal diameter = 56mm.
External diameter = 60mm.

$$\text{Torque (Nm)} = \frac{60 \times \text{Power (kW)}}{2\pi \times \text{Speed (Rpm)}}$$

T= 967.6N-M
Shear stress

$$\tau_{max.} = \frac{16}{\pi d_o^3 (1-C^4)} \sqrt{\left[(M_b)^2 + (M_t)^2\right]}$$

=313.5 N/M^2

E. 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.

1) 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.
L10h = 150 hrs
Radial load, F_r = 3707.1N
Life of bearing = 10000 hours

$$L_{10} = \frac{60 \times n \times L10h}{10^6}$$

= 13.8 m_{rev}
Dynamic load capacity
$C = P(L10)^{\frac{1}{3}}$
C = 13771.26N
According to dynamic load bearing no 3006 is suitable.

Shaft	Diameter	Bearing no.
Input shaft	20mm	3006
Intermediate shaft	25mm	2205
Output shaft	60mm	61812

Table 4: dimension of shaft and bearing numbers

Fig. 2: Cad model of gear train.

F. FEA of Gears

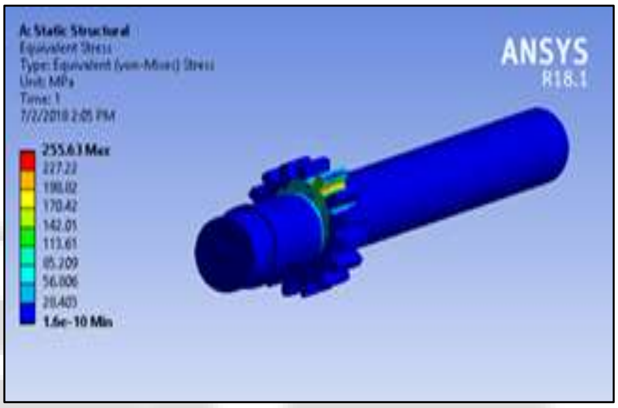


Fig. 3: analysis of input shaft.

Stress = 255.63Mpa
Deformation = 0.012mm
FOS= 2.15

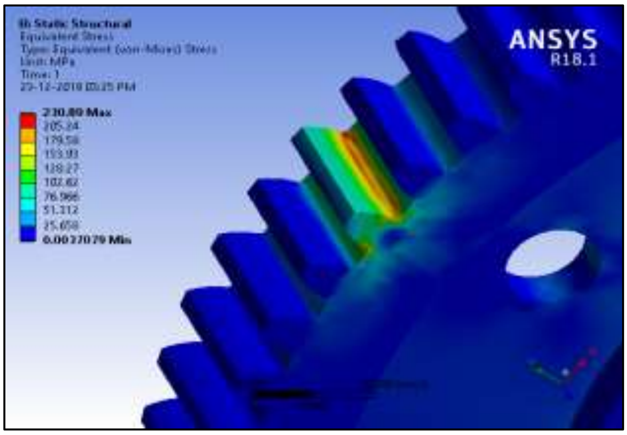


Fig. 4: FEA of output gear.

Stress=230Mpa
FOS=2.5

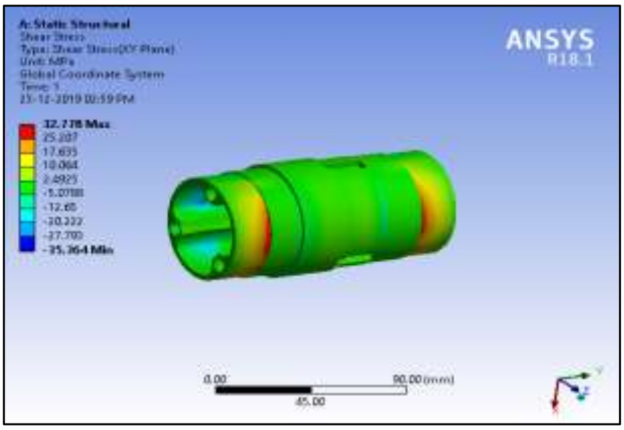


Fig. 5: FEA of output shaft.

stress = 340Mpa
FOS = 3.5


Fig. 6: manufactured gearbox.

V. CONCLUSION

This paper illustrates the comprehensive design methodology for a gearbox with integrated CV joints. gear were designed by using solidworks and analysis was carried out by Ansys. An appropriate reduction was chosen for drive adaptability. system shows no fatigue and run over more than 500km.

Achieved targets

- Efficient power and torque transferred.
- Increase in vehicle acceleration.
- Reduced jerk in power transmission.
- Serviceability and maintainability is increased.
- Reduced overall weight.

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