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Design and Analysis of Gearbox of an All-Terrain Vehicle

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Abstract - *This is a detailed report about the powertrain system of the ATV vehicle designed for SAE BAJA India. Powertrain system designed with the objective to utilize the power available at the engine output shaft by reducing transmission losses comprises of main components such as engine, gearbox, CVT, half shaft, joints, etc. CVT is tuned to its ultimate efficient output by confirming its changing values, i.e. from 3.9 to 0.9. Gears are designed in a way to provide an aggressive launch with appropriate output torque while the optimized maximum speed could be obtained within time. Gearbox is optimized in its dimensional view reducing its weight and space required to mount it. Joint is selected to accommodate the extreme value of travel for the vehicle with considering its transmission efficiency according to varying articulation angle. Powertrain system is designed and manufactured to attain vehicle's top acceleration as well as drawbar pull capacity and gradeability.*

Key Words: **Efficiency, Power, Gearbox, CVT, Torque, Acceleration, Speed.**

1. INTRODUCTION.

The powertrain system includes engine as a power producing device. CVT assists in torque multiplication to produce the required output torque. Gearbox is Customized as per the ATV requirement as a single Speed 2 stage reduction system. The joint used is OEM with some tolerable travel and allowing varying articulation. The transmission system multiplies the Torque of the engine such that power peak is achieved to the maximum potential. The design of Components is done considering the losses of every Component both individually and as a part of the System wherever it was necessary. All the resistances were mathematically formulated in order to set the design requirements. Material for every component was selected giving preference to its design as well as performance requirement.

1.1 Engine Specifications.

We Used Briggs and Stratton engine which is 4 stroke air cooled petrol engine.

Torque	**18.3 Nm.**
Engine Displacement	**305 cc.**
No of Cylinders	**Single.**
Engine Configuration	**Horizontal Shaft.**
Engine Technology	**OHV.**
Mass (Kg)	**28 kg.**
Bore (in)	**3.12.**
Stroke (in)	**2.44.**
Oil Capacity (dry)	**24 ounces.**
Rpm	**3800.**
Compression ratio	**8.1 to 1.**
Power (HP)	**10 hp.**
Fuel Type	**87 Octane.**

1.2 CVT Specifications.

We used Gaged CVT gx9 Model of center to center distance 8.5 inches with high tunability options.

- Low ratio = 3.9.
- High Ratio = 0.9.
- Weight = 5kg.
- Cost = 100K.

2. Design Targets.

- To obtain an output speed of 50 Km on road.
- To reduce the overall weight and cost of the System.
- To avoid losses to the maximum possible extent and increase the efficiency of the transmission system.
- To achieve maximum acceleration.
- To minimize the vibrations.
- To reduce the maintenance, increasing serviceability.
- To design components considering all parameters so they don't undergo any failure after manufacturing.

2.1 Design Consideration.

- Coefficient of friction between tyre and road = 0.31.
- Coefficient of Rolling Resistance = 0.31.
- CVT Ratio – 1. Low ratio – 3.9.
  2. High ratio – 0.9.
- Vehicle Mass = 220 Kg.

- Tyre Size = 23”-7”-10”.

3. Selection of Gear-Ratio.

Rolling Resistance = µmg
= 0.3*9.81*200
= 588.6 N.

Air Drag = 1/2*Cd*A*ρ*V^2
= 1/2* 0.23*0.838*1.225*229.129
= 25.05N.

Gradient Resistance = mg*sinΘ
= 200*9.81*sin (35)
= 1125.356 N.

Total Resistance = Rolling Resistance + Air drag + Gradient Resistance
= 588.6 + 25.05 +1125.356
= 1739.006 N.

Radius = 0.2921 m (23” x 7” x 10” in)

Torque = Total Resistance * Radius
= 1739.006 * 0.2921
= 507 Nm.

Powertrain Efficiency to be considered as 96%
507/0.96 = 526.5 Nm.

The above obtained value is for maximum resistance on loose sand. Hence, the torque obtained from the above value will be the maximum torque. But in practical that value is never achieved. Hence, we choose a value (for torque) that nearest to the above calculated value.

Selected Gear Ratio = 526.5 / (18.3*3.9)
= 7.36.

Speed = [RPM (engine)* 2π * Radius]/ [60*CVT ratio* Gear ratio]
= [3600 * 6.28 * 0.292] /[60*0.9*7.36]
= 16.6 m/s
= 60 km/hr.

Total Tractive Force = T*G*r /R
Where T = Engine Torque
G = Gear Ratio
r = CVT low ratio.
R = Radius
Total Tractive Force = 18.3*7.36*3.9 /0.2921
= 1799 N.

Acceleration = (Tractive Force-Resistances) / m
= (1799 –613.65) / 190
= 6.23 m/s.

Gradeability =
M*a = Tractive Force – (Rolling Resistance + Air Resistance) – Gradient Resistance

190*0=1799–(588.6+25.05) – 190*9.81*sinΘ
Θ =39.419.

Gradeability % = tanΘ*100
= tan (39.41) * 100
= 82.17%.

4. Gear Calculations

For First Pair:

Gear Geometry Data	**Data**
Gear Mesh type	External
Helix type	Single Helix Type
Normal Pressure angle PHI(n)	20
Standard helix Angle beta	15
Required gear ratio u	2.411

Note: Dimensions are in mm, all angles in degrees, and all stresses in N/mm2.

Materials / Heat Treatment Data	**For Both Pinion and Gear.**
Material (Pinion)	EN-24
Material (Gear)	EN-24
Heat Treatment	Toughened
Surface Hardness	HRC 55

Load Data	**Data**
Transmitted Power (kw)	6.19
Pinion Speed (rpm)	821

Required Life (hrs)	1000
Overload factor (Ko)	1.75
Dynamic Factor (Kv)	1.0858
Load Distribution factor (Km)	1.2
Pitting safety factor (SH)	1.5
Bending Fatigue Safety Factor (SF)	2.5
Reliability	99%
Driving	PINION
Number of contacts per revolution	1

Stress cycle factors, curve chosen, figs. 17 & 18 = Lower (For Critical application).

*****DESIGN OPTIONS*****

Operating Centre Distance: By Input.

Operating Centre Distance (mm) a = 60.

Last Precise Solution	Input	Output
Pinion Operating Pitch Dia (mm)	34.622	34.622
Face Width (mm)	15	15
Normal Module (mm)	2	2

Selection of Variants:-

Z1	Z2	Ratio	Ratio	Wt.	Wn.	w
16	36	2.25	3.211	25.897	25.064	15.575
17	36	2.118	-2.86	23.525	22.779	15.295
17	41	2.411	-0.162	20.904	20.248	15.024

Tooth number Combination chosen (Z1, Z2):- 17, 41.

Geometry Summary	Pinion	Gear
Tooth Number (Z1,Z2)	17	41
Net Face Width (b1,b2)	15	15
Outside Diameter (do, Do)	39.199	88.800
Normal Module (mm)	2	
Normal Pressure Angle (Φ)	20	
Standard Helix Angle (β)	15	
Operating Centre Distance (a)	60	

FOR SECOND PAIR:

Gear Geometry Data	Data
Gear mesh type	External
Helix type	Single helical gear
Normal pressure angle, (φ)	20°
Standard helix angle, (β)	15°
Required gear ratio, (μ)	3.06

Materials / Heat Treatment Data	For Both Pinion and Gear.
Material (Pinion)	EN-24
Material (Gear)	EN-24
Heat Treatment	Toughened
Surface Hardness	HRC 55

***** DESIGN OPTIONS *****

Operating centre distance: By Input

Operating centre distance (mm) a= 100

Load Data	Data
Transmitted power (kW)	6.19
Pinion speed (rpm)	279.14
Required life (HRS)	1000
Overload (or application) factor	1.75
Dynamic factor	1.0309
Load distribution factor	1.2
Pitting safety factor	1.5
Bending fatigue safety factor	2.5
Reliability	99%
Driving:	PINION
Number of contacts per revolution	1

Last Precise Solution	Input	Output
Pinion operating pitch dia. (mm)	49.2611	49.2611
Face width (mm)	15	15
Normal module (mm)	2.5	2.5

Selection of Variants:-

Z1	Z2	Ratio	Ratio	Wt.	Wn	w
18	56	3.111	1.67	26.347	25.498	15.632
18	57	3.167	3.486	24.738	23.948	15.433
19	56	2.947	-3.681	24.738	23.948	15.433
19	57	3	-1.961	23.024	22.295	15.24
19	58	3.053	-0.241	21.18	20.515	15.051

Tooth number Combination chosen (Z1, Z2):- 19, 58.

GEOMETRY SUMMARY	Pinion	Gear
Tooth number	19	58
Net face width	15	15
Outside diameter	54.166	155.8242
Normal module (mm)	2.5	
Normal pressure angle	20°	
Standard helix angle	15°	
Operating centre distance	100	

5. Shaft Calculations:-

- **Input Shaft**

Weight of pinion: 1.4 N

Vertical force analysis

$\sum MA = 0$

(3663.81×14) - (RB×45) =0

RBV=1139.85N

$\sum Fy = 0$

RA+RB=3663.81 N

RAV=2523.96 N

Bending moment acting in vertical plane

MAV= 0 Nmm
MCV = 35335.44 Nmm
MBV = 0 Nmm

Horizontal force analysis

$\sum MB = 0$

(1216.61×197) - (RAH×45)
19303.19 + (1380.0314×31) =0

RAH=5847.78 N

$\sum Fy = 0$

1216.61-5847.78+1380.0314-RBH = 0

RBH = -3251.14 N

Bending moment acting in horizontal plane

MDH = 0 Nmm
MAH = 184924.22 Nmm
MCL= 120088.34 Nmm
MCR = 100785.15 Nmm
MB = 0 Nmm

Maximum bending moment

Resultant B.M at D

= 0 Nmm

Resultant B.M. at A

= √184294.722

= 184924.72 Nmm

Resultant B.M. at C

= √120088.342 + 35335.432

= 125179.0826 Nmm

Resultant B.M. at B

= 0 Nmm

Maximum Bending moment from above

Mmax = 184924.72 Nmm

Then the permissible shear stress is,

$\tau per = \tau$ 1.64 = 142.39 N/mm2

From Combined bending moment and torsion equation

D = 22 mm

- **Intermediate Shaft**

Pinion and Gear mounted on intermediate shaft have a weight of 2.19 N and 11.33N respectively.

Vertical force analysis

$\sum MA$ = 0

(-3663.03×16.5) - (8328.57×33.5) + (RBV×50) =0

RBV=6788.94N

$\sum Fy$ =0

RAV+RBV=11991.6 N

RAV=5202.66 N

Bending moment acting in vertical plane

MAV= 0 Nmm
MCV =-85706.57 Nmm
MDV =-111931.5 Nmm
MBV = 0 Nmm

Horizontal force analysis

$\sum MB$ =0

(1380.26×16.5) + (54866.179) -(3138.64×33.5) (RBH×50) =0

RBH=-550.079N

$\sum Fy$ =0

RAH+RBH= -1758.38N

RAH = 2308.459 N

Bending moment acting in horizontal plane

MAH = 0 Nmm
MCL= -38041.32 Nmm
MCR = 16829.07 Nmm
MDL = -45829.11Nmm
MDR = 9034.58 Nmm
MBH = 0Nmm

Maximum bending moment

Resultant B.M at A

= 0 Nmm

Resultant B.M. at C

= 93769.71 Nmm

Resultant B.M. at D

= 120950.34 Nmm

Resultant B.M. at B

= 0 Nmm

Maximum Bending moment from above

Mmax = 120950.34 Nmm

Then the permissible shear stress is,

$\tau per = \tau$ 2.26 = 103.3743 N/mm2

From Combined bending moment and torsion equation

D = 26 mm.

- **Last Shaft**

Gear is mounted on last shaft which have a weight of 19.01212 N.

Vertical force analysis

$\sum MA$ =0

(8348.69289 ×31) - (RB×45) =0

RBV=5751.3217N

$\sum Fy$ =0

RA+RB= 8348.69289 N

RAV=2597.3711 N

Bending moment acting in vertical plane

MAV= 0 Nmm

MCV = 80518.5047 Nmm

MBV = 0 Nmm

Horizontal force analysis

$\sum MB$ =0

(2231.93×69.88) + (3138.7046×31) - (RBH×45)

=0

RBH=5628.158 N

$\sum Fy$ =0

RAH + RBH = 0

RBH = -2489.4534 N

Bending moment acting in horizontal plane

MAH = 0 Nmm

MCR = 78794.212 Nmm

MCL= -77173.056 Nmm

MBH = 0 Nmm

Maximum bending moment

Resultant B.M. at C

=√ (78794.212)2 + (80518.5047)2

= 112657.7003 Nmm

Maximum Bending moment from above

Mmax = 112657.7003 Nmm

Then the permissible shear stress is,
$\tau per = \tau$ 1.7 = 137.64 N/mm2

From Combined bending moment and torsion equation

D = 32mm

- **Bearing Calculations:-**

Standard factors:

L10= 1000 hrs.

a = 3

61904-2RS1

Loads acting due to the gear

Pa = 981.3294 N

Pr = 1380.017 N

Pt = 3662.3113 N

N = 820.51 rpm

Rated life:

L10 = 49.2306 million revolutions

Effective Dynamic Load P = 1745 KN.

Dynamic Load Capacity C = 8.39 KN.

62/22-2RS1

Loads acting due to the gear

Pa = 2437.8684 N Pr = 3428.309 N

Pt = 9098.226 N

N = 288.812 rpm Rated life:

L10 = 17.3288 million revolutions

Effective Dynamic Load P = 4358 KN.

Dynamic Load Capacity C = 6.5658 KN.

61906-2RS1

Loads acting due to the gear

Pa = 2231.635 N Pr = 3138.2886 N Pt = 8328.5768

N = 101.6944 rpm

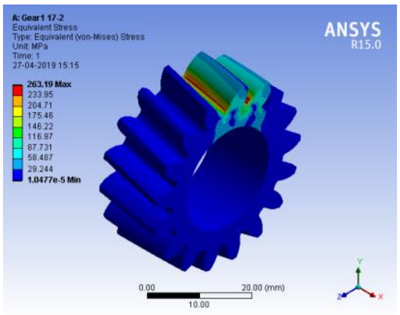
Rated life:

L10 = 6.1017 million revolutions

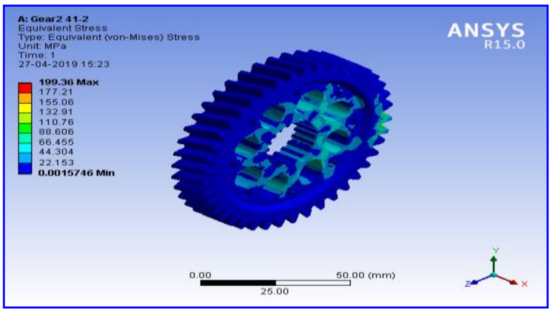
Effective Dynamic Load P = 3989 KN

Dynamic Load Capacity C = 4.591 KN.

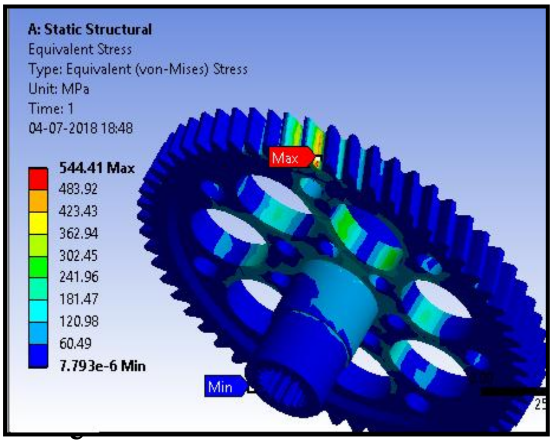
• **Analysis of Components.**



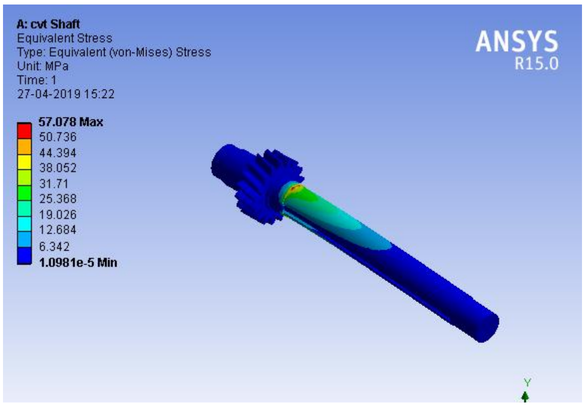
Gear 1



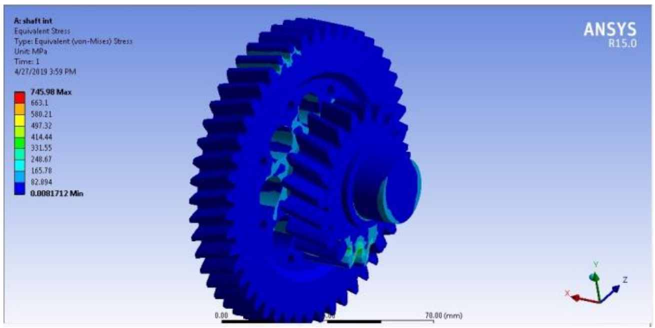
Gear 2.



Gear 3 with output shaft.



Gear 1 with input shaft.



Gear 3 and Intermediate Shaft.

• **Results.**

1. Stress generated in gear 1 is 263.19 MPA and FOS is 3.22.
2. Stress generated in Gear 2 is 199.36 and FOS is 3.51.

3. Stress Generated in 3rd Gear is 454.1 MPA and FOS is 2.5.

4. Stress generated in input shaft is 281.2MPA and FOS is 3.

5. Stress generated in intermediate shaft is 457.3MPA and FOS is 1.853.

For above results material's YTS is 848 MPA.

Conclusion.

The agenda for designing gearbox was to increase the efficiency and torque of an ATV. The Gears were designed by using Gearcalc software and cad with the help of CatiaV5 and analysis was carried out by the software 'Ansys'. Considering the efficiency of gears helical gears were selected. The speed and torque was selected optimum according to the event and thus gearbox of 2 stage single speed was designed and analyzed successfully.

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