Design, Development And Performance Analysis of Two Stage Reduction Gear Box For An ATV

Rishikesh Shelke[1], Shubham Alladwar[2], Hemang Pancholi[3], Dilip Choudhari[4]
[1, 2, 3] Dept of Mechanical Engineering
[4] Asst.Professor, Dept of Mechanical Engineering
[1, 2, 3, 4] Dr. D. Y. Patil Institute of Technology Pimpri, Pune, India

Abstract*- Transmission is the most important unit in ATV which allows it to propel through the odds of the terrain. The main purpose of this project is to design and develop a two stage reduction gear box for an ATV which will provide a sufficient torque conversion and speed reduction with enhanced performance. Gearbox is a set of gears and gear train which is use to transmit power from a rotating power source to another device. It allows torque conversion and speed variation. Spur gears are designed according to AGMA standards while the shafts are design as per the ASME standards. Bearing selection was done according to its load carrying capacity and life. Modelling was carried out in CATIA V5R21 and analysis was done in ANSYS software. Testing of gearbox was carried by running it on an ATV.*

Keywords*-* AGMA, ASME, ATV, Bearing, Gearbox, Spur gear, Shaft, Torque.

I. INTRODUCTION

In Automobile, to transmit power from engine to the road wheels a transmission unit is required. It plays a crucial role in performance of the automobile on various track conditions. Transmission unit of a vehicle includes clutches, torque converters, CVTs, IVTs (in case of automatic transmission), Multispeed Gearbox, Differential unit, Axels. In this Gearbox plays an important role. The Gearbox is the secondary element of the power train in an automobile. A transmission is a machine in a power transmission system, which provides controlled application of the power. Often the term transmission refers simply to the gearbox that uses gears and gear trains to provide speed and torque conversions from a rotating power source to another device.

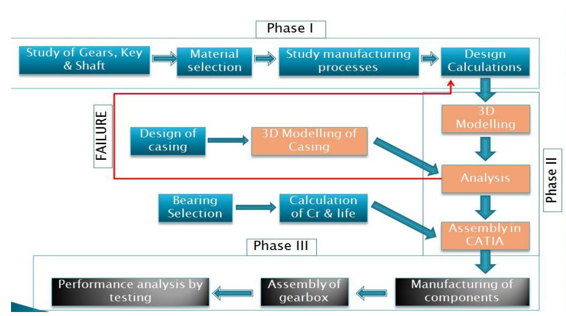
Main functions of automotive gear box are as follow:

1. Provide the torque needed to move the vehicle under a variety of road and load conditions. It does this by changing the gear ratio between the engine crankshaft and vehicle drive wheels.
2. Be shifted into reverse so the vehicle can move backward.
3. Be shifted into neutral for starting the engine.

II. PROBLEM STATEMENT

In every vehicle, transmission unit plays an important role in its performance. The engine along with its power train components should be very efficient so as to provide an enhance performance in any road condition which will certainly give feel of joy to the rider or driver. So, to provide maximum torque and to transmit power from the engine a two stage reduction gearbox is being designed. Also it should be light weight to increase the performance of the vehicle. And gearbox should also serve the purpose for inboard braking of our ATV.CVT is the primary drive which provides variable gear ratio but not the sufficient torque to let the vehicle crawl. So, we design a constant reduction gear box to multiply the torque output from CVT, so that the vehicle can perform in any road condition.

III. METHODOLOGY



It consists of three phases:

Phase I: Study of various components of gearbox such as shaft, gears and keys. After that study and selection of material.

Material selection based on:

- Working condition i.e. power, speed & torque.
- Working environment i.e. temperature, vibration.
- Ease of manufacturing and cost of material.

So considering above parameter EN24 is selected for gears and shaft having an yield strength of 680 N/mm^2 and ultimate tensile strength = 850 N/mm^2. Study of manufacturing process and design calculations.

Phase II: It consists of designing a 3D model in CATIA V5 and its analyses ANSYS WORKBENCH. If a component fails then again the design calculations are done.

Phase III: Manufacturing components of gearbox such as gears, shaft, key and casing. Assembly and and testing of gearbox under off road terrain.

IV. DIMENSION CHART

Dimension of Pair 1:

Pair 1:-

Parameter	Gear	Pinion
Teeth	60	18
Module	2mm	2mm
PCD	120mm	36mm
Pressure Angle	20 deg	20 deg
Addendum	2mm	2mm
Deddendum	2.5mm	2.5mm
Base circle	112.76mm	33.8289mm
Face width	20mm	20mm
Circular Pitch	6.28mm	6.28mm
Diametral Pitch	.5mm	.5mm
Tooth thickness	3.14mm	3.14mm
Center to center distance	78mm	

Dimension of Pair 2:

Pair 2:-

Parameter	Gear	Pinion
Teeth	62	18
Module	2mm	2mm
PCD	124mm	36mm
Pressure Angle	20 deg	20 deg
Addendum	2mm	2mm
Deddendum	2.5mm	2.5mm
Base circle	116.52mm	33.8289mm
Face width	20mm	20mm
Circular Pitch	6.28mm	6.28mm
Diametral Pitch	.5mm	.5mm
Tooth thickness	3.14mm	3.14mm
Center to center distance	80mm	

V. NOMENCLATURE

P = Diametral pitch.
Y = Lewis form factor.
F = Face width (3 to 5 times pitch).
Wt = Tangential transmitted load.
Ko = Overload factor.
Kv = Dynamic factor.
Ks = Size factor.
Pd = Transverse diameteral pitch.
F = Face width.
Km(KH) = Load distribution factor.
KB = Rim thickness factor.
J = Geometry factor.
mt = Transverse metric module.
Temperature factor KT = 1
Surface condition factor = Cf = 1
Load sharing factor = mN = 1
Surface strength geometry factor = I
Cp = Elastic coefficient
SFP = AGMA factor of safety for Pinion
SFG =AGMA factor of safety for Gear
Peff = Effective load.
RA = Radial load.
X = Radial load factor = 1.
Y = axial load factor = 0.
Fr = Radial load.
Fa = Axial load.
Ka = Application factor.

VI. CALCULATIONS

Spur Gear design by AGMA code:

i.Bending stress(σb):
σb =Bending moment(M)/Section modulus(I/c)
= Wt P/FY
σ = Wt*Ko*Kv*Ks*(Pd/F)*(KmKB/J)
ii. Pinion tooth bending:
σp = Wt*Ko*Kv*Ks*(Pd/F)*(KmKB/J)
= 386.68*1.243*0.9963*12.7/0.63*1.0957
= 33594.4 psi.
SFP = (St*YN/(kt.kr.σ))
= (43720*0.9767/33594.4*0.85)
= 1.49539 (Safety Factor)
iii. Gear tooth bending:
σG = 386.68*1.243*1.0047*12.7/0.63*1.095/0.3
= 35531.64 psi
SFG = (38309*0.09981)/0.85*35531.64
= 1.266 (Safety Factor of Gear)
iv. Pinion tooth wear:
(σc)p = cp(wt*ko*kv*ks*km/dPF*CF/I)^(1/2)
=2300(386.68*1*1.243*0.9963*1.095/1.9173 *0.63*0.1245)^(1/2)
=157963.06 psi
(SH)p = (157900*0.9484/1*0.85*157963.06)
= 1.115 (Safety Factor)
v. Gear tooth wear:
(σc)G = [(ks)G/(ks)p]^(1/2)(σc)p
=[1.0047/0.9963]^(1/2)*157963.06
=158627.571psi
(SH)G = 135360*0.9758*1.0063/0.85*158627.57
=0.98578 (Safety Factor)

Gear tooth is failing in wear strength, so to compensate this failure, we have to increase the hardness of gear and pinion.

vi. Beam strength (Sb):
Sb = m*b*σb*Y
b=8m
σb =Sut/3=1110/3=370N/mm^2
Zp=18 Y=0.308 (from data handbook)
Sb = m*8m*370*0.308
= 911.68m^2 N
vii. Wear Strength(Sw):
Sw=bQdp'k
Q = (2zg/zg+zp) = 2*60/60+18
=1.5385
dp = m*zp = 18m
k = 0.16(BHN/100)^2
= 0.16(419/100)^2 (AISI 4340 Hardened to 45 HRC)
= 2.81
Sw = 8m*1.5385*18m*2.81
= 622.5386m^2 N
Therefore, Sw<Sb
Taking module = 2
Beam strength,
Sb = 911.68 m^2
= 911.68*4
=3646.72 N
Wear strength,
Sw = 622.5386m^2
=622.5386*4
=2490.1544 N
Effective Load
Peff = Cs/Cv*Pt
V = πdp'np/60000 = π*18*2*8837.21/60000
= 16.6577m/s
Cv = 6/6+V = 0.2648
Mt = 60*10^6*7.457/2π *8837.21
= 8057.87Nmm
Pt = 2Mt/dp
= 2*8057.87/18*2
= 447.66N
Peff = Cs/Cv*Pt
=1.2/0.2648*447.66
=2028.67 N.
To Avoid Failure.
Sb = Peff *FOS
3646.72 = 2028.67*FOS
(FOS)b = 1.8
Sw = Peff *FOS
2490.1544 = 2028.67*FOS
(FOS)w = 1.2275
Therefore design is safe for wear strength.

Shaft Design

Shaft are design as per ASME standard. It is based on maximum shear stress theory. According to this code the permissible shear stress τ_{per} for shaft is considered as 30% of yield strength.

Kb = Shock and fatigue factor for bending moment
Kt = Shock and fatigue factor for torsional moment

For solid shaft -
$$\sqrt{(KbM)^2 + (KtT)^2} = \frac{\pi}{16} \times d^3 \times \tau_{max}$$
For hollow shaft -
$$\sqrt{(KbM)^2 + (KtT)^2} = \frac{\pi}{16} \times d^3 \times (1-k^4) \times \tau_{max}$$

By above code shaft diameter were found as below:

Shaft	Diameter (mm)
Shaft 1	25
Shaft 2	25
Shaft 3	30

Bearing Selection

For shaft 1 (Bearing no - 6205 is selected from SKF bearing catalogue).
RA = (RAH^2+ RAV^2)^1/2
= 2140.74 N.
Axial Load = 0.
Ka= 1.2.
Equivalent dynamic load (Pe)
Pe = (X*V*Fr+Y*Fa)*Ka
= 2568.88 N.
Bearing life
L_{10} = (L_{10} *60*n)/10^6
L_{10} = 120 million revolution.
Basic dynamic capacity
$L_{10} = (C/Pe)^a$
a = 3 for ball bearing.
C = (L_{10})^1/a*Pe.
= 12.67 KN.
C = 14.80 KN from SKF catalogue.
Hence, C = 12.67KN < 14.80 KN.
Therefore, Bearing 6205 is suitable.

For Shaft 2 (Bearing no - 6305 is selected from SKF bearing catalogue).
RA = (RAH^2+ RAV^2)^1/2
= 1918.65 N.
Pe = (X*V*Fr+Y*Fa)*Ka
= 2302.38 N.
L_{10} = (L_{10} *60*n)/10^6
L_{10} = 399.6 million revolution.

$L_{10} = (C/Pe)^a$
$C = (L_{10})$^1/a*Pe.
C = 16.95 KN < 23.40 KN from catalogue.
Hence bearing 6305 is safe.

For shaft 3 (Bearing no **-** 6306 is selected from SKF bearing catalogue).
RA = (RAH^2+ RAV^2)^1/2
= 3143.60 N.
Pe = (X*V*Fr+Y*Fa)*Ka
= 3772.32 N.
$L_{10} = (L_{10} *60*n)/10^6$
L_{10} = 412.8 million revolution.
$C = (L_{10})$^1/a*Pe.
= 28.08 KN < 29.60 KN.
Hence bearing 6306 is safe.

VII. ANSYS RESULT

Gear 1

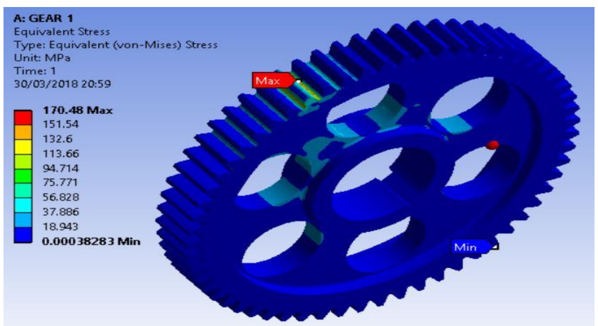


Fig 1.Stress Analysis

Shaft 1

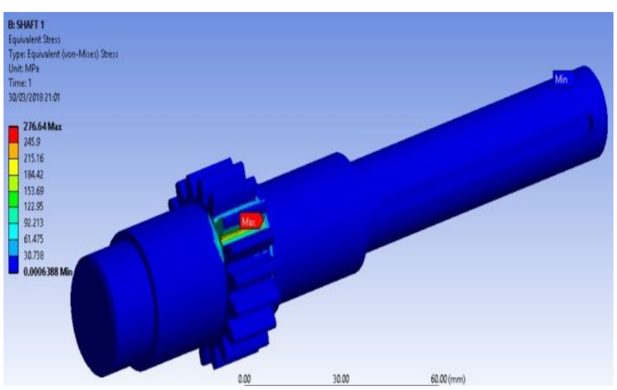
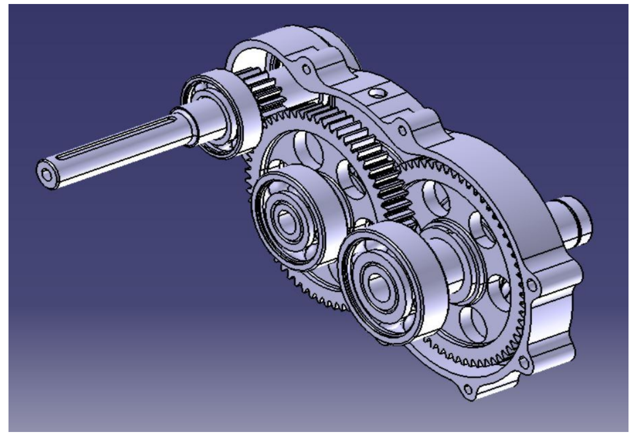


Fig 2 .Stress Analysis

Gearbox Assembly



VIII. CONCLUSION

A project report on Design, analysis, Optimization of gearbox reflects the fundamental aspects of reduction gearbox implemented for the generation of maximum torque required for the vehicle. From the various papers that we have referred for the study of the project conclude that the gearbox which we have proposed to get manufacture has many advantages over the standard gearbox available in the market.

We are also successful in reducing the weight as well as the cost of the gearbox comparing with the standard gearboxes available in the market without affecting the properties and other safety factors of the gearbox.

The gearbox is successfully tested and examined in all terrains without any damage to any component of the gearbox. We also successfully completed the BAJA SAE INDIA competition held at Indore.

REFERENCES

[1] Aditya Patankar, Rohit Kulkarni, Sanketkothawade And Sameer Ingle, '*Design And Development Of A Transmission System For An All Terrain Vehicle'*,International Journal Of Mechanical Engineering And Technology (Ijmet) Volume 7, Issue 3, May–June 2016
[2] Eric T. Payne, '*Design Of An Sae Baja Racing Off-Road Vehicle Powertrain'*, The University Of Akron In Akron, Ohio, Usa, Spring 2015
[3] Chetan Wadile , Rohan Dubal , Roshan Kolhe , Versharangaswamy, Aqleem Siddiqui & Nitin Gurav, *"Selection, Modification And Analysis Of Power Transmission And Braking System Of An Atv"*, Issn (Print): 2321-5747, Volume-1, Issue-1, 2013.
[4] Abhinav Sharma, Jujhar Singh And Ashwani Kuma, '*Optimum Design And Material Selection Of Baja*

Vehicle', International Journal Of Current Engineering And Technology,Vol.5, No.3 (June 2015).

[5] V.B.Bhandri *"Design Of Machine Elements"*,Third Edition Tata McGraw Hill Edu,2010.

[6] Shigley J.E.,Shigley's *"Mechanical Engineering Design"*, Eighth Edition,Tata McGraw Hill Edu,2011.

[7] SingiReddy Ravindra , Ramesh Banothu, M.tech Students, Mechanical, Vathsalya Institute of Science and Technology , Nalgonda Dist. Telangana. *"Design and Analysis of Gear Shafts"*.