Design and Analysis of Two Stage Reduction Gearbox

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ABSTRACT- *Gearbox designed is safe affordable of performance efficient with respective speed and torque required to clear the dynamic event satisfactorily. It is 2 stage reduction gearbox along with reversal transmission system induced with efficient output and comparatively lesser weight. All this achieved by studying the gear ratio and various resistance acting on vehicle calculation of gear, shaft, bearing etc. with respect to force, speed, density, and other parameters. It also required specification of vehicle, engine and CVT. The design and its simulation is covered by SOLIDWORKS and ANAYS.*

KEY WORD:- Design, Transmission System, All Terrain Vehicle Or Handicap Vehicle, Gearbox, Engine, CVT, Designing And Analysis.

1. INTRODUCTION

Vehicle transmission is to convert the energy provide by engine into traction of vehicle as efficiency to overcome the speed, thrust, bumps, climbs, etc. By simulation of adjustment of gear speed, acceleration of vehicle, CVT reduction ratio, weight and fuel consumption. The reverse gear system to ensure the vehicle to get rid during uneven situation where vehicle is restricted to move due to muddy terrain to achieve all the dynamic events and endurance race.

2. LITERATURE SURVEY

Design and analysis of two stage reduction of gearbox for all terrain vehicles(OCT-2010)[1] .The parameters such as various road resistance ,air resistance, gradient resistance etc can affect the reduction ratio of gearbox.

Design and development of trans system for an ATV(May-2017)[2] To increase the efficiency with respective materials and to achieve required reduction ratio with constant and continuous power transmission.

Design of reverse gear mechanism in two wheeler for physically challenged person(JULY-2016)[3] We studied reversed mechanism which consists of 3 shaft with keyway cutting and four Spur gear mechanism shifted from R-L moves in forward direction and L-R reverse direction for physically challenged person.

Design and manufacturing of 2-stage speed reduce for BAJAATV(OCT.-2017)[4]The parameters such as various road resistance , air resistance, gradient resistance etc can effect the reduction ratio of gearbox.

Design and fabrication of reverse gearbox (MARCH-2017) [5] here we get to know about the reverse mechanism i.e used in ATV to overcome the bumps and the uneven surface road. With the help of shifter lever the forward drive is converted in reverse gear mechanism

3. PROBLEM DEFINATION

1. Lack of reversal system in two stage reduction gearbox.
2. In order to tackle heavy weight of the existing reversal system in gearbox.
3. Reduction ratio of gearbox is high. The current reduction ratio is 12.89.
4. Costly
5. Space

4. OBJECTIVE

1. To design and optimize reversal gearbox for all-terrain vehicle.

2. To reduce the heavy weight of the gearbox comprising of reversal system in vehicles such as auto rickshaw, wheelchairs.

3. To reduce the reduction ratio of the gearbox in reverse system.

4. To reduce the cost of the the gearbox by using efficient materials.

5. To reduce the space required for the gearbox by designing an integrated design model with respect to the other components in the assembly of the vehicle.

5. REQUIRED SPECIFICATION

INPUT POWER	7.5kw
HORSE POWER	10hp
MAXIMUM TORQUE	20Joules=20Nm

ENGINE SPEED(RPM)	3600rpm
WHEEL SPEED	ph
MAXIMUM OUTPUT TORQUE	560Nm
ENGINE IDLE SPEED	1750RPM
TYRE DIMENSION	22×8×12(INCHES)
MAXIMUM WEIGHT OF VEHICLE	220KG
OPERATING TEPM	108^0
WEIGHT OF DRIVER	50KG
CVT REDUCTION RATIO	Min-3 OR Max -0.5

6. GEAR RATIOS CALCULATION

1. Rolling Resistance = Weight ×Rolling Resistance Coefficient
Fr=270×9.81×0.16=423.92N
2. Gradient Resistance=Weight×Gradient Coff.
Fa=270×9.81×0.60=1589.22N
3. Total Resistance (TR) =F_r+F_a=2013.012N
4. Reactive Torque On Wheels=TR×TYRE RADIUS =2013.012×0.294=562.435Nm
5. Reduction Ratio Is Given By:
Reactive Torque On Wheels =Engine torque × CVT Ratio× reduction ratio
562.435=18.5 × 3 × R.R
Therefore R.R=9.99=10.

7. GEAR-

A gear is a rotating machine part having teeth which mesh with another toothed part to transmit torque. Geared devices can change the speed, torque and direction of power source.

GEAR 2	**PCD=90mm**	**Z_2=36**
GEAR 3	**PCD=250mm**	**Z_3=100**
GEAR 4	**PCD=50mm**	**Z4=20**

7.1 DESIGN OF GEAR

MATERIAL SELECTION:- WHITE CAST IRON
Density=7.77×10^3 kg/cm^3
Young's modulus=175 ×10^3N/mm^2
Tensile Strength =700N/ mm^2
BHN=415, FOS=4
Poisson Ratio=0.27-0.28
Bulk Modulus= 58-107 *10^3N/mm^2
1.Bending Stress =700/FOS=1750kgf/cm^2
2.Crushing Stress=2.8(BHN)-70=10920 kgf/cm^2

CALCULATION FOR FORWARD MOTION OF GEARBOX: FIRST STAGE

A] Specification:

Gear 1 module(m)=2.5 Z_1=12
1. Blank Diameter =m(Z+2)=2.5(18+2)=35mm
2. Pitch Circle Diameter (D)=mz_1=30mm
4. Addendum=2.5mm
5. Deddendum =1.25×m=3.125mm
6. Tooth Depth =2.25×m=5.625mm
7. Clearance =0.625mm
8. Radius Fillet(R) =0.4×m =1mm
9. Diametrical Pitch (P_d)=Z_1/D=0.4mm
10 .Circular Pitch=Π×D/P_d=234mm
11. Root Diameter =Z-2/P_d=25mm
12. Base Diameter =D×Cos20=28.19mm
13. Circular Thickness=1.57/Pd=3.925mm

B] GEAR INDEXING CALCULATION:-

.Index Crank 1Moment =40/Z_1=3.33=4
2.No Of Rotation Required To Cut One Tooth =
40/Z_2=1.11=1, 40/Z_3=0.4=0.5

C] FORCE ANALYSIS:-

D_1=30mm , D_2=90mm, D_3=250mm
N_1=1200rpm N_2=400rpm N_3=148rpm
Therefore V=πDN/60
V_1=2m/s V_2=2m/s V_1=2m/s
For Gear 1:
Tangential force on gear 1 and gear 2
$T_{f1}=P/V_1$=7457/2.83=3728.5Nm
$V_1=V_2=V_3$, Therefore $T_{f1}=T_{f2}=T_{f3}$=3728.5Nm
Radial force
$R_{f1}= T_{f1}$×tan20=1357.09N
$R_{f1}=R_{f2}=R_{f3}$=1357.09N
Resultant force (F)
$F_1= T_{f1}$/cosϕ=2634.98/cos20=3967.7N
$F_1=F_2= F_3$=3967.7N
……………………….Forces for gear1, 2, and 3.

D] Design Procedure for spur gear based on gear life power=7.5kw=7457watts

N_1=1200rpm N_2=400rpm N_3=148rpm
P=2πNT/60
T_1=59.34 T_2=197.80 T_3=593.40
CHECKING STRESS:
1. Bending Stress =700/FOS=1750kgf/cm²
2. Crushing Stress=2.8(BHN)-70=10920 kgf/cm²
3. Design Torque[Mt]=K_t×K_d×M_t
$M_t=97420\frac{KW}{N1}$…………………………KW=P×S.F=8.9kw
Therefore M_t=72646 Nmm
[M_t]=94439.8 Nmm
4. Centre Distance(a)=$\frac{m(z1+z2)}{2}$=60mm
5. Selection of Teeth on Pinion Z_2=36
Therefore Z_2=i× Z_1
Z_1=36/3=12 teeth.
6. Calculation of Module (m)=$\frac{2a}{Z1+Z2}=\frac{2\times60}{12+36}$=2.5mm
7. Calculation of b, d, v,φ.
Face width(b)= φ×a=0.3×60=18mm
Pitch Diameter of Pinion (d_1) =mZ_1 =30mm
Pitch Velocity (v)=2m/s
φ_p =b/ d_1=0.65
1.**Checking for Bending** σ_t
$\sigma_t=\frac{i[Mt]}{amby}$=136.01 N/mm²
2.**Checking for Crushing**
$\sigma_c=0.74\times\frac{i+1}{a}\times\sqrt{E\times Mt}$ =891.304 N/mm²
Hence design is safe

Similarly for: - SECOND STAGE.

CALCULATION FOR REVERSE MOTION OF GEARBOX

MATERIAL SELECTION FOR REVERSE GEAR:
Material=40Ni10Cr_3Mo6
Density=7.8-8 kg/cm³
Young's Modulus=215×10³N/mm²
Tensile Strength =1350N/ mm²
BHN=401, FOS=5
Poisson Ratio=0.3
Bulk Modulus=140×10³N/mm²
1. Bending Stress =1350/FOS=270N/mm²
2. Crushing Stress=2.8(BHN)-70=1052.8 N/mm²

NOTE-(Calculate the parameters for reverse motion of gearbox, mentioned below.)

A] GEAR INDEXING CALCULATION:
B] FORCE ANALYSIS:
C] Design Procedure for spur gear based on gear life power=7.5kw=7457watts
CHECKING STRESS:

7.2 DESIGN PROCEDURE BASED ON AGMA METHOD: 1ST STAGE AND 2ND STAGE.

A]Material Selection :- white cast iron
B]Calculation of Z_1 and Z_2
Z_2=36 teeth $Z_1=Z_2$/I, therefore Z_1=12 teeth
C]Tangential Load (Wt):-P×Ko/V_1 , V_1= πDN/60
Therefore V_1=2m/s and Wt=7500N

D]Critical Dynamic load(Wd)=Wt×Cv Cv=(6+Vm)/6=1.471
Therefore Wd=11032.5N
E] Beam Strength (Wbs) = $\sigma_b \pi mby$, where y=0.4568 by calculation from data book.
Wbs_1=286984.5N, Wbs_2=308508.32N Wbs_3=3135181.84N
F] Wear Resistance(Ww)=Qbd_1K where
Q=2i/(i+1)=1.5 , d_1=30mm ,b=25mm K=$\sigma_c{}^2$sinα(2/E)/1.4=3.32
Ww1=3735N, Ww2=11205N, Ww3=31125N

Similarly:-FOR REVERSE

7.3 ANYSIS OF STAGE 1 AND 2

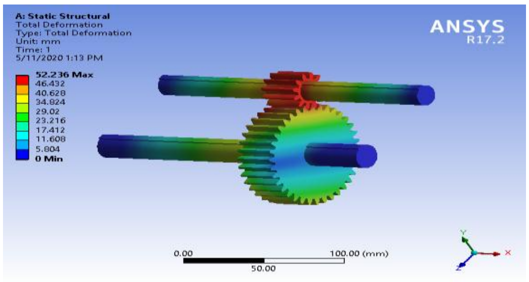
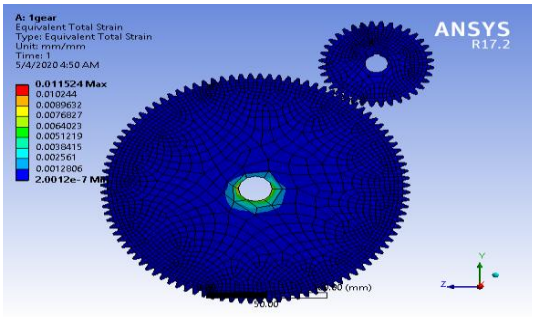
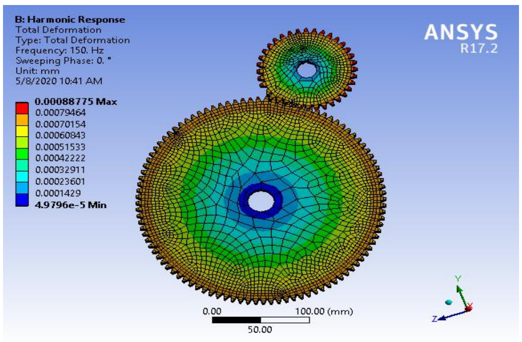


FIG.7.3.1 STAGE 1 SIMULATION



(A)



(B)

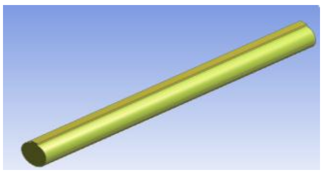
FIG.7.3.2. A, B, STAGE 2 SIMULATION

8. SHAFT–

A shaft is a rotating member/machine element, which is used to transmit power from one place to another.

8.1 DESIGN OF SHAFT

SHAFT 1:
1. Radial load on tooth surface (pinion)(Wr) W_r=W_t*tanϕ=7500*tan20=2729.7N
2.Bending moment on shaft: M=W_r*r =163788Nmm
3.Twisting moment on shaft: T=W_t×d/2=112500N
4.Equivalent torque on shaft: T_e=$\sqrt{m^2+T^2}$=198702.68Nmm
5.Shaft diameter(d)=3$\sqrt{\frac{T_e \times 16}{\pi \times \tau}}$=19mm



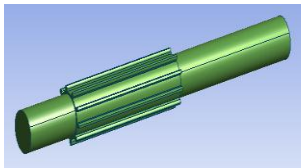


FIG.8.1.1.SHAFT FIG.8.1.2 SLINDING GEAR SHAFT

8.2 ANALYSIS OF SHAFT:

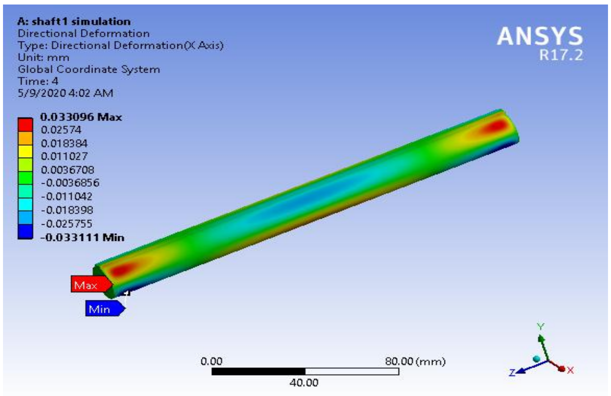


FIG.8.2.1SHAFT 1 SIMULATION

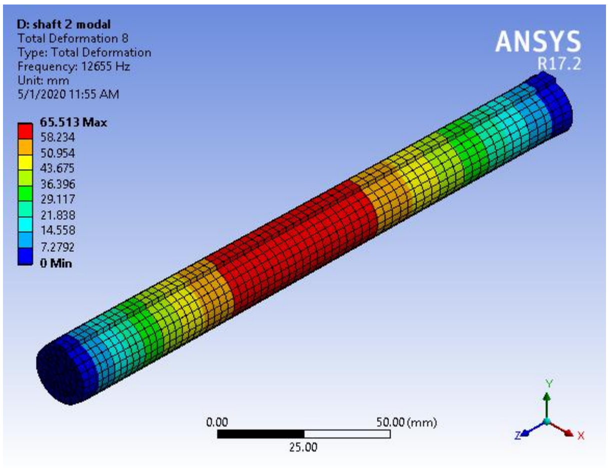


FIG.8.2.2 SHAFT 2 SIMULATION

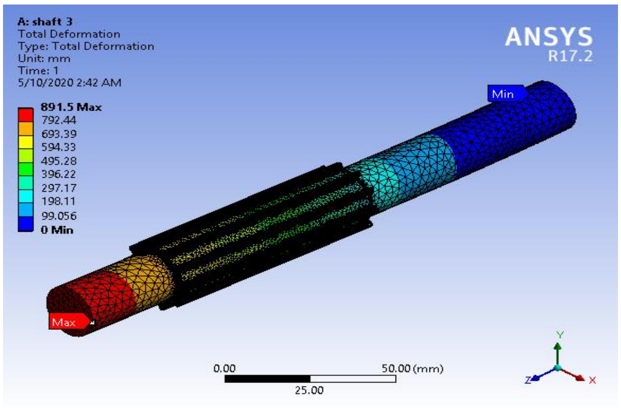


FIG.8.2.3 SHAFT 3 SIMULATION

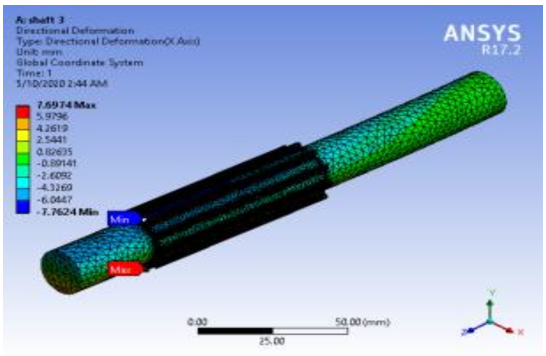


FIG.8.2.4 SHAFT 3 SIMULATION

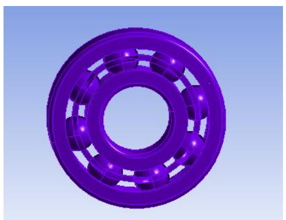
9. BEARING–

Mechanical bearings are used between two automotive parts that allow for rotation or liner movements. These bearings will enhance the vehicles performance, bear heavy loads and reduce friction.

SIMILARLY FOR SHAFT 2 & 3:	
1. W_r=2729.7N	1. W_r=2729.7N
2. M=163788Nmm	2. M=163788Nmm
3. T=122836.5N	3. T= 341212.5N
4. T_e=204732.3Nmm	4. T_e=78487Nmm
5. d =20mm.	5. d=25mm.

9.1 BEARING SELECTION:

1. Resistance force(R)= $\sqrt{T_f^2 + R_f^2}$=5634.25N
2. L_{08}= $\frac{L_{hr} \times N \times 60}{10^6}$ consider where L_{08}=92%survivial, $L_{hr} = 1000$
Therefore L_{08}=72mr
3 .$\frac{L_{08}}{L'_{10}}$= $[\frac{In(^1/_{P08})}{In(^1/_{P10})}]^{\frac{1}{b}}$= therefore L'_{10}=85.73mr
4.'c' Required =C=$[\frac{L'_{10}}{L_{10}}]^{1}/_{3}$×P
Therefore C=11813.42N=1181.342kgf.
B.NO6304
d=20mm,B=15mm,C=1250kgf,C_0=765kgf
6.checking : L'_{10} =$(c/p)^k.L_{10}$,where P=R.v.fr.s.Kt=7099.155 Fr=R=5634.35,s=1.2 kt=1.05
L'_{10}=91,58=92mr or L'_{10} =105mr
Hence we selected bearing no1.6304.
Similarly: Bearing no.2, 4,6 =6304
Bearing no. 3, 5 =6405



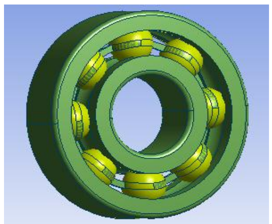


FIG.9.1.1. BEARINGNO.6304 **FIG.9.1.2. BEARING 6405**

10. CASING-The gear housing is the casing that surrounds the mechanical components of a gear box. It provides mechanical support for the moving components.

10.1CASING MODEL

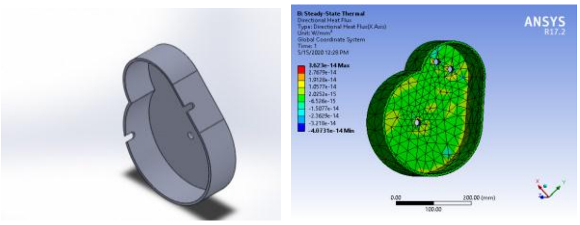
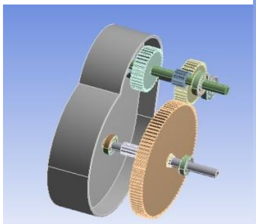


FIG 10.1.1 CASING AND ANALYSIS OF CASING

11. DESIGN CONCEPT

In forward transmission the power is driven to input shaft, through which it drives G_1 in clockwise direction which drives G_2 in anticlockwise G_2 mesh with G_3 so G_2 drives G_3 in clockwise.

In Reverse transmission, G_5 power is driven to input shaft through which it drives G_1 with $G_{4.}$ The lever connected to G_3toward to mesh with G_4. Here G_4 rotates clockwise direction whereas it drives G_3 in anticlockwise direction.



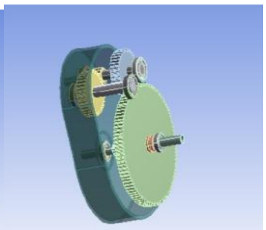
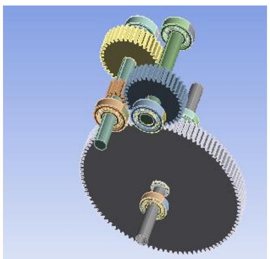


FIG.11.1 FIG.11.2
FORWARD DRIVE REVERSE DRIVE

12. SOLIDWORK MODEL



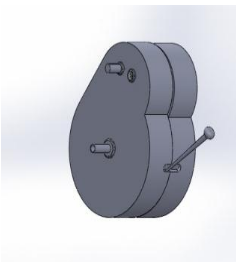


FIG12.1 FIG 12.2
GEAR SYSTEM GEARBOX ASSMEBLY

13. CONCLUSION

Our project "**Design and Analysis of Two Stage Reduction Gearbox**" gave us an opportunity to learn multiple knowledge of upgrading the problem solving and decision making concept in every steps of the project.

Such an mechanism of gearbox, its component and the parameters required for Designing as well as various boundary condition to be applied an simulation.

The project is mainly introduced to achieve the forward as well as reverse drive with an optimal requirement of speed and torque to enhance in the dynamic events.

These project will be very useful in setting a very efficient transmission drive

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