Design and Analysis of a gearbox for an all terrain vehicle

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Abstract- **The ATV's are generally small sized, single-seated motor vehicles used for off-road travelling. A two stage reduction gearbox is a part of a transmission assembly which reduces the rotational speed at the input shaft to a slower rotational speed at output shaft. Due to reduction in output speed, the torque of the system is increased. The objective of this paper is to design and analyze the various stresses acting on the gearbox on critical conditions of an ATV vehicle. The gearbox failure reasons are predicted with proper understanding and accordingly analysis is carried out. The gearbox design is modelled on CatiaV5 software. The calculations related to the boundary conditions upon which the analysis was performed using Ansys16.0 software are further verified with theoretical values.**

Index Terms- **ATV (All terrain vehicle); two stage-reduction gearbox; Torque; stress; Catia; Ansys**

1. INTRODUCTION

As ATV is a vehicle deigned for off-roading it has to tackle all the muddy areas, rocks, hills and all the obstacles in its way. Due to off-road terrain the friction is low as compared to normal roads. It requires high torque when climbing hills as well as while starting the vehicle. Also when running at high speeds at level road, high torque is not required because of momentum. For achieving this a compact and lightweight two stage-reduction gearbox is required which initially provides high torque by reducing the speed of the shaft. Thus the output shaft has lower rpm than the input shaft. The gearbox also restricts the power flow to the gear train by maintaining a neutral position.

The gearbox is designed with spur gears having involute tooth profile as they are having highest efficiency and ease in design considerations and manufacturing cost. During our study, we found that generally failure occurs in the gearbox when the tooth stress exceed the safe limit, thus it became necessary to calculate the maximum stresses acting on the gear under the applied boundary conditions. To prevent these failures analysis is performed on the gears.

The gear tooth fails in a number of ways such as pitting, stucking, scuffing, corrosion, scoring, etc. but the main causes are due to bending stresses and contact stresses. Thus based on our survey we performed analysis on two of the gear materials namely CI20 which is commonly used and AISI1060 in an attempt to suggest a better material according to the situation. After that we calculated theoretically the bending stresses by Lewis equation, the contact stress by hertz equation and finally the deflection by Castigliano's theorem. Both results which are obtained by analytical and theoretical methods are compared and finally the conclusions are made.

2 DESIGN CONSIDERATIONS

In the design procedure we first targeted the transmission line wherein the dimensions of the OEM parts were recorded.

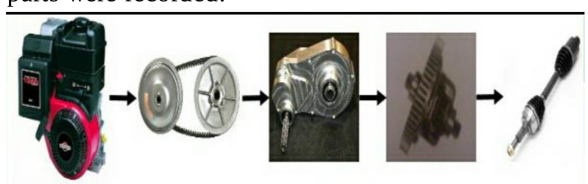


Fig2.1: Transmission line of the vehicle, where the power is transmitted from engine to cvt to gearbox to shaft through coupling.

From the above figure we observe that the power coming from the engine is transmitted to the CVT then to the gearbox and from coupling, finally to the shaft. The engine used in the vehicle has a maximum torque of 18.9Nm at 2800rpm and max power of 10hp at 3200rpm. The cvtech cvt used has a low gear ratio of 3:1. The diameter of Tyre specifications (22-7-10) with 22" radius and Static friction: 0.75 as well as rolling friction: 0.014 – 0.03. The total mass of the vehicle is taken to be 260kg. However the weight distribution is taken as 65:35.

3 DESIGN METHODOLOGY AND CALCULATIONS

Now the gearbox is designed on the critical condition where the vehicle is running at an incline of about 30 degrees.

3.1 MASS CALCULATIONS

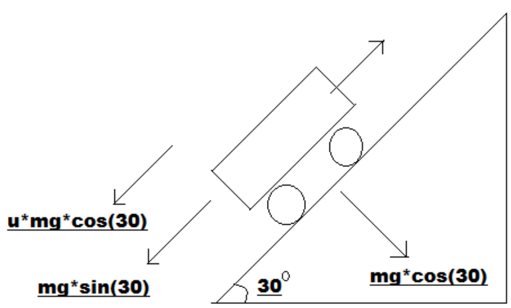


Fig3.1.1: Geometry of vehicle

Thus calculating mass 'm' acting on the rear wheels. So the actual weight or percentage of weight acting on the rear wheels for traction is 65% of total mass of vehicle 'M'.

m=0.65*M

=0.65*260

= 169kg

3.2 TRACTIVE FORCE AND GEAR RATIO

Total tractive force = Force req. to overcome Static friction + Force req. to overcome rolling friction

F(t) = F1 + F2 (1)

Now,

F1 = (u*mg*cos(30)) + mg*sin(30) (2)

where u = coefficient of static friction = 0.75

F1 = 1905.7761 N

Also,

F2 = U(r)*(u*mg*cos(30) + mg*sin(30)) (3)

U(r) = coefficient of rolling friction = 0.03

F2 = 57.1732 N

Hence,

Total Tractive force = 1905.7761 + 57.1732

F(t) = 1970 N (approx.)

Now this tractive force will give us the torque developed along shaft through tire

Radius of tyre = 11"

Torque = F(t)* r (4)

T = 1970*11*0.0254

T = 550.418 N-m

Thus taking an approx. torque of 600Nm

But max torque supplied by the engine

T(e) = 19.8 N-m

Taking low ratio of CVT as 3:1. In order to achieve required torque, the gear reduction ratio required is G.

G = 600/19.8*3

G = 10.10 ……………..Final Gear ratio

3.3CALCULATING MODULE AND VALIDATING THE DESIGN

The staging of gearbox is done, taking in to consideration the gear hunting tooth phenomenon and optimum spacing.

Hence, 1st staging = 3.388:1

2nd staging = 3.388:1

Thus, velocity ratio V.R. = 3.388:1, N=2800, T=19.8Nm

tp = 18

D_p=18m

$P_r = \frac{2\pi * N * T}{60}$ = 5.541 KW (5)

$V_p = \frac{\pi * Dp * Np}{60}$ =2.638 m/sec. (6)

$P_d = P_r*K*l$ = 5.541*1.25 =6.962 KW. (7)

$F_t = \frac{Pd}{Vp} = \frac{2.639}{m}$ KN. (8)

F_b = So*Cv*b* Y*m (9)

Where,

S_{yt} = 450 Mpa

So = 300

Cv = 0.3

Y = 0.325

b = 8m.

F_B = 292.99 m^2 N

Since $F_t = F_B$ (10)

$\frac{2.639}{m}$= 292.99 m^2

m=2.080 ≈ 2.5 ………on standardizing

F_t =1055.6 N

D_p =18m = 49mm

D_g = 61m = 152.5mm

V_p = 2.638m = 6909 m/s

b = 8m =20 mm

F_B = 6408 N

As $F_B > F_t$ SO DESIGN IS SAFE (11)

3.4 CVT PULLEY LOADS

Now the CVT used in the vehicle consist of driver and driven pulleys with diameter 200mm and 182.4mm respectively with a center distance of 260mm. The CVT is set at an angle of 30^0. The belt used is of cast iron. Now, as the cvt is mounted on the input shaft of gearbox thus the max power will be transmitted to cvt at max power of 10hp @3628rpm

Transmission torque

$T_{in} = \frac{30*P}{\pi N} = \frac{30*7.547*10^3}{\pi*3628} = 19.9$ Nm (12)

V-belt Pulley Loads

$T_{in} = (F_{max} - F_{min})*\frac{Dpulley}{2}$ (13)

$(F_{max} - F_{min}) = \frac{2*Tin}{Dpulley} = 199.2$ N

We also know

$\frac{Fmax}{Fmin} = e^{\mu\theta}$ (14)

Where $\mu = 0.35$

$\theta = \pi + \frac{D1-D2}{C}$ (radian) (15)

$= \pi + \frac{200-182.5}{260}$

= 3.208 rad

$\frac{Fmax}{Fmin} = e^{0.35*3.208}$

= 3.073 ≈ 3

So Fmax = 3* F_{min}

F_{min} = 99.6 N

Now, total force applied by belt on the shaft is given by $F_B = F_{max} + F_{min}$

$= 4F_{min}$

= 4*99.6

F_B = 298.8 N

3.5 GEAR TRANSMISSION 2D LAYOUT

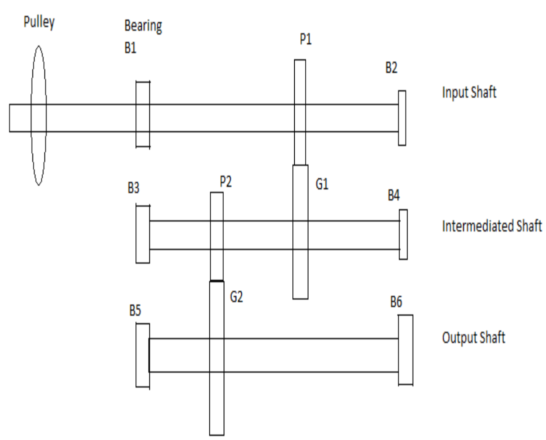


Fig 3.5.1: Line diagram of gearbox assembly

In the above diagram it is clear that the input torque provided by the engine is transmitted to the input shaft through low cvt ratio of 3:1. Further this torque is again increased due to first stage reduction and thus the torque at the intermediate shaft is obtained. In the end again the torque is increased due to second stage reduction and the final torque is obtained at the output shaft.

3.6 Calculations of Radial and Tangential forces for Bending Moment and Diameter of shafts

For calculating the moment and diameters, the forces required are obtained from above calculations are listed below.

Here,

PCD = pitch circle diameter

T = Torque

Ft = Tangential force

Fr = Radial force

Calculations for the Torque on input shaft

Now $T = T_{engine}$ * CVT low gear ratio

$T = T_{engine}*3$

= 18.9*3

T = 56.7 Nm

1. For Input shaft on Pinion (P1)

PCD = 45 mm

T = 56.7 Nm

F_t = (2*T/PCD) = 2520 N

$F_r = F_t * \tan(20)$ = 917.20 N

2. For intermediate shaft on gear (G_1)

PCD = 152.5 mm

$T = T_{input\ shaft} * 1^{st}$ gear reduction

T = 56.7*3.388 = 192.096 Nm

F_t = (2*T/PCD) = 2520 N

$F_r = F_t * \tan(20)$ = 917.20 N

3. For intermediate shaft on pinion (P_2)

PCD =45 mm

T =192.096 Nm

F_t = (2*T/PCD) = 8540 N

$F_r = F_t * \tan(20)$ = 3108 N

4. For output shaft on gear (G_2)

PCD = 61*2.5 =152.5 mm

$T = T_{intermediate\ shaft} * 2^{nd}$ gear radius

= 192.15*3.388

T = 651.175 Nm

$F_t = (2*T/PCD) = 8540$ N
$F_r = F_t * \tan(20) = 3180$ N

3.6.1 FREE BODY DIAGRAM OF INPUT SHAFT AND CALCULATING THE MOMENT THROUGH THE RADIAL FORCES

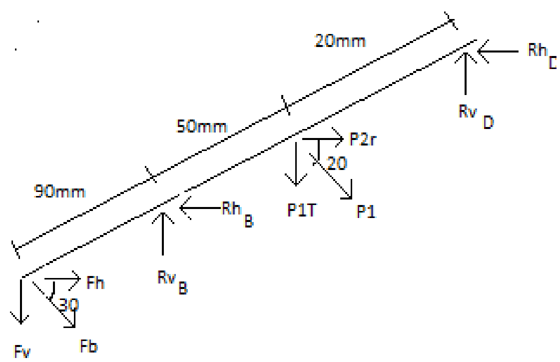
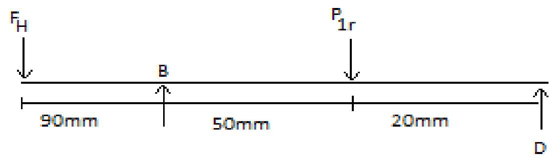


Fig3.6.1: FREE BODY DIAGRAM OF THE INPUT SHAFT

For calculating the moments of Radial Forces



- Now balancing the vertical forces

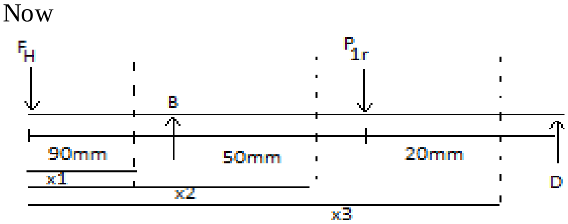
$\sum f(H) = R_{HB} + R_{HD} - F_H - P_{1r} = 0$

$R_{HB} + R_{HD} = F_H + P_{1r} = 344.6 + 917$
$= 344.6 + 917$
$= 1261.6$ N

- Calculating the moment about point D

$\sum Mhd = F_H*0.16 - R_{HB}*0.07 + P_{1r} * 0.02 = 0$
$R_{HB} = 1030$ N
$R_{HD} = 231.6$ N

Now



Taking the Moment using section formulae to calculating the moment for the section between $0<x_1<90$

$M_A = F_x * x_1$ ....general equation
$M_A = 0$
$M_B = 344.6 * 0.09 = 31$ Nm

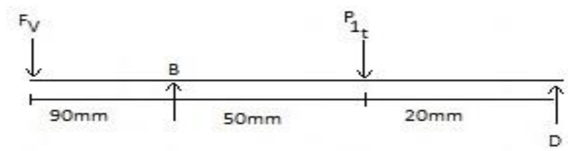
- Calculating the moment for the section between $90<x_2<140$

$M=F*x_2-F_H(x_2-0.09)$.......general equation
$M_B = 31$Nm
$M_{P1r} = 344.6*0.14 -1.03*0.05$
$= -3.256$ Nm

- Calculating the moment for the section between $140<x_3<100$

$M = F_H*x_3-R_{HB}(x_3-0.09) + P_{1r}(x_3-0.014)$ .....general equation
$M_{P1r} = -3.256$ Nm
$M_D \approx 0$

3.6.2 FREE BODY DIAGRAM OF INPUT SHAFT FOR CALCULATING THE MOMENT THROUGH THE TANGENTIAL FORCES



- Now balancing the vertical forces

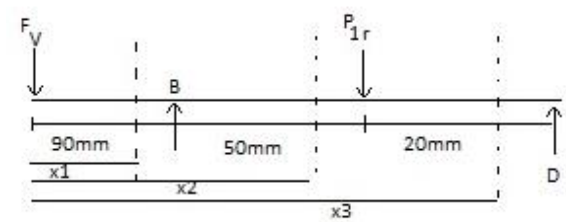
$\sum Fv = R_{VB} + R_{VD} - F_v - P_{IT}$

$R_{VB} + R_{VD} = 0.199+2.52$
$= 2719$ N

- Calculate the moment about point D

$\sum Mvd = F_{v\ *0.16} - R_{VB}*0.07+ P_{IT}*0.02$
$R_{VB} = 1170$ N
$R_{VD} = 1549$ N

Now



Taking the Moment using section formulae to calculating the moment for the section between $0<x_1<90$

$M = F_v* x_1$.................general equation
$M_A = 0$
$M_B = F_v*0.09 = 17.9$ Nm

- Calculating the moment for the section between $90<x_2<140$

$M= F_v*x_2- R_{Bv}(x_2-0.09)$ ....general equation
$M_B = 17.9$ Nm
$M_{P1T} = 149.2*0.14-199.2*0.05$
$= -30.64$ Nm

- Calculating the moment for the section between $140<x_3<160$

$M = F_H * x_3 - R_{VB}(x_3 - 0.09) + P_{1r}(x_3 - 0.014)$…general equation

$M_{P1T} = -30.64$ Nm

$M_D = 149.2*0.16 - 593.82*0.07 + 2520*0.02$

$M_D \approx 0$

4.6.3 Finding the critical section on the input shaft and obtaining the diameter

$M_A = 0$

$M_B = \sqrt{31^2 + 173^2} = 35.79$ Nm

$M_{P1} = \sqrt{-3.25^2 + (-30.64)^2} = 30.8$ Nm

$M_D = 0$

The critical section is at point B

Now calculating the equivalent torque on the input shaft,

Hence we know

$$T_{eq} = \sqrt{M^2 + T^2} \quad (16)$$

$T_{eq} = 67.05083$ Nm

$= 67050.83$ Nmm

$$\text{Also, } T_{eq} = \frac{\pi}{16} * ds^3 * \tau_{max} \quad (17)$$

$$\text{where……. } \tau_{max} = \frac{Syt}{2*FOS} \quad (18)$$

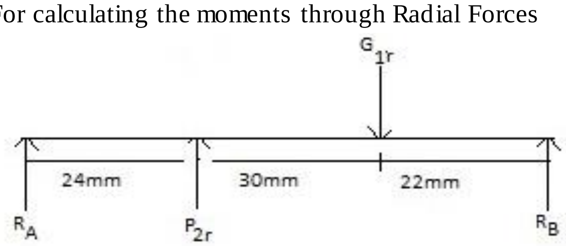
$\tau_{max} = \frac{680}{2*2}$

$\tau_{max} = 170$ MPa

Putting the value of τ_{max} in the above equation we get the diameter of the shaft as $d_s = 14.11$mm. But the diameter is taken as 15mm on the basis of the standard bearing size.

3.6.4 FREE BODY DIAGRAM OF INTERMEDIATE SHAFT AND CALCULATING THE MOMENT THROUGH THE RADIAL FORCES

For calculating the moments through Radial Forces



- Now balancing the vertical forces

$R_A + R_B + P_{2r} = G_{1r}$

$= 917.20 - 3108$

$= -2190.8$ N

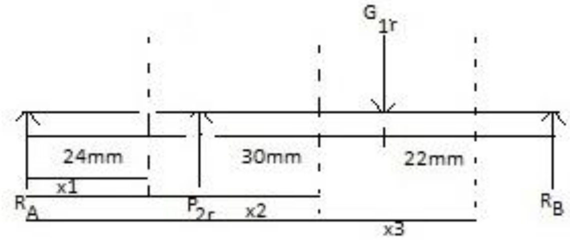
- Calculating the moment about point B

$-R_A * 76 - 3108*52 + 917.2*22 = 0$

$R_A = -1861.02$ N

$R_B = -329.78$ N

Now



Taking the Moment using section formulae to calculating the moment for the section between $0<x_1<24$

$M = R_A * x_1$ ………………general equation

$M_A = -0.0073$Nm ≈ 0

$M_{P2r} = 44.64$Nm

- Calculating the moment for the section between $24<x_2<54$

$M = F*x_2 - F_H(x_2 - 0.09)$…….general equation

$M_{P2r} = 44.64$Nm

$M_{G1r} = 7.25$Nm

- Calculating the moment for the section between $54<x_2<76$

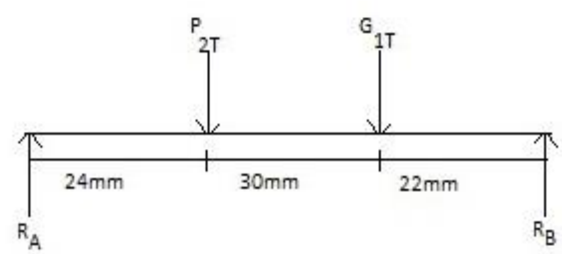
$M = F*x_2 - F_H(x_2 - 0.09)$…….general equation

$M_{G1r} = 7.25$Nm

$M_B = -0.00008$Nm ≈ 0

3.6.5 FREE BODY DIAGRAM OF INTERMEDIATE SHAFT AND CALCULATING THE MOMENT THROUGH THE TANGENTIAL FORCES

For calculating the moments through Tangential Forces



- Now balancing the vertical forces

$R_A + R_B = P_{2T} + G_{1T}$

$= 8540 + 2520$

$= 11060$ N

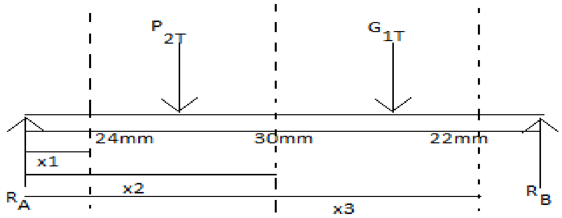
- Calculating the moment about point

$-R_A*76 + 8540*52 + 2520*22 = 0$

$R_A = 6572.63$ N

$R_B = 4484.36$ N

Now



Taking the Moment using section formulae to calculating the moment for the section between $0<x_1<24$

$M=R_A * x_1$ ………………general equation

M_A = -0.228Nm

M_{P2T} = 157.74Nm

- Calculating the moment for the section between $24<x_2<54$

$M=F*x_2-F_H(x_2-0.09)$…….general equation

M_{P2T} = 157.74Nm

M_{G1T}= 98.65Nm

- Calculating the moment for the section between $24<x_2<54$

$M=F*x_2-F_H(x_2-0.09)$…….general equation

M_{G1T} = 98.65Nm

M_B = 0.00012Nm ≈ 0

3.6.6 FINDING THE CRITICAL SECTION ON THE INTERMEDIATE SHAFT AND OBTAINING THE DIAMETER

$M_A = \sqrt{-0.228^2 + -0.0073^2}$ = 0.2281Nm

$M_{P2} = \sqrt{(157.74)^2 + (44.64)^2}$ = 163.93Nm

$M_{G1} = \sqrt{(98.65)^2 + (7.25)^2}$ = 98.916Nm

M_B = 0

The critical section is at point P2

Now calculating the equivalent torque on the input shaft,

Hence we know

$T_{eq} = \sqrt{M^2 + T^2}$

$= \sqrt{(163.93)^2 + (192.15)^2}$

= 252.57Nm

Also

$T_{eq} = \frac{\pi}{16} * ds^3 * \tau_{max}$

Where……. $\tau_{max} = \frac{Syt}{2*FOS}$

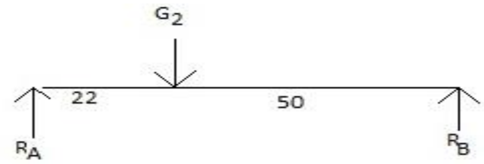
$\tau_{max} = \frac{680}{2*2}$

τ_{max} = 170 MPa

Putting the value of τ_{max} in the above equation we get the diameter of the shaft as d_s = 21.973 mm. But the diameter is taken as 25mm on the basis of the standard bearing size

3.6.7 FREE BODY DIAGRAM OF OUTPUT SHAFT AND CALCULATING THE MOMENT THROUGH THE RADIAL FORCES

For calculating the moments through Radial Forces



- Now balancing the vertical forces

$R_A + R_B$ = 3108 N

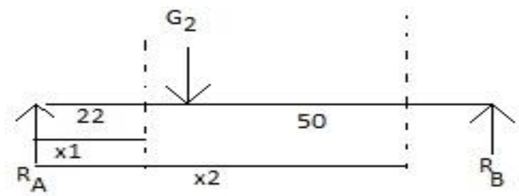
- Calculating the moment about point B

$-R_A*72 + 3108*50 = 0$

R_A = 2158.3 N

R_B = 949.66 N

Now



Taking the Moment using section formulae to calculating the moment for the section between $0<x_1<22$

$M=R_A * x_1$ ………………general equation

M_A ≈ 0Nm

M_{G2r} = 47.482Nm

- Calculating the moment for the section between $22<x_2<72$

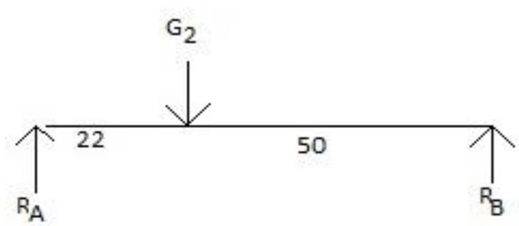
$M=F*x_2-F_H(x_2-0.09)$…….general equation

M_{G2} = 47.48Nm

M_B≈ 0Nm

3.6.8 FREE BODY DIAGRAM OF OUTPUT SHAFT AND CALCULATING THE MOMENT THROUGH THE TANGENTIAL FORCES

For calculating the moments through Tangential Forces



- Now balancing the vertical forces

$R_A + R_B$ = 8540 N

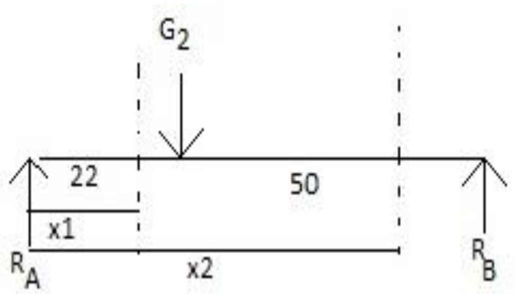
- Calculating the moment about point B

$-R_A*72 + 8540*50$

$R_A = 5930.96$ N

$R_B = 2609.44$ N

Now



Taking the Moment using section formulae to calculating the moment for the section between $0<x_1<22$

$M=R_A* x_1$ ………………general equation

$M_A \approx 0$Nm

$M_{G2T} = 130.472$Nm

- Calculating the moment for the section between $22<x_2<72$

$M=F*x_2–F_H(x_2–0.09)$…….general equation

$M_{G2} = 130.472$Nm

$M_B \approx 0$Nm

3.6.9 FINDING THE CRITICAL SECTION ON THE INTERMEDIATE SHAFT AND OBTAINING THE DIAMETER

$M_A = 0$

$M_B = 0$

$M_{G2} = \sqrt{(47.482)^2 + (130.472)^2}$

$= 138.84$Nm

The critical section is at point G2

Now calculating the equivalent torque on the input shaft,

Hence we know

$T_{eq} = \sqrt{M^2 + T^2}$

$= \sqrt{138.84^2 + 651.17^2}$

$= 665.806$ Nm

Also

$T_{eq} = \frac{\pi}{16}*ds^3*\tau_{max}$

Where…..… $\tau_{max} = \frac{Syt}{2*FOS}$

$\tau_{max} = \frac{680}{2*2}$

$\tau_{max} = 170$ MPa

Putting the value of τ_{max} in the above equation we get the diameter of the shaft as d_s = 30.33 mm. But the diameter is taken as 35mm on the basis of the standard bearing size

In similar fashion, by using same standard procedure we obtained the values for the gear material CI grade20 which are listed below

Properties	AISI 1060	CI20
Module	2.5	4
Pressure angle	20	20
Diameter of pinion	49	72
Diameter of gear	152.5	244
Face width	20	32
Number of teeth on pinion	18	18
Number of teeth on gear	61	61
Diameter of input shaft(mm)	15	15
Diameter of intermediate shaft(mm)	25	20
Diameter of output shaft (mm)	35	30

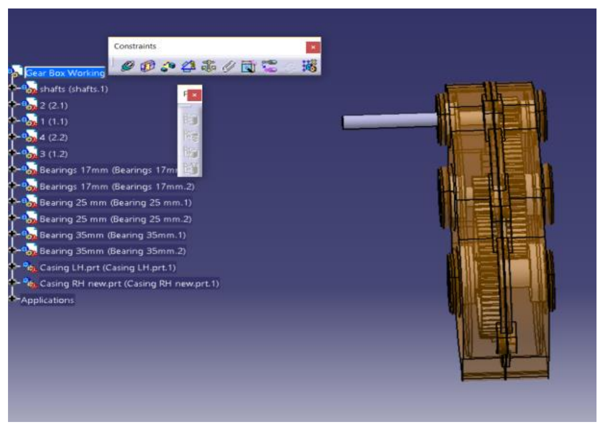


Fig3.6.9.1: The gearbox with visible transmission through casing.

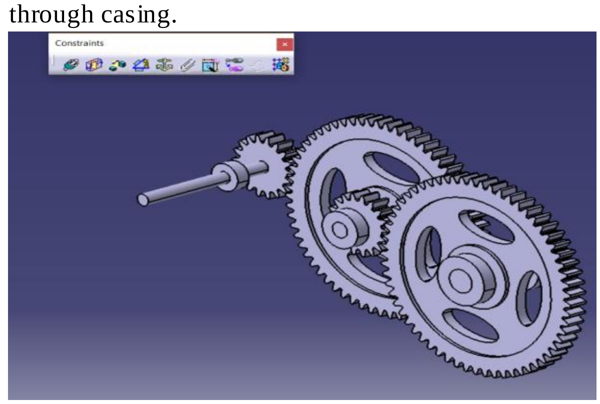


Fig3.6.9.2: The gearbox mechanism assembly.

4 ANALYSES BY FEM

The gearbox is designed on the basis of the above theoretical values on CatiaV5 software. Further the failure analysis is carried out on Ansys16 software where the boundary conditions are applied accordingly.

4.1 ANALYSIS PROCEDURE

The assembly generated on CatiaV5 is then imported to Ansys16 software in IGES format which becomes as a single body and thus analysis is performed.

The necessary material properties are added to the file such as density, young's modulus, Poisson's ratio etc

Material	Density(g/cm^3)	Poisson's ratio	Young's modulus (Gpa)	Yield strength(Mpa)	Ultimate strength (Mpa)
AISI 1060	7.8	0.3	200	485	620
CI20	7.5	0.3	180	98	160

The body imported is then meshed with coarse definitions and boundary conditions of the particular failure modes are applied. Thus we obtain the analytical values.

After study we found out that the gearbox fails majorly due to bending failures, contact tooth failure and due to deflection values that exceed the safe limit.

4.2 CALCULATION OF BENDING STRESSES

Bending failures in the gear are calculated by using the equation derived by Wilfred Lewis. It is considered as the basic equation in designing the gears. However while deriving the equation Lewis considered certain assumptions which also became our boundary conditions while performing analysis.

Lewis considered that the gear tooth which acts as the cantilever beam itself. The further assumptions while deriving are stated as follows

- The forces are applied at the tip of the gear tooth in static loading.
- The load is uniformly distributed across the face width of the gear tooth.
- The radial component is negligible.
- Forces due to tooth sliding friction are negligible.

LEWIS FACTOR - Y

NO. OF TEETH	14 1/2° INVOLUTE	20° INVOLUTE
10	0.176	0.201
11	0.192	0.226
12	0.210	0.245
13	0.223	0.264
14	0.236	0.276
15	0.245	0.289
16	0.255	0.295
17	0.264	0.302
18	0.270	0.308
19	0.277	0.314
20	0.283	0.320
22	0.292	0.330
24	0.302	0.337
26	0.308	0.344
28	0.314	0.352
30	0.318	0.358
32	0.322	0.364
34	0.325	0.370
36	0.329	0.377
38	0.332	0.383
40	0.336	0.389
45	0.340	0.399
50	0.346	0.408
55	0.352	0.415
60	0.355	0.421
65	0.358	0.425
70	0.360	0.429
75	0.361	0.433
80	0.363	0.436
90	0.366	0.442
100	0.368	0.446
150	0.375	0.458
200	0.378	0.463
300	0.382	0.471
RACK	0.390	0.484

Fig4.2.1: The relationship between Lewis form factor(Y) and number of teeth (Z), for Z=61 we have Y=0.42

Calculation of Bending stress by Lewis formulae is given by $\sigma_b = \frac{F_t}{m*b*Y}$ (19)

where $F_t = \frac{2*T}{d}$ (20)

For the material AISI 1060 and CI20

$F_t = \frac{2*19.7*10^3}{152.5}$

$F_t = 258.36$ N

$\sigma_b = \frac{258.36}{2.5*20*0.42}$

$\sigma_b = 12.3028$ Mpa

$F_t = = \frac{2*19.7*10^3}{244}$

$F_t = 161.479$ N

$\sigma_b = \frac{161.479}{4*40*0.42}$

$\sigma_b = 3.00362$ Mpa

4.3 CALCULATION OF CONTACT STRESSES

The contact stress failures occurs while mating between the gears takes place. The contact stresses are calculated on the basis of Hertz equation. In application of this equation the two mating gears are having a line contact as in the case of cylinders. Thus maximum contact stresses along the line contact is given by $\sigma_c = \frac{2F}{\pi*b*L}$ (21) where F is the force applied by the gears at the specific torque, L is the length of contact and b is calculated by b

$$= \sqrt{\frac{4F\left(\frac{1-V_1^2}{E_1}+\frac{1-V_2^2}{E_2}\right)}{\pi * L *\left(\frac{1}{R_1}+\frac{1}{R_2}\right)}}$$ (22) now combining both the equations we get

$$\sigma_c = \sqrt{\frac{F\left(1+\frac{R_1}{R_2}\right)}{R_1 * B * \pi *\left(\frac{1-V_1^2}{E_1}+\frac{1-V_2^2}{E_2}\right)}}$$ (23) hence calculating the contact stresses for AISI1060 and CI20

$$\sigma_c = \sqrt{\frac{2520 *\left(1+\frac{22.5}{76.25}\right)}{22.5 * 20 * \pi *\left(\frac{1-0.3^2}{190}+\frac{1-0.3^2}{190}\right)}}$$

σ_c = 15.52 MPa.

And similarly

$$\sigma_c = \sqrt{\frac{1575 *\left(1+\frac{36}{122}\right)}{36 * 32 * \pi *\left(\frac{1-0.29^2}{180}+\frac{1-0.29^2}{180}\right)}}$$

σ_c = 7.4419 MPa.

4.4 CALCULATION OF DEFLECTION

By using the Castigliano's theorem the total deformation in the gear tooth can be calculated with minimum errors. The equation is give as

$$\delta = \frac{16 * F_t * h^3}{E * b * t^3}$$ (24) where F_t is the force applied at that torque, E is the stiffness constant, b is face width, h is height of the tool and t is the tooth thickness.

Now to calculate h, we have Y = t^2/6*h*m where m is the module and Y is the constant Lewis form factor and t is tooth thickness. Thus calculating the values for AISI1060

t = tooth thickness = 5.997mm and m = 2.5

For calculating h

We have Y = t^2/6*h*m……….for Z=61 (25)

Thus h = 5.71mm

Hence the value of deflection is found to be

$$\delta = \frac{16 * 258 * 5.71^3}{200000 * 20 * 5.997^3}$$

$\delta 1$ = 0.0008903 mm.

Similarly calculating the values for CI20

t = tooth thickness = 10mm and m = 4

For calculating h

We have Y = t^2/6*h*m……..y = 0.42 for Z=61

Thus h = 9.92mm

Hence the value of deflection is found to be

$$\delta = \frac{16 * 161.475 * 9.92^3}{200000 * 32 * 10^3}$$

$\delta 2$ = 0.000394 mm

5 RESULTS AND DISCUSSIONS

6

From the above data, analysis is performed on every failure mode and data obtained is as follows

- The bending stress analysis

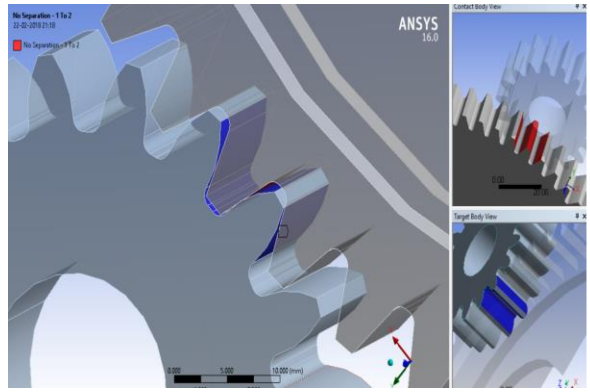


Fig5.1: Boundary conditions for applying the forces

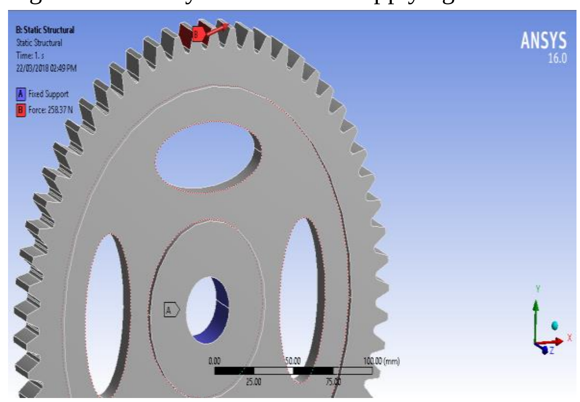


Fig5.2: Boundary condition for material AISI1060 wherein the bore is fixed and the force of 258.36N is applied

Material	Therorетical Bending stress(Mpa)	Analytical value(Mpa)	% Error
AISI1060	12.302	12.294	0.065.
CI20	3.0036	2.946	1.914

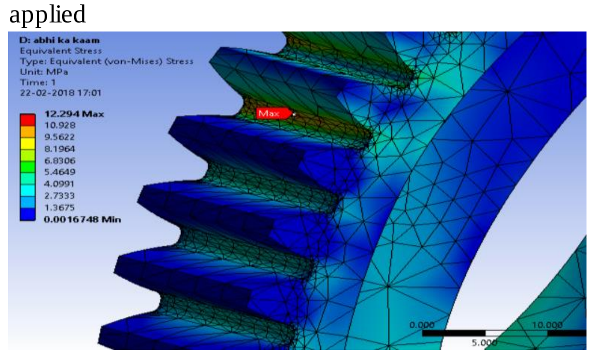


Fig5.3: Bending stress analysis is performed on material AISI1060 where maximum of 12.29Mpa stress is obtained

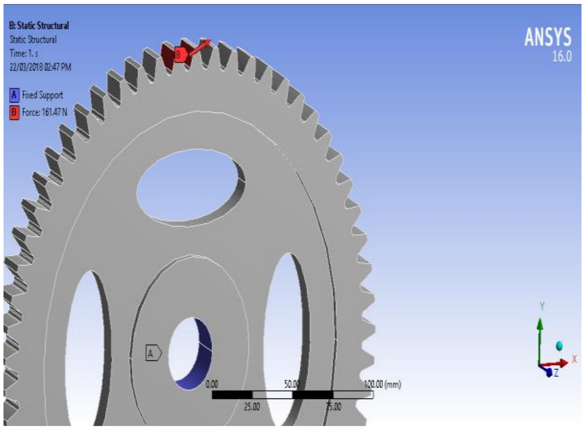


Fig5.4: Boundary condition for material CI20 wherein the bore is fixed and the force of 161.47N is applied

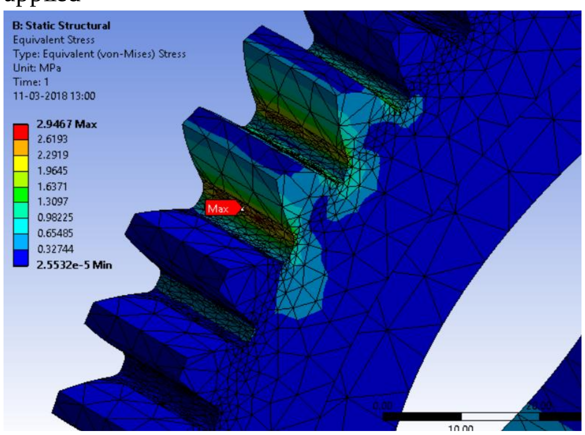


Fig5.5: Bending stress analysis is performed on material CI20 where maximum of 2.9467Mpa stress is obtained

- The contact stress analysis

In case of contact stresses the boundary conditions are applied on the area in contact in the direction of motion accordingly

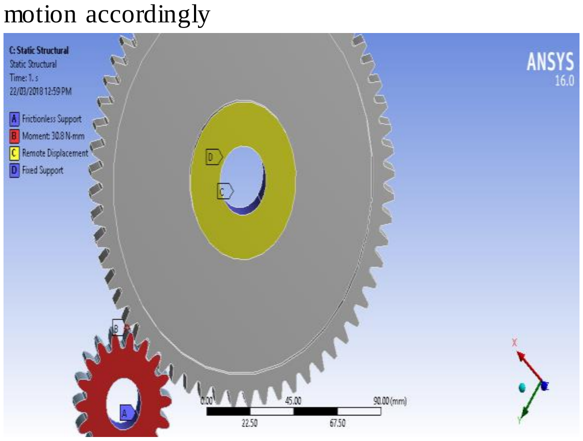


Fig5.6: Boundary condition for finding contact stresses by applying frictionless and fixed support as shown and moment of 30.8Nm

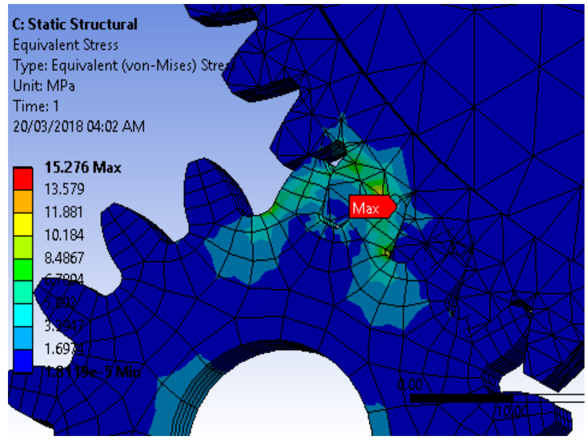


Fig5.7: The contact shear stress obtained is 15.276Mpa which is maximum at the root of the gear

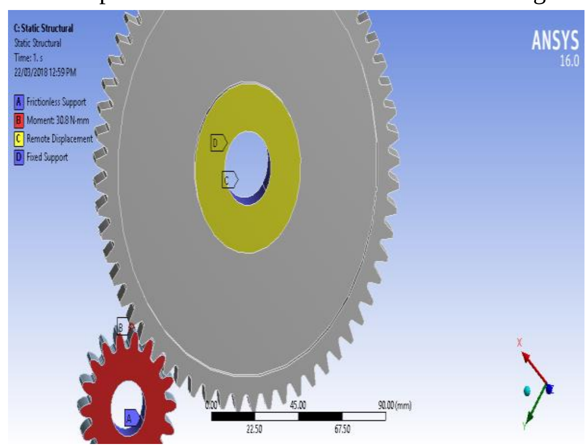


Fig5.8: Boundary condition for finding contact stresses by applying frictionless and fixed support as shown and moment of 17.41Nm

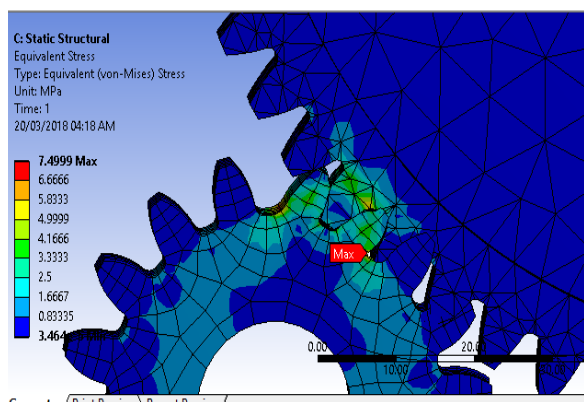


Fig5.9: The contact shear stress obtained is 7.499Mpa which is maximum at the root of the gear.

Material	Theoretical contact stress value(Mpa)	Analytical value(Mpa)	%Error
AISI1020	15.52	15.27	1.610
CI20	7.441	7.499	0.773

- The deformation analysis

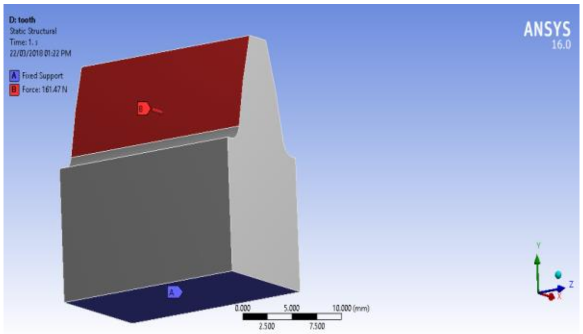


Fig5.10: Boundary condition is obtained by applying the max Force of 161.47N (2*T/PCD) and fixing the bottom of the tooth

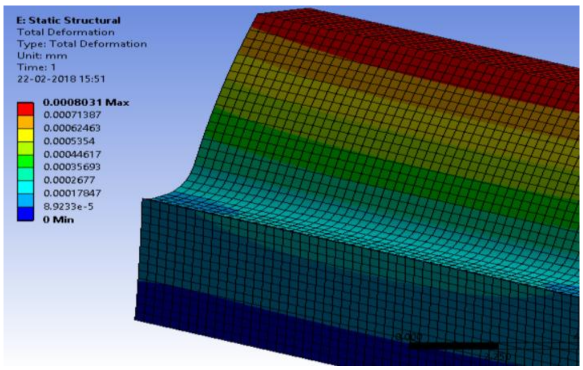


Fig5.11: This deflection is obtained as 0.0008031mm

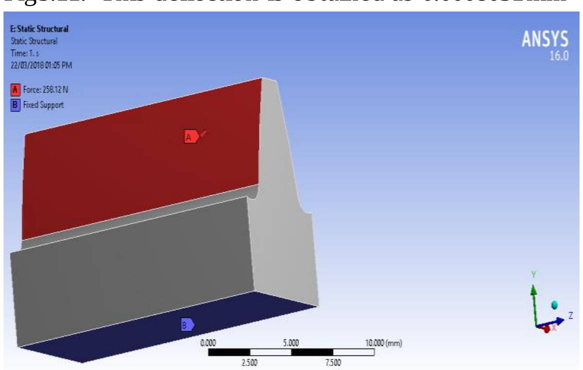


Fig5.12: Boundary condition is obtained by applying the max Force of 258.36N (2*T/PCD) and fixing the bottom of the tooth

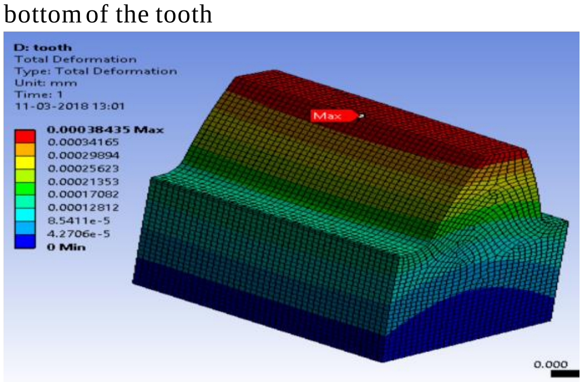


Fig5.13: This deflection is obtained as 0.0008031mm

Material	Theoretical Deflection (mm)	Analytical Deflection (mm)	%Error
AISI1060	0.000890	0.000803	9.775
CI 20	0.000394	0.000384	2.538

- From the above comparison the error for bending stress is in the range of 0 – 2 % and contact stress is 0 – 2% and for Deflection 2 – 10%. The value of deflection observed are 0.000803mm and 0.000394mm, which is very negligible to cause the failure. Because usually when the deformations are in range of 2-5mm it would be severe problem. Hence we can say that the design is safe.

6 CONCLUSIONS

- The assembly designed on CATIA V5 software was imported to ANSYS16.0 where structural analysis has been performed by applying the proposed material properties, boundary conditions and loads as discussed in previous sections.
- The theoretical maximum contact stress calculated by using hertz equation, maximum bending stress calculated by Lewis equation and the deformation calculated by castigliano's theorem as well as the finite element analysis done on ANSYS16.0 was found in good agreement.
- By viewing the results obtained in the project, it can be said that the gears can withstand the proposed load with a factor of safety as 2. So, thereafter the designed model can be manufactured or fabricated with optimum testing.

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