DESIGN AND VALIDATION OF TWO STAGE DIFFERENTIAL GEARBOX WITH SHIFTER MECHANISM FOR AN ELECTRICALLY DRIVEN VEHICLE

Rushikesh Kumbhar[1], Ashutosh Kumbhar[2], Tushar Bhoge[3]

[1,2,3]*Student, Mechanical Engineering, RMD Sinhgad School of Engineering, Maharashtra, India*

Abstract – *The Government of India has recently announced the National Electric Mobility Mission Plan, which sets ambitious targets for electric vehicle development in India. Electric rickshaw have been becoming more popular in some cities since 2008 as an alternative. In study about the Electric vehicles in market, it is generally seen that when the vehicle stars to climb the gradient road it requires more torque than rated torque. This tends consume more energy of the battery which further lowers the performance of the vehicle. During inclined road conditions, reduction ratio of recent vehicle reaches to 8:1 but in actual practice, but in actual practice more torque is required to maintain the performance of the vehicle. In order to minimize this problem of conventional differential gearbox, design of an alternative gearbox without hypoid gear will be more beneficial. This arrangement will include parallel arrangement of motor shaft and differential having total reduction ratio of 16:1.*

Key Words: Reduction ratio, helical gear, bevel gear, helix angle, module.

1. INTRODUCTION

Zero emission vehicles or Electric vehicle have been the one prominent answer to the rising pollution in this century. It has main source of onboard batteries and does not emit any of the hazardous gases. Development of Evs are mainly due to impact of fuel based vehicle on the environment along with the rising crude oil prices. Considering these factors, Central Government of India which announces the National Electric Mobility Mission Plan. This provides the guide-way for development of electic vehicles in Indian Automobile Market which aims for 5-7 millions of EVs on road by 2020.

BEVs are not widely accepted by the public now, because of the high price, limited driving range and long charging time. Currently as we have seen due to limited conventional fuels, there is need of an alternative. Electric vehicles seem to be a good option. In order to increase the acceptance of BEVs many researches are being done and they focus on overcoming technical barriers such as battery technology limitations, charging infrastructure problems and improving electric motor and power train performance. To improve power train performance conventional differential gear box are to be modified with high torque transmission ratio.

1.1 Problem Statement

When we studied about recent electric vehicles in market, we found that at the time when the vehicle starts to climb on inclined road it requires more torque than rated torque. This tends to consume more energy of battery. So it also tends to lower the performance of the vehicle. During inclination the reduction ratio of recent vehicle reaches almost up to 4:1 but actually, to maintain the performance of the vehicle the torque requirement is more.

In order to minimize this problem of conventional differential, we have decided to design an alternative arrangement of differential with higher reduction ratio and torque transmission capacity which does not need hypoid gear. This will increase the space for battery which will act as energy storage unit for this electric vehicle. The method chosen for manufacturing of gears will be cost reducing. Also the motor used for this vehicle will be less in costing. This in turn will lead to reduce the total cost of vehicle which is our main aim.

1.2 Objective

- To design and manufacture differential gear box for electrically driven vehicle with reduction ratio more than conventional differential gear box.
- To optimize the size of differential gear box.

1.3 Methodology

- Problem Definition:
  During inclination the reduction ratio of recent vehicle reaches almost up to 4:1 but actually, to maintain the performance of the vehicle the torque requirement is more.

- Literature survey
  To refer the Research papers based on similar design topics in order to clarify the basic concepts and methodology.

- Mechanism
  To study the basic working mechanisms using research papers which will lay the guide-way for the successful design.

- Design

To design the two stage reduction gearbox based on wear strength or beam strength theory which also includes material and gear selection.

- Prototype Manufacturing and Testing
  To manufacture the prototype of designed gearbox from the vendor using 3D printing technique. Furthermore, shaft RPM should be tested using digital device to verify whether the successful reduction is achieved.

- Conclusion

1.4 Input Parameters

Specification of an electrically driven Vehicle

- Motor Specification = 900 W
- Input speed = 3000 RPM
- Input Torque = 2.864 Nm
- Output RPM on gradient track = 56.25 RPM
- Desired Gear Ratio on gradient track = 16:1
- Output RPM on non-gradient track = 225 RPM
- Desired Gear Ratio on non- gradient track = 4:1

2. MATERIAL SELECTION

Table 1 -: Material Properties

Material Properties	15Ni4Cr1	EN 19	20MnCr5
Yield Strength (MPa)	1125	755	750
Ultimate Tensile Strength (Mpa)	1500	1075	1000
Hardness (BHN)	650	277	400

While selecting material we have considered various factors such as Strength, Hardness, Weight, Cost and availability of material. Our requirements match with the properties of the material 20MnCr5, Henceforth material 20MnCr5 is selected for further processes.

3. GEAR SELECTION

The first step in the design of the gear drive is the selection of a proper type of gear for a given application. The factor that are consider for deciding the type of gear and generally layout of shaft, speed reduction, power to be transmitted, input speed and cost. Spur and helical gear are used when the shaft are parallel. When the shaft intersects at right angles, bevel gears are used. Worm gear is recommended when the axes of shaft are perpendicular and non-intersecting.

The speed reduction or velocity ratio for a single pair of spur and helical gear is normally taken as 6:1. On rare occasions, this can be raise to 10:1. When the velocity ratio increase, the size of the gear increase. This results in increase in the size of the gearbox and the material cost increase. For high speed reduction, two stage or three stage construction are used. The normally velocity ratio for the pair of bevel gear is 1:1, which can be increased to 3:1 under certain circumstances. For high speed reduction, worm gear offer the best choice the velocity ratio in their case is 60:1, which can be increased to 100:1. They are widely used in material handling equipment due to this advantage.

In our proposed system we are going to use three pairs of helical gears and two pairs of bevel gears. Out of all these gears, first helical gear will be mounted on motor shaft and it will mesh to another helical gear. This will be the first stage reduction. Next two helical gears will mesh for second stage reduction, Pinion is compound with Gear of 1st stage and gear of the second stage will include two pairs of bevel gears inside it. The dimensions of gears are finalized from a number of iterations; those iterations were done considering all standard values of modules, pitch diameters, materials, etc. After performing some iteration, suitable values as per our requirement were finalized for manufacturing. By using the above mentioned gears we can get our desired.

4. DESIGN CALCULATION

General design methodology is as follows

- Calculation of Lewis form factor for pinion and gear.

- Calculation of Bending strength and wear strength in term of module (m) for the pinion or gear whichever has the less value of above calculated Lewis form factor.

- Further design must be based on wear strength or bending strength whichever has lower calculated value. (based on bending strength in our case)

- Calculation of pitch line velocity.
  $v = \frac{(\pi * dp * Np)}{(60000)}$, where dp = diameter of pinion (in terms of module.)

Np = Pinion RPM

- Calculation of tangential force Pt. (in terms of module)

$$Pt = \frac{Input\ Power}{Pitch\ line\ Velcoity}$$

- Calculation of effective load Peff.

$$Peff = \frac{Cs \times Pt}{Cv}$$

Where, Cs=1.25(assumed value for heavy loading)

$$Cv = \sqrt{\frac{5.6}{5.6+\sqrt{2.7303\ m}}}$$

(for fine grinding finishing)

- Module Calculation using the following equation

$$Beam\ Strength = FOS\ (assumed) \times Peff$$

Design of certain components such as cam, followers, gears is based on calculation of contact stresses by Hertz theory. Hence, recommended factor of safety is between 1.5 to 2.5.

- Calculations of addendum, dedendum, pitch circle diameter, gear width, tangential force, beam strength using calculated module.

- Calculation of actual effective load.

$$Peff = (Cs \times Pt) + Pd$$

Where, $Pd = \frac{21\ v\ [b\ C \cos^2 \psi + Cs\ Pt]\cos\psi}{21\ v + \sqrt{b\ C \cos^2 b\ C \cos^2 \psi + Cs\ Pt}}$

- Calculation of actual FOS.

$$Beam\ Strength = FOS\ (actual) \times Peff$$

For safe design, $FOS_{actual} > FOS_{assumed}$

4.1 First Stage Reduction

Number of teeth on pinion= Z_p= 16

Reduction ratio G_1= 4:1

$G_1 = Z_g / Z_p$

Number of teeth on Gear = Z_g = 64

Helix angle = $\Psi = 23^0$

Pressure angle= $\alpha_n = 20^0$

Sut = 1000 N/mm^2

BHN = 400

Virtual Number of teeth on Pinion=$Z_p' = (\frac{Zp}{\cos^3 \Psi})$ = 18.88

Virtual Number of teeth on Gear=$Z_g' = (\frac{Zg}{\cos^3 \Psi})$ = 75.53

Permissible Stresses for pinion and Gear

$\sigma b_g = \sigma b_p$ = (1000 / 3) = 333.333 MPa.

Lewis Form factor

i. For Pinion

$Y_p' = 0.484 - (\frac{2.87}{Zp'}) = 0.3473$

ii. For Gear

$Y_g' = 0.484 - (\frac{2.87}{Zg'}) = 0.449$

$(\sigma_b Y')_p$ = 115.766 N/mm^2 and $(\sigma_b Y')_g$ = 149.666 N/mm^2

$(\sigma_b Y')_g > (\sigma_b Y')_p$

Bending Strength

$S_b = m_n Y_p' \sigma_b b$

$b = 10 * m_n$

Y_p' =0.331

σ_b = 333.333 MPa

$S_b = 1103.32\ m_n^2$ N

Wear Strength

$S_w = (\frac{Q\ K\ dp\ b}{cos^3 \Psi})$

$K = 0.16\ [\frac{BHN}{100}]^2 = 2.56$

$Q = (\frac{2\ Zg'}{Zp'+Zg'}) = 1.6$

$d_p = m\ Z_p = 16 m_n$

$b = 10\ m_n$

$S_w = 773.44\ m_n^2$ N

$S_b < S_w$

Since Wear strength is lesser than Bending strength. So our design is based on Wear strength theory.

$v = \frac{(\pi * dp * np)}{(60000)} = 2.51\ m_n$

$P_t = (P / v) = (358.09 / m_n)$ N

$$C_v = \sqrt{\frac{5.6}{5.6+\sqrt{2.7303\ mn}}}$$

(∵It has high accuracy, finished by shaving followed by fine grinding)

$Peff = \frac{Ka \times Km \times Ft}{Kv}$ N

Table 2: First Stage Design Summary

Sr. No.	Parameters	Pinion	Gear
1	Module	2 mm.	2 mm.
2	Number of Teeth	16	64
3	Helix Angle	23^0	23^0
4	Pitch Circle Diameter	35 mm.	139 mm.
5	Width	20 mm.	20 mm.

Now, assuming Ka=Km=1 for moderate speed operation

Kv = 3 / (3+v)

Thus,

Peff = $\frac{1074.27+898.8mn}{3mn}$

S_b = FOS × P_{eff}

1157.665 m_n^2=1.5 × $\frac{1074.27+898.8mn}{3mn}$

$m_n = 1.09 \approx 1.25$ mm

$d_p = \approx 22$ mm

$d_g = \approx 87$ mm

b = 12.5 mm

$\phi_p = m_n + 0.25\sqrt{dp}$

$\phi_g = m_n + 0.25\sqrt{dg}$

$e_p = 3.2 + 0.25\ \phi_p = (3.8)\ \mu m$

$e_g = 3.2 + 0.25\ \phi_g = (4.09)\ \mu m$

C = 11400 e = 89.74 N/mm

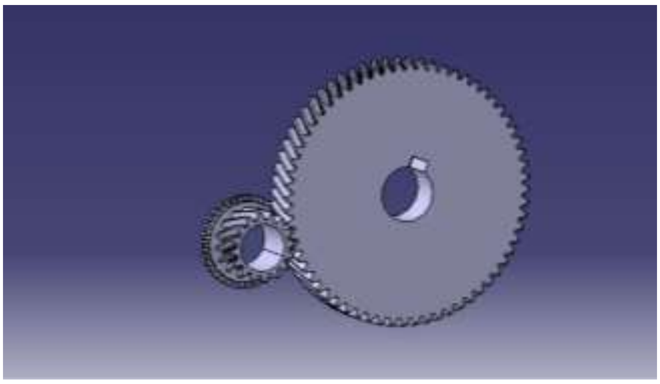

Fig -1: First Stage Gear Assembly

$P_d = \frac{21\ v\ [b\ C \cos^2 \psi + Cs\ Pt\ ] \cos \psi}{21\ v + \sqrt{b\ C \cos^2 b\ C \cos^2 \psi + Cs\ Pt}}$

P_d = 743.17 N

P_{eff} = Pt + Pd = 1029.64 N

S_b = FOS × P_{eff}

∴ FOS = 1.17

∵ FOS = 1.17 < 1.5

Since calculated FOS is less than assumed FOS. Hence, design is not safe. Hence, we will now increase the normal module for safe design. Consider the module 2 mm. Performing the above calculation with module value of 2 mm, we get results as per table 2.

4.2 Second Stage Reduction

By following the similar methodology as done earlier, calculations for second stage have been performed. Following are the specification obtained after the design.

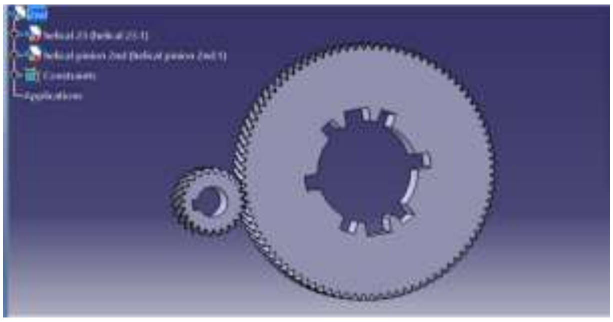

Fig 2: Second Stage Gear Assembly

Table 3: Second Stage Design Summary

Sr. No.	Parameters	Pinion	Gear
1	Module	1.25 mm.	1.25 mm.
2	Number of Teeth	23	92
3	Helix Angle	23^0	23^0
4	Pitch Circle Diameter	32 mm.	125 mm.
5	Width	12.5 mm.	12.5 mm.

4.3 Bevel Gear Selection

Bevel gears have been selected as per our requirement from the standard size available from the market for the design purpose. Following are the specification of the selected bevel gear pair.

Table 4: Bevel Gear Specifications

Sr. No.	Parameters	Pinion	Gear
1	Module	4 mm.	4 mm.
2	Number of Teeth	10	14

3	Pitch Circle Diameter	18 mm.	70 mm.
4	Width	20 mm.	20 mm.

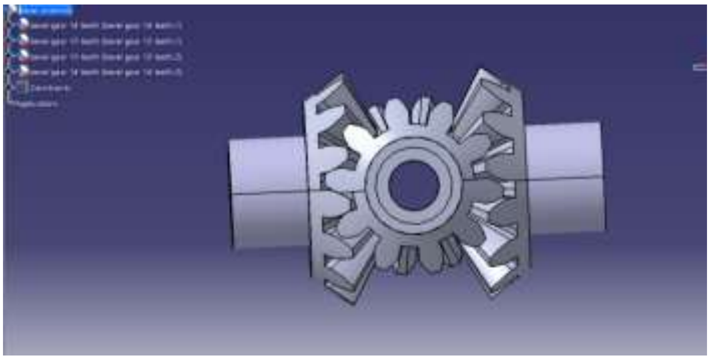

Fig 3: Bevel Gear Assembly

4.4 Hub and Sleeve Calculation

By following the similar methodology as done earlier, calculations for hub and sleeve design have been performed. Following are the specification obtained.

Table 5: Hub and Sleeve Design Summary

Sr. No.	Parameters	Hub	Sleeve
1	Module	1 mm.	1 mm.
2	Number of Teeth	36	36
3	Pitch Circle Diameter	23.8 mm.	23.8 mm.
4	Width	10 mm.	20 mm.

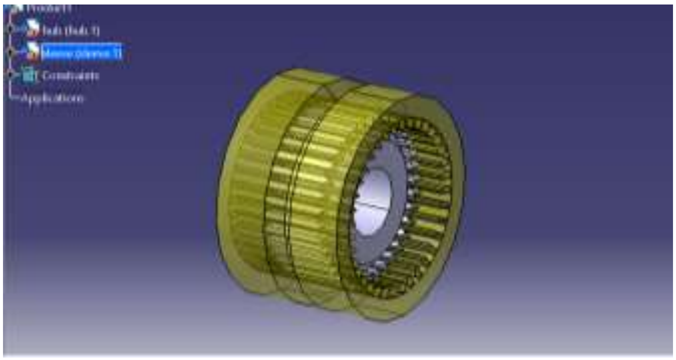

Fig 4: Hub and Sleeve Assembly

4.5 '1:1' Gear Pair Calculation

By following the similar methodology as done earlier, calculations for this gear pair have been performed. Following are the specification obtained.

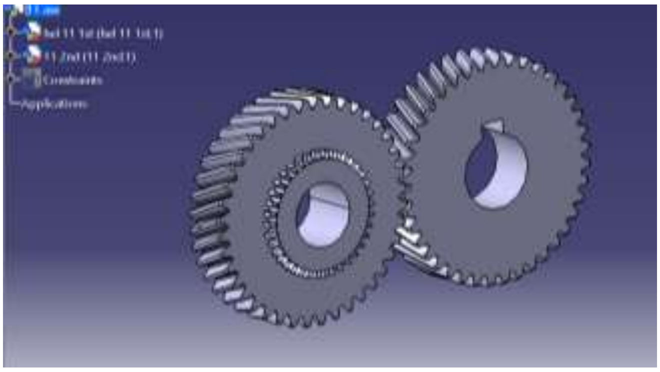

Fig 5: '1:1' Gear Pair Assembly

Table 6: Bevel Gear Specifications

Sr. No.	Parameters	Pinion	Gear
1	Module	1 mm.	1 mm.
2	Number of Teeth	40	40
3	Pitch Circle Diameter	43.8 mm.	43.8 mm.
4	Width	10 mm.	10 mm.

5. FORCE ANALYSIS

Stage I (A)

1. Tangential Load

$P_t = \frac{P}{v} = \frac{900}{2.73032} = 329.632$ N

2. Radial Load

$P_r = P_t \tan \emptyset = P_t \frac{\tan \emptyset n}{\cos \mu} = 130.337$ N

3. Axial Load

$P_a = P_t \tan \Psi = 139.920$ N

Stage I (B) (1:1)

1. Tangential Load

$P_t = \frac{P}{v} = \frac{900}{2.73032} = 329.632$ N

2. Radial Load

$P_r = P_t \tan \emptyset = P_t \frac{\tan \emptyset n}{\cos \mu} = 130.337$ N

3. Axial Load

$P_a = P_t \tan \Psi = 139.920$ N

Stage I (C) (Hub)

1. Tangential Load

$P_t = \frac{P}{v} = \frac{900}{3.6914} = 243.8118$ N

2. Radial Load

$P_r = P_t \tan \emptyset = P_t \frac{\tan \emptyset n}{\cos \mu} = 96.4039$ N

3. Axial Load

$P_a = P_t \tan \Psi = 103.4914$ N

Stage II

1. Tangential Load

$P_t = \frac{P}{v} = 733.7916$ N

2. Radial Load

$P_r = P_t \tan \emptyset = P_t \frac{\tan \emptyset n}{\cos \mu} = 267.078$ N

3. Axial Load

$P_a = P_t \tan \Psi = 311.476$ N

6. SHAFT CALCULATIONS

6.1 Methodology for Shaft Calculations

- Calculation of allowable stresses considering the key way effect and selected material for the shaft.
- Calculation of the length of shaft according to following formula.
  $L = B + (2l_2) + (2\partial) + b + b_1 + c$
  Where,
  B = Bearing Width
  l_2 = Distance of bearing from wall
  ∂ = Distance of rotating part from inner wall
  b = face-width of pinion
  b_1= face-width of gear
  c = Distance between adjacent rotating parts
- Drawing of loading diagram, shear force diagram and bending moment diagram for the respective shafts.
- Calculation of horizontal reactions, vertical reactions, resultant bending moment, torque calculations and finally diameter based on above drawn diagram.

6.2 Input Shaft Calculations

Total Length of Shaft

L = 10 + 2(10) + 2(10) + 20 + 12 + 10 = 92 mm.

Hub Shaft Reaction Forces:-

Vertical Loading

$\sum M_A = 0$

7798.5922 – 96.4039 + 131.868(21.5+21.5+24.5) + 92 R_B = 0

R_B = 180.4704 N

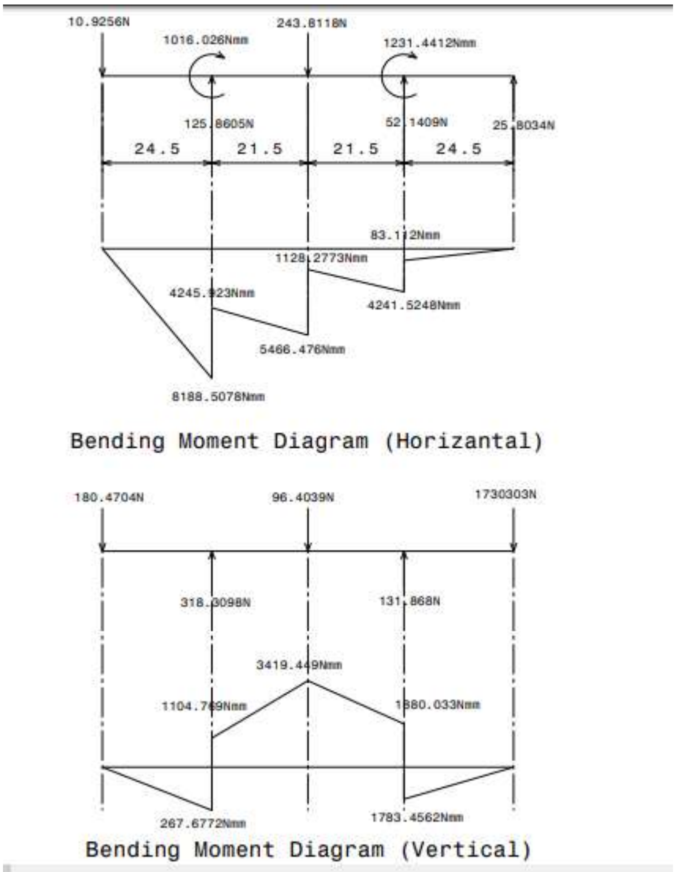

Fig 6: BMD for Input Shaft

$\sum F_A = 0$

$\therefore R_A = 173.303$ N

Vertical Bending Moment:-

$BM_A = BM_B = 0$

$BM_{CL} = -R_A (24.5) = - 4245.923$Nmm.

$BM_{CR} = -R_B(67.5) + 131.868(43) - 96.4039(21.5) = -8188.578$ Nmm

$BM_{DL} = -R_A (46) + 318.3098 (21.5) = - 1128.2773$ Nmm

$BM_{DR} = -R_B(46) + 131.868 (21.5) = -5466.4764$ Nmm

$BM_{EL} = -R_A(67.5) + 318.3098 (43) - 96.4039(21.5) = -83.11245$ Nmm

$BM_{ER} = -R_B(24.5) = -4421.5248$ Nmm.

Horizontal Loading

$-R_A + R_B + 125.8605 - 243.8118 + 52.1409 = 0$

$\sum M_A = 0$

$92\ R_B = 7059.717$

$R_B = 76.73605$ N

$R_A = 10.9256$ N

Horizontal Bending Movement:-

$BM_A = BM_B = 0$

$BM_{CL} = -R_A\,(24.5) = -267.06772$ Nmm.

$BM_{CR} = R_B(70.5) - 1231.4412 + 52.1409(43) - 243.8118(21.5) = 1104.769$ Nmm.

$BM_{DL} = -R_A(46) + 1216.026 + 125.860(21.5) = 3419.44915$Nmm.

$BM_{DR} = R_B(46) + 52.1409(21.5) - 1231.4412 = 3419.44915$Nmm.

$BM_{EL} = -R_A(67.5) + 125.8605(43) + 1216.026 - 243.8118(21.5) = 1783.4562$Nmm

$BM_{ER} = R_B(24.5) = 1880.033$Nmm.

Hub Torque Calculations:-

Torque = T = 2864.7889 Nmm

$M = BM_C = \{ (BM_C)_v^2 + (BM_C)_H^2 \}^{(1/2)} = 6.030 \times 10^3$ Nmm.

$T_e = \{ (K_bM)^2 + (K_tT)^2 \}^{(1/2)} = 9.045 \times 10^3$ Nmm

$\tau_{all} = \frac{16}{\pi d^3} (T_e) = 46066.7227 / d^3$ N/mm² .

For Material – 20MnCr5

Shear Stress

S_{yt} - 750 N/mm²	S_{ut} - 1000 N/mm²
$\tau = 0.75 \times 0.3 \times S_{yt}$	$\tau = 0.75 \times 0.18 \times S_{ut}$
$\tau = 168.75$ N/mm²	$\tau = 135$ N/mm²

Taking minimum value of shear stress,

$\therefore \tau = 135$ N/mm² .

$\tau_{all} = \tau_{min} = 135 = 46066.7227 / d^3$

$\therefore d = 6.9879 \approx 9$ mm.

Thus, diameter for input shaft is calculated. Similar methodology is followed to obtain diameter of intermediate and output shaft.

7. BEARING SELECTION

7.1 Methodology for Bearing Selection

- Calculations of bearing reactions.
- Selection of the bearing on the trial and error basis.
- Calculation of the radial and tangential forces considering the loading conditions and bearing mounting type. (face to face or back to back)
- Check for the selected bearing based on equivalent dynamic load and dynamic load capacity.
- Selected bearing is safe if the calculated dynamic load capacity is less than that of standard dynamic load capacity of the selected bearing.

7.2 Bearing Selection for Input Shaft

d = 9 mm

n = 3000 rpm

$L_{10h} = 15000$ hrs

$L_{10} = \frac{60 \times n \times L10h}{10\hat{}6} = 2700$ million revolutions.

Reactions on the bearing due to shaft

$R_{AV} = R_{BV} = 48.2019$ N

$R_{AH} = R_{BH} = 121.9059$ N

Resultant of respective Reaction forces,

$\therefore R_A = R_B = 131.0896$ N

Bearing required in Radial reaction,

$R_A = Fr_A = 131.0896$ N

$R_B = Fr_B = 131.0896$ N

Random bearing selected for shaft is 699Z single row deep groove ball bearing Specifications of bearing 699Z

Bearing load rating (C) = 2480 N

Factors X = 1 & Y=0

e = 0.22

As there is no application of thrust force,

$\therefore K_a = F_{aA} = F_{aB} = 0.$

As, $(F_a / F_r) < e$.

$\therefore P_a = Fr_A = 131.0896$ N and $P_b = Fr_B = 131.0896$ N

$\therefore P = P_a = P_b = 131.0896$ N

$L_{10} = \left(\frac{C}{P}\right)^3$

$2700 = \left(\frac{C}{131.0896}\right)^3$

$\therefore C = 1825.3922$ N

C = 1825.3922 N < 2480 N , ($\because$ 2480 N is bearing load rating of selected bearing)

Hence selected bearing is safe.

Thus, bearing is selected for the input shaft. Similar methodology is followed in order to select the bearing for intermediate and output shaft.

8. CASING DESIGN

Characteristic dimension N of the casing,

$N = \frac{(2L+B+H)}{4}$

Where,

N: Characteristic dimension;

L: Overall length of the casing = 217 mm

B: Overall width of the casing = 92 mm;

H: Overall height of the casing =72 mm.

$\therefore$ N = 0.1495 m

Since Characteristic dimension (N) < 0.75 m

$\therefore$ Thickness of casing is,

t = 8 mm

9. EXPERIMENTAL VALIDATION

Since, we have manufactured 3D prototype of our designed differential gearbox, we have used motor input RPM as 200.

9.1 Results for Gradient Way without Turn

On the gradient way, we basically require high torque and low RPM. This can be achieved by shifting the sleeve towards pinion of the input shaft thereby achieving the total reduction of 16:1.

Fig 7: Tachometer reading showing reduction of 16:1

9.2 Results for Gradient Way with Turn

Under the differential action while cornering, outer wheel rotates with higher rpm than that of inner wheel in order to cover relatively larger distance. To achieve this, we held one of the ends of output shaft which resulted into higher rpm of the other end.

Fig 8: Tachometer reading showing higher RPM at one end

9.3 Results for Non-Gradient Way without Turn

For the normal way, high speed and low torque is required for comfortable ride. This can be achieved by shifting the sleeve towards the gear of the input shaft thereby achieving the total reduction of 4:1.

Fig 9: Tachometer reading showing reduction of 4:1

9.4 Results for Non-Gradient Way with Turn

As discussed earlier, while cornering under differential action, the outer wheel must rotate with high rpm covering larger distance than that of inner wheel. For experimental validation, we held one of the ends of output shaft which resulted into higher rpm of the other end.

Fig 10: Tachometer reading showing higher RPM at one end

10. CONCLUSION

As per the idea of the project to increase the torque transmission capacity on gradient way we have designed the differential gear box. After the case study of various Research papers, various parameters were considered and calculations were processed. After that market survey was done and we selected 20MnCr5 material for our gearbox. The selection of material and the manufacturing processes has been decided using the data collected in the survey, as we have completed the calculation, design and modeling of Differential Gear Box and further we headed towards prototype manufacturing of differential gear box.

As per our problem statement, required torque on gradient way should be greater than torque required on a normal way. Since we are using the same differential gear box on both gradient and normal way, we introduced a shifter mechanism which will give us the different rotations of wheels and torque requirement on gradient and normal way. So, with this we conclude our differential gearbox for our problem statement and advanced mechanism as shifter.

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