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DESIGN OF DIFFERENTIAL GEAR BOX

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Abstract: This project presents the design of a differential gearbox for the TATA ACE small commercial vehicle. The gearbox is designed to transmit power smoothly while allowing the rear wheels to rotate at different speeds during turning. Design calculations were carried out using standard equations for strength, wear, and load capacity based on engine specifications and material properties collected from Tata Motors data and published research. The gearbox components were designed using 20MnCr5 steel due to its high strength and wear resistance. A 3D model was developed in SolidWorks 2025, and static structural analysis was performed in ANSYS to evaluate stress and deformation. The analysis results showed that the maximum stress and deformation were within safe limits, confirming the structural safety and reliability of the design. The study also indicates that the gearbox is slightly overdesigned, which improves durability and operational safety. Future improvements may include thermal analysis, lightweight manufacturing methods, and adaptation for electric and hybrid vehicles.

Index Terms - Solid works, Tata Ace, Differential gear box, Bevel gear, Static Analysis Novelty: The novelty of this project is designing a differential gearbox specifically for the TATA ACE using real vehicle data, verified through static structural analysis to ensure practical strength, reliability, and manufacturability.

I.INTRODUCTION

The differential gearbox is an important component of a vehicle transmission system that transmits power from the engine to the wheels while allowing them to rotate at different speeds during turning. During a turn, the outer wheel travels a greater distance than the inner wheel, and the differential mechanism enables this speed variation to ensure smooth vehicle operation and reduce tyre wear. In small commercial vehicles such as the Tata Ace, the differential gearbox plays a significant role in maintaining stability and load-carrying performance under varying road and traffic conditions. Since the vehicle frequently operates in urban areas with continuous stops, turns, and changing loads, the gearbox must be compact, durable, and efficient.The objective of this work is to design a differential gearbox suitable for the Tata Ace using actual engine power and torque data. The gearbox components, including gears, shafts, and housing, were designed and analysed using SolidWorks and ANSYS to evaluate strength, stress distribution, and overall performance. The design was verified through static structural analysis to ensure safe and reliable operation under working conditions. This study focuses specifically on the Tata Ace vehicle, providing a practical and application-oriented gearbox design for small commercial vehicles.

II.LITERATURE

R. Karthick, V. Mohankumar, et al. (2018). “Design and Analysis of Differential Gear Box.” This paper designed and tested a differential gearbox for small utility vehicles. The gearbox model was made in SolidWorks and analyzed in ANSYS. Different materials like stainless steel, aluminum, magnesium, and structural steel were compared. The study found that magnesium alloy is the best choice because it gives good strength and reduces weight [1].Patoju S. S., Pudi T. K., et al. (2022). “Design and Analysis of Differential Gear in Automobile.” This paper focuses on designing and analyzing a differential gear system for a four-wheeler vehicle. The gearbox was modeled using SolidWorks and tested in ANSYS for different torque levels. Materials such as grey cast iron, aluminum alloy, titanium alloy, and structural steel were compared. The results showed that titanium alloy gave the best strength and safety factor, making it suitable for lightweight and durable differential design [2]. U. V. Mane, S. S. Choure (2021). “Modeling, Design & Analysis of Differential Gear Box and Its Housing through FEM, SolidWorks & ANSYS Benchwork 14.0.” This paper explains how a differential gearbox and its housing were designed and tested using SolidWorks and ANSYS software. The goal was to study how vibration and stress affect the gearbox when used in heavy vehicles. The authors found the natural frequencies and stress levels to make sure the gearbox can handle real driving conditions without damage. The study showed that with a good design and suitable material, vibration problems and gear failures can be reduced. Overall, the research helps in making the gearbox more strong, reliable, and long-lasting [3].

Jain R., Goyal P. (2016). “Design and Analysis of Gear-Box Using Spur Gear and Eliminating the Differential Unit.” This paper focuses on designing a gearbox using spur gears instead of bevel gears and removing the differential unit to make the system simpler and lighter. The gearbox was modeled in Creo Parametric and analyzed in ANSYS 14.0 using two materials — 15NiCr1Mo15 and SCM415. The stress and deformation results showed that both materials are strong enough and perform safely under working conditions. By eliminating the differential, the design reduces weight and improves efficiency without affecting strength. The study proves that spur gears can replace differential systems in some applications, leading to easier manufacturing and better fuel efficiency [4].Nikhil K. Toke, Girish C. Kurkure, et.al (2018). “A Review on Design and Development of Modified Differential Gearbox,” explains that using Aluminium alloy and spur gears can make the gearbox lighter and cheaper. They also found that composite materials like glass-filled polyamide improve strength and reduce friction. Overall, they suggest combining different differential designs to solve common problems and improve vehicle performance [5].Sumair Sunny, Siddhesh Ozarkar, et. al (2014). “Design of an Automotive Differential with Reduction Ratio Greater Than 6,” explains how a worm and bevel gear setup can deliver more power while keeping the system compact and strong. Using Altair Hyper Mesh analysis, they showed the design is safe and efficient for heavy vehicles. Although it has slightly lower efficiency due to friction, it’s well-suited for trucks and buses that need more strength and durability [6].Subhajit Konar and Vijay Gautam (2019). “Design and Analysis of an Open Differential” explains how an open differential helps vehicles turn smoothly by allowing wheels to rotate at different speeds. The authors designed the system using CAD software and tested it with ANSYS to check its strength. They used AISI 4340 steel for good durability and found that the design could handle stress safely. The study showed that the results from computer analysis matched well with theoretical calculations, proving the design is strong, safe, and reliable for real vehicle use [7]. So, referring to them, in this paper, we taking effort to design a gearbox for a given vehicle, say TATA ACE. We will be referring TATA’s official website for all the numerical data.

III.RESEARCH GAP

3.1 Focus on small heavy vehicles

The previous studies mostly focus on a heavy vehicle. This study focuses on a Tata Ace, which is a small commercial vehicle, and designs a gearbox that suitable its real engine and torque needs.

3.2 Practical verification

Many old projects stopped at theory or simulation. This study checks the calculated results with the actual Tata Ace gearbox, proving that the design is safe and realistic.

3.3 Complete design study

Previous work mostly studied gears only. Here, the team analyzed gears, shafts, and bearings together using real materials like 20MnCr5 steel to make the design stronger and more reliable.

3.4 Industry related approach

Instead of only academic design, this project uses methods and materials that industries actually use, making the gearbox ready for real manufacturing and testing. From these gaps, we came up with the problem statement, which is explained in the next section.

IV.PROBLEM STATEMENT

Many of the gearbox designs are made for a heavy vehicle and they don't suitable for small vehicle like the Tata Ace. There's also a lack of practical designs that are tested with real vehicle data.SolutionThis study designs a differential gearbox for the Tata Ace using real engine and torque values. The design was checked for strength and matched with actual Tata Ace dimensions, making it safe, reliable, and suitable for real use.

V.OBJECTIVES

To design the differential gearbox and that can be used in a small vehicle like the TATA ACE.
To make sure the gearbox allows the rear wheels to rotate at different speeds while turning.
To calculate and check the strength of gears and shafts so they can handle the engine's torque safely.
To choose suitable materials and sizes for the gears and shafts to avoid failure during operation.
To compare our design and results with the actual TATA ACE gearbox and see how accurate our design is it.
To select the suitable bearings and confirm that the overall design is safe, strong, and reliable.

VI.METHODOLOGY

In this project, we have designed a differential gearbox for a light weight commercial vehicle, we are taking about the TATA ACE as a reference vehicle. The design process is included in the following several stages, including data collection, calculations & modeling and simulation.

FIG1 TATA ACE Input data

SrNo.	Property	Value
1.	Vehicle	TATA ACE
2.	Engine Power	26 KW @ 3750 RPM
3.	Max torque	85Nm @ 2250 RPM
4.	Gear material	20 Mn Cr 5
5.	Material yield strength	760 MPa
6.	Brinell hardness no.	402
7.	Factor of safety	2
8.	No. of teeth on pinion	12

For this project, we have chosen the TATA ACE vehicle as a reference vehicle for designing the differential gearbox. And for that purpose, we have considered required input data such as engine power, torque, and material properties were collected from reliable sources, including Tata Motors' official website and published research papers. The gears were assumed to be made of 20MnCr5, a material widely used in automotive applications because of its high strength and wear resistance. A factor of safety of 2 was applied throughout the design to ensure durability and reliability. All gear dimensions, tooth profiles, and modules were selected based on standard values available in the market to keep the design practical and manufacturable.

Calculations

6.2.1 Design for pinion and ring bevel gear

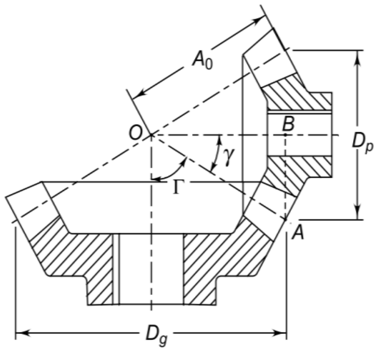


Fig. 1 Schematic of mating bevel gear

$$\tan\gamma = \frac{Z_p}{Z_g}$$

$$\tan\emptyset = \frac{Z_g}{Z_p}$$

Also, cone distance (A_0)

$$A_0 = \sqrt{(\frac{D_p}{2})^2 + (\frac{D_g}{2})^2}$$

Actual no. of teeth on the pinion = $Z_p = 12$ (Asumption)

Let, no. of teeth on formative gear be $Z_p{}'$

$$\frac{Z_p{}'}{Z_p} = \frac{2r_b}{D}$$

Tangential Force component, $P_t = \frac{2M_t}{D_p}$

Radial Force component, $P_r = P_t \tan \propto \cos\gamma$

Axial Force component, $P_a = P_t \tan \propto \sin\gamma$

Separating Force component, $P_s = P_t \tan \propto$

$$P_t = \frac{2M_t}{D_p} = \frac{85 \times 10^3 \times 2}{D_p}$$

$$D_p = m\,Z_p = 12\text{ m}$$

$$P_t = \frac{85 \times 10^3 \times 2}{12\text{ m}}$$

$$P_t = \frac{14.16 \times 10^3}{m}$$

Considering Service Factor $C_s = 2$

$$C_v = \frac{6}{6 + v}$$

$$v = \frac{\pi\,D_p\,N}{60} = \frac{\pi\,(12\text{ m})\,2250}{60}$$

Substituting,

$$C_v = \frac{6}{6 + \frac{\pi\,(12\text{ m})\,2250}{60}}$$

Effective Force,

$$P_{eff} = \frac{P_t\, C_s}{C_v} = \frac{170 \times 10^3 + 40 \times 10^3 m}{6\ m}$$

Simplifying,

$$P_{eff} = \frac{28.3\ \times 10^3}{6\ m} + 6.67\ \times 10^3$$

6.2.1.1 **Beam Strength**

$$S_b = m\, b\, \sigma_b\, Y\, [1 - \frac{b}{A_0}]$$

Considering, $\frac{b}{A_0} = \frac{1}{3}$ & $b = 10\ m$

$\sigma_b = \frac{S_{yt}}{FOS} = \frac{760}{2} = 380$ MPa

Lewis Form Factor, Y = 0.358

Substituting in beam strength equation, we get,

$$S_b = m\ (10\ m)\ (380)\ (0.358)\ \left[1 - \frac{1}{3}\right]$$

Simplifying,

$$S_b = 1432\ m^2$$

$$S_b = P_{eff} \times FOS$$

$$1432\ m^2 = \left[\frac{28.3\ \times 10^3}{6\ m} + 6.67\ \times\ 10^3\right] \times 2$$

Multiplying by m^2 *and rearranging,*

$$6.67 \times 10^3\, m^2\ + 28.3 \times 10^3\, m - 716 = 0$$

Solving above quadratic equation for *m*, we get,

m = 4.26 & -0.025

Ignoring negative value of m, module m = 4.26 mm

Considering nearest standard value of module available in the market, module is 5 mm

Module,

m = 5 mm

From empirical relations,

Diameter of pinion gear $[D_p] = m\ Z_p = 5 \times 12 = 60$ mm

Face width of pinion gear $[b] = 10\ m = 50$ mm

Ratio between pinion and bevel ring gear [G] = 4.4

Number of teeth on ring bevel gear $[Z_G] = Z_p \times 4.4 = 54$

Diameter of teeth on ring bevel gear $[D_G] = m\ \times Z_G = 270$ mm

Pitch angle of pinion gear $[\gamma] =\ \tan -1 \left[\frac{Z_G}{Z_p}\right] = 12.5°$

Pitch angle of bevel ring gear $[\emptyset] = \tan -1 \left[\frac{Z_p}{Z_G}\right] = 77.47°$

Cone Distance $[A_0] = \sqrt{\left(\frac{D_p}{2}\right)^2 + (\frac{D_g}{2})^2}$

$$A_0\ = \sqrt{(\frac{60}{2})^2 + (\frac{270}{2})^2} = 137.3\ mm$$

$$\text{Checking } [\gamma] + [\emptyset] = 90°$$

So, our comes to be 89.97°, So, our design is correct

6.2.1.2 Wear strength

Wear strength for bevel gears is given by following relation:

$$S_w = \frac{0.75\ b\ Q\ D_p\ k}{\cos\gamma}$$

Ratio Factor Q is given by,

$$Q = \frac{2\ Z_G}{Z_G + Z_p\ \tan\gamma}$$

$$Q = \frac{2\ \times 12}{12 + 54\ \tan 12.5} = 1.28$$

Material constant is given by,

$$K = 0.16\ (\frac{BHN}{100})^2 = 1.3$$

Substituting the values in main equation,

$$S_w = \frac{0.75\ (50)(1.28)(1.3)\ (60)}{\cos 12.5} = 38341.5\ N$$

6.2.1.3 Dynamic load

Pitch line velocity of gear,

$$V = \frac{\pi\ D_p\ N}{60 \times 10^3} = 7.068\ m/s$$

Assuming, $C = 11400\ N/mm^2$ & $P_t = 2.83\ \times\ 10^3\ N$

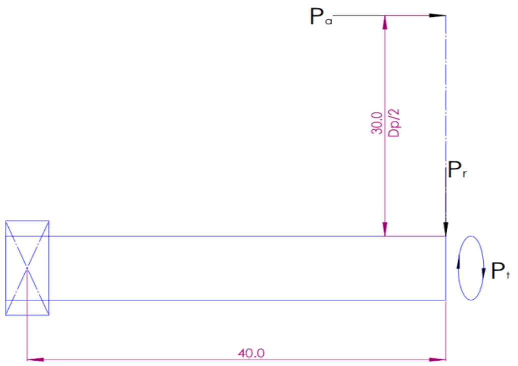
$$P_d = \frac{21\ v(Ceb + P_t)}{21\ v + \sqrt{(Ceb + P_t)}}$$

$$P_d = \frac{21\ (7.068)[(11400)(0.0125)(20) + (2.83 \times 10^3)]}{21\ (7.068) + \sqrt{[(11400)(0.0125)(20) + (2.83 \times 10^3)]}} = 3767.2\ N$$

6.2.1.4 Effective load

$$= P_d + C_s \times P_t$$
$$= 3767.2 + 2\ (170 \times 10^3)$$
$$= 37767.2\ N$$

From above value we can see that the Pd < Peff, Hence the design is safe.



6.2.2 Shaft design for pinion

Fig. 2 Free body diagram of loading conditions of the pinion shaft

Considering shaft diameter as a d,

Force acting on the gear,

$$P_t = \frac{2M_t}{D_p} = 2.83 \times 10^3 \text{ N}$$
$$P_a = P_t \tan \propto \sin\gamma = 222.9 \text{ N}$$
$$P_r = P_t \tan \propto \cos\gamma = 1005 \text{ N}$$

Equivalent bending moment on the shaft,

In horizontal plane:

Moment generated due to axial and radial force at the bearing end of the shaft in vertical plane

$$[m_v] = (P_r \times 40) + (P_a \times 30) = 43.887 \text{ kN.mm}$$

In vertical plane:

Moment generated due to tangential force at the bearing end of the shaft in horizontal plane

$$[M_h] = (P_t \times 30) = 84.9 \text{ kN.mm}$$

Equivalent bending moment:

Taking vectorial sum of $[Mv]$ & $[Mh]$ we can get Equivalent bending moment.

$$[M_{eq}] = \sqrt{[M_v]^2 + [M_h]^2}$$
$$[M_{eq}] = \sqrt{[43.887 \times 10^3]^2 + [84.9 \times 10^3]^2} = 95.57 \text{ kN.mm}$$

According to flexural equation,

$$\sigma_b = \frac{M_{eq}}{I} Y$$

Moment of inertia $\sigma_b = \frac{\pi d^4}{64}$

Substituting and rearranging,

$$\sigma_{b_{eq}} = \frac{3058240}{\pi d^3}$$

Tensile stress due to axial force,

$$\sigma_t = \frac{P_a}{\frac{\pi}{4} d^2}$$
$$\sigma_t = \frac{891.6}{\pi d^2}$$

Total normal component,

$$\sigma_x = \sigma_t + \sigma_{b_{eq}}$$
$$\sigma_x = \frac{891.6}{\pi d^2} + \frac{3058240}{\pi d^3}$$
$$\sigma_x = \frac{891.6\, d + 3058240}{\pi d^3}$$

Shear stress in shaft,

$$\tau = \frac{T_{eq}}{\pi d^3}$$
$$\tau = \frac{85 \times 10^3 \times 16}{\pi d^3}$$

As 20 Mn Cr 5 material is ductile, we will consider maximum shear stress theory for diameter calculation.

According to maximum shear stress theory,

$$\frac{0.5 \times S_{yt}}{2} = \sqrt{(\frac{\sigma_x}{2})^2 + (\tau)^2}$$
$$\frac{0.5 \times 760}{2} = \sqrt{(\frac{891.6\, d + 3058240}{2\,\pi\, d^3})^2 + (\frac{85 \times 10^3 \times 16}{\pi d^3})^2}$$

solving for d, we get,

$$d = 33.4 \text{ mm}$$

Rounding off to the nearest standard shaft size, we get, diameter of shaft to be 35 mm.

6.2.3 Design for planet & sun gear

The ratio between ring and pinion gear [G] = 4.4

So the torque at ring gear = G x pinion torque = 4.4 x 85 x10^3 = 374 x 10^3

Also, $Ao = \sqrt{(\frac{D_p}{2})^2 + (\frac{D_G}{2})^2}$

Diameter of sun gear = DP = 100 mm {Assumption}

Tangential force component, $P_t = \frac{2M_t}{D_p}$

Radial force component, $Pr = Pt \tan\alpha \cos\gamma$

Axial force component, $Pa = Pt \tan\alpha \sin\gamma$

Separating force component, $Ps = Pt \tan\alpha$

$$P_t = \frac{2M_t}{D_p} = P_t = \frac{2 \times 374 \times 10^3}{D_p}$$

$$D_p = m Z_p$$

$$Z_p = \frac{D_p}{m} = \frac{100}{m}$$

$$P_t = \frac{2 \times 85 \times 10^3}{100}$$

$$P_t = 1.7 \times 10^3$$

Considering Service factor, Cs=2

$$C_v = \frac{6}{6 + v}$$

$$v = \frac{\pi \times D \times N}{60}$$

$$= \frac{\pi (100)(2250)}{60}$$

$$= 11.77 \text{ m/s}$$

Substituting,

$$C_v = \frac{6}{6 + 11.77} = 0.34$$

Effective force,

$$P_{eff} = \frac{P_t C_s}{C_v}$$

$$P_{eff} = \frac{1.7 \times 10^3 (2)}{0.34}$$

Simplifying,

$$P_{eff} = 1.156 \text{ kN}$$

6.2.3.1 Beam Strength

$$S_b = m\, b\, \sigma_b\, Y\left[1 - \frac{b}{A_o}\right]$$

Considering b/AO=1/3 & b = 10 m

$$\sigma_b = \frac{S_{yt}}{FOS} = \frac{760}{2} = 380 \text{ Mpa}$$

Lewis's form factor, Y = 0.358

Substituting in beam strength equation, we get,

$$S_b = m\,(10m)(380)\,(0.358)[1 - \frac{1}{3}]$$

Simplifying,

$$S_b = 1432\, m^2$$
$$S_b = P_{eff} \times FOS$$
$$1432\, m^2 = 1.156 \times 10^3 \times 2$$

Solving for m and rearranging,

$module\ (m)$=3.23 mm

Considering the nearest standard module available in the market, module is 3 mm.

Module

$m = 4\ mm$

From empirical relations,

Diameter of pinion gear $[Z_p] = \frac{D_p}{m} = \frac{100}{4} = 25$

Face width of pinion gear [b] = 10× 4 =

Ratio between pinion & bevel ring gear [G] = 1

Number of teeth on ring bevel gear $[Z_G] = [Z_p]$ = 25

Diameter of teeth on ring bevel gear $[D_G] = m\, Z_G$ = 100 mm

Pitch angle of pinion gear [γ] = tan −1 $\left[\frac{Z_G}{Z_p}\right]$ = 45°

Pitch angle of bevel ring gear [φ] = tan −1 $\left[\frac{Z_p}{Z_G}\right]$ = 45°

Cone distance Ao = $\sqrt{(\frac{100}{2})^2 + (\frac{100}{2})^2}$ = 70.7 mm

Checking γ+ φ=90°

Our comes to be 90°, so the design is correct.

6.2.3.2 **Wear strength**

Wear strength for bevel gears is given by following relation,

$$S_w = \frac{0.75\, b\, Q\, D_p\, k}{\cos\gamma}$$

Ratio factor Q is given by,

$$Q = \frac{2\, Z_G}{Z_G + Z_p\ \tan\gamma}$$

$$Q = \frac{2\,(25)}{25 + 25\ \tan 12.5} = 1.65$$

Material constant is given by,

$$k = 0.16(\frac{BHN}{100})^2 = 1.3$$

Substituting the values in main equation,

$$S_w = \frac{0.75\ (40)(1.65)(1.3)(60)}{\cos 45}$$

$$= 5428.5\ \text{N}$$

6.2.3.3 Dynamic Load

Pitch line velocity of gear,

$$v = \frac{\pi\ D\ N}{60 \times 10^3} = 11.77\ \text{m/s}$$

Assuming C = 11400 N/mm2 and Pt = 1.7 x 103 N

$$P_d = \frac{21\ v(Ceb + P_t)}{21\ v + \sqrt{(Ceb + P_t)}}$$

$$P_d = \frac{21\ (11.77)(11400 \times 0.0125 \times 20 + 1.7 \times 10^3)}{21\ (11.77) + \sqrt{(11400 \times 0.0125 \times 20 + 1.7 \times 10^3)}}$$

$$P_d = 3572.84\ \text{N}$$

6.2.3.4 Effective load,

$$P_{eff} = P_d + C_s P_t$$

i.= $3572.84 + 2 \times 1.7 \times 10^3$

b. = 5272.84 N

From above value we can see that the Pd < Peff, Hence the design is safe

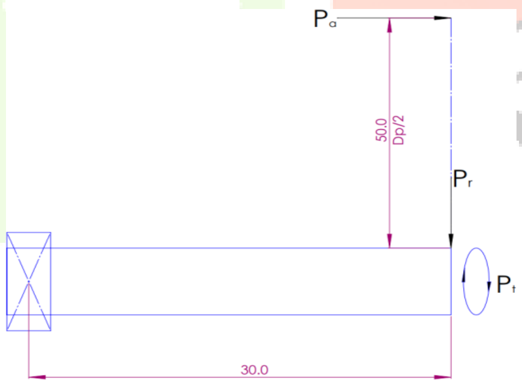
6.2.4 Shaft design for planet and sun gear

Considering the shaft diameter as d.

Force acting on the gear,

$P_t = \frac{2M_t}{D_p} = 1.7 \times 10^3$

$Pa = Pt \tan\alpha \sin\gamma = 437.5\ N$



$$P_r = P_t \tan \propto \cos\gamma = 437.5\ N$$

Fig. 3 Free body diagram of loading conditions of the sun gear shaft

Equivalent bending moment on the shaft,

In horizontal plane:

Moment generated due to axial and radial force at the bearing end of the shaft in vertical plane

$$[m_v] = (P_r \times 60) + (P_a \times 60) = 52.5\ \text{kN.mm}$$

In vertical plane:

Moment generated due to tangential force at the bearing end of the shaft in horizontal plane

$$[M_h] = (P_t \times 60) = 102\ \text{kN.mm}$$

Equivalent bending moment:

Taking vectorial sum of $[Mv]$ & $[Mh]$ we can get Equivalent bending moment

$$[M_{eq}] = \sqrt{[M_v]^2 + [M_h]^2}$$

$$[M_{eq}] = \sqrt{[52.5 \times 10^3]^2 + [102 \times 10^3]^2} = 114.7 \text{ kN.mm}$$

According to flexural equation,

$$\sigma_b = \frac{M_{eq}}{I} Y$$

Moment of inertia, $\sigma_b = \frac{\pi d^4}{64}$

Substituting and rearranging,

$$\sigma_{b_{eq}} = \frac{2220224}{\pi d^3}$$

Tensile stress due to axial force,

$$\sigma_t = \frac{P_a}{\frac{\pi}{4} d^2}$$

$$\sigma_t = \frac{1750}{\pi d^2}$$

Total normal component,

$$\sigma_x = \sigma_t + \sigma_{b_{eq}}$$

$$\sigma_x = \frac{1750}{\pi d^2} + \frac{2220224}{\pi d^3}$$

$$\sigma_x = \frac{1750 \text{ d} + 2220224}{\pi d^3}$$

Shear stress in shaft,

$$\tau = \frac{T_{eq}}{\pi d^3}$$

$$\tau = \frac{374 \times 10^3 \times 16}{\pi d^3}$$

As 20 Mn Cr 5 material is ductile, we will consider maximum shear stress theory for diameter calculation. According to maximum shear stress theory,

$$\frac{0.5 \times S_{yt}}{2} = \sqrt{(\frac{\sigma_x}{2})^2 + (\tau)^2}$$

$$\frac{0.5 \times 760}{2} = \sqrt{(\frac{891.6\, d + 3058240}{2\,\pi\, d^3})^2 + (\frac{374 \times 10^3 \times 16}{\pi d^3})^2}$$

$$190^2 = (\frac{891.6\, d + 3058240}{2\,\pi\, d^3})^2 + (\frac{374 \times 10^3 \times 16}{\pi d^3})^2$$

Rearranging and solving for d, we get,

d = 35.23 mm

Rounding off to the nearest standard shaft size, we get, diameter of shaft to be 36 mm.

6.2.5 Bearing selection

$Pa = Pt \tan\alpha \sin\gamma = 222.9$ N

$$P_r = P_t \tan \propto \cos\gamma = 1005 \text{ N}$$

Initially selected bearing,

ISO 16007 With Co = 6950 and C = 12400

Calculating fractions,

$$\frac{F_a}{Co} = \frac{222.9}{6950} = 0.032$$

Similarly,

$$\frac{F_a}{Fr} = \frac{222.9}{1005} = 0.22$$

For above values of fractions, and referring the standard table from V. B. Bhandari, we get values of, X = 1 & Y = 0 From X and Y dynamic load (P) = XFr + YFa = 1005 N

$$L_{10} = \frac{60\, n\, L_{10h}}{10^6}$$

$$L_{10} = \frac{60\,(125)(10000)}{10^6} = 75$$

Dynamic load rating,

$$C = P\,(L_{10})^{\frac{1}{3}} = 1005\,(75)^{\frac{1}{3}} = 4238.25$$

The dynamic load rating of ISO 16007 is 12400 which is more than the calculated load rating.
So, the design is safe.

VII.3D CAD MODEL

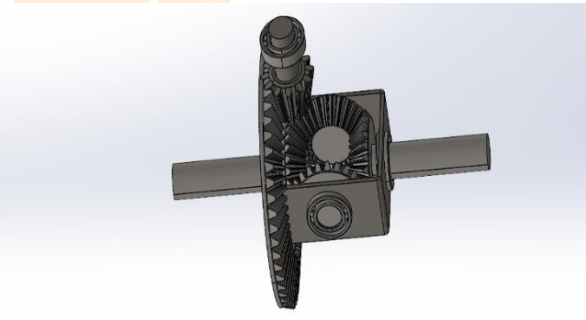


Fig. 4 Material properties

7.1 STATIC STRUCTURAL ANALYSIS

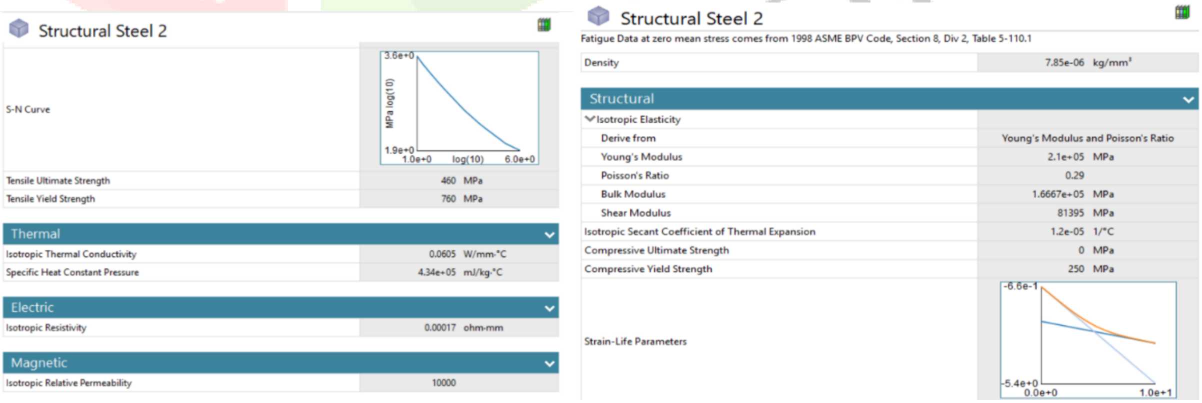


Fig. 5 Material properties

7.2 Equivalent von-mises stress

We did a static structural analysis of the open differential gearbox to see how much stress and deformation it would have when the maximum torque was applied. The parts were made from 20MnCr5 steel, which has a yield strength of 760 MPa. The analysis included the rotational load, fixed supports at the bearings, and frictional contact between the bevel gears.

Equivalent Stress From the stress plot, the highest von-Mises stress was found to be 176.2 MPa, mainly on the bevel gear teeth where the load is applied and at the joint between the shaft and gear. These are the points that naturally carry most of the torque and bending force. Since this value is much lower than the material's yield strength, the factor of safety comes out to be around 4.3, which means the design is safe. The stress pattern also looks normal, with no unwanted high-stress areas, which shows that the setup and contacts in the model are correct.

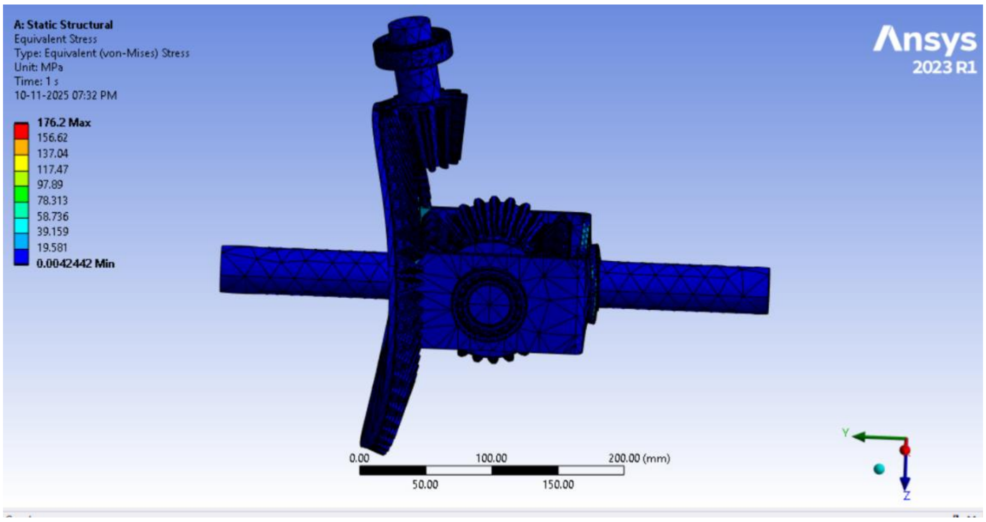


Fig.5 Equivalent von-mises stress

FOS = 760/176.2 = 4.3

FOS > 2 design is safe.

7.3 Total deformation

Total Deformation The total deformation plot (Figure Y) illustrates the overall displacement of the assembly under the same loading condition. The maximum deformation obtained was 0.0939 mm, occurring near the input bevel gear where torque was applied. The deformation gradually decreases toward the fixed supports, indicating smooth load distribution through the shafts and housing.

This deflection is significantly below the recommended alignment tolerance of 0.2 mm for small commercial vehicle differentials, confirming that the structure possesses adequate stiffness to maintain accurate gear meshing during operation. The low deformation also supports the finding that the assembly is neither under-designed nor excessively rigid, but well balanced in terms of strength and stiffness.

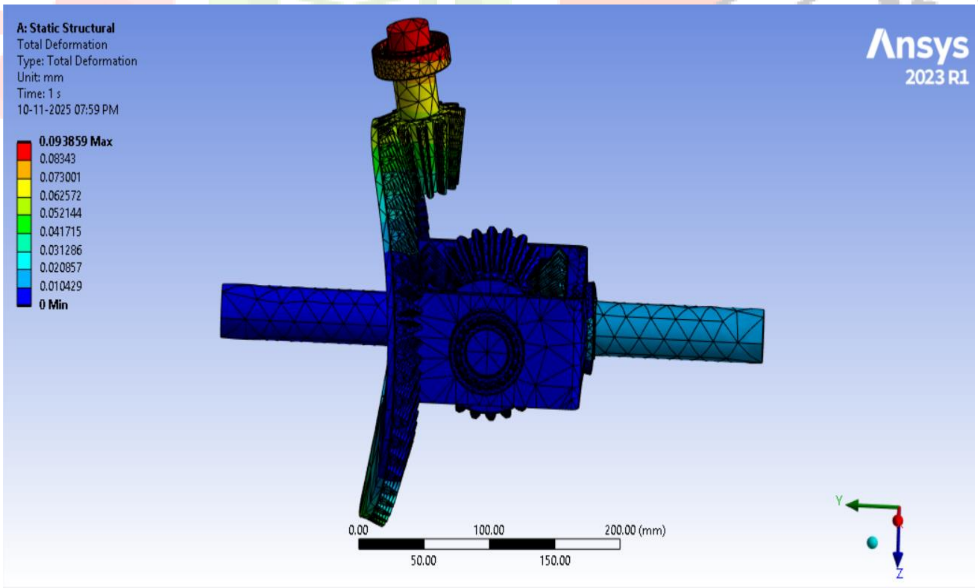


Fig.6 Total deformation

From above calculations, we can say that the design made is safe under the loading conditions i.e., the torque applied by the engine transmission shaft. Also, if we compare it with the gearbox used in the TATA ACE, we can see that the values calculated by us are similar with small deviations. Also, we can infer from the values that the design is overdesigned.

(Table no.: 4: Summary of calculations.)

Pinon gear		
1	Module	5
2	No. of teeth	12
3	Pitch angle	12.5
Crown Gear		
1	Module	5
2	No. of teeth	54
3	Pitch angle	74.77
Planet and sun gear		
1	Module	4
2	No. of teeth	25
3	Pitch angle	45

(Table no.: 5: FEA Results)

Parameter	Result	Allowable / Reference	Remarks
Maximum von-Mises Stress	176.2MPa	760 MPa (Yield of 20MnCr5)	Safe
Factor of Safety	4.3	≥ 2 (Recommended)	Satisfactory
Maximum Total Deformation	0.0939mm	≤ 0.2 mm (Allowable)	Acceptable

Overall, the finite element results demonstrate that the designed differential gearbox is structurally safe, sufficiently stiff, and suitable for the torque and operating conditions of the Tata Ace vehicle. The stress and deformation levels confirm the reliability of the selected material and dimensions, providing strong validation for the analytical design calculations.

*VIII.*CONCLUSION

In this project we have tried to design a differential gearbox considering the torque input coming from the engine transmission shaft. The dimension so calculated matched with what TATA ACE is currently using, with small deviations. A static structural analysis was also performed in ANSYS to verify the strength and deformation of the designed model. The results showed that the maximum stress and the total deformation were well within safe limits, confirming that the gearbox design is structurally safe and reliable for real operating conditions.

IX.FUTURE SCOPE

The project can be improved and expanded in several ways to make the design more advanced, efficient, and suitable for modern vehicles. Future work can focus on:

Thermal Analysis
To check how the gearbox handles heat distribution during operation.
Additive Manufacturing
To create lightweight and precise components using 3D printing technology.
Application in Electric and Hybrid Vehicles
By changing torque and speed ratios to suit different powertrains.
Design Optimization
To reduce material, use while maintaining strength and performance.

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